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Acoustic journal bearing – Performance under various load and speed conditions

T.A. Stolarski [a,b,*], R. Gawarkiewicz [a], K. Tesch [a]

[a] Gdansk University of Technology, Gdansk, Poland
[b] Brunel University London, United Kingdom

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ABSTRACT

The paper presents results of experimental testing aiming at finding out what effect system of piezoelectric actuators (PZTs) attached to an aerodynamic journal bearing has on the magnitude of shaft's motion within the bearing operating at specified speed and load. The results clearly demonstrate effectiveness of PZTs in mitigating the shaft's motion thus contributing to the increased stability of the bearing. This stabilizing effect is especially pronounced for lightly loaded bearing running at speed.

Three bearings, each having a different geometry, were tested and their dynamic performance recorded using fast response data acquisition system.

1. Introduction

Traditional non-contact bearings, such as air bearings (aerostatic and aerodynamic) and magnetic bearings are commonly used in a number of practical specialist applications. However, magnetic bearings are unacceptable where a strong magnetic flux is harmful to the surrounding environment while a continuous supply of a large volume of clean air, never sufficiently clean for food and medical applications, from external auxiliary devices is required for air bearings (for example batteries of drilling spindles used in mass scale manufacturing), which increases the cost of their use. Also, aerodynamic air bearings are known to become unstable when lightly loaded and operating at high speeds. Therefore, alternative and radically new concepts, such as acoustic bearings, offer an exciting solution.

Acoustic levitation, being at the heart of an acoustic bearing concept, uses an acoustic wave to exert a force on objects immersed in the wave field. These forces are normally weak but can become quite large when using high frequency and high intensity waves, large enough to suspend substances against gravity force. Ultrasonic levitation (frequencies higher than 20 kHz) has been used initially for levitating small objects using a standing wave field between a sound radiator and a reflector. In standing wave acoustic levitation, the size of the levitated object is limited to less than a wavelength. Standing wave type ultrasonic levitators have been designed for applications in various scientific disciplines such as material processing and space engineering [1].

Kundt's in his tube experiment [2] in 1866, in which small dust particles moved towards the pressure nodes of a standing wave created in a horizontal tube, was the first to observe standing wave levitation. The first detailed theoretical description of standing wave levitation was given by King [3] in 1934, which was extended by Hasegawa [4] to include the effects of compressibility. Embleton [5] adopted King's approach to fit to the case of a rigid sphere in a progressive spherical or cylindrical wave field. Westervelt [6–8] derived a general expression for the force owing to radiation pressure acting on an object of arbitrary shape and normal boundary impedance to show that a boundary layer with a high internal loss can lead to forces that are several orders of magnitude greater than those predicted by the classical theory.

Another type of levitation is the squeeze-film (or near field acoustic) levitation in which an object (not limited by wavelength) is brought very close to a radiation surface vibrating at high frequency. The gap is smaller than the wavelength of the generated sound, which means that the standing wave is replaced by a gas film with pressure that varies according to the motion of the radiation surface. Squeeze film levitation can carry higher loads than standing wave levitation and has been widely investigated for building non-contact linear and rotational bearings [9–13]. A contribution to the analysis of squeeze film gas non-contact suspensions was made by one of the authors of this paper [14–19].

Earlier research fully described in [18], laid out foundation for the current acoustic bearing concept. The early device operated in an audible range of frequencies (a few kHz) generated by the

* Corresponding author at: Brunel University London, United Kingdom.

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piezoelectric transducers attached to at the bearing structure and for that reason was unacceptable and in the need of improvement, therefor an acoustic journal bearing system utilizing acoustic levitation phenomenon was proposed [19–21]. The bearing shell was specially configured to secure its flexibility through the use of “elastic hinges”. It could be elastically deformed with desired frequency by three piezo-electric actuators (PZT).

In principle, acoustic bearing should have most of the advantages of aerostatic bearings but external pressurized air supply is not needed and the bearing interface can be as simple as two plain surfaces; although design consideration is needed to create high frequency vibration of the bearing surfaces. From experiments [22] it is evident that acoustic pressure was sufficient to support a bearing spindle system even at its stationary state during the start and stop operation under a load. This system was tested experimentally for various geometry configurations of the bearing shell to determine its static load supporting capacity [22]. It was found that the static load capacity of the bearing depends, to a large extent, on the geometry of the bearing shell. Studies, which results are presented in this paper, mainly aimed at finding out how effective is the squeeze-film acoustic levitation in mitigating shaft's motion within the bearing of a given geometry and operating with specified rotation speed and load.

2. Apparatus, tested bearings, and procedure

The search for the most effective geometry of the acoustic journal bearing and its outcomes are presented elsewhere [22]. As a result of that three configurations of the acoustic journal bearing were identified and selected for further experimental testing to determine they performance under various load and speed conditions. Nominal bore diameter of 30 mm was used all bearings tested experimentally. They had length of 50 mm. Two of the bearings were made of aluminium and required the use of three foil type PZTs (piezo-electric actuators) arranged around their circumferences. The foil-type PZTs were of rectangular shape (12×10 mm) and had thickness of 0.5 mm. They were attached to the bearing outer wall at specially machined flats, spaced 120°. The third bearing was made of alloy steel and six rod-type PZTs were employed. The rod-type PZT had a square cross-section (5×5 mm) and the length of 18 mm.

Since there is a complete separation of interacting surfaces by an air film therefore it was thought that any special microscopic examination of their state and conditions was not critically important. However, both shaft and the bearings were machined in accordance with strict precision and tolerances. Surface roughness Ra was equal to or less than 0.32 μm. Dimensional tolerances of the bearing bore and the shaft were imposed in such a way as to secure effective radial clearance of 20 μm. Routine check of roundness confirmed that both the shaft and the bearing bore were within prescribed limits.

The details of the three bearings used are elaborated on below where all three geometries are presented and discussed.

2.1. Geometry of tested bearing

Fig. 1 shows the geometry and main dimensions of the first bearing (called G1). This particular geometry was arrived at on the basis of studies presented in [22]. The radial clearance used was 20 μm. Aluminium was used as the bearing's material because it has low coefficient of energy absorption.

Fig. 2 depicts the geometry together with main dimensions of the second bearing (denoted G2). This bearing has a slightly different geometry, comparing to the previous one, which was arrived at through finite element modelling with the aim to obtain

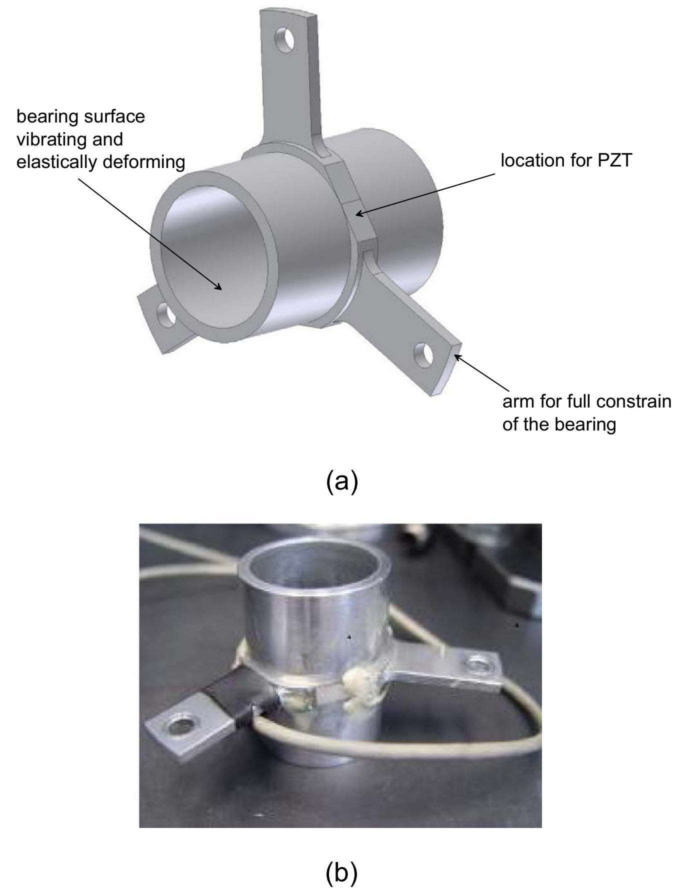

Fig. 1. (a) Solid modelling image of the bearing; (b) photograph of the bearing with PZTs.

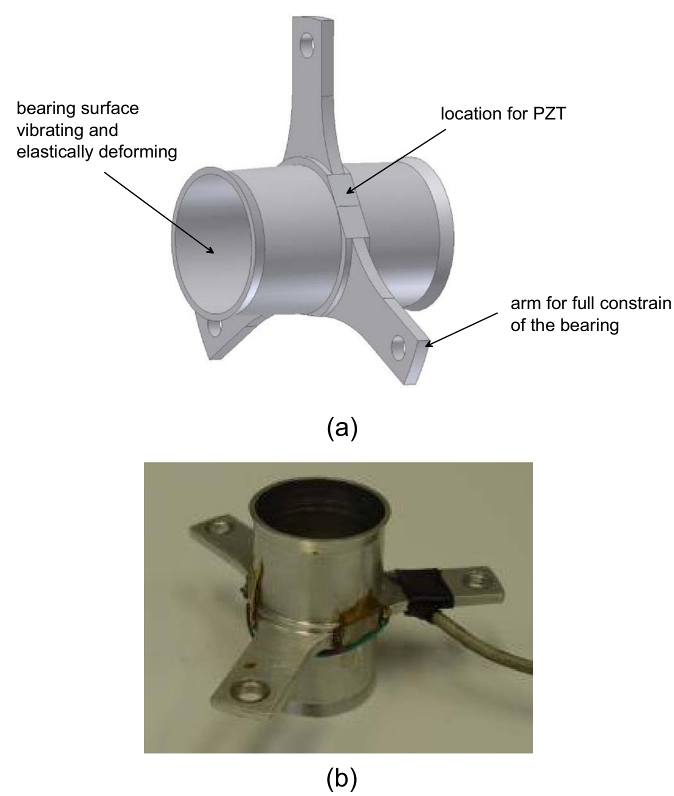

Fig. 2. (a) Solid modelling image of the bearing; (b) photograph of the bearing with PZTs.