Printed in U.S.A.

79-GT-58

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Distinctions Between Two Types of Self Excited Gas Oscillations in Vaneless Radial Diffusers

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Contributed by the Gas Turbine Division of The American Society of Mechanical Engineers for presentation at the Gas Turbine Conference & Exhibit & Solar Energy Conference, San Diego, Calif., March 12-15, 1979. Manuscript received at ASME Headquarters December 18, 1978.

Copies will be available until December 1, 1979.

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ABSTRACT

Experiments were conducted to determine the characteristics of oscillating flows in a centrifugal compression system with vaneless diffusers. The system was operated without a diffuser and with eight different diffuser configurations to determine the effects of diffuser diameter and width ratios on the unsteady behaviour of the system. Mean and fluctuating velocity and static pressure measurements were carried out in the time and frequency domains. The system without a diffuser was found to be stable at all operating conditions. The installation of any of the eight diffusers resulted in the generation of self-excited oscillations at some operating conditions. It was found that the critical flow coefficient at which onset of oscillations was observed increased as the diffuser width ratio was decreased and as the diameter ratio was increased. Comparison between the characteristics of the oscillations observed in the present study and those observed by other investigators indicate that rotating stall in two geometrically similar diffusers can be an order of magnitude different in the non-dimensional rotational speed and level of unsteady pressure fluctuations. These differences point towards the possibility of existence of more than one set of flow conditions which could lead to the occurence of the unsteady phenomena.

NOMENCLATURE

D_i	Diffuser inlet diameter
D_o	Diffuser outlet diameter
f_B	Blade passage frequency
m	Number of lobes in disturbance pattern
M_{tip}	Mach number at impeller tip
p_i	Fluctuating pressure at location i
R_2	Diffuser inlet radius
x	Axial distance in the diffuser measured from the shroud wall
t	Time
U	Impeller tip velocity
$V_{\theta 2}$	Component of velocity in tangential direction at diffuser inlet
$v_{i,x}$	Fluctuating velocity at location (i,x)
W	Diffuser width
θ	Angular separation
ρ	Density of air
σ	Phase angle
ϕ	Flow coefficient
ψ	Pressure rise coefficient
Ω_p	Disturbance pattern speed
Ω_i	Impeller rotational speed

INTRODUCTION

Operating a centrifugal compressor system at Low flow rates is in many cases characterized by the existence of self-excited fluid oscillation. The nature of the oscillations has been extensively investigated in recent years and it is firmly believed now that the oscillations may be of a local character like those associated with rotating stall in impellers and diffusers or may involve every component in the system as is the case with compression system surge.

The information reported by most investigators of the stability of flows in centrifugal compression systems [1-9]*deals mainly with the determination of the time-averaged flow conditions which lead to the onset of oscillations and the designation of these conditions on the compressor map or its equivalent. The identification of the cause of the oscillations is not usually attempted possibly because it would require very expensive and elaborate dynamic measurement of the fluctuating flow properties. The more detailed but very scarce data reported by some investigators [10-13] who measured the flow fluctuation in the time domain suggest that impellers and diffusers can become unstable on their own or as a result of their dynamic interaction. To date, however, the spatial and temporal characteristics of the oscillations and their correlation with component geometry and operating conditions have not been put in a form which would be useful to compressor designers.

In this paper the characteristics of gas flow oscillations in a centrifugal blower fitted with a vaneless diffuser are presented. Systematic measurements of steady and fluctuating flow velocity and pressure at various locations in the gas flow path were undertaken. Eight different diffusers were tested to determine the effects of diffuser geometry on the characteristics of the oscillations. The measured data have been non-dimensionalized and compared with previously reported results [10,13]. The comparison indicates distinct differences between the characteristics of the oscillations measured in the present study and those reported earlier. Plausible explanations of the reasons behind these differences are discussed.

EXPERIMENTAL TEST FACILITY

A schematic diagram of the test facility used in the present investigation is shown in figure 1. The diffuser inlet diameter was 235mm, and the impeller had ten backward leaning blades with an exit angle of 27 degrees from the radial direction. The impeller tip height to diameter ratio was 0.076. The rate of volume flow was controlled by the axial motion of the intake center-body. The volume flow rate was calculated from measurements of the static and total pressure upstream of the impeller. The geometrical configuration of the diffuser was varied by changing the diffuser width and/or exit to inlet diameter ratio. The performances of the impeller and the diffuser were determined from measurements of the static and total pressures at impeller and diffuser exits. The fluctuating static pressure and velocity were detected by four miniature piezo-electric dynamic pressure transducers and a hot wire system respectively. The locations of holes for the pressure transducers on the shroud walls of the impeller and the diffuser are shown in figure 2 and are given in table 1. The hot wire probe whose axis was normal to the diffuser walls was mounted at the same locations of the pressure transducer holes in the diffuser through the use of a suitable fitting. The hot wire probe was rotated around its own axis and also moved axially to traverse the diffuser fluctuating velocity field from hub to shroud. One pressure transducer hole, location 12 in figure 2, was located at the hub wall just outside the impeller exit opposite to location 1 on the shroud wall.

Signals from transducers in locations 3, 5 and 6 were used to calculate the lobe number and rotational

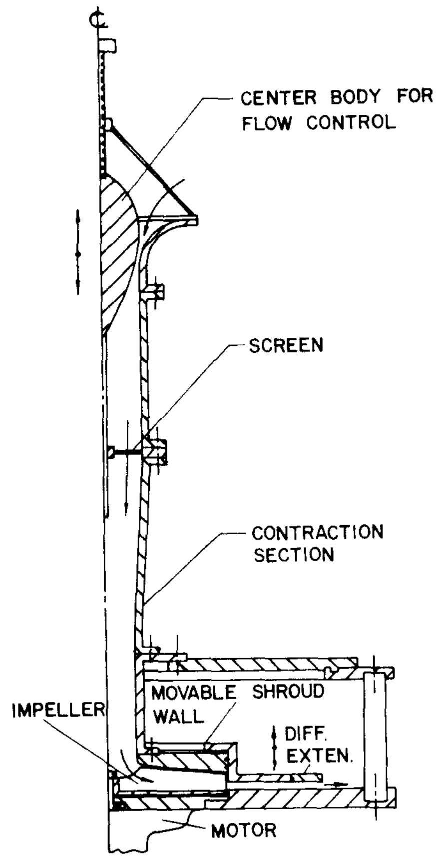

Fig.1 Schematic of the experimental facility

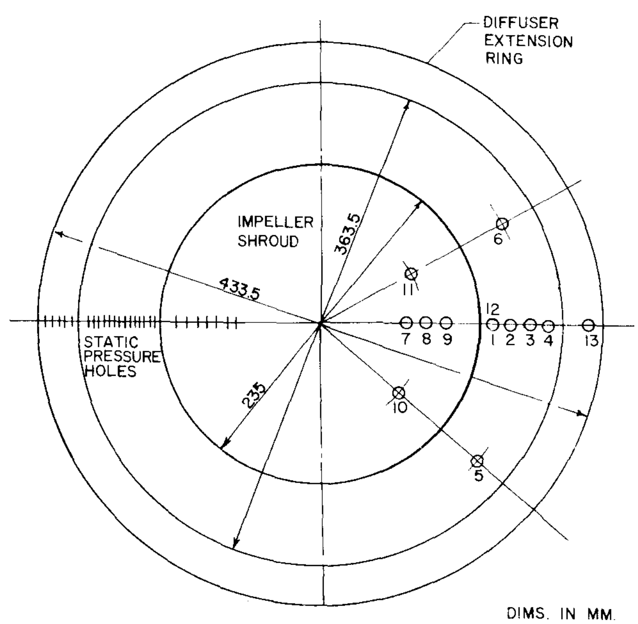

Fig.2 Locations of pressure transducers and hot wire probe in the vaneless diffuser and impeller shroud

* Numbers in brackets indicate references at end of paper.

Location Number	Diameter Ratio
1	1.08
2	1.21
3,5,6	1.35
4	1.48
7	0.54
8	0.67
9	0.81
10	0.67
11	0.67
12	0.94
13	1.71

Table 1

speed of any possible rotating disturbance pressure patterns in the diffuser. The blade to blade pressure variation and the lead or lag of the pressure fluctuations in the impeller relative to those in the diffuser were determined from signals from transducers in locations 1, 7, 8 and 9. The hot wire probe in location 1 was used to measure the velocity fluctuations at the diffuser inlet.

Typically a signal from a pressure transducer or the hot wire system was passed through a Spectral Dynamics Real Time Analyzer type SD 330-20 and a Gould Digital Storage Scope type OS 4000 to determine the power spectrum and the waveform of the fluctuations respectively. When two signals were analyzed simultaneously the signals were passed through the Gould Digital Storage Scope and a DEC PDP11/10 data acquisition and analysis system to obtain the relative phase shift between the two signals. The amplitude and phase of the cross spectrum between the two signals was determined by implementing a Fast Fourier Transform routine on the PDP11/10 system.

RANGE OF TEST OPERATING CONDITIONS AND PROCEDURE

The thrust of the present investigations has been directed towards the determination of the effects of diffuser geometry on the generation of self-excited oscillations in the system. Once the oscillations were detected their characteristics in space and time were measured. The variation of these characteristics with flow conditions in the system was then determined.

The system was operated first without a diffuser but with a sleeve at impeller exit to cover part of the blades and thus change the impeller exit area. Then the system was operated with two diffusers with diameter ratio of 1.55 and 1.83 respectively. For each of the three cases considered the height of exit area at impeller discharge was successively adjusted to 0.063, 0.054, 0.045 and 0.038 of diffuser inlet diameter. The speed of the impeller was varied from 4000 to 6000 rpm which resulted in impeller tip Mach number range of approximately 0.14 - 0.22.

For each geometrical configuration and speed of the system the mass flow rate was gradually reduced and the performance of the system was evaluated. After onset of the oscillations the waveform frequency and lobe number of the fluctuating pressure and velocity were measured for every mass flow rate. This procedure was repeated until further axial movements of the center-body shown in figure 1 produced no further changes in the flow rate.

RESULTS AND DISCUSSIONS

Most of the data in this paper are presented in a non-dimensional form. The flow coefficient ϕ is based on the impeller tip speed (U) and diffuser inlet area. The pressure rise coefficient of the system, ψ, is the difference between the diffuser exit static pressure and the impeller inlet total pressure divided by $\rho U^2/2$ were ρ is the density of air. Fluctuating velocity data are normalized by U while those of the fluctuating pressure are normalized by $\rho U^2/2$. All length dimensions are normalized by the diffuser inlet diameter.

The fluctuating pressure data are designated p_i, where i takes values between 1 and 13 to identify the location at which the pressure was measured. The velocity data are designated by $V_{i,j}$ because for each location i the wire can be moved in the axial direction. Values of j therefore indicate the ratio between the distance of the wire from the shroud wall and diffuser width.

1. Performance Characteristics of the System With and Without Vaneless Diffusers

Figures 3(a) to 3(d) show the variation of the pressure rise coefficient ψ with flow coefficient ϕ for the four different diffuser widths investigated

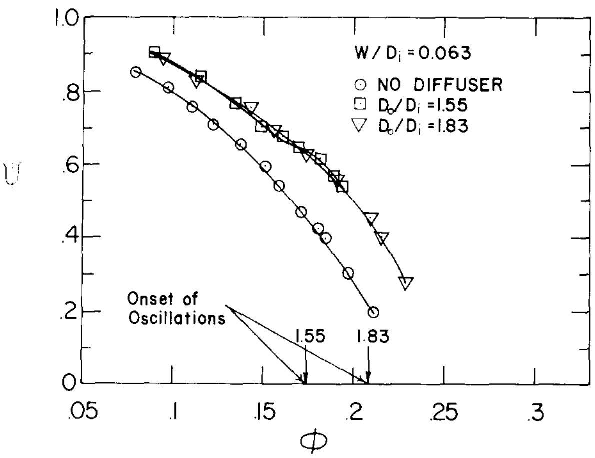

Fig. 3a

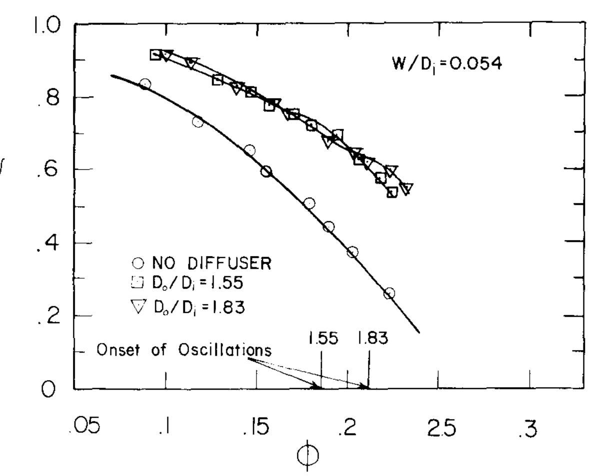

Fig. 3b

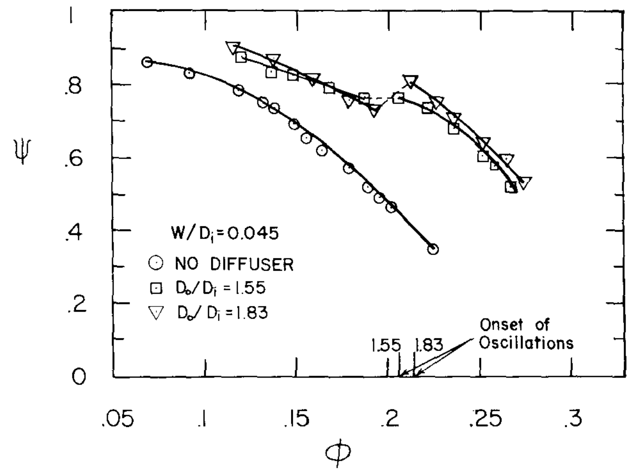

Fig. 3c

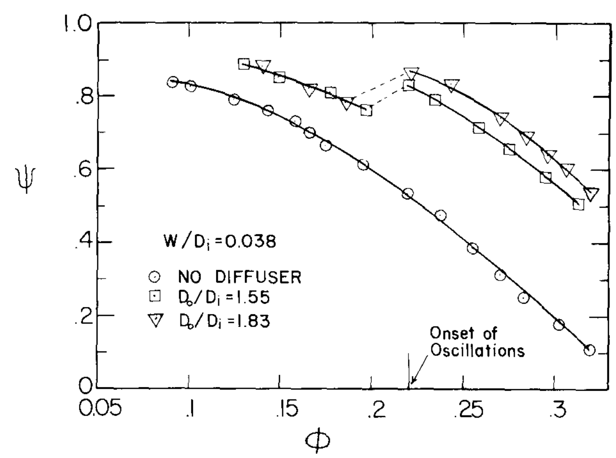

Fig. 3d

Fig.3 Effects of diffuser geometry on system performance and limits of instability. $M_{tip} = 0.18$

in the present study. Indicated on these figures also are the flow conditions at which the onset of gas flow oscillations was detected. No oscillations were observed under any flow conditions when the system was operated without a diffuser. The installation of any of the vaneless diffusers resulted in the generation of oscillations at some flow coefficients.

At higher values of diffuser width ratio the onset of the oscillations was intermittent with an on - off ratio which increased with further reduction of flow rate until the oscillations became on all the time time. At the smaller values of width ratio, however, the onset of the oscillation was abrupt and very well defined and was associated with a discontinuity in the performance curves. At onset of the oscillations in these cases both the flow and pressure coefficients were reduced.

It should be noted here that in all the cases considered in the present study, the onset of the oscillations occurred when the slope of the ψ-ϕ curve of the system was negative. This observation is not in agreement with the generally accepted criteria which says that the instability occurs when the slope of the system ψ-ϕ curve is greater than or equal to zero. Earlier experimental investigations however, for example references 13 and 14 demonstrated that self excited oscillations can be generated in centrifugal and axial compression systems at an operating point with a negative slope on the ψ-ϕ curve.

The critical flow coefficients below which the oscillations were first detected, were dependent on the diffuser diameter and width ratios. As shown in figure 4 for the same diffuser width ratio the critical flow coefficient was larger for the system with

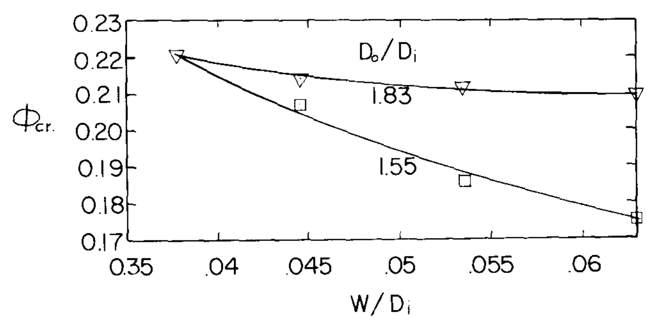

Fig.4 Variation of critical flow coefficient with diffuser width ratio. $M_{tip} = 0.18$

a diffuser diameter ratio of 1.83 than that with a diameter ratio of 1.55. The difference, however, was reduced to zero at the smallest diffuser width ratio investigated here. It is conceivable, therefore, that for a diffuser width ratio less than 0.038 the critical flow coefficient for the system with diffuser diameter ratio of 1.83 would be less than that for a diffuser diameter ratio of 1.55. Another aspect of the data shown in figure 4 is that for both diffuser diameter ratios the critical flow coefficient increased monotonically as the diffuser width ratio was decreased. The system with the small diameter ratio diffuser, however, was more sensitive to diffuser width ratio than the system with the large diameter ratio diffuser.

2. Characteristics of the Oscillations in the Vaneless Diffuser

To be able to determine whether or not the oscillations observed in the present study are similar to those observed by other investigators it is essential to identify the oscillations by all of their measurable characteristics. In this section the variation of the waveform, frequency, level and spatial distribution of the oscillations with diffuser geometry and flow conditions are discussed.

2.1 Waveform at Onset of Oscillations.

The fluctuating velocity and pressure at locations 1 and 12 were used to detect the onset of the oscillations. Figure 5(a), 5(b), 6(a) and 6(b) show typical traces of the hot wire and pressure transducer output just before and just after the onset of the oscillations. The diffuser diameter and width ratios were 1.83 and 0.038 respectively, and the impeller tip Mach number was 0.18. The hot wire was located half way between the hub and the shroud for this particular trace. The orientation of the wire was adjusted such that time average of the output signal from the wire, i.e. the D.C. component, was maximum. The peak to peak velocity fluctuation was about 0.75 which is almost

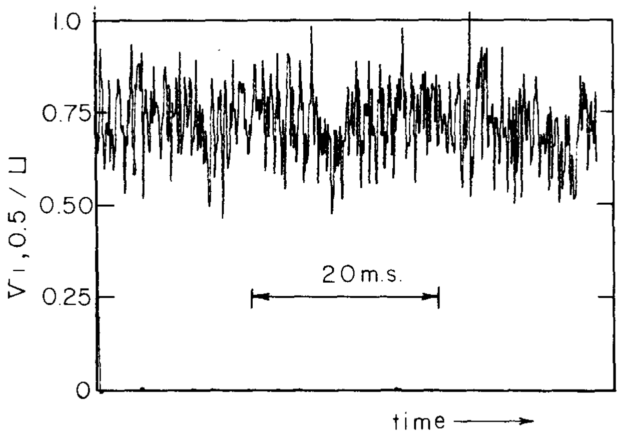

Fig. 5a

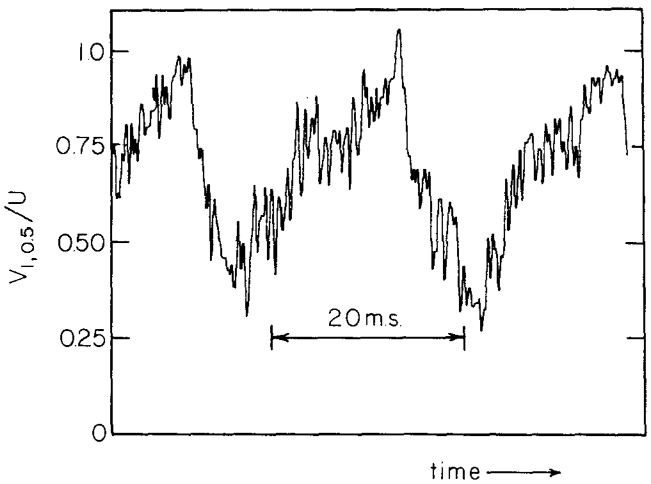

Fig. 5b

Fig.5 Diffuser inlet velocity before and after instability. D_o/D_i = 1.83, W/D_i = 0.038, M_{tip} = 0.18

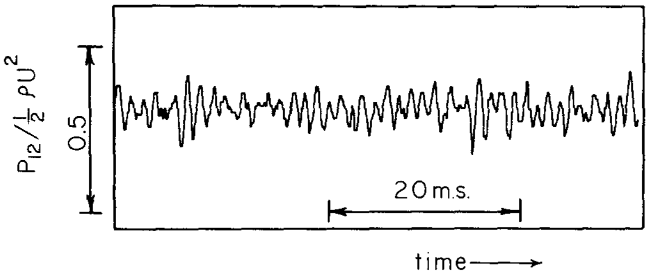

Fig. 6a

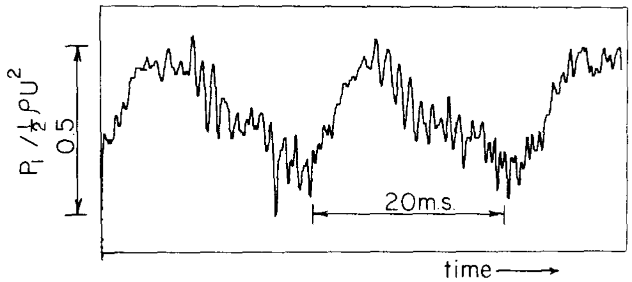

Fig. 6b

Fig.6 Diffuser inlet pressure fluctuations before and after instability. D_o/D_i = 1.83, W/D_i = 0.038, M_{tip} = 0.18

equal to the time average velocity at this location. The peak to peak pressure fluctuation at diffuser inlet was approximately 0.3. The high frequency fluctuations shown in figures 5 and 6 are, of course, due to blade passage.

The waveform of the fluctuations was dependent on the diffuser geometry. For example, figure 7 shows a

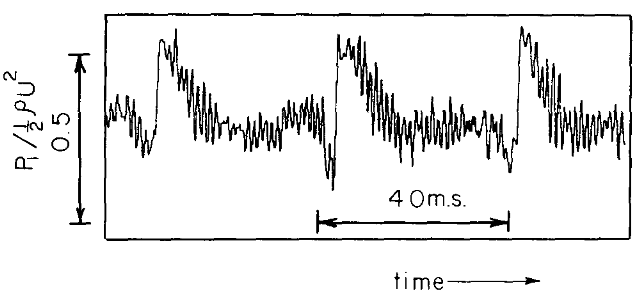

Fig.7 Pressure waveform at diffuser inlet. D_o/D_i=1.55 W/W_i = 0.054, M_{tip} = 0.18, ϕ = 0.156

trace of the pressure fluctuations at location 12 for the system with a diffuser diameter ratio of 1.55 and width ratio of .054 operating at a tip Mach number of 0.18 and a flow coefficient of 0.156. The trace is clearly different from the one shown in figure 6(c). In one period the pressure first decreased slightly then sharply increased to its mazimum value then relaxed gradually to its time average value and remained at this level until the next cycle. The percentage of time in which the pressure was equal to its time average value was approximately equal to 60%.

2.2 Spatial Extent of the Oscillations. Figure 8 shows traces of the hot-wire output at location 1 for different axial locations of the wire between the hub and the shroud for diffuser diameter and width ratios of 1.55 and 0.038 respectively just after onset of insatbility. The orientation of the wire in each case was adjusted such that the D-C output from the wire was maximum. Figure 9 shows traces of signals from pressure transducers positioned at locations 1, 4 and 13 for a diffuser diameter and width ratios of 1.83 and 0.038 respectively also just after onset of

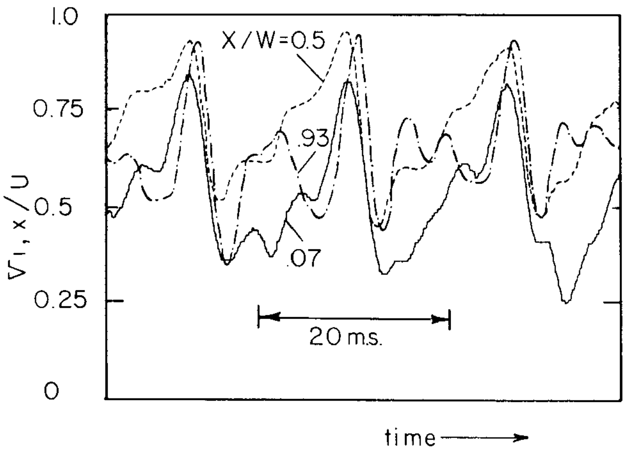

Fig.8 Variation of fluctuating velocity with axial distance at diffuser inlet. $D_o/D_i = 1.55$, $W/D_i = 0.038$, $M_{tip} = 0.18$, $\phi = 0.196$

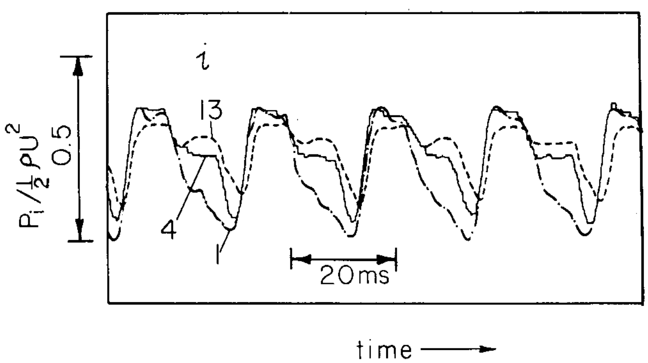

Fig.9 Variation of pressure fluctuation with radius. $D_o/D_i = 1.83$, $W/D_i = 0.038$, $M_{tip} = 0.18$, $\phi = 0.186$

instability. The signals in these figures were passed through 250 H_z low pass filters to eliminate the fluctuations due to blade passage from the traces. The phenomena clearly existed throughout the entire diffuser. Furthermore, figure 10 shows traces of outputs from pressure transducers at locations 1 and 12 for the same operating conditions corresponding to figure 9. The traces are exactly in phase which means that the static pressure fluctuations in the diffuser were two dimensional and did not vary from hub to shroud.

2.3 Lobe Numbers and Frequency of Oscillations. The outputs from pressure transducers at locations 3 and 5 were not in phase as shown for example in figure 11. The diffuser diameter and width ratios were 1.55 and 0.045 respectively and the flow coefficient was 0.122 at an impeller tip Mach number of 0.18. It should be noted here that the traces are inverted because the signals from these two pressure transducers were passed through an amplifier with an inherent reverse in polarity. The signals from the

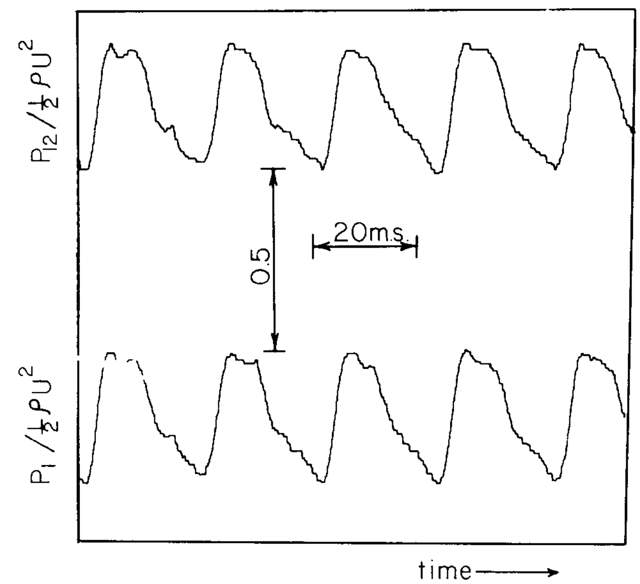

Fig. 10 Hub and shroud pressure fluctuations at diffuser inlet. $D_o/D_i = 1.83$, $W/D_i = 0.038$, $M_{tip} = 0.18$, $\phi = 0.196$

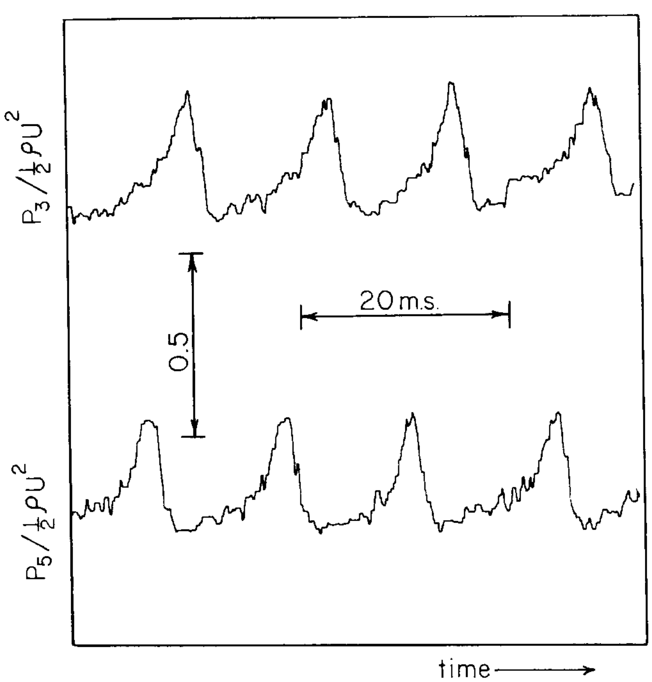

Fig.11 Variation of pressure fluctuations with circumferential separation. $D_o/D_i = 1.55$, $W/D_i = 0.045$, $M_{tip} = 0.18$, $\phi \doteq .122$

two transducers were analyzed in the complex frequency domain and the cross spectrum between the two signals were obtained. The phase angles at the dominant frequencies were examined to determine the lobe numbers and speed of any rotating pressure pattern in the diffuser. Table 2 shows the rotational speeds in revolutions per second of the measured patterns corresponding to all diffuser configurations investigated

W/D_i	D_o/D_i = 1.55				D_o/D_i = 1.83		
	m=1	m=2	m=3	m=4	m=1	m=2	m=3
0.063	26.0	-	22.7	21.0	-	20.5	18.0
0.054	27.0	27.0	-	-	22.0	22.0	-
0.045	28.0	28.0	26.3	-	22.0	22.0	19.3
0.038	-	29.5	27.3	-	-	22.5	-

Table 2

in this study. The impeller rotational speed for all the cases shown in table 2 was 83.3 revolutions per seconds. The existence of more than one lobe number for every diffuser geometry shown in table 2 does not mean the simultaneous occurrence of more than one pattern. It is only an indication that as the flow rate in the system was reduced after onset of the oscillation the lobe number of the disturbance pattern changed. Up to four lobes were observed at the largest width ratio. The results indicate that the rotational speeds of the disturbance patterns were strongly dependent on the diffuser diameter ratio and weakly dependent on the diffuser width ratio. A decrease of 15% in diameter ratio resulted in an increase of 31% in the rotational speed of the unsteady pressure pattern while a decrease of 40% in diffuser width ratio resulted in only 10% increase of the pattern rotational speed. Also for a given diffuser geometry, the rotational speeds of the patterns decreased as the number of lobes in the pattern increased above two. The rotational speeds of the patterns of a certain lobe number in a given diffuser did not vary measurably with flow coefficient. The data shown in table 2 was obtained from measurements taken while the impeller was rotating at a speed of 5000 rpm. When the impeller speed was changed the amplitude and frequency of the oscillations also changed. Figure 12(a) shows the power spectrum of

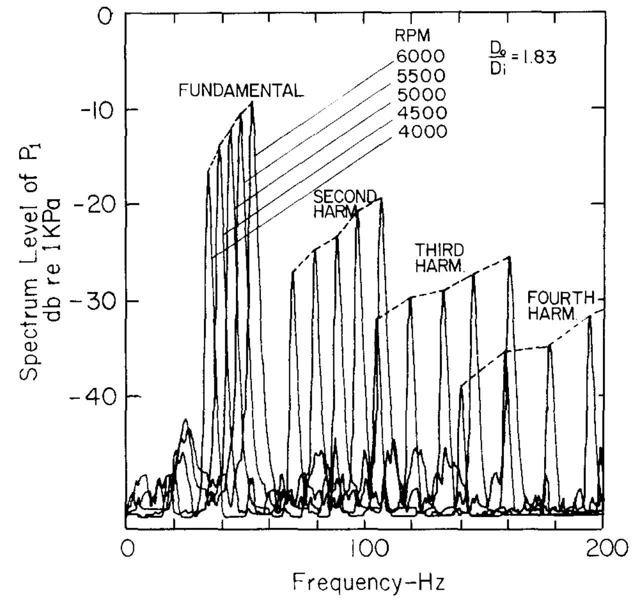

Fig. 12b

Fig.12 Spectra of diffuser inlet pressure fluctuations. W/D_i = 0.038

the pressure fluctuations at location 1 for different impeller speeds. The diffuser geometry in this case was a diameter ratio of 1.55 and a width ratio of 0.038. The number of lobes in the pattern was two. Similar results are shown in figure 12(b) for the system with a diffuser of diameter ratio 1.83 and width ratio 0.038. When the frequency and pressure were normalized by the impeller speed and $\rho U^2/2$ respectively as shown in figure 13 the non-dimensional

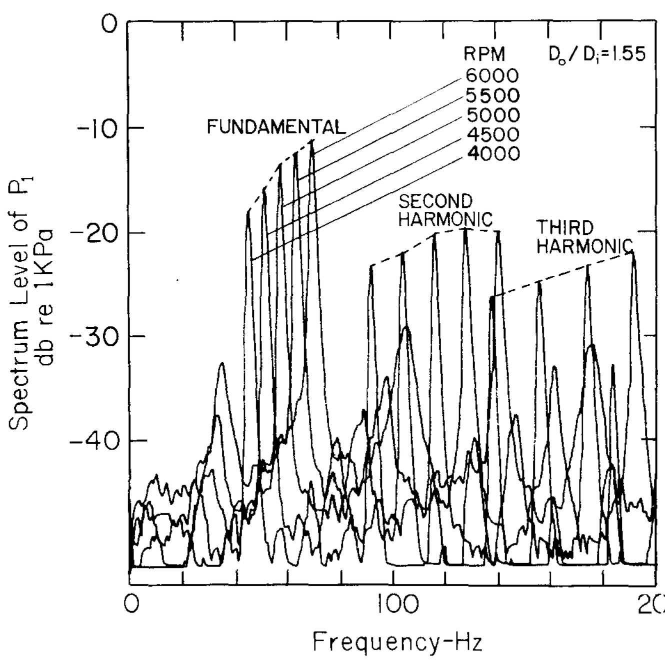

Fig. 12a

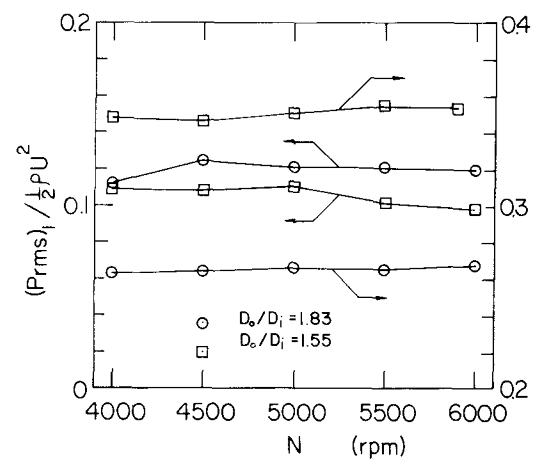

Fig.13 Effect of impeller speed on amplitude and speed of disturbance patterns. W/D_i = 0.038, m = 2

rotational speed of the oscillation patterns and fluctuating pressure amplitude were fairly constant for each diffuser. Two-lobe patterns were rotating at 26.6% of the impeller speed for the system with large diffuser and at 35% for the system with small diffuser. The magnitudes of these non-dimensional rotational speeds are similar to the previously measured non-dimensional rotational speeds of rotating stall in axial and circular cascades [15]. Whether or not this correlation is an indication that the flow processes associated with the unsteady phenomena in the circular cascade [15] and the compression system of the present investigation is not certain. What is certain and is demonstrated by the results of the present investigation is the importance of the interaction between the impeller and the diffuser on the generation of flow oscillations. The critical flow coefficient and the rotational speed of the stall cells are affected by such interaction. This idea of the effects of the coupling between a compressor and a diffuser has not been explored in reference [15] but has been investigated analytically and experimentally in reference [14] for an axial flow machine.

The level of the pressure oscillations was slightly smaller for the system with small diffuser than for the system with large diffuser. The root mean square level of the pressure oscillations was between 10 and 12% of $\rho U^2/2$. It should be mentioned at this point that for a given impeller speed, diffuser geometry and lobe number the peak to peak variation of the pressure fluctuations did not vary with the flow coefficient. The oscillation level did however, vary very slightly when the number of lobes in the pattern changed as shown in figure 14.

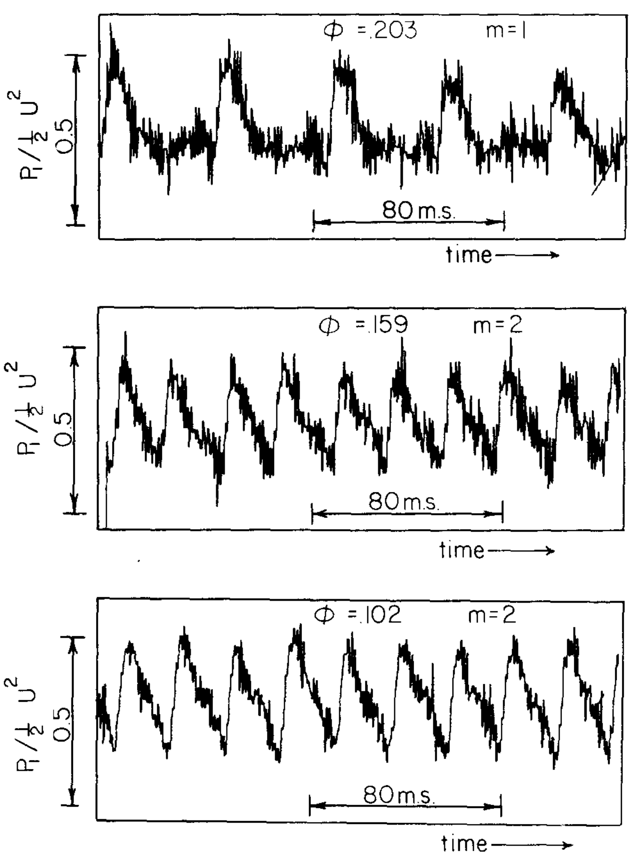

Fig.14 Variation of flow coefficient on pressure fluctuation waveform at diffuser inlet. D_o/D_i = 1.83, W/D_i = 0.054

2.4 Phase Relationships Between Pressure and Velocity Fluctuations. Figure 15 shows traces of the

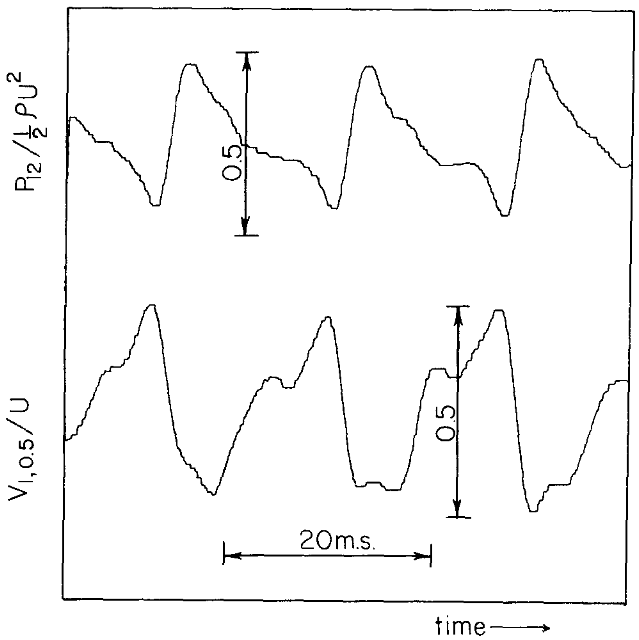

Fig.15 Phase relationship between pressure and velocity at diffuser inlet. D_o/D_i = 1.55, W/D_i = 0.038, M_{tip} = 0.18, ϕ = 0.196

output of the hot wire positioned at location 1 midway between the hub and the shroud and the pressure transducer at location 12 for diffuser diameter and width ratios of 1.55 and 0.038 just after onset of oscillations. The wire was oriented in a direction normal to the mean flow direction at its position. Both signals were passed through a matched pair of low pass filters set at 250 Hz. Maximum velocity occurred exactly when the pressure was minimum. The sharp rise in pressure followed by the gradual relazation was associated with a sharp reduction in velocity followed by gradual increase towards the mean value. The slight inconsistency in the velocity waveform at the minimum velocity may be attributed to the sensitivity of the hot wire to velocity magnitude and direction simultaneously. At the minimum velocity which, as will be shown in section 3, corresponded to the passage of an impeller channel with no flow and as a result some random changes in the velocity vector are likely to happen and this affect the trace.

3. Description of the Unsteady Flow in an Impeller Channel

The outputs of pressure transducers at locations 7, 8 and 9 were used to synthesize the flow variation in any one channel with time. Because of the rotation of the disturbance patterns the frequency of the flow oscillations that any one channel experienced was the difference between the impeller shaft frequency and the pattern rotational frequency. The frequencies were subtracted because the patterns were found to rotate in the same direction as the impeller. Events which occurred at time t in one channel were repeated in a following channel after a time $\Delta t_c = \Delta\theta/(\Omega_i - \Omega_p)$ where

$\Delta\theta$ is the angular separation between the channels and Ω_i and Ω_p are the angular velocities of the impeller and the disturbance pattern respectively. Relative to a stationary observer however, the time separation between the passage of the two channels would be $\Delta t_o = \Delta\theta/\Omega_i$. An event recorded by the stationary observer at the time of passage of the first channel would be the same as the event which would occur in the second channel after a time $[\Delta t_c - \Delta t_o]$ from the passage of the second channel. Therefore the successive blade to blade pressure variations measured by a pressure transducer at location 7, 8 and 9 after onset of the oscillations would represent the blade to blade variation of a single channel at different times. Figures 16(a) and 16(b) show traces of the

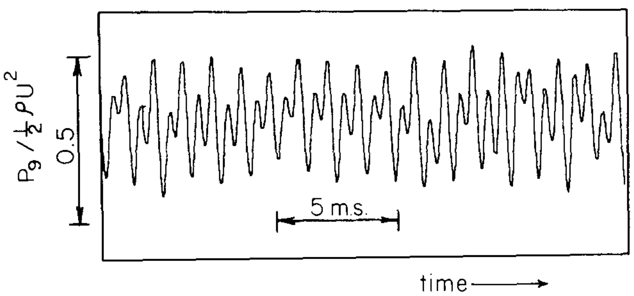

Fig. 16a

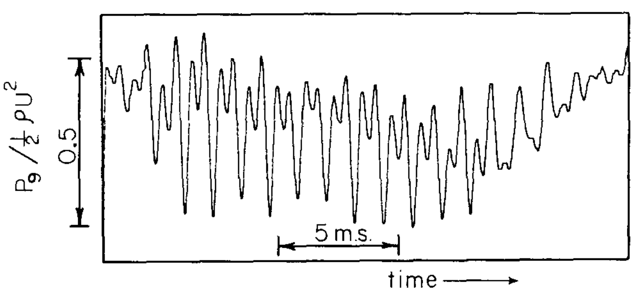

Fig. 16b

Fig.16 Impeller pressure fluctuations before and after instability. $D_o/D_i = 1.83$, $W/D_i = 0.038$, $M_{tip} = 0.18$

output of a pressure transducer at location 9 just before and just after onset of instability. The amplitude of the blade to blade pressure variation after onset of instability changed from almost zero to a level higher than the one shown in figure 16a just before onset of oscillations. In addition to the blade to blade variation, the mean pressure in the channels also varied with time with a peak to peak variation of about 25% of $\rho U^2/2$.

The blade to blade pressure variation in one channel is related to the mass rate of flow in the channel. The steady flow relationship for the particular system used in the present study was obtained by operating the system without a diffuser and measuring the average blade to blade pressure variation for different mass flow rates in the system. The results are shown in figure 17. If the unsteady flow in the channels is assumed to be quasisteady the trace shown in figure 16(b) can now be used to describe the flow

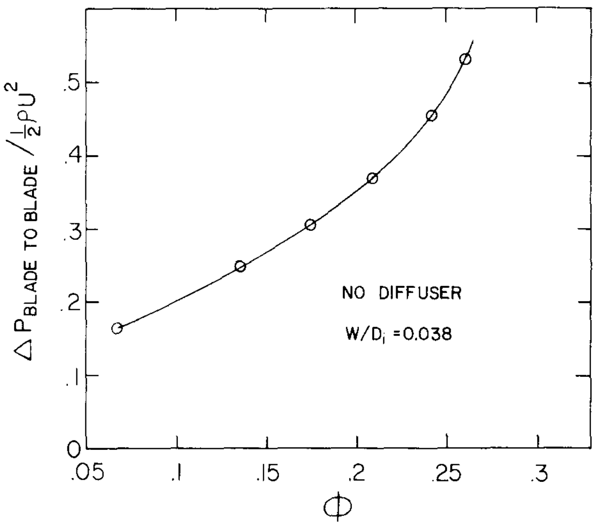

Fig.17 Relation between blade-to-blade pressure difference and flow coefficient. No diffuser, $W/D_i = 0.038$, $M_{tip} = 0.18$

in any one particular channel. Relative to the channel the positive sense of time would be from right to left. The successive blade to blade variations are $(1/f_B) \times [\Omega_p/(\Omega_i - \Omega_p)]$ seconds apart where f_B is the blade passage frequency. At time of maximum average pressure in the channel the level of the blade to blade pressure variation was minimum which meant that the flow rate in the channel was also minimum. The mean pressure then gradually decreased towards the minimum value and the flow rate in the channel gradually increased. The pressure reached its minimum value and started to increase while the flow rate continued to increase. The channel flow rate then reached its maximum level and started to decrease while the pressure continued to increase. Finally the pressure and the flow rate reached their values at the start of the cycle and the cycle was repeated.

The phase angle between the signals from pressure transducers at locations 1, 7, 8 and 9 was measured. The results are shown in figure 18 for the system with diffuser diameter and width ratios of 1.55 and 0.038 respectively for a disturbance pattern with two lobes. The pressure variations in the impeller lead the pressure variation at diffuser inlet. The magnitude of the phase angle increased as the radial separation between the two transducers was increased.

4. Comparison with Previously Reported Experimental Results

Detailed dynamic measurements of self-excited oscillations in vaneless diffusers have been reported before [10,13]. The geometrical parameters of the diffuser used in reference [10] were diameter and width

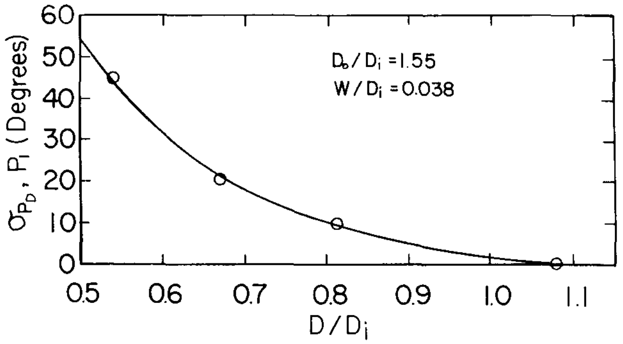

Fig.18 Phase angle between impeller fluctuations in the impeller and at diffuser inlet. D_o/D_i = 1.55, W/D_i = 0.038, M_{tip} = 0.18, ϕ = 0.196

ratios of 2.93 and 0.068 respectively. The diffuser flow was generated through the use of a blower which fed the flow to a plenum and a rotating screen at diffuser inlet. Two diffusers were investigated in reference [13]. The geometrical parameters of the first diffuser were diameter and width ratios of 1.51 and 0.032 while those of the second diffuser were 1.52 and 0.065 respectively. Each diffuser was a component of a test compressor which was investigated in a closed loop tunnel.

The characterisitcs of the oscillations presented in references [10,13] were different from those described in the present investigation in many respects. Most prominent of the differences is the one associated with the non-dimensional rotational speed of the pattern. Figure 19 shows the results presented in references [10] and [13]. Although the geometrical

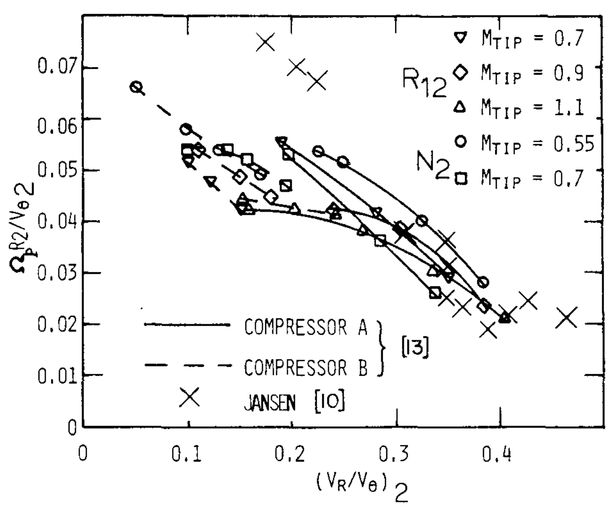

Fig.19 Variation of non-dimensional rotational speed with diffuser inlet angle reported in references 10 and 13

parameters of the diffusers testes in reference [13] are very similar to those of some of the diffusers investigated in the present study, the non-dimensional speeds measured in the present investigation were an order of magnitude larger than those measured in reference [13]. Furthermore the rotational speeds measured in this study did not change as the flow coefficient was reduced after onset while in both of the other references the rotational speed of the patterns increased as the flow rate was decreased.

At the onset of the oscillations the waveforms of the velocity and pressure fluctuation reported in references [10,13] were sinsoidal or almost sinsoidal while the waveforms measured in the present study indicated sharp variations as shown in figures 7 and 14. The non-dimensional level of the pressure and velocity fluctuations measured in the present investigation was much higher than those reported in reference [13] and reference [10] respectively. The root mean square value of pressure fluctuations at diffuser inlet was approximately 11% of $\rho U^2/2$ in this investigation while it was only about 0.4 - 2% in reference [13]. In the present investigation the level of the oscillations remained constant as the mass flow rate was decreased while in the data shown by references [10,13] the levels of the oscillations increased as the mass flow rate was decreased. The number of lobes in the patterns measured in the present investigation varied between 1 and 4 while in references [10] and [13] only patterns with two lobes were observed when the experimental rigs were operated with a through flow.

All of these differences indicate that the nature of the phenomena measured in the present investigation was different from those of the phenomena reported on by either of references [10] or [13]. The latter ones, however, seem to belong to the same group. Now recalling that the swirling flow in the diffuser of the test rig used in reference [10] was generated by a rotating screen with a through flow, it must be concluded then that the existence of an impeller is not necessary for the generation of oscillations of the type reported on in reference [10]. Furthermore since the characteristics of the oscillations reported on in references [10] and [13] were the same, it is very likely that the impeller in the test facility used in reference [13] had unsteady flow characteristics similar to those of a rotating screem, while the oscillations measured in the present study was connected to an impeller characteristic which could not be simulated by the flow through a rotating screen.

One characteristic which could have an effect on the generation of the oscillation is the variation of the absolute velocity vector at impeller exit with the rate of flow. At high flow rates a small reduction of flow generally results in a small increase in the tangential component of the velocity. As the flow rate is further decreased without the occurrence of the instability of the type reported in reference [10] a flow reguime is reached in which a small reduction in flow rate results in a substantial reduction in the tangential velocity component. Preliminary results from an analytical investigation of the unsteady flow coupling between an impeller and a vaneless diffuser show that this flow condition can result in self-excited flow oscillations. The larger the decrease in the tangential velocity component for a given reduction in flow rate the more unstable the system becomes. Further experimental work is currently underway to determine the variation of the diffuser inlet flow velocity with flow rate in the apparatus used in the present investigation. The objective being the establishment of whether or not the flow conditions at the diffuser inlet at the onset of the flow oscillations are in fact similar to those predicted analytically.

CONCLUSIONS

1. The generation of rotating stall in a centrifugal impeller-vaneless diffuser combination has been shown to depend on diffuser diameter and width ratios. For diffuser width ratios greater than 0.038 the larger the diameter ratio the higher the critical flow coefficient. For a given diameter ratio, the smaller the width ratio the higher the critical flow coefficient.
2. The rotational speeds of the oscillation patterns were found to depend strongly on the diffuser diameter ratio and weakly on the diffuser width ratio. The patterns rotated in the same direction as the impeller but at a fraction of its speed. The larger the diffuser diameter ratio the smaller the rotational speed of the pattern. Patterns with larger number of lobes rotated at a lower speed than those with fewer lobes.
3. The characteristics of the phenomena measured in the present study seem to differ from those of previously reported investigations in many respects. These differences are very distinct and point towards the possibility of existence of more than one set of flow conditions which could lead to the occurrence of the unsteady phenomena in an impeller diffuser combination.

ACKNOWLEDGEMENTS

The authors would like to acknowledge the financial support of the National Research Council of Canada under Grant No. A7429. Also the authors would like to thank Mr. S. Ahmed and Mr. M. Abouzeed who contributed to the design and operation of the experimental facility.

REFERENCES

1 Emmons, H.W., Pearson, C.E., and Grant, H.P., "Compressor Surge and Stall Propagation", TRANS. ASME, Vol. 77, 1955, pp. 455-467.

2 Dussourd, J.L. and Putman, W.G., "Instability and Surge in Dual Entry Centrifugal Compressors", Symposium on Compressor Stall, Surge and Systems Response, ASME, New York, 1960, p. 6.

3 Amann, C.A., and Nordenson, G.E., "The Role of the Compressor in Limiting Automotive Gas Turbine Acceleration", Centrifugal Compressors, Technical Progress Series, Vol. 3, ASME, New York, 1961, p. 74.

3 Dean, R.C. Jr., "Fluid Dynamic Design of Advanced Centrifugal Compressors", Creare TN-185, 1974.

5 Kenny, D.P., "A Comparison of the Predicted and Measured Performance of High Pressure Ratio Centrifugal Compressor Diffusers", ASME Paper No. 72-GT-54.

6 Rodgers, C., "Impeller Stalling as Influenced by Diffusion Limitations", ASME Journal of Fluids Engineering, Vol. 99, Series I, March 1977, pp. 84-97.

7 Whitfield, A., Wallace, F.J., and Atkey, R.C., "The Effect of Variable Geometry on the Operating Range and Surge Margin of a Centrifugal Compressor", ASME Paper No. 76-GT-98, March 1976.

8 Senoo, Y., and Kinosjita, Y., "Influence of Inlet Flow Conditions and Geometrics of Centrifugal Vaneless Diffusers on Critical Flow Angle for Reverse Flows", ASME Journal of Fluids Engineering, Vol. 99, March 1977, pp. 98-103.

9 Senoo, Y., and Kinoshita, Y., "Limits of Rotating Stall and Stall in Vaneless Diffuser of Centrifugal Compressors", ASME paper No. 78-GT-19, April 1978.

10 Jansen, W., "Rotating Stall in Radial Vaneless Diffusers", ASME Journal of Basic Engineering, Vol.86, Dec. 1964, pp. 750-758.

11 Lennemann, E., and Howard, J.H.G., "Unsteady Flow Phenomena in Rotating Centrifugal Impeller Passages", ASME Journal of Engineering for Power, Vol. 92, January 1970, pp. 65-72.

12 Toyama, K., Runstandler, P.W. Jr., and Dean R.C. Jr., "An Experimental Study of Surge in Centrifugal Compressors", ASME Journal of Fluids Engineering, Vol. 9, March 1977, pp. 115-131.

13 Abdelhamid, A.N., Colwill, W.H., and Barrows, J.F., "Experimental Investigation of Unsteady Phenomena in Vaneless Radial Diffusers", ASME Paper 78-GT-23.

14 Greitzer, E.M., "Coupled Compressor - Diffuser Flow Instability", AIAA Journal of Aircraft, Vol. 14, No. 3, 1977, pp. 233-238.

15 Stenning, A.H. and Kriebel, A.R., "Stall Propagation in a Cascade of Airfoils", ASME Paper No. 57-SA-29.