An Experimental Study on the Development of a Reverse Flow Zone in a Vaneless Diffuser*

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This study presents the measured rotating stall signal patterns in a vaneless diffuser in a centrifugal compressor with a radial bladed impeller. Unsteady flow and rotating stall in the vaneless diffuser were investigated by measuring unsteady velocity fluctuations at several different diffuser radius ratios and axial distances using a hot wire anemometer. The flow characteristics in terms of the radial and tangential velocity components and the flow angle distribution in the vaneless diffuser during rotating stall were investigated using phase-locked averaging techniques. The results clearly identified abrupt rotating stall at several different impeller rotational speeds. According to the experimental results, two different mechanisms exist for the development of the reverse flow zone in a vaneless diffuser. One is dominated by the extension of the reentering flow from the diffuser exit, and the other is dominated by the growth of the local flow separation zone on the hub and shroud side. The fluctuation in the flow direction increases as the diffuser radius ratio increases and is dominated by the strength of the reverse flow. At the onset of rotating stall, the radial velocity for one period of the rotating stall slowly increased to a maximum value and then decreased quickly to a minimum value with an intermediate peak. However, at lower flow rates, this intermediate peak did not occur.

**Key Words:** Centrifugal Compressor, Vaneless Diffuser, Rotating Stall, Local Reverse Flow, Reentering Flow, Relative Amplitude

1. Introduction

The performance of a centrifugal compressor is typically characterized by its pressure ratio, flow rate and efficiency. Stability can also be considered another important performance factor. Stall is a drop of lift coefficient on an airfoil in external flow or a drop of the static pressure recovery in internal flow such as that in diffusers or in cascades. In both cases, stall results from boundary layer separation. Rotating stall in centrifugal compressors occurs at the impeller and/or diffuser with pressure and velocity fluctuations, and its rotating direction is either the same as that of the impeller or the opposite. Flow instability causes not only the deterioration of compressor performance, but also mechanical damage due to dynamic excitations, especially in high pressure, high mass flow applications\(^{1-3}\). Rotating stall is a limiting factor that determines the stable operating range. Therefore studies on rotating stall have been focused on reducing the occurrence of rotating stall and expanding the stable operating range.

Experimental studies on rotating stall in vaneless diffusers have been reported by many investigators. Frigne and Van Den Braembussche\(^{4}\) reported rotating stall characteristics in terms of the propagation speed of the stall cell, the cell number and the amplitude. Abdelhamid et al.\(^{5}\) reported the behavior of pressure fluctuations. The transient process on the development of reverse flow was studied by Watanabe et al.\(^{6}\). Tsurusaki et al.\(^{7}\) proposed a simple correlation for the critical inlet flow angle from experimental data. Kinoshita and Seno\(^{8}\) and Otugen et al.\(^{9}\) investigated the critical inlet flow angle from experimental data for several different widths of diffuser. However

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these studies on rotating stall did not include detailed flow behavior in vaneless diffusers under rotating stall.

This investigation focuses on periodic flow in vaneless diffusers of centrifugal compressors under rotating stall. The flow characteristics such as phase-averaged radial and tangential velocity fluctuations and the absolute flow angle were measured to understand the complex flow physics during rotating stall. The extensions of the reverse flow region at the onset of rotating stall and at further reduced flow rates are also discussed.

**Nomenclature**

- \( A \): relative amplitude of the velocity fluctuation
- \( b \): diffuser axial width
- \( C_p \): static pressure rise coefficient
- \( D \): diameter
- \( p \): static pressure
- \( Q \): flow rate
- \( r \): radius

**Table 1** Dimensions of the test impeller and diffuser

- Impeller exit diameter: 418 mm
- Impeller inlet hub diameter: 110 mm
- Impeller inlet tip diameter: 240 mm
- Number of blade: 17 (no splitter)
- Exit blade angle: 90 degrees from tangential direction
- Diffuser inlet diameter: 420 mm (parallel wall)
- Diffuser exit diameter: 720 mm
- Diffuser width: 19.6 mm

- \( U \): impeller circumferential velocity
- \( V \): phase averaged velocity
- \( z \): diffuser axial velocity from the hub side
- \( a \): absolute flow angle from tangential direction
- \( \phi \): flow coefficient
- \( \varphi \): diffuser peripheral angle

**Subscripts**

- \( 1 \): impeller inlet
- \( 2 \): impeller exit or diffuser inlet
- \( id \): inlet duct
- \( k \): inlet plenum
- \( r \): radial component
- \( t \): tangential component

2. Test Rig and Instrumentation

2.1 Compressor test rig

The test compressor has a single stage with an unshrouded radial impeller and a parallel wall vaneless diffuser. The compressor test rig is shown in Fig. 1, and the dimensions of the impeller and diffuser are summarized in Table 1. The test compressor was driven by a 15 kW electric motor with a frequency inverter and operated as an open loop type. The air comes through a suction duct from an inlet plenum. The air from the vaneless diffuser was delivered to a collecting chamber, which is followed by a discharge duct. The flow rate was controlled by the throttle valve at the end of the discharge duct and measured using the orifice plate in the discharge duct. The rotational speed of the impeller was maintained at 3000 rpm.

2.2 Instrumentation and measurement techniques

Total pressure, temperature and wall static pressure were measured at the inlet plenum, impeller inlet and exit, diffuser exit, and discharge duct. For the pressure measurement a pressure scanner (PSI Sys-
system 8400) was used. The holes for hot wire probes to measure the propagation speed of the stall cell, its rotating direction, and the flow behavior in the vaneless diffuser during rotating stall were also prepared as shown in Fig. 1. The measurement positions were located at diffuser radius ratios of 1.02, 1.159, 1.293, and 1.561, with a circumferential interval of 90°. The trigger signal for phase-locked sampling was obtained from a rotating disk installed on the impeller shaft.

A single hot wire probe, TSI 1210·T1.5, was used to measure the instantaneous velocity fluctuations and the flow angle. Radial and tangential velocity components were measured. The axial component was negligible in magnitude compared with the other two components. To calibrate a hot wire probe, the technique of Schmidt and Okishi(10), which was used for three-dimensional measurement, was applied to the two-dimensional calibration. The wire angle (θw) and the probe pitch angle (θp) are eliminated. The calibration was conducted using a nozzle which delivers jet flow with 0.8% turbulence intensity. From the preliminary measurement it is estimated that the flow angle in the vaneless diffuser varies from −30 to +60 degrees at the diffuser inlet and from −45 to +45 degrees at the diffuser exit. Therefore, the hot wire probe was positioned at their corresponding angles to get two velocity components, and after that these components were converted into radial and tangential directions. Reverse flow could also be detected with this method. The signal from the hot wire anemometer was processed with a waveform analyzer (Analogic D6500E).

The accuracy of the hot wire measurements was estimated to be about 1.4 m/s in magnitude and 3° in direction. The time-averaged flow rates were compared with those obtained from the orifice plate. At the maximum flow rate, the difference between the measured flow rate using the hot wire and orifice was less than 6.6%. To obtain the phase-locked averaged data, fifty data sets were sampled with sampling periods of 0.6 milliseconds.

3. Results and Discussion

3.1 Compressor performance

The performance characteristics of the tested compressor are shown in Fig. 2. The static pressure rise for the impeller was measured from the inlet plenum to the impeller exit, and for the diffuser, from the diffuser inlet to the discharge duct. The flow coefficient and the static pressure rise coefficient are defined by Eqs. (1) and (2) as

\[ \phi = \frac{Q}{\pi D_b U_2} \]

\[ C_p = \frac{\Delta p}{\frac{1}{2} \rho \omega U_2^2} \]  

As the flow coefficient decreases from the maximum value to 0.19, the coefficient of static pressure rise of the impeller increases monotonically from 0.55 to 0.81, and that of the diffuser shows almost the same trend from 0.3 to 0.4. However, at all the measured

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**Fig. 2** Compressor performance map

**Fig. 3** Peripheral distribution of static pressure at diffuser inlet and outlet
rotational speeds, a sudden drop in the static pressure rise coefficient appears in the diffuser after the peak value ($\phi=0.19$). Once the sudden drop occurs, the total compressor performance is dominated by the static pressure rise in the diffuser. This effect was found to be independent of the impeller rotating speed, and the rotating stall occurs at the flow rate where the static pressure rise coefficient suddenly drops. It is believed that the observed rotating stall causes pressure loss that results in a decrease in the pressure rise coefficient. From these characteristics of compressor performance, the measured stall phenomenon is judged to be an abrupt rotating stall$^{(1)}$.

The results of static pressure measurements in the peripheral direction at the diffuser inlet ($r/r_2=1.02$) and outlet ($r/r_2=1.70$) are shown in Fig. 3. The flow in the vaneless diffuser is non-uniform, and the pressure difference at the diffuser inlet is larger than at the exit. The maximum pressure ratio was measured near 90° position, and the minimum pressure ratio was detected near 300°. At the flow coefficient ($\phi=0.16$) where rotating stall occurs, the pressure variation in the peripheral direction is nearly constant.

The critical inlet flow angle for rotating stall is 13.7°, which is in the range of Van Den
3.2 Instantaneous velocity measurement

The instantaneous measurements of the periodic velocity fluctuation were taken from the diffuser inlet to the exit at four different radius ratios. The phase-averaged radial and tangential velocity components and the absolute flow angle distribution at the rotating stall onset point \( \phi = 0.16 \) are shown in Figs. 4(a)-(d). They are plotted for three different axial positions, i.e., hub side \( (z/b = 0.05) \), middle \( (0.5) \), and near shroud side \( (0.95) \), and for four diffuser radius ratios, i.e., diffuser inlet \( (r/r_2 = 1.02) \), 1.159, 1.293, and diffuser near exit \( (1.561) \). The periodicity of the velocity pattern was clearly observed with a period of 0.0630 seconds at all measured positions. The number of stall cells is calculated as one in this case using Frigne's relationship, and the rotating direction of the stall cell corresponds to that of the impeller.

The magnitude of the radial velocity for all the measured radius ratios is higher along the diffuser middle section than near walls, and the magnitude of the velocity near the shroud is higher than that near the hub. The radial velocity for one period of the rotating stall increases slowly to a maximum value and then decreases rapidly to a minimum value with an intermediate peak. Reverse flow regions exist on

![Fig. 5 Phase-averaged radial, tangential velocity and absolute flow angle distributions with diffuser radius ratio and axial distance during rotating stall \( (\phi = 0.06) \)]
the hub and shroud side at all measured diffuser radius ratios except the diffuser inlet \( r/r_2 = 1.02 \). The magnitude of the reverse flow is stronger on the shroud side than on the hub side at radius ratios 1.159 and 1.293.

The distribution of the tangential velocity component is slightly different from that of the radial velocity. The intermediate peak value increases relatively as the diffuser radius ratio increases, whereas the maximum value decreases. The distribution of the tangential velocity becomes flatter with diffuser axial distance as the diffuser radius ratio increases.

The distribution of the absolute flow angle is very similar to that of the radial velocity. A detailed discussion of the flow angle will be presented later.

The phase-averaged radial and tangential velocity components and the absolute flow angle distribution at a flow coefficient of 0.06 are shown in Figs. 5 (a) - (d). The periodicity of the velocity pattern can also be clearly observed with a period of 0.0522 seconds at all measured positions. This result coincides with the general trend of rotating stall; the propagation speed of the stall cell increases as the flow rate decreases. The stall cell number and the rotating direction of the stall cell are the same as for the onset point of the rotating stall. The distribution of radial velocity components for one period of the rotating stall is different from the case in which the flow coefficient is 0.16. The distribution has no intermediate peak, and the time for which the reverse flow lasts is longer than that for \( \phi = 0.16 \). The distribution of the tangential velocity components also shows no intermediate peak.

The detailed information on the flow angle is shown in Fig. 6. As the diffuser radius ratio increases, the maximum value of the flow angle at \( z/b = 0.5 \) changes from 23° to 19° for \( \phi = 0.16 \) and from 16° to 15° for \( \phi = 0.06 \). Whereas the minimum value of the flow angle changes from 15° to -18° for \( \phi = 0.16 \) and from 2° to -19° for \( \phi = 0.06 \). This indicates that the maximum angle of flow is nearly constant as the diffuser radius ratio increases, but the minimum flow angle decreases; in other words, the strength of the reverse flow increases. Consequently the fluctuation in the flow direction becomes more severe as the diffuser radius ratio increases, which is dominated by the strength of the reverse flow.

As the flow rate is reduced from \( \phi = 0.16 \) to 0.06 and the diffuser radius ratio increases, the strength of the reverse flow and the stall cell size, which can be defined as the ratio of the time for which the reverse flow lasts to the duration of the stall period, increase simultaneously.

3.3 Distribution over diffuser width

The five distributions of the radial and tangential velocity components with a diffuser axial distance of \( \phi = 0.16 \) are shown in Figs. 7 (a) - (e). The following five moments during one stall period are taken for further discussion, i.e., the maximum value of the radial velocity component (moment A), the decreasing region (moment B), the minimum value (moment C), the recovery region (moment D), and the increasing region (moment E). The measurements were taken at seven axial positions between the hub and the shroud side, i.e., \( z/b = 0.05, 0.15, 0.3, 0.5, 0.7, 0.85, \) and 0.95. The shaded areas in Fig. 7 indicate reverse flow regions.

The distribution of radial velocity components at all selected moments is asymmetric with the diffuser axial distance and demonstrates that the location of the maximum velocity changes as it goes downstream.

At moment A (the maximum radial velocity region), the through flow from the impeller, which preserves higher dynamic energy, extends through the whole region. The radial velocity distribution with the diffuser axial distance at the exit becomes nearly symmetric. At moment B, locally reversed flow regions occur on the hub and shroud side. Strong reverse flow is also observed at the diffuser exit \( r/r_2 = 1.561 \) on the hub side, which is considered the reentering flow from the diffuser exit. At the moment of the minimum radial velocity region C, the reversed flow region due to the incoming flow from the diffuser exit spreads over all the entire diffuser except the core region, where stronger dynamic energy exists. From these results, it is believed that the extension of the reverse flow at \( \phi = 0.16 \) is dominated not by the growth of the local flow separation zone, but by the

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**Fig. 6** Maximum and minimum flow angles with diffuser radius ratio at \( z/b = 0.5 \) (\( \phi = 0.16 \) and 0.06)

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flow that reenters the diffuser exit. At the moment D, the radial velocity recovers gradually from the diffuser entrance and at the center plane of the diffuser width by the through flow coming from the impeller. However, the reverse flow regions still remain along the hub and shroud side and the influence of the through flow does not reach the diffuser exit region. At moment E, the radial velocity component increases continuously and the influence of the through flow expands over almost the entire region of the diffuser except for some small regions on the hub side. The radial velocity begins to recover on the shroud side rather than on the hub side because of the influence of the through flow with higher dynamic energy. This means the magnitude of the radial component of the through flow at the diffuser width is higher near the shroud than near the hub.

The tangential velocity distributions from hub to shroud also clearly demonstrate that the locations of the maximum velocity fluctuate as it moves to the

Fig. 7 Phase-averaged radial and tangential velocity distributions with diffuser radius ratio and axial distance ($\phi=0.16$)
diffuser outlet. The distributions show more distortion near the hub side than near the shroud side as the diffuser radius ratio increases. The distribution over the diffuser width at moment C becomes flat in the fully reversed flow region of the diffuser exit. The tangential velocity distribution shows a steep rise at the hub side, while it grows gradually as the diffuser radius ratio increases. It is believed that this effect is caused by the nature of the spiral vortex and that the rotating direction of the spiral vortex is clockwise.

It is necessary to conduct 3-D measurements at more positions inside the diffuser to get more detailed information about the spiral vortex in the vaneless diffuser.

The five distributions of the radial and tangential velocity components with diffuser axial distance at a further reduced flow rate ($\phi=0.06$) are shown in Figs. 8(a)–(e). The following five moments during one stall period are also taken for further discussion, i.e., the near maximum radial velocity region (moment A), the maximum radial velocity region at $r/r_2=1.561$

Fig. 8 Phase-averaged radial and tangential velocity distributions with diffuser radius ratio and axial distance ($\phi=0.06$)
(moment B), the decreasing region (moment C), the minimum value (moment D), and the radial velocity recovering region (moment E).

The distributions of radial velocity at all selected moments are also asymmetric with the diffuser axial distance and also demonstrate that the locations of the maximum velocity fluctuate as it moves downstream. At moment A, the radial velocity distributions with diffuser width are severely distorted, and localized reverse flow regions are observed on the hub and shroud side at the diffuser exit. Especially on the hub side at the diffuser inlet, a local flow separation zone almost exists during one stall period. At moment B, the local reverse flow regions are diminished at the diffuser exit because of the influence of the through flow from the diffuser midsection. However, strong reverse flow regions are observed alternately on the hub side at a diffuser radius ratio \( r/r_2 = 1.159 \) and on the shroud side at \( r/r_2 = 1.293 \). As the radial velocity components further decrease, the local reverse flow regions grow extensively and simultaneously on both sides at moment C. However, the reentering flow which was observed at the diffuser exit (\( r/r_2 = 1.561 \)) for moment B of the previous flow rate (\( \phi = 0.16 \)) is not detected in this case. Consequently, the local reverse flow regions grow up further and merge into one region, and finally the entire width of the diffuser is filled with the reverse flow at the minimum radial velocity (moment D). From these observations, it is believed that the extension of the reverse flow at \( \phi = 0.06 \) is dominated, not by the reentering flow from the diffuser exit, but by the growth of the local flow separation zone generated on both sides. The flow characteristics at the radial velocity recovering region (moment E) are different than those of \( \phi = 0.16 \). The local reverse flow regions on both walls still remain for a relatively long time when compared to the previous flow rate (\( \phi = 0.16 \)) and the strong reverse flow region at the diffuser exit is also maintained. The dynamic energy of the through flow coming from the impeller is not strong enough to overcome the local reverse flow in the diffuser. The distribution is nearly symmetric with the diffuser width at the diffuser exit.

The flow behavior of the tangential velocity shows nearly the same trend as that of the previous flow rate. However, the magnitude and the profiles of the tangential velocity at each diffuser radial positions are smaller and more uniform than those at \( \phi = 0.16 \). The figures also clearly demonstrate that the fluctuations in the locations with maximum velocity are more severe than those at \( \phi = 0.16 \).

### 3.4 Relative amplitude

The relative amplitude of the velocity fluctuations with the diffuser width and the radius ratio at the rotating stall onset point (\( \phi = 0.16 \)) are shown in Fig. 9. The relative amplitude can be defined by Eq. (3) as

\[
A = \frac{V_{\text{max}} - V_{\text{min}}}{2 \cdot V_{\text{rms}}}
\]  

(3)

The relative amplitude of the radial velocity fluctuations is larger at both walls than that at the center of the diffuser width and at the hub side than at the shroud. The magnitude of the radial components (\( V_{\text{rms}} \)) is smaller than the difference between their maximum and minimum values at both walls. For the tangential components, the relative amplitude with the diffuser width is nearly constant when compared to that of the radial components. At \( r/r_2 = 1.561 \), the amplitude is higher at the diffuser center than at both walls, which is due to the influence of the reentering flow from the diffuser exit. Especially at the center of the diffuser width, the relative amplitude of the radial component is two times higher than that of the tangential component.

The variation in the mean relative amplitude of the total velocity fluctuations with the diffuser radius ratio is shown in Fig. 10. As the diffuser radius ratio...
increases, the relative amplitude also increases, which is independent of the flow rate. However, the rate of the increase becomes small as the radius ratio increases.

4. Conclusion

An experimental investigation has been performed of the unsteady flow in a vaneless diffuser of a centrifugal compressor during rotating stall. The measurements were taken to investigate the flow characteristics inside the vaneless diffuser at two different flow rates at which rotating stall occurred. The results of this study lead to the following conclusions:

(1) Two different mechanisms are recognized for the development of the rotating stall, in other words, for the extension of the reverse flow into the whole diffuser region in a vaneless diffuser. One is dominated by the extension of the reentering flow from the diffuser exit, and the other is dominated by the growth of the local flow separation zone on the hub and shroud side. The former is observed at the relatively higher flow coefficient, $\phi = 0.16$, and the latter is observed at a decreased flow rate, $\phi = 0.06$.

(2) The maximum angle of through flow for one stall period is nearly constant as the diffuser radius ratio increases, but the minimum flow angle decreases. Therefore the fluctuation of the flow direction increases as the diffuser radius ratio increases; it is dominated by the strength of the reverse flow. As the flow rate is reduced from $\phi = 0.16$ to 0.06 and the diffuser radius ratio increases, the stall cell size increases.

(3) The relative amplitude of the radial velocity fluctuations is larger at both walls than that at the center of the diffuser width, and at the hub side than at the shroud. For the tangential components, the relative amplitude with the diffuser width is nearly constant when compared to that of the radial components.

(4) The radial velocity for one stall period at $\phi = 0.16$ increases relatively slowly to a maximum value and then decreases to a minimum value with an intermediate peak. However, when the flow rate decreases further, the radial velocity component decreases to a minimum value without an intermediate peak.

(5) The distributions of radial and tangential velocity are asymmetric with the diffuser axial distance, and the locations of the maximum velocity fluctuate as they move downstream. The tangential velocity distribution with diffuser axial distance becomes flat in the fully reversed flow region at the diffuser outlet.

References


