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UNBALANCED RESPONSE OF ROTOR USING ANSYS PARAMETRIC DESIGN FOR DIFFERENT BEARINGS

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ABSTRACT

This paper presents an alternative procedure called harmonic analysis to identify frequency of a system through critical speed, amplitude and phase angle plots using ANSYS. The unbalance that exists in any rotor due to eccentricity has been used as excitation to perform such an analysis. ANSYS parametric design language has been implemented to achieve the results. In first case, two un damped isotropic bearings were located at positions four and six. In second case, two symmetric orthotropic bearings were located at positions four and six, and in third case two identical fluid film bearings were located at positions four and six. The result shows to finding a critical speed of rotor using different bearing. The accuracy of the model and the solution technique has been demonstrated by comparison with results of previous publications. Very good agreement has been obtained.

KEYWORDS: *ANSYS Parametric Design Language, unbalanced response, Critical Speed, frequency, undammed isotropic bearings, symmetric orthotropic bearings, fluid film bearings etc.*

I. INTRODUCTION

Vibration control of turbo machinery is very important for the integrity of industrial plants. In this regard it is very important to predict the dynamic behavior of rotating machinery which operates above the first critical speed accurately. In fact, rotating motion and critical speeds are design criteria of rotating machinery and play an important role in diagnosis and control of rotors.

Harmonic response analysis is a technique used to determine the steady-state response of a linear structure to loads that vary sinusoidally (harmonically) with time and is used to predict the sustained dynamic behavior of structures to consistent cyclic loading. Thus, it can be verified whether or not a machine design will successfully overcome resonance, fatigue, and other harmful effects of forced vibrations.

ANSYS is a common tool for finite element analyses and it is widely used in research and development of rotating machinery. It has rotating beam elements such as BEAM188, BEAM4 element and PIPE16 elements which can be used to model the shaft. For a rotating beam element, the gyroscopic effect can be taken into consideration. Also, the effects of rotary inertia, shear deformation, axial load and internal damping can be included. However, this paper like most others doesn’t set specific elements for modeling rotating disks and bearings . This paper shows how COMBIN14, COMBI214 and MATRIX27 element is used to model rotating disks and bearings. The geometry of this arbitrary element is undefined, but its mechanism can be specified by stiffness, damping or mass matrix. Specifications for the elements, descriptions and the critical speed calculations of rotor-bearing system are included in this paper.

The use of flexible supports was strongly developed during last few years due to some practical advantages offered by this design. A number of authors [1,6,7,8,10,11,12,17,20,21,22] have addressed the problem, highlighted the importance of this component, and triedto use experimental procedures and data in order to undertake the influence of the support flexibility of rotor machinery. Effectively, the dynamic behavior of rotating machines may be drastically affected by the characteristics of the support flexibility of rotor machinery. Then, one of the main objectives of the researchers and designers was to be able to obtain fundamental mathematical models, adequate to the observed physical phenomena, in order to predict numerically the dynamic behavior of rotor systems and the influence of the support flexibility of rotors. In recent year there has been an important research activity in the field of modeling and analysis of the dynamic behavior of rotating machinery in order to adjust some system parameters and to obtain the most suitable design within the speed range of interest. Then, the utilization of finite element models in the area of rotor dynamics was applied to develop suitable models and has yielded highly successful results [1, 10]. These numerical models are now used to design machinery to operate within acceptable limits.

Taplak[9] in his paper studied a program named Dynrot was used to make dynamic analysis and the evaluation of the results. For this purpose, a gas turbine rotor with certain geometrical and mechanical properties was modeled and its dynamic analysis was made by Dynrot program. Gurudatta[4] in his paper presented an alternative procedure called harmonic analysis to identify frequency of a system through amplitude and phase angle plots. The unbalance that exists in any rotor due to eccentricity has been used as excitation to perform such an analysis. ANSYS parametric design language has been implemented to achieve the results

Sinou[16] investigated the response of a rotor’s non- linear dynamics which is supported by roller bearings. He studies on a system comprised of a disk with a single shaft, two flexible bearing supports and a roller bearing. He found that the reason of the exciter is imbalance. He used a numerical method named Harmonic Balance Method for this study. Chouskey[13] et.al studied the influences of internal rotor material damping and the fluid film forces (generated as a result of hydrodynamic action in journal bearings) on the modal behavior of a flexible rotor-shaft system. It is seen that correct estimation of internal friction, in general, and the journal bearing coefficients at the rotor spin-speed are essential to accurately predict the rotor dynamic behaviour. This serves as a first step to get an idea about dynamic rotor stress and, as a result, a dynamic design of rotors.

Whalley and Abdul-Ameer[18], calculated the rotor resonance, critical speed and rotational frequency of a shaft that its, diameter changes by the length, by using basic harmonic response method. Gasch[15], investigated the dynamic behavior of a Laval (Jeffcott) rotor with a transverse crack on its elastic shaft, and developed the non-linear motion equations which gave important clues on the crack diagnosis.

Das et al[2]. aimed to develop an active vibration control scheme to control the transverse vibrations on the rotor shaft arising from imbalance and they performed an analysis on the vibration control and stability of a rotor- shaft system which has electromagnetic exciters.

Villa et al.,[5] studied the non-linear dynamic analysis of a flexible imbalanced rotor supported by roller bearings. They used Harmonic Balance Method for this purpose. Stability of the system was analyzed in frequency term with a method based on complexity. They showed that Harmonic Balance Method has realized the AFT strategy and harmonic solution very efficiently. Lei and Palazzolo[19] have analyzed a flexible rotor system supported by active magnetic bearings and synthesized the Campbell diagrams, case forms and eigen values to optimize the rotor-dynamic characteristics and obtained the stability at the speed range. They also investigated the rotor critical speed, case forms, frequency responses and time responses.

This paper presents an alternative procedure called harmonic analysis to identify frequency of a system through critical speed, amplitude and phase angle plots using ANSYS. The unbalance that exists in any rotor due to eccentricity has been used as excitation to perform such an analysis. ANSYS parametric design language has been implemented to achieve the results. In first case, two undamped isotropic bearings were located at positions four and six. In second case, two symmetric orthotropic bearings were located at positions four and six, and in third case two identical fluid film bearings were located at positions four and six.

II. METHODOLOGY

A. Model

The model considered is a Nelson rotor[14] which is a 0.355(m) long overhanging steel shaft of 14 different cross1sections .The shaft carries a rotor of mass 1.401(kg) and eccentricity 0.635(cm) at 0.0889(m) from left end and is supported by firstly two bearings at a distance of 0.1651(m) and 0.287(m) from the left end respectively. Six stations are considered during harmonic analysis as shown in Fig.1, where station numbers denote different nodes in the model1(1)Left extreme of shaft, (2)Disc, (3)First bearing node, (4) Between the two bearings, (5) Second bearing node and (6)Right extreme of shaft.

A density of 7806 kg/m^3 and elastic modulus 2.078E11 n/m^2 were used for the distributed rotor and a concentrated disk with a mass of 1.401 kg , polar inertia 0.002 kg.m^2 and diametral inertia 0.00136 kg.m^2 was located at station five. The following three cases of bearings were analyzed:

a) Undamped isotropic bearings
b) Symmetric orthotropic bearings
c) Fluid film bearings.

B. Geometric Modeling and Finite Element Modeling Using APDL

In critical speed calculations of rotor-bearing systems, BEAM188, COMBIN14, COMBI214 and MATRIX27 elements are adopted.

1) Beam188

The multisection shaft has been modeled in ANSYS using Beam 188 which is a linear/quadratic two node beam element in three dimensions with six degrees of freedom at each node: translations in the nodal x, y and z directions and rotations about nodal x, y and z axes. This element facilitates the meticulous definition of all the cross1sections of the shaft. The rotor and the its Real Constants include: AREA, SPIN , ADDMAS. SPIN is an important item in the critical speed calculations, which defines the rotational speed of the shaft. ADDMAS defines added masses along the shaft. The nodes, elements, material properties, real constants, boundary conditions and other physical system1defining features that constitute the model have been created by exclusively using APDL commands such as ET, MAT, K, N, LSTR, R, RMORE, LATT, LESIZE and E.

2) Mass21

The disk of the rotor has been modeled using element MASS21. The its Real Constants include: IZZ , IYY , IXX.

3) Combin14

The undammped isotropic bearing has been modeled in ANSYS using COMBIN14 which has longitudinal or torsional capability in 1-D, 2-D, or 3-D applications. The longitudinal spring-damper option is a uniaxial tension-compression element with up to three degrees of freedom at each node: translations in the nodal x, y, and z directions. No bending or torsion is considered. The torsional spring-damper option is a purely rotational element with three degrees of freedom at each node: rotations about the nodal x, y, and z axes.

4) Combi214

The symmetric orthotropic bearing has been modeled in ANSYS using COMBI214 which has longitudinal as well as cross-coupling capability in 2-D applications. It is a tension-compression element with up to two degrees of freedom at each node: translations in any two nodal directions (x, y, or z). COMBI214 has two nodes plus one optional orientation node. No bending or torsion is considered.

5) Matrix27

The fluid film bearing has been modeled in ANSYS using MATRIX27. Its represents an arbitrary element whose geometry is undefined but whose mechanism can be specified by stiffness ,damping , or mass matrix coefficients. The matrix is assumed to relate two nodes, each with six degrees of freedom per node: translations in the nodal x, y and z directions and rotations about the nodal x, y and z axes. There are three options to use the MATRIX27 to define coefficients, which is very useful to model linear cross coupling bearing characteristics and gyroscopic damping matrix for rotating disks.

III. SOLUTION AND POST/ PROCESSING

Once the finite element model has been prepared, harmonic analysis is performed by applying an unbalance force at the rotor (assuming an eccentricity of 0.00635(mm)). The system is solved using frontal solver to find response of the system in terms of amplitude and phase angle plots. Response is determined at 6 stations (1) Left extreme of shaft, (2)Disc, (3)First bearing node, (4) Between the two bearings, (5) Second bearing node and (6) Right extreme of shaft. The resulting graphs are exported as jpeg files.
In general, any rotating critical speed is associated with high vibration amplitude. When the rotating speed is close to or away from a critical speed, vibration amplitude increases or decreases abruptly and phase becomes unsteady as figure 1 shows. For rotating machinery, rotor unbalance mass is a kind of synchronous excitation, and induces vibration.

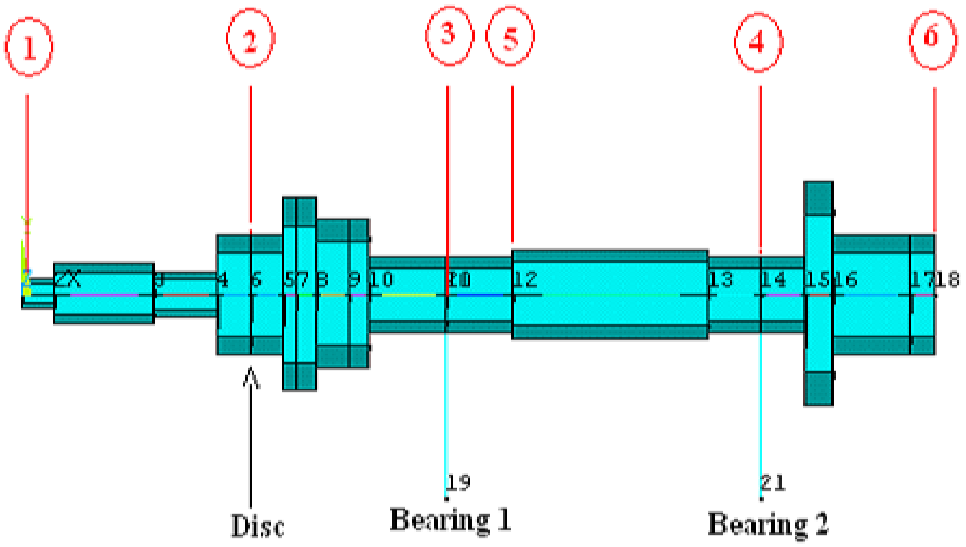

Figure 1: Model of Nelson rotor with various sections, disc and bearings. Numbers in red indicate station numbers

The Harmonic Response Analysis module of ANSYS is applied to calculate unbalance synchronous response of the rotor-bearing system, and a Bode plot can be obtained .From the Bode plot, rotating speeds with peak vibration are defined as critical speeds.

IV. RESULTS AND DISCUSSION

4.1 Case 1: Undamped Isotropic Bearings

The shaft is supported by two identical undamped isotropic bearings of stiffness Kxx = Kyy = 4.378E7 N/m. As explained earlier, detailed harmonic analyses have been carried out on the considered model to study the unbalance response of the system. The unbalance response for a disk mass center eccentricity of 0.0635cm at station two was determined for speed range 4800-28800 rpm. The first critical speed is found around 266 Hz. Fig. 2 through 7 displays the variation of amplitude of vibration of the system at the six identified stations respectively. The maximum value of amplitude obtained was 0.784223E-7. It can be observed that the amplitude reaches a maximum value at one particular excitation frequency. Fig.8 displays the typical variation of phase angle with excitation frequency at all the stations. The results are compared to that published by Nelson for a first critical speed. Very Good agreement is obtained.

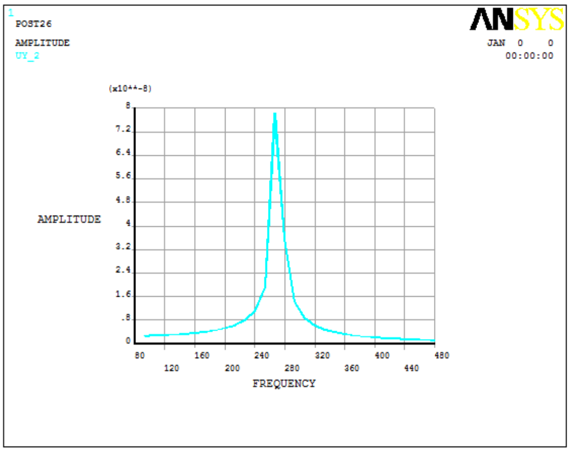

Figure 2: Variation of amplitude of vibration (m)(on Y axis) at station 1 with excitation frequency (Hz)(on X axis

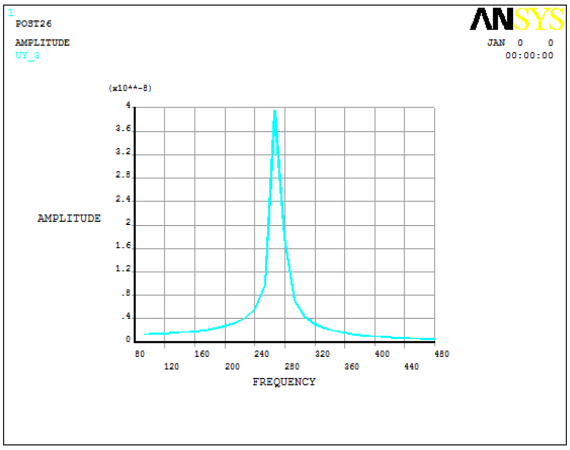

Figure 3: Variation of amplitude of vibration (m)(on Y axis) at station 2 with excitation frequency (Hz)(on X axis

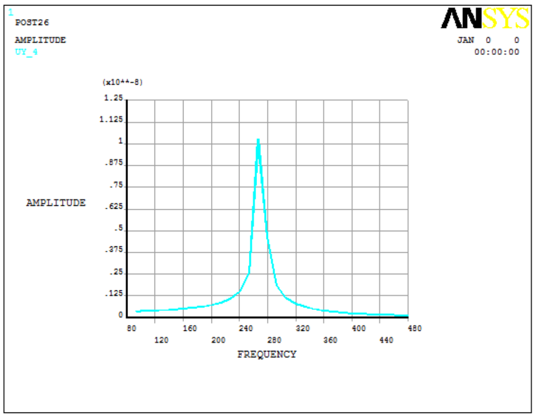

Figure 4:Variation of amplitude of vibration (m)(on Y axis) at station 3 with excitation frequency (Hz)(on X axis)

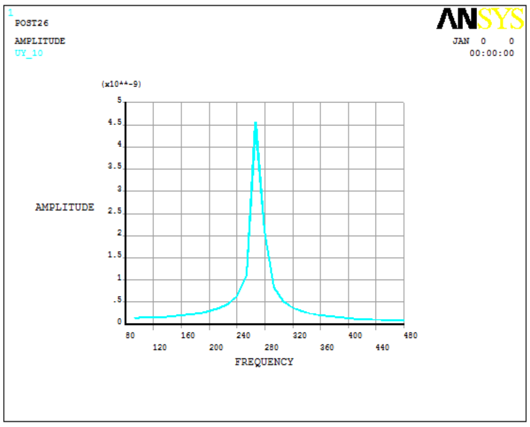

Figure 5: Variation of amplitude of vibration (m)(on Y axis) at station 4 with excitation frequency (Hz)(on X axis)

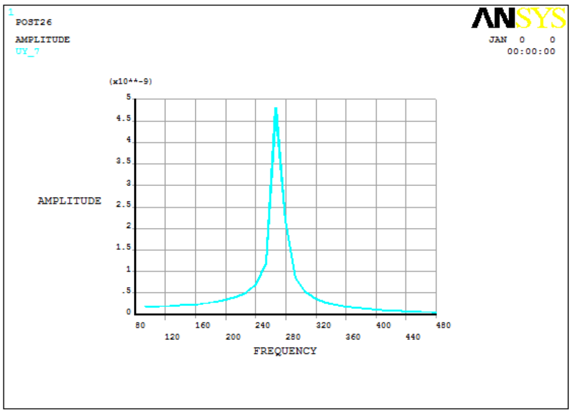

Figure 6: Variation of amplitude of vibration (m)(on Y axis) at station 5 with excitation frequency (Hz)(on X axis)

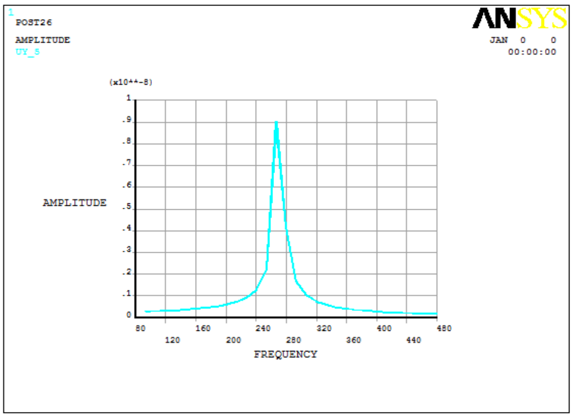

Figure 7:Variation of amplitude of vibration (m)(on Y axis) at station 6 with excitation frequency (Hz)(on X axis)

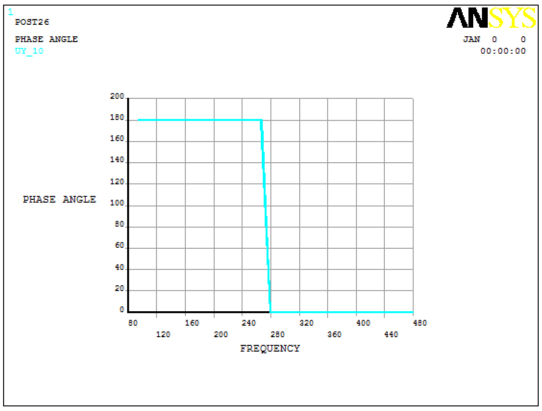

Figure 8: Typical variation of phase angle with excitation frequency.

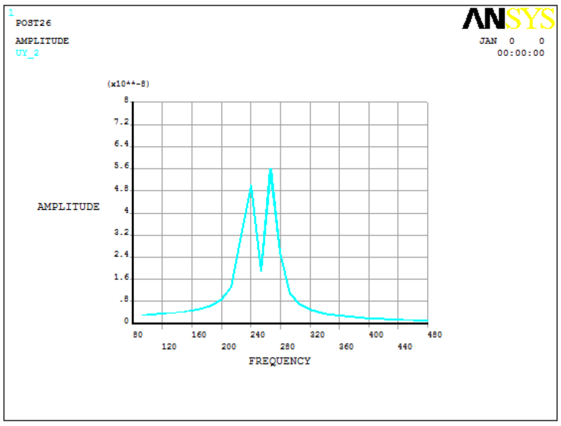

Figure 9:Variation of amplitude of vibration (m)(on Y axis) at station 1 with excitation frequency (Hz)(on X axis)

4.2 Case 2: Symmetric Orthotropic Bearings

The shaft is supported by two Symmetric Orthotropic Bearings of stiffness components of the bearings are Kxx=Kyy= 3.503E7 N/m and Kxy=Kyx = -8.756E6 N/m. The unbalance response for a disk mass center eccentricity of 0.0635cm at station two was determined for speed range 4800-28800 rpm. The first critical speed is found around 240 Hz. Fig. 9 through 14 display the variation of amplitude of vibration of the system at the six identified stations respectively. The maximum value of amplitude obtained was 0.50077E-7. It can be observed that the amplitude reaches a maximum value at one particular excitation frequency. Fig. 15 displays the typical variation of phase angle with excitation frequency at all the stations. The numerical results of the first critical speeds are well compared to those published by Nelson. In this case also very good agreement is obtained.

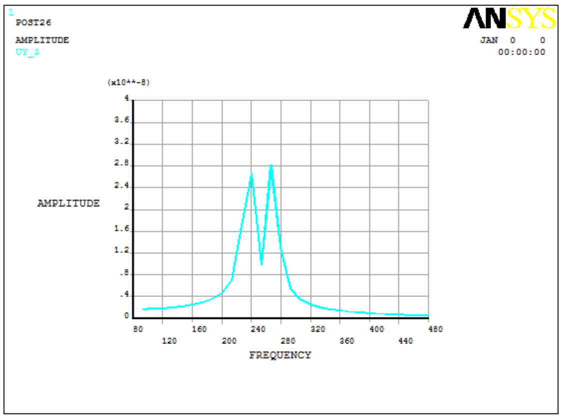

Figure 10:Variation of amplitude ofvibration (m)(on Y axis) at station 2 with excitation frequency (Hz)(on X axis)

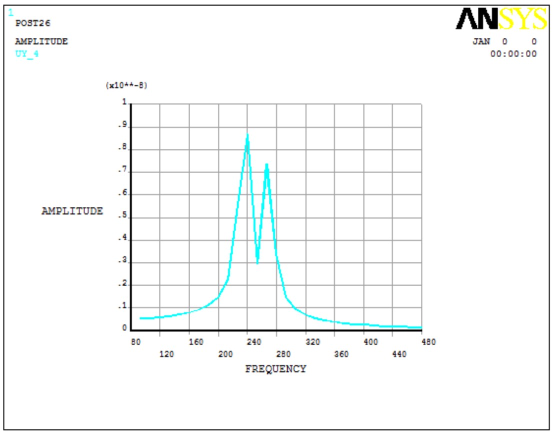

Figure 11: Variation of amplitude of vibration (m)(on Y axis) at station 3 with excitation frequency (Hz)(on X axis)

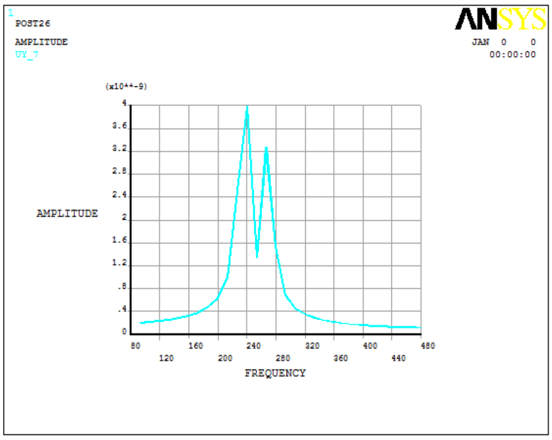

Figure 12: Variation of amplitude of vibration (m)(on Y axis) at station 4 with excitation frequency (Hz)(on X axis)

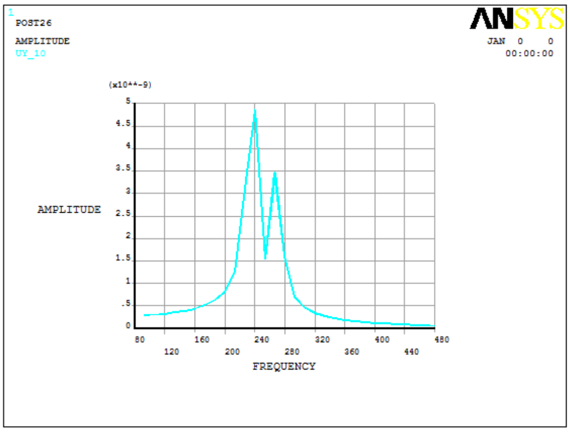

Figure 13:Variation of amplitude of vibration (m)(on Y axis) at station 5 with excitation frequency (Hz)(on X axis)

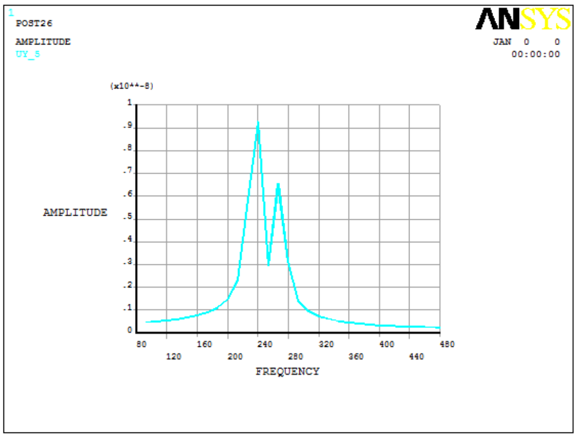

Figure 14:Variation of amplitude of vibration (m)(on Y axis) at station 6 with excitation frequency (Hz)(on X axis)

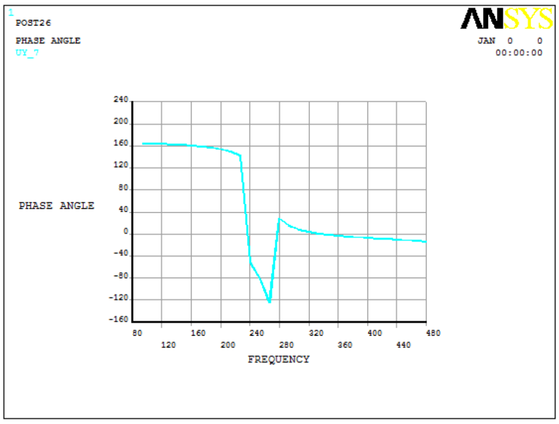

Figure 15: Typical variation of phase angle with excitation Frequency.

4.3 Case 3: Fluid film bearings

The shaft is supported by two fluid film bearings of stiffness component the bearings are Kxx = Kyy = 4.378E7 N/m. while damping components are Czz=Cyy=1752(Ns/m). The unbalance response for a disk mass center eccentricity of 0.0635cm at station two was determined for speed range 4800-28800 rpm. We therefore present the results that correspond to 268.9Hz, which is the first critical speed of the system as calculated by B. Gurudatt, and Vikram Krishna [3]. Fig. 16 through 21 display the variation of amplitude of vibration of the system at the six identified stations respectively. The maximum value of amplitude obtained was 0.5572E-7. It can be observed that the amplitude reaches a maximum value at one particular excitation frequency. Fig. 22 displays the typical variation of phase angle with excitation frequency at all the stations.

Using the above three bearings harmonic analysis were performed and finding the different critical speed for different bearings. Similarly the variation of amplitude response with frequency is also obtained.

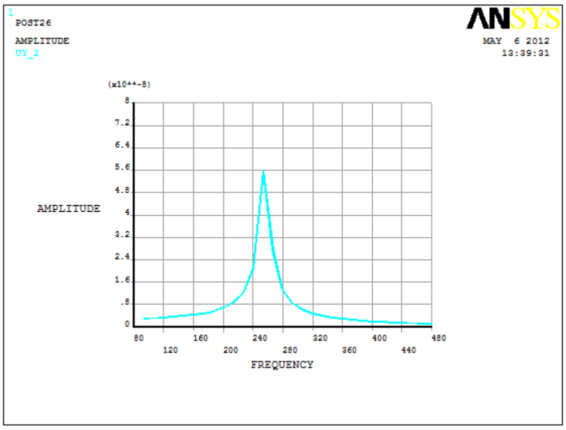

Figure 16: Variation of amplitude of vibration (m) (on Y axis) at station 1 with excitation frequency (Hz)(on X axis)

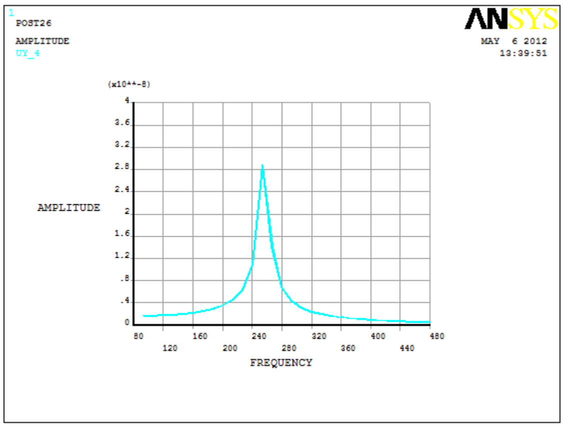

Figure 17:Variation of amplitude of vibration (m)(on Y axis) at station 2 with excitation frequency (Hz)(on X axis)

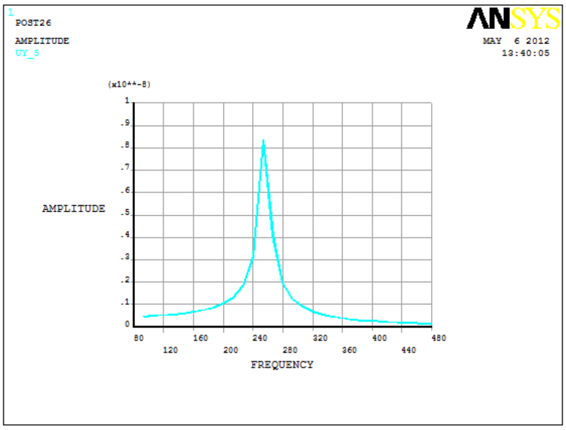

Figure 18:Variation of amplitude of vibration (m)(on Y axis) at station 3 with excitation frequency (Hz)(on X axis)

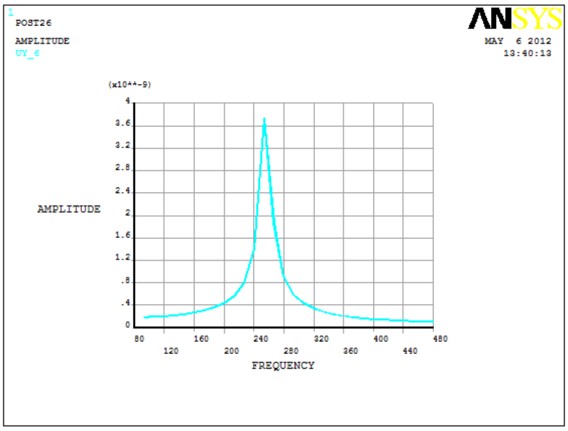

Figure 19:Variation of amplitude of vibration (m)(on Y axis) at station 4 with excitation frequency (Hz)(on X axis)

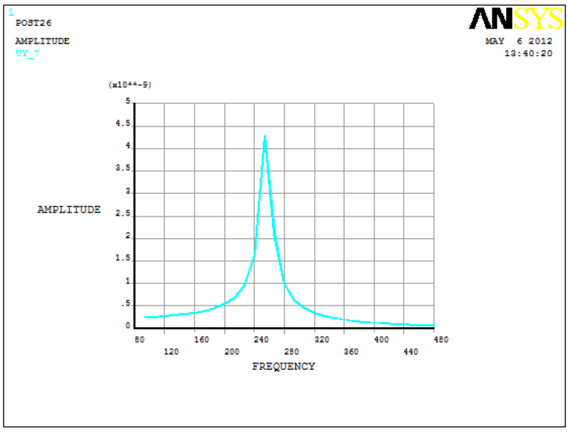

Figure 20:Variation of amplitude of vibration (m)(on Y axis) at station 5 with excitation frequency (Hz)(on X axis)

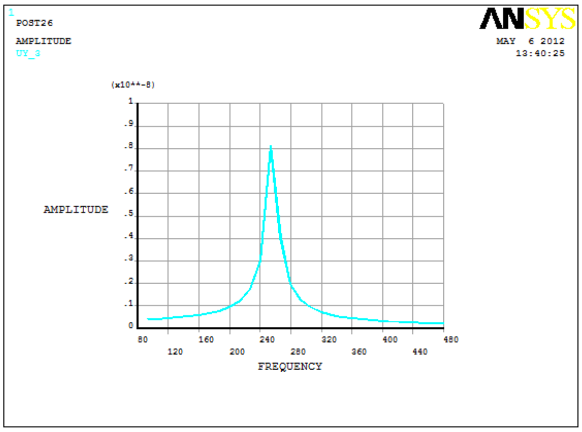

Figure 21: Variation of amplitude of vibration (m)(on Y axis) at station 6 with excitation frequency (Hz)(on X axis)

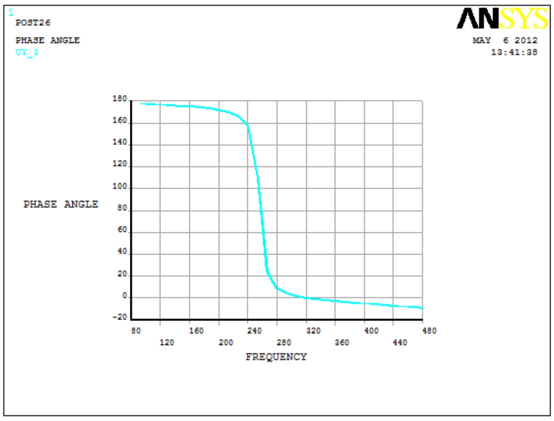

Figure 22: Typical variation of phase angle with excitation frequency

Table I shows the comparison of three bearings used in this study. The excitation frequency is corresponding to the maximum amplitude of rotor. The excitation frequency of the rotor is found for different critical speeds. It is seen that, combin14 and Matrix27 bearings gives the more critical speed than the combi214 bearing.

Table 1. Comparison of different bearings

Bearings	Max. Amplitude	Excitation frequency
Undamped isotropic bearings (COMBIN14)	0.78423E-7	266.2
Symmetric orthotropic bearings(COMBI214)	0.50077E-7	240
Fluid film bearings. (MATRIX27)	0.5572E-7	267

V. CONCLUSION

A finite element model of multi-bearing rotor system using ANSYS is presented in this paper. The effects of rotary inertia, and internal damping were included in the analysis. Comparison of three bearings included in the study, shows combin14 and Matrix27 bearings gives the more critical speed than the combi214 bearing. Study also obtained the variation of amplitude response with frequency as it is important for minimizing the noise of the rotor. The increasing amplitude increases the noise of the rotor. This analysis gives the alternate procedure of finding the critical speed that is harmonic analysis.

Table.2 Geometric Data of Rotor-Bearing Element

Element Node No	Node Location (cm)	Bearing and Disk	Inner Diameter (cm)	Outer Diameter (cm)
1	0.0		0.0	0.51
2	1.27		0.0	1.02
3	5.08		0.0	0.76
4	7.62		0.0	2.03
5	8.89	Disk	0.0	2.03
6	10.16		0.0	3.30
7	10.67		1.52	3.30
8	11.43		1.78	2.54
9	12.70		0.0	2.54
10	13.46		0.0	1.27
11	16.51	Bearing	0.0	1.27
12	19.05		0.0	1.52
13	22.86		0.0	1.52
14	26.67		0.0	1.27

15	28.70	Bearing	0.0	1.27
16	30.48		0.0	3.81
17	31.50		0.0	2.03
18	34.54		1.52	2.03

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