Study on critical speed of three-watt tilting pad bearing-rotor system

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Abstract. By establishing the model of three-watt tilting tile bearing and rotor system and studying the critical speed of three-watt tilting tile bearing and rotor system, the factors affecting its critical speed are obtained. The oil film stiffness coefficient and damping coefficient of sliding bearing and the mass of the counterweight plate added in the mixed bearing supporting rotor system all have a certain influence on the overall critical speed of the system. In this paper, a rotor model was established using DyRobes rotor dynamics analysis software. The critical rotational speed of the three-watt tilting tile bearing/rotor system was investigated by means of the oil film stiffness of the three-watt tilting tile bearing at different rotational speeds and changing the weight of the counterweight plate.

1. Introduction
In the study of rotor dynamics, the influence of bearings on the system must be considered. The oil film of sliding bearings has the advantages of bearing load, reducing friction and reducing vibration etc. which is widely used[1]. The dynamic performance of rotating machinery supported by sliding bearings is jointly affected by the non-linearity of bearing oil film and rotor system[2]. The nonlinear phenomenon of the rotor system is prominent when the rotating machinery works in the range of the first critical speed, and it is easy to produce large vibration phenomenon in engineering practice. Ma Wensheng found that there was a critical speed when studying the resulting oil film pressure characteristics, and when the speed was lower than a certain critical value, the critical speed had a greater impact on the maximum oil film pressure[3]. Xie Youbai studied the influence of nonlinear oil film force and bearing external elastic damping on the vibration characteristics of hydrodynamic sliding bearing rotor system[4].

According to the standard of correlator, it is required that there should be a certain avoidance rate between the running speed and the critical speed, and the calculation of the critical speed has become the key problem in the design of rotating machinery. There are many factors that affect the critical speed, such as the structure, size, material and supporting stiffness of the rotor. The early research mainly focused on the analysis of the geometric structure of the rotor, ignoring the influence of the support bearing on the critical speed. Therefore, it is of great engineering significance to study the critical speed of bearing-rotor system. In this paper, DyRobes[5-6] dynamics analysis software is used
to build a model that is closer to the actual model with high computational accuracy and is suitable for calculation of dynamic characteristics of complex rotor systems.

2. Dynamic characteristics of three-watt tilting pad sliding bearings

The movement mechanism of the three-watt tilting tile sliding bearing is shown in Figure 1. The tile swings freely with the change of the rotor load during operation, forming an oil wedge around the journal, and at the same time, the tile is subjected to the action of oil film force, forming a stress field. Tilting segment of the sliding bearing shaft axial uniform distribution in the bearing bracket, the ends of the bearing support each have a pair of bearing end cover, supports and end cover are in the upper and lower half fraction, tile and axis formed between a sliding contact surfaces, and within the segment of the arc radius greater than with the radius of the axis of the body, outside the arc radius less than the inner arc radius of bearing suppor. It can meet the eccentricity value of the shaft body to ensure the clearance of the oil wedge and the formation of the oil film, and it can greatly reduce the phenomenon of heating caused by the contact friction between the shaft and the bearing bush when working.

Figure 1. Movement mechanism diagram of three-watt tilting pad sliding bearing

Import the parameters of 3-watt tilt-tile sliding bearings into the DyRobes-beperf module as shown below: Bearing length is 50mm, bearing radius is 30mm, radial clearance is 0.06mm, number of bearing bush is 3, tile length is 56mm, radian is 115°, calculation speed of 3000r/min~8000r/min, increase variable of each speed is 1000rpm, the oil film pressure distribution of tilting tile sliding bearing under different speeds. As can be seen from Figure 2, the stress at the fulcrum of each tile of the radial bearing is the largest, and the oil film pressure of the lowest tile is greater than that of the rest tiles at low rotating speed. With the increase of rotating speed, the oil film pressure difference of each tile decreases.

(a) 3000, 4000, 5000 Transfer oil film pressure distribution
The minimum oil film, stiffness and damping values of three-watt tilting tile sliding bearings at different speeds are shown in Table 1 below.

| r/ (r·min⁻¹) | h min/ (mm) | Kxx/ (N/mm⁻¹) | Kyy/ (N/mm⁻¹) | Cxx/ (N·s·mm⁻¹) | Cyy/ (N·s·mm⁻¹) |
|--------------|-------------|----------------|---------------|-----------------|-----------------|
| 3000         | 4.15E-02    | 1.07E+05       | 1.47E+05      | 4.60E+02        | 5.61E+02        |
| 4000         | 4.26E-02    | 1.48E+05       | 1.87E+05      | 4.70E+02        | 5.44E+02        |
| 5000         | 4.33E-02    | 1.88E+05       | 2.28E+05      | 4.76E+02        | 5.35E+02        |
| 6000         | 4.38E-02    | 2.29E+05       | 2.68E+05      | 4.81E+02        | 5.29E+02        |
| 7000         | 4.41E-02    | 2.70E+05       | 3.09E+05      | 4.84E+02        | 5.25E+02        |
| 8000         | 4.44E-02    | 3.11E+05       | 3.50E+05      | 4.86E+02        | 5.23E+02        |

According to the data in the table, a diagram of the influence of rotational speed on the dynamic characteristics of the numerical changes of the minimum oil film, oil film, stiffness and damping of the three-watt tilt-tile sliding bearing is made, as shown in Figure 3.
It can be seen from the above figure that the minimum oil film thickness and stiffness coefficient of the bearing at different speeds increase with the increase of the speed, while the damping coefficient increases or decreases with the increase of the speed. The oil film shows a strong nonlinear characteristic, which makes the calculation accuracy of the critical speed more consistent with the actual situation.

3. Calculation of rotor critical speed

The critical speed is the characteristic speed of the main response of the bearing rotor system at resonance. The critical speed is related to the elasticity and mass distribution of the rotor. For a discrete rotor system with finite lumped masses, the number of critical speeds is equal to the number of lumped masses. For an elastic rotating system with continuous mass distribution, there are infinitely many critical speeds.

For a rotor system, the critical speed is the speed at which the dynamic deflection \( r \) reaches the maximum, expression as follow:

\[
\frac{dr}{d\Omega} = 0
\]  

By substituting the expression of dynamic deflection \( r \) into formula (1), we get another expression:

\[
\frac{dr}{d\Omega} = \frac{2\xi \Omega \left[ 2\xi^2 - 1 \right] \Omega^2 + \omega_n^2}{\omega_n^4 \left[ 1 + \left( \frac{\Omega}{\omega_n} \right)^4 + \frac{2 (2\xi^2 - 1) \Omega^2}{\omega_n^2} \right]} = 0
\]

Solution as follow:

\[
\Omega_c = \omega_n \sqrt{\frac{1}{1 - 2\xi^2}}
\]

Where \( \xi \) is the relative damping of the system, \( \xi = \frac{c}{2m\omega_n} \); \( \omega_n \) is the transverse undamped natural frequency of the system, \( \omega_n = \sqrt{\frac{k}{m}} \); \( c \) is viscous damping; \( \Omega_c \) is the critical speed.

4. Critical speed analysis of bearing rotor system

4.1. Rotor modeling

Using Dyrobes software, the model of three pad tilting pad bearing rotor system and the position of support bearing are established in Dyrobes rotor module, analyzing the rotor critical speed, as shown...
in Figure 4. Combined with theoretical analysis and experiment, the calculation of critical speed of rotor system is explored, and according to the actual situation, the parameter characteristics that affect the critical speed of rotor system are obtained.

![Figure 4. Model of three-watt tilting tile bearing-rotor system](image)

The center of the three pad tilting pad bearing in the figure is represented by node 5, and the critical speed of the rotor under the initial condition is 28086 rpm.

4.2. Influence of counterweight on critical speed of rotor system
In this paper, 10kg, 20kg, 30kg and 40kg turntables are respectively applied to the rotor system at node 8, and the influence of counterweight selection on the critical speed is analyzed.

After 10kg, 20kg, 30kg and 40kg counterweight disks were applied to the rotor system, the critical speeds of rotor system were 21645rpm, 16142rpm, 13428rpm and 11741rpm respectively. According to the critical speed data of different counterweight plate mass, as shown in Figure 5, it can be found that the first-order critical speed gradually decreases with the increase of the turntable and other conditions unchanged.

![Figure 5. Trend chart of first critical speed with counterweight](image)

4.3. Influence of stiffness of three pad tilting pad bearing on critical speed of rotor system
In this paper, the hybrid bearing is used to support the rotor. The angular contact rolling bearing with inner diameter of 30mm is used for the rolling bearing, and the three pad tilting pad bearing with inner diameter of 60mm is used for the sliding bearing, the stiffness and damping of the rolling bearing are kept unchanged, and the influence of the stiffness of the supporting bearing on the critical speed of the rotor system is studied by changing the stiffness of the three pad tilting pad bearing.

Input the calculated bearing stiffness coefficient in the rotor module, in this paper, only the meridional stiffness is considered to calculate and analyze the critical speed of the transfer system, and the critical speed score obtained is shown in Table 2:
Table 2. First order critical speed under different stiffness

| Bearing stiffness (N/m) | 1.07E+05 | 1.48E+05 | 1.88E+05 | 2.29E+05 | 2.70E+05 | 3.11E+05 |
|------------------------|----------|----------|----------|----------|----------|----------|
| First critical speed (rpm) | 11454    | 16332    | 17725    | 18833    | 19671    | 20373    |

According to the data in the table, the diagram of the first critical speed changing with the bearing stiffness is made, as shown in Figure 6.

5. Conclusion
In this paper, the critical speed of rotating machinery is analyzed by DyRoBeS dynamics software. Through calculation and analysis, the critical speed diagram and critical speed of rotor system are output. It can avoid resonance on the working speed, and the calculation process of the critical speed of the bearing rotor system is presented by a case.

By analyzing the critical speed data under different counterweights, it can be found that the first critical speed gradually decreases with the increase of counterweight mass and other conditions unchanged.

In the analysis of the critical speed of the rotor system, it is found that the critical speed of the rotor increases with the increase of the bearing stiffness in a certain range.

Acknowledgments
Authors wishing to acknowledge assistance or encouragement from colleagues, In addition, the author would like to thank the Guangdong Provincial Innovation Strong School Project (H320202) for supporting this paper.

References:
[1] Zhang Zhiming, Zhang Yanyang, Xie Youbai, et al. Hydrodynamic Lubrication Theory of Plain Bearing [M]. Beijing: Higher Education Press, 1986:275-305.
[2] Philosophical Maga. (in Chinese with English abstract) Geoffrey H H. The Lateral Vibration of Loaded Shafts in The Neighbourhood of a Whirling Speed [J] Zine, 1919, 6 (37): 304-314.
[3] Ma Wensheng, Chen Zhaobo, Jiao Yinghou et al. Journal of Vibration and Shock, 2014, 33(05): 8-13.
[4] Xie Youbai, Tang Yudi. Journal of Xi ’an Jiaotong University, 1987(04):93-104. (in Chinese)
[5] Kirk R G, Alsaeed A, Gunter EJ. Stability Analysis of Highspeed Automotive Turbocharger [J]. Tribology Transactions, 50, 3, 2007:427-434.
[6] Kirk R G, Alsaeed A. Liptrap J, et al. Experimental test results for vibration of A high speed diesel engine turboC Harger [J]. Tribology Transactions, 2008,51 (4) :422-427