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EXPERIMENTAL IDENTIFICATION OF LINEARIZED OIL FILM COEFFICIENTS OF CYLINDRICAL AND TILTING PAD BEARINGS

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ABSTRACT

The dynamic behavior of the rotating machinery supported by the hydrodynamic journal bearings is significantly influenced by the dynamic characteristics of the oil-film. In the present work an efficient identification method is used to identify the stiffness and damping coefficients of the tilting pad and cylindrical journal bearings of flexible rotor-bearing system. The method uses FRFs (Frequency Response Functions) obtained by the measurements and the finite element method. The accuracy and feasibility of the method were tested and demonstrated by theoretical simulation. The possible effects of oil-film inertia is also verified by the theoretical simulation. The method can be further extended to identify twelve linearized oil-film coefficients.

NOMENCLATURE

Cp	machined radial clearance of the tilting pad
Cr	radial clearance of the cylindrical bearing
W	steady state load on the test bearing
b	width of the bearing
c	bearing damping coefficient
d	diameter of the bearing
k	bearing stiffness coefficient
Ω	rotative speed
ω	perturbation frequency

Subscripts:

xx	horizontal direction
yy	vertical direction

INTRODUCTION

The dynamic characteristics of modern turbomachines, generally operating at beyond few critical speeds, have to be predicted accurately to avoid the possibility of operating the machine at near the critical speeds or the unstable speed range and further to identify the sources of vibration of the machine during operation. The finite element modeling of dynamics of rotating machinery is more accurate and reliable compared to transfer matrix method (Firoozian and Stanway, 1989). If the machine is supported by fluid-film bearings, the dynamic behavior is significantly influenced by the stiffness and damping characteristics of the oil-film of the bearing but the exact values of stiffness and damping coefficients are not known. The stiffness and damping characteristics are greatly dependent on many physical and mechanical parameters such as temperature, load, speed, misalignment etc., of the system. Hence, to simulate the dynamic characteristics of the rotor-bearing system, the stiffness and damping characteristics of the fluid film bearings have to be identified accurately.

Several different time-domain and frequency-domain techniques have been developed for determining the oil-film bearing coefficients. Burrows and Sahinkaya (1982), assessed the relative merits of each method and showed that frequency domain techniques were less susceptible to noise. Many works have dealt with identification of bearing coefficients and rotor-bearing system parameters using impulse, step change in force, synchronous and non-synchronous excitation techniques. Zang et al (1992), and Rouvas et al (1992) have used impact excitation to identify the bearing coefficients. Impulse testing may lead to under estimation of input forces when applied to a rotating shaft as a result of the generation of friction-related tangential force components (Muszynska et al, 1992) and, further, is prone to poor signal to noise ratios because of the high crest factor. Also the force input phase is important as it may lead to over/under estimation of ratio of displacement to force. Different methods have been developed (Stanway, 1984, Sahinkaya and Burrows, 1984, Hong and Lee, 1992) to identify the bearing coefficients from the unbalance response of the system. A simulation study carried out by Hong and Lee (1992) shows that in the regions except near the vicinities of the critical speeds the errors of identified values of bearing coefficients are significant due to poor signal to noise ratio. Muszynska et al (1989a, 1989b, 1992) have applied sweep frequency rotating force perturbation method to rotating systems for dynamic stiffness identification and showed that excellent signal to noise ratio data can be obtained for identification of the rotor-bearing system parameters. In the present work unidirectional sweep frequency excitation using

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electromagnetic exciter is applied to obtain the Frequency Response Functions (FRFs).

The method of structural joint parameter identification proposed by Wang and Liou (1991) has been extended to identify the eight linearized oil-film coefficients utilizing the experimental FRFs and the theoretical FRFs obtained by finite element modeling. This method has been used to identify the linearized oil-film coefficients of tilting pad and cylindrical journal bearings.

THEORETICAL FORMULATION

The equation of motion of a rotor-bearing system can be written as

$$[M]\{\ddot{\alpha}\} + [C + C_b]\{\dot{\alpha}\} + [K + K_b]\{\alpha\} = \{F\} \tag{1}$$

where [M], [C] and [K] are mass, damping and stiffness matrices of the rotor, respectively, $[C_b]$ and $[K_b]$ are damping and stiffness matrices, respectively,of the bearings for which the dynamic coefficients have to be identified, $\{F\}$ denotes external force vector and $\{\alpha\}$ represents displacement vector. The dynamic stiffness matrix of the rotor-bearing system can then be expressed as

$$[D_t] = -\omega^2 [M] + i\omega [C + C_b] + [K + K_b]$$

$$= [H_t]^{-1} \tag{2}$$

where $[H_t]$ represents the matrix of FRFs of the rotor-bearing system including the bearings for which the dynamic coefficients have to identified. $[H_t]$ should be determined experimentally. The dynamic stiffness matrix of rotor-bearing system excluding the bearings for which the dynamic coefficients have to be identified can be expressed as

$$[D_r] = -\omega^2 [M] + i\omega [C] + [K]$$

$$= [H_r]^{-1} \tag{3}$$

where $[H_r]$ is the FRFs of the rotor excluding the bearings for which the dynamic coefficients have to be identified. $[H_r]$ can be obtained using finite element technique without considering the bearings and taking free support conditions in place of them.

Using equations (2) and (3) one can obtain the following equation

$$[D_t] - [D_r] = [H_t]^{-1} - [H_r]^{-1} = [K_b] + i\omega[C_b] \tag{4}$$

Hence, if the FRF matrices $[H_t]$ and $[H_r]$ are known, then the bearing parameters can be obtained from equation (4). Note that if the values of $[H_t]$ and $[H_r]$ are exact, then theoretically one can obtain the exact dynamic coefficients of the bearings since there is no approximation in the formulation of the equation (4). To overcome the problem of measurement noise, it is necessary to consider the possible existence of measurement noise and to avoid the inverse operation of the FRF matrix, the following method (Wang and Liou, 1991) is applied. From equation (4) one can obtain the following,

$$[H_t]\left([H_t]^{-1} - [H_r]^{-1}\right)[H_r] = [H_t]\left([K_b] + i\omega[C_b]\right)[H_r]$$

or

$$[H_r] - [H_t] = [H_t]\left([K_b] + i\omega[C_b]\right)[H_r] \tag{5}$$

Let $[H] = [H_r] - [H_t]$, r_{ij} and b_{ij} be the coefficients of the FRF matrices $[H_r]$ and $[H_t]$ and c_{ij} and k_{ij} be the coefficients of the bearing matrices $[K_b]$ and $[C_b]$, respectively, where i,j = 1,2,...n., then, $h_{ij} = r_{ij} - b_{ij}$ and $d_{ij} = k_{ij} + i\omega c_{ij}$. With these notations the equation (5) can then be written as

$$\begin{bmatrix} h_{11}\ h_{12} \dots h_{1n} \\ h_{21}\ h_{22} \dots h_{2n} \\ \cdot\ \ \cdot\ \ \cdot\ \ \cdot \\ \cdot\ \ \cdot\ \ \cdot\ \ \cdot \\ h_{n1}\ h_{n2} \dots h_{nn} \end{bmatrix} = \begin{bmatrix} \sum_{j=1}^{n}\sum_{i=1}^{n} b_{1i}r_{j1}d_{ij} & \sum_{j=1}^{n}\sum_{i=1}^{n} b_{2i}r_{j1}d_{ij} & \dots & \sum_{j=1}^{n}\sum_{i=1}^{n} b_{ni}r_{j1}d_{ij} \\ \sum_{j=1}^{n}\sum_{i=1}^{n} b_{1i}r_{j2}d_{ij} & \sum_{j=1}^{n}\sum_{i=1}^{n} b_{2i}r_{j2}d_{ij} & \dots & \sum_{j=1}^{n}\sum_{i=1}^{n} b_{ni}r_{j2}d_{ij} \\ \cdot & \cdot & & \cdot \\ \cdot & \cdot & & \cdot \\ \sum_{j=1}^{n}\sum_{i=1}^{n} b_{1i}r_{jn}d_{ij} & \sum_{j=1}^{n}\sum_{i=1}^{n} b_{2i}r_{jn}d_{ij} & \dots & \sum_{j=1}^{n}\sum_{i=1}^{n} b_{ni}r_{jn}d_{ij} \end{bmatrix} \tag{6}$$

From equation (6) one can obtain n x n number of equations. It can be written in matrix form as follows,

$$\{A\}_{n^2 \times 1} = [D]_{n^2 \times n^2} \{E\}_{n^2 \times 1} \tag{7}$$

where A is

$$[h_{11}\ h_{21} \dots h_{n1}\ h_{12}\ h_{22} \dots h_{n2}\ h_{13}\ h_{23} \dots h_{nn}]^T_{n^2 \times 1}$$

D is

$$\begin{bmatrix} b_{11}r_{11} & b_{12}r_{11} \dots & b_{1n}r_{11} & b_{11}r_{21} & b_{12}r_{21} \dots & b_{11}r_{n1} \dots & b_{1n}r_{n1} \\ b_{11}r_{12} & b_{12}r_{12} \dots & b_{1n}r_{12} & b_{11}r_{22} & b_{12}r_{22} \dots & b_{11}r_{n2} \dots & b_{1n}r_{n2} \\ \cdot & \cdot & \cdot & \cdot & \cdot & \cdot & \cdot \\ \cdot & \cdot & \cdot & \cdot & \cdot & \cdot & \cdot \\ b_{11}r_{1n} & b_{12}r_{1n} \dots & b_{1n}r_{1n} & b_{11}r_{2n} & b_{12}r_{2n} \dots & b_{11}r_{nn} \dots & b_{1n}r_{nn} \\ b_{21}r_{11} & b_{22}r_{11} \dots & b_{2n}r_{11} & b_{21}r_{21} & b_{22}r_{21} \dots & b_{21}r_{n1} \dots & b_{2n}r_{n1} \\ b_{21}r_{12} & b_{22}r_{12} \dots & b_{2n}r_{12} & b_{21}r_{22} & b_{22}r_{22} \dots & b_{21}r_{n2} \dots & b_{2n}r_{n2} \\ \cdot & \cdot & \cdot & \cdot & \cdot & \cdot & \cdot \\ \cdot & \cdot & \cdot & \cdot & \cdot & \cdot & \cdot \\ b_{21}r_{1n} & b_{22}r_{1n} \dots & b_{2n}r_{1n} & b_{21}r_{2n} & b_{22}r_{2n} \dots & b_{21}r_{nn} \dots & b_{2n}r_{nn} \\ b_{31}r_{11} & b_{32}r_{11} \dots & b_{3n}r_{11} & b_{31}r_{21} & b_{32}r_{21} \dots & b_{31}r_{n1} \dots & b_{3n}r_{n1} \\ b_{31}r_{12} & b_{32}r_{12} \dots & b_{3n}r_{12} & b_{31}r_{22} & b_{32}r_{22} \dots & b_{31}r_{n2} \dots & b_{3n}r_{n2} \\ \cdot & \cdot & \cdot & \cdot & \cdot & \cdot & \cdot \\ \cdot & \cdot & \cdot & \cdot & \cdot & \cdot & \cdot \\ b_{n1}r_{1n} & b_{n2}r_{1n} \dots & b_{nn}r_{1n} & b_{n1}r_{2n} & b_{n2}r_{2n} \dots & b_{n1}r_{nn} \dots & b_{nn}r_{nn} \end{bmatrix}_{n^2 \times n^2}$$

and E is

$$[d_{11}\ d_{21} \dots d_{n1}\ d_{12}\ d_{22} \dots d_{1n} \dots d_{nn}]^T_{n^2 \times 1}$$

with $d_{ij} = k_{ij} + i\omega c_{ij}$.

Note that the above equation is formed directly from the elements of equation (6). The equation (7) contains the n x n parameters as unknowns. Hence, theoretically, this equation can be solved to get exact values of linearized dynamic coefficients provided the FRFs are exact. The bearings and seals are usually modeled as four stiffness and four damping coefficients. Hence, in actual practical cases the number of unknowns would be less, one can use those equations which has the transfer function between same points as coefficients. The real and imaginary part of the unknown parameter, $k + i\omega c$, can be separated and then the equation (8) can be written as follows.

$$\{A\}_{n^2\times 1} = [[D] + i\omega[D]]_{n^2\times 2n^2}\begin{Bmatrix}\{K\}\\ \{C\}\end{Bmatrix}_{2n^2\times 1} \quad (8)$$

The equation (8) has $2n^2$ unknowns, but has only one set of n^2 simultaneous equations. However, the vector {A} and matrix [D] are functions of frequency ω and if the FRFs are known at m number of discrete frequencies, $\omega_1, \omega_2, \ldots \omega_m$, then for each frequency one can have a set of n^2 simultaneous equations. To smooth the random measurement error the FRFs at more than two frequency points can be taken and least squares method can be used to obtain the unknown dynamic parameters as:

$$\begin{Bmatrix}\{K\}\\ \{C\}\end{Bmatrix} = \left(\begin{bmatrix}[D]\omega_1\\ [D]\omega_2\\ \cdot\\ [D]\omega_m\end{bmatrix}^T \begin{bmatrix}[D]\omega_1\\ [D]\omega_2\\ \cdot\\ [D]\omega_m\end{bmatrix}\right)^{-1} \begin{bmatrix}[D]\omega_1\\ [D]\omega_2\\ \cdot\\ [D]\omega_m\end{bmatrix}^T \begin{Bmatrix}\{A\}_{\omega_1}\\ \{A\}_{\omega_2}\\ \cdot\\ \{A\}_{\omega_m}\end{Bmatrix} \quad (9)$$

The above equation can be used to find not only the bearing parameters but also the full stiffness and damping matrices of the system provided the complete FRFs of the system are known.

THEORETICAL SIMULATION

The identification algorithm presented here can identify the bearing coefficients exactly, if the FRFs are free from noise. However, measurement noise is unavoidable in practice. Further, the inertia coefficients of oil-film, which are not being identifeid in the present method, may affect the accurcy of the identified coefficients. Hence the simulation studies have been carried out to know the effect of measurment noise and the inertia coefficients of the oil-film.

Effect of Measurement Noise

In this simulation study, a random noise with normal distribution (zero mean, variance σ^2) was added to the FRFs to simulate the measurement noise. If the $H_{ij}(\omega)$ denotes the frequency response function between the i^{th} and the j^{th} points of the rotor bearing system, then the variance of the random noise σ_r^2 and σ_i^2, for real and imaginary parts of the FRFs, are defined as:

$$\sigma_r^2 = e_n^2(\mathrm{Re}H_{ij}(\omega)^2_{max}) \text{ and } \sigma_i^2 = e_n^2(\mathrm{Im}H_{ij}(\omega)^2_{max})$$

where subscript max denotes the maximum value in the frequency range of interest and e_n represents the noise level. If the noise level is given then a set of random number can be generated by a computer program.

The simulated system is a rotor supported on two bearings. The length of the shaft is 1260 mm and the diameter is 50 mm. A disk of 200 mm diameter and 420 mm long is centrally mounted on the shaft. The internal viscous loss factor and hysteretic loss factor are assumed to have the values $1.0\ 10^{-4}$s and 0.0, respectively. The stiffness and damping coefficients of the left end bearing, assumed to be isotropic, are $1.75\ 10^7$ N/m and $3.0\ 10^5$ Ns/m, respectively. The characteristics of the right end bearing are identified using the proposed algorithm. The detailed prcedures of the simulation are summarized here:

Find the equation of motion of the rotor-bearing system without the bearing for which the dynamic coefficients are to be identified using finite element technique, i.e., considering free end conditon at the right end bearing location.

Find the equation of motion of the rotor-bearing system including the right end bearing.

Compute the necessary FRF matrices from the equation of motion.

Determine the frequency range of interest, ω_{min} and ω_{max}, and the frequency resolution $\Delta\omega$.

Determine the number N and the the noise level e_n, then generate one set of N discrete random values with normal distribution to add to the real part of the discrete FRFs and another set of N random values to add to the imaginary part of the discrete FRFs.

Identify the bearing coefficients from the noisy FRFs using equation (9).

In the present example the number N was set to be 400 and the frequency resolution is 0.125 Hz. The noise level of 3 persent is considered. The results are shown in Table 1. The identified values of coefficients are more or less accurete, the error is less than 8 percent. However, the error of the damping coefficients may increase if the values are very small compared to the values of the stiffness coefficients. This is because of the characteristics of the least square method. In general, the smaller parameter may have larger percentage error by using the least square method. (Wang and Liou, 1991).

TABLE 1 EFFECT OF MEASUREMENTS NOISE

Coefficients	Exact	Identified	Error (%)
Kxx (N/m)	$1.5\ 10^5$	$1.3815\ 10^5$	7.9
Kyy (N/m)	$1.5\ 10^5$	$1.4775\ 10^5$	1.5
Cxx (Ns/m)	$5.0\ 10^3$	$4.7763\ 10^3$	4.5
Cyy (Ns/m)	$5.0\ 10^3$	$5.1973\ 10^3$	-3.9

One can improve the accuracy of the smaller parameters by adding the identified values of the larger parameter into the equation of motion and repeating the identification algorithm to identify the smaller parameters alone.

Effect of Inertia Coefficients

The hypothetical rotor-bearing system taken in the previous simulation study is considered in this simulation study also. 10 % (11.5 kg) of the rotor mass is taken as the value of the inertia coefficients of the oil-film. The simuation procedure is as follows:

Find the equation of motion of the rotor-bearing system without the bearing for which the dynamic coefficients are to be identified using finite element technique, i.e., considering free end conditon at the right end bearing location.

Find the equation of motion of the rotor-bearing system including the right end bearing (with the stiffness, damping and inertia coefficients).

Compute the necessary FRF matrices from the equation of motion.

Determine the frequency range of interest, ω_{min}, and ω_{max}, and the frequency resolution $\Delta\omega$ and then determine the number N

Identify the bearing coefficients from the FRFs using equation (9).

The results are shown in the Table 2. From the result it is clear that the proposed method is not affected by the inertia coefficients even if their values are significant.

TABLE 2 EFFECT OF INERTIA COEFFICIENTS

Coefficients	Exact	Identified	Error (%)
Kxx (N/m)	$1.5\ 10^6$	$1.5018\ 10^6$	.0012
Kyy (N/m)	$1.5\ 10^6$	$1.5018\ 10^6$	.0012
Cxx (Ns/m)	$5.0\ 10^4$	$5.0030\ 10^4$	.0006
Cyy (Ns/m)	$5.0\ 10^4$	$5.0030\ 10^4$	.0006

EXPERIMENTAL SYSTEM

Test-Rig

The proposed method is used to identify the dynamic coefficients of bearings in a rotor-bearing test rig. The test rig (Fig. 1) is a single mass flexible rotor supported by a self aligning ball bearing at one end and by the test bearing (hydrodynamic tilting pad/cylindrical bearing) at the other end. The rotor is coupled to the motor through an electromagnetic coupling. Three different bearings have been tested. A brief description to the parameters of the test rig is as follows:

Rotor parameters:
Shaft length between bearing centers 825 mm;
Shaft diameter 30 mm; Rotor mass 24.5 kg;
Bearing parameters of Test bearings: Bearing diameter 25 mm;
i Tilting pad bearing; 4 pads; b/d = 0.3; C_P = 54 μm
ii Tilting pad bearing; 4 pads; b/d = 0.4; C_P = 42 μm
iii 360° cylindrical bearing; b/d = 1; C_r = 65 μm
where C_P is the machined radial clearance of the tilting pad bearing and C_r is the radial clearance of the cylindrical beaimg.

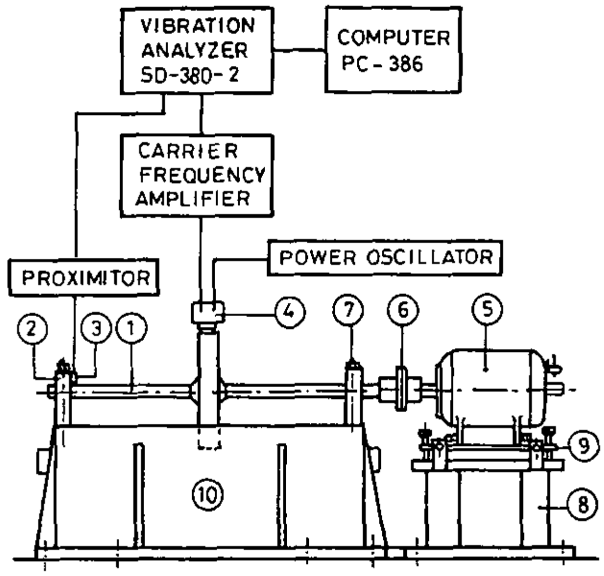
Instrumentation

Exicter and Transducers. A Noncontact electromagnetic exciter (Designed at University of Kassel) is used for excitation. The exciter has builtin calibrated force transduser for measuring the excitation force applied to the rotor. The displacement responses are measured using eddy current proximity probes with proximitor (eddy probe drivers).

Analyzer. The measured input and output signals are analyzed in a two channel signal analyzer. The analyzer has 12 bits A/D converter for each channel, channel phase match of ±1.5° and 800 lines of frequency baseband resolution. The frequency range used in the present analysis is 0 - 100 Hz and the corresponding frequency resolution is 0.125 Hz.

Measurements

The rotor mass is excited by unidirectional sine sweep perturbation force using the electromagnetic exciter and the displacement response is measured at the test bearing (3 mm away from the inside edge of the bearing) to get the Transfer function (FRF). Typical values of forcing level and the associated displacement response



1 Flexible shaft rotor
2 Test bearing
3 Eddy probe
4 Electromagnetic Exciter
5 Motor
6 Electromagnetic coupling
7 Rolling element bearing
8 Motor base
9 Adjustable base plate
10 Main frame cum oil sump

FIG. 1 ROTOR-BEARING TEST RIG (schematic)

level are shown in Fig. 2. In Fig. 2 the SPEC A shows the frequency spectrum of the sine sweep perturbation force applied to the rotor and SPEC B shows the frequency spectrum of the displacement response at the test bearing. The excitation force level varies from 0.2 N to 0.5 N in the frequency range of 10 - 60 Hz. In the displacement response spectrum the marked peak is the resonance peak of 2.09 μm occuring at 31.000 Hz. The other two peaks occuring at the 37.5 Hz (1ω) and 75.0 Hz (2ω, where ω is the rotating frequency of the rotor) are due to unbalance and misalignment force excitations. The amplitude ranges used in the present anlysis are 0.0 - 4.0 N and 0.0 - 4.0 μm for the excitation force and the displacement response, respectively.

The analyzer directly gives the transfer function (FRF) between the excitation force and the displacement response for the chosen locations and the same can be transfered to the computer for further analysis/storage. The FRF curve of a typical case is shown in Fig. 3. In the Fig. 3 numerator of EU/EU (Engineering Unit/Engineering Unit) refers to the displacement unit μm and denominator refers to the force unit N. The displacement to force ratio at the resonance frequency of 31.124 Hz is found to be 4.67 μm/N with the phase angle of -104.0°. From the Fig. 3 it is clear that the FRF is more or less accurate particularly in the frequency range of 20 - 40 Hz. FRFs in this frequency range only are used for the indetification of bearing coefficients. A typical cross coupled FRF i.e., the FRF obtained by applying excitation at the rotor mass in the vertical direction and measuring the displacement response in the horizontal direction of the bearing or vice versa, is shown in Fig. 4 to illustrate the response level and the phase error in the FRF.

For the case of tilting pad bearings the FRFs are measured only in the horizontal direction for excitation in the horizontal and in the vertical direction for excitation in the vertical direction because the

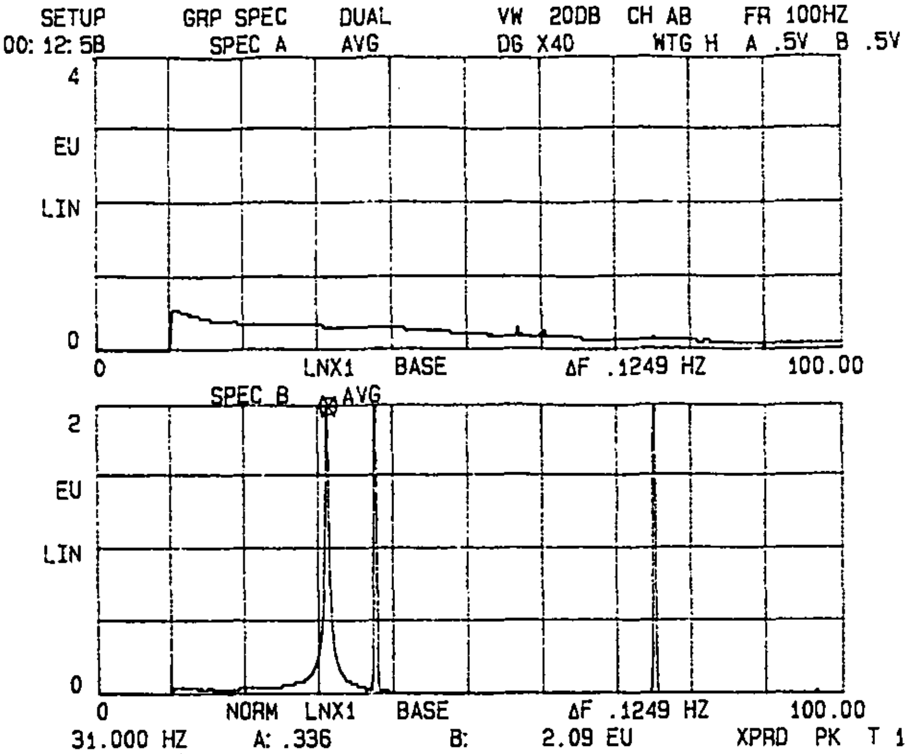


FIG. 2 FREQUENCY SPECTRUM OF THE PERTURBATION FORCE AND THE DISPLACEMENT RESPONSE, TILTING PAD BEARING, b/d = 0.4, LOAD BETWEEN PAD, SPEED = 2250 rev/min.

cross coupled stiffness and damping coefficients are zero (theoretically) for the tilting pad bearings. But in the case of cylindrical bearing the FRFs are measured in both the directions (vertical and horizontal) for excitation in the each direction. Thus two FRFs are measured for tilting pad bearings and four FRFs are measured for cylindrical joumal bearing.

Identification

The identification algorithm presented here needs FRFs of the rotor-bearing system and also the FRFs of the rotor without those bearings for which the coefficients are to be identified, i.e., with free end condition at that bearings locations. The latter FRFs are difficult to obtain experimentaly. Hence finite element technique is used to obtain the equation of motion of the rotor considering free end condition at the location of test bearing in the rotor. The rotor is supported by the self-aligning ball bearing. Since the rolling elelment bearing is new the clearance in the bearing may not be sufficient enough to behave non-linearly. Further, since the stiffness of the ball bearing is higher than the shaft as well as the hydrodynamic bearing, the bearing characteristics do not significantly influence either the first critical speed or the response of the rotor-bearing system. In the present analysis the stiffness of the rolling element bearing is estimated using the approximate realtion given by Gargiulo JR., E. P., (1980), and the same is added in finite element model of the rotor at the appropriate nodal degrees of freedom. The bearing dynamic coefficients are identified using equation (9)

RESULTS AND DISCUSSIONS

The stiffness and damping coefficients are non-dimentioanlized as given below:

$$K_{ij} = (C' k_{ij}) / W; \quad C_{ij} = (C' \Omega c_{ij}) / W \quad \text{with } (i,j = x,y)$$

where W =124.5 N is the steady state load on the bearing and $C' = C_r$ for cylidrical bearing and $C' = C_p$ for tilting pad bearings.

The non-dimensional coefficients of stiffness and damping versus the speed of the rotor are shown in Figs. (5-8). The estimates , with the exception of C_{yy} of tilting pad bearings of b/d = 0.4 (Figs. 5&6), exhibits less scattering. The scattering of the estimate is because not only the speed of the rotor is changing but also the viscosity of the lubricating oil. The scattering of the results may also be due to the excitation of both the forward and reverse modes simultaneously since the applied perturbation does not discriminate on direction of the applied force. The results of load between configuration of tilting pad bearings show that the bearing is not performing like isotropic bearing. This is due to variations of pad clearance from pad to pad both in the assembled and machined clearances and the misalignment in the bearing. Further the bearing pedestals are not isotropic. The anisotropic behaviour of the tilting pad bearings with load between pad configuration is also reported in the works of Someya (1989). Figure (8) shows stiffness and damping coefficients of cylindrical bearings. In this case cross coupled damping coefficients are not shown since they are very small.

The effectiveness of the identification algorithm is tested by demonstrating the ability of the identified model to predict the first natural frequency (in bending) and FRFs of the rotor-bearing system. Figure (9) shows the comparison of FRFs predicted by finite elment method using the identified model and the FRFs acually measured on the rotor-bearing system. Figures (10&11) show the comparison of natural frequency(in bending) obtained experimentally and predicted by finite element method using the identified stiffness and damping coefficients of bearings. The comparisons are satisfactory.

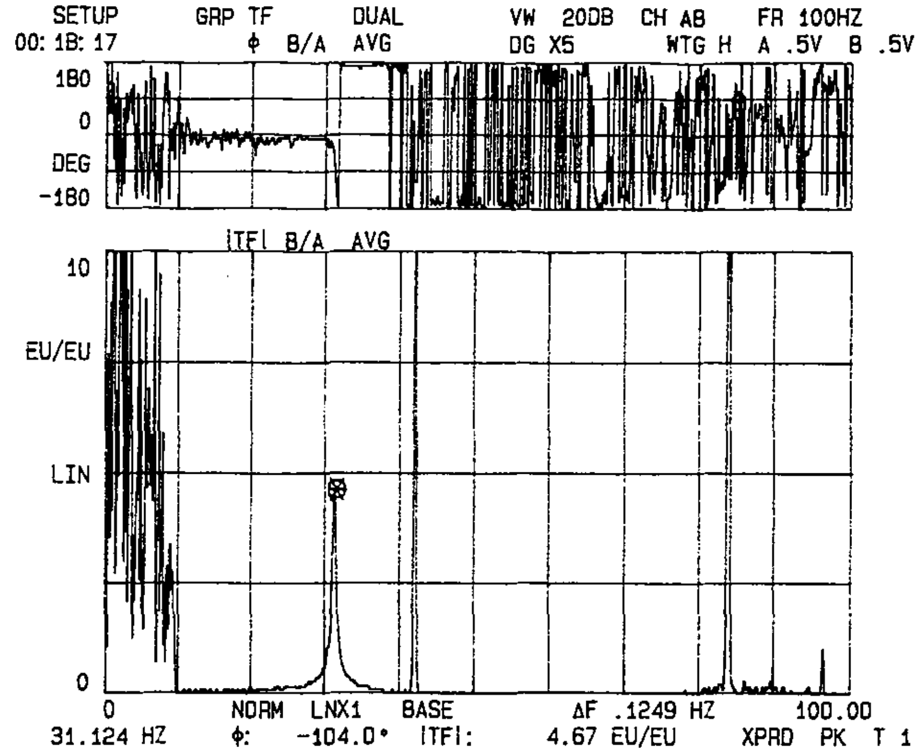


FIG. 3 THE MEASURED FRFs OF THE ROTOR-BEARING SYSTEM, TILTING PAD BEARING, b/d = 0.4, LOAD DN PAD, SPEED =2500 rev/min.

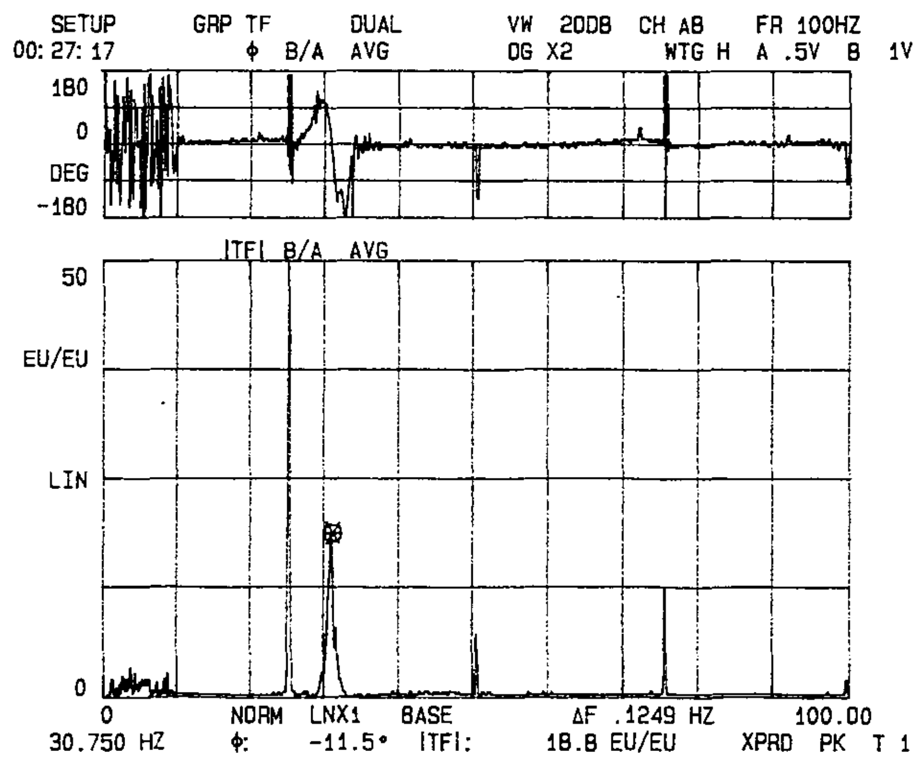


FIG. 4 THE MEASURED CROSS COUPLED FRFs OF THE ROTOR-BEARING SYSTEM, CYLINDRICAL JOURNAL BEARINGS, b/d = 1.0, SPEED = 1500 rev/min.

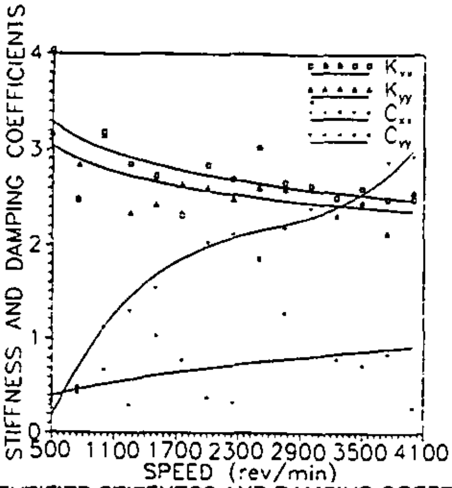


FIG.(5) IDENTIFIED STIFFNESS AND DAMPING COEFFICIENTS OF TILTING PAD BEARING, LOAD BETWEEN PAD CONFIGURATION (NO.OF PADS 4, b/d = 0.4, C_p= 42.0 μm)

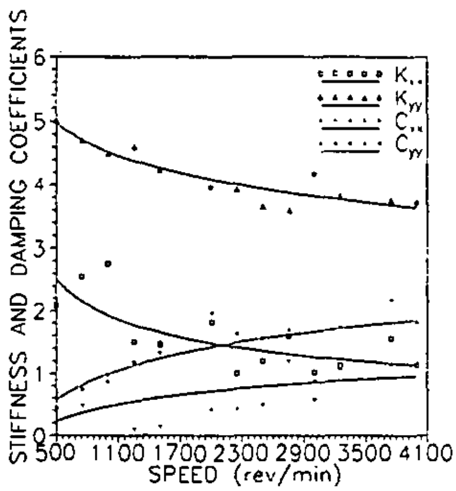


FIG.(6) IDENTIFIED STIFFNESS AND DAMPING COEFFICIENTS OF TILTING PAD BEARING, LOAD ON PAD CONFIGURATION (NO.OF PADS 4, b/d = 0.4, C_p= 42.0 μm)

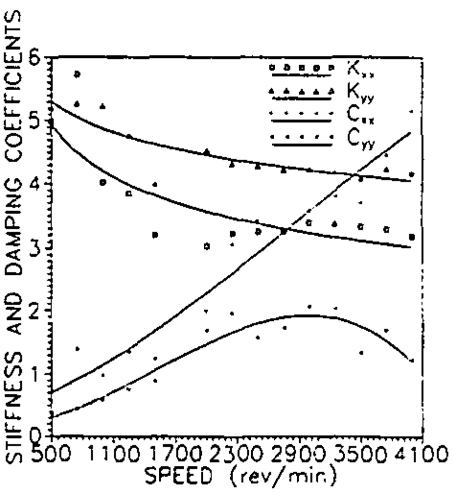


FIG.(7) IDENTIFIED STIFFNESS ANO OAMPING COEFFICIENTS OF TILTING PAD BEARING, LOAD BETWEEN PAD CONFIGURATION (NO. OF PAOS 4, b/d = 0.3, C_p= 54.0 μm)

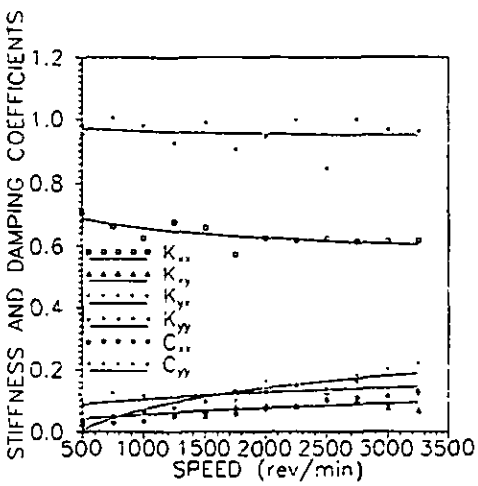


FIG.(8) IDENTIFIED STIFFNESS AND DAMPING COEFFICIENTS OF CYLINDRICAL JOURNAL BEARING (b/d = 1.0, C_r= 65.0 μm)

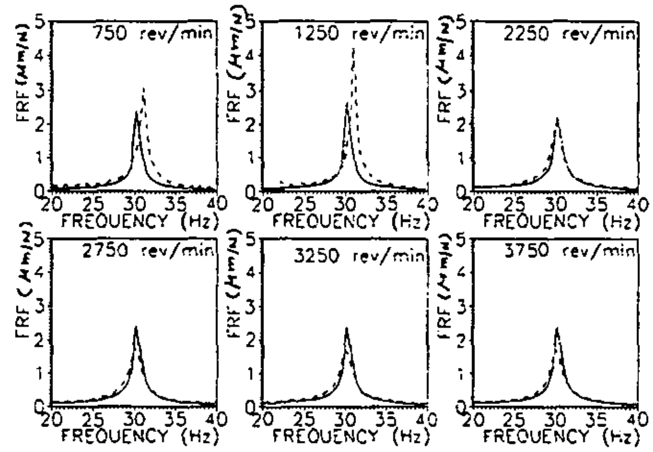


FIG. 9 COMPARISON OF FRFs MEASURED AND PREDICTED BY FINITE ELEMENT ANALYSIS USING IDENTIFIED BEARING COEFFICIENTS; TILTING PAO BEARING (b/d = 0.3, LOAD BETWEEN PAD), —— PREDICTED, ----- MEASURED

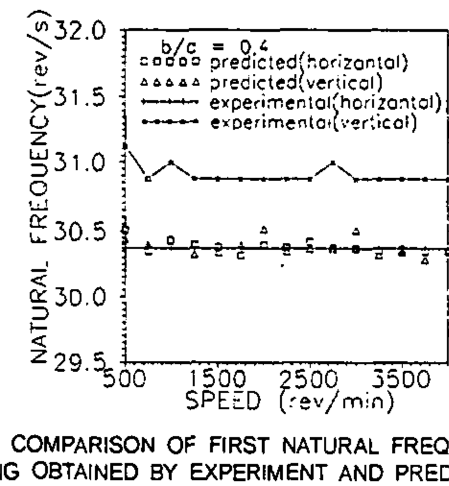


FIG. 10 COMPARISON OF FIRST NATURAL FREQUENCY IN BENDING OBTAINED BY EXPERIMENT AND PREDICTED BY FINITE ELEMENT ANALYSIS USING IDENTIFIED BEARING COEFFICIENTS, TILTING PAO BEARINGS (LOAD BETWEEN PAO)

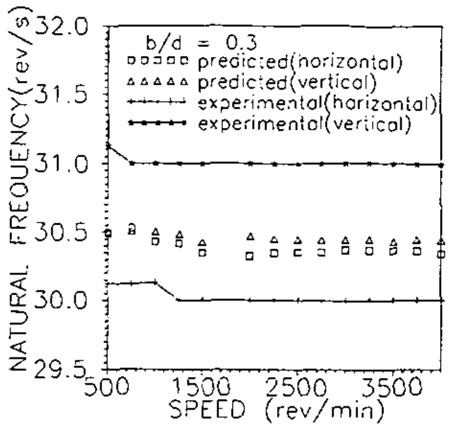


FIG. 11 COMPARISON OF FIRST NATURAL FREQUENCY IN BENDING OBTAINED BY EXPERIMENT AND PREDICTED BY FINITE ELEMENT ANALYSIS USING IDENTIFIED BEARING COEFFICIENTS, TILTING PAD BEARING (LOAD BETWEEN PAD)

CONCLUSIONS

In the present work an extended method has been utilized for the identification of linearized oil-film coefficients of cylindrical and tilting pad journal bearings. This method can also be used for the identification of all the stiffness and damping coefficient of the system if the complete frequency response function of the system is known.

The effect of inertia coefficients of the oil-film on the identification method is practically nil. This is demonstrated in the simulation study.

The method presented here can be extented find to all the twelve coefficients of the bearing.

Linearized oil-film coefficients of flexible rotor-bearing system are estimated experimentally. Comparison of FRFs predicted using finite element method with the measured FRFs shows that the identified oil-film coefficients are fairly accurate.

The number of frequency response functions needed for the estimation are only two per bearing for tilting pad bearings and four per bearing for cylindrical bearing.

This method can be easily applied to field cases since it is easy to implement for on-line identification and only a few measurements are needed at each speed.

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