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Variable-Speed Centrifugal Chiller Control for Variable Primary Flow (VPF) Applications

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ABSTRACT

Variable speed operation offers the potential of increased centrifugal compressor efficiency at off-design conditions compared to variable geometry compressor control. However, this advantage of variable speed operation is offset by early (=low head) compressor surge. A combination of variable-speed and variable-geometry control is required to take advantage of the variable speed efficiency improvement without sacrificing low-flow high-head capability. Once two flow control methods are possible, optimal control becomes more problematic, where a change in one parameter may be theoretically beneficial for efficiency improvement but detrimental to stable operation. Understanding the physics and inter-relation of these parameters is critical to proper chiller control, Compounding this issue is the fact that building system control is sometimes optimized by variable primary water flow and the fact that most chiller systems are not equipped with water flow measuring devices, which, with the water temperature change, is the direct way of measuring compressor flow. Conventionally, with the know compressor mass flow, condenser to evaporator saturated temperature difference and water temperature rise, the position on the performance map can be determined, and in the case of chiller control, the margin from surge to the current operating point. This paper describes the aerodynamic reasons for the necessity of a multi-input, multi-output compressor control system with model-based feed-forward element if part-load performance is to be improved through the use of variable speed operation. The model-based feed-forward element of the controller determines as a function of required operating condition (pressure ratio and flow rate) which combination of speed and variable geometry will result in maximum compressor efficiency. Since compressor peak efficiency occurs close to the surge an accurate definition of the surge line as well as a precise measurement of the actual compressor operating point are required. Actual compressor operating conditions are normally determined from head and flow measurements. Compressor head can be calculated fairly precise from suction and discharge pressure measurements through instrumentation already available on the machine. Compressor flow rate is obtained indirectly through a heat balance over the evaporator, requiring knowledge of chilled water side flow rate and temperature drop over the cooler. The water flow rate for the traditional primary/secondary chilled water flow systems installations is constant, making the water-side temperature drop in the cooler the indicator of compressor flow. Since chillers are equipped with entering and leaving chilled water temperature sensors, the actual operating point of the compressor can then be determined without the need of additional water-side flow measurements. For VPF systems, the chilled water flow rate changes necessitates an additional flow measurement, which also has proven to be less reliable and accurate in actual field installation. A new control system that circumvents this problem and consequently is not affected by the variable primary flow changes is described in this paper.

1. INTRODUCTION

As energy cost rises and end users become more sophisticated in there application of cooling on the building system level, variable speed motor and inverter technology is making ever stronger inroads into products that were traditionally operating in fixed speed mode and purchased on first cost considerations only. Equipment life cycle cost reduction has been the driving force behind this trend. With ever increasing energy costs and a focus on low carbon emissions, this trend towards more energy efficient operation but higher initial costs is only more likely to accelerate. With payback reaching 1 to 3 years for these upgrades, the attractiveness of variable speed operation is only greater.

The HVAC industry is experiencing this transition from fixed-speed to variable-speed operation, especially for commercial size equipment. Variable-speed fan/blower operation of commercial air-handling products is currently the standard product offering while every year a larger percentage of chillers have variable-speed compressors. The

centrifugal compressors used on large water-cooled chillers are no exception to this trend. The savings on annual operating cost can be as high as 40% and justify the additional cost of the variable speed option on the chiller.

Variable speed operation has also been proposed for the other elements of the chiller system, i.e. the chilled water pumps, the condenser water pumps and the cooling tower fans. Reduced speed operation of the condenser water pump and the cooling tower fan will result in a higher condenser entering water temperatures and larger condenser water temperature rise over the condenser, thus resulting in a higher condenser saturation pressure of the refrigerant and therefore the power required by the compressor. If the chiller is equipped with a variable speed centrifugal compressor, the best overall system efficiency is normally obtained by getting maximum compressor speed reduction. The energy savings obtained by reduced speed condenser pump operation and reduced speed cooling tower fan operation are normally less than the energy penalty the main compressor will experience under those circumstances.

Variable speed operation of the chilled water flow pump does not suffer from this disadvantage. The leaving water temperature of the cooler is fixed by the user requirements and has to remain constant independent of load in order to enable proper dehumidification. Given this requirement for constant leaving chilled water temperature the evaporator saturation temperature does not change much with load conditions whether the chilled water flow rate has its constant high flow value or varies with load and there is an opportunity to optimize overall power consumption by reducing the primary water flow rate, thus reducing pump power with significantly increasing compressor power.

In traditional primary-secondary (PSS) systems, this was achieved by having a constant speed primary pump forcing a constant water flow rate through the cooler and a variable-speed secondary pump supplying the required (=variable) amount of water flow to the air handling units. A bypass line decouples the primary flow from the secondary flow (see Figure 1).

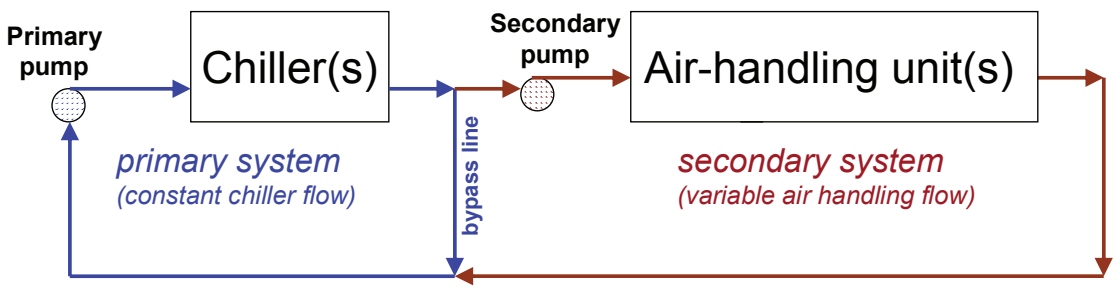

Figure 1. Primary/secondary chilled water flow pumping system

The constant chilled water flow rate through the cooler guarantees optimum water-side heat transfer and prevents excessive fouling of the tubes. Reducing the chilled water flow rate will obviously save on pumping power. The question is: what will be the effect on compressor power?

The effect of having less water flow go through the chiller tends to reduce the water-side heat transfer coefficient. That would cause a drop on evaporator saturation temperature thus increasing the required head the compressor has to deliver and therefore increase compressor power consumption. On the other hand, since for the same chiller capacity this reduced water flow rate means that the water returning from the air-handlers enters the cooler now at a higher temperature, the temperature difference (LMTD) between water and refrigerant increases. This will increase the saturation temperature of the refrigerant in the cooler thus reducing the head the compressor has to deliver. The larger LMTD helps heat transfer more than the lower water-side flow rate hurts it. The main energy savings comes from a reduction in pump power requirement but net effect is a savings in overall plant power consumption. However, there is a minimum value of the chilled water flow rate below which excessive fouling will occur. The industry recommends a minimum water side velocity inside the tubes of 3 ft/s. Depending on the specific chiller selection this allows maximum chilled water flow reductions between 30 to 60% at lower load conditions. It is important not to operate below these minimum chilled water flow rates.

Replacing the fixed speed pump in the primary loop with a variable speed pump also allows the elimination of the secondary chilled water pump that serves the air handlers, thus resulting in a first cost reduction. In order to guarantee the minimum chilled water flow rate the chiller requires to prevent excessive fouling a by-pass control valve is required for low-load operations. Figure 2 shows the basic arrangement of a variable primary flow (VPF)

system including the bypass control valve used to decouple the chiller flow rate from the air-handling chilled water flow rate at low-load conditions.

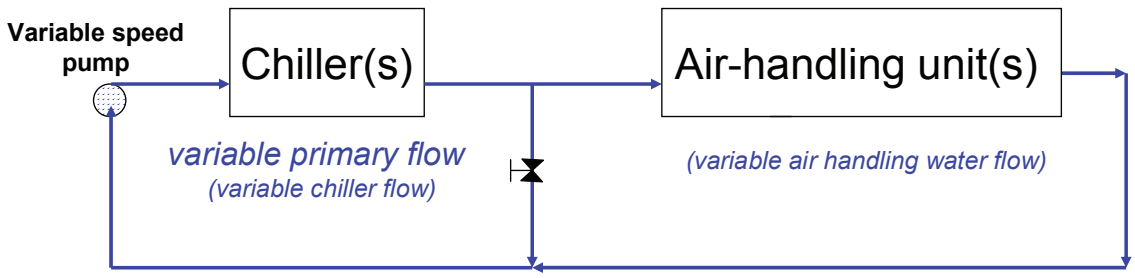

Figure 2. Variable primary flow (VPF) system

2. WORK INPUT LIMITATION OF CENTRIFUGAL COMPRESSORS

One of the main operational distinctions between positive displacement and turbo compressors is that the pressure rise or output head at a certain flow rate and for that matter the work input (change in enthalpy per unit mass) of a turbo compressor, being a dynamic machine, is inherently limited for a given rotor speed. Work input is dictated by the Euler equation:

$$\Delta h = u_2 c_{\theta 2} - u_1 c_{\theta 1} \qquad (1)$$

where Δh = the change in enthalpy [J/kg]
u_1 = the wheel speed at rotor inlet [m/s]
$c_{\theta 1}$ = the tangential component of the incoming flow [m/s]
u_2 = the rotor speed at compressor exit [m/s]
$c_{\theta 2}$ = the tangential velocity of the flow exiting the rotor [m/s]

Multiplying the rise in enthalpy or input head given by Equation (1) with the mass flow rate gives the amount of power required to drive the compressor. This amount of power cannot be exceeded, no matter what the system conditions encountered by the compressor are. In other words, this one-to-one mapping of flow and head means a change of system conditions (head) will force a corresponding change in flow, and therefore a change in power. This feature can be considered an inherent safety advantage of centrifugal compressors over positive displacement compressors.

The other effect of the work input limitation of centrifugal compressors is much better known. Limited work input necessarily means limited work output which for compressors means limited head or pressure ratio capability. All turbo compressors will surge as soon as the imposed head or pressure ratio (for a given IGV setting) exceeds the maximum head or pressure ratio the compressor can deliver (dictated by the compressor-system stability). The compressor will enter an operational mode of large flow and pressure fluctuations with corresponding variations in power consumption. Though surge is protecting the drive system from power overload, the compressor is not functioning properly anymore and unit controls will shut down the compressor after a number of surge cycles to prevent mechanical damage. In comparing the pro's and con's of positive displacement versus turbo compressors, surge-free operation is considered an inherent operational advantage of positive displacement compressors.

Since the work input of a centrifugal compressor is fixed by the Euler equation, centrifugal compressor peak efficiency is achieved close to or at maximum pressure ratio. Fixed speed centrifugal compressors will have excellent efficiency at their design point (close to maximum pressure ratio) but suffer at lower head conditions since

the work input stays essentially constant. The efficiency of positive displacement compressors, being in general somewhat less than that of their centrifugal counterparts at design conditions, tends to suffer less from operation at lower pressure ratio conditions.

A fixed-speed, fixed-geometry compressor has a unique relationship between head and flow, or in chiller terminology: temperature lift (condenser saturation temperature – evaporator saturation temperature) and cooling capacity (see Figure 3). This is a fixed, one-to-one mapping of lift to flow.

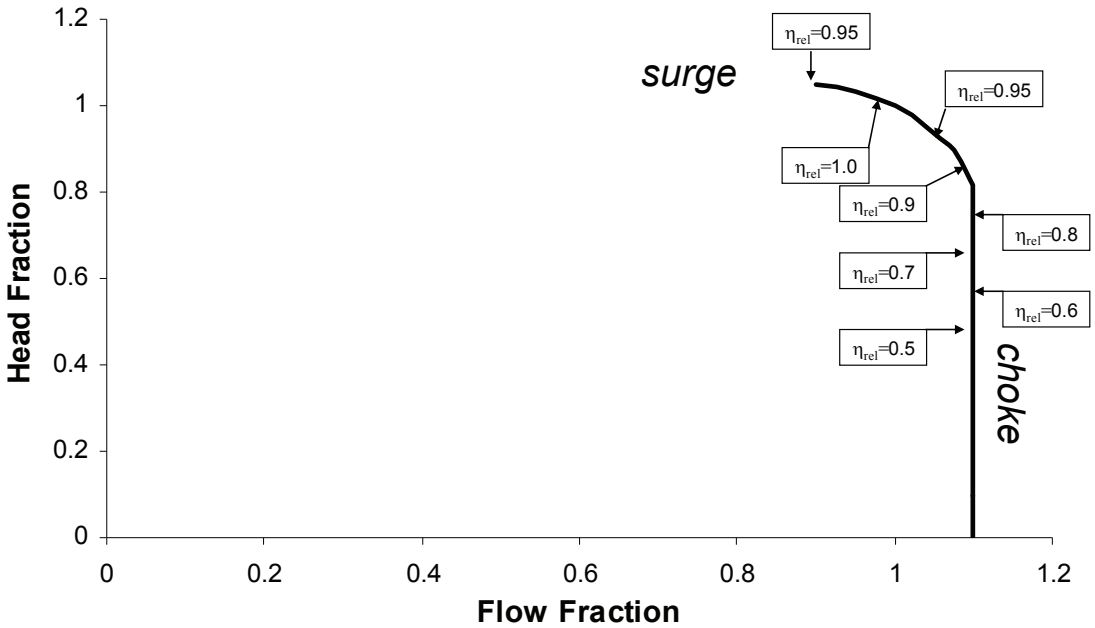

Figure 3. Head/Flow relationship for a fixed-speed fixed geometry centrifugal compressor

3. USE OF VARIABLE GEOMETRY FOR CENTRIFUGAL COMPRESSOR FLOW CONTROL

Variable geometry of stationary compressor components, e.g. inlet guide vanes giving pre-swirl to the flow entering the impeller or diffuser vanes that can rotate and narrow the flow passage, allows capacity control of fixed speed centrifugal compressors. This effect is quantified in equation 1, where a change in the tangential component of inlet velocity reduces the head of the compressor. A set of variable inlet guide vanes is typically employed for centrifugal compressor capacity control on water-cooled chillers. This variable geometry allows the compressor to independently select lift and capacity based on system requirements. The single compressor line relating head to flow is now replaced by a two-dimensional area of possible head/flow combinations, known as the compressor map and shown in Figure 4.

Each possible head/flow combination has a unique inlet guide vane angle position and a unique efficiency. The compressor performance map is limited in capacity by the fully open inlet guide vane performance curve. The surge line of the compressor map determines the maximum head the compressor can achieve for a given flow rate at a particular IGV setting (A fact that will become essential to the VPF control system later described in this paper). The compressor surge line is the connection of individual surge points of the head/flow characteristics for different inlet guide vane setting angles. Lines of various inlet guide vane setting angles (measured from the tangential direction, i.e. 90^0 is fully open and 0^0 is fully closed) can be drawn on the fixed-speed compressor performance map. Note the very nonlinear relationship between inlet guide vane setting angle and flow rate.

The compressor surge line of fixed speed centrifugal compressors has a slope that increases with flow rate. The reason is that at higher flow rates, when the inlet guide vanes are more opened, pre-swirl is created without much throttling action, the net result being an increase in the second term of the right hand side of Equation (1) and therefore a reduction in the compressor work input and as a result a reduction in the head. The head- flow curve moves both to a smaller head and a smaller flow, but the difference between maximum and minimum flow (choke

flow and surge flow), the so-called compressor operating range for that guide vane setting angle, stays more or less constant.

At lower flow rates, the guide vanes become more closed and start - besides giving more pre-swirl - acting like a throttling device. The resistance of a throttling device, if in close proximity to a centrifugal compressor, is known to increase its stable operating range, allowing lower flow rates before surge occurs. This phenomenon shows itself in the shape of the head/flow characteristics of the centrifugal compressor when the inlet guide vanes are more closed. As a result the compressor surge line gets a more horizontal slope at smaller flow rates.

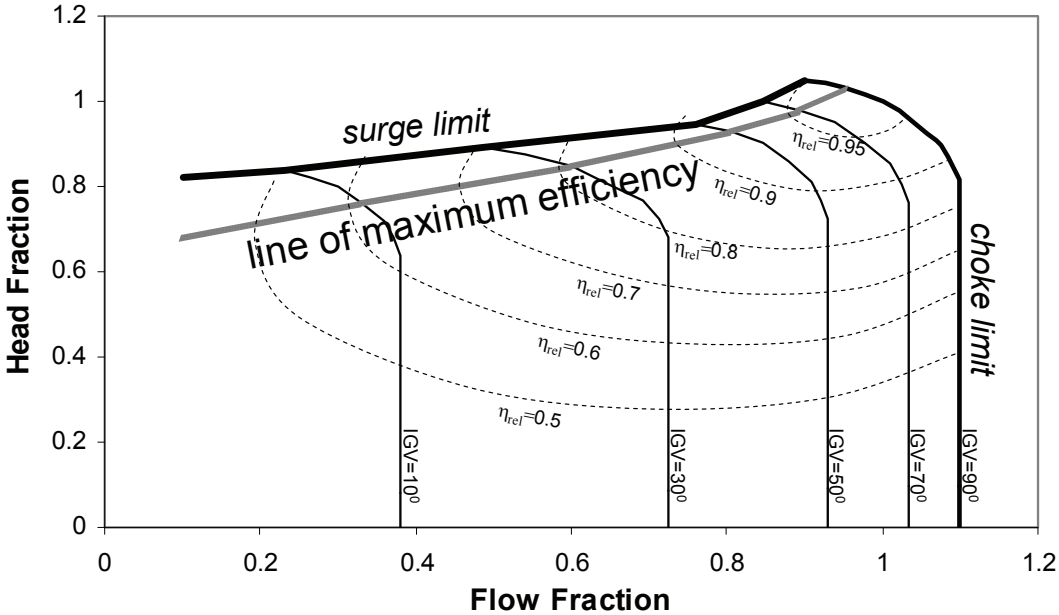

Figure 4. Performance map of a fixed-speed centrifugal compressor with variable inlet guide vanes (IGV's) for capacity control.

The efficiency at the various head/flow combinations is normally of more interest than the corresponding guide vane opening angle. This efficiency is required for the calculation of seasonally-averaged compressor performance under a variety of possible operating conditions. Centrifugal compressor performance maps show lines of constant efficiency, often referred to as efficiency islands as a result of the shape of these contours.

Control of fixed-speed centrifugal compressor with variable inlet guide vanes in actual applications is straightforward. This will be illustrated for the example of a centrifugal compressor on a water-cooled chiller. The function of the centrifugal chiller is the delivery of chilled water of at a given temperature. If the actual leaving chilled water temperature is higher than its set point value, compressor capacity needs to be increased. This is achieved by opening the inlet guide vanes more. Guide vanes will be more closed if less capacity is required which is indicated by a leaving chilled water temperature lower than the set point value. Stable leaving chilled water temperatures with minimal deviation from its set point value can be obtained with the selection of appropriate time constants and controller action in the inlet guide vane feedback control loop.

The efficiency islands shown on the compressor map also indicate what head, for each flow rate, the maximum efficiency occurs. From inspection of the performance map (Figure 4) it can be seen that for fixed-speed inlet-guide-vane controlled compressors the highest efficiency for a given flow rate occurs close to surge. A line connecting the points of maximum efficiency for each flow rate can be drawn on the fixed speed compressor map. This line is constructed by a connecting of the points with vertical slope of the efficiency islands.

The line of maximum compressor efficiency as a function of flow rate is located close to the surge line. The reason for that location is again given by Equation (1). The work input is fixed by this equation. The highest head for the given work input obviously results in the highest efficiency. Lower heads do not need the work input of Equation (1) and will therefore result in lower compressor efficiency. The predicament of the fixed speed centrifugal compressor is that its peak efficiency is limited to a narrow operating range close to its surge line.

4. EFFICIENCY ADVANTAGE AND OPERATIONAL LIMITS OF VARIABLE SPEED CENTRIFUGAL COMPRESSOR OPERATION

Achieving higher compressor efficiency at lower head conditions requires a reduction in work input. Variable compressor speed operation is the mechanism that allows the reduction in work input required to achieve higher efficiency at lower head.

Variable speed operation is more effective in head reduction than in flow reduction creating a compressor map that does not allow operation at low flow high head conditions, needed for many applications. Figure 5 shows a typical performance map of a fixed geometry variable speed centrifugal compressor.

For example, variable-speed centrifugal compressor operation without variable geometry would cause the compressor to surge at low-flow high-head conditions occasionally encountered during water-cooled chiller operation. Head fractions up to 85% of full-load head might be required at low flow conditions. Therefore, adding inverters to achieve variable speed operation of the centrifugal compressors used on water-cooled chillers does not eliminate the need for inlet guide vanes.

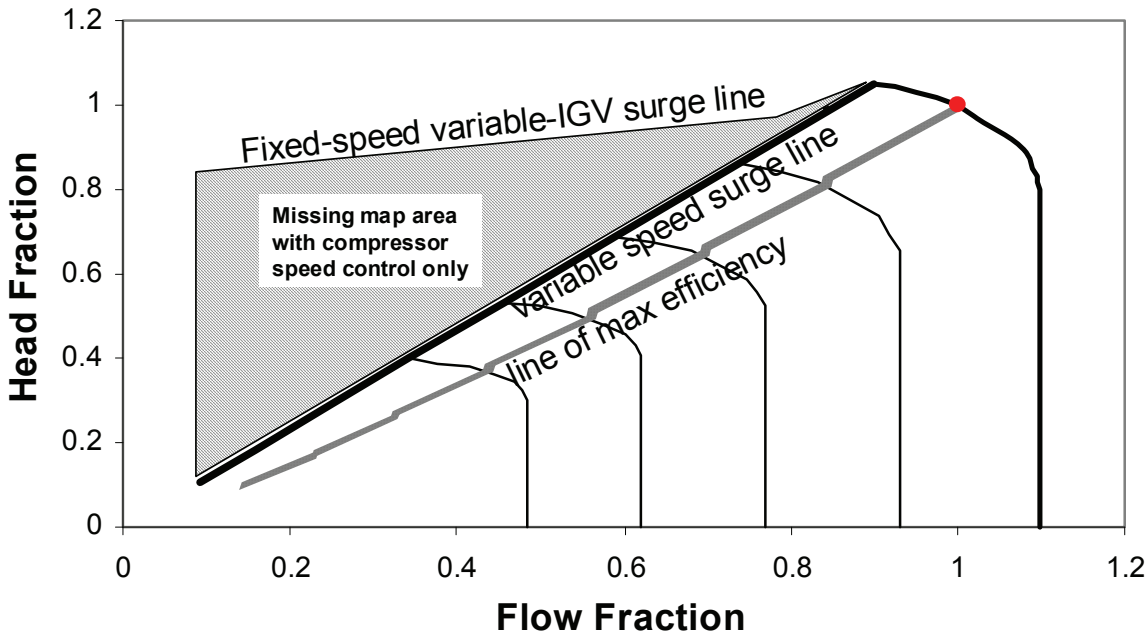

Figure 5. Performance map of a fixed-geometry centrifugal compressor with a variable frequency drive (VFD) for capacity control.

It is important to note that both the surge line and the line of peak efficiency on the performance map of a variable speed centrifugal compressor are almost straight lines. This behavior does not follow the so-called "fan laws" that dictate that flow, F, is proportional to speed, N, and head, H, proportional to speed squared:

$$F \sim N \tag{2}$$

$$H \sim N^2 \tag{3}$$

The fan laws apply satisfactory for pumps and fans (incompressible flow) but do not accurately describe the lower speed performance of compressors. Figure 6 shows test data at different lower speed for 2.5:1 pressure ratio centrifugal compressors. Flow reduces faster with speed and head reduced somewhat slower with speed than indicated by the fan laws. The physical explanation for this deviation is based on the effect of compressibility. The gas leaving the impeller of a 2.5:1 pressure ratio centrifugal compressor has a larger density than the gas entering the impeller due to the change in static pressure with impeller radius as a result centrifugal effect of the rotating impeller. To maintain optimum impeller incidence the impeller inlet flow rate wants to reduce proportional with speed (fan law characteristic). Optimum diffuser incidence would occur if the diffuser inlet volumetric flow rate

reduced proportional with speed as well. The smaller inlet flow rate at reduced impeller speed reduces the exit mass flow rate proportional to speed. However, the reduction in impeller exit density at reduced impeller speed increases the impeller exit volumetric flow rate partly offsetting the reduction in volume flow rate caused by the speed reduction. The net effect is that for optimum diffuser incidence the compressor flow rate has to reduce more than proportional to impeller speed. For the specific compressor shown in Figure 4 flow rate is proportional with speed to the power 1.6. Best approximation with test data was obtained by the following power law relations:

$$F \sim N^{1.6} \tag{4}$$

$$H \sim N^{1.9} \tag{5}$$

as is illustrated in Figure 6.

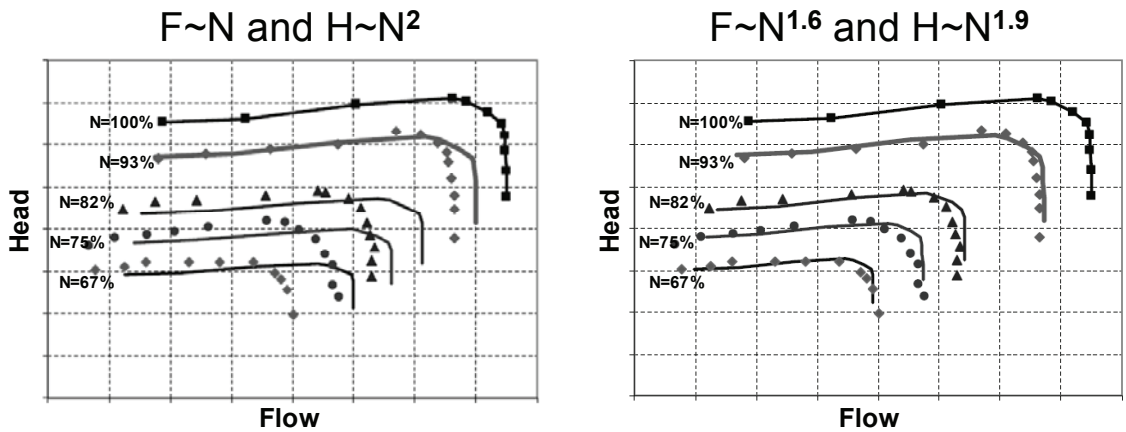

Figure 6. Measured deviation of "fan law" behavior (F~N and H~N^2) for a 2.5:1 variable speed compressor. The points are the measurements while the solid lines are predictions based on power law assumptions.

5. VARIABLE-SPEED/VARIABLE-GEOMETRY CENTRIFUGAL COMPRESSOR OPERATION.

With both speed and inlet guide vane position to influence compressor performance the control logic becomes more complicated. Most head/flow points on the compressor map can be realized by an infinite number of speed/inlet guide vane combinations with different efficiency. The objective of the variable speed control logic is to always find that speed/inlet guide vane combination that results in maximum compressor efficiency for that head/flow combination. As opposed to just opening or closing the inlet guide vanes based on the difference between the leaving chilled water temperature and its set point value, compressor speed as well as inlet guide vane setting angle have to change in response to a change in head and/or flow.

The higher flow/lower head combinations (the area in Figure 5 that is not hatched) can be covered by variable speed fixed geometry. Best compressor efficiency for those operating conditions is obtained by changing speed only and leaving the guide vanes completely open. The hatched area of the map of Figure 5 can only be achieved by a combination of inlet guide vane closure and speed reduction. Figure 7 shows the speed lines and the IGV positions of the combined variable-speed variable-IGV map and the optimum part-load efficiency that result. A remarkable difference in part-load efficiency is shown between the fixed- and variable speed compressor performance, despite the equal efficiencies shown at the map boundaries (surge and choke). Compressor applications with part-load conditions requiring a proportional relationship between flow and head, as encountered in many water-cooled chiller applications, show the largest benefit from variable speed operation. Compressor applications that require an almost constant head (with variable capacity) or an almost constant flow (with variable head) would obviously not benefit much from variable speed operation, since the compressor efficiency on the surge (maximum head) and choke (maximum flow) lines is identical since it requires full-speed compressor operation..

6. ACHIEVING OPTIMUM EFFICIENCY THROUGH CONTROL

In order to realize peak efficiency, the head/flow combination the compressor is required to deliver has to be known. This information then has to be used to determine the optimum speed/inlet guide vane combination, shown in Figure 7 that gives the best efficiency for that operating point. For head/flow combinations that can be achieved without closing the guide vanes, variable speed alone should be used for capacity control and the inlet guide vanes should stay fully opened. For head/flow combinations above the variable speed fully open IGV surge line, the combination of speed and IGV position indicated in Figure 7 should be applied for capacity control.

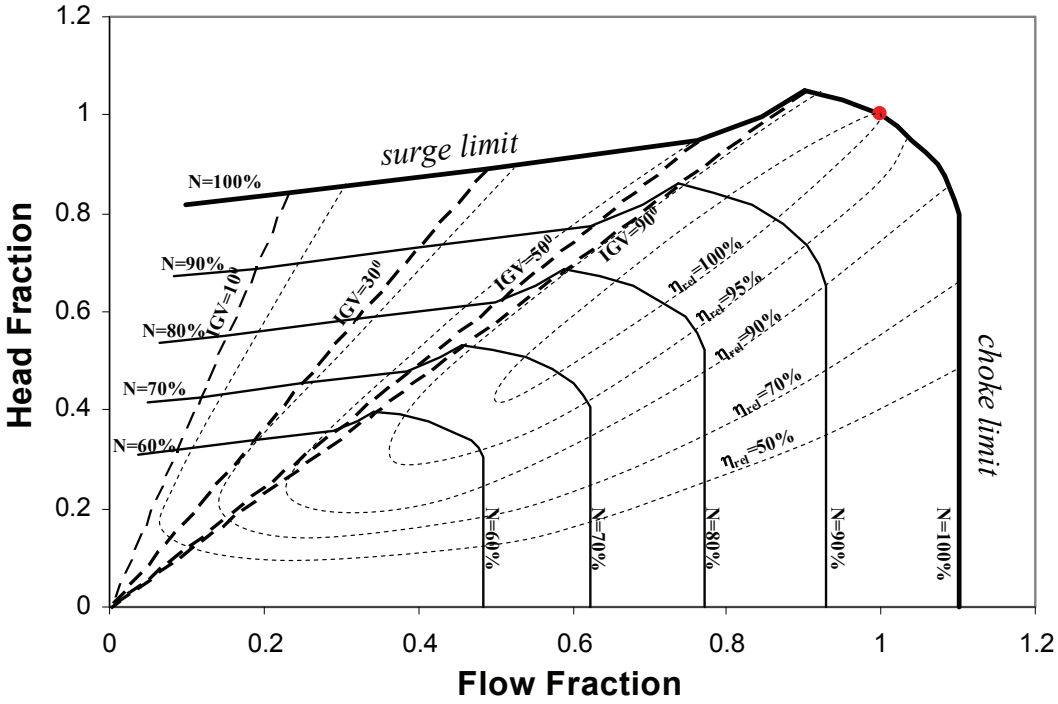

Figure 7. Performance map of a variable-speed, variable-IGV compressor with IGV/speed combinations for optimum efficiency.

For centrifugal chiller applications, head can be determined from evaporator and condenser saturation pressure measurements that are readily available. The refrigerant flow of the compressor is not measured directly and has to be determined from a heat balance over the heat evaporator. Measured chilled water flow rate and entering and leaving chilled water temperatures determine the heat being absorbed by the refrigerant in the evaporator. Knowing the thermodynamic state points of the refrigerant entering and leaving the evaporator, the compressor flow rate can then be determined from the heat balance between the water and refrigerant side of the evaporator. The speed/inlet guide vane combination that gives the highest compressor efficiency for that specific head/flow condition should be known from previous compressor testing and can then be selected by the controller.

From the above discussion it becomes clear that the variable-speed compressor control system becomes more complicated and requires detailed compressor performance information as well as knowledge of the actual operating point the compressor is supposed to run at. In contrast, no information at all is needed for fixed-speed centrifugal compressors using variable geometry to control capacity.

Changes in chilled water flow rate, encountered with VPF (variable primary flow) systems, have become more popular recently. The variable speed control of the chilled water pumps achieves an additional chilled water plant power savings of around 5% [2]. These systems put even more demands on the variable speed chiller control system, while not affecting the fixed-speed control system at all. However, the difference in compressor map efficiency between fixed and variable speed operation is worth the extra complication.

7. COMPRESSOR PERFORMANCE MAPPING METHOD ELIMINATING THE NEED FOR FLOW MEASUREMENTS

Centrifugal compressor performance is traditionally represented by a two-dimensional performance map with head, pressure rise, pressure ratio or lift (=condenser/evaporator saturation temperature difference) on the vertical axis and volume flow rate, mass flow rate or capacity on the horizontal axis. Figure 4 showed the non-dimensional representation of this map with head and flow as a fraction of their full-load peak efficiency design values. Inlet guide vane setting angle is a parameter on these performance maps. It is possible to reverse the role of flow rate and inlet guide vane setting angle. Following that approach a performance map can be created with head fraction again on the vertical axis, inlet guide vane setting angle on the horizontal axis and flow fraction as a parameter. Figure 6 makes a side–by-side comparison of these two compressor performance maps.

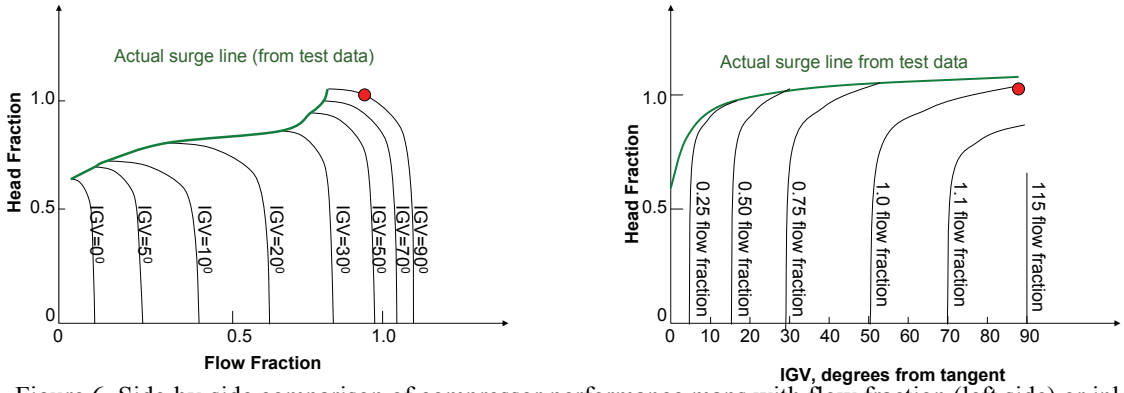

Figure 6. Side-by-side comparison of compressor performance maps with flow fraction (left side) or inlet guide vane setting angle (right side) along the horizontal axis. The point highlighted in red is the full-load design point.

One surprising result of compressor performance mapping with inlet guide vane setting angle as opposed to the flow fraction on the horizontal axis has been the more consistent shape of the surge-line for different compressor builds. Figure 7 shows the width of the uncertainty band typically encountered when the maps of a number of different compressors are plotted in non-dimensional form. It can be seen by comparing the uncertainty bands of these two maps, that the performance map with inlet guide vane setting angle on the horizontal axis results in a more predictable surge line.

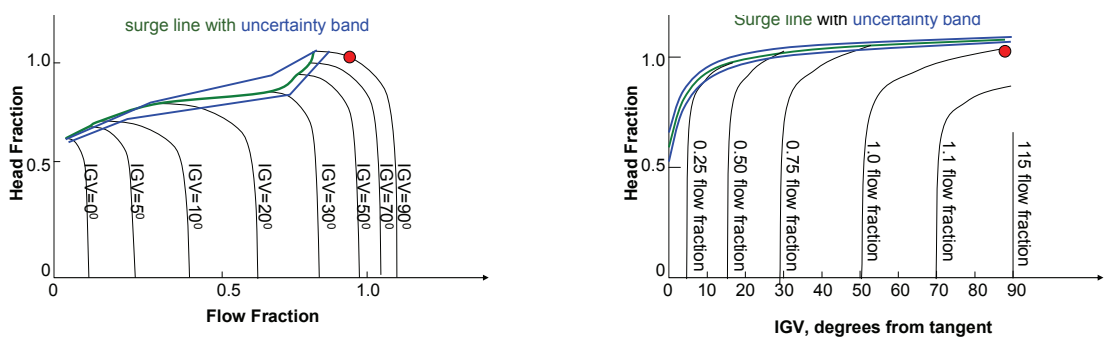

Figure 7. Side-by-side comparison of surge-line uncertainty band of compressor performance maps with flow fraction (left side) or inlet guide vane setting angle (right side) along the horizontal axis.

The reason for the more consistent prediction of surge when head is plotted versus inlet guide vane position can be understood from the shape of the constant IGV line in the head/flow map (Figure 3 or the left figure in Figure 7). The constant IGV-curve is almost horizontal close to surge meaning that uncertainty in surge point locations is much more an uncertainty in flow at surge than in head at surge. Since flow is not an independent parameter when the map is plotted in terms of head versus IGV position, the uncertainty in surge on that map is limited to an uncertainty in maximum head for a given inlet guide vane position which was the smaller uncertainty.

8. SURGE LINE REPRESENTATION FOR CONTROL

Since the controller has to guarantee surge free operation, compressor performance is only allowed for operation below the surge line. Compressor peak efficiency occurs close to surge, so prohibiting variable speed compressor operation at certain map areas close to the surge line as a results of uncertainty in the current operating point will result in reduced compressor efficiency. Therefore, because control surge lines are not generic but are specific for individual compressors, each specific compressor will have a different surge line that must be tested in-situ.

The limited uncertainty in surge line location of the head/IGV map representation allows the definition of a generic surge line. A good approximation using an exponential curve of the shape:

$$H(IGV) = H_{IGV=90^0} - (H_{IGV=90^0} - H_{IGV=0^0})e^{-x*IGV} \qquad (6)$$

where IGV = the setting angle of the IGV's measured from the tangential direction

$H_{IGV=90^0}$ = the full-load design head, obtainable from the compressor data release

$H_{IGV=0^0}$ = the minimum head at 10% flow obtainable from the data release

x = an adjustable shape parameter with a default value of 0.08

for small x is shown in Figure 8. This equation has some inherent inaccuracies (for example $H_{IGV=90^0}$ =.984 instead of 1.0), but for the purpose of discussion, equation 6 will be used. For accurate modeling, a more complicated equation to describe this curve should be used.

Figure 8 shows the close approximation of the surge line as described by Equation (6) to the experimentally determined surge line for a Carrier 19XR4P6 compressor.

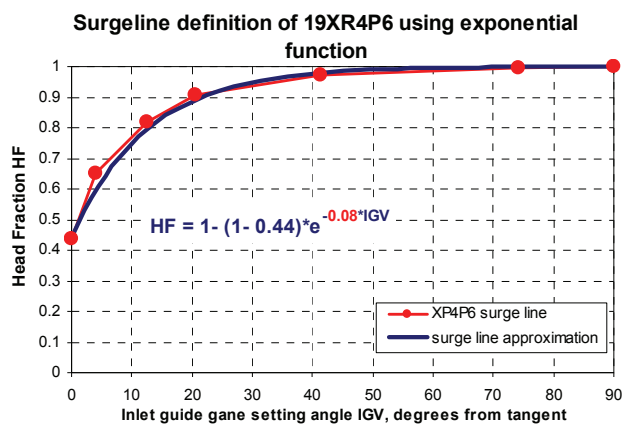

Figure8.. Comparison between actual surge line test data of the 19XR4P6 compressor (red bullets) and the surge line approximation using the exponential function (Eq.(6)).

If needed the adjustable shape parameter can be used to accommodate machines with different surge lines as illustrated in Figure 9. The change in surge line with speed for a centrifugal compressor with its performance represented by a head/flow map was shown in Figure 6. If the compressor performance is represented by a head/IGV performance map the location of the surge line at reduced speed is obtained by using the head speed relationship given in Equation 5.

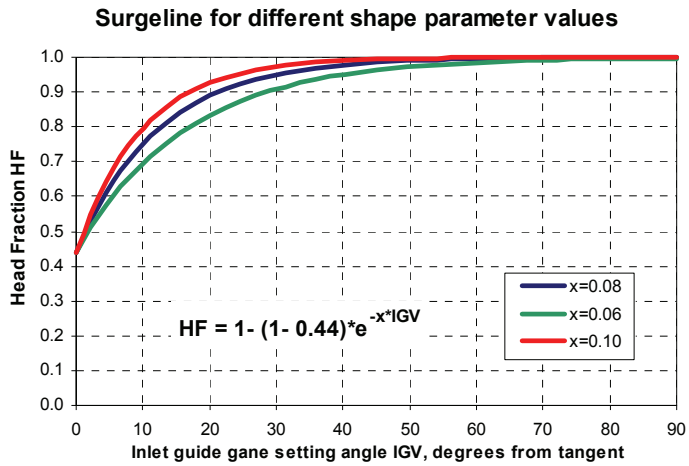

Figure 9. Surge line shape change for different shape parameters.

Figure 10 shows the surge line predictions at reduced speed with performance now shown on the new head/IGV map.

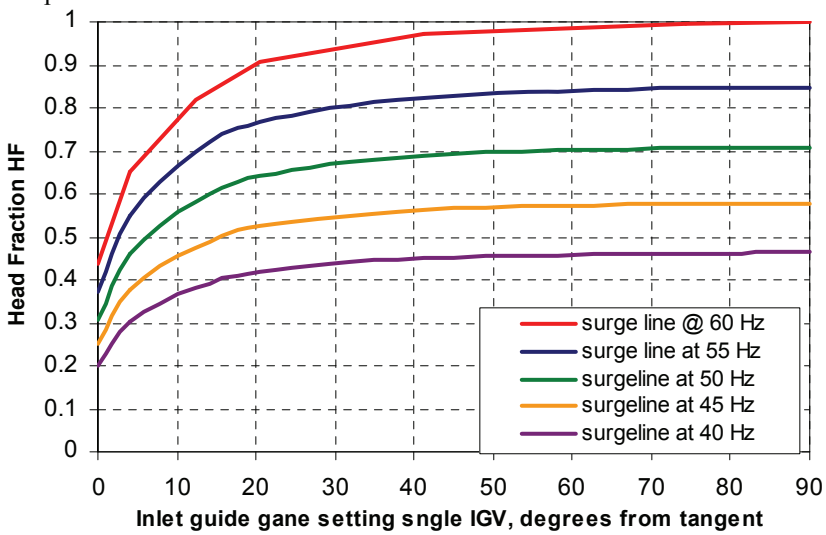

Figure 10. Variable-speed surge line definition derived from 60 Hz surge line using the head/speed power law $HF(f)=HF(f=60)*(f/60)^{1.9}$

9. CONCLUSIONS

Variable-speed compressors have the potential of substantially improving compressor part-load efficiency and have become the drive option of choice for many applications with the availability of low cost inverters.

The application of variable speed to compressors requires additional knowledge of compressor behavior since - contrary to pumps, fans and blowers - compressor behavior changes substantially at lower speed due to compressibility effects.

In general, compressor flow decreases faster than a linear relationship with speed while compressor head reduces somewhat slower then the square of speed, violating the simple "fan laws". The exact relationship between compressor flow and head with speed depends on compressor details such as impeller backsweep, choice of diffuser (vaneless versus vaned), number of stages.

Variable speed alone is not a control option for centrifugal compressor applications that require substantial head at part-load conditions such as encountered in water-cooled chillers. Variable speed control has to be complimented by variable geometry control. To obtain the full benefit of variable speed compressor operation the

compressor speed always has to be reduced down to an operational point close to surge where peak efficiency occurs.

To find the optimum control values for speed and inlet guide vanes the variable speed controller needs – contrary to the fixed-speed variable geometry controller – both head and flow information. Therefore many control schemes for variable-speed centrifugal compressors are more prone to surge and do not realize their maximum efficiency potential.

The compressor surge line is more accurately described in terms of head fraction versus inlet guide vane position than following the conventional approach of describing compressor performance in terms of head versus flow.

Mapping the compressor in terms of head versus IGV setting angle has the added advantage of allowing chiller control schemes that do not depend on flow information. Such control schemes are well suited for variable primary flow systems where the chilled water flow rate is not constant but changes with operating conditions.