Experimental Identification of Bearing Stiffness in a Rotor Bearing System

Sharad Shekhar Palariya, M. Rajasekhar and J. Srinivas

Department of Mechanical Engineering, National Institute of Technology Rourkela, Rourkela
E-mail: mr_sekhar21@yahoo.co.in

ABSTRACT

This paper presents the dynamic analysis of a dual-disk rotor supported over a double-row ball bearing system. Finite element model of the rotor is employed to know the approximate range of first few bending frequencies and the corresponding resonance amplitudes as a function of bearing stiffness coefficient. A non-parametric model based on back-propagation neural network is developed in order to assess the bearing stiffness for known values of frequency shifts and resonance amplitudes with respect to rigid bearing support conditions. A scaled prototype of the rotor dynamic test rig with the disks is fabricated and experimental modal analysis is conducted using electrodynamic excitation technique. Frequency modulated sine-sweep tests are conducted in two different conditions: first by laying it over the ball-bearing system whose stiffness has to be ascertained and then by mounting it over rigid supports. The non-dimensional frequency shifts and amplitude ratios from the experimentally obtained frequency response curves are supplied as inputs to previously formulated neural network estimator model and the corresponding stiffness of the bearing is reported.

Keywords: Dual-disk rotor, Finite element modeling, Neural network estimator, Bearing stiffness coefficient

INTRODUCTION

Rotating machinery is commonly used in many mechanical systems, including electrical motors, machine tools, compressors, turbo machines and aircraft gas turbine engines. Typically these systems are affected by exogenous or endogenous vibrations produced by unbalance, misalignment, resonances, material imperfections and cracks. Vibration caused by mass unbalance is a common problem in rotating machinery. Rotor unbalance occurs when the principal inertia axis of the rotor does not coincide with its geometrical axis and leads to synchronous vibrations and significant undesirable forces transmitted to the mechanical elements and supports. Although the controlled imbalance response measurements commonly performed during shop testing, provide information about the rotating system critical speed and the amplification factors, it is often desirable to confirm the support rotor dynamic properties in the test stand and in the field, giving rise to the need for appropriate identification methods. Such methods prove useful for the verification of bearing property-prediction as well as for eventual condition monitoring and troubleshooting. Experimental identification of bearing impedances requires measurements of the rotor response and force excitation. Bearing impedances are complex functions, with real parts representing stiffness and imaginary parts being proportional to the viscous damping coefficients. Some of the experimental techniques are available for identification of bearing impedances and are classified in terms of the type of measuring equipment required, time available to carry out the testing and the reliability of the results. Therefore, several works aimed at experimental analysis of rotor dynamic test rigs to predict the internal nonlinear dynamics due to various components.

Chen *et al*.[1] modeled the rubbing fault, by considering the stator motion, the flexible support and squeeze film damper and the nonlinear factors of ball bearing, such as the clearance of the bearing, the nonlinear Hertzian contact force between balls and races and the varying compliance vibration because of the periodical varying contact position between balls and races, Here experimental work was conducted on an aero engine rotor with stator using eddy current probes and data acquisition system. Cheng *et al*.[2] investigated the non-linear dynamic behaviors of a rotor–bearing–seal coupled system and the influence of parameters, such as the rotation speed, seal clearance and eccentricity of rotor are analyzed by state trajectory, Poincare maps, frequency spectra and bifurcation diagrams here experimental work is conducted on a test rotor supported on two oil lubricated bearings. Patel and Darpe [3] aimed to examine vibration response of the cracked rotor in presence of common rotor faults such as unbalance and rotor stator rub. Numerical and experimental investigations are carried out and steady-state vibration analysis is presented. Bai *et al*.[4] investigated the nonlinear dynamic behavior of a flexible rotor supported by ball bearings using experiments and the numerical approach. Wenhui *et al*. [5] illustrated the analysis of dual-disk flexible rotor supported over oil-film bearings using a simple experimental analysis. Han *et al*. [6] showed experimentally the periodic motions occurring in a rotor with two disks, where the rub-impact occurs at fixed limiter. Recently, importance of experimental analysis work in rotor dynamics area is tremendously increasing. Even the experiments are standard; some novelty has been illustrated in each of the experiments. Hariharan and Srinivasan [7] presented an experimental analysis on a rotor shaft with flexible couplings to predict the effect of rotor unbalance. Dickmen *et al*. [8] also recently described an experimental set-up for testing multiphysics effects on high speed mini-rotors. Determination of bearing stiffness properties from vibration response is of vital importance in rotor dynamics. Marsh and Yantek [9] illustrated a method of measurement for dynamic bearing stiffness value from experimental data. Sadeghi and Zehsaz [10] shown the effect of bearing stiffness on frequency response in a case study of a turbopump system.

Present paper deals with the modeling and analysis of a dual disk rotor shaft supported over double-row ball bearing system. The rotor has two rigid disks placed symmetrically from the supports. Initially, the analysis is conducted using finite element model by discritizing with one-dimensional, Timoshenko beam elements in order to obtain the frequency response curves at different values of bearing stiffness. A neural network model that relates the relative natural frequencies (called frequency shifts) and corresponding amplitude ratios with the bearing stiffness is developed using the finite element simulations. Two sets of experiments are conducted on the prototype system to know the effect of bearing stiffness on dynamic response. Approximate linear stiffness of the bearing is finally predicted from the experimentally measured frequency shifts and amplitude ratios using the neural network estimator model. Organization of the paper is as follows: Section 2 explains the neural network based stiffness estimation model generated from the basic finite element modeling of the rotor. Section-3 describes the experimental testing procedure for obtaining the frequency response on the prototype and prediction of approximate bearing stiffness from experimental frequency response.

DYNAMIC ANALYSIS OF ROTOR

The rotor having two disks considered in the present study is shown in Fig.1. Diameter of the shaft is 20 mm and two disks are identical having diameters of 124 mm each. Both the disk and shaft are made up of alloy steel having following material properties: Elastic modulus E=207e11 GPa, density r=7840 kg/m^3, Poisson ratio =0.3 and an approximated shaft damping of 0.1% is also considered.

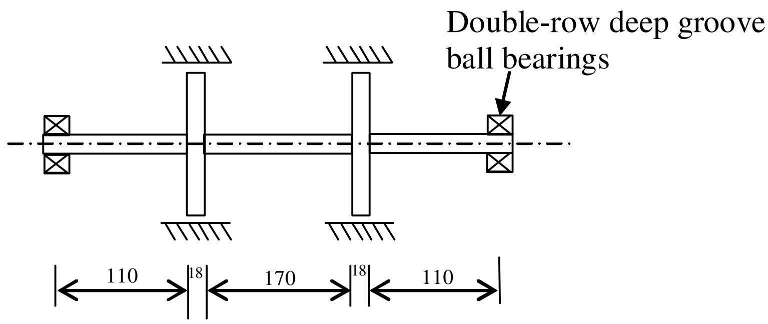


Fig. 1. Rotor supported over ball bearings (total supporting length = 426 mm)

The flexible shaft is discritized into six, two-noded Timoshenko beam elements. The assembled matrices are formulated from nodal connectivity data and the matrices are condensed to eliminate rotational degrees of freedom. Computer program is developed in MATLAB for the basic formulation and to solve the eigenvalue program as well to obtain the time histories at the various nodes. The disk masses are lumped at appropriate nodes in the mass matrix and stiffness value of the support bearings is included as the boundary conditions. Here the bearing is considered to have an equal linear stiffness in both y and z directions with no cross-coupled terms. Table 1 shows the effect of support stiffness on the first four bending modes of the rotor using the present model along with a solution obtained from ANSYS software under three different values of bearing stiffness parameter (k_b). During the ANSYS simulations, two-noded beam elements (beam 188) are used along with 3D-mass and 2D-combination elements (two each at one bearing along y and z directions). It is seen that the present six-element model gives close results with ANSYS and would approximate the dynamics of rotor for further analysis. As present program adopts the Timoshenko formulation, the frequencies are observed always on the lower side of ANSYS outputs.

Table 1. Theoretical modal frequencies of the rotor (Hz)

Mode. No.	*k_b=1e15 N/m*		*k_b=1e9 N/m*		*k_b=1e7 N/m*	
	Program	*ANSYS*	*Program*	*ANSYS*	*Program*	*ANSYS*
1(FW-Y)	100.018	101.5	99.97	101.46	96.04	97.36
2(BW-Z)	100.018	101.5	99.97	101.46	96.04	97.36
3(FW-Y)	323.083	346.79	322.8	346.44	297.5	315.52
4(BW-Z)	323.083	346.79	322.8	346.44	297.5	315.52

Fig.2 shows the variation of first two Y-bending modes as a function of bearing stiffness parameter. As seen a decrease in bearing stiffness reduces the natural frequencies considerably in a nonlinear manner.

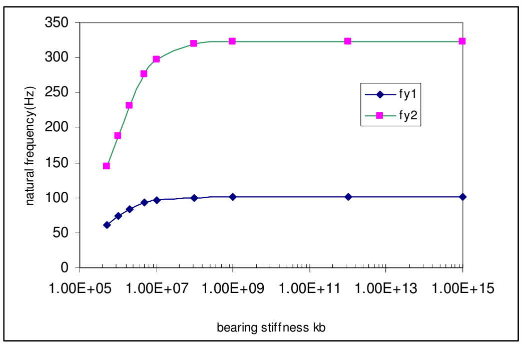


Fig. 2. Effect of bearing stiffness on first two bending modes

The frequency response curves of the system are also obtained at different bearing stiffness values so as to measure the resonance amplitudes. Fig.3 shows the harmonic response of the system in Y-direction at a node in-between bearing and disk.

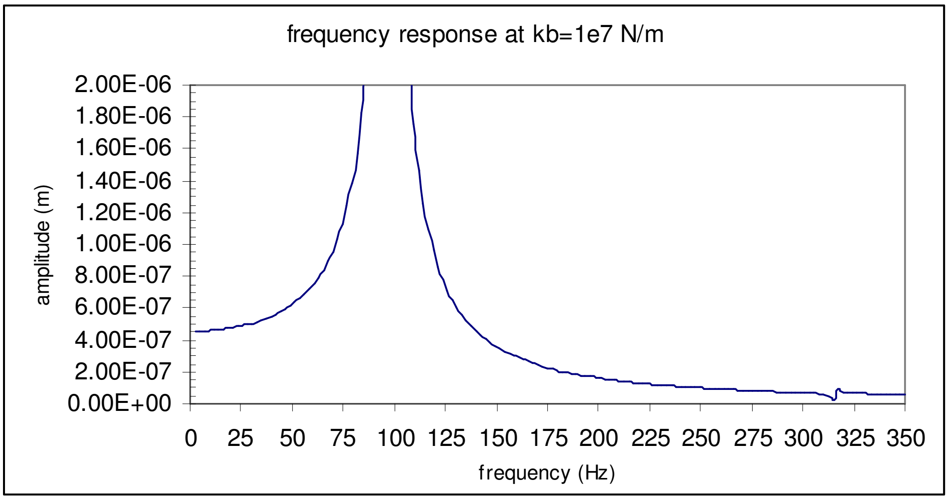


Fig. 3. Harmonic response in Y-direction at node-2

The relationship between the bearing stiffness with resonance frequencies and corresponding modal amplitudes is utilized to get the unknown bearing stiffness from experimentally measured harmonic response. As the first resonance amplitude is much higher than that of second and higher modes, only one amplitude ratio is considered in present analysis. Also, it is considered that the bearing stiffness is less than the shaft stiffness which is of order 1e7 N/m in the present case. The bearing stiffness values are therefore varied from 1e10 N/m to 1e6 N/m in certain steps. In each case, amplitude ratio, the frequency shifts are recorded and is depicted in Table 2.

Table 2. Effect of bearing stiffness on the modes

k_b (N/m)	Δf_1 (Hz)	Δf_2 (Hz)	A_0/A_1
1.00E+08	0.415	2.752	6.67E-01
5.00E+07	0.835	5.49	1.33E+00
2.00E+07	2.045	13.354	2.29E-01
1.00E+07	3.972	25.5	2.67E-01
8.00E+06	4.897	31.15	1.14E+00
7.00E+06	5.545	35.02	2.00E-01
6.00E+06	6.385	39.99	9.70E-01
5.00E+06	7.528	46.58	7.62E-02
4.00E+06	9.175	55.753	4.27E-01
3.00E+06	11.735	69.38	4.10E-01
2.00E+06	16.285	91.71	1.00E+00
1.00E+06	26.675	135.21	1.33E-01

The data is normalized and trained in a back-propagation neural network. A wide variety of feed-forward neural network architectures were used in literature for bearing parameter estimations [11-13]. Back propagation neural network is quite popular and is used in present task. Inputs and outputs of the proposed model are shown in Fig.4.

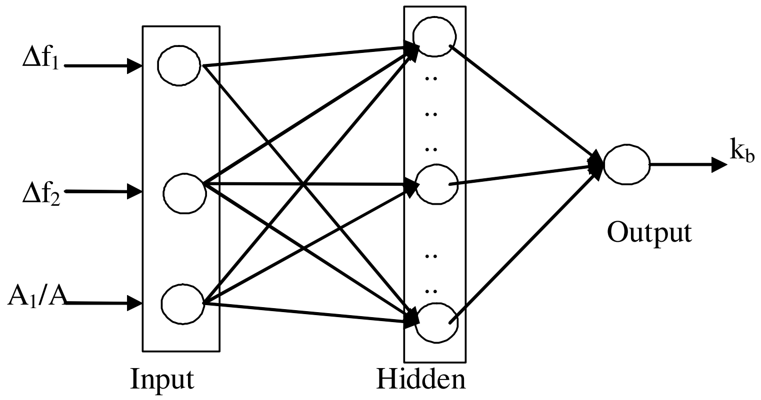
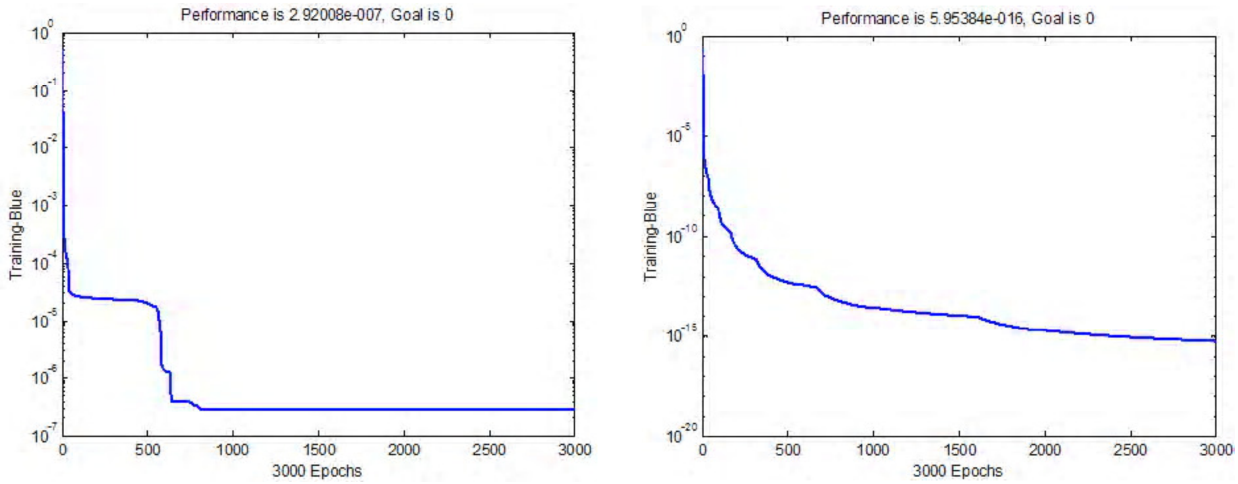


Fig. 4. Three-layer neural network model for identification of bearing stiffness

Dynamic state of the system with respect to simply-supported rotor condition is shown by first two frequency shifts, Δf_1 and Δf_2 and resonance amplitude ratio A_1/A_0. Using finite modeling, such a data points are obtained for different k_b values. Following neural network parameters are employed: sigmoid activation function $f(h_j) = \dfrac{1}{1+\exp(-h_j)}$, where input of each successive layer $h_j = \sum_{i=1}^{3} w_{ji} I_i$, with w_{ji} as connection weights and I_i is input vector elements for each layer. A learning parameter of 0.8 is selected during training. The weight adjustment in each learning cycle is according the conventional back-propagation learning law that minimizes the mean square error between the target or given k_b values and those computed from sigmoid functions. Fig.5 shows the training trend as a function of number of epochs.



(a) 3-3-1 Architecture (b) 3-5-1 Architecture

Fig. 5. Neural network training Process

Total 12 data sets are used; 10 for training and 2 for testing. Table-2 shows the output prediction error for different number of hidden layer neurons. It is seen that the 3-5-1 architecture with 5 hidden nodes is approximating nicely the system dynamics and is adopted as a system model in present case.

Table 2. Trail networks tested

Architecture	*Training Error*	*Testing Error*
3-2-1	1.55e-6	2.00e-4
3-3-1	2.92e-7	8.88e-5
3-4-1	1.71e-12	4.53e-6
3-5-1	5.95e-12	8.45e-8
3-9-1	5.61e-18	8.20e-9

EXPERIMENTAL SET-UP

Vibration exciter tests are conducted on the prototype rotor to predict frequency-response and natural frequencies. As shown in Fig. 6, the set-up has a signal generator (supplying sine excitation of variable frequency) driving the exciter via a power-amplifier to control and amplify the input signal amplitudes.

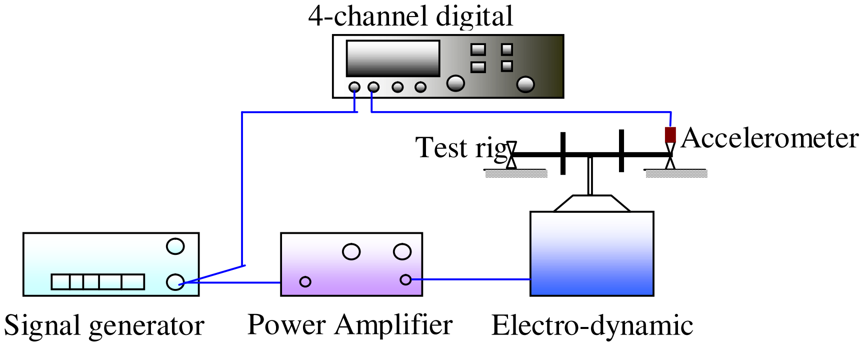


Fig. 6. Line diagram of experimental set up

An accelerometer mounted on the test object near the bearings measures the output signal. Both the input and output signals are observed using a four channel digital storage oscilloscope. Fig.7 shows the actual experimental set-up making use of above equipment.

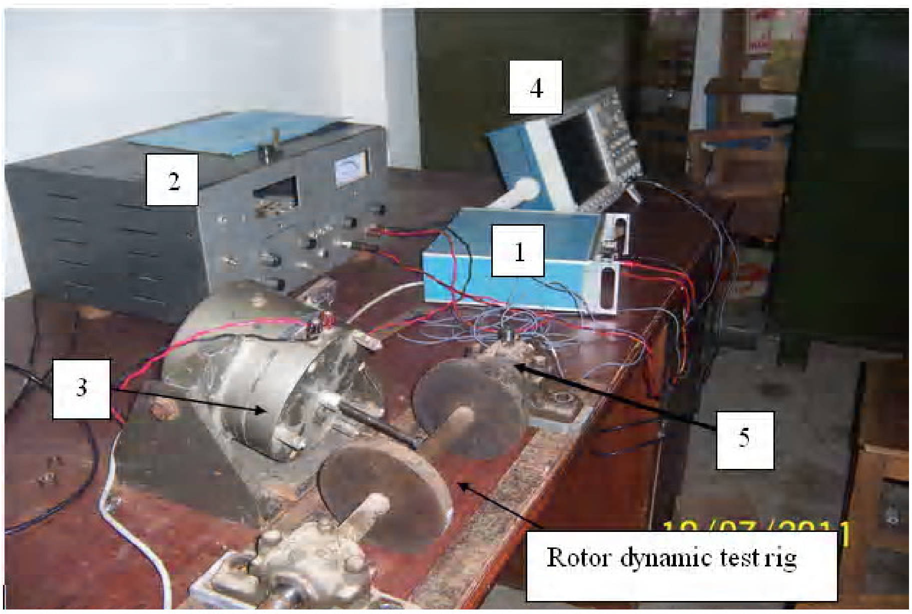


Fig. 7. Experimental set-up employed

Care is taken to avoid the frequency-inaccuracies of signal generator by observing the input signal in the oscilloscope simultaneously. Constant amplitude of 0.4 Amperes is maintained through-out the experiment. Fig.8 shows the screen shot of the oscilloscope illustrating the applied sine signal along with output response obtained from accelerometer. Using oscilloscope settings, we can measure the amplitude of the periodic signal from accelerometer at each frequency. This amplitude data is plotted with the frequencies manually so as obtain the frequency response curves.

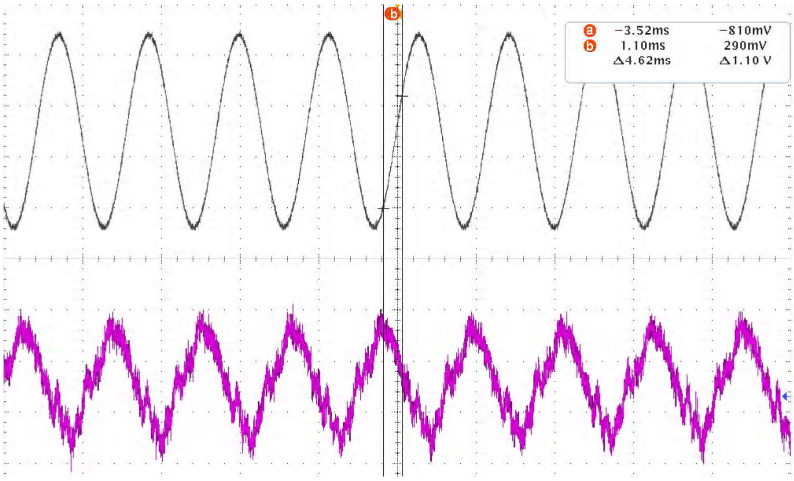


Fig. 8. Screen shot of oscilloscope with signals from the input and output channels

Two test conditions are implemented: (i) rigid simply supported conditions and (ii) supporting over a ball bearing system of unknown stiffness characteristics. Figure 9 shows the output frequency response curves of the rotor in two cases. It is seen that there are two peaks in solid curve (at around 151 and 172 Hz) along with two Y-bending modes (one at 70 Hz and other at 190 Hz). These are due to the internal bearing dynamics. For the rigidly supported rotor, there are two dominant modes one at 95 Hz and other at 320 Hz as seen from the dotted curve. Further, additional modes are also seen due to inaccuracies in simulating rigid rotor supports. Using this data, non-dimensional values of frequency shift and relative amplitudes are approximately evaluated and provided to the trained neural network model formulated earlier.

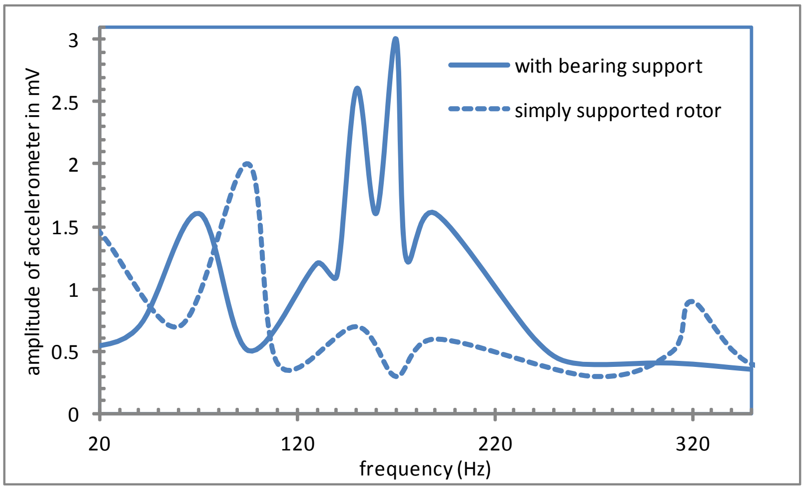


Fig. 9. Frequency response curves of the rotor

The output stiffness of the bearing corresponding to the input frequency shifts of 25 and 130 Hz along with amplitude ratio of 0.78 is estimated as 1.3 MN/m. During training process a learning rate of 0.4 is employed and learning is based on Levenberg–Marquardt algorithm.

CONCLUSIONS

A methodology of prediction of bearing stiffness of unknown supports from experimentally measured frequency-response has been presented in this work. The approach was illustrated with a dual-disk rotor system supported over two-row deep-groove ball bearings. Finite element model was employed to obtain an approximate dynamic response data of the system for different bearing stiffness values. The data of relative amplitude ratios and frequency shifts using harmonic response were used to interpolate with the corresponding bearing stiffness parameters by means of a neural network model. Experimental modal analysis was conducted on the scaled prototype rotor mounted over ball bearings of unknown stiffness and using the obtained frequency response and identified system model, the bearing stiffness values were approximately reported. In real practice, as the ball-bearing support is a nonlinear dynamic system, the stiffness parameters vary as a function of time. Thus, the model must be improved and generated using a time-delay neural network architecture rather than back-propagation model. Work is under progress.

REFERENCES

G.Chen, C.G.Li. and D.Y. Wang , *Nonlinear Dynamic Analysis and Experiment verification of Rotor-Ball Bearings-Support- Stator Coupling System for Aeroengine With Rubbing Coupling Faults*, Journal of Engineering for Gas Turbines and Power, Trans. ASME, Vol. 132,(2010),pp.022501-1-9,2010.

M.Cheng. M, Guang M. and Jianping J,*Numerical and experimental study of a rotor–bearing–seal system,* Mechanism and Machine Theory, Vol.42, Issue 8,(2007), pp.1043-105, 2007.

T.H.Patel, and A.K. Darpe, *Vibration response of a cracked rotor in presence of rotor–stator rub,* Journal of Sound and Vibration, Vol. 317, (2008), pp. 841–865, 2008.

C.Bai,Z. Hongyan and X.Qingyu, *Experimental and Numerical Studies on Nonlinear Dynamic Behavior of Rotor System Supported by Ball Bearings,* Journal of Engineering for Gas Turbines and Power, Trans. ASME, Vol. 132, (2010),pp. 082502-1-5, 2010.

X.Wenhei, T.Yougag and C.Yushu, *Analysis of motion stability of the flexible rotor-bearing system with two unbalanced disks*, Journal of Sound and Vibration, Vol.310 (2008), pp.381-393.

Q.Han, Z.Zhang and B.Wen, *Periodic motions of dual-disk rotor system with rub-impact at fixed limiter*, Journal of Mechanical Engineering Science, Part-C, Proc. IMechE., Vol.222 (2008), pp.1935-1946.

V.Hariharan and P.S.S.Srinivasan, *Vibration Analysis of parallel misaligned shaft with ball-bearing systems*, Songklanakarin Journal of Science and Technology, Vol.33/1 (2011), pp.61-68.

E.Dikman, P.J.M.Vander Hoogt, A.deBoer, R.G.K.M.Aarts and B.Jonker, *Design of an experimental set-up for testing multiphysics effects on high-speed mini rotors*, Journal of Mechanical Design, Trans. ASME, Vol.133 (2011), pp.0510091-9.

E.R.Marsh and D.S.Yantek, *Experimental measurement of precision bearing dynamic stiffness*, Journal of Sound and Vibration, Vol.202 (1997), pp.55-66.

M.H.Sadeghi and M.Zehsaz, *The effect of bearing stiffness on the natural frequency response function of a turbo-pump shaft-comparative study*, European J.Scientific Research, Vol.39 (2010), pp.381-395.

K.B.Lee, Y.B.Kim and K.Y.Jeong, *Identification of rotor bearing parameters using orbit data*, Journal of Multi-body dynamics, Part-K, Proc. IMechE., Vol.219 (2005), pp.249-257.

Y.Kang, C.C.Huang, C.S.Lin, P.C.Shen and Y.P.Chang, *Stiffness determination of angular contact ball bearings by using neural network*, Tribology International, Vol.39 (2006), pp.461-469.

Y.H.Kim, B.S.Yang and A.C.C.Tan, *Bearing parameter identification of rotor-bearing system using clustering-based hybrid evolutionary algorithm*, Structural and Multidisciplinary Optimization, Vol.33 (2007), pp.493-506.