Experimental Research and Analysis of Rotordynamic Characteristics for a New Kind of Tilting-pad Gas Seal

Hao Cao¹, Xiaoyu Cao²*, Fei Chen¹, Ming Li¹, Bolin Zhang¹, Jilong Wei¹

¹Hunan Province Key Laboratory of High Efficient & Clean Thermal Power Technology, State Grid Hunan Electric Power Corporation Research Institute, Changsha, 410007, China
²The Library of Hunan Agricultural University, Changsha, 410128, China

*Corresponding author e-mail: caohao45@126.com

Abstract. This paper presents a new kind of tilting-pad gas seal. This design is introduced to reduce the tangential seal force and to improve the stability of rotor system finally. A seal test rig is set up. The paper compares the leakage between tilting-pad seal and fixed pad seal. The result shows that the leakage ratio of the tilting-pad seal is close to the leakage ratio of the fixed pad seal. The work done by seal force on the cylinder system is calculated as an index of comparison between these two seals. Result shows that the work done by the fixed pad seal is greater than the work done by the tilting-pad seal. Moreover, system damping factor is used to compare the stabilities of these two seals. The impact tests on the cylinder system are done under different conditions. The system damping factors are calculated from the damped waves of system vibration. Test results show that the damping factor of the tilting pad seal is higher than that of the fixed pad seal in both the vertical and the horizontal directions.

1. Introduction

Annular seals are widely used in turbomachines such as compressors, pumps and gas-turbines to limit the leakage of fluid from a high-pressure region to a low-pressure region. It has also been confirmed as one of the major source of destabilizing forces resulting in rotor dynamic instability problems [1]. Many researches show that high pressure gas flow in the small clearances produces tangential force on the rotor. In order to enhance the energy transformation efficiency and reduce the pollution, steam turbines with supercritical and ultra-supercritical steam parameters become more and more popular. As the increment of fluid pressure, the force induced by gas flow in seals becomes large. How to reduce the tangential force of seals become more and more important.

People have done a lot of work to reduce the tangential force of conventional labyrinths in the past 20 years. It has been proved that spiral flow of fluid along seals is the main cause of tangential force. Thus, many research focused on the new kind of seals that can cut down the spiral flow. Honeycomb seal, pocket damper seal, helically grooved seal, brush seal and so on were put forward [2-3].

This paper presents a new way to reduce the tangential force. The traditional labyrinth seals with fixed pad are improved to the seal with tilting pad. The tilting pad idea was first used in journal bearing to solve the self-excited oil whirl and oil whip problems. Lund noted that the force acting on
each tilting-pad point to the pad pivot if the effects of pad inertia and pivot friction are neglected [4]. Memmott firstly proposed the concept of tilt-pad oil seal. The tilting pad was applied into the outer oil-film rings of the casing end seals of centrifugal compressor, and it effectively eliminated and prevented the detrimental bisynchronous vibration [5].

In this paper, a new design of gas seal named tilting-pad gas seal is proposed. The traditional fixed seal is divided into two parts: tilting seal pad and pocket type seal base. The tilting pad can swing automatically on circumferential direction. The leakage and stability characteristics are compared between the fixed pad seal and the tilting pad seal.

2. Design of tilting-pad seal

The test tilting-pad seal is shown in Figure 1. It is comprised of a seal pad and a pocket type seal base. These two parts are connected with a pivot. The seal pad can swing around the pivot. In the back of seal base, there are two bolts installed to limit the swing angle of seal pad. As shown in Figure 2, h is the gap between seal pad and rotor surface, and g is the gap of two adjacent seal pads. The ratio of g/h is less than 0.2. The gap between two adjacent seal pads is far smaller than the gap between seal and rotor surface. If the swing angle is set to zero, the tilting-pad seal is turned into traditional fixed pad labyrinth seal. Thus, the two kinds of seal have the same dimensions in this test, the specific design parameters are shown in table 1.

![Figure 1. The tilting-pad seal photo](image1)

![Figure 2. The tilting-pad seal schematic diagram](image2)

| Table 1. The designed tilting-pad seal parameters |
|-----------------------------------------------|
| Pad number                                    | 6 |
| Ring number                                   | 3 each side |
| Pad arc angle                                 | 59.5 deg |
| Pad axial length                              | 30.00 mm |
| Seal axial length                             | 31.00 ±0.01 mm |
| Axial clearance between pad and seal          | 0.05 mm |
| Rotor diameter                                | 179.00±0.01 mm |
| Radial clearance between rotor and seal       | 0.50±0.01 mm |
| Mass of cylinder                              | 53.4 kg |
3. Test rig description

![Schematic Diagram of the Seal Test Rig](image)

The tilting-pad seal test rig is illustrated in Figure 3. It contains the following major components: A base, two supporting oil bearings, a slender steel shaft, a pressure cylinder holding the seals, two disks, spring system supporting the test section, gearbox and drive motor. The middle of the shaft holds a steel sleeve coupled with labyrinth seals and with a diameter of 180 mm. The rotor is driven by a 15 KW variable-speed electric motor through a 4:1 ratio speed-increasing gearbox. Six rings of tilting-pad seals are fixed on the interior wall of the cylinder. The pressurized air is pumped into the cylinder in the middle and exhausts from two sides of cylinder. This arrangement can eliminate axial force. In order to enlarge the influence of seal force, test is done near the resonance region of the cylinder. The cylinder is hung by springs in the vertical and the horizontal directions. The resonance frequency of cylinder can be regulated by changing the stiffness of these springs.

Two sets of eddy current sensors, 90 deg. apart, are secured through the wall of the cylinder and face the outer diameter of the steel sleeve from horizontal and vertical directions. The sensors measure the relative displacements between the rotor and cylinder along two orthogonal directions. 4 magnetoelectric velocity transducers are located on the two ends of the cylinder to monitor the absolute vibrations from horizontal and vertical directions on each end. One eddy sensor is fixed on the bearing box to work as a key phasor transducer. The phases of all vibration signals are based on this key phasor signal. All measurements are conducted at room temperature (28°C). Table.2 details the operating conditions for measurements.

| Table 2. Operating conditions |
|-------------------------------|
| Fluid | compressed air |
| Pressure downstream(discharge) | 101 kpa |
| Pressure upstream(supply) | 101~600 kpa |
| Rotor speed | 1500~2800 rpm |

4. Work analysis model for cylinder-seal system

In order to compare the stability performance of the tilting-pad seal and the fixed pad seal, we put forward here the concept of work done by seal. The energy generated by seal force is imported to the cylinder and cause cylinder vibrating and eventually unstable.

First, the seal force can be expressed as
\[
\begin{bmatrix}
\hat{F}_x \\
\hat{F}_y
\end{bmatrix} =
\begin{bmatrix}
H_{xx} & H_{xy} \\
H_{yx} & H_{yy}
\end{bmatrix}
\begin{bmatrix}
\hat{X}_x \\
\hat{Y}_x
\end{bmatrix}
\] (1)

Where, \( H_y = k_y + j \omega c_y \), \( \omega \) is the excitation frequency. \( \hat{X}_x \) and \( \hat{Y}_y \) are the relative displacements between the cylinder and rotor.

The work done by seal force during one period can be presented as,

\[
W = \int_0^{2\pi} [\hat{F}_x(t)\hat{X}_x(t) + \hat{F}_y(t)\hat{Y}_y(t)]dt
\] (2)

Where, \( \hat{X}_x \) and \( \hat{Y}_y \) is the absolute displacements of cylinder. \( \hat{F}_x \) and \( \hat{F}_y \) are the seal forces acting on cylinder. The seal forces are calculated by Eq.3. Submitting Eq.3 to Eq.5, we have,

\[
W = \int_0^{2\pi} [(k_{xx} + j\omega c_{xx})\hat{X}_x(t) + (k_{yx} + j\omega c_{yx})\hat{Y}_y(t)]\cdot\dot{X}_x(t)dt + \int_0^{2\pi} [(k_{yx} + j\omega c_{yx})\hat{X}_x(t) + (k_{yy} + j\omega c_{yy})\hat{Y}_y(t)]\cdot\dot{Y}_y(t)dt
\] (3)

We can assume,

\[
\begin{align*}
\hat{X} &= X \cos(\omega t + \theta_x) \\
\hat{Y} &= Y \cos(\omega t + \theta_y) \\
\hat{X}_x &= X_x \cos(\omega t + \varphi_x) \\
\hat{Y}_y &= Y_\gamma \cos(\omega t + \varphi_\gamma)
\end{align*}
\] (4)

Where, \( X, Y, X_x \) and \( Y_\gamma \) are the amplitudes, and \( \theta_x, \theta_y, \varphi_x \) and \( \varphi_\gamma \) are the phases of corresponding forces. Submitting Eq.7 to Eq.6, we can get

\[
W = \pi XX_x [k_{xx} \sin(\varphi_x - \theta_x) + \omega c_{xx} \cos(\varphi_x - \theta_x)] + \pi YY_\gamma [k_{yy} \sin(\varphi_\gamma - \theta_\gamma) + \omega c_{yy} \cos(\varphi_\gamma - \theta_\gamma)]
\] (5)

5. Test Procedure

In each set of supporting springs, the system resonance frequency is determined. In the range of \( \pm 5\% \) of the system resonance frequency, the cylinder vibration is too sensitive to seal force. It will cause large identification error. Meanwhile, the cylinder is less sensitive to seal force if rotating speed is beyond \( \pm 30\% \) of the system resonance frequency. So, we choose the test speed in the range of \( -30\% \sim -5\% \) and \( 5\% \sim -30\% \) of the system resonance frequency.

The impedance matrix tests were conducted at zero rotor speed without the inlet of compressed air. The tests were performed by shaking the stator in the x and y directions on two side planes independently. The excitation frequency of the shakers changes from 15Hz to 50Hz in 1.67Hz (100rpm) increment.

The tests were done from 1500 rpm-2800rpm. The vibration signal was recorded at every 100 rpm increment automatically. At each speed, air inlet pressure changes from 0.1 MPa to 0.5 Mpa. Impulse excitation was done simultaneously at the middle of cylinder to obtain the transient damped signal and to calculate the system damping factor for stability analysis.

The bolts on the back of tilting pad seal were crewed after the first experiment. The tilting pad seal was turned to the traditional fixed pad seal. The dimensions of the two kinds of seal were the same. The difference between the dynamics of two kinds of seal can thus be compared and analyzed.
6. Test Results

6.1. Comparison of Leakage

Figure 4 shows the comparison of leakage between the tilting-pad seal and the fixed pad seal versus inlet/outlet pressure ratios. The leakages of two seals increase almost linearly with the increasing inlet/outlet pressure ratio. During the full inlet pressure range, the relative deviations of leakages are less than 5%. Leakages of two seals change little with rotating speed. It means that the leakage ratios of the two kinds of seals are almost the same.

![Figure 4. Leakage comparison result](image1)

6.2. Comparison of work done by seal on cylinder

Figure 5 shows the comparison of work done by two kinds of seals versus inlet/outlet pressure ratios under two rotor unbalance conditions. The work done by two seals both increase with increasing pressure ratio. In case of small inlet pressure, the work done by two seals are almost the same. But with the increase of inlet pressure, the difference becomes large. Work done by the tilting pad seal is smaller. The phenomenon is the same under two rotor unbalance conditions. It means that the energy imported to cylinder system by tilting-pad seal is smaller. Thus, the tilting-pad seal is more stable.

![Figure 5. Comparison of work done by two seals](image2)

6.3. Comparison of damping factor for Cylinder-Seal system

The cylinder-seal system’s damping factor was measured by hammer testing. We hammered the cylinder from vertical and horizontal directions respectively. 10 seconds long data, which contained about 10 damped waves, had been recorded for each hammer test. Figure 6 shows the result of processed signals. The black dot line is the chosen reliable decrement waves processed by random decrement method. And the red line is the exponential fitting result. We can get the values of system damping factor through fitting calculation.

![Figure 6. Damping factor comparison result](image3)
Figure 6. Damped waves before and after fitted

(a) Fixed seal ($\omega=2800$ rpm)  (b) Tilting-pad seal ($\omega=2800$ rpm)

Figure 7. Cylinder-seal system damping factor comparison for two seals

(a) y-direction (Pr=5)  (b) x-direction (Pr=5)

7. Conclusion

In this paper, we present a new kind of tilting pad seal. Experimental research on leakage, rotodynamic characteristics, work and stability analysis are done and compared between tilting-pad seal and fixed-pad seal.

The leakage of tilting-pad seal is nearly the same as the fixed pad seal. Work done by the tilting pad seal on cylinder is lower than that of fixed pad seal. It means that the energy generated by the fixed pad seal is higher than that of tilting-pad seal. The system damping factor is used as the stability criterion for two seals. It is found that the cylinder system with tilting-pad seal have higher damping factor. It means the tilting-pad seal has higher stability than then fixed-pad seal.

With the increasing inlet pressure, the difference between two kinds of seals becomes large. In the modern power plant, inlet pressure reaches 22 MPa, the difference may be larger than the test rig. This tilting pad seal provides a new way for solving the problem of flow induced vibration in turbomachinery.
References

[1] S. Greathead and P. Bastow. Investigations into Load Dependent Vibrations of the High-Pressure Rotor on Large Turbo-Generators, in: Proceedings of Conference on Vibrations in Rotating Machinery, The Institution of Mechanical Engineers, University of Cambridge, 1976, pp. 279–285.

[2] Zeidan F, Perez R, Stephenson E. The Use of Honeycomb Seals in Stabilizing Two Centrifugal Compressors[C] // 22nd Turbomachinery Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, TX, Sept. 1993: 14-16.

[3] Childs, D., Elrod, D., and Hale, K., 1989, "Annular Honeycomb Seals: Test Results for Leakage and Rotordynamic Coefficients; Comparisons to Labyrinth and Smooth Configurations," ASME Journal of Tribology, Vol. 111, pp. 293-301.

[4] Childs, D., and Elrod, D., and Ramsey, C, 1990, "Annular Honeycomb Seals:Additional Test Results for Leakage and Rotordynamic Coefficients," in IFToMM, Proceedings of the Third International Conference on Rotordynamics,Lyon, France, pp. 303-312.

[5] Memmott, E. A. The Stability of Centrifugal Compressors by Applications of Tilt-Pad Seals [C]. IMechE, 8th International Conference on Vibrations in Rotating Machinery, Swansea, 2004, 81-90.