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CONDITION MONITORING OF ROTOR USING ACTIVE MAGNETIC ACTUATOR

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ABSTRACT

It has been widely recognized that the changes in the dynamic response of a rotor could be utilized for general fault detection and monitoring. Current methods rely on the monitoring of synchronous response of the machine during its transient or normal operation. Very little progress has been made in developing robust techniques to detect subtle changes in machine condition caused by rotor cracks. It has been demonstrated that the crack-induced changes in the rotor dynamic behavior produce unique vibration signatures. When the harmonic excitation force is applied to the cracked rotor system, nonlinear resonances occur due to the nonlinear parametric excitation characteristics of the crack. These resonances are the result of the coexistence of a parametric excitation term and different frequencies present in the system, namely critical speed, the synchronous frequency, and excitation frequency from the externally applied perturbation signals. This paper presents the application of this approach on an experimental test rig. The simulation and experimental study for the given rig configuration, along with the application of active magnetic bearings as a force actuator, are presented.

1 INTRODUCTION

Many critical rotating machines such as compressors, pumps, and gas turbines continue to be used beyond their expected service life despite the associated potential for failure due to damage accumulation. Therefore, the ability to monitor the structural health of these systems is becoming increasingly important, and the term *structural health monitoring* can be defined as the process of implementing a damage-detection strategy. This strategy involves the observation of a structure over a period of time, the identification of features from measurements, and the analysis of these features to determine the current damage state of the system.

The area of damage detection and location using measured low frequency vibration data has attracted considerable attention recently. Doebling et al. [1] have presented an extensive survey of the field which should be consulted for further details. Health monitoring of structures and machinery is a major concern of the engineering community. Early damage detection, flaw identification, and failure prevention have far reaching implications in the management and preservation of any equipment. Machinery failure prevention is a complex activity that requires the interaction of several concurrent factors. Critical among them is the ability to detect the appearance and propagation of structural damage in vital machinery parts. The detection of damage (e.g., cracks, de-laminations, de-bonding, etc.) is crucial in any failure prevention technology [2].

Today, monitoring and diagnosis systems are normally not an integral component of turbomachinery. These fault detection and diagnosis systems mainly measure the output signals; the relative and/or absolute motions of the rotor. After signal processing, certain features (threshold values, orbits, frequency spectra, *etc.*) are presented from the measured data. Based on the deviations of these features from a non-faulty initial state, faults are detected. Subsequently, the diagnosis attempts to recognize possible faults. The difficulty with these procedures is that the causes of the variations in the output signals cannot be detected clearly. The reason might either be a change of the process (i.e., input) or a modification of the system itself (e.g., shaft crack). In addition, the measured vibration signals fail in revealing a minor change in system parameters because all the excitations are due to shaft rotation. Therefore, it should not be a surprise that a carefully designed externally applied excitation can extract more information of the system than a *natural*, rotation-related, synchronous excitation.

Morton [3] was one of the first who identified dynamic properties of a full operating turbomachine by means of a broadband excitation technique. The rotating shaft was preloaded with a static force via an additional foil bearing. The excitation force was created by a sudden release of the preload due to the use of a breaking link, while the applied forces were measured with strain gauges. The method was used for the identification of rotordynamic coefficients of oil film bearings. Nordmann and Schollhorn [4] and Tonnesen and Lund [5] excited a rotating Jeffcott rotor supported on oil film bearings by a hammer. The impact force and displacement signals were used for identification of the modal parameters of the rotating system. One of the disadvantages of the broadband excitation in general is the distribution of the energy over a certain frequency range, which may cause a poor signal to noise ratio. Iwatsubo *et al.* [6] used external excitation technique to analyze the response of the cracked shaft. He showed the presence of combination harmonics due to interaction between impact force and rotation of shaft as the crack indicators.

The topic of cracked rotor vibrations has been analyzed in a great number of published works, for example [7-14]. They have been focused on the study of dynamic behavior of rotors with the so-called breathing type of crack during the passage through a critical speed at constant angular acceleration or deceleration. For example, Sawicki et al. [7-10] studied the dynamics of accelerating cracked rotors, the crack-induced coupling between the torsional and lateral vibrations, and the application of nonlinear dynamics tools for crack diagnosis. Gasch [10] provided a comprehensive investigation of the stability behavior of a cracked Jeffcott rotor. The cracked shaft analysis of Gasch assumed a constant rotational speed and a forced vibration resulting from the system's residual unbalance.

Recently active magnetic bearings (AMBs) have been proposed as actuators to apply force to the shaft of a rotating machine [15-18]. This paper discusses some of the issues to be addressed to enable this approach to become a robust condition monitoring technique for cracked shafts. The modeling of cracked rotor is presented along with the results of the simulated cases involving the configuration of experimental test rig employed for crack detection study. The description of the rig, experimental approach and experimental results for undamaged and damaged rotors are presented.

EQUATIONS OF MOTION OF CRACKED ROTOR

The analysis of a rotor-bearing system may be performed in fixed or rotating co-ordinates. If neither the bearings and foundations nor the rotor is axi-symmetric then the resulting differential equations, whether described in fixed or rotating coordinates, will be linear equations with harmonic coefficients. Typically foundations of a large machine will be stiffer vertically than horizontally and in this case the cracked rotor will not be axi-symmetric when the crack is open. Thus there is no compelling reason to used fixed or rotating coordinates for the analysis. To determine the stiffness of the rotor as the crack opens and closes it is easier to work in co-ordinates that are fixed to the rotor and rotate with it. The reduction in stiffness due to a crack is then calculated in directions perpendicular to and parallel to the crack face, and these directions will rotate with the rotor. Having determined the rotor stiffness in rotating coordinates we transform the stiffness matrix to fixed coordinates and join the stiffness to the system inertia to obtain the equation of motion in fixed coordinates.

Let the stiffness matrix in rotating co-ordinates for the un-cracked rotor be $\tilde{\mathbf{K}}_0$ and the reduction in stiffness due to a crack be $\tilde{\mathbf{K}}_c(\theta)$, where θ is the angle between the crack axis and the rotor response at the crack location and determines the extent to which the crack is open. Thus the stiffness of the cracked rotor is

$$\tilde{\mathbf{K}}_{cr} = \tilde{\mathbf{K}}_0 - \tilde{\mathbf{K}}_c(\theta). \tag{1}$$

This stiffness matrix is transformed from rotating to fixed co-ordinates using the transformation matrix $\mathbf{T}(\Omega t)$, to give, assuming the un-cracked rotor is axi-symmetric,

$$\mathbf{K}_{cr} = \mathbf{T}^{\mathrm{T}}\tilde{\mathbf{K}}_0\mathbf{T} - \mathbf{T}^{\mathrm{T}}\tilde{\mathbf{K}}_c(\theta)\mathbf{T} = \mathbf{K}_0 - \mathbf{K}_c(\theta, t). \tag{2}$$

Let the deflection of the system be $\mathbf{q} = \mathbf{q}_{st} + \mathbf{q}_{dy}$ where $\mathbf{q}_{st}$ is the static deflection of the un-cracked rotor due to gravity, and $\mathbf{q}_{dy}$ is the dynamic deflection due to the rotating out of balance and the effects of the crack. Thus, $\dot{\mathbf{q}} = \dot{\mathbf{q}}_{dy}$ and $\ddot{\mathbf{q}} = \ddot{\mathbf{q}}_{dy}$, and the equation of motion for the rotor in fixed co-ordinates is

$$\mathbf{M}\ddot{\mathbf{q}}_{dy} + (\mathbf{D}+\mathbf{G})\dot{\mathbf{q}}_{dy} + \left(\mathbf{K}_0 - \mathbf{K}_c(\theta,t)\right)\left(\mathbf{q}_{st} + \mathbf{q}_{dy}\right) = \mathbf{Q}_u + \mathbf{W} \tag{3}$$

where $\mathbf{Q}_u$ and $\mathbf{W}$ are the out of balance forces, and the gravitational force respectively. Damping and gyroscopic effects have been included as a symmetric positive semi-definite matrix $\mathbf{D}$ and a skew-symmetric matrix $\mathbf{G}$, although they have little direct bearing on the analysis. If there is axi-symmetric damping in the rotor then there will also be a skew-symmetric contribution to the undamaged stiffness matrix, $\mathbf{K}_0$. We refer to Equation (3) as the "full equations".

The steady state deflection of the rotor varies over each revolution of the rotor since $\mathbf{K}_c$ varies. However, the stiffness reduction due to the crack is usually small, and we may make the reasonable assumption that $\|\mathbf{K}_0\| \gg \|\mathbf{K}_c(\theta,t)\|$. With this assumption the steady state deflection is effectively constant and equal to the static deflection, $\mathbf{q}_{st}$, given by

$$\mathbf{K}_0\mathbf{q}_{st} = \mathbf{W}\,. \tag{4}$$

Equation (3) then becomes

$$\mathbf{M}\ddot{\mathbf{q}}_{dy} + (\mathbf{D}+\mathbf{G})\dot{\mathbf{q}}_{dy} + (\mathbf{K}_0 - \mathbf{K}_c(\theta,t))\mathbf{q}_{dy} = \mathbf{Q}_u\,. \tag{5}$$

The second approximation commonly used in the analysis of cracked rotors is weight dominance. If the system is weight dominated it means that the static deflection of the rotor is much greater than the response due to the unbalance or rotating asymmetry, that is $|\mathbf{q}_{st}| \gg |\mathbf{q}_{dy}|$. For example, for a large turbine rotor the static deflection might be of the order of 1 mm whereas at running speed the amplitude of vibration is typically 50 μm. Even at a critical speed the allowable level of vibration will only be 250 μm. In this situation, the crack opening and closing is dependent only on the static deflection and thus $\theta = \Omega t + \theta_0$, where Ω is the rotor speed and θ_0 is the initial angle. Thus Equation (5) becomes

$$\mathbf{M}\ddot{\mathbf{q}}_{dy} + (\mathbf{D}+\mathbf{G})\dot{\mathbf{q}}_{dy} + (\mathbf{K}_0 - \mathbf{K}_c(t))\mathbf{q}_{dy} = \mathbf{Q}_u \tag{6}$$

where $\mathbf{K}_c$ is now independent of θ.

MODELS OF A BREATHING CRACK

The breathing crack was initially studied by Gasch [10] who modeled the crack as a "hinge". In this model the crack is open for one half and closed for the other half a revolution of the rotor, and the transition from open to closed (and vice-versa) occurs abruptly as the rotor turns. Mayes and Davies [11-12] developed a similar model except that the transition from fully open to fully closed is described by a cosine function, that will be used in this paper. Penny and Friswell [13-14] compared the response due to different crack models and considered the effect on the dynamic response of the rotor.

In this paper the fully open crack is modeled by reducing the element stiffness in orthogonal directions (parallel and perpendicular to the crack face). The stiffness matrix of the machine when the crack is open, in rotating co-ordinates, is then $\tilde{\mathbf{K}}_1$. If weight dominance is assumed, then the opening and closing of the crack is periodic at the rotor spin speed. In the Mayes model the time dependent stiffness matrix in rotating coordinates is

$$\tilde{\mathbf{K}}_c(t) = 0.5 \times \left(1 - \cos(\Omega t + \theta_1)\right)\left[\tilde{\mathbf{K}}_0 - \tilde{\mathbf{K}}_1\right] \tag{7}$$

where θ_1 depends on the crack orientation and the initial angle of the rotor. When $\cos(\Omega t + \theta_1) = 1$ the crack is fully closed and $\tilde{\mathbf{K}}_{cr}(t) = \tilde{\mathbf{K}}_0$, the un-cracked rotor stiffness, where $\tilde{\mathbf{K}}_{cr}$ is defined in Equation (1). Thus the rotor is axi-symmetric when the crack is closed. When $\cos(\Omega t + \theta_1) = -1$ the crack is fully open so that $\tilde{\mathbf{K}}_{cr}(t) = \tilde{\mathbf{K}}_1$. Note that when the crack is open the rotor is asymmetric.

Converting from rotating to fixed co-ordinates is performed using the transformation given in Equation (2). The stiffness matrix in stationary co-ordinates, $\mathbf{K}_c(t)$, is a periodic function of time only and the full non-linear Equation (3) becomes a linear parametrically excited equation. Penny and Friswell [14] showed that the model generates a constant term plus 1X, 2X and 3X rotor angular velocity components in the stiffness matrix.

CONDITION MONITORING USING ACTIVE MAGNETIC ACTUATORS

Active magnetic bearings (AMB) have been used in high-speed applications or where oil contamination must be prevented, although their low load capacity restricts the scope of applications. Recently a number of authors have considered the use of AMBs as an actuator that is able to apply force to the shaft of a machine [15-18]. If the applied force is periodic, then the presence of the crack generates responses containing frequencies at combinations of the rotor spin speed and applied forcing frequency. The excitation by unbalance and AMB forces produces combination resonances between critical speed of the shaft, the rotor spin speed and the frequency of the AMB excitation. The key is to determine the correct excitation frequency to induce a combination resonance that can be used to identify the magnitude of the time-dependent stiffness arising from the breathing mode of the rotor crack.

The force applied on the rotor by the AMBs must be included in the equations of motion. Thus Equation (6) becomes

$$\mathbf{M}\ddot{\mathbf{q}}_{dy} + (\mathbf{D}+\mathbf{G})\dot{\mathbf{q}}_{dy} + (\mathbf{K}_0 - \mathbf{K}_c(t))\mathbf{q}_{dy} = \mathbf{Q}_u + \mathbf{Q}_{\text{AMB}} \quad (8)$$

where $\mathbf{Q}_{\text{AMB}}$ is the external forces applied to the rotor by the active magnetic bearing. This force will probably be chosen to be harmonic, either in one or two directions. Other waveforms would be possible if they were perceived to offer some advantage.

The key aspect of the analysis is that the system has three different classes of frequencies, namely the natural frequencies (or critical speeds), rotor spin speeds, and the forcing frequencies from the AMB. The parametric terms in the equations of motion (or non-linear terms in the full equations) cause combinational resonances in the response of the machine. Mani et al. [16, 18] and Quinn et al. [17] used a multiple scales analysis to determine the conditions required for a combinational resonance, which occurs when

$$\Omega_2 = |n\Omega - \omega_i|, \qquad \text{for} \quad n = \pm1, \pm2, \pm3 \quad (9)$$

where Ω is the rotor spin speed, Ω_2 is the frequency of the AMB force, and ω_i is a natural frequency of the system. This analysis was based on a two degree of freedom Jeffcott rotor model with weight dominance, equivalent to that described in this paper. Mani et al. [18] also considered the effect of detuning, that is when the excitation is close to this exact excitation frequency for resonance, and investigated the effect on the magnitude of the primary resonance close to the natural frequency of the machine. In the examples the running speed of the machine was five times higher than the natural frequency. This ratio is not practical since there is likely to be a second un-modeled resonance below the running speed. Indeed, the fact that higher resonances are not modeled is a serious omission, particularly as the combinational resonances are likely to excite any higher frequency resonances.

THE TIME SIMULATION

The analysis thus far has indicated the combinational resonances that are likely to occur in a machine with a breathing crack, excited by a magnetic actuator. Most of the analysis in the literature has been performed on simple two degree of freedom models of the machine, with simplifying assumptions concerning the crack model, gyroscopic effects, higher modes and so on. In order to check the robustness of the frequency content of the machine response a time simulation will be performed on a detailed model of the machine. This will allow realistic features of the real machine to be easily incorporated. To ensure the transient response decays within a reasonable time, damping is added to the bearings and/or disks. The equations of motion are integrated using the Runge-Kutta method. However the number of degrees of freedom of a detailed finite model is likely to be large, requiring a long computational time to simulate the response. Thus the equations of motion in the rotating frame are reduced using the lower mode shapes of the undamped and undamaged machine, neglecting gyroscopic effects. A sufficient number of modes should be included to simulate the range of excitation frequencies, and also any significant combinational resonances. This reduction has two beneficial effects; not only are the number of degrees of freedom reduced, leading to a lower computational cost per time step, but also the higher frequencies are removed, thus allowing a larger time step.

The reduction procedure is to calculate the eigenvectors of the undamped and undamaged machine as

$$\left[-\omega_{0i}^2\mathbf{M} + \mathbf{K}_0\right]\boldsymbol{\phi}_i = \mathbf{0} \quad (10)$$

where ω_{0i} and $\boldsymbol{\phi}_i$ are the ith natural frequency and mode shape. If the lower r modes are retained then the reduction transformation is

$$\mathbf{T}_r = \left[\boldsymbol{\phi}_1 \ \boldsymbol{\phi}_2 \dots \boldsymbol{\phi}_r\right]. \quad (11)$$

The reduced equations of motion, assuming the mode shapes are mass normalized, are then

$$\ddot{\mathbf{q}}_r + (\mathbf{D}_r + \mathbf{G}_r)\dot{\mathbf{q}}_r + (\boldsymbol{\Lambda}_0 - \mathbf{K}_{rc}(t))\mathbf{q}_r = \mathbf{Q}_{ru} + \mathbf{Q}_{r\text{AMB}} \quad (12)$$

where

$$\begin{aligned} &\mathbf{q}_{dy} = \mathbf{T}_r\mathbf{q}_r, \quad \mathbf{D}_r = \mathbf{T}_r^T\mathbf{D}_r\mathbf{T}_r, \quad \mathbf{G}_r = \mathbf{T}_r^T\mathbf{G}_r\mathbf{T}_r, \\ &\mathbf{K}_{rc} = \mathbf{T}_r^T\mathbf{K}_c\mathbf{T}_r, \quad \mathbf{Q}_{ru} = \mathbf{T}_r^T\mathbf{Q}_u, \quad \mathbf{Q}_{r\text{AMB}} = \mathbf{T}_r^T\mathbf{Q}_{\text{AMB}}, \end{aligned} \quad (13)$$

and $\boldsymbol{\Lambda}_0 = \text{diag}\left(\omega_{01}^2, \omega_{02}^2, \dots, \omega_{0r}^2\right)$ is the diagonal matrix of undamped and undamaged eigenvalues.

The equations are integrated until a steady state has been established and then the FFT is calculated. The steady state response should only contain the excitation and rotor spin frequencies, and the combinational resonances, and therefore the spectrum of the response should only contain discrete frequencies. However, leakage is likely to occur because of the difficulties in choosing a sample period so that every sinusoidal component in the response has an integer number of cycles in the sample. The effect of leakage may be reduced by using time window functions. Furthermore the sample period must be sufficiently long to ensure that the frequency increment is small enough to distinguish the individual frequency components.

CASE STUDY

The approach is demonstrated on a machine with the rotor supported on two ball bearings and one active magnetic actuator. The simulated machine is based on the experimental test rig described in the next section. Fig. 1 shows a schematic of the machine and indicates the 30 finite elements used to model the rotor. The shaft diameter is 15.875 mm and the shaft length is 0.659 m. The diameter of the active magnetic bearing rotors and radial actuator is 47.625 mm. The disk has a diameter of 127 mm and a thickness of 12.7 mm. The shaft is running on single row deep groove ball bearings which are modeled as constant stiffness bearings with stiffness 3 MN/m at nodes 2 and 29, as shown in Fig. 1. Although the rotor is configured to be supported by conical AMBs, in this study for simplicity these are not activated. Fig. 2 shows the first two pairs of mode shapes at 10000 rpm, and Fig. 3 shows the Campbell diagram up to a rotor spin speed of 25000 rpm.

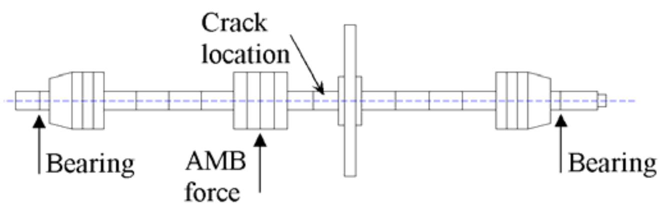

Figure 1. The finite element model of the test rig.

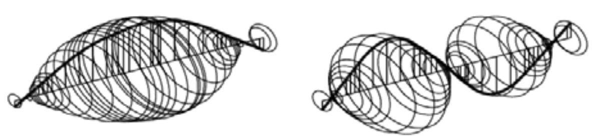

Figure 2. The first two flexible modes pairs of the undamaged machine at 10000 rpm.

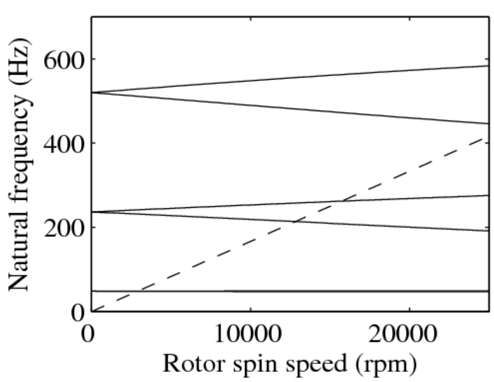

Figure 3. The Campbell diagram of the undamaged machine.

An unbalance of 1 kg-mm is assumed on the disk. The sinusoidal force from the magnetic actuator is assumed to be 200 N peak-peak. The crack is located between the disk and the active magnetic actuator, and in the simulation is assumed to be within the element nearest the disk. The stiffness reduction for the fully open crack is taken to be 30% and 20% in the two orthogonal directions. The rotor spin speed is assumed to be 5400 rpm (90 Hz). To ensure the transient response decays within a reasonable time, the bearings are assumed to have a damping of 10 Ns/m, and, in addition, a damper with coefficient of 100 Ns/m is simulated between the disk and ground. The equations of motion in the rotating frame are reduced using the lower 12 modes. Fig. 4 shows the response of the rotor at the disk when the machine is undamaged, and the magnetic actuator is turned off. As expected, the steady state response is solely due to unbalance and occurs only at the rotor spin speed of 90 Hz.

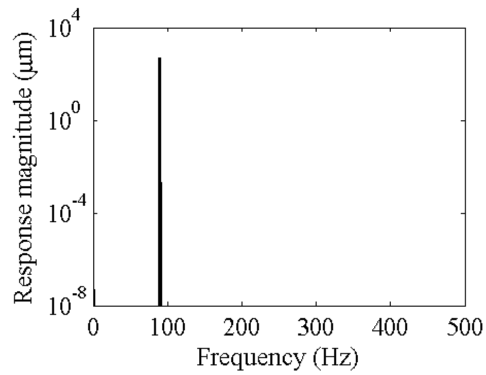

Figure 4. The response of the undamaged machine.

When the crack is introduced the harmonics of rotor spin speed also occur, as shown in Fig. 5. As expected, the crack induces frequencies being multiples of rotor speed. Fig. 6 shows the response of the undamaged machine with the magnetic actuator excited the rotor at 55 Hz, which is close to the first resonance frequency of 48 Hz.

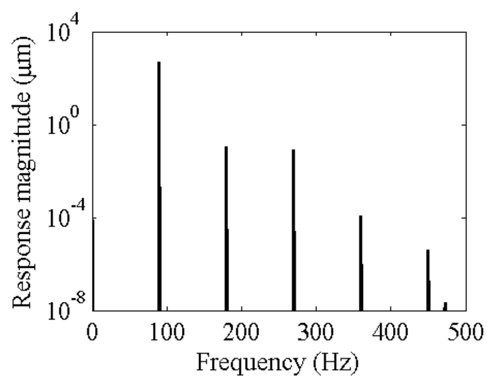

Figure 5. The response of the damaged machine.

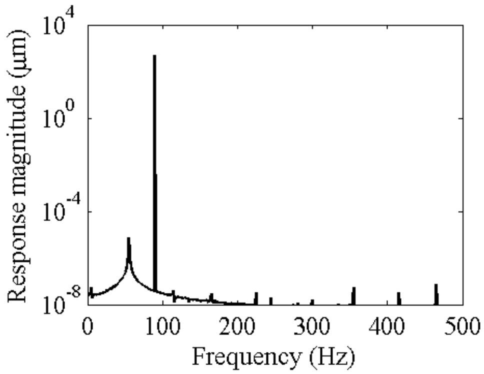

Figure 6. The response of the undamaged machine with a magnetic actuator frequency of 55 Hz.

Suppose now that the magnetic actuator is used to excite the machine with the cracked rotor. The first resonance occurs at approximately 48 Hz, and for a rotor spin speed of 90 Hz, and $n = 1$ in Equation (9), the excitation frequency is 42 Hz. Fig. 7 shows the response in this case. The increased response at the actuator frequency of 42 Hz is barely discernable due to the response at the first resonance at approximately 48 Hz. The

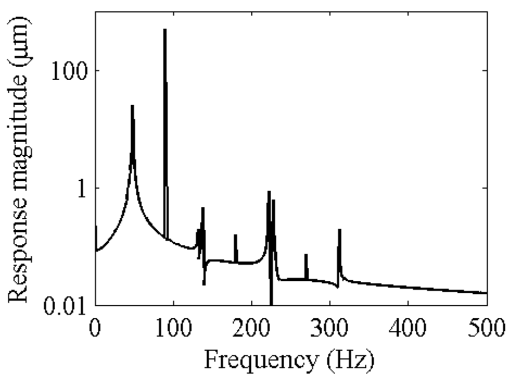

Figure 7. The response of the damaged machine with a magnetic actuator frequency of 42 Hz.

response at the rotor spin speed (90 Hz) and its harmonics (180 and 270 Hz) are clear. There are also peaks at 132, 138, 222, 228 and 312 Hz due to combinational resonances.

Figure 8 shows the response at the disk if the rotor is excited by the magnetic actuator with a frequency of 132 Hz ($n = 2$ in Equation (9)). Again the increased response at the actuator frequency of 132 Hz is barely discernable. The response at the rotor spin speed and its harmonics is again clear. In this case there are also clear peaks at 42, 138, 222 and 312 Hz due to combinational resonances.

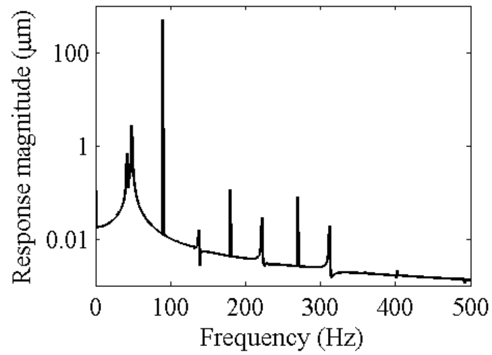

Figure 8. The response of the damaged machine with a magnetic actuator frequency of 132 Hz.

EXPERIMENTAL FACILITY

The Center for Rotating Machinery Dynamics and Control (RoMaDyC) at Cleveland State University has recently acquired a rotor crack detection test rig, Fig. 9, with active magnetic bearings (AMB) shown in Fig. 10. The system was built by Revolve Magnetic Bearings Inc., a subsidiary of AB SKF, Sweden. Each AMB is an 8-pole radial or conical heteropolar design and is equipped with four variable reluctance type position sensors. Each magnetic actuator provides actuation in two perpendicular axes, which are rotated 45° from the vertical. In addition, using the conical design allows an axial and/or radial excitation to be imparted to the rotor, which might be essential for the development of health monitoring techniques for cracked disks. The actuators are axially adjustable, i.e., can be placed at almost any axial location. The actuators are calibrated and applied currents and/or fluxes are monitored to determine applied forces. The waveform of external excitation is introduced in MATLAB/SIMULINK and transferred to hardware via dSPACE DS1103 board.

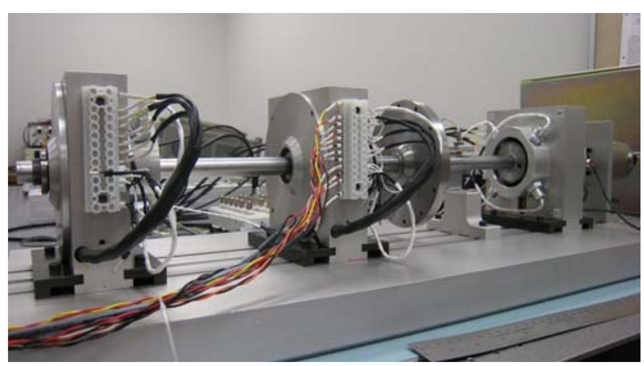

Figure 9. Rotor crack detection rig.

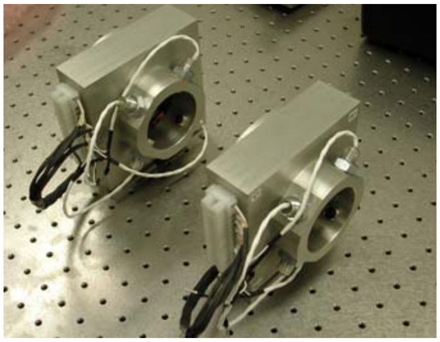

Figure 10. Active magnetic force actuator.

The 48 V DC, 0 to 15,000 rpm brush type motor is controlled by a digital controller. Rotor speed and motor torque are monitored and the controller will shut off the power to the motor in the event of a fault or alarm. The shaft is connected to the motor using a lightweight, flexible coupling that allows radial and axial movement of the shaft. The experimental test stand is designed to operate either on ball bearing shaft supports or on AMBs. The ball bearings are oil lubricated, deep groove precision ball bearings rated for full test rig speed of 15,000 rpm. They are mounted in support housing, similar to the touchdown bearings, and the support housing can be bolted to the side of either radial or conical magnetic bearing pedestals. The static load capacity of radial and conical AMBs is 266 N (60 lbf) with a dynamic load capacity of 48 N (11 lbf) at 1,000 Hz. The nominal air gap of the magnetic bearings is 381 µm (0.015 in) and the design gap between the auxiliary bearings and the landing sleeves on the rotor shaft is 190 µm (0.0075 in). The balance disks with threaded holes drilled for balancing are mounted onto the shaft using a hydraulic mount hub.

Two digital controllers provide control loop updates at a 10 kHz frequency and they essentially control and power the magnetic bearings while allowing the user to view useful information about the system under operation. The dSPACE platform is used to have complete control over the experiment by providing access to every sensor signal, user defined control command, and operational parameter.

Figure 11 shows the measured transfer function between the current disturbance and the rotor response. The force excitation was applied to the non-rotating rotor by magnetic actuator at node 13 (see Fig. 1) as a result of injecting the current sweep over the frequency from 0 to 1,000 Hz. The rotor response was measured by sensors located at the node 15. The measured resonances are 47, 225, 435, and 636 Hz, and they agree well with rotordynamic predictions presented in previous section.

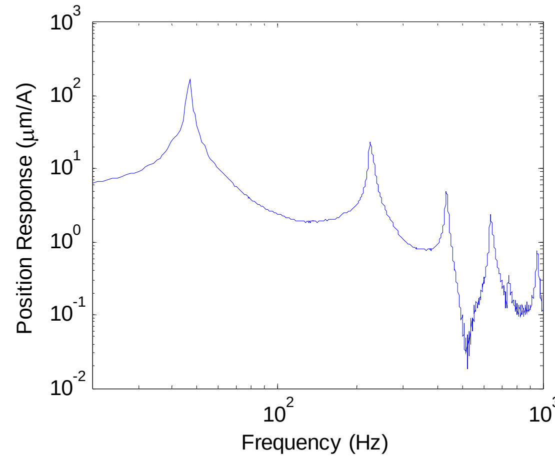

Figure 11. The measured transfer function of the undamaged machine (no rotation).

To approximate the effect of a crack, a notch was cut at the location of the shaft between the magnetic actuator and the unbalance disk, as denoted in Fig. 1. The notch had a width of 0.94 mm (0.037 inch) and a depth of 40% of shaft diameter. Figure 12 shows the measured response at the node 15 of the machine with the damaged (notched) and healthy rotor due to the residual unbalance. The response is characterized by major peak at the synchronous frequency 36.7 Hz. The noise-induced harmonics of the spin speed (2X, 3X, …) can also be seen in the response but their magnitude is very small compared to 1X frequency component (note the logarithmic scale). It can be observed that the response magnitude is higher almost at all frequencies for the damaged machine.

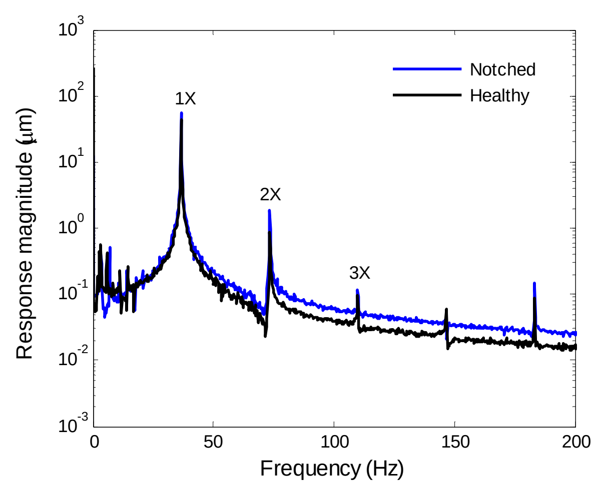

Figure 12. The measured response of the damaged and healthy machine, Ω=2200 rpm (36.7 Hz), no external excitation.

Figure 13 compares the measured response at node 15 of the machine with the damaged (notched) and healthy rotor due to the residual unbalance and the externally applied at the node 13 magnetic force excitation having frequency $\Omega_2 = 63$ Hz ($n = 3$ in Equation (9)) and amplitude of 200 N. Figure 14 presents the simulated results for the same conditions for the damaged rotor. Both figures clearly show the rotor spin speed and its harmonics (1X, 2X, 3X, …). Also, the response at the actuator frequency of 63 Hz can be noted. In addition to these responses, in this case, there are visible peaks of combinatorial resonances at 26.3, 63, 83.6, 99.7, 120.3 and 136.3 Hz.

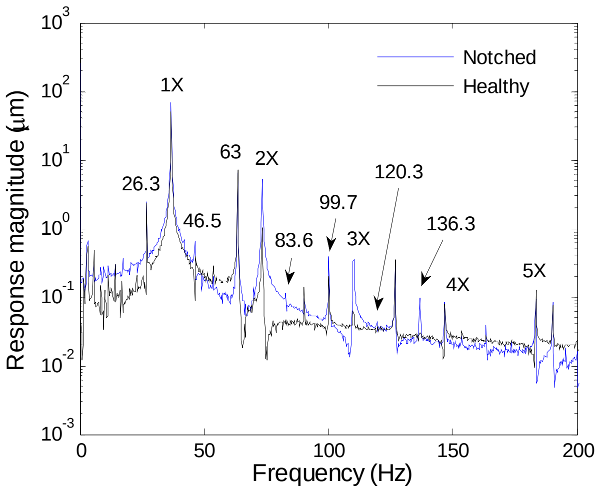

Figure 13. The measured response of the damaged and healthy machine, Ω=2200 rpm (36.7 Hz), magnetic actuator frequency 3780 rpm (63 Hz).

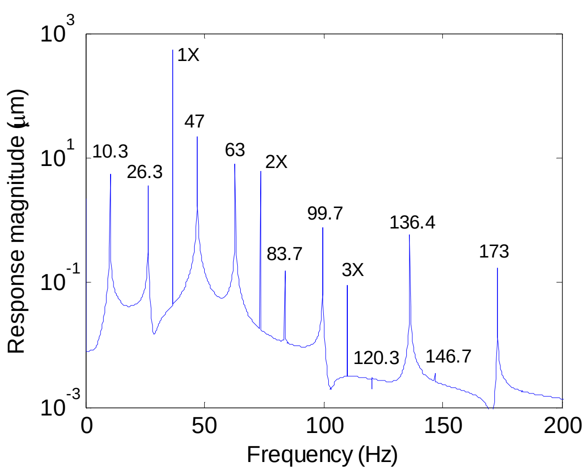

Figure 14. The simulated response of the damaged machine, Ω=2200 rpm (36.7 Hz), magnetic actuator frequency 3780 rpm (63 Hz).

All these frequencies are the result of the application of external force excitation which produces combination resonances based on the magnetic actuator frequency, the rotational speed, and the natural frequencies. The combination frequencies present a unique crack signature that could be used to detect cracks in the rotor.

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CONCLUSIONS

The modeling and experimental approach has been presented in attempt to use the changes in the dynamics of a rotor to identify and possibly locate crack (and other faults) in a rotor at an early stage in their development. The presented simulation and experimental results show that the use of an active magnetic bearing (AMB) to excite the rotor with a harmonic force at an appropriate frequency produces components in the system response at many frequencies which are combinations of the rotor speed, the AMB excitation frequency and the system natural frequency. These combination frequencies have a potential to be used to detect cracks in the rotor. The presented approach has some merit, but further work is needed to produce a robust condition monitoring technique.

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