3 GPM/TON CONDENSER WATER FLOW RATE-- DOES IT WASTE ENERGY ?

Articles and ASHRAE transaction papers published in the last several years have been unanimous in concluding that the standard rule of thumb for condenser water flow rate through an electric centrifugal chiller of 3 gpm/ton (0.054 mL/J) is too high. These publications have concluded that the optimum condenser water flow rate per ton is, depending on the publication, 2 gpm (0.036 mL/J), 1.5 gpm (0.027 mL/J), or the minimum recommended by the chiller manufacturer. To wit:

1. Waller (ASHRAE Journal, January 1988) recommended a condenser water flow rate of 1.5 gpm/ton for centrifugal chillers to achieve "the lowest life cycle cost with no energy penalty when compared to 3 gpm/ton". He studied a 300 ton (1056 kW), .631 kW/ton chiller initially with approximately a 25 BHP (18.7 kW) condenser water pump at 3 gpm/ton and 41 BHP (31 kW) cooling tower fans before down-sizing them at lower condenser water flow rates. Electricity cost was 8 cents/KWH.

2. Shelton and Weber (1991 ASHRAE Transactions) found that equipment selected for a condenser water flow rate of 2 gpm/ton saved 3.5% in system peak demand and 10.5% in annual energy consumption compared to 3 gpm/ton. They studied a 500 ton chiller (1760 kW) consuming .629 kW/ton at 3 gpm/ton condenser water flow rate that rose to .654 kW/ton at 2 gpm/ton--a 4% increase.

3. Shelton and Joyce (June 1991 ASHRAE Journal) concluded that "the conventional practice of designing chilled water plants with condenser flow rates of 2.8 to 3.0 gpm/ton results in unnecessarily high condenser flow rates". Their study indicated that "the chiller manufacturer's minimum of 1.5 gpm/ton is optimum under a wide range of conditions". Optimality was measured in terms of lowest construction cost and annual energy cost.

4. Kinter-Myer and Emery (April 1995 ASHRAE Journal) recommended reducing water flow through the condenser to the manufacturer's minimum to save pumping costs. (They cautioned that their results were strictly valid only for the hermetic chiller modeled in their study.) Their chiller model was based on formulae used in the *DOE-2.1d* simulation program.

My analysis does not support the generalization that a condenser water flow rate design of 2 or 1.5 gpm/ton will result in the lowest system full load power consumption for chiller, condenser water pump and cooling tower. I conclude that a 3 gpm/ton (0.054 mL/J) condenser water flow rate will most often lead to a smaller **full load power draw** for system components compared to 2 or 1.5 gpm/ton, and, although a lower condenser water flow rate can often result in lower **annual energy cost**, this result is highly dependent on the specific site utility rates, pumping head, and the chiller load profile and so cannot be generalized without specifying these factors.

Framing the Question--

Reducing condenser water flow rate has the effect of reducing the power consumption of the condenser water pump and cooling tower while increasing the power consumption of the chiller.

The question is: Which is greater--the annual energy cost savings for the pump and tower, or the energy cost penalty for the chiller? There is also a question of capital cost savings for down-sizing the system components at the lower condenser water flow rate. I'll briefly address the potential for capital cost savings after focusing on the energy cost question.

Let's look at how the power requirements of each of the three equipment elements change if the condenser flow rate is reduced from 3 to 2 gpm/ton (0.054 to 0.036 mL/J). Later I'll address a 1.5 gpm/ton (0.027 mL/J) condenser water flow rate.

The Chiller

Lowering the flow rate across the condenser of a chiller increases the temperature rise of the water flowing through it. At 3 gpm/ton (0.054 mL/J) through the condenser, the condenser water temperature rise, "ΔT", for a .60 kW/ton hermetic chiller (C.O.P = 5.86) is 9.4°F (5.2°C) calculated like so:

$$\Delta T\ (°F) = \frac{12{,}000\ \text{Btuh/ton} + [\ .6\ \text{kW/ton x } 3413\ \text{Btuh/kW}\ ]}{500 \text{ x } 3\ \text{gpm/ton}} = 9.36°F$$

(SI)

$$\Delta T\ (°C) = \frac{1 + 1/\ \text{C.O.P.}}{4.187\ \text{J/g-°C x } .054\ \text{mL/J x } 1.0\ \text{g/mL}} = 5.18°C$$

With 85°F (29.4°C) entering condenser water, leaving water temperature, then, is 94.4°F (34.6°C). At 2 gpm/ton, via the same formula, condenser water ΔT increases to 14.05°F (7.81°C). If, in addition, we account for the fact that chiller kw/ton will increase slightly due to the warmer leaving condenser water, say by 10%, then ΔT = 14.25°F (7.92°C) so that leaving condenser water temperature increases almost 5.0°F (2.8°C) to 99.3°F (37.4°C).

Reduced flow rate through the condenser along with the resulting higher leaving condenser water temperature effects condenser heat transfer in two ways.

1. The film heat transfer coefficient "h_w" on the water side of the condenser tubes is reduced.

2. The mean temperature difference between the condensing refrigerant and the condenser water (shown diagramatically in **Figure 1a**) is reduced, or would be, if the condensing temperature did not adjust upward. The mean temperature difference is calculated via the log mean temperature difference, or "LMTD".

If the chiller is to handle the same cooling load at 2 gpm/ton condenser water flow rate as at 3 gpm/ton, then the refrigerant condensing temperature must adjust upward to restore the LMTD driving potential and compensate for the reduced heat transfer coefficient across the condenser tubes as shown in **Figure 1b**. In terms of formulae--

$$Q = U\ A\ LMTD$$

$$Q = U_{3gpm}\ A\ LMTD_{3gpm} = U_{2gpm}\ A\ LMTD_{2gpm}$$

so $$U_{3gpm}\ LMTD_{3gpm} = U_{2gpm}\ LMTD_{2gpm}$$

Temperature Difference Sketches for 2 & 3 GPM/Ton

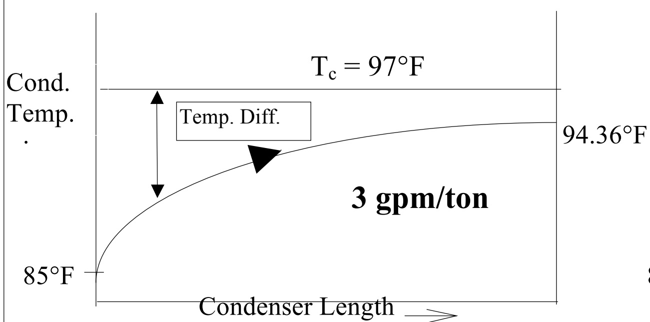

Figure 1a
Approach conditions of condenser water to refrigerant condensing temperature at 3 gpm/ton condenser water flow rate. $T_c = 97°F$ is typical for the chiller under consideration.

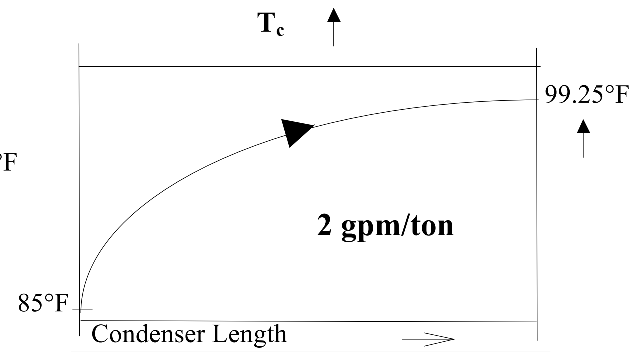

Figure 1b
Condensing temperature T_c must rise at 2 gpm/ton condenser water flow rate to restore LMTD.

$$LMTD_{3\ gpm/ton} = \frac{(97 - 85)\ -\ (97 - 99.25)}{\ln[(97 - 85) / (97 - 99.25)]} = 6.18°F$$

$$LMTD_{2\ gpm/ton} = \frac{(T_c - 85)\ -\ (T_c - 99.25)}{\ln[(T_c - 85) / (T_c - 99.25)]}$$

A rise in the chiller's condensing temperature and hence condensing pressure requires the chiller's compressor to work harder to produce slightly less refrigerating effect per pound of refrigerant circulated. Thus, efficiency, and to some extent chiller capacity, is reduced.

By solving for the change in condensing temperature, "T_c", for a reduction in condenser water flow rate, the theoretical decrease in chiller efficiency can be determined thereby relating decreased condenser water flow rate to increased chiller energy consumption. The equation below Figure 1b permits one to solve for "T_c" once $LMTD_{2\ gpm/ton}$ is known. From above--

$$LMTD_{2gpm} = (U_{3gpm} / U_{2gpm})LMTD_{3gpm}$$

-- and from Figure 1a, $LMTD_{3gpm} = 6.18°F$ (3.43°C). "U" in the equation above is the inverse of "R_t", the total resistance to heat transfer across condenser water tubes. "R_t" consists of four resistances in series: the water film resistance "$1/h_w$", the waterside fouling factor "R_f", the refrigerant side film resistance "$1/h_f$", and the resistance of the copper tubes and fins. In formula form, based on the inside surface area of the tubes and omitting the relatively negligible resistance of the tubes and fins:

$$R_t = 1/h_w + R_f + \frac{A_i}{A_o\, n_f}\, 1/h_f$$

where "n_f" represents an effective area multiplier for refrigerant side fins.

The water film resistance is the only term which changes if the condenser water flow rate is reduced. The water film heat transfer coefficient "h_w" is proportional to Reynolds number to the 0.8 power. At two-thirds the water flow, velocity and consequently Reynolds number is reduced by two-thirds so that--

$$(h_w)_{2\ gpm/ton} = (2/3)^{0.8}\ (h_w)_{3\ gpm/ton} = .72\ (h_w)_{3\ gpm/ton}$$

Therefore the resistance of the water film to heat transfer increases to 1/.72, or 1.38 of its former value. The **table below** shows that the water film represents, for a typical set of chiller conditions, around 31% of the total heat transfer resistance across the condenser tubes.

Table 2.
Ballpark Estimates for
Resistance to Heat Transfer Across Condenser Water Tubes
at ARI Conditions

	Formula Term	Typical "R" Value $h \cdot ft^2 \cdot °F/$ Btu $(m^2 \cdot °C/kW)$	% Total
Water Film	$1/h_w$	.00031 (0.055)	31%
Fouling Factor	R_f	.00025 (0.044)	25%
Refrigerant film	$(A_i/A_o n_f)1/h_f$	.00045 (0.079)	45%
Total		.00100 (0.177)	

Therefore, increasing "$1/h_w$" by 38% has the effect of increasing the total resistance across the condenser tubes by .38 x .31 = 12%. Put another way: $R_{2gpm}/R_{3gpm} = 112\%$

From before, then:

$$LMTD_{2gpm} = \frac{U_{3gpm}}{U_{2gpm}} LMTD_{3gpm} = \frac{R_{2gpm}}{R_{3gpm}} LMTD_{3gpm} = 1.12\ (6.18°F) = 6.92°F$$

Knowing the value $LMTD_{2\ gpm}$, we solve for the condensing temperature "T_c" at 2 gpm/ton condenser water flow using the condensing water temperatures shown in Figure 1b:

$$LMTD_{2\ gpm} = 6.92°F = \frac{(T_c - 85) - (T_c - 99.25)}{\ln[(T_c - 85) / (T_c - 99.25)]}$$

$$\therefore \quad T_c = 101.33°F\ (38.52°C)$$

Thus, the condensing temperature at 2 gpm/ton must rise 4.33°F (2.41°C) --from 97.0°F to 101.33°F-- in order to motivate the same heat transfer per unit area in the condenser as would be the case at 3 gpm/ton condenser water flow.

Knowing the rise in condensing 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 the simple vapor compression cycles for both a single stage and two stage chiller in **Figures 2a and 2b** below.

Simple Cycle
Pressure-Enthalpy Diagrams

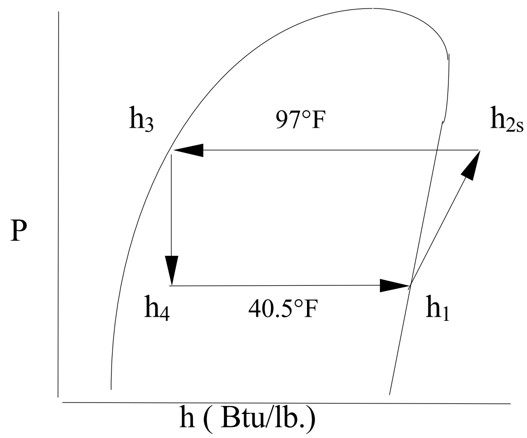

Figure 2a
Single Stage Chiller

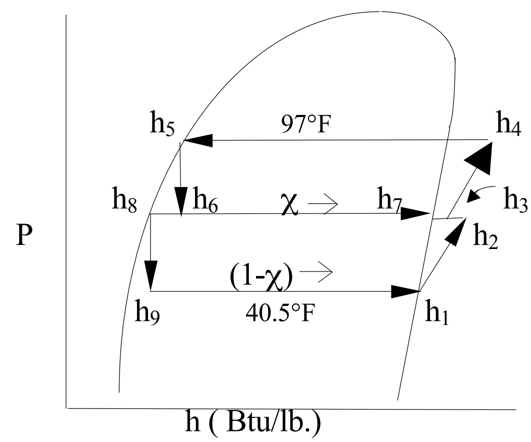

Figure 2b
Two Stage Chiller

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

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

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

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.

Table 3 below tabulates the theoretical percentage change in isentropic power consumption based on refrigerant enthalpy tables for a 4.33°F (2.41 °C) rise in condensing temperature. Several refrigerants and the multi-stage compressors in common use are represented.

Table 3
Isentropic Change in kW/ton for 4.33°F (2.41 °C) Increase in Condensing Temperature

Refrigerant	#Compressor stages	$(KW/ton)_s$ @ Cond. Temperature = 97°F (36.1°C)	101.33°F (38.52°C)	%(KW/ton) Increase
R134a	Single Stage	.469	.514	9.6%
R22	"	.469	.512	9.2%
R123	"	.442	.480	8.8%
R123	Two Stage	.418	.453	8.2%
R123	Three Stage	.412	.445	8.0%
			Average	8.8%

*Evaporating temperature is 40.5°F (4.72°C), no sub-cooling, or superheat.

From Table 3 above, the isentropic power penalty per ton of cooling for a 4.33°F (2.41 °C) increase in condensing temperature ranges from **8.0 to 9.6%--**depending on the chiller design. *That's roughly a 2% change in kw/ton per °F change in condensing temperature*. Single stage machines suffer a little higher penalty, multi-stage machines suffer a little less[1].

There are three other factors which affect chiller efficiency -- the compressor adiabatic efficiency, "η_c", the motor efficiency, "η_{mot}", and the drive train efficiency "η_{mech}", so that kW/ton could be expressed as:

$$kW/ton = \frac{(kW/ton)_s}{\eta_c \, \eta_{mot} \, \eta_{mech}}$$

"η_{mech} " and "η_{mot} " should not generally change for a 4.33°F change in condensing temperature.
"η_c "-- the compressor efficiency -- will generally vary. Which direction it varies depends on where the actual operating points happen to fall on the selections' compressor map (or maps). Compressor efficiency could improve, degrade, or remain the same. I would argue that more often compressor efficiency will tend to degrade if for no other reason that as the condensing temperature/pressure rises, the refrigeration effect per pound of refrigerant decreases. (See Figure 4.) Hence for the same tonnage, more pounds of refrigerant must be circulated at a 2 gpm/ton condenser water flow rate thereby increasing refrigerant flow losses through the refrigerant system. I'm told, however, by Bill Landman, Manager of Applications Engineering Group at the Trane Company, that this is a small, if not negligible, effect.

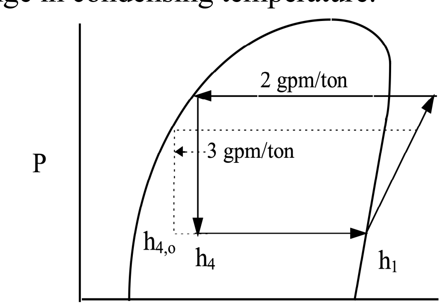

Figure 4
Cooling effect per # refrigerant is greater @ 3 gpm/ton (= h_1 - $h_{4,o}$) than at @ 2 gpm /ton (= h_1 - h_4)

[1]Strictly speaking, % kW/ton figures in Table 3 are only valid for the evaporating and condensing temperatures cited in Table 3 although variation is small within the range of typical evaporating and condensing temperatures for water cooled chillers used for HVAC.

The upshot is that for an actual chiller selection, a change in compressor efficiency could increase or diminish the power penalty for a 4.33°F (2.41 °C) rise in condensing temperature compared to that shown in Table 3. To check the degree of variation, data from chiller selections made by two manufacturers are presented below. The manufacturers were asked to make their most economical selection to achieve a power input of approximately .60 kW/ton for a specific tonnage at ARI conditions. Then make a second selection in which condenser flow was decreased from the ARI condition of 3 gpm/ton (0.054 mL/J) to 2 gpm/ton (0.036 mL/J) while holding shells constant, but allowing the compressor impeller and speed to vary. Table 4 demonstrates the resulting % change in kW/ton for the selections.

Table 4: Manufacture's Data
KW/ton Increase at 2 and 3 gpm/ton CW Flow

Chiller Type	Tons	Refrig	Evap. Temp°F	Cond. Temp. (°F) 3 gpm/tn	Cond. Temp. (°F) 2 gpm/tn	kW/ton @ 3 gpm/tn	kW/ton @ 2 gpm/tn	% Increase kW/ton
Single Stage	400	R123	40.7	96.7	101.1	.618	.683	**+ 10.5%**
"	500	"	40.8	96.6	100.9	.606	.667	**+ 10.1%**
Two Stage	600	R123	37.8	97.4	101.9	.582	.637	**+ 9.5%**
Three Stage	500	"	37.6	99.3	103.8	.596	.644	**+ 8.1%**
"	500	"	40.3	96.4	100.8	.582	.618	**+ 6.2%**

Other than the last selection, the increases in kW/ton predicted by the manufacture's selection programs exceed the theoretical estimate of 8.0 to 9.6 % tabulated in Table 3 for an increase in condensing temperature due to the same reduction in condenser water flow. Therefore, I adjudge the theoretical estimate to be, in general, reasonable and probably conservative more often than not.

The Pump

Assume the condenser water pump is a base mounted centrifugal pump serving a close coupled cooling tower and chiller. Assume pump head at 3 gpm/ton (0.054 mL/J) is 50' (150 kPa) of which 10' (30 kPa) is devoted to lift at the cooling tower. I can select an 82% efficient 6 x 6 x 11 pump with a 92% efficient motor running at 1750 rpm so that pumping power per *full load* ton is:

$$\frac{3 \text{ gpm/ton} \times 50'}{3960 \times .82} \times \frac{.746 \text{ kw/hp}}{.92} = .0375 \text{ kW/ton}_f$$

(SI)

$$\frac{.054 \text{ mL/J} \times 150 \text{ kPa} \times .001 \text{ m}^3\text{/L}}{.82 \times .92} = 0.0107 \text{ kW/ kW}_{chiller,f}$$

If the condenser water flow rate is reduced to 2 gpm/ton (0.036 mL/J), the frictional portion of the pumping head falls by approximately the 1.85 power (per the Hazen-Williams correlation), and I can make a pump selection at 84% efficiency. (Actually, it's the same 6 x 6 x 11 pump run at 1,150 rpm in stead of 1,750 rpm.) Then pumping horsepower per full load ton is calculated like so:

$$\frac{2\ \text{gpm/ton} \times [(2/3)^{1.85} \times 40' + 10']}{3960 \times .84} \times \frac{.746\ \text{kw/hp}}{.92} = .0143\ \text{kw/ton}$$

(SI) $$\frac{.036\ \text{mL/J} \times [(2/3)^{1.85} \times 120 + 30']\ \text{kPa} \times .001\ \text{m}^3/\text{L}}{.82 \times .92} = 0.0040\ \text{kW/ kW}_{\text{chiller,f}}$$

The difference in pumping power is **.023 kW/ton$_f$** (0.0067 kW/ kW$_{\text{chiller,f}}$). For a .60 kW/ton chiller (1/C.O.P. = .1706), this amounts to a savings of **3.9%** of the full load chiller power for the lower condenser water flow rate.

The Cooling Tower

Assume a crossflow cooling tower with draw through fan. Reducing the chiller's condenser water flow rate increases the condenser water ΔT and hence the temperature of the entering tower water. With warmer entering water, the tower experiences a larger temperature difference, or more precisely, enthalpy difference, between entering air and water. The increased enthalpy difference, "Δh", between air and water provides more driving potential for heat transfer in the cooling tower. The ratio of enthalpy differences for a 2 versus 3 gpm/ton (0.036 versus 0.054 mL/J) flow rate, assuming a 78° WB (25.6 °C) design condition and the same leaving water temperatures computed earlier, is:

$$(h_{w,99.25} - h_{a,78wb}) / (h_{w,94.36} - h_{a,78wb}) = 26.4\ \text{Btu/\#} / 21.4\ \text{Btu/\#} = 123\%$$

Thus. at 2 gpm/ton compared with a tower selected for 3 gpm/ton, there's 23% more "driving potential" available to the tower to transfer heat. The designer can take advantage of the increased heat transfer driving potential at 2 gpm/ton (0.036 mL/J) by either:

1. Decreasing tower size to save first cost,
2. Lowering the CFM drawn through the same size tower, thereby lowering the tower fan break horsepower requirement, or
3. Achieving a closer approach to design wet bulb temperature by leaving tower size and motor horsepower unchanged.

Decreasing tower size can save about one case sizing. Most of the potential energy cost saving for reducing the condenser flow rate is forfeited, however.

The benefit of decreasing tower airflow is its impact on tower fan horsepower. According to the tower model solved in Shelton and Weber's 1991 paper, reducing the water flow rate through a tower from 3 to 2 gpm/ton allows the tower to achieve the same leaving water approach at an air flow rate of 85% of the 3 gpm/ton design value[2]. This figure is in good agreement with several

[2] Lowering the air flow through the tower causes the remaining air to approach the enthalpy of the hot water more rapidly thus reducing the mean enthalpy difference Δh, and hence heat transfer "driving potential" between air and water. The degree of air flow reduction is such that it approximately offsets the increase in mean enthalpy difference gained by the reduction in water flow rate. The film heat transfer coefficient "h" between air and the film of water flowing on the tower fill is also reduced due to reduced air velocity. Because h is proportional to Reynolds Number to the 0.5 or 0.6 power, this is a relative weak effect compared to the effect of the reduction in air mass flow. *(Omit footnote)*

selections I checked via a manufacture's computerized tower selection program. Cooling tower fan break horsepower varies as roughly the cube of CFM drawn through the tower. A reduction in water loading also slightly reduces fill static pressure and hence fan horsepower. Shelton and Weber report a correlation of fan break horsepower to the air flow rate raised to the 3.2 power. Baltimore Aircoil Company Engineers use a 3.175 power as a rule of thumb, according to Richard Harrison, Chief Engineer at BAC. At 85% airflow, then, tower fan break horsepower falls to $(.85)^{3.2}$, or approximately 60%, of its former value. That's a savings of 40% of fan power. For a typical cooling tower fan motor selected at .05 HP/ton_f of cooling (0.011 kW/ $kW_{chiller,f}$), a 40% savings would amount to **.0166 kW/ton_f** (0.004 kW/$kW_{chiller,f}$) assuming a 90% efficient electric motor. This amounts to **2.8%** of the full load chiller power of .60 kW/ton_f.

A better cooling tower selection in terms of life cycle cost in most southern states is to pick an oversized tower at .025 HP/ton (0.005 kW/ $kW_{chiller,f}$). At .025 HP/ton, savings for reduction from 3 to 2 gpm/ton would be half that calculated above.

The third option cited above is to leave the tower size and motor horsepower unchanged to achieve a lower approach. According to the *Marley Cooling Tower Selection Program (version 94.11c)*, this results in about a 1.7°F (0.9°C) improvement in approach to the design wet bulb temperature at full load. The improved approach allows both entering and leaving water temperature at the chiller condenser to drop by 1.7°F (0.9°C) causing the chiller's condensing temperature to fall by virtually the same amount. Based on the average penalty per °F calculated previously, a reduction of 1.7°F condensing temperature would result in approximately a **3.4%** decrease in full load chiller power.[3] This is a greater power savings than can be attained by reducing the fan horsepower of the cooling tower. At part load, however, the improvement in approach diminishes. At 50% part load, for example, approach improves only 0.87°F (0.5°C) for one selection checked. In fact, the improvement in cooling tower approach varies approximately linearly in proportion to chiller part load. Since we have expressed power savings in term of percent of *full load chiller power*, and full load chiller power itself varies approximately linearly with part load, the per cent chiller power reduction due to improved cooling tower approach is roughly proportional to the square of chiller part load, i.e.--
:

$$\text{\% Cooling Tower Power Reduction for 2 vs. 3 gpm/ton} \sim (\text{P.L.})^2 \times 3.4\%.$$

(*as % of chiller full load power*)

So, comparing the two cooling tower strategies which save energy for 2 vs. 3 gpm/ton (0.036 vs. 0.054 mL/J) design--

1. Leaving tower motor horsepower unchanged in order to attain a lowered approach saves more power at chiller full load, but power savings falls off rapidly (as a square) with chiller part load.

2. Reducing cooling tower fan motor horsepower saves less power at chiller full load but does not diminish at chiller part loads. For the numbers developed above for the per cent savings for the

[3]If cooling tower and chiller were considered together-- instead of the chiller suffering a 4.33°F rise in condensing temperature for 2 versus 3 gpm/ton operation, the condensing temperature rise would be rolled back to 2.63°F .

two strategies, below 89% part load, the power savings for reducing fan horsepower exceeds that for reducing cooling tower approach.[4]

Recapping Penalty and Savings for the Three Components.

If a chilled water system is designed for 2 gpm/ton (0.036 mL/J) condenser water flow instead of the customary 3 gpm/ton (0.054 mL/J):

1. A .60 kw/ton (C.O.P. = 5.86) chiller selection (with compressor impeller and speed allowed to vary) will increase power consumption typically from **8.0 to 9.6%** at full load depending on chiller design. At part load, the power penalty will fall off in approximately linear proportion to the load.
2. The condenser water pump's power consumption will be reduced by an amount equivalent to **3.9%** of chiller full load power. (This assumes 50' (150 kPa) of pumping head. If pumping head exceeds 50', savings will be proportionally greater). Condenser water pump savings is constant at all operating part loads assuming no variable speed operation.
3. Cooling tower savings can be, depending on the strategy adoped, either **3.4%** of full load chiller power falling off rapidly at part load, or **2.8%** (for a .05 hp/ton selection at 3 gpm/ton) at all part loads at which the fan continues to run and assuming no variable speed operation.

Figures 5a and 5b show the above relationships as a function of chiller part load.

Conclusions

Comparing a 2 gpm/ton (0.036 mL/J) condenser water design with a 3 gpm/ton (0.054 mL/J) design for a typical new chiller water system:

1. *At full load*, the chiller kW penalty will generally exceed savings for the pump and cooling tower if the pump, chiller, and cooling tower are close coupled.

2. *At decreasing part load*, the magnitude of the energy savings for the condenser water pump and cooling tower will catch up with and outstrip the chiller penalty. At 50 % part load, for example, the chiller kW penalty will have fallen to approximately 4.0 to 4.8%, while pump and cooling tower savings, if the cooling tower design strategy is followed that reduces fan horsepower, will remain constant at 6.66%. (If the cooling tower had been selected at .025 hp/ton (0.019 kW/ton$_f$) at the 3 gpm/ton design condition, combined pump and tower savings would equal 5.3% which would generally still exceed the chiller penalty, but not by much.)

So, the question as to which condenser water flow rate results in the least total energy cost resolves itself into a second question: Which condition has the predominant impact on the annual utility cost--full load power consumption or part load energy consumption?" The answer depends primarily on two factors: the structure of the electric rate schedule, and, to a lesser extent, the load profile which the chiller will serve.

If, as is the case in the South, peak electric demand charges make up a significant portion of annual

[4] % Cooling Tower Power Reduction for 2 vs. 3 gpm/ton ~ $(.894)^2$ x 3.45% = 2.76%.

electric cost and they occur coincident with chiller full load output, then minimizing power consumption at full load and high part loads is generally paramount, i.e. a 3 gpm/ton (0.054 mL/J) design would be favored.

If, on the other hand, electric charges are fairly uniform at all chiller part loads, then the distribution of ton-hours produced and hours operated at various part loads must be examined to see if the accumulated pump and tower energy savings will out strip the accumulated chiller energy penalty.

Either way, it appears that in most cases the net energy advantage is small since the accumulated penalty and savings at high and low part loads tend to offset one another.

<u>What about design at even lower condenser water flow rates?</u>

At 1.5 gpm/ton (0.027 mL/J), I calculate that the chiller's condensing temperature would rise to 105.9°F (41.1°C) causing its isentropic kW/ton at full load to increase by 17 to 20% (relative to a chiller operating at 41.5°F (5.1°C) evaporating and 97°F (36.1°C) condensing temperature). The savings for the pump and cooling tower would be about 9.0 % at all part loads assuming the pump is close coupled and the cooling tower fan break horsepower is reduced to save energy. Thus, the chiller penalty continues to increase linearly at about 2% per degree rise in condensing temperature while the savings for pump and cooling tower fan horsepower approaches an asymptotic limit.

<u>What about Capital Cost Savings?</u>

Reducing condenser water flow rate from 3 gpm/ton to 2 gpm/ton should permit reductions in pipe size, pump size, and cooling tower size, right? No, I don't think so. Not if the goal is to achieve at least equal or better energy efficiency for each component. Let me address the likely cost impact on each equipment component individually for a reduction in condenser water flow rate from 3 to 2 gpm/ton.

1. Installed chiller cost will generally rise at least slightly. The chiller's motor size will generally increase and the chiller selection could jump a compressor frame size although this is the exceptional case. Chiller electrical installation cost will probably increase slightly due to increased motor amp draw which will typically cause an increase of one wire size in conductors. The electrical cost increase could be more if a circuit breaker frame size is exceeded.
2. If, as I have assumed in this analysis, the goal is to save energy, then the cooling tower casing size cannot be decreased at reduced flow. To do so would sacrifice most of the energy savings needed to make up for the chiller penalty at reduced flow. The cooling tower motor size can be reduced, but this is a minor savings.
3. Condenser water piping cannot be reduced in size without sacrificing a portion of the energy saving claimed for the condenser water pump.
4. One would think that the condenser water pump could be reduced in size without penalty, but that's not necessarily true either. A centrifugal pump selection equal in efficiency to that assumed -- 82% efficient for a 1500 gpm (26.9 mL/s) pump @ 50' head (150 kPa) -- cannot be made for a 1000 gpm (17.9 mL/s) selection at a motor rpm of 1750. Pump efficiency drops to

72.5%. The reason is that the pump's *specific speed*[5] starts to approach the limit for a good centrifugal pump selection. Lowering the motor rpm to a nominal 1150 rpm design, however, puts the pump selection back in the sweet spot and allows a pump selection at 84% efficiency. It is not a coincidence, but rather a consequence of the pump laws, that the optimum pump selection at 2 gpm/ton is the same pump optimally chosen for 3 gpm/ton operation, but driven by a motor spinning at two-thirds the nominal rpm. So, if pump efficiency is to be maintained, the same pump body would be selected in this example and there would be no first cost savings for the pump.

From the above reasoning I would conclude that there is no significant capital cost savings for reducing condenser water flow rate from 3 to 2 gpm/ton *if* the designer's goal is to retain at least the same overall system energy efficiency as in the 3 gpm/ton design.

Note that this conclusion does not speak to the proposition put forward in two of the publications cited at the beginning of this article that a judicious *sacrifice* of energy efficiency by some system components may be justified by first cost savings which improve overall system life cycle cost. This is a tougher generalization to address because it's so sensitive to local utility rates and load profile.

Fouling

A consideration not discussed in this analysis so far is the effect of reduced flow rate on fouling. Note in Table 2 that condenser water side fouling at a .00025 fouling factor accounts for around 25% of the total resistance to heat transfer across condenser tubes. Thus, an increase in fouling factor would substantially impact the condensing temperature and subsequent energy consumption of a chiller. Manufacturer's recommend that condenser water tube velocity be maintained above 3.3 fps (1 m/s) to avoid fouling. There is no research of which I'm aware that correlates fouling with water velocity, but it stands to reason that reducing flow rate *might* result in a greater degree of fouling. If, for example_fouling factor rose from .00025 to .0005 (0.044 to 0.088 $m^2\cdot°C$ /kW) as a result of lowering the condenser water flow rate from 3 to 2 gpm/ton, the isentropic chiller power penalties listed in Table 3 would increase by about 28% from an average of 8.9% to 11.4%. More research is needed on this issue to know if increased fouling will result from reduced condenser water flow rates.

Recommendations

In deciding what condenser water flow rate to design for, here's what I would do.

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[5] Specific speed is a measure of the suitability of an impeller design to serve a set of pump conditions. It's defined by the equation below:

$$n_s = \frac{rpm\ (GPM)^{.5}}{H^{.75}}$$

For a centrifugal pump, the specific speed should fall between 650 and 5000. At 3 gpm/ton, 50' head, 500 tons and 1750 rpm. n_s = 3,604. At 2 gpm/ton, it was previously calculated that pump head would drop from 50' to 29'.. Thus, at 1750 rpm, n_s = 4428. At 1150 rpm, n_s = 2910.

1. Have the chiller and cooling tower manufactures make alternate selections at 3 and 2 gpm/ton (0.054 and 0.036 mL/J) to accurately gauge the chiller penalty and cooling tower advantage. Estimate the pump head and resulting pump horsepower for each flow rate.
2. Mentally at least, construct a chart like that in Figure 5 and solve for the break-even part load.
3. Estimate the mean part load at which the chiller's annual ton-hours will be produced. Ask yourself-- Is it likely to be above or below the break-even part load solved for in 2. above?
4. Factor in the effect of the electric rate schedule. High demand costs incurred when the chiller is operating at full load favor 3 gpm/ton. Uniform electric rates favor lower condenser water flow.

If your evaluation shows it's about a wash for the competing designs, go with the higher condenser water flow rate to minimize the risk of increasing condenser tube water side fouling.

SYMBOL KEY

A	Heat transfer area
A_i	Inside heat transfer area
A_o	Outside heat transfer area
C.O.P	Coefficient of Performance
h	enthalpy Btu/# (J/ kg·K)
h_1	enthalpy at 1.
h_{2s}	enthalpy at 2 for isentropic increase
$h_{a,78wb}$	enthalpy of air at 78 wet bulb
h_f	enthalpy of saturated water
h_w	enthalpy of water
$h_{w,99.25}$	" at 99.25°F
$kW_{chiller,f}$	SI unit of chiller refrigeration capacity at full load
LMTD	log mean temperature difference
$LMTD_{3gpm}$	LMTD at 3 gpm/ton condenser water flow
n_f	heat transfer area multiplier for fins
P	Pressure
P.L.	part load
Q	Heat Transfer, tons of refrigeration
R_f	Refrigerant resistance to heat transfer
R_t	Total Resistance to heat transfer
T_c	Condensing temperature, °F (°C)
ton_f	full load chiller tons
U	Overall heat transfer coefficient
U_{3gpm}	U for 3 gpm/ton condenser water flow
ΔT	Temperature difference, °F(°C)
χ	Refrigerant quality, i.e. weight fraction which is saturated vapor

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BRIEF VITA

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