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Vibration reduction in a hollow-shaft rotor using flexibly-mounted internal-stator magnetic bearings

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Abstract

A topology composed of a hollow-shaft rotor on rolling element bearings coupled with internal-stator magnetic bearings mounted on independently supported flexible shafts is considered for the purposes of vibration reduction. Details of a bespoke test rig constructed to assess this topology are presented. The rig makes use of custom-designed internal-stator magnetic bearings fabricated from Soft Magnetic Composite material. Results from standard numerical analysis of the rig are given to demonstrate the expected behaviour of the system, with a particular focus on its potential to reduce synchronous vibration seen while passing critical speeds. These behaviours are then confirmed via both impulse frequency response tests and rotating run-down tests. These results show that the proposed topology can be used effectively to alter the system vibration behaviour.

Key words: Magnetic bearings, Rotor dynamics, Control systems, Homopolar, Soft magnetic composite

1 Introduction

In the field of rotor system design, the occurrence of vibrations is one of the most significant factors limiting the available machine performance. Of the many techniques that have been proposed to address this limitation, one of the most capable and adaptable options is the inclusion of Active Magnetic Bearings (AMBs) in the system design. A considerable body of literature exists exploring the use of magnetic bearings in a variety of applications and with specific focus on various aspects of the design and control of such devices.

Early work in the field was focused on using a magnetic actuator as a "supplementary" bearing in a system already fully constrained on tradition bearings, for example oil-film bearings. It was found in this work (Kasarda et al., 1990; Nikolajsen et al., 1979) that a very substantial reduction in vibration amplitudes could be achieved through the used of these "Active Magnetic Dampers" (AMDs).

In the early research, meaningful full shaft levitation (i.e. with useful stiffness and damping characteristics) was challenging and expensive, due to lack of availability of key technologies: transistors for power amplification, microprocessors for real-time adaptive control etc. The rapid developments in these fields over the 1970s and 1980s provided a huge stimulus to development in the magnetic bearing field, and a number of works over those years explored the possibilities AMBs offer to rotor system performance (Bleuler and Schweitzer, 1983; Burrows et al., 1989; Schweitzer and Lange, 1976).

In subsequent work, various authors have addressed specific tasks and applications relating to the use of magnetic bearings: Clark et al. (2004) summarises work that has been done towards the use of AMBs in gas turbine engines; self-sensing (sensorless) techniques have been explored by Vischer and Bleuler (1993) , and Noh and Maslen (1997); control of rotor/stator contact events have been addressed by Keogh et al. (Cole and Keogh, 2003; Keogh and Cole, 2003; Keogh et al., 2004); Komori et al. consider systems incorporating superconducting magnetic bearings (Komori and Shiraishi, 2003; Komori et al., 1996, 1998).

However, a commonality among the vast majority of this work is the basic design and construction of the magnetic bearings, and the topology of the rotor system. The magnetic bearings are usually large, heteropolar, laminated steel constructions fixed rigidly to the machine base. This work considers a more uncommon topology.

The authors have presented (Lusty et al., 2016) a novel concept for a rotor system involving a hollow-shaft rotor together with a concentrically aligned secondary shaft running through the centre. The two shafts are supported entirely independently, the rotor on rolling element (or other conventional) bearings, and the secondary shaft either clamped or pinned. The secondary shaft is used to support one or more internal stator magnetic bearings, which may be used to couple and decouple the rotor to the secondary shaft as required. Such a system presents a number of advantages over traditional magnetic bearing systems, for example, no space is taken up on the outside of the rotor by the AMBs, potentially allowing shorter rotors to be employed, and freedom of placement for working components (turbines, gears etc.). In addition, as they are not used for levitation, the AMBs may be turned on only when vibration control is specifically required (e.g. when passing critical speeds), making for a more energy efficient system. In principle, however, the passive rotor bearings could be removed to yield a levitated rotor.

2 Test Rig

A design for a bespoke test rig demonstrating this principle has been numerically modelled and presented by the authors (Lusty et al., 2015) - an illustrative view of this test rig is shown in Fig. (1).

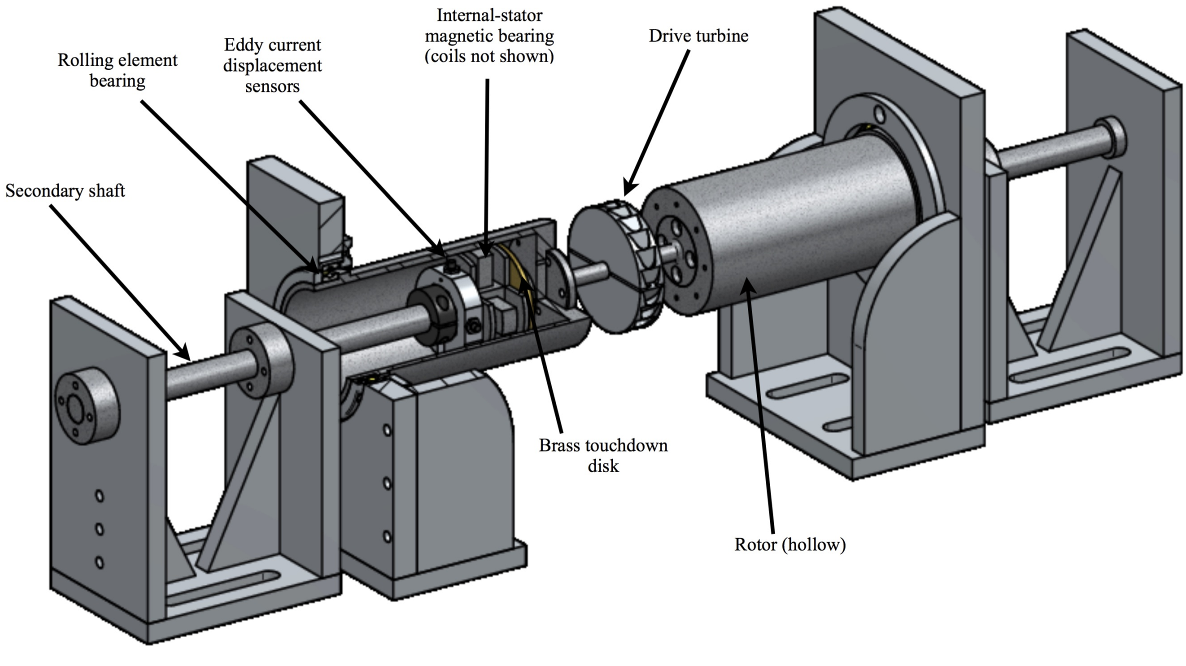


Fig. 1: Cut-away section of test rig demonstrating internal-stator magnetic bearing layout

The test rig is based on a rotor constructed from two hollow sections of tube connected in the middle by a thinner, solid bar. The purpose of the thin central section is to give the rotor flexible characteristics at low enough speeds to be practical in the laboratory. This central section also supports an air-powered impulse turbine, used to rotate the shaft. The rotor is 0.5 m long, with rolling element bearings at either end.

Two secondary shafts are inserted concentrically into the hollow sections of the rotor - one at either end. These shafts are supported entirely independently of the rotor, and do not themselves rotate. The shafts form cantilever beams, and mounted at the free ends are the magnetic bearings, displacement sensors and a touchdown surface.

The magnetic bearings are of a four-pole-pair homopolar design - see Fig. 2. The homopolar design allows the omission of the customary laminated collar on the rotor, as eddy current losses in the rotor are minimised by this pole configuration. Experimental comparison of homopolar vs. heteropolar power losses is provided by Kasarda et al. (1999). In order to manufacture the bearing stator in the most compact way possible with the most complete insulation against stator-based eddy currents, the core is fabricated from Soft Magnetic Composite (SMC). A good overview of the properties and uses of SMCs is provided by Shokrollahi and Janghorban (2007), and a specific example of their use in a magnetic bearing application made by Fleischer and Hofmann (2011, 2015). A brass touchdown disk is mounted adjacent to the magnetic bearing to protect it in the event of excessive vibration or instability.

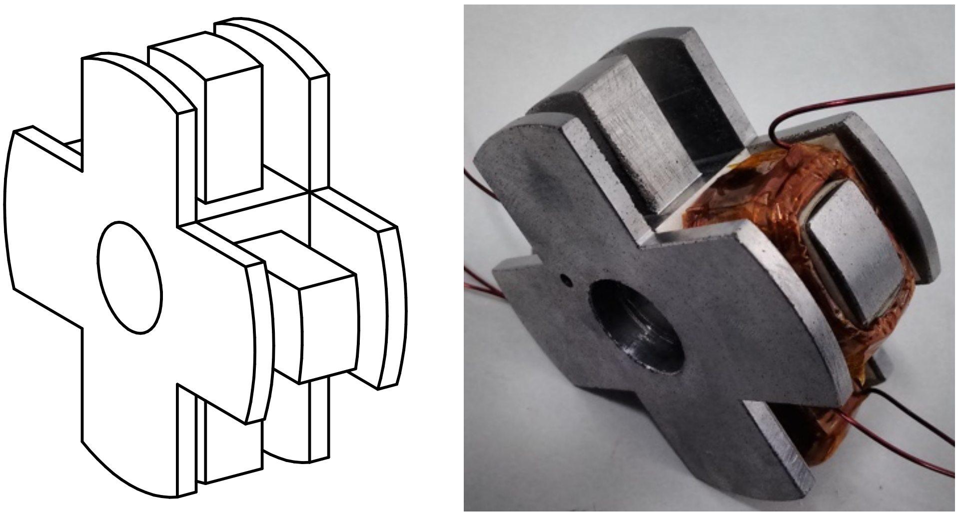


Fig. 2: Illustrative and photographic view of internal-stator homopolar magnetic bearing stator used in test rig. The bearing is fabricated from a Soft Magnetic Composite material

Eddy current displacement sensors are also mounted on the secondary shaft at this location, proving the feedback used in the magnetic bearing control loop.

3 Numerical Modelling Results

The initial analysis consists of a finite element model of the rig, on which eigenvalue and mode shape predictions can be made. For the test rig consturucted, the first system natural frequency when the magnetic bearings are inactive is predicted to occur at 47 Hz, and is associated with a classic first bending mode shape in the rotor as expected from standard beam theory, shown in Fig. (3). When the magnets are activated according to a proportional/derivative (PD) controller with parameters as shown in Table (1), the systems first natural frequency rises to 52 Hz, and the mode shape associated with this is shown in Fig. (4). Importantly, deflection of the flexible secondary shafts is now expected.

Parameter	Value
Bias Current (A)	3
Gain K_P	10,000
Gain K_D	20

Table 1: PD Controller Parameters

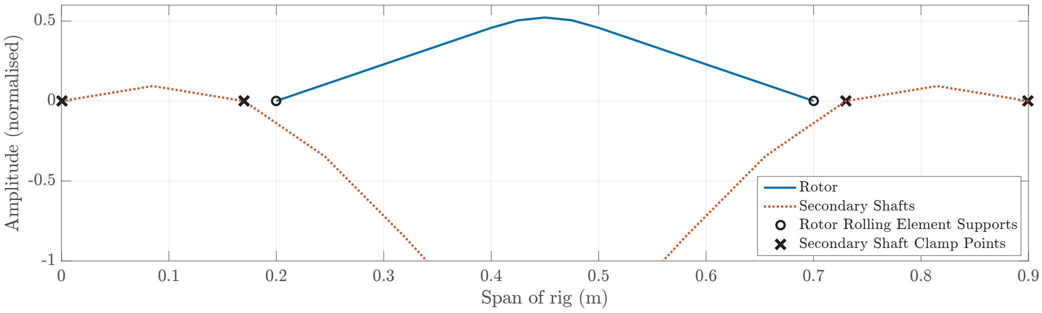


Fig. 3: Shape of first bending mode of system without magnetic bearings active

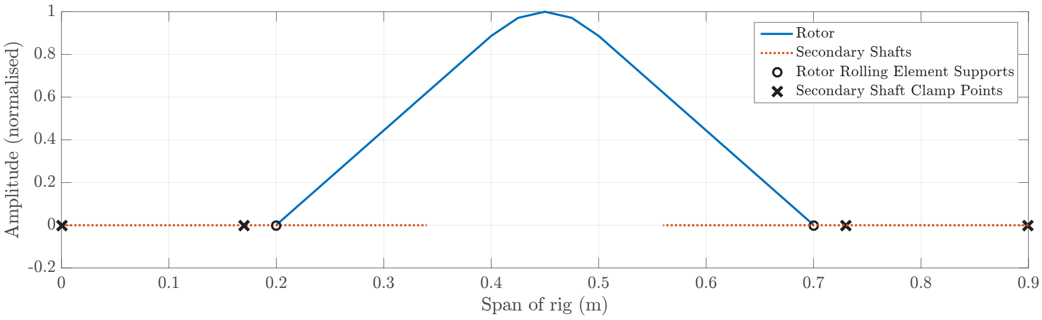


Fig. 4: Shape of first bending mode of system with PD control in magnetic bearings

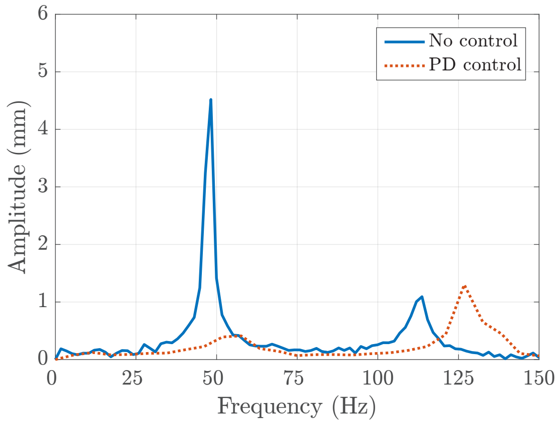
4 Impulse Testing

In order to verify practically the results of the natural frequency analysis, the test rig was subject to impulse response frequency analysis. To do this, the magnetic bearings were used to deliver an impulse input to both the rotor and the secondary shaft simultaneously, with eddy current sensors at two locations used to capture the response. The first sensor is mounted on the secondary shaft and observes the internal surface of the rotor. Thus this sensor captures a response of the relative motion of between the rotor and the secondary shaft. The second sensor used is mounted rigidly to the rig base plate, and observes the external surface of the rotor.

The impulse response is recorded in two scenarios - firstly with the magnetic bearings inactive, and then again with the PD controller (gains as above) active. Fourier transforms of the results are presented in Fig. (5).

In the first case, it is seen that the sensor mounted on the secondary shaft detects resonances at around 48 Hz and 112 Hz - i.e. one resonance associated with each of the rotor and the secondary shaft, as predicted by the analysis. In contrast, the externally mounted sensor only detects a resonant peak at around 48 Hz, which is the predicted rotor first bending frequency.

It is observed clearly that a shift in the system first natural frequency is caused by the use of PD control in the magnetic bearings, to a value around 55-60 Hz. This matches well to the analytically predicted change. There is also a substantial decrease in the amplitude of the response at this frequency, indicating a significant increase in system damping.



(a)

(b)

Fig. 5: Fourier transforms of the impulse response of the rotor system presented illustrating the effect of a PD controller; sensors used for the readings are (a) an internal sensor mounted on the flexible secondary shaft and (b) a rigidly mounted sensor monitoring the external surface of the rotor only

5 Rotating Tests

Run-down tests were performed to assess the ability of the test rig to reduce vibration amplitudes in the rotor. The rotor was spun up to a speed of 3150 rpm before the gas supply was shut off, and the run-down behaviour recorded. This speed is slightly below the first critical speed of the rotor, and without the magnetic bearings active, touchdown activity would occur if a higher speed is used.

Time series displacement responses are shown in Figs. (6) and (7) for the run-down tests. These responses are provided by the rigidly mounted external sensor, and thus show the absolute displacement behaviour of the rotor.

The response in Fig. (6) shows the behaviour of the rotor when the magnetic bearings are completely inactive. It is observed that the maximum amplitude of vibration (seen at $t = 0$) is a little over 0.3 mm. The large amplitudes decay as the rotor spins down, the speed moving further from the critical speed. As the run-down proceeds, some sub-synchronous peaks are seen in the response, particularly at around $t = 35$ and $t = 50$.

In Fig. (7), when the PD control scheme is active, some clear reductions in vibration amplitude are observed. The maximum amplitude of vibration seen in the case is 0.2 mm - a 33% reduction compared to the case with no control. It is seen further that the sub-synchronous peaks are substantially reduced by the control action.

6 Conclusions

This work has presented a test rig designed to illustrate a novel geometry of a system using magnetic bearings for vibration reduction. Specifically, the system involves using internal-stator magnetic bearings mounted on flexible shafts to alter the rotor frequency response.

Results from finite element modelling are presented, showing an increase in the frequency of the predicted first flexible mode of the system caused by the magnetic bearings under a PD control scheme, and a the resulting change in the system mode shape (eigenvector).

Two sets of tests have been presented to verify experimentally these analytical results. Firstly, a frequency response to an impulse clearly shows both a change in system natural frequencies and reduction in vibration amplitudes (around the first natural frequency). Rotating run-down tests then demonstrate conclusively the effectiveness of the of the

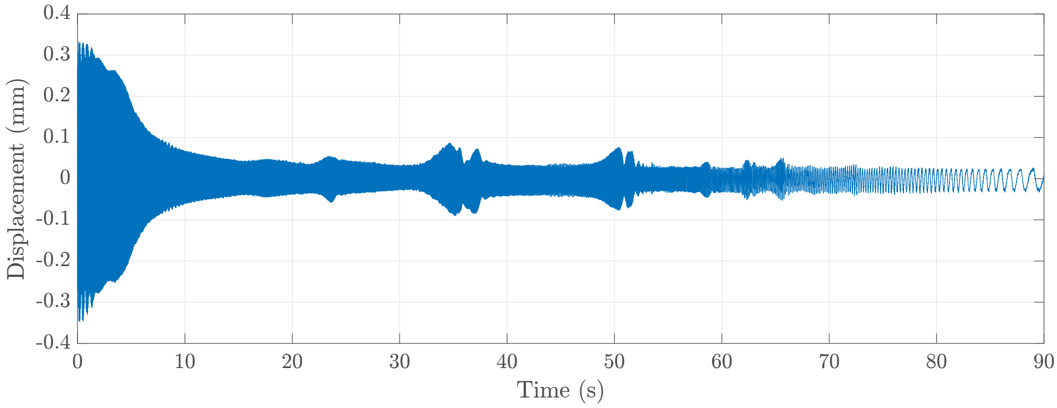


Fig. 6: Run-down without magnetic bearings active (external sensor)

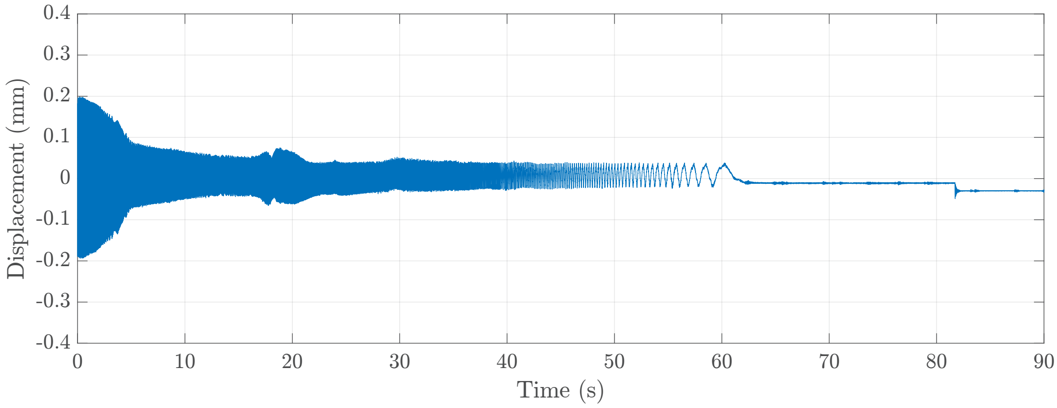


Fig. 7: Run-down with PD control applied (external sensor)

design, providing evidence of significant reduction in vibration amplitudes for both synchronous and sub-synchronous excitation.

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