1-Dof Bearing Arrangement with Passive Radial Bearings and Highly Efficient Integrated Electrodynamic Dampers, EDD

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Abstract

This paper briefly describes a novel 1-DOF bearing arrangement with integrated passive electrodynamic dampers. A general problem with passive magnetic bearings is that sufficient damping in the radial direction is difficult to provide, without using additional rubber bushings or squeeze film dampers. Electrodynamic damping schemes are sometimes proposed, but they tend to have either weak or nonlinear properties, or they contribute to additional bearing instability in terms of negative stiffness.

The proposed bearing arrangement is inherently stable in terms of positive stiffness in radial direction. For dynamic stability it comprehends short circuit coils with geometries that allow for both damping coefficient and cut-off frequency to be chosen at will.

A discussion on alternative coil winding schemes and possible active control is done, and finally some preliminary test results from the first prototype are presented, indicating both a stable and a linear behavior.

1 Introduction

A general problem with passive and electrodynamic magnetic bearings is that sufficient damping in the radial direction is difficult to provide. Some inventors have achieved sufficient damping and dynamically stable levitation using electrodynamic dampers [1], [2]. Others use separate additional arrangements like rubber bushings or squeeze film dampers which are provided in order to supply adequate bearing damping functionality [3], [4], [5].

A general object of the present research project has been to improve damping of a rotating bearing arrangement in a passive and yet efficient manner. A further object for preferred embodiments is to provide possibilities to actively control the damping properties.

These objects are achieved by combining a conventional active magnetic thrust bearing with a reluctance type permanent magnet radial bearing. This has been done in such a way that an electrodynamic damper can be integrated into the magnetic circuit without violating the action of the active magnetic bearing. The electrodynamic damper further comprises at least one permanent magnet inducing a magnetic flux through a magnetic circuit that comprises the stator magnetic circuit and the rotor magnetic circuit. The rotor magnetic circuit is essentially rotationally symmetric with respect to the axis. The rotor and the stator surfaces exhibit varying reluctance in a radial direction. Thus any discrepancy from alignment between the stator actuator part and the rotor actuator part gives rise to large flux changes which will induce currents in the short-circuit coils enclosing the flux. In this way an efficient vibration damping action is achieved. However, should the stiffness or the damping action need to be altered, it is possible to connect these coils to an external control circuit.

One advantage with the 1-DOF-EDD system is that improved damping is provided to rotating bearing systems in a simple and energy efficient manner. The damping can be utilized as a passive damping, but allows also for different active damping solutions. The axial control is not necessary if axial stability is already achieved in the system. Thus the device can be used as a standalone damper. Due to the “magnetic switch” function of the

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reluctance bearing the damping coefficient is substantially higher than compared to conventional eddy current dampers found in literature consisting of permanent magnets and conducting plates/coils in relative motion.

2 Bearing Description

The bearing arrangement can be described as a conventional active magnetic axial bearing with permanent magnet bias which has been combined with a reluctance type radial bearing with two perpendicular sets of damping coils operating in the two radial directions. The teeth provide a pronounced reluctance force variation in radial direction, which is used for bearing stiffness and can be either stable or unstable depending on the teeth structure. By series connecting and short circuiting the coils two by two the optimum damping properties are achieved.

The axial active magnetic bearing itself will not be described in this paper since it is not novel art.

2.1 Bearing Parts

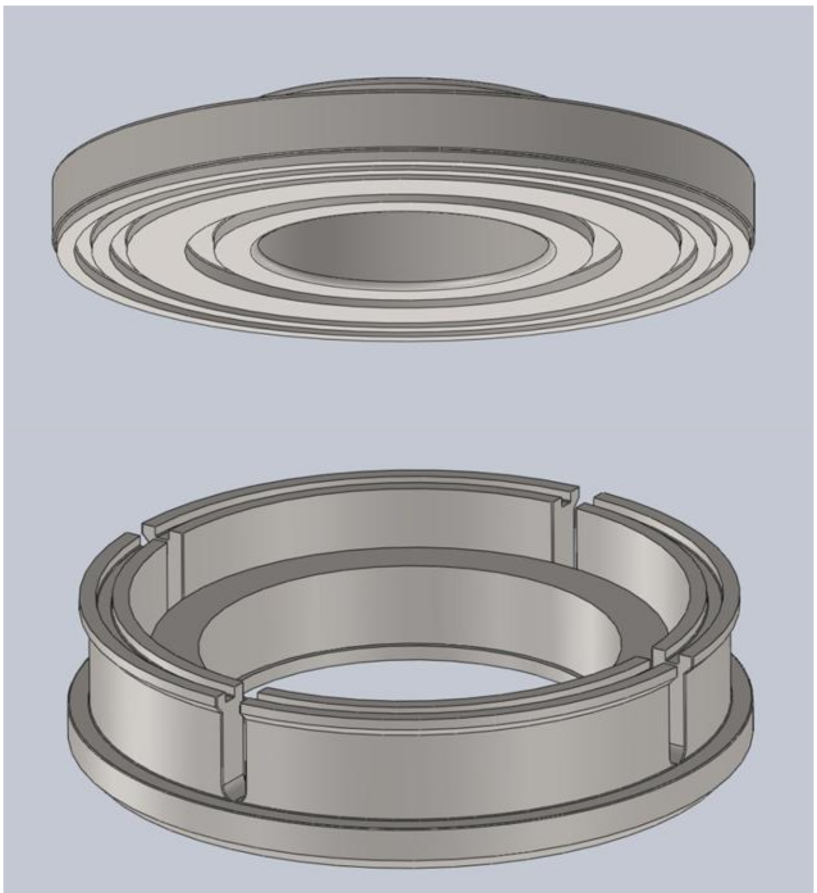

Figure 1. Iron rotor above and stator core without coils and magnet below.

Figure 1 shows the two main parts of the bearing: the solid iron rotor and the stator core which is a modified C-core. In the first prototype, which is mounted on a shaft in Figure 2, the core is made of solid construction steel, but the material might be changed in later designs. Inside the core an axially magnetized permanent magnet and/or an electromagnet is placed to magnetize the circuit. The prototype contains both, and can be described as an active axial bearing with permanent magnet bias. However, this coil is up to date only used for flux measurements.

The teeth are divided into four parts, each with slots for individual damper coils. The slots can be arbitrarily deep, within some limitations set by flux leakage, which gives the designer more freedom to choose damping coefficient than is possible with simple magnet/conductor damping arrangements.

2.2 Stator Winding

The prototype comprises five coils, one for the axial control and four for radial damping. The damper coils can be individually short circuited, but this would result in a cross coupling between radial and axial damping, which might or might not be desired. To avoid this they can be series connected two by two. Depending on coil polarity, the damping will be either radial or axial. Axial is not necessary, since the axial control has a separate coil.

For best damping response over a wide bandwidth, a single turn with large diameter wire would be best since it has low inductance and low resistance. However, since the coils in the prototype will also be used for measurements and later also for tests with active control, it is desired to increase the number of turns for better accuracy. 8 turns per coil was chosen. If a certain low pass filter cut-off frequency is desired, the bandwidth can be reduced by increasing the number of turns, and thus the ratio L/R.

2.3 Teeth Structure

The stator poles have a structure consisting of several teeth producing a variable reluctance in radial direction. The geometry of these affects both the reluctance force and the flux change through the coils. By the law of energy conservation, these effects are totally magnetically coupled to each other, but for better understanding of the stiffness and damping properties, we treat them separately. Thus the reluctance forces provide stiffness while the flux change results in induced voltage and damping currents. The relative teeth structure between the stator and the rotor decides whether

a) the arrangement is stable or unstable,
b) the stiffness is positive or negative,
c) the damping is linear or nonlinear.

For the prototype the goal was to find a teeth structure that provided positive linear stiffness and linear induced voltage, which for low frequencies will result in a linear damping/current relationship. If this can be achieved the arrangement will resemble a mechanical system with a linear spring and a viscous damper.

In literature several reluctance type bearings are presented, all of them having symmetrical rotor and stator teeth [2]. This solution was first tried, but turned out to give very nonlinear and weak damping properties, and the solution was discarded. Instead a suitable misalignment was introduced to linearize the properties, [6]. For the prototype the rotor teeth have a slightly smaller diameter than the stator rings. Figure 3, 4, and 5 shows that this is enough to achieve almost perfectly linear properties.

3 Test Rig

The prototype bearing was mounted on a shaft, Figure 2, in a pendulum arrangement. On the opposite side of the bearing, a small spherical ball bearing was mounted that allowed angular/radial movements of the magnetic bearing, but did not allow axial flexibility. Axial movements were instead provided by lowering the ball bearing which was mounted in a housing attached to an AEP transducer (wire gauge) for force measurements, which in turn was mounted in a milling machine. Axial measurements have this far been done at stand still and at low speed.

Radial movements were measured directly on the shaft with an eddy current sensor, and axial movements were measured with the built in scales of the milling machine. Radial forces have been measured in standstill with calibrated weights, and will later be measured more accurately with the use of the test rig described in [5], which also allows for measurements at very high rotational speed.

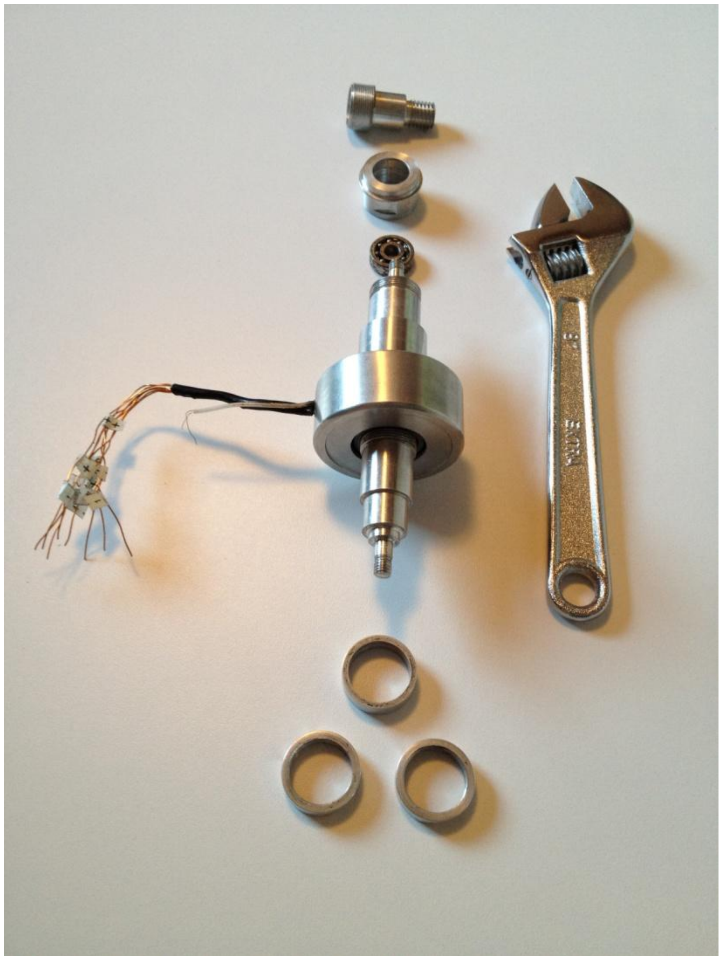

Figure 2. Test rig parts.

4 Results

Figure 3 shows the measured axial force versus displacement. Normally the axial displacement is limited to ±0,2 mm, but for this measurement the touchdown surface was removed on one side. Minimum gap on the other side remained 0.2 mm. Normal operating range is 0.2 to 0.6 mm, where it is relatively linear, but the measurements were extended to 0.2 to 3.3 mm.

Figure 4 shows the first very preliminary results regarding the force versus current relationship. The coils are series connected two by two. The result is not yet very accurate, but good enough to verify that the relationship is linear. A least square fit shows proportionality constant of 0.27 N/A with just a negligible second order term. The results also show some hysteresis, and the plotted data are mean values.

Hysteresis has a damping effect, but is less predictable than resistive losses. It also violates the position accuracy of the passive bearing.

Figure 5 shows the radial force versus displacement with zero coil current. It is very linear. The negative slope gives a positive (stable) stiffness of 24 N/mm.

Figure 6 shows the accumulated flux change through both series connected coils. This is directly proportional to the damping coefficient, [7], and it is also linear, which was one of the main goals with the research.

More results from ongoing experiments will be prepared and be presented in a coming journal paper.

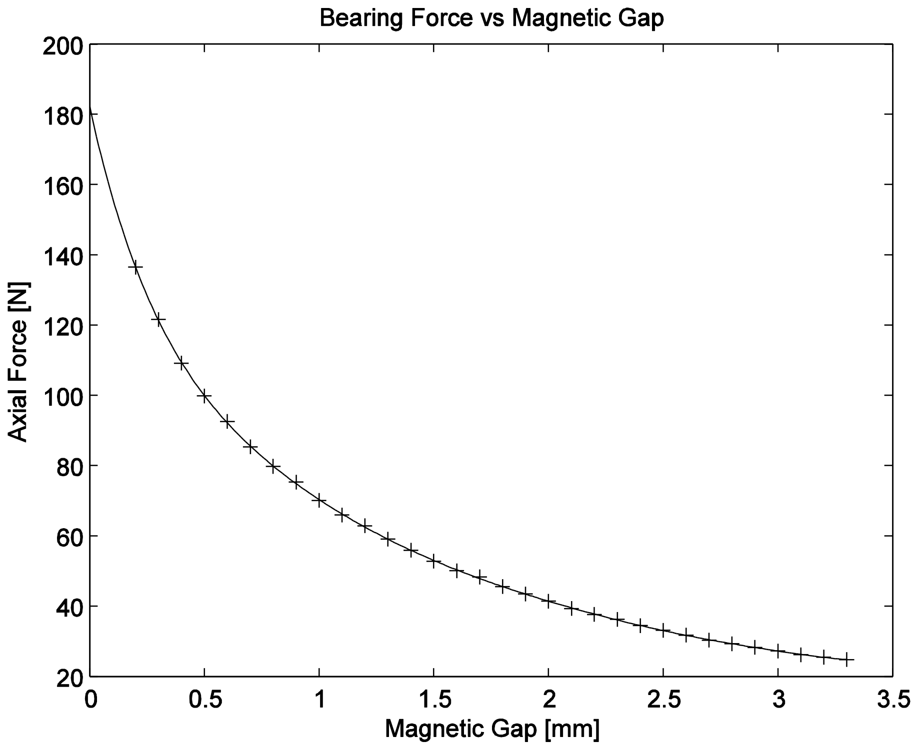

Figure 3. Attracting force versus air gap.

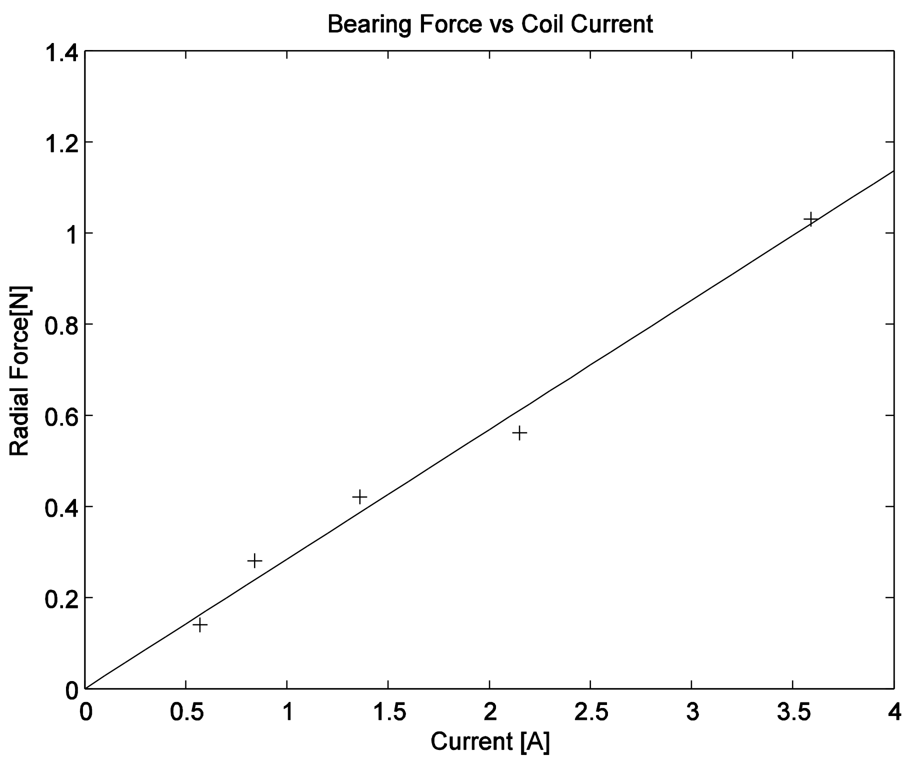

Figure 4. Measured force/current relationship.

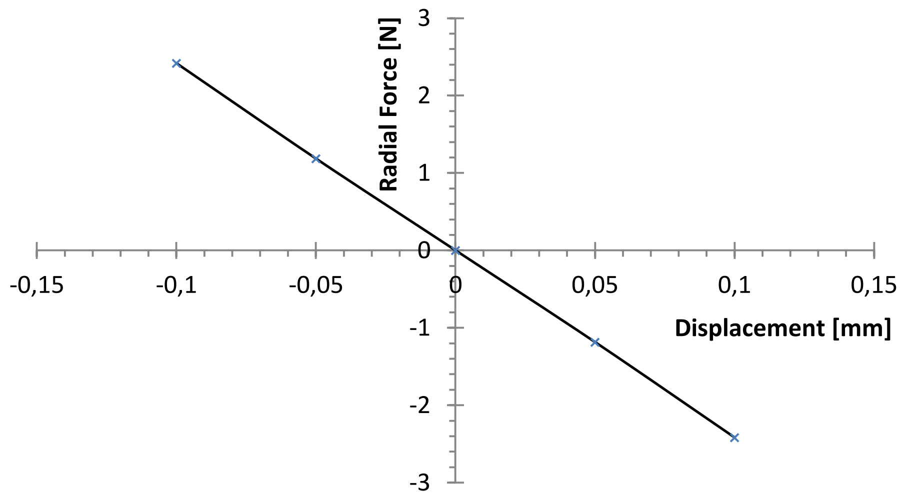

Figure 5. Radial force versus radial displacement using 2D-FEM.

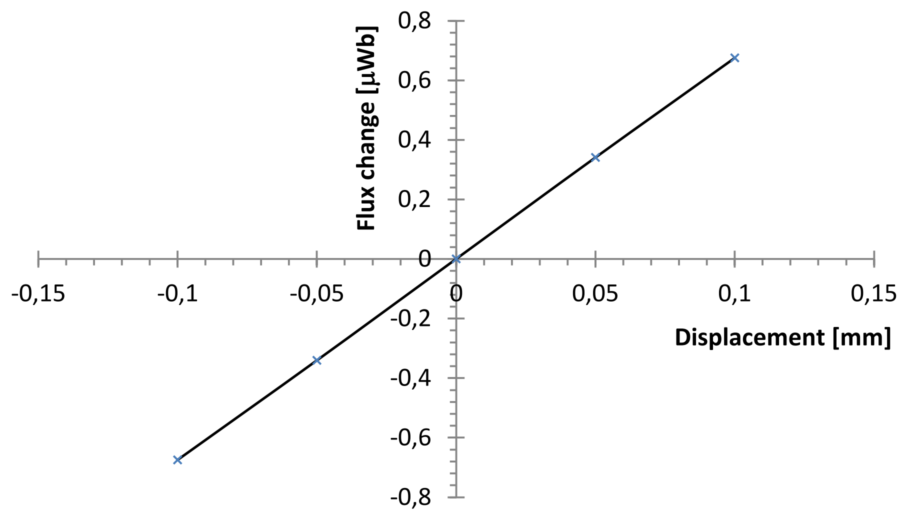

Figure 6. Accumulated flux change through damper coils.

5 Discussion on passive versus active control

This bearing was designed for passive radial operation, but it is obvious that the bearing might have advantages when operated as an active bearing. Suppose the bearing is designed to give the same positive stiffness as a particular conventional unstable active magnetic bearing, AMB. Suppose also that the winding has the same force current characteristic. Typically the AMB controller would be designed to “keep the natural frequency of the system” which means that the controller stiffness is double as high as the negative stiffness. Now suppose this controller is connected to the proposed stable bearing. The reluctance stiffness and the regulator stiffness would now add, not subtract, and the proposed bearing would have stiffness three times higher than compared to the active bearing, using the same controller. If this is realistic or not remains to be seen, but it certainly opens up for a new interesting research area.

6 Conclusion

A novel 1-DOF bearing with integrated EDD is proposed. Preliminary results show that the bearing is stable and that both stiffness and damping properties are linear. These results are very encouraging and suggest that further research on this bearing should be considered. Also active control in radial direction seem applicable, and the bearing will likely be the first reported bearing which is not unstable in open loop configuration, or even in case of controller breakdown.

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