1996/Vol. 25, No. 1

Engineers Newsletter

How Can This Be?

Introduction

In two previous *Engineers Newsletters*, we addressed the issue of chiller performance under the title "The Search For Chiller Efficiency." The linchpins of these discussions were the laws of physics and thermodynamics and how these laws offer specific and defined limitations on maximum, real-world chiller efficiency.

Recent HVAC industry advertisements appear to contradict some of these earlier-cited limitations. Were the laws of physics and thermodynamics recently repealed? Are they being misinterpreted? Or is there something funny going on? If so, exactly **what is the source of confusion?**

The objective of this newsletter is to answer each of these questions and provide substantive, fundamental technical background for those system designers who wish to go beyond the headlines and interpret chiller efficiency claims and facts for themselves.

The Latest Volley

The performance curve shown in Chart 1 models an advertisement headline proclaiming a particular chiller's ability to consume 0.20 kW/ton at 60 percent of its design capacity and 0.26 kW/ton at full capacity. Clearly, this is sensational performance. But is it real? Surprisingly, the answer is **yes!**

Performance at 0.20 kW/ton seems so much better than anything published and ARI-rated today. How can this be? There's some trickery involved, but here's how it's done.

Back To The Basics

First, it's important to build a foundation of fact. Overall chiller efficiency continues to depend on four, and only four, elements:

- Water-to-refrigerant heat transfer
- Refrigerant-cycle thermodynamics
- Power conversion and transmission efficiency
- Centrifugal compressor efficiency

Chart 1: Effect of decreased condenser water temperature on chiller efficiency (York's claim)

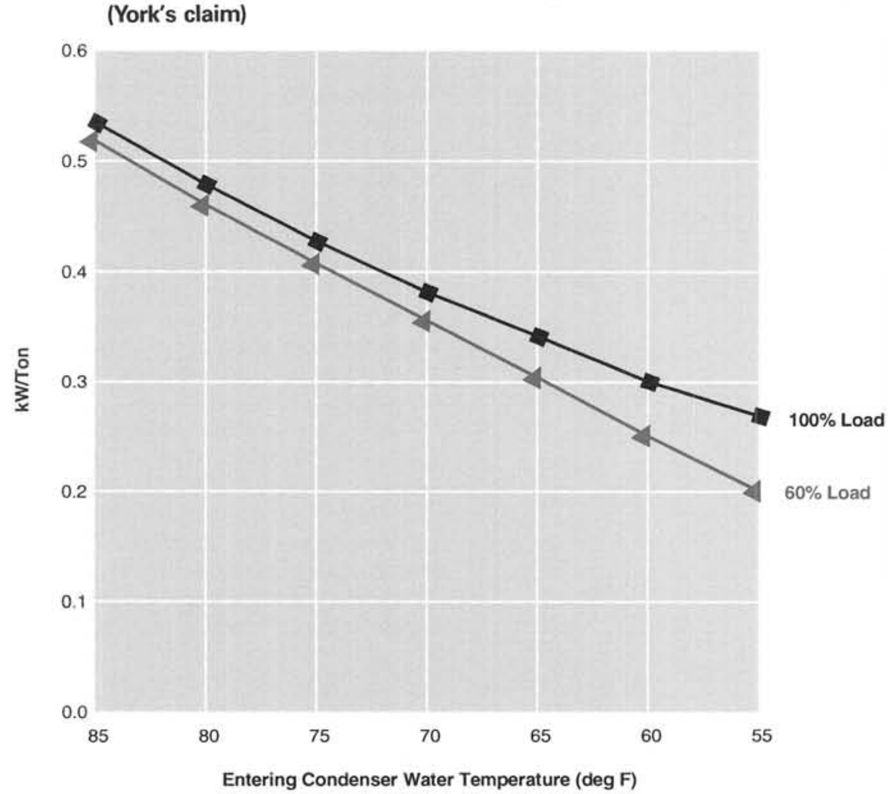


Table 1: Theoretical isentropic efficiencies for commonly used refrigerants at 40 F saturated suction and 100 F saturated condensing temperatures

Refrigerant	Theoretical hp/ton	Theoretical kW/ton
11	0.624	0.465
12	0.667	0.497
134a	0.679	0.507
123	0.634	0.473
22	0.676	0.504

Table 2: Effect of "economizers" on refrigerant cycle efficiency (see Table 1)

	Theoretical hp/ton			Theoretical kW/ton		
Refrigerant	Simple	1-econ	2-econ	Simple	1-econ	2-econ
134a	0.679	0.652	— [1]	0.507	0.487	— [1]
123	0.634	0.609	0.590	0.473	0.454	0.440

[1] No such compressor design is commercially available

While the differences from one manufacturer's design to another aren't great in any of these four elements, they are significant enough to merit inspection. The *Engineers Newsletter* focused on this twice in the past; once in 1981 and again in 1994. Good news! The basics are unaltered and the laws of physics and thermodynamics remain unchanged, so the fundamentals we all learned in our physical science classes still hold true.

That being the case, let's examine the latest chiller performance claims in light of each of these elements.

Heat Transfer

Thanks to advanced technology, and the additional research and development activity prompted by the advent of new alternative refrigerants, heat transfer continues to improve in both evaporators and condensers. The fruits of this labor are evident, as we now observe refrigerant-to-water "approach" temperatures in the 2 to 4 F range. For evaporators, this means it's now economically feasible to obtain refrigerant gas temperatures roughly 3 F lower than the chilled water temperature produced there. Likewise, saturated condensed liquid in the condenser can often be within 3 F of the leaving condenser water temperature. Twenty years ago, normal approach temperatures were 7 to 10 F.

Refrigeration-Cycle Thermodynamics

"Simple cycle" efficiencies based on alternative refrigerants are a trifle lower than their CFC predecessors in HVAC centrifugal chiller applications. As Table 1 indicates, R-123 is about two percent less efficient than the R-11 it replaces, while R-134a is about three percent less efficient than R-12.

Various alterations can be made to improve "simple-cycle" refrigeration performance. One of the most common compressor adaptations is the addition of an **economizer**, also known as an **interstage flash chamber**. Since this device is physically placed between stages of compression, its use is limited to multistage compressors. Of course, multiple economizers can be used if more than two stages of compression are involved. Table 2 compares the cycle efficiencies for common compressor/economizer arrangements with the "simple cycle" values presented in Table 1.

Liquid subcooling benefits some refrigerant cycles. Unfortunately, its potential benefits are insignificant in water-cooled chiller applications because of the small temperature difference between the heat sink (condenser water) and saturated condensed liquid. Heat transfer improvements further diminish this temperature differential. Not surprisingly, subcooling cycles are seldom employed with this kind of equipment.

All commonly used refrigerants respond to changes in "thermodynamic head." As is the case for an ideal refrigerant, less theoretical power is required when the pressure differential between the evaporator and the condenser is reduced. Charts 2 and 3 show the effect of different "heads" on the theoretical power consumption of R-123 and R-134a cycles, respectively. Clearly, head reduction affects each of these refrigerants in a very similar manner.

Power Conversion And Transmission Efficiency

There is **almost** nothing new to report on these technologies. (We say "almost" because we'll examine the effects of frequency variation later in this article.) Induction motor efficiencies remain in the 93 to 96 percent range for three-phase, 60-Hz motors in the size ranges used by centrifugal compressors.

This performance holds true for both hermetic and open motors; apart from the cooling system, their designs are virtually identical. (The power sacrificed by open motors to drive cooling-air fans is minimal and unworthy of analysis here, although the effect on equipment room cooling and ventilation can be significant.) Both motor designs, for example, use two journal bearings to support the rotor. Motor performance ratings reflect the frictional losses associated with these bearings.

(Editor's Note: The analysis presented in this issue is based on a motor with an overall efficiency of 95 percent.)

Chart 2: Effect of reduced "head" on the refrigerant-cycle efficiency of R-123

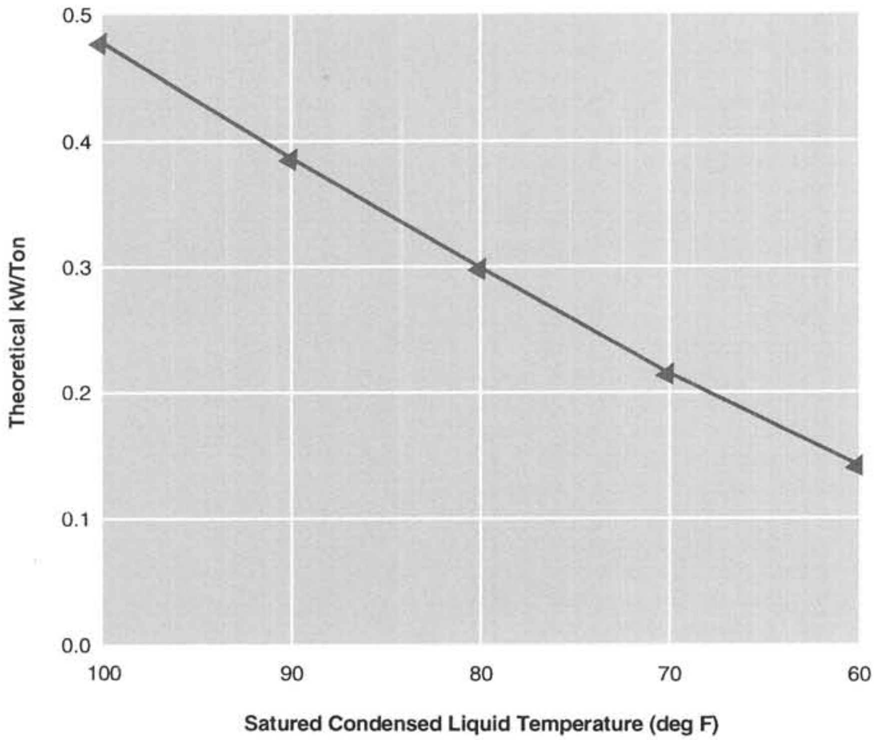
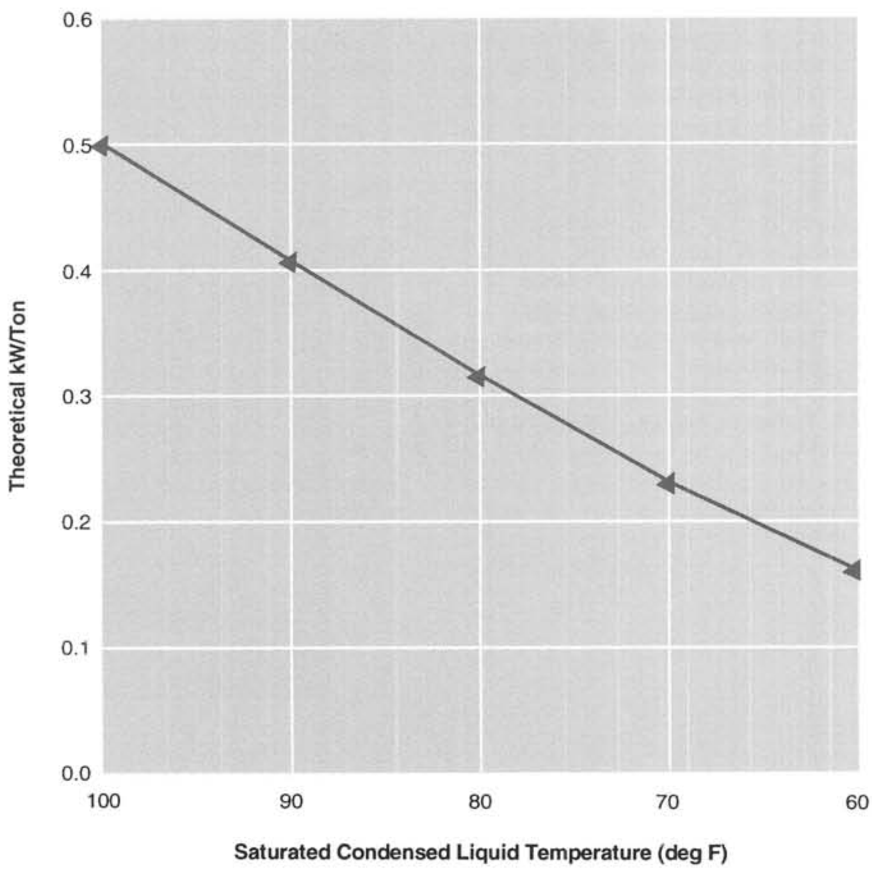


Chart 3: Effect of reduced "head" on the refrigerant-cycle efficiency of R-134a



Compressors operating at speeds higher than the motor's synchronous speed (3600 rpm for 60-Hz, two-pole motors) require some kind of "step-up" transmission. Various arrangements are used, each with additional gears and bearings. By themselves, the gears absorb about one percent of their power transmission output in heat-producing friction. Efficiency losses attributable to friction typically amount to about 1 hp (2,545 Btu/hr) per bearing.

Consider the performance impact on a common transmission design that employs two gears and five bearings: frictional bearing losses are roughly 5 hp (12,726 Btu/hr) or nearly 4 kW which, in turn, amounts to about one percent of the power output of a 540-kW motor. Thus, total gear and bearing losses in the transmission of an efficient 1,000-ton chiller are about two percent, leaving a transmission efficiency of 98 percent. Of course, chillers without transmissions suffer no such loss.

Compressor Efficiency

Incremental improvements in compressor efficiency continue to be achieved with the ongoing competition between single- and multiple-stage compressor technologies. Both designs are highly refined and well-suited to water chiller applications.

As evidenced by the typical centrifugal compressor "map" in Chart 4, all constant-speed centrifugal compressors lose efficiency at part-load conditions. Consequently, manufacturers generally select compressors at their "sweet spot" for the rated design conditions. Normal chilled water system unloading not only results in lower pressure differential demand, but proportionally less volume as well; i.e., fewer tons of capacity.

Chart 4: Open inlet vane compressor performance at full (100%) speed

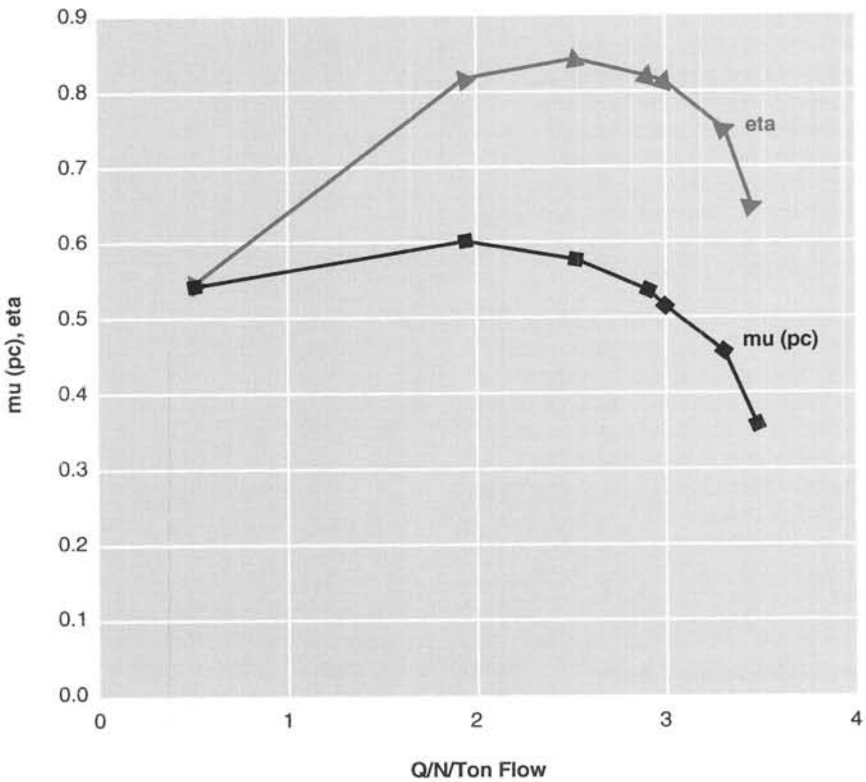
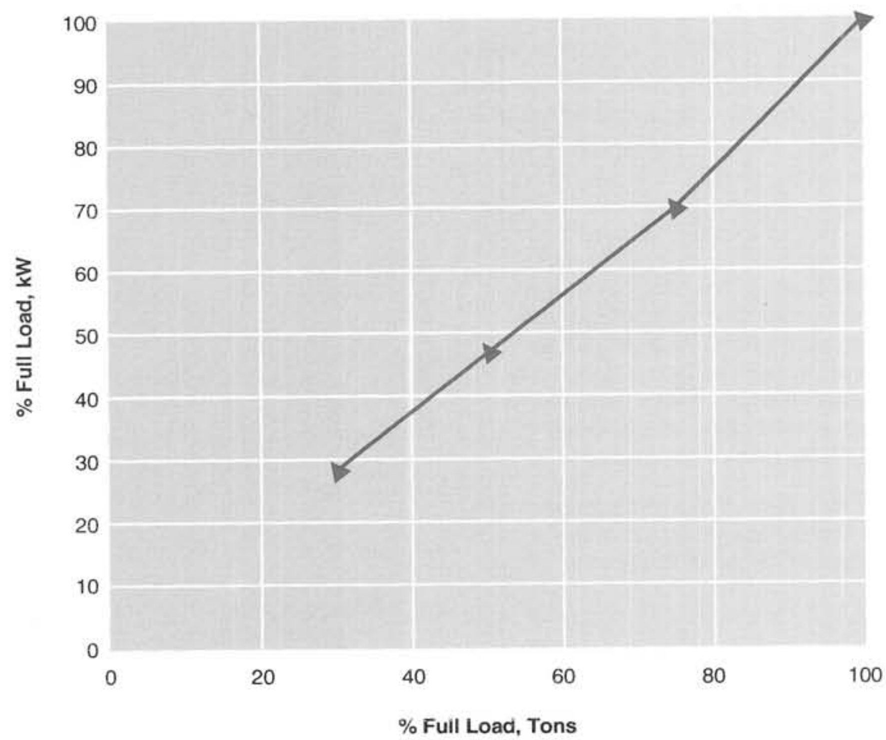


Chart 5: Chiller performance, % power vs % load



Pressure differential, on the other hand, basically follows saturated refrigerant pressure/temperature values between the condenser and evaporator. These values track closely with the **leaving** water temperature for each of these heat exchangers.

Therefore, as a system unloads, we can expect a **lower temperature rise** in the condenser water, as well as a potentially **lower condenser water supply temperature**. These two affects combine to reduce the pressure differential significantly, leading to a much more efficient refrigerant cycle . . . but at the expense of reduced compressor efficiency. The net result is better overall chiller part-load efficiency . . . to a point. Chart 5 shows this relationship. Points plotted on this chart originate from the data contained in Charts 4 and 2, and incorporate the ARI unloading factor for decreasing condenser water temperature.

Clearly, this performance doesn't approach anything close to 0.20 kW/ton at any load condition, regardless of entering condenser water temperature. What did we overlook?

The Missing Ingredient

Chart 2 contains our first clue. Notice the relative theoretical "simple cycle" efficiencies (0.47 and 0.25 kW/ton) at thermodynamic heads of 98 F and 73 F, respectively. These two values represent normal saturated condensed liquid temperatures for a fully loaded chiller with a design entering-condenser water temperature of 85 F and for the same chiller when the condenser is supplied with 60 F water. These extremes are plotted on Chart 1 and correspond to performance of 0.54 kW/ton and 0.26 kW/ton for a fully loaded chiller. The ratio of these two values is 0.26/0.54, or 0.48. In other words, a chiller operating with this cycle consumes 48 percent of its design power when the entering-condenser water is 60 F.

Chart 6: Variable-speed compressor performance at 100% and 70% speeds

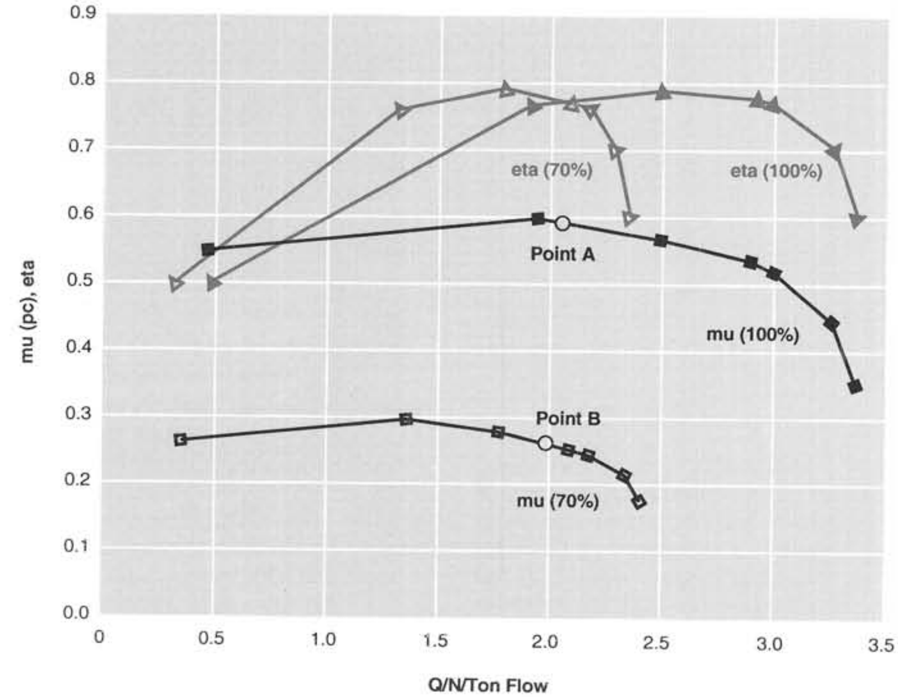
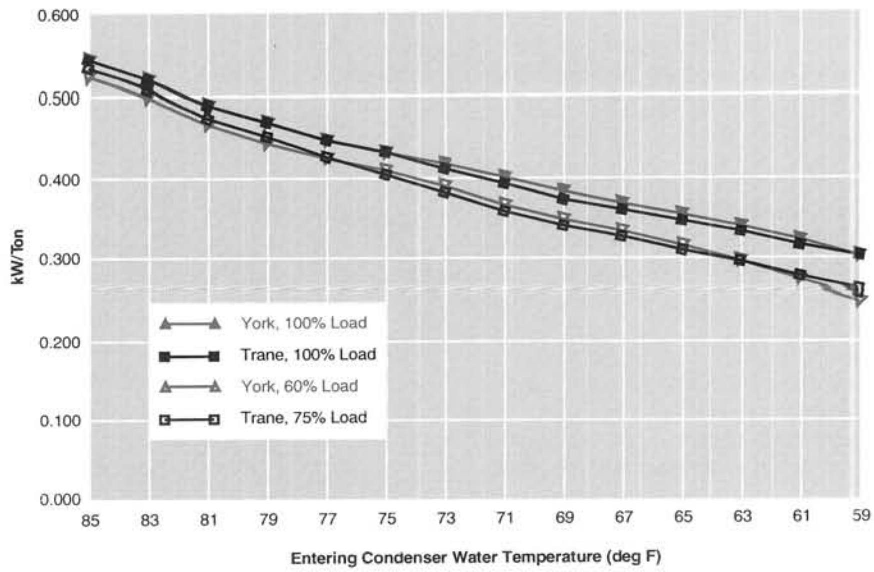


Chart 2 shows the relative theoretical efficiency values of 0.45 kW/ton and 0.24 kW/ton at 98 F and 73 F, respectively. The ratio of these two values is 0.53 . . . amazingly similar to the 0.48 ratio calculated above. Theoretically, then, the power consumed at a 73 F saturated condensing temperature is 53 percent of that consumed at 98 F. This means that **all other chiller components must operate with very similar efficiencies at both conditions.**

How can this be, if compressor efficiency falls at decreased thermodynamic head (Chart 4)? We need to find some way for the compressor to produce **less thermodynamic head without losing volumetric capacity or efficiency**. Said another way, we must develop a scheme to "kill" compressor head without sacrificing available cooling tons or performance. One possible way to perform this feat is to operate the compressor at a lower speed. Since head is proportional to *(velocity)*^{2}, we can theoretically produce 49 percent head at 70 percent speed. But slower velocities also reduce volume capability. How can we obtain full volume at reduced speed?

The answer: **Ride the compressor curve** to a higher relative volume at reduced speed. Chart 6 shows two different speed curves for the compressor represented in Chart 4. While any point on either of these curves is a possible operating point, we want to find two points that differ only in thermodynamic head . . . not volume. Points A and B qualify.

Chart 7: Comparison of York and Trane centrifugal chiller performance



Notice that it's important to select the original point (A) away from any sort of compressor volume limit since we must "ride" the volume curve far enough to reach design gas volume. Failing to do so will result in a capacity shortfall. Also significant is this simple fact: Operating a centrifugal compressor at reduced speed "kills" thermodynamic head without sacrificing efficiency.

Speed Control

Reaching high chiller performance values requires a twofold strategy: **Variable compressor speed and controls that respond to thermodynamic head demand**. Specialized frequency inverters can implement this strategy. Adding such an inverter to the drive train means it **is** possible to reach 0.26 kW/ton at full capacity and 0.20 kW/ton at reduced (60 percent) capacity conditions. (Bear in mind that inverters do lose small amounts of power in the form of heat. These losses are continuous unless the inverter is electrically bypassed when the motor must operate at full speed.)

So far, everything we've seen is based on the practical application of fundamentals. Is there any test data that reflects the actual performance of a centrifugal chiller at these specific off-design conditions? The answer is **yes**. Chart 7 illustrates the performance of a Trane CVHF-080 operating with an Adaptive Frequency™ drive (inverter) and compares it with the advertised claims modeled in Chart 1. It's evident that the addition of an inverter enables exceptional performance at part-load conditions.

So why doesn't everyone use frequency inverters? For the same reason that everyone doesn't own a $40,000 automobile. They can't afford it. Inverters are very expensive. As Table 3 indicates, large and continuous operating cost savings are required to amortize the first cost of a $40,000 frequency inverter purchased to drive the compressor motor. In reality, very few systems can generate operating cost savings of this magnitude.

Table 3: Amortization table for a $40,000 inverter

Year	Savings	Cost of Money — Annual Percentage Rate						
		0%	4%	5%	6%	7%	8%	9%
1	annual	$40,000	$41,600	$42,000	$42,400	$42,800	$43,200	$43,600
	kWh*	571,429	594,286	600,000	605,714	611,429	617,143	622,857
2	annual	$20,000	$21,208	$21,512	$21,817	$22,124	$22,431	$22,739
	kWh*	285,714	302,969	307,317	311,678	316,052	320,440	324,839
3	annual	$13,333	$14,414	$14,688	$14,964	$15,242	$15,521	$15,802
	kWh*	190,476	205,913	209,833	213,777	217,744	221,733	225,746
4	annual	$10,000	$11,020	$11,280	$11,544	$11,809	$12,077	$12,347
	kWh*	142,857	157,423	161,150	164,909	168,702	172,526	176,382
5	annual	$8,000	$8,985	$9,239	$9,496	$9,756	$10,018	$10,284
	kWh*	114,286	128,358	131,986	135,655	139,366	143,118	146,910

* @ 7 cents per kWh

Let's take a closer look at Table 3 and what it tells us. The "0%" column represents simple payback values. More likely, money has an associated cost to any owner and is expressed as the "annual percentage rate."

If electricity costs 7 cents per kWh, the necessary dollar payback values are translated into kWh savings required to pay off the inverter. Naturally, the longer an owner can wait for payback, the smaller the threshold for required savings becomes. For example, if an owner requires payback within three years and assigns six percent to the cost of money, an annual energy savings of 213,777 kWh is required.

Assume a difference of 0.27 kW/ton between 0.48 kW/ton (normal 60 F, 60 percent, part-load operation **without** an inverter) and 0.21 kW/ton (60 F, 60 percent, part-load operation with an inverter). For an 800-ton chiller, 0.27 kW/ton amounts to a decrease in power consumption of 216 kW at this particular off-design condition.

Under these circumstances, it would take 990 hours of operation annually to generate the necessary 213,777 kWh. The likelihood of a combined system load and environmental climate providing this condition over such an extended number of hours seems remote.

To qualify as a candidate for a variable-speed operating strategy, the chiller's load profile must exhibit many hours of operation at condenser water temperatures between 55 and 70 F. Little savings are generated at temperatures above 70 F. Below 55 F, **free cooling** is a better alternative. Of course, this narrows the "window of opportunity" for economic payback. In any serious economic analysis, the relative cost of obtaining 55 to 70 F condenser water must be weighed against the cost of 85 F water.

0.0 kW/Ton

Face it: **Free cooling is difficult to beat . . . if the associated capital costs are reasonable.** Several forms of free cooling make sense; one option is the "sidecar" strategy explained in an earlier *Engineers Newsletter* (Vol. 20, No. 3). Illustrated in Figure 1, the "sidecar" heat exchanger permits a measure of free cooling whenever the cooling tower water temperature is lower than the system **return** water temperature. This scheme greatly expands the number of hours during which some 0.0 kW/ton cooling is obtained. Amortization is often quite rapid.

Just Kidding?

Definitely not. Seemingly unbelievable performance headlines usually contain at least a shred of truth. In this case, we find that 0.20 kW/ton chiller performance is very possible, as is 0.0 kW/ton. There's no end to one-upmanship in advertising!

Under the harsh light of informed scrutiny, we find that the simple laws of economics are the only real obstacle to achieving spectacular performance within a narrow range of operating conditions. Economics forces us to broaden the scope of study beyond the headlines.

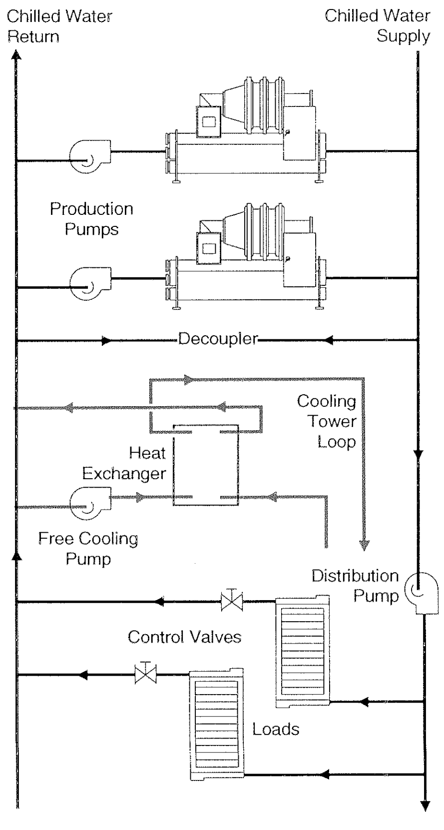


Figure 1: "Sidecar" free cooling

1996/Vol. 25, No. 1A

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How Can This Be?

Volume 25, No. 1, of the *Engineers Newsletter* examined the claims of several recent HVAC industry advertisements that seem to contradict the laws of physics and thermodynamics. To clarify the effect of these laws on real-world chiller efficiency, the newsletter provided technical background to aid system designers wishing to evaluate such assertions themselves.

Two graphs—**Chart 4** and **Chart 6**—were presented as part of the discussion on **centrifugal compressor efficiency** as one of the four basic elements of overall chiller efficiency. Regrettably, the legends were inadvertently omitted from both illustrations. Both charts are reproduced here, along with more descriptive axis and curve labels. Explanatory notes follow.

Compressor Performance

Charts 4 and 6 at right describe three facets of centrifugal compressor performance:

- **Adiabatic efficiency,** or **eta (η_s),** refers to mechanical efficiency, and is the ratio of isentropic work, W_s, to actual work, $h_2 - h_1$; that is, $W_s \div (h_2 - h_1)$.
- **Overall work coefficient,** or **mu (μ),** is determined by multiplying gravitational constant g by polytropic work W_p and dividing the result by the sum of the per-stage impeller blade tip speed u_i squared—i.e., $(g \times W_p) \div \sum u_i^2$. The relationship between polytropic head H_p and polytropic work is expressed as $H_p = W_p \div g$.
- **Flow coefficient** is dimensionless, and is calculated by dividing the per-ton volumetric flow rate, Q, by the product of impeller rotational speed N and impeller diameter D cubed—i.e., $Q \div (N \times D^3)$.

Chart 4: Open inlet vane compressor performance at full (100%) speed

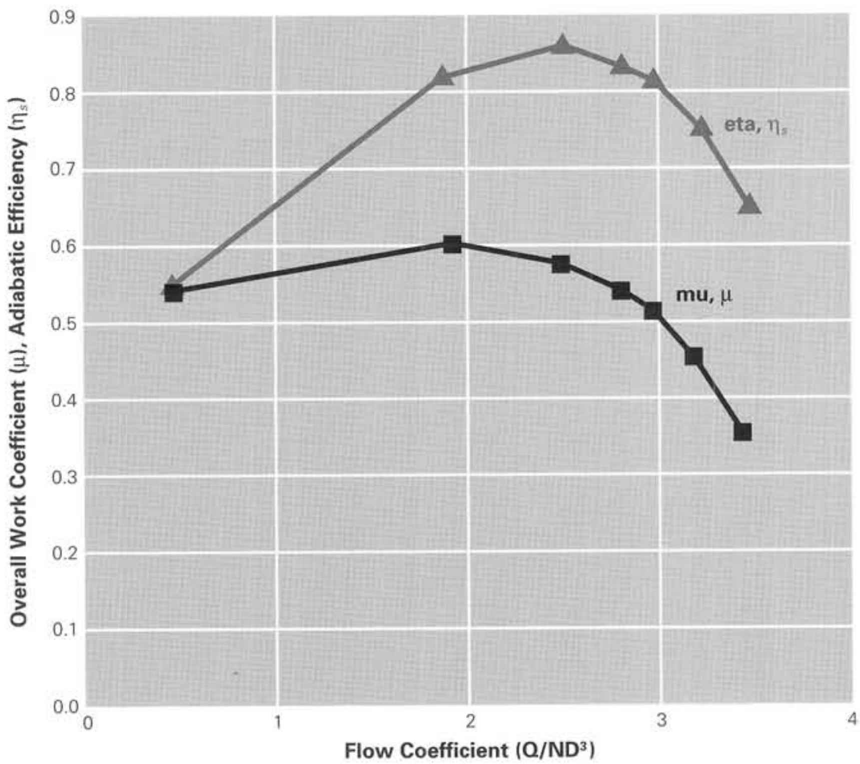

Chart 6: Variable-speed compressor performance at 100% and 70% speeds