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Increasing the Stable Operating Range of a Fixed-Geometry Variable-Speed Centrifugal Compressor

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ABSTRACT

Centrifugal compressors used on water-cooled chillers require stable operation over a wide range of flows at greatly varying pressure ratios. These operational requirements are dictated by variations in cooling demand and ambient conditions. Variable-speed centrifugal compressors are known to maintain their peak efficiency at the varying operating conditions much better than fixed-speed compressors. Replacing a fixed-speed centrifugal compressor with a variable speed one can reduce the annual energy consumption of a chiller by 40-45%. The majority of centrifugal chillers sold today are therefore inverter driven. Lower speed operation maintains and sometimes even increases compressor efficiency along a wide band of capacity and head combinations which fits quite naturally with most of the chiller operating requirements. However, the variable speed compressor will eventually surge when forced to operate at lower capacity while maintaining head. Some variable-geometry compressor features are necessary to enable stable compressor operation at these conditions. Variable-geometry inlet-guide-vanes and/or variable-geometry diffusers have to be added to variable speed centrifugal compressors to allow stable operation at all possible centrifugal chiller operating conditions. The inherent mechanical complexity of variable-geometry hardware has a negative effect on compressor cost and reliability. What is less appreciated is that compressor efficiency also suffers from variable geometry hardware. The inlet guide vanes introduce additional flow blockage and frictional losses at compressor inlet while the clearances needed for the movement of the variable geometry diffuser hardware introduce flow leakage passages resulting in parasitic flow leakage losses. Moreover, these losses affect compressor performance under all operating conditions, even those where variable speed control without variable geometry flow passage reduction results in stable compressor operation.

This paper describes the application of the newly developed IntraFlow™ technology on a recently introduced two-stage variable-speed centrifugal refrigeration compressor. The concept will be explained in detail and test results will be shown. The compressor is stabilized and surge is postposed by injecting a small amount of flow upstream of the throat area of the vaned diffuser of the first stage compressor. The increase in stable operating range using this technique is substantially larger than what can be obtained with variable geometry inlet guide vanes. Using this technology the compressor also achieves higher efficiency due to the elimination of the blockage, friction and leakage losses that accompany the variable mechanical geometry surge/capacity control concepts. The amount of flow to be injected is controlled by an externally mounted flow control valve which increases reliability and serviceability.

1. INTRODUCTION

Fixed-speed multistage centrifugal compressors were the compressors of choice in the early days of large capacity water-cooled chillers. Multistage compression allows high pressure ratios at moderate rotational speeds and the potential to include the economizer cycle to boost overall chiller performance. The large heat exchanger approach temperatures of the centrifugal chillers in the 1950's and 1960's resulted in low cycle efficiencies that benefitted substantially from economizing. Advancement in heat exchanger/transfer technology has resulted in much smaller approach temperatures. Those smaller approach temperatures combined with increased refrigerant subcooling improved the refrigeration cycle efficiency while at the same time reducing the relative benefit of the economizer. It also resulted in lower compressor head requirements enabling the use of less-expensive single-stage centrifugal compressors without economizers. Moreover, fixed-speed single-stage compressors with variable inlet guide vanes showed part-load efficiency advantages over their two-stage counterparts. As a result, the centrifugal chiller industry saw a move towards single-stage compression. Multistage centrifugal compressors were limited to special high-lift chiller applications and in low pressure centrifugal chillers where direct-drive operation at 50/60 Hz was

possible. In this last case the additional cost of the multistage compressor and its economizer could be partly offset by the better cycle efficiency and elimination of the cost and complication of a transmission system.
Fixed-speed centrifugal compressors, both single and two-stage, experience a dramatic efficiency penalty at lower head and lower flow part-load operating conditions. This drop in off-design efficiency is stronger than that of screw compressors. As a result, fixed-speed screw compressors show higher relative part-load efficiencies than fixed-speed centrifugal compressors [Brasz, 1996].
Variable speed compressor operation has dramatically improved compressor efficiency at off-design conditions, even much more so for centrifugal compressors than for screw compressors, especially at the prevailing lower-capacity lower-lift conditions where the reduction in impeller speed reduces the specific work input. The built-in fixed compression ratio of screw compressors results in over-compression at these lower head conditions causing inferior part-load efficiency relative to variable-speed centrifugal compressors.
Centrifugal compressors have a larger stable operating range at lower pressure ratio. They also maintain higher relative efficiencies at off-design conditions at lower pressure ratio. That is the reason that for a given pressure ratio two-stage compressors, consisting of two lower pressure ratio compressors in series, have a wider stable operating range and less variation in efficiency compared to single-stage compressors.
Figure 1 illustrates the differences in part-load efficiency discussed above by showing the location of the 95% relative efficiency islands for the following three compressors:

1. Variable-speed screw compressor
2. Variable-speed single-stage centrifugal
3. Variable-speed two-stage centrifugal compressor

The relative compressor efficiency is larger than 95% of the compressor efficiency at the full-load design point for the head/flow combinations falling within these boundaries. It can be seen from this figure that - starting from the same full-load design efficiency - the variable speed two-stage centrifugal compressor is the preferred compression concept for maximum off-design efficiency, even in non-economized chiller applications.

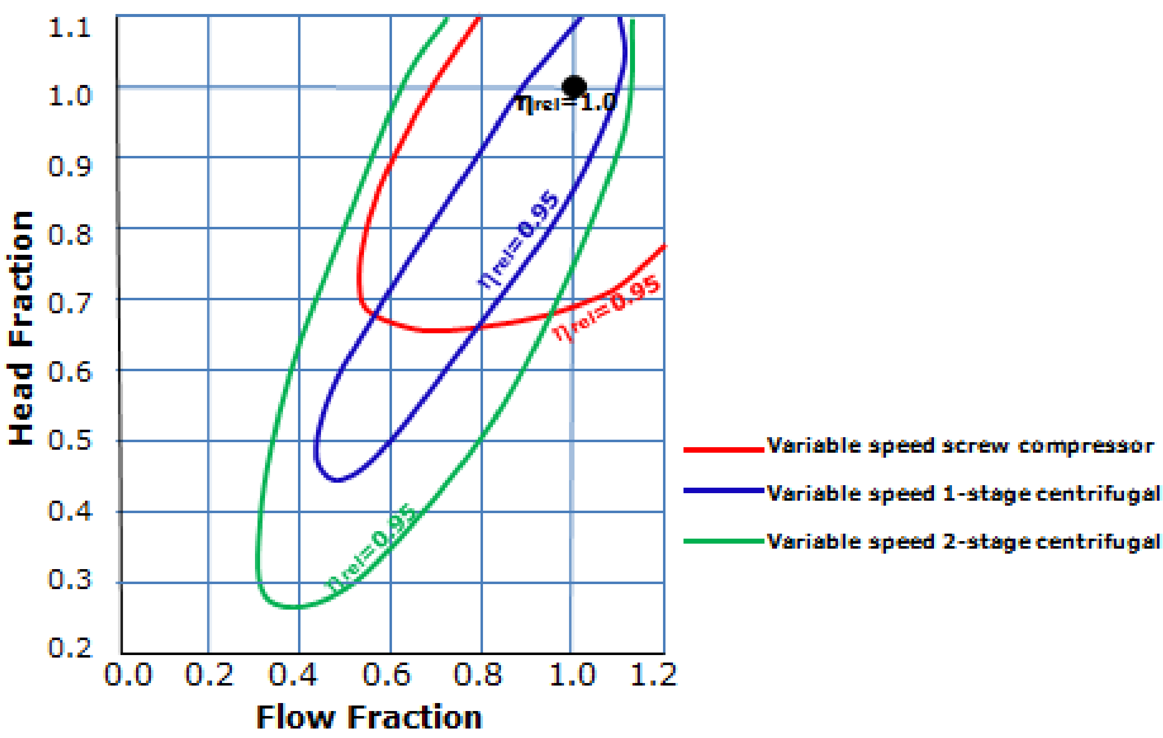


Figure 1: The 95% relative efficiency islands on the performance map for three compressor concepts

2. IMPROVING COMPRESSOR EFFICIENCY WITH VANED DIFFUSERS

The diffuser is the compressor element directly downstream of the impeller. Its function is to transfer the high velocity of the flow leaving the impeller into additional static pressure. The impeller transfers mechanical shaft energy into fluid flow energy. Only slightly more than half the mechanical energy supplied to the impeller results directly in pressure rise (potential energy) while the remaining energy leaves the impeller in the form of high velocity (kinetic energy). A very important and technically challenging element of a centrifugal compressor design is therefore the diffuser downstream of the impeller that transforms this velocity into additional pressure rise. Vaneless diffusers have traditionally been the preferred diffuser technology for centrifugal refrigeration compressors and are still used on the majority of centrifugal chillers. They provide the compressor with a simple design that results in a wide stable operating range. Combined with variable inlet guide vanes, the centrifugal compressor can satisfy the predominant capacity and lift conditions a water-cooled chiller will encounter. The flow path of fluid particles in a vaneless diffuser follows essentially a log spiral profile maintaining its relative flow angle as shown in Figure 2. This limits the reduction in velocity to the radius ratio of the diffuser. For example, a vaneless diffuser with a radius ratio (diffuser outlet radius/diffuser inlet radius) of 2 would see only a 50% reduction in velocity and a corresponding rise in static pressure. A vaned diffuser on the other hand would achieve a larger reduction in velocity over a smaller flow path length as shown in Figure 3. The corresponding increase in diffuser pressure recovery has the potential to improve overall compressor efficiency by roughly 5%.

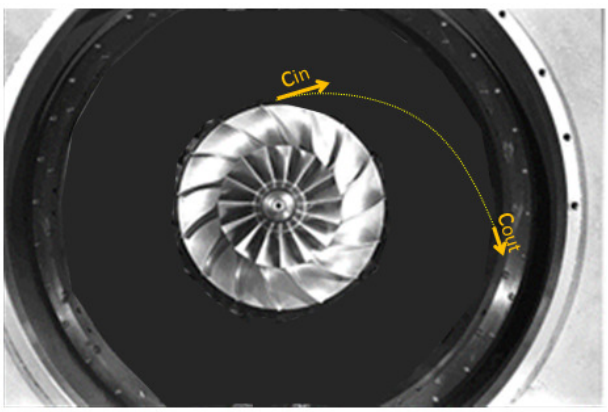
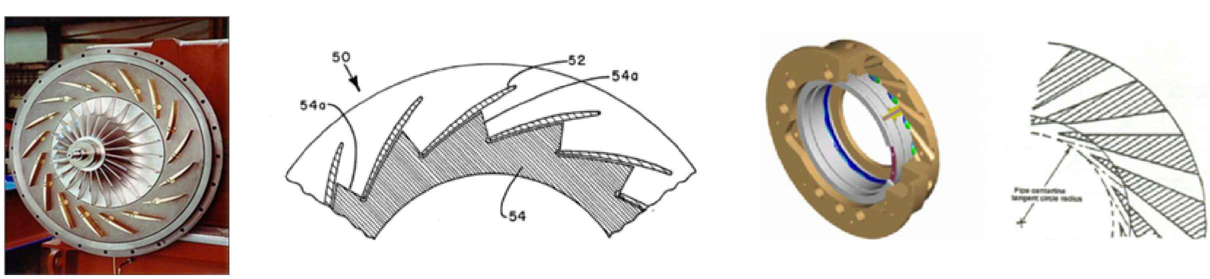


Figure 2: Vaneless diffuser log spiral streamline with a velocity reduction $c_{out}/c_{in} \sim r_{out}/r_{in}$

Figure 3: Vaned diffuser mean streamline showing a larger velocity reduction and a shorter passage length

Various fixed-geometry vaned diffuser concepts have been used successfully in the aerospace industry for the centrifugal compressors of smaller gas turbines used on APU's and small jet engines, e.g. (Kenny, 1968). The more stringent part-load stability requirements of water-cooled chillers (80% head at 20% of full-load capacity) demands variable geometry vaned diffusers. Making the diffuser geometry variable has been tried in many different ways such as rotatable diffuser vanes (Ubben and Niehuis, 2005), movable diffuser walls (Sishtla, 1996) or circumferential rotation of an inner diffuser versus an outer ring (Salvage, 1999). The common element in all these mechanisms is the adjustment of the diffuser inlet throat area thus preventing diffuser stall at off-design conditions.



a. Rotatable vanes b. Axially movable wall element c. Split ring pipe diffuser

Figure 4: Examples of variable geometry diffuser concepts employed for centrifugal chillers

Clearances are needed for the movement of the variable geometry diffuser hardware. These clearances introduce flow leakage passages resulting in parasitic flow leakage losses. The compressor design point efficiency gain of a fixed geometry vaned diffuser is typically reduced 20 – 25% as a result of these flow leakage losses, taking 1 to 1.5 points away from the 5 point efficiency improvement possible with fixed diffuser geometry. Figure 5 shows the compressor efficiency variation with flow rate that at a given compressor speed for a compressor with different type of diffusers.

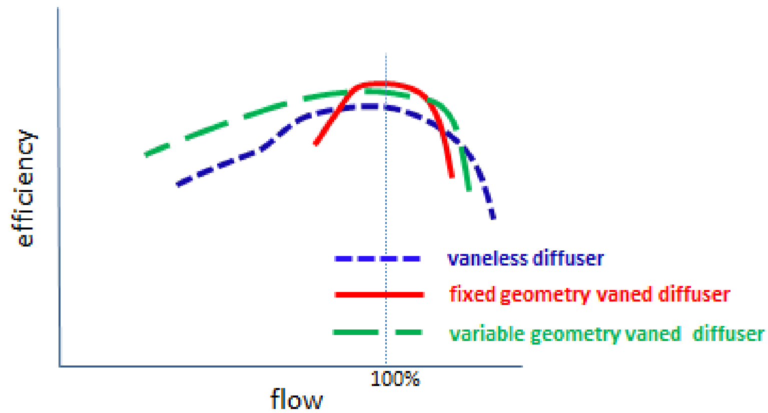


Figure 5: Efficiency variation with flow rate at a given speed for a compressor with different diffuser types

3. IMPROVING COMPRESSOR EFFICIENCY OF A TWO-STAGE COMPRESSOR

The complexity of variable diffuser geometry, needed for compressor off-design stability when vaned diffusers are required for efficiency improvement creates additional compressor failure modes reducing overall reliability. Unacceptable field failure rates have forced some centrifugal chiller manufacturers to discontinue the variable geometry diffuser option of their single-stage compressors after initial market release. Applying variable geometry diffusers to two-stage compressors would double the cost/complexity relative to single-stage compressors causing even more reliability issues. It is therefore no surprise that two-stage centrifugal refrigeration compressors have stayed with their fixed geometry vaneless diffusers. The variable geometry of two-stage compressors has been limited to the inlet guide vanes.

The variable diffuser geometry prevents diffuser stall at lower flow conditions by reducing the inlet cross-sectional flow area of the diffuser. This prevents the sudden excessive flow deceleration at reduced flow rates that would result in diffuser stall. Excessive flow deceleration can also be prevented by aerodynamically instead of mechanically reducing the throat area at part-load conditions. This can be accomplished by taking some of the compressed gas from the compressor exit and injecting it at the throat area of the vaned diffuser reducing the throat area available for the main flow. The net effect is similar to that of a movable wall vaned diffuser. The metal blockage achieved by the movable wall that narrows the diffuser flow passage is replaced by aerodynamic blockage from injected flow that in a similar way reduces the remaining diffuser flow passage area available for the main flow. This approach would be especially useful for a two-stage compressor since the flow injected at the throat area of the first stage vaned diffuser would add to the main flow in the second stage compressor and improves its stability at the same time. With this concept it becomes possible to eliminate the requirement of mechanically variable diffuser geometries. It even allows elimination of the variable inlet guide vanes, as has been demonstrated on single-stage variable-speed centrifugal compressors with variable geometry diffusers by Ubben and Niehuis 2005. Flow recirculation allows the use of fixed geometry vaned diffusers that do not suffer from leakage losses along the clearances needed for variable-geometry diffusers. As a consequence the compressor efficiency at full-load design conditions - when no flow recirculation is required - will be that of a compressor with a fixed geometry diffuser which is higher than that of a variable geometry diffuser.

This flow recirculation technology has resulted in the ability to improve two-stage compressor efficiency by increasing the pressure recovery through the use of vaned diffusers instead of vaneless diffusers while eliminating the mechanical complexity of the variable inlet guide vanes and or adding variable geometry diffusers (Brasz and Rasmussen, 2014).

The capacity of the compressor is controlled by speed variation at higher capacities enhanced by flow recirculation at the lower capacities where variable geometry was needed in the past to prevent surge. Figure 6 shows the two stage compressor with its externally mounted flow control valve that increases the stable operating range of the compressor. Disassembly of the compressor suction housing from its main base shows the nozzle assembly sitting in the suction housing (Figure 7) and the first-stage diffuser vanes as part of the return channel component sitting in the main housing (Figure 8). Figure 9 shows how the individual nozzles injecting the recirculation flow into the vaned diffuser throat areas are lined up angle-wise in order to minimize flow losses.

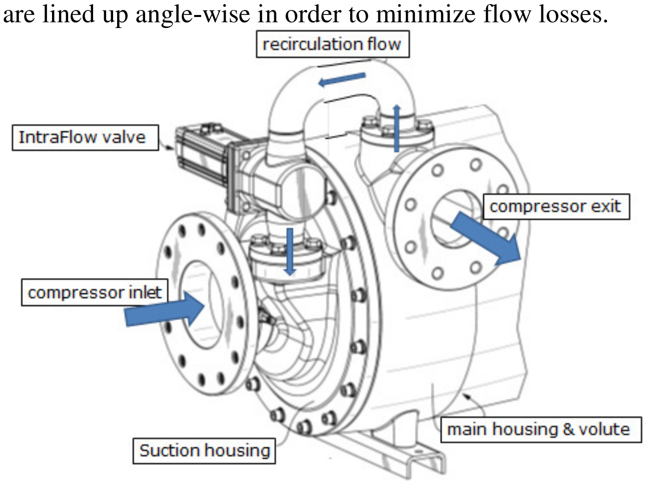


Figure 6: Model of the two-stage compressor with the externally mounted intraflow valve for part-load capacity control and surge prevention

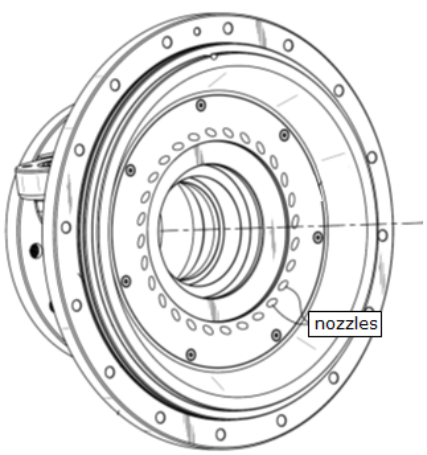


Figure 7: Compressor suction housing showing array of nozzles for the recirculation flow

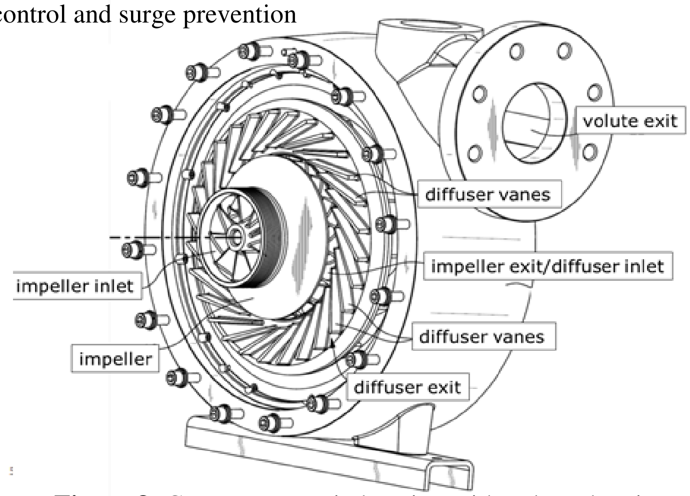


Figure 8: Compressor main housing with volute showing the 1st stage impeller and vaned diffuser array

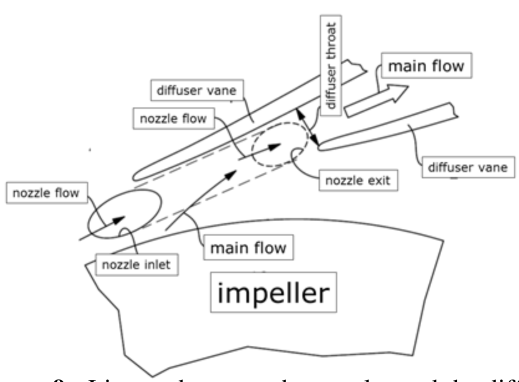


Figure 9: Line-up between the nozzles and the diffuser vanes

4. DIFFERENCE BETWEEN INTERSTAGE FLOW RECIRCULATIONAND HOT GAS BYPASS

It is important to discuss the difference between the inter-stage flow recirculation concept described in this paper and the conventional hot gas bypass method used to increase the stable operating range of a centrifugal compressor at lower flow rates. Both concepts start with extracting some flow from the compressor discharge.
In case of hot gas bypass, the condenser is the only component that the hot gas bypasses. The hot gas bypass flow fraction is throttled to suction pressure before entering the evaporator. This bypass flow does not contribute to any cooling. The compression work done on this flow is therefore completely lost. Moreover, the hot gas bypass flow enters the evaporator with a substantial amount of superheat. This superheat is removed by flashing off some of the liquid refrigerant in the evaporator increasing the vapor fraction of the two-phase flow mixture. This further reduces the coefficient of performance of the chiller. Figure 10 shows the pressure-enthalpy diagram of the hot gas bypass arrangement. The throttling over the hot gas bypass valve causes an isenthalpic expansion process that is represented by the vertical line on the pressure-enthalpy diagram going from compressor exit to evaporator pressure (line a-b in Figure 10). The enthalpy reduction needed to de-superheat the hot gas bypass (line b-c in Figure 10) occurs at the expense of an increase in enthalpy of the two-phase flow refrigerant entering the evaporator (line d-e in Figure 10). These enthalpy changes are opposite in direction. They are equal in size if half the compressor flow is bypassed through the hot gas bypass valve.

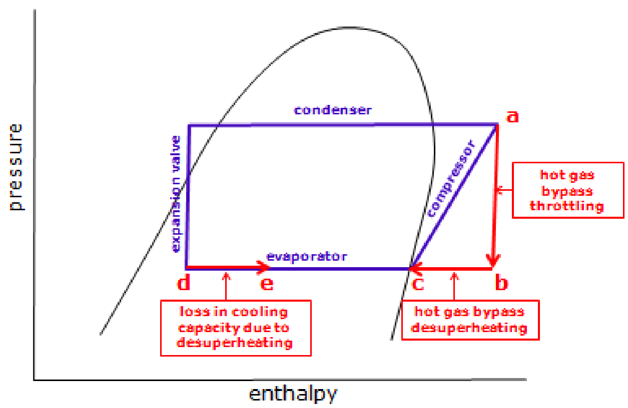


Figure 10: Pressure enthalpy diagram of a simple refrigeration cycle with hot-gas bypass

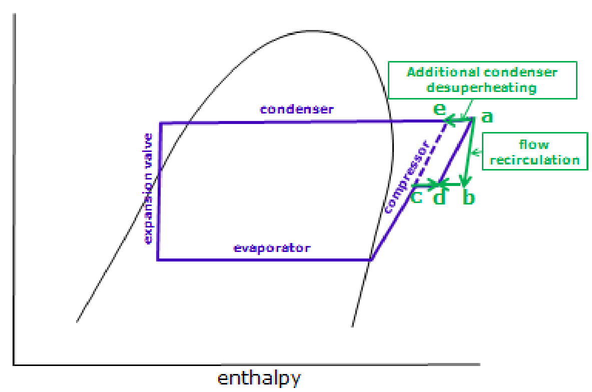


Figure 11: Pressure enthalpy diagram of a two-stage compressor with inter-stage flow recirculation

In the inter-stage flow recirculation arrangement some gas is also extracted from the discharge of the compressor. However, it bypasses not only the condenser, but also the evaporator and the first-stage impeller. Also, the bypass flow is not completely throttled. The recirculation flow enters inlet of the 1st stage vaned diffuser with an energy that is somewhat higher than the flow leaving the first stage impeller in order to take possession of part of the throat area in the vaned diffuser and thus reducing amount of the main flow entering the diffuser. Figure 11 shows the pressure-enthalpy diagram of the flow recirculation concept: The recirculated flow experiences some loss in total pressure as a result of pressure drop over the control valve but it still enters the first stage vaned diffuser inlet with somewhat higher total pressure than the core flow it is joining because it has been compressed by the second stage impeller. The recirculated flow leaving the volute and exiting the nozzles is represented by line a-b in Figure 11. It is also hotter than the core it is merging with at 1st stage vaned diffuser entry. As a result the enthalpy of the merged flow ends up at point d being between the enthalpy of the core flow leaving the first stage impeller (point c) and the recirculated flow (point b). All fluid temperatures downstream of the recirculation nozzles including the compressor exit flow (point e) are somewhat higher when flow recirculation is applied. The additional superheat of the flow leaving the compressor is removed by the condenser and does not cause a reduction in the cooling capacity of the chiller. The only performance penalty of flow recirculation is due to the extra work done by the second stage impeller, part of which is recovered when this flow energizes the core flow. More importantly, the resulting larger second stage flow rate moves that stage closer to its peak efficiency.

5. COMPRESSOR PERFORMANCE MAP

The aerodynamic efficiency of a variable speed centrifugal compressor is a function of volume flow and head (compressor isentropic enthalpy rise). It is typically shown in a plot with volume flow on the horizontal axis and head on the vertical axis (ASHRAE Handbook of Equipment, 2012). When points of equal compressor efficiency are connected they form the so-called efficiency islands on the map (the thin dashed lines in Figure 12) and connecting points of equal rotational speed create the so-called speed lines (the solid lines in Figure 12).

Part-load off-design relative efficiency is similar whether inlet guide vanes or inter-stage recirculation is used. Closing the inlet guide vanes reduces the first-stage impeller incidence losses. This comes at the price of some throttling losses over the inlet guide vanes. Moreover the components downstream of the first stage impeller still run at lower-efficiency off-design flow conditions. The inter-stage recirculation off-design control concept will have higher first stage impeller incidence losses and higher second stage impeller work input due to the compression of the recirculated flow. However, the flow recirculation creates a higher flow rate downstream of the first stage impeller resulting in more efficient and more stable compressor operation in that part of the compressor. The net result has been very similar relative part-load efficiencies of these two off-design control concepts. Actual design as well as off-design compressor efficiency of the two-stage compressor with flow recirculation is higher as a result of the replacement of the vaneless diffuser with a vaned diffuser. The relative compressor part-load efficiency is substantially higher for inter-stage flow recirculation than for hot gas bypass.

At reduced flow a centrifugal compressor will eventually surge. Figure 12 shows the compressor performance map with three surge lines. The first surge line which occurs at the highest part-load flow is for a variable-speed fixed-geometry compressor, the second one is for a variable-speed centrifugal compressor with variable-geometry inlet guide vanes and the third surge line occurs when the fixed-geometry variable-speed centrifugal compressor uses inter-stage flow recirculation.

The compressor map also shows the location of the four test points defined by ANSI/AHRI Standard 550/590 to determine the integrated part-load value of a centrifugal chiller. As can be seen from the map, the inter-stage flow recirculation concept is the only technique that can reach these four points comfortably.

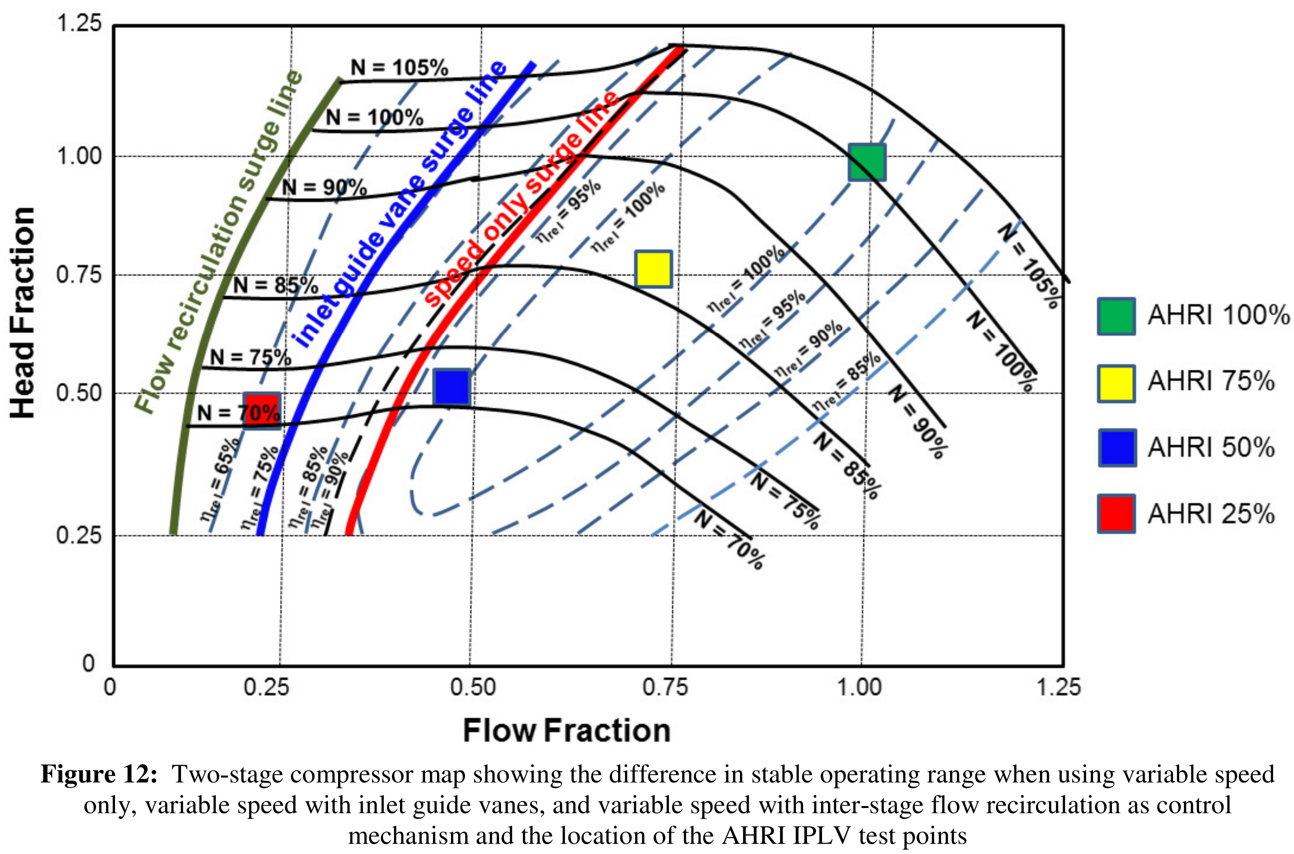


Figure 12: Two-stage compressor map showing the difference in stable operating range when using variable speed only, variable speed with inlet guide vanes, and variable speed with inter-stage flow recirculation as control mechanism and the location of the AHRI IPLV test points

6. CONCLUSIONS

The inter-stage flow recirculation concept described in the paper allows

a. Higher full-load two-stage compressor efficiency as a result of the use of fixed geometry vaned diffusers instead of vaneless diffusers, or variable-geometry vaned diffusers.
b. Better turndown / larger stable operating range than two-stage centrifugal compressors with variable geometry inlet guide vanes.
c. Replacement of internal variable-geometry hardware (inlet guide vanes and/or variable diffusers) with an externally mounted valve, reducing mechanical complexity and thus improving reliability.

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