Experimental Study and Theoretical Analysis of the Rotating Stall in a Vaneless Diffuser of Radial Flow Pump

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Abstract: This paper reports an experimental and theoretical study of rotating stall in a vaneless diffuser which is coupled with a radial impeller. The experiments were conducted at 22 flow rates for two rotating speed: 1200rpm and 1800rpm. The measurements have consisted of: i/ unsteady pressure measurements delivered by two microphones flush mounted on the vaneless diffuser, ii/ 9 steady pressure taps mounted in one radial line on the diffuser to measure the pressure recovery in the vaneless diffuser. The stability of each stall mode was also studied by a 2D linear analysis; and the theoretical prediction was compared to experimental observations. The capabilities and limits of such an approach to predict the development of rotating stall have been evaluated. A non-dimensional analysis of the pressure losses at outlet was conducted to evaluate the effect of the instability development on the performance of the diffuser. It has shown that the arising of rotating stall has a positive effect on the diffuser performance.

Keywords: Rotating stall, Vaneless diffuser, Pressure recovery, 2D linear analysis, Spectrum analysis, Experimental research

Nomenclature

| Greek letters | Diameter | Blade angle |
|---------------|----------|-------------|
| β             |          |             |
| ω             |          | Propagation velocity |
| ρ             |          | Density |
| Q             | Flow rate |
| Qₜ            | Design flow rate |
| f             | Frequency |
| Z             | Blade number |
| S             | Mean blade thickness |
| R             | Radius |
| P             | Pressure |
| n             | Number of cell |
| U             | Circumferential velocity |
| V             | Absolute velocity |

Subscripts

| Subscripts | Impeller inlet |
|-----------|----------------|
| 1         |                 |
| 2         | Impeller outlet |
| 3         | Diffuser inlet |
| 4         | Diffuser outlet |
| b         | Blade |
| imp       | Impeller |
| rs        | Rotating stall |
| d         | Design flow rate |

1. Introduction

One important limit in the operating range of turbomachinery is the occurrence of unstable phenomena in their internal flows. One of them is the rotating stall which is occurring in diffusers of radial compressors or pumps operating at partial flow rates [1, 2]. Several negative effects are associated with rotating stall, such as noise, vibration, mechanical damage and performance reduction. Therefore, characterizing, understanding and predicting this phenomenon is important and necessary. A lot of studies of rotating stall have thus been carried out and the research methods could be summarized as follows:

(1) Theoretical analysis, which includes three dimensional analysis (Jansen [3], Senoo et al [4, 5], Frigne [6], Dou et al [7]) and two dimensional analysis (Tsujimoto [8], Moore [9] and Abdelhamid et al [10]). For two dimensional analysis, it was suggested the 2D core flow instability might be responsible for rotating stall. Generally, it is admitted that the effect of
boundary layer can be ignored when \( \frac{B_3}{R_2} > 0.1 \) (wide diffuser), then the two dimensional analysis is enough to predict the onset of rotating stall.

(2) Experimental researches and numerical analyses (Ljevar et al [11], Dazin et al [1,12], Dodds and Vahdati [13,14], Ferrara et al [15,16], Abdelhamid et al [17-19], Abidogun [20], Nishida and Kobayshi [21,22], Cellai et al [23], Tsurukski et al [24], Suzuki et al [25], Caignaert et al [26]. The main research contents of these experimental studies were focused on several aspects:

i. The effect of flow conditions and geometries of impeller/diffuser on the rotating stall, such as diffuser width, pinch shapes, diffusion ratios, Mach number, Reynolds number, radius ratios of diffuser.

ii. The detection and analysis of inception of rotating stall, and the determination of the characteristics of the flow field, such as the critical flow angle.

iii. Analysis of the characteristics of rotating stall itself: such as the cell size, propagation velocity, frequency and number of stall cells.

Although rotating stall was studied in many aspects listed above, some of the mechanisms of the instability and their effect on the diffuser performance are still unclear. Therefore, the present work reports studies managed with the support of an experimental test rig which was used to observe the rotating stall in the vaneless diffuser. 22 flow rates were tested for two rotating speed: 1200 and 1800 rpm. A dedicated spectrum analysis was applied to spectrally identify and characterize the rotating modes which were observed in the vaneless diffuser: number of cells, amplitude and propagation velocity of the stall. The result has been compared to existing two-dimensional theory and its capabilities and limits have been discussed. In order to determine the effect of rotating stall on the diffuser performance, the experimental and theoretical pressure recovery in the vaneless diffuser have been calculated. The impact of the rotating stall on the diffuser performance is then discussed.

2. Experiment Apparatus

The experiment test rig (figure 1) includes a radial impeller which was coupled with a vaneless diffuser downstream, and a suction pipe with a tank upstream. In order to ensure axisymmetric outlet boundary conditions, no volute was set downstream of the vaneless diffuser. The configuration was equipped with 9 steady pressure taps which were flush mounted on the diffuser wall in one radial line. To ensure the stability of the inlet flow, a tank, equipped with a honeycomb flow straightener, was set before the suction pipe. A set of changeable diaphragms is available to be installed at the tank inlet to adjust the flow rate.

![Figure 1](image)

Two Brüel & Kjaer condenser microphones (Type 4135) were used to get the unsteady pressure information, and the location of them was set at the same radius on the diffuser wall but with an angle difference of \( \Delta \theta = 75^\circ \). The data were acquired by LMS. Test Xpress. The sampling time and frequency were 600 seconds and 4096Hz, respectively.
The experiment was made in air and the impeller was rotating in clockwise direction. The main parameters of impeller and vaneless diffuser are given in table 1. 22 flow rates ($Q/Q_d=0.26 \rightarrow 1.53$) were performed for two rotating speeds: 1200rpm and 1800rpm. The pump performance (in terms of static pressure rise) is plotted in figure 1.

| Impeller | Inlet Diameter | 282.2mm |
|----------|----------------|---------|
| $D_1$    | Outlet Diameter | 513.2mm |
| $D_2$    | Outlet Width    | 38.5mm  |
| $B_2$    | Number of Blades | 7 |
| $\beta_2$ | Outlet Blade Angle | 22.5° |
| $S$      | Mean Blade Thickness | 9mm |
| $Q_d$    | Design Flow Rate(1200rpm) | 0.236m$^3$/s |

| Vaneless Diffuser | Inlet Diameter | 514.2mm |
|-------------------| Outlet Diameter | 768.0mm |
| $D_3$             | Diffuser Width  | 38.5mm  |
| $R_e=\rho V_d D_h \mu$ | Reynolds Number($Q/Q_d=1$) | 9.08×10$^4$ |

3. Experimental results

3.1. Characteristics of rotating stall

The spectrum information which was acquired by two microphones allows to detect the development of rotating stall in the vaneless diffuser and also to analyze its characteristics [12, 25, 26], as shown in figure 2, where $\psi$ was defined as:

$$\psi = \frac{P}{\rho U_z^2}$$ (1)

According to the spectrum analysis, rotating stall was observed at the five partial flow rates: $Q/Q_d=0.26$, 0.36, 0.47, 0.56, 0.58, and cannot be observed when the flow rate ratio is higher than 0.6. For a given flow rate, rotating stall characteristics are similar for the rotating speed investigated in the present work. When the flow rate ratio is higher than 0.6 (a typical example is proposed figure 3a), the dominant frequency was the blade passing frequency. Besides, the impeller frequency and its harmonics are also significant.
On the contrary, for the five lower partial flow rates, the spectra are dominated by several low frequency peaks due to rotating stall, as shown in figure 3b, c, d, e and f.

Some peaks may correspond to different modes of rotating stall, but others to harmonics or nonlinear interaction fundamental phenomena. Therefore, a dedicated spectrum analysis was applied to identify harmonics and the different modes of rotating stall. Considering the amplitude, the result of flow rate \( Q/Q_d = 0.26 \) (figure 3b) are shown in table 2.

**Table 2. Rotating stalls and harmonics at flow rate \( Q/Q_d = 0.26 \)**

| Frequency(Hz) | Amplitude(Pa\(^2\)) | Mode                  |
|--------------|----------------------|-----------------------|
| \( f_1 = 16.5 \) | 410.1                | Stall mode 1          |
| \( f_2 = 13 \)     | 87.2                 | Stall mode 2          |
| \( f_3 = 20 \)     | 82.6                 | Impeller frequency    |
| \( f_4 = 29.5 \)   | 82.1                 | Nonlinear interaction(\(=f_1+f_1\)) |
| \( f_5 = 33 \)     | 69.8                 | Harmonic(\(=2f_1\))  |
As it can be seen, several frequencies corresponding to different rotating stall modes can be identified. A synthesis of the unstable modes that can be evidenced for each flow rates is proposed in table 3, in which \( n \) is the number of rotating cells that could be identified through the analysis of the phase difference of the pressure signals of the two microphones.

Table 3. Different modes of rotating stall

| \( Q/Q_{d} \) | Dominant | Second | Third | Fourth |
|---------|----------|--------|-------|--------|
| 0.26    | \( n=3 \) | \( n=2 \) | no    | no     |
| 0.36    | \( n=2 \) | \( n=3 \) | \( n=1 \) | no     |
| 0.47    | \( n=4 \) | \( n=2 \) | \( n=3 \) | \( n=1 \) |
| 0.56    | \( n=4 \) | no     | no    | no     |
| 0.58    | \( n=4 \) | no     | no    | no     |

It can be seen that with the flow rate decrease, the dominant stall mode starts from 4 cells mode (\( Q/Q_{d}=0.47, 0.56, 0.58 \)), then becomes 2 cells mode (\( Q/Q_{d}=0.36 \)), and changes to 3 cells mode (\( Q/Q_{d}=0.26 \)) at last, this trend agree well with previous study on the same experimental configuration (Dazin et al \[11\]). Based on the rotating stall frequency, the propagation velocity of the stall cell-\( \omega \), can be estimated as follow:

\[
\omega = 2\pi \cdot f_{n} / n
\]

Then the amplitude and propagation velocity for different modes are drawn in figure 4.

![Figure 4](image)

Figure 4. The amplitude and propagation velocity of stall cells for 1200rpm

Concerning the amplitude, the general tendency is an increase of the intensity of the unstable flow with the flow rate decrease. For the propagation velocity shown on the right side, it can be seen that the propagation velocity of each mode always increases with flow rate decreases. This could be linked to the increased circumferential velocity at the inlet of the vaneless diffuser. In addition, it seems that the mode with fewer cells propagates faster than the mode with more cells (Except mode \( n = 1 \), but this mode is never the dominant mode).

3.2. Stability analysis of stall modes

Considering that the diffuser is wide (\( B_{v}/R_{3}=0.15 \)), 2D linear stability analysis in the core flow of the diffuser was applied to study the stability of the flow. The analysis is based on the assumptions of a 2D, non-viscous flow with axisymmetric boundary conditions at inlet. The detail of the methodology used to predict the flow stability is available in Tsujimoto \[8\]. The results of this analysis are compared to the present experimental work to identify if such a methodology is able to predict the unstable modes developing. According to linear stability analysis, for a given diffuser radius ratio \( R_{4}/R_{3} \) and a given number of unstable modes, rotating stall occurs when the absolute flow angle at diffuser inlet is below a limit which is called the critical flow angle. In present experiment the radius
ration is \( R_4/R_3 = 1.5 \). In figure 5, the stall mode can be judged to be stable or unstable by comparing the diffuser inlet flow angle to the critical flow angle predicted theoretically. For example, the stall modes with 1, 2, 3 and 4 cells were experimentally observed at flow rate \( Q/Q_d = 0.47 \). At this flow rate, the inlet flow angle is estimated to be \( \alpha_3 = 4.4 \, ^\circ \), by the calculation of the velocity triangle and diffuser inlet. The analysis of figure 5 shows that, for this inlet flow angle, the mode \( n=1 \) should be linearly stable, whereas the other modes are unstable. The final results of stability analysis are given in table 4 for each flow rate. The theoretical prediction agree well with experimental observation at \( Q/Q_d = 0.47 \). But some differences are noticeable for the other flow rates. This could be attributed to two major reasons:

1. The flow angle is not measured in the experiments, but estimated by the determination of the velocity triangle at impeller outlet.
2. The theoretical prediction is limited in this study to the critical angle determined by linear stability analysis. Its extension to angles lower than the critical one is needed in order to compute the growth rate of each mode which could give more accurate conclusions. Moreover, a nonlinear stability analysis could also give more realistic predictions on the competition between different unstable modes. These two tasks are postponed to a future work.

![Figure 5. Critical flow angle versus diffuser ratio](image)

**Table 4. Stall modes: theoretical prediction and experimental observation**

3.3. **Effect of rotating stall on the diffuser performance**

Since the presence of rotating stall in the vaneless diffuser is confirmed, it is possible to determine the effect of rotating stall on the diffuser performance. Through the 9 pressure taps along the diffuser wall, the pressure evolution from diffuser inlet to outlet was measured experimentally, and then the performance curve was plotted in figure 6. In this figure, the pressure recovery in the diffuser in scaled by the kinetic energy at the diffuser inlet. (The absolute velocity at diffuser inlet has been evaluated by the determination of the velocity triangle at impeller outlet).
In normal conditions (with no rotating stall), it can be seen that the diffuser performance is decreased with the flow rate, but is increased at the onset of rotating stall, and then decreased again when rotating stall structure full developed.

To determine the effect of rotating stall on the diffuser performance, the isentropic increase of pressure due to the velocity decrease in the diffuser has been calculated theoretically compare to the experimental pressure recovery (figure 7). At the diffuser outlet, the difference between theoretical and experimental result is then defined as “Losses”.

If the flow in vaneless diffuser is stable, the losses are supposed to be mainly due to friction along the length $L$ of a streamline:

$$\text{Losses} = \frac{\lambda}{D_h} \frac{\rho V^2}{2} L$$

Where $D_h$ is the hydraulic diameter, $\lambda$ is the friction factor, and in the present vaneless diffuser,

$$D_h = \frac{4S}{C}$$

Where $S$ is the cross sectional area which is equal to $2\pi r^* B$, and $C$ is the perimeter of the cross-section which is equal to $4\pi r^*$.

If the losses are scaled by $\frac{1}{2} \rho V^2$, the quantity: $\text{Losses} = (\frac{\rho V^2}{2}) = \frac{\lambda}{D_h} L$, is depending only on the length of streamline $L$. The streamline is assumed to be a logarithmic spiral.
Its length $L$ can be estimated by equation (5). It is depending only on the value of the absolute flow angle $\alpha$ at diffuser inlet and is thus increasing with the flow rate decrease (figure 9).

$$L = a\sec(\alpha + \frac{\pi}{2})(e^{\cot(\alpha + \frac{\pi}{2})} - 1)$$

(5)

\[\text{Figure 9. Length of streamline versus flow rate}\]

The evolution of the losses versus the length of streamline was drawn in figure 10. As expected, the losses increase linearly with the streamline length for small value of $L$, that is, for operating conditions without rotating stall. What is particularly notable in this figure is that the arising of rotating stall is corresponding to a clear drop of the losses. This is confirmed by figure 11, presenting the evolution of the losses as a function of the flow rate. In this figure, the losses drop at the arising of rotating stall is also obvious. After this drop, the losses are increasing again with the flow decrease but with a slope smaller than the exponential tendency observed in the operating range with no rotating stall. It seems that the arising of the instability could be a way, for the flow, to reduce its losses when the flow angle is very low at diffuser inlet.

\[\text{Figure 10. Losses at diffuser outlet versus length of streamline}\]

\[\text{Figure 11. Losses at diffuser outlet versus flow rate ratio } Q/Q_d\]

4. Conclusions

The rotating stall in a vaneless diffuser was experimentally and theoretically analyzed. A dedicated spectrum analysis was applied to characterize rotating stall. Based on 2D linear theory, the stability of different stall modes was analyzed. The evolution of the losses in the diffuser with the flow rate was studied to determine the effect of rotating stall on the diffuser performance. Several conclusions were drawn as follows:

1. 22 flow rates were tested in experiment, and rotating stall was observed at 5 partial flow rates: $Q/Q_d=0.26$, 0.36, 0.47, 0.56 and 0.58. It was found that the dominant stall mode was characterized by 4 cells at $Q/Q_d=0.47$, 0.56, 0.58, 2 cells at $Q/Q_d=0.36$ and 3 cells at
Moreover, the experimental result shows that the characteristics of rotating stalls are the same for the two rotating speed investigated in the experiment.

(2) By the comparison between theoretical and experimental result, it is verified that 2D linear analysis could predict the inception of rotating stall, but unable to give the amplitude of each mode to determine the dominant mode, and the non-linear interaction between different modes also unsolved. It should be noticed that, in the present work, the flow angle at the inlet of diffuser was estimated and not measured. The exact flow angle will be measured experimentally in the next future. Moreover, a non-linear stability analysis will be also conducted to try to improve the theoretical prediction.

(3) The diffuser performance and losses were analyzed. The result shows that, for operating points with no rotating stall, the losses increase linearly with the length of streamline in the diffuser. But, it has also been observed that the development of rotating stall is corresponding to a decrease of the losses. The arising of instability has thus a positive effect on the diffuser performance.

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