An Integrated Active Rotor/Active Magnetic Bearing System

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Abstract
The concept of using internal bend actuation in a rotor supported by active magnetic bearings is proposed. Since the bend occurs within the rotating frame, static bend actuation is equivalent to synchronous excitation from an actuation component based in an inertial frame of reference. This has the advantage that low bandwidth Active Magnetic Bearing (AMB) levitation can be combined with low bandwidth bend actuation for synchronous vibration control. The design of the bend actuator is presented together with the principle of operation. The bend actuation is commanded wirelessly, with low power required to operate lead screw motors from an onboard battery. This is demonstrated experimentally in static tests in which rotor deformation under bend actuation is measured using a laser tracker. Consideration is then given to vibration reduction during rotating tests. The rotor was levitated on AMB's and the synchronous vibration control achievable by static internal bending is demonstrated. The optimal level and orientation of bend actuation was achieved by applying search techniques. These were applied at increments of rotational speed in run-up tests and shown to be particularly effective in compensating for unbalance response at the rotor mid-point.

Keywords : Active rotor, Active magnetic bearing, Low bandwidth control, Integrated system, Wireless rotor command, Rotor bend control.

1. Introduction

Rotating machines are generally balanced to run smoothly over a specified operating speed range. Balancing procedures are well developed according to API 684 (2005) and ISO 21940-11 (2016) balance quality G grades. If the balance condition changes during operation, either by rotor added/subtracted mass, or thermal variations, it may be necessary to take a machine out of service for rebalancing. In the case of hot or cold state variations, a compromise will be necessary to achieve an acceptable balance condition. In optical disc drives, the use of automatic ball balancers is implemented (Huang et al., 2002), however, their application is limited to rigid rotors. For rotors operating in the flexible regime, the balls may not adopt to stable orientations. For flexible rotors, Olsson (2004) considered the limitations of auto-balancing. Discs with heavy spots have also been proposed to adjust the rotor balancing conditions during operation (Van de Vegte and Lake, 1978, Van de Vegte, 1981). This technique has been applied to industrial rotor systems (Pardivala et al., 1998, Alauze et al., 2001, Fan et al., 2014).

Passive bearings can also be used to reduce rotor vibrations (Bonello et al., 2004). However, oil whip is a limiting factor at high rotating speed (Newkirk and Taylor, 1925). Due to reduced viscosity operation, air bearings can operate at very high speeds (Belforte et al., 2006), but are susceptible to wear at low speeds or high loads, which can be overcome through the use of solid lubricants (DellaCorte et al., 2004). For oil operated journal bearings, the pressure can be adjusted to make the bearings active (Santos, 1994). Estupinan and Santos (2012) had success applying this approach to combustion engines. Rolling element bearings have also been made active by using shape memory alloy springs to control their stiffness (He et al., 2006a, 2006b). High frequency low amplitude piezoelectric actuators have also been used effectively (Rong et al., 1990, Heinfel et al., 2017).

Active Magnetic Bearings (AMB's) provide another path to active support control with well-documented design methods (ISO 14839-2, 2004, Bleuler et al., 2009), together with advanced control (Defoy et al., 2014). However, AMB's

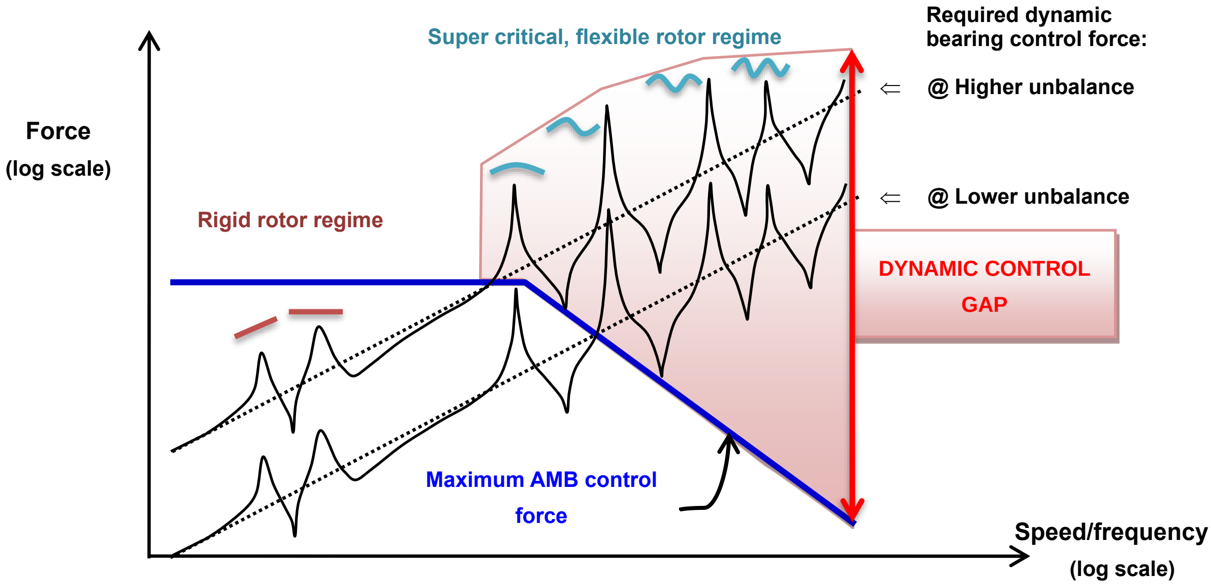

Fig. 1 Schematic showing why active magnetic bearings are unable to apply forces for precise rotor position control above threshold speeds (log scales assumed). Even if the dynamic control gap could be bridged to cover all flexural mode resonances, resulting transmitted forces could be excessive.

have constrained bandwidth and force capacity due to power amplifier limits and magnetic flux saturation. Hence, the level of rotor dynamic control of synchronous vibrations is speed limited. Piezoelectric stacks provide a much larger force capacity, hence, are more effective in controlling rotor dynamics (Tang et al., 1995) though remaining as conceptual external devices due to complexity and additional cost.

On-shaft control can be thermally driven, with slow scale localised shaft heating demonstrated to control the bend of a rotor and minimise vibration (Olsson and Wildheim, 1977). Although not taken up by industry due to the added complexity of implementation, it is a novel concept. The bandwidth limitation of external devices, such as bearings, for rotor dynamic control may be overcome by applying on-shaft control internal to the rotor. Static forcing applied onboard a rotor is equivalent to synchronous forcing from an inertial non-rotating frame of reference. Piezoelectric material has been applied to a rotor in the form of external patches bonded to its surface (Przybylowicz, 2002). Control of resonance has been demonstrated experimentally by Horst and Wolfel (2004) and Sloetjes and De Boer (2008). However, force capacity is very limited by the two-dimensional nature of a piezoelectric patch.

State-of-the-art rotors are usually composed of passive metallic shafts on which are mounted working sections (motors, generators, turbine/compressor discs, flywheels, laminations, machine tools, etc). It is impossible to manufacture these with absolute precision and operational issues may also cause tolerance deviations due to thermal distortion or mass loss/deposition. Design procedures for rotors equipped with AMB's are covered by recognised standards (ISO 14839, 2002, API 617, 2014). However, for a given grade of rotor balance, there will inevitably be a threshold speed above which the limited force capacity of AMB's will prevent complete control of rotor position. Even if control capacity is available at high rotor speeds, holding the rotor position may result in excessive force transmission through the AMB's. For these reasons, flywheels are typically considered as rigid and allowed to spin about their principal axes to allow zero force transmission. However, this may not be possible for other rotor types running in the flexible rotor regime.

The limitations for unbalance control are captured in Fig. 1. AMB's allow feedback control of rotor position or transmitted force. Rotor position control is limited at high speeds due to the inability to compensate for the dynamic control gap shown in Fig. 1. Bandwidth limitations at high frequencies arise from power amplifier constraints. Nonetheless, AMB's can operate in a vacuum and hot environments (Provenza et al., 2005). A rotor surface tangential speed limitation arises from material strength considerations, typically around 300 m/s (Schweitzer and Maslen, 2009, Ch. 6). Hence design changes to increase force capacity by increase of rotor/bearing diameter will decrease maximum rotational speed capability.

The contribution of this paper lies in an innovative design of a bend actuator that is mounted internally within the shell of a hollow rotor. Viewed externally, the rotor resembles a standard passive rotor, however, it is possible to apply

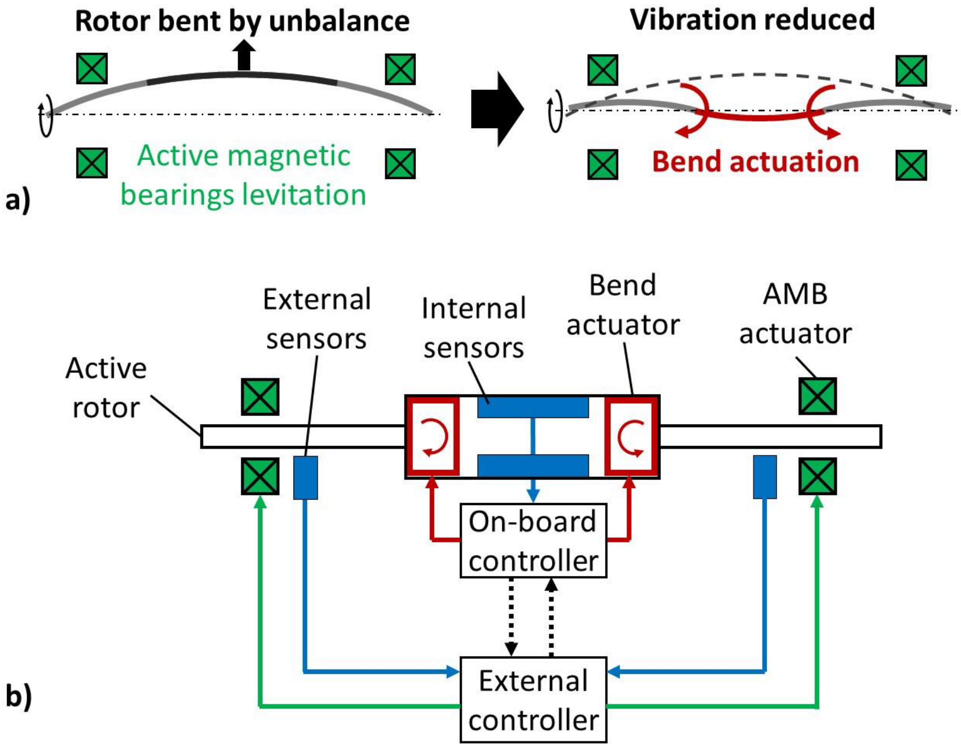

Fig. 2 Schematic of integrated active rotor AMB concept. a) Shows a rotor levitated by AMBs and deformed by rotating unbalance (left) and under bend actuation (right). b) Shows integrated control of AMBs via displacement sensors for levitation and by wireless commands to the onboard bend actuator. Internal sensors are incorporated to detect strains.

internal actuation to bend the rotor is any circumferential orientation. Commands may be sent to the rotor wirelessly. This adaptability of the rotor has potential advantages for controlling the rotor dynamic response. The details of the bend actuator are presented together with its manufacture as a prototype system. The bend actuation is demonstrated statically and when implemented in a spinning laboratory rotor dynamic system levitated by AMBs.

2. Integrated active rotor active magnetic bearing concept

The proposition is to control higher frequency rotor flexural modes by introducing internal bend actuation inside a rotor. In this way, AMB control at a frequency that is synchronous with the rotational speed may be replaced by internal bend actuation that is static in the rotating frame of reference. The potential then exists for low bandwidth control for levitation and control of rigid body modes by AMB's, and low bandwidth internal rotor control for flexible modes. The overall system would therefore have no requirement for high bandwidth amplifiers. Figure 2 shows the conceived layout of the integrated active rotor AMB system.

2.1 Internal bend actuator

The principle of achieving bend actuation in a single plane is shown in Fig. 3. A lead screw motor, located on the axial centerline of a rotor, drives a small double acting piston of area, A_p, within a chamber containing pre-pressurised hydraulic fluid at pressure, p_0. When the piston is centralised, both hydraulic lines are at the same pressure, hence, the hydraulic forces applied to the four rams are equal and the bending moments, M_C, are zero. If the piston moves to the right as shown, the top hydraulic line has increased pressure, $p_0 + \Delta p$, and the bottom line has decreased pressure $p_0 - \Delta p$. In this case, $M_C = 2\Delta p A_r L_r$, where A_r is the ram area and L_r is the ram radial offset. Reaction tubes are used to prevent relative movement of the ram chambers. The reaction tubes may also have strain gauges attached to provide an indication of the forces applied by the rams to the flanges shown. An equivalent system may be implemented to achieve bend actuation in an orthogonal plane.

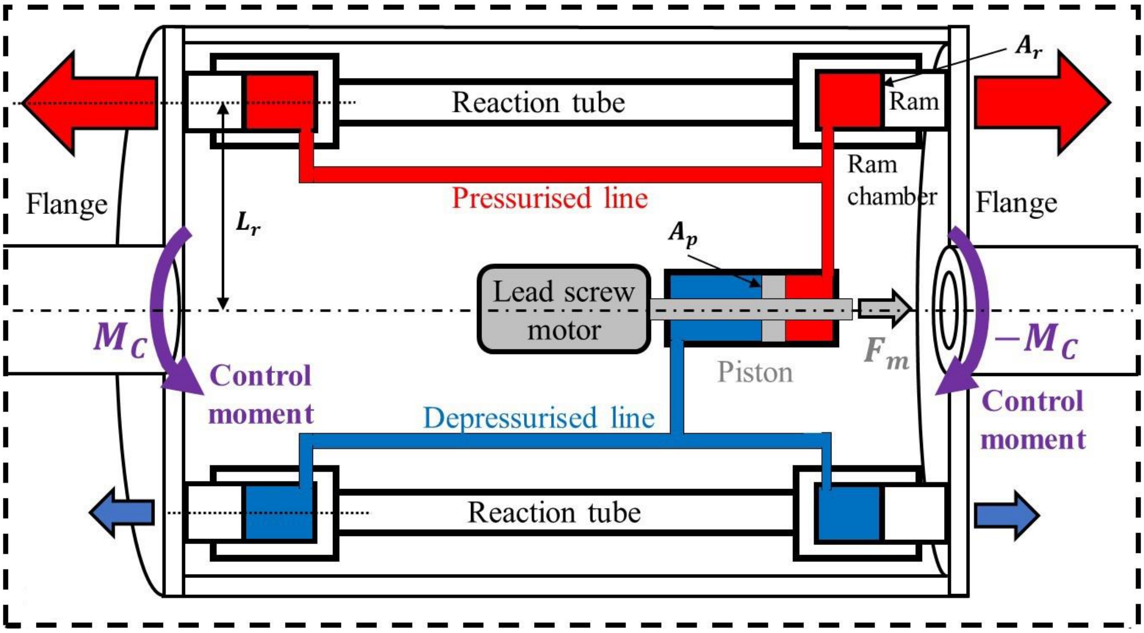

Fig. 3 Schematic of the bend actuation in a single plane.

The intention is for the bend actuation to be slow acting or pseudo static so that minimal power is demanded by the lead screw motors during operation. Hence, it is possible to achieve effective synchronous control using an onboard battery. Other pertinent features include:

(i) Pre-pressuring the hydraulic lines allows a differential driving mode of operation without causing hydraulic fluid cavitation.
(ii) There is a need to use hydraulic fluid with minimal dissolved air to achieve good force transmission in the hydraulic lines. Dissolved air may be removed using a vacuum chamber.
(iii) Carbon fibre reaction tubes may be used to minimise added mass offset radially.
(iv) Seals on the piston and rams are required to prevent hydraulic fluid leakage and loss of system pressure.

The bend actuator design parameters chosen are presented in Table 1.

Table 1 Design parameters associated with the bend actuator system

Maximum lead screw motor force	222 N
Piston inner diameter	6 mm
Piston outer diameter	8 mm
Ram outer diameter	20 mm
Hydraulic fluid pre-pressure	101 bar
Ram/piston area ratio	14.3
Maximum ram force	3171 N
Ram radial offset	60 mm
Maximum bending moment	190 Nm
Maximum piston/lead screw motor displacement	10 mm
Maximum ram displacement	0.35 mm

The bend actuator was fabricated and is shown in Fig. 4. It consists of two systems of the type shown in Fig. 3 arranged orthogonally, with a central disc for additional support. The strain gauged reaction tubes are labelled, e.g. DE, LE, and the high strength copper alloy hydraulic lines are visible. The manifold blocks at either end contain the rams,

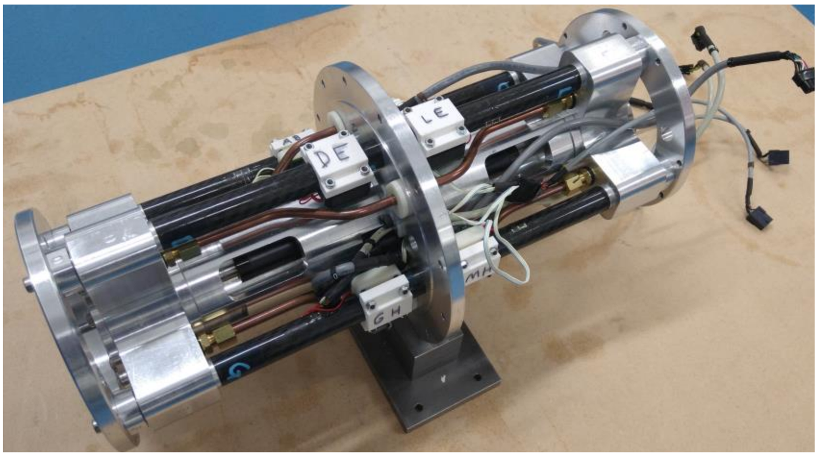

Fig. 4 The fabricated internal bend actuator.

which are bolted to annular push discs. Motor power cables and strain gauge cables are also shown. The manifold blocks, motor housings and discs were of aerospace grade aluminium alloy construction.

2.2 Bend actuator integrated into the rotor/AMB system

The bend actuator was fitted into the hollow section of the rotor as shown in Fig. 5 before being bolted up. The rotor is segmented and consists of steel supporting shafts on which AMB cores and touch-down bearing landing sleeves are mounted. The supporting shafts are also hollow to allow cabling from the bend actuator to be passed through. The central shell segments enclosing the bend actuator were machined from aerospace grade aluminium alloy. Overall, the rotor was 1.2 m long and had a total mass of 19.3 kg, of which 5.5 kg was from the bend actuator.

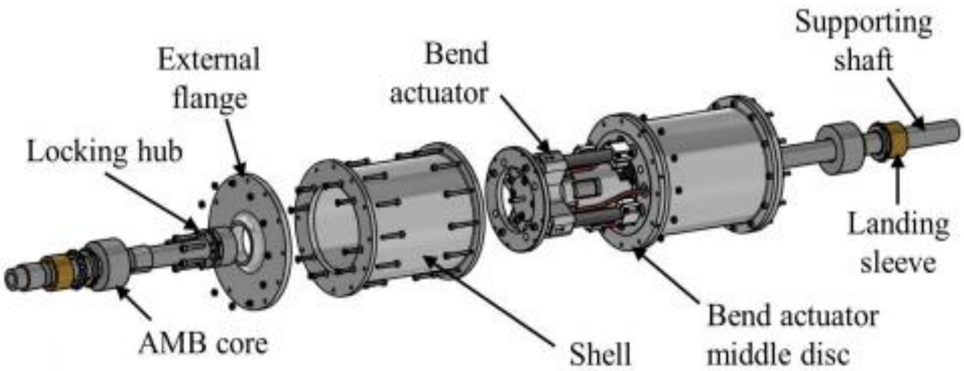

Fig. 5 Exploded view of the rotor and its internal bend actuator system.

Prior to assembly in the AMB's, a static test was undertaken with the rotor supported on chocks as shown in Fig.6. A wireless communication module with battery is shown in yellow. Fourteen laser tracker targets were located axially on the top surface of the rotor and the bend actuator was commanded to activate at various angles involving both orthogonal planes. The square-like deflections recorded by the laser tracker indicate central deflections of $\pm 20\ \mu$m while at the support points are effectively zero, as expected.

The AMB's were of a standard 8-pole radial type and for the purpose of this system were under PID control only. Relevant parameters are shown in Table 2. Figure 7 shows the assembled rotor on its AMB's. It also includes an auxiliary rotor, flexibly coupled to the main rotor. This contains a battery to power the lead screw motors of the bend actuator, together with a wireless communication module to transmit strain data and receive commands from an external controller for the bend actuation. The first two free-free natural frequencies of the rotor were evaluated by finite element analysis to be 135 Hz and 222 Hz. Each AMB had a radial stiffness of 1.9×10^5 N/m and damping coefficient of 460 Ns/m.

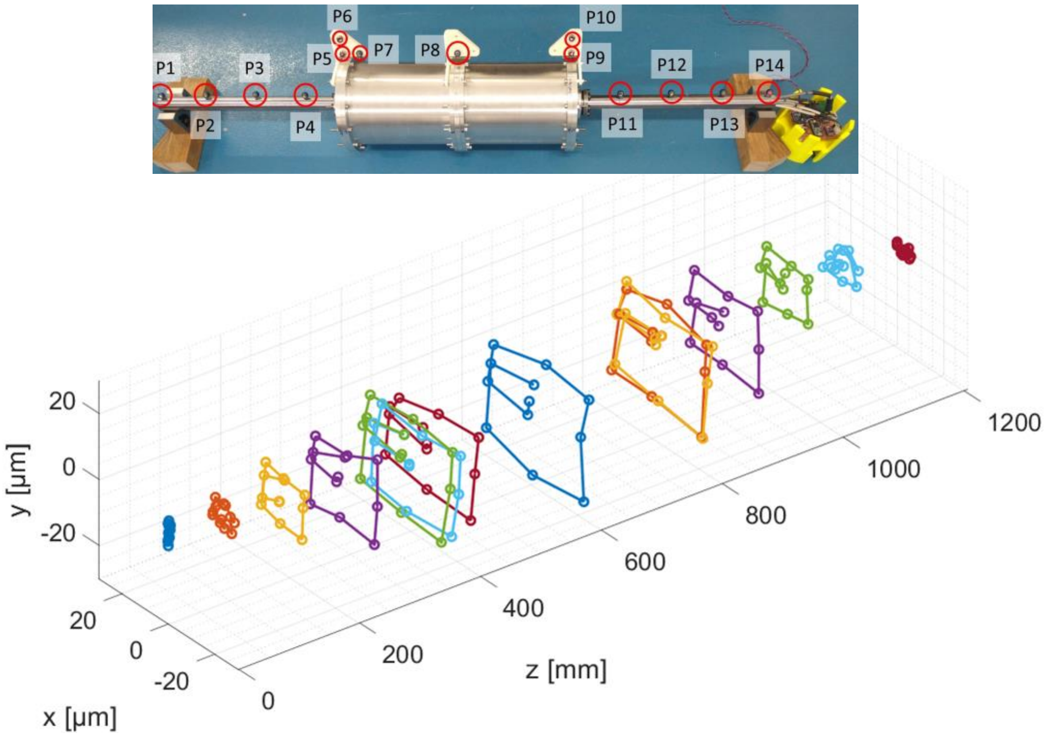

Fig. 6 Rotor deflection measured by a laser tracker under commanded wireless actuation.

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Table 2 AMB parameters.

Radial magnetic gap	0.8 mm
Touch-down bearing radial clearance	0.4 mm
Pole area	540 mm^2
Number of pole coil turns	200
Bias current	2.5 A
Maximum current	5 A
Maximum force	980 N
Proportional gain	4500 A/m
Integral gain	1000 A/s/m
Derivative gain	5 As/m

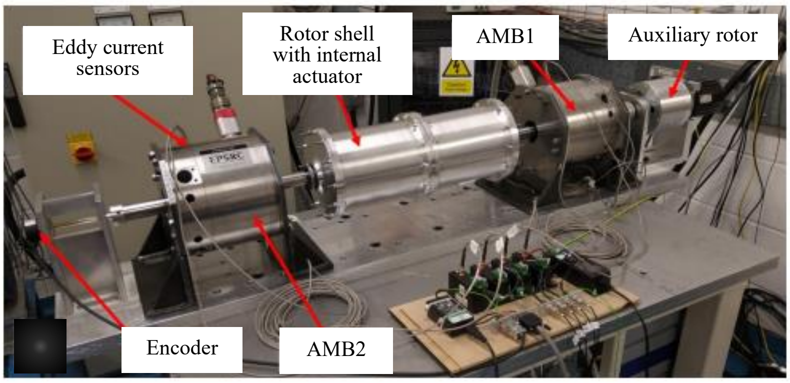

Fig. 7 Assembled active rotor mounted on its AMB’s.

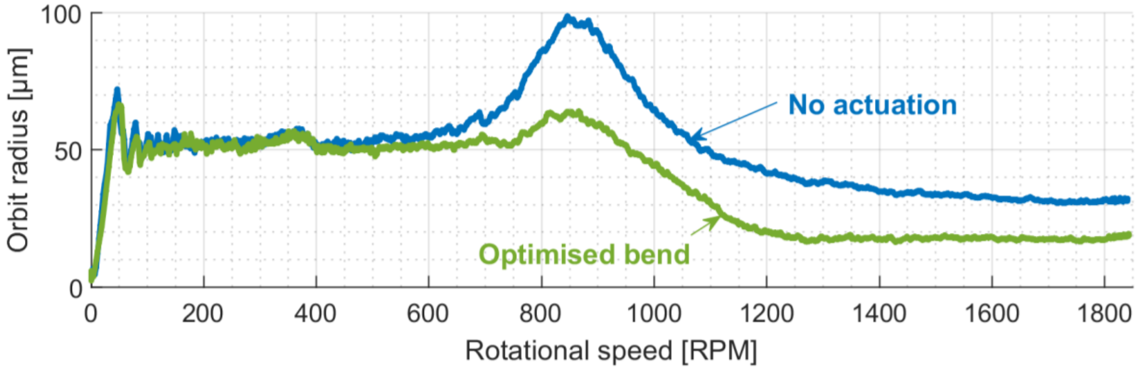

Fig. 8 Rotor orbit radii measured at AMB2 location with and without bend actuation.

3. Rotating tests

The rotor was initially levitated on its AMB's. This was with the specified PID gains of Table 2. The rotor was then run up to 1800 rpm, with the bend actuation switched off, and the eddy current displacement signals at each AMB were recorded. At each speed, effective orbit radii were evaluated. The test was then repeated with the bend actuation switched on. The orientation and level of bend actuation was then varied manually at each speed, effectively following a search process, to find the condition for optimal orbit reduction. Figure 8 corresponds with the AMB2 location of Fig. 7 and notable reductions of orbit radii are evident.

In another test, a laser vibrometer was used to measure rotor vibration amplitudes at the midpoint of the rotor shell that contains the bend actuator. After initial balancing, a 4 g mass was added to the rotor shell and without bend actuation, the vibration amplitude reached almost 45 μm at 2000 rpm as shown in Fig. 9. The test was then repeated and at approximately 100 rpm intervals an optimum bend angle and level were sought to minimise vibration amplitudes. The effectiveness of the bend actuation is clear.

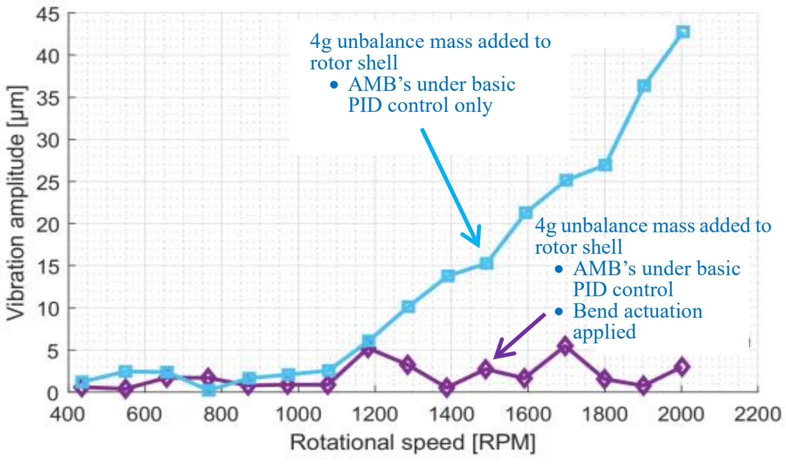

Fig. 9 Demonstration of combined AMB/bend actuator control when a 4 g unbalance mass is added to the middle of the rotor shell.

4. Conclusions

The concept of using internal bend actuation in a rotor supported by active magnetic bearings has been demonstrated. Since the bend occurs within the rotating frame, static bend actuation is equivalent to synchronous excitation from actuation within an inertial frame of reference. Hence low bandwidth AMB levitation can be combined with low bandwidth bend actuation for synchronous vibration control. Since the bend actuator is onboard the rotor, power supply may be thought of as an issue. However, the concept presented was able to be operated with a battery contained in an auxiliary rotor to power lead screw actuators intermittently. The design of the bend actuator and its principle of operation allowed its functionality to be achieved by wireless commands. The performance of the integrated system was demonstrated experimentally in static tests in which rotor deformation under bend actuation was measured using a laser tracker. Consideration was then given to vibration reduction during rotating tests. The rotor was levitated on AMB's and optimal levels and orientations of bend actuation were achieved by easy to apply search techniques. These were applied at increments of rotational speed in run-up tests and shown to be particularly effective in compensating for unbalance response at the rotor mid-point.

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