Extended duration running and impulse loading characteristics of an acoustic bearing with enhanced geometry

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1. Introduction

Acoustic levitation has no commonality with vibratory bowl feeders and conveyors, which are conventional contact devices based upon a vibrating track.

The standing-wave levitation phenomenon , in which small dust particles moved towards the pressure nodes of a standing-wave created in a horizontal tube was first observed in Kundt’s tube experiment [1] in 1866. The first detailed systematic theoretical description of standing-wave levitation was given by King [2] in 1934, which was extended by various authors until Westervelt [3] 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 existing theory. There have been improvements since.

An acoustic wave can exert a force on objects immersed in the wave field. These forces are normally weak, but they can become quite large when using high frequency (ultrasonic) and high intensity waves. The forces can even be large enough to suspend substances against gravity force. This technique is called acoustic levitation. Since the sound waves used are often in the ultrasonic frequency range (higher than 20 kHz), it is more often called as ultrasonic levitation.

Ultrasonic levitation has been firstly used for suspending small particles by creating a standing wave field between a sound radiator and a reflector that is standing wave ultrasonic levitation. Standing wave type ultrasonic levitators with various features were designed for applications in different scientific disciplines such as contactless conveying of light objects, material processing and space engineering [4]. Another well-known type of ultrasonic levitation is squeeze film ultrasonic levitation. It happens when a flat surface is brought to a conformal radiation surface which vibrates with high frequency.

To overcome the difficulties that are inherent to the existing air non-contact bearings (aerostatic and aerodynamic), squeeze film levitation has been widely investigated for designing non-contact linear and rotational bearings [5, 6, 7]. In principle, squeeze film bearing should have most of the advantages of aerostatic bearings but instead of a pressurized air being fed through orifice or porous carbon/metal in aerostatic bearings, the load-carrying air film is generated by high frequency vibrations and the corresponding squeeze actions between two surfaces. Externally pressurized air supply is no longer needed. This feature allows the bearing interface to be as simple as two plain surfaces. The additional effort needed in this type of bearing is to induce high frequency vibration of the bearing shell and, simultaneously, to deform it in such a way as to facilitate squeeze-film action in the space between the shaft and the bearing’ bore. A number of prototype non-contact suspension and transportation systems based on squeeze film levitation have been reported.

Yoshimoto [8] introduced the use of elastic hinges in squeeze film bearing design and presented a squeeze-film gas bearing with elastic hinges for linear motion guide. Stolarski [9] followed Yoshimoto’s idea and developed a self-lifting linear air contact using elastic hinges in order to increase flexibility of the structure. Both Yoshimoto [8] and Stolarski [9] devices were operated at several kHz, out of the resonance of the piezoelectric transducer or the bearing structure; therefore they were, however, quite noisy during operation. Vibration amplitudes generated were also small, in the order of 4 to 5 micrometres. This resulted in a strict machining tolerance of the bearing and a limited load capacity. In 2007, Yoshimoto [10] introduced a new design of the earlier presented linear motion guide. The bearing plate was excited to vibrate in a bending mode at ultrasonic frequency by two piezoelectric transducers placed on top of the plate. The newly designed bearing was found to be quiet and have higher load capacity up to 10 N. Ha [11] presented an aerodynamic journal bearing capable of self-lift using squeeze film pressure. The basic principle was the same as Yoshimoto [8] and Stolarski design [9], using elastic hinges and piezoelectric actuators to deform the bearing surface. The difference was that the bearing has a cylindrical inner surface. During the start and stop, the squeeze film pressure was generated by oscillating the bearing clearance to lift up the shaft. When the shaft reached sufficiently high rotation speed, three-lobe geometry of the bore was formed by deforming the bearing’s inner surface using bias voltage applied to the transducers. The bearing could then work as a normal aerodynamic bearing. The load capacity of this bearing was found to be limited at 2.18 N, with bearing clearance and frequency of 0.45 μm and 1400 Hz respectively. More detailed description of the squeeze-film ultrasonic acoustic bearing operation can be found elsewhere [12].

This paper presents results of experimental testing carried out on an acoustic bearing with enhanced geometry. This enhanced geometry was arrived at through the analysis of results contained in previous two papers [13, 14] dealing with the search for an adequate geometry and the performance of acoustic bearings under various load and speed conditions.

2. Apparatus, tested bearing and procedure

The search for the most effective geometry of the acoustic journal bearing and its outcomes together with the results of performance testing under various load and speed conditions are presented elsewhere [13, 14]. As a result of that the bearing with enhanced geometry was identified and selected for further long duration experimental testing to determine its running characteristics.

2.1 Apparatus

Experimental testing of the acoustic bearing was carried out using bespoke designed apparatus. The configuration of the apparatus was that such that the load on the bearing could have been precisely controlled. Also, to drive the shaft at speed an air turbine principle was chosen as that guaranteed no additional load on the bearing resulting from applied torque. Figure 1 illustrates experimental apparatus. With the vertical shaft supported by an aerostatic thrust bearing (no-load on the test bearing) it was possible, by tilting the whole apparatus, to impose a load on the test bearing of a desired magnitude. The apparatus facilitated tests at various loads on the acoustic bearing as well as at various rotational speeds of the shaft – both parameters important for determining overall performance of tested bearing and speeds at which bearing’s instability was triggered. The shaft, supported by an aerostatic thrust bearing, was powered by an air turbine and the set of “buckets” specially machined on its circumference. In order to ensure uniform supply of pressure and to secure that no force apart from a pure torque was acting on the shaft three air nozzles were used. The nozzles were fixed to the housing of the apparatus in such a way as to direct air jets tangent to the shaft’s surface.

The apparatus was clamped to the base plate and could be tilted by desired angle Θ creating, in consequence, a loading on the test bearing in a controlled way. The load on the bearing (F) was calculated from the expression,

where m is the mass of the shaft, g is gravitational acceleration, and Θ is the angle of tilt.

During experimental testing three different loads on the bearing were used namely: 0 N (corresponding to Θ = 0 deg), 0.31 N (Θ = 3 deg) and 0.62 N (Θ = 6 deg).

An amplifier and frequency generator were used to operate PZTs at required frequency. The force produced by PZTs, and hence the magnitude of bearing’s deformation, was controlled by the offset voltage and amplitude voltage. Motion of the shaft within the bearing was continuously monitored by a pair of contactless sensors positioned and mutually perpendicular planes. They were of Eddy current type with the frequency response of up 10 kHz. Data acquisition system was deployed to register shaft’s motion in real time. Tested bearing was loaded by tilting the apparatus from the vertical position by a required angle. The effective load on the bearing was a function of the angle of tilt and the weight of the shaft.

The main goal of experiments was to obtain prolonged duration running characteristics of the acoustic bearing for two test conditions, that is when PZTs action was switched on (squeeze film ultrasonic levitation) and when PZTs were switched off (tested bearing was running as a standard aerodynamic journal bearing). In this way it was possible to ascertain the effectiveness of the squeeze film ultrasonic levitation in stabilizing the operation of the bearing.

2.2 Geometry and characteristic features of the bearing

Judging by performance test results for the bearing with conventional geometry [14] it was clear that the main issue with rather a low load capacity is the bearing’s overall stiffness and the inability of PZTs stacks to deform the bearing sufficiently. In a search for a way to make the bearing more flexible the idea of an “elastic hinge” was utilised. The concept of the bearing geometry is based on the notion of an elastic hinge advanced by Yoshimoto [15] for a linear bearing and adopted for the journal bearing by Stolarski [11, 16 ]. So, the bearing with enhanced geometry was arrived at not through a formal design optimisation by resulted for the analysis of results generated for bearings with different geometries and the need to reduce bearing’s stiffness.

The geometry of the test bearing, shown in Figure 2, is a novel design comprising of a number of distinct features. Its main feature is the use of “elastic hinges” arranged in such a way as to make the bearing flexible in the circumferential direction while preserving its considerable stiffness in the longitudinal direction. Nominal bore diameter of the bearing was 30 mm and its length equal to 50 mm. The nominal radial clearance was set to 30 µm. The bearing was made of an alloy steel. The choice of bearing’s material was dictated by the need to find out if it is acceptable material for an acoustic bearing. Besides, some manufacturing reasons, such as ability to machine the bearing with prescribed dimensional accuracy, also played a role in the choice of the material.

The bearing was fitted with multi-layered piezoelectric stack elements. They were made of lead zirconate titanate, which is an intermetallic compound. The rod-type PZT elements used had a square cross-section (5 x 5 mm) and the length of 18 mm. In order to deform the bore of the bearing by desired amount (usually 2 to 3 microns) six PZTs were deployed (see Fig.2b)

The bearing had increased length resulting in the L/D ratio of 1.6. The rationale behind that was the need to minimise misalignment of the shaft relative to the bearing.

The outcome of computer simulation of elastic deformation of the bearing under the action of PZTs is shown in Fig.2e. It can be seen that the bearing deforms into favourable shape facilitating creation of three-lobe geometry known to contribute to a stable running of an aerodynamic journal bearing.

The shaft was made of carbon steel, with nominal diameter of 30 mm and the length of 110 mm. It was dynamically balanced and at its both ends had machined “buckets” for air turbine drive (see Fig. 2c).

2.3 Experimental procedure

Testing was carried out in accordance with the procedure which involved a number of steps.

At the beginning of each test it was ensured that the shaft was in a true vertical position and floated freely on air cushion created by the aerostatic bearing fitted into the base of testing apparatus. Next, the offset voltage, Voff, was set to a required value (usually 50 V) so that the bearing was pre-deformed. Adjusting the running voltage, Vamp, (usually to 40 V) to create the amplitude of cyclic elastic deformations was the next step. After that the PZTs were switched on and run at frequency corresponding to resonant frequency of the bearing. Rotation of the shaft at the speed prescribed by a given test was executed when it was in a vertical position (no load on the bearing). Finally, the load on the bearing was exerted by tilting the apparatus by required angle Θ.

The first test was with switched-off PZTs. This enabled to measure the movement of the shaft within the bearing with the help of two non-contact probes located at two mutually perpendicular directions. Using computerised data acquisition system, continuous recording of the shaft movement took place. These data, recorded over a substantial period of run-time allowed construction of diagrams illustrating movement of the shaft’s centre within the bearing operating at specified speed and load. The magnitude of the shaft motion in X and Y directions provided information whether operation of the bearing was stable or not. Using the same procedure, the influence of PZTs switched-on and operating at a certain frequency on the movement of the shaft was ascertained. Data recorded by the acquisition system showed the magnitude of shaft motion in the X and Y directions for a given rotational speed and load for both PZTs switched-on and PZTs switched-off. In this way it was possible to assess the influence of squeeze-film acoustic levitation on the performance of the bearing during a prolonged run.

Furthermore, performance of the acoustic bearing with enhanced geometry under impulse loading was assessed. This type of test provided information on the ability of the bearing to cope with a sudden impulse loading - quite common situation in many practical applications. The assessment was carried out for the bearing running with switched-on PZTs and also when PZTs were switched-off. The impulse load was applied on the rotating shaft at speed with the simple device of a pendulum. The length of a flexible cord from which the steel ball with mass = 14 g was suspended equalled 420 mm. The angle at which the ball was released to hit the shaft was 35 degrees to the vertical. Knowing the mass of the ball and the length of the pendulum it was possible to precisely define the impulse load applied. Also, this way applying impulse load guaranteed repeatability of impulse loading for successive tests. Movement of the shaft due to an impulse loading was recorded in real time. Subsequent analysis of the data enabled assessment of the bearing response to the impulse loading when PZTs were switched-on and also when they were switched-off.

3. Results and their discussion

3.1 Extended duration tests

The main purpose of the extended duration tests was to find out how the acoustic bearing would perform under specified load and rotational speed over a period of time. A typical test consisted in observing and recording shaft’s displacement in X and Y directions for a given rotational speed and the load on the bearing when the PZTs were switched off and switched on. Figure 3 shows the pattern of shaft’s positions when the load on the bearing was 0.31 N. It is easy to notice that when the PZTs is off (indicated by the bottom diagram – PZT status) the displacements of the shaft in the X and Y directions are quite substantial although its position is quite stable. Quantitatively, shaft’s displacement in X direction was changing between 11 µm and 19 µm resulting in shaft movement of 8 µm. Displacement in Y direction was between 11 µm and 23 µm giving shaft movement of 12 µm. These changes were recorded at the shaft’s rotational speed of 20000 rpm. However, when PZTs are switched on shaft’s displacements are significantly reduced as shown in Figure 3. Displacements in X direction were between 13 µm and 18 µm giving shaft movement of 5 µm. Similarly, in Y direction recorded displacements were between 16 µm and 23 µm resulting in shaft movement of 7 µm. So, the overall reduction in shaft’s movement in X and Y directions is 37.5% and 42% respectively. This is a clear evidence of the benefits offered by the acoustic bearing. Extended duration performance of the acoustic bearing under the load of 0.62 N is shown in Figure 4. Using recorded data shown in Figure 4 it is possible to find out that at the increased load on the bearing the shaft’s movements were reduced for both PZTs off and PZTs on. Resulting movement of the shaft in both X and Y directions is 6 µm for the case of PZTs switched off. For PZTs switched on shaft’s movement is equal to 5 µm in both directions. Therefore, the overall reduction of shaft’s displacement is 17%. Again, benefits offered by acoustic bearing are clearly visible. When the PZTs are switched on, shaft’s displacements in the X and Y directions are considerably reduced which points to the stabilising effect of the active PZTs.

3.2 Impulse force tests

For any air bearing it is quite important to know its response to impulse loading which can occur at any time of the bearing operation. Therefore, tests of the acoustic bearing were carried out during which an impulse force has been applied to the shaft. All tests were performed when the shaft was in a vertical position which means no load on the bearing.

Figure 5 illustrates the response of the bearing to the application of an impact load when PZTs are switched off. The bearing was at the rotational speed of 30000 rpm when the impulse load of 0.12 N was applied to the shaft. The moment of impulse load application is clearly visible as the rotational speed decreased and shaft’s displacements momentarily increased. After impulse load application there was no immediate return to the previous shaft’s dynamic. Displacements in the X and Y directions continue to be greater than those before the impulse load application and the record has characteristic serrated appearance.

For comparison, Figure 6 shows impact force test when PZTs were switch on. The reaction of the bearing to the application of the impulse force is quite different than that depicted in Figure 5. First of all the stabilizing effect of active PZTs is clearly noticeable. The moment of impact force application is marked by sharp momentary increase of the shaft’s displacements. However, they rapidly decayed within the next 2 seconds and the shaft return to its dynamics before the impact force application. Undoubtedly, the stabilizing effect of the acoustic pressure generated within the clearance space of the bearing is the main factor responsible for the observed behaviour. While impact force could eventually cause dynamic instability for the bearing operating with PZTs switched off (see Figure 5) there is no such a danger for the case when PZTs are switched on. This is yet another important feature of the acoustic bearing presented in this paper. Moreover, the ability of the acoustic bearing to quickly recover from a randomly applied impact force (a case quite often encountered in practical applications) means that PZTs could be active only for a short period of time. As it can be seen in Figure 6 after a period of recovery from instability the bearing returns to a stable running. Therefore, it could be suggested that PZTs ought to be active only when it is required to ensure bearing’s stable operation.

The ability of the bearing with operating PZTs stacks to stabilise the shaft’s motion and prevent a catastrophic failure due to dynamic instability is undoubtedly due to the squeeze-film acoustic pressure generation within the bearing. This pressure, substantially assisted by the deformed shape of the bearing’s bore, creates conditions preventing continuous growth of shaft’s orbit and rapidly stabilise the system. This explanation is fully justified by the commonly accepted theory of instability occurring in lightly loaded, high-speed air bearings [17].

4. Conclusions

The results presented in this paper justify the following conclusions:

(i) Acoustic bearing with PZTs switched on runs stable under a light load (0.31 and 0.62 N) and the rotational speed of 30000 rpm over a period of 10 minutes which indicates a steady-state of the system. The same is true for the acoustic bearing operating under similar load and speed conditions but with PZTs switched off.

(ii) The benefit of running a bearing with PZTs switched on are clear as, quantitatively, reduction of shaft’s motion is between 40% and 17% depending on the magnitude of the load applied to the bearing. For very light loads (0.31 N) the reduction is around 40% and for the increased load (0.62 N) this comes to 17%.

(iii) On the application of an impulse force to the shaft rotating at speed of 30000 rpm, acoustic bearing with PZTs switched on mitigates the impulse force effect rapidly and ensures that the dynamics of the shaft quickly return to the state before the impulse force application. This is not the case for the acoustic bearing with PZTs switched off.

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6. References

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