Numerical and experimental analysis of the rotor eccentric effect on the labyrinth seal

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Abstract. Labyrinth seals are widely used in turbomachines. In this paper, a transit simulation method is introduced to model the relationship between the journal offset in Labyrinth seals and the system stability based on Fluent UDF. A special dynamic mesh method has been adopted to ensure fine mesh quality during the arbitrary motion of the journal. In each time step, the transient fluid seal force on the journal surface is obtained by the solving Navier-Stokes equations. Then the displacement of the journal in the next step can be numerically solved by the FEM model of the rotor system. For every transient time step, the computed fluid seal force and the rotor motion displacement were coupled by data exchange. The experiment is carried out in a steam turbine with a position adjustable labyrinth seal. The experimental results show that the amplitude of the vibration decreases after the change of the relative position between the journal and the labyrinth seals and the numerical results show that the locus of the journal under different labyrinth seal offsets has some half-frequency components. Both the simulation and the experiment validates that the relative position between the journal and the labyrinth seals can greatly affect the stability in the coupled rotor-seal system.

1. Introduction
Labyrinth seals, which are widely used in turbomachines to reduce leakage, have close relationship with the stability of the rotor system due to self-excited vibrations induced by the seal fluid force under certain conditions. Therefore, it is necessary to evaluate the forces in labyrinth seals in a precise rotor dynamic analysis. A linear model is first introduced in reference [1-3] to illustrate the relationship between the motion of the journal and the seal fluid force. The general form can be seen in Eq(1). If the journal moves in a small area, the seal fluid force is proportional to the displacement as well as the velocity of the rotor and adequately accurate solution can be got under the condition. However, if the vibration amplitude is large, this linearized equation of motion will fail to give reliable results.

\[
\begin{bmatrix}
-F_x \\
-F_y \\
\end{bmatrix} = \begin{bmatrix}
K & k \\
-k & K \\
\end{bmatrix} \begin{bmatrix}
X \\
Y \\
\end{bmatrix} + \begin{bmatrix}
C & c \\
-c & C \\
\end{bmatrix} \begin{bmatrix}
\dot{X} \\
\dot{Y} \\
\end{bmatrix}
\]

(1)

A nonlinear dynamic model [4-6] is then introduced as the Eq(2) shows. The equation couples the fluid film radial stiffness, damping and inertia effects, and has been applied in many calculations [7-9]. However, the accurate solution of these equation requires reliable coefficients. Although the rotordynamic coefficients of labyrinth seals have been calculated or tested by many researchers [10-12], empirical corrections are still needed to allow calculations matching the experimental results.
because the detailed structure can not be included in the model. This greatly limits the application of these models.

With the development of the computer technology, more and more researchers chose to use CFD to calculate the seal fluid force and leakage in Labyrinth seals [13-16]. As more geometric details can be considered in CFD, more reliable results can be got.

In this paper, in order to simulate the transit flow field in a labyrinth seal, the commercial CFD software Fluent and a dynamic mesh method similar in journal bearing calculation [17] are adopted. The method can ensure fine mesh quality during the arbitrary motion of the journal so that the transit analysis of the coupled rotor-seal system with an offset labyrinth seal becomes feasible.

2. Method

In order to simulate the effect of the journal offset in the labyrinth seal, the coupled rotor-seal system is built numerically. The system involves two parts. The CFD model is used to calculate the seal fluid force on the journal. The single disk FEM model is used to calculate the transit displacement of the rotor.

2.1. CFD model

The Figure 1 shows the geometric model and the mesh model of the labyrinth seal, they are both built in the GAMBIT. From the mesh section of the geometric, it is obvious that the mesh can be divided into two parts, the chamber part and the clearance part. In the simulation, the clearance part is used as the dynamic mesh zone. This means the UDF macro DEFINE_GRID_MOTION is applied in this area to ensure fine mesh quality during the arbitrary motion of the journal. The detailed dynamic method description can be seen in reference[17] and is not given here for brevity. The number of elements corresponds to the Figure 1 is listed in the Table1.

![Figure 1. The Geometric dimensions and the mesh of the labyrinth seal](image)

| Label | Number of elements |
|-------|--------------------|
| n1    | 5                  |
| n2    | 10                 |
| n3    | 10                 |
| n4    | 5                  |
| n5    | 500                |

The $k$-$\varepsilon$ model is selected to simulate the turbulent flow in the chamber. The inlet total pressure is specified to be 1MPa and the outlet static pressure is specified to be 101325Pa. The back flow turbulent intensity and viscosity ratio are specified to be 4% and 4 separately in both the inlet and
outlet. For a transient simulation, as the center of the journal changes over time, the UDF macro DEFINE_PROFILE is applied to specify the rotational speed of the journal according to the reference [17].

The SIMPLEC method is selected to solve the discrete equations. The first order implicit format is chosen as the temporal discretization format. The detailed spatial discretization formats are listed in the Table 2. The time step is 0.0001s and the iterations in each step is 20.

### Table 2. The spatial discretization format

| Item                           | Spatial discretization format          |
|--------------------------------|---------------------------------------|
| Gradient                      | Least squares cell based              |
| Pressure                      | Standard                              |
| Density                       | Second order upwind                   |
| Momentum                      | Second order upwind                   |
| Turbulent kinetic energy      | First order upwind                    |
| Turbulent dissipation rate    | First order upwind                    |
| Energy                        | Second order upwind                   |

2.2. Coupled rotor-seal system

The Figure 2 shows the single disk model, the length of the axis is 0.63m and the mass of the axis is ignored. The radius of the axis is 0.03m. The weight of the disk is 0.7351kg and the eccentric distance on the disk is 1.2mm. The bearing clearance is 3e-4m and the bearing length is 0.3m.

![Figure 2. The rotor model description](image)

The equation of the rotor system can be seen in Eq(3).

\[
\begin{bmatrix}
M & 0 \\
0 & M
\end{bmatrix} \begin{bmatrix}
\dot{X} \\
\dot{Y}
\end{bmatrix} + \begin{bmatrix}
C & 0 \\
0 & K
\end{bmatrix} \begin{bmatrix}
X \\
Y
\end{bmatrix} + \begin{bmatrix}
\dot{F}\_x \\
\dot{F}\_y
\end{bmatrix} + \begin{bmatrix}
f\_x \\
f\_y
\end{bmatrix} + \begin{bmatrix}
Q\_x \\
Q\_y
\end{bmatrix} = \begin{bmatrix}
F\_x \\
F\_y
\end{bmatrix}
\]

(3)

Where \( M \) is a mass matrix; \( C \) and \( K \) are the damping matrix and stiffness matrix respectively; \( F \) is the oil film force calculated by the short bearing theory; \( f \) is the seal fluid force calculated by the CFD model in Fluent; \( Q \) is the unbalanced force.

The whole coupled rotor-seal system calculation procedure can be seen in the Figure 3. In order to consider the journal offset in the seal, before the transient calculation, a static calculation with the given offset should be done as the initialization for the following simulation. The detailed description of the relative position between the journal and the seal can be seen in the Figure 2. Then, like the flow diagram shows, in every time step, the UDF macro first calculate the seal fluid force on the journal...
surface to write into the .txt files and call the Matlab code to process the generated files. The Matlab program reads in the seal fluid force and begins to solve the Eq(3). After the Newmark integration, the Matlab code writes the displacement for the temporary time step into .txt files. Then, when the UDF code judge the Matlab program has been successfully executed, it reads in the disk displacement to update the mesh in DEFINE_GRID_MOTION macro and also update the journal surface velocity in DEFINE_PROFILE macro. After all of these have been done, the Fluent records all the data you want and starts to iterate. When the iterations of the temporary time step is over, if the iteration demand is not fulfilled, the above procedure will be repeated, else the whole calculation will be ended.

![Figure 3. The flow diagram of the coupled rotor-seal system](image)

### 3. Calculation results

The whole calculation results can be divided into two parts. The first part focuses on the relationship between the rotor-seal system stability and the journal offset. The second part reveals the transit characteristics of the seal fluid field under different axis orbits.

#### 3.1. Stability description

The Figure 4 shows the rotor axis orbits under different journal offset, obviously, with the change of the offset, the axis orbits under different offsets are quite different. When the offset is 0μm and 100μm, the axis orbit is a circle, its amplitude is influenced by the unbalanced force. Ordinarily, this kind of axis orbit is considered stable. However, when the offset is -80μm and -100μm, the half-frequency vibration can be seen from the axis orbits. This kind of axis orbit is viewed unstable. What’s more, when the offset shifts from -80μm to -100μm, the half-frequency vibration seems more obvious. So it is easy to draw the conclusion that the relative position between the journal and the labyrinth seals can greatly affect the stability in the coupled rotor-seal system. Often, especially the huge turbinemachine rotor systems can easily become unstable due to the bad uniformity of seal fluid force in the labyrinth seals. So if the seals relative position with the rotor can be adjusted properly, the whole coupled rotor-seal system stability may be improved.
Figure 4. The axis orbits under different journal offset

(a) Offset=100μm  (b) Offset=0μm

(c) Offset=-80μm  (d) Offset=-100μm

The Figure 5 shows the temporal change of the seal fluid force and the leakage under the offset 100μm and -100μm, which represents two distinguished state of the rotor system as described above. It is obvious to see that the seal fluid force under the offset of 100μm changes with time like the sine curve. While under the offset of -100μm, the sine curve of the seal fluid force are coupled with some half-frequency components. The similar change can also be seen in the leakage transit curve. This means the relative position between the journal and the seal can cause different seal fluid force components, which greatly affect the system stability and further affect the leakage. So the leakage changes in a similar way.

3.2. Transit characteristics
The Figure 6 shows the circumferential distribution of the total pressure at different position with different offsets. The distribution curves are similar to the sine curve at fixed positions marked in the small picture of the axis orbits. But the amplitude and the phase of the curves varies with different positions. This may have close relationship with the relative position between the journal and the labyrinth seal. When the journal offset is -100μm, the journal orbit has some half-frequency components as the figure shows. For the four points on the outer circle of the journal bearing locus, point C has the maximum peak-to-peak total pressure value while point A has the minimum value. This is probably because the point A is closer to the center of the seal, thus bringing a more even circumferential distribution of the total pressure. While for the point C, the clearance between the journal and the seal is the smallest, which makes the total pressure differs along the circumferential direction of the journal. Similarly, for the four points on the inner circle of the journal bearing locus, point B’ is closer to the seal center and has a higher peak-to-peak total pressure value as well as a smaller clearance than point C’. When the journal offset is 100μm, the shape of the journal orbit is ellipse. With the change of the offset, point B’’ has the smallest clearance and point D’’ is closest to the center of the seal. As a result, the curve for the point D’’ is more flat.

![Figure 6. The circumferential distribution of the total pressure at different position](image)

The axial distribution of the total pressure at different position can be seen in Figure 7. From the curves, the pressure of the values decreased and are divided into several segments due to the effect of the labyrinth seal. Similar to the circumferential distribution of the total pressure, the total pressure distribution is different with the change of the journal position. For example, when the journal offset is 100μm, for the point B, the clearance between the journal and the seal is small, thus increasing the pressure drop of the seal.
Figure 7. The axial distribution of the total pressure at different position

From the above discussion, the pressure distribution changes regularly with the relative position between the journal and the seal. As a result, the nonlinear gas sealing force acted on the journal surface makes the journal move in a nonlinear orbit. In return, the nonlinear orbit will also result in the nonlinear change of the gas sealing force. The coupling effect will finally determine the stability of the system.

4. Experimental results

The Figure 8 shows the experimental vibration of a coupled rotor-seal system for the steam turbine. In the experiment, the labyrinth seal is designed to be allowed to adjust to a proper position. From the figure, when the time is around 70 min, the amplitude of the vibration decreases due to the adjustment of the labyrinth seal, which changes the eccentricity of the journal.

Figure 8. The experimental transit vibration amplitude
However, when the labyrinth seal position is adjusted to the original position at about 100 min, the vibration of the journal become severe. Similar phenomenon has been repeated at the time 120 min and 250 min. The experimental results have proved that the coupled rotor-seal system stability has close relationship with the relative position between the journal and the labyrinth seal and this conclusion agrees with the calculated results.

5. Conclusions
In this paper, a transit simulation method based on Fluent is introduced to model the coupled rotor-seal system. This method which involves a dynamic mesh method similar in journal bearing calculation can ensure fine mesh quality during the arbitrary motion of the journal. Experiment has been done to validate the conclusion. Some conclusions can be summed as followed:
1. The relative position between the journal and the labyrinth seal has close relationship with the gas sealing force, the leakage and the stability of the coupled rotor-seal system.
2. The circumferential distribution of the total pressure is similar to the sine curve. The amplitude of the curve increases with the eccentricity of the journal.

References
[1] Guinzburg, A., et al., The effect of inlet swirl on the rotordynamic shroud forces in a centrifugal pump. Journal of engineering for gas turbines and power, 1993. 115(2): p. 287-293.
[2] Kirk, R.G., Evaluation of Aerodynamic Instability Mechanisms for Centrifugal Compressors—Part II: Advanced Analysis. Journal of Vibration and Acoustics, 1988. 110(2): p. 207-212.
[3] Picardo, A. and D.W. Childs, Rotordynamic coefficients for a tooth-on-stator labyrinth seal at 70 bar supply pressures: measurements versus theory and comparisons to a hole-pattern stator seal. Journal of engineering for gas turbines and power, 2005. 127(4): p. 843-855.
[4] Muszynska, A., Improvements in lightly loaded rotor/bearing and rotor/seal models. Journal of Vibration and Acoustics, 1988. 110(2): p. 129-136.
[5] Muszynska, A. and D.E. Bently, Frequency-swept rotating input perturbation techniques and identification of the fluid force models in rotor/bearing/seal systems and fluid handling machines. Journal of Sound and Vibration, 1990. 143(1): p. 103-124.
[6] Muszynska, A., Whirl and whip—rotor/bearing stability problems. Journal of Sound and vibration, 1986. 110(3): p. 443-462.
[7] Li, S., Q. Xu and X. Zhang, Nonlinear dynamic behaviors of a rotor-labyrinth seal system. Nonlinear Dynamics, 2007. 47(4): p. 321-329.
[8] Shen, X., et al., Experimental and numerical analysis of nonlinear dynamics of rotor—bearing—seal system. Nonlinear Dynamics, 2008. 53(1-2): p. 31-44.
[9] Wang, W.Z., et al., A nonlinear model of flow-structure interaction between steam leakage through labyrinth seal and the whirling rotor. Journal of mechanical science and technology, 2009. 23(12): p. 3302-3315.
[10] Childs, D.W. and J.K. Scharrer, Theory versus experiment for the rotordynamic coefficient of labyrinth gas seals: Part II—A comparison to experiment. Journal of Vibration and Acoustics, 1988. 110(3): p. 281-287.
[11] Iwatsubo, T., Evaluation of instability forces of labyrinth seals in turbines or compressors. NASA. Lewis Res. Center Rotordyn. Instability Probl. in High-Performance Turbomachinery p 139-167(SEE N 80-29706 20-37), 1980.
[12] Scharrer, J.K., Theory Versus Experiment for the Rotordynamic Coefficients of Labyrinth Gas Seals: Part I—A Two Control Volume Model. Journal of Vibration and Acoustics, 1988. 110(3): p. 270-280.
[13] Arghir, M. and J. Fre`ne, Rotordynamic Coefficients of Circumferentially-Grooved Liquid Seals Using the Averaged Navier-Stokes Equations. Journal of tribology, 1997. 119(3): p. 556-567.
[14] Arghir, M. and J. Fre`ne, Rotordynamic Coefficients of Circumferentially-Grooved Liquid Seals Using the Averaged Navier-Stokes Equations. Journal of tribology, 1997. 119(3): p. 556-567.
[15] Hirano, T., Z. Guo and R.G. Kirk, Application of computational fluid dynamics analysis for rotating machinery—part ii: labyrinth seal analysis. Journal of engineering for gas turbines and power, 2005. 127(4): p. 820-826.

[16] Ishii, E., et al., Prediction of Rotordynamic Forces in a Labyrinth Seal based on Three-Dimensional Turbulent Flow Computation. JSME International Journal Series C Mechanical Systems, Machine Elements and Manufacturing, 1997. 40(4): p. 743-748.

[17] Li, Q., et al., A new method for studying the 3D transient flow of misaligned journal bearings in flexible rotor-bearing systems. Journal of Zhejiang University SCIENCE A, 2012. 13(4): p. 293-310.