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Effect of Oil Viscosity over Critical Frequencies in Rotor Bearing Systems

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Abstract

This paper describes the effect of bearing oil viscosity over critical frequencies in rotor bearing systems. A finite element technique is used to study the behavior of the system for different ranges of operation. Critical frequencies and vibrations are studied by varying stiffness and damping coefficients for different oil grades. Results for various conditions have been discussed.

Keywords: Frequency, rotor bearing, vibration, stiffness and damping coefficient

INTRODUCTION

Any motion that repeats itself after an interval of time is called as vibration oscillation. A vibration system generally includes a means for storing potential energy (spring or elasticity), a means for storing kinetic energy (mass or inertia) and a means which energy is gradually dissipated (damped). Hence a perfect way to represent a system is through spring mass, related natural frequency is undamped in nature i.e system will be oscillating throughout.

In this condition the equation of motion is given by equation 1.

$$m\ddot{x} + kx = 0 \qquad \text{Eq. 1}$$

$$\omega_n = \sqrt{\frac{k}{m}} \qquad \text{Eq. 2}$$

Where

ω_n : Undamped natural frequency in Hz

m : Mass of the system in kg

k : Stiffness of the system in N/mm

If vibration energy is lost then vibration corresponding to it is known as damped vibration and corresponding natural frequencies are known as damped natural frequencies.

In this condition the equation of motion is given by equation 1.

$$m\ddot{x} + c\dot{x} + kx = 0 \qquad \text{Eq. 3}$$

Then system response for under damped condition is

$$x(t) = e^{-\xi\omega_n t}(x_0 \cos\omega_d t + \frac{\dot{x}_0 + \xi\omega_n x_0}{\omega_d} \sin\omega_d t) \qquad \text{Eq. 4}$$

$$\omega_d = \omega_n \sqrt{1 - \xi^2} \qquad \text{Eq. 5}$$

Where

ω_d : Damped natural frequency in Hz

ξ : Damping ratio

t : Time in Seconds

If damping is viscous in nature then mechanical vibration will be in fluid medium such as air, gas, water, oil etc. The resistance offered by the fluid to the moving body leads to absorption of energy. This dissipation of energy depends on many factors such as size, shape of vibrating body, viscosity of supporting fluid, the frequency of vibration and velocity of vibrating body.

In viscous damping the damping force is proportional to the velocity of the vibrating body.

$$\tau = \mu \frac{du}{dy} = \mu \frac{v}{h} \qquad \text{Eq. 6}$$

$$F = A\mu \frac{du}{dy} = \mu A \frac{v}{h} = cv = c\dot{x} \qquad \text{Eq. 7}$$

Where

τ : Shear force between oil layers in N/m^2

μ : Fluid viscosity in N-s/m^2

c : Damping coefficient in N-s/m

F : Damping force in N

v : Velocity in m/s

h : Fluid film thickness in meters.

A: Area in m^2

If the system is excited by some external source $F(t)$ then

Equation of motion is given by equation 8.

$$m\ddot{x} + c\dot{x} + kx = F(t) \qquad \text{Eq. 8}$$

Then system response for under-damped condition is

$$x(t) = X\sin(\omega t + \phi) \qquad \text{Eq. 9}$$

Where ,

X : Amplitude in meter

ϕ : Phase angle of the response in radians

ω : Angular frequency in rad/second

Amplitude ‘X’ can be obtained from equation 10.

$$X = \frac{F_0 / K}{\sqrt{(1 - r^2)^2 + (2\xi r)^2}} \qquad \text{Eq. 10}$$

Where,

$r = \omega / \omega_n$: Frequency ratio

F_0 : Static force in Newton

From the above equations of response it is observed that harmonic motion is a function of damping ratio and frequency ratio which is in turn a function of damping oil viscosity.

In present paper the results of system response with respect to operating speed range for various oil grades are analyzed and discussed.

FINITE ELEMENT ANALYSIS OF ROTOR –BEARINGS SYSTEMS

In present analysis a simple rotor bearing arrangement is simulated and analyzed by using finite beam elements as proposed by Nelson and Vaugh [7]. Considered simply supported rotor bearing system is as shown in figure 1.

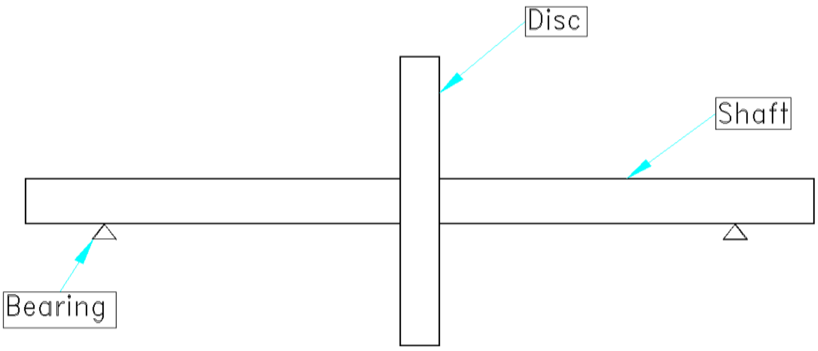


Figure 1. Simply Supported Rotor Bearing System

Conditions for rotor bearing systems are as follows:

1) A shaft diameter of 0.125m is considered over a length of 2m.
2) A Disc diameter of 500 kg with a diameter 0.5m which is located at center of shaft and bearing are equidistant with respect to center line of shaft.
3) Bearings are considered as natural lubricated RENK make horizontal journal bearing. Details of mass, stiffness and gyroscopic matrixes are discussed as per [7][8].

The equation of motion for considered rotor bearing system can be represented by equation 11.

$$[M]\{\ddot{x}\}+[C]\{\dot{x}\}+[K]\{x\}=[F] \qquad \text{Eq.11}$$

Where

[M] : Mass matrix

[C] : Damping matrix

[K] : Stiffness matrix

[F] : External excitation force matrix

Damping is introduced into the system through bearing oil film which is viscous in nature. Stiffness and damping coefficients were calculated through RENK bearing calculations [6] for given loading and rotational speed. As shaft-bearing center eccentricity is varied with respect speed and loading hence stiffness and damping coefficients are also varied proportionally.

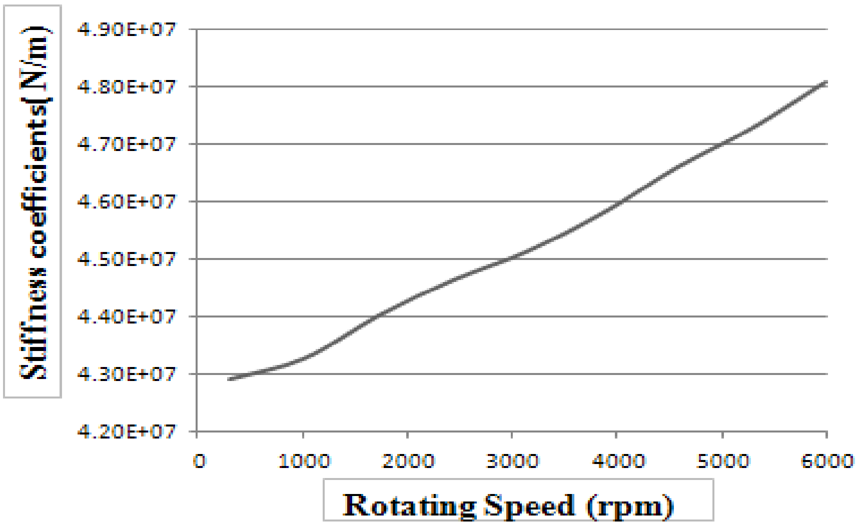
Variation of stiffness and damping coefficients in particular direction for a given loading with respect to speed are shown in graph1 and graph 2.

Stiffness variation with respect to speed is shown in graph-1.

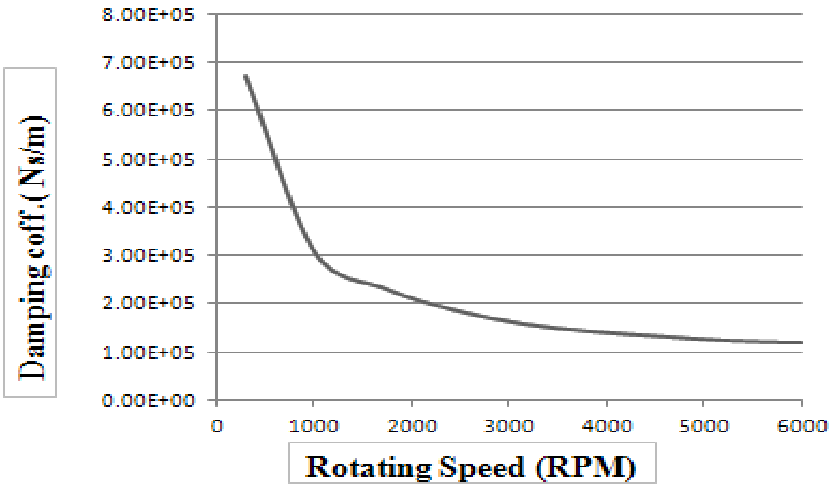
Stiffness increase with increase in speed, this is due to the fact that clearance between bearing shell and shaft increase. This is because there is lift in rotor from lower speeds to higher speeds.

Damping variation with respect to speed is shown in graph-2.

Damping decreases with increase in speed, this is due to the fact that at higher speeds there is more friction between the layers in oil film, this leads to temperature rise of oil. As oil viscosity decreases with increase in temperature, damping offered by oil film at higher speeds is less than that of offered at lower speeds.



Graph1. Stiffness coefficients variation with speed



Graph 2. Damping coefficients variation with speed

An external two-plane unbalance is introduced into the system as per American Petroleum Industries standard (API) 541[5]. As per standard API-541[5] the maximum residual unbalance that can be introduced an a rotating system can be calculated by equation 12.

$$Ub = 4\frac{W}{N} \qquad \text{Eq.12}$$

Where,

Ub : Unbalance in kg-m

W : Journal static loading in kg

N: Maximum continuous speed in rpm.

The API-541 standard also describe the limit for vibration level. As per API-541 any vibration with reference to amplification factor greater than 2.5 can be considered as critical frequency. Operation of the rotating rotary system at this frequency may lead to system distraction.

PRESENT ANALYSIS

For considered rotor bearing system different grades of oil are selected as per ISO VG grade, and are introduced as viscous dampers through the bearing to rotating system. Standard RENK made flange mounted, cylindrical bore, natural cooled journal bearings (FLNB 11-125) are considered to the system. Stiffness and damping coefficients were calculated for a particular oil grade and are introduced into the system as shown in Figure-II from RENK bearing calculations software officially available in RENK website [6].

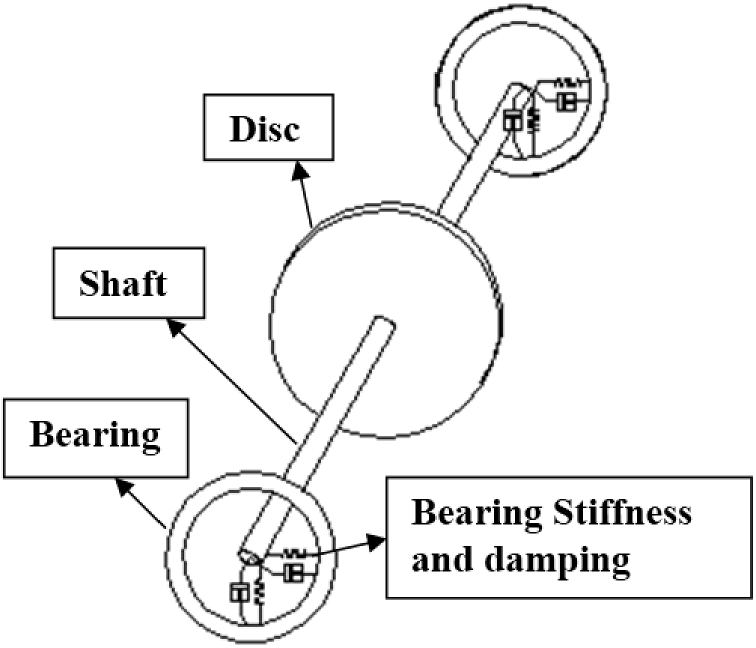
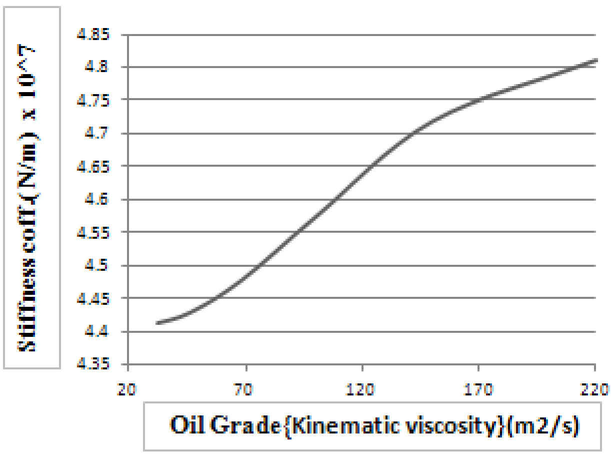
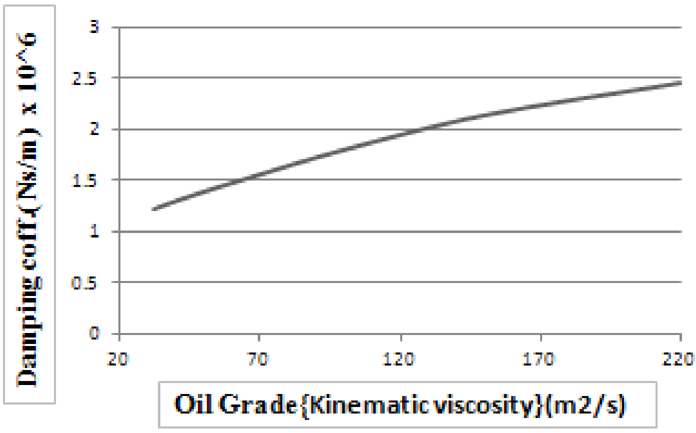


Figure 2. Rotor bearing system with stiffness and damping

Stiffness and damping parameters are calculated for particular oil grade for range of speeds. These parameters are introduced to the rotating system. Unbalance response graph is plotted for a range of rotational speeds. The same procedure is carried out for various oil grades. Standard oil grades that were considered in present analysis are ISO-VG 22, 32, 46, 68, 100, 150, 200. Variation of stiffness and damping coefficients for a particular speed for various grades of oils are as shown in graph 3 and graph 4.



Graph 3. Stiffness coefficients variation with Oil viscosity

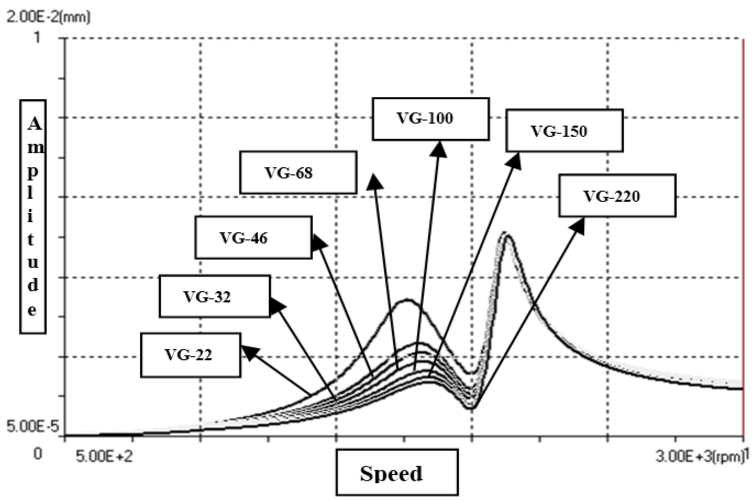


Graph 4. Damping coefficients variation with Oil viscosity

From above graphs the stiffness and damping coefficients are increased with the increase in oil viscosity. For a given speed the damping offered by high viscous fluid is more when compared to low viscous fluid. Unbalance response graph is plotted for a range of speeds with different grades of oils. Observations regarding critical frequencies and vibration amplitude are presented in Graph 5, Graph 6 and Graph 7.

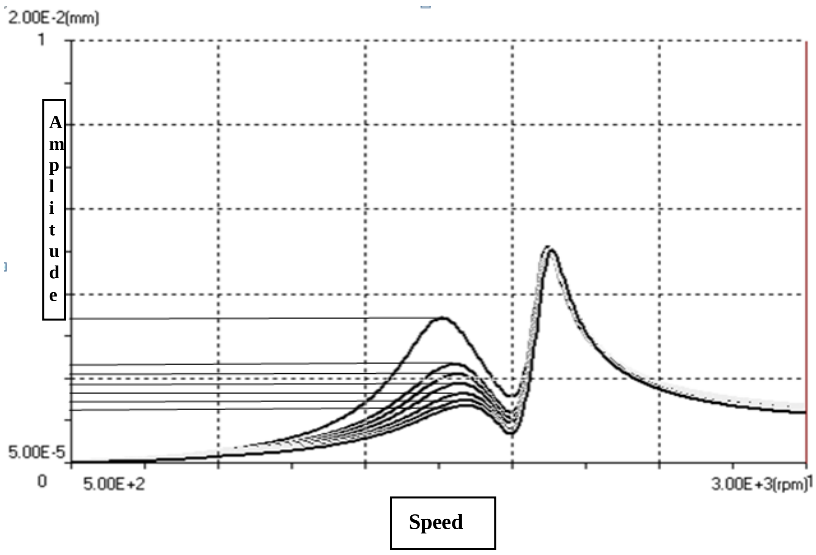
RESULTS AND DISCUSSIONS

System response graphs are generated for different grades of oils individually. Graph 5 shows the system response comparison for various oil grades. It is observed that both amplitude and speeds corresponding to maximum amplitudes are varying with respect to oil grades.



Graph 5. System responses variation for different oil grades

With increase in oil viscosity the damping offered by oil to the system increases there by the amplitude of vibration decreases. Response of variation is presented in graph 6.



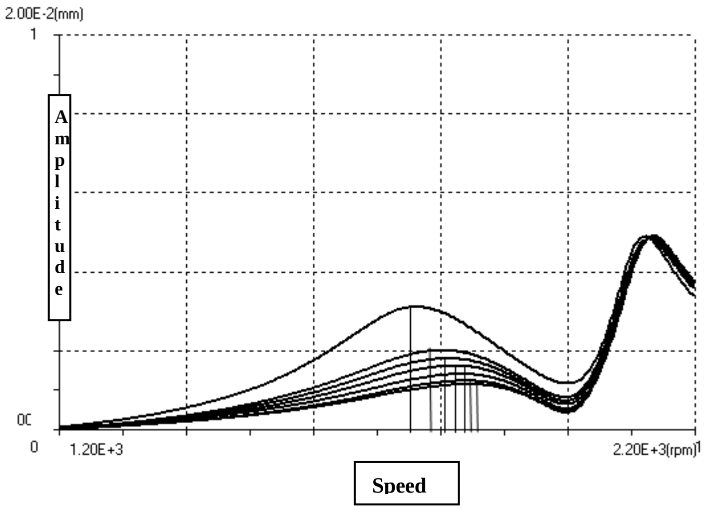
Graph 6. Variation of amplitude with different oil grades

The magnitude of variation in amplitude is as shown in Table1.

Table 1. Amplitude variation with respect to Oil Grade

AMPLITUDE VARIATION	
Oil grade ISO-VG	**Amplitude in µm**
22	6.91
32	4.73
46	4.26
68	3.82
100	3.33
150	3.01
220	2.74

With increase in oil viscosity the stiffness offered by oil to the system increases there by the damped frequencies corresponding to maximum amplitude also increases. Response of variation is presented in graph 7.



Graph 7. Variation of damped natural frequencies with different oil grades

The magnitudes of variation in damped frequencies corresponding to maximum amplitude are as shown in table2.

Table 2. Frequency variation with respect to Oil Grade

DAMPED FREQUENCIES VARIATION		
Oil grade ISO-VG	**Speed (RPM)**	**Frequency (Hz)**
22	1761	29.4
32	1802	30.0
46	1810	30.2
68	1821	30.4
100	1832	30.5
150	1838	30.6
220	1843	30.7

CONCLUSIONS

A finite element model of rotor bearing system was presented for a particular oil viscosity. Corresponding stiffness and damping coefficients were calculated and introduced. System response graph was plotted with certain residual unbalance which acts as external excitation and the results corresponding to vibration amplitude and frequency are noted.

Stiffness and damping coefficient were calculated for different viscous grade of oils and are introduced to system and corresponding system response graphs were generated with the same residual unbalance. It was observed that with increased of oil viscosity, damping offered by oil to the system has increased, this effect has decreased the amplitude of vibration and increased system's damped natural frequencies.

REFERENCES

[1] M. Lalanne, G. Ferraris, Rotor dynamics Prediction in Engineering, John Wiley & Sons, Inc., New York (1990).

[2] R. L. Eshleman and R. A. Eubanks, On the critical speeds of a continuous shaft disc system. ASME J. Engng Ind. 89, 645-652(1967).

[3] A. S. Sekhar and B. S. Prabhu, Unbalance response of rotor considering the distuted stiffness and damping. Page no. 94-GT-128, ASME Int.Gas Turbine and Aero engine Congr. And Expo., The Hague, Netherlands (1994).

[4] B. S. Rao, A. S. Sekhar and B. C. Majumdat, Analysis of rotors considering distributed bearing stiffness and damping, Department f mechanical engineering, IIT Kharagpur(1995).

[5] Form-wound Squirrel-Cage Induction Motors-500Horse power and Larger, ANSI/API STANDARD 541/2013, FOUTTH EDITION, JUNE (2004).

[6] RENK Bearing Calculation, www.renk-hannover.de/iks/

[7] H. D. Nelson and J. M. McVaugh, The dynamics of rotor-bearing system using dinite element. ASME J. Engng Ind. 98, 596-600(1977).

[8] E. S. Zorzi and H. D. Nelson, Finite element simulation of rotor-bearing system with internal damping. ASME J. Power 99, 71-76(1977).

[9] Chris A. Papadopoulos a, Pantelis G. Nikolakopoulos a, George D. Gounaris Mechanism and Machine Theory 43 (2008) 411–426

[10] B. Sudhakara Rao, A. S. Sekhar and B. C. Majumdar Compurers & Smcrvres Vol. 61, No. 5, pp. 951-955, 1996.

[11] Kasarda MEF , Allaire PE , Humphris RR , Barret LE . A magnetic damper for the first-mode vibration reduction in multimass flexible rotors. J Eng Gas Turbines Power 1990;112(4):463–9

[12] A. Muszynska, P. Goldman, Chaotic responses of unbalanced rotor bearing stator systems with looseness or rubs, Chaos, Solit. Fractals 5 (1995) 1683e1704.