Vibration Analysis of Rotor Shaft System using Journal Bearing

A thesis submitted in partial requirements for the degree of

Bachelor of Technology

In

Mechanical Engineering

By

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Tophan Nial (109ME0397)

Gogireddy Ravindra Reddy (109ME0409)

National Institute of Technology

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Under the guidance

Of

Prof. Tarapada Roy

National Institute of Technology

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National Institute Of Technology

Rourkela

CERTIFICATE

This is to certify that the thesis entitled, "**Vibration Analysis of Rotor Shaft System using Journal Bearing**" submitted by Mr. Kosish Gaihre (109ME0536), Mr. Tophan Nial (109ME0397), Mr. Gogireddy Ravindra Reddy (109ME0409) in partial fulfilments for the requirement of the award of Bachelor of Technology Degree in Mechanical Engineering at National Institute of Technology, Rourkela is an authentic work carried out by them under our guidance. To the best of our knowledge, the matter embodied in the thesis has not been submitted to any other University/ Institute for the award of any Degree or Diploma.

DATE

Prof. Tarapada Roy

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Date

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Abstract:

The present work deals with the development of finite element modeling for the analysis and study of vibration of rotor shaft system using journal bearing. The composite shaft is mounted an isotropic rigid disk and is supported by journal bearings at the ends those are modeled as springs and viscous dampers. The journal bearing for finite element model has been used based on the classical Reynolds equation for oil lubricated journal bearings. In this model the transverse shear deformation, rotor inertia and gyroscopic effects due to the lamination of composite layers have been incorporated. The equation of motion of composite rotor shaft-bearing system has been derived using the finite element method and the rotor shaft has also been modeled and analyzed using Timoshenko beam theory. To verify the present model, the critical speeds of composite shaft system are compared with those available in the literature. Graphs were also plotted to illustrate the frequency and displacement response of composite rotor shaft system.

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Chapter 1

Introduction

Introduction:

A composite is a material system consisting of two or more phases on a macroscopic scale, whose mechanical properties are designed to be superior to those of the constituent materials acting independently. One of the phases is stronger which is discontinuousis called reinforcement whereas the less stiff and weaker phase is continuous and is called matrix. Sometimes,interphase exists between the reinforcement and the matrix because of chemical interactions or other processing effects. The properties of a composite material depend on the properties of the geometry,constituents and distribution of the phases[1].

1.1 Classification of composites

A. Based on Matrices:

i. Polymer Matrix Composites(PMCs):

Polymers are the macromolecule built up by the linking tougher of large no of much smaller molecule monomers.

Properties

a. High tensile strength.

b. High stiffness, fracture toughness.

c. Good abrasion resistance, corrosion resistance and low cost.

ii. Metal Matrix Composites(MMCs): When the matrix is a metal it is known as metallic matrix composite.

Properties

a. Has improved strength stiffness, improved creep and fatigue resistance and hardness, wear and tear resistance.

iii. Ceramic Matrix Composites(CMCs): The fiber are usually impregnated with a slurry of fine glass powder and subsequently hot pressed the glass ceramic material goes through the stage called ceramming process, it is the final heat treatment to convert glass to fully dense ceramics.

Properties

a. These are usually used in high temperature application in aerospace industries

b. These are used where a need for reliable economic, good mechanical properties and reasonable wear and corrosion resistance in addition to adequate impact and thermal shock resistance.

B. Based on reinforcements: It is a condition or analysis for a process of strengthening a direct measurable dimension of behavior such as rate, duration magnitude. Reinforcement is only said to have occurred if the delivery of the stimulus is directly caused by the response. Two types of reinforcement are done

i. **Fiber reinforcement**: This is advanced type of composite group. It consist of fiber as a dispersed and discontinuous phase, the matrix as a continuous phase, the fiber interphase region.

ii. **Particle reinforcement**: Particle reinforcement is more attractive due to their cost effectiveness,isotropic properties. This kind of reinforcement is mainly done on metal matrix composite [2].

1.2 Advantages

Why are composites materials used?

a. Lower density (20 to 40%)

b. Higher directional mechanical properties(ratio ofmaterial strength to density 4 times greaterthan that of steel and aluminium, higher stiffness to density ratio).

c. Higher fatigue endurance.

d. Highertoughnessthan ceramics and glasses.

e. Versatility and tailoring by design.

1.3 Uses of fiber composite

i. Aircraft: doors skin on the stabilizer box, fin elevators,landing gear, tail spoiler.

ii. Aerospace: space shuttle, space station.

iii. Automotive: body frame, chassis components, engine components, drive shaft,etc. so as to get high stiffness and damage tolerance, good surface finish and appearance, weight reduction hence higher fuel efficiency.

iv. Sporting goods: tennis and racquet ball, racquets golf club shaft, heads of bicycle frame.

v. Electrical: printed circuit board, computer housing insulators, battery plates[3].

1.4 Rotating shaft:

A shaft is a rotating machine element which is used to transmit power from one element to another. The power is supplied to the shaft by someresultant torque, tangential force set up within the shaft permits the power to be transferred to various machines linked up to the shaft. Various members such as pulleys,gears etc. are mounted on the shaft in order to transfer the power from one shaft to another. These members along with the forces exerted upon them causes the shaft to bending. Shafts are manufactured by hot rolling and finished to size by turning or cold drawing and grinding. The hot rolled shafts are weaker than the cold rolled shafts. We are using graphite composites shaft. They have unique properties of relatively high strength at high temperatures coupled with low thermal expansion and low density.

1.5 Bearing:

A fluid film bearing is defined as a bearing in which the opposing or mating surfaces are completely separated by a layer of fluid lubricant.A widely used bearing type is the plain journal bearing has application in compressors, turbines, pumps, electric motors and electric generators. Journal Bearings consists of two cylinders rotating relative to each other. The outer one is stationary (bearing) and the inner ring (shaft) rotating with an angular velocity is called the Journal. The main purpose of the journal bearing is to support the rotating machinery providing sufficient lubrication to separate the moving parts and to minimize the friction due to rotation [4]. The high-pressure fluid film in the clearance between the journal and the bearing due to rotation of the journal provides the hydrodynamic film lubrication, and the load capacity to the bearing. The displacement of the shaft with respect to the bearing center is known as eccentricity.

Terms used in journal bearing

1. Diametral clearance:

Diameters difference between the bearing and the journal is diametral clearance.

2. Radial clearance**:**

Radii difference between the bearing and the journal is radial clearance.

3. Diametral clearance ratio**:**

Diametral clearance to the diameter of the journal ratio is diametral clearance ratio.

4. Eccentricity**:**

Radial distance between the center of the bearing and the displaced center of the bearing under load is eccentricity.

5. Minimum oil film thickness:

Under complete lubrication condition it is theminimum distance between the bearing and the journal.

Chapter 2

Literature survey

2.1 Literature survey

In the study of Hydrodynamic Fluid Film Bearings and Their Effect on the Stability of Rotating Machinery by Luis San Andres [5]deals with the static and dynamic performance characteristics of short length cylindrical journal bearings, with application to the dynamic forced performance of a rigid rotor supported on plain bearings. In a radial bearing, the Sommerfield number defines a relationship between the static load and the journal eccentricity within the bearing. These fluid film bearings also introduce viscous damping that aids in reducing the amplitude of vibrations in operating machinery. Reynolds equation for journal bearing was derived which depicts the amount of hydrodynamic pressure within the journal bearing.Kinematics of journal motion is studied which depicts about the Journal eccentricity which may vary with load on the bearing. The journal eccentricity cannot exceed the bearing clearance, otherwise solid contact and potential failure may occur.

In the study of a simple spinning laminated composite shaft modelby Min- Yung Chang, et.al [6]. The transverse shear deformation, gyroscopic effects, rotary inertia as well as the coupling effect due to the lamination of composite layers have been incorporated. Based on first order shear deformable beam theory, strain energy of the shaft was obtained by considering three dimensional constitutive relation of the material with the use of coordinate transformation. The critical speeds of composite shaft system are compared with those available in the literature. The composite shaft contains discrete isotropic rigid disks and is supported by bearings that are model as springs and viscous dampers.

The study of Foil Bearing Design Guidelines for Improved Stabilityby was studied by J.Schiffmann, et al[7]. A reduced order foil bearing model, coupled with a rigid-body,linear rotor dynamic model, was used to investigate the underlying rotor dynamic mechanisms and the onset speed of instability of a foil bearing-supported by the rotor. Introducing a critical mass parameter as a measure for stability, whirl instability onset was proposed. A sensitivity analysis demonstrates that structural damping does not significantly alter the onset of sub-synchronous whirl. It is shown, however, that the orientation of the axial feed line of the top foil can strongly influence the bearing load capacity and rotor dynamic performance.

The study of fundamentals of fluid film journal bearing operation and modeling was carried outby Minhui He etal[8]. All the important theoretical aspects of journal bearing modeling, such as film pressure, film pad and pad temperatures, thermal and mechanical deformations, and turbulent flow are reviewed. The bearing is running with sufficient load capacity and acceptable vibration response helps to ensure long term reliability of the bearing. For relatively vibrations like those normally encountered, bearings dynamic properties can be represented by linear springs and dampers. The direct stiffness, direct damping and cross-coupled stiffness coefficients all have significant implications.

In study of thedynamics of rotor bearing systems using finite elements[9]a finite element model including the effects of rotor inertia, gyroscopic moments and axial load is developed. The bearings may be nonlinear however only the linear stiffness and viscous damping is considered. Natural whirl speed and modes were calculated by using both the fixed and rotating frame formulations. The finite element model can easily be utilized to model rotor bearing systems for purposes of determining critical speeds, stability, unbalance response, transient response etc. the rotor element can be generalized to include the effects of shear deformation, axial torque and various forms of internal damping.

2.2 Objective:

In this study, a bearings supported shaft is modeled using finite element modeling method. The model is used for dynamics analysis. The shaft used in this design is hollow. The shaft is supported by journal bearing at both ends. Here, journal bearing is modeled by using classical Reynolds's equation for oil lubricated journal bearing. Variation of stiffness and damping coefficients journal bearing with respect to rotor speed can be obtained for vibration analysis. Whirl frequency with respect to rotating speed of shaft will be obtained and from those results, the most stable system can be concluded.

Chapter 3

Material and methods

3.1 Finite element modeling

The rotor bearing system is modeled using finite element modeling for both composite shaft and the hydrodynamic journal bearings. The implementation steps of the finite element procedure are described below.

3.2 Rotor shaft modeling

Figure 1depicts a schematic view of a composite rotor supported on fluid-film journal bearings. The bearings are represented by their stiffness and damping direct coefficients in x-axis and y-axis. The cross-coupled bearing coefficients are omitted in picture, but they will be considered in the modeling.

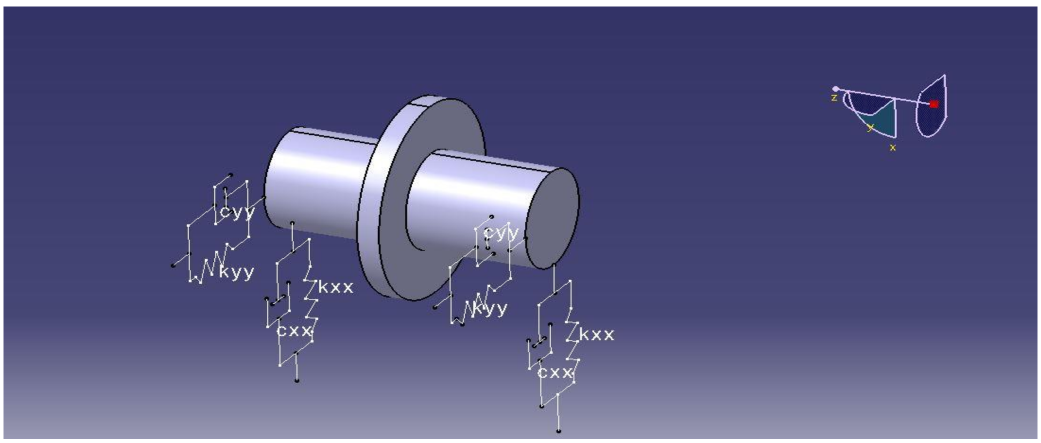

Figure 1. Composite rotor shaft supported on fluid-film journal bearings

The rotating system model includes a composite shaft carrying an unbalance disk. The finite element shaft modeling implemented in this work has been based onthe shape functions using Timoshenko beam theorywith three nodded element beams each node having 4 degree of freedom. The shaft model takes into account the shaft shear deformation effects, gyroscopic moments, and rotatory inertia[10]. The finite element procedure gives the following global equation of motion:

$$[M]\{\ddot{q}\} + ([C] + \Omega[G])\{\dot{q}\} + [K]\{q\} = \{F\}$$

Where *[M]* represents the mass matrix,*[G]* is the shaft gyroscopic effects matrix,*[K]*the shaft and bearing stiffness matrix, *[C]* is the generalized shaft and bearing damping matrix,*{q}* and *{F}* denotes the displacement and external force vectors respectively.

Elemental mass matrix

$$\left[M_e\right]=\begin{bmatrix} \left[M_{11}\right]_{3\times3} & 0 & 0 & 0 \\ 0 & \left[M_{22}\right]_{3\times3} & 0 & 0 \\ 0 & 0 & \left[M_{33}\right]_{3\times3} & 0 \\ 0 & 0 & 0 & 0 \quad \left[M_{44}\right]_{3\times3} \end{bmatrix}_{12\times12}$$

In which[6]

$$M_{ij}^{11}=\int_{x_a}^{x_b} I_m\psi_i\psi_j dx+\int_{x_a}^{x_b}\sum_{k=1}^{n_D} I_m^D \psi_i\psi_j\Delta(x-x_{Dk})dx$$

$$M_{ij}^{22}=\int_{x_a}^{x_b} I_m\psi_i\psi_j dx+\int_{x_a}^{x_b}\sum_{k=1}^{n_D} I_m^D \psi_i\psi_j\Delta(x-x_{Dk})dx$$

$$M_{ij}^{33}=\int_{x_a}^{x_b} I_m\psi_i\psi_j dx+\int_{x_a}^{x_b}\sum_{k=1}^{n_D} I_d^D \psi_i\psi_j\Delta(x-x_{Dk})dx$$

$$M_{ij}^{44}=\int_{x_a}^{x_b} I_m\psi_i\psi_j dx+\int_{x_a}^{x_b}\sum_{k=1}^{n_D} I_d^D \psi_i\psi_j\Delta(x-x_{Dk})dx$$

Gyroscopic matrix

$$\left[G_e\right]=\begin{bmatrix} [0]_{3\times3} & [0]_{3\times3} & [0]_{3\times3} & [0]_{3\times3} \\ [0]_{3\times3} & [0]_{3\times3} & [0]_{3\times3} & [0]_{3\times3} \\ [0]_{3\times3} & [0]_{3\times3} & [0]_{3\times3} & \left[G_{34}\right]_{3\times3} \\ [0]_{3\times3} & [0]_{3\times3} & \left[G_{43}\right]_{3\times3} & [0]_{3\times3} \end{bmatrix}_{12\times12}$$

In which

$$G_{ij}^{34}=\int_{x_a}^{x_b} I_p\psi_i\psi_j dx+\int_{x_a}^{x_b}\sum_{k=1}^{n_D} I_p^D\psi_i\psi_j\Delta(x-x_{Dk})dx$$

$$G_{ij}^{43}=\int_{x_a}^{x_b} -I_p\psi_i\psi_j dx+\int_{x_a}^{x_b}\sum_{k=1}^{n_D} -I_p^D\psi_i\psi_j\Delta(x-x_{Dk})dx$$

Damping matrix of bearing

$$\left[C_e^B\right]=\begin{bmatrix}[C_{11}]_{3\times3} & [C_{12}]_{3\times3} & [0]_{3\times3} & [0]_{3\times3}\\ [C_{21}]_{3\times3} & [C_{22}]_{3\times3} & [0]_{3\times3} & [0]_{3\times3}\\ [0]_{3\times3} & [0]_{3\times3} & [0]_{3\times3} & [0]_{3\times3}\\ [0]_{3\times3} & [0]_{3\times3} & [0]_{3\times3} & [0]_{3\times3}\end{bmatrix}_{12\times12}$$

In which

$$C_{ij}^{11}=\int_{x_a}^{x_b}\left\{\sum_{k=1}^{n_b} C_{xx}^{bk}\psi_i\psi_j\vartriangle\left(x-x_{bk}\right)\right\}dx$$

$$C_{ij}^{12}=\int_{x_a}^{x_b}\left\{\sum_{k=1}^{n_b} C_{xy}^{bk}\psi_i\psi_j\vartriangle\left(x-x_{bk}\right)\right\}dx$$

$$C_{ij}^{21}=\int_{x_a}^{x_b}\left\{\sum_{k=1}^{n_b} C_{yx}^{bk}\psi_i\psi_j\vartriangle\left(x-x_{bk}\right)\right\}dx$$

$$C_{ij}^{22}=\int_{x_a}^{x_b}\left\{\sum_{k=1}^{n_b} C_{yy}^{bk}\psi_i\psi_j\vartriangle\left(x-x_{bk}\right)\right\}dx$$

Stiffness matrix

$$[K]=\begin{bmatrix}[K_{11}]_{3\times3} & [K_{12}]_{3\times3} & [K_{13}]_{3\times3} & [K_{14}]_{3\times3}\\ [K_{21}]_{3\times3} & [K_{22}]_{3\times3} & [K_{23}]_{3\times3} & [K_{24}]_{3\times3}\\ [K_{31}]_{3\times3} & [K_{32}]_{3\times3} & [K_{33}]_{3\times3} & [K_{34}]_{3\times3}\\ [K_{41}]_{3\times3} & [K_{42}]_{3\times3} & [K_{43}]_{3\times3} & [K_{44}]_{3\times3}\end{bmatrix}_{12\times12}$$

In which

$$K_{ij}^{11}=\int_{x_a}^{x_b}\left\{k_s(A_{55}+A_{66})\frac{\delta\psi_i}{\delta x}\frac{\delta\psi_j}{\delta x}+\sum_{k=1}^{n_b}K_{xx}^{bk}\psi_i\psi_j\Delta(x-x_{bk})\right\}dx$$

$$K_{ij}^{12}=\int_{x_a}^{x_b}\left\{\sum_{k=1}^{n_b}K_{xy}^{bk}\psi_i\psi_j\,\vartriangle\!\left(x-x_{bk}\right)\right\}dx$$

$$K_{ij}^{13}=\int_{x_a}^{x_b}-\frac{1}{2}k_sB_{16}\frac{\delta\psi_i}{\delta x}\frac{\delta\psi_j}{\delta x}dx$$

$$K_{ij}^{14}=\int_{x_a}^{x_b}-k_s(A_{55}+A_{66})\frac{\delta\psi_i}{\delta x}\psi_jdx$$

$$K_{ij}^{12}=\int_{x_a}^{x_b}\left\{\sum_{k=1}^{n_b}K_{yx}^{bk}\psi_i\psi_j\,\vartriangle\!\left(x-x_{bk}\right)\right\}dx$$

$$K_{ij}^{22}=\int_{x_a}^{x_b}\left\{k_s(A_{55}+A_{66})\frac{\delta\psi_i}{\delta x}\frac{\delta\psi_j}{\delta x}+\sum_{k=1}^{n_b}K_{yy}^{bk}\psi_i\psi_j\Delta(x-x_{bk})\right\}dx$$

$$K_{ij}^{23}=\int_{x_a}^{x_b}k_s(A_{55}+A_{66})\frac{\delta\psi_i}{\delta x}\psi_jdx$$

$$K_{ij}^{24} = \int_{x_a}^{x_b} -\frac{1}{2} k_s B_{16} \frac{\delta\psi_i}{\delta x} \frac{\delta\psi_j}{\delta x} dx$$

$$k_{ij}^{31} = \int_{x_b}^{x_b} -\frac{1}{2} k_s B_{16} \frac{\delta\psi_i}{\delta x} \frac{\delta\psi_j}{\delta x} dx$$

$$K_{ij}^{32} = \int_{x_a}^{x_b} k_s (A_{55} + A_{66}) \psi_i \frac{\delta\psi_j}{\delta x} dx$$

$$K_{ij}^{33} = \int_{x_a}^{x_b} \left\{ k_s \left(A_{55} + A_{66} \right) \psi_i \psi_j + D_{11} \frac{\delta\psi_i}{\delta x} \frac{\delta\psi_j}{\delta x} \right\} dx$$

$$K_{ij}^{34} = \int_{x_a}^{x_b} \frac{1}{2} k_s B_{16} \frac{\delta\psi_i}{\delta x} \psi_j - \frac{1}{2} k_s B_{16} \psi_i \frac{\delta\psi_i}{\delta x} dx$$

$$K_{ij}^{41} = \int_{x_a}^{x_b} -k_s (A_{55} + A_{66}) \psi_i \frac{\delta\psi_j}{\delta x} dx$$

$$K_{ij}^{42} = \int_{x_a}^{x_b} -\frac{1}{2} k_s B_{16} \frac{\delta\psi_i}{\delta x} \frac{\delta\psi_j}{\delta x} dx$$

$$K_{ij}^{43} = \int_{x_a}^{x_b} -\frac{1}{2} k_s B_{16} \frac{\delta\psi_i}{\delta x} \psi_j + \frac{1}{2} k_s B_{16} \psi_i \frac{\delta\psi_j}{\delta x} dx$$

$$K_{ij}^{44} = \int_{x_a}^{x_b} \left\{ k_s \left(A_{55} + A_{66} \right) \psi_i \psi_j + D_{11} \frac{\delta\psi_i}{\delta x} \frac{\delta\psi_j}{\delta x} \right\} dx$$

3.3 Fluid-film Bearing Modeling

The journal bearing finite element model is developed based on the classical Reynolds equation for oil-lubricated journal bearings. Journal eccentricities in the vertical and horizontal directions are expressed as e_x and e_y respectively. The eccentricity ratio is defined as ε= e/c, where $e^2 = e^2{}_x + e^2{}_y$. From Reynolds equation the relationships for the stiffness and damping are determinedwhich is given as.

$$K_{ij} = \left(\frac{F}{h}\right)K_{ij}^*$$

$$C_{ij} = \left(\frac{F}{\Omega h}\right)C_{ij}^*$$

Where K_{ij}^* and C_{ij}^* are the dimensionless stiffness and damping coefficients [11] respectivelyand are given by,

$$K_{11}^* = \left[2\pi^2 + \left(16 - \pi^2\right)\varepsilon^2\right]\psi\left(\varepsilon\right)$$

$$K_{12}^* = \frac{\pi}{4}\left(\frac{\pi^2 - 2\pi^2\varepsilon^2 - \left(16 - \pi^2\right)\varepsilon^4}{\varepsilon\left(1 - \varepsilon^2\right)^{1/2}}\right)\psi\left(\varepsilon\right)$$

$$K_{21}^* = \frac{\pi}{4}\left(\frac{\pi^2 + \left(32 + \pi^2\right)\varepsilon^2 + \left(32 - 2\pi^2\right)\varepsilon^4}{\varepsilon\sqrt{\left(1 - \varepsilon^2\right)}}\right)\psi\left(\varepsilon\right)$$

$$K_{22}^* = \left(\frac{\pi^2 + (32 + \pi^2)\varepsilon^2) + (32 - 2\pi^2\ )\varepsilon^4}{(1 - \varepsilon^2)}\right)\psi(\varepsilon)$$

$$C_{11}^* = \pi\frac{(1 - \varepsilon^2)^{1/2}}{2\varepsilon}\left[\pi^2 + (2\pi^2 - 16)\varepsilon^2\right]\Psi(\varepsilon)$$

$$C_{12}^* = C_{21}^* = -\left[2\pi^2 + (4\pi^2 - 32)\varepsilon^2\right]\Psi(\varepsilon)$$

$$C_{22}^* = \frac{\pi}{2}\frac{\pi^2 + (48 - 2\pi^2)\varepsilon^2 + \pi^2\varepsilon^4}{\varepsilon(1 - \varepsilon^2)^{1/2}}\Psi(\varepsilon)$$

Where,

$$\psi(\varepsilon) = \frac{4}{\left[\pi^2 + (16 - \pi^2)\varepsilon^2\right]^{3/2}}$$

Chapter 4

Results and discussion

4.1 Numerical Example

This investigation discusses the results of the analyses performed in this project work. For composite rotor-shaft system mechanical properties and geometric dimensions are shown in Table 1 and Table 2. Based on the above formulation a MATLAB program has been proposed. The recent developed code has been validated with the results available in literatures. Various type of analysis for composite shaft have been studied and presented in the following sections. Firstly, it is shown that the variation of dimensionless stiffness and dimensionless damping with change in eccentricity variation and secondly, variation of dimensionless stiffness and dimensionless damping with change in rotor speed variation. Thirdly, it is shown that how the v-displacement and w- displacement changes for different oil thickness values. Finally, Campbell diagram is studied by plotting graph between natural frequency vs. rotor speed.

4.2 Validation

In order to verify the FE developed code the following dimensions and mechanical properties were considered from [6] (details of which are given in Table 1 and Table 2). In order to convergence study of the result, it has been observed that result from the present code has been achieved an excellent agreement with the already published results[6] and thus validates the correctness of the developed code. It is shown in figures.

4.3 Dynamic analysis of rotor shaft

The shaft being analyzed is made of the graphite epoxy material and has the lamination $\left(90/45/-45/0/0/0/0/0/0/90\right)$. Composite shaft is supported by bearings which are modeled as springs and dampers and has discrete isotropic rigid disk attached to it. By using these properties we have plotted graph between dimensionless stiffness vs. eccentricity, dimensionless damping vs. eccentricity, dimensionless stiffness vs. rotor speed and dimensionless damping vs. rotor speed.

Table 1. Material properties used in analysis

Material properties	Graphite/Epoxy
E_{11} (GPa)	139.0
E_{23} (GPa)	11.0
$G_{12}=G_{13}$ (GPa)	6.50
G_{23} (GPa)	3.78
shear correction factor	0.56
density (Kg/m^3)	1578.0

Table 2. Dimension of shaft, disk and properties of bearing

	Shaft	Disk	Bearing
Length(m)	0.72	--	--
Inner diameter(m)	.028	--	--
Outer diameter(m)	.048	--	--
Mass(Kg)	--	2.4364	--
Polar moment of inertia(Kgm^2)	--	0.3778	--
Diameter moment of inertia(Kgm^2)	--	0.1901	--
$K_{yy}=K_{zz}$(10^7 N/m)	--	--	1.75

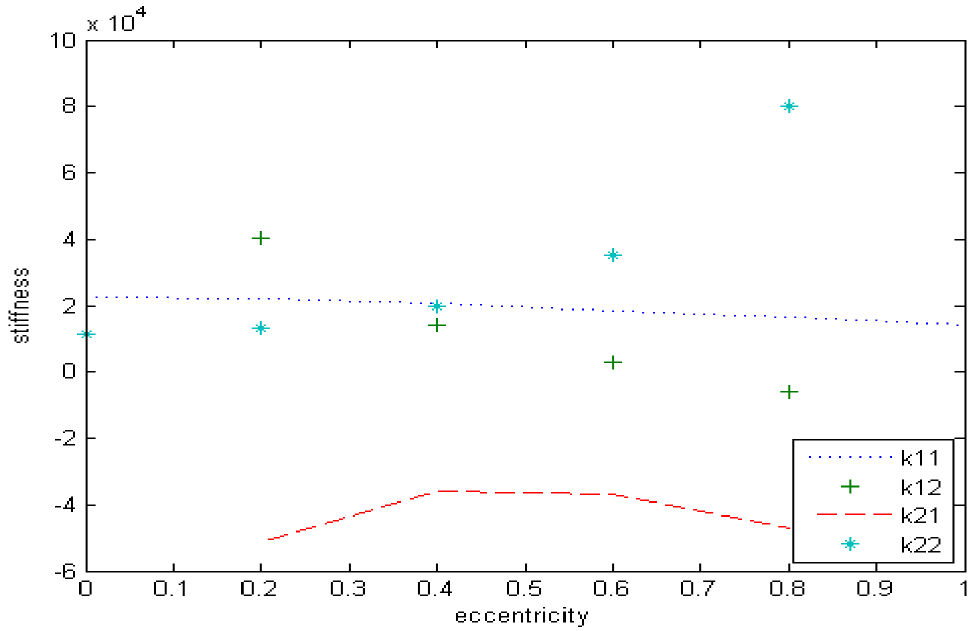

Figure 2. Dimensionless stiffness vs. eccentricity

The above graph between stiffness and eccentricity brings out the variation of k_{11}, k_{12}, k_{21} and k_{22}. Stiffness k_{22} increases steadily with eccentricity whereas k_{12} decreases. k_{21} first raises then gets stable then decreases.

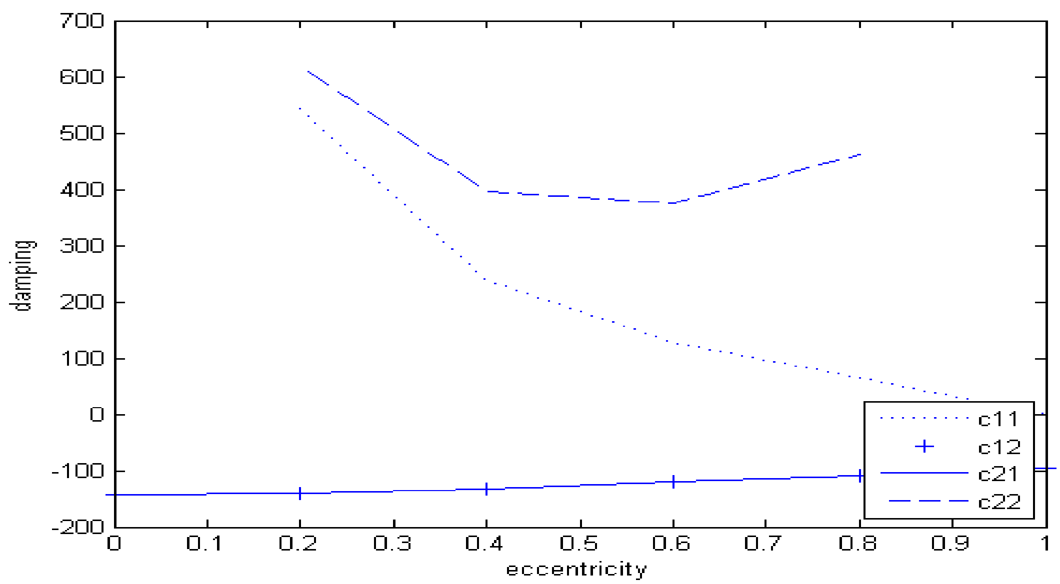

Figure 3. Dimensionless damping vs. eccentricity

The above graph plotted between damping and eccentricity depicts the variation of different damping coefficient with respect to eccentricity ratio. In the above graph, c_{11} decreases abruptly with increasing eccentricity ratio, c_{22} first decreases then becomes stable and then again increases. c_{21} and c_{12} increases steadily and coincide as they hold same equation.

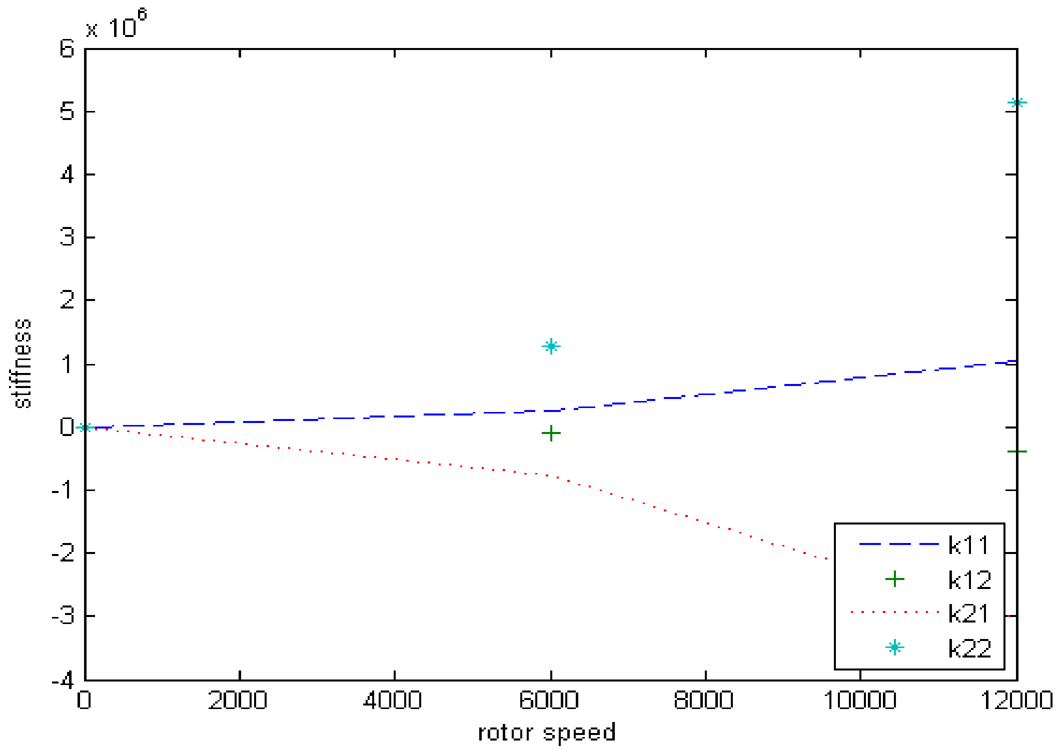

Figure 4. Dimensionless stiffness vs. rotor speed

Above figure show the plots of the dimensionless stiffness coefficients versus the eccentricity ratio respectively. In the above graph k_{21} and k_{11} varies with rotor speed but k_{21} varies more steadily hence it can be used for analysis.These coefficients can now be inserted into the equation of motion of the journal and then a vibration analysis can be performed.

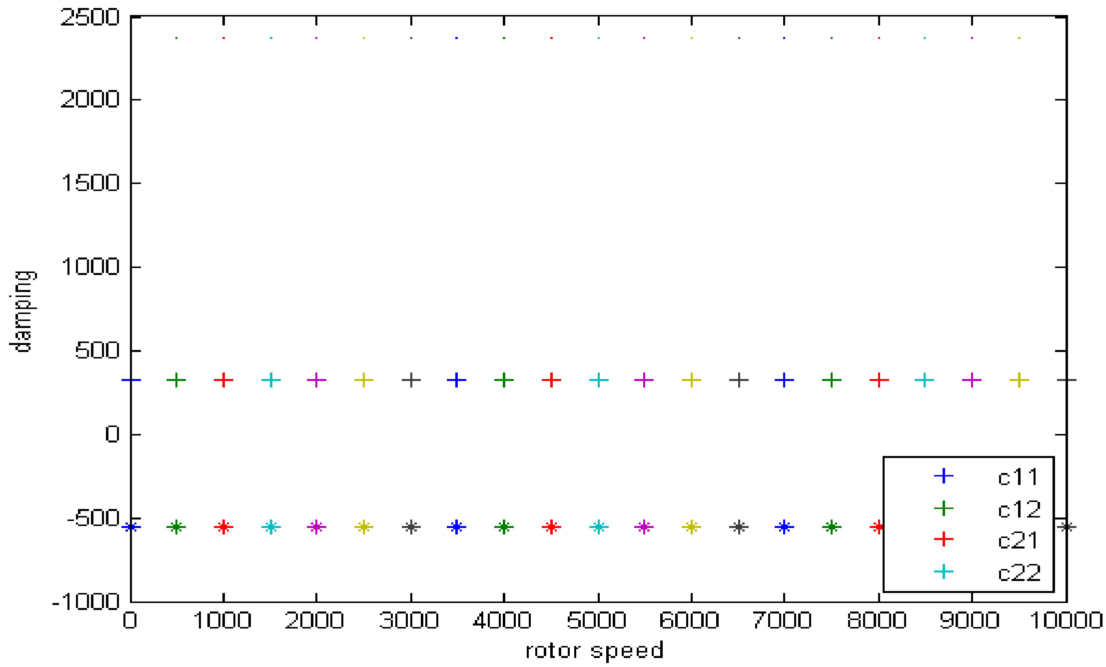

Figure 5. Dimensionless damping vs. rotor speed

4.4 Vibration response of rotor shaft system with oil film thickness.

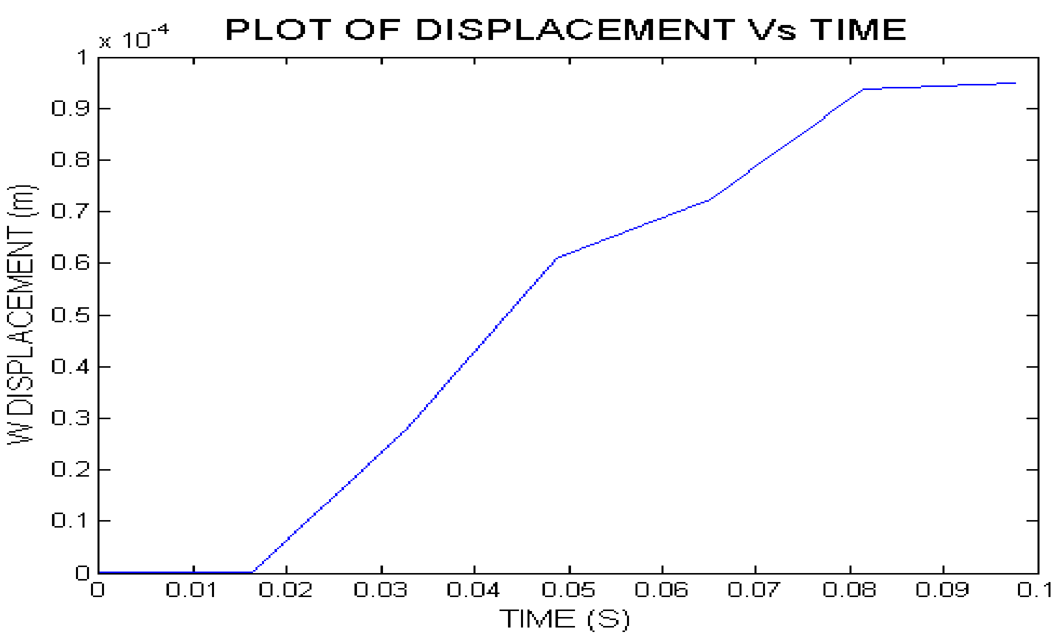

Figure 6. Displacement in *W* direction vs time

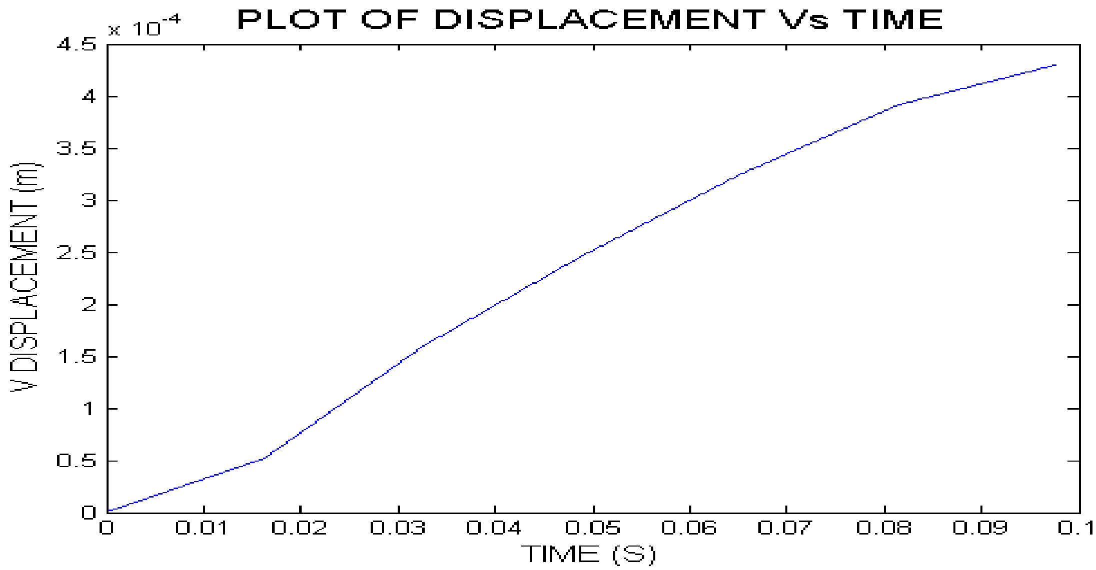

Figure 7. Displacement in *V* direction vs. time

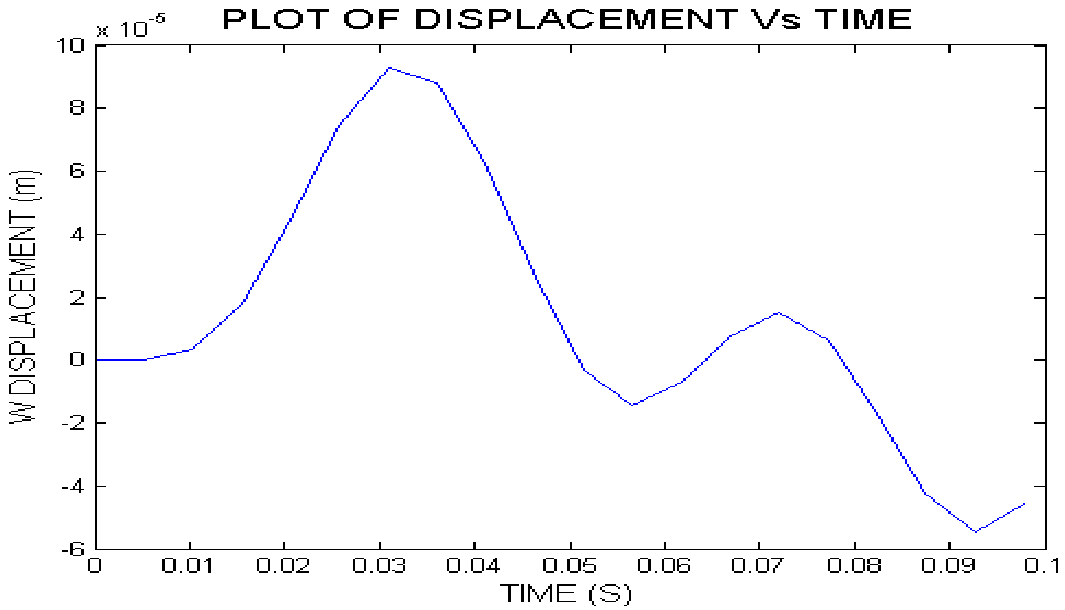

Figure 8. Displacement in *W*direction vs. time

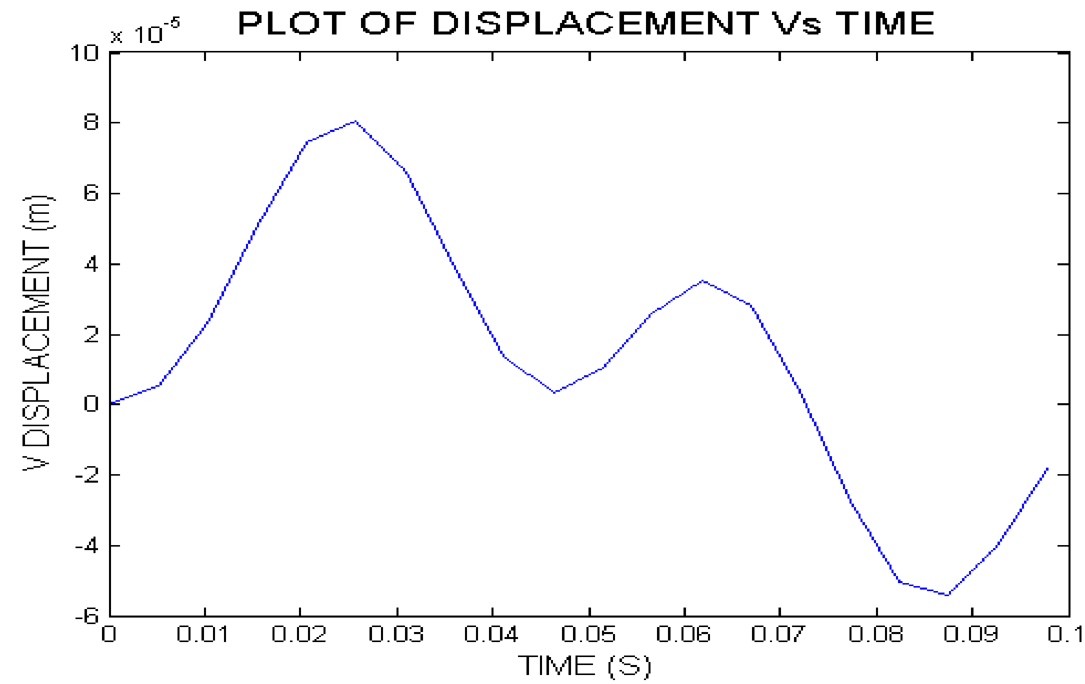

Figure 9. Displacement in V direction vs. time

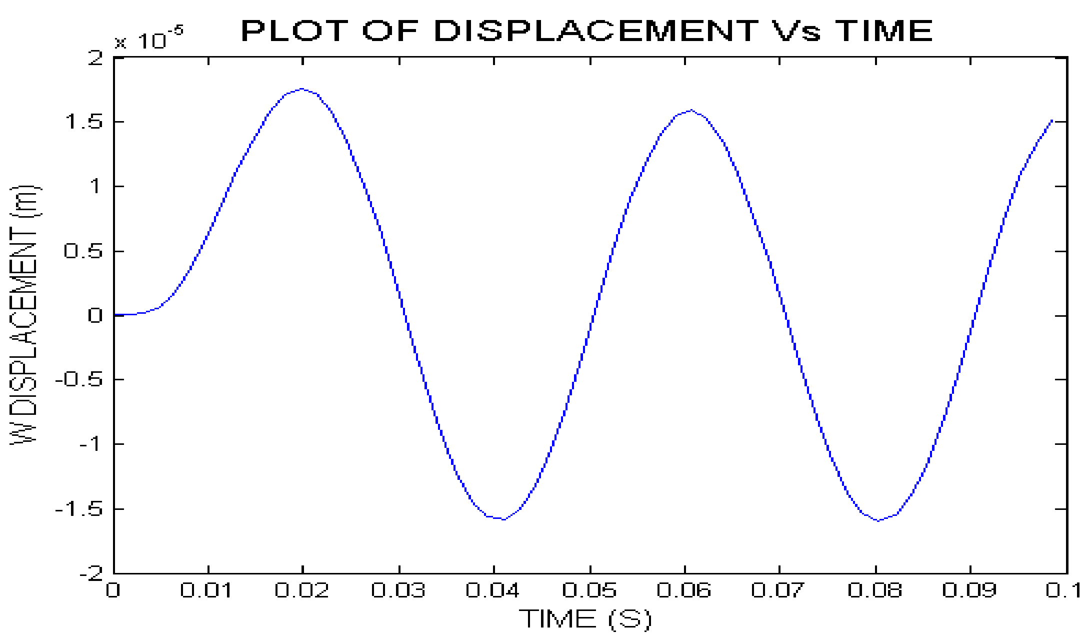

Figure 10**.** Displacement in *W* direction vs. time

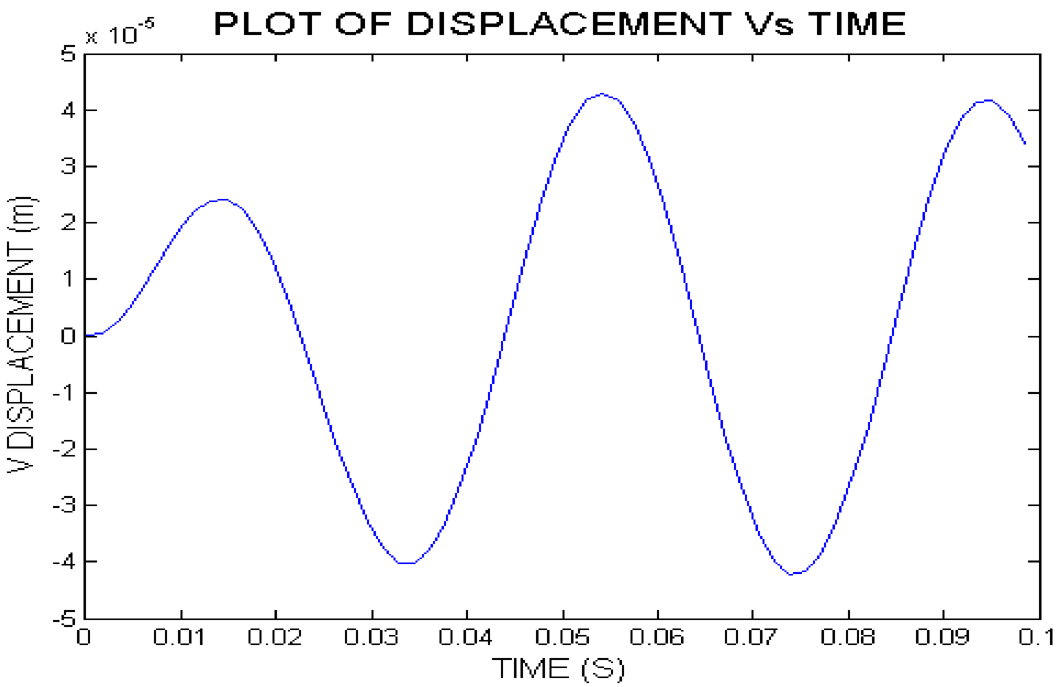

Figure 11. Displacement in *V* direction vs. time

Figure 6 and figure 7 show the maximum displacement for oil film thickness h_0 =0.02mm (higher) and figure 10 and figure 11 show the minimum displacement for oil film thickness $h_0 = 0.0002mm$ (lowest).

4.5 Vibration response of rotor shaft system

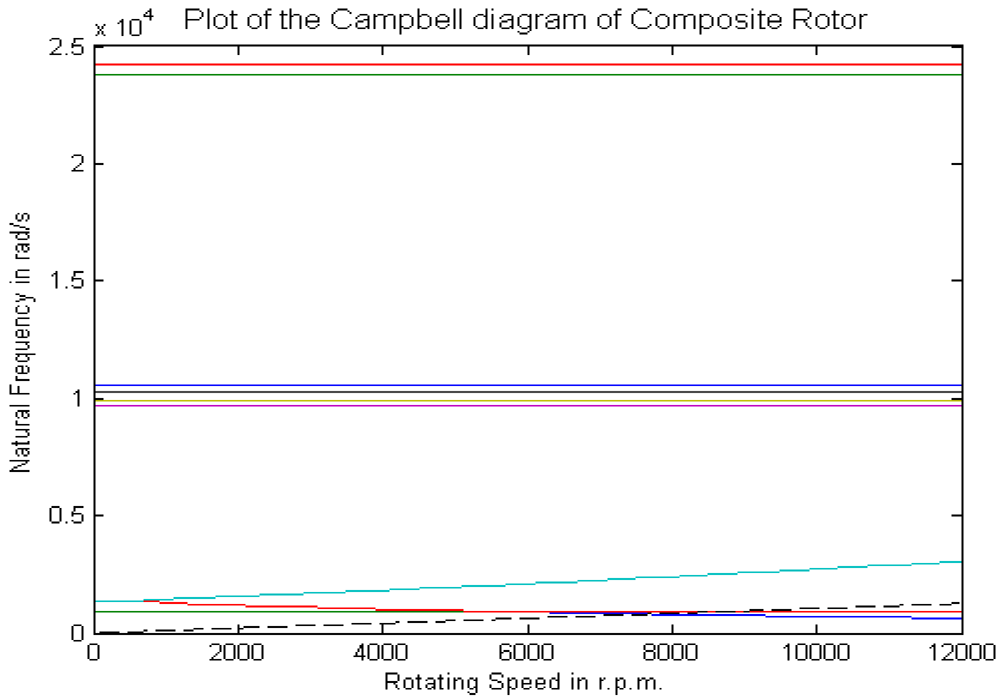

Figure 12. Campbell diagram of the rotor shaft system

The above figure shows the Campbell diagram containing the frequencies of the first five pairs of bending whirling modes of the composite system. When the whirl speed and rotor speed are in the same direction then we get forward modes and else we get backward modes. The intersection point of the line γ=1 with the whirling frequency curves indicate the speed at which the shaft will vibrate violently i.e. the critical speed. The evolutions of the natural frequencies corresponding to a mode are drawn in function of the rotation speed of the shaft.

Chapter 5

Conclusions & scope of future work

5.1 Conclusions

In this work, an analytical method for computing the stiffness and damping coefficients of short bearing has been summarized. The concept of stiffness and damping coefficients for journal bearings has been proven and modern rotor dynamics calculations for unbalance response, damped natural frequencies and stability are based on this concept. The FEM procedure can generate important technical information about the behavior of composite shafts supported on oil lubricated journal bearings that can help technicians, engineers in the analysis and development of industrial rotating machines. The rotor bearing model which accounts for the shaft model based on the Timoshenko beam theory and the bearing coefficients. These coefficients can be predicted for journal bearing with any value of operating load condition. The vibrational amplitude is less in journal bearing shaft system than simply supported shaft. In other words the vibration is predominant in case of simply supported shaft than journal bearing shaft system.

5.2 Scope of future work

1. The study could be performed on various composite materials available of better strength, cheaper and less weight as compared to Graphite/Epoxy composite material.

2. Better performance could be optimized through using different types of bearings having good stiffness coefficient and damping coefficient (like foil bearings).

3. Various analyses could be carried out to obtain more results and study various effects on the nonlinear rotor-shaft system.

4. Temperature effect can be analyzed for composite rotor shaft system.

References

[1] Isaac M.daniel,Engineering Mechanics of Composite Materials, new york oxford university press, (1994).

[2] Tudu Prakash, Processing and Characterization of Natural Fiber Reinforced Polymer Composites, nit Rourkela (2009), 2-13.

[3] Sanjay K.Mazumdar,Composites manufacturing: materials, product, and process engineering, (2001).

[4] Balupari Raja Shekar, Validation of Finite Element Program for Journal Bearings – Static and Dynamic Properties, Paper 325,(2004).

[5] Luis San Andres, Hydrodynamic Fluid Film Bearings and Their Effect on the Stability of Rotating Machinery, Turbomachinery Laboratory, Texas A&M University, (2006).

[6] Min-Yung Chang, Jeng-Keag Chen, Chih-Yung Chang, A Simple Spinning Laminated Composite Shaft Model, International Journal of Solids and Structures ,(2004), 637–662.

[7] J.Schiffmann and Z. S. Spakovszky, Foil Bearing Design Guidelines for Improved Stability,January 2013, Vol. 135.

[8] Minhui He, James M.Byrne and C.Hunter Cloud,Fundamentals of Fluid Film Journal Bearing Operation and Modeling, (2005), 156-174.

[9] Nelson H.D.andMcVaugh J.M, The Dynamics of Rotor-Bearing Systems Using Finite Elements, Journal of Engineering for Industry , (May 1976), 593-600.

[10] Miranda W.M.,Faria M.T.C, Lateral Vibration Analysis of Flexible Shafts Supported on Elliptical Journal Bearings, Tribol Lett,(2012), 217-227.

[11] Yucel Ugur, Calculation of Dynamic Coefficients For Fluid Film Journal Bearings, Journal of Engineering sciences,(2005), 335-343.