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Vibration Analysis of a Support Excited Rotor System with Hydrodynamic Journal Bearings

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Abstract

This paper presents modelling and dynamic analysis of an on-board rotor with base excitation. A flexible shaft and rigid disc model is considered for idealizing a turbocharger rotor. The nonlinear hydrodynamic journal bearing forces are computed and the discretized equations of motion using finite element method are solved with time integration scheme. The disc unbalance forces are also accounted. The instability regions are identified by inducing varying sinusoidal support excitations. The work gives insights in design of the automobile engine turbocharger rotor mounted on hydrodynamic journal bearings supported with viscoelastic springs.

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Keywords: Finite element method, Transient analysis, Short-Bearing forces, Frequency response, Support Excitation, Nonlinear forces.

1. Introduction

In several instances rotating machinery are found to be situated on a moving supports. For example the vehicle turbocharger and gas turbine rotors are having such a support motion along with internal unbalance forces. The dynamic behavioral study of such a system is therefore very important in vibrations perspective. Several works reported on the behavior of rotors in such situations. In early 1980's the effect of random excitations of the support on the rotor response were studied [1-3]. Lee et al. [4] studied the effect of shock excitation on the support of a rotor. Elsaeidy et al. [5] proposed the dynamics of rigid rotor mounted on flexible bearing subjected to support excitation.

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Most of these earlier works simplified the modelling issues by considering either rotor as a Jeffcott model or analyzing it with approximate methods without considering time varying bearing forces and flexibility of the bearing support system. For the dynamic analysis of systems like rotors, finite element method and transfer matrix method are quite common. Due to its simplicity, transfer matrix technique along with its improved versions has become famous in rotor dynamics. Few of the earlier works are reported first.

Mzaki et al. [6] described the nonlinear dynamics of a support-excited rotor with hydrodynamic journal bearings. Nelson et al. [7] explained about the using of the finite element method for the dynamics of rotor-bearing system. Peletan et al. [8] proposed a comparison of stability computational methods for periodic solution of nonlinear problems with application to rotor dynamics. Dakel et al. [9] explained the dynamic analysis of harmonically excited on-board rotor-bearing system. Kang et al. [10] proposed an investigation in stiffness effects on dynamics of rotor-bearing-foundation systems. Lee et al. [11] proposed a finite element transient response analysis of a rotor-bearing system to base shock excitation using the state-space Newmark scheme and compared with experiments. Han et al. [12] described the parametric instability of flexible rotor-bearing system under time-periodic base angular motions. Several other recent works [13, 14] also focused on dynamics of rotors supported on journal bearings. In all above works, the bearings were directly mounted over the support without considering support elasticity.

Present paper focuses on the dynamic modelling of dual disc flexible rotor idealizing a turbocharger rotor system supported on hydrodynamic journal bearings with viscoelastic supporting conditions. The base platform is subjected to excitation which will be transmitted to the rotor and the response levels at different locations in the rotor are estimated. Effect of base excitation frequency and amplitude on the dynamics of rotor is illustrated, with time-histories, phase diagram and frequency responses.

2. Dynamic Modeling

The proposed rotor model (R) considered is shown in Fig. 1. It has two discs (D_1 and D_2) at each end and supported by hydrodynamic journal bearings (HDB_1 and HDB_2) with an oscillating support (S), through viscoelastic springs. During the modelling, the bearing forces are functions of relative displacements of the shaft Centre with respect to the support motion. Further, the disc unbalances, gravity as well as the viscoelastic force components are treated as external forces. The base motion is approximated as a sinusoidal signal acting in the vertical direction.

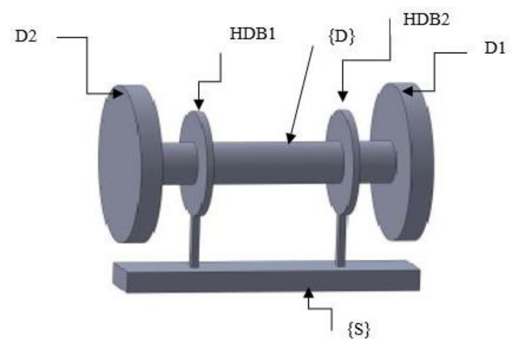


Fig. 1. Components of the on-board rotor model

A simplified finite element model of the rotor system is established using Timoshenko beam theory. As shown in Fig. 2. There are three shaft elements and four nodes altogether to discretize the simplified rotor. Each node has four degrees of freedom (DOF) including two translations (u_x, u_y) and two rotational displacements (θ_x, θ_y). Each journal bearing is supported by viscoelastic spring elements k_1, k_2 on an oscillating base. Total there are 16 degrees of freedom along with two vertical degrees each at the two base joints.

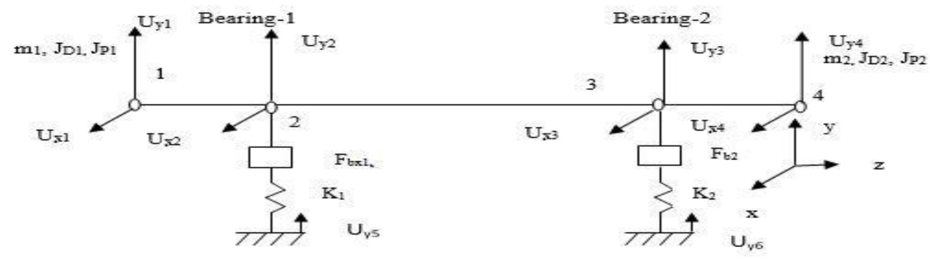


Fig. 2. Finite element formulation considered

The two discs are represented as lumped masses (rigid bodies) located at the nodes 1 and 4, while the nodes 2 and 3 represent the journal bearings. In addition, the component unbalance forces and gravity due to discs act at nodes 1 and 4 as external excitations. Taking the bending and shearing effect into consideration, the kinetic energy and potential energy of a rotating shaft element can be written as:

$$T_e = \int \frac{1}{2}\rho\left\{A\left(\dot{u}_x^2 + \dot{u}_y^2\right) + I_d\left(\dot{\theta}_x^2 + \dot{\theta}_y^2\right) + I_p\left[\Omega^2 + \Omega\left(\dot{\theta}_x\theta_y - \dot{\theta}_y\theta_x\right)\right]\right\}ds \tag{1}$$

$$U_e = \int \frac{1}{2}\left\{EI\left(\theta_x'^2 + \theta_y'^2\right) + kGA\left[\left(\theta_y - u_x'\right)^2 + \left(\theta_x + u_y'\right)^2\right]\right\}ds \tag{2}$$

The kinetic energy of the each disk can be written as:

$$T_d = \frac{1}{2}m_d\left(\dot{u}_x^2 + \dot{u}_y^2\right) + \frac{1}{2}J_d\left(\dot{\theta}_x^2 + \dot{\theta}_y^2\right) + \frac{1}{2}J_p\left[\Omega^2 + \Omega\left(\theta_x\theta_y - \theta_y\theta_x\right)\right] \tag{3}$$

The work caused by the mass eccentricity of the disk is:

$$W_d = m_d r_d \Omega^2\left(u_y \cos\Omega t + u_x \sin\Omega t\right) \tag{4}$$

According to the finite element method, the translation displacements (ux, uy) and rotational displacements (θx, θy) of a typical cross section of the shaft unit can be approximated by the following equations:

$$\begin{Bmatrix} u_x \\ u_y \end{Bmatrix} = Nq_e, \qquad \begin{Bmatrix} \theta_y \\ \theta_x \end{Bmatrix} = Dq_e \tag{5}$$

Where, N and D are translational and rotational shape function matrices. By introducing the above equations into the kinetic and potential energy expressions and carrying out the integrations over the element length and applying the Hamilton's principle, the following matrix equations of motion for shaft unit and disk are generated:

$$M_e\ddot{q}_e + \Omega G_e\dot{q}_e + K_e q_e = F_e + G_{ge}, \qquad M_d\ddot{q}_d + \Omega G_d\dot{q}_d = F_d + G_{gd} \tag{6}$$

$$[M_e] = \int_0^\ell \rho A[N]^T[N]ds + \int_0^\ell \rho I_d[D]^T[D]ds, \qquad [G_e] = \int_0^\ell \rho I_p[D]^T\begin{bmatrix} 0 & 1 \\ -1 & 0 \end{bmatrix}[D]ds \tag{7}$$

$$[Ke] = \int_0^{\ell} EI[D']^T[D']ds + \kappa GA\int_0^{\ell}\left\{[N]^T[N]+[D]^T[D]+2[N']^T\begin{bmatrix}0 & -1\\ 1 & 0\end{bmatrix}[D]\right\}ds \tag{8}$$

$$[M_d] = \begin{bmatrix} m_d & & & \\ & m_d & & \\ & & J_d & \\ & & & J_d \end{bmatrix},\ [G_d] = \begin{bmatrix} 0 & & & \\ & 0 & & \\ & & 0 & J_p \\ & & -J_p & 0 \end{bmatrix} \tag{9}$$

Considering the damping effect on rotor, the matrix equation of motion for the whole rotor system can be described as follows:

$$[M]\{\ddot{q}\}+\left[[C]+\Omega[G]\right]\{\dot{q}\}+[K]\{q\}=\{F\}+\{G_g\} \tag{10}$$

Where:

$$\{q\}=[u_{x1}\quad u_{y1}\quad \theta_{x1}\quad \theta_{y1}\quad \ldots\ldots\quad \theta_{y4}]^T \tag{11}$$

The system of equations are thus obtained by assembling equations of shaft, disk and bearing together with supports. The resulting equations with non-proportional damping are written as:

$$[A]\{\dot{Z}\}+[B]\{Z\}=\{\hat{F}\} \tag{12}$$

Where

$$[A]=\begin{bmatrix}[0] & [M]\\ [M] & [C]\end{bmatrix},[B]=\begin{bmatrix}[M] & [0]\\ [0] & -[K]\end{bmatrix},\{Z\}=\begin{Bmatrix}\{\dot{q}\}\\ \{q\}\end{Bmatrix},\{\hat{F}\}=\begin{Bmatrix}[0]\\ \{F\}\end{Bmatrix} \tag{13}$$

2.1. Bearing forces

Figure 3 shows hydrodynamic bearing consisting of a fixed bearing, rotating shaft and oil film separating the two. Let R, L and C_r =(R-r) are respectively the radius, length and clearance of the bearing.

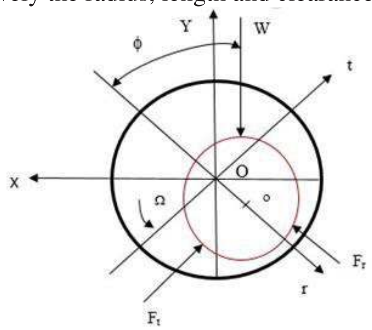


Fig. 3. Bearing coordinate system and notation.

At a constant static load due to rotor weight, the shaft Centre attains an equilibrium position with displacements u_{x0} and u_{y0} or equivalently by a static eccentricity e_0 and static attitude angle ϕ_0 between static load direction and the line of centers. The oil film pressure distribution is governed in a dynamic zone by Reynolds equation and hydrodynamic forces acting on the shaft are obtained by integrating over the bearing. Due to dynamic forces, including mass unbalance, bearing forces and support excitation, the shaft center position displaces at every time second. The dynamic forces with short bearing theory (L/R≤1/4) are given by

$$F_t = \frac{\mu R L^3 \varepsilon}{2C_r^2\left(1-\varepsilon^2\right)^2}\left[4\dot{\varepsilon} + \frac{\pi}{2}\sqrt{1-\varepsilon^2}\left(\Omega - 2\dot{\phi}\right)\right], \quad F_r = \frac{-\mu R L^3}{2C_r^2\left(1-\varepsilon^2\right)^2}\left[\frac{\pi\dot{\varepsilon}\left(1+2\varepsilon^2\right)}{\sqrt{1-\varepsilon^2}} + 2\varepsilon^2\left(\Omega - 2\dot{\phi}\right)\right] \tag{14}$$

Where

$$\varepsilon = \frac{e}{C_r} \quad e = \sqrt{\frac{x^2}{y^2}} \quad \tan\phi = \frac{u_x}{u_y}$$

Also, μ is dynamic viscosity and ω is the speed of rotation. The final component forces are given as follows in x and y directions.

$$F_x = -F_t \cos\phi - F_r \sin\phi, \qquad F_y = -F_r \cos\phi + F_t \sin\phi \tag{15}$$

2.2. Viscoelastic support

Viscoelastic supports connecting the hydrodynamic bearings with oscillating base are assumed to have an equivalent stiffness coefficients $K_v^* = K_v \times (1 + j\eta)$, where K_v is the in-phase stiffness of the support and η is the damping loss factor of the supporting material. The effective forces at the bearing due to viscoelastic supports connecting the bearing with the support platform are given by

$$F_{bx} = F_{by}, \qquad F_{by} = F_{by} + K_v^* \times \left(u_{yb} - y_s\right) \tag{16}$$

Where u_{yb} is the y displacement at the bearing and $y_s = Y_0 \sin(\varpi t)$ is support excitation in y direction.

3. Results and discussion

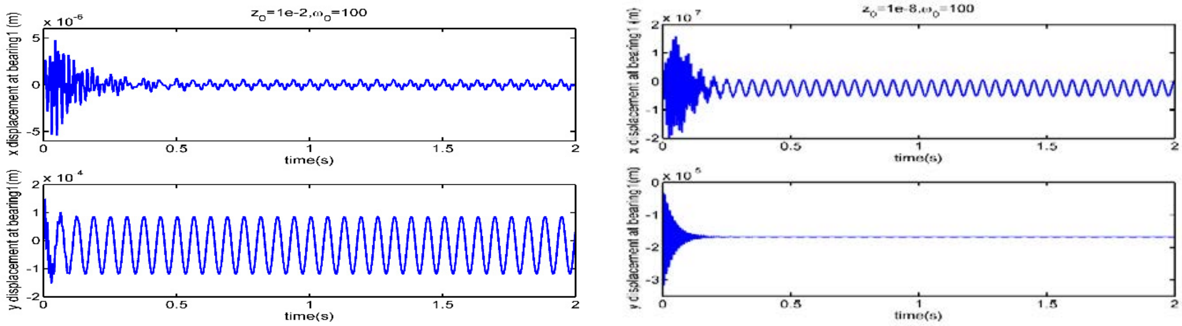
The equations of motion can be solved conveniently with zero initial conditions using fourth-order Runge-Kutta method or by using ode45 solver in MATLAB. The material and geometric data considered for dynamic modelling of rotor and bearing is shown in the Table 1.

Table 1. Rotor dynamic system data

Properties	Value
Density of shaft material (kg/m^3)	7800
Radius of disc, R_D (m)	0.15
Thickness (m)	0.03
Radius of shaft, R_{sh} (m)	0.04
Length of the shaft (m)	0.4
Young's modulus, E (GPa)	200
Shear correction factor (κ)	0.88
Bearing radius (m)	0.04
Bearing length(m)	0.01
Bearing radial clearance (microns)	200

The base platform is subjected to excitation which will be transmitted to the rotor and the response levels at different locations in the rotor can be estimated. The base excitation considered is a translation $Y = Z_0 \, cos(\omega_0 t)$. Effect of base excitation frequency and amplitude on the dynamics of rotor are illustrated at particular speed of 1200 rpm. Figure 4 shows the time histories at the left bearing node corresponding to the following bearing parameters: radius R=0.04 m, L=0.01m, radial clearance C=200×10^{-4} m. It shows the time histories of first bearing for different values of the base amplitude (Z_0), at k_v =1×10^5 N/m. and base frequency (ω_0) =100 rad/sec.

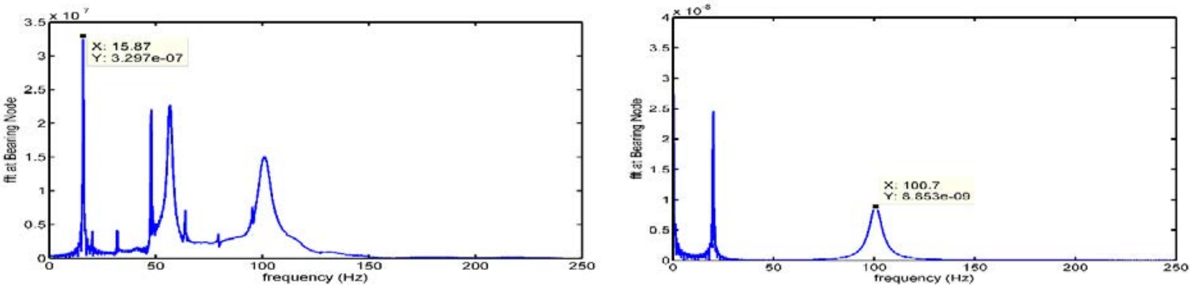
3.1. Effect of base excitation amplitude



(a) Z_0= 1×10^{-2}m (b) Z_0= 1×10^{-8} m

Fig. 4. Time histories at bearing-1

It is seen that in all the cases, the system is periodic and as the base excitation amplitude reduces, the steady state reaches must faster. Figure 5 shows the corresponding frequency responses.



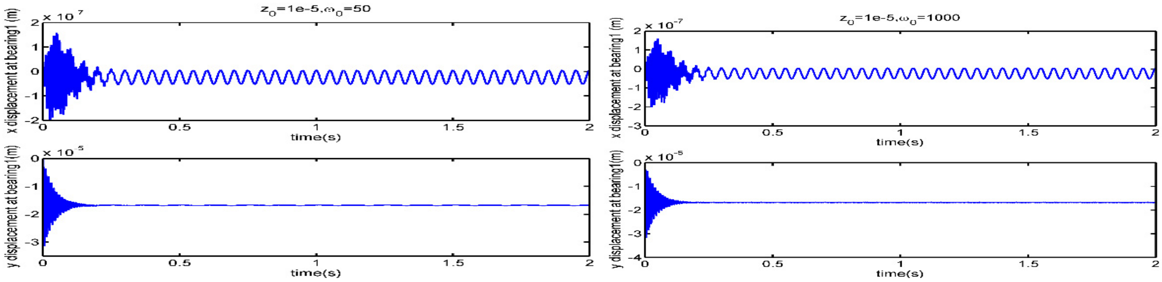
(a) Z_0 =1×10^{-2} m, ω_0=100 rad/sec., k_v=1×10^5 N/m (b) Z_0=1×10^{-8} m, ω_0=100 rad/sec, k_v=1×10^5 N/m

Fig. 5. Frequency response at bearing-1

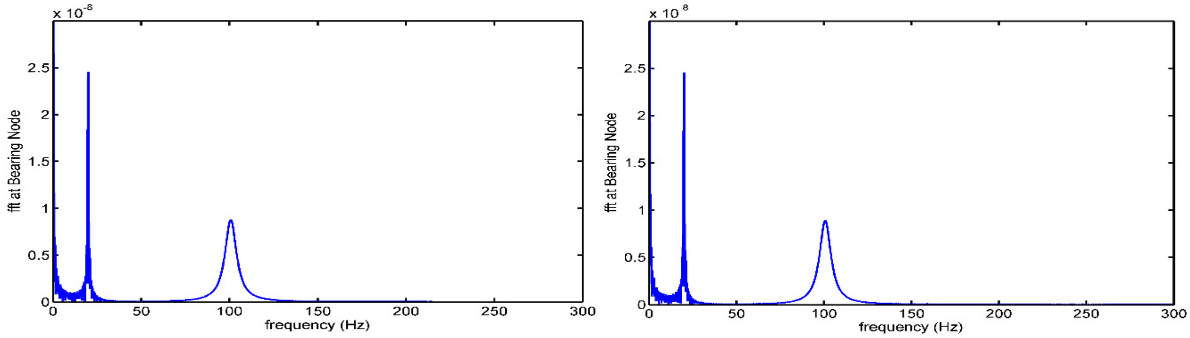
It is seen that due to high amplitudes of base excitation, there multiple peaks in FFT diagram indicating an unstable condition of the rotor.

3.2. Effect of base excitation frequency

Figure 6 shows the time histories, and FFT at two different values of excitation frequency ω_0. There is no significant effect observed in these figures.



(a) $Z_0 = 1\times10^{-5}$ m, ω_0=50 rad/sec, $k_v = 1\times10^5$ N/m (b) $Z_0 = 1\times10^{-5}$ m, ω_0=1000 rad/sec, $k_v = 1\times10^5$ N/m

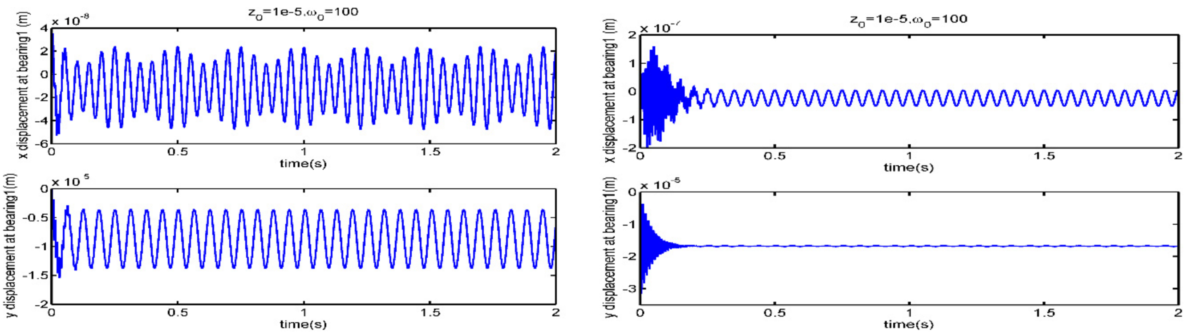


(c) $Z_0 = 1\times10^{-5}$ m, ω_0=50 rad/sec, $k_v = 1\times10^5$ N/m (d) $Z_0 = 1\times10^{-5}$ m, ω_0=1000 rad/sec, $k_v = 1\times10^5$ N/m

Fig. 6. Time histories and FFT at bearing -1

3.3. Effect of support stiffness

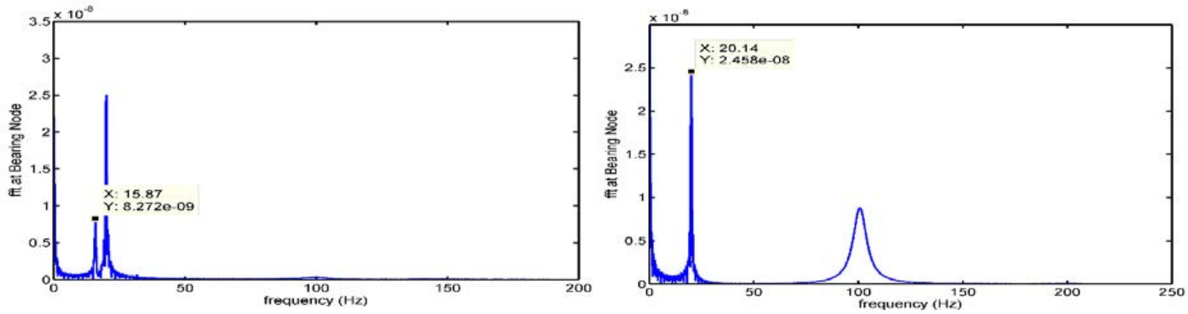
Figure 7 shows time histories at different values of support stiffness.



(a) $Z_0 = 1\times10^{-5}$ m, ω_0=100 rad/sec, $k_v = 1\times10^7$ N/m (b) $Z_0 = 1\times10^{-5}$ m, ω_0=100 rad/sec, $k_v = 1\times10^5$ N/m

Fig. 7. Time histories at bearing -1

Figure 8 shows the corresponding FFT in X-direction at bearing node-1. It is seen that at lower values of support stiffness, there are two distinct peaks corresponding to the two modes of the shaft bearing system. Further, due to the higher value of support stiffness, the two peaks have come close to each other leading to the avoidance of resonance effect.



(a) Z_0=1×10^{-5} m, ω_0=100 rad/sec, k_v =1×10^7 N/m (b) Z_0=1×10^{-5} m, ω_0=100 rad/sec, k_v=1×10^5 N/m

Fig. 8. Frequency response at bearing -1

4. Conclusions

In the present work the dynamic modelling of dual disc flexible rotor idealizing a turbocharger rotor system supported on hydrodynamic journal bearings with viscoelastic supporting conditions is considered and it was analyzed by finite element modeling using Timoshenko beam theory. The rotor supported on short length hydrodynamic journal bearings possesses nonlinear dynamic behavior. The base platform supporting bearing pedestal is subjected to excitation which will be transmitted to the rotor and the response levels at three different parameters of the system were estimated using time integration scheme. It can be concluded that the effect of base amplitude is much higher compared to the base excitation frequency. Also the viscoelastic support considered as a hysteresis damper has appreciable effect on the overall dynamics of rotor. The results have to be validated with an experimental analysis.

References

[1] R. Subbiah, R. B. Bhat, T.S. Sankar, Response of rotor subjected to random support excitations. ASME Journal of vibration acoustics. 107 (1985) 453– 459.

[2] B. Samali, K.B. Kim, J.N. Yang, Random vibrations of rotating machine under earthquake excitations. ASCE Journal of Engineering Mechanics. 112 (1986) 550–565.

[3] V .Srinivasan, A.H. Soni, Seismic analysis of a rotor bearing system. Earthquake Engineering and structural dynamic. 12 (1984) 287–311.

[4] A.S. Lee, B.O. Kim, Y.C. Kim, A finite element transient response analysis method of a rotor bearing system to base shock excitations, Journal of sound and vibrations. 297 (2006) 595-615.

[5] F.M.A. Elseidy, F. Sticher, Dynamic of rigid rotor linear/nonlinear bearings system subjected to rotating unbalance and base excitation, Journal of vibration and control. 16 (2010) 403-438.

[6] M. Dakel, S. Baguet, R. Dufour, Nonlinear dynamics of a support –excited flexible rotor with hydrodynamic journal bearings, Journal of sound and vibrations. 333 (2014) 2774-2799.

[7] H.D. Nelson, J.M. McVaugh, The dynamics of rotor bearing system using finite elements, ASME Journal of Engineering for industry. 98 (2010) 593-600.

[8] L. Peletan, S. Baguet, M. Torkhani, G. Jacquet-Richardet, A comparison of stability computational methods for periodic solution of nonlinear problems with application to rotor dynamics, Nonlinear dynamics.72 (2013) 671-682.

[9] M. Dakel, S. Baguet, R. Dufour, Dynamic analysis of a harmonically excited on-board rotor-bearing system, in: 10th International Conference on Vibrations in Rotating Machinery, Sep. 11–12, London, UK, 2012, pp. 57–68.

[10] Y. Kang, Y.P. Chang, J.W. Tsai, L.H. Mu and Y.F. Chang, An investigation in stiffness effects on dynamics of rotor bearing foundation systems, Journal of sound and vibrations. 231 (2000) 343-374.

[11] A. Lee, B.O. Kim, Y. C. Kim, A finite element transient response analysis method of a rotor of a rotor bearing system to base shock excitation using the state space Newmark scheme and comparisons with experiments, Journal of sound and vibrations. 297 (2006) 595-615.

[12] Q. Han, F. Chu, Parametric instability of flexible rotor-bearing system under time-periodic base angular motions, applied mathematical modelling. 39 (2015) 4511-4522.

[13] K. Avramov, M. Shulzhenko, O. Borysiuk, C. Pierre, Influence of periodic excitation on self-sustained vibrations of one disk rotors in arbitrary length journal bearings, International journal of non-linear mechanics. 77 (2015) 274-280.

[14] M. Cha, S. Glavatskih, Nonlinear dynamic behavior of vertical and horizontal rotors in compliant liner tilting pad journal bearings: some design considerations, Tribology International. 82 (2015) 142-152.