Better Than Oil-Free:

Why Screw Chillers with Pure Speed Capacity Control Save More Energy Than Magnetic Bearing Chillers

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INTRODUCTION

The proposed benefits of oil-free, magnetic bearing chillers have been widely marketed, and many current trade journals include multiple references to these chillers. Interestingly, the benefits of chillers that offer variable speed screw technology with pure speed capacity control[1] have not received the same attention. This paper will compare and contrast the two technologies and illuminate a simple, logical approach to higher energy savings. See Table 1 for a comparison summary.

ACHIEVING HIGHER EFFICIENCY

In the case of oil-free magnetic bearing chillers, eliminating energy losses due to friction and reducing resistance to heat transfer are marketed as contributing to higher efficiency. In the case of variable speed screw chillers with pure speed capacity control, the elimination of mechanical unloaders and surge are, in contrast, marketed primarily as reliability and operational flexibility benefits. Equipped with these explanations, and provided only with the chillers' efficiencies as shown in Figure 1, can you tell which of the chillers shown is oil free?

The answer may surprise you. The chiller on the left, with the higher kW/ton value, is the oil-free chiller; the more efficient chiller on the right is the variable speed screw chiller. This result is consistent across a wide range of capacities in which the two technologies are offered, as shown in Table 2.

Table 1 – Comparison of Claims

Centrifugal Chillers with Oil-Free Magnetic Bearings	Screw Chillers with Pure Speed Capacity Control
• Frictionless bearings* • Improved heat transfer† • Less service and maintenance** • Low Sound	• No losses due to mechanical unloaders • No use of hot gas bypass to prevent surge • Less service and maintenance • Low sound

* Main bearings require power for the levitation system, operating at higher motor speeds may increase in windage losses. Touchdown (back up) bearings are anti-friction bearings.
† The impact of the studied benefit is still less than 0.1 – 1% depending on load, ignoring the benefit of oil "wetting" on the upper rows of evaporator tubes.
** Magnetic bearings include multiple sensors and coils as well as the electronics to control and power them.

[1] The use of mechanical unloaders of any kind is not required under normal operating conditions.

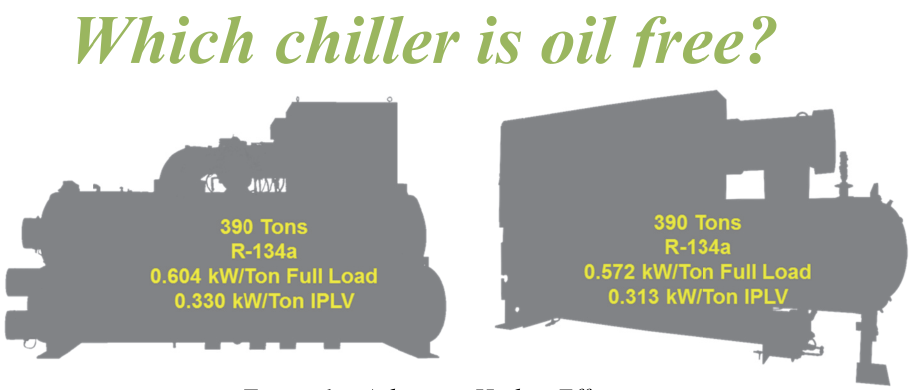


Figure 1 – Achieving Higher Efficiency –
Oil-Free Magnetic Bearings vs Pure Speed Capacity Control

Table 2 – Chiller Efficiency Values

Capacity (Tons)	Oil-Free Chiller Claim (kW/ton)*	23XRV Chiller* (kW/ton)
250	0.633 / 0.357	0.592 / 0.327
290	0.634 / 0.328	0.572 / 0.313
360	0.576 / 0.327	0.574 / 0.303
390	0.604 / 0.330	0.575 / 0.315
450	0.590 / 0.346	0.578 / 0.307

*Oil-free chiller efficiency values are presented in public domain, 23XRV efficiency values are per AHRI certified selections.

TO BUILD A BETTER CHILLER, BUILD A BETTER COMPRESSOR

Given the well-publicized explanation surrounding the benefits of oil-free chillers, it is worth noting that the most efficient chillers in the industry are actually oiled chillers. To understand why, we need to understand how each chiller component influences efficiency. The pie chart shown in Figure 2 displays the contribution of losses within a water-cooled chiller at 100% load, operating at AHRI (Air-Conditioning, Heating and Refrigeration Institute) conditions. The results were similar at reduced loads except the compressor and VFD contributions increased while heat exchanger losses dropped from 14% at full load to less than 3% at 25% load.

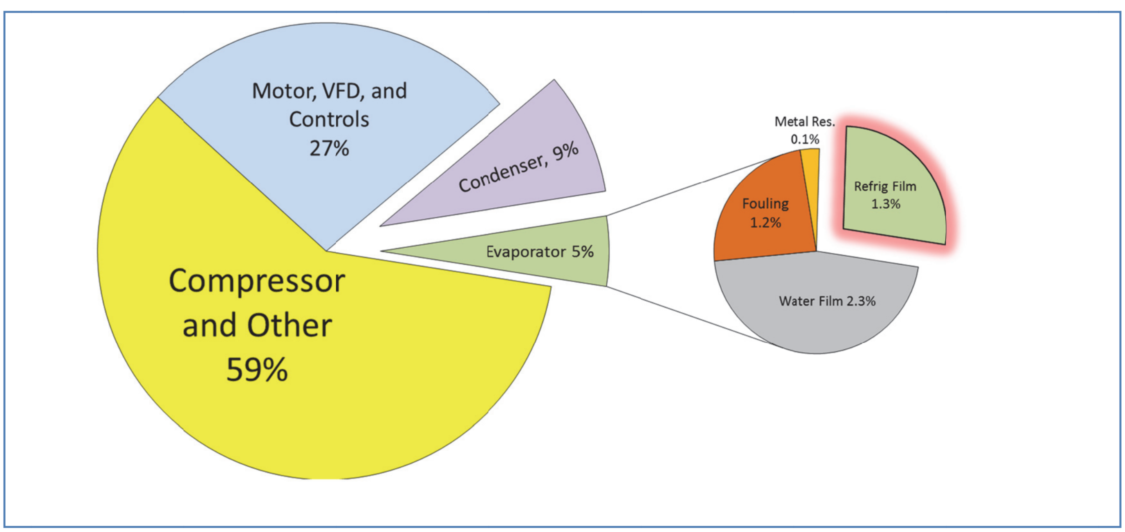


Figure 2 – Chiller Losses by Component

The outcome suggests that if you want to build a better chiller, start with a better compressor. The losses within the compressor represent over half of the losses in the chiller.[2]

A screw chiller with pure speed capacity control eliminates all losses associated with mechanical unloaders such as guide vanes, providing significant improvement to the component that contributes to efficiency most. In contrast, most of the oil free marketed benefit surrounds the bearing friction and impact of oil on heat transfer. These losses are minor in modern production chillers.

Bearings

While marketed as frictionless, the magnetic bearings in oil-free chillers actually do consume power.

In screw compressor designs, rolling element bearings offer long bearing life and simplicity.[3] The rolling element bearing losses represent a small mechanical loss in a low speed compressor. These bearing losses further decline with speed as the compressor unloads and are included in the published kW/Ton of the chiller. Furthermore, rolling element bearings do not require capacitors or electronic boards to operate, eliminating these potential modes of failure.

Many oil-free compressors use magnetic bearings instead of mechanical bearings. (Many oil-free compressors operate at over 30,000 rpm, influencing the selection of magnetic bearings.) While these bearings are marketed as "frictionless," they actually do consume power. Power is required for the positioning sensors and control. Electrical losses associated with this function are generally emitted as heat, which does not advance the purpose of chilling a fluid and would therefore be categorized as a loss. In addition, the operation of compressors at such high speed may increase the windage losses in the electric motor. Since windage losses are associated with the movement of the cooling medium around the motor as it spins, in general, increasing speed will increase windage losses.

Even in compressors that use magnetic bearings, mechanical bearings are not eliminated from the design. To ensure the shaft is supported in the event that there is an inability to control levitation (power or component failure) the compressor is equipped with a mechanical bearing called a touchdown bearing.

In summary, the AHRI ratings include bearing losses. No additional benefit should be assumed.

OIL-FREE CHILLERS AND THE IMPACT OF OIL ON HEAT TRANSFER

The net result of oil on chiller efficiency is a fraction of a fraction of a fraction, essentially equivalent to 0.1% to 1% (ignoring the wetting benefit of oil).

ASHRAE (American Society of Heating, Refrigerating, and Air-Conditioning Engineers) Research Projects 601 and 751 pertain to oil concentration and heat transfer. Using the value from RP 751 to determine the impact on the full chiller, the net result of oil on chiller efficiency is a fraction of a fraction of a fraction, essentially equivalent to 0.1% to 1% impact depending on load.[4]

NOT IMPACTED BY OIL

Rw	0.000469	46%
Rf	0.000250	24%
Rm	0.000029	3%
Rr	0.000277	27%
Rt	0.00011025	100%

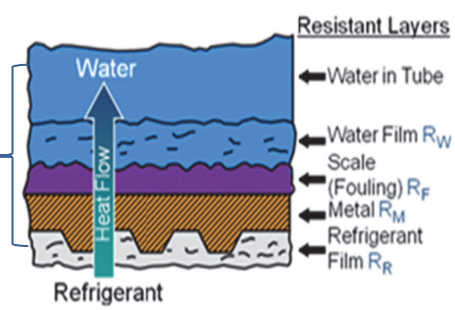


Figure 3 – Impact of Oil on Heat Transfer is Minimal

[2] The "other" in the compressor and other category represented miscellaneous small refrigerant side pressure drops and bypass in the system piping. The compressor mechanical inefficiency alone was over 50% of the chiller's total kW losses.

[3] When operated at AHRI conditions, the lowest L10 life of any bearing in a 23XRV compressor is over 50 years.

[4] Measurements in actual full bundle tests show oil concentration is greatest near the liquid level. At the top of the bundle, oil may improve heat transfer by wetting the tube surface. In the lower parts of the bundle the oil concentration is less than 0.5%, too low to impact heat transfer.

These works are quite technical and it is important that references to these publications are cited properly. For example, Research Project 751 references a “22-25% reduction in refrigerant side heat transfer coefficient due to average oil effects.” This should not be misinterpreted as a 22 to 25% reduction in chiller efficiency. Research Project 751 deals only with the refrigerant side heat transfer in the evaporator. Electric cartridge heaters were used to create the load. The test bundle rig used an over feed and drain system thereby assuring all tubes were covered in liquid refrigerant, a set up not possible in an actual chiller. In general, quantitative comparisons of oil to oil-free in real bundles may not translate reliably from studies using bundle rigs.

As there was no water in the tubes, there could not be a water film resistance, or fouling. Water film, fouling and metal resistance are the same whether a chiller has oil in it or not. These represent 73% of the overall resistance to heat transfer in an evaporator tube. The remaining 27% resides in the refrigerant film and is the only opportunity for oil to impact evaporator heat transfer. (See Figure 3.) As shown in the pie chart in Figure 2, the evaporator accounts for only 5% of the losses in the system.[5] The impact of oil is therefore 22 to 25% of 27% of 5% of the losses in the chiller.

Another commonly cited reference is ASHRAE Research Project 601. This paper cites oil concentrations in samples taken from 10 chillers on a college campus. These were older chillers, not of current design. Half of the ten chillers were manufactured prior to 1970 and all of the chillers used R-11 refrigerant. Without effective oil management, oil retards the refrigerant-oil vapor pressure in low pressure chillers. It is well understood that old negative pressure chillers commonly had oil recovery challenges. But these are not issues with positive pressure R-134a centrifugal and screw chillers in production today. The inherent improvement in oil recovery due to higher differential pressures associated with R-134a chillers is one of the significant advantages of the modern design. It would be incorrect to imply results from this study are applicable to modern R-134a positive pressure chillers built after 3 to 4 decades of improvements.

[5] The total kW losses in an evaporator were determined by comparing a model of a chiller with a normal approach (leaving chilled water temperature less saturated suction) with a model of a chiller with a zero degree approach (leaving chilled water temperature equal to saturated suction temperature, signifying a perfect theoretical evaporator).

Given the well-publicized explanations of how oil-free design improves efficiency, it would be easy to overestimate the impact in the absence of actual data. As outlined above, the impact is actually quite small. In the case of the variable speed screw chillers, it may be obvious that reducing losses by eliminating mechanical unloaders improves performance. It may also be easy to underestimate the dramatic impact this has on performance.

ELIMINATING MECHANICAL UNLOADERS IMPROVES PERFORMANCE

To better illustrate the compression benefits of pure speed capacity control, it may be helpful to realize the advantages of eliminating mechanical unloaders. A discussion of speed and refrigerant gas flow will provide ample opportunity to observe the impact of mechanical unloaders used to restrict flow.

Refrigerant flows from the evaporator through the compressor and is discharged into the condenser. Along this path, the refrigerant may be compressed, turned and accelerated. Mechanical unloaders serve to restrict, recirculate or bypass flow. Examples of recirculation are hot gas bypass (bypassing the condenser) or hot gas recirculation (bypassing condenser, evaporator, and first stage impeller).

Imagine a pipe with a large volume of fluid moving through it. Now place a restriction in the pipe to reduce flow and observe what happens to power per unit of throughput. In a chiller, inlet guide vanes are used to reduce flow. While reducing flow, the inlet guide vanes in a centrifugal compressor (Figure 4) create a pressure drop of 1% to 13%.

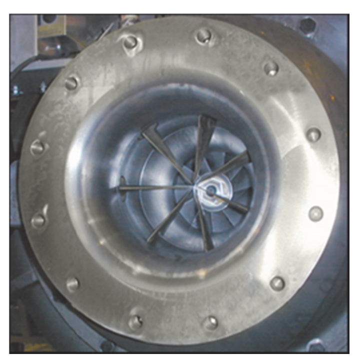


Figure 4 – Inlet Guide Vane Mechanical Losses Increase as the Vanes Close

This increases the head requirements on the compressor thereby increasing power. The actual impact observed in compressor performance is 4% to 50%.[6] Note, the impact is significantly more than the <1% from the oil free effect discussed earlier.

IDEAL FAN LAWS

This section will compare the mechanical unloader approach (described above) to a compressor that uses only speed to control flow. By conceptualizing the ways in which eliminating mechanical unloaders impact efficiency, we will understand the magnitude of the savings. Oil-free centrifugal chillers also employ limited speed control, leveraging the benefit of speed control where possible. However, all centrifugal compressors (regardless of bearing type, refrigerant type, motor type, oiled or oil free) must adhere to the ideal fan laws, illustrated in Figure 5.

Significant marketing has been done to educate the industry on the 3rd law, relationship between speed and power. Unfortunately, the 2nd law, which plays a crucial role in equipment efficiency (and in the ability for the chiller to stay on line), is commonly omitted.

1st Law: Mass flow (capacity) is linear with speed

2nd Law: Lift (ability to generate head) is the square of speed

3rd Law: Power is the cube of speed*

*It may be less than cubic due to the proximity of guide vanes to the impeller inlet

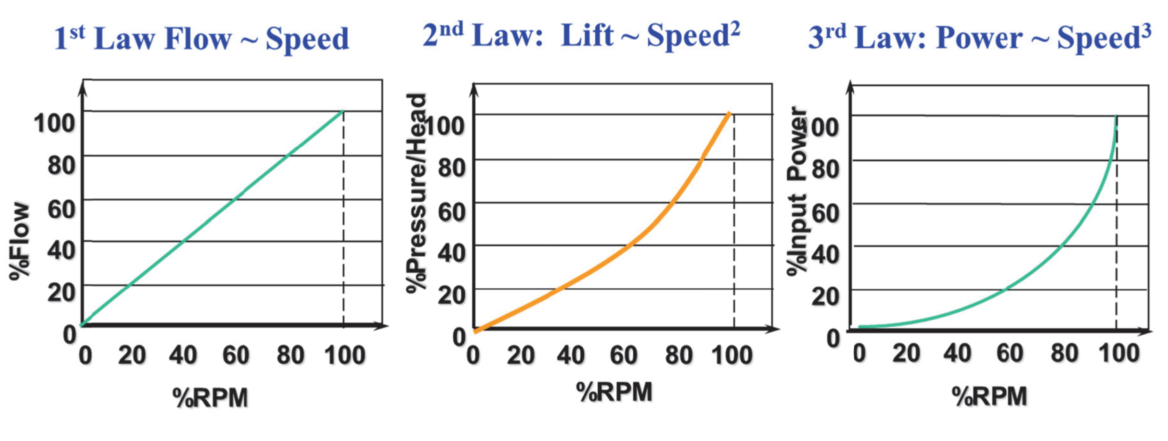


Figure 5 – Ideal Fan Laws

[6] In the paper "Aerodynamics of Rotatable Inlet Guide Vanes for Centrifugal Compressors" 1996, JJ Brasz notes that the impact of inlet guide vanes on efficiency, 4% to 50% in a centrifugal compressor, is far higher than the increase in head alone accounts for. This observation suggests that changes in aerodynamic efficiency due to changes in the operating point on the compressor map are also a factor.

In simple terms, the ability to generate head in a centrifugal compressor is related to the square of the speed. At 80% speed, a centrifugal will generate 80% x 80% = 64% of its design head. If the operating condition requires more than 64% of design head, the centrifugal compressor will surge.[7]

To avoid surge, the centrifugal compressor must either speed up or engage hot gas bypass (also called load balance valves or recirculation flow on many magnetic bearing machines). So while we may be inclined to believe that compressor speed is typically based on capacity, it is actually often based on the head required. For example, if a chiller is operating at 50% load and 75 F entering condenser water, the first law would suggest we could operate at 50% speed, but the 2nd law requires us to run at 84% speed. (See Figure 6.)

This raises an interesting question: If the centrifugal compressor must operate at 84% speed to develop sufficient head, how do we limit the capacity to 50%? This requires an unloading device such as inlet guide vanes, discharge flow restrictions, load balance valves (hot gas bypass) or some other means of diverting or restricting flow. Thinking back to our simple pipe example, it becomes obvious that inlet guide vanes will increase the energy consumption and reduce efficiency. If guide vanes or variable geometry diffusers are removed, hot gas bypass or recirculation flow will be required. Hot gas bypass or recirculation flow allows higher flow to move through the compressor than is needed to meet the load. This flow bypasses the evaporator (hot gas bypass) or the condenser, evaporator and first stage of the compressor (recirculation flow). The net result is the compressor is compressing more refrigerant than is required to meet the load, which represents an increase in power per unit of cooling at the operating point.

$$\%\ \text{Lift} = \frac{\text{Entering Condenser Water Temp}^{\text{Actual}} + \text{Condenser } \Delta T^{\text{Actual}} - \text{Leaving Chilled Water Temp}^{\text{Actual}}}{\text{Entering Condenser Water Temp}^{\text{Design}} + \text{Condenser } \Delta T^{\text{Design}} - \text{Leaving Chilled Water Temp}^{\text{Design}}}$$

$$\%\ \text{Lift} = \frac{75F + 5F - 44F}{85F + 10F - 44F} = \frac{36F}{51F} = 71\%$$

$$\textbf{\% Speed} = \sqrt{71\%} = 84\%$$

Figure 6 – Determining Compressor Speed Required to Develop Sufficient Head

7 Centrifugal compressors move gas through an open path between the low pressure evaporator to the higher pressure condenser. If the pressure difference (head) is too large, flow reversal will occur (surge).

Warm Condenser Water Operation

Now consider this same operating point with a screw chiller. Screw chillers provide the same amount of head at any speed and are not bound by the ideal fan laws. So if the screw chiller needs to provide 50% capacity at 75 F entering condenser water, it will essentially run at 50% speed. This represents 34 points of speed reduction relative to a centrifugal chiller. To understand how the physics favors the screw compressor just remember: Any speed reduction is good speed reduction and more is better. Figure 7 illustrates this principle by comparing centrifugal and screw performance at varying loads with constant entering condenser water temperature. The variable speed screw chiller with pure speed capacity control reduces speed as the load is reduced even with constant 85 F (29.4 C) entering condenser water temperature. Efficiency increases until the chiller is less than half load.[8] While the centrifugal chiller enjoys the same reduction in load, because speed reduction is limited by the 2nd ideal fan law, it must employ mechanical means to unload, and the decrease in efficiency is observed (higher kW/ton represents more power consumed for a given cooling output). In the graphic below it is obvious that mechanical unloaders are required and therefore in warmer climates pure speed capacity control screw chillers will provide an enormous efficiency advantage.

Any speed reduction is good speed reduction and more is better.

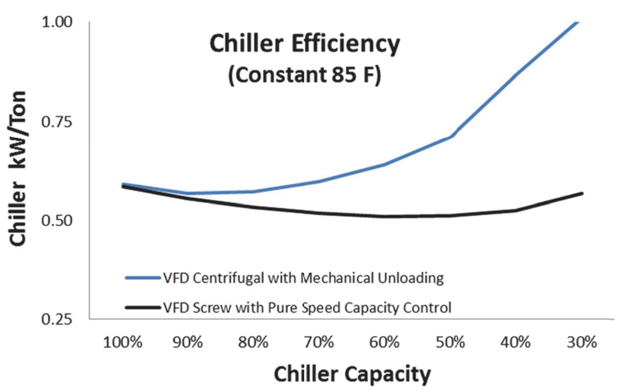


Figure 7 – Chiller Efficiency at Constant 85 F Condenser Entering Water Temperature

[8] Maintaining high efficiency above 40% is highly valued in multiple chiller plants since only the last chiller on line is expected to operate below 40% load and this only occurs at a building loads below 40% x 1/N, where N is the number of chillers needed to meet building full load (for example 13% in a 3-chiller plant).

Cold Condenser Water Operation

Is it possible to compare the two technologies in an operating range where the mechanical unloaders will not come into play? It may seem that a data center application, with chillers running at 100% load continuously, would provide a case where inlet guide vanes, variable geometry diffusers, hot gas bypass and flow recirculation would not be required. Figure 8 depicts chillers operating at 100% capacity over a wide variety of entering condenser water temperatures. Note, that while the data center load may be fixed, the outdoor ambient temperature will vary throughout the data center's 24x7 operating profile; significant hours will be spent at 100% of tons with significantly reduced entering condenser water temperature.[9]

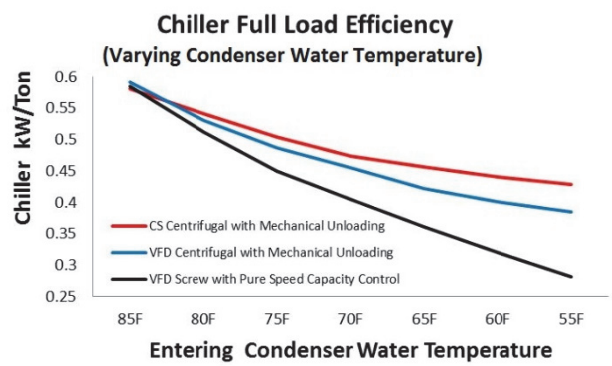


Figure 8 – Chiller Full Load Efficiency at Varying Condenser Entering Water Temperature

A variable speed screw chiller with pure speed capacity control performs at higher operating efficiencies overall, leveraging the reduction in entering condenser water temperature even more than constant or variable speed centrifugal chillers.

[9] Unless there is constant entering condenser water, on the equator for example.

As the condenser water temperature declines, the sub cooler in the chiller becomes more effective thereby improving refrigerant quality. As refrigerant moves through the orifice from condensing pressure to evaporator pressure, some refrigerant will flash to vapor. This vapor will move through the refrigeration cycle, including the compressor, but will not contribute to evaporator tons as it has already changed state. At design conditions, the compressor may be moving a volume of refrigerant of 120% (100% due to boiling in the evaporator, and 20% due flashing across the orifice). At lower entering condenser water temperatures the condenser and sub coolers in chillers provide colder denser refrigerant liquid to the orifice. The result is less flashing across the orifice. The compressor may now be moving a volume of 110% (100% due to boiling in the evaporator, and 10% due flashing across the orifice). While the chiller capacity remains constant at 100%, the compressor is actually unloading from 120% to 110% of mass flow from design head to a reduced head condition. The compressor mass flow required to achieve 100% chiller capacity has been reduced.

But even as the speed needed for mass flow drops, the speed required to generate head declines by a wider margin. Therefore, as the entering condenser water temperature is reduced, the 2nd ideal fan law becomes irrelevant. Since the 2nd ideal fan law is not at play, further insight as to the impact of mechanical unloaders becomes visible when comparing the constant and variable speed centrifugal curves. Speed is now dictated exclusively by mass flow and the impact of a centrifugal operating with excess tip speed can be observed. The impeller will impart energy onto the refrigerant flow throughout the entire length of the blade, from center to tip. The faster the impeller spins, the more energy is transferred, as illustrated in Figure 9 below. Since the impeller geometry is fixed and the speed is greater than required to satisfy the 2nd ideal fan law, the impeller blade will impart more energy on the gas flow than is required for head at the operating point. The centrifugal is over compressing and performing more work than required.

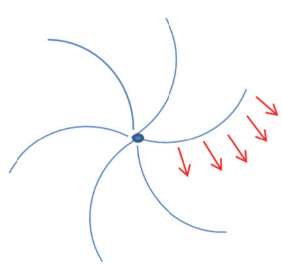


Figure 9 – Impeller Blade

Screw compressors develop the same amount of head at any speed, and so by definition are also over compressing at the high capacity, low head operating points. What may be unexpected is that the variable speed screw with pure speed capacity control performs at higher operating efficiencies overall, leveraging the reduction in entering condenser water temperature even more than constant or variable speed centrifugal chillers. In this operating scenario, the performance advantage cannot be attributed to the mechanical unloaders alone. Both the screw and centrifugal compressors are over compressing. The explanation is beyond the scope of this paper, but in simple terms, the resulting isentropic efficiency of the screw compressor is higher than the aerodynamic efficiency of the centrifugal compressor at the high flow, low head conditions.

The Ideal Centrifugal Speed and Less Than Ideal Operating Conditions

The further the operating point is from the ideal speed curve, the greater the impact will be.

The ideal fan laws define a minimum speed requirement for flow and a separate minimum speed requirement for the generation of head in a centrifugal compressor. Earlier, we observed that when the speed required for head exceeded the flow requirements, mechanical means were required to unload a centrifugal compressor, which resulted in lower efficiency than a screw compressor using speed control alone. Next, in the case of the data center, we discovered that the speed required for flow often exceeds the speed necessary to generate head. In this case, both compressor types over compress, but the screw compressor over compresses with less penalty.

Combining these observations, it becomes apparent that the ideal speed for a centrifugal compressor occurs when the speed required for mass flow exactly matches the speed required to generate head. Above this speed, the high head forces the centrifugal to operate at higher speeds than needed for capacity and mechanical unloaders are employed. Below this speed, the speed for mass flow exceeds the head requirements resulting in over compression. In either case variable speed screw chillers with pure speed capacity control operate at an advantage. The further the operating point is from the ideal speed curve, the greater the impact will be.

Consider over sizing for example. It may be common practice to oversize the chiller by 10 to 15%, but this may come at a price. At peak load, chillers would be operating at 85 to 90% load[10] but the entering condenser water would be at design. The net effect is that the operating point moves to the left from A to A1 (Figure 10). The chiller capacity has been reduced, but the head required remains very close to design.[11] Without head reduction, further speed reduction is severely limited and mechanical unloaders must be used. The more the chiller is oversized, the bigger the impact will be. This effect will be observed until the building load allows a chiller to be taken off line.

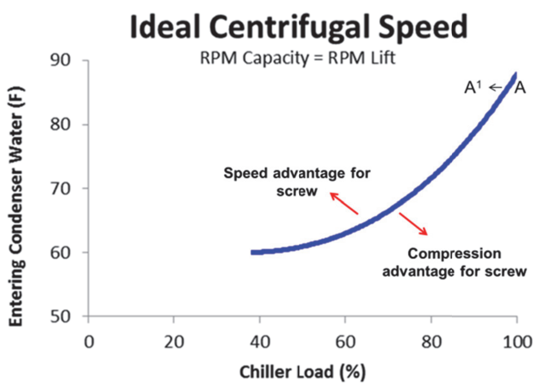


Figure 10 – Ideal Centrifugal Speed

Once a chiller is taken off line, the remaining chillers will ramp up to high load. These chillers would therefore be running at 100% load (subject to low delta T and other operating realities). Since the building is at part load, it is expected that the condenser water temperature has dropped. Ironically, the chiller went from operating at a point where the speed required to generate head exceeded the speed required to generate flow (thus requiring the use of mechanical unloaders) to an operating point where speed required for flow exceeds the speed required for head (thus resulting in over compression).

THE MISSING FACTOR IN COMMON INDUSTRY METRICS

The reality of actual operation is much less elegant than IPLV, resulting in operating points where the use of mechanical unloaders or over compression are more likely to occur.

The practice of oversizing chillers provides a good example of how the most common metrics used to compare chillers may not reflect the differences between centrifugal and screw compressors. The two most commonly referenced metrics are full load kW/ton and IPLV (integrated part load value). If the chillers are oversized, the submittal full load point does not accurately depict actual operation. However, this reality is not acknowledged and the differences between compression types will not be seen in the submittal data. The IPLV metric considers four discrete operating points:

Load	Entering Condenser Water Temperature	Weight
100%	85F	1%
75%	75F	42%
50%	65F	45%
25%	65F	12%

These four points assume the entering condenser water temperature is dropping with load. From an ideal fan curve perspective, this operating profile allows speed requirements for flow and head to decline together reducing the need for mechanical unloaders or over compression. If chillers ran only at these four discrete operating points this would be appropriate, but this is rarely the case (if ever). As shown in the above chart the assumption is that at 12% and 45% of the time, the chiller will be operating at 65 F. The weather in warmer climates will not support 65 F entering condenser water temperatures for 57% of the ton hours.[12] Cooler climates will experience colder condenser water temperatures even at high loads while air and water side economizers reduce the number of chiller operating hours at low loads. AHRI 550/590 advises that IPLV "was derived to provide a representation of the average part load efficiency for a single chiller only."[13] Most chilled water plants employ more than one chiller to meet the load. The manner in which chillers are sequenced in multiple

[10] If chillers are commonly oversized, emphasizing or prioritizing full load efficiency seems illogical.

[11] Technically the head dropped by 1.5 F due to the reduction in delta T across the condenser water at 85% load versus of 100% load.

[12] AHRI 550/590 recommends local weather to be used when conducting a life cycle analysis.

[13] AHRI 550/590 section D.2 further explains that a more comprehensive tool should be used for life cycle analysis.

chiller plants will typically result in 2/3 of the ton hours occurring in the 75% load range. The IPLV rarely depicts an actual chiller's load profile because most chiller plants have more than one chiller and the local weather likely does not match the standard AHRI profile. The reality of actual operation is much less elegant than IPLV, resulting in operating points where the use of mechanical unloaders or over compression are more likely to occur. This is because the actual operating points are further from the ideal speed curve than the operating points in the AHRI IPLV load profile. While the VFD screw chillers with pure speed capacity control already possess industry leading IPLVs, the benefits observed in actual operation is expected to be even greater.

Consider for example a three-chiller plant. The operating profile of chillers 1, 2 and 3 are represented on the chart in Figure 11 by the blue, yellow and black lines.

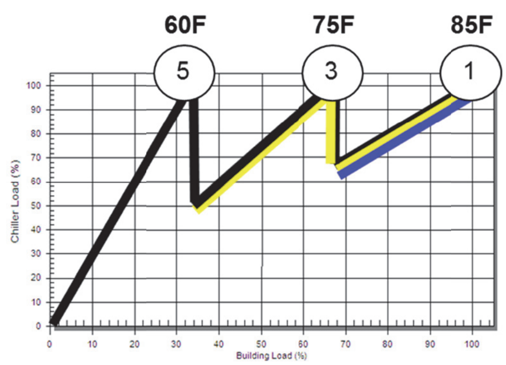


Figure 11 – Chiller Operating Profiles

We observe that when the building has unloaded to 67% (two thirds load) the needed capacity can be met by operating just 2 of the 3 chillers. After the blue chiller is turned off, the yellow and black chillers will load up to 100% to meet set point.[14] While the chillers are at 100% load, the building itself is just 67% loaded, so the entering condenser water temperature would not be expected to be 85 F as listed as the 100% load point on the submittal. For this example we will assume 75 F. This occurs again when the building operates at 33% load and thus requires only one chiller. Comparing these 100% load points to the AHRI submittal reveals that not all 100% load points are the same.

Operating Condition	Load	Entering Condenser Water Temperature	Design Flow	Design Head
AHRI Condition	100%	85 F	~ 100%	~ 100%
Actual Condition (#3)	*100%*	*75 F*	*~ 100%*	*~ 80%*
Actual Condition (#5)	*100%*	*65 F*	*~ 100%*	*~ 60%*

The screw chiller further outperforms the centrifugal chiller at operating points 3 and 5, but this benefit will not be seen in either the submitted full load point or the IPLV.

The AHRI point will fall very close to the ideal speed curve for centrifugal compressors. The actual operating point does not, as the speed needed for flow and head are now quite different. In this case over compression occurs, impacting chiller efficiency as seen previously on the Ideal Centrifugal Speed graph in Figure 10. The screw chiller outperforms the centrifugal chiller at these operating points, but this benefit will not be seen in either the submitted full load point or the IPLV. It should be noted, that far more ton hours are expected at operating points 3 and 5, than at operating point 1. This observation raises the question as to why so much emphasis is placed on the full load submittal data when a majority of the high load operation hours occur at off peak temperatures. In addition, if we accept the idea that chillers are commonly oversized, most chillers will experience zero hours of operation at 100% load with design entering condenser water temperature.

The BIN Method's Unintended Impact

The impacts of local weather and staging are large enough that it is recommended that when hourly analysis cannot be performed, the IPLV equation be modified such that temperatures and weightings reflect local weather and chiller staging. Most manufacturers have simple modeling programs that will provide these values based on your building load profile, use of economizers, etc. It is important to recognize that IPLV or modified IPLV is a BIN metric, and BIN metrics use average values. Using

[14] Subject to low delta T issues and other operating realities.

average values can be convenient but it can also oversimplify. Consider a spring day with a cool morning and a warm afternoon. In the morning the building requires some heat and in the afternoon some cooling. On average, the building requires neither heating nor cooling but you should not expect your utility bill to reflect this average. The same is true for BIN data.

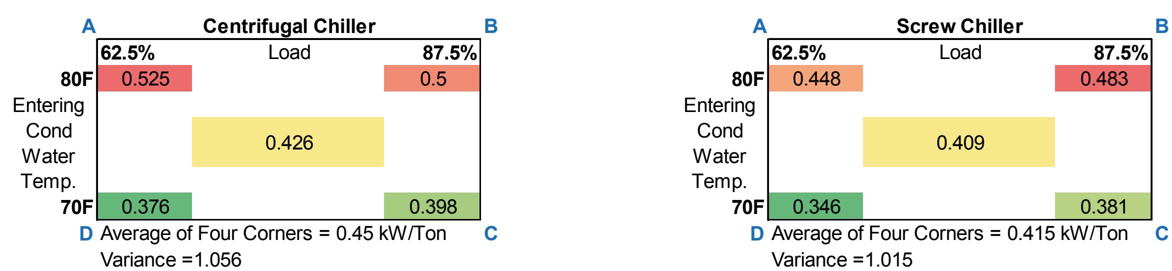
Within each load BIN, chillers operate over a wide range of conditions. On average, the chiller may be operating at 75% load with 75 F entering condenser water temperature. In actual operation, the loads and condenser water temperatures vary above and below the average. Figure 12 compares chillers of similar full load kW/ton to evaluate the variance between the 75% point published and the surrounding operating points. We find that IPLV uses the average operating point, not the average efficiency for the range of operating points.

For the variable speed centrifugal chiller evaluated, the average of operating points A,B,C,D is 5.6% less efficient than the rating point. For the variable speed screw chiller evaluated, this variance is just 1.5%. These results are consistent with the compressor behavior presented earlier. For centrifugal chillers operating at warmer temperatures, the head limits speed reduction and mechanical unloaders must be used. At colder temperatures, the centrifugal compressor is over compressing. The average efficiency resulting from the use of mechanical unloaders and over compression is not equal to the IPLV rating point, but is actally 5.6% lower!

Earlier we presented AHRI certified data showing that oiled screw chillers have better IPLVs than centrifugal chillers. Here we see that in actual operation, the advantages are likely larger than the full load and IPLV metrics would indicate. As we move from IPLV to actual operation, the impact of averaging alone accounts for a large variance in operation. This observation leads to the following question: What other common issues experienced in actual practice impact chiller performance, and do they impact screw chillers differently than centrifugal chillers?

The Counterintuitive Energy Retrofit Case

A facility manager once boasted that his load had been significantly reduced by installing a new roof, new lighting and other changes. Based on his new lower cooling demands, he intended to upgrade his chillers to new units that specifically featured great efficiencies at part load. The solution was counter intuitive. The chiller that had the higher kW/ton (worse) values listed at the 50% and 25% load points at AHRI operating conditions was actually better. Reviewing in more detail we find that while he lowered the cooling demand, he could not lower the outdoor wet bulb. The AHRI IPLV efficiencies listed at 50% and 25% load are based on 65 F entering condenser water. His new chillers would be operating at these new lower loads, but with the warmer condenser water temperatures normally associated with chillers operating at higher loads. In essence, he had shifted his operating profile, creating oversized chillers. Only after accounting for both load and condenser water temperature did the answer become clear. Centrifugal chillers operated at high speed due to head requirements and used mechanical unloaders to get to 50% and 25% load. Screw chillers could use speed control alone, even with high entering condenser water temperatures, yielding true energy savings.



IPLV reflects the average operating point, but the numerical average of operating at points ±5ºF and ±12.5% capacity above and below the IPLV point can be 5.6% worse for centrifugal chillers.

Figure 12 – Comparison of Actual Operating Loads to Average Loads

Warm Climates

In warmer climates, the speed of a centrifugal chiller is more likely to be dictated by the head required. This increases the use of mechanical unloaders and increases the expected relative operating savings of using variable speed screw compressors with pure speed capacity control.

CHILLER PLANT OPERATION: IDEAL vs ACTUAL OPERATION

A typical life cycle analysis often assumes that a chilled water plant will not only achieve ideal operating conditions, but sustain them over a 25-year period. Ignored are the less than stellar maintenance practices, wear and tear, imperfect controls, low delta T syndrome, the fact that the equipment is oversized by 10 to 15% and, of course, unusual override requirements. What impact will these factors have on the chiller operating condition? The operating point is moved further from the ideal speed curve of a centrifugal chiller. As we move away from this point, the difference in compression matters more.

An interesting exercise to demonstrate this impact would be to perform a sensitivity analysis on a life-cycle evaluation. Change the simulation assumptions to reflect an actual load of 85% of the nominal chiller nameplate capacity. To simulate poor maintenance, use a tower approach of 10 F instead of 7 F. The results will show the effects of actual operating conditions on chiller performance.

Performance Degradation over Time

The concept of performance degradation over time occasionally sparks some debate. Some have suggested oiled chillers may worsen over many years of operation due to an assumed increase in oil concentration. There is no evidence to support this occurrence on modern positive pressure chillers. With respect to oil concentrations in the field, an ASHRAE subcommittee noted that, "the data is extremely sparse and statistically insignificant."[15] Any nominal impact of oil is already included in the AHRI submittal data and reflected in the approach temperature both on the submittal and in test results.

Despite the seemingly large numbers often referenced in claims of oil impact on heat transfer, the actual impact of oil is a fraction of a fraction of a fraction, totaling just 0.1 % to 1% of the total losses in a chiller, as shown earlier in Figure 3. Oil-related losses are included in the AHRI certified performance. Comparisons of AHRI submittals reveal oiled chillers may have lower approach temperatures than oil-free chillers. Much is made of the impact of oil, but it is machine design and heat exchanger surface area that drive performance.

The approach temperatures of chillers and the instances requiring corrective action are commonly logged. Common causes of increasing approach temperatures are tube fouling, low refrigerant charge or issues with the metering device between the condenser and evaporator. Low refrigerant levels may contribute to oil accumulation in the evaporator and therefore, oil is often blamed; however, the actual cause of the change in performance is the low refrigerant charge. If the refrigerant charge level is adjusted to the appropriate level, the approach temperature will decrease. If all of the oil is removed from the evaporator and the refrigerant level is not corrected, the approach may actually get worse because the refrigerant level is still not covering the tubes and without the wetting benefit of oil, further degradation may occur.

Rather than focus on any one hypothetical cause of performance degradation for marketing gain, we should instead recognize that the maintenance of high performance variable speed chillers is more valuable than ever. Poor maintenance practices of many kinds have a common symptom. The head required of the compressor will rise. Poor tower maintenance, poor water treatment, skipping tube brushing, low refrigerant charge and many other issues all result in higher approach temperatures which increase the head on the compressor. In the case of centrifugal compressors, raising the head will raise the speed, and now the poor maintenance practice has elevated the power by the cube. A variable speed screw chiller speed is largely unaffected as lift increases. So while power will also increase, it is less sensitive to such issues.

[15] RTAR from Subcommittee 8.2 "Oil Concentration of Field-Installed Chillers with Flooded Type Evaporators."

SUMMARY

Screw compressors are not bound by the ideal fan laws and are not subject to surge, so speed can be precisely matched to the operating point without the use of mechanical unloaders, hot gas bypass or flow recirculation. This provides a significant compression efficiency advantage, improving the component that accounts for over 50% of the total losses in a chiller.

Not all components and subsystems influence chiller efficiency equally. Oil-free chillers seek to reduce refrigerant film resistance associated with boiling. However, evaporator losses represent just 5% of the total losses in a chiller, with refrigerant film accounting for just roughly a quarter of all evaporator losses. The vast majority of evaporator losses (water film, fouling and metal resistance) do not change whether the chiller is oiled or oil-free. Despite the seemingly large numbers often quoted with respect to the impact to refrigerant side heat transfer coefficient, the resulting benefit is between 0.1 and 1% depending on load (ignoring the wetting benefit of oil).

COMPARISON CHART

Oil-Free Magnetic Centrifugal	Pure Speed Control Screw
Oil-free seeks to improve heat transfer	Pure speed control improves compression
Refrigerant side of evaporator represents <1 ½ % of all chiller losses	Compressor represents >50% of all chiller losses
Oil-free benefit is <1%	Compression benefit can exceed 40%
Bound by Ideal Fan Laws	Free of Ideal Fan Laws
Surge line worsens exponentially as speed is reduced	Surge not possible
To avoid surge, mechanical unloaders are required to reduce load	Mechanical unloaders eliminated
Inlet guide vanes introduce 1-13% pressure drop at impeller inlet	Inlet guide vanes not required to unload
Hot Gas Bypass ("load balance valves", "flow recirculation") waste energy, often employed well above minimum load	Hot Gas Bypass (of any kind) not required above minimum load
Bearing "friction" losses exchanged for new added bearing control losses	Eliminates losses due to mechanical unloaders
Good IPLV, relative performance declines further from Ideal Speed Curve	Best in Class IPLV, performance advantage increases away from Ideal Speed Curve

In contrast, variable speed screw chillers garner high efficiency by focusing on the compressor itself, where over half of the losses in a chiller reside. Centrifugal compressors (oiled or oil-free) may need to operate at higher speeds to prevent surge, requiring inlet guide vanes (flow restriction), hot gas bypass or flow recirculation (compressing more refrigerant than needed) to reduce capacity for the operating point. Screw compressors are not bound by the ideal fan laws and are not subject to surge, so speed can be precisely matched to the operating point without the use of mechanical unloaders, hot gas bypass or flow recirculation. This provides significant compression efficiency advantage, improving the component that accounts for over 50% of the total losses in a chiller.

While screw chillers often provide better efficiency at common industry metrics like IPLV, the benefit is likely much higher in reality. The IPLV load profile reduces head and capacity in tandem, limiting the need for mechanical unloaders, hot gas bypass or flow recirculation at the four points that comprise the metric. The reality of actual operation is less elegant. Chillers operate at a wide variety of head and capacity conditions. Most chillers are installed in multiple chiller plants that are unlikely to follow the IPLV load profile (and weighting) due to the staging (on/off) of chillers as the building load is reduced. Furthermore, the weather profiles in most locations will not match the IPLV assumptions. Warmer weather results in the higher head conditions that require centrifugal chillers to operate with mechanical unloaders and their resulting losses. Cooler weather may contribute to over compression. The average of over compression and the use of mechanical unloaders is not equal to operating without either.[16] The further off of the IPLV load points, the more significant the impact. As we consider the impact of actual operation due to everyday occurrences such over sizing chillers, energy upgrades such as lighting and insulation, changes in occupancy or use, varying weather, and less than ideal maintenance and operation, it becomes a great comfort that variable speed screw chillers with pure speed capacity control operate with such consistency.

We place great value on people and tools we can count on, not just on their best day, but when put under pressure. The same can be said of mechanical equipment. Chillers should be evaluated in totality and in recognition of our imperfect chilled water plants.

[16] IPLV uses average operating points, but the numerical average of operating ±5ºF and ±12.5% capacity above and below the 75% IPLV load point can be 5.6% worse for centrifugal chillers. In contrast, screw chillers operate consistently, with the numerical average falling within 1.5% of the IPLV point.

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