Rotor vibration control via active magnetic bearings and internal actuation

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Abstract

Designing rotor systems for high rotating speeds requires careful consideration of vibration signatures. Active Magnetic Bearings (AMBs) improve the vibration response by removing physical contact between the rotor and the stator. Their performance is, however, limited by magnetic flux saturation and the amplifier switching frequency. This paper develops a rotor bend-control concept with an experimental test rig. It shows that it is possible to reduce the amplitude of oscillation of the demand control current of an AMB under PID control by up to 65% by applying a bend that counteracts the effect of unbalance. The optimal orientation of the bend is found using a phase sweep. It shows that the bend control has promising capabilities to control the vibration level and reduce the synchronous demand from the AMB.

Keywords: Active Magnetic Bearing, Vibration control, Active rotor

1. Introduction

Rotating machines are used for a wide variety of applications, from transport to energy and manufacturing. Rotor vibrations arise from manufacturing tolerances, operating conditions, and thermal effects (Keogh & Morton 1994). They degrade the performance of the machine, either by forcing larger clearances to accommodate rotor orbits or stressing the material which can lead to premature failure. The vibration condition of a passive rotor can be improved via balancing as described in the ISO standards (ISO 2004). More advanced balancing methods such as disks with heavy spots (Van de Vegte & Lake 1978) or automatic ball balancers (Thearle 1932) can mitigate variable balancing conditions during operation. However, balancing ultimately reaches performance boundaries at high rotation speeds. This is due to the onset of dynamic instabilities and increased difficulty to locate precisely the inertial spin axis. Passive rolling elements bearings are often used with squeeze film dampers (Bonello et al. 2004). They are, however, limited in speed and load by the oil whip phenomenon (Newkirk & Taylor 1925), which generates instabilities. Gas/foil bearings can reach higher speeds and become self-sustaining (Belforte et al. 2006) but are subject to wear at low speed or high loads (DellaCorte et al. 2004). Active Magnetic Bearings (AMBs) are capable of offering controllable stiffness and damping, and the application of finer additional control to filter out unwanted excitations (Bleuler et al. 2009). They can, however, only apply a force at discrete locations and are limited at higher frequencies by magnetic flux saturation (Kang & Palazzolo 2012) and amplifier switching frequency (Maslen et al. 1989). Application of forces in the rotating frame has been achieved using macro fibre composites (Vadiraja & Sahasrabudhe 2009), functionally graded material (Alexander et al. 2007), or external piezoelectric patches (Przybylowicz 2002). In these cases, a voltage is applied to the material or through the patches to generate a bending moment. However, the achievable force magnitude limits their efficiency away from resonant frequencies.

This paper presents a novel control method using an active rotor featuring wireless, low-frequency, internal bend control, supported by AMBs. This has the advantage that low-frequency actuation in the rotating frame, which effectively transforms to high-frequency synchronous control in the stationary frame of reference. The concept is presented, along with some preliminary results. A finite element model, modal response, expected deformation and a controller concept are presented in depth in Fieux et al. (2022).

2. Methodology

A hydraulic system with lead screw actuators is used to generate large internal rotor bending moments. The aim of this prototype illustrated in Figure 1 a) is to counter the bending generated by an unbalance on a rotor supported symmetrically by two AMBs. As the rotor is spun, a bowed shape is generated by the unbalance force. The bending mechanism is then used to straighten the rotor using symmetrical bending moments. Figure 1 b) shows a schematic of the actual test rig setup, with a main rotor supported by two AMBs and an auxiliary rotor containing all the electronics to control the bending actuator.

The main rotor is composed of a larger cross-section module in the centre and two smaller cross-section shafts on either side. Forces are applied on the flanges that connect the different cross-sections. These forces can be varied in magnitude, resulting in an equal and opposite internal bending moment applied to the rotor.

The rotor used for this test has a mass of 20 kg and a length of 1.2 m. The larger central section has an outer diameter of 166 mm and an inner diameter of 150 mm. The outer shafts are connected to the larger central module via flanges and couplings and they have an outer diameter of 30 mm and an inner diameter of 18 mm. The bending actuator uses lead screw actuators and a hydraulic circuit to distribute the forces to the flanges. The maximum bending moment that the actuator can apply is 190 N.m. The rotor is balanced via the influence coefficient method (Darlow 1987), and was limited to run below 2400 RPM.

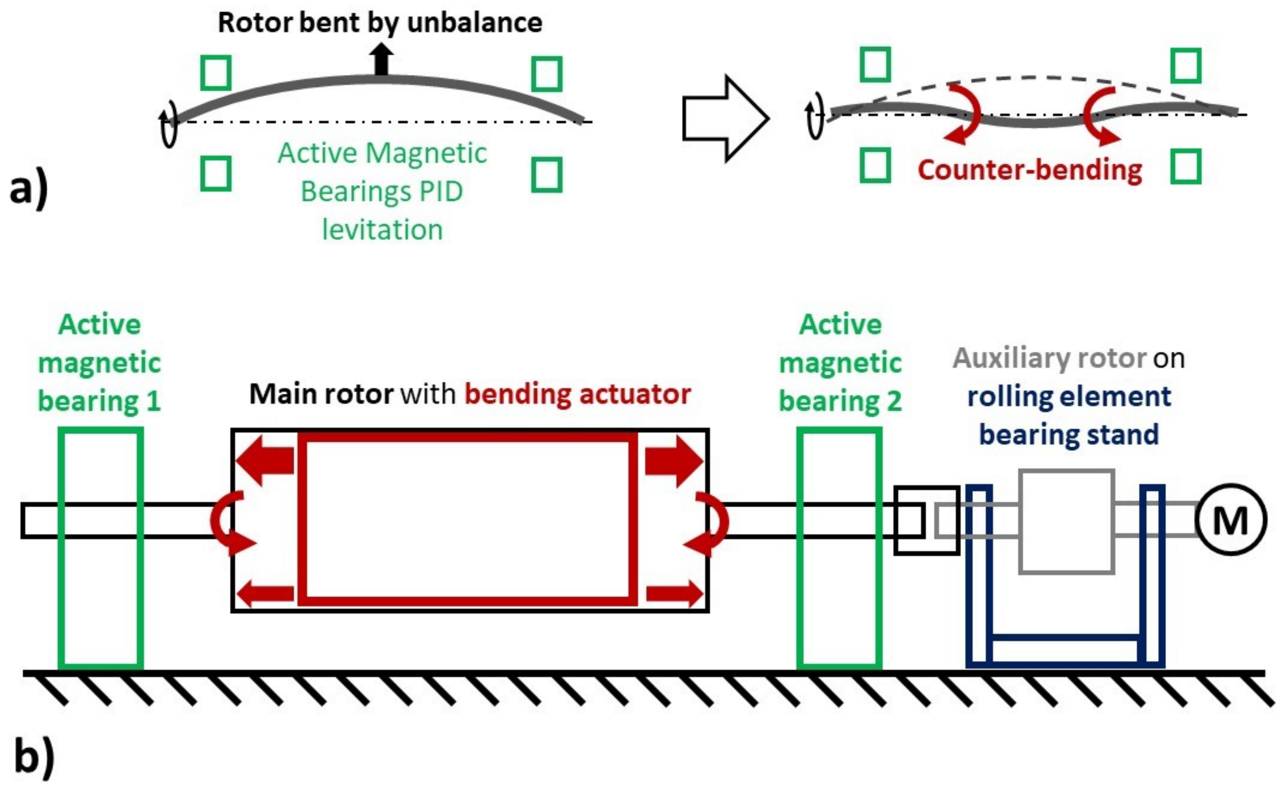

Figure 1: Rotor combined control concept.

3. Results

The rotor was spun at 800 RPM, supported by the AMBs under PID control. The current demand was monitored, isolating the contributions of the proportional and derivative controller action, since they are associated with the dynamic effects of rotor vibrations. The integral contribution of the control contributes only to the weight cancellation. To make the oscillatory current more readable, it is expressed in terms of a Root Mean Square

(RMS) value. Then, as the phase of the unbalance was unknown, the optimal bending phase was sought by applying rotor bend at $45°$ increments between $0°$ and $315°$.

Figure 2 illustrates the current response of one axis of an AMB to this phase sweep at 800 RPM. The initial level of 700 mA between 0 s and 40 s is without any bend actuation, as is the final level between 220 and 250 s. Between 40 s and 160 s, the phase of the control leads to an increase in control current, whereas between 160 s and 220 s the phase decreases the current required. From a base level of 700 mA, the current required for the levitation can be reduced to 350 mA, which is a 50% reduction.

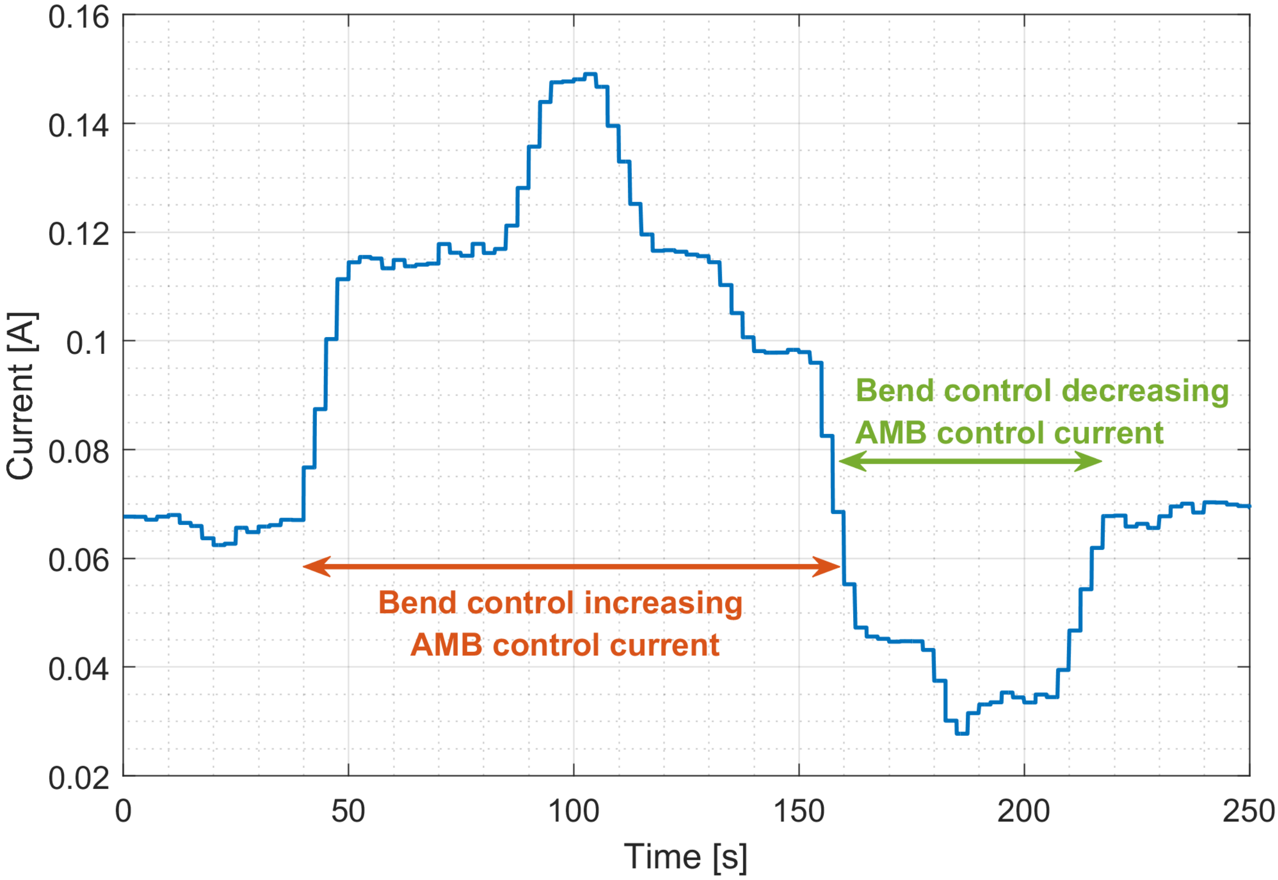

Figure 2: RMS of AMB current to applied bend phase sweep at 800 RPM with the AMBs under PID control. One axis of AMB1 displayed only.

Figure 3 displays the total control current required to maintain the levitation, at three different phases of bending. The recordings are over 100 ms at 5 s, 100 s and 190 s. Figure 3 a) shows the current associated with axis 1 from AMB1. The current for the uncontrolled, reference case oscillates between 340 mA and 515 mA (peak-to-peak amplitude of 175 mA). For the optimal bend control phase, it oscillates between 400 mA and 460 mA (peak-to-peak amplitude of 60 mA). This corresponds to a 66% reduction of the AMB1 control current amplitude of oscillation. In the axis 2 direction (Figure 3 b)), in the nominal case the current oscillates between 200 mA and 520 mA (peak-to-peak amplitude of 320 mA). For the optimal bend control phase, it oscillates between 260 mA and 435 mA (peak-to-peak amplitude of 175 mA). This corresponds to a 45% reduction of the control current amplitude of oscillation. Figure 3 c) and d), shows the corresponding results for AMB2. This time, the AMB current oscillation peak-to-peak amplitude is reduced from 310 mA to 150 mA in the axis 1 direction (-52%) and from 824 mA to 385 mA (-53%) in the axis 2 direction.

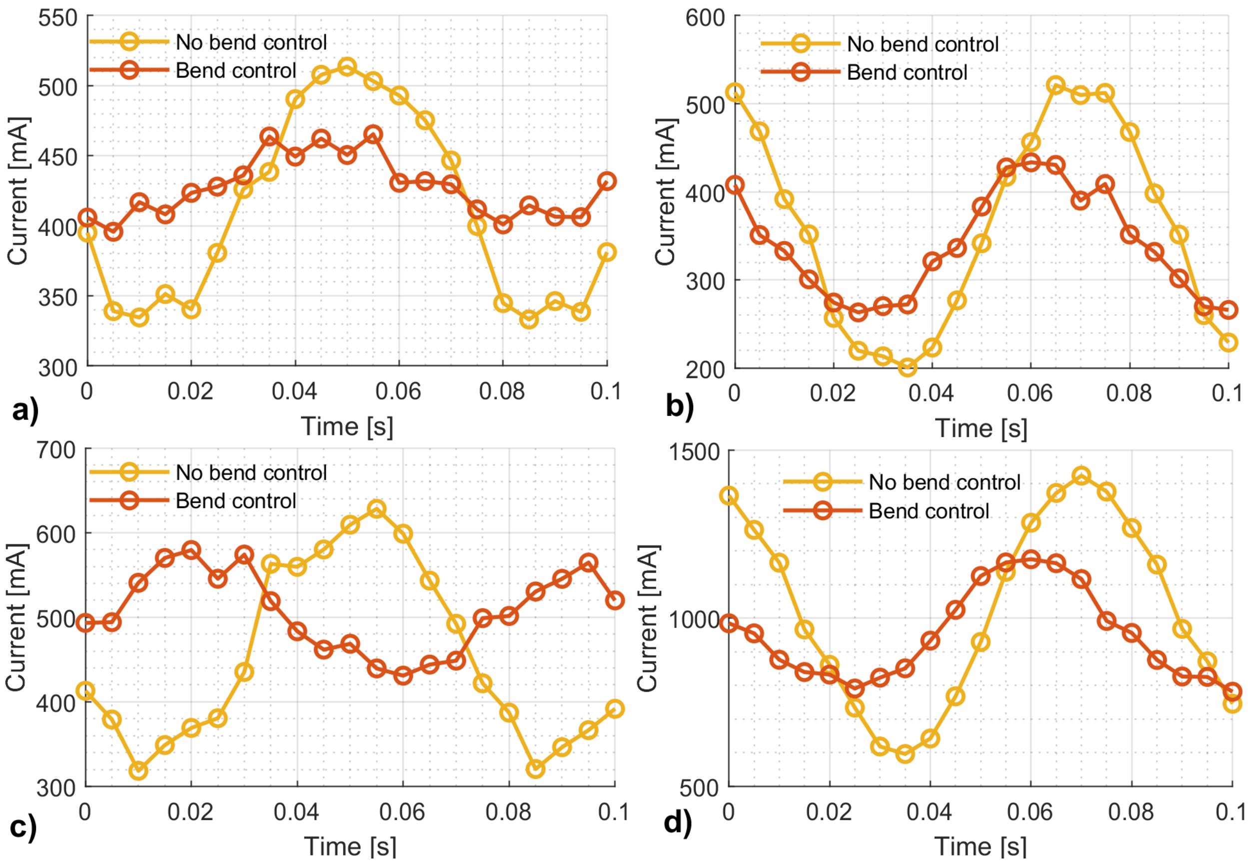

Figure 3: AMB control current response at 800 RPM comparing nominal, uncontrolled response to the controlled response at the optimal bend phase. a) shows AMB1 axis 1, b) AMB1 axis 2, c) AMB2 axis 1 and d) AMB2 axis 2.

4. Conclusions

This work has shown the potential to reduce the amplitude AMB control current by additional symmetrical internal rotor bend control. A prototype rotor has been manufactured and demonstrated that it was capable of reducing the dynamic part of the control current required for levitation by up 65% at 800 RPM. Reducing the amplitude of AMB current oscillation is of interest, considering that amplifier limitations at high frequencies are linked to the amplitude of the required current oscillation.

Further work is ongoing to characterise the rotor response in terms of rotor vibrations, as well as expanding these results for higher rotating speeds and different sources of vibration excitation.

References

Alexander, P. W., Brei, D. & Halloran, J. W. (2007), 'DEPP functionally graded piezoceramics via micro-fabrication by co-extrusion', *Journal of Materials Science* **42**(14), 5805 – 5814.
URL: *http://dx.doi.org/10.1007/s10853-007-1793-5*

Belforte, G., Raparelli, T., Viktorov, V., Trivella, A. & Colombo, F. (2006), 'An experimental study of high-speed rotor supported by air bearings: test rig and first experimental results', *Tribology International* **39**(8), 839 – 845.
URL: *http://dx.doi.org/10.1016/j.triboint.2005.07.013*

Bleuler, H., Cole, M., Keogh, P., Larsonneur, R., Maslen, E., Nordmann, R., Okada, Y., Schweitzer, G. & Traxler, A. (2009), *Magnetic bearings: Theory, Design, and Application to Rotating Machinery*, Springer.

Bonello, P., Brennan, M. J. & Holmes, R. (2004), 'A study of the nonlinear interaction between an eccentric squeeze film damper and an unbalanced flexible rotor', *Journal of Engineering for Gas Turbines and Power* **126**(4), 855 – 866.
URL: *http://dx.doi.org/10.1115/1.1787503*

Darlow, M. (1987), 'Balancing of high-speed machinery: Theory, methods and experimental results', *Mechanical Systems and Signal Processing* **1**(1), 105–134.
URL: *https://www.sciencedirect.com/science/article/pii/0888327087900872*

DellaCorte, C., Zaldana, A. R. & Radil, K. C. (2004), 'A systems approach to the solid lubrication of foil air bearings for oil-free turbomachinery', *Journal of Tribology* **126**(1), 200 – 207.
URL: *http://dx.doi.org/10.1115/1.1609485*

Fieux, G. A., Bailey, N. Y. & Keogh, P. S. (2022), 'Internal rotor actuation and magnetic bearings for the active control of rotating machines', *Actuators* **11**(2).
URL: *https://www.mdpi.com/2076-0825/11/2/57*

ISO (2004), Mechanical vibration – vibration of rotating machinery equipped with active magnetic bearings - Part 2: Evaluation of vibration, ISO 14839-2:2004, International Organization for Standardization, Geneva, Switzerland.

Kang, K. & Palazzolo, A. (2012), 'Homopolar magnetic bearing saturation effects on rotating machinery vibration', *IEEE Transactions on Magnetics* **48**(6), 1984 – 1994.
URL: *http://dx.doi.org/10.1109/TMAG.2012.2182776*

Keogh, P. & Morton, P. (1994), 'The dynamic nature of rotor thermal bending due to unsteady lubricant shearing within a bearing', *Proceedings of the Royal Society A: Mathematical, Physical and Engineering Sciences* **445**(1924), 273 – 290.
URL: *http://dx.doi.org/10.1098/rspa.1994.0061*

Maslen, E., Hermann, P., Scott, M. & Humphris, R. (1989), 'Practical limits to the performance of magnetic bearings: Peak force, slew rate, and displacement sensitivity', *Journal of Tribology* **111**(2), 331 – 336.

Newkirk, B. & Taylor, H. (1925), 'Shaft whipping due to oil action in journal bearings', *General Electric Review* **28**(8), 559–568.

Przybylowicz, P. M. (2002), 'Near-critical behaviour of a rotating shaft actively stabilised by piezoelectric elements', *Systems Analysis Modelling Simulation* **42**(4), 527 – 537.
URL: *http://dx.doi.org/10.1080/02329290290031305*

Thearle, E. L. (1932), 'Dynamic balancing machine'.

Vadiraja, D. & Sahasrabudhe, A. (2009), 'Vibration analysis and optimal control of rotating pre-twisted thin-walled beams using mfc actuators and sensors', *Thin-Walled Structures* **47**(5), 555 – 567.
URL: *http://dx.doi.org/10.1016/j.tws.2008.10.004*

Van de Vegte, J. & Lake, R. (1978), 'Balancing of rotating systems during operation.', *Journal of Sound and Vibration* **57**(2), 225 – 235.
URL: *http://dx.doi.org/10.1016/0022-460X(78)90581-3*