Performance Analysis of a Two-Stage Refrigeration Centrifugal Compressor With Variable Inlet Guide Vanes on Both Stages

M. Cambio

The Trane Company Engineering Technology

Follow this and additional works at: https://docs.lib.purdue.edu/icec

https://docs.lib.purdue.edu/icec/1370

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.
Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html
Performance Analysis of a Two-Stage Refrigeration Centrifugal Compressor
With Variable Inlet Guide Vanes on Both Stages

Matt Cambia, Project Engineer, Compressor Technology
The Trane Company Engineering Technology, La Crosse, WI 54601

ABSTRACT

A two-stage refrigeration centrifugal compressor with variable inlet guide vanes on each stage is analyzed to determine an optimum unloading control strategy. The optimization is based on minimizing annual operating costs of the compressor using an industry accepted load line definition and performance analysis method. This report documents the analytical methods used as well as applicable assumptions and constraints. Alternative guide vane control strategies are investigated and practical methods for achieving the best result are reported.

NOMENCLATURE

- $U_1$: Impeller blade speed at inlet
- $U_2$: Impeller blade speed at exit
- $V_1$: Absolute velocity at impeller inlet
- $V_2$: Absolute velocity at impeller exit
- $V_{ao}$: Absolute axial velocity upstream of IGV
- $V_{r1}$: Axial velocity at impeller inlet
- $V_{r2}$: Radial velocity at impeller exit
- $V_{o1}$: Tangential velocity at impeller inlet
- $V_{o2}$: Tangential velocity at impeller exit
- $W_1$: Relative velocity at inlet
- $W_2$: Relative velocity at exit
- $\Delta h_{ideal}$: Ideal enthalpy rise per unit mass
- $\alpha$: Angle of attack

INTRODUCTION

Variable guide vanes at the inlet of a centrifugal compressor stage are used as an effective means of unloading an impeller. For each guide vane setting the impeller will have a unique pressure-flow characteristic. In a two-stage compressor, inlet guide vanes (IGV) on the 1st stage have virtually no effect on the 2nd stage impeller characteristic. The application of 2nd stage IGV requires developing a strategy that optimizes the refrigeration cycle efficiency. Considering a refrigeration cycle with an economizer, if you assume equivalent stage efficiency, this intuitively suggests keeping the economizer centered in the cycle.

As a water chiller is unloaded in the refrigeration cycle there are industry standard head requirements at part load, referred to as load lines. There are two basic load line schedules. The first is without entering condenser water relief and the second is with entering condenser water relief. Each load line has an optimum 2nd stage IGV schedule. Industry standard weighting factors are assigned to part load points and a single part load value can be calculated [1]. This paper will examine both load lines and determine an optimum vane schedule for each based on maximizing the part load values.
Inlet guide vanes (IGV) are widely used as an effective means of unloading a centrifugal compressor. Their function is to impart a tangential velocity to the incoming flow. Fig. 1a) depicts an axial inlet guide vane where the incoming axial velocity \( V_{\alpha 0} \) enters a cascade and the resultant velocity vector \( V_1 \) has axial and tangential components, \( V_{a1} \) and \( V_{\alpha 1} \), respectively. Fig. 1b) represents the velocity triangle at the impeller inlet. The impeller inlet velocity triangle is composed of \( V_1 \) in addition to the impeller blade speed \( U_1 \) and the resultant relative velocity vector \( W_1 \). Fig. 1c) shows the outlet velocity triangle for a centrifugal impeller with a backward swept blade. The resultant relative velocity vector \( W_2 \) leaves at roughly the impeller exit blade angle and is composed of two components: the impeller exit blade speed \( U_2 \) and the absolute velocity \( V_2 \). \( V_2 \) can be broken down further into radial and tangential components, \( V_{r2} \) and \( V_{\alpha 2} \).

The velocity triangles described above define the work input of a centrifugal impeller. The enthalpy rise, or work input is governed by the Euler equation [2].

\[
\Delta h_{\text{ideal}} = \frac{1}{g_c} (U_2 V_{r2} - U_1 V_{\alpha 1})
\]  

By creating an inlet tangential velocity component \( V_{\alpha 1} \), or prewhirl, the work input of the impeller can be reduced. Without \( V_{\alpha 1} \), the quantity \( U_1 V_{\alpha 1} \) is equal to zero and work input is simply a function of \( U_2 V_{r2} \). By not using some means introducing prewhirl, the only way of taking work out of the impeller would be through reducing \( U_2 \). This would require a variable speed compressor, a higher cost approach.

Further examination of Fig. 1c) shows a unique feature of backward swept impellers. When volume flow is reduced through the impeller the component \( V_{\alpha 2} \) decreases proportionally. Because the component \( W_2 \) leaves at the blade exit angle, the component \( V_{r2} \) becomes a larger proportion of the \( U_2 \) component. Therefore, for a backward swept blade, \( V_{r2} \) is inversely proportional to \( V_2 \) and volume flow through the impeller. Thus, impeller work increases with decreasing flow rate.

Fig. 2 shows the inverse relationship between head and flow described above for an impeller with leanback. As the volume flow through the impeller increases the work input decreases. Point 1 in the figure indicates the full load operating point. If prewhirl is introduced upstream of the impeller inlet, impeller work input will be reduced. This will change the operating point from pt. 1 to pt. 2. If the same
amount of prewhirl is introduced and head remains constant then the volume flow through the impeller must be reduced, which shifts the operating point to pt. 3. For the vane setting which creates this amount of prewhirl a new impeller characteristic is generated called a vane curve.

The introduction of prewhirl is the initial effect of IGV rotation. If the guide vanes are continually rotated, at some point the angle of attack (α in fig. 1a) will become so great that separation will occur. Most likely at this point the pressure drop across the vanes becomes much greater and they become a throttling device. When throttling occurs, there is a pressure drop across the vane along with a drop in density. As mass flow through the cascade is reduced, the drop in density will recover a portion of the volume flow rate. This mechanism allows the stage to be pushed to lower suction volumetric flow rates.

The Two Stage-Refrigeration Cycle With Economizer

Fig. 3 is a sketch of the refrigeration cycle on a pressure-enthalpy (P-h) diagram. Pt. 1 is the compressor suction state point. Refrigerant vapor is compressed through the first stage impeller to state pt. 2. Isentropic compression is shown by state pt. 2'. Economizer vapor is injected from pt. 8 upstream of the 2nd stage compression and the mixed out state point is shown by state pt. 3. The 2nd stage of compression ends at state pt. 4 and the compressor discharge gas dumps into the condenser. The vapor is condensed from state pt. 5 to state pt. 6 where it enters the condenser orifice. The liquid expands through the condenser orifice to state pt. 7 and the flash vapor at state pt. 8 is vented to the interstage of the compressor. The liquid is expanded further from state pt. 9 through the evaporator orifice to state pt. 10. In the evaporator the flash gas goes directly to the compressor and the liquid refrigerant is boiled off to create the compressor suction vapor at state pt. 1 and the cycle is repeated.

The value of the economizer is that a portion of the flash gas generated from the expansion process is routed through only one stage of compression. In addition, the refrigeration effect of the evaporator is increased due the quality at state pt. 10. Optimum refrigeration cycle efficiency is achieved when the economizer is roughly centered in the cycle. At part load however, compressor stage efficiencies will be changing at different rates due to stage components being off design, including IGV. Optimizing a single part load value based on power and capacity (kW/ton) will be determined by some combination of economizer location and compressor stage efficiency.
Compressor Unloading with Second Stage Guide Vanes

Fig. 4 is a cross section of a typical two-stage compressor with dual IGV's. The 2nd stage IGV's are located in the return channel. The variable vane is the second blade in a tandem radial cascade. The flow field through the cascade has an accelerating tangential component associated with conservation of angular momentum.

One advantage to not having 2nd stage guide vanes is obvious. The vane assembly adds parts which increases material and assembly costs. Leak paths are also introduced which create the need for additional seals. There are also wear concerns of seals and bearings associated with the linkage. Therefore, if the second stage guide vanes were not necessary, a substantial amount of assembly time and material cost could be saved.

A test program was conducted to evaluate variable 2nd stage guide vanes. The test vehicle was a two-stage, fixed-speed compressor with inlet guide vanes on both stages. The compressor was assembled so that IGV's could be controlled independently. This allowed the 2nd stage IGV to be held fixed while the 1st stage was variable, in addition to varying 2nd stage along with 1st stage IGV. During testing, compressor vane curves were taken with both a fixed and variable configuration.

The variable 2nd stage vane schedule was constrained at two points. When the 1st stage vanes were wide open, the 2nd stage IGV must be at the design setting. The second constraint was when the 1st stage vanes are closed, the 2nd stage vanes should also be closed. During testing the vane schedule varied linearly between these two constraints.

Turbomachinery performance is typically presented in a map showing head rise and efficiency as a function of flow rate. In this paper, head rise is expressed relative to the full load rating point. The flow axis is defined by volume flow rate, also relative to the full load. Performance maps in the form of head vs. volume flow rate are shown in Fig. 5. In the maps in Fig. 5, the parameter shown with the heavier lines is the first stage IGV setting. Islands of constant efficiency (relative to full load rating point efficiency) are shown with the lighter lines. Fig. 5a is an overall compressor map with variable first stage and fixed second stage guide vanes (called the fixed geometry case). Fig. 5b is a map for variable first
and second stage IGV's (called the variable geometry case). The guide vane curves in Fig. 5 represent the same first stage vane settings.

Overlaid on the compressor vane curves are two load lines. The load lines originate at the compressor's full load rating point. In addition to full load requirements, capacity modulation is required along industry standard a load lines. Along load line #1 entering condenser water remains constant as the water chiller cooling capacity is reduced.

Load line #2 is an example where condenser water relief is applied. As the water chiller is unloaded, the entering condenser water temperature is reduced. This is a typical scenario for a single chiller in a comfort cooling application. As outside ambient temperature drops, the cooling load is reduced. The relief schedule is specified in reference [1]. Operating points are marked for 100%, 75%, 50% and 25% load. Performance for a load line can be characterized by a weighted average as defined in reference [1]. A single number, called integrated part load value (IPLV), expressed in terms of kW per ton of refrigeration, is calculated. Reference [1] does not address the unloading schedule defined by load line #1. In this study, for load line #1, a part load value (PLV) will be calculated in the same manner as IPLV.

In this section part load values will be expressed as relative to full load kW/ton. The PLV of load line #1 is 1.279 for the fixed geometry while the variable geometry has PLV of 1.226, a 4.1% improvement over the fixed geometry. The IPLV of load line #2 is .952 with fixed geometry and .921 with variable geometry. Variable geometry improves IPLV by 3.3% along load line #2 when compared to the fixed geometry. It is interesting to note the IPLV improvements along load line #2 in the variable geometry case even though the compressor efficiency is lower. Compressor efficiency and cycle performance are discussed in more detail in the remainder of this section.

Comparing load line #1 for the two cases we see that the variable geometry operates at higher efficiency at each load point than the fixed geometry. The variable geometry also requires less 1st stage vane closure than the fixed geometry to reach each load point. The remaining IGV travel will allow the variable geometry to get to lower loads. An examination of load line #2 shows slightly different characteristics. Compressor efficiency for the variable geometry is higher at 75% load, but down at 50% and 25% loads relative to the fixed geometry. Again, the variable geometry shows more range than the fixed geometry case.
Looking at what is taking place in the cycle will offer insight into the efficiency differences along load line #2. Fig. 6 is a contour plot of percent 1st stage pressure ratio relative to overall pressure ratio. Fig. 6a) shows that for the fixed geometry at 50% and 25% load, the 1st stage is making less than 10% of the overall pressure rise of the cycle. To a lesser extent the exact opposite is occurring in the variable geometry case. At the 50% load point along load line #2 the first stage is making roughly 70% of the overall pressure rise.

![Fig. 6: First Stage Pressure Ratio as a Percentage of Overall Pressure Ratio](image)

**a) Fixed Geometry**  
**b) Variable Geometry**

Fig. 6 shows what is happening in the cycle at these extremes. Fig. 7 a) shows the large pressure drop across the 1ST stage IGV that needs to occur in the fixed geometry case in order to take out the work input of the 1ST stage impeller. Work can not be taken out of the 2ND stage impeller and as flow is reduced the backward swept blade causes work input to go up. This drops the economizer pressure near evaporator pressure. The 1ST stage of compression, which includes the IGV, has very little pressure rise but generates entropy which results in a stage efficiency that can be zero or negative.

![Fig. 7: Refrigeration Cycle at Part Load](image)

**a) Fixed Second Stage IGV**  
**b) Variable Second Stage IGV**

Fig. 7 b) shows the other extreme where there is more throttling done by the second stage guide vanes. This causes the economizer pressure to rise in the cycle and approach the condensing pressure. With a large pressure differential between the economizer and evaporator more flash gas will be generated in the expansion process through the orifice and this will reduce the refrigeration effect of the cycle. Based on the two test cases presented it seems that an optimization should occur between compressor efficiency and economizer location in order to maximize cycle efficiency.
Revisiting Fig. 7a), a drawback to dropping the economizer pressure near evaporator pressure is that the lack of a pressure drop across the evaporator orifice will hold liquid refrigerant in the economizer, which can cause liquid carryover to the 2nd stage. In fact, this is what actually happened in the laboratory. The two vane curves in Fig. 5a) at the lowest volume flows pulled the economizer near enough to evaporator pressure to induce liquid carryover into the 2nd stage. This caused problems with accurately measuring the 2nd stage discharge temperature. When the carryover gets severe enough liquid refrigerant will make it to the discharge of the compressor and come in contact with the total temperature probes. This will give an artificially low discharge temperature and in turn 2nd stage adiabatic efficiency will be calculated artificially high. This is the case in Fig. 5a) where the overall efficiency level for the fixed geometry at 50% and 25% load is being effected by second stage carry over.

Optimization of Second Stage Guide Vanes

The test program generated 1st and 2nd stage compressor maps that are independent of each other. The compressor maps can be loaded into a computer program that calculates chiller performance. Guide vane schedule can then be manipulated to achieve different results. Two approaches to adjusting IGV schedule will be discussed further. The first approach will be to keep the economizer centered in the cycle and the second approach will be to maximize overall compressor efficiency.

![Fig. 8: Optimized IGV Schedules](image)

Fig. 8 is a chart of 2nd stage vane schedule for the two approaches. For all cases the 2nd stage schedule is more open than the original linear schedule. The chart shows that each load line requires a different vane schedule in order to center the economizer or maximize compressor efficiency. The original linear schedule is plotted for reference. In order to center the economizer the 2nd stage guide vanes have a much slower rate of closure compared to trying to achieve maximum efficiency. The 2nd stage guide vanes close the slowest for load line #1 without condenser water relief. At 50% and 25% load maximum efficiency is achieved very near the original linear vane schedule.

Table 1 lists the kW/ton (relative to the full load rating point) for the two load lines and all four schedules. The linear schedule shows 4.1% and 3.3% improvements in part load values over the fixed vane schedule for load line #1 and #2, respectively. Centering the economizer in the cycle improved part
load values by 2.3% and 2.9% for load line #1 and #2, respectively. However, it did not make improvements over the original linear schedule. The vane schedule that optimized overall compressor efficiency demonstrated the greatest improvement over the fixed geometry improving part load values by 4.2% and 4.5% along load lines #1 and #2, respectively.

<table>
<thead>
<tr>
<th>Load (%)</th>
<th>Fixed</th>
<th>Linear</th>
<th>Cent. Econ.</th>
<th>Best η</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>1.000</td>
<td>1.000</td>
<td>1.000</td>
<td>1.000</td>
</tr>
<tr>
<td>75</td>
<td>1.093</td>
<td>1.069</td>
<td>1.079</td>
<td>1.061</td>
</tr>
<tr>
<td>50</td>
<td>1.354</td>
<td>1.287</td>
<td>1.311</td>
<td>1.294</td>
</tr>
<tr>
<td>25</td>
<td>2.170</td>
<td>1.915</td>
<td>2.058</td>
<td>1.912</td>
</tr>
<tr>
<td>PLV</td>
<td>1.279</td>
<td>1.226</td>
<td>1.249</td>
<td>1.225</td>
</tr>
<tr>
<td>Improvement</td>
<td>BASE</td>
<td>4.1%</td>
<td>2.3%</td>
<td>4.2%</td>
</tr>
</tbody>
</table>

Table 1: kW/Ton Analysis of Inlet Guide Vane Schedules.

CONCLUSION

Variable 2nd stage guide vanes have the capability of improving part load efficiency values by a significant amount. This study shows more than a 4% improvement when compared to a fixed geometry approach. Refrigeration cycle efficiency seems to be maximized when overall compressor efficiency is maximized. This does not necessarily coincide with centering the economizer in the refrigeration cycle. The variable geometry also provides added range during unloading down to 10% of full load, which can be an additional customer requirement.

2nd stage variable vanes also provide a better-balanced system, which minimizes the occurrence of economizer carryover. Carryover will dramatically increase power consumption and possibly have detrimental effects on reliability. Another feature of 2nd stage variable vanes not mentioned is the benefit of shutting off economizer vapor flow during start-up. This reduces the requirement of motor starting torque or eliminates the need for valves.

The best method for achieving maximum cycle efficiency would be to independently control 1st and 2nd stage IGV. However, an acceptable compromise could be reached between optimum schedules which would produce near maximum cycle efficiency.

ACKNOWLEDGMENTS

The work presented here is based on experimental work performed at The Trane Company of La Crosse, Wisconsin. Permission to publish the results is greatly appreciated. The author would also like to thank Mr. Jack Sauls of The Trane Company for offering his technical and editorial assistance.

REFERENCES
