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Thermohydrodynamic analysis and thermal management of hybrid bump–metal mesh foil bearings: Experimental tests and theoretical predictions

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ABSTRACT

Hybrid bump–metal mesh foil bearings (HB-MFBs) are novel gas foil bearings (GFBs) composed of foil strips and metal mesh blocks (MMBs) in bearing substructure. HB-MFBs provide more advantages than traditional foil bearings, such as high structural damping, assembly accuracy, and ability to work at high temperatures. A thermohydrodynamic model of HB-MFBs was proposed to predict the bearing temperature field with various bearing loads and rotational speeds. The theoretical model considered the complex thermal boundary of bearing substructure in this high-performance foil bearing, including the top foil, bump foil, and MMBs. A detailed thermal-transfer model of hollow rotor was introduced to calculate the heat energy transferring through the rotor shell to the surrounding ambient. Both thermal and centrifugal growth of hollow rotor were considered because of the thin-air film of GFBs to avoid bearing failure. A test rig used to measure the bearing temperature distribution with static load and various rotational speeds was built to validate the proposed thermohydrodynamic model. Different thermal managements with cooling air flow in hollow rotor and bearing substructure were applied to decrease the bearing temperature. The influence of load carry coefficients on bearing peak temperature was investigated with respect to rotational speed. A comparison of heat carry ratios between hollow rotor and bearing substructure was also conducted. The heat energy conducted through the bump foil region and MMB region was analyzed to validate the high-temperature capacity of HB-MFBs.

1. Introduction

Gas foil bearings (GFBs) enable advance oil-free turbomachinery operating at extreme conditions, such as in high rotational speed and high temperature rotor-bearing system. The nickel-base superalloy material of typically foil structures in GFBs can maintain high elastic modulus in the expected wide range of operating temperatures [1]. The solid lubrication on top foil or rotor surface can decrease the friction of moving surfaces between rotor and top foil. Examples of lubrication are polytetrafluoroethene, $MoS_2$and PS304, and the solid lubrications can withstand a high temperature of 250 °C, even up to 800 °C [2–4]. The rotor-bearing systems which employ GFBs as rotor support are compact units with reduced maintenance costs and operate with better mechanical efficiency and improved reliability [5]. Current applications, commercialised or under development, include aircraft gas turbine engines, auxiliary power units, microgasturbines, cryogenic turboexpanders, turbochargers and so on [6].

The estimation of bearing temperature field and the thermal management in rotor-bearing system are essential during the design of high speed, high temperature turbomachinery. The high rotational speed of rotor makes for large velocity gradient of thin-air film along bearing radial direction that leads to foil bearing temperature rises because of the viscous dissipation energy [1]. Another main factor which can influence the bearing temperature is the heat energy conducted from surrounding environment, especially in the high-temperature turbomachinery. For instance, the viscous dissipation energy and the heat conducted from a hot turbine would degrade the material properties of bearing substructure and change the bearing operating clearance [7,8]. Note that the thermal growth of rotor may equal to or even greater than bearing clearance, and it may lead to an excessively thin film which can result in surface-to-surface rub damage between top foil and rotor surface [9]. DellaCorte [10] found that increasing ambient temperature from 23 °C to 538 °C degraded the maximum load capacity at various rotational speeds. At a low frequency excitation of merely 20 Hz, the

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direct stiffness of bearing decreases by 50% when the rotor temperature increases from 22 °C to 188 °C [11]. Particularly, at high-temperature conditions, reliable operation of GFB supported rotor systems relies on adequate thermal management. A cooling gas flow aids to carry away heat and prevent GFBs from encountering thermal seizure thus maintaining an adequate load capacity and thermal stability [12].

Salehi et al. [13,14] conducted an initial analysis in which the Couette flow approximation on the simplification of energy equation was utilized to analyze the temperature magnitude and the effect of pressure gradients on the bearing temperature was ignored. Peng and Khonsari [15] introduced a thermohydrodynamic model coupled with the Reynolds equation and energy equation of air film that considers the compressibility and the temperature-dependent viscosity relationship to predict the steady state temperature fields of GFBs. The predicted results agree well to test data in Ref. [14], with an assumptive bearing clearance. However, the neglect of heat energy which transfer to the rotor and bearing housing will lead to overestimate of temperature field. Feng and Kaneko [16,17] introduced thermohydrodynamic models of multi-wound foil bearings (MWFBs) and bump-type foil bearings (BFBs) that use the Lobatto point quadrature algorithm to accelerate the computation of bearing temperature fields. The predicted results were verified by the experimental data. San Andres et al. [18,19] developed a theoretical model that considers the thermoelastic deformation, thermal expansion and centrifugal growth of rotor. The rotor growth at high temperature and high rotational speed significantly influences the air film because of its thinness. To reasonably predict the temperature field, the rotor growths from thermal and centrifugal effect cannot be ignored.

Experimental investigations of the thermal performance of GFBs have been conducted for decades. Ruscitto et al. [20] first conducted a measurement of temperature rise of BFBs with respect to different bearing loads and rotational speeds. Salehi et al. [14] measured the temperature profile of GFBs with cooling air flow through the foil structure. Test results show that most of the heat energy generated in the film is absorbed by the cooling air flow through conduction and convection at the bearing housing and rotor surface. Radil and Zeszotek et al. [1] conducted experimental investigation of temperature distribution in a generation III GFB at room temperature with respect to a series of bearing loads and rotational speeds. The experimental results show that the peak temperature is measured along the bearing midplane and at the loading region rather than at the bearing side edges where the film thickness is minimal. Measurements also show significant axial temperature gradients as the rotor speed increases.

Metal mesh foil bearing (MMFB) can offer high structural damping to rotor system compare with BFBs but it also has some disadvantages such as low moulding precision, material creeping and complex heat management. Feng, Liu and Zhao [21] proposed a high-performance hybrid bump–metal mesh foil bearing (HB-MFB) which has high structural stiffness and damping coefficient by combining the advantages of BFBs and MMFBs. The HB-MFBs utilised the bump foil and metal mesh blocks (MMBs) to support the top foil. The proposed foil bearing has cooling channel inside the bearing substructure to carry away heat energy easily and thus can work at high temperature rotor-bearing system. Feng, Zhao and Huo [22] redesigned the bump foil of HB-MFBs by adding circular bump between MMBs to obtain uniform support and avoiding excessive sagging of the top foil compared with the original bearing. Fig. 1 shows the schematic view of HB-MFBs which has larger diameter than that in previous literature. HB-MFBs have four main components: top foil, bump foil, MMBs and bearing housing. The single pad top foil is circular and it is fixed at one end by inserting it to the slot inside bearing housing. The bump foil consists of circular bump and trapeziform bump arranged alternatively along the circumferential direction. It is placed under the top foil in three pads to decrease the mechanical error on the accuracy of bearing assembly. The MMBs are inserted into the grooves inside the bearing housing under the trapeziform bump region. The improvements on the bearing compliant structure are essential for HB-MFBs with large diameter which works at high bearing load and high-temperature system.

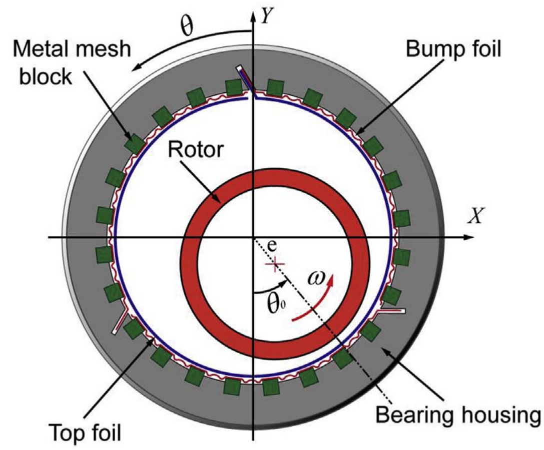

Fig. 1. Schematic view of HB-MFBs.

This study proposes a thermohydrodynamic model of HB-MFBs to predict the temperature field with various bearing loads and rotational speeds. The theoretical model considers the complex-bearing substructure of this high-performance foil bearing. A detailed thermal-transfer model of hollow rotor is introduced to calculate the heat energy which transfers through the rotor shell to surrounding ambient. Both thermal and centrifugal growths of hollow rotor are considered because of the thin-air film of GFBs to avoid bearing failure. A test instrumentation to measure the bearing temperature distribution with static load and various rotational speeds is built to valid the proposed thermohydrodynamic model. Different thermal managements with cooling air flow in hollow rotor and bearing substructure are applied to decrease the bearing temperature. The influence of load carry coefficients on bearing peak temperature is investigated with respect to rotational speed. The comparison of heat carry ratios both between hollow rotor and bearing substructure and between through bump foil region and MMBs region are analyzed to validate the high-temperature capacity of HB-MFBs.

2. Thermohydrodynamic model and boundary conditions

2.1. Reynolds equation and energy equation

In the light of the coordinate system showed in Fig. 1, the dimensionless steady-state Reynolds equation for the compressible gas in Cartesian coordinates can be presented as

$$\frac{\partial}{\partial\theta}\left(\overline{p}\,\overline{h}^3\frac{\partial\overline{p}}{\partial\theta}\right)+\frac{\partial}{\partial\overline{y}}\left(\overline{p}\,\overline{h}^3\frac{\partial\overline{p}}{\partial\overline{y}}\right)=\Lambda\frac{\partial}{\partial\theta}(\overline{p}\,\overline{h}) \tag{1}$$

where $\theta=\frac{x}{R}$, $\overline{y}=\frac{y}{R}$, $\overline{p}=\frac{p}{p_a}$, $\overline{h}=\frac{h}{C}$, $\Lambda=\frac{6\mu(T)\omega}{p_a}\left(\frac{R}{C}\right)^2$. x and y denote the bearings circumferential and axial direction, respectively. $\mu(T)$ is the viscosity of air and it is a function of local temperature T. The dimensionless film thickness is

$$\overline{h}=1+\varepsilon\cos(\theta-\theta_0)+\frac{\delta_f}{C} \tag{2}$$

where ε is eccentricity ratio, θ_0 is attitude angle of rotor centre and δ_f is the deflection in the top and bump foil. Then, the gas pressure distribution in bearing surface can be calculated by coupling the deflection of top foil and the Reynolds equation with given bearing parameters.

The simplified energy equation of air film is provided as [15].