Vaneless diffuser core flow instability and rotating stall characteristics

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ABSTRACT

Since rotating stall in centrifugal compressors limits the operating range and causes damage and noise nuisance it must be avoided. To widen the operating range more knowledge on flow dynamics of rotating stall mechanism is required. To study the vaneless diffuser rotating stall two-dimensional flow model is used where the influence of the wall boundary layers is neglected. At the diffuser inlet rotating jet-wake velocity is prescribed and at the outlet constant static pressure is assumed. Under these conditions a two-dimensional rotating instability is observed, which is studied in terms of the rotating stall frequencies and pressure fluctuations.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>( N )</td>
<td>number of impeller blades</td>
</tr>
<tr>
<td>( r )</td>
<td>radius</td>
</tr>
<tr>
<td>( t )</td>
<td>time</td>
</tr>
<tr>
<td>( u )</td>
<td>radial velocity component</td>
</tr>
<tr>
<td>( v )</td>
<td>tangential velocity component</td>
</tr>
<tr>
<td>( v_{\text{tip}} )</td>
<td>impeller tip speed</td>
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</tbody>
</table>

Greek letters

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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</thead>
<tbody>
<tr>
<td>( \alpha )</td>
<td>flow angle</td>
</tr>
<tr>
<td>( \theta )</td>
<td>circumferential position</td>
</tr>
<tr>
<td>( \sigma )</td>
<td>slip factor</td>
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\[ \omega \] angular velocity

**Subscripts**
- \( cr \) critical
- \( i \) impeller
- \( m \) mean
- \( 2 \) diffuser inlet
- \( 3 \) diffuser outlet

## 1 INTRODUCTION

When centrifugal compressors are operating at low mass flows unsteady flow phenomena such as surge and rotating stall will occur. These instabilities can cause critical operating conditions with strong dynamical loading on the blades. Therefore, these instabilities are not tolerated during compressor operation and compressors are operated at reduced pressure ratios to keep a safety margin to the stability limit.

Main goal of this research is to book progress in development of stall and surge control systems in centrifugal compressors. Willems (1) has designed and implemented a model for active surge control on a small scale centrifugal compressor. The active surge control in his setup resulted in occurrence of rotating stall when surge was postponed. Therefore, to increase the region of operation in centrifugal compressors better understanding of rotating stall flow dynamics is required. The understanding of basic effects that can lead to the flow breakdown is important for successfully delaying or controlling it.

In the literature different approaches have been used to investigate vaneless diffuser rotating stall. Jansen (2), Senoo (3), Frigne (4) and Dou (5) have used the wall boundary layer theory and they generally hold the effect of the three-dimensional wall boundary layers responsible for the occurrence of rotating stall in vaneless diffuser. On the other hand Abdelhamid (6) and Tsujimoto (7) have applied a two-dimensional diffuser flow approach where the effect of wall boundary layers is not taken into account. They suggest the existence of the two-dimensional core flow instability at the onset of rotating stall. Abdelhamid (8), Dou (9) and You Hwan (10) found that the performance of the vaneless diffuser is different for narrow and wide diffusers and suggested that different flow mechanisms might exist that can lead to the occurrence of rotating stall.

In this analysis it is assumed that the core flow instability can be one of the mechanisms that causes rotating stall in vaneless diffusers. Therefore, rotating stall inception in vaneless diffuser is investigated from the point of view that it can be a two-dimensional flow instability.

In this paper an instability analysis of the two-dimensional vaneless diffuser core flow is performed using CFD. In the following section numerical model and modeling aspects are explained. Third section introduces the obtained two-dimensional rotating instability and describes its characteristics and similarity with rotating stall. Fourth section studies the influence of the diffuser radius ratio on the two-dimensional rotating instability and
pressure fluctuations. Fifth section discusses the value of the pressure fluctuation signals and their interpretation while the final section discusses the application of the numerical model and summarizes conclusions.

2 NUMERICAL MODEL

To study the vaneless diffuser core flow instability two-dimensional incompressible flow model is developed. A two-dimensional core flow is modeled in the plane parallel to the diffuser walls. Influence of the wall boundary layers is not taken into account and no diffuser width is modeled. Therefore, this model applies only to the wide vaneless radial diffusers with parallel diffuser walls where the core flow dominates and the influence of the wall boundary layers can be neglected.

For the numerical analysis a commercial software package Fluent is used. The governing integral equations for the conservation of mass and momentum are solved using the finite-volume approach. Second-order implicit unsteady formulation is used to model the time dependent terms and for discretization of the convection terms QUICK scheme as proposed by Leonard (11) is used.

Although the studied fluid flow is turbulent incompressible laminar viscous flow solver is applied. This is done to avoid excessive numerical dissipation of turbulence models. Turbulence models capture the diffusion-like character of turbulent mixing associated with many small eddy structures. It is assumed that the two-dimensional core flow instability is large structure instability of approximately the same order as the jet-wake at the diffuser inlet. Turbulence models damp out large eddy structures and result in delay of instability inception.

No experimental results are discussed in this paper, but to understand the texture of the numerical model a few words are spent on the relation between the experimental and numerical model. For experimental study of rotating stall a water model of the centrifugal compressor stage is build. The hydrodynamic analogy shows that water model of the same geometry must operate at much lower fluid velocities and shaft speeds than the air compressor configuration. Deceleration of the process allows plain visualization of the unsteady flow phenomena using particle image velocimetry. Therefore, scaling of the existing air compressor stage into a water model of the centrifugal compressor stage is performed and applied. Operating condition near the stability limit of an existing air compressor is scaled to obtain near stall water model operating conditions. Since the numerical model is destined to support the experimental study it is based on the scaled diffuser geometry and fluid operating flow conditions.

The reference geometry of the modeled vaneless diffuser has the diffuser radius ratio of \( r_3/r_2 = 1.52 \) where the inlet radius ratio equals 0.3225 [m] and the outlet radius ratio 0.4908 [m]. At the reference operating condition the tangential and the mean radial velocity component at the diffuser inlet are equal to \( v = 7.7564 \) [m/s] and \( u_m = 0.9281 \) [m/s] respectively. A simple two-dimensional quadrilateral grid consisting of 750 by 62 elements is applied to this geometry and the convergence criterion of \( 10^{-3} \) is applied to the continuity, \( x \)-velocity and \( y \)-velocity residual.
At the diffuser outlet constant static pressure is assumed. At the diffuser inlet jet-wake alike velocity pattern is prescribed that is rotating in the clockwise direction. Tangential velocity component is constant and the radial velocity component is described by the square hyperbolic tangent function, $\tanh(N \cdot \theta + \omega_i \cdot t)$. The rotational velocity of the jet-wake $v_{tip}$ is coupled to the tangential velocity component by the slip factor, which is assumed to be constant, $v = \sigma \cdot v_{tip}$. At reference condition the jet-wake pattern exists of 17 jet-wake lengths around the circumference corresponding to 17 impeller blades, $N = 17$. Reference ratio of the jet-to-wake intensity equals 5.5.

3 TWO-DIMENSIONAL ROTATING INSTABILITY

With this two-dimensional model of the vaneless diffuser core flow a two-dimensional rotating instability similar to rotating stall is obtained. This instability develops within four to eight impeller revolutions and it consists of a number of rotating cells that are propagating with approximately 40% of the impeller speed.

By alternately varying the tangential velocity component $v$ and mean radial velocity component $u_m$ the stability limit is determined. The two-dimensional rotating instability turned out to occur when $v/u_m$ ratio at the diffuser inlet exceeds a certain value. For $v/u_m < (v/u_m)_{cr}$ the steady stable operating condition is obtained, which is shown in figure 1a. For $v/u_m > (v/u_m)_{cr}$ the two-dimensional rotating instability similar to rotating stall occurs, which is shown in figure 1b. Solutions in figure 1 are represented by the contours of velocity magnitude where the rotating cells are high velocity regions.

![Figure 1: a) stable operating condition and b) two-dimensional rotating instability](image)
The \( \frac{v}{u} \) ratio and thus also the core flow stability can be expressed in terms of the flow angle \( \alpha \) since the flow angle \( \alpha \) is the angle between the absolute velocity and tangential velocity component. The mean flow angle can be written as \( \alpha_m = \tan^{-1}(\frac{u_m}{v}) \).

According to the core flow stability criterion two-dimensional rotating instability will occur as soon as the mean flow angle at the diffuser inlet becomes smaller than the critical flow angle.

Because in the past years the vaneless diffuser rotating stall is investigated using pressure transducers dynamic pressure fluctuation signals are compared with the earlier measurements found in the literature. Dynamic pressure signal is monitored during the gradual mass flow decrease at the time that stable operating condition starts to become unstable and rotating instability occurs. In the model dynamic pressure is monitored at one point, which is located exactly in the middle between the inlet and the outlet radius. The monitored pressure signal from the two-dimensional vaneless diffuser core flow model is shown in figure 2 and the measured pressure signal from the literature is shown in figure 3.

Comparison between the figures 2 and 3 implies that also the obtained pressure signals near the stability limit look similar to the measured pressure fluctuations in the real vaneless diffuser at stall inception.

![Figure 2: pressure signal from the model during the instability inception](image-url)
The other numerical results, such as the critical flow angle and the diffuser radius ratio influence, also agree well with the measurements found in the literature, which is shown in Ljevar (13). Because of high similarity with rotating stall and good agreements with the measurements found in the literature it is presumed that this instability might contribute to the vaneless diffuser rotating stall.

4 DIFFUSER RADIUS RATIO

To study the influence of the diffuser radius ratio on the vaneless diffuser core flow stability the radius ratio is varied. Major influences of the diffuser radius ratio on the core flow stability are already discussed in Ljevar (13). The diffuser radius ratio is varied by changing the outlet radius and leaving the inlet radius unchanged.

In this section monitored pressure fluctuations for different radius ratios are analyzed and discussed. Pressure signals are monitored for different diffuser radius ratios during the instability inception. They are monitored each time at $r = (r_3 + r_2)/2$ while the mean radial velocity component is gradually being decreased. Decrease of the mean radial velocity component corresponds to the decrease of the mass flow which finally leads to

Figure 3: pressure signals measured in the vaneless diffuser during rotating stall inception – source Abdelhamid (12)
core flow instability. Monitored pressure signal for \( r_3/r_2 = 2 \) is shown in figure 4 and for \( r_3/r_2 = 1.52 \) and \( r_3/r_2 = 1.2 \) in figures 5 and 6 respectively.

Figure 4: pressure signal during instability inception for \( r_3/r_2 = 2.0 \) and corresponding stable and unstable operating condition
In figures 4, 5 and 6 gradual transition from the stable steady operating condition into the two-dimensional rotating instability can be clearly noticed. In figure 4 stable operating condition ranges from $t = 0$ to $1$ [s] and then the instability starts to develop. The instability is fully developed at $t = 2$ [s] approximately. The corresponding stable and unstable operating conditions are given in figure 4 below. In figure 5 the stable steady operating
condition at \( t = 0.5 \) [s] starts immediately to turn into the two-dimensional rotating instability, which is fully developed at approximately \( t = 0.75 \) [s]. The stable and unstable operating condition for \( r_3/r_2 = 1.52 \) are given below in figure 5.
Finally for $r_3/r_2 = 1.2$ the pressure signature is given in figure 6 where the stable steady operating condition starts to fall apart at $t = 2$ [s] and the two-dimensional rotating instability is fully developed at $t = 2.2$ [s] approximately. The corresponding stable and unstable operating condition for $r_3/r_2 = 1.2$ are given below in figure 6.

After the two-dimensional rotating instability is fully developed the amplitude of the pressure fluctuations and the average pressure remain constant for some time. But after a while the average pressure starts to decrease towards zero which can be explained by shifting of the rotating cells towards the diffuser inlet. As the mass flow decreases rotating cells move closer to the diffuser inlet because they are not being blown away by the radial velocity component. Since the pressure transducer position remains unchanged the highest pressure fluctuations move out of the measurement point. This effect is higher for larger diffuser radius ratios because in that case the diffuser space within which the cells are able to move around is larger. Not only the average pressure decreases as the mass flow is being reduced but also the amplitude of the pressure fluctuations slightly decreases. This is also due to the fact that rotating cells move towards the diffuser inlet but besides that also because rotating cells become somewhat smaller as the mass flow is being decreased.

For $r_3/r_2 = 2.0$ the amplitude of the fluctuations at stable operating condition is smaller than the amplitude during the instability. But as the radius ratio decreases the amplitude of the fluctuations at stable steady operating condition becomes larger relative to the amplitude at instability, which is also due to the available diffuser space. At higher diffuser radius ratios diffuser space is larger and the alternating pattern near the diffuser outlet, as shown in figure 1a, is relatively smaller than for small diffuser radius ratios. For $r_3/r_2 = 1.2$ the amplitude of the pressure fluctuations is almost the same for steady stable operating condition as well as for the two-dimensional rotating instability.

5 PRESSURE SIGNALS

Figure 10 shows the pressure fluctuations measured in the stalled vaneless diffuser by Ferrara (14). In this figure the stall changing pattern in terms of the pressure signal is shown. They show that any changes in the rotating stall pattern can be noticed on the measured pressure signal. What the exact changes are within the rotating stall pattern during these measurements, they could only assume.

It is known that rotating stall pattern is not only influenced by the configuration type, but that it also changes within the same configuration when the operating conditions change. This makes the study on rotating stall even more complex. Therefore, it could be very useful to be able to read the pressure signals and to translate the pressure transducer signature into the terms of physical rotating stall. Several techniques are developed to determine the number of rotating stall cells and their propagation speed from the pressure signals, but when the pressure signal is changing one could only suggest what is going on within the diffuser. Therefore, use of the numerical models to investigate the instability behavior together with corresponding pressure signals can be very helpful in the future study on rotating stall. Pressure signatures can reveal not only the number and propagation
speed of rotating stall cells, but also their size and location in the diffuser as well as their merging or splitting behavior.

Figure 10: stall change – source Ferrara (14)

6 CONCLUSIONS

To study the core flow instability within the wide vaneless radial diffusers of centrifugal compressors two-dimensional viscous incompressible flow model is developed within Fluent. The influence of the wall boundary layers is neglected and no diffuser width is modeled. Using this model two-dimensional rotating instability associated with rotating stall in wide vaneless diffusers is found to exist.
The two-dimensional rotating instability is similar to rotating stall since it develops within four to eight impeller revolutions and consists of a number of rotating cells that propagate with a fraction of the impeller speed. Also the dynamic pressure fluctuations are similar to the pressure transducer measurements found in the literature. It is found that the core flow stability limit can be expressed in terms of the flow angle.

The diffuser radius ratio has strong influence on the two-dimensional rotating instability. As the two-dimensional rotating instability differs in shape, size and propagation speed for different diffuser radius ratios also the dynamic pressure signature changes. Physical differences between rotating instabilities for different radius ratios can be noticed in the dynamic pressure signature.

Not only diffuser geometry influences the rotating stall pattern but also the different operating conditions can cause changes within the same compressor configuration. For better understanding of stall behavior it would be very useful to be able to read the pressure signals. The pressure transducer fluctuations tell something about the physical behavior and characteristics of rotating stall within the diffuser space. Several techniques are developed to determine the number of rotating stall cells and their propagation speed from the measured pressure fluctuations, but when the pressure signal is changing from one pattern to the other one can only suggest what is happening within the diffuser. Therefore, numerical models can be very useful in the future study on rotating stall since they can help to investigate the relation between the instability characteristics and corresponding pressure signals. Pressure signatures can reveal not only the number and propagation speed of rotating stall cells, but also their size, location in the diffuser and their merging or splitting behavior.

Study of the pressure fluctuations together with other numerical results can contribute to the approximation and prevention of noise and damage in some centrifugal compressors and systems in which they are implemented.

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