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Relationship between geometric errors of thrust plates and error motions of hydrostatic thrust bearings under quasi-static condition

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ABSTRACT

A new model using approximate formulas is established to predict the error motions of hydrostatic thrust bearings. Three different types of geometric errors of thrust plates are listed in this paper including tilt errors, saddle shaped errors and petal shaped errors. The influences of them on lateral tilt error motion, longitudinal tilt error motion and axial error motion are discussed. Definitions of averaging coefficients are made based on the approximate formulas. It is found that the time-varying tilt errors are the main reason for the error motions of hydrostatic thrust bearings. The thrust bearings with six pairs of recesses have priority over the thrust bearings with four and three pairs of recesses in the view of rotation accuracy. Experiments are done using a hydrostatic rotary table with an outer diameter of 2 m. It is found that the second harmonic errors are the main component of the radial run-out and the results agree well with the results calculated from the approximate formulas.

1. Introduction

Heavy rotary tables play an important role in vertical lathes and vertical grinding machines used in energy, transportation, ship building, aerospace and other national key industries [1]. Hydrostatic thrust bearings are widely used in heavy rotary tables for its higher accuracy, higher damping and longer life than rolling element bearings [2]. The high accuracy of hydrostatic bearings comes from the error averaging effect of pressured oil film [3]. In recent years, many researchers have done research on the accuracy of hydrostatic bearings.

Park et al. utilized a transfer function method (TFM) model to predict motion errors of hydrostatic guideways by analyzing the relationship between motion errors and geometric errors of guide rails [4]. Xue et al. investigated the mechanism and affecting factors of the error averaging effect for the hydrostatic guideways using the average film thickness based on the volume of oil film is equal on single pad [5]. Wang et al. used the finite element method (FEM) model to predict the influence of speed on the motion errors of hydrostatic guideways [6]. Cai et al. analyzed the influence of profile errors on axial and angular motion errors for hydrostatic rotary tables by solving the static equilibrium of the closed and constant flow rotary table [7]. Hiroshi et al. studied the static and dynamic running accuracy of an externally pressurized gas thrust bearing taking consideration of the perpendicularity of rotor end surface and the size deviation of the gas supply holes [8–10]. Kashchenevsky et al. got the approximate formulas for the relationship between run-out of the individual parts and run-out of the thrust bearing in a spindle [11]. Cappa et al. used a modified steady-state model to predict the error motions of an aerostatic journal bearing and found that the error motions are mainly influenced by the geometric errors of the shaft compared with the geometric errors of the bush [12]. Similarly, it can be presumed that the geometric errors of thrust plates have a greater impact on the error motions of hydrostatic thrust bearings than the geometric errors of bush.

However, it seems that there are no systematic studies on the influence of the geometric errors of thrust plates on the error motions of hydrostatic thrust bearings. Both the FEM and TFM models are complicated and not convenient for practical engineering applications. In addition, the number of oil chamber needs to be determined first in the design process. Design principles for hydrostatic bearings are discussed in many books [13–15]. However, there are no principles to point out the relationship between the recess pair number and rotation accuracy of hydrostatic thrust bearings. The purpose of this paper is to try to solve these problems.

In this paper, a new model with approximate formulas is established to predict the influence of the geometric errors of thrust plates on the error motions of hydrostatic thrust bearings. The approximate formulas are convenient for practical engineering

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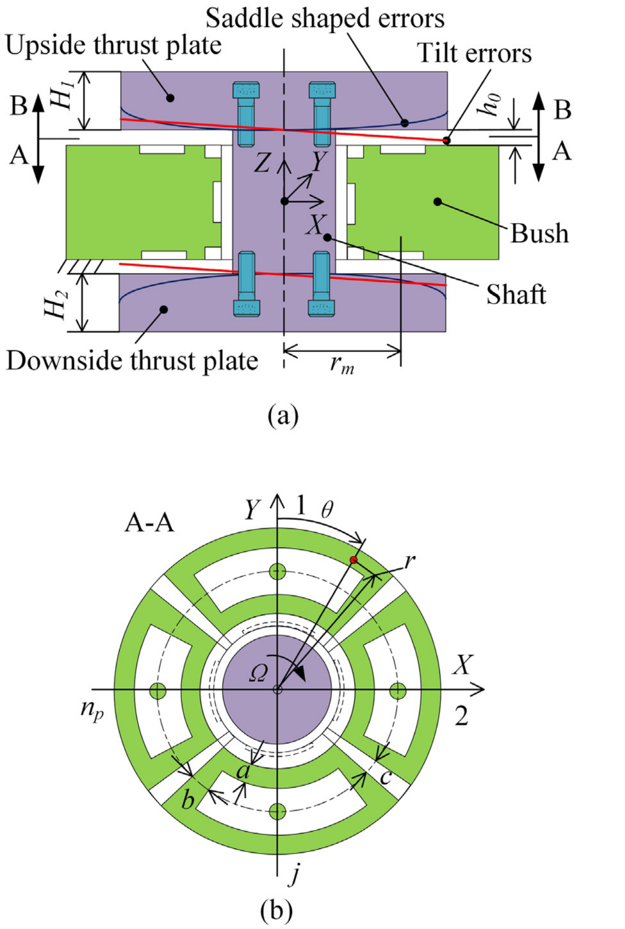

Fig. 1. Structure of hydrostatic bearings: (a) cutaway view, and (b) structure of pads.

applications. Three different types of geometric errors of thrust plates are discussed for hydrostatic thrust bearings with four, three, and six pairs of recesses. The geometric errors include the tilt errors, the saddle shaped errors and the petal shaped errors. Then definitions of averaging coefficients are made. Finally, experiments are carried out to verify the correctness of theory using a hydrostatic rotary table with an outer diameter of 2 m.

2. Theory and calculation

In order to analyze the relationship between geometric errors of thrust plates and error motions of the hydrostatic thrust bearings, the error model should be established in the first step.

2.1. Structure of hydrostatic thrust bearings

As shown in Fig. 1(a), the hydrostatic rotary table consists of a pair of thrust bearings and a journal bearing. The thrust bearings consist of a bush and two thrust plates. The two thrust plates are connected to the shaft by several screws. As shown in Fig. 1(b), the bush has several pads separated by the exhausted grooves. Frequently used situations are discussed in this paper when the bush has just four, three and six pair of pads. Initially, the two thrust plates are symmetrical with respect to the bush. When the shaft is rotating at angular velocity Ω, the thrust plates are deviated from the symmetrical positions causing the vertical and angular displacements of shaft center. A Cartesian coordinate system *XOY* is established at the center of bush. Then, the error motions of shaft center can be expressed by the axial error motion *Z*, lateral tilt error motion β and longitudinal error motion α. The positive direction of lateral tilt error motion β is around *Y* and the positive direction of longitudinal error α motion is around *X*. The structure of single pad is shown in Fig. 2 when the bush is expanded along the circumferential direction. A coordinate system *xoy* is established at the center of pad. The study in this paper is based on the condition that the journal bearing is an ideal bearing which means the journal bearing has no error motions.

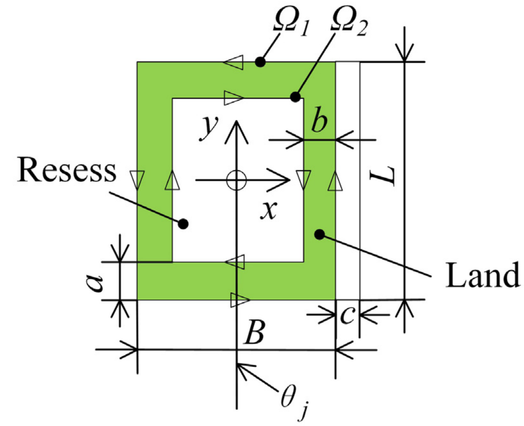

Fig. 2. Structure of single pad after coordinate transformation.

Additionally, r_m is the middle radius of thrust plates and L is the length of single pad. B is the width of single pad which equals $\left(2\pi r_m - n_p c\right)/n_p$, where n_p is the number of recesses and c is the exhausted groove width. a and b are the land width. θ is the position angle and r is the radius for film thickness calculation.

2.2. Types of geometric errors

The error motions are mainly influenced by the geometric errors of thrust plates. Therefore, three different types of geometric errors of thrust plates are focused in this paper. As shown in Figs. 1 (a) and 3 (a), the first case is that the thrust plates have tilt errors. As shown in Fig. 3(b), the second case is that the thrust plates have saddle shaped errors. As shown in Fig. 4, the third case is that the thrust plates have petal shaped errors which are the superposition of the bowel shaped errors and sinusoidal errors. In order to simplify the analysis, the situation that only the upside thrust plate has geometric errors is discussed in theory.

2.2.1. Tilt errors of thrust plates

When the two thrust plates have parallelism errors, the upside thrust plate has a tilt angle β_1 with respect to the downside thrust plate. Therefore, the film thickness change Δh_1 of the upside clearance h_1 can be expressed as follows:

$$\Delta h_1 = -r\sin\left(\theta - \Omega t\right)\tan\beta_1 \tag{1}$$

where the tilt angle β_1 has a constant part β_{10} and a time-varying part β_{1k} in Eq. (2). The constant part is mainly caused by the parallelism errors between the upside and downside surfaces on the shaft which can be expressed as the angle between the two thrust plates. The time-varying part of the tilt angle is caused by the deformation of thrust plates under the pressures of recesses. As shown in Figs. 1 (a) and 3 (a), the two thrust plates and shaft form the "H" shaped gap. Since the pressures of different pairs of pads can't be equal in the adjustment, the gap becomes larger when the smallest part of the "H" shaped gap rotates to the pad pair with maximum pressure. The gap becomes smaller when the smallest part of the "H" shaped gap rotates to the pad pair with the minimum pressure. Therefore, the changing velocity of the time-varying tilt angle is