ON THE USE OF FLEXIBLY-MOUNTED, INTERNAL-STATOR MAGNETIC BEARINGS FOR VIBRATION CONTROL OF A FLEXIBLE ROTOR

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ABSTRACT

Active magnetic bearings (AMBs) present a host of potential advantages over other bearing types in the design of rotor systems. An important category of such advantages is the ability to use the active nature of AMBs to influence the rotor-dynamics, and thus to control and reduce vibration exhibited by a rotor. The majority of work on this topic employs the same basic system geometry: the magnetic bearings are external to the rotor, and they tend to be large and rigidly mounted. In some situations it may be necessary or desirable to use a more compact arrangement, with AMBs mounted inside a hollow rotor, and mounted on a flexible structure. A system with such a topology is considered in this paper.

1 INTRODUCTION

The use of magnetic bearings for vibration control has been an active field of research for over half a century, and a number of authors have undertaken to summarise the progress in the field at various stages with significant review papers [1–3].

Throughout this period, however, the fundamental geometry of AMBs and of the overall rotor systems they are employed in has remained relatively stationary. From the use of magnetic bearings as active magnetic dampers (AMDs) [4–6] to the development of competent and efficient vibration controllers [7, 8], AMBs have tended to be large, externally and rigidly mounted components. However, in systems where space on the rotor surface or in its vicinity are limited, it may be advantageous to employ an alternate system geometry. In particular, the prospect of mounting magnetic bearings within a hollow-shaft rotor is substantially interesting. In many cases, such a geometry dictates that the structure supporting the AMB will be flexible, which complicates the system dynamics, as well as raising interesting new possibilities for vibration control.

2 TEST RIG

To allow an experimental investigation into the behaviour of a rotor system with flexibly-mounted, internal-stator magnetic bearings, a customised test rig has been constructed. A schematic showing the fundamental nature of the test rig is shown in Figure 1, while a photographic view of the finished test rig is shown in Figure 2.

The rotor is constructed with a variable cross-section, with large diameter, hollow ends, and a thin and solid centre. The thin section is present purely to lower the rotor's natural frequencies to allow supercritical speeds to be reasonably achievable in the laboratory. The magnetic bearings (marked "Active Coupling" in Figure 1) are mounted at the end of cantilever beams. In this example, the magnetic bearings are not used for levitation of the rotor, which is mounted at its ends on traditional passive bearings. Instead, the magnetic bearings are in place solely for the purpose of vibration control. In theory, however, the bearings could be used for a combination of both levitation and vibration control.

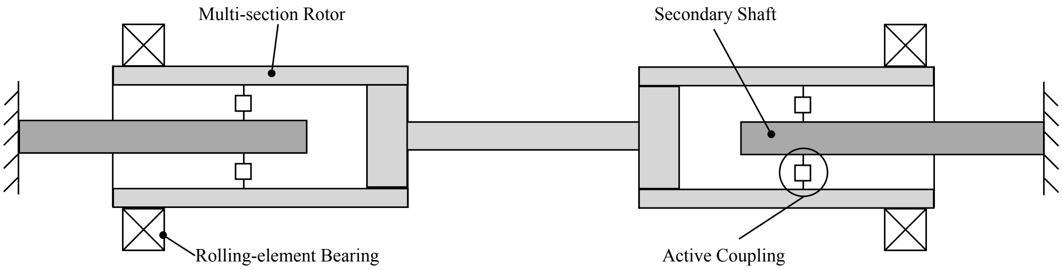


Figure 1: Schematic diagram illustrating layout of flexibly-mounted magnetic bearing test rig

3 CONTROL AND RESULTS

A key challenge is reducing the rotor vibration without exciting the magnetic bearing supports to vibrate excessively, and especially to avoid the combined vibration of rotor and magnetic bearing shaft causing contact between the rotor and the magnetic bearing to occur. From a system design point of view, it is therefore of great importance to ensure a substantial difference between the natural frequencies of the separate shafts.

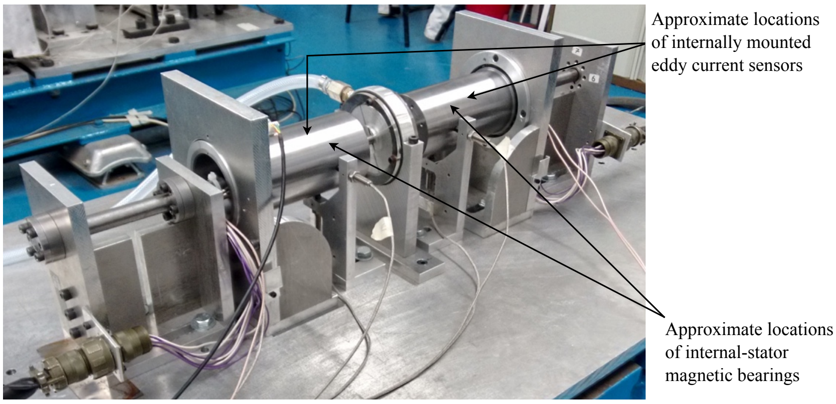


Figure 2: Photograph of flexibly-mounted magnetic bearing test rig

As the design under consideration uses the magnetic bearings purely for vibration reduction (and not for levitation), it is possible to only activate the bearings when excessive vibration behaviour is occurring in the rotor - for example when passing critical speeds.

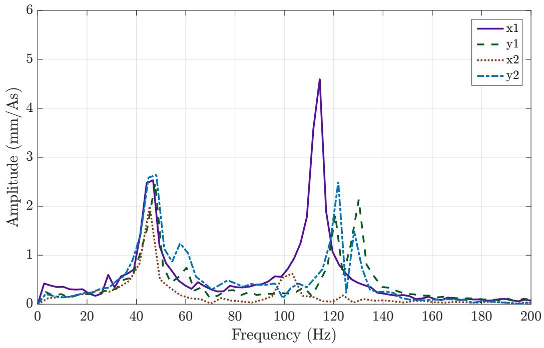
In terms of particular control strategy, a variety of options are available. With carefully selected gains, a traditional PD controller can be employed to couple to rotor to the secondary shafts, thus altering the rotor behaviour. The controller may be devised, for example, to add stiffness to the rotor, thus shifting its natural frequencies, and therefore critical speeds. An interesting alternative to this may be not to use a closed loop controller at all, and instead just apply a bias current to all poles of the magnetic bearings, effectively reducing the rotor stiffness. An interesting study of such a techniques has been presented by Mahfoud and Der Hagopian [9].

A broader range of options are opened up by considering the use of model based controller for such a geometry. This allows the expression of specific goals; for instance, a controller could be designed to minimise the absolute vibration of the rotor as far as possible, while allowing the flexible magnetic bearing shaft to vibrate *provided* such motion will not cause a contact event. While more complex to design, such control schemes offer powerful capabilities to a system with flexibly mounted magnetic bearings.

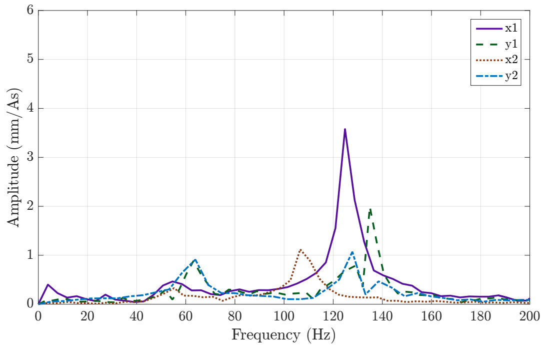
By way of illustrating the capability of such a system topology for reducing vibration, some results from impulse testing on the rig in Figure 2 are shown in Figure 3. The impulses are measured by eddy current sensors (four in total) mounted adjacent to the magnetic bearings on the flexible support shafts. Thus they contain data relating both to the rotor (the target), and the shafts they are mounted on. The data presented is a Fourier transform of the time responses. The peaks around 45 Hz pertain to the rotor behaviour, while those around 120 Hz relate to the support shaft natural frequency. It is seen that the use of PD control in the bearings (3b) substantially reduces the vibration amplitudes compared to the uncontrolled system (3a), as well as increasing the frequencies at which they occur. These frequency increases are a result of the additional stiffness contributed by the magnetic bearings when operated under PD control. The extent of the frequency shift is governed by the value of the proportional gain. It is also possible to cause negative frequency shifts, achieved by using the magnetic bearings to add negative stiffness to the system, as done by Mahfoud and Der Hagopian [9].

4 CONCLUDING REMARKS

An atypical topology for a rotor system using magnetic bearings for vibration reduction has been presented, involving mounting internal-stator magnetic bearings within a hollow-shaft rotor. The AMBs are mounted on flexible beams. Options for using such a system for vibration control are considered, and example results illustrating the potential of such a topology are



(a)



(b)

Figure 3: Fourier transforms of the impulse response of the rotor system comparing the behavour without control (a) and with a PD controller (b)

included.

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