A Study on Rotordynamic Characteristics of Swirl Brakes for Three Types of Seals

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Abstract. In order to understand swirl brakes mechanisms and their influence on rotordynamic characteristics for different types of seals, a three-dimensional flow numerical simulation was presented. Three typical seals including labyrinth seal, fully partitioned damper seal and hole-pattern seal were compared under three inlet conditions of no preswirl, preswirl and preswirl with swirl brakes. FAN boundary condition was used to provide inlet preswirl. A modified identification method of effective damping was proposed. Feasibility of the swirl brakes on improving performance of damper seals was discussed. The results show that the swirl brakes influence the seal stability characteristics with whirl frequency. For the labyrinth seal the swirl brakes reverse the sign of effective damping at low frequency and improve the seal stability performance in a wide frequency range. The swirl brakes also improve the damper seals’ stability performance by increasing the low frequency effective damping and reducing their crossover frequency. Further results indicate the swirl brakes affect the rotational direction of the maximum (minimum) pressure positions and enhance the stability of the seals by reducing tangential force in each cavity.

1. Introduction

Annular seals in turbomachinery are necessary components to restrict leakage flow through rotor-stator clearances between different pressure regions [1]. These components, while crucial for minimizing leakage of process fluid or lubricant, also drive instability in a rotor system [2]. The unfavorable excited force produced in seals has been recognized as a major destabilizing factor [3]. Stable seals are critically required with equipment’s development towards high power. Many new types of seals were proposed by researchers. The honeycomb seal was one of the most representative inventions. The excited force in honeycomb is effectively reduced due to the special hexagon structures. Subsequently hole-pattern seal and pocket damper seal were invented. The three seals are commonly called damper seals which can increase the damping of the inside fluid [4].

Besides damper seals, swirl brakes were the first proposed method for improving the stability of labyrinth seal [5]. Swirl brakes are a series of vanes provided at seal entrance to impede or direct the entrance flow of circumferentially rotating fluid, as shown in figure 1. Practical applications of swirl brakes have been reported, such as high-pressure fuel turbopump seal [6], high pressure oxygen turbopump turbine seal [7] and centrifugal-compressor impeller eye seal [4]. Benckert and Wachter [5] firstly demonstrated that introducing swirl brakes at the seal entrance can greatly reduce the destabilizing force and destabilizing cross-coupled stiffness. Sivo [8] investigated the effects of swirl brakes on shrouded centrifugal pumps and found installing swirl brakes at the leakage entrance...
significantly reduced the destabilizing tangential force. Kwanka [9] experimentally showed that a small amount of swirl brakes (8 swirl brakes) effectively reduced the induced force to a quarter of the initial value. Kimura [10] carried out a numerical simulation of leakage flow in a centrifugal impeller. The swirl brakes suppressed the swirl of the leakage flow which caused rotor instability. The axial thrust on the impeller can be controlled by adjusting the shape of swirl brakes. Nielsen [11] compared aerodynamic swirl brakes and nonaerodynamic swirl brakes by numerical simulation. The nonaerodynamic swirl brakes had better performance than the aerodynamic one because of strong separation vortices within the swirl brakes. Brown [12] tested a hole-pattern seal with negative preswirl. High negative inlet preswirl produced negative cross-coupled stiffness over a frequency range brand and improved the stability of the seal. Childs [13] investigated negative-swirl brakes for labyrinth seal. The negative-swirl brake changed the sign of cross stiffness and produced a marked increase in effective damping.

Although damper seals are considered as the alternative to traditional labyrinth seal, researchers have found damper seals have the shortcoming of potential instability at low frequency [14]. Studies on optimizing damper seals performance have been carried out for ten years. Smalley [15] compared the stability between a straight honeycomb seal and a diverging taper honeycomb seal. The diverging taper improved seal effective damping over a wide frequency range. Childs [16] investigated rotordynamic characteristics of hole-pattern seals with varying hole depth axially. The various depth patterns significantly improved seal performance in terms of both increasing the seal’s effective damping and reducing its crossover frequency. Vannarsdall [17] tested a hole-pattern seal with large diameter holes and found the seal had high and positive effective damping without crossover frequency for all tested frequencies.

Swirl brakes were originally proposed for labyrinth seals. To author’s knowledge, applying swirl brakes on damper seals has not been reported before. In order to further improve the stability of damper seals, the paper studied the stability of damper seals by considering swirl breaks installed at seal entrance. A three-dimensional flow numerical analysis was presented to compare the performance of three typical seals including labyrinth seal, fully partitioned damper seal and hole-pattern seal. The three seals were compared under three inlet conditions of no preswirl (NPS), preswirl (PS) and preswirl with swirl brakes (PSSW). The purpose is to understand the swirl brakes rotordynamic mechanisms and to explore the feasibility of the swirl brakes on improving performance of damper seals.

2. Methods

2.1. Numerical Calculation Methods

Many successful applications of fluid software for solving seal rotordynamic coefficients have been reported [18-21]. In the paper, fluid software FLUENT 14.0 is used to solve the full eccentric three
dimensional seal model. Rotating reference frame method [18] is employed to simulate the whirl movement of rotor. The entire fluid zone rotates with the reference frame with speed $\Omega$. The rotor walls rotate with speed $\omega - \Omega$ and the stator walls rotate with speed $-\Omega$. Standard k-\(\varepsilon\) turbulence model and standard wall function are selected. The working fluid is treated as ideal gas. Convergence residuals are set to $10^{-5}$.

The rotor is placed at the positive Y axis direction for eccentric and rotated in a clockwise direction, as shown in figure 2. In this case, the sign of the tangential force $F_t$ primarily determines the seal stability. If the sign is positive, the tangential force points to positive $X$ direction and accelerates the whirl movement of the rotor. The seal system is unstable under the situation.

Compared with another fluid software CFX, FLUENT can directly specify inlet preswirl with FAN boundary condition. In CFX, specifying inlet preswirl usually needs a special approach. Pugachev [21] proposed that a velocity boundary condition set at seal inlet and a pressure boundary condition set at a bypass. This approach sometimes could lead to improper flow condition when the area or velocity of the boundary condition is set to an unreasonable parameter. In FLUENT the FAN boundary condition is an interior face representing a virtual fan with known characteristics. The FAN boundary condition is placed at 1/3 length of seal entrance away from inlet for current analysis.

2.2. Identification methods of effective damping

Rotor of rotating machinery usually works in synchronous mode or asynchronous mode. Rotor self-excited vibration often occurs in asynchronous mode [4]. In order to comprehensively evaluate the stability of seals, the effective damping $C_{\text{eff}}$ is used as evaluation criterion. It is defined as [17]

$$C_{\text{eff}} = C - k / \Omega$$  \hspace{1cm} (1)

The effective damping is a function of rotor whirl frequency $\Omega$ and thus it can evaluate the seal stability with different $\Omega$.

The relation between tangential force and rotordynamic coefficients, $C$ and $k$, is expressed as [18]

$$F_t = F_{t, e} = k \delta - \Omega C \delta$$  \hspace{1cm} (2)

To obtain the effective damping, the rotordynamic coefficients, $C$ and $k$, usually need to be obtained firstly. In simulation, the identification method of the rotordynamic coefficients is to solve Eq. (1) with different $\Omega$. It implies the simulation should be carried out at least two times. However, the rotordynamic coefficients are generally frequency dependent, especially for damper seals [22]. This method may introduce error. For avoiding solving rotordynamic coefficients, the effective damping is directly obtained by substituting $C$ and $k$ from Eq. (2) into Eq. (1), as follows:

$$C_{\text{eff}} = C - k / \Omega = -F_t / (\Omega \cdot \delta)$$  \hspace{1cm} (3)

All the seals are evaluated with the effective damping for current analysis.

2.3. Seals and swirl brakes parameters

Labyrinth seal and fully partitioned damper seal are both chosen from literature [14]. Hole-pattern seal (2668 holes) is chosen from literature [23]. Figure 3 and 4 show parameters of swirl brakes and meshes of the three seals. 72 swirl brakes are uniformly distributed along the circumference at seal entrance. Seal body mesh and seal entrance mesh are separately generated with hexahedral grids. Then the two independent meshes are connected with the help of the interface (see figure 4). The radial clearance at tooth tip is divided into 10 layer nodes for all seals. The circumferential direction is divided into 350 layer nodes for the labyrinth seal and the fully partitioned damper seal. For the hole-pattern seal, the mesh is generated with 1 mm node distance and its mesh number is approximate 7 million which is 8 times of the other two seals. Compared with Hirano's mesh density [18], the present mesh density increases circumferential number and reduces elements' aspect ratio.
Figure 3. Model and parameters of swirl brakes for labyrinth seal.

Figure 4. Meshes of the three seals.

The labyrinth seal and fully partitioned damper seal are calculated with 6.9 bar total pressure inlet, 7000 rpm rotational speed and 60 m/s preswirl. The hole-pattern seal is calculated with 7 MPa total pressure inlet, 3.15 MPa pressure outlet, 20200 rpm rotational speed and 72.5 m/s preswirl.

3. Numerical method validation
Figure 5 shows the comparison of the effective damping between present results and literature data for the three seals. It can be seen that the effective damping is generally well predicted by the present numerical
method for all three seals. The error mainly comes from the different eccentric ratio of seals. The eccentric ratio in experiment is not a constant but a changing value with time, while the eccentric ratio in simulation always keeps a constant. The eccentric ratio affects the amplitude of the tangential force and hence introduces error into the results.

Figure 5. Comparison of the effective damping between present results and literature data for the three seals.

4. Results and discussion

Figure 6 shows the effective damping of the three seals. Each graph displays results with NPS, PS and PSSW. Obviously, the highest effective damping of the three seals all appears at NPS condition. That is to say, NPS is relatively stable compared to PS and PSSW. However, NPS doesn’t exist in reality because partial admission or rotating blades can generate inlet preswirl velocity at seal entrance in rotating machineries.

In PS condition the effective damping becomes high as whirl frequency increases for all seals. It implies the rotor tangential force becomes low and seal system becomes stable. The working fluid undergoes compression and expansion when entering the minimum radial clearance (see figure 2), which causes non-uniform pressure distribution in circumferential direction (see figure 10). If considering rotor whirl movement, the minimum clearance position changes with time in circumferential direction, and the compression and expansion of fluid are delayed. The non-uniform pressure distribution is reduced by the delayed effect and thus the effective damping is increased.

The labyrinth seal possesses negative effective damping at low frequency under PS condition and positive effective damping throughout the entire calculated frequency range under NPS and PSSW condition. The negative effective damping is unstable for seal system. The preswirl velocity is the main reason of causing flow excited force. The swirl brakes reverse the sign of effective damping at low frequency and an improved stability in a wide frequency range is obtained. Installing swirl brakes at labyrinth seal entrance is encouraged. Unlike the labyrinth seal, the damper seals yield negative effective damping at low frequency and transform to positive effective damping as frequency increases under all three inlet conditions. The swirl brakes are also helpful for improving the damper
seals’ stability performance by increasing the low frequency effective damping and reducing their crossover frequency. Installing swirl brakes at damper seals entrance is also encouraged.

Figure 6. Comparison of the effective damping for the three seals with NPS, PS and PSSW.

Figure 7. Comparison of the effective damping for the labyrinth seal with PSSW and the fully partitioned damper seal with PS.
**Figure 8.** Velocity vectors in the center plane of the labyrinth seal with NPS, PS and PSSW.

- Fully partitioned damper seal
- Hole-pattern seal

**Figure 9.** Pressure distribution and velocity vectors in the center plane of damper seals with PSSW for whirl frequency 50 Hz.

- NPS condition
It is noted that the effective damping of the labyrinth seal is positive over the whole frequency, while that of the damper seals is negative at low frequency under NPS and PSSW condition. Different from labyrinth seal, damper seals contain baffles in seal cavity. The fluid cannot rotate through the entire circular seal cavity smoothly. At low frequency their pressure distribution inside seal cavity may be more sensitive to rotor rotational speed which causes the negative effective damping (see results of literature [14]).

The effects of swirl brakes on the three seals are not strictly consistent, but the benefits can be regarded as improving the seal stability between NPS condition and PS condition. The function of swirl brakes is to decrease the flow circumferential velocity at seal entrance. Therefore, swirl brakes cannot bring more stability improvement than seal with NPS.

Figure 7 shows the comparison of the effective damping for the labyrinth seal with PSSW and the fully partitioned damper seal with PS. The labyrinth seal with swirl brakes is more stable than the damper seal at low frequency; however, its mid-high frequency stability performance is not desirable, which is at least 10 times lower than that of the damper seal. Optimizing the stability of damper seal is significant. Similar results were described by literature [22].

Figure 8 shows the velocity vectors in the center plane of the labyrinth seal with NPS, PS and PSSW. In NPS condition, the circumferential velocity is produced by the rotor rotational movement. The velocity value is weak and decreases with radial direction rapidly. The FAN boundary condition successfully provides stronger circumferential velocity at seal entrance for PS and PSSW. The circumferential velocity is approximately linearly distributed in radial direction. Compared to the PS, the swirl brakes partly reduce the inlet circumferential velocity by their flow guiding function. Similarly, the swirl brakes have the same effects on the fully partitioned damper seal and hole-pattern seal, as shown in figure 9.

To explain how the swirl brakes suppress the spiral flow of fluid at low frequency, figure 10 shows the pressure distributions on the rotor surface at 1st, 7th and 13th cavity center planes of the labyrinth seal for whirl frequency 50 Hz. It is noted that the displayed pressure is acting on rotor surface and the displayed surface thickness is amplified for facilitate observation. Maximum and minimum pressure always exists in

**Figure 10.** Pressure distributions on the rotor surface at 1st, 7th and 13th cavity center planes of the labyrinth seal for whirl frequency 50 Hz.
The tangential force is negative and points to the negative $\theta$ to 116.9° and to 160.2°. The difference of the rotational direction is consistent with the sign of effective damping. The effective damping of the seal with NPS and PSSW at 50 Hz is positive (see figure 6(a)). It means that the tangential force is negative and points to the negative $X$ direction (see figure 2). Therefore, most of the maximum pressure positions should be distributed between 180° and 360°, which are shown in figure 10(a) and (c). Correspondingly, most of maximum pressure positions of the seal with PS are distributed between 0 and 180° as shown in figure 10(b). It can be seen that the swirl brakes suppress the spiral flow of fluid by reversing the rotational direction of extreme pressure positions at low frequency.

Table 1 lists the tangential force of the labyrinth seal for whirl frequency 50 Hz. The tangential force decreases along the main flow direction for the seal with PS. The maximum tangential force appears at the first cavity and accounts for 20.2% of the total tangential force. This is because the first seal cavity possesses the strongest circumferential velocity. The tangential force of the seal with NPS and PSSW is negative and uniformly distributed in each cavity. The swirl brakes drop the tangential force in each cavity to enhance the stability of the seals.

| Seal with NPS | Cavity 1 | Cavity 2 | Cavity 3 | Cavity 4 | Cavity 5 | Cavity 6 | Cavity 7 |
|---------------|---------|---------|---------|---------|---------|---------|---------|
| $F_t$ (N)     | -0.70   | -0.62   | -0.72   | -0.77   | -0.77   | -0.72   | -0.66   |
| Percent       | 8.8%    | 7.9%    | 9.1%    | 9.8%    | 9.8%    | 9.1%    | 8.3%    |
| Seal with PS  | $F_t$ (N) | 6.41    | 4.25    | 3.52    | 3.01    | 2.63    | 2.30    | 2.03    |
| Percent       | 20.2%   | 13.4%   | 11.1%   | 9.5%    | 8.3%    | 7.3%    | 6.7%    |
| Seal with PSSW | $F_t$ (N) | -0.15   | -0.14   | -0.16   | -0.16   | -0.15   | -0.16   | -0.15   |
| Percent       | 9.6%    | 9.0%    | 10.3%   | 10.3%   | 9.6%    | 10.3%   | 9.6%    |

| Seal with NPS | Cavity 8 | Cavity 9 | Cavity 10 | Cavity 11 | Cavity 12 | Cavity 13 | Total |
|---------------|---------|---------|-----------|-----------|-----------|-----------|-------|
| $F_t$ (N)     | -0.60   | -0.53   | -0.46     | -0.41     | -0.39     | -0.39     | -7.88 |
| Percent       | 7.6%    | 6.7%    | 5.8%      | 5.2%      | 5.0%      | 5.0%      | 1     |
| Seal with PS  | $F_t$ (N) | 1.78    | 1.56     | 1.35      | 1.14      | 0.95      | 0.77   | 31.72 |
| Percent       | 5.6%    | 4.9%    | 4.3%      | 3.6%      | 3.0%      | 2.4%      | 1     |
| Seal with PSSW | $F_t$ (N) | -0.11   | -0.10   | -0.09     | -0.08     | -0.08     | -0.03   | -1.56 |
| Percent       | 7.1%    | 6.4%    | 5.8%      | 5.1%      | 5.1%      | 1.9%      | 1     |

5. Conclusions
The paper investigated the effects of swirl brakes on rotordynamic performance for three types of seals. Feasibility of swirl brakes on improving performance of damper seals was focused. Evaluation of effective damping and development of pressure distribution were discussed.

The swirl brakes suppress the spiral flow of fluid and enhance the stability of the three typical seals. For labyrinth seal the swirl brakes reverse the sign of effective damping at low frequency and improve seal stability performance in a wide frequency range. Similarly, the swirl brakes also improve the stability performance of fully partitioned damper seal and hole-pattern seal by increasing the low frequency effective damping and reducing their crossover frequency. Installing swirl brakes at damper seals entrance is encouraged.

Nomenclature
- $C$: damping coefficient
- $C_{eff}$: effective damping
- $h_1$: minimum radial clearance
- $h_2$: maximum radial clearance
- $F_r$: radial force
- $F_t$: tangential force
- $F_x$: force in $X$ direction
$F_y$: force in $Y$ direction
$k$: stiffness coefficient
$\delta$: eccentric
$\theta$: circumferential coordinate
$\omega$: rotor rotational speed
$\Omega$: rotor whirl frequency

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