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# Control of Self-Excited Flow Oscillations in Vaneless Diffuser of Centrifugal Compression Systems

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*Experiments were conducted to evaluate the effectiveness of axisymmetric diffuser exit throttle in delaying the occurrence of self-excited flow oscillation in vaneless diffusers. Sharp edge rings were installed at diffuser exit in order to change the exit flow area. Tests were carried out with the rings attached to one or both of the diffuser walls. Steady and unsteady flow measurements were used to determine the flow field in the diffuser at the onset of the flow oscillations. Results showed that the occurrence of flow oscillation was continuously delayed as the diffuser exit flow area was reduced for all these configurations and impeller speeds. Comparison between the performance of the compression system with and without diffuser exit blockage indicated that although large losses occur at high flow rates, the use of diffuser exit rings resulted in overall diffuser performance improvement at low flow rates. Retractable diffuser exit rings would therefore be ideal for centrifugal compression systems with vaneless diffuser.*

## NOMENCLATURE

$A_2$ Impeller exit flow area

$p_2$ Static pressure at diffuser inlet

$p_3$ Static pressure at diffuser exit

$p_{t_1}$ Total pressure at impeller inlet

$q$ Equal to $1/2\ \rho\ U^2$

$U$ Impeller wheel speed at exit

$V_2$ Velocity magnitude at diffuser inlet divided by U

$\alpha$ Flow angle at diffuser inlet measured from the tangential direction

$\phi$ Flow coefficient, equal to rate of volume flow/$UA_2$

$\psi$ Pressure rise coefficient; equal to $(p_3-p_{t_1})/q$

$\nu$ Diffuser static pressure rise coefficient; equal $(p_3-p_2)/q$

$\rho$ Fluid density

## INTRODUCTION

The occurrence of self-excited periodic flow fluctuations in centrifugal compression systems under some operating conditions is of major concern to compressor manufacturers and users because of its impact on the system's operating range and level of mechanical vibration. Two types of such flow oscillations have been observed and reported on by previous investigators. The first is a phenomenon in which the flow fluctuations are mainly confined to the impeller and the diffuser of the compressor, with one or more wave fronts which rotate around the compressor axis at a fraction of the impeller speed. The second type of flow oscillations involves every component of the system and is usually termed compression system surge. The fundamental characteristics of surge are self-induced fluctuations in the fluid rate of mass flow through the entire compression system and the axisymmetric phase distribution of the resulting flow fluctuations in the impeller and the diffuser.

Rotating patterns of flow non-uniformity in vaneless radial diffusers have been recently investigated experimentally and analytically [1-5]*. The experimental results (1,2,4,5) showed the following aspects: (a) flow oscillations occur in the entire diffuser at onset; (b) the rotational speed of the pattern is strongly dependent on radius ratio but weakly dependent on diffuser width ratios; (c) for the same diffuser geometry the rotational speed may vary from one experimental set-up to another; and (d) the critical flow angle at onset of unsteady flow patterns increased with diffuser radius ratio. Results of the

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* Numbers in square brackets indicate references at end of paper

analysis [3] were in good qualitative agreement with the experimental measurement and are discussed in detail in the next section.

In this paper a new adaptive technique to eliminate the occurrence of rotating patterns of flow fluctuations in vaneless radial diffusers is presented. The technique was tested experimentally and not only did it eliminate the oscillations but also resulted in significant improvement in the diffuser performance.

## Development of the Control Technique

Two models have been proposed in the past to explain the occurrence of flow oscillations in radial vaneless diffusers. The first [6] linked the existence of the oscillation to the occurrence of reversed flow zones at the diffuser walls while the second [3] demonstrated that flow oscillations can be due to hydrodynamic instability of the core flow in the diffuser. The conditions of flow instability in the second model were shown to be dependent on the flow angle at diffuser inlet, the diffuser radius ratio, the dynamic coupling parameters between the impeller and the diffuser and the diffuser exit boundary conditions.

The idea of the first model at the time of its publication was easy to accept since both flow reversal and fluctuations have been observed to occur as the flow angle at diffuser inlet is gradually reduced. Recent detailed measurements of the phenomena however [1-5] revealed some flow characteristics which contradicts the results of the first model. For example, the occurrence of flow reversal in the diffuser is not a sufficient condition for the onset of flow oscillation; the onset of the oscillations is strongly dependent on diffuser radius ratio; and for the diffuser geometry there are two possible rotating patterns of flow fluctuations with widely different rotational speed. All of these characteristics are in qualitative agreement with the results of the second model [3].

Several techniques are being used to delay the occurrence of flow oscillations in centrifugal compression systems. A moving throttle ring at diffuser inlet is being used on some of Carrier's centrifugal water chillers and a moving diffuser wall is used by Westinghouse on some of their chillers. In the first case it was thought that the oscillations occur when reversed flow in the diffuser reaches the impeller and therefore a throttle ring at diffuser inlet would prevent these conditions from occurring. The moving diffuser wall is used to increase the flow angle at diffuser inlet and thus delay the occurrence of reversed flow in the diffuser. The onset of the oscillations is therefore delayed since the prevailing understanding at the time of implementation of the idea linked flow oscillations to flow reversal. Both of these techniques also affect the core flow distribution in the diffuser and therefore their success is not contradictory to the ideas of the second model.

The idea of the present technique is derived from the results of the earlier investigations [3,5]. Modification of the core flow distribution and or the dynamic conditions at diffuser inlet or exit should affect the onset of the flow oscillations in the diffuser. A movable ring at diffuser exit is used in the present investigations to modify the core flow and the diffuser exit conditions. The use of the ring as demonstrated in this paper stabilized the diffuser flow despite the occurrence of excessive reversed flow zones on the diffuser walls. The use of a moving ring at diffuser exit is advantageous to a moving diffuser wall since the flow velocity at exit is smaller than the flow velocity anywhere else in the diffuser and the mechanical complexity of implementing an exit ring in practice is certainly less than that of a moving diffuser wall.

## Experimental Facility

A schematic diagram of the test facility used in the present investigation is shown in figure (1). The shrouded impeller overall geometry was

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|---|---|
| Impeller exit diameter | 233mm |
| Height to exit radius ratio | 0.116 |
| Inlet radius to exit radius ratio | 0.287 |
| Number of blades | 24 |
| Exit angle of the backward leaning blades measured from the radial direction | $60^{\circ}$ |

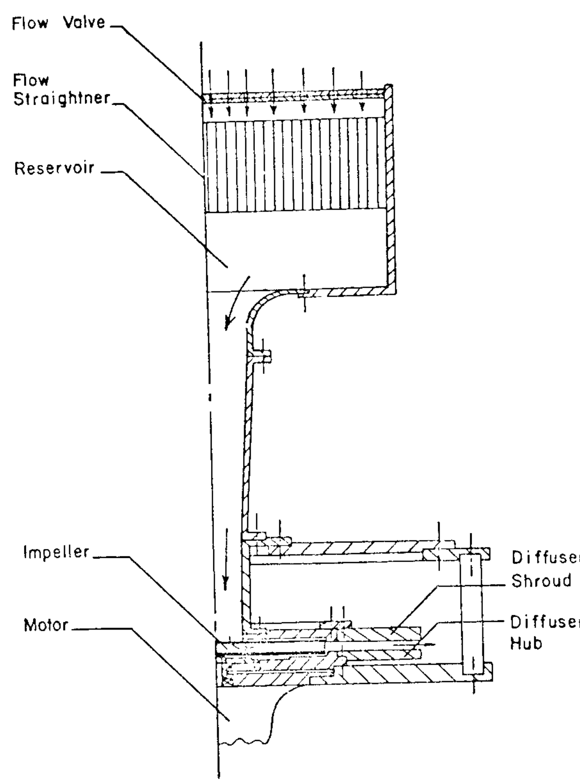


Figure 1: Schematic of experimental test facility.

Throughout the entire test program the impeller was run at a speed of 5000 rpm. The geometry of the vaneless diffuser was

| | |
|---|---|
| Diffuser radius ratio | 1.75 |
| Difference width to inlet radius ratio | 0.116 |

The flow area at diffuser exit was changed using the arrangements shown in figure (2). Rings were attached to the diffuser walls individually or simultaneously. The rings were adjusted such that the diffuser exit area blockage was 0.25, 0.50, 0.75 and 0.85. In the case of rings attached simultaneously to the hub and shroud, the extent of the blockage was identical on both sides.

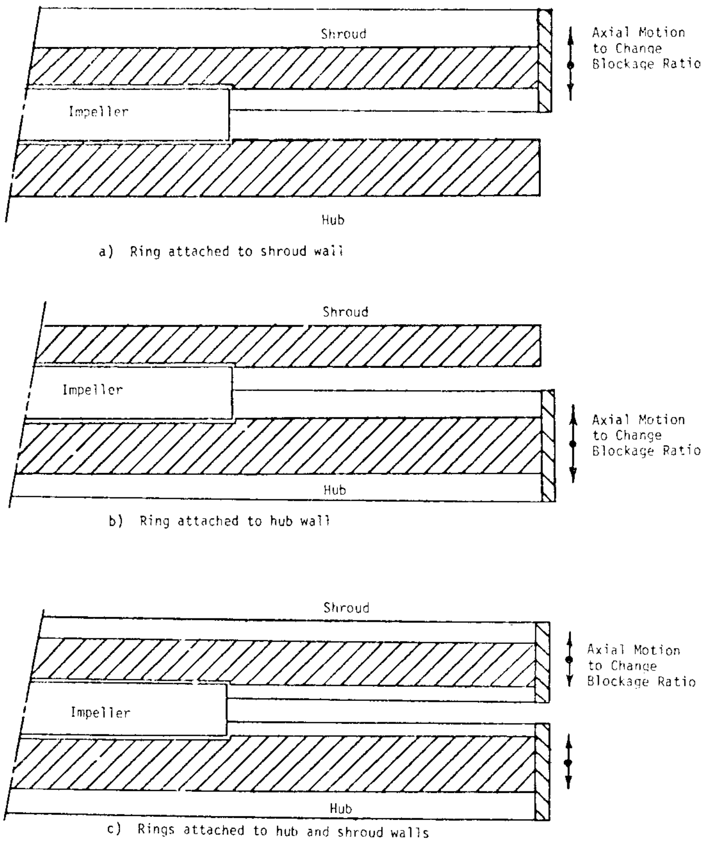


Figure 2: Modes of diffuser exit blockage used in the present study.

The volume flow rate through the system was calculated from measurements of the total and static pressure upstream of the impeller. The flow angle at diffuser inlet was measured using a traversed 3-hole cobra probe which was located at a radius ratio of 1.05. The magnitude and direction of the flow velocity was measured using the probe at seven different locations between the hub and the shroud and a mass average flow angle was calculated from the data. The performances of the impeller and the diffuser were determined from measurements of the static and total pressures at impeller and diffuser inlets and exits.

The onset of self-excited oscillations in the vaneless diffuser was determined from the outputs of two miniature piezo-electric dynamic pressure transducers. The transducers were flush mounted with the shroud wall at the diffuser inlet. The angular separation between the transducers was $41^{\circ}$. The signals from the two transducers were passed through a Hewlett-Packard two channel spectrum analyzer type 3582. Several output functions were recorded from the analyzer such as coherence and amplitude and phase of the cross spectrum between the outputs of the two pressure transducers.

The use of the coherence function to detect the onset of the oscillations proved to be very satisfactory since prior to the generation of the oscillations there was virtually no coherence between the pressure fluctuations in the frequency range of interest to the phenomena. At onset of the oscillations values of the coherence function at the frequency of the oscillation were found to be 0.8 and higher. The phase spectrum at onset of oscillation was used to determine the periodicity of the unsteady flow pattern in the circumferential direction and the rotational speed of the pattern.

For each diffuser exit blockage ratio considered in the present investigations the system was first operated at the maximum flow rate. The flow rate was then gradually decreased and the performances of the system and the diffuser were evaluated at successive values of the rate of mass flow. The procedure continued until self-excited flow fluctuations were generated in the diffuser. The flow conditions just before onset of the oscillations were clearly identified and are termed here "the critical flow conditions". The mass flow rate was further decreased and the performances were evaluated even when the oscillations were present. Compression system surge did not occur under any of the flow conditions investigated in the present study.

## Results and Discussions

Figure (3) shows a comparison between the overall system performance without and with the diffuser exit ring. The horizontal axis is the flow coefficient $\phi$ and the vertical axis is the pressure rise coefficient $\psi$. The ring was attached to the hub-side of the diffuser. Very similar results were obtained when rings were attached to the shroud-side or simultaneously to the hub and the shroud.

The critical flow coefficient at onset of the oscillations for the system without the ring was 0.115. At flow rates higher than 0.115 the use of the ring resulted in degradation of the performance as would be expected. At low flow rates, however, the use of the ring became quite viable. For example, between a flow coefficient of 0.115 and 0.093, the use of 0.25 blockage ratio at diffuser exit increased the pressure rise coefficient $\psi$ and delayed the occurrence of flow oscillations. With 0.50 blockage ratio the critical flow coefficient reduced to 0.066 and with further increase in blockage ratio to 0.75 the system was operated without oscillations all the way down to a flow coefficient of 0.039.

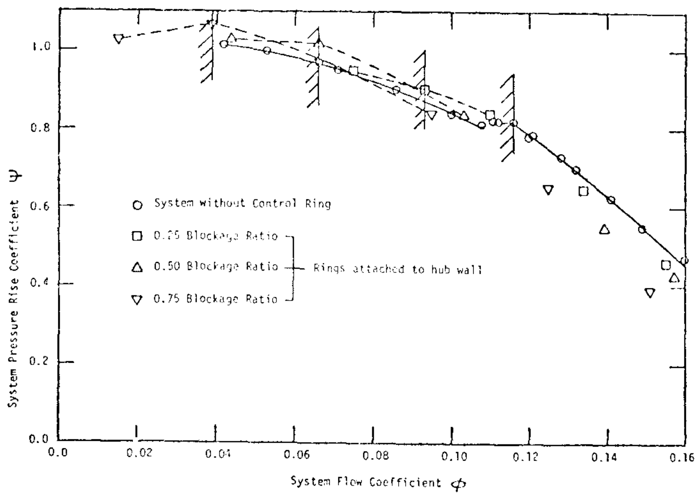


Figure 3: Effect of diffuser exit blockage on compression system performance

The dotted line shown in figure (4) represents the envelope of the effect of diffuser ring on the system performance. Along the envelope no flow oscillations existed in the diffuser. The envelope can be extended to zero flow since when the system was operated with a blockage ratio of 0.85 there was no oscillation observed even at zero through flow in the system.

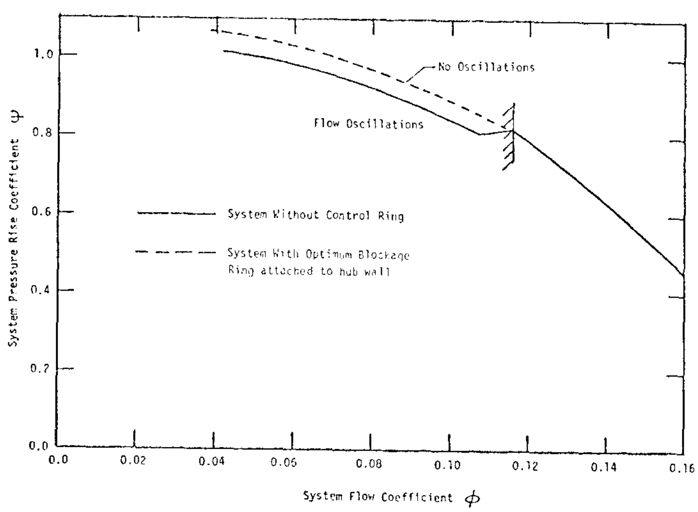


Figure 4: Optimum compression system performance without oscillations.

The effects of the control ring on diffuser performance is shown in figure (5). at low flow rates the effects of the control ring on the stable range and the static pressure rise coefficient were quite significant. Flow oscillations were progressively delayed as the blockage ratio was increased and the diffuser static pressure rise coefficient at some operating conditions was increased by more than 50%. The data also shows that there is an optimum inlet angle at which the static pressure rise coefficient is maximum. A reduction of the flow angle below the optimum value with the appropriate amount of diffuser exit blockage resulted in reduction of the static pressure rise coefficient but the flow in the diffuser was stable.

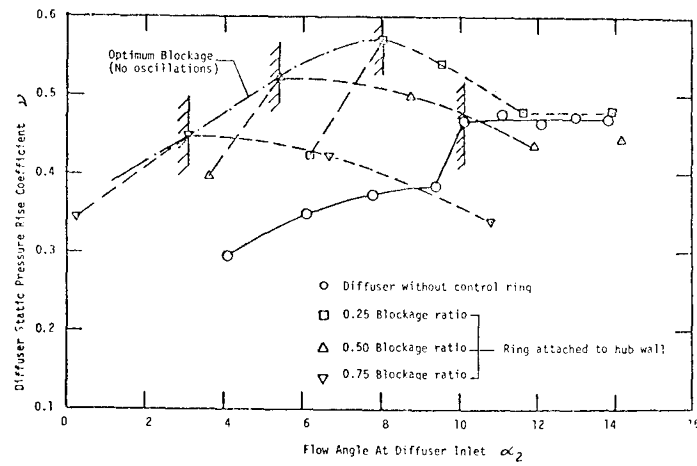


Figure 5: Effect of diffuser exit blockage on vaneless diffuser performance.

The variation of velocity ratio $V_2$ and direction α at diffuser inlet without the use of control rings is shown in figure (6) for three different flow conditions. A mass average flow angle of $10.1^{o}$ was the critical flow angle for the diffuser without rings. The data shows that as the flow rate was decreased the conditions at the shroud wall of the diffuser deteriorated with reversed flow zones reaching the diffuser inlet at the lowest flow angle shown in figure (6).

The effects of the control ring on the velocity profile at diffuser inlet is shown in figure (7). The mass average flow angles for the flow conditions shown were within one degree. The blockage ratio for the data with control rings was 0.75. When the control ring was attached to the shroud side the velocity profile did not change significantly from the one without the ring. The effect of the shroud ring on the diffuser performance however was quite substantial. Without the ring the flow was on the verge of becoming unstable whereas with the ring there was a mass average flow angle margin of $6^{o}$. The effects of the hub ring on the velocity profile is evident. Such an effect was expected however since without the ring most of the through flow in the diffuser was concentrated on the hub side. It should be noted here that despite the major differences between velocity profiles with shroud-side ring and with hub-side ring the stability margin of the diffuser in terms of mass average flow angle at diffuser inlet was almost the same.

The difference between the velocity profiles with hub-side ring and with shroud-side ring decreased as the flow rate was decreased further. Figure (8) shows the flow angle profile at diffuser inlet for two flow conditions with hub-side ring and two other

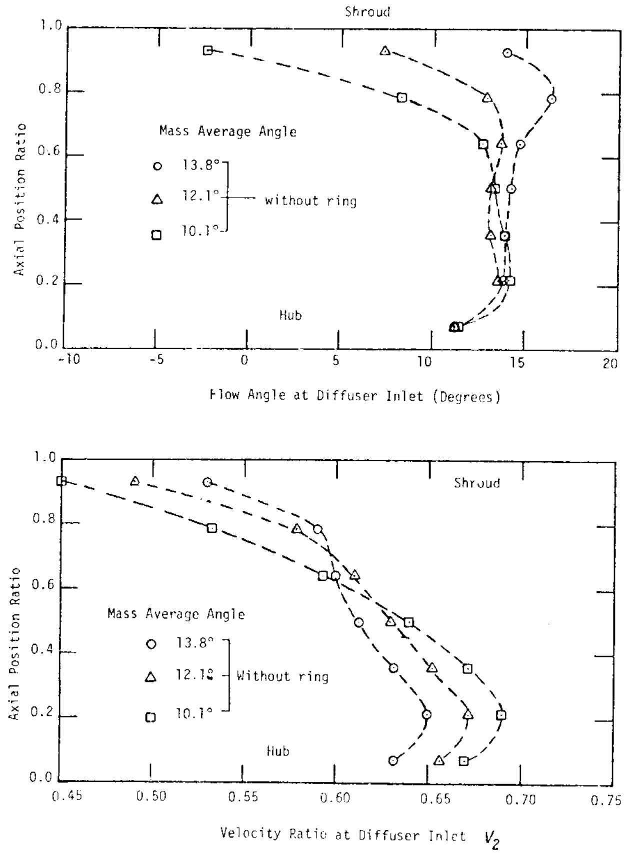


Figure 6: Velocity profile at diffuser inlet without diffuser exit ring

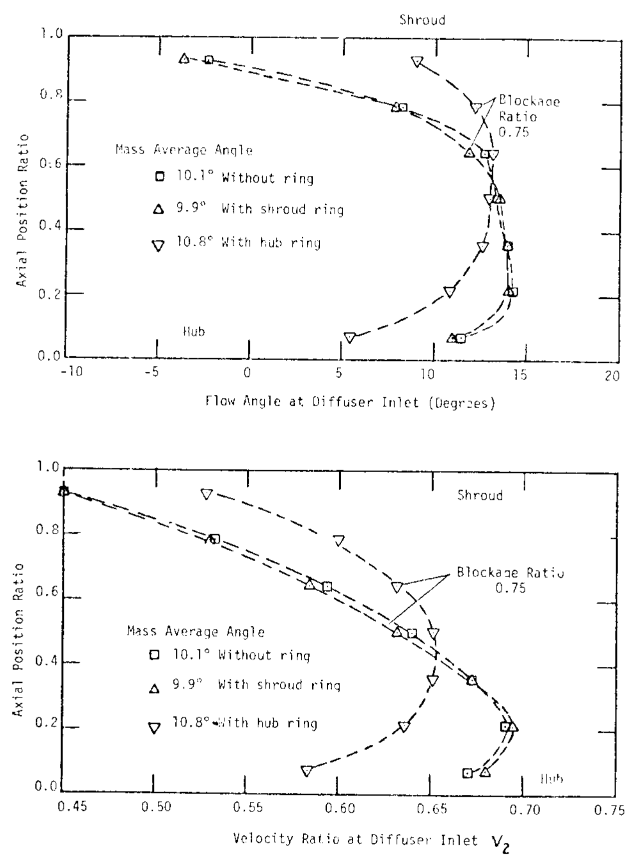


Fig. 7: Effect of diffuser exit blockage on velocity profiles at diffuser inlet

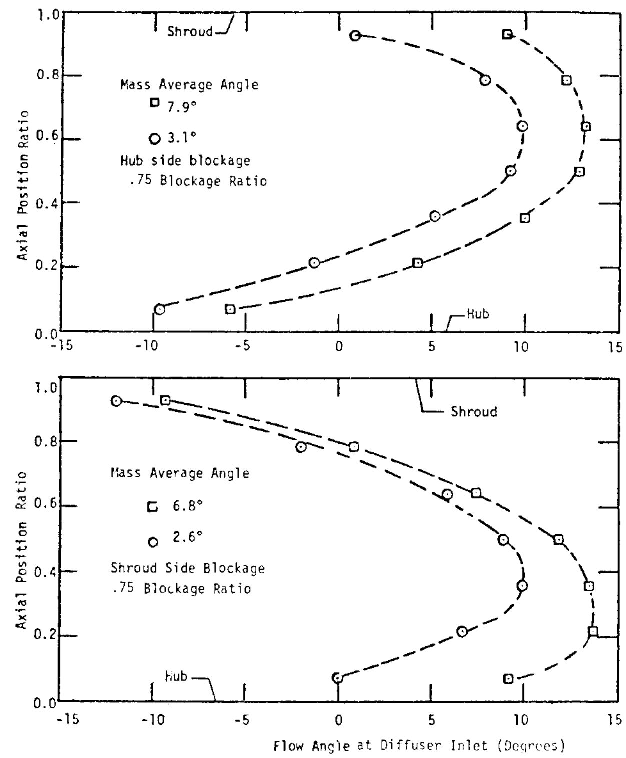


Fig. 8: Comparison between the diffuser inlet velocity profiles with hub-side ring and shroud side ring

reasonably close flow conditions with shroud-side rings. All of these conditions were without flow oscillation in the diffuser. Major flow reversal occurred at the diffuser wall to which the control ring was attached and at the lower flow angles covered more than 20% of the diffuser width at inlet. This result is quite significant because it clearly dismisses the ideas that flow oscillations occur when reversed flow exists in the diffuser or when the reversed flow reaches the impeller exit.

The variation of the critical mass average flow angle at diffuser inlet with blockage ratio is shown in figure (9) for hub-side blockage, shroud-side blockage and simultaneous hub-side and shroud-side blockage. No significant difference between the three cases exist. The trend is monotonic and the rate of change of the critical angle with blockage ratio is almost constant.

It is not certain at this time whether the changes in diffuser performance were mainly due to core flow modification or due to exit condition modification. The diffuser flow with the control ring is rather complicated with skewed boundary layers at the hub and the shroud. However if in the present situation one assumes that the development of the flow in the diffuser is only dependent on the velocity profile at inlet the results of figure (7) for the cases of no control ring and shroud-side control ring would lead to the conclusion that the effects of the

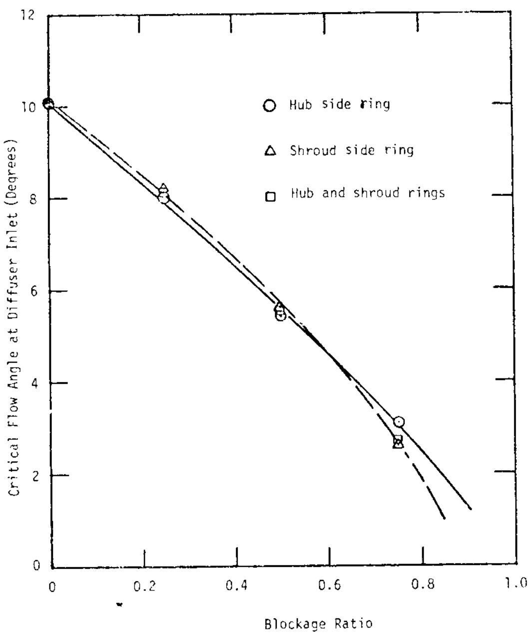


Figure 9: Variation of critical flow angle at diffuser inlet with diffuser exit blockage ratio.

ring on the diffuser performance should be mainly attributed to the exit condition modification.

The reason for the uncertainty with regard to this conclusion is twofold. First, the development of flow in the diffuser is dependent on the boundary condition at inlet and exit. Second, other tests, which are yet to be reported on, showed that when a tapered shroud wall diffuser was used to reduce the diffuser exit area by the same amount as the control ring the effects of the use of tapered shroud on the flow stability were similar to those of the control ring at diffuser exit. Further detailed testing is required to fully explore this aspect of the phenomena.

## Conclusions

The performance of radial vaneless diffusers is favorably affected by the use of control rings at diffuser exit at lower values of inlet flow angles. The control rings result in stabilizing the flow in the diffuser and improve the static pressure rise coefficient characteristics. The rings may be attached to the hub-side, the shroud side or both with very similar effects. The technique is relatively easy to implement in practical applications and should have a significant impact on centrifugal compressor range and efficiency.


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## REFERENCES


1. Abdelhamid, A.N., Colwill, W.H. and Barrows, J.F., "Experimental Investigation of Unsteady Phenomena in Vaneless Radial Diffusers", ASME Journal of Engineering for Power, Vol. 101, pp. 52-60, January 1979.
2. Abdelhamid, A.N. and Bertrand, J., "Distinctions Between Two Types of Self-Excited Gas Oscillations in Vaneless Radial Diffusers", Canadian Aeronautics and Space Journal, Vol. 26, No. 2, 1980. pp. 105-117. Also ASME Paper No. 79-GT-58.
3. Abdelhamid, A.N., "Analysis of Rotating Stall in Veneless Diffusers of Centrifugal Compressors", Canadian Aeronautics and Space Journal, Vol. 26, No. 2, 1980. pp. 118-128. Also ASME Paper No. 80-GT-184.
4. Van Der Braembussche, R.A., Frigne, P. and Roustan, M., "Rotating Non-Uniform Flow in Radial Compressors", AGARD Conference reprint 282, page 12, May 1980.
5. Abdelhamid, A.N., "Effects of Vaneless Diffuser Geometry on Flow Instability in Centrifugal Compression Systems", ASME Paper No. 81-GT-10.
6. Jansen, W., "Rotating Stall in Radial Vaneless Diffusers", ASME Journal of Basic Engineering, Vol. 86, pp. 750-758, December 1964.