Experimental Research on Influence of Seal Fluid Excitation Force on Rotor Bearing System

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Abstract: A test rig has been built to study the influence of seal fluid excitation force on rotor-bearing system. The vibration change of rotor near the critical frequency and the stability change of bearing oil-film under the seal fluid excitation force are studied in this paper. The test result demonstrates that the vibration amplitude of rotor is affected by the seal force obviously during the region near the critical rotor speed, the damping effect induced by fluid excitation force is in dominant position near the rotor critical speed on this rotor bearing system. The influence of seal fluid excitation force on stability of bearing oil film is also be studied. The result shows that the begin speed of half frequency component appearance is earlier than the speed of compressed fluid open test. It means the seal fluid excitation force can induce the bearing oil film whirling in advance, and reduce the stability of this rotor bearing system.

1. Introduction
Annual seal is widely applied in turbomachineries to minimize the fluid leakage and improve the unit efficiency. With the improvement of turbine operating parameters, the system instability problem resulting from seals, which is usually called as fluid induced vibration, becomes more and more serious [1]. The earliest experimental research for annual seal can trace back to Benckert and Wachter’s research on the excitation force of labyrinth seal in 1978, they studied the influence of preswirl and rotor whirling on seal fluid excitation force [2]. More theoretical and experimental studies were carried out by Childs and Pelletti [3-5]. Kwanka built a seal test rig to measure the dynamic coefficients of seal [6]. Because of the restriction of field test for practical annual seal, it is a valuable work to study the influence of seal fluid excitation force on rotor bearing system by experimental method. So, in this paper considering the above experiences, we also built a test rig to study the influence of seal fluid excitation force on rotor-bearing system.
2. Mechanism of Rotor Instability Caused by Seal fluid Excitation Force

The mechanism of rotor instability caused by seal fluid force is analyzed firstly. Assume that the rotor is in steady-state concentric whirling motion in circle by the radius of $\delta$. $\Omega$ is the angular velocity of rotor whirling motion and $\omega$ is the auto-rotation angular velocity of rotor. As the Figure. 1 shows. The X axis expresses horizontal direction and the Y axis expresses vertical direction. Under this motion, the flow parameters are equally distributed along the circumferential in the seal, so the direct coefficients are considered approximately equal and the cross coefficients are considered having the equal absolute value with opposite sign in X and Y directions. The seal fluid force can be written as Equation.1.

$$\begin{bmatrix} F_x(t) \\ F_y(t) \end{bmatrix} = \begin{bmatrix} K & k \\ -k & K \end{bmatrix} \begin{bmatrix} x(t) \\ y(t) \end{bmatrix} + \begin{bmatrix} C & c \\ -c & C \end{bmatrix} \begin{bmatrix} \dot{x}(t) \\ \dot{y}(t) \end{bmatrix}$$

(1)

In Equation.1, The $x$, $y$ mean the perturbation displacements of rotor corresponding to the equilibrium position in X and Y directions, and $\dot{x}$, $\dot{y}$ mean the perturbation velocities. $K$, $C$, $k$, $c$ mean direct stiffness coefficient, direct damping coefficient, cross stiffness coefficient and cross damping coefficient respectively. In concentric whirling motion, the rotor motion can be expressed by the Equation.2.

$$\begin{cases} x(t) = \delta \cos \Omega t \\ y(t) = \delta \sin \Omega t \end{cases}$$

(2)

Assuming the rotor position in Figure.1 is in the moment of $t = 0$, some conditions can be got as follow: $x(0) = 0$, $y(0) = -\delta$, $\dot{x}(0) = \delta \cdot \Omega$, $\dot{y}(0) = 0$. Plugging this conditions into Equation.1 and Equation.2, the fluid induced forces $F_x$ and $F_y$ on X and Y directions can be obtained. Generally, the X direction force is perpendicular to the direction of rotor motion, it is known as tangential force $F_t$, and the Y direction force is parallel to the direction of rotor motion, it is known as radial force $F_r$. Finally, they can be expressed as,

$$\begin{cases} F_t = F_x(0) = (k - C\Omega) \cdot \delta \\ F_r = F_y(0) = (-K - c\Omega) \cdot \delta \end{cases}$$

(3)
Consequently, the work done by the seal fluid induced forces during one cycle of rotor whirling motion can be expressed as,

\[
W_f = -\int_{0}^{2\pi} \left( F_x \dot{x} + F_y \dot{y} \right) dt = 2\pi \delta^2 \left( k - \Omega C \right)
\]  

(4)

See from the Equation 4, the elastic force corresponding to the direct stiffness coefficient \(K\) is a conservative force, so the work done by this force during one cycle of rotor whirling motion is 0. And the damping force corresponding to the direct damping coefficient \(C\) always do negative work, in other words, consuming energy during the whole process. However, the non-conservative elastic force corresponding to the cross stiffness coefficient \(k\) and the damping force corresponding to the cross damping coefficient \(c\) do positive or negative work during one motion cycle, the sign of work is determined by the shape of rotor whirling track, the value and sign of dynamic coefficients. In one motion cycle, if the energy inputting to the system is less than the energy consuming by the damper, the rotor whirling will going to be smaller and smaller and vanishing in the end, it means the system is stable. Conversely, if the energy inputting to the system is more than the energy consuming by the damp, the whirling motion will growing and the system will become to the state of unstable finally.

In Equation 4, when \(k < \Omega C\), gets \(W < 0\), means the energy of rotor system is consumed by flow, the result is weaker whirling motion and stabler system. Conversely, when \(k > \Omega C\), gets \(W > 0\), means the flow inputs the energy to the system, the result is stepped-up whirling motion and unstable system. When \(k = \Omega C\), \(W = 0\), it is an equilibrium state, the strength of whirling motion remains unchanged. So, in order to enhance the stability of rotor-seal system, it is necessary to reduce the cross stiffness coefficient and increase the direct damping coefficient. Finally, a whirling coefficient \(\Omega_f\) is introduced to judge the stability of rotor system.

\[
\Omega_f = k / \Omega C
\]

(5)

From the above analysis, the value of \(\Omega_f\) is more smaller and the rotor stability is more better. From the Equation 3, the tangential force \(F_t\) is determined by the cross stiffness coefficient \(k\) and the direct damping coefficient \(C\) directly, so the reason which induce the rotor whirling motion and the rotor system instability is exactly the tangential fluid excitation force.

3. Design of Test Rig

A test rig has been built. The structure of test system is shown in Figure 2. Compressed air is used as the test fluid. The rotor length is 1200 mm, diameter is 70 mm. In the middle of the test rotor, an aluminium cylinder is set to hold the test seals, it is shown in Figure 3. In order to make a structure of labyrinth seal, a steel sleeve with 180 mm diameter and 430 mm length has been installed in the rotor section to coordinate with the cylinder. The cylinder is supported by a flexible supporting system consist of tension springs as shown in Figure 3. The flexible supporting system can enlarge the influence of tiny fluid excitation force on cylinder. The cylinder is divided into upper half and bottom half. Four air inlets are symmetrically placed along the circumferential in the middle of cylinder. The test fluid flow into the cylinder on the middle and flow out from two end side of cylinder, a flowmeter is installed on the main inlet pipe to test the flow rate across the seals. Two sliding bearings are used to support the test rotor, the bearing parameters are as follow: length-to-diameter ratio 0.436, diameter 70 mm, clearance-to-radius ratio 0.002. The type of bearing lubricating oil is #32 turbine oil. 8 rings of dovetail grooves are set on the inner wall of cylinder, and the test seals are installed on the cylinder as show in the Figure 4.
Figure 2. Structure drawing of test rig for seal-bearing-rotor system
(1 Compressor, 2 Oil System, 3 Motor, 4 No.1 Bearing, 5 No.1 Balance Disk, 6 springs, 7 Cylinder, 8 No.2 Bearing, 9 No.2 Balance Disk, 10 Exciter; 11 Air Tank)

In the test, 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. 2 magnetoelectric velocity transducers are located on the cylinder to monitor the absolute vibrations from horizontal and vertical directions. An eddy sensor is fixed on the bearing box to work as a keyphasor transducer. The phases of all vibration signals are based on this keyphasor signal. All measurements are conducted at room temperature (28°C).

4. Influence of Seal Fluid Excitation Force on Rotor Vibration near the Critical Frequency
The change of rotor absolute vibration is tested under the fluid excitation force. Comparison results of rotor vibration amplitudes and phases with or without test fluid are shown in Figure.5. The rotor speed is from 800 r/min to 3800 r/min, when the test with fluid in, the fluid pressure is kept to 0.5 Mpa all over one test.

In Figure.5, the critical speed of rotor is know as 2400 r/min approximately. The vibration amplitude of rotor is affected by the seal force obviously during the rotor speed region from 2000 r/min to 3700 r/min as show in Figure. 5a, but the vibration phase is basically not affected by the seal force as show in Figure.5a. Near the critical rotor speed, the rotor vibration amplitude with fluid in is greater than the one without fluid in. It is demonstrated that the damping effect induced by fluid excitation force is in dominant position near the rotor critical speed, so the rotor vibration amplitude is restrained when the seal force joint to the test rotor bearing system.
5. Influence of Seal Fluid Excitation Force on Stability of Bearing Oil Film

In addition to affecting the rotor directly, the seal fluid excitation force can also exert influence on the bearings through the rotor. Generally, when the bearing oil film begins to whirling, the half frequency component appears. In order to study the influence of seal fluid excitation force on stability of bearing oil film, a speed-up test is carried out. First, a rotor speed-up test without compressed fluid in has been done, speeding up the rotor until the half component of rotation frequency occurs, and the oil film unstable speed region of No.1 and No.2 bearings have been determined respectively. Second, speeding up the rotor again with the compressed fluid in, and the fluid pressure is kept on 0.5 Mpa all over the test. The changes of each vibration frequency components have been tested. Figure.6 and Figure.7 show the test results of bearing’s vibration amplitude changes as rotor speed and frequency before and after the compressed fluid flow into the seals.
From the Figure.6 and Figure.7, we can see that the vibration amplitude in the initial stage is dominated by the fundamental frequency component. In the compressed fluid close test, the half frequency component begins to appear at 5000 r/min approximately, and the amplitude of half component becomes bigger and bigger as the rotor speed up. It means the bearing oil film begins to whirling and becomes unstable from 5000 r/min in this rotor-bearing system. Compared to the result of compressed fluid close test, the begin speed of half frequency component appearance is earlier than the speed of compressed fluid open test. As show in Figure.6, the begin speed of No.1 bearing half-frequency vibration is brought forward to 4800 r/min obviously, and the similar situations exist on No.2 bearing vibration as show in the Figure.7. Furthermore, in the test of compressed fluid open, the increasing trend of half frequency vibration amplitude is more obvious as the rotor speed goes up than the trend in the test of compressed fluid close.

![Waterfall curve of No.1 bearing’s vibration as rotor speed and frequency](image)

Figure 6. Waterfall curve of No.1 bearing’s vibration as rotor speed and frequency
Figure 7. Waterfall curve of No.2 bearing’s vibration as rotor speed and frequency

6. Conclusion
In this paper, a test rig has been built to study the influence of seal fluid excitation force on rotor-bearing system. The test result demonstrates that the vibration amplitude of rotor is affected by the seal force obviously during the rotor speed region near the critical rotor speed, the damping effect induced by fluid excitation force is in dominant position near the rotor critical speed on this rotor bearing system. The influence of seal fluid excitation force on stability of bearing oil film is also be studied. The result shows that the begin speed of half frequency component appearance is earlier than the speed of compressed fluid open test. It means the seal fluid excitation force can induce the bearing oil film whirling in advance, and reduce the stability of this rotor bearing system.

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