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Research paper

The thermodynamic properties of a new type catcher bearing used in active magnetic bearings system

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HIGHLIGHTS

- The DDCB is a more suitable catcher bearing for AMBs.
- Compared to SDCB, using DDCB, the temperature rise can decrease in the same states.
- A lower viscosity of lubricant may induce a lower temperature rise.
- The inner raceway temperature of the first layer bearing is the highest.
- Reducing the unbalance mass of the rotor is a method to decrease the temperature rise.

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ABSTRACT

Normally a rotor levitated by active magnetic bearings (AMBs) system would rotate without contacting with any stator component, but the possibility still remains that the supporting force might lose temporarily or permanently, thus requiring the Catcher bearings (CBs) to provide backup protection in case of the failure of AMBs. A new type CB with two separate rolling element bearing series could have the speed distribution between the inner race and intermediate race according to certain ratio, in which the speed of each roller element bearing decreases with the limit speed of the whole CB increasing, offering high capability to sustain its initial rotation speed. Based on the theory of heat transfer, tribology, and rotor dynamics, this paper analyzes the thermal structure of double-decker catcher bearing (DDCB) and single-decker catcher bearing (SDCB), respectively. Through this structure, the thermal resistances and equations of heat transfer can be obtained. Then we calculate the friction heat and temperature distribution in the various CBs upon rotor's dropping on SDCB or DDCB, followed by the discussion on the CBs temperature rise's effects on lubrication conditions and rotor dynamics parameters. Finally various experiments are carried out to measure the temperature rise of different CBs. The results obtained validate the theoretical analysis and also provide main methods to reduce heat generation. Using DDCB is proved to be effective to reduce the temperature rise.

1. Introduction

AMBs support rotors by electromagnetic forces rather than mechanical forces which require lubricated fluid films or contact of rolling element bearings, possessing several advantages over mechanical bearings such as low friction, no need for lubrication and high attainable rotating speed [1,2]. Therefore, AMBs have been used in turbo machinery, energy storage as well as other industrial fields [3,4]. The catcher bearings (CBs) (auxiliary bearing, emergency bearing or backup bearing) are indispensable for AMBs for protecting the stators from contact with the rotor directly. They can temporarily support the rotor during operation and prevent the system damage resulted from AMB failure or excessive transient loads. Most conventional CB is a rolling element bearing with a fixed clearance, which is approximately one-half of the AMB air gap. However, in most cases, conventional CB cannot undertake the ultra-high speed, excessive vibration and impacts when the rotor drops [5]. To overcome these defects, a new type CB which has two

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separate rolling element bearing series is proposed in this paper. On one hand, because of the shared speed of the intermediate race, the speed limit of this new type CB is thus improved, providing higher capability to sustain initial rotating speed. On the other hand, with extra bearing added to the traditional CB, the rotor vibrations after rotor drop can be absorbed significantly.

Once the AMB fails or excessive transient loads occur, the inner race will instantaneously swift to the high-speed rotation from stationary state. As the heat caused by collision and friction result in a dramatically high temperature rise, conventional CB would be burned out quickly, thus yielding a serious damage. Hence Some researches were performed on the thermodynamics of the conventional CBs. Houghton and Carslaw [6,7] pointed out that the heat was mainly determined by the bearing operating parameters, lubrication condition and mechanical structure. Carslaw et al. [7] gave a method to calculate the heat generated in the contact area and proposed a simple formula to estimate its surface temperature. Jorgensen and Shin [8] developed a quasi-static bearing model including thermal expansions for a high-speed spindle system. The detailed CB model was determined based on its material, geometry, speed and preload using the nonlinear Hertz load-deflection formula while the thermal growth of bearing components during the rotor dropped was estimated through a one dimensional thermal model. The empirical formula for the drag torque due to external load was derived by Palmgren [9]. Jiang et al. [10] established the spindle bearing model based on quasi-dynamics analysis considering the friction heat and preloading. Mohsen and Hooshang [11] presented the structural and thermal analysis of a zero clearance auxiliary bearing for magnetic bearing systems. Tedric et al. [12] calculated the bearing temperature and thermal resistance of each node and then obtained the steady-state temperature distribution of the bearing using a 3D heat transfer model. The optimal prediction of bearing heating was given by the numerical analysis with comparison with experimental results. Sun et al. [13,14] studied rotor dropping dynamic properties and simulated corresponding thermal growth, which was also validated by experiments. The friction coefficients, support damping, and side loads were critical parameters to prevent backward (super) whirl prevention and friction-induced heat reduction as shown in Ref. [13]. Ref. [14] studied a ball bearing with a variety of heat sources, and selected rotor/CB mechanical rub and drag torque as the two major sources during the rotor dropped. Using fixed and rotating coordinate systems, Patrick et al. [15] studied the contact problem between the CB and rotor under different initial excitations. Based on the Hertzian Contact Stress Theory, the calculation formulas to estimate the surface temperature of contact area were obtained. In Ref. [16], rotor dynamic contact forces were predicted for a range of initial conditions. The transient thermal response of a CB was assessed for a range of dynamic contact conditions. The localized contact temperatures may rise from each contact event, which would accumulate for multiple contact cases. A nonlinear auxiliary bearing model was developed in Ref. [17], in which the Hertzian contact bearing model was used to predict the fatigue life of the CB during the touchdown. Chen et al. [18,19] established the models of both the ball bearing and the high-speed cylindrical roller bearing considering moving heat source and calculated each local heat generation of each heat source. Zheng et al. [20] established a new thermal structure of the double-decker auxiliary bearing and analyzed its thermal characteristics with comparison with traditional bearings. The rainflow counting algorithm to the sub/surface shear stress-time history was applied to calculate the fatigue life of the CB by the number of drop occurrences by Jung et al., in 2012 [21]. A one-dimensional thermal model of the ball bearing composed of heat transfer network and heat sources based on heat transfer equations was established in the system of the HTR-10 helium turbine generator electromagnetic bearings. The results revealed that the axial contact force was critical to the bearing heat generation, and the ceramic balls with superior thermal properties were recommended [22].

In this paper, we present a new type CB to improve the rotational speed of the intermediate race (formed by outer race of the inner ball bearing, retaining race and the inner race of the outer groove ball bearing) and the carrying capacity of CB. Based on the thermal transfer theory, tribology of rolling-element bearing and rotor dynamics, this paper conducts the thermal structure analysis of the DDCB and SDCB. The effect on temperature rise of the CB regarding lubrication conditions and rotor dynamics parameters are discussed.

The remainder of the paper is organized as follows. Section 2 describes the structure of AMB systems together with various CBs and the simplification of the heat energy loss formulae. The thermal model and processing of DDCB based on the thermal transfer theory is presented in section 3. Section 4 presents the simulations with major CB design parameters. Section 5 shows experimental studies in order to validate the proposed model. Conclusions are drawn in Section 6.

2. Mechanical model

Fig. 1 shows the studied AMBs system structure. δ_1 is the radial clearance between the CB inner race (commonly in rolling element bearing) and the rotor and δ_2 is the air gap of AMBs. Generally, the CBs do not work when the AMBs operate properly. However, they play a protective role in preventing the rotor from contact with the AMBs upon AMBs failure or excessive transient loads. For this new double-layer CB, two deep groove ball bearings (61801) are mounted together to constitute the first layer of DDCBs, while an outer deep groove ball bearing (61805) is mounted in the bearing housing acting as the second layer, depicted in Fig. 2(a). The structure of SDCB is shown in Fig. 2(b). The advantage of this structure can greatly improve the limit speed since part of speed inner race will be allocated to the intermediate race according to a certain speed ratio.

When the rotor is out of control and falls onto the CBs with high speed, the loss energy caused by collision and friction between the CB and the rotor is huge and will scattered in the form of heat, resulting in dramatic temperature rise. If the temperature exceeds the extreme limit, the CB will be damaged. To simplify the problem, it is assumed that there always exist two inner balls and one outer ball vertically below the rotor contact point. The models of rotor dropping onto the DDCB and SDCB are illustrated in Fig. 3(a) and (b), respectively.

Based on the Hertzian Contact Stress Theory for two spheres with limited impact velocity below 500 mm/s, the normal contact force F_n is a function of the contact penetration δ and the penetration velocity $\dot{\delta}$ which can be written as follows [8]:

$$F_n = \begin{cases} K\delta^p\left(1+\dfrac{3}{2}k\dot{\delta}\right) & \delta>0 \\ 0 & \delta\le 0 \end{cases} \tag{1}$$

where the K is the contact stiffness between the rotor and the inner race, which is decided by the material property and contact geometry; in this paper, the rotor and CB are made of steel. Compared with the point contact, a variety of calculation models are available for the contact stiffness of line contact. We choose the empirical formula given by Palmgren to calculate approximately the contact stiffness [23].