The Testing and Analysis of Water Lubricated Bearing Test Rig In Seawater Desalination High Pressure Pump

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Abstract

The test machine of the water lubricated journal bearing was designed based on the structure of seawater desalination high pressure pump. Compute the dynamics Fluid-structure interaction (FSI) of the water lubricated bearing-rotor system and test the journal orbit and the journal pressure pulsation. The correctness of numerical simulation is by comparing numerical simulation with the results of test. The test mainly test the pressure spread and the journal orbit at steady condition. By comparing the main parameters, the hydrodynamic lubrication and system stability could be found at different rotating speed. As a result, the following conclusions were obtained. In low rotating speed, the pressure is not stable and distribute discontinuously, so the pressure curve is not stable. The increase of rotating speed would expand high pressure area and pressure value, therefore, the carrying capacity was improved and the pressure distribution and curve is stable, so the system tends to stable. The rotating speed has no obvious influence on pressure pulsation amplitude, this is because the high pressure pulsation amplitude concentrate in the low frequency area. Based on the numerical simulation and test analysis of the journal whirling orbit, increasing the rotating speed would increase the centrifugal force and improve the journal whirling amplitude. The increase of water film depth cause the journal whirling center lift obviously, the sensitivity of system to the external exciting force and unbalancing force improved. Therefore, in order to improve the system stability margin, dynamic balancing accuracy should be improved. The result shows that the combination of numerical simulation and test to analyze the dynamic characteristic of the rotor-bearing system can is useful and important.

Keywords

Seawater desalination pump; water lubricated bearing test rig; journal whirling orbit; FSI; pressure pulsation.

1. Introduction

At present, the typical three calculation methods of journal whirling orbit which are based on fluid lubrication theory are Hahn, Holland and Mobility. These methods are based on some certain assumptions. They sometimes have many imperfections and very far from the actual situation. With the rapid development of the Computing Fluid Dynamics (CFD), the researchers begin to solve the journal whirling orbit based on CFD. Huiping Liu adopt the ADINA to research the whole elastic hydrodynamic lubrication of the complex bearing-rotor system, the cavitation in the oil film was simulated through the phase transition boundary condition. The result was compared and analyzed to the result in the half Sommerfeld boundary condition. In addition, the influence of bearing elastic deformation and periodic unbalancing loads to the journal orbit was analyzed. The numerical simulation result was compared to the experimental result. The comparison indicts that phase transition boundary condition can handle the cavitation in the oil film. The ADINA system’s the whole

Fluid-structure interaction technology can have a good lubrication analyses about bearing-rotor system. V. Meruane obtained the free response in different rotating speed based on the bearing-rotor system built by the ADINA. The oil film pressure distribution in 1000rpm is compared to both the axial pressure distribution under three different eccentricity and the expressions for short bearing, long bearing and their harmonic combination. The numerical solution is very similar to the one obtained with the harmonic combination. Rocket engine turbo pump for low viscosity medium lubricated bearings were studied on stress, high-speed and light load conditions on XI'AN Jiao tong University by Xiao yang Yuan, and designed for water-lubricated bearing test rig trial under such conditions. Before and after the end of the bearing test rig are arranged two eddy current displacement sensor at right to test water film thickness at horizontal and vertical directions, and then measured the overall water film thickness, and then measured the water film thickness of the whole. Pressure sensor and flow sensor was arranged through the pipeline of the measurements of inlet pressure and flow. Under a given pressure, change rule between the water film thickness and pressure of water supply and changes between water pressure and power consumption of the system and speed relations can be detected.

This paper uses the combined research method of CFD method and experimental test to research the journal orbit and the journal Pressure distribution about the water lubricated bearing–rotor system. The research mainly contents the journal orbit, the journal pressure distribution and pressure pulsation of the water lubricated bearing-rotor system in stable operation about sever different rotational speed. Firstly the paper introduces a new type of FSI calculation method, rigid body, to calculate the journal orbit under different speeds. And a test rig was designed to measure water lubricated bearing journal orbit and the journal pressure distribution. It is the successful for the journal orbit testing, and data processing.

2. CFD Method for Solving the Journal Orbit

2.1 Rigid Body

The calculation of bearing-rotor system transient flow field requires the use of FSI calculation method. Water film force acts on the rotor in CFX, to correspond the journal orbit. In turn, the moving of journal will have an impact on water film thickness distribution, resulting in changes in their water film pressure distribution so that the water film bearing capacity will change accordingly. The water film capacity changes will cause changes in journal orbit, followed by cycle, the interaction between the rotor movement and water film thickness, resulting in variation of journal orbit and other key parameters in the calculation of the time.

CFX was used in this paper provided for CFD software, CFX provides two calculation methods of Two-Way FSI and Rigid Body, when solving, Two - Way FSI provide CFX solver and ANSYS solver at the same time, which has very high requirements about the computational memory and computing time. And this method is only suitable for application in solid structure deformation or smaller mesh deformation of flow field, the flow field due to its role in a variety of loads will have a significant amount of deformation of the mesh, when the mesh be pulled, the mesh quality will be worse and worse until it affect the calculation. The Rigid Body calculation methods simply call CFX solver, which cause a shorter computation time. And the mesh remeshing can be combined. When the mesh appears relatively large deformation or the mesh quality is reduced by pulling, by calling the mesh remeshing, can always updated mesh to ensure that the mesh could in a high state.

2.2 Setup Process

(1)Setting up the water lubricated bearing-rotor system model to get water film 3D model and rotor 3D model. Six rotor inertia and mass was obtained by Pro/E because the need of rotor properties when create rigid body.

(2) The water film 3D model import ICEM to mesh and record mesh replay file. The structure mesh with high density was used to mesh the water film. The optimum number of layers is divided according

to mesh-independent analysis. It is very important for frequent operation to ensure the correctness of the t steps as far as possible, and reduce the modification operations.

(3) Setting boundary conditions to rigid body. The inlet and outlet water lubricated bearing for axial. Therefore, the water film axial ends are set for the import and export. But the bearing is short, to avoid reflux, the export face was set as opening. And reference pressure is set to 0MPa. The axial imported for pressure, the inlet pressure change according to simulation. Water film outer wall is set to static wall, and inner wall is set to rotate wall.

(4) Water film set to mesh motion, no rotational mo. The turbulence model is the $k-\varepsilon$ Model. Flow medium is water.

(5) In this paper, water film has four surfaces. We assume that the wall is stationary in addition to the inner wall surface.

(6) Create rigid body. Establish a fixed coordinate system before creating rigid body. Coordinate origin is the center of the outer wall of water film. In this paper, the rotating cylinder inside the water film was assumed as a rigid rotor. Input mass and moment of inertia of rigid rotor. The axis of rotation to the Z axis, involves only rigid body translation, so the rotational degree of freedom is constraint only the movement of the X and Y direction. The direction of gravity is the -Y direction.

(7)Interface Settings. Coupling form for Interface does not need special setting. We just need to set up. In this paper, the uncoupled forms for interface need to be set and set the movement to rigid body solution. And the rigid body in step 6 is set as rigid body.

(8) CEL Description. Because the water lubricated bearing-rotor system under different load, thus the writing of the external load by CEL was required, and then input it into the calculation program. In the mesh remeshing, you need to write the mesh quality monitoring standards, which adopted in the numerical simulation is the minimum orthogonality angle. Its expression is as follows: minVal (Orthogonality Angle Minimum) @ REGION: SOLID < 10 [deg]. When a mesh minimum computational domain orthogonality angle is less than 10, CFX will suspend calculation and update mesh remeshing. CFX will back to calculate when the mesh is updated completely. In rotation speed Settings, the journal moment in the axis of disturbed state makes the position changes unceasingly, CFX can’t set a rotational speed without a fixed coordinate system, so CEL is used to read the axis position in the process in each moment. The axis position is used to load the tangential velocity. The rotational speed is synthesized by the tangential velocity. Tangential velocity formula IS shown in the following formula:

$$v_x = (x - x_d) \cdot \omega$$

$$v_y = (y - y_d) \cdot \omega$$

x, y Represents a point’s coordinate value of the journal in some time; x_d, y_d stands for the axis displacement. The rotational speed is set by this method does not require a fixed coordinate system.

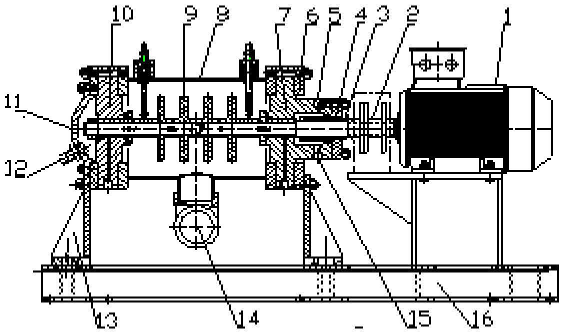
(9)After rigid body is created, we set the mesh remeshing. Setting calculation interrupt condition; Selecting mesh reconstruction method; Selecting start conditions; Selecting mesh reconstruction position; importing geometry file; Importing replay file.

(10) Monitoring and extraction of the data. After setting mesh remeshing can be numerical simulation. In the solver data can be monitored. Monitoring data are mainly axial displacement, velocity, acceleration, load capacity and minimum water film thickness of the water film. Export the data into the origin to screen and process.

3. Test Rig Design and Test Plan

3.1 Test Rig Overall Design

According to the structure characteristics of the desalination high pressure pump, at the same time, considering the safety of loading system and measurement system, designs the water lubricated bearing test rig as shown.



1- inverter motor 2- couplings 3- shaft 4- cartridge mechanical seal 5- drive end taps 6- flange 7- drive end bearings 8- barrel 9- disc 10- non-drive end bearing 11- non-drive end taps 12- non-drive end into the drain 13- bracket 14- big drain15 Drive end into the drain 16- base

Figure 1. Test bed structure

Mechanical structure of the test rig is mainly composed of barrel, on both ends of the sliding bearing, the loading disk, support, and a base. The main shaft is supported by sliding bearings. At the drive end bearing, cartridge mechanical seal, the main shaft and plain bearings formed a smaller cavity. External cavity has a pressure sensor mounting holes and one into the drain. At the non-drive end bearing, cap, the main shaft and sliding bearing also constitute a relatively large volume of cavity, the same cavity in vitro has a pressure sensor and into the drain. At the ends of the sliding bearing by flange connected to cylinder, respectively. On the other side of the cylinder and the combination of the sliding bearing, through the base fixed on the bracket. In the barrel right there is a big drain connection. Loading ways made largely simplified, is placed directly on the main shaft through the key connection four disc, through the disc with the rotation of the spindle radial load, used to simulate the multistage process of multistage pump running desalination bear radial load on the pump. This way of loading advantage is to adjust the size of the load by adjusting the number of disk, and the adjustment is more convenient and simple operation. The disadvantage is that the precision of the load cannot guarantee.

3.2 Test Plan

The test bearing d = 45 mm diameter, bearing width is 45 mm, B = c = 0.2 mm radius clearance. Take the dry rotor test method, that is, from at both ends, the water from a drain drainage at the bottom of the barrel. Supply pressure choice for 0.1 MPa and 0.15 MPa, USES frequency conversion motor to change speed, speed range from 1500 to 3000 RPM, every 500 RPM increases on a test. Experimental study on the object for the spindle axis of the system is stable operation trajectory and sliding bearing circumferential pressure distribution. Axis direction of rotation is clockwise, so along the direction of rotation, in turn, set up six pressure pulse sensor, numbered from 1 to 6, according to the hydrodynamic pressure lubrication theory of sliding bearing, the six pressure sensor only measured pressure distribution area for the water film sliding bearing dynamic pressure area. On the cylinder arrangement of a set of mutually vertical eddy current displacement sensor, Numbers for 7 and 8. The schematic diagram as shown in the figure below.The installation of data acquisition device

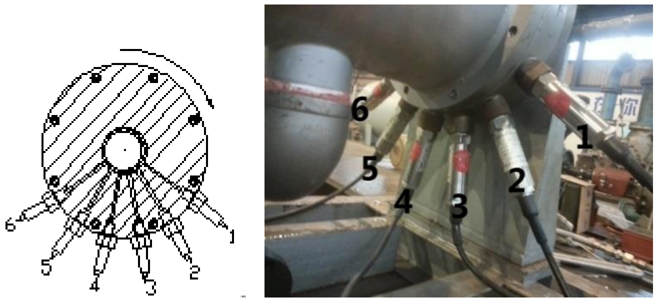


Figure 2. Schematic diagram of pressure sensor arrangement

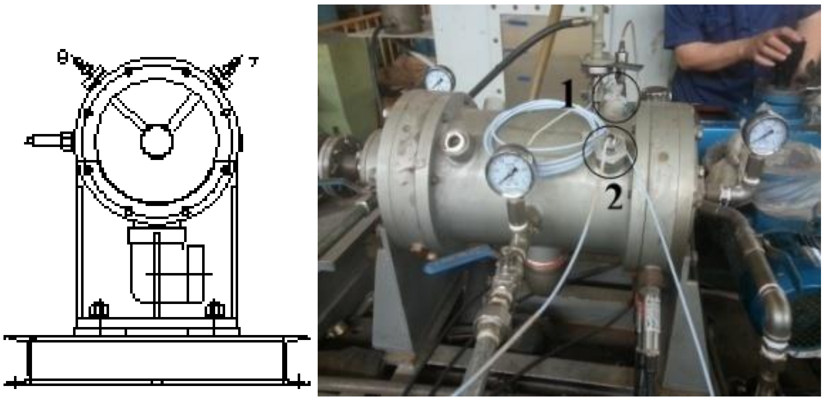
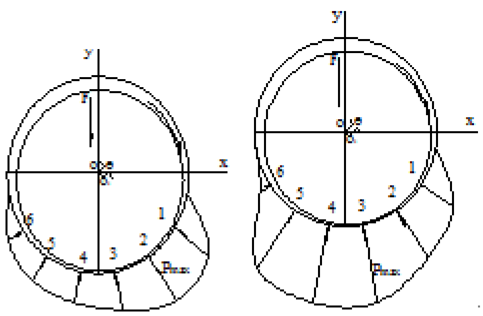


Figure 3. Schematic diagram of displacement sensor installation

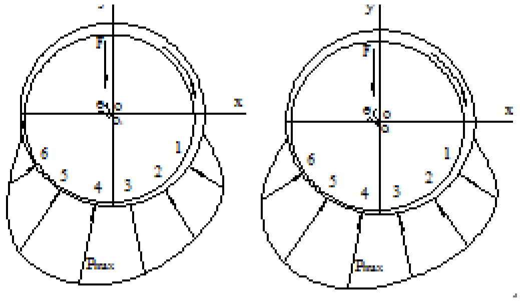
4. Test Results and Analysis

4.1 When the Feed Water Pressure of 0.1 Mpa Bearing Under Different Rotational Speed Circumferential Pressure Distribution Is Analyzed

In figure 4, the circumferential stress distribution under different rotational speed bearing have a common law, namely the circumferential stress from the water to water location after the first increased, high middle and low on both sides, forming a similar to parabolic distribution. But from a low speed to high speed bearing circumferential pressure distribution has an obvious change trend. Low speed, the maximum pressure in the pressure at point 2, with the increase of rotational speed, the maximum pressure in the point 2 to point 3 final stability in point 4, and its amplitude is becoming more and more big, finally came to a stable state in 2500 and 3000 RPM. Because of low viscosity of water at low speed when the water film forming effect is poorer, water film bearing effect is poorer, with the increase of rotational speed, water film bearing effect increased, leading to high expansion and the increase of the pressure amplitude, from 2500 r/min and 3000 r/min can be seen that the pressure distribution and pressure amplitude change is not big, explain bearing rotor system has been in a relatively stable state.



1500rpm 2000rpm



2500rpm 3000rpm

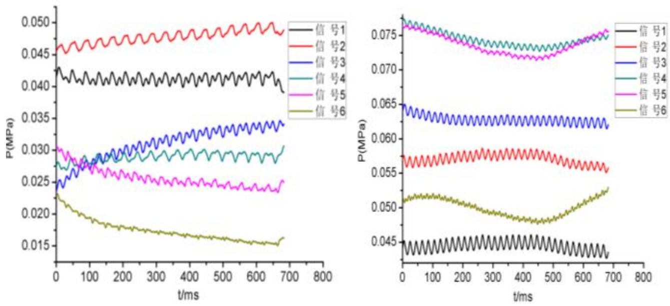
Figure 4. Feed water pressure of 0.1 MPa when bearing circumferential stress distribution under different rotational speed

Figure 4 for feed water pressure of 0.1 MPa when bearing circumferential stress distribution under different turn.

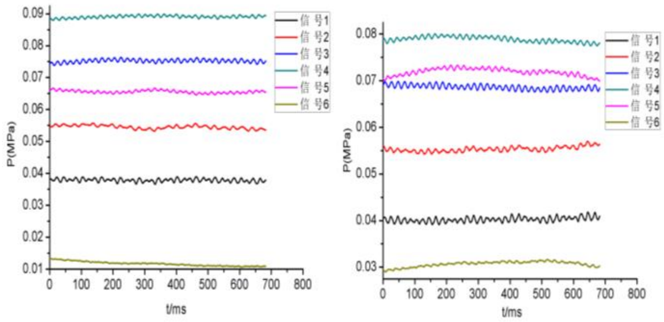
4.2 When the Feed Water Pressure of 0.1 Mpa Bearing Under Different Rotational Speed Circumferential Stress Time Domain Analysis

Figure 5 for feed water pressure of 0.1 MPa when the bearing under different turn circumferential pressure time-domain diagram. Can be seen from the figure 9 in pressure at point 1, the pressure amplitude change is not big, between 0.04 to 0.045 MPa, and the wave shape and wave period has a certain change, 1500 RPM, the wave shape is composed of two peaks of one large and one small, and has obvious fluctuation cycle, with the increase of rotational speed, the wave shape tends to be stable, pressure signal curve, tend to be stable. The pressure of the pressure at point 2 amplitude increased from 0.045 MPa to 0.055 MPa when rotating speed is 3000 RPM, change is more obvious its wave shape, at low speed when the volatility is relatively severe, its fluctuation flatten out after 2500 RPM. Pressure 3 pressure amplitude change is more obvious, increased from 0.03 MPa at 1500 RPM to 3000 RPM 0.07 MPa, combined with the circumferential stress distribution, the bearing are results of increased speed spindle to left shift static equilibrium position, thus makes the pressure at point 3 of stress amplitude increases. Pressure measuring point and pressure amplitude change is more intense, increased from 0.027 MPa at 1500 RPM to 3000 RPM 0.08 MPa, and its in low speed, wave shape is irregular, and the pressure curve have some volatility, speed increase after its wave character and the pressure curve tends to be stable. Pressure of 5 points has been 0.05 MPa and 0.06 MPa pressure amplitude, wave shape, cycle and the change of the pressure curve is not particularly evident. Pressure point 6 different pressure amplitude change is not big, this is due to it’s at the end of the wedge gap, speed of change on the stress amplitude, the impact of the fluctuation cycle and shape are small.

Integrated pressure curve under four kinds of rotational speed, low speed 1500 RPM, the pressure measuring point of the pressure curve has certain fluctuations, slowly with the increase of rotational speed fluctuations is abate, 2500 RPM began in a relatively stable state, until 3000 RPM system is in a stable state.



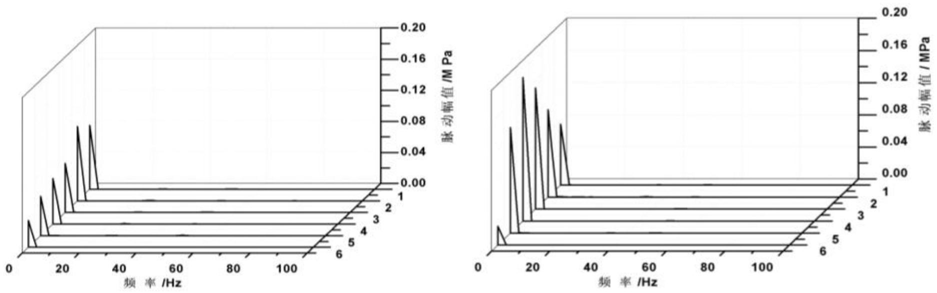
1500rpm 2000rpm



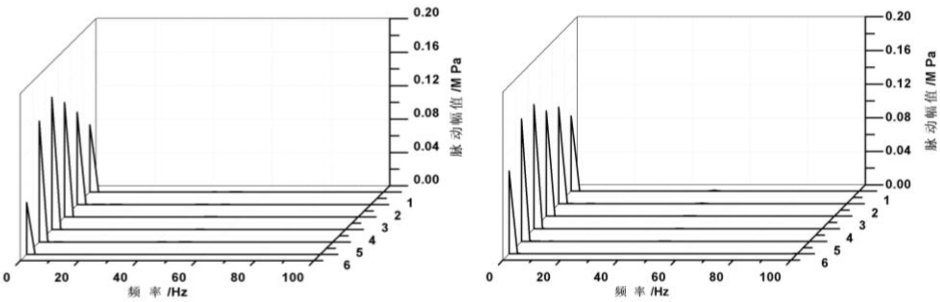
2500rpm 3000rpm

Figure 5. Feed water pressure of 0.1 MPa when bearing axial pressure time-domain diagram under different rotational speed

4.3 When the Feed Water Pressure of 0.1 Mpa Bearing Circumferential Stress Spectrum Analysis under Different Rotational Speed



1500rpm 2000rpm



2500rpm 3000rpm

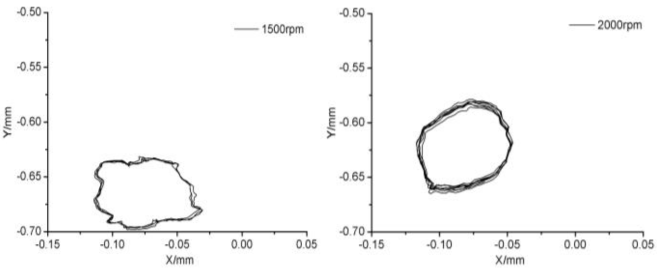
Figure 6. When the feed water pressure 0.1 MPa bearing under different turn circumferential pressure pulsation spectrum diagram

In order to analyze the bearing axial pressure pulsation amplitude frequency characteristics, using the fast Fourier transform (FFT) spectrum analysis, the pressure pulsation of sliding bearing six monitoring sample values in sequence FFT, get six monitoring points under different rotating speed of pressure pulsation spectrum diagram. Spindle speed is 1500 RPM, 1500 RPM, 2000 RPM, 3000 RPM, turn the corresponding frequency of 25 Hz, 33.3 Hz, 41.6 Hz, 50 Hz, as a reference, through the analysis of the pressure pulsation amplitude energy of different monitoring sites to study the influence of different rotation speed to its analysis.

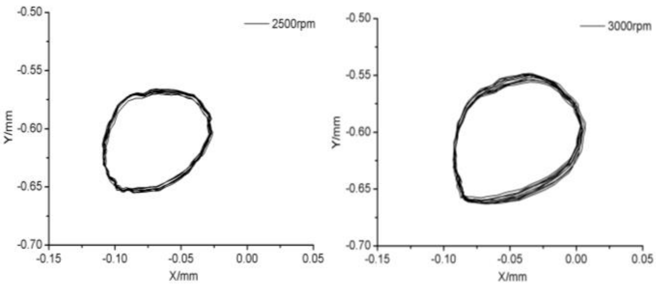
Figure 6 for feed water pressure 0.1 MPa when the bearing under different turn circumferential pressure pulsation Spectrum diagram, can be seen from the figure 6 under different rotational speed of the pressure pulsation amplitude have one thing in common, is in the low frequency (about 1.4 Hz) when the pulsation amplitude is the largest, and its value is significantly greater than other frequency, and the pressure pulsation amplitude distribution and circumferential pressure distribution almost unanimously, 1500 RPM at low frequency of pressure pulsation amplitude is relatively small, speed increase of the pressure pulsation amplitude increase, but in the 2000 RPM to 3000 RPM, the low frequency of pressure pulsation amplitude is almost the same, it also illustrates the system began to stabilize. Under different rotational speed, in addition to have a larger low-frequency pressure pulsation amplitude and frequency in turn and 50 Hz has a relatively weak pulse amplitude. Through the analysis, different speed on bearing axial pressure pulsation, because affect the pressure pulsation amplitude is mainly for the low frequency area, speed changes the pressure pulsation amplitude is not very obvious.

4.4 Numerical Simulation and Experimental Study on the Axis Of Vortex Trajectory Analysis

4.4.1 Track Feed Water Pressure Of 0.1 Mpa When Axial Vortex Trajectory Analysis Under Different Rotational Speed



1500rpm 2000rpm



2500rpm 3000rpm

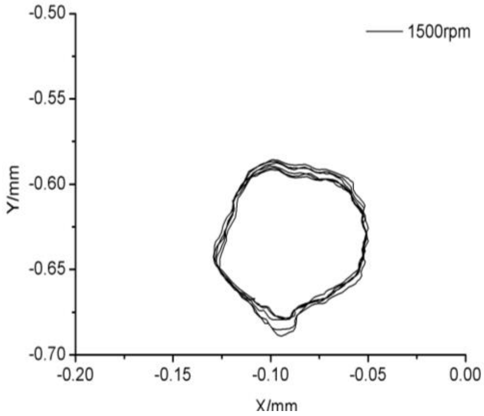
Figure 7. Feed water pressure of 0.1 MPa when axial vortex trajectory diagram under different rotational speed

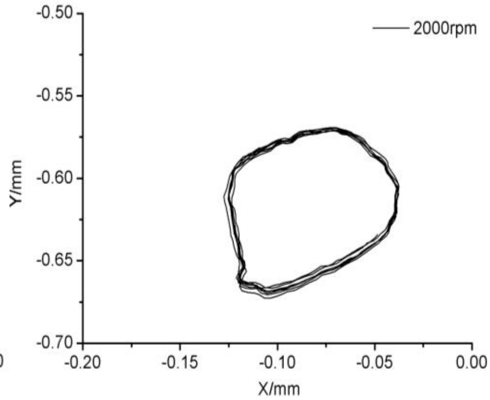
Table 1. When the feed water pressure of 0.1 MPa journal eddy center with varied with the change of rotating speed

Speed/rpm	1500	2000	2500	3000
Xo/mm	-0.07	-0.07	-0.06	-0.04
Yo/mm	-0.66	-0.62	-0.61	-0.59
X /mm	0.051	0.059	0.079	0.113
Y /mm	0.053	0.061	0.081	0.125

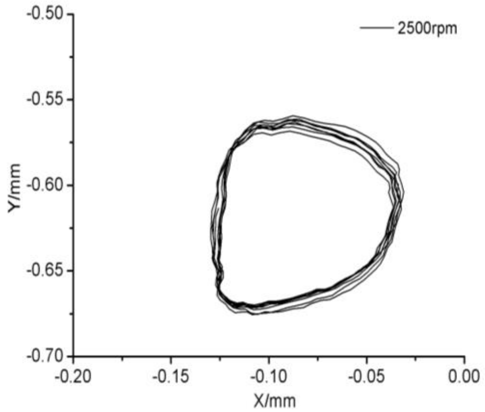
Figure 7for in the feed water pressure is 0.1 MPa, the axial vortex trajectory under different rotational speed figure, can be seen from the figure 10, when supply pressure is constant, when the working condition of low speed because of the relative speed between shaft and bush is small, and therefore at low speed when its hydrodynamic pressure lubrication effect is poor, at the same time by the circumferential pressure bearing part of the figure, the low speed when the value of dynamic pressure, the lower, under external load effect is poorer, extreme cases may cause water film lubrication of incomplete, cause the direct contact of shaft and bush, destabilize the vortex shape as shown. Results of increased speed centrifugal force increases, the vortex center gradually to the upper right deviation, vortex amplitude increase gradually. Speed increase into the wedge gap lubrication medium water, the more water film thickness increases, the dynamic pressure value increases the load effect, the better, also slowly approximation in the elliptical shape of vortex, thus bearing rotor system gradually stabilized.

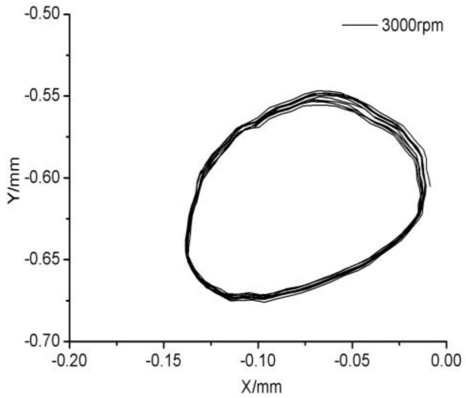
4.4.2 Feed Water Pressure 0.15 Mpa When Axial Vortex Trajectory Analysis Under Different Rotational Speed





1500rpm 2000rpm





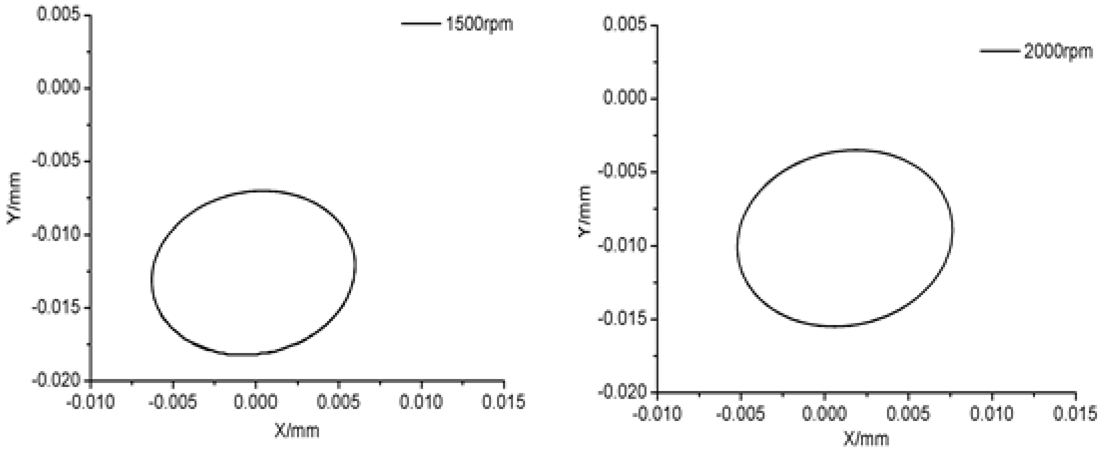
2500rpm 3000rpm

Figure 8. Feed water pressure of 0.15 MPa when axial vortex trajectory under different rotational speed

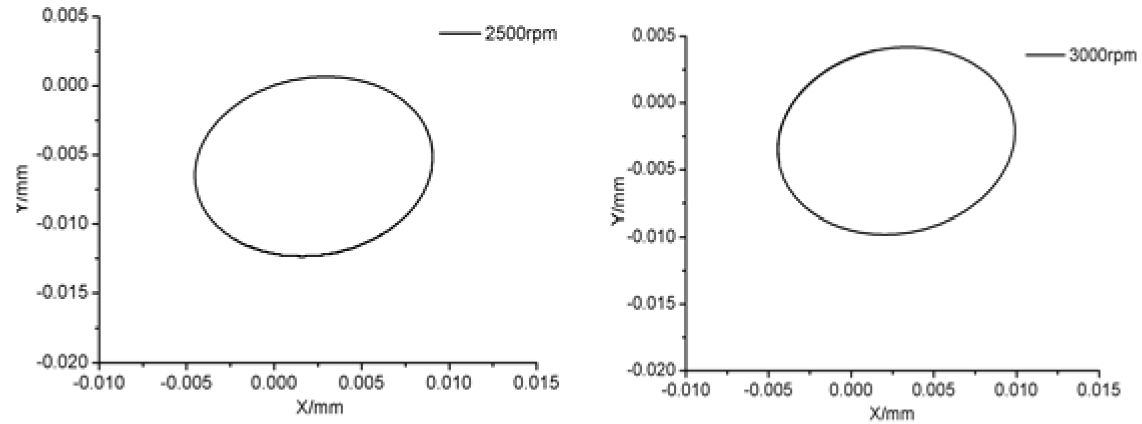
Table 2. Feed water pressure of 0.15 MPa when the journal eddy center with varied with the change of rotating speed

speed/rpm	1500	2000	2500	3000
Xo/mm	-0.07	-0.07	-0.06	-0.04
Yo/mm	-0.66	-0.62	-0.61	-0.59
X /mm	0.062	0.071	0.093	0.13
Y /mm	0.059	0.073	0.096	0.128

Figure 8 for the feed water pressure is 0.15 MPa, the axial vortex trajectory under different rotational speed figure, can be seen from figure 8, the change trend of the vortex trajectory and feed water pressure is 0.1 MPa, when there are two main difference. (1) the same speed, the increase of the feed water pressure makes the journal eddy center also upward migration, this is because the feed water pressure would increase bearing axial dynamic pressure value, increase its bearing capacity. (2) the same speed, feed water pressure increase Journal eddy amplitude also increased accordingly.



1500rpm 2000rpm



2500rpm 3000rpm

Figure 9. Journal obit of dry rotor in different rotating speed in feed water pressure 1 MPa

Figure 9 for feed water pressure 1 MPa nowadays under different rotational speed axial vortex trajectory figure. Can be seen from the figure 9, as the speed is reduced, journal eddy amplitude decreases, its level of vortex center position to negative x direction deviation, and the offset is more, vertically downward shift. Can be seen from the figure, the journal eddy amplitude increased with the increase of rotational speed increases, this is due to the rotor of the centrifugal force caused by the larger, the vortex center was also clear up with the increase of rotational speed, this is caused by liquid film thickness increase, with the increase of rotational speed, causing the rotor - bearing system is sensitive to external excitation force, unbalanced force higher degree, therefore, in order to improve the stability margin of rotor system, the higher the speed of the rotor dynamic balancing precision of the higher level requirements.

Comprehensive CFD numerical simulation results and the test results of axial vortex trajectory that: (1) journal eddy amplitude increased with the increase of rotational speed increases, this is due to the rotor of the centrifugal force caused by the larger, the vortex center was also clear up with the increase of rotational speed, this is caused by liquid film thickness increase, with the increase of rotational speed, causing the rotor - bearing system is sensitive to external excitation force, unbalanced force higher degree, therefore, in order to improve the stability margin of rotor system, the higher the speed of the rotor dynamic balancing precision of the higher level requirements. Although (2) the numerical simulation on flow field after a certain simplification, lubrication can not fully reflect the real bearing - lubrication situation of rotor system, but the results have certain reference value, through the comparison of experiment research and numerical simulation, using numerical simulation is verified for bearing rotor system dynamic characteristics is of certain reference and reference value.

5. Summarizes

(1) This paper introduces a new type of CFD fluid-structure coupling calculation method to solve the rigid body, and connecting with the grid reconstruction technology of water lubricated bearing - the coupling rotor system is calculated and analyzed under different rotational speed again, journal eddy amplitude with the changing rule of the vortex center, and the object, and the results were analyzed.

(2) this paper introduces a self-developed design of water lubricated bearing test rig, and has carried on the experimental research, the research object is mainly measured the stable operating condition of spindle axis neck vortex trajectory, bearing circumferential stress distribution and the change of the pressure pulsation.

(3) Sliding bearing circumferential pressure distribution and pressure fluctuation analysis, low speed, due to the low viscosity of water, water film pressure distribution is discontinuous and size is low, so the monitoring of pressure curve is not stable. The increase of rotational speed makes the dynamic pressure lubrication effect increased, high expansion and the increase of the pressure amplitude makes water film bearing

(4) Load effect also strengthen, pressure distribution and pressure curves are stable, so the whole bearing rotor system gradually stabilized. Rotational speed of bearing axial pressure pulsation amplitude impact is not big, because of high pressure pulsation amplitude mainly concentrated in the low frequency region.

(5) Under the comprehensive CFD numerical simulation results and the experimental test results of axial vortex trajectory can be concluded that the journal eddy amplitude increased with the increase of rotational speed increases, this is due to the rotor of the centrifugal force caused by the larger, the vortex center was also clear up with the increase of rotational speed, this is caused by liquid film thickness increase, with the increase of rotational speed, causing the rotor - bearing system is sensitive to external excitation force, unbalanced force higher degree, therefore, in order to improve the stability margin of rotor system, the higher the speed of the rotor dynamic balancing precision of the higher level requirements. Results show that the combined method of numerical simulation and experimental research for bearing rotor system dynamic characteristics is of certain reference value.

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