Edition No. 14 December 2002

ENGINEERING

SYSTEM SOLUTIONS

This issue of Engineering System Solutions is on chilled and condenser water temperature ranges. There is a fair amount of information circulated these days on "the best" temperature ranges to use. Some of the information is good, some not, as it was true in the "refrigerant wars" of the 1990s. The topic is complicated; changing the chilled water temperature will affect every component in the cooling HVAC system.

This article attempts to show the relationships between the components that make up the air conditioning system so that the reader can appreciate the nuances. There is no one solution, no perfect operating condition that will yield the best result for any and all buildings. The only "absolute" answer is that, unless it is backed up with an energy analysis specific to your application, care should be exercised.

For more information on air conditioning system design, contact your local McQuay representative or visit www.mcquay.com.

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Why Change the Chilled Water Temperature Range?

To answer this question, let's review the primary goal of HVAC design. The goal is to provide the customer with a safe, comfortable environment at an acceptable life cycle cost. Focusing on life cycle cost brings up the necessary balance between capital cost and operating cost. It is here that changing the chilled water temperature range becomes an issue. To consider the capital cost, let's look to the following formula:

Load (tons) = Flow (US gpm) x Temperature range (°F)/24
or
Flow (US gpm) = Load (tons) x 24/ Temperature Range (°F)

As the chilled water temperature range is increased, the flow rate is decreased for the same capacity. Smaller flow means smaller pipes, pumps, insulation (area not R value), etc. This equates to capital savings. Notice that the chilled water supply temperature is *not* part of the equation. It has nothing to do with these savings!

To evaluate the operating cost, let's consider the following formula:

Pump power (hp) = [Flow (US gpm) x Head (ft)] / [3960 x Pump efficiency]

As the chilled water flow is decreased, the pump work is also decreased, assuming constant head. This equates to operating savings. Notice again that chilled water supply temperature has *nothing* to do with the savings.

The pump head will remain approximately constant because the pipe sizes will be decreased. ASHRAE recommends that piping design be based on a 4-foot pressure drop per 100 feet of pipe. Maintaining this requirement will let the designer downsize piping as the flow decreases.

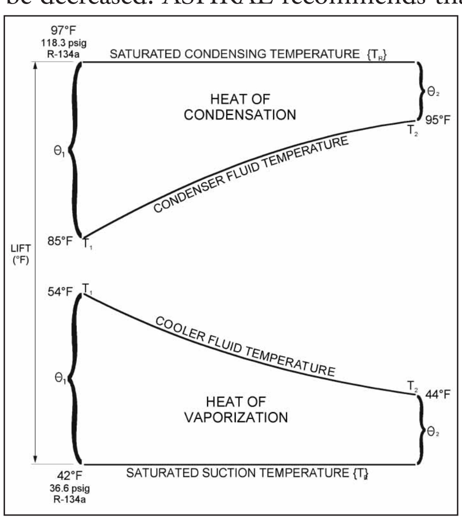

Figure 1 – Chiller Heat Exchangers at ARI Conditions

Consider the Entire System

The two considerations discussed above are two very strong arguments for increasing the chilled water temperature range. These considerations, however, are only focused on the pumps and piping. Designers must consider the entire HVAC system and changing the chilled water temperature range will affect the entire system!

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Starting with the Chiller

The purpose of a chiller is to collect heat from the chilled water loop and reject it in the condenser water loop. This process takes work, so a compressor is required. Figure 1 shows the chiller's two heat exchangers. The bottom part of the figure shows the evaporator process. Here the heat leaves the chilled water in a sensible cooling process that lowers the chilled water temperature. The heat enters the refrigerant, which boils (changes from a liquid to gas) at a constant pressure and temperature. The refrigerant must be colder than the water (heat flows downhill) and the difference is referred to as the approach.

The top portion of Figure 1 shows the condenser process. Here the condenser water is warmed by the refrigerant. The refrigerant condenses from a gas to a liquid at a constant pressure and temperature. In this case, the refrigerant must be warmer than the condenser water.

The difference between the condition the refrigerant boils at (evaporator), and the condition the refrigerant condenses at (condenser), is the compressor lift. Compressor designers measure compressor lift in enthalpy change (Btu/lb). For this discussion, temperatures will be used as the measurement. Increasing the lift will increase the compressor work and lower the chiller efficiency. This is physics and it occurs regardless of compressor types, refrigerant type or the manufacturer. The specifics will define the size of the efficiency penalty, but there will always be a penalty.

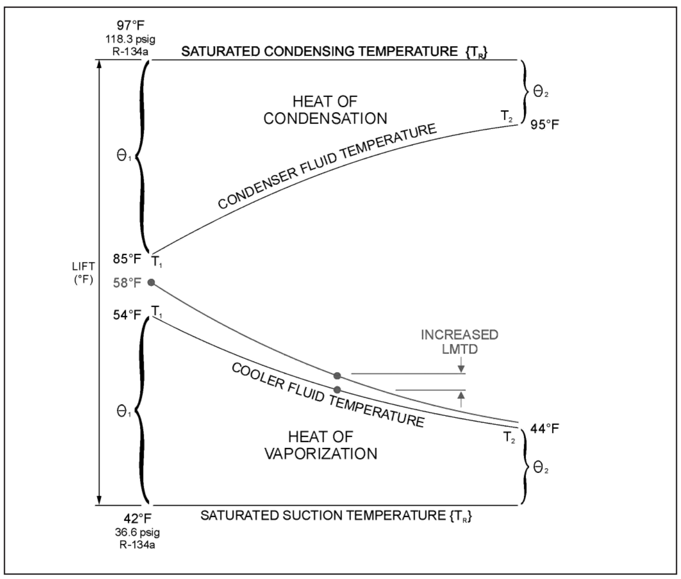

Figure 2 – Chiller Heat Exchangers at a 14°F Temperature Range

Figure 2 includes a curve for a 14°F chilled water temperature range. Heat transfer engineers describe temperature approaches in terms of Log Mean Temperature Differences (LMTD). This discussion will use average fluid temperatures. Notice the "average" water temperature for the 14°F temperature range is higher, so the chiller actually likes increasing the chilled water temperature range, if the supply water temperature is held constant. However, if the supply water temperature is lowered, then the refrigerant boiling temperature will have to be lowered. This will increase the compressor lift (See Figure 3) and lower the chiller performance.

The Effect on Cooling Coils

The cooling coils collect heat (both latent and sensible) from the air and transfer it into the chilled water, raising the water temperature. A cooling coil is a heat exchanger as well. It, too, will transfer energy proportional to its LMTD. In this case, increasing the chilled water range while maintaining the same supply water temperature will hurt the coil performance. The designer has two choices, either increase the coil area to offset the decrease in LMTD, or lower the chilled water supply temperature to maintain the LMTD.

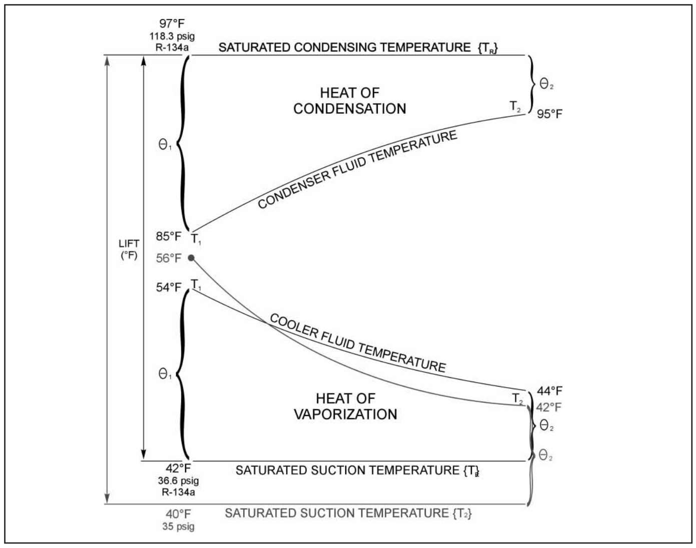

Figure 3 – Chiller Heat Exchangers at a 14°F Temperature Range, 42°F Supply Water

What About Condenser Water Temperature Ranges?

Building on the discussion so far, increasing the condenser water temperature range will decrease the pipe and pump size and the operating cost of the condenser pump. It will also require the condensing temperature to increase, which will increase the compressor lift and decrease the compressor performance. The higher "average" condenser water temperature (or increased LMTD) will improve the cooling tower performance, allowing for a smaller cooling tower.

Putting It All Together

Increasing the chilled water temperature range will reduce pipes, pumps, insulation, etc. It will also save on pump operating costs because the pump motor will be smaller.

Increasing the coil area (adding rows and fins) will increase the coil cost and increase air pressure drop. The air pressure drop increase will lead to larger fan motors and more fan work. Lowering the chilled water supply temperature will result in some combination of increased chiller cost and reduced chiller performance.

Increasing the condenser water temperature range will reduce the condenser pump, pipe and cooling tower sizes, saving capital costs. On the other hand, it will result in some combination of increased chiller cost and reduced chiller performance.

The correct answer brings the discussion full circle. The best answer is the combination of components that provides the best life cycle performance. This can only be found by performing an annual energy analysis, followed by a life cycle analysis.

Time for an Example

Let's consider an eight-story, 160,000-square foot office building in New York City. This example building uses VAV with reheat and a single chiller plant with constant flow. The chilled water pump head is 100 ft and the condenser water pump head is 50 ft. Note that a constant flow system will allow the best benefit from increasing the chilled water temperature range. The condenser loop is fixed at 3.0 US gpm per ton and 85°F entering water temperature.

Table 1 – Design Conditions Varying Chilled Water Temperature Range

Run	Chiller Capacity Tons	Perform kW/ton	Chilled Water Temp Range (°F)	Pump HP	Coil APD (in.w.c.)	Rows/fins	TSP (in.w.c.)	Fan Motor size (HP)	Total Power (HP)
1	400	0.546	10	38.5	0.62	5/10	3.0	94.8	426.1
2	400	0.546	12	32.1	0.66	5/11	3.04	96.0	420.9
3	400	0.547	14	27.5	0.70	6/10	3.08	97.3	417.6
4	400	0.547	16	24.0	0.79	6/12	3.15	99.5	416.3
5	400	0.543	18	21.4	0.87	8/9	3.25	102.7	415.3
6	400	0.543	20	19.2	0.94	8/11	3.32	104.9	415.3
7	400	0.543	22	17.5	1.10	10/10	3.48	109.9	418.6
8	400	0.543	24	16.0	1.25	12/10	3.63	114.7	421.9

Table 1 shows the design performance based on a fixed supply chilled water temperature of 44°F and increasing chilled water temperature range. As expected, the chiller performance improves, but only marginally. The chilled water pump, as well as the piping, gets smaller. The chilled water coils were selected based on an 80/67 °F (db/wb) entering air condition and a 55/54.9 °F leaving air condition. To maintain the supply air conditions, the coil area is increased through rows and fins, which increases the air pressure drop. This in turn increases the supply fan brake horsepower. Reviewing just the design performance would indicate that, until the chilled water temperature range is very large (over 20°F), there is not any need to consider lowering the chilled water supply temperature.

Table 2 – Annual Energy Analysis with VAV

Run	C.W. Range (°F)	Chiller ($/yr)	Pumps ($/yr)	Tower Fan ($/yr)	S.A. Fan ($/yr)	Total ($/yr)
1	10	26,074	15,175	1,591	28,275	71,115
2	12	26,096	13,784	1,593	28,560	70,033
3	14	26,167	12,792	1,594	28,846	69,399
4	16	26,211	12,055	1,597	29,350	69,213
5	18	26,081	11,489	1,601	30,070	69,241
6	20	26,126	11,034	1,604	30,574	69,338
7	22	26,259	10,784	1,619	31,726	70,388
8	24	26,358	10,487	1,625	32,810	71,280

Table 2 shows the annual performance in dollars rather than kWh/yr. New York City was chosen because of its reputation for high demand charges (the demand is tiered and starts at over $90/kW). This location should favor HVAC systems with the lowest peak power performance. Again, the annual performance suggests the chilled water temperature ranges can be increased a very large amount with little to no effect on operating cost, without lowering the supply water temperature at all. In essence, the capital cost can be lowered without affecting the operating cost.

Considering Constant Volume Systems

VAV systems minimize the impact of increased total static pressure. Most of the time, the fans are not running near full load. For instance, a 20% drop in airflow will drop the static pressure by 35% and power draw by 49%. Switching to constant volume with reheat will make the fans operate at design flow and power whenever the building system is operating.

Table 3 – Annual Energy Analysis with Constant Volume

Run	C.W. Range (°F)	Chiller ($/yr)	Pumps ($/yr)	Tower Fan ($/yr)	S.A. Fan ($/yr)	Total ($/yr)
1	10	40,035	19,842	2,821	70,957	133,655
2	12	40,034	18,013	2,821	71,954	132,822
3	14	40,224	16,728	2,831	72,396	132,179
4	16	40,327	15,765	2,839	73,657	132,588
5	18	40,174	15,025	2,852	75,455	133,506
6	20	40,285	14,429	2,863	76,715	134,292
7	22	40,526	13,963	2,884	79,595	137,193
8	24	40,772	13,692	2,912	82,283	139,659

Reviewing Table 3 shows that the fan penalty in a constant volume system catches up and overtakes the pump savings more quickly than in a VAV system. All the operating costs increased dramatically because constant volume with reheat operates at design air volume all the time and has significant simultaneous heating and cooling.

Adjusting Supply Water Temperature

Table 4 – Annual Energy Analysis with VAV and Declining Supply Water Temperature

Run	C.W. Range (°F)	C.W.S.T. (°F)	Chiller ($/yr)	Pumps ($/yr)	Tower Fan ($/yr)	S.A. Fan ($/yr)	Total ($/yr)
1	10	44	26,074	15,175	1,591	28,275	71,115
2	12	44	26,096	13,784	1,593	28,560	70,033
3	14	42	27,733	12,790	1,593	28,573	70,689
4	16	42	27,779	12,039	1,593	28,570	69,981
5	18	40	29,371	11,462	1,594	28,584	71,011
6	20	40	29,351	11,002	1,596	28,872	70,081
7	22	38	30,365	10,623	1,596	28,881	71,465
8	24	38	30,403	10,313	1,598	29,242	71,556

The goal should be to strike a balance between the fan work penalty and the chiller work penalty. Some combination of slowly lowering the chilled water supply temperature and increasing the coil size (rows and fins) should yield the best balance. Table 4 shows the annual energy analysis based on popular supply water temperature vs. temperature ranges. The overall operating cost scarcely changed at all. The operating costs shifted from the fans to the chillers when compared with Table 2.

Changing the Condenser Water Temperature Range

In some cases, it may be possible to reduce the capital cost of the condenser water system by increasing the condenser water temperature range. Since the condenser pump is small, it raises the question whether raising the temperature range will save operating cost. This answer is not obvious, however. The condenser pump is a fixed flow pump (in most cases), so it is either on or off. When it is on, it draws a fixed amount of power.

The chiller, however, varies its power draw with the cooling load. At high load, the chiller is dominant relative to the condenser pump, so it is important to optimize the chiller by providing a small temperature range. As the cooling load drops, the condenser pump becomes a more dominant power user. In this case, it may pay to increase the temperature range and reduce the pump power. The goal is to save enough on the pump power to offset the chiller performance penalty. At some part load operating point, optimizing the pump performance over the chiller performance will save energy and a higher temperature range is justified. The controlling factor here is pump head. Where this condition occurs can be determined through annual energy analysis.

Table 5 – Annual Energy Analysis with Increasing Condenser Temperature Range

Run	C.W. Range (°F)	Chiller ($/yr)	Pumps ($/yr)	Tower Fan ($/yr)	S.A. Fan ($/yr)	Total ($/yr)
1	10	26,074	15,175	1,591	28,275	71,115
2	11	27,084	14,562	1,592	28,283	71,521
3	12	27,517	14,049	1,592	28,286	71,444
4	13	28,094	13,616	1,592	28,290	71,592
5	14	28,527	13,245	1,592	28,293	71,657
6	15	29,057	12,923	1,593	28,297	71,870

Table 5 shows the results in increasing the condenser water temperature range one degree at a time from ARI conditions (54/44 °F chilled water, 85/95 °F condenser water). In this example, the penalty to the chiller outweighs the pump savings and it simply costs more to operate. The only argument for doing this would be the possible capital savings. To minimize the performance erosion, a better performing but more expensive chiller was used. On the capital side, enough money has to be saved to pay for the more expensive chiller. Regardless of the capital savings, this chiller will cost more to operate.

Increasing the condenser pump head will raise the part load point where the chiller/condenser pump work starts to increase. The higher the point, the more operating hours where the condenser pump is dominant. Increasing the condenser water temperature range will offset this effect. The amount of head required to justify increasing the condenser temperature range has to be found through analysis, but generally, if the cooling towers are anywhere near the chillers, it isn't worth it.

A Review So Far

Between tables 2 and 4, we designed a VAV office building 16 ways and could only manage a 3% improvement from worst to best. In this example, the best was a 16°F temperature range with 44°F supply water temperature.

Moral: There is not a huge amount of operating cost to be saved by varying temperature and range parameters. The best operating conditions in this example was only 2% better than the conventional ARI design conditions.

Moral: ARI conditions (54/44 °F chilled water, 85/95 °F condenser water) work very well for many applications.

Reviewing the design performance (Table 1) and the annual performance (Table 2) gives us different answers. Design performance is often very misleading when trying to determine the annual operating efficiency of an HVAC system. Even choosing a city with a high demand charge didn't make a huge difference.

Moral: There is no substitute for annual energy analysis. It doesn't have to be difficult or time consuming; all the analysis for this example only took a few hours using the McQuay Energy Analyzer™ program. Similar results can be obtained using other software in the same amount of time.

Reviewing the constant volume with reheat system versus the VAV with reheat system shows constant volume to be much more sensitive to increased coil area. Because VAV systems modulate the airflow, a 20% increase in design static pressure only resulted in a 14% increase in operating cost.

Moral: The whole HVAC system must be considered when designing a chilled water system. It is not possible to do a "one time" analysis and come up with the optimum design criteria.

All the designs analyzed in the above tables were based on constant chilled water flow. This clearly shows the pump savings in the best light since the pump operates at design head and flow any time there is a need for chilled water. Converting Run 1, from Table 1, the ARI conditions (54/44 °F chilled water, 85/95 °F condenser water) run, to variable primary flow yields the following results.

Table 6 – Annual Energy Analysis with Variable Primary Flow

Run	C.W. Range (°F)	Chiller ($/yr)	Pumps ($/yr)	Tower Fan ($/yr)	S.A. Fan ($/yr)	Total ($/yr)
1	10	26,072	9,915	1,591	28,273	65,851

Converting to variable primary flow reduced the pump work by 35%.

Moral: One of the best ways to reduce pump work is to use variable flow.

Reviewing Table 5, which shows increasing the condenser water temperature range, reveals that it actually drives the operating cost up. Condenser loops generally have small head requirements, so it is not advantageous to increase the range.

Moral: Always optimize the (expensive) chiller over the (inexpensive) cooling tower.

Alternatives Not Considered

To demonstrate the relationships shown in this article, not all avenues were explored. For instance, we did not consider chiller pressure drops; the higher the tube velocity, the better the chiller performance. In most cases, the chillers will be reselected to optimize chiller

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performance so it can be expected that the chiller pressure drops will not decrease with increased temperature ranges. In addition, the passes will increase (two pass to three pass) as the range increases.

Chiller types, differentiated by compressor or manufacturer, were not considered. While it is true that all chillers will behave generally the same, the specific properties of a chiller will change the outcome.

Chiller plant types will also affect the outcome. As the chilled water temperature range is increased, series chillers will outperform either a single large chiller or parallel chillers. Series counterflow chillers will outperform series chillers; however, the savings may be lost when considering the condenser pump work.

Pump head is a key parameter. Increasing the pump head will make the pump work more dominant and change the outcome.

Conclusion

Temperature ranges are a complicated issue. There is no single best solution that can be used for every case. Each project will have its own optimum operating conditions. To further complicate matters, different chillers (either types or manufacturers) will have their own optimum operating conditions. Said another way; the best McQuay solution may not necessarily be the best solution for other chiller vendors and vice-versa.

All chillers are negatively affected by increased compressor lift. Some chiller types (screw chillers, for instance) are less affected. Casually increasing the compressor lift for any chiller type or manufacturer will increase the operating cost. Changing the temperature ranges and supply temperatures requires careful analysis. The following are some points to consider:

- The traditional ARI operating conditions work very well for many buildings.
- Unnecessary reduction of the chilled water supply temperature should be avoided because it increases chiller work.
- When using standard products such as fan coils and unit ventilators, maintain the chilled water temperature range between 10 and 12°F where they are designed to operate.
- Increasing the chilled water temperature range is a good way to reduce the capital and pump operating cost, particularly if the pump head is large or the piping runs long.

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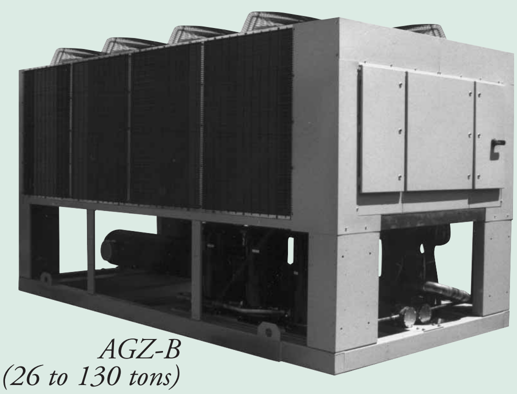
AGZ-A
(10 to 39 tons)



AGZ-B
(26 to 130 tons)

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- With larger chilled water temperature ranges, it may be necessary to lower the supply water temperature to find a balance between coil and fan performance vs. chiller performance. Annual energy analysis using the McQuay Energy Analyzer™ program is recommended.
- If the chilled water supply temperature is reduced, consider oversizing the cooling tower to reduce the condenser water temperature and minimize the affect on the chiller.
- Always take into account the actual design ambient drybulb or wetbulb conditions when designing a chiller plant. If the location is arid, then lower the wetbulb design as indicated by ASHRAE design weather data, and select both the cooling tower and chiller accordingly.
- For very large chilled water ranges, use series chillers, possibly with series counterflow condenser circuits, to optimize chiller performance.
- Increasing the condenser water range should only be considered for projects where the piping runs are long and the pump work high. When it is required, optimize the flow to the actual pipe size that is selected and select the chillers accordingly. In the example in Table 5, the temperature ranges 10 through 12°F would all need 8-inch pipe, while the ranges 13 through 15°F would need 6-inch pipe. There are little capital savings to go beyond the 13°F temperature range. Consider oversizing the cooling towers to minimize the effect on the chiller. Remember that cooling towers are significantly less expensive than chillers.

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