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The Relative Benefit of Variable Speed Operation for Various Centrifugal Compressor Concepts Used on Water-Cooled Chillers

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Summary
The performance characteristics of the three most common compression concepts used with centrifugal compressors on water-cooled chillers, viz. single-stage vaneless, single-stage vaned and two-stage vaneless are compared with fixed-speed and variable speed drives in order to determine the relative benefit of VFD for each of these compression concepts. The analysis presented in this paper shows that adding variable speed capability tends to benefit chillers differently depending on the aerodynamic compressor design concept selection, both from a full-load design selection and a part-load operating point of view.

1. Introduction

The HVAC industry developed before the advent of variable-speed motor and drive technologies. As a result HVAC capacity control relies on other methods than speed variation. On-off compressor operation is most commonly used with positive displacement compressors to modulate the capacity of residential and light commercial HVAC equipment. Capacity control of water-cooled chiller systems is traditionally realized through the use of centrifugal compressor variable inlet geometry, viz. rotatable inlet guide vanes. Fixed-speed operation has been the most common practice for compression equipment in the HVAC industry.

The recent drop in inverter prices has caused a dramatic increase in variable speed drives on centrifugal compressors. The question to be addressed in this paper is whether variable speed compressor operation has identical benefits for each of the three centrifugal compressor design concepts originally developed for fixed speed operation or if one of these design concepts profits fundamentally more from variable speed operation. Based on the outcome of this study we will also speculate about the required difference in aerodynamic characteristics of water-cooled chiller centrifugal compressors designed for variable speed versus fixed speed. The answer to this question will guide the development of next generation centrifugal compressors for water-cooled chillers.

The first major entry of variable-speed drive technology in the HVAC industry was in commercial air handling equipment. This application is a perfect match for variable speed operation. The variation in fan head H $(=\Delta P/\rho)$ and fan flow F as a function of rotational fan speed N ($H \sim N^2$ and $F \sim N$) means that variable speed control is optimal for applications where the head is proportional to the square of the flow rate ($H \sim F^2$). For turbulent flow - the condition encountered in air handling systems - this condition is perfectly matched. A fan selected for optimum efficiency of the air handling system at design speed will exhibit optimum efficiency at all reduced speeds. Under those conditions the fan power P is proportional to the to the cube of the flow rate $P \sim HF \sim F^3$.

Thus speed reduction allows substantial savings in fan power during part-load operation compared to the power savings achieved by inlet guide vanes and inlet throttling devices used previously on variable capacity air handling units.

Figure 1. Air handler system head/flow requirements (system load line)

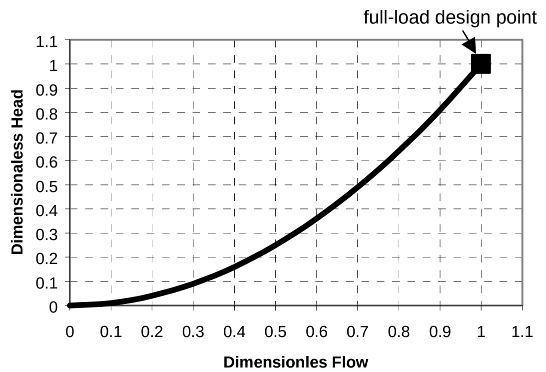
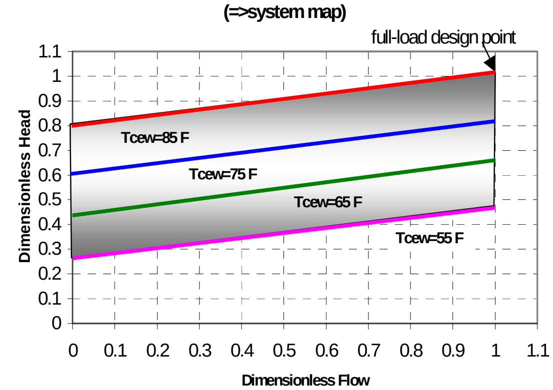


Figure 2. Chiller system head/flow requirements for different entering condenser water temperatures Tecw (=>system map)



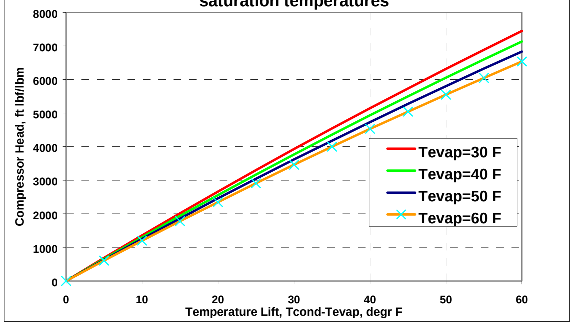
Variable speed operation of centrifugal chillers is not as straightforward as that of air-handling equipment. Water-cooled chillers do not have a single parabolic load line ($H\sim F^2$) like air handlers. The compressor head requirement at part-load flow conditions can vary substantially. The compressor has to operate over a two-dimensional system map as opposed to the air-handling fan that has to satisfy a one-dimensional system load line (see Figures 1 and 2). The required chiller system head depends mainly on the cooling tower return water temperature. The chilled water temperature leaving the chiller stays typically constant under varying load conditions to maintain the dehumidification capability of the air handlers receiving the chilled water. A constant entering condenser water temperature *Tecw* (=cooling tower return water temperature) results in a much higher relative head requirements for the centrifugal compressor under part-load conditions than the parabolic load line shown in Figure 1 that worked so well for variable speed fan operation with air-handling equipment.

Chiller head-flow requirements at part load

With constant water flow applications there is always, even at constant entering condenser water temperature, a certain head relief with reduced capacity due to the reduction in condenser water temperature rise under part-load conditions. For example, if the water temperature rise in the condenser is 10 ^{0}F at full load (say from 85 ^{0}F to 95 ^{0}F) it will reduce roughly in half at 50% capacity (from 85 ^{0}F to 90 ^{0}F). If the condenser refrigerant saturation temperature was 2 ^{0}F above the saturation temperature at full load (97 ^{0}F in our example) it will approach the leaving chilled water temperature (90 ^{0}F in our example) somewhat closer at 50% chiller capacity, ending up at around 91 ^{0}F. In the evaporator (or the “cooler” as it is referred to in the centrifugal chiller world) there is not much difference in refrigerant saturation temperature between full- and part-load since the leaving chilled water temperature is normally kept constant. The only head-relief comes from the fact that the saturation temperature will approach the leaving chilled water temperature somewhat closer at part-load. For example, a 2 ^{0}F leaving temperature difference at full-load might reduce to 1 ^{0}F at 50% chiller capacity.

In this example of constant entering condenser water temperature *Tecw* and constant leaving chilled water temperature at constant water flow rates the difference in condenser and evaporator saturation temperature (sometimes called temperature lift) reduces from 55 ^{0}F to 48 ^{0}F when reducing the capacity in half. This analysis shows a linear reduction in temperature lift with capacity. In our example the lift reduces linearly from 55 ^{0}F to 41 ^{0}F when going from 100% to 0% capacity. Compressor maps plot “head” and not “lift” versus flow rate. How much will the head reduce for a given reduction in lift? It turns

Figure 3. Relationship between compressor head and temperature lift for R-134a at various evaporator saturation temperatures

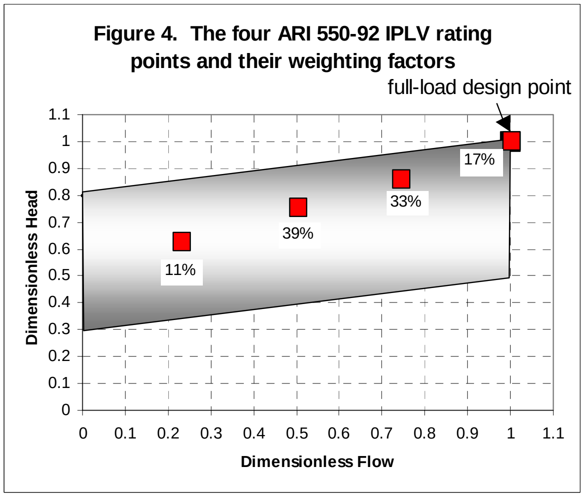


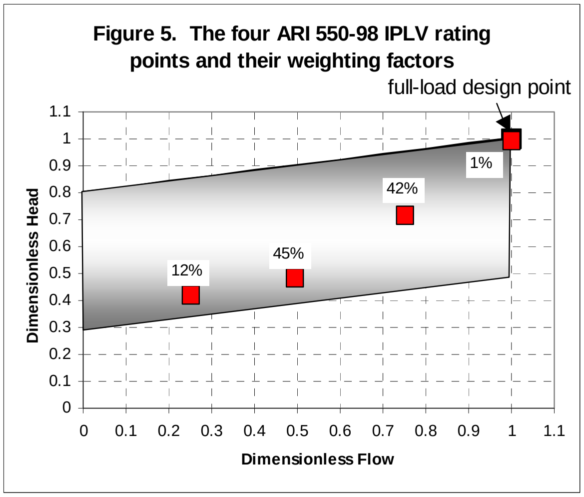
out that for all refrigerants temperature lift is in first approximation directly proportional head as shown for R-134a in Figure 3. The proportionality factor changes slightly with the evaporator saturation temperature. However, this temperature stays within a few degrees for chiller applications. For a given evaporator saturation temperature, temperature lift and compressor head are within engineering accuracy directly proportional to each other.

Since compressor head is proportional to temperature lift, the 23% temperature lift reduction derived earlier will result in a 23% compressor head reduction going from full-load to zero percent capacity. This results in the head flow curves shown in Figure 2.

When used for air conditioning applications, the chiller load requirement is dependent on ambient temperatures. The lower load caused by lower ambient temperatures will often also lower the cooling tower return water temperature, thus causing a stronger reduction in head at part-load than shown by the lines in Figure 2.

In order to characterize the part-load performance of the chillers made by the various manufacturers a part-load system load line has been defined by the American Refrigeration Institute. This load line accounts for the fact that on average the lift at part-load is less than the lift corresponding to a constant entering condenser water temperature. Chiller performance at the 25%, 50%, 75% and 100% capacity points on this load line is averaged using certain weighting factors, representing the relative contribution of operation under these conditions to annual energy consumption. The end result is a so-called IPLV number, or integrated part load value, that can be used to estimate annual operating expenses. The location of this load line on the compressor map and the weighting factors of the rating points are a topic of constant debate and have changed dramatically over time. This is illustrated in Figures 4 and 5. These figures show the location and weighting factors of the four part-load rating points of two successive ARI Standards.





A centrifugal compressor with variable speed capability will perform much better on the rating points of Figure 5 than on those of Figure 4 since head reduction favors variable speed operation.

However, the compressor still has to have a part-load lift capability (or in centrifugal compressor nomenclature a surge line) corresponding to the constant entering water temperature load line since such lift conditions might occasionally be required. Variable speed operation of a centrifugal compressor will not provide this part-load lift capability. The compressor will surge at low flow, high head/lift conditions. Even for fixed-speed, inlet-guide-vane controlled centrifugal compressors this part-load head requirement is hard to realize. Rotating impeller or diffuser stall and compressor-system surge are easily encountered under those conditions and much effort has been directed towards design adaptations preventing these phenomena to occur.

It follows from the above mentioned arguments that, in contrast with air handling capacity control where variable speed control replaces IGV's or inlet throttling devices, speed variation alone can never be used for chiller capacity control. The variable geometry inlet guide vanes, the predominant mechanism to control capacity of fixed speed chillers, are still required with variable speed VFD controlled chillers to reach areas of low flow/high head on the compressor map. Centrifugal chiller variable-speed control is therefore not a replacement of but instead in addition to inlet guide vane control.

The addition of variable speed to variable geometry complicates the control problem. Each point on the head/flow map can be reached by an infinite combination of speeds and inlet guide vane positions. The control objective becomes to optimize compressor efficiency for each point on the map by selecting the appropriate combination of speed and inlet guide vane setting angle. Determining those combinations seems an almost impossible task. However, centrifugal compressors are known to reach their peak efficiency close to surge. Knowledge of the location of the compressor surge line and how it changes with speed variation is the control strategy for compressor performance optimization for all operating points on the original fixed speed compressor map. Optimum efficiency being close to surge means that the surge line location has to be known accurately to prevent unintended surge system trips.

The various compressor design concepts

All existing centrifugal compressors currently in use on water-cooled chillers were designed with fixed-speed compressor operation in mind. The three most common compression concepts used with centrifugal compressors on water-cooled chillers, viz. single-stage vaneless, single-stage vaned and two-stage vaneless

The two fundamental distinctions in aerodynamic design of centrifugal compressors used on water-cooled chillers are

1. the number of compression stages used and
2. the diffuser concept selection.

Both single- and two-stage compressor designs are commercially available. Single-stage compressors are offered with both vaneless and vaned diffusers. The term vaned diffuser is used here for any type of discrete-passage-diffuser, e.g. pipe diffuser, vane-island diffuser, wedge diffuser as well as diffusers with single-thickness and airfoil vanes.

Location of map peak efficiency for different design concepts

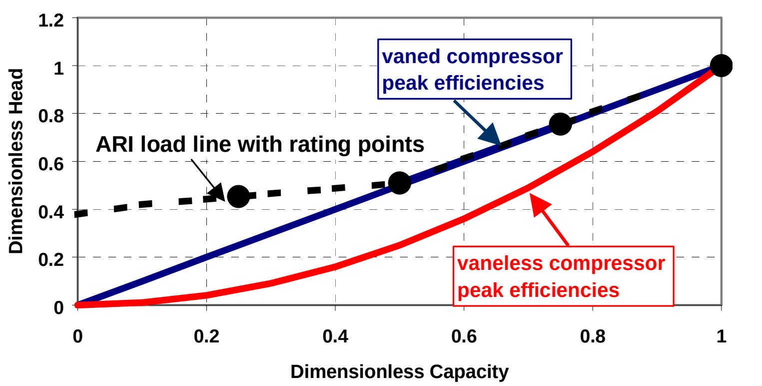
Vaneless diffuser compressors (both single-stage and two-stage) have a performance characteristic that share some similarity with that of a fan. Compressor head H is proportional to the square impeller speed N, while the flow F varies linear with impeller speed: $H \sim N^2$ and $F \sim N$. The locus of optimum efficiency points for different compressor speeds on the compressor map is in first approximation a parabolic curve $H \sim F^2$, similar to the variable speed fan characteristic.

Vaned diffuser compressors show a different flow dependence on speed. Compressor head H is still proportional to the square of impeller speed N. The Euler turbine equation dictates that behavior. However, the flow F varies now proportional to the square of impeller speed. The reason is that vaned diffuser inlet flow angles as opposed to impeller inlet flow angles have to remain constant to guarantee peak efficiency of vaned diffuser compressors. The tangential component of the absolute velocity leaving the impeller is in first approximation proportional to impeller speed. If the radial component also reduced proportional to impeller speed, flow would vary proportional to speed. However, the radial component of the velocity leaving the impeller does not reduce proportional with impeller speed due to compressibility effects. At reduced impeller speed the refrigerant gas leaving the impeller is less compressed and has therefore a smaller density or larger specific volume, causing a relative increase in radial velocity over the velocity reduction proportional to speed that would occur with incompressible fluids. A lower flow rate than the one corresponding to a linear dependence between flow and speed is therefore required for variable speed operation of centrifugal compressors with vaned diffusers. A flow dependence proportional to the square of impeller speed is a good first approximation for vaned diffuser compressors. Since both head and flow are proportional to the square of impeller speed for vaned compressors: $H \sim N^2$ and $F \sim N^2$ the locus of peak efficiency points on the variable speed compressor map becomes a straight line $H \sim F$.

Compressor peak efficiency and ARI load line location

As was shown in Figure 5, the latest ARI standard 90 defined three of the four IPLV rating points with an accumulated weighting factor of 88% on a straight line where head

Figure 6. Matching of compressor peak efficiency with ARI load line



is proportional to flow. This means that the performance benefit of variable speed operation for chillers with centrifugal compressors with vaned diffusers is larger than for chillers with vaneless diffusers. Figure 6 shows how well for vaned diffuser compressors the locus of compressor peak efficiencies matches the ARI load line. Vaneless compressors would benefit from a system load line with steeper reduction in head with reduced flow than specified by ARI standard 90.

Conclusions

1. Although the ARI load line used for the IPLV calculation should only be used as a first estimate of chiller part-load efficiency and a detailed APLV (Applied Part Load Value) calculation is recommended using appropriate chiller system and location weather data, the tendency of variable speed vaned diffuser compressors to have their peak efficiencies at higher heads for the same reduction in flow rate than variable speed vaneless compressors favors the use of vaned diffuser compressors for variable speed applications.

2. Reaching areas on the compressor map above the ARI load line requires higher speed in combination with the use of variable geometry. Vaned diffuser geometry variation is preferable over inlet guide vane geometry variation since it will allow lower impeller speeds (the main efficiency booster) for identical map positions.

3. Variable geometry vaned diffuser concepts that allow choke-free compressor operation at high-flow low-head map locations are required to be competitive with variable speed vaneless compressors at those operating points.