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Active bump-type foil bearing with controllable mechanical preloads

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ABSTRACT

This paper presents an active bump-type foil bearing (ABFB), in which mechanical preloads can be actively adjusted by controlling the driving voltages of piezoelectric actuators (PZTs) to create a variable multiple-lobe clearance and enhance bearing performance. A theoretical model that couples active and compliant foil structures is proposed, and static load–deformation tests and measurements of airborne drag torque on the basis of a prototype bearing are conducted at different driving voltages. Tests results show that the proposed active structure is feasible and effective in controlling mechanical preload. Moreover, the static and dynamic performances of ABFB are discussed. This work lays foundation for the further study of active control methods in gas foil bearings.

1. Introduction

Gas foil bearings (GFBs) are compliant hydrodynamic fluid film bearings that support high-speed rotors through a thin gas film generated by hydrodynamic viscous pumping of ambient gas into the wedge-shaped space between the spinning rotor and bearing surface [1]. GFBs use ambient gas as a lubricant and do not require a complex oil lubrication system [2,3], thus, they are considered good substitutes for oil-lubricated and rolling-element bearings in high-speed, high-temperature applications [4]. By utilizing this oil-free technology, turbomachinery acquires several advantages, including compactness, high-speed, high--temperature capacity, low power loss and reduced maintenance. Various studies have demonstrated the potential application of GFBs in air cycle machines [3], micro-turbo generators [5], vehicle turbo charges, and turbo expanders [6].

The compliant foil structure in GFBs works in series with the gas film to support the rotating shaft [7–9]. The structural stiffness and damping of the foil structure are critical for load-carrying capacity and stability. Thus, numerous studies have presented ingenious structures of GBFs to achieve large load-carrying capacity and good stability [10–12]. The development of coating technology and design of compliant structures have resulted in advanced bearings, in which the compliance of the support structure is tailored, and whose load capacities are up to five times those of simple designs [10]. Santos et al. [13] studied the mechanical behavior of the GFB by using numerical analysis and experimental investigation. Iordanoff et al. [14] studied the effect of internal friction on the dynamic behavior of the GFB.

However, the relatively poor stability of GFBs severely hinders their extensive application in micro turbomachinery. The damping characteristics of GFBs remain a problem in stabilizing rotor–bearing systems and have elicited much attention. San Andrés et al. [15,16] directly employed damping materials as compliant structures and developed metal mesh foil bearings (MMFBs). They measured the structural stiffness and damping coefficients of the MMFBs. The MMFBs exhibited excellent damping characteristics, although, their low molding precision and unstable structure remain core issues in their industrial application. Lee et al. [17] proposed a viscoelastic foil bearing, in which a viscoelastic foil is inserted between the top and bump foils to achieve good damping characteristics. Similarly, Sim and Park [18] inserted a polymer layer between the bearing cartridge and bump foil. Feng et al. [8] proposed a hybrid bump-metal mesh foil bearing (HB-MFB) with a bump foil and metal mesh blocks in the bearing substructure. The HB-MFBs demonstrated good assembly accuracy and stability and operated at high temperatures. Feng et al. [19] proposed a novel nested compression spring GFB, in which series of nested compression springs are used as compliant support structures.

Common GFBs suffer from extremely strong nonlinearity effects during high-speed rotation, and these effects influence the stability of rotor–bearing systems and may cause severe subsynchronous motion amplitudes. Heshmat et al. [20] proposed three pad bump-type foil bearings (BFBs). They reduce the stiffness of the bump foil from the weld to the free end along each pad. San Andrés [21] inserted three metal

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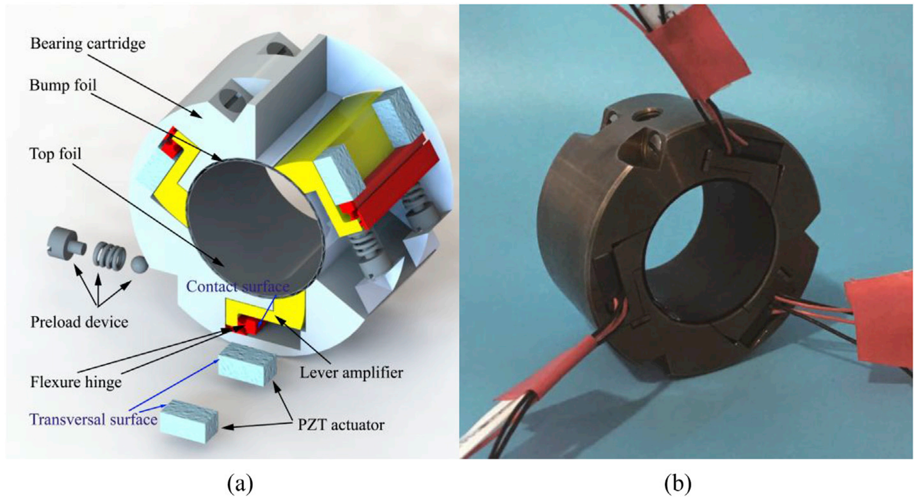

Fig. 1. Schematic of the ABFB and the test bearing.

Table 1
Main parameters of the ABFB.

Parameters	ABFB
Bearing cartridge axial length	30.00 mm
Top foil thickness	0.1 mm
Bump foil height	0.51 mm
Bump number	25
Foil material	X-750
Journal diameter	30.00 mm
Nominal clearance	35, 40, 45 μm
PZTs dimension	3 × 3 × 10 mm
Groove number	3
Type of PZTs	PTJ1500303101
Width of flexure hinge A	0.4 mm
Width of flexure hinge B	0.4, 0.5, 0.6 mm

shims into BFBs to create a multiple-lobe clearance profile. Kim [22] parametrically analyzed preloaded BFBs, and the bearing cartridges were processed as three-lobe profiles. Hoffmann and Liebich [23] studied the effects of the metal shim on the dynamic behavior of a BFB by using experimental and numerical analysis. The main idea in these studies was to utilize the stiffness gradient or apply a mechanical preload to induce hydrodynamic wedges, generate more supporting gas pressure, and enhance the operational stability of BFBs at high rotor speeds.

However, these solutions induce new problems. Adding a mechanical preload reduces bearing performance at low rotor speeds because of the reduction in film thickness and increases the wear of the bearing and rotor. Traditional solutions are passive and cannot meet the requirements of current research. Increasing availability means enhancing the capability of machines to adapt to different operational environments, minimizing or eliminating service intervals, and extending the useful life of rotating machinery. The use of "active" mechatronic elements that consist of transducers, actuators, and control systems provides innovative solutions to such demands and issues above.

The goal of this study is to develop an active BFB (ABFB) with adjustable mechanical preloads to overcome the limitations of common passive GFBs. Piezoelectric actuators (PZTs) are used as active mechatronic elements to control the radial clearance of the proposed ABFB. PZTs can induce hydrodynamic wedges between the top foil and rotor to enhance bearing performance. The deformations of PZTs are much less than the radial displacements required to adjust the bearing radial preload. Flexure hinges and a lever amplifier are utilized to create an amplification system of deformations. A theoretical model of the active foil structure is developed in the consideration of PZTs, flexure hinges, and the lever amplifier system. The model is then coupled with the Reynolds equation to calculate the ABFB performance. Static load–deformation tests

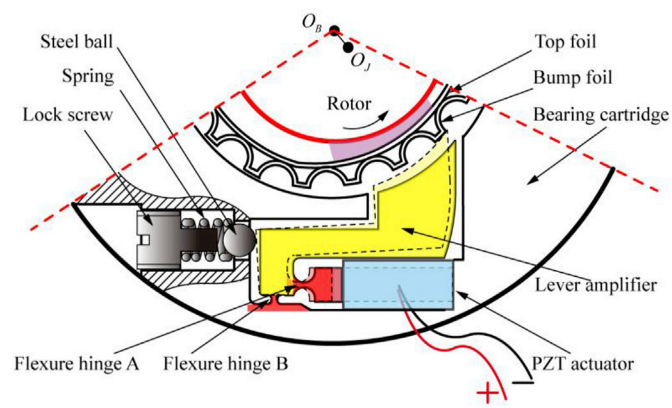

Fig. 2. Schematic of the working principle for generating the radial displacement of the lever amplifier.

are conducted to measure the stiffness coefficients and radial clearance of the proposed ABFB when the PZTs are electrified with different driving voltages to prove the feasibility and effectiveness of the structure. Drag torque is measured to validate bearing lift-off performance at different driving voltages. The static and dynamic performance of ABFBs with different parameters and electrifying methods is predicted and analyzed with the proposed theoretical model. This work proposes a new means for active control of GFBs and lays a theoretical foundation for the further study of active control methods in rotor–bearing systems.

2. Description of ABFB

The mechanical preload of GFB, which creates a multiple lobe clearance on the bearing inner surface, can induce a hydrodynamic wedge between the top foil and rotor to enhance the GFB performance [20]. The structure of the proposed ABFB is shown in Fig. 1. The lever amplifier consists of a rigid body and two types of flexure hinges. Flexure hinge A is a circular flexure hinge and placed next to the PZTs. It is used to ensure parallelism between the transversal surface of the pair of PZTs and the contact surface of the flexure hinge A. Flexure hinge B is a corner-filleted flexure hinge and used as the rotational pivot for the lever amplifier. Six flexure hinges including three flexure hinges A and three flexure hinges B, three grooves, and three lever amplifiers are machined by wire cutting and uniformly aligned along the circumferential direction in the bearing cartridge. The inner surface of the bearing cartridge consists of three flexible lever amplifiers and a rigid body. The main