Experiments on ROLAC to Recover Rotor Position Following Contact*

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***Abstract*— There is growing interest in magnetic bearing applications because of their advantages for contactless support, high-speed operation, vibration control, condition monitoring and fault diagnostics. However, they have limited force capacity and have to incorporate retainer bearings to protect the laminations when vibration levels reach the clearance level. Retainer bearings can take a relatively small number of rotor drops followed by machine shutdown. There are many potential applications where transient events or sudden changes in operating conditions could drive the rotor into contact with retainer bearings. These include land, sea and air transport applications, which are exposed to external inputs and disturbances, and system shut down is not acceptable.**

This paper describes the application of a Recursive Open Loop Adaptive Controller (ROLAC), an adaptive and fast acting controller, to prevent contact between a flexible rotor and retainer bearing. If however contact does occur, then the control action will minimize the damage and recover the rotor position without the need to shut down the system. It is assumed that the magnetic bearings are fully operational and contact is due to external disturbances such as sudden unbalance, base excitation, blade loss or damage. Experimental results are presented to demonstrate the effectiveness of the control algorithm.

***Index Terms*— Magnetic bearings, contact dynamics, recursive Fourier transform, adaptive control.**

I. Introduction

The future of magnetic bearing applications relies on addressing the safety and reliability issues in critical applications [1]. One of the complex dynamic failure modes of active magnetic bearings (AMBs) is when a rotor contacts its retainer bearing. In applications such as land, sea and air transportation, the contact problem may occur due to transient or abrupt disturbances. Understanding the contact dynamics is essential for designing controllers to prevent contact, minimise damage or recover the rotor position.

The conventional approach to modelling contact forces utilizes Hertzian theory, where the contact forces are expressed in terms of the penetration depth through stiffness and damping coefficients [2]–[4]. In some applications, these parameters vary exponentially [5], [6]. It is difficult to predict the contact modelling parameters, which may change during the contact as a function of temperature, surface damage etc. Most importantly, these models have to use large stiffness coefficients to simulate material properties and to restrict the penetration depth, causing computational inefficiencies and difficulties due to high frequency oscillations. A constrained Lagrangian formulation has been proposed to simulate the contact dynamics in a computationally efficient manner [7], [8]. This method does not require the contact forces to be modelled. When the radial displacement reaches the bearing clearance, the simulation is switched to a constrained model, where the contact is treated as a constraint on the system. This constraint is included in the system equations through a Lagrangian multiplier. The constraint, and hence the contact forces, are automatically calculated during the simulation and the zero crossing of the radial constraint force indicates a return to a non-contact mode of operation. This method can easily be extended to cover multi contact cases including backward whirl rolling motion [9]. In this study, the model is used to develop, test and tune controllers before real-time implementation.

Controllers designed to be used for normal non-contact operation of magnetic bearings usually do not function well under contact conditions. Various passive, active, adaptive and modern control techniques have been developed to control active magnetic bearings under normal operations, to minimize rotor vibrations, and also to minimize transmitted forces [10]–[14]. A simple, but effective method of controlling synchronous vibration under varying operating conditions has been introduced by Burrows and Sahinkaya [10], [15], [16]. This open loop adaptive control strategy, also referred as *automatic balancing*, can also be used to attenuate multi-frequency vibrations of the rotor [17]. However, it is not fast enough to prevent contact as it relies on the steady state response and a Fourier transform of measured displacements. Subsequent work to improve the reaction speed of the controller to transient changes has shown the potential of the technique [18], [19], but the phase of the measured vibration response under contact may be changed to such an extent that synchronous controllers may worsen the rotor response, resulting in contact with higher forces [20]. A phase modification is proposed to restore the rotor position.

In this paper, a recursive open-loop adaptive control scheme is implemented. This is a modified version of the open loop adaptive control (OLAC) method and utilises a recursive version of the Fourier transform to update optimum control force components at every sampling interval.

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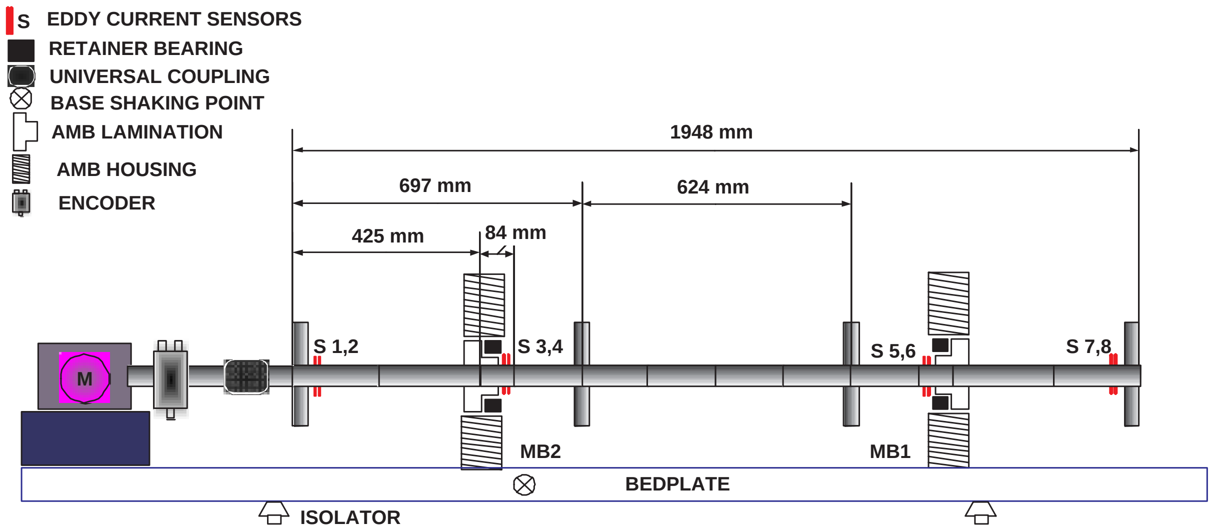


Fig. 1. Experimental flexible rotor/magnetic bearing system

II. Experimental Setup

The experimental system consists of a uniform flexible steel shaft of length 2 m and radius 0.025 m, with four 10 kg disks of radii 0.125 m. The rotor of total mass 100 kg is mounted horizontally on two radial magnetic bearings each of which has a radial force capacity of 1.25 kN with a bandwidth of 100 Hz. The magnetic bearings have an air gap of 1.2 mm and each is protected by a retainer bearing having a 0.75 mm clearance. Eight eddy current displacement sensors are placed in four planes at a 45° angle with the vertical line. A schematic view of the experimental setup showing the rotor, magnetic bearings, retainer bearings and sensors is provided in Fig. 1.

The rotor considered in the present paper is flexible. Critical speeds were measured experimentally as 10 Hz, 17 Hz, and 28 Hz. The rotor displacements were measured relative to the base frame with four pairs of eddy current displacement transducers. At each transducer location, a precision stainless steel collar is mounted on the shaft to minimise the measurement errors due to material imperfection. The control current of the magnetic coils were supplied through eight amplifiers with a bias current of 4.3 A for the lower poles and 5.7 A for the upper poles, giving a negative stiffness of 2×10^{6} N/m and a static lifting to support the rotor. The rotor-bearing system stability is ensured through local PID controllers, which were digitally implemented with a sampling frequency of 4 kHz using dSPACE.

III. Recursive Open-Loop Adaptive Control (ROLAC)

The OLAC calculates the optimum frequency response of the control force by a Least Square Estimator as follows [10]:

$$\begin{aligned}\Delta\mathbf{U}(j\Omega) &= [\mathbf{R}(j\Omega)^{T}\mathbf{R}(j\Omega)]^{-1}\mathbf{R}(j\Omega)^{T}\mathbf{Q}(j\Omega)\\ &= \mathbf{H}(j\Omega)\mathbf{Q}(j\Omega)\end{aligned} \tag{1}$$

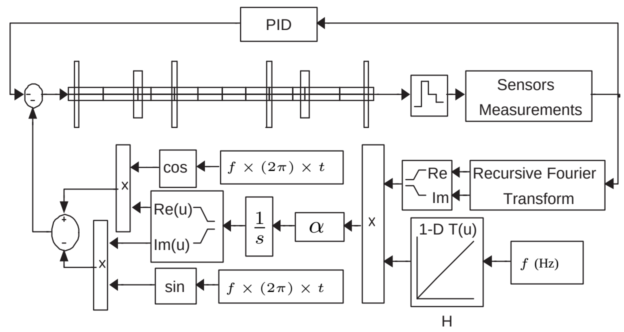


Fig. 2. Block diagram of the recursive open-loop adaptive controller (ROLAC).

where $\mathbf{Q}(j\Omega)$ is the frequency response vector containing all eight measurement locations, $\mathbf{H}(j\Omega)$ is the complex control gain matrix, and $\mathbf{R}(j\Omega)$ is the receptance matrix between the measured and control signals.

The partial receptance matrix $\mathbf{R}$ can either be calculated from the rotor bearing model, or estimated experimentally without any knowledge of system parameters. The estimation involves sending small amplitude test signal from each AMB axis one at a time, and measuring the changes in the frequency response of the displacements [16]. Frequency responses can be calculated by a Fourier transform using one or more synchronous periods of steady state data. Since the $\mathbf{R}$ matrix is not a function of the external disturbances and unbalance distribution, it can be identified during system commissioning. However repeated online identification during the normal operation of the system may be beneficial for condition monitoring and fault diagnostics if there is a change in the measured response [21]

The recursive version updates the Fourier coefficients at each sampling time, and hence the optimised open loop control parameters are updated at each sampling time. ROLAC acts as soon as the orbit size starts to increase, and hence it achieves fast adaptation of the control force in accordance with changing operating conditions and sudden external disturbances.

Since equation (III) gives the optimum modification of the control force, the recursive version is implemented through an integrator with an integrator constant of α as shown in Fig. 2.

$$\mathbf{U}(j\Omega, t) = \alpha \int \mathbf{H}(j\Omega)\mathbf{Q}(j\Omega, t)dt \tag{2}$$

where $\mathbf{Q}(j\Omega, t)$ is the recursive Fourier transform of the displacements and can be calculated as follows:

$$\mathbf{Q}(j\Omega, t) = \mathbf{J}(j\Omega, t) - \mathbf{J}(j\Omega, t - 2\pi/\Omega) \tag{3}$$

where the integral $\mathbf{J}(j\Omega, t)$ is defined as:

$$\mathbf{J}(j\Omega, t) = \frac{\Omega}{2\pi}\int_{\tau=0}^{t} \mathbf{q}(\tau)\; e^{-j\Omega} d\tau \tag{4}$$

where $\mathbf{q}(t)$ is the time domain measurement vector. This gives a control force in the time domain

$$\begin{aligned} \mathbf{u}(t) &= \mathrm{Re}[\mathbf{U}(j\Omega, t)e^{j\omega}] \\ &= \mathrm{Re}[\mathbf{U}]\cos(\Omega t) - \mathrm{Im}[\mathbf{U}]\sin(\Omega t) \end{aligned} \tag{5}$$

The simulation model is used to find an optimum value for the integration constant $\alpha = 7$. Higher values give faster but oscillatory transient response, whereas low values make the controller slow acting. Setting $\alpha = 0$ after the steady state response is established is equivalent to using the OLAC.

The $\mathbf{H}(j\Omega)$ matrix is estimated online at selected running speeds and stored in a file for subsequent use. A linear interpolation is used to obtain the $\mathbf{H}$ matrix between the stored speed values during runup and rundown of the system. The real-time implementation allows a user to update or measure the $\mathbf{H}$ matrix during steady state operation of the system at any speed. The speeds at which the $\mathbf{H}$ matrix is stored are carefully selected by using computer simulations so that its elements show piecewise linear behaviour.

IV. Experimental Results

Various experiments were carried out to demonstrate the effectiveness of ROLAC under single and multiple contacts with the retainer bearings. In all cases, a sudden out-of-balance disturbance is applied to the rotor through the magnetic bearings to initiate the contact. The ROLAC plus PID results are compared with PID only controllers. The first experimental result in Fig. 3 shows the performance of the ROLAC under normal operating conditions. The system was operated at a running speed of 10Hz with PID controllers that support the rotor and ensure system stability. The ROLAC control algorithm was switched on after the 10^{th} synchronous cycle as indicated by arrows on Fig. 3. It clearly demonstrates the capability of ROLAC to attenuate the inherent vibration from approximately 150μm to 50μm at the 7^{th} and 8^{th} sensors and 75μm to 30μm at the 5^{th} and 6^{th} sensors. This is similar to the OLAC performance.

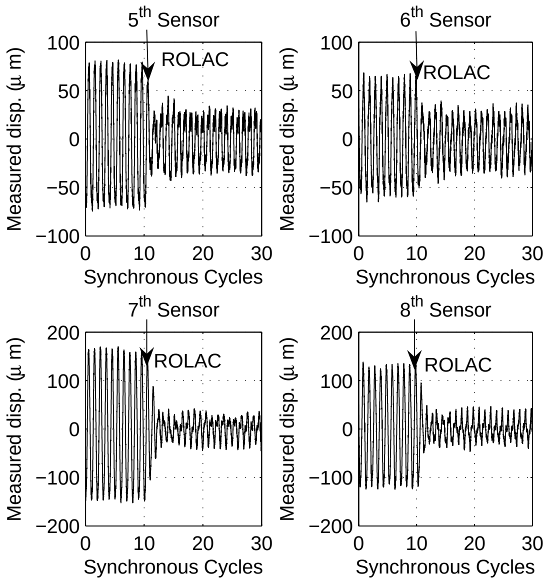
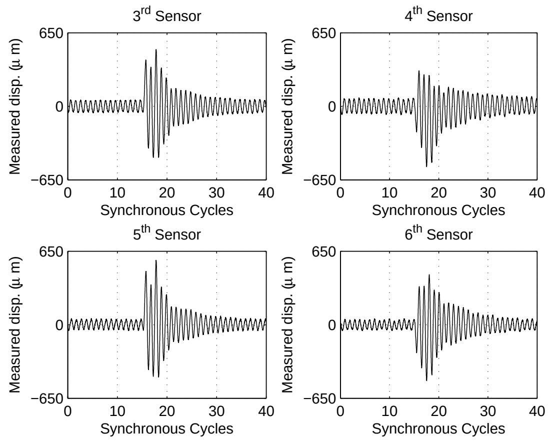


Fig. 3. Vibration attenuation using ROLAC at $\Omega = 10Hz$

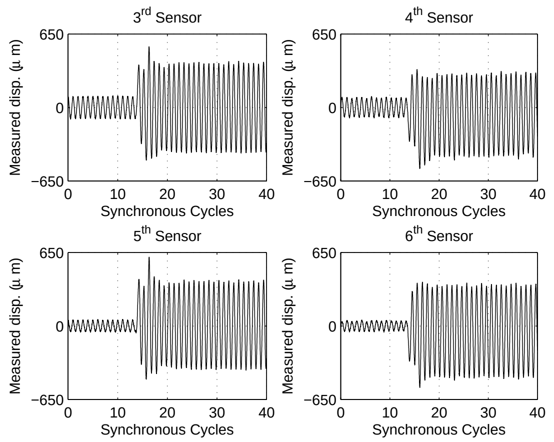
The experimental results in Fig. 4 show the speed at which ROLAC responds to a sudden change in unbalance forces. The rotor running speed was set to Ω=35Hz, which is well above the first flexural frequency. An unbalance of 10.34gm and 8.27gm was introduced through MB1 and MB2, respectively. Fig. 4(a) shows the reduction in vibration level at both AMB locations under ROLAC. The same measurements with a PID only controller is shown in Fig. 4(b). The corresponding orbits are shown in Fig. 5(a) and Fig. 5(b), respectively.

The capability of ROLAC to prevent rotor contact with retainer bearings is presented in Figs. 6 and 7. The contact was initiated by injecting an out of balance force of 105.37gm through MB1 at the 5^{th} synchronous cycle at the running frequency of Ω=12.5Hz. The rotor orbits with a PID controller is compared with those with ROLAC in Fig. 6. The measured radial displacements under ROLAC are shown in Fig. 7. Under PID control in Fig. 6(a), contact occurs at the MB2 location and the rotor remains in contact with no chance of recovery. However, ROLAC not only prevents the contact, but also reduces vibration levels to a very low level.

It may not be possible to prevent contact if the sudden change of unbalance is significant. A contact case was initiated at a running frequency of 10Hz with a sudden out of balance of 202.64 gm applied through both magnetic bearings. The rotor orbits with PID only control is shown in Fig. 8(a), where contact occurs at both magnetic bearings and the rotor remains in contact for the rest of the operation. This makes it necessary to run down the rotor. The performance of ROLAC for the same case is shown in Fig. 8(b). Although contact was not prevented,

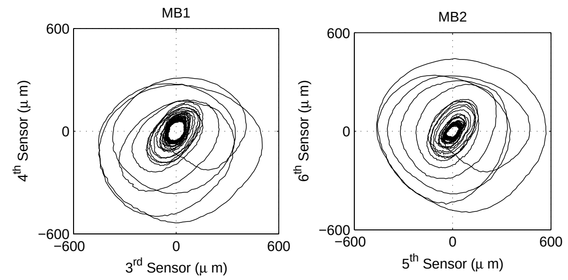


(a) PID +ROLAC

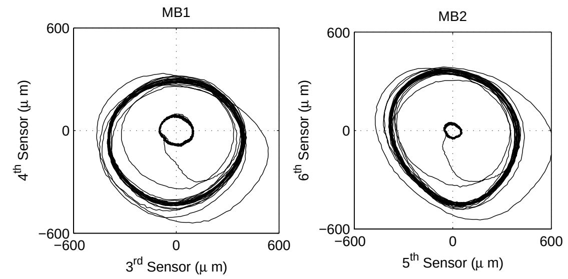


(b) PID only controller

Fig. 4. Experimental displacements at both magnetic bearing locations when sudden out of balance is introduced at $\Omega = 35Hz$.

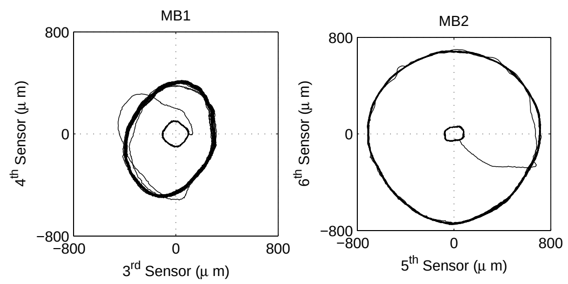


(a) PID +ROLAC

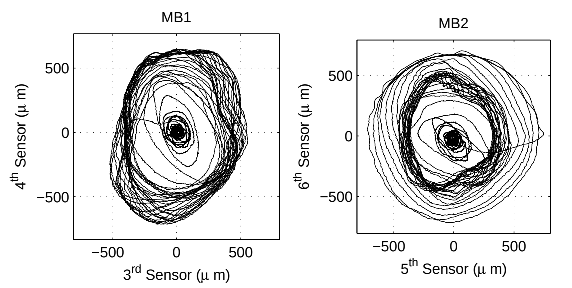


(b) PID only controller

Fig. 5. Experimental rotor orbits at both magnetic bearing locations when sudden out of balance is introduced at $\Omega = 35Hz$.



(a) PID only controller



(b) PID + ROLAC

Fig. 6. Experimental rotor orbits at both magnetic bearing locations when sudden out of balance is introduced at $\Omega = 12.5$Hz.

the rotor position was recovered very quickly (less than two synchronous cycles), then returning to a low vibration steady state operation. The measured radial displacements of the rotor at both magnetic bearing locations is shown in Fig. 9 thereby demonstrating the effectiveness of ROLAC under severe contact conditions.

V. Conclusions

The capacity for contactless support offered by magnetic bearings makes them potentially significant for applications that require high standards of safety during abnormal condition without shutting down the system. A ROLAC controller is introduced to increase the speed of the controller action to changes in the vibration levels. A recursive Fourier transform is utilised and implemented with an integrator to update the optimum force amplitude and phase at every sampling interval. Experimental work is described using a flexible rotor/magnetic bearing system to demonstrate the ability of ROLAC to prevent contact or to recover the rotor position quickly with minimum impact damage if contact does occur. It is shown that the

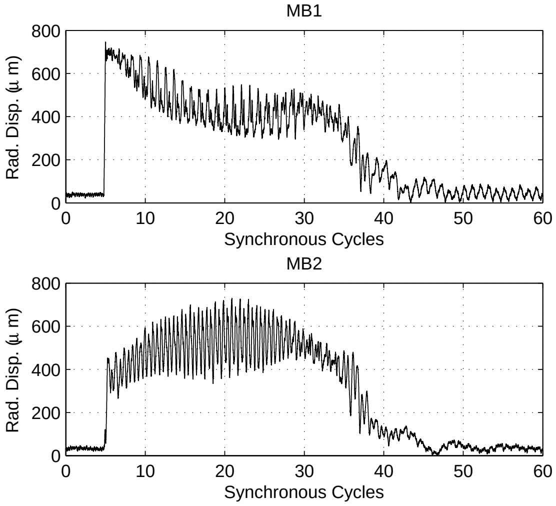
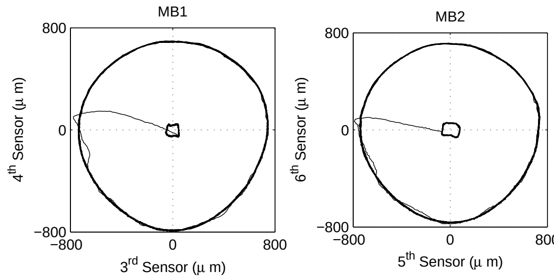
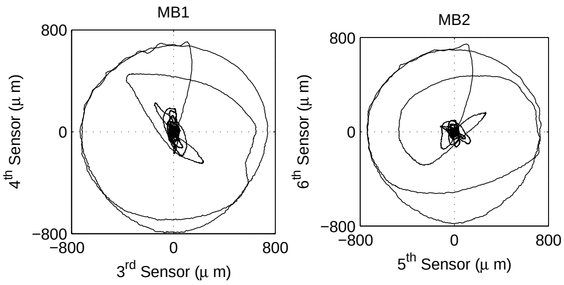


Fig. 7. Measured radial displacements of the rotor with the application of ROLAC at $\Omega = 12.5Hz$.



(a) PID only controller



(b) PID + ROLAC

Fig. 8. Experimental rotor orbits at both magnetic bearing locations when sudden out of balance is introduced at $\Omega = 10$Hz.

rotor can resume its linear and stable mode of operation after contact without the need to shut down the system. ROLAC is capable of attenuating vibration due to inherent out of balance, sudden changes of out of balance and abrupt disturbances.

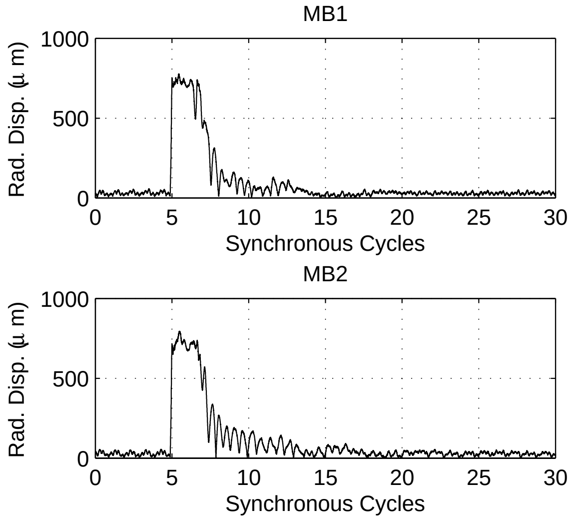


Fig. 9. Measured radial displacements with the application of ROLAC at $\Omega = 10$Hz.

REFERENCES

[1] G. Schweitzer, "Safety and reliability aspects for active magnetic bearing applications - a survey," *Proceedings of the Institution of Mechanical Engineers Part I-Journal of Systems and Control Engineering*, vol. 219, no. I6, pp. 383–392, 2005.

[2] H. Xie, G. T. Flowers, L. Feng, and C. Lawrence, "Steady-state dynamic behavior of a flexible rotor with auxiliary support from a clearance bearing," *Journal of Vibration and Acoustics-Transactions of the ASME*, vol. 121, no. 1, pp. 78–83, 1999.

[3] G. Gilardi and I. Sharf, "Literature survey of contact dynamics modelling," *Mechanism and Machine Theory*, vol. 37, no. 10, pp. 1213–1239, 2002.

[4] J. L. Lawen and G. T. Flowers, "Interaction dynamics between a flexible rotor and an auxiliary clearance bearing," *Journal of Vibration and Acoustics-Transactions of the ASME*, vol. 121, no. 2, pp. 183–189, 1999.

[5] R. G. Kirk, K. V. S. Raju, and K. Ramesh, "Modeling of AMB turbo-machinery for transient analysis," in *Proceedings of MAG 97*. Alexandria, USA, 1997, pp. 139–153.

[6] G. von Groll and D. J. Ewins, "A mechanism of low subharmonic response in rotor/stator contact - measurements and simulations," *Journal of Vibration and Acoustics-Transactions of the ASME*, vol. 124, no. 3, pp. 350–358, 2002.

[7] M. N. Sahinkaya, A. G. Abulrub, and P. S. Keogh, "On the modeling of flexible rotor/magnetic bearing systems when in contact with retainer bearings," in *The Ninth International Symposium on Magnetic Bearings (ISMB9)*. Kentucky, USA, 2004.

[8] A. G. Abulrub, M. N. Sahinkaya, P. S. Keogh, and C. R. Burrows, "A constrained Lagrangian approach for modelling multiple contacts in flexible rotors," in *IFTOMM 7th International Conference on Rotor Dynamics*. Vienna, Austria, 2006.

[9] M. N. Sahinkaya, A. G. Abulrub, P. S. Keogh, and C. R. Burrows, "Multiple sliding and rolling contact dynamics for a flexible rotor/magnetic bearing system," *IEEE-ASME Transactions on Mechatronics, Submitted*, 2006.

[10] C. R. Burrows and M. N. Sahinkaya, "Vibration control of multimode rotor-bearing systems," *Proceedings of the Royal Society of London Series A- Mathematical Physical and Engineering Sciences*, vol. 386, no. 1790, pp. 77–94, 1983.

[11] C. R. Knospe, R. W. Hope, S. J. Fedigan, and R. D. Williams, "Experiments in the control of unbalance response using magnetic bearings," *Mechatronics*, vol. 5, no. 4, pp. 385–400, 1995.

[12] P. Schroder, B. Green, N. Grum, and P. J. Fleming, "On-line genetic auto-tuning of mixed H_2/H_∞ optimal magnetic bearing controllers," in *Procedings of Control'98*. Swansea, UK, 1998.

[13] S. K. Hong and R. Langari, "Robust fuzzy control of a magnetic bearing system subject to harmonic disturbances," *IEEE Transactions on Control Systems Technology*, vol. 8, no. 2, pp. 366–371, 2000.

[14] S. Yu, S. Wu, and T. Lee, "Optimal fuzzy control of radial active magnetic bearing systems," in *Proceedings IEEE International Sym-*

posium on Computational Intelligence in Robotics and Automation. Kobe, Japan, 2003.

[15] C. R. Burrows, M. N. Sahinkaya, and S. Clements, "Active vibration control of flexible rotors - an experimental and theoretical-study," *Proceedings of the Royal Society of London Series A- Mathematical Physical and Engineering Sciences*, vol. 422, no. 1862, pp. 123–146, 1989.

[16] M. N. Sahinkaya and C. R. Burrows, "Control of stability and the synchronous vibration of a flexible rotor supported on oil-film bearings," *Journal of Dynamic Systems Measurement and Control-Transactions of the ASME*, vol. 107, no. 2, pp. 139–144, 1985.

[17] P. S. Keogh, M. O. T. Cole, and C. R. Burrows, "Multi-state transient rotor vibration control using sampled harmonics," *Journal of Vibration and Acoustics-Transactions of the ASME*, vol. 124, no. 2, pp. 186–197, 2002.

[18] C. R. Burrows, P. S. Keogh, and R. Tasaltin, "Closed-loop vibration control of flexible rotors - an experimental-study," *Proceedings of the Institution of Mechanical Engineers Part C-Journal of Mechanical Engineering Science*, vol. 207, no. 1, pp. 1–17, 1993.

[19] P. S. Keogh, C. Mu, and C. R. Burrows, "Optimized design of vibration controllers for steady and transient excitation of flexible rotors," *Proceedings of the Institution of Mechanical Engineers Part C-Journal of Mechanical Engineering Science*, vol. 209, no. 3, pp. 155–168, 1995.

[20] P. S. Keogh, M. O. T. Cole, M. N. Sahinkaya, and C. R. Burrows, "On the control of synchronous vibration in rotor/magnetic bearing systems involving auxiliary bearing contact," *Journal of Engineering for Gas Turbines and Power-Transactions of the ASME*, vol. 126, no. 2, pp. 366–372, 2004.

[21] M. N. Sahinkaya, M. O. T. Cole, and C. R. Burrows, "Fault detection and tolerance in synchronous vibration control of rotor-magnetic bearing systems," *Proceedings of the Institution of Mechanical Engineers Part C- Journal of Mechanical Engineering Science*, vol. 215, no. 12, pp. 1401–1416, 2001.