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Analysis on Dynamic Characteristics of Inter-shaft Bearing

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Abstract. The method to analyze the dynamic characteristics of the inter-shaft bearing in the dual rotor system is presented. Under different radial loads, the distributions of minimum oil film thicknesses and contact loads between rollers and raceways are obtained. The results show that the rollers' spinning angular speed decreases and the number of loaded roller increases with the increase in the radial loads of inter-shaft bearing when the rotational directions of outer ring and inner ring are same. The minimum oil film thickness between each roller and inner raceway is thinner than that between each roller and outer raceway in loading region. The rollers' spinning angular speed increases with the increase in the rotational speeds of outer ring, and it decreases with the increase in the rotational speeds of inner ring.

Introduction

The modern aviation engine shaft is the dual rotor structure, in which the low-pressure rotor and high-pressure rotor are in the state of the mechanical coupling, and the bearing connected by the low-pressure rotor and high-pressure rotor is called the inter-shaft bearing[1-2].Study on the dynamic characteristics of the rolling bearing has gone through a process of continuous development. Jones[3] first proposed the raceway control theory, then established a rolling bearing statics analysis model, but in the model the role of lubrication and the effect on bearing torque did not be considered. Aramaki[4] developed a bearing dynamics analysis software BRAIN which can be used to analyze the multitude bearing types. But lubrication and friction resistance did not be considered. Stacke [5] developed the 3D analysis software Beast, but the calculation of the film lubrication and squeeze damping yet need to be improved.

Although comprehensive factors are considered in dynamic analysis model , it is still not very perfect due to the difficulties in getting the mathematics solutions[6]. In view of this, the quasi statics analysis method are used in this paper to research the dynamic characteristics of the inter-shaft bearing.

Basic assumptions

When the inner and outer rings of the inter-shaft bearings rotate in the same direction, the force and movement are more complex compared to the case in which one of the ring is fixed. The following assumptions will be taken in solving load distribution:

(1) The EHL film will be formed between the roller and the inner (or outer) raceway.

(2) Only the deformation of the roller is considered while ignoring the inner and outer raceway deformation.

(3) Contact loads between the rollers and the inner (or outer) ring are symmetrical to radial load line and are evenly distributed.

Quasi static Model

In order to understand the dynamic characteristics of inter-shaft bearing, the quasi static model is established.

Movement analysis of bearing roller. The average speed of the roller and the inner ring, the average speed of the roller and outer ring are as follows:

$$\begin{cases} U_{ij} = \frac{D_m}{4}\left[(1-\gamma)(\omega_i - \omega_c) - \gamma\omega_{rj}\right] \\ U_{ej} = \frac{D_m}{4}\left[(1-\gamma)(\omega_\varepsilon - \omega_c) + \gamma\omega_{rj}\right] \end{cases} \tag{1}$$

The average speed of the roller and outer ring raceway in the non-loading zone:

$$U_{eu} = \frac{D_m}{4}\left[(1-\gamma)(\omega_e - \omega_c) + \gamma\omega_{ru}\right] \tag{2}$$

The relative sliding velocity between the roller and the inner ring , the relative sliding velocity between the roller and the outer ring are as follows:

$$\begin{cases} V_{ij} = \frac{D_m}{2}\left[(1-\gamma)(\omega_i - \omega_c) - \gamma\omega_{rj}\right] \\ V_{ej} = \frac{D_m}{2}\left[(1+\gamma)(\omega_\varepsilon - \omega_c) + \gamma\omega_{rj}\right] \end{cases} \tag{3}$$

Stress analysis for the inter-shaft bearing roller. The simplified force and moment balance equation of the roller in loading area are as follows:

$$\begin{aligned} &(f-1)T_{ej} + f\left(P_{ej} - P_{ij}\right) + (f+1)T_{ij} - fF_{oa} = 0 \\ &(j = 0,1,2\cdots,\mathrm{N}) \end{aligned} \tag{4}$$

Where "+" can be taken when maintaining the cage to promote the roller to move, otherwise "-"can be taken. Where f is the friction factor.

The moment equilibrium equation can be obtained as follows:

$$\left(F_{co} + 2\sum_{j=1}^{N} F_{cj}\right)\frac{D_m}{2} - \left[z - (2N+1)\right]F_{cu}\frac{D_m}{2} = 0 \tag{5}$$

The force balance equation of the roller in the non-loading area is as follows:

$$F_{ou} - F_{cu} - P_{eu} - T_{eu} = 0 \tag{6}$$

Each parameters in equations (1)~(6) are illustrated as follows:

(1) Q_{ij} and Q_{ej} are the contact load between each roller and the inner ring raceway, and the contact load between each roller and outer ring raceway, respectively. The load distribution on inter-shaft bearing can be obtained under equilibrium conditions:

$$F_r - Q_{i0} - 2\sum_{j=1}^{N} Q_{ij}\cos\psi_j = 0 \tag{7}$$

The amount of displacement of the outer ring to the inner ring:

$$\Delta_j = \Delta_0\cos\psi_j = \delta_j + \frac{1}{2}u_r - h_{ij} - h_{ej} \tag{8}$$

where the total elastic deformation amount of the j the roller can be reads:

$$\delta_j = \delta_{ij} + \delta_{ej} \tag{9}$$

The bearing compatible equations in loading area can be obtained by the literature[7] and (8)~(9):

$$[(\frac{Q_{i0}}{K_n})^{0.9} + (\frac{Q_{e0}}{K_n})^{0.9} + \frac{1}{2}u_r - 1.13(h_{i0} + h_{e0})]\cos\psi_j - [(\frac{Q_{ij}}{K_n})^{0.9} + (\frac{Q_{ej}}{K_n})^{0.9} + \frac{1}{2}u_r - 1.13(h_{ij} + h_{ej})] = 0 \tag{10}$$

Equations (7) and (10) can be solved for the N +1 unknowns $Q_{i0}, Q_{i1} \cdots Q_{in}$. The numbers of the rollers in loading area is $M = 2N + 1$. Considering a symmetrical distribution, it only needs to solve N +1 unknowns to simplify the calculation.

(2) P_{ij} 、P_{ej} and P_{eu} are tangential dynamic pressure.

Lubricant between the roller and the inner ring(or outer ring) are squeezed to result in a certain pressure.

$$P_{ij} = \overline{P}_{ij} l_e E_0 R_i \tag{11}$$

$$P_{ej} = \overline{P}_{ej} l_e E_0 R_e \tag{12}$$

The tangential momentum in Non-loading area is as follows:

$$P_{eu} = \overline{P}_{eu} l_e E_0 R_i \tag{13}$$

(3) T_{ij}, T_{ej} and T_{eu} are the tangential frictions of inner ring raceways, outer ring raceways and roller , they can be reads:

$$T_{ij} = \overline{T}_{ij} l_e E_0 R_i \tag{14}$$

$$T_{ej} = \overline{T}_{ej} l_e E_0 R_e \tag{15}$$

$$T_{eu} = \overline{T}_{eu} l_e E_0 R_e \tag{16}$$

$$\overline{T}_{ij} = -9.2 G^{-0.3} \left(\frac{\eta_o U_{ij}}{E_0 R_i} \right)^{0.7} + \frac{\eta_o V_{ij} I_{ij}}{E_0 R_i H_{ij}} \tag{17}$$

$$\overline{T}_{ej} = -9.2 G^{-0.3} \left(\frac{\eta_o U_{ej}}{E_0 R_e} \right)^{0.7} + \frac{\eta_o V_{ej} I_{ej}}{E_0 R_e H_{ej}} \tag{18}$$

$$\overline{T}_{eu} = -9.2 G^{-0.3} \left(\frac{\eta_o U_{eu}}{E_0 R_e} \right)^{0.7} \tag{19}$$

where I_{ij} and I_{ej} are the dimensionless integral; H_{ij} and H_{ej} are the dimensionless quantity of thickness of the oil film between the roller and the inner (or outer) ring. $H_{ij}=h_{ij}/R_i$,$H_{ie}=h_{ie}/R_e$, le is the effective length of the roller; the E_O equivalent modulus of elasticity; R_i and R_e are equivalent radius of the inner ring and outer ring,respectively. G is material parameters; α is lubricant pressure-viscosity index.

The lubricant film is formed between the elastic contact surfaces. In the case of high-speed rotation, thermal effects on the film thickness can not be ignored. Therefore, you must take into account the thermal effect of the calculation of the film thickness. Minimum film thickness formula:

$$h_{i(e)j} = 2.65 \frac{C^*_{i(e)j} \alpha^{0.54} (\eta_0 U_{i(e)j})^{0.7} R^{0.43}_{i(e)} l_e^{0.13}}{E_0^{0.03} Q^{0.13}_{i(e)j}} \tag{23}$$

where $C_{i(e)j}$ can be referred to literature[8].

(4) F_m is the the single roller centrifugal force, which can be calculated as follows:

$$F_m = \frac{1}{2} m_b D_r \omega_c^2 \quad (24)$$

where m_b is the roller quality.

(5) F_{oa} and F_{ou} are the roller resistance result from the fuel-air mixture flow

$$F_{oa} = F_{ou} = \frac{1}{8} C_d \rho D_b l \left(D_m \omega_c\right)^2 \quad (25)$$

where C_d is resistance coefficient of the flow ; p is the density of the fuel-air mixture.

The above equations can be calculated by Newton-Raphson method.

The dynamic characteristics analysis for the inter-shaft bearing

When the rotation speed of the inner ring is 5000r/min, and the outer ring speed is 1000r/min ,the contact loads between the inner ring and roller increase as shown in Figure 1. When load increases from 1000N to 2000N , the number of bearing rollers increases .

It can be seen from Fig. 2 that the roller rotation angular velocity changes with radial force, the rotation angular velocity of each roller in the loading area decreases with the increasing radial force.With the increase of the radial force, the contact load between the roller and the inner ring also increases.But due to the acting of the centrifugal force, the increase amount of contact force between the roller and the inner ring is less than that between the roller and the outer ring. Under a constant radial force, the closer roller next to the line of the radial force, the smaller friction torque inner and outer ring raceways applied on, which caused the rotation angular velocity of roller is smaller. When the radial force reached 1700~2000N, due to the increase of the bearing rollers, the reduction of the contact force and friction force between the inner ring and the roller is less than that of the outer ring and the roller, which caused the increase of the friction torque of the inner and outer rings on the roller and the roller rotation angular velocity.

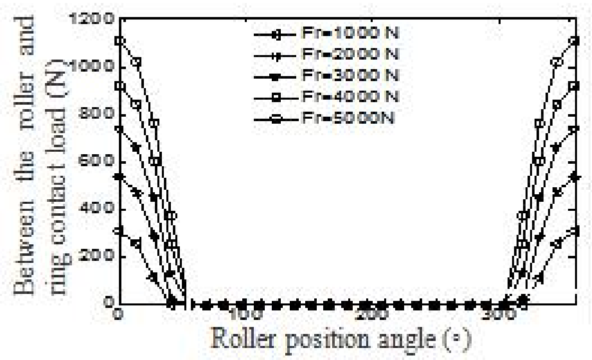

Fig.1 Contact load and roller position angle between the roller and ring under different loads

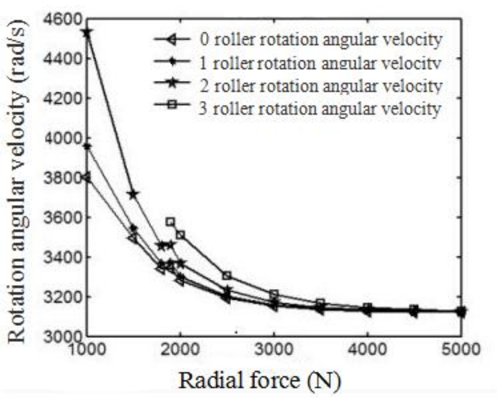

Fig.2 Rotation angular velocity versus radial force

The result showed that :the rotation angular velocity of the rollers increased with the increasing inter-shaft bearing outer ring speed . the rotation angular velocity of the rollers decreases with the increase of inter-shaft bearing inner ring speed . This is because with the increase of the inner and outer rings speed , the traction force of the ring on the roller is reduced. When the traction force between the outer ring and the roller is reduced, the friction torque of the inner and outer rings on the roller is increased, which caused the increase of the rotation speed of the roller; When the traction force between the inner ring and the roller is reduced, the friction torque of the inner and outer rings on the roller is reduced, which results in the decrease of the roller rotation speed.

The film thickness decreases with the increasing of radial force ,with the same roller in the bearing area. And the oil film thickness between the inner ring raceway and roller is less than that of the outer ring raceway and roller.

The bearing cage speed under different radial load can be measured by double rotor testing machine . Compared calculation results with the experimental results, the maximum error of 4.28%, it's visible that the inter-shaft bearing dynamic characteristics analysis model established in this paper is correct, It can guide the inter-shaft bearing design.

Conclusions

In this paper, the inter-shaft bearing dynamic characteristics of the dual-rotor system were studied, the main conclusions are as follows:

(1) An dynamic characteristics analysis model for inter-shaft bearing was established, which can provide theoretical guidance for the inter-shaft bearing design.

(2) The rollers' spinning angular speed decreases and the number of loaded roller increases with the increase in the radial loads of inter-shaft bearing when the rotational directions of outer ring and inner ring are same.

(3) The rollers' spinning angular speed increases with the increase in the rotational speeds of outer ring, and it decreases with the increase in the rotational speeds of inner ring.

(4) The minimum oil film thickness between each roller and inner raceway is thinner than that between each roller and outer raceway in loading region.

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