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NUMERICAL AND EXPERIMENTAL RESEARCH ON PERIODIC SOLUTION STABILITY OF INCLINED ROTOR JOURNAL BEARING SYSTEM

Dashuai Qian, Zhansheng Liu, Jiajia Yan, Liquan Sun and Yongliang Wang
School of Energy Science and Engineering, Harbin Institute of Technology
Harbin, Heilongjiang, People's Republic of China

ABSTRACT

Rotor bearing systems on ships usually work in inclined states when ships are swaying in wave and wind. The inclined status will affect the lubricant condition of journal bearing and bring about changes of the dynamic characteristics of the rotor system. To study the periodic solution stability of inclined rotor journal bearing system, Capone's short bearing model is employed to describe the journal bearing support properties. Considering the inclination induced change of bearing radial load, the dynamic equation of inclined rotor system is established by using finite element method. The periodic solution stability is discussed based on bifurcation and response analysis. With the increase of rotating speed, instability of period-1 motion happens and oil whirl occurs. The motion then develops into a kind of quasi-periodic motion. Two special cases of inclined rotor system, the horizontal and the vertical cases, are compared and discussed. Both of the numerical and the experimental results show that the periodic solution unstable threshold decreases with the increase of rotor inclination angle. At last, some experimental results about influences of experiments conditions on rotor system stability are given.

INTRODUCTION

Horizontal and vertical rotor bearing systems are the two common cases in rotordynamic research and engineering applications. But in ship equipment, rotor axis usually diverge from its designed direction and sways with its foundations when the ships are swaying in wind and wave. The inclined status is a kind of off-design condition because rotor system should be naturally in horizontal or vertical states.

Few studies focus on the inclined states and the subsequent change of dynamics and stability of rotor bearing system. Research on rotor axis direction induced problems mainly concentrate on comparisons of horizontal and vertical rotor systems. Kelzon et al. [1] studied the stability of a vertical rotor system based on the Korovchinskii's nonlinear oil film force model and obtained the curves of unstable threshold. The authors stated that appropriate supports can improve the stability noticeably. White et al. [2] studied a vertical rotor tilting pad bearing system and concluded that the bearing clearance is significant to rotor stability. The results from nonlinear analyses with a small clearance are coinciding with test results. Kirk et al. [3,4] compared the dynamic characteristics of horizontal and vertical rotor systems and presented an analytic method to predict the unstable areas. There are more than one unstable threshold speeds of vertical rotor journal bearing system and the stable and unstable areas exist alternatively. The authors stated that instability can be avoided by using appropriate damping because damping has obviously effect on the low order threshold speeds. Cavalca et al. [5,6] studied the horizontal and vertical rotor journal bearing systems based on Capone's nonlinear bearing model and discussed the effects of damping and rotor parameters. The numerical and experimental results show that the sub-synchronous motion is more evident in vertical rotor. Peng et al. [7] studied the stability problems of a journal-bearing-supported vertical rotor based on linearized oil force model. An analytic method was given and the effects of parameters on unstable areas were discussed based on a generalized Routh-Hurwitz stability criterion. These comparison research indicate that the lateral vibration and stability characteristics of vertical and horizontal rotor systems differ greatly.

Although the differences in dynamic characteristics of horizontal and vertical rotors are apparent, a vertical rotor does not have a especial different dynamic model. In existing papers and approaches, the only difference is that no gravity term exist in the dynamic equation of vertical rotor system. All the research methodology and tools, including the lubrication theory and various nonlinear oil film force models, can apply to the horizontal and vertical rotor systems indistinguishably. Thus, the gravity, and more generally, the radial load on journal bearing can affect the dynamic characteristics of rotor systems. Different radial load brings different lubricating conditions and support status in journal bearing. For a inclined rotor journal bearing system, the most obvious inclination induced change is the vary of radial load because the gravity direction does not incline with the rotor foundation. The gravity component perpendicular to the axis line and the journal bearing load changed with the inclination. In this case, the inclined rotor system may have different stability characteristics.

The journal bearing related rotor instability generally refers to oil whirl and oil whip phenomena. The oil whirl and whip are typical self-excited oscillations caused by the interaction between rotor journal and oil film. The small gap circumferential flow in bearing clearance is the ultimate cause of oil whirl and whip. Anything affecting the circumferential flow can affect the oil whirl or whip, and ultimately, affect the stability of rotor system. Based on experimental results, Hashimoto and Ochiai [8] verified that integrated circumferential flow is the necessary condition of oil whirl and oil whip. Muszynska has done plenty of works [9] to discuss the oil whirl and whip phenomena. She suggested a parameter "λ" to study the related problems. "λ" represents the circumferential average velocity ratio and bears on the journal eccentricity ratio. She explained that with a higher eccentricity ratio, the flow pattern inside the clearance is modified from a dominantly circumferential type to a dominantly axial type, accompanied with increasing oil film stiffness and decreasing average velocity ratio "λ". The ultimate result of enlarging the eccentricity ratio is the delay of oil whirl and the improvement of stability.

In rotor journal bearing system, the period-1 motion usually refers to the unbalance-excited synchronous motion in accordance with the pure rotating motion of the shaft. The pure rotating motion and low-amplitude synchronous motion is desired in rotor operations. But in real machines, the synchronous motion is usually accompanied with other undesired motion modes. There are two kinds of non-synchronous motions, the sub-synchronous motion and the super-synchronous motion. The sub-synchronous motion frequency is smaller than the synchronous motion. A typical sub-synchronous motion is oil whirl induced by journal bearing. On the contrary, the super-synchronous motion frequency is bigger than the synchronous motion. The periodic solution stability discussed in this paper refers to the stability of the period-1 motion, i.e., the pure synchronous motion. The rotor system is periodically stable when there is no non-synchronous motion and only the period-1 motion exists. If the sub-synchronous or the super-synchronous motion emerge, we state that the period-1 motion is unstable and the changing speed is called "unstable threshold".

To study the periodic solution stability of inclined rotor system, numerical and experimental research are performed in this paper. The dynamic equation is established based on nonlinear oil film force model. Based on the analyses of rotor dynamic responses, the periodic motion features are detected and the unstable thresholds are obtained. Two special cases of inclined rotor, the horizontal rotor and vertical rotor are compared and the effects of parameters are discussed. Then experimental results are presented. The experimental unstable thresholds are obtained from the waterfall plots, the frequency spectra and the journal orbits. At last, some experimental results about effects of oil orifice positions and oil pressure are given.

NOMENCLATURE

θ	Inclination angle
$[M]$	Mass matrix
$[C]$	Damping matrix
$[J]$	Gyroscopic matrix
$[K]$	Stiffness matrix
Ω	Rotating speed
$\{F_{un}\}$	Unbalance force vector
$\{F_{oil}\}$	Oil film force vector
$\{F_r\}$	Reaction force vector of ball bearing
$\{G\}$	Gravity vector
x, y	Displacement coordinates
$x_i, y_i, \theta_{xi}, \theta_{yi}$	Displacements at node i
$\{u\}$	Displacement vector
f_x, f_y	Dimensionless oil film force
U, V, S	Intermediate variables in Capone's model
α	Attitude angle of journal bearing
k_{rx}, k_{ry}	Main stiffness of ball bearing
f_{rx}, f_{ry}	Reaction force of ball bearing
$\Delta x, \Delta y$	Displacement increment
β, γ	Algorithm parameters of Newmark method
C	Radial clearance of journal bearing
L	Length of journal bearing
D	Journal diameter
RR	Speed rising rate
DR	Speed drop rate
P	Oil pressure

THE INCLINED ROTOR BEARING SYSTEM AND DYNAMIC EQUATION

In this paper we study a Bently RK4 rotor system installed on a rotatable plate. The inclination angle is defined as the intersection angle of the rotor axis and the horizontal line, as depicted in **Figure 1**. The rotor system contains a shaft with two disks, a thrust ball bearing and a cylindrical journal bearing. The shaft diameter is 10 mm and the length is 460 mm.

In this paper, we mainly focus on the performance of journal bearing and do not consider the stability characteristics induced by ball bearing. When the rotor is inclined, the most significant difference from the usual horizontal or vertical state is the dependence of journal bearing radial load on inclination angle. The radial load of journal bearing can greatly affect the dynamic response and the stability characteristics. Thus, the inclination states can be expressed as variational gravity at each node in the dynamic equation.

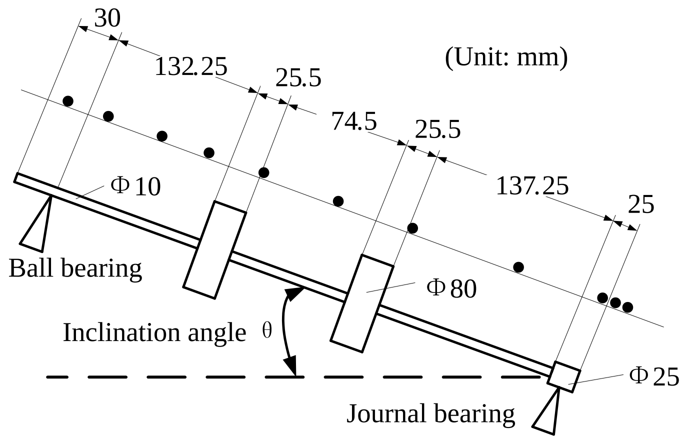


Figure 1 Schematic of inclined rotor system

Based on finite element method, the dynamic model of rotor system can be given and the dynamic equation can be written as

$$[\mathbf{M}]\{\ddot{u}\}+([\mathbf{C}]+\Omega[\mathbf{J}])\{\dot{u}\}+[\mathbf{K}]\{u\} \\ =\{F_{un}\}+\{F_{oil}\}+\{F_r\}+\{G\}cos\theta \tag{1}$$

Here, $[\mathbf{M}],[\mathbf{C}],[\mathbf{J}],[\mathbf{K}]$ are the mass matrix, damping matrix, gyroscopic matrix and stiffness matrix, respectively. The mass matrix, damping matrix, and stiffness matrix are symmetric band matrices, while the gyroscopic matrix is anti-symmetric. rotor system On the right hand, $\{F_{un}\},\{G\}$ are the unbalance force vector and the gravity vector. $\{F_{oil}\}$ and $\{F_r\}$are the force vectors containing oil film force and ball bearing reactive force. $\{u\}$ is the displacement vector. The whole rotor is divided into 10 elements modeled by Euler-Bernoulli beam, leading to totally 11 nodes. Each node has 4 degrees of freedom, including two lateral displacements x_i, y_i, and two bending angle displacements θ_{xi},θ_{yi}.

The nonlinear oil film force is applied at the node 2, corresponding to the midpoint position of journal bearing. Capone's short bearing model [10] is employed to describe the nonlinear force on the shaft journal. In Capone's model, the dimensionless oil film force can be expressed as

$$\begin{Bmatrix} f_x \\ f_y \end{Bmatrix}=-\frac{\sqrt{(x-2\dot{y})^2+(y+2\dot{x})^2}}{1-x^2-y^2}\begin{Bmatrix} 3xV-Usin\alpha-2Scos\alpha \\ 3xV+Usin\alpha-2Scos\alpha \end{Bmatrix} \tag{2}$$

Where

$$U=U(x,y,\alpha) \\ =\frac{2}{\sqrt{1-x^2-y^2}}\left[\frac{\pi}{2}+\arctan\frac{y\cos\alpha-x\sin\alpha}{\sqrt{1-x^2-y^2}}\right] \tag{3}$$

$$V=V(x,y,\alpha)=\frac{2+(ycos\alpha-xsin\alpha)U(x,y,\alpha)}{1-x^2-y^2} \tag{4}$$

$$S=S(x,y,\alpha)=\frac{xcos\alpha+ysin\alpha}{1-(xcos\alpha+ysin\alpha)^2} \tag{5}$$

The reactive force of ball bearing is given at node 10 in a simplified linear expression, as shown in equation (6). The dynamic stiffness of ball bearing is not considered here and only the static main stiffness coefficients k_{rx} and k_{ry} are employed. k_{rx} and k_{ry} are also set to be the same for simplify.

$$\begin{cases} f_{rx}=k_{rx}\Delta x \\ f_{ry}=k_{ry}\Delta y \end{cases} \quad \text{with } k_{rx}=k_{ry} \tag{6}$$

The Newmark method is used to integrate the equation (1), with the algorithm parameters $\beta=0.5$ and $\gamma=0.25$.

NUMERICAL REASEARCH

In this paper, the stability characteristics of inclined rotor system are discussed based on the dynamic responses. Periodic motion and quasi-periodic motion of inclined rotor system can be detected from Poincare maps, frequency spectra and map bifurcation diagrams. Poincare map [11,12] is a useful tool to reduce the stability analysis of periodic trajectories in phase space to the analysis of the behavior of map points in a low order state space. The map points are the intersections of the periodic trajectories and the selected Poincare section transversal to the orbits. If the rotor has a periodic motion, the map will consist of a set of repeating points. To put it simply, there are N map points in the Poincare map of period-N motion. Change of the number and topology of map points usually means a change of motion mode.

Two methods can be used to calculate the map points of a non-autonomous system (e.g., a forced nonlinear oscillator, or a unbalance-excited rotor system), the definition method and the stroboscopic method. The definition method is accurate but sometime complicated because the trajectories may be irregular and it is not easy to select a suitable Poincare section in the phase space, especially for high order systems. Unlike the calculating of spatial positions in the definition method, the relation of between the trajectory positions and the time course, i.e., the periodicity of the motion, is employed in the stroboscopic method and the "map points" can be obtained directly from the time waveform results. A periodic motion can be anticipated because the inclined rotor system is a forced non-autonomous system. Thus, the trajectories will return to the suitably-selected Poincare section by a time interval corresponding to the period of unbalance excitation, i.e., the rotating period of the inclined rotor. So we can extract values of state parameters x, $\dot{x}$ from the steady-state response at the time points $t_0, t_0+T, ..., t_0+NT$. All the extracted points are plotted in the $x-\dot{x}$ plane and the Poincare map are obtained.

The numerical results of case 0 °at the journal bearing node 2 are given in **Figure 2** to **Figure 5**. Base frequency (69.16Hz) is the only component existing in the frequency spectra at 4150 rpm (**Figure 2**(a)). Correspondingly, the time waveform is simple harmonic and displacement and velocity values at the time points $t_0, t_0+T, ..., t_0+10T$ are almost the same (**Figure 3**(a)). Thus, the map points in the $x-\dot{x}$ plane at these time points coincide with each other and only one map point exists in the Poincare map (see **Figure 4**(a)). This means the rotor is experiencing a period-1 motion, i.e., the synchronous motion. This period-1 motion at 4150 rpm is stable because no other motion mode exist. In the time waveforms in **Figure 3**, the horizontal dashed line indicates the value line of the first or the second selected map point.

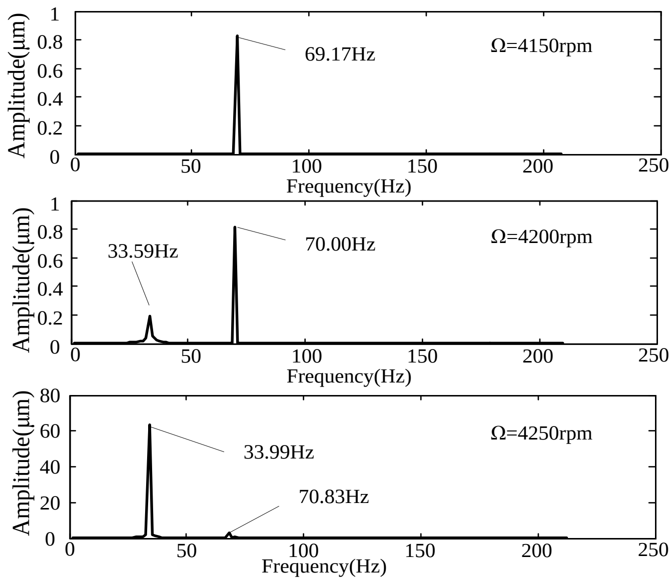
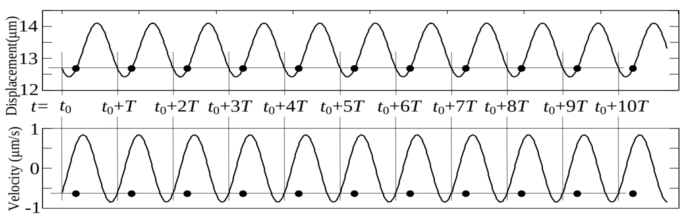
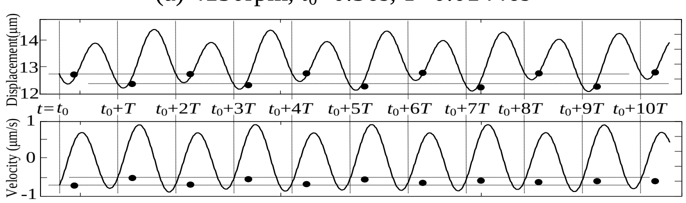


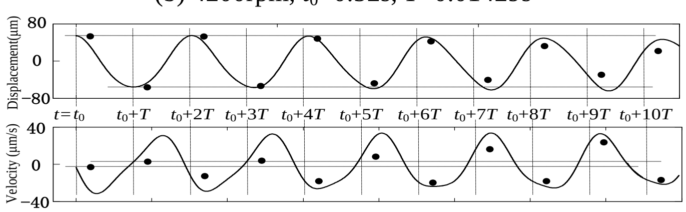
Figure 2 Frequency spectra of case 0 °at node 2



(a) 4150rpm, t_0=0.56s, T=0.01446s

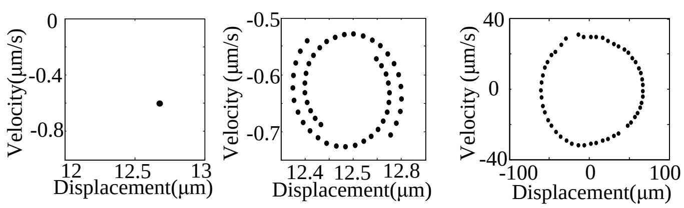


(b) 4200rpm, t_0=0.52s, T=0.01429s



(c) 4250rpm, t_0=0.52s, T=0.01412s

Figure 3 time waveform of case 0 °at node 2 and selection of map points by stroboscopic method



(a) 4150rpm (b) 4200rpm (c) 4250rpm

Figure 4 Poincare maps of case 0 °at node 2

Then oil whirl emerge at 4200rpm (**Figure 2**(b)). The sub-synchronous frequency (33.59Hz) is about 0.48 times of the base frequency (70.00Hz). Two aliquant frequencies dominate the rotor motion and the waveform is no longer a simple harmonic curves (**Figure 3**(b)). At the time points $t_0, t_0+T, ..., t_0+10T$, the selected points do not coincide in the Poincare map (**Figure 4**(b)). The map points form two intertwined but disjoint spiral lines. The motion at 4200 rpm is a kind of gradually changed motion from period-1 to quasi-periodic motion because the synchronous and the sub-synchronous motion dominate the motion together. The mapping points of 4250 rpm form a closed loop in Poincare map (see **Figure 4**(c)). This means the rotor motion becomes a quasi-periodic motion. Here the oil whirl dominates the rotor motion and the base frequency (70.83Hz) can be neglected compared with the oil whirl frequency (33.99Hz). The frequency ratio is 0.48, indicating that the whirl here is a "half frequency whirl".

The motion between 4150 rpm and 4250 rpm is transition motion from period-1 motion to quasi-period motion. The period-1 motion becomes unstable when the oil whirl occurs because the synchronous motion coexist with the sub-synchronous motion. The Poincare map at every rotating speed between 4150 rpm and 4250 rpm contains two intertwined but disjoint spiral lines as **Figure 4**(b). If all these spiral lines are plotted together, two intertwined lines originating from the period-1 motion point at 4150 rpm will extend as close as the closed mapping circle at 4250 rpm.

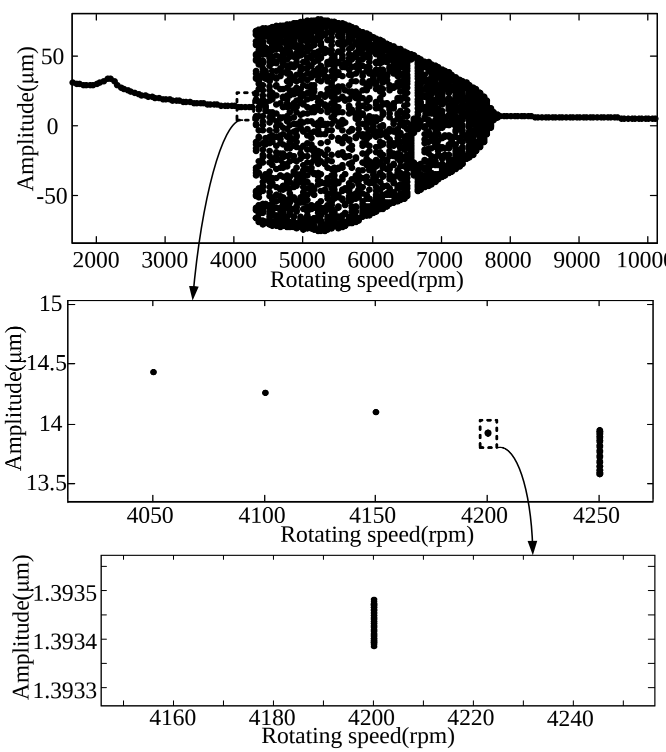


Figure 5 Map bifurcation diagram of case 0 °at node 2

Bifurcation of Poincare map can also reveal the aforementioned transition from period-1 motion to quasi-periodic motion. If we plot the map points according to the displacement values at each rotating speed, the map bifurcation diagram will be obtained, as **Figure 5** shown. The rotor system experiences synchronous motion before 4150 rpm. The map points coincide and only one bifurcation point exists at each rotating speed. Then sub-synchronous motion occurs and the map points do not coincide at 4200 rpm. More than one bifurcation points exists in the bifurcation diagram although the coordinates of the bifurcation points differ little. Thus, bifurcation occurs at 4200 rpm and the period-1 motion becomes unstable. It can also be seen that the unstable area ends at 7900 rpm and the period-1 motion recurs.

When the inclination angle increases, the characteristics of periodic motions and the changes can be obtained from such similar results and analyses. However, differences exist in the morphology of result figures. In bifurcation diagrams of large inclination angle cases (from 50° to 90°), the bifurcation points form two branches of bifurcation point sets after the bifurcation of period-1 motion and then get together. (See the example of case 90° in **Figure 6**). Another important feature of **Figure 6** is that high order whirl occurs at 9600 rpm. The high order whirl does not occur in the horizontal rotor system. This reflects the poor stability of vertical rotor system.

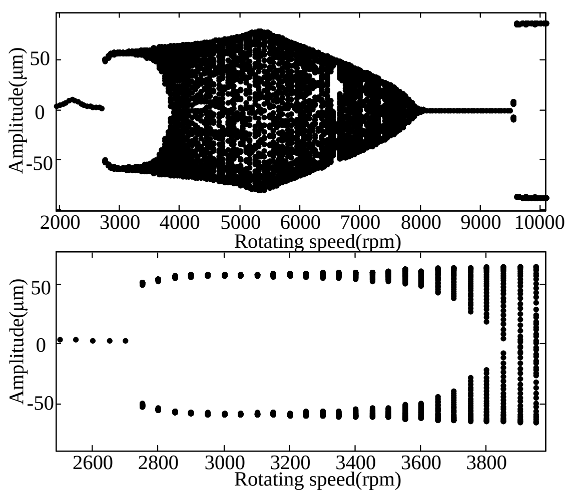


Figure 6 Map bifurcation diagram of case 90°

Oil whip happens only at the rotating speed which is twice the first critical speed, while oil whirl is not limited by this rule. Smooth transitions from oil whirl to oil whip exist in all the inclination cases, and there is no obvious different feature between Poincare maps of oil whirl and oil whip.

The amplitude curves can also reflect some differences of different inclination cases. In **Figure 7**, amplitude curves of all the ten inclined cases are plotted together. Critical speeds in different inclination states are almost the same because the shaft stiffness is smaller than the oil film stiffness although the increase of inclination angle can bring about a smaller oil film stiffness of journal bearing. We can see that the vibration amplitude increases sharply after the occurrence of oil whirl. This means the transition from period-1 motion to quasi-periodic motion usually comes with a soaring of vibration which is undesirable in rotor system operations. But the rotating speed where soaring happens does not always agree accurately with the speed where the period-1 motion transits to other motions. Unlike the unstable threshold describing a transition of periodic motion, the "soaring speed" describes a practical amplitude instability, which can be regarded as "soaring threshold". The soaring threshold can be the same as the unstable threshold, or a little later after the unstable threshold. Based on the above analyses, we identify that the unstable threshold of case 0° is 4200 rpm and the soaring threshold is 4250 rpm.

With the increase of inclination angle, the soaring threshold decreases. This is consistent with the change of unstable threshold. However, the whirl or whip amplitude in all the inclination cases are identical. This depicted that different inclined states and radial load have little effect on amplitude in the unstable area.

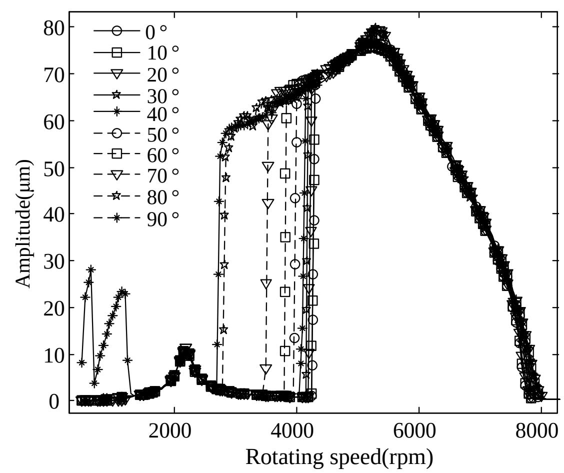
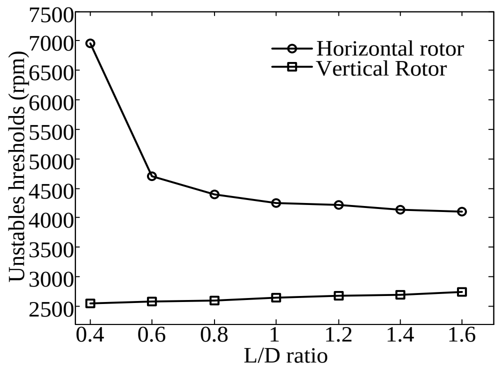


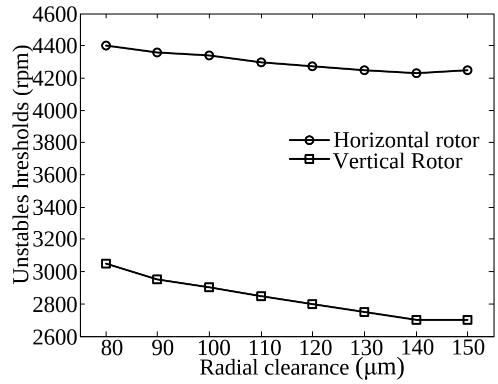
Figure 7 Amplitude curves of different inclined cases at node 2

COMPARISONS OF HORIZONTAL AND VERTICAL ROTOR BEARING SYSTEMS-NUMERICAL RESULTS

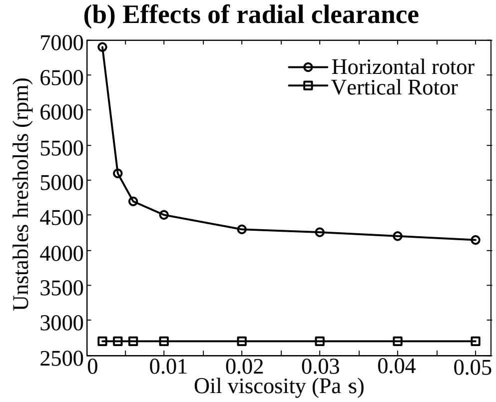
Horizontal and vertical rotors are the two special cases of inclined rotor and rotor bearing system is usually designed and arranged in horizontal or vertical status. Comparison research of horizontal and vertical rotor systems have concluded that the stability characteristics differ greatly [3-5]. The main reason of the differences is that rotor gravity takes no effect on the radial load of journal bearing in vertical rotor system. With a light load journal bearing, vertical rotor is much more sensitive to disturbances and its unstable threshold is much less than that of horizontal rotor. At the same time, effects of parameters on stability are different in horizontal and vertical cases. The effects of length to diameter ratio (L/D), radial clearance, oil viscosity and unbalance are given in **Figure 8**.



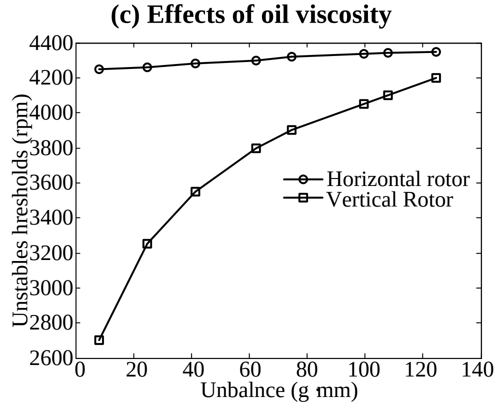
(a) Effects of L/D ratio



(b) Effects of radial clearance



(c) Effects of oil viscosity



(d) Effects of rotor unbalance

Figure 8 Effects of parameters on stability thresholds of horizontal and vertical rotor systems: numerical results

The L/D ratio has a noticeable effect on the unstable threshold of horizontal rotor system, especially when the L/D ratio is less than 0.6, as shown in **Figure 8(a)**. In comparison, the L/D ratio has less effect on the unstable threshold of vertical rotor. The difference mainly results from the fact that a larger L/D ratio or a long bearing leads to abatement of radial load on unit length bearing. But for a vertical rotor system, the radial load, such as gravity, is not directly undertaken by journal bearing. When the L/D ratio or bearing length increases, the radial load of vertical bearing changes little and the unstable threshold does not decrease as in horizontal rotor system. Instead, the unstable threshold may increases with an increasing L/D ratio.

Radial clearance of journal bearing may be the most important parameter in rotor design and analysis because its significant influences on bearing performance. A small radial clearance can provide large stiffness and improve the critical speeds. A small radial clearance can also block the development of circumferential flow in the bearing clearance, and thus lead to a better stability. The effect of radial clearance on circumferential flow is almost the same in horizontal and vertical rotor systems. The unstable thresholds of rotor system decrease when the radial clearance increases (**Figure 8(b)**).

Oil viscosity has a noticeable effect on the threshold of horizontal rotor system as the L/D ratio, as shown in **Figure 8(c)**. But for a vertical rotor system, the unstable threshold is almost unaffected by the change of oil viscosity. With a larger viscosity, the friction between oil and journal can provide bigger tangential friction force and the circumferential flow can develop easily. For a horizontal rotor, larger oil viscosity leads to smaller unstable threshold. Note that the formation of friction force needs normal force on the contact interface. In vertical rotor bearing system, the friction force between journal and oil is much smaller due to the light journal bearing. The change of oil viscosity cannot significantly bring about the change of friction force. Thus, oil viscosity has little effect on unstable threshold of vertical rotor system.

Rotor Unbalance can also provide a radial load to the journal bearing. But the radial load is alternate and increases with rotating speed. A bigger unbalance results in a better stability and both of the unstable thresholds of horizontal and vertical rotor systems increase when unbalance increases (**Figure 8(d)**). The rotor unbalance has greater impact on vertical rotor unstable threshold than that of horizontal rotor.

EXPERIMENTAL RESEARCH

The inclined Rotor bearing test rig is shown in **Figure 9**. The Bently RK4 rotor system is installed on a rotatable plate which can be fixed to brackets at different inclination angles. The rotor system parameters and experiment conditions are listed in Table 1.

Due to the restriction of journal bearing structure, we cannot measure the journal vibration directly. Instead, we measure the shaft vibration beside the bearing end cover. So the

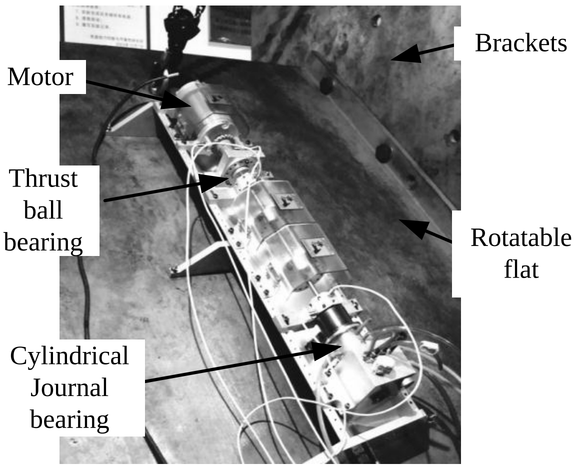


Figure 9 The inclined Rotor bearing test rig

Table 1 Parameter settings in experiments

Parameter	Symbol, Unit	
Radial clearance	C, μm	140
Journal diameter	L, mm	25
Bearing length	D, mm	25
Lubricating oil grade		L-TSA32
Speed rising rate	RR, (rpm)/min	2000
Speed drop rate	DR, (rpm)/min	2000
Oil supply pressure	P, MPa	0.16

vibration amplitude given next may be larger than the bearing clearance value.

In experimental research, the periodic motions of rotor system are also analyzed by oil whirl or whip phenomena which can be distinguished from the journal orbits, the frequency spectra and the waterfall plots. The typical experimental results are presented in **Figure 10**~**Figure 12**.

In **Figure 10**, the waterfall plots describe the base frequency spectra and the sub-synchronous frequency spectra during the speed rising process. Before 4142 rpm, only the base frequency motion exists in frequency spectrum and the rotor is in period-1 motion. Then the oil whirl occurs and dominates the motion rapidly. The frequency spectrum depicts a sub-synchronous motion at 4142 rpm. The frequency ratio of the sub-synchronous motion to rotating motion is about 0.46875 (**Figure 11(a)**). This aliquant frequency ratio represents a quasi-periodic motion, which is coinciding with numerical results.

Changes of the periodic motion can also be demonstrated by the journal orbits at different rotating speeds. In **Figure 12** we present journal orbits in a time interval of four rotating periods at different rotating speeds. At 4077 rpm, the orbit includes 4 circles and the rotor motion is the unbalance induced synchronous motion, i.e. the period-1 motion. But when the oil whirl occurs, the period-1 motion becomes unstable and another motion mode progresses. At 4142 rpm, the journal orbit also includes 4 circles but the orbit shows an “inner eight” shape, implying a change of motion mode.

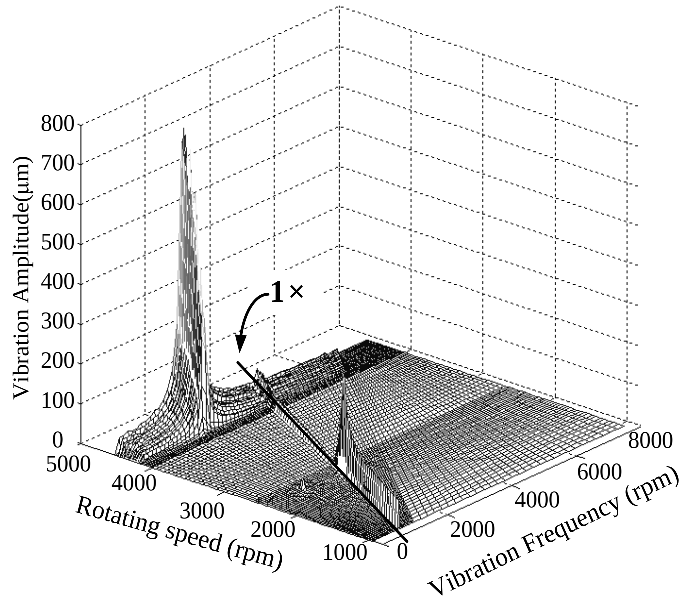
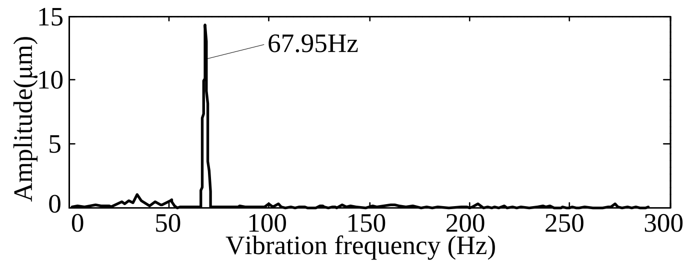
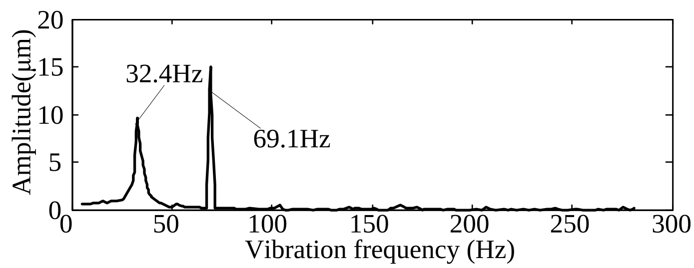


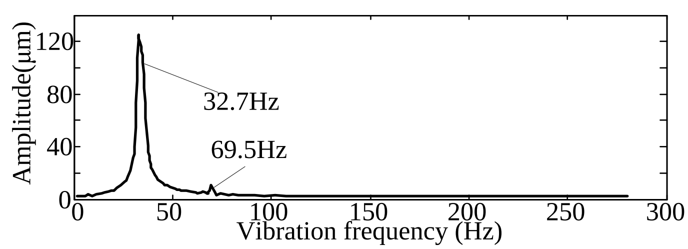
Figure 10 Experimental waterfall plots of case 0 °



(a) Frequency spectra of 4077 rpm



(b) Frequency spectra of 4142 rpm



(c) Frequency spectra of 4171 rpm

Figure 11 Experimental frequency spectra at different rotating speeds

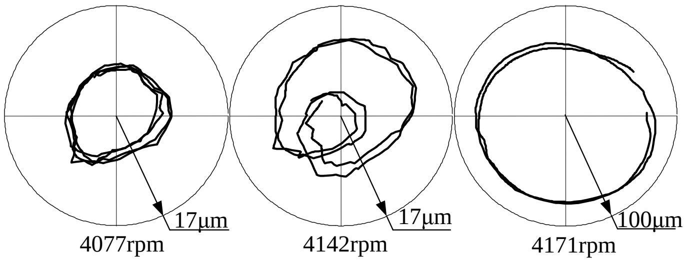


Figure 12 Journal orbits at different rotating speeds

As shown in numerical results, the soaring threshold is usually later than the unstable threshold. This is more apparent in experimental results. The vibration amplitude (peak-peak value) at the unstable threshold (4142 rpm) is 30.9μm. With the increase of rotating speed, the oil whirl developments and the vibration amplitude accretes. At about 4160 rpm, the vibration begins to soar and the overall amplitude (peak-peak value) at 4171 rpm is 211.9μm. So the soaring threshold is 4160 rpm.

Figure 13 presents the numerical and experimental unstable and soaring thresholds of each inclination case. Despite the differences, the experimental results validate the basic regularity that the unstable threshold decreases when the inclination angle increases. In the cases from 0° to 30°, the experimental unstable thresholds agree well with the numerical unstable thresholds. But for cases from 40° to 90°, there are differences between the experimental and numerical results. In numerical results, the difference between unstable and soaring threshold keeps less than 100 rpm in each inclination case. But in experiments, the difference is much larger, especially in large inclination angle cases. The difference in case 90° can reach 360 rpm.

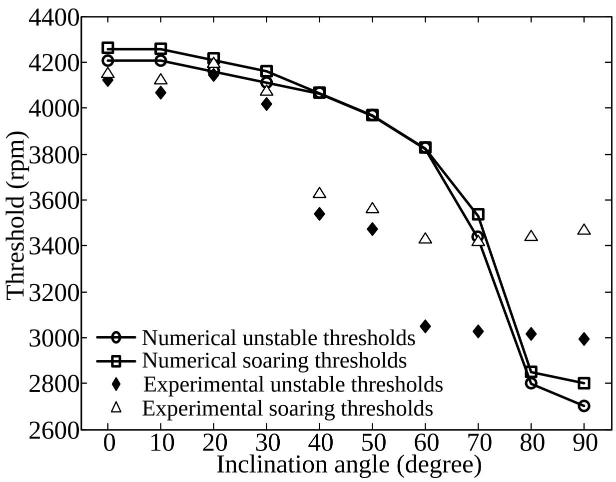


Figure 13 Comparisons of numerical and experimental results about soaring thresholds and unstable thresholds

These differences between numerical and experimental results mainly come from the ideal hypotheses and simplifies in the dynamic model and the experimental errors. There are several hypotheses in the short bearing model employed and the journal deflection in the bearing is not considered. In the inclined rotor system, the journal deflection and the induced misalignment are inevitable and can bring significant influences on journal bearing performance and thus the rotor system stability. In experiments, many factors, such as the rubbing of journal, the misalignments from bearing and shaft coupling and the uneven hold of thrust ball bearing, can affect the stability of rotor system. Disturbance of the oil supply conditions call either be neglected. In our experiments, the oil pressure is higher and the oil is supplied to the bearing through one or more orifices. The throttle effect of the orifices may affect the normal oil film shape and the stability characteristics. With a higher oil pressure, the oil from the orifices can form oil film even when the rotor is not rotating. In this case, the journal bearing seems like a hybrid bearing which have hydrodynamic and hydrostatic effects at the same time. The hybrid bearing has better stability than a pure hydrodynamic bearing.

Here we also present some experiments results about influences of experimental conditions on stability of rotor system. These experimental conditions refer to the position of oil supply orifice on journal bearing and the oil pressure.

The lubricating oil film in cylindrical journal bearing clearance contains a convergence wedge and a divergence wedge. If oil is supplied through different orifices corresponding to different positions of the oil film structure, the circumferential flow in bearing clearance will be strengthened or weakened, and oil whirl or whip will be affected. The cross section of the journal bearing employed in experiments is shown in **Figure 14**. There are four oil supply orifices symmetrically positioned along the journal circumference, numbered by 1 to 4. When the rotor system is positioned as a horizontal rotor, the journal bearing and the orifices' positions is actually set as **Figure 14**.The lubricating oil is supplied to the bearing by a pump.

Figure 15 presents the experimental amplitude curves of the horizontal case with different orifice combinations. The arrows on the amplitude curves signify speed rising operation. When the orifices are used separately, the stability characteristics differ greatly. When No.3 or No.4 orifice is used, the soaring threshold is larger than that when No.1 or No.2 orifice is used. This can be explained by the circumferential flow theory. **Figure 14** also presents the velocity directions of circumferential flow by red arrows and the inlet velocity of oil from the orifices by black arrows. Due to the special oil film shape and the whirl direction of the journal, the circumferential flow velocity direction at the inlet position is not perpendicular to velocity direction of the orifice oil. Besides the pressure component perpendicular to the journal surface, the inlet oil from No.1 or No.2 also provides a tangential velocity component on the similar direction of the circumferential flow. This promotes the full development of circumferential flow and accelerates the occurrence of oil whirl, thus lead to a small threshold. On the contrary, the inlet oil from the No.3 or No.4 orifice has a tangential velocity component opposite to the circumferential flow direction and delayed the occurrence of oil whirl. The delay effect of No.3 or No.4 orifices is similar to the anti-swirl technique [9].

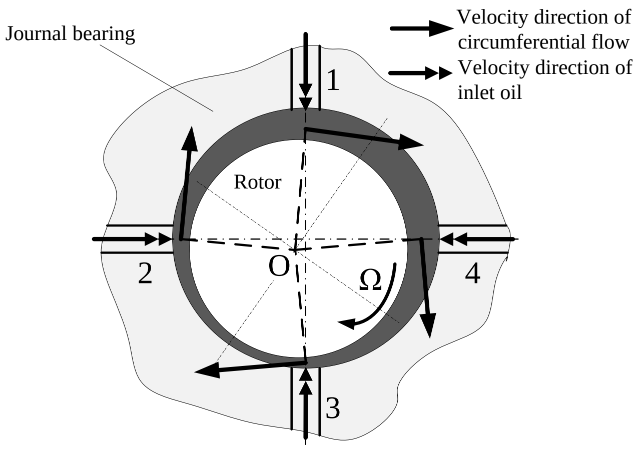


Figure 14 Oil orifice positions in journal bearing and velocity directions of circumferential flow in bearing clearance and oil flow in orifices

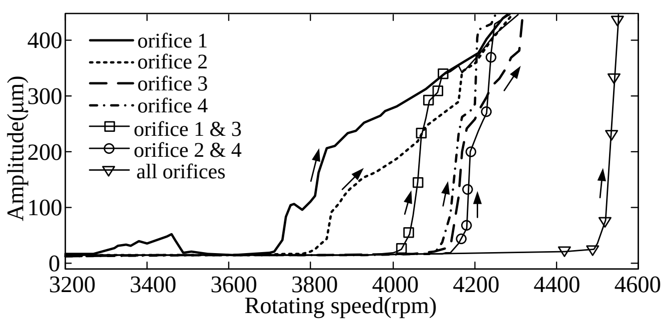


Figure 15 Experimental amplitude curves of horizontal rotor with different orifices

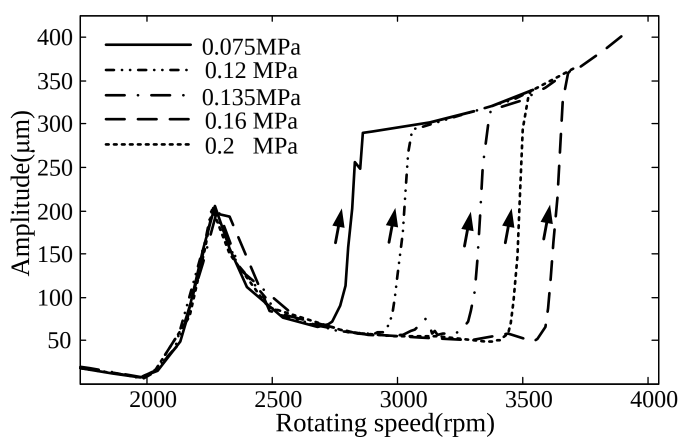


Figure 16 Experimental amplitude curves of vertical rotor under different oil pressure

Figure 16 presents the effects of oil pressure on rotor stability. The lubricating oil is pumped to the bearing through the No.3 orifice. The soaring threshold increases when the oil pressure increases from 0.075MPa to 0.16MPa. We cannot give a conclusion that high oil pressure leads to better stability because the soaring threshold decrease when the oil pressure continue increasing to 0.2MPa. This indicates that there is an optimal oil pressure which can lead to a biggest unstable threshold. Muszynska has studied the effects of oil pressure on unstable threshold in [10]. She concluded that a higher pressure cause an increase of the oil film radial stiffness which can lead to an increase of unstable threshold. But our experiments indicate that when the oil pressure is large enough, the threshold may decrease with an increasing pressure. The oil pressure in [9] (2 to 6psi, about 0.01379 to 0.041369 MPa) is much less than that in our experiments.

CONCLUSIONS

Numerical and experimental research were performed to investigate the periodic solution stability of inclined rotor journal bearing system. In the dynamic model, the inclination induced load change of journal bearing is considered. The transition from period-1 motion to quasi-periodic motion and the unstable threshold and the soaring threshold can be identified from the response results, including the Poincare maps, map bifurcation diagrams, and frequency spectra of each inclination case.

The stability characteristics of rotor bearing system are different in different inclined cases. The differences mainly result from the change of support status of journal bearing. Although we only considered the change of the radial load on the journal bearing, the differences between different inclined cases are obvious.

Due to the difference in load conditions, the effects of parameters are different in horizontal and vertical rotor systems. While having little effect on the unstable threshold of vertical rotor system, the L/D ratio and oil viscosity have noticeable effect on unstable threshold of horizontal rotor. But the vertical rotor is more sensitive to rotor unbalance. The radial clearance has significant impact on both of the horizontal and vertical rotors.

The experiments validate the numerical results and the transitions of periodic motions are observed in the test results. The differences between numerical and experimental results mainly results from the ideal hypotheses and the experimental errors. A more accurate oil force model considering the journal defection and misalignment are needed in the future work.

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