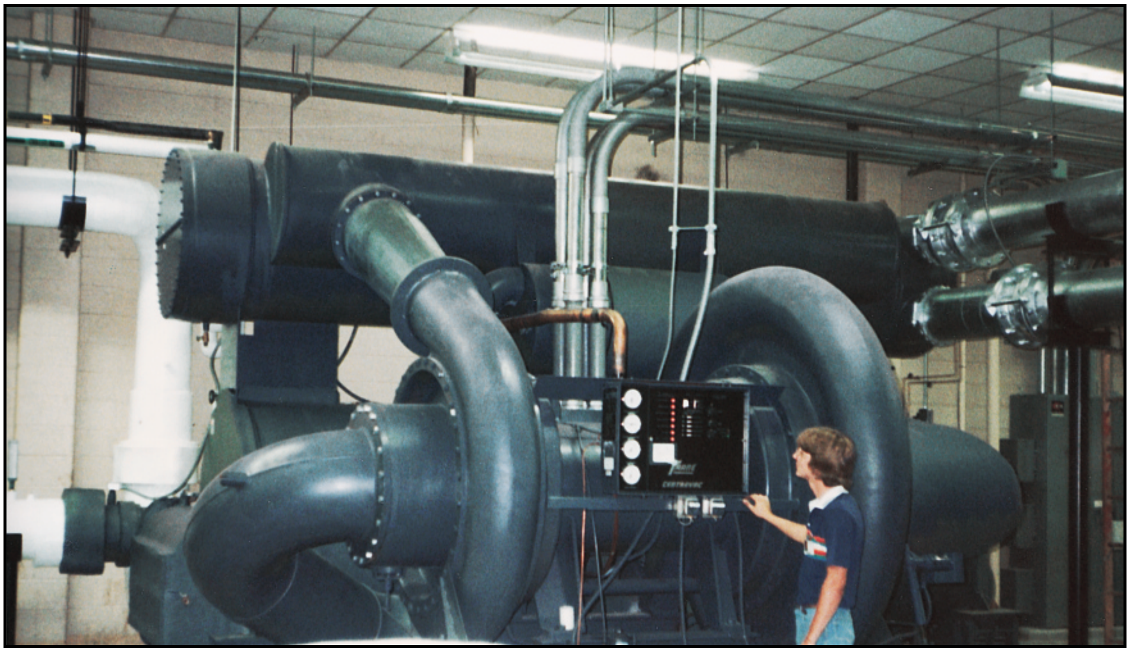

Reducing CHS temperature penalizes chiller efficiency but can save pumping power.

# Designing for 42°F Chilled Water Supply Temperature— Does It Save Energy ?

**By Wayne Kirsner, P.E.**
Member ASHRAE

In an article published in the February 1996 issue of *ASHRAE Journal*, I posed the question: "3 GPM/Ton Condenser Water Flow Rate: Does It Waste Energy?" The article examined the assertion made in several preceding ASHRAE articles and transactions that chiller/cooling tower systems designed for 2 or 1.5 gpm/ton (0.036 or 0.026 mL/J) are more energy efficient than those designed for the rule-of-thumb 3 gpm/ton (0.054 mL/J) condenser water flow rate. Lowering the condenser water flow rate raises the chiller's condensing temperature, thereby increasing the head over which the compressor must lift the refrigerant.

This article deals with the other variable which increases compressor head—lowering evaporator temperature. It examines the assertion that there is a net energy advantage for producing and circulating lower chilled water supply (CHS) temperature water at an expanded ΔT as compared to 44°F or 45°F* CHS temperature at a 10°F ΔT. Specifically, this article compares the relative power consumption of a chilled water system designed for 45°F CHS at a 10°F ΔT across cooling coils with a similar system designed for 42°F chilled water at a 14.3°F ΔT. In either case, it is assumed that the same leaving air temperature is desired across the cooling coils. The only variables are the CHS temperature and ΔT.

The advantage of supplying colder chilled water to cooling coils is that *less flow* is required to achieve the same heat transfer through identical coils. *Table 1* shows that a typical chilled water coil requires about 70% of the chilled water flow to achieve the same leaving air temperature when supplied with 42°F CHS as compared with 45°F CHS. If an entire system requires only 70% of the chilled water flow (CHW) because it was designed for lower CHS temperature, pumping power can be reduced by roughly the cubic power of 0.70 to about one-third the base pumping power. This can amount to a significant horsepower reduction when pumping a large system.

* *All temperatures are expressed in Farenheit °F. 42°F = 5.56°C, 45°F = 7.22°C, 10°F ΔT = 5.56°C ΔT.*

**About the Author**

**Wayne Kirsner, P.E.**, is a consulting engineer in Atlanta. He received his masters degree in physics in 1973 and his BSME in 1980 from Georgia Tech. He has written a number of papers on steam systems and chilled water system design.

| EDB,EWB °F | LDB/LWB °F | FPM | Row/FPI | GPM at ΔT 45°F CHST | GPM at ΔT 42°F CHST | % GPM at 42 vs 45 |
|---|---|---|---|---|---|---|
| 80/67 | 56.0/55.0 | 500 | 6/8 | 148 @ 10°F | 103 @ 14.3°F | 69.6% |
| 80/67 | 54.7/54.0 | 500 | 6/9.6 | 238 @ 10°F | 172 @ 13.8°F | 72.1% |
| 80/67 | 53.2/52.7 | 500 | 6/12 | 173 @ 10°F | 122 @ 14.3°F | 70.3% |
| 78/64 | 56.0/55.0 | 500 | 4/9 | 13.5 @ 10°F | 9.1 @ 14.7°F | 67.4% |
| | | | | | Average = | **70.0%** |

***Table 1: Coil GPM Requirement for 42 vs 45°F CHST.***

*Each row represents a coil selection at two different entering chilled water supply temperatures. The last column compares the GPM required at 42°F versus 45°F entering water temperature and a 10° ΔT to achieve the same leaving wet bulb condition.*

The disadvantage of supplying colder chilled water temperature is that the chiller's evaporating temperature must be depressed to achieve the lower temperature. This increases the compressor's work per ton of cooling. The question is: *which is greater?* Is the chiller energy penalty (produced by decreasing the evaporating temperature to provide 42°F CHW) greater or less than the pumping energy saved by circulating only 70% of the chilled water flow at an expanded ΔT?

## The Chiller Energy Penalty

The chiller energy penalty arises from the decrease in evaporator temperature necessary to produce 42°F CHS compared with 45°F CHS. The evaporator temperature is determined by the requirement to maintain sufficient mean temperature difference between the chilled water and the evaporating refrigerant in order to transfer the heat removed by the chiller. The log mean temperature difference, or LMTD, for the case of 45°F CHS and 10°F ΔT is shown in *Figure 1*. Its value is 8.55°F (4.75°C). *Figure 2* depicts the LMTD for the alternate chilled water supply condition of 42°F CHS and 14.3°F (8.2°C) ΔT. The required LMTD changes slightly. By determining the necessary LMTD for this lower temperature condition, the necessary change in evaporator temperature can be calculated. The change in evaporating temperature, in turn, will determine the theoretical change in kW/ton for an alternate chiller selection at the lower chilled water supply condition.

So, for the same load $Q$ and the same heat transfer surface area $A$, the LMTD must remain the same except for a small correction due to the change in the heat transfer coefficient $U$ across the evaporator tubes due to reduced water velocity. This is demonstrated in the following equations.

To transfer the same cooling load:

$$Q_{42°,14\Delta T} = Q_{45°,10\Delta T}$$

$$\text{or} \quad U_{42°,14} \cdot A \cdot \text{LMTD}_{42°,14} = U_{45°,10} \cdot A \cdot \text{LMTD}_{45°,10}$$

$$\text{so} \quad \text{LMTD}_{42°,14} = (U_{45°,10}/U_{42°,14})\ \text{LMTD}_{45°,10}$$

or, equivalently, in terms of heat transfer resistance "$R$" = $1/U$:

$$\text{LMTD}_{42°,14} = (R_{42°,14}/R_{45°,10})\ \text{LMTD}_{45°,10}$$

To execute this calculation so that the new $T_{evap}$ may be calculated, the ratio $R_{42°,14}/R_{45°,10}$ must first be determined. The overall resistance across an evaporator $R$ consists of the sum of the refrigerant side film resistance, copper wall resistance, fouling factor, and the waterside film resistance. Each of these resistances remain unchanged except for the waterside film resistance

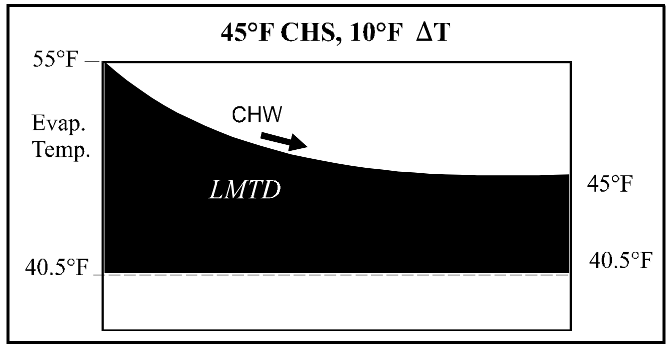


***Fig. 1:*** $\text{LMTD}_{45°,10} = \dfrac{(55-40.5)-(45-40.5)}{\ln[(55-40.5)/(45-40.5)]} = 8.55°\text{F}$

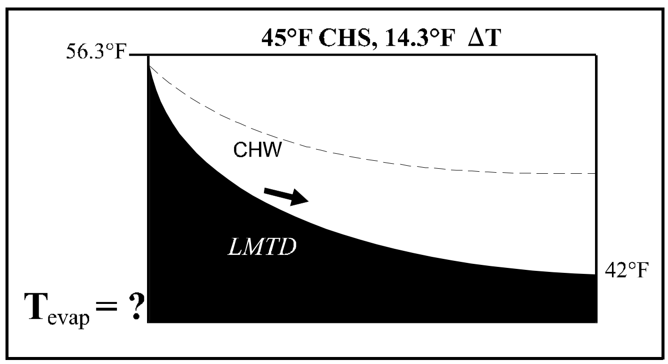


***Fig. 2: Evaporating temperature $T_{evap}$ must drop in order to restore LMTD to drive the same heat transfer as Figure 1.***

which increases due to the reduction in water flow velocity at 14.3°F (7.9°C) ΔT. The waterside film heat transfer coefficient, $h_w$ is proportional to flow velocity to the 0.8 power so that at two-thirds of the flow, $h_w$ falls to $(0.70)^{0.8}$ = 75% and water film resistance increases to 1/.75 = 133% of its full flow value.[1]

Typically, from 35 to 40% of the overall resistance to heat transfer across the evaporator tubes is due to the resistance of the water film. Thus, the increase in water film resistance causes the overall resistance to heat transfer to increase by about $(0.375 \times 0.33)$ = 12.4%. If resistance to heat transfer increases 12.4%, then the mean temperature difference must also increase to offset it. Thus, substituting 1.124 for $R_{42°,14}/R_{45°10}$ into the previous formula:

$$\text{LMTD}_{42°,14} = 1.13\ \text{LMTD}_{45°,10}$$

and for $\text{LMTD}_{45°,10}$ = 8.55°F:

$$\text{LMTD}_{42°,14} = 1.124 \times (8.55)°\text{F} = 9.61°\text{F}\ (5.34°\text{C})$$

## Simple Cycle Pressure – Enthalpy Diagrams

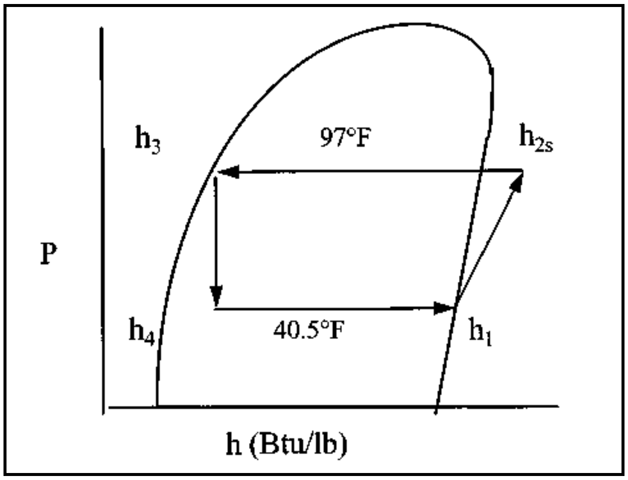


*Fig. 2a: Single stage chiller.*

$$(\mathrm{kW/ton})_s = \frac{h_{2s} - h_1}{h_1 - h_4} \times \frac{12,000\ \mathrm{Btu/\#}^2}{3,413\ \mathrm{Btu/kWh}}$$

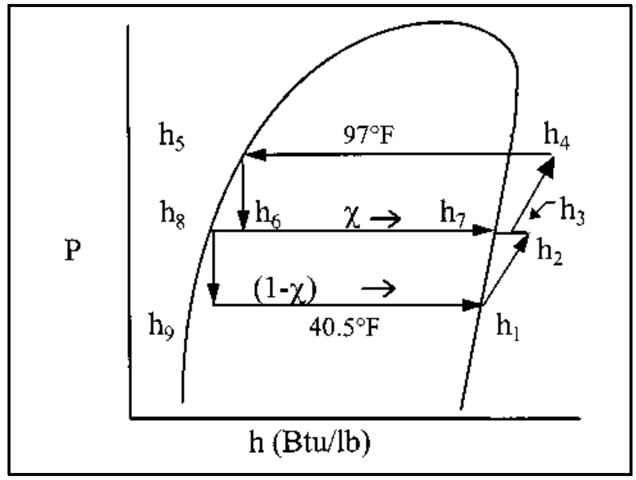


*Fig. 2b: Two-stage chiller.*

$$(\mathrm{kW/ton})_s = \frac{(h_4 - h_3) + (1 - \chi)(h_2 - h_1)}{(1 - \chi)(h_1 - h_9)} \times \frac{12,000}{3,413}$$

$$\text{where} \quad \chi = \frac{h_6 - h_8}{h_7 - h_8}$$

*In Figure 2a, the isentropic work per pound of refrigerant required by the single stage compressor is $h_{2s}$–$h_1$. The refrigerating effect is $h_1$–$h_4$. Thus, the isentropic adiabatic work required per unit cooling effect received is simply $(h_{2s}–h_1)/(h_1–h_4)$, or expressed in kW/ton, 3.52 times this ratio. Figure 2b shows the simple refrigeration cycle for a two-stage compressor and economizer with the formula for isentropic kW/ton below.*

| | | (KW/ton) @ Evap. Temperature = | | (KW/ton) | (KW/ton) |
|---|---|---|---|---|---|
| **Refrigerant** | **# Compressor stages** | **40.5°F** | **37.7°F** | **Penalty** | **°F** |
| R134a | Single Stage | – | – | 0.038 | 0.014 |
| R22 | " | 0.585 | 0.621 | 0.036 | 0.013 |
| R123 | " | 0.554 | 0.587 | 0.033 | 0.012 |
| R123 | Two Stage | 0.547 | 0.578 | 0.032 | 0.011 |
| R123 | Three Stage | 0.539 | 0.570 | 0.031 | 0.011 |
| | | | | **Average** | **0.012** |

***Table 2: Theoretical change in kW/ton for a reduction in evaporating temperature.***
*Based on simple refrigerant cycle with subcooling as indicated and assuming: 82% compressor efficiency, 95% motor efficiency, 1% gear loss (single-stage only). * Sub-cooling equals 9°F at 97°F on single-stage chillers only. Multi-stage chillers do not incorporate sub-coolers. Condensing temperature is 40.5°F.*

Knowing the value $\mathrm{LMTD}_{42°,14\Delta T}$ allows us to now solve for the evaporating temperature $T_{evap}$ at 42°F CHS and 14.3°F ΔT:

$$9.61°\mathrm{F} = \frac{(56.3 - T_{evap}) - (42 - T_{evap})}{\ln[(56.3 - T_{evap})/(42 - T_{evap})]}$$

$$\text{so} \quad T_{evap} = 37.83°\mathrm{F}\ (3.24°\mathrm{C})$$

Thus, the evaporating temperature falls 2.67°F (1.48°C) from 40.5°F (4.7°C)—the value assumed for 45°F CHS and 10°F ΔT—in order to drive the same heat transfer per unit area across the evaporator tubes.

Knowing the change in evaporating temperature, the theoretical chiller power penalty can be determined from refrigerant enthalpy tables and the simple vapor compression cycle for a refrigeration machine. The applicable formulae are shown beneath *Figures 2a* and *2b.* These figures show the simple vapor compression cycles for both a single stage and two stage chiller.

*Table 2* tabulates the theoretical change in power consumption based on refrigerant enthalpy tables and the formulae in *Figure 2* for a drop in evaporating temperature. Each of the refrigerants and multi-stage compressors in common use are represented. The kW/ton figures are adjusted for compressor efficiency, motor efficiency and drive train efficiency as shown in the next formula so as to closely approximate the actual chiller penalties.

$$\mathrm{kW/ton} = \frac{(\mathrm{kW/ton})_s}{\eta_c \eta_{mot} \eta_{gears}}$$

Compressor adiabatic efficiency $\eta_c$ and motor efficiency $\eta_{mot}$ are presumed to be 82% and 95%. Drive train efficiency $\eta_{gears}$ is presumed to be 99% for single-stage chillers and 100% for multi-stage chillers since the major manufacturer of multi-stage machines uses a direct-driven design. Bearing and shaft seal losses are neglected since these are essentially fixed losses which cancel out when examining changes in kW/ton for two operating conditions.

*Table 2* shows that the power penalty per ton of cooling for an increase in evaporating temperature averages 0.012 kW/ton penalty per 1°F (0.006 $kW_{input}/kW_{load}$ per 1°C) change in evaporating temperature**.** However, this figure assumes no *change* in motor, transmission or compressor efficiency at the lower temperature condition.

$\eta_{gears}$ and $\eta_{mot}$ generally should not change for a small shift in evaporating temperature. On the other hand, the compressor efficiency, $\eta_c$, will vary depending on where the new operating point falls on its compressor curve relative to where the base point fell on its compressor curve. For optimally selected compressors, the efficiency will most often decrease slightly for compressors selected to operate at higher head.

The effect of a decrease in compressor efficiency on the kW/ton penalty is significant. It can be calculated by multiplying the percent change in compressor efficiency times the chiller kW/ton at the new condition. For example, a 1% loss in compressor efficiency for an 80% efficient compressor in a 0.60 kW/ton chiller will cause an increase in the kW/ton penalty of:

$$\begin{aligned}\text{Increase in kW/ton Penalty per \% } \eta_c &= \frac{1\%}{82\%} \times 0.60 \text{ kW/ton} \\ &= 0.007 \text{ kW/ton } (0.0023\ kW_{input}/kW_{loc}\end{aligned}$$

Thus, the net ΔkW/ton penalty for a typical chiller, including a modest 1% decrease in compressor efficiency, would be (2.67°F × .012 kW/ton – °F) + 0.007 kW/ton = .039 kW/ton, or about **.015 kW/ton per °F** (0.0077 $kW_{input}/kW_{load}$/°C). Compressor efficiency will typically degrade from 0.5 to 3% for selection at reduced chilled water supply temperature, so the bold figure above is a conservative estimate of the actual penalty for a specific chiller selection. To get a sense of the ΔkW/ton penalty for some actual chiller selections, *Table 3* in the sidebar lists data from two manufacturers' selections programs.

### *The Chilled Water Pump*

At a 14.3°F (7.94°C) ΔT, the required chilled water flow will

## Manufacturers Rate 42°F vs. 45°F

| Chiller | Tons | Evap. Temp. (°F) @ | | Δ Evap. T | kW/ton @ | | Δ kW/ton | Δ kW/ton per°F | Inferred $\Delta\eta_c$% |
|---|---|---|---|---|---|---|---|---|---|
| | | 45°CHS | 42°CHS | | 45°CHS | 42°CHS | | | |
| #1 | 500 | 41.09 | 38.54 | 2.55 | 0.584 | 0.618 | 0.034 | 0.0133 | 0.3 |
| #2 | 750 | 43.07 | 40.39 | 2.68 | 0.544 | 0.585 | 0.041 | 0.0154 | 1.1 |
| #3 | 1000 | 40.01 | 37.20 | 2.81 | 0.574 | 0.617 | 0.043 | 0.053 | 1.1 |
| #4 | 500 | 41.20 | 38.79 | 2.41 | 0.610 | 0.664 | 0.054 | 0.0224 | 3.0 |
| | | | **Averages** | **2.61°F** | | | **0.043** | **0.0165 kW/tn/°F** | |

***Table 3: Manufacturers' data kW/ton increase at 42° vs. 45° CHS temperature R-123 refrigerant.***

Manufacturers were asked to make an economical selection for a specific tonnage at 45°F CHS temperature to achieve a power input of approximately 0.60 kW/ton (C.O.P. = 5.87). Then the manufacturers made a second selection at 42°F CHS with a 14.3°F (7.94°C) ΔT while holding shells constant, but allowing the compressor, impeller, gears and motor to be optimized. The ΔkW/ton column shows the resulting change in kW/ton for the selections at the two conditions. All selections are for R-123 refrigerant.

The manufacturers' ratings for the selections in *Table 3* indicate a ΔkW/ton penalty in excess of the 0.031 kW/ton predicted by the refrigerant cycle analysis alone. The difference can be explained by accounting for the following factors:

1. The decrease in evaporating temperature shown in the fifth column is different than 2.67°F (1.48°C), which was calculated based on the conservative assumption that the water film resistance made up only 37.5% of the total resistance to heat flow across evaporator tubes. If the resistance proportion of the water film is smaller than presumed, the change in evaporating temperature will be less. This is the case with chiller #4 in which tube velocity is very high and super enhanced tubes were selected.

2. Because of their higher power requirement per ton, the selections at 42°F CHS reject slightly more motor heat through their condenser. This requires a slightly higher condensing temperature to increase the rate of heat transfer through the condenser. The higher condensing temperature contributes from 0.001 to 0.002 kW/ton to the power consumption of the chiller and hence to the ΔkW/ton penalty.

3. The remaining differences not accounted for by the earlier corrections are because of compressor efficiency degradation. The degradation inferred by the difference in the theoretical ΔkW/ton prediction versus the manufacturer's ratings with the earlier corrections taken into account is estimated in the last column. (The theoretical ΔkW/ton prediction is calculated by multiplying the actual evaporating temperature difference times .012 or .011 kW/ton as applicable from *Table 2* and then adding the small correction for the increase in condensing temperature.) The range of compressor efficiency degradation shown in the table is from 0.3 to 3.0%.

be 70% of that required for a 10°F ΔT. If all system components remain unchanged (i.e., pipe, valves, coils, strainers etc.), system head will be reduced in approximately half, calculated like so:

$$\text{Head}_{42°-15\Delta T} = (0.70)^{1.85}(\text{Head})_{45°-10\Delta T}$$
$$= 0.52(\text{Head})_{45°-10\Delta T}$$

Pumping horsepower, which equals flow times head, will be reduced to one-third, calculated like so:

$$\text{HP}_{42°-15\Delta T} = (0.70)(\text{Flow})_{45°-10\Delta T} \times (0.52)(\text{Head})_{45°-10\Delta T}$$
$$= 0.36\ \text{HP}_{45°-10\Delta T}$$

Thus, a reduction of 64% of the full flow base pumping horsepower can be achieved. Does a savings of almost two-thirds of the initial pumping power exceed the chiller kW penalty for 42°F CHS operation? The answer depends on the pump head for the system under consideration.

### The "Break-Even" Head

In terms of an equation, the question can be expressed as follows on a per ton basis:

Is .64 [Pumping HP/ton at 45°CHS – 10Δ*T*]
> Chiller ΔkW/ton Penalty at 42°CHS – 15°Δ*T*?

Pumping horsepower/ton is a function of the system head alone if we assume 80% pumping efficiency and 93% motor efficiency. Then the inequality becomes:

$$.64 \times \frac{2.4\ \text{gpm/ton} \times (\text{Head})_{45°-10\Delta T}}{3960 \times 0.80} \times \frac{0.746\ \text{kw/hp}}{0.93}$$
$$> \text{Chiller } \Delta\text{kW/ton Penalty}$$

$$(\text{SI})\quad .64 \times \frac{0.054\ \text{mL/J} \times (\text{Head})_{45°-10\Delta T} \times 0.001\ \text{m}^3/\text{L}}{0.80 \times 0.93}$$
$$> \text{Chiller kW/kW}_{chiller,\ f}$$

Solving for the Head at which the pumping HP savings will exceed the chiller penalty:

$$\text{Head}_{42°-14.3\Delta T} > 2571 \times \Delta\text{kW/ton}_{42°\ \text{vs}\ 45°}$$

By substituting 0.015 kW/ton/°F × 2.67°F = .040 ΔkW/ton calculated earlier for the average chiller penalty, the "Break-Even" head equals 103 ft of water gage (308 kPa). For each of the manufacturers' selections shown in *Table 3*, the break-even head is tabulated in *Table 4* using the formula above.

For a *typical* chiller selection suffering a 1% compressor efficiency degradation, then, the break-even head is about 100 ft of water gage (300 kPa). So, in general, designing for 42°F chilled water supply temperature and 15°F ΔT below 100 ft of pumping head will be a net energy loser. Above 100 ft, it could be an energy winner, but one must examine part load operation.

### Part Load

When dealing with part load, the chiller penalty, in terms of kW (not kW/ton) for supplying 42 vs. 45°F CHS will diminish linearly and in direct proportion to part load. Pumping horsepower, in contrast, if applied in a variable flow system, will diminish at a rate more quickly than linearly with part load. The functional relationship of pumping horsepower to part load CHW flow can vary anywhere from slightly greater than linear

| Chiller | Tons | ΔkW/ton | Δηc% | Break-even Head |
|---|---|---|---|---|
| #1 | 500 | .034 | .3 | **87 ft.** |
| #2 | 750 | .041 | 1.1 | **105 ft.** |
| #3 | 1000 | .043 | 1.1 | **111 ft.** |
| #4 | 500 | .054 | 3.0 | **139 ft.** |

***Table 4:** ΔkW/ton and break-even head for 42°F vs. 45°F CHS selection.*

to faster than a cubic power. The relationship depends on the relative response of each terminal throttling valve at part load and hence cannot be generalized. (The family of curves in *Figure 3* popularized by Burt Rishel represent the various functional relationships by which *system head* could vary with flow.)

Because pumping horsepower equals flow × head, pumping horsepower will fall off, at a minimum, at a rate slightly greater than linearly, and thus, *more rapidly than the chiller penalty.* Thus, if the chiller kW penalty exceeds the pumping kW savings at full load, it will exceed it at all part loads too.

### Conclusion

We may conclude that unless system pumping head at 2.4 gpm/ton (0.043 mL/J) is well in excess of 100 ft (300 kPa), there is an energy *penalty* to design chillers for 42°F CHS at 14.3°F ΔT as opposed to 45°F at 10°F ΔT. If total head is well above 100 ft there will be energy savings *at full load* for designing for lower CHW supply temperatures at expanded ΔT's. However, whether or not this energy advantage is sustained at part loads will depend on the response of the particular pumping system. Therefore, if the pumping head is below 100 ft, it is probably advantageous from an energy standpoint to maximize chiller efficiency by choosing a high CHS temperature—such as 45°F.

#### Consideration of Low ΔT Syndrome

In the November '96 and February '95 issues of *Heating/Piping/Air Conditioning,* I wrote about the prevalence of "Low ΔT Syndrome" in central chilled water plants—that is, the failure of chilled water systems to return chilled water to the central plant at a temperature approaching the design ΔT. If

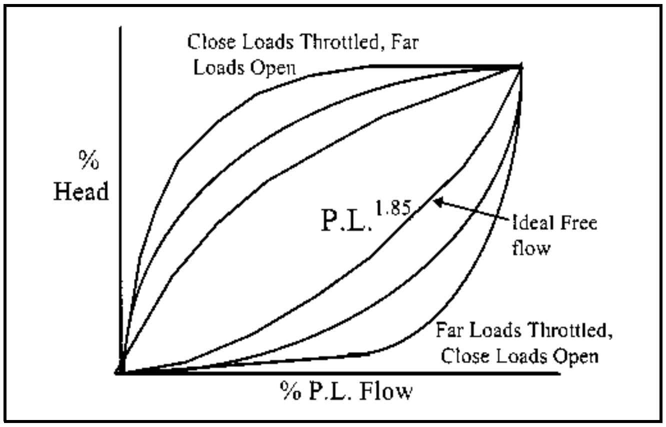


***Fig. 3:** Family of % Head vs. % Flow Curves for variable flow CHW system with multiple throttled loads.*

the prevalence of "Low ΔT Syndrome" were considered in this analysis, would it undermine the expectation of pumping savings for the 14.3°F ΔT alternative? The answer is—not really. Although "Low ΔT Syndrome" is most often associated with large distributed chilled water systems that usually are designed for large ΔTs, it is as likely to occur, if not more so, in a system of comparable size and complexity designed for 10°F ΔT. Thus, either design is equally likely to suffer.

### *What about Capital Cost Savings?*

Reducing chilled water flow rate to 70% *could* permit reductions in pipe, fittings, and pump sizes. The resulting capital cost savings may be another reason to design for 42°F vs 45°F CHS. However, reductions in pipe size will diminish the pumping horsepower savings needed to make up for the chiller penalty suffered at lower evaporating temperature. This approach necessitates a life-cycle cost analysis to determine if the energy sacrifice is worth the capital cost savings and so, is beyond the scope of the simpler question addressed in this article.

Of course, everything else being equal, there likely would be some savings for a reduction in pump sizes due to the reduced flow. However, this might be partially offset by a penalty for a larger chiller motor and electrical service.

### *Fouling*

At reduced water flow rates, increased fouling can be an issue in chiller condenser tubes. This is not an issue, however, in the evaporator. Research has shown that evaporator fouling even at low flow rates with "dirty" water is much smaller than that presumed by the current ARI standard for rating chiller performance.

## Note

1. If a three-pass evaporator is provided in the chiller selected for 42°F CHS, there is no flow velocity reduction and hence no heat transfer degradation at 70% flow. Compared with full flow through a two-pass evaporator, two-thirds flow through a three-pass evaporator will experience the same velocity but 1.5 times the pressure drop. The heat transfer penalty in the evaporator is avoided, but at the expense of increased pumping power. Analysis shows that the energy benefit to chiller efficiency approximately offsets the extra pumping energy. ■

Please circle the appropriate number on the Reader Service Card at the back of the publication.

| | |
|---|---|
| Extremely Helpful | 458 |
| Helpful | 459 |
| Somewhat Helpful | 460 |
| Not Helpful | 461 |