CONTROL DEVICE FOR HVAC SYSTEMS WITH INLET AND OUTLET FLOW CONTROL DEVICES

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ABSTRACT
A process for controlling operation of a heating, ventilation and air conditioning system broadly comprises providing a heating, ventilation and air conditioning system having an evaporator, a condenser, a compressor having an inlet and an outlet, and at least one flow control device; measuring a performance parameter of the heating, ventilation and air conditioning system; determining a performance parameter measurement of the flow indicative of onset of surge; determining a surge line of the heating, ventilation and air conditioning system based upon the performance parameter measurement; and independently controlling the at least one flow control device based upon the performance parameter to mitigate an onset of the surge.
FIG. 1
**FIG. 3**

**FIG. 4**
FIG. 7
**FIG. 9A**

**FIG. 9B**
Inlet guide vane setting angle $IGV$, degrees from tangent

\[ HF = 1 - (1 - 0.44)e^{-0.08IGV} \]

**FIG. 10**
HF = 1 - (1 - 0.44) * e^{-x/IGV}

FIG. 11
FIG. 12
FIG. 13A

FIG. 13B
CONTROL DEVICE FOR HVAC SYSTEMS WITH INLET AND OUTLET FLOW CONTROL DEVICES


BACKGROUND

[0002] The disclosure relates to heating, ventilation and air conditioning (“HVAC”) systems of buildings and, more particularly, relates to a control device for HVAC systems of buildings.

[0003] As energy cost rises and end users become more sophisticated in their 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.

[0004] 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 or 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.

[0005] 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 known 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.

[0006] There exists a need to improve part-load performance through the use of both fixed speed and variable-speed operations.

[0007] There exists further a need for a multi-input, multi-output compressor control system for both fixed speed and variable-speed operations.

[0008] There exists further yet a need for a model-based feed forward element for the controller to determine the combination of speed and variable geometry necessary to achieve maximum compressor efficiency for both fixed speed and variable-speed operations.

SUMMARY

[0009] In one aspect of the present disclosure, a process for controlling operation of a heating, ventilation and air conditioning system broadly comprises providing a heating, ventilation and air conditioning system having an evaporator, a condenser, a compressor having an inlet and an outlet, and at least one flow control device in communication with the inlet or the outlet or both the inlet and the outlet; measuring a performance parameter of the heating, ventilation and air conditioning system; determining a performance parameter measurement indicative of onset of surge; determining a surge line of the heating, ventilation and air conditioning system based upon the performance parameter measurement; and independently controlling the at least one flow control device based upon the performance parameter to mitigate an onset of the surge.

[0010] In another aspect of the present disclosure, a heating, ventilation and air conditioning system broadly comprises an evaporator; a condenser; a compressor having an inlet and an outlet; at least one flow control device in communication with the inlet or the outlet or both the inlet and the outlet; and a controller programmed with information corresponding to performance parameter measurement indicative of onset of surge, and adapted to independently control at least any one of the following: the compressor or at least one flow control device or both the compressor and the at least one flow control device, and mitigate an onset of surge.

[0011] The details of one or more embodiments are set forth in the accompanying drawings and the description below. Other features, objects, and advantages will be apparent from the description and drawings, and from the claims.

BRIEF DESCRIPTION OF THE DRAWINGS

[0012] FIG. 1 is a representation of a heating, ventilation and air conditioning system;

[0013] FIG. 2 is a representation of a centrifugal compressor of FIG. 1;

[0014] FIG. 3 is a representation of a head/flow relationship for a fixed-speed fixed geometry centrifugal compressor;

[0015] FIG. 4 is a representation of a performance map of a fixed-speed centrifugal compressor with variable inlet guide vanes;

[0016] FIG. 5 is a representation of a performance map of a fixed-geometry centrifugal compressor with a variable frequency drive (VFD) for capacity control;

[0017] FIGS. 6A and 6B are representations of a measured deviation of “fan law” behavior for a 2:5:1 variable speed compressor;

[0018] FIG. 7 is a representation of a performance map of a variable-speed, variable-IGV compressor with IGV/speed combinations for efficiency;

[0019] FIGS. 8A and 8B are representations of a side-by-side comparison of compressor performance maps indicating the full load design point using (A) flow fraction versus (B) inlet guide vane setting angle;
FIGS. 9A and 9B are representations of a side-by-side comparison of surge-line uncertainty bands of compressor performance maps using (A) flow fraction versus (B) inlet guide vane setting angle;

FIG. 10 is a representation of a comparison between actual surge line test data of an 19XR4P6 compressor and the surge line approximation using the exponential function;

FIG. 11 is a representation of a surge line shape change for different shape parameters;

FIG. 12 is a representation of surge line predictions at reduced speed with performance shown a head/IGV map; and

FIGS. 13A and 13B are test results of a VPF chiller using the exemplary controller.

Like reference numbers and designations in the various drawings indicate like elements.

DETAILED DESCRIPTION

Referring to FIGS. 1-13, a multi-input, multi-output compressor control system with model-based feed-forward element for both fixed speed and variable-speed operation is disclosed and described herein. The model-based feed-forward element of the exemplary controller described herein may be implemented with various HVAC systems utilizing any combination of single input/output and/or multi-input/output, any type of compressor, and any type(s) and number of input and output flow control devices. For example, compressors for use herein may include but are not limited to fixed-speed compressors and variable-speed compressors. In addition, such compressors may also include centrifugal compressors and axial compressors. Likewise, various inlet/outlet flow control devices may be utilized such as variable geometry diffusers, dampers, extractors, grilles, valves, inlet guide vanes, and the like as known to one of ordinary skill in the art. The exemplary controller of the controller described herein is designed to determine the onset of surge based upon a performance parameter and calculate a surge line. The controller then independently controls the system to mitigate the onset of surge.

For use herein, the terms “performance parameter” or “performance parameter measurement” is defined as any parameter or parameter measurement describing the enthalpy difference or difference in saturation properties (e.g., temperature, pressure) or pressure ratio from some inlet to some outlet of a compressor of the system.

For purposes of illustration and not to be taken in a limiting sense, the exemplary controller disclosed herein will be described using a heating, ventilation and air conditioning system 100 (“HVAC system 100”) having a chiller 102, an air handling unit 104, a pump 106 and a valve 108. The direction of water flow, water temperatures and configuration of components is representative of an exemplary embodiment of the system 100. The chiller 102 includes an evaporator 103, a compressor 110, a condenser 105 and a controller 107. The compressor 110 of the HVAC system 100 of FIG. 1 is shown in FIG. 2. The components of interest from inlet to exit are the inlet guide vanes (IGV’s) 112, typically composed of a plurality, preferable a set of seven, uncambered vanes, a back-sweep twenty-two (22) bladed compressor (11 main, 11 splitters), a small vaneless space 114 to a pipe diffuser 116, a constant cross-sectional area collector 118, and an impeller 120. The working gas is pulled from an evaporator 103, is compressed, and then discharged to a condenser 105. Pressure measurements can be made in the evaporator 103, condenser 105 and a plenum adjacent and connected to vaneless space 114 before the diffuser (see FIG. 1). Of course, it should be appreciated that the VPF system and compressor are given as non-limiting examples only, and other configurations would certainly fall well within the broad scope of the present disclosure.

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:

\[ \Delta H = \alpha_{2} - \alpha_{1} \]  

(Equation 1)

where \( \Delta H = \) the change in enthalpy [J/kg];

\( u_{1}= \) the wheel speed at rotor inlet [m/s];

\( c_{01} = \) the tangential component of the incoming flow [m/s];

\( u_{2} = \) the rotor speed at compressor exit [m/s]; and

\( c_{02}= \) 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.

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.

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, that is, 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, e.g., condenser saturation temperature and/or evaporator saturation temperature, and cooling capacity (see FIG. 3). This is a fixed, one-to-one mapping of lift to flow.

Variable geometry of stationary compressor components, e.g. inlet flow control devices imparting 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. Although other inlet flow control devices may be utilized, variable inlet guide vanes are 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 FIG. 4.

Again, for purposes of illustration and not to be taken in a limiting sense, the compressor map will be described using a compressor system utilizing an inlet guide vane ("IGV") as the inlet flow control device. Each possible head/flow combination has a unique inlet guide vane angle position and a unique efficiency. The compressor performance map may be 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. 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, e.g., measured from the tangential direction, i.e. 90° fully open and 0° fully closed, can be drawn on the fixed-speed compressor performance map. The inventors of the present disclosure observed a 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 open, 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 thus also the head. The head-flow curve moves both to a smaller head and a smaller flow, but the difference between maximum and minimum flow, that is, choke flow and surge flow, the approximate 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.

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 (FIG. 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 connecting 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 may be that its peak efficiency may be limited to a narrow operating range close to its surge line.

The model-based feed-forward element of the controller determines as a function of required operating condition, e.g., 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 precisely 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 HVAC systems (See FIG. 1), the chilled water flow rate changes necessitate an additional flow measurement, which also has proven to be less reliable and accurate in actual field installation. The exemplary control system described herein circumvents this obstacle and is not affected by such variable primary flow changes.

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. FIG. 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.

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:

\[
F = N \quad \text{(Equation 2)}
\]

\[
H = N^2 \quad \text{(Equation 3)}
\]

The fan laws apply satisfactorily for pumps and fans (incompressible flow) but do not accurately describe the lower speed performance of compressors. FIGS. 6A and 6B show test data at different lower speeds 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, a 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.

With both speed and inlet guide vane position to influence compressor performance the control logic becomes more complicated. Most head/flow points on a 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 FIG. 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 FIG. 5 can only be achieved by a combination of inlet guide vane closure and speed reduction. FIG. 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, e.g., 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.

In order to realize peak efficiency, the head/flow combination the compressor is required to deliver has to be known. This information may then be used to determine the optimum speed/inlet guide vane combination, shown in FIG. 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 can be used for capacity control and the inlet guide vanes can remain fully opened.

For centrifugal chiller applications, both the saturation temperatures and 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 exchanger. 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 can be known from previous compressor testing as known to one of ordinary skill in the art, and can then be selected by the controller. The variable-speed compressor control system requires detailed compressor performance information as well as knowledge of the actual operating point the compressor is supposed to run at. Changes in chilled water flow rate, encountered with VPF 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 8%. These systems put even more demands on the variable speed chiller control system.

Centrifugal compressor performance is traditionally represented by a two-dimensional performance map with head, pressure rise, pressure ratio or lift, i.e., condenser evaporator saturation temperature difference, on the vertical axis and volumetric flow rate, mass flow rate or capacity on the horizontal axis. The inlet guide vane setting angle is a parameter on such performance maps. The inventors of the present application discovered the role of flow rate and inlet guide vane setting angle can be reversed. Following that approach an exemplary 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. FIGS. 8A and 8B illustrate a side-by-side comparison of these two compressor performance maps.

One surprising result of compressor performance mapping with inlet guide vane setting angle as opposed to the flow fraction on the horizontal axis is the more consistent shape of the surge-line for different compressor builds. FIGS. 9A and 9B show the width of the uncertainty band typically encountered when the maps of a number of different compressors are plotted in non-dimensional form. When comparing the uncertainty bands of these two maps, the performance map with inlet guide vane setting angle of FIG. 9B results in a more predictable surge line.

The reason for the more consistent prediction of surge when head is plotted versus inlet guide vane position (as shown in FIG. 9B) can be understood from the shape of the constant IGV line in the head/flow map (See FIG. 9A). 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 FIG. 9B is limited to an uncertainty in maximum head for a given inlet guide vane position which was the smaller uncertainty.

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 result 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 determined in situ.

The uncertainity 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:

$$H_{IGV} = H_{IGV_{max}} - H_{IGV_{min}} e^{-x/10}$$  \hspace{1cm} (Equation 4)

where IGV is the setting angle of the IGV’s measured from the tangential direction;

where $H_{IGV_{max}}$ is the full-load design head, obtainable from the compressor data release;

where $H_{IGV_{min}}$ is the minimum head at 10% flow obtainable from the data release; and

x is an adjustable shape parameter with a default value of 0.08 (See FIG. 10). Although the exemplary curve is used for demonstration purposes here, one of ordinary skill in the art recognizes and appreciates any appropriate curve fit equation may be used to approximate the surge line.

If needed the adjustable shape parameter can be used to accommodate machines with different surge lines as illustrated in FIG. 11. The change in surge line with speed for a centrifugal compressor with its performance represented by a head/flow map was shown in FIGS. 6A and 6B. 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 3. FIG. 12 shows the surge line predictions at reduced speed with performance now shown on an exemplary head/IGV performance map. FIGS. 13A and 13B show test results of a VPF chiller using the exemplary controller described herein. The VPF chiller operated surge free while exhibiting a 50% VPE reduction over the course of approximately ten (10) hours (See FIGS. 13A and 13B).

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 backswep, choice of diffuser (vaneless versus vane), 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/outlet flow control device the variable speed controller needs 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, or some equivalent performance parameter or measurement as contemplated herein, 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.

One or more embodiments have been described. Nevertheless, it will be understood that various modifications may be made without departing from the spirit and scope of the disclosure. Accordingly, other embodiments are within the scope of the following claims.

What is claimed is:

1. A process for controlling operation of a heating, ventilation and air conditioning system, comprising:
   providing a heating, ventilation and air conditioning system having an evaporator, a condenser, a compressor having an inlet and an outlet, and at least one flow control device in communication with the inlet or the outlet or both the inlet and the outlet;
   measuring a performance parameter of the heating, ventilation and air conditioning system;
   determining a performance parameter measurement indicative of onset of surge;
   determining a surge line of the heating, ventilation and air conditioning system based upon the performance parameter measurement and independently controlling the at least one flow control device based upon the performance parameter to mitigate an onset of the surge.

2. The process of claim 1, wherein measuring the performance parameter comprises the steps of:
   measuring a first pressure of the heating, ventilation and air conditioning system at a location upstream from the compressor;
   measuring a second pressure of the heating, ventilation and air conditioning system at a location downstream from the compressor; and
determining a pressure ratio.

3. The process of claim 1, wherein measuring the performance parameter comprises the steps of:
   calculating a first saturation temperature at a location upstream from the compressor;
   calculating a second saturation temperature at a location downstream from the compressor; and
determining a saturation temperature difference.

4. The process of claim 1, wherein measuring the performance parameter comprises calculating a head of the compressor.

5. The process of claim 1, wherein measuring the performance parameter comprises the steps of:
calculating a first saturation pressure at a location upstream of the compressor;
calculating a second saturation pressure at a location downstream of the compressor; and
determining a saturation pressure difference.
6. The process of claim 1, wherein independently controlling the heating, ventilation and air conditioning system comprises controlling the operation of any one of the following: a compressor or at least one flow control device or both the compressor and at least one flow control device.
7. The process of claim 6, wherein the at least one flow control device comprises at least one inlet flow control device or at least one outlet flow control device or both at least one inlet flow control device and at least one outlet flow control device.
8. The process of claim 6, wherein independently controlling comprises controlling the speed of the compressor.
9. The process of claim 6, wherein independently controlling comprises increasing the speed of the compressor or decreasing the speed of the compressor or shutting off the compressor.
10. The process of claim 7, wherein independently controlling comprises independently controlling the at least one outlet flow control device comprising a hot gas bypass.
11. The process of claim 7, wherein independently controlling comprises independently controlling a position of at least one inlet flow control device.
12. The process of claim 11, wherein independently controlling comprises independently controlling a position of any one of the following inlet flow control devices comprising: dampers, extractors, grilles, valves, and inlet guide vanes.
13. A heating, ventilation and air conditioning system, comprising:
an evaporator;
a condenser;
a compressor having an inlet and an outlet;
at least one flow control device in communication with the inlet or the outlet or both the inlet and the outlet; and
a controller programmed with information corresponding to a performance parameter indicative of onset of surge, and adapted to independently control at least any one of the following: the compressor or at least one flow control device or both the compressor and the at least one flow control device, and mitigate an onset of surge.
14. The heating, ventilation and air conditioning system of claim 13, wherein the controller is programmed to:
measure a first pressure of the heating, ventilation and air conditioning system at a location upstream from the compressor;
measure a second pressure of the heating, ventilation and air conditioning system at a location downstream from the compressor;
determine a pressure ratio indicative of the onset of surge; determine a surge line of the heating, ventilation and air conditioning system based upon the pressure ratio; and independently control the at least one flow control device based upon the pressure ratio to mitigate the onset of the surge.
15. The heating, ventilation and air conditioning system of claim 13, wherein controller is programmed to:
calculate a first saturation temperature at a location upstream from the compressor;
calculate a second saturation temperature at a location downstream from the compressor; and
determine a saturation temperature difference indicative of the onset of surge;
determine a surge line of the heating, ventilation and air conditioning system based upon the saturation temperature difference; and
independently control the at least one flow control device based upon the saturation temperature difference to mitigate the onset of the surge.
16. The heating, ventilation and air conditioning system of claim 13, wherein the controller is programmed to:
calculate a head of the compressor;
determine a head measurement indicative of the onset of surge;
determine a surge line based upon the head; and
independently control the at least one flow control device based upon the head to mitigate the onset of the surge.
17. The heating, ventilation and air conditioning system of claim 13, wherein the controller is programmed to:
calculate a first saturation pressure at a location upstream of the compressor;
calculate a second saturation pressure at a location downstream of the compressor; and
determine a saturation pressure difference indicative of the onset of surge;
determine a surge line based upon the saturation pressure difference; and
independently control the at least one flow control device based upon the saturation pressure difference to mitigate the onset of the surge.
18. The heating, ventilation and air conditioning system of claim 13, wherein the compressor comprises any one of the following: a fixed-speed compressor or a variable-speed compressor.
19. The heating, ventilation and air conditioning system of claim 13, wherein the compressor comprises any one of the following: a centrifugal compressor or an axial compressor.
20. The heating, ventilation and air conditioning system of claim 13, wherein at least one flow control device comprises any one of the following: an inlet flow control device or an outlet flow control device or both an inlet flow control device and an outlet flow control device.
21. The heating, ventilation and air conditioning system of claim 20, wherein the inlet flow control device comprises any one of the following: dampers, extractors, grilles, valves, and inlet guide vanes.
22. The heating, ventilation and air conditioning system of claim 20, wherein the outlet flow control device comprises any one of the following: a variable geometry diffuser or a hot gas by-pass.