THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47th St., New York, N.Y. 10017

94-GT-92

ACTIVE CONTROL OF DYNAMIC BEARING LOADS IN ROTATING MACHINERY USING NON-INVASIVE MEASUREMENTS

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ABSTRACT

A new technique for measuring and actively controlling dynamic bearing loads in rotating machinery is presented. Bearing loads are estimated using the Deflection-Coefficient Method, a technique which does not rely on a full system model, and which applies commonly-used shaft-deflection measurement equipment to obtain estimates of bearing loads. The estimated bearing load is used as an error signal in an adaptive feedforward disturbance rejection controller. The result is a control system which can selectively minimize dynamic bearing loads in real time in rotating machinery systems. The method is applied to a numerical model of a typical rotating machinery system to suppress dynamic reaction forces at bearing supports.

INTRODUCTION

The availability of powerful computing equipment and precision actuators and sensors has led to an increased interest in the use of active control techniques for suppressing vibrations in rotating machinery. Much of that effort has focused on suppressing shaft displacements [Burrows et al., 1989, Stanway and Burrows, 1981, Ulsoy, 1984], and for good reason, since a primary objective in rotating machinery operation is to prevent interference between rotating and stationary components. An issue that is often seen as secondary is that of dynamic loads in bearings which are a result of the vibrations. Prolonged exposure to excessive loads can lead to reductions in bearing life, and ultimately increased maintenance and down-time for machinery [Collacott, 1977, Bradfield et al., 1991]. For that reason, suppression of bearing loads can be just as important to the life of a rotating machine as can controlling vibration amplitudes.

Active control of bearing loads has seen little attention in the literature for two primary reasons. Most importantly, bearing loads are very difficult to measure, especially as compared to shaft deflections, and without measurement they are difficult to control. The general line of reasoning used in much of the literature has been, then, that by controlling shaft deflections, dynamic bearing loads will be held in check. Recent studies have shown, however, that decreased shaft deflections do not necessarily coincide with decreased bearing loads, and can, in fact, result in increased loads [Clark et al., 1993, Kim et al., 1993].

This paper presents a method whereby dynamic bearing loads in a rotating machine are actively controlled. At the heart of the control process is a method for estimating bearing loads, called the Deflection-Coefficient Method [Marangoni, 1990]. The method relies on standard rotating machinery displacement sensors and beam theory to infer the dynamic loads in a bearing support. No detailed model of the system is required, other than information about the shaft properties and its boundary conditions. The bearing load information is then incorporated into a feedforward vibration control method to minimize the reaction forces at the bearing. The result is a straightforward, non-invasive method which uses standard sensors and a common control approach that can be retrofitted to existing rotating machinery systems or incorporated into new systems.

The paper first presents the Deflection-Coefficient Method (DCM) for estimating bearing loads, which involves determining shear forces in the shaft from a set of closely-spaced deflection measurements. The technique is verified by comparing its results to the exact analytical reaction forces found in a simply-supported and a clamped-pinned beam. Next, the controller is presented. The control technique involves applying a controlling force to the system that cancels the effects of a disturbance which may arise from rotating unbalance or shaft misalignment. The magnitude and phase of

Presented at the International Gas Turbine and Aeroengine Congress and Exposition
The Hague, Netherlands — June 13–16, 1994

the control force is adapted (using a Least Mean Square algorithm) so that an error signal, which in this case is the bearing load signal, is minimized.

After the bearing-load measurement and the control techniques are presented, the method is applied to a model of a rotating machinery system. A rotor-coupler-motor system is modeled as a flexible beam with lumped masses which vibrates in two uncoupled planes. To simplify the analysis, the control is applied in one plane only. Simulations are carried out to show the performance of this technique in minimizing dynamic reaction forces. Bearing loads are determined using the deflection-coefficient method, and the control force is generated to minimize that signal. The results show that the bearing loads can be greatly reduced using this technique, particularly when the first system mode dominates the response.

DYNAMIC BEARING LOAD ESTIMATION–THE DEFLECTION-COEFFICIENT METHOD

The basic idea presented here for determining bearing loads in a rotating machinery system is that the load in a supporting bearing is related to the shear and moment in the shaft at the location of the bearing. These can be approximated using several displacements of the shaft, measured very close to the bearing. The technique for making that approximation is described below, and the results are presented for a pinned-pinned and for a clamped-pinned beam.

If gyroscopic forces are small in a rotating machinery system, then the two planes of vibration can be uncoupled, and the vibration problem in either plane can be treated as a beam problem. From elementary flexural beam theory [Weaver et al., 1990], the general deflection equation for a beam such as those shown in Fig. 1 is

$$\frac{d^4 y(x)}{dx^4} - \beta^4 y(x) = 0 \quad (1)$$

which has a solution of [James et al., 1990]

$$y(x,t) = \sum_{i=1}^{\infty} \Phi_i(x) q_i(t) \quad (2)$$

or, for a steady-state harmonic forcing function,

$$y(x,t) = \sum_{i=1}^{\infty} \Phi_i(x) Q_i \sin(\omega t + \phi_i) \quad (3)$$

The contribution of any given mode at some point in time is

$$y(x) = C_1 \sin\beta_i x + C_2 \cos\beta_i x + C_3 \sinh\beta_i x + C_4 \cosh\beta_i x \quad (4)$$

In Eqs. 1 and 4, β_i is a function of the natural frequencies of the beam,

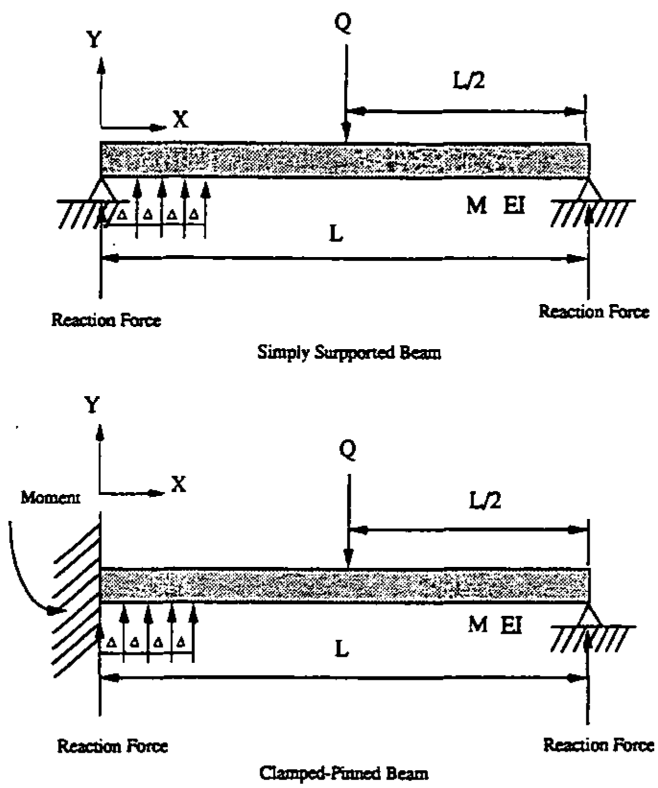

Figure 1. Simply-supported and clamped-pinned beams.

$$\beta_i = \sqrt[4]{\frac{\rho\omega_i^2}{EI}} \quad (5)$$

where ρ is the mass per unit length of the shaft, E is its Young's modulus, I is its moment of inertia, and ω_i is the natural frequency of the i th mode of the shaft. The coefficients C_j in Eq. 4 depend on the boundary conditions and the loading of the beam. Since the loading of a shaft in a rotating machinery system is due to the reactions at the bearings and to any disturbances applied to the shaft, the bearing loads may be inferred if the coefficients of Eq. 4 can be found. By obtaining four measurements of the shaft deflections at locations near the beam, the four coefficients of the general deflection expression can be determined for the loading conditions at any time from the following set of equations

$$\begin{bmatrix} y_1 \\ y_2 \\ y_3 \\ y_4 \end{bmatrix} = \begin{bmatrix} \sin\beta x_1 & \cos\beta x_1 & \sinh\beta x_1 & \cosh\beta x_1 \\ \sin\beta x_2 & \cos\beta x_2 & \sinh\beta x_2 & \cosh\beta x_2 \\ \sin\beta x_3 & \cos\beta x_3 & \sinh\beta x_3 & \cosh\beta x_3 \\ \sin\beta x_4 & \cos\beta x_4 & \sinh\beta x_4 & \cosh\beta x_4 \end{bmatrix} \begin{bmatrix} C_1 \\ C_2 \\ C_3 \\ C_4 \end{bmatrix} \quad (6)$$

or

$$\mathbf{Y} = \mathbf{AC} \tag{7}$$

where in the matrix $\mathbf{A}$ only the contribution of the fundamental vibration mode is considered, so that $\beta = \beta_1$. The vector of coefficients may be solved as

$$\mathbf{C} = \mathbf{A}^{-1}\mathbf{Y} \tag{8}$$

From Eq. 4 the expression for shear may be found by recalling that shear is proportional to the third derivative of deflection, or

$$V(x) = EIy'''(x) = EI\left(-C_1\beta^3 \cos\beta x + C_2\beta^3 \sin\beta x + C_3\beta^3 \cosh\beta x + C_4\beta^3 \sinh\beta x\right) \tag{9}$$

To find the reaction force on the left end of the beam, Eq. 9 can be evaluated at $x = 0$, or

$$V(x=0) = EI\beta^3(C_3 - C_1) \tag{10}$$

In a rotating machinery system, Eqs. 8-10 may be used to approximate, in real time, the reaction forces at supports (which are the bearing loads) for diagnostic or control purposes. The steps involved are: 1) measure deflections near the bearings, 2) use Eq. 8 to determine the coefficients of the deflection equation (note that for shear at the end of a shaft, only two coefficients need to be evaluated), and 3) from Eq. 10 determine the shear.

The above discussion presents a general approach to using the deflection-coefficient method for approximating reaction forces at bearings. A few comments are in order concerning its applicability. No assumptions have been made about boundary conditions. The boundary conditions can be thought of as loading conditions on the shaft, and as such, are incorporated into the solution of the deflection coefficients in Eq. 8. Note also that the deflection-coefficient method only provides approximate information about the bearing loads because the deflection shape is assumed to take on a first-mode fit. In general, since there are four coefficients to be evaluated in Eq. 4, four displacement measurements are necessary. Equation 10, however, shows that only two are necessary for end-bearings, so the other two could be used to provide information about higher modes, and thus provide a more accurate approximation of bearing load. As will be seen from the following examples, the technique is quite accurate when first mode is the dominant mode in the displacement response.

Examples

As an example of this technique, reaction forces will be determined for two beams with different boundary conditions. The first case is the simply-supported beam shown in Fig. 1, and the second case is the clamped-pinned beam shown in Fig. 1. In each case the shear (at $x = 0$) is approximated by Eq. 10. The beams have the following characteristics: $\rho = 7861 kg/m^3$, $E = 2.0 \times 10^{11} N/m^2$, and $l = 1m$. The area moments of inertia were chosen to give fundamental frequencies of 56.2 Hz for each beam ($I = 3.00 \times 10^{-8} m^4$ for the pinned-pinned beam and $I = 5.06 \times 10^{-9} m^4$ for the clamped-pinned beam).

Numerical simulations were run in order to determine the validity of the deflection-coefficient method for bearing-load estimation discussed above. In each case, a disturbance with frequency of 10 Hz and magnitude of 10 N was applied to the beam (at mid-span). Numerical simulations were carried out to obtain time-deflection information so that reaction forces could be determined three ways. First, the deflection-coefficient method was used where the simulation provided deflection information for the beam locations specified in Fig. 1, then Eq. 8 was used to find the unknown deflection coefficients, and Eq. 10 was used to determine the reaction forces. The separation distance between deflection "measurements" (see Δ in Fig. 1) was set at $0.01l$. It was found that the best accuracy in reaction-force measurement was obtained when $\Delta \leq 0.02l$. The second method used was to obtain the reaction forces directly from the finite-element model of the systems. Finally, the previous two methods were compared to the analytical reaction forces, a derivation for which can be found in Weaver et al. [1990], for example. Each of the reaction-force results are plotted as a function of time in Figs. 2, 3, and 4, for the simply-supported and clamped-pinned cases, respectively. In each case, there is clearly no noticeable difference between any of the three techniques.

DEFLECTION COEFFICIENT METHOD USED IN ACTIVE BEARING-LOAD CONTROL

An adaptive feedforward control method is used in this work to suppress bearing reaction forces. The basic idea is to produce, on the structure, a control input whose response exactly cancels the response of the unwanted disturbance, as measured by an appropriate sensor. The response to be canceled is bearing load as determined by the deflection coefficient method.

The difficulty in carrying out feedforward disturbance cancellation is two-fold. In most cases, as in the case of rotating machinery, the disturbance force is not available for measurement, and even if it were, the optimal location for applying the control force is not available (the most straightforward method would be to apply an equal-and-opposite force at the location of the disturbance, but in a rotating machine that is not possible since the disturbance usually occurs somewhere within the rotor). The result is a control force which is applied at some available location, but which follows a different path than the disturbance force to the point of response in the structure, resulting in a different magnitude and phase shift. For example, the lateral force in a bearing due to a rotating

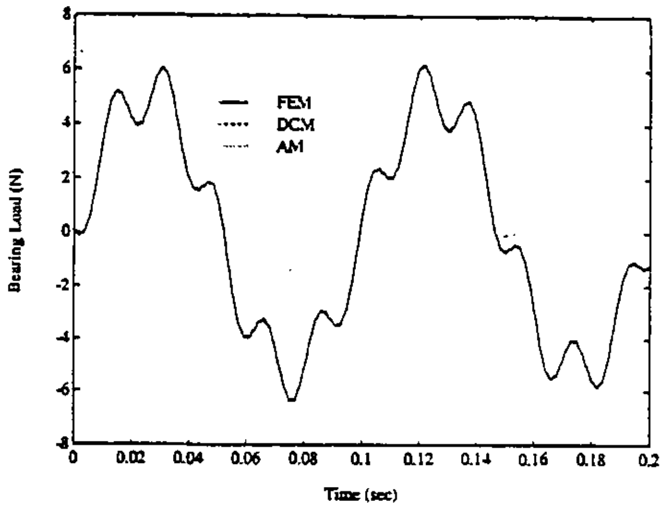

Figure 2. Reaction force in simply-supported beam.

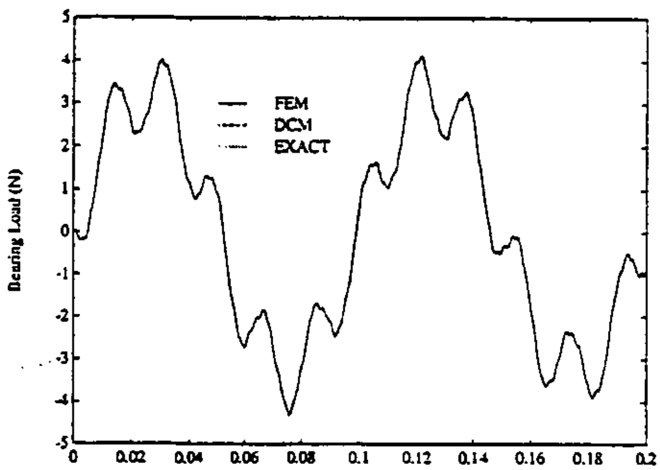

Figure 3. Shear force in pinned end of clamped-pinned beam.

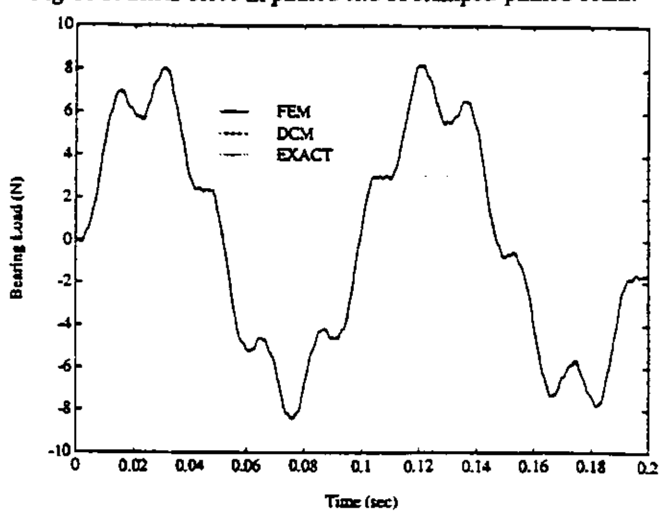

Figure 4. Shear force in clamped end of clamped-pinned beam.

unbalance disturbance applied at mid-span of the rotor may be

$$F_d(t) = F_d \sin(\omega_d t + \phi_d) \tag{11}$$

and that due to the control force may be

$$F_c(t) = F_c \sin(\omega_d t + \phi_c) \tag{12}$$

so the total dynamic bearing force is

$$F(t) = F_d \sin(\omega_d t + \phi_d) + F_c \sin(\omega_d t + \phi_c) \tag{13}$$

In order for the controller to cancel the effect of the disturbance, we need $F_c = -F_d$ and $\phi_c = \phi_d$. This does not happen, in general, but can be brought about by passing the control force through a filter whose magnitude and phase are properly adjusted so that the added responses of the disturbance and control cancel. This is a common technique in vibration and noise control [Fuller et al., 1989, Widrow et al., 1975]. In this paper the filter adjustment will be done on-line using the Least Mean Square (LMS) algorithm [Widrow and Stearns, 1985].

Model Development

The system to be controlled in this work is a model of a typical rotating machinery system shown in Fig. 5. The rotor could be a compressor, pump, fan, or some other common piece of equipment. The rotor is driven by a motor or turbine, which is connected to the rotor through a coupling device. In this work, the coupler will also serve as the location for a control actuator. In many instances this is the only feasible location for such a control input, since the rest of the shaft length (inside the motor and inside the rotor) is not accessible. In this model, the actuator has been chosen to be a noncontacting force actuator, such as a magnetic actuator. The system is supported by four bearings, numbered in Fig. 5. Bearings 1 and 2 support the rotor while 3 and 4 support the motor.

The system shown in Fig. 5 is modeled using a finite element approach. Twelve Bernoulli-Euler beam elements are used in modeling the rotor and shaft to represent the translational and

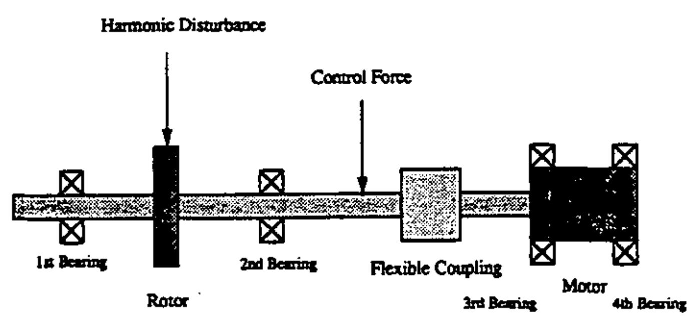

Figure 5. Illustration of rotating machinery system.

rotational displacements. The bearings are modeled as pinned supports. The resulting model is a set of second-order differential equations describing the motions of the rotating machinery system which can be written in matrix form as

$$\mathbf{M}\ddot{\mathbf{w}} + \mathbf{C}\dot{\mathbf{w}} + \mathbf{K}\mathbf{w} = \mathbf{F}_d f_d + \mathbf{F}_c f_c \qquad (14)$$

where f_d is the disturbance force, and f_c is the control force, and the matrices $\mathbf{F}_d$ and $\mathbf{F}_c$ describe the force input dynamics and locations. The fundamental frequency of the modeled system is 56.2 Hz.

The system model represented by Eq. 14 includes no gyroscopic terms and no damping. The most prominent damping term in flexural vibrations of a rotating system is structural damping, which is assumed to be negligible in this application. (A small amount of viscous damping is included in the model to eliminate transients in the numerical simulations and to allow the system to reach steady-state faster.) Gyroscopic effects are also neglected because of the symmetry provided by the bearing supports and the rotor placement.

Based on these assumptions, the motions in the vertical and horizontal planes are uncoupled. In actual implementation, the approaches discussed here will be duplicated, once in the vertical and once in the horizontal plane, to achieve full vibration control.

The disturbance force, f_d in Eq. 14, is assumed to result from an unbalance in the rotor. The cause of this disturbance force is a non-symmetric distribution of mass around the rotor, so as the shaft rotates, an unbalanced centrifugal force is applied to the shaft. The frequency of the disturbance is equal to the rotating shaft frequency, and the force (in the vertical plane) can be written as

$$f_d(t) = me\omega^2 \cos(\omega t) \qquad (15)$$

where m is the unbalanced mass, e is the effective eccentricity, and ω is the shaft rotational speed. The horizontal component is shifted by 90°.

Controller Development

The control force, f_c, is applied near the coupler, as shown in Fig. 5. It is made up of a harmonic signal whose frequency is set equal to that of the disturbance (this is easily done in practice since the disturbance usually occurs at the shaft speed, or multiples thereof, which can be measured directly). To obtain the proper magnitude and phase, the control signal is passed through a finite impulse response (FIR) filter so that the actual control force applied to the structure is

$$f_c(t) = (w_0 + w_1 z^{-1})u(t) \qquad (16)$$

where w_0 and w_1 are the filter weights, z^{-1} represents a unit time delay, and u is a control-reference input of the form

$$u(t) = u_o \cos(\omega t) \qquad (17)$$

The reference input can be obtained directly from the rotating shaft. The LMS algorithm recursively calculates new weights, based on the current values for the weights, the control input, and the "error" signal, according to the equation [Widrow and Stearns, 1985, Widrow et al., 1975]

$$\mathbf{w}_{k+1} = \mathbf{w}_k + 2\mu\varepsilon_k \mathbf{u}_k \qquad (18)$$

where $\mathbf{w}_k$ is the vector of filter weights at the current time k, $\mathbf{u}_k$ is the vector of current and previous control inputs, and μ is a convergence parameter. The term ε_k is the current output value, and is called the error. This is the system variable that is to be minimized. In this work the error signal is chosen to be the load carried by a given bearing in the system, so from Eq. 10, the error can be written as

$$\varepsilon_k = EI\beta^3 (C_{3_k} - C_{1_k}) \qquad (19)$$

When both the disturbance and control forces are applied to the system, the LMS algorithm recursively adapts the filter weights using Eq. 18 to minimize the error signal, which is bearing load.

RESULTS OF ACTIVE BEARING-LOAD CONTROL

The feedforward control technique described above was used in conjunction with the deflection coefficient method to control bearing loads in the system shown in Fig. 5 under persistent excitation. Three cases were simulated. In each case a harmonic disturbance (with amplitude 22.2 N) was applied at mid span of the rotor section, and had frequencies of 10, 30, and 65 Hz, respectively. The control force was applied at the flexible coupling. The controller was formulated such that the load in bearing 1 was minimized, that is, the error in Eq. 19 was the estimate of load in bearing 1 from the deflection-coefficient method. The simulated results are shown in Figs. 6-14. Only the results for vibrations in the vertical plane are shown, as the horizontal plane roughly duplicates these results with a 90° phase shift. Figures 6, 9, and 12 show the bearing load in each case as determined by the finite element model of the system. The error signal, which is a deflection-coefficient-method approximation to the bearing load is shown in Figs. 7, 10, and 13. In each plot both an uncontrolled and a controlled portion is plotted. The first 20% of each plot shows the bearing load response to the disturbance with no control applied. Then the control force is allowed to enter the system, and it is adapted, starting with weights of $\begin{bmatrix} 0 & 0 \end{bmatrix}^t$, to minimize the error. Figures 8, 11, and 14 show the FIR filter weights (convergence

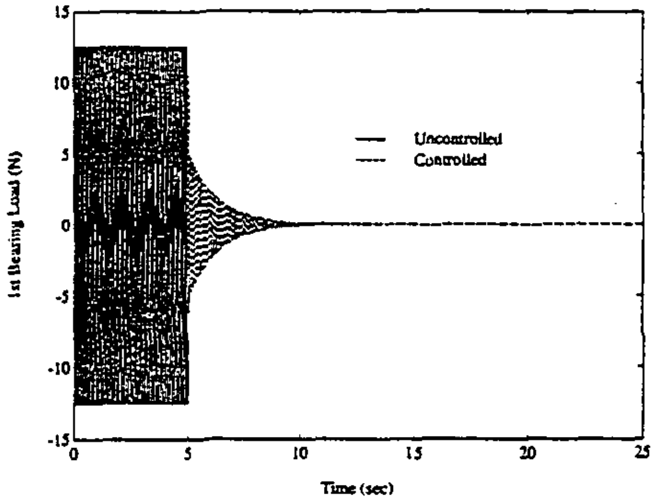

Figure 6. 1st Bearing load from FEM (10 Hz case).

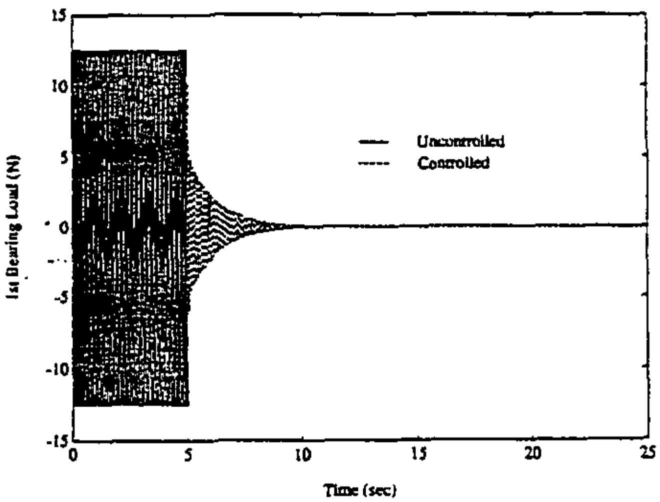

Figure 7. 1st Bearing load from DCM (10 Hz case).

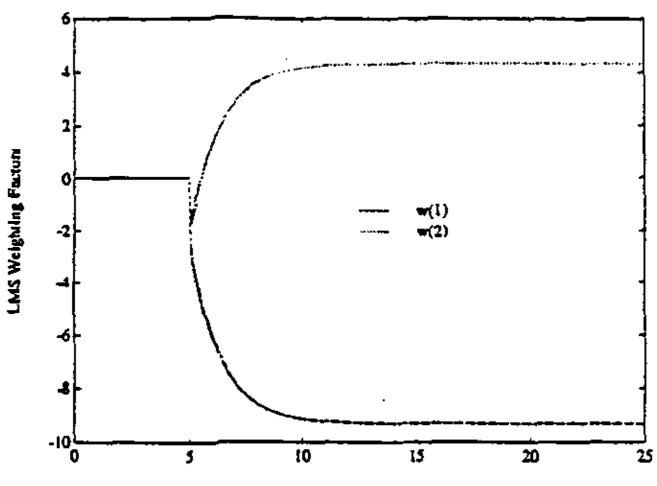

Figure 8. FIR filter coefficients (10 Hz case).

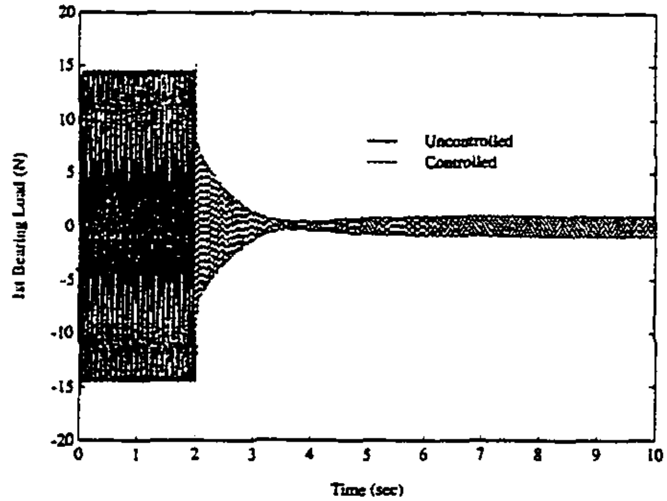

Figure 9. 1st Bearing load from FEM (30 Hz case).

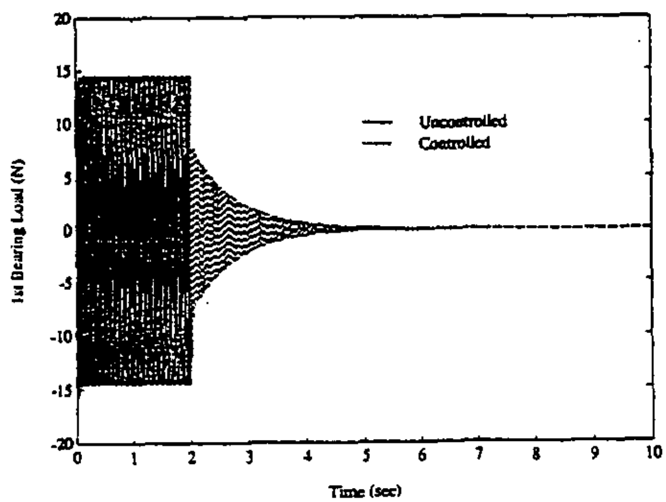

Figure 10. 1st Bearing load from DCM (30 Hz case).

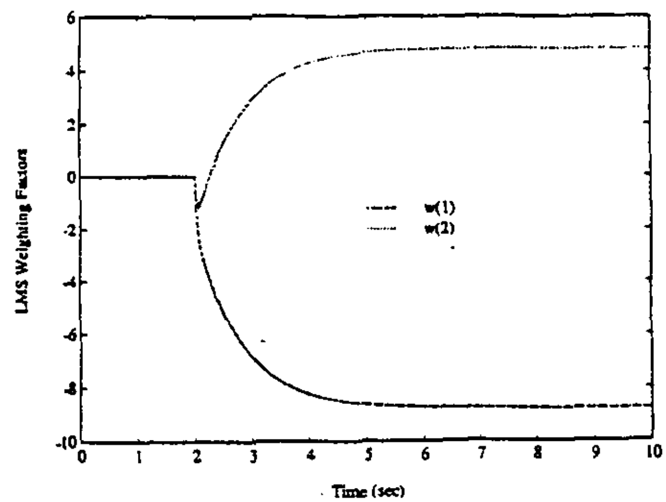

Figure 11. FIR filter coefficients (30 Hz case).

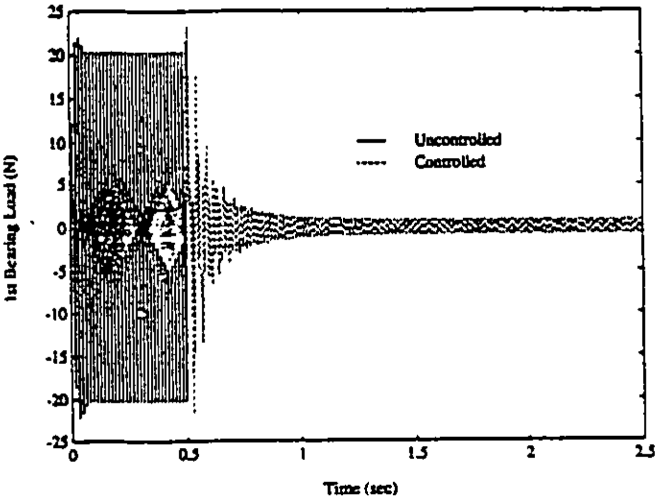

Figure 12. 1st Bearing load from FEM (65 Hz case).

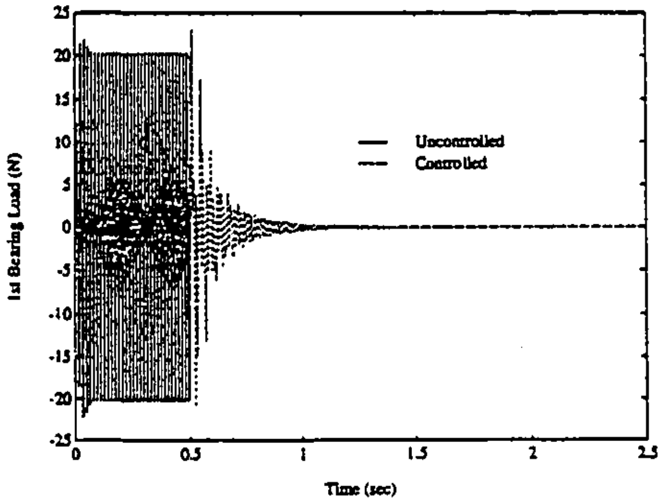

Figure 13. 1st Bearing load from DCM (65 Hz case).

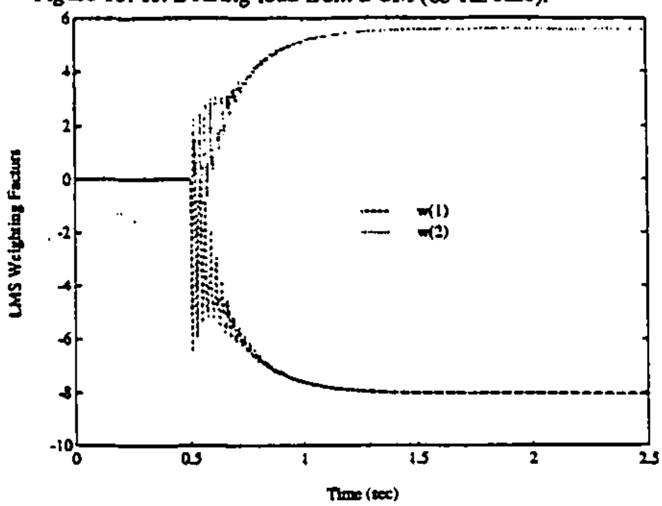

Figure 14. FIR filter coefficients (65 Hz case).

parameters of 0.1, 0.03, and 0.05 were used for the 10, 30, and 65 Hz cases, respectively).

The 10 Hz case shows almost complete suppression of the bearing loads, while the 30 and 65 Hz cases show 93% and 96% reduction, respectively. In comparing Figs. 9 and 10, and Figs. 12 and 13, it can be seen that even though the error approaches zero as the filter weights converge, the true bearing load does not. The reason for this stems from the fact that the deflection-coefficient method provides an approximation to the load based on a first-mode fit of the deflections. While this is very accurate for the disturbance response, the control response sees more of a third mode contribution. The generalized forces for the disturbance input, normalized to the first-mode value, are 1.0, .06, and .03 for first, second, and third modes, respectively. Those for the control input are 1.0, .03, and 0.4, showing an order of magnitude increase for mode three. When the control is applied, the deflection-coefficient method mistakes some of third mode as a first-mode contribution to bearing load, and therefore differs slightly from the true bearing load. (This is not a control spillover phenomenon whereby the higher modes are excited through feedback of a truncated set. The control force is an independent forcing function on the system and is not generated by feedback. The modal contributions to the true error are only dependent on forcing frequency and location.) By accounting for the higher mode contributions to the error signal, improvements can be made in the controller's performance.

CONCLUSIONS

This paper has presented a technique, the Displacement Coefficient Method, for estimating dynamic loads in support bearings of a rotating machinery system. The technique requires very little information about the dynamics of the system, only basic characteristics of the shaft, such as the mass (ρ) and stiffness (EI) properties and knowledge of the fundamental mode, and it is easily implemented, requiring only displacement measurements which are readily available in rotating machinery. The technique was implemented on two flexible beam systems with differing boundary conditions to demonstrate the accuracy with which it can be used to estimate reaction forces. The method was then used as a means of obtaining an error signal for use in an adaptive feedforward control system to minimize dynamic bearing loads. That active control technique, combining feedforward disturbance rejection with the deflection-coefficient method for determining bearing loads, was shown to be very effective in minimizing bearing loads which are first-mode dominated. The control technique is relatively free from requirements of a system model, is easily implemented in an actual rotating machinery system since it uses commonly-used displacement measurement equipment, and it is adaptable to changes in disturbance magnitude or location or changes in the system characteristics. Even though a significant reduction (greater than 93%) in bearing load was shown in the examples given, more studies need to be done to determine the importance of first-mode dominance in the response.

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