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update

Reduced Condenser-Water Flow Rate: Energy-Saving Miracle or Mirage?

The conventional design of air-conditioning plants using large-tonnage chillers calls for a condenser-water flow rate of 3 gallons-per-minute-per-ton-of-refrigeration (GPM/TR). For years, this standard has been thought to offer a good balance between first cost and energy cost.

Recently, this conventional design has been challenged by proponents of a 2 GPM/TR design. These proponents suggest that reducing condenser-water flow rate to 2 GPM/TR will save both first cost and energy cost. They posit that the cooling tower, condenser-water pump and piping can all be downsized, thus reducing first cost. Furthermore, although the lower flow rate will penalize energy performance of the chiller, the proponents suggest that the energy savings on the tower and pump will be greater than the energy penalty on the chiller — and that these savings are available in most applications.

The legitimacy of these claims was recently scrutinized by Wayne Kirsner, P.E., an independent consulting engineer based in Atlanta, Ga. He conducted an impartial analysis of the impact of 3 versus 2 GPM/TR condenser-water flow rate, which was published in the February 1996 issue of the ASHRAE Journal. His analysis forms the basis upon which this discussion is founded.

Fallacies of First-Cost Savings

The 2 GPM/TR proponents imply that a lower flow rate simultaneously offers first-cost and energy-cost savings. We'll examine those claims separately, starting with first cost.

The first-cost claim is this: lowering the condenser-water flow rate will allow cost-effective reductions in the condenser-water piping, condenser-water pump, and cooling tower. But is this assumption generally correct? Relying on Kirsner's analysis, the following examination shows the answer is "no." That's because either smaller equipment can't be specified, or where it can, the first-cost savings are too small to offset greater energy usage.

Piping First-Cost Savings?

In many cases, there can be no size reduction in condenser-water piping, because piping is only available in specific sizes. Often the same diameter pipe used for a 3 GPM/TR system must be used for 2 GPM/TR.

For example, we can refer to the 500-TR chiller used in the example offered by some 2 GPM/TR proponents. At 3 GPM/TR, the water flow is 1500 GPM. With pipe sized according to the ASHRAE guideline of 1 to 4 feet-of-pressure-drop-per-100-feet-of-pipe, this would require 8-inch pipe. At 2 GPM/ton, water flow is 1000 GPM. Can 6" pipe be used? No, because friction loss would exceed the ASHRAE-recommended upper limit of 4 feet, as shown in Table 1. So 8-inch pipe must still be used. Thus, in this and many other instances, there can be no pipe-size reduction and no first-cost savings.

Table 1: Pipe Sizing

GPM	6"	8"	10"
1500	13.57'	**3.37'**	1.07'
1000	6.19'	**1.56'**	.50'

Even if the pipe size could be reduced (imagine that pipe components came in an infinite range of sizes), it is important to realize that the increase in frictional pressure drop caused by smaller piping will negate 30 to 60% of the potential pump energy-savings contribution, depending upon the percentage that the piping-system friction contributes to the total pump head — the greater the magni-

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tude of the piping WPD, the greater the potential energy-savings contribution lost by downsizing the pipe.

Tower First-Cost Savings?

In most cases, maximum downsizing of the cooling tower eliminates all of the tower energy savings from a 2 GPM/TR design. It is true that reducing condenser-water flow rate will improve a tower's thermal efficiency, because the entering water will be warmer and will increase its thermal "driving potential." This will allow the airflow on the original tower to be reduced 12 to 15% to do the same load and approach, and result in 30 to 40% less fan energy for the same duty. However, if the tower is downsized to save first cost, the increased thermal "driving potential" will be offset by the reduction in heat-transfer surface, and the airflow must be increased through a smaller face area. The increased air-pressure drop will eliminate the energy savings gained by a 2 GPM/TR design.

Pump First-Cost Savings?

Theoretically, a condenser-water pump handling 33% less flow could be reduced in size to provide first-cost savings without running into an energy-cost penalty. But the theory usually doesn't hold up in practice.

For example, a pump handling a condenser-water loop with 1500 GPM (3 GPM/TR for a 500-TR chiller) at 45' of pressure drop would be selected at 1750 RPM and 81% pump efficiency. (All selections are based on Aurora pumps).

At 1000 GPM (2 GPM/TR) with the same size pipe and condenser-pass arrangement, the head falls to 26.5', and the least expensive selection would be a smaller 1750 RPM pump, resulting in a first-cost savings of about $400. However, the pump efficiency would fall to 70%, resulting in a net energy-cost increase of $600/year (assuming $.08/kWh and 5000 hours annual operation).

The best pump to handle 2 GPM/TR would be the same pump used to handle 3 GPM/TR, but selected at 1150 RPM. That's because it would perform at 84% efficiency, paying back the added first-cost of $400 (compared to the smaller, less-efficient pump) in less than a year.

This performance phenomenon is not isolated to this case. Pump efficiency is a function of the specific speed of centrifugal impellers, and lower heads usually result in lower efficiencies. Thus, there is no first-cost benefit to switching to a smaller pump. (In fact, in this example, pump cost would rise by $100 because of the more expensive 1150 RPM motor.)

Chiller First-Cost Savings?

The first-cost of the chiller does not decrease with a 2 GPM/TR design. In fact, it generally increases. A larger motor, starter, and, in some cases, a larger compressor are required. Higher amp draw of a larger motor may also increase electrical installation costs by requiring larger wiring, breakers, and other components.

Even if upsizing is limited to only the motor and starter, it could offset 30% of the first-cost savings available from downsizing the piping, tower and pump.

First-Cost Conclusions

You cannot simultaneously reduce first costs and energy costs with a 2 GPM/TR design. If the goal is to maximize first-cost savings, then 70 to 83% of the energy-cost savings from the tower and pump will be lost. And without the energy-cost savings to offset the chiller penalty, 2 GPM/TR does not normally make good economic sense. Therefore, if the goal is to maximize energy savings, the piping, pump, and cooling tower cannot be downsized, and first-cost savings will be lost.

The next section of the analysis will assume that the goal is to maximize energy-cost savings, so no first-cost opportunities were assumed.

Fallacies of Energy-Cost Savings

Proponents of 2 GPM/TR claim that while reducing water flow increases chiller energy consumption, there will be compensating reductions in tower and pump energy. Tower and pump savings supposedly exceed the chiller penalty, resulting in net chiller-plant energy savings. But is this premise true? The following real-world analysis shows the answer is most often "no". First, we'll examine the impact on energy at design conditions, then we'll follow with an analysis at off-design conditions.

We'll start by citing the example offered by some proponents of 2 GPM/TR. Then, we'll do an in-depth analysis of that example using both Kirsner's study and our own research.

Proponents' Example at Design

The example begins with a midsize office building having a 500-TR cooling load. The chiller is a 500-TR, three-stage, direct-drive chiller with a design leaving-chilled-water temperature of 44°F. A 500-TR tower supplies a design entering-condenser-water temperature of 85°F to the chiller. The condenser-water pump handles 1500 GPM at 3 GPM/TR, and 1000 GPM at 2 GPM/TR.

A comparison of the energy performance is shown in Table 2. With 2 GPM/TR, the chiller performance is penalized by 17 kW. However, the tower and pump performances save 19 kW combined, resulting in a system savings of 2 kW.

We believe this analysis is flawed. The chiller selection is misleading, making its energy penalty artificially low, while the energy consumption of the 3 GPM/TR pump and tower are higher than current practice for most building designs, making their savings artificially high. To provide a sound basis for analysis of this example, we'll start with Kirsner's work.

Energy Analysis of Pump at Design

Condenser-water pumps are typically sized for approximately 45 feet of head, or .05 HP/TR. The proponents' example uses 26 kW, which equates to .07 HP/TR. Why so high?

Probably because the higher the pump energy is at 3 GPM/TR, the greater the savings at 2 GPM/TR. Let's use Kirsner's more impartial example.

Since the proponents' example did not specify all the design details, Kirsner makes some assumptions. He assumes the condenser-water pump is a base-mounted centrifugal pump serving a close-coupled cooling tower and chiller (that is, a tower within 100 feet of the chiller and at the same elevation).

The chiller condenser is a 2-pass design. The pump's head requirement is assumed to be 45', consisting of 10' static head, 15' for the condenser, and 20' for the pipe, strainers, valves, etc.

To serve a 500-TR chiller with 3 GPM/TR, he chose an 81% efficient, 6 x 6 x 11 pump with a 92% efficient motor running at 1750 RPM. The pumping power per full-load TR is .034 kW/TR (or .046 HP/TR), as calculated below (the energy calculations are made on a kW/TR basis so as to be independent of system capacity):

$$\frac{\text{3 GPM/TR x 45'}}{\text{3960 x .81 pump eff.}} \times \frac{\text{.746 kW/HP}}{\text{.92 motor eff.}} = \text{.034 kW/TR}$$

If the condenser-water flow rate is reduced to 2 GPM/TR, and all pipe components remain unchanged, the frictional portion of the pumping head (35') falls by approximately the square, and the tower static head (10') remains the same. Thus, by selecting an 84% efficient pump (the same pump running at 1150 instead of 1750 RPM, because a smaller pump running at 1750 RPM has an efficiency of only 70%, as explained earlier), Kirsner calculated the pumping power to be:

$$\frac{\text{2 GPM/TR x [((2/3)}^{1.85}\text{ x 35') + 10']}}{\text{3960 x .84 pump eff.}} \times \frac{\text{.746 kW/HP}}{\text{.92 motor eff.}}$$

$$= \text{.013 kW/TR}$$

Thus, Kirsner calculated the difference between the pumping power at 3 versus 2 GPM/TR to be .021 kW/TR (.034-.013). However, this savings ignores the effect of reduced water flow on chiller-tube fouling.

If the condenser-water flow rate is cut by 33%, but the condenser passes and size remain the same, tube fouling will increase, and we calculate energy costs will rise $2500/year (based on 5000 hours/year of operation and $.08/kWh). If tube-cleaning frequency is increased to maintain heat-transfer efficiency, then maintenance costs will increase by $2000/year.

To avoid increased fouling, we will switch to a three-pass condenser to maintain the same water velocity in the condenser tubes. Doing so results in a chiller pressure drop that's 50% higher, or 22.5'. This will reduce the pump energy savings by $1600, but is the least of the three evils. The only pressure-drop reduction now is in the interconnecting piping system (20' of the original 45'). The pump efficiency is 80%, and the pumping horsepower is 0.021 kW/TR, as shown below:

$$\frac{\text{2 GPM/TR x [((2/3)}^{1.85}\text{ x 20') + 32.5']}}{\text{3960 x .80 pump eff.}} \times \frac{\text{.746 kW/HP}}{\text{.92 motor eff.}}$$

$$= \text{.021 kW/TR}$$

The pump energy savings with 2 GPM/TR in a 3-pass condenser is now .013 kW/TR (.034-.021) or 6.5 kW for 500 TR — a far cry from the 11 kW savings in the proponents' example.

Table 2: Proponents' Example

	Power Consumption (kW)		Savings or (Penalty)		
	3 GPM/TR	2 GPM/TR	kW	kW/TR	%
Chiller	272	289	(17)	(0.034)	(6.3)
Tower	24	16	8	0.016	33.3
Pump	26	15	11	0.022	42.3
Total	322	320	2	0.004	0.6

Energy Analysis of Tower at Design

In his energy analysis of the tower, Kirsner assumed a crossflow cooling tower with a draw-through propeller fan and a gravity-fed distribution header (the most common design today). Reducing condenser-water flow rate from 3 to 2 GPM/TR allows the tower to achieve the same leaving-water temperature with 15% less airflow, if it is not downsized.

Because the cooling-tower-fan horsepower varies roughly as the cube of airflow through the tower, at 85% airflow the tower-fan horsepower falls to (85% CFM)$^{3.2}$ = 58% of its former value, which translates to a 42% energy savings.

For a typical tower-fan motor selected at .05 HP/TR and assuming a 90% efficient motor, a 42% reduction in horsepower would amount to .017 kW/TR saved, as shown below:

$$\frac{\text{.05 HP/TR x .40 x .746 kW/HP}}{\text{.90 motor efficiency}} = \text{.017 kW/TR}$$

The savings equate to 8.5 kW for 500 TR, similar to the proponents' example.

Summary of Pump and Tower Energy

The preceding calculations show that with 2 GPM/TR, the pump saves 0.013 kW/TR and the tower saves 0.017 kW/TR, for a total savings of 0.030 kW/TR at design conditions. For a 2 GPM/TR design to make economic sense, these savings must outweigh the chiller penalty, which is examined next.

Energy Analysis of Chiller at Design

Everyone agrees that reducing the condenser-water flow rate has a negative impact on chiller efficiency. The dispute is over the actual numbers. Fortunately, Kirsner has done an impartial analysis with solid theoretical underpinnings.

Table 3: Isentropic Chiller Penalties with 2 GPM/TR at Design

Refrig.	# Compr. Stages	Drive Types	kW/TR with		kW/TR Penalty	% kW/TR Penalty
			3 GPM/TR	2 GPM/TR		
R-22	Single-Stage	Gear	.585	.632	.047	8.0
R-134a	Single-Stage	Gear	.571	.616	.045	7.9
R-123	Single-Stage	Gear	.554	.596	.042	7.8
R-123	Two-Stage	Direct	.541	.586	.045	8.3
R-123	Three-Stage	Direct	.534	.578	.044	8.2
				Average	0.45	8.0

Theoretical Energy Penalties — Kirsner did isentropic power calculations based on commercially available, .60 kW/TR chillers, both single-stage and multi-stage units, using a variety of refrigerants. He assumed 82% compressor efficiency, 95% motor efficiency, and, for gear-driven compressors, a gear efficiency of 99%. The results are summarized in Table 3.

It is interesting to note that all the theoretical chillers have penalties near .045 kW/TR, or 8.0%, for going to 2 GPM/TR, while the chiller analyzed by the 2 GPM/TR proponents had a penalty of only .034 kW/TR, or 6.3%, as shown in Table 2. This is our first indication that this chiller selection might be an unusual one.

The one factor Kirsner couldn't calculate theoretically was the change in compressor efficiency. He hypothesized that switching from 3 to 2 GPM/TR would have no effect on motor or gear efficiencies, but it might have an effect on compressor efficiency. To examine this effect, he looked at a number of actual chiller selections.

Actual Chiller Energy Penalties — Kirsner gathered data from two chiller manufacturers. They were asked to make the most economical selection for a specific tonnage at ARI conditions to achieve power input of approximately .60 kW/TR. Then, a second selection was requested to accommodate a condenser-water-flow reduction from 3 to 2 GPM/TR. The shells were held constant, but the compressor-impeller size and gears (if available) were allowed to vary. All selections were made for R-123. The results are shown in Table 4.

Actual Versus Theoretical Penalties — With the actual selections, all except one showed greater penalties than the .045 kW/TR and the 8.0% averages predicted with isentropic calculations. In fact, compared with the theoretical efficiencies, actual performance averaged 1% worse at 2 GPM/TR. As Kirsner predicted, compressor efficiency got worse at 2 GPM/TR.

One selection, however, exhibited only a 6.2% kW/TR penalty compared to the 8% average predicted by the isentropic calculations. For that selection, compressor efficiency was approximately 2% ***better*** at 2 GPM/TR, which is unusual.

It is interesting that this is virtually the same chiller performance used by the 2 GPM/TR proponents — a three-stage, direct-drive chiller using R-123, which suffered only a 6.3%

Table 4: Actual Chiller Penalties with 2 GPM/TR at Design

# Compr. Stages	Drive Types	TR	kW/TR with		kW/TR Penalty	% kW/TR Penalty
			3 GPM/TR	2 GPM/TR		
Single-Stage	Gear	400	.618	.683	.065	10.5
Single-Stage	Gear	500	.606	.667	.061	10.1
Single-Stage	Gear	485	.559	.606	.047	8.4
Two-Stage	Direct	600	.582	.637	.055	9.5
Two-Stage	Direct	500	.596	.644	.048	8.1
Three-Stage	Direct	500	.582	.618	.036	6.2
Three-Stage	Direct	600	.582	.637	.055	9.5
				Average	.052	8.9

energy penalty. The similarity is striking.

Since this is the only chiller analyzed with improved compressor efficiency at 2 GPM/TR, and because this selection is the linchpin of the analysis done by the proponents of 2 GPM/TR, it is worth trying to understand what makes this chiller different. Doing so will also show whether or not the overall analysis is valid. Kirsner did not take his study this far, but we can.

Analysis of an Exceptional Chiller

To analyze this exceptional chiller selection, we will look at how centrifugal compressors are designed and selected.

Centrifugal-Compressor Design — For a centrifugal compressor to meet a given head-pressure requirement, the tip speed of the impeller(s) must meet or exceed a given minimum speed. The compressor is most efficient if it operates at exactly that speed, without any overspeeding. This is true regardless of the number of compression stages.

In a direct-drive compressor, tip speed can be adjusted only by changing impeller diameter. Because impellers are expensive, there are limited sizes, which limit the tip-speed flexibility. In a gear-drive compressor, tip speed is a combined function of impeller and gear ratio, so more combinations are possible, yielding more tip-speed-tuning capability.

When the tip speed is higher than the required minimum, a higher compressor head is generated, which forces the leaving-chilled-water temperature (LCHWT) to go below design. The LCHWT sensor will detect this condition and close the compressor pre-rotation vanes slightly (even at design conditions) in order to compensate. Thus, the chiller literally starts off operating at part-load. This erodes compressor efficiency by varying amounts, depending upon how much the impeller(s) are overspeed.

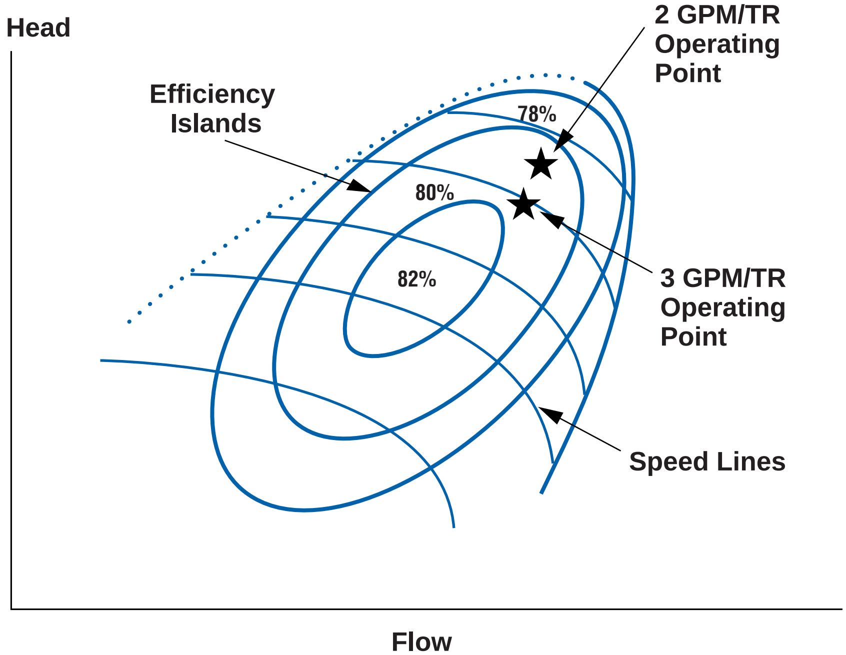
Centrifugal-Compressor Selection — To ensure efficient operation, a properly selected compressor at 3 GPM/TR will have a design operating point above and to the right of the peak efficiency island on a compressor map, as shown in Figure 1. This selection will ensure adequate range to avoid surge and optimum performance at off-design conditions, where most operating hours are spent.

At 2 GPM/TR, the compressor head increases, moving the operating point up farther from the peak efficiency island. As a result, there is an additional compressor efficiency loss at design conditions, which averages approximately 1%, as Kirsner's analysis showed in Tables 3 and 4.

Now consider the situation where a selection at 3 GPM/TR has an impeller tip speed that is too high, because the tip speed of the next smaller impeller would be too slow to handle the head at design conditions. In this situation, the vanes partially close, creating an efficiency loss — let's say 3%.

Now let's envision this same chiller operating with 2 GPM/TR. Lower condenser-water

Figure 1: Centrifugal Compressor Map



flow increases the head-pressure requirement, requiring a faster impeller tip speed, which this compressor already has. As a result, the vanes open up and the 3% efficiency lost at 3 GPM/TR is regained at 2 GPM/TR. So on this chiller, the combination of the 1% loss (because of the higher head-pressure requirement at 2 GPM/TR) and the 3% gain (because of vanes opening up) results in a net efficiency gain of 2%.

This is precisely what happened with the exceptional chiller — and why it had a 2% improvement in compressor efficiency at 2 GPM/TR, while all the other chillers had a loss of 1% or more. What appeared to be a good selection at 2 GPM/TR was actually the result of a poor selection at 3 GPM/TR, because the impeller was not machined to the correct diameter for the duty.

Realistic Energy Penalty for Entire Chiller Plant at Design Conditions

To correctly evaluate the 2 vs. 3 GPM/TR issue, it is more realistic to use the performance of the other 2 GPM/TR chillers, which had an average energy penalty of 9.4% (up from 8.9% because we dropped the exceptional selection out of the average). If the exceptional chiller had the proper tip speed at 3 GPM/TR and a more realistic energy penalty of 9.4% at 2 GPM/TR, then it would have required a design energy input of 272 kW x 1.094 = 297 kW, which is a penalty of 25 kW.

With more realistic chiller, pump, and tower selections, the chiller-plant energy at design conditions would *increase* by 10 kW, as shown in Table 5 (next page). A 2 GPM/TR design no longer makes economic sense.

Table 5: Proponents' Example with Realistic Performances

	Power Consumption (kW)		Savings or (Penalty)		
	3 GPM/TR	2 GPM/TR	kW	kW/TR	%
Chiller	272	297	(25)	(0.050)	(9.4)
Tower	20	11.5	8.5	0.017	42.5
Pump	17	10.5	6.5	0.013	38.2
Total	309	319	(10)	(0.020)	(3.2)

However, this is just one operating point — design conditions. What is the impact of 2 GPM/TR at off-design conditions?

Energy Analysis at Off-Design

The proponents' example looked only at design conditions. But off-design conditions prevail for nearly 99% of operating hours, so performance at off-design conditions is more important than at design. To evaluate off-design performance, energy savings of the pump and the tower will be compared to the energy penalty of the chiller at various off-design conditions. Single-chiller systems will be examined first, with multiple-chiller systems discussed later.

Energy Analysis of Pump at Off-design — During off-design operation, pump energy savings resulting from 2 GPM/TR will be constant because of constant water flow (this assumes that no special energy-saving technology is applied, such as variable-speed drive or two-speed motors, which would actually reduce the savings of a 2 GPM/TR design). However, on a kW/TR basis, the savings will increase, because the TR are decreasing. So a .013 kW/TR savings at 100% load will be a savings of roughly .026 kW/TR at 50% load, and so on.

Energy Analysis of Tower at Off-design — During off-design conditions, tower energy savings on a kW/TR basis will rise with load reduction, just as the pump savings did. However, the savings will be mitigated by the need to control the minimum entering-condenser-water temperature (ECWT) sent to the chiller.

When the ECWT reaches the minimum that the chiller can handle, the tower fan will begin to cycle, flattening out the savings (when the fan cycles off, there is no relative savings between 2 and 3 GPM/TR — in both cases, energy usage is zero).

Assuming the only tower control is minimum ECWT (an energy-saving technology such as variable-speed would actually reduce the savings of a 2 GPM/TR design), systems with chillers that are limited to 75°F minimum ECWT will start cycling the tower fan below 70% load, reducing the tower savings. On systems with chillers that can utilize ECWT as low as 55°F, the full tower savings are available at all off-design conditions (assuming that an air-side economizer takes over the cooling duty below 55°F outdoor DB). This method of control is the simplest, but most inefficient.

Figure 2: Typical Chiller ARI Off-design Performance

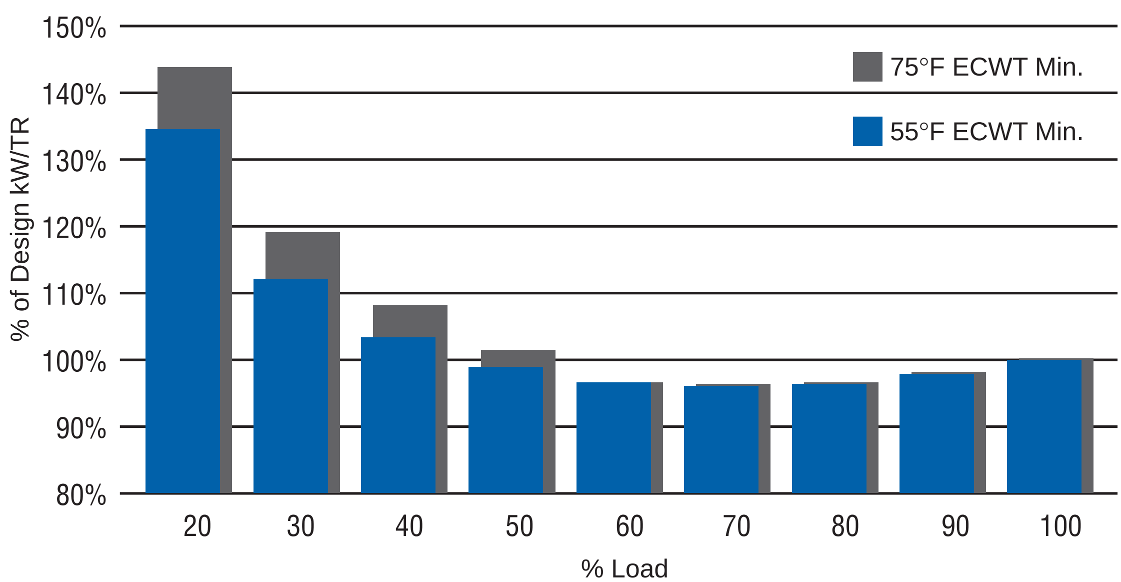
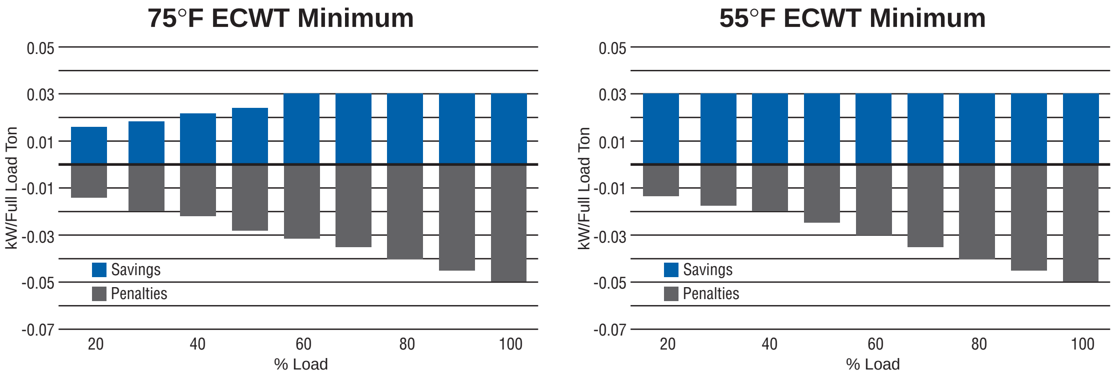


Figure 3: Energy Analysis of 2 GPM/TR with Constant-Speed Chiller System



Therefore, it provides the most favorable conditions for a 2 GPM/TR design.

Energy Analysis of Chiller at Off-design — Figure 2 shows the typical ARI performance of a constant-speed centrifugal chiller at off-design conditions. This performance is independent of condenser-water flow, but it is dependent upon the minimum entering-condenser-water-temperature (ECWT) that a chiller can utilize. At lower loads, a 75°F-minimum-ECWT chiller uses more energy, because the ECWT limit holds the compressor head artificially high.

The energy consumption of both chillers initially decreases, and then increases as the pre-rotation vanes close further. This means that a 55°F-minimum-ECWT chiller with an energy penalty of .052 kW/TR at 100% load will have an energy penalty at 70% load of (.052 kW/TR x .97) = .050 kW/TR, and an energy penalty at 20% load of (.052 kW/TR x 1.34) = .070 kW/TR.

Summary of Energy Savings and Penalties at Off-design

It is easier to understand the net effect of the varying energy penalties and savings by graphing them and drawing some inferences. The graphs use kW/Full-Load-TR to properly weight the effect of load on each kW/TR point (i.e., a .01 kW/TR difference at 100% load has twice the effective value as a .01 kW/TR difference at 50% load).

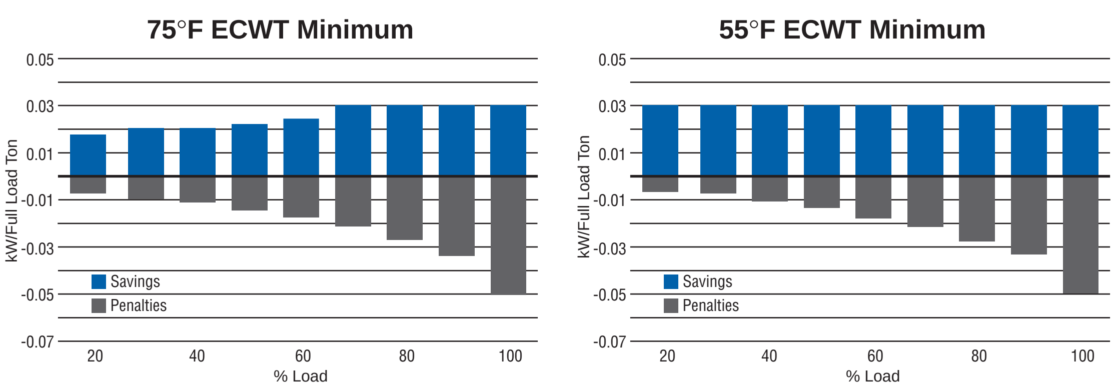
The left side of Figure 3 illustrates a system with a 75°F-minimum-ECWT chiller, and it shows that 2 GPM/TR only starts to save energy below approximately 30% load. This cross-over point is defined as the % load below which the savings *begin* to outweigh the penalty. This is not to be confused with the point where net savings begin, because there must be sufficient kWh savings below the cross-over point to counter-balance the kWh penalty above it. Everything depends on the loadline for building and the number of hours of operation at each load.

The right side of Figure 3 depicts a system with a 55°F-minimum-ECWT chiller. It shows that 2 GPM/TR starts to save energy below approximately 60% load. However, the kWh saved is normally insufficient below 60% load to counterbalance the kWh penalty above it. It is interesting to note that the better a chiller's off-design performance (i.e., able to effectively use 55°F minimum ECWT), the higher the cross-over point and the greater the opportunity for 2 GPM/TR designs to deliver real energy-cost savings.

Are there any other types of chiller systems where a 2 GPM/TR design might make sense? A variable-speed chiller might be a possibility, because it has the best off-design performance in existence. Its energy penalty falls off more rapidly than that of a constant-speed chiller at off-design conditions, as seen in Figure 4.

With a variable-speed chiller utilizing 55°F

Figure 4: Energy Analysis of 2 GPM/TR with Variable-Speed Chiller System



minimum ECWT in a 2 GPM/TR system, energy savings start below 83% load. A chiller in this configuration might make economic sense if a 2 GPM/TR design is to be used. A word of caution, however — the chiller must react well to lower ECWTs, not just be capable of handling them. Some designs don't reduce energy with ECWT reductions below 70-75°F. In fact, some even use more energy!

Where Does 2 GPM/TR Make Sense?

Based on the above analysis, where does it make sense to use 2 GPM/TR? From a first-cost standpoint, a system with very long condenser-water piping runs and/or an over-abundance of valves would favor 2 GPM/TR. The first-cost savings of reducing the pipe and valve sizes could be substantial enough to outweigh the energy-cost penalty. However, if a life-cycle-cost analysis is done over the life expectancy of the chiller plant, it will almost always pay to remain with the 3 GPM/TR design and ignore any first-cost savings available from the 2 GPM/TR design.

A somewhat similar situation where the first-cost savings might make sense is where an existing piping system is being connected to a larger replacement chiller. Rather than replace the piping to accommodate the larger duty, reduce the water flow. The avoided first-cost savings may offset any energy penalties, although this should be analyzed each time.

From an energy-cost viewpoint, single-chiller systems with lots of low-load hours will favor 2 GPM/TR. However, multiple-chiller systems probably will not, because the individual chiller loads remain high. As the building load goes down and a chiller cycles off, the average load on the remaining chillers rises, and higher loads on chillers equate to higher penalties, favoring 3 GPM/TR.

Finally, any situation which might favor 2 GPM/TR can be made even more attractive with variable-speed chillers that can also utilize 55°F ECWT. Cross-over points are in the vicinity of 83% load, a situation that favors 2 GPM/TR systems. Again, the ability to effectively utilize lower ECWT is paramount. On a variable-speed chiller, there must be energy savings of 2 to 3% per °F of condenser-water reduction below 75°F for this strategy to favor 2 GPM/TR. In addition, any other engineering design which attempts to reduce pump or tower energy (with intelligent control strategies, for instance) will work against 2 GPM/TR.

Conclusions about 2 GPM/TR

Finding real savings is the goal of every owner and operator. From time to time, new ideas emerge (or old ideas re-emerge, as is the case here), which must be tested in the harsh light of reality. Here, we have examined the claim that a 2 GPM/TR system produces net savings compared to the conventional 3 GPM/TR design.

Analysis shows, however, that reductions in first cost and energy cost are not available simultaneously with a 2 GPM/TR design. If the goal is to maximize energy savings, then the same size cooling tower, condenser-water pump and piping must be maintained. If first-cost savings are the goal, then energy costs will increase, usually exceeding the first-cost savings on a life-cycle basis.

There are a few particular situations that favor a 2 GPM/TR design, but not as many as proponents imply. The flaw in their analysis is to use a misleading chiller selection whose 2 GPM/TR performance looked good only because improper impeller sizing made its 3 GPM/TR performance look bad. They also used pump and tower designs that used more energy than is in keeping with current design practice.

In short, there are no miracle savings available. While there are times when a 2 GPM/TR design may be advantageous, most of the time, the old standard of 3 GPM/TR will result in a more economic design.