

1996

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J. J. Brasz  
*Carrier Corporation*

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Brasz, J. J., "Aerodynamics of Rotatable Inlet Guide Vanes for Centrifugal Compressors" (1996). *International Compressor Engineering Conference*. Paper 1196.  
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# Aerodynamics of Rotatable Inlet Guide Vanes for Centrifugal Compressors

Joost J. Brasz  
Carrier Corporation  
Syracuse, NY 13221

## Abstract

*Fixed speed motors are the most common drivers for centrifugal compressors encountered in large air-conditioning and refrigeration units. Capacity is altered by changing the swirl entering the impeller. This is accomplished by rotatable inlet guide vanes. However, part-load operation of constant-speed centrifugal compressors with rotatable inlet guide vanes suffers from a substantial drop in compressor efficiency. When variable speed is used as a part-load control mechanism, centrifugal compressors suffer a much smaller efficiency reduction during part-load operation. The inlet guide vanes are therefore often blamed for the drop in efficiency of fixed speed compressors under part-load conditions. In order to verify this hypothesis the aerodynamic efficiency of rotatable inlet guide vanes at various setting angles was investigated. The measured mass-average total pressure loss over the inlet guide vanes was very moderate under all but the most extreme guide vane setting angles. This small pressure loss was not enough to explain the compressor efficiency drop encountered under part-load operating conditions. Modifying inlet guide vane geometry will therefore not help in improving part-load compressor efficiency.*

## Introduction

Figures 1 and 2 show the performance maps of identical compressors with different control mechanisms. The map resulting from variable speed compressor control is shown in Figure 1 and the map resulting from fixed-speed variable inlet guide vane control is shown in Figure 2. Comparison of these two performance maps shows two important differences:

1. The substantial drop in part-load efficiency of the variable inlet guide vane compressor compared to the almost constant compressor efficiency at reduced flow rate of the variable speed compressor. For example, at 25 percent flow rate the variable speed compressor can still achieve peak design point efficiency while the variable inlet guide vane compressor suffers at least a 45% reduction in efficiency.
2. The much steeper surge line of the variable-speed controlled compressor compared to the variable-inlet-guide-vane controlled machine. For example, at 25 percent flow rate the variable-speed compressor surges at 55% of the full-load head while the same compressor with variable-inlet-guide-vanes can reach 80% of full-load head.

Application of variable speed on centrifugal chillers is limited to less than 5% of the installed base of centrifugal chillers. The three main reasons are:

1. The inherently steep surge line of a variable speed centrifugal compressor prohibits installation of centrifugal chillers with only variable speed control. Inlet guide vanes are still required to meet most part-load operating conditions. Figures 1 and 2 show the constant condenser inlet temperature load line of a centrifugal chiller. This load line represents applications with a predominantly internal load in humid climates. For this application the variable speed controlled machine would go into surge at 60% capacity while the variable inlet guide vane controlled compressor could be turned down 40% capacity before reaching surge. If turn-down to 20% capacity were required for this application with constant condenser water inlet temperature, the full-load compressor selection point has to be at 90% full-load head, causing a 4% drop in full-load efficiency. If a similar turndown is required with variable speed the full-load compressor selection has to be (according to Figure 1) at 60% full-load head, causing a

substantial (15-20%) drop in full-load efficiency. Obviously, variable speed control is not practical for the constant condenser water inlet temperature applications.

2. The cost of the inverter is relatively high. Inverter costs in excess of total compressor costs are not uncommon.
3. The improved aerodynamic part-load efficiency does not translate into an equally large reduction in operating cost. Variable speed introduces a full-load performance penalty due to the inevitable losses of the inverter (inverter efficiencies range from 93-97%). Also, induction motor efficiency typically drops slightly (1-2%) due to the non-sinusoidal shape of the wave form output coming from the frequency converter. Therefore, roughly 5% of compressor efficiency improvement is required to overcome the added losses due to inverter inefficiency and added motor inefficiency. As can be seen from comparing the efficiency islands of Figures 1 and 2, much more than 5% efficiency improvement can be achieved with variable speed at part load. However, the 5% performance penalty of variable speed at full-load operating conditions requires for those parts of the country where the electrical rate structure is such that annual demand charges equal or exceed annual energy consumption charges, that the first 5% in operating savings is needed to compensate for the increase in demand charges. Consequently, the first 10% of the seasonal improvement in compressor efficiency is required to cover the higher demand charges and the losses of the inverter and the reduced motor efficiency.

The almost constant aerodynamic efficiency obtained under part-load conditions with variable speed control compared to the relatively poor part-load efficiency obtained with variable inlet guide vane control load line suggests that the guide vanes are to blame for the observed poor part-load efficiency. It is often stated that after more than 45 degrees of turning, the guide vanes start acting as a throttling valve. The suggestion then becomes that centrifugal compressor part-load efficiency can be improved by redesigning the inlet guide vanes.

When efforts at Carrier to improve part-load efficiency by redesigning the inlet guide vanes were not successful, a more fundamental program was undertaken to experimentally determine the contribution of the inlet guide vanes on part-load efficiency deterioration. This paper describes that study.

### **Experimental Set-Up and Procedure**

The total pressure drop from the evaporator till after the inlet guide vanes will indicate the loss incurred by the inlet guide vanes. Eight rakes with a total of 25 Kiel probes custom made by United Sensor were installed after the 7 inlet guide vanes (see Figure 3). The installation took place on an 500 ton HCFC22 chiller.

Assuming circumferential uniformity of the inlet flow this arrangement allows an area-averaged measurement of the total pressure after the inlet guide vanes (see Figure 4). The rakes were mounted at 4 different setting angles (90, 60, 30 and 0 degrees from tangent). For each of the four rake setting angles the compressor was run with the following 5 different inlet guide vane setting angles (90, 70, 50, 20 and 5 degrees from tangent). At each inlet guide vane setting angle a data point was taken at choke, midpoint and surge. The end result is a set of  $4 \times 5 \times 3 = 60$  data points. Fifteen identical map points (choke, midpoint and surge with the compressor inlet guide vane setting angle at 90, 70, 50, 20 and 5 degrees from tangent) were taken four times, viz. with the rakes at 90, 60, 30 and 0 degrees from the tangent.

The main interest during these tests was the total pressure drop over the inlet guide vanes at different vane positions. Given the relative angle insensitivity of the Kiel probes ( $\pm 30$  degrees) total pressure drop was defined as the smallest pressure drop measured for any of the four rake setting angles.

Overall compressor efficiency was not considered important. The additional blockage of the probes (see Figure 4) will obviously have a negative influence on overall compressor performance.

## Results

Figures 5 and 6 show the type of results obtained for each of the 60 tests. Figure 5 shows the total pressure profiles after the inlet guide vanes at an inlet guide vane setting angle of 70 degrees from tangent (20 degrees closed) at choke conditions as measured with the rakes installed at a setting angle of 90 degrees from tangent. The vertical axis shows the total pressure measurement of each of the 25 probes on the eight rakes, non-dimensionalized by dividing by the measured evaporator pressure. Pressure measurements at the same radius are connected by solid lines. The open circles connect the eight outer diameter pressure measurements, the triangles the eight medium diameter pressure readings, the plusses the eight inner diameter readings and the star symbol shows the single reading at the center of the duct (see Figure 4). Figure 6 shows the same total pressure profiles at the same inlet guide vane setting angle and the same operating condition but with the rakes installed at a setting angle of 60 degrees from the tangent.

The wake due to the presence of the inlet guide vane could be recognized in all tests by roughly a 1% reduction in total pressure at the corresponding circumferential position.

As soon as swirl was given to the flow (i.e. all measurements except 90 degree inlet guide vane setting) a radial total pressure gradient was observed, becoming stronger with further closure of the inlet guide vanes. This radial total pressure gradient is a consequence of the forced vortex constant flow angle swirl pattern created by the inlet guide vanes.

Figures 7-10 show the area-averaged total pressures at various rake setting angles for different inlet guide vane settings. The drop in total pressure at different inlet guide vane setting angles can be found by using the highest readings from Figures 7 - 10. Consequently, we find a total pressure drop of 1%, 1.3%, 6% and 13% at inlet guide vane setting angles of 90, 50, 20 and 5 degrees. Since compressor power is proportional to compressor head, an identical compressor efficiency drop of 1%, 1.3%, 6% and 13% would be expected at these guide vane settings if the inlet guide vanes were fully responsible for part-load performance deterioration. However, Figure 2 shows substantially larger efficiency drops at these guide vane settings: at least 4% at 50 degrees, 30% at 20 degrees, 50% at 5 degrees, where our experiments show that the inlet guide vanes can only be blamed for an efficiency drop of 0.3% at 50 degrees, 5% at 20 degrees and 12% at 5 degrees. Overall, less than 25% of the part-load efficiency deterioration of centrifugal compressors controlled by rotatable inlet guide vanes can be explained in terms of the loss in total pressure generated by these vanes.

The results from figures 7 - 10 can also be used to determine to what extent the inlet guide vanes keep guiding the flow under more and more closed conditions. In each of these four cases the highest area averaged total pressure is found at a rake setting angle closest to the inlet guide vane setting angle suggesting excellent guidance of the flow.

## Conclusions

1. Aerodynamic losses over the inlet guide vanes contribute for less than 25% to the compressor efficiency drop experienced under off-design conditions.
2. Other components such as the impeller and the diffuser are more responsible for part-load efficiency deterioration than the inlet guide vanes.
3. The inlet guide vanes are guiding the flow with a flow swirl angle close to the inlet guide vane setting angle from fully open (90 degrees setting angle) to almost fully closed (5 degree setting angle).
4. These tests have shown that the intuitive reasoning that inlet guide vanes are acting more and more as throttling devices when further and further closed and therefore cause the observed dramatic drop in part-load efficiency, is incorrect. Modifying inlet guide vane geometry will therefore not help in improving part-load compressor efficiency.

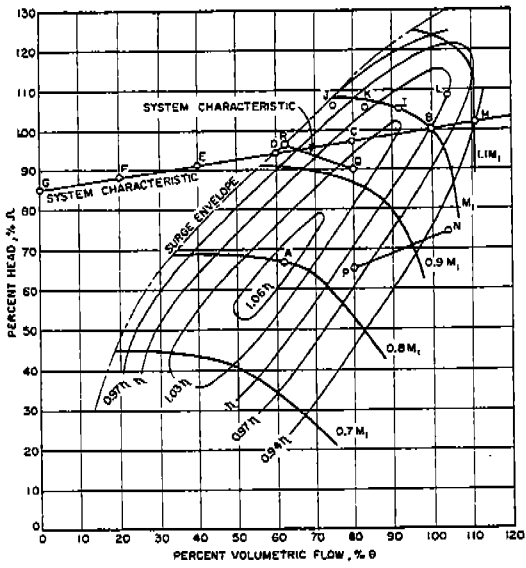


Figure 1. Typical compressor performance at variable speed

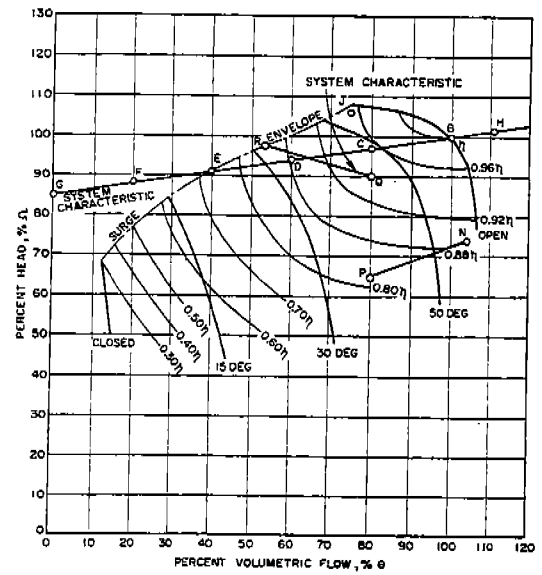


Figure 2. Typical compressor performance with variable inlet guide vanes

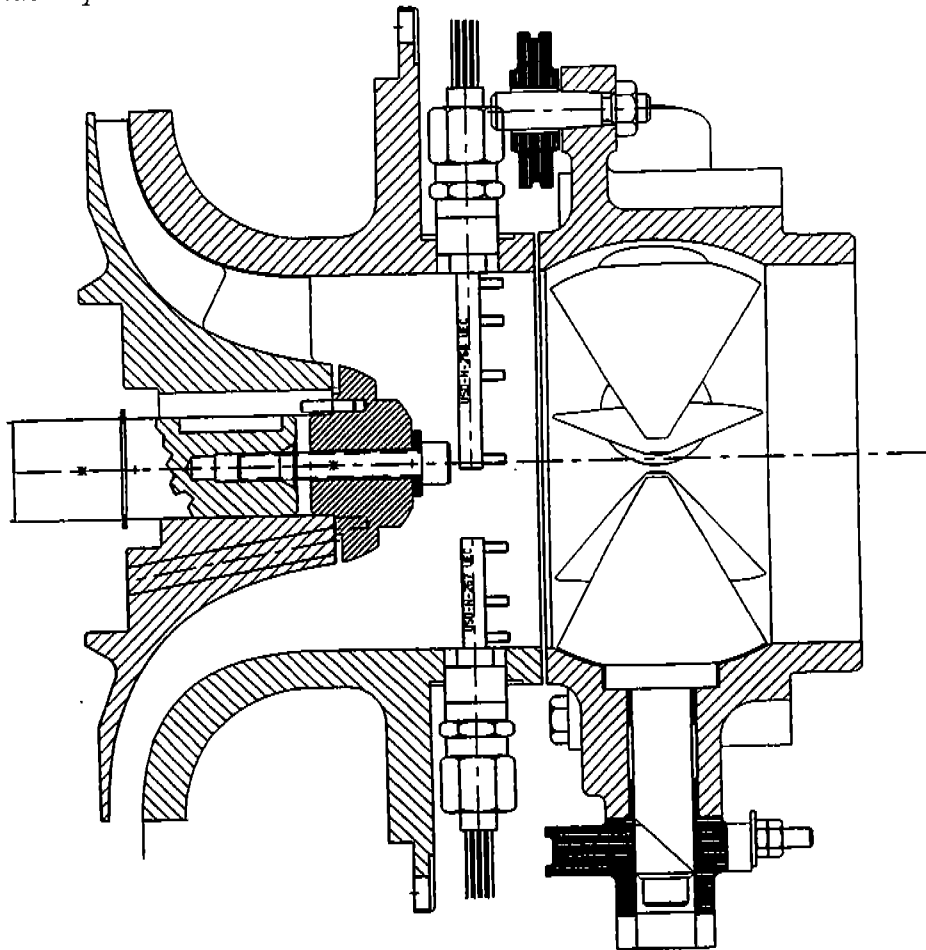
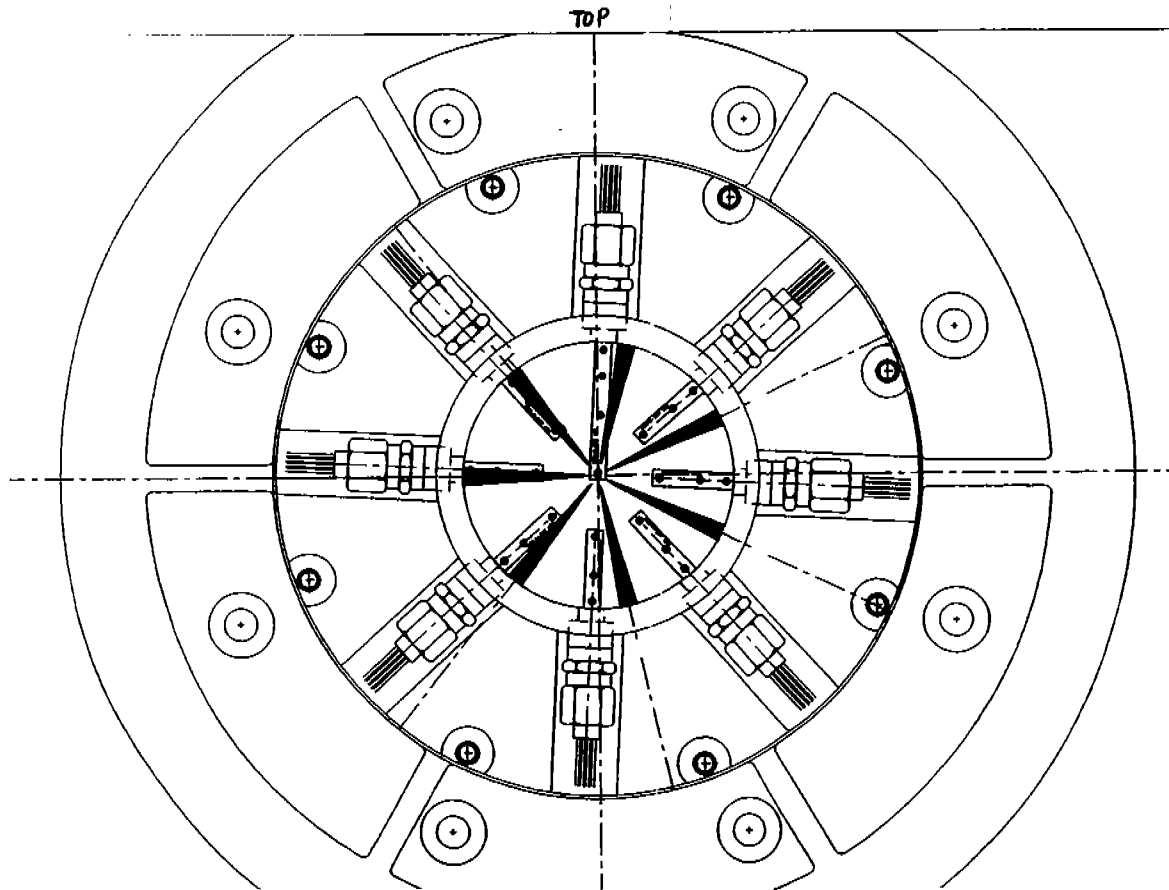
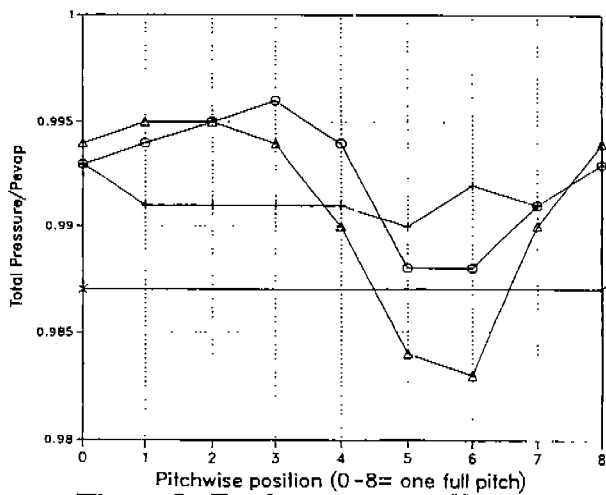


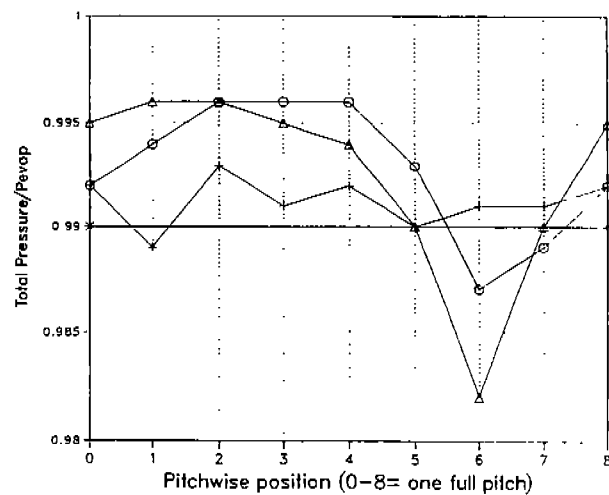
Figure 3. Axial location of total pressure rakes between the variable inlet guide vanes and the impeller.



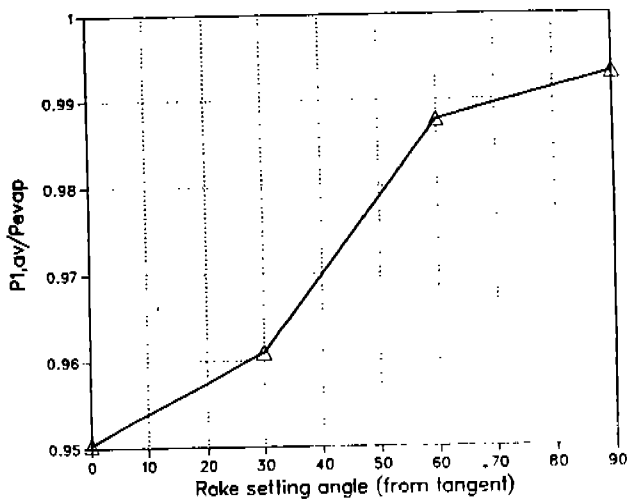
**Figure 4.** Circumferential location of the 8 total pressure rakes relative to the inlet guide vanes.



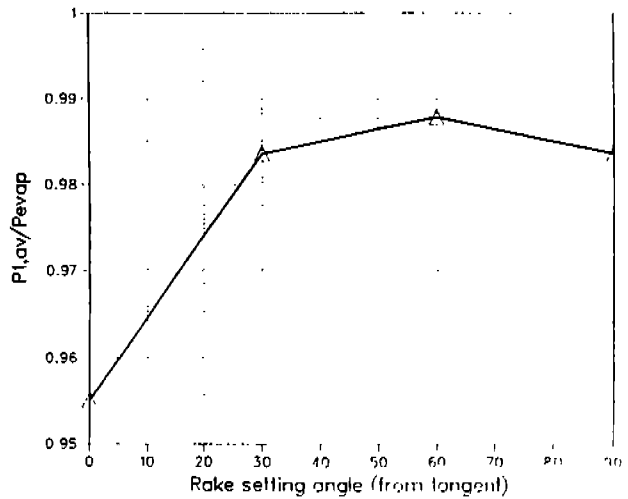
**Figure 5.** Total pressure profile  
Inlet guide vanes 70 degrees  
Rake angle 90 degrees



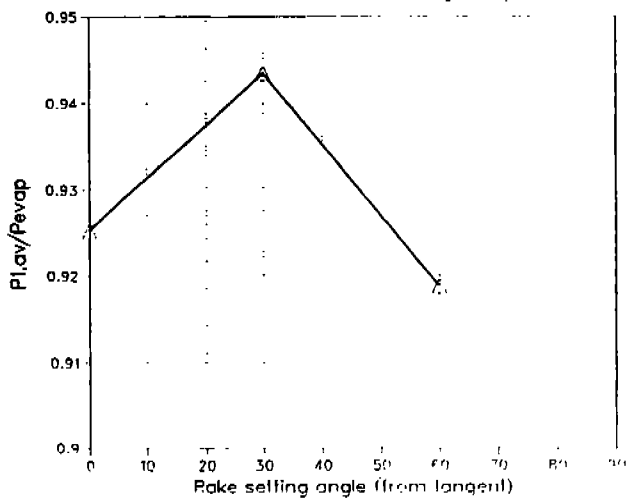
**Figure 6.** Total pressure profiles  
Inlet guide vanes 70 degrees  
Rake angle 60 degrees



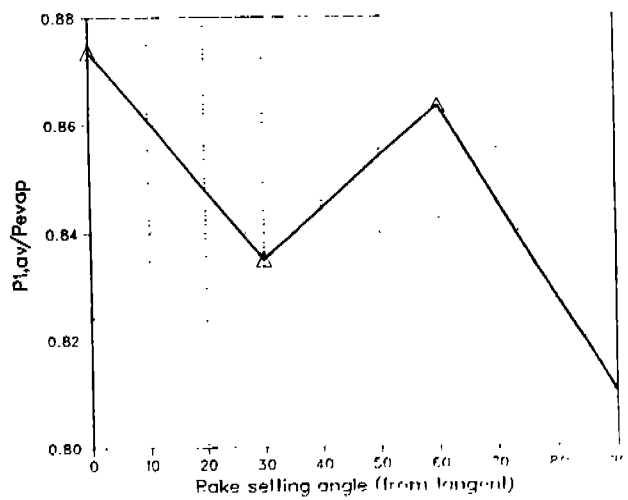
**Figure 7.** Total pressure drop over inlet guide vanes at various rake setting angles  
Inlet guide vanes 90 degrees



**Figure 8.** Total pressure drop over inlet guide vanes at various rake setting angles  
Inlet guide vanes 50 degrees



**Figure 9.** Total pressure drop over inlet guide vanes at various rake setting angles  
Inlet guide vanes 20 degrees



**Figure 10.** Total pressure drop over inlet guide vanes at various rake setting angles  
Inlet guide vanes 5 degrees