CFD Predictions of Induced Fluid Forces and Dynamic Coefficients for a Compressor Eye Seal

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Abstract—Labyrinth seals are widely used in gas compressors to reduce internal leakage and increase the compressor efficiency. Due to the eccentricity between the rotating impeller and the stationary part as well as the shaft whirling motion, forces are generated when the leakage flow passing through these seals. These driving forces act on the rotor and may modify its dynamic stability. Thus, accurate predictions of these forces become a very important task for compressor rotordynamic designs. This paper presents tri-dimensional CFD investigations to simulate driving forces and dynamic coefficients of an eccentric labyrinth seal generally used as a compressor eye seal. The seal has five tapered teeth fixed on the stator and the working fluid is air. The model accuracy has been compared to previous Bulk Flow and CFX results on the same seal. These comparison results concern the pressure distribution along the seal as well as radial force components at zero preswirl. Moreover, results about pressure distribution in the circumferential direction and velocity vectors are presented. Additionally, a parametric study has been conducted to show the effects of the whirl frequency along with three inlet swirl velocities (negative, zero and positive) and three high shaft speeds on the rotordynamic characteristics of the seal. Obtained results show that zero or moderate negative inlet swirl velocities are beneficial from a point of view dynamic stability of the seal and the rotordynamic coefficients have to be considered whirling frequency-dependent at high rotor speed rates.

Index Terms—Labyrinth seal, Dynamic stability, Rotor dynamic coefficients, CFD, Inlet swirl, Whirl frequency

1 INTRODUCTION

In response to the constantly increasing demand for higher levels of productivity, modern turbomachines are designed to be more efficient. They should run at higher speeds, higher pressures, tighter clearances, and last longer. As a particular component improving performance of such machines, annular gas seals are key elements to reduce leakage flows between rotating and stationary parts of turbomachines. These elements are effective not only on the system damping but even on the system stiffness, strongly affecting both the rotor stability and the position of the rotor first natural frequency [1, 2]. It has been proven that annular seals dynamic instability can lead to serious damage if not properly controlled.

Labyrinth seals, the most common type of annular gas seals, are widely used in turbomachinery including gas turbines, turbo pumps and compressors. Figure 1 illustrates typical labyrinth seals integrated in a multi-stage centrifugal compressor. These seals are composed of a series of circular blades and annular grooves which present a tortuous path for flow of the process fluid. The working principle of a labyrinth seal consists in separating regions at different pressures through the clearance where the conversion of pressure energy into kinetic energy occurs. The latter can then be dissipated into thermal energy in the cavity downstream the clearance or transferred to the next gap through the kinetic carryover [3]. The major advantages of labyrinth seals are their simplicity, reliability, and tolerance to large thermal and pressure variations [4]. Although labyrinth seals have been confirmed to be a major source of driving forces resulting in rotor instability problems, they remain the common sealing solution in compressors and turbines as balance drum, interstage, center, shaft end, and impeller eye seals. Due to the many geometrical and physical influence factors on the rotordynamic characteristics of labyrinth gas seals and their decisive influence on the dynamic stability of turbomachines, worldwide researchers have conducted a large number of experimental and theoretical investigations focused on measuring and predicting the rotordynamic coefficients for this kind of seals. However, various disagreements have been reported between predictions and real measurements on the rotor dynamic behaviour of these seals [5 - 7]. Thus, accurate prediction of fluid induced forces and rotor dynamic coefficients remains a demanding and challenging task for labyrinth gas seals [8].

Eye seal
Balance drum seal

Fig.1. Typical labyrinth seals in a multi-stage centrifugal compressor
Currently, there are two known solutions to predict dynamic characteristics of labyrinth gas seals, the bulk flow and CFD methods.

The bulk flow models were developed in the early 1980s to predict dynamic characteristics of annular seals and these models continue to be used in the industry [9]. Iwatsubo [10] is among the first authors which developed bulk-flow approach. Subsequently, to assess its predictions accuracy, Childs and Scharrer [11, 12] compared bulk flow calculations within labyrinth seals to experimental measurements. Due to the complex geometry of labyrinth seals, the bulk-flow method needs to simplify the physical problem models and governing equations as well to obtain approximate and quick results. Therefore, this approach yields good results for plain annular seals but poor predictions when recirculation is present in the flow field [13]. To improve the bulk flow model, multiple control volume techniques have been used that divide the geometry of the seal [14]. These techniques associated dominant flow behavior into different control volumes which are then linked by appropriate boundary conditions. However, a priori knowledge of the flow required parameters is not always known and the interface conditions change for different seal operating conditions. In this context, to improve bulk flow predictions in labyrinth seals, Kirk [15] has developed a method to calculate inlet swirl velocity which it has been proven that it affects strongly the dynamic stability of the rotor. Furthermore, these models require some empirical relationships such as Hirs and Moody friction factor relationships to quantify shear stress in a developed turbulent pipe flow [16, 17]. These empirical coefficients have been the subject of many investigations [18, 19] to formulate an accurate model for them. However, it is difficult to capture the full nature of the friction factor through bulk flow models without an experimental measurement for each seal [20].

Unlike bulk flow model, computational fluid dynamic “CFD” does not rely on empirical wall and interface constants that may change for varying applications and geometry. In addition to this, the exact geometry of the seal may be modelled allowing optimization of the teeth profile. However, the obvious drawback of CFD compared to bulk-flow is increased computational requirement. The present research attempts to calculate the induced forces acting on the rotor and the dynamic characteristics of an eccentric labyrinth gas seal based on tridimensional CFD techniques solving the general Reynolds Averaged Navier-Stokes equations along with appropriate turbulence model.

2 SEAL GEOMETRY AND COMPUTATIONAL MODEL

2.1 Seal Geometry

The labyrinth gas seal object of this work has five tapered teeth fixed on the stator lateral surface. This kind of seals is generally used as impeller eye seal in multi-stage centrifugal compressors to limit secondary flows. The seal’s mean radial clearance is $C = 0.292$ mm and its relative eccentricity is $\varepsilon = \varepsilon/C = 0.1$ where $\varepsilon$ is the orbital motion radius. The 2-D seal geometry is shown in figure 1.

The seal working fluid is air. A cut section of the 3-D fluid computational domain is shown in figure 2. The teeth are represented by circumferential grooves in this computational domain.

2.2 Governing Equations

The turbulent flow through the seal is governed by the continuity and momentum equations.

Continuity equation:

$$\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u_i)}{\partial x_i} = 0 \quad (1)$$

Momentum equation:

$$\frac{\partial \rho u_i}{\partial t} + \frac{\partial (\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ \mu_{eff} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] + S_M \quad (2)$$

where

- $S_M$ is the sum of body forces
- $\mu_{eff}$ is the effective viscosity accounting for turbulence.
- $\mu_{eff} = \mu + \mu_t$ where $\mu$ is the turbulence viscosity and $\mu_t$ is a constant.

These equations are completed by the $(k, \omega)$ SST turbulence model.
2.3 Meshing Procedure

Hexahedral mesh elements were used to create three dimensional non-uniform structured meshes in the entire computational domain. An adequate mesh refinement is allowed to the clearance area and boundary layers to accurately calculate desired physical parameters of the flow within the seal. Fig. 5 shows a cut section of generated computational grids in the 3-D computational domain.

2.4 Frame Transfer from Stationary to Rotating

Observing the motion of rotor-seal system from a stationary frame, the rotor is spinning at the speed ω while also whirling at the speed Ω at the same time, which means that the location of rotor and thus the shape of mesh are changing all the time. So, it is actually a transient problem involved with mesh moving. To avoid a transient analysis and moving mesh, a rotating frame with the speed Ω was applied as shown in figure 5. In the rotating frame, the rotor itself spins at the speed (ω - Ω), while the stator spins at the speed Ω in the opposite direction to the frame. Thus it becomes a steady state problem and there is no mesh moving.

2.5 Induced Forces and Rotordynamic Coefficients

The fluid driving forces exerted on the rotor can be obtained at each whirl frequency Ω by integration of the static pressure along and around the seal rotor surface.

\[ F_y = R \int \int p \cos \theta \, d\theta \, dx \]
\[ F_z = R \int \int p \sin \theta \, d\theta \, dx \]

For small motion of the rotor center about a centered position, the relation between the reaction-force components and the shaft motion is defined by the following linearized dynamic model where the added mass terms are neglected.

\[
\begin{bmatrix}
F_y \\
F_z
\end{bmatrix} = 
\begin{bmatrix}
K & k \\
k & K
\end{bmatrix}
\begin{bmatrix}
y \\
z
\end{bmatrix} + 
\begin{bmatrix}
D & d \\
d & D
\end{bmatrix}
\begin{bmatrix}
y' \\
z'
\end{bmatrix}
\]

where (y, z) define the lateral motion of the rotor center relative to the stator, (Fy , Fz) are the components of the reaction force acting on the rotor. (K, k) and (D, d) represent the direct and cross-coupled stiffness and damping coefficients, respectively. The cross-coupled terms (k, d) arise from the fluid’s circumferential velocity component. The rotor whirling motion is referred to as self-excited rotordynamic instability. It has been proven that cross-coupling stiffness effects tend to sustain whirling motion in annular seals at a subsynchronous natural frequency when insufficient damping is present [21].

If the shaft center moves in a circular orbit with radius e, then the rotation displacement vector to the shaft center has coordinates:

\[ y = e \cos (\Omega \, t) \]
\[ z = e \sin (\Omega \, t) \]

The radial force component \( F_r \) and the tangential force component \( F_t \)can be formulated as follows:

\[ \frac{F_r(\Omega)}{e} = \frac{F_y(t = 0)}{e} = -d \, \Omega - K \]
\[ \frac{F_t(\Omega)}{e} = \frac{F_z(t = 0)}{e} = k - D \Omega \]

The direct stiffness coefficient \( K \) and the cross-coupled damping coefficient \( d \) can be obtained employing a linear regression of the calculated radial force for two different values of \( \Omega \) in equation (9). The direct damping coefficient \( D \) and the cross-coupled stiffness coefficient \( k \) can be determined from linear regression of the calculated tangential force for two different values of \( \Omega \) in equation (10).

3 RESULTS AND DISCUSSIONS

3.1 Assessment of the CFD Predictions Accuracy

The developed model has been solved in the given seal
respecting the boundary and operating conditions used by [22 and 23] and summarized in table 1. The static pressure distribution along the seal has been locally computed in the seal mid-cavities and theradial and tangential force components have been also calculated for five whirling ratios (Ω/ω=0, 0.25, 0.5, 0.75, and 1).

Figure 6 shows a comparison between the current CFD predictionsof the axial pressure distribution andprevious results of Kirk and Gao [23] on the same seal using separately bulk Flow and CFX models. This figure shows a good agreement between CFD predictions and calculation results of Kirk and Gao. Additionally, this figure shows that the pressure drops from the inlet pressure at left to the outlet pressure at right. The pressure drop mainly occurs at the tips of teeth and the pressure is almost equal in the same cavity interior.

3.2 Flow Structure Details in the Seal

Figure 9 shows contours of the static pressure in an axial radial plane of the seal. One can easily see that pressure drop starts at the seal inlet and continues at each tooth throttling to rich the outlet pressure at the seal exit. Furthermore, the quasi same color in each cavity interior confirms that the pressure is quasi constant in each cavity interior.

<table>
<thead>
<tr>
<th>TABLE 1</th>
<th>BOUNDARY AND OPERATING CONDITIONS OF THE SEAL</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotating speed, ω</td>
<td>11097 rpm</td>
</tr>
<tr>
<td>Inlet pressure, P_in</td>
<td>3.447 MPa</td>
</tr>
<tr>
<td>Outlet pressure, P_out</td>
<td>1.724 MPa</td>
</tr>
<tr>
<td>Roughness, Ra</td>
<td>1 μm</td>
</tr>
<tr>
<td>Inlet swirl velocity, W_in</td>
<td>0 m/s</td>
</tr>
<tr>
<td>Eccentricity ratio, ε = e/C</td>
<td>10%</td>
</tr>
<tr>
<td>Working fluid</td>
<td>Air</td>
</tr>
</tbody>
</table>

Fig. 6. CFD compared to previous predictions of the pressure distribution in the axial direction of the seal

Fig. 7. Radial force versus whirling ratio

Fig. 8. Compared tangential force versus whirling ratio
3.3 Effects of Inlet Swirl Ratio and Whirling Ratio on Dynamic Coefficients

Figure 12 represents the direct stiffness coefficient versus whirling ratio with the inlet swirl ratio as a parameter. This figure shows that direct stiffness decreases with increasing whirling ratio but it decreases slightly with preswirl ratio increasing.

Figure 13 shows that cross-coupled stiffness coefficient is practically not influenced by whirling ratio variations. Namely, cross stiffness coefficient is whirling frequency-independent. However, it is of interest to note that the cross-coupled stiffness changes from the positive sign to the negative one when the preswirl ratio changes also from positive value (0.25) to negative one (-0.25). Since negative cross stiffness becomes a stabilizing influence, it can be stated that negative inlet swirl is beneficial in dynamic stability point of view for this seal.
Figure 14 shows that direct damping coefficient increases with increasing preswirl ratio but it increases weakly with increasing whirling ratio. Direct damping is a stabilizing influence, so maximum values are desirable for this coefficient. Namely, high preswirl ratios provide high direct damping to keep the seal dynamically more stable.

This ratio provides an assessment for the dynamic stability of the seal. More the whirl frequency ratio is smaller more the seal is dynamically stable. Figure 16 shows that the whirl frequency ratio is quite small for no preswirl and minimal for the negative preswirl case. Therefore, integration of swirl brakes at the seal inlet is more important to improve the dynamic stability of this kind of seals.

Figure 15 shows that cross-coupled damping coefficient increases slightly with increasing preswirl ratio but decreases weakly with increasing whirling ratio. Furthermore, it is proven that this coefficient has only little influence on the rotor stability. Thus, small fluctuations in this coefficient are without big interest.

3.4 Effects of Rotor Speed and Whirling Ratio on Dynamic Coefficients

As shown in the last section (3.3), at the given rotor speed ($\omega = 11097$ rpm) and for different preswirl velocities, the direct stiffness coefficient is sensitive to the whirl frequency variations. However, this ratio has a slight influence on the direct and cross-coupled damping coefficients and only a very weak influence on the cross-coupled stiffness. Namely, except for direct stiffness, direct and cross damping and cross stiffness can be considered whirl frequency-independent in the given predictions range. So, it’s of interest to show the effect of whirl frequency ratio on dynamic coefficients at high rotor speed cases.

Figures 17, 18, 19, and 20 represent at zero preswirl respectively direct stiffness, cross-coupled stiffness, direct damping, and cross-coupled damping coefficients versus the whirling ratio with the rotor speed as a parameter. It can be easily seen that at high rotor speeds ($\omega \geq 15500$ rpm), these dynamic coefficients become more influenced by the whirl frequency and they have to be considered whirl frequency-dependent.

To compare the destabilizing influence of cross-coupled stiffness to the stabilizing influence of direct damping, the whirl frequency ratio $\sigma$ is introduced as follows:

$$\sigma = \frac{k}{D\Omega}$$  \hspace{1cm} (11)
A CFD model has been developed to predict induced fluid forces and dynamic coefficients in labyrinth gas seals. The model solves the general Reynolds Averaged Navier-Stokes equations along with the k-ω SST turbulence model respecting given boundary and operating conditions. The model predictions accuracy has been compared to previous results of Kirk and Gao on the same seal using separately Bulk Flow and CFX methods to calculate pressure distribution and driving force components at zero preswirl. Then, the effects of inlet swirl (negative, zero, and positive), whirl frequency, and high rotor speed have been investigated on the dynamic characteristics of the seal. Through the in-depth research and analysis, some conclusions are summarized as follows:

- Pressure drops from the inlet to the outlet pressures. The
pressure drop mainly occurs at the tips of teeth and the pressure is almost equal in the same cavity interior.

- The high pressure drop occurs in the first cavity where a strong flow jet is generated making the flow more turbulent in this cavity. We note the presence of recirculation zones in the seal cavities. These vortexes act as brakes to stop the axial velocity of the flow through the seal.

- With no preswirl or moderate negative preswirl, the seal is dynamically more stable. Therefore, integration of swirl brakes at the seal inlet is more important to improve the dynamic stability of this kind of seals.

- At the rotor speed ($\omega = 11097$ rpm), direct and cross-coupled damping and cross-coupled stiffness coefficients are very slightly influenced by the whirling ratio variations and can be considered whirling frequency-independent.

- At high rotor speeds ($\omega \geq 15500$ rpm), stiffness and damping dynamic coefficients become more influenced by the whirl frequency and they have to be considered whirling frequency-dependent.

References


