TECHNICAL FEATURE

Optimizing Design & Control Of Chilled Water Plants

Part 3: Pipe Sizing and Optimizing ΔT

By **Steven T. Taylor, P.E.,** Fellow ASHRAE

This is the third of a series of articles discussing how to optimize the design and control of chilled water plants. The series will summarize ASHRAE's Self Directed Learning (SDL) course called *Fundamentals of Design and Control of Central Chilled Water Plants* and the research that was performed to support its development. See sidebar, Page 34, for a summary of the topics to be discussed. The articles, and the SDL course upon which it is based, are intended to provide techniques for plant design and control that require little or no added engineering time compared to standard practice but at the same time result in significantly reduced plant life-cycle costs.

A procedure was developed to provide near-optimum plant design for most chiller plants including the following steps:

1. Select chilled water distribution system;
2. Select chilled water temperatures, flow rate, and primary pipe sizes;
3. Select condenser water distribution system;
4. Select condenser water temperatures, flow rate, and primary pipe sizes;
5. Select cooling tower type, speed control option, efficiency, approach temperature, and make cooling tower selection;
6. Select chillers;
7. Finalize piping system design, calculate pump head, and select pumps; and
8. Develop and optimize control sequences.

Each of these steps is discussed in this series of five articles. This article discusses Steps 2 and 4.

Optimizing chilled and condenser water temperatures and flow rates depends significantly on the impact flow rates have on piping sizes since piping costs are a significant part of plant first costs. So we will start by discussing optimized pipe sizing.

Pipe Sizing

Traditionally, most designers size piping using rules of thumb, such as limiting friction rate (e.g., 4 ft per 100 ft [1.2 m per 30 m] of pipe), water velocity (e.g., 10 fps [3.0 m/s]), or a combination

About the Author

Steven T. Taylor, P.E., is a principal at Taylor Engineering in Alameda, Calif.

Operating Hours/Year	≤2,000 Hours/Year		>2,000 and ≤4,400 Hours/Year		>4,400 and ≤8,760 Hours/Year	
Nominal Pipe Size (in.)	Other	Variable Flow/ Variable Speed	Other	Variable Flow/ Variable Speed	Other	Variable Flow/ Variable Speed
2 1/2	120	180	85	130	68	110
3	180	270	140	210	110	170
4	350	530	260	400	210	320
5	410	620	310	470	250	370
6	740	1,100	570	860	440	680
8	1,200	1,800	900	1,400	700	1,100
10	1,800	2,700	1,300	2,000	1,000	1,600
12	2,500	3,800	1,900	2,900	1,500	2,300
Maximum Velocity for Pipes Over 12 in. Size	8.5 fps	13.0 fps	6.5 fps	9.5 fps	5.0 fps	7.5 fps

Table 1: Piping system design maximum flow rate in gpm (Table 6.5.4.5 from ASHRAE Standard 90.1-2010).

of the two. These methods are expedient and reproducible, but they seldom result in an optimum design from either a first-cost or life-cycle cost perspective.

A better way to size piping is to use life-cycle cost analysis (LCCA) as discussed in a 2008 Journal article[1] and easily performed using a free spreadsheet[2] developed as part of the CoolTools Chilled Water Plant Design Guide funded by California utility customers through Energy Design Resources (www.energydesignresources.com). The spreadsheet is easy to use, but it is primarily intended to analyze piping systems that are completely laid out with all valves, fittings, and other appurtenances fully identified. It is not as handy to use during the early design phase when these details are not yet known.

An easier tool to use in early design is a simple look-up table showing maximum flow rates for each pipe size, such as *Table 1*. This table, which was extracted from ASHRAE Standard 90.1,[3] was developed from the LCCA spreadsheet assuming a "typical" distribution system and Standard 90.1 life-cycle cost parameters (see Reference 1 for details). The flow rates listed are the maximum allowed by Standard 90.1 for each pipe size using the prescriptive compliance approach.

Pipe Diameter (in.)	Non-Noise Sensitive			Noise Sensitive		
	2,000	4,400	8,760	2,000	4,400	8,760
1/2	7.8	5.9	4.6	1.8	1.8	1.8
3/4	18	14	11	4.6	4.6	4.6
1	29	22	17	8.9	8.9	8.9
1 1/4	51	39	30	15	15	15
1 1/2	88	67	52	24	24	24
2	120	84	67	51	51	51
2 1/2	160	120	91	81	81	81
3	270	210	160	140	140	140
4	480	360	290	280	280	280
5	670	510	390	490	490	390
6	1,100	800	630	770	770	630
8	1,800	1,400	1,100	1,500	1,400	1,100
10	2,900	2,200	1,800	2,700	2,200	1,800
12	4,400	3,300	2,600	4,200	3,300	2,600
14	6,000	4,600	3,600	5,400	4,600	3,600
16	7,400	5,700	4,500	7,200	5,700	4,500
18	10,000	8,000	6,300	9,200	8,000	6,300
20	11,000	8,800	7,000	11,000	8,800	7,000
24	17,000	13,000	11,000	17,000	13,000	11,000
26	21,000	16,000	13,000	20,000	16,000	13,000

Table 2: Piping system design maximum flow rate in gpm for variable flow, variable speed pumping systems (developed from LCCA spreadsheet assuming "green" life-cycle cost parameters).

Tables 2 and *3* are similar tables that the author uses for preliminary design of variable flow, variable speed, and constant flow, constant speed pumping systems, respectively. They were also developed from the LCCA spreadsheet assuming

California utility rates and fairly aggressive life-cycle cost parameters for discount rates, energy rate escalation, etc., which the author feels is appropriate for "green" buildings ($0.15/kWh, 1% energy escalation rate, 5% discount rate, 30-year life). The maximum flow rates are a bit lower than those in *Table 1* accordingly. *Tables 2* and *3* also include a set of flow limits for piping located in acoustically sensitive areas; again, see Reference 1 for details. *Tables 2* and *3* are useful for selecting pipe sizes in the early design phase; once the design is more complete, the LCCA spreadsheet can be used to select final pipe sizes.

Pipe Diameter (in.)	Non-Noise Sensitive			Noise Sensitive		
	2,000	4,400	8,760	2,000	4,400	8,760
1/2	5.0	3.9	3.0	1.8	1.8	1.8
3/4	12	9.0	7.0	4.6	4.6	4.6
1	19	14	11	8.9	8.9	8.9
1 1/4	34	26	20	15	15	15
1 1/2	57	43	34	24	24	24
2	73	55	44	51	51	44
2 1/2	100	77	60	81	77	60
3	180	140	110	140	140	110
4	320	240	190	280	240	190
5	430	330	260	430	330	260
6	700	530	420	700	530	420
8	1,200	900	720	1,200	900	720
10	1,900	1,500	1,200	1,900	1,500	1,200
12	2,900	2,200	1,700	2,900	2,200	1,700
14	4,000	3,000	2,400	4,000	3,000	2,400
16	4,900	3,800	3,000	4,900	3,800	3,000
18	7,000	5,300	4,200	7,000	5,300	4,200
20	7,700	5,800	4,600	7,700	5,800	4,600
24	12,000	8,900	7,100	12,000	8,900	7,100
26	14,000	11,000	8,500	14,000	11,000	8,500

Table 3: Piping system design maximum flow rate in gpm for constant flow, constant speed pumping systems (developed from LCCA spreadsheet assuming $0.15/kWh, 1% energy escalation rate, 5% discount rate, 30-year life).

	ΔT		
	Low		High
Typical Range	8°F	to	25°F
First Cost Impact	Smaller Coil		Smaller Pipe Smaller Pump Smaller Pump Motor
Energy Cost impact	Lower Fan Energy		Lower Pump Energy

Table 4: Impact on first costs and energy costs of chilled water temperature difference (assuming constant chilled water supply temperature).

Optimizing Chilled Water Design Temperatures

Table 4 shows the typical range of chilled water temperature difference (commonly referred to as delta-T or ΔT) and the general impact on energy use and first costs. The table shows that there are significant benefits to increasing ΔT from a first-cost standpoint, and there may be a savings in energy cost as well depending on the relative size of the fan energy increase versus pump energy decrease as ΔT increases. Chiller energy use is largely unaffected by ΔT for a given chilled water supply temperature. The leaving chilled water temperature drives the evaporator temperature, which in turn drives chiller efficiency; entering water temperature has almost no impact on efficiency.

Intuitively, one might think that fan energy would dominate in the energy balance between fan and chilled water pump since fan energy is so much larger than pump energy annually and the fan sees the coil pressure drop under all conditions while the chilled water pump typically only runs in warmer weather (assuming the system has an air-side economizer). But detailed analysis has shown that not to be the case: the impact on the airside of the system is seldom significant. *Table 5* shows a typical cooling coil's performance over a range of chilled water ΔTs. While the example in the table will not be true of all applications, it does suggest that airside pressure will not increase very much as chilled water ΔT rises, while waterside pressure drop falls significantly. For variable air volume systems, the impact on annual fan energy is even less significant because any full load airside pressure drop penalty will fall rapidly as airflow decreases.

Figure 1 shows the impact of chilled water ΔT on energy use for a typical Oakland, Calif., office building served by a variable air volume air distribution system with variable speed drive and an air-side economizer. Fan energy rises only slightly as ΔT increases. If pipe size is left unchanged as ΔT increases, chilled water pump energy will fall substantially due to reduced flow and reduced piping losses. In real

Chilled Water ΔT (°F)	10	13	16	19	22	25
Coil Water Pressure Drop, (ft of Water)	23.5	13.9	9.1	8.3	6.7	4.7
Coil Airside Pressure Drop (in. of Water)	0.48	0.50	0.52	0.60	0.63	0.78
Rows	6	6	6	8	8	8
Fins Per Inch (fpi)	7.4	8.3	9.4	7.7	8.6	11.6

Cooling coil pressure air- and water-side drops were determined from a manufacturer's ARI-certified selection program assuming 500 fpm (2.54 m/s) coil face velocity, smooth tubes, maximum 12 fpi fin spacing, 43°F (6°C) chilled water supply temperature, 78°F/63°F (26°C/17°C) entering air and 53°F (12°C) leaving air temperature.

Table 5: Typical coil performance vs. chilled water temperature difference.

applications, pipe sizes are generally reduced to decrease first costs, but pump energy will still fall due to reduced flow rates, although not as dramatically as in *Figure 1*.

Reducing chilled water temperature can eliminate the fan energy penalty. *Figure 2* shows the same system as *Figure 1* but instead of holding chilled water temperature constant, chilled water temperature is lowered to keep airside pressure drop (and therefore fan energy) constant as ΔT increases. Dropping chilled water temperature increases chiller energy but pump energy savings more than make up the difference. As with *Figure 1*, pump energy shown in *Figure 2* assumes that pipe sizes remain constant, which is not always the case.

Table 6 compares three cooling coils with four, six and eight rows that result in about 10°F, 18°F, and 25°F (5.6°C, 10°C, and 13.9°C) ΔT, respectively. Pipe sizes were selected from *Table 2* assuming acoustically sensitive location with about 2,000 hours per year of operation. Coil first costs were obtained from the manufacturer's representative and piping costs (including typical valve train and 20 ft [6 m] of branch piping) were obtained from the LCCA spreadsheet piping cost data. The table shows that the added cost of the deeper coil is more than offset by the savings in the cost of piping the coil. And there will be additional first cost savings from the reduced piping mains, pumps, pump motors, and variable speed drives.

So increasing ΔT reduces both first costs and energy costs. Clearly life-cycle costs will be lower, the higher the ΔT. We were unable in our analysis to find a point where the negative impact on fan energy or coil costs caused life-cycle costs to start to rise with increasing ΔT; within the range of our analysis (up to 25°F [13.9°C] ΔT), bigger ΔT was always better. Energy savings from high ΔT are even greater with systems that have water-side economizers[4] or chilled water thermal energy storage. To reiterate: our analysis suggests that it *never* makes sense to use the traditional 10°F or 12°F (3°C or 4°C) ΔTs that are commonly used in standard practice.

As ΔT is increased, eventually the ever-deepening coil will run into the Standard 62.1[5] coil pressure drop limit. Standard 62.1 uses dry coil pressure as a surrogate for the cleanability of the coil. Section 5.11.12 of the standard requires that dry coil pressure drop at 500 fpm (2.54 m/s) face velocity must not exceed 0.75 in. (187 Pa). This is roughly the pressure drop of an eight-row, 12 fpi coil (30 f/cm) (depending on the details of the fin and tube design).

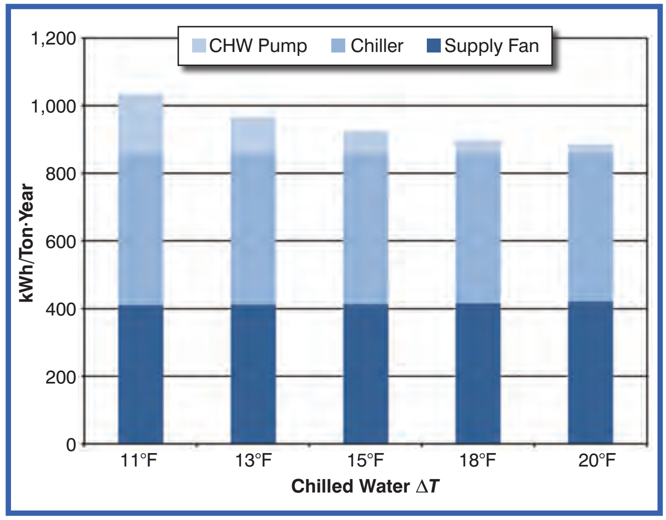

Figure 1: Typical annual energy use vs. chilled water ΔT with a constant chilled water supply temperature and constant pipe sizes.

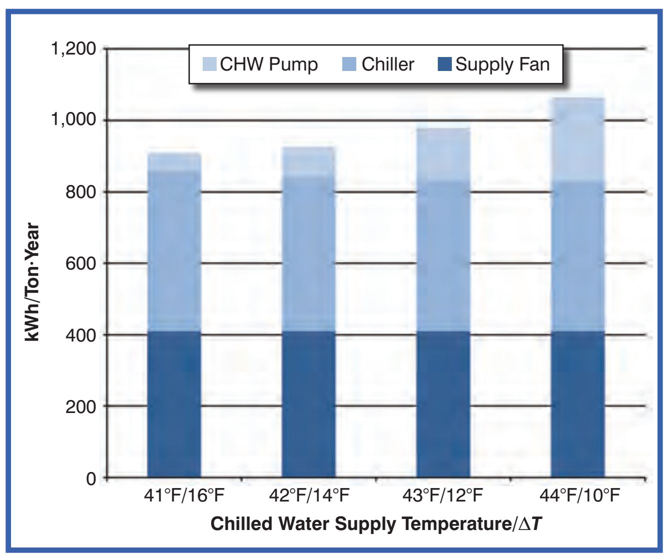

Figure 2: Typical annual energy use with coils selected for constant airside pressure drop.

So the design procedure for selecting chilled water coils is simple: rather than arbitrarily selecting chilled water temperatures and then selecting coils that deliver those temperatures,

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Coil							Piping		
FPI	Rows	Air Pressure Drop (in. of Water)	Fluid ΔT (°F)	Fluid Flow (gpm)	Fluid Pressure Drop (ft of Water)	Coil Cost	Pipe Size (in.)	Coil Connection	Total Cost
10	4	0.70	10.1	118.7	9.1	$3,598	3.0	$4,551	$8,149
11	6	0.65	18.2	66.0	7.6	$4,845	2.5	$3,581	$8,426
10	8	0.80	24.9	47.0	5.7	$5,956	2.0	$2,101	$8,057

Table 6: Cooling coil and associated piping costs for 20,000 cfm (9439 L/s) coil sized at 500 fpm (2.54 m/s), 42°F (5.5°C) chilled water supply temperature, 78°F (26°C) entering dry-bulb temperature, 62°F (17°C) entering wet-bulb temperature, and 53°F (12°C) leaving dry-bulb temperature.

reverse the logic: always use the biggest (highest effectiveness) coil available without exceeding the Standard 62.1 pressure drop limits and let the chilled water ΔT be determined by the coil and design air conditions. This will typically be an eight-row/12 fpi coil (30 f/cm).†

Based on this logic, here is the recommended procedure for sizing chilled water coils and selecting chilled water design temperatures. The intent of this procedure is to achieve all of the piping first cost savings resulting from a high ΔT but with as warm a chilled water temperature as possible to improve chiller efficiency.

1. Calculate the chilled water flow rate for all coils assuming a 25°F (13.9°C) ΔT.*

2. Pick primary pipe sizes (at pumps, headers, main risers, main branch lines) in the "critical circuit" (that which determines pump head) using *Table 2* or LCCA spreadsheet.

3. With pipe sizes selected, use *Table 2* or LCCA spreadsheet backwards to find the maximum flow for each pipe size and then recalculate the ΔT in each pipe using these flow rates. This is the minimum average ΔT for this segment of the circuit.

4. Use the coil manufacturer's selection program to find the maximum coil size that complies with the Standard 62.1 cleanability limit, typically eight-row/12 fpi (30 f/cm).† Use this for all coils so that ΔT is maximized. (For some smaller fan-coils, eight-row coils may not be an option. If so, use the largest coil available but no less than six rows. If that is not an option with the selected manufacturer, find another manufacturer.)

5. With the coil manufacturer's selection program, iterate on coil selections to determine what chilled water supply temperature results in selected ΔT *on average* for each leg of the critical circuit, starting with the coil at the end of the circuit and working back to the plant. It is not necessary that all ΔTs be the same (and, in fact, they definitely will not be the same with this approach) just that the flow through the circuit equals the maximum flow determined in Step 3. The recommended minimum chilled water supply temperature is 42°F (5.5°C).‡

6. The lowest required chilled water supply temperature for any coil in the circuit is then the design temperature.

7. Determine actual flow and ΔT in other coils in other circuits using the coil selection program with this design chilled water supply temperature, again maximizing coil size within Standard 6.1 limits (e.g., eight rows, 12 fpi [30 f/cm]) and letting the program determine return water temperature.

8. The plant flow is then calculated using the diversified (concurrent) load and the gpm-weighted average return water temperature of all coils.

Sound too complicated? Here is a shortcut procedure. Skip Steps 1 to 5 and just assume a chilled water supply temperature of 42°F (5.5°C) in Step 6. This will provide basically the same result except that the design chilled water temperature may be lower than needed to achieve the pipe size savings from high ΔT, so the chiller design efficiency may be worse

† Here is a simple way to test a coil for Standard 62.1 compliance with a manufacturer's coil selection program: Start with the desired coil including desired rows, fin type, and fin density; adjust the airflow rate up or down until the face velocity is 500 fpm (2.54 m/s); reduce the entering dry-bulb temperature to 60°F (16°C) to ensure a dry coil; then run the selection. To comply, the pressure drop under these conditions must be 0.75 in. w.g. (187 Pa) or less.

* Some engineers may be concerned that a 25°F (13.9°C) ΔT is "non-conservative" and reduces future flexibility for load changes. Designing around large ΔTs results in large coils and small pipes and pumps. Designing around small ΔTs results is the opposite, small coils and large pipes and pumps. The author has found that both are equally forgiving with respect to possible coil load changes; one is no more "conservative" than the other. If excess capacity is desired for future flexibility, the author believes it should be explicitly built into the design rather than relying on accidental flexibility from design parameters.

‡ In our analyses, the lowest chilled water supply temperature resulting from this technique was about 42°F (5.5°C); we do not know if lower chilled water temperatures will start to affect the life-cycle cost due to reduced chiller efficiency. So unless the designer performs additional life-cycle cost analysis, we suggest limiting the design chilled water supply temperature to no colder than 42°F (5.5°C). For most applications, this low temperature will not be required to achieve the target 25°F (13.9°C) ΔT. Limiting the supply temperature to 42°F (5.5°C) also provides some conservatism in the design; should there be a miscalculation in loads or unexpectedly high loads at a certain coil, chilled water temperature can be lowered below 42°F (5.5°C) to increase coil capacity, although with a resultant loss in overall chiller capacity and efficiency.

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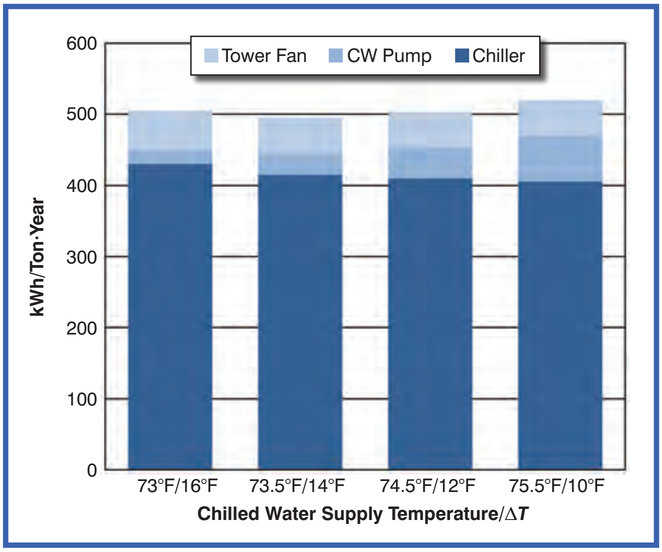

Figure 3: Oakland, Calif., office building annual energy use vs. with condenser water supply temperature and Δ*T* selected for constant tower size and constant pipe sizes.

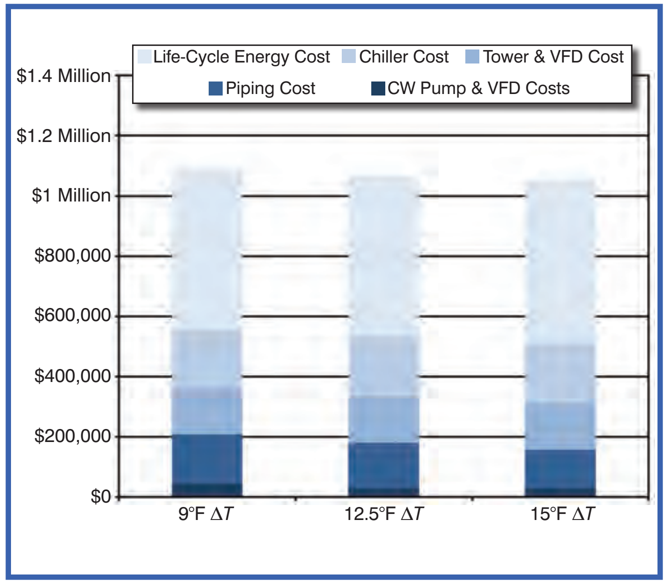

Figure 4: Life-cycle costs of 1,000 ton (3,517 kW) chilled water plant in Chicago as a function of condenser water Δ*T*.

than it needed to be. The energy impact will be minimal, however, since chilled water temperature will be aggressively reset as will be discussed in the fifth article of this series. This simplified approach also results in a somewhat lower chilled water flow rate so pump size and power will be reduced.

Optimizing Condenser Water Design Temperatures

Selecting optimum condenser water temperatures is more complex than selecting chilled water temperature due to the complex interaction between cooling towers and chillers. As with chilled water, there can be significant first-cost savings using high condenser water Δ*T*s (also known as cooling tower range). But with chilled water, the supply fan energy impact was small so increasing Δ*T* was found to always reduce total system energy costs. With condenser water, the energy impact on the chiller of increasing Δ*T* and return condenser water temperature is not small (in fact, it is very large), and Δ*T* also significantly affects the energy used by the cooling tower. So, optimum condenser water temperatures are not as easily determined as they were for chilled water.

	Δ*T*		
	Low		High
Typical Range	8°F	to	18°F
First Cost Impact	Smaller Condenser		Smaller Pipe Smaller Pump Smaller Pump Motor Smaller Cooling Tower Smaller Cooling Tower Motor
Energy Cost impact	Lower Chiller Energy		Lower Pump Energy Lower Cooling Tower Energy

Table 7: Impact on first costs and energy costs of condenser water temperature difference assuming constant condenser water supply temperature.

Table 7 shows the first-cost and energy impacts of condenser water temperature difference within the ranges commonly used in practice. Higher Δ*T*s will reduce first costs (because pipes, pumps, and cooling towers are smaller), but the net energy-cost impact may be higher or lower depending on the specific design of the chillers and towers.

Figure 3 shows chiller, tower and condenser water pump energy use for the example Oakland office building introduced in *Figures 1* and *2*. The condenser water temperature and Δ*T* were selected so that the cooling tower size and fan power do not change. As the Δ*T* decreases, the temperature of the water returning to the cooling tower decreases and the tower becomes less efficient. This requires the condenser water temperature leaving the tower to rise (or the tower size must be increased). The most energy efficient combination in this case was a 14°F (7.8°C) Δ*T*. But this assumes pipe sizing is constant; the pipe sizes could have been reduced for the larger Δ*T* designs, reducing first costs but increasing pump energy costs.

Figure 4 shows life-cycle costs for a large office building chilled water plant that was analyzed as part of the ASHRAE SDL that is the basis of this series of articles. Utility costs and life-cycle cost assumptions are those used in the evaluation of energy conservation measures for Standard 90.1 ($0.094/kWh average electricity costs and 14 scalar ratio[6] [the scalar ratio is essentially the simple payback period]). The plant was modeled in great detail

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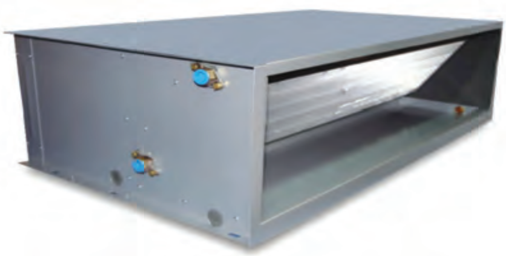

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1	2	3	4	5	6	7
Piping Section	**Application**	**gpm at 15°F ΔT**	**Pipe Size Per *Table 3***	**Maximum gpm Per *Table 3***	**ΔT at Maximum gpm**	**gpm at MaximumΔT**
Common Pipe	Constant Flow/Constant Speed, 2,000 Hours, Non-Noise Sensitive	1,850	10	1,900	14.6	1,900
To Each Equipment	Constant Flow/Constant Speed, 2,000 Hours, Non-Noise Sensitive	925	8	1,200	11.6	950

Table 8: Example condenser water pipe sizing.

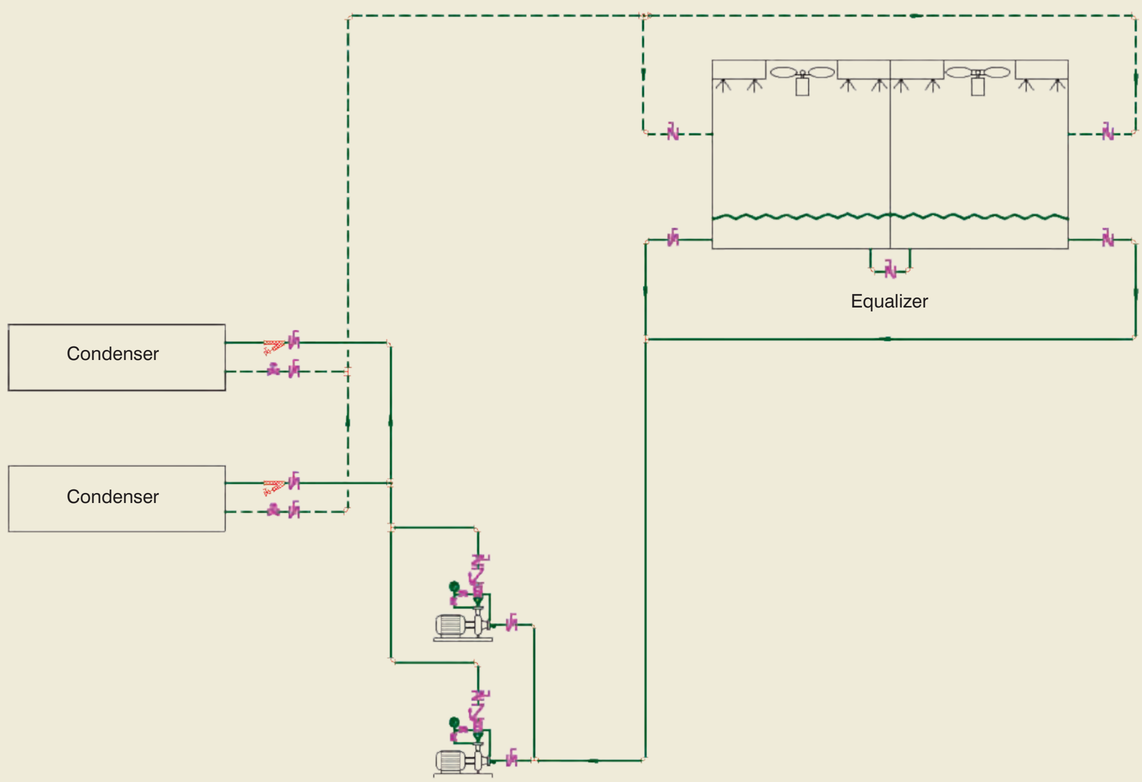

Figure 5: Condenser water piping schematic of 1,000 ton chilled water plant.

Example: Condenser Water Pipe Sizing

Figure 5 is a schematic of a 1,000 ton (3,517 kW) plant serving an office building in Oakland, Calif. Each pump, chiller, and tower is sized for half the load.

1. At the initial chiller selection efficiency of 0.56 kW/ton (6.3 COP), the rejected heat is about 13.9 million Btu/h (4 million W). So at a 15°F (8.3°C) ΔT, the total condenser water flow rate is about 1,850 gpm (117 L/s) and the flow rate to each individual piece of equipment is 925 gpm (58 L/s) (*Table 8*, Column 3).
2. For an Oakland office served by a system with an air-side economizer, the chiller plant will operate for about 2,000 hours per year. Assuming constant flow/constant speed, we can use *Table 3* for pipe sizing. Since the chiller room is not noise sensitive, the pipe sizes on the left side of the table are used. The selected pipe sizes for each of the two piping sections are shown in *Table 8*, Column 4.
3. Next, the selected pipe sizes are "maxed out" using the maximum flow rates for each from *Table 3* (shown in Column 5 of *Table 8*) and the ΔT for each piping section is recalculated using this flow rate.
4. The highest ΔT in Column 6 is selected (14.6°F [8.1°C]) and flow rates recalculated using this ΔT (*Table 8*, Column 7). These flow rates would be used to select chillers, towers, and pumps.

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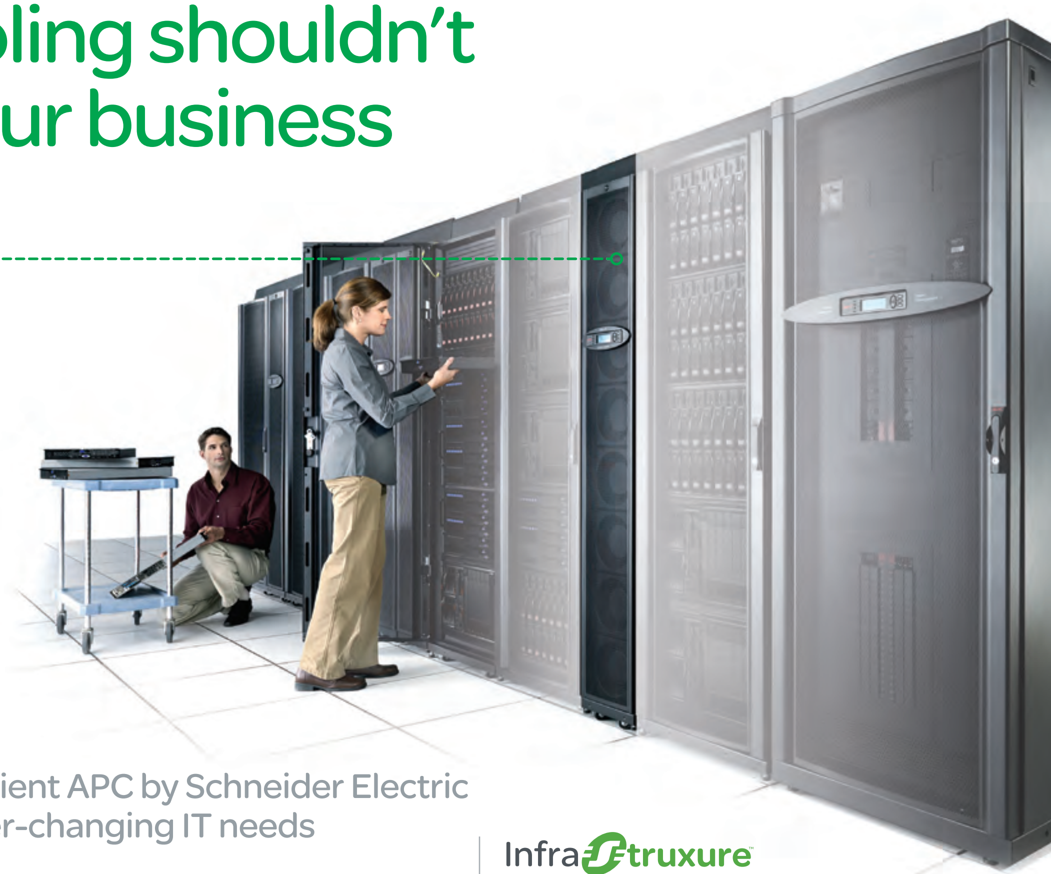

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Racks are packed with more and more equipment, driving the highest-ever rack power densities. The result: unprecedented heat levels, row by row. Meanwhile, virtualization is everywhere, leading to more dynamic loads and shifting hot spots. Tackling this challenge with traditional raised floors and perimeter cooling alone poses a real struggle: How can you locate enough cooling exactly where it's required? And how can you streamline cooling efficiency to minimize energy costs? APC by Schneider Electric™ has the efficient and effective solution you need: InRow™ cooling. InRow units come in two forms: floor-mounted units and, now, our new overhead units, both of which deliver cooling targeted at heat sources.

Variable-speed fans target heat and improve efficiency

Rack-mounted sensors monitor the temperature, giving you real-time information on where heat is hiding. As heat loads shift around the room, unique variable-speed fans automatically adjust to meet the demand. By closely matching cooling with the heat load, you use the cooling that's required in the right place at the right time, reducing waste by preventing hot and cold air from mixing and eliminating hot spots. You improve efficiency and avoid overcooling. And the overhead units give you the cooling you need within your existing floor print.

Modular design delivers maximum flexibility

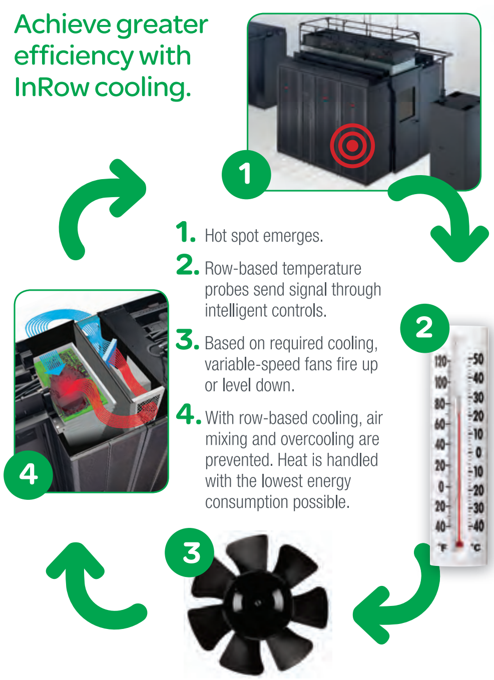
Scalable, modular InRow cooling units can be easily and quickly deployed as the foundation of your entire cooling architecture or in addition to current perimeter cooling for a high-density zone within an existing data center. With this kind of hybrid environment, there is no need to start over, and installation is quick and easy, allowing you to align your IT "on-demand" to your business needs.

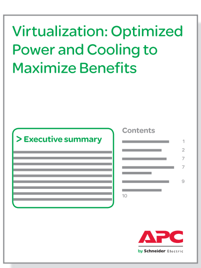
Next Generation InfraStruxure

InRow cooling is part of the Next Generation InfraStruxure™ solution, the APC by Schneider Electric one-of-a-kind scalable, adaptable, and "on-demand" data center architecture.

InfraStruxure™

Achieve greater efficiency with InRow cooling.





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Central Chilled Water Plants Series

This series of articles summarizes the upcoming Self Directed Learning (SDL) course called Fundamentals of Design and Control of Central Chilled Water Plants and the research that was performed to support its development. The series includes five parts. Part One: "Chilled Water Distribution System Selection" was published in July and Part Two: "Condenser Water System Design" was published in September. Parts Four and Five are forthcoming.

Chiller and cooling tower selection. This article will address how to select chillers using performance bids and how to select cooling tower type, control devices, tower efficiency, and wet-bulb approach.

Optimized control sequences. The series will conclude with a discussion of how to optimally control chilled water plants, focusing on all-variable speed plants.

The intent of the SDL (and these articles) is to provide simple yet accurate advice to help designers and operators of chilled water plants to optimize life-cycle costs without having to perform rigorous and expensive life-cycle cost analyses for every plant. In preparing the SDL, a significant amount of simulation, cost estimating, and life-cycle cost analysis was performed on the most common water-cooled plant configurations to determine how best to design and control them. The result is a set of improved design parameters and techniques that will provide much higher performing chilled water plants than common rules-of-thumb and standard practice.

(including real cooling tower and piping costs) for three climates: Oakland, Calif., Albuquerque, N.M., and Chicago. *Figure 4* shows results for Chicago but the trend was the same in all three climate zones: life-cycle costs were minimized at the largest of the three ΔTs analyzed, about 15°F (8.3°C).[§] This was true for both office buildings and data centers and for both single-stage centrifugal chillers and two-stage centrifugal chillers. It was also true for low, medium, and high approach cooling towers. (The optimum approach temperature will be discussed in the next article in this series but it had no impact on the optimum ΔT). In all cases, pipe, pump, pump motor, and pump variable frequency drive (VFD) sizes reduced as ΔT increased, and these cost differences were the primary driver in life-cycle cost differences as shown in *Figure 4*. The differences in energy use among the options is not as significant since savings in pump and tower energy largely (though not completely) offset the increase in chiller energy use. Other studies have also found that 15°F (8.3°C) condenser water ΔT is optimum and can even reduce annual energy costs.[7,8] The plant analyzed (shown schematically in *Figure 5*) had a relatively short distance between the towers and chillers; high ΔTs would have an even larger life-cycle cost advantage for plants that have a large distance between the two, such as a plant with chillers in the basement and towers on the roof.

Based on this analysis, the following procedure is suggested to pick the condenser water ΔT (cooling tower range):

1. Calculate the condenser water flow rate for all pipe sections assuming a range of 15°F (5°C).
2. Pick primary pipe sizes (at pumps, headers, main risers, main branch lines) in the "critical circuit" (that which determines pump head) using *Table 2* or LCCA spreadsheet.
3. With pipe sizes selected, use *Table 2* or LCCA spreadsheet backwards to find the maximum flow for each pipe size and then recalculate the ΔT in each pipe using these flow rates.
4. The largest calculated ΔT in any pipe segment is the design plant ΔT. Recalculate all flow rates using this ΔT.

This procedure attempts to minimize cost by reducing pipe size as much as possible, but then taking full advantage of the resulting pipe size to minimize ΔT to reduce chiller energy. Pump energy will be a bit higher than if a 15°F (8.3°C) ΔT were simply used, but pump energy is small relative to the impact of high ΔT on chiller energy use. An example of this technique is shown in the sidebar on Page 32.

Summary

This article is the third in a series of five that summarize chilled water plant design techniques intended to help engineers optimize plant design and control with little or no added engineering effort. In this article, optimum pipe sizing and optimum design chilled and condenser water temperature selection were discussed. In the next article, cooling tower and chiller selection will be addressed.

References

1. Taylor, S. and M. McGuire. 2008. "Sizing pipe using life-cycle costs." *ASHRAE Journal* 50(10):24 – 32.
2. Taylor Engineering. 2011. "EDR Cool-Tools Pipe Size Optimization Spreadsheet." Available for free at: www.taylor-engineering.com/publications/design_guides.shtml.
3. ASHRAE Standard 90.1-2010, *Energy Standard for Buildings Except Low-Rise Residential Buildings.*
4. Stein, J. 2009. "Waterside economizing in data centers: design and control considerations." *ASHRAE Transactions* 115(2).
5. ASHRAE Standard 62.1-2010, *Ventilation for Acceptable Indoor Air Quality.*
6. McBride, M. 1995. "Development of Economic Scalar Ratios for Standard 90.1." *Proceedings of Thermal Performance of the Exterior Envelopes of Buildings VI.* Atlanta: ASHRAE.
7. Trane. 2009. *Trane Applications Manual, Chiller System Design and Control,* SYS-APM001-EN.
8. Trane. 2005. *Trane Engineers Newsletter*, Volume 34 – 1. January. http://tinyurl.com/7flevx8.

[§] It is possible that an even larger ΔT is life-cycle cost optimum—our analysis did not look at ΔTs higher than about 15°F (8.3°C).

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