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DYNAMIC ANALYSIS OF A CRACKED ROTOR BEARING DISC SYSTEM –FINITE ELEMENT INVESTIGATION

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ABSTRACT

The vibration in rotating machinery is generally comes due to unbalance, misalignment, shaft crack, mechanical looseness and external forces. In this paper, the influence of transverse cracks in a rotating shaft is analyzed. The paper addresses the three distinct issues –Effect of crack, changes in position of crack and changes in speed of cracked rotor on the dynamic response. In order to conduct this study, a model of steel shaft having a single disc mounted at its center and supported by two bearing is developed in CATIAVSR20. The dynamic response of a rotor with and without crack is evaluated by using ANSYS workbench 15. Initially, the signature data was obtained without crack and thereafter, an artificial crack of 4mm depth was introduced at 275mm distance from the right end of the shaft at a speed of 1500 rpm. Thereafter, in different cases, same crack is introduced at different locations and different speeds of the shaft. Firstly, a significant change in peak amplitude was examined, when a crack of 4mm depth is introduced as compared to no crack condition. Secondly, peak amplitude of the shaft is more in case crack nears the disc than crack at any other positions. This paper also reflects that, at higher speed, the peak amplitude of the shaft is decreased in both the cases i.e. with and without crack.

Keywords: *peak amplitude, crack, crack location, rotor speed*

I. INTRODUCTION

Severely stressed rotors particularly in power plant sector, fatigue cracks in several locations where stresses (both mechanical and thermal) can increase are always a possibility. The presence of a crack may lead to a dangerous and catastrophic effect on the dynamic behavior of rotating structures and cause serious damage to rotating machinery. There are two approaches used to identify the presence of a crack in rotating structures. The first approach is based on the fact that the presence of a crack in rotating shaft reduces the stiffness of the structure, hence reducing the natural frequencies of the original uncracked shaft. Another approach of crack identification is based on the modification of the dynamic responses of the crack rotor during its rotation. Indeed, dynamic analysis of the cracked rotor based on theoretical and experimental studies has been a subject of great interest for the last decades. Several researchers have therefore conducted extensive investigations on the response of cracked rotor. J-J Sinou and A.W.Lees [1] extracted the influence of crack in rotating shaft and showed the change of the shaft frequencies, harmonic component of the dynamical system response, and the evolution of the orbits are the principal effects due to the presence of a crack in a rotating shaft. V. Sudheer Kumar and Ch. Harikrishna [2] analyzed a major increase in displacement curve was observed after the crack depth has reached to 4mm. The results were well focused with the aid of displacement curves, and subcritical curves. These curves helped in better understanding the influence of crack location on displacement of rotor and also in the identification of crack. Anuj Kumar Jain, Vikas Rastogi, Atul Kumar Agrawal [3] experimentally investigated and validated through analytical equation of crack rotor system. He concluded that with the increase in crack depth stiffness varies whereas the stiffness of uncracked rotor is optimum. However, when the depth of crack increases, the stiffness of the shaft drastically reduces. After creating the second crack, stiffness also reduces but the effect of second crack on the stiffness is found marginal. It has also been observed that larger crack depth has the more significant effect on the shaft. Sekhar and Prabhu [4] extracted adequate number of natural frequencies by using a finite element model analysis. It was observed that this method is applicable in cases of shafts with low slenderness ratio L/D, where the changes in natural frequencies due to a crack were found to be significant. Mohammad A. AL-Shudeifat , Eric A. Butcher, Carl R. Stern [5] introduces the general harmonic balance solution of the cracked rotor-bearing-disk system with breathing crack for studying the behavior of the system. The results of this method show important observations of the behavior of the whirl orbits, vibration amplitudes and frequencies of a damaged rotor-disk-bearing system. This behavior may help in detecting the crack at the beginning of its growth. Changping Chen , Liming Dai , Yiming Fu [6] studied and addressed the nonlinear governing equations to make it available for the numerical investigations on the effects of a variety of crack depths, shafts rotational speeds and the locations of the crack and the disc to the dynamical response of the cracked rotor system subjected to the external excitations. Oh Sung Jun [7] use the additional slope and bending moment at crack position to analyze the dynamic behavior of the cracked rotor. Huichun Peng , Qing He, Pengcheng Zhai ,Yaxin Zhen [8] examine the stabilizing effects of both rotational and stationary damping on the dynamic vibration of a rotating rotor with a transverse open crack on the surface of the shaft. H. D. Nelson and C. Nataraj [9] developed a procedure to simulate the dynamic characteristics of a rotor-bearing system with a transverse crack using a FEM representation for the rotating assembly and a perturbation parameter to represent the local change in stiffness due to the crack. A computer code, crack, was developed which implements the analytical procedure and includes a static condensation option to reduce the system degrees-of-freedom. Fangyi Wan, Qingyu Xu, Songtao Li [10] investigated the vibration of a cracked rotor sliding bearing system with rotor–stator rubbing using harmonic wavelet transform (HWT). Three non-linear factors, non-linear oil film forces, rotor–stator rubbing and the presence of crack, was taken in this investigation. Jean-Jacques Sinou [11] analyzed the dynamic characteristics and stability of the non-linear periodic solutions for a cracked rotor. The stability analysis is carried out by applying a perturbation to the non-linear periodic solution, previously computed using the harmonic balance method. Chaozhong Guo, JihongYan, WeichengYang [12] experimentally investigated and verifies the theoretical results of the dynamic behavior and the EMD based crack detection method for the cracked rotor. The breathing crack in the rotor is simulated by a real fatigue crack. As a comparison, the fast Fourier transform method is used to derive the amplitude variation of the high order frequencies from the frequency spectra of the experimental vibration signal.

Hamid Khorrami, Subhash Rakheja, Ramin Sedaghati [13] modified harmonic balance method to compute the vibrational properties of a cracked rotor disc-bearing system using the Timoshenko beam theory. He found the presence of the second crack intensifies the effect of the first crack and the small depth cracks are more sensitive to the propagation of the second crack. He also found that the detection of the two cracks on a rotating shaft is more feasible considering the frequency spectrum of the lateral vibration while the crack has been modeled using the softly-clipped cosine breathing function. Alireza Ebrahimi,Mahdi Heydari, Mehdi Behzad [14] developed a new continuous model for flexural vibration analysis of a simply supported rotor with an open edge crack and obtained governing equation of motion for this rotor solved by a modified Galerkin projection method. Obtained results of free vibration analysis were compared with finite element results for the first natural frequencies and critical speeds. Yanli Lin,Fulei Chu [15] examine the stiffness changes for a cracked rotor and stiffness matrix is obtained by the strain energy release approach and concluded that whether for the slant crack or the transverse crack, obvious appearance of the double frequency is the common characteristic in the transversal responses of the cracked rotor system as long as the crack is not totally close. The slant crack can cause coupling stiffness of bending–torsion, bending–tension and torsion–tension on the shaft and the transverse crack can only cause coupling stiffness of bending–tension on the shaft. The eccentricity can cause coupling vibration of bending–torsion. Ashish K. Darpe [16] compared a rotor with slant crack and with transverse crack regard to the flexibility/stiffness coefficients and the unbalance response characteristics. The rotor with slant crack is found to be stiffer in lateral and longitudinal directions, but more flexible in torsion, compared to the one with transverse crack. Robert Gasch [17] analysed Dynamic behavior of the Laval rotor with a transverse crack and discussed the forced vibrations due to crack and unbalance. Zbigniew Kulesza [18] adopted the multisine technique for the detection of a shaft crack in a rotor. The approach is illustrated with the numerical results of the flexible rotating shaft modeled with the rigid finite element method. Jerzy T. Sawicki , Michael I. Friswell, Zbigniew Kulesza, Adam Wroblewski, John D. Lekki[19] used auxiliary active magnetic bearing (AMB) to help identify crack in the rotor and concluded a sinusoidal force from the AMB produces combinational frequencies based on the AMB frequency, and the rotational speed, that could be used to detect cracks in the rotor. Mohammad Hadi Jalali , Mostafa Ghayour, Saeed Ziaei-Rad, Behrooz Shahriari [20] performed the dynamic behavior of the rotating system by obtaining the Campbell diagram and critical speeds with the use of both beam model and 3D FE model and the unbalance response with beam model. The results were compared and satisfactory agreement between them indicated that one-dimensional beam finite element method which is simple in modeling can be used for rotor dynamic analysis with acceptable accuracy.

II. EQUATION OF MOTION

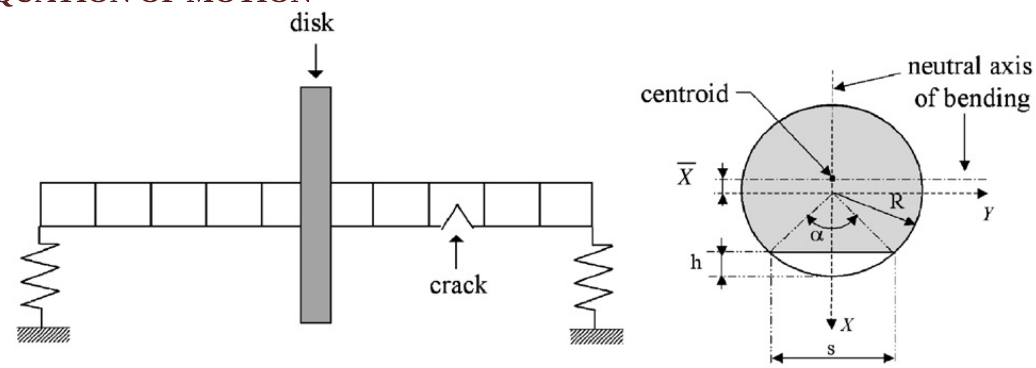


Figure 1: Finite-element model of the rotor and the cracked-beam section.

Using a finite element method, the rotor is discretized into 10 Timoshenko beam finite elements, having four degrees of freedoms at each node. The axial and torsional degrees of freedom are not considered here. Each Timoshenko beam finite element has four degrees of freedom at each node:

$$(M_T^e + M_R^e)\ddot{X}^e + (\eta K_B^e - \omega G^e)\dot{X}^e + (K_B^e + \eta\omega K_C^e)X^e = F_e$$

Where ω is the rotational speed, M_T^e and M_R^e are the translational and rotary mass matrices of the shaft element, K_B^e is the stiffness matrix and K_C^e is the circulatory matrix which describes internal shaft damping. G^e is the gyroscopic matrix. η defines the coefficient of damping, associated with modal damping, for the first mode of the system at rest *(*ω= 0*)*. Fe includes the influence of gravitational forces.

Modeling of disc

The disk is modeled as a rigid disk and may be written as

$$(M_T^d + M_R^d)\ddot{X}^d - \omega G^d \dot{X}^d = F^d$$

Where $M_T^d, M_R^d,$ *and* G^d are respectively the translational mass, the rotary mass and the gyroscopic matrices, F^d defines the unbalance and gravitational forces.

Modeling of uncracked rotor

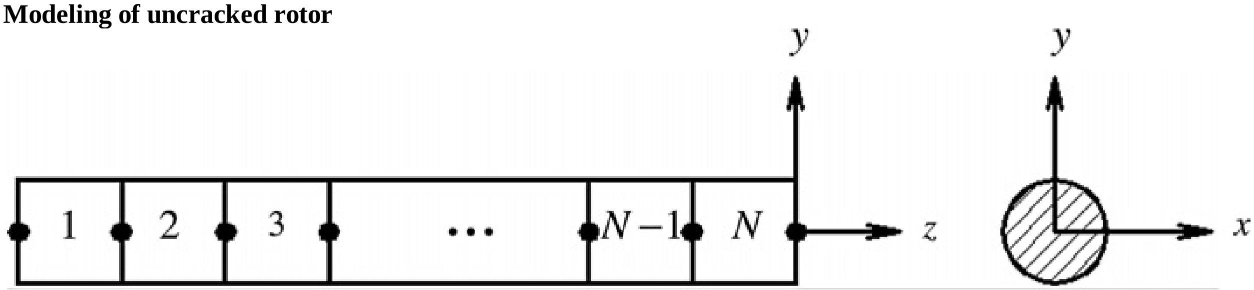


Figure 2 Finite element model of the rotor

Finally, discrete bearing stiffness components are located at either end of the shaft, after assembling the various shaft elements and the rigid disc. The Equation of motion of uncracked rotor considered by jean jacqes sinou is

$$M\ddot{X} + D\dot{X} + KX = g(t)$$

$$M\ddot{X} + D\dot{X} + KX = F_u + F_g$$

Where $g(t)$ contain unbalance force and gravitational force ,M include the mass matrix of the shaft and disc, D combine the effect of stiffness of shafts, internal damping, damping of support and gyroscopic moment K include stiffness matrix of shaft and support. F_u and F_g Are unbalance force and gravitational force respectively.

Modeling of cracked rotor

Equation of motion of the cracked shaft with a breathing crack model considered by Injean Jacqes Sinou

$$M\ddot{X} + D\dot{X} + [K - K_C(t)X] = g(t)$$

$$M\ddot{X} + D\dot{X} + [K - K_C(t)X] = F_u + F_g$$

Where M and K are the mass and stiffness matrices of the complete uncracked rotor. g*(t)* contains the unbalance and gravitational forces. It may be noted that the global stiffness matrix of the rotor consists of a constant component K and a time dependent component

$$K_C = f(t)K_{crack}$$

Equation of motion of the cracked shaft with open crack model in which the stiffness matrix is assumed to be constant (K - Kc) is given by Mohammad A. AL-Shudeifat

$$M\ddot{X} + D\dot{X} + [K - K_C]X = F_1 cos\Omega t + F_2 sin\Omega t + F_g$$

System description

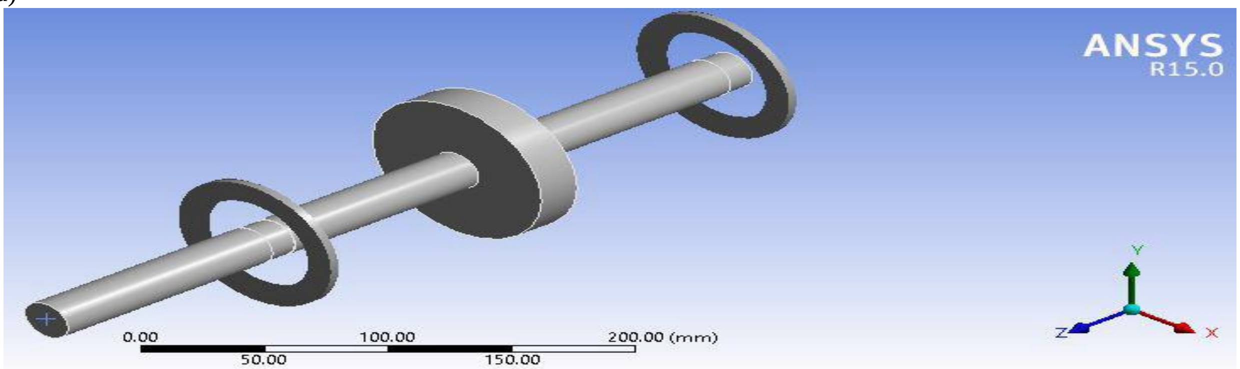
Value of the physical parameters used in the analysis

Notation	Description	Value
D	Diameter of the rotor shaft	25mm
L	Length of the rotor shaft	400mm
T	Torque	31.7N-m
d	Diameter of the disk	100mm
M	Mass of the disc	1.2kg
t	Thickness of the disk	20mm
$\boldsymbol{\rho}$	Density of shaft and disc	7800 kg/m^3
$\boldsymbol{F_R}$	Frequency range	10-1kHz

III. HARMONIC ANALYSIS

Different model designed in ANSYS

a)



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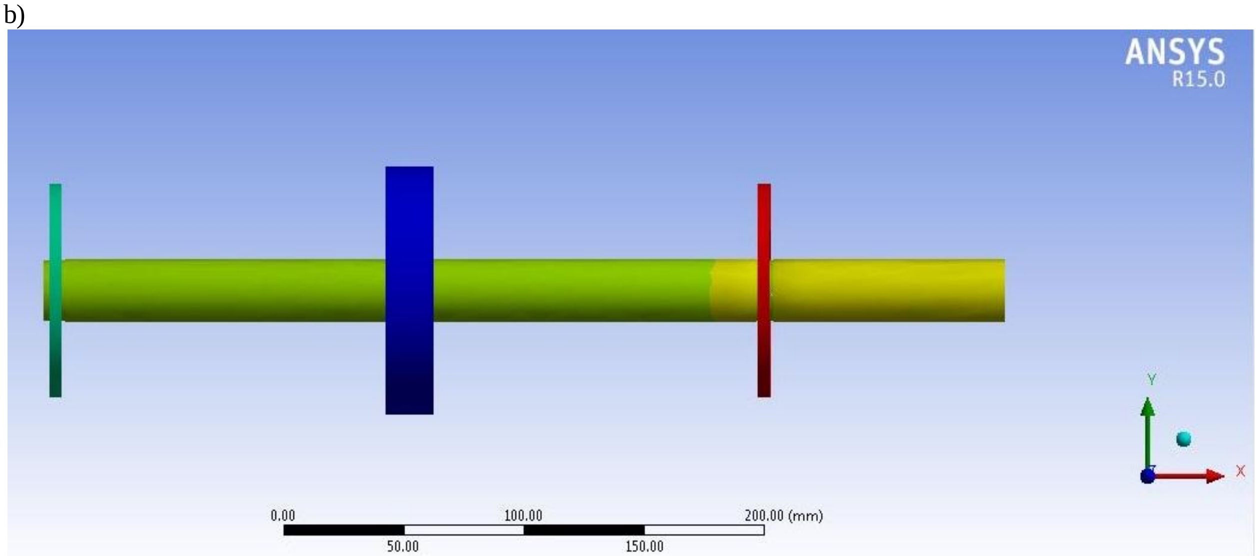

c)

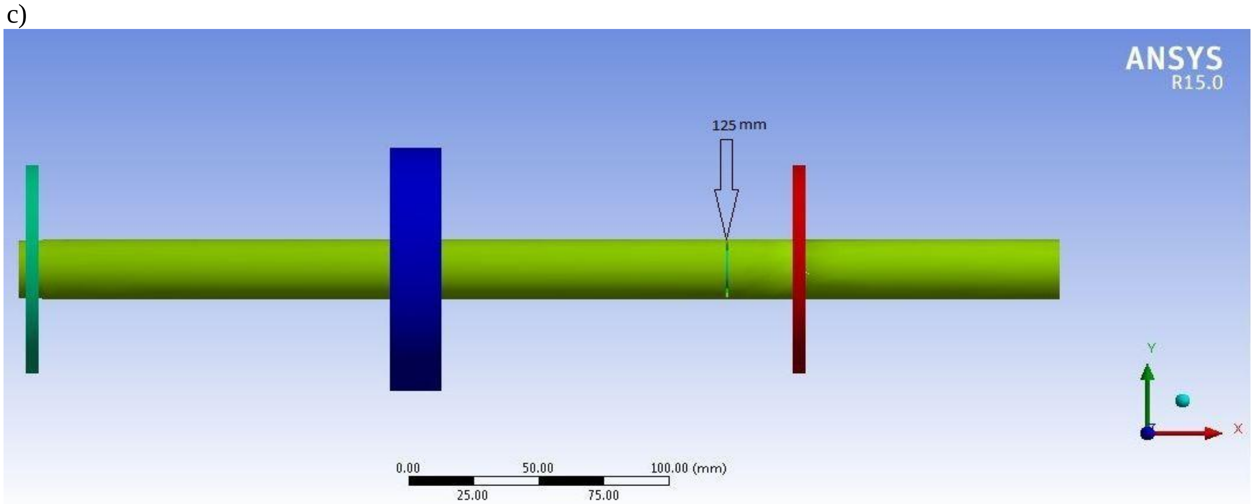

Figure 3: Analytical model of rotor in ANSYS. (a) no-crack; (c) no- crack (c) crack at 125mm

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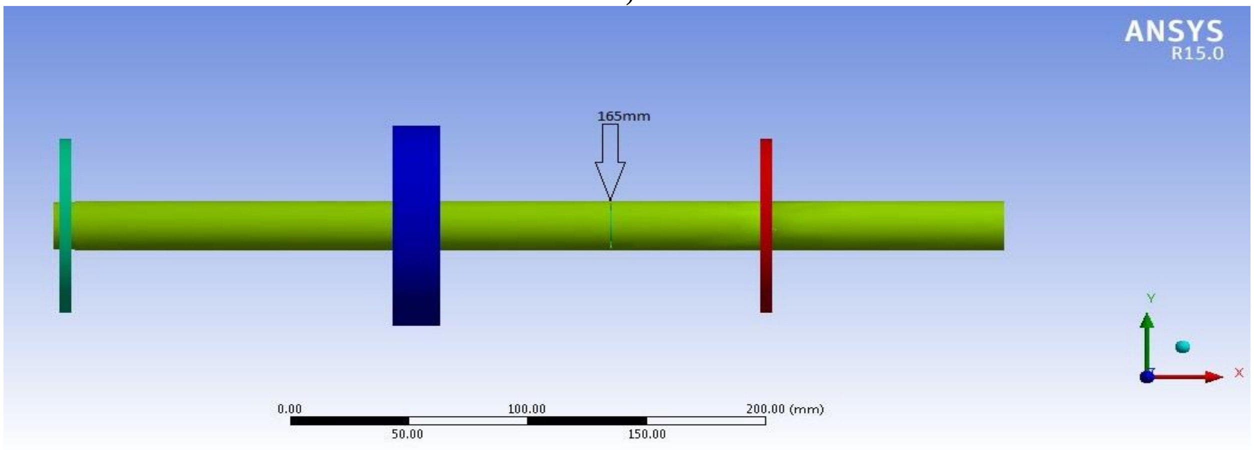
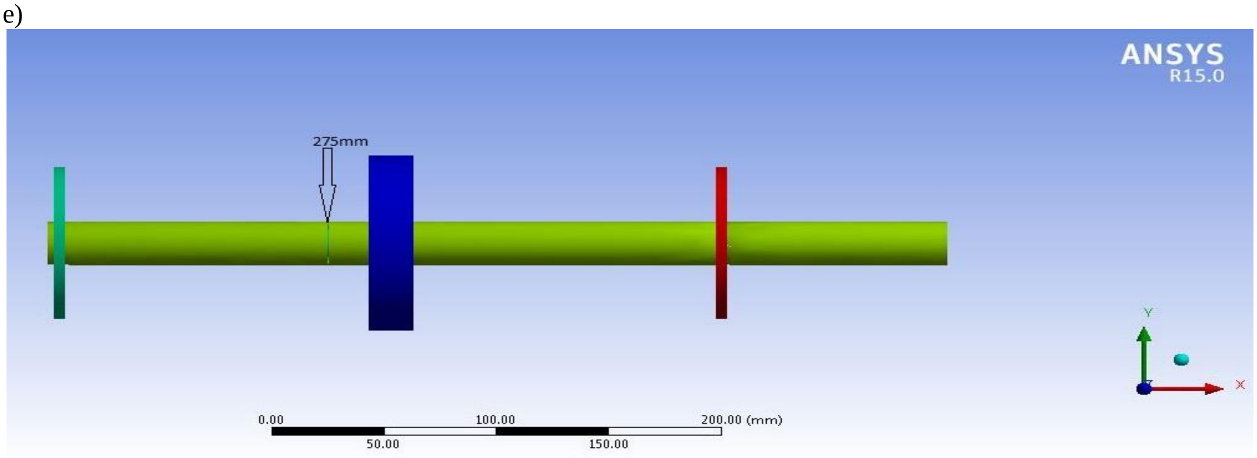
d)



e)



f)

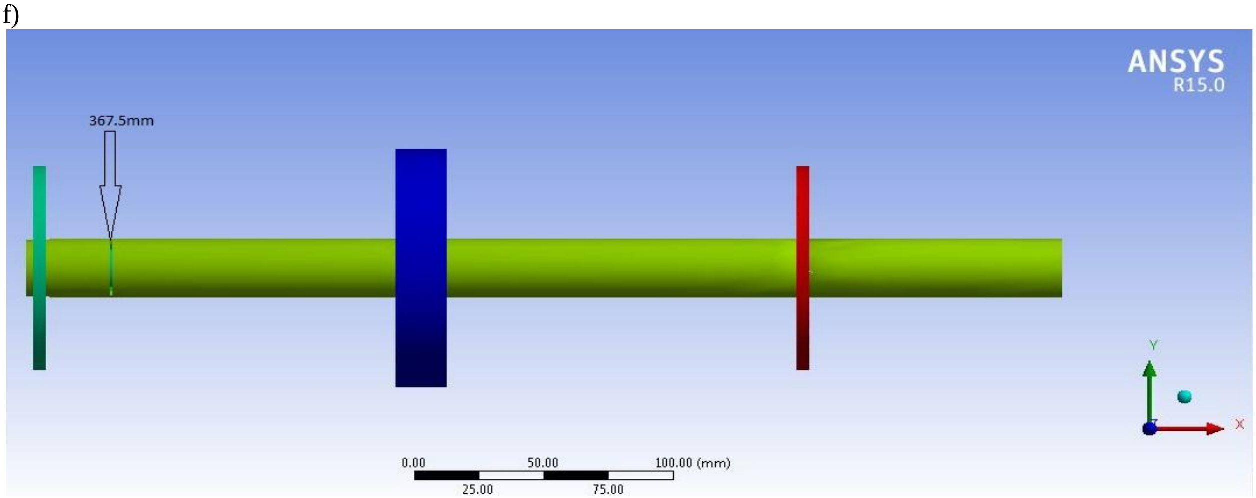


Figure 4: (d) crack at 165mm (e) crack at 275mm (f) crack at 367.5mm

Harmonic analysis was performed for the model presented in Fig.3 and Fig.4 in ANSYS with different combinations of rotor speed and locations. There after displacement plots were drawn and results were discussed.

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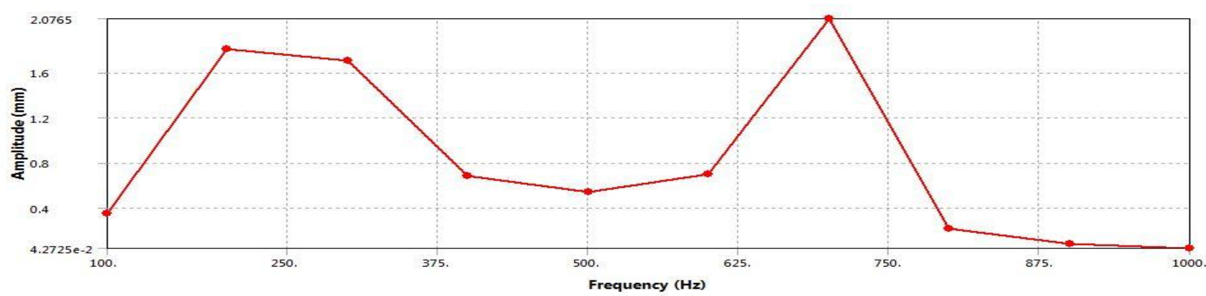
IV.RESULT AND DISCUSSION

From the results of harmonic analysis, the peak amplitude of shaft without crack was 2.07mm at 1500 rpm. When a crack of depth 4 mm was introduced at a distance 275 mm from the right end of the shaft, peak amplitude was 4.36mm. To study the effect of crack positioning and rotor speed on the peak amplitude of shaft, cracks of depth 4 mm was introduced one by one at a distances 125mm, 165mm and 367.5mm from right end of the shaft at a speed of 1500rpm, 2000rpm, 2500rpm, 3000rpm and at 3500rpm. Behavior of displacement curves were illustrated in (Fig.5 and Fig.6) By observing the displacement curves it was clear that peak amplitude of 4mm cracks near disc that is at 275mm was significantly more than 4mm crack at any other position, at any speed. It was also observing that at higher speed, peak amplitude were significantly less when compared to lower speed at any crack position. Peak amplitude of no-crack condition was very less when compared to other cases at all speed considered here. By this significant change in displacement curves it is evident that there is presence of crack in rotor.

a)



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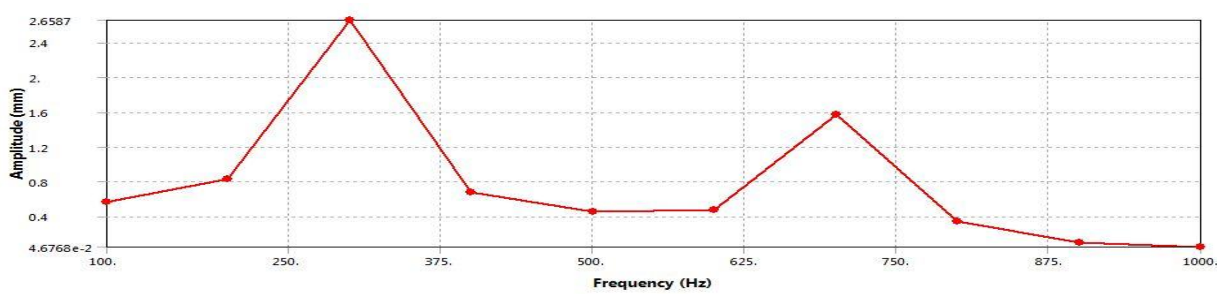


c)

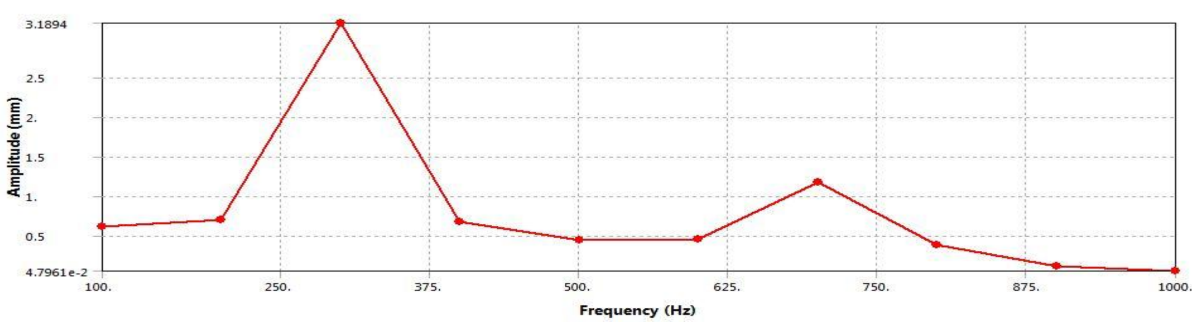
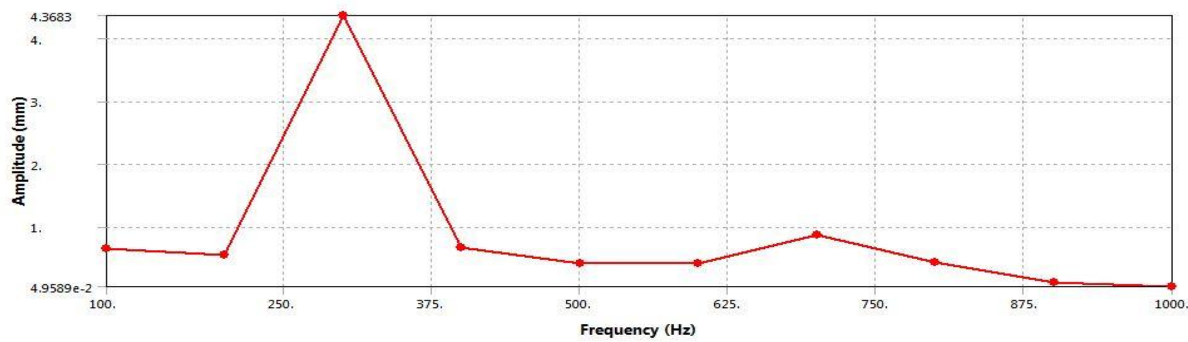


Figure 5: graphs generated by ANSYS at 1500rpm (a) no crack (b) crack at 125mm (c) crack at 165mm

d)



e)

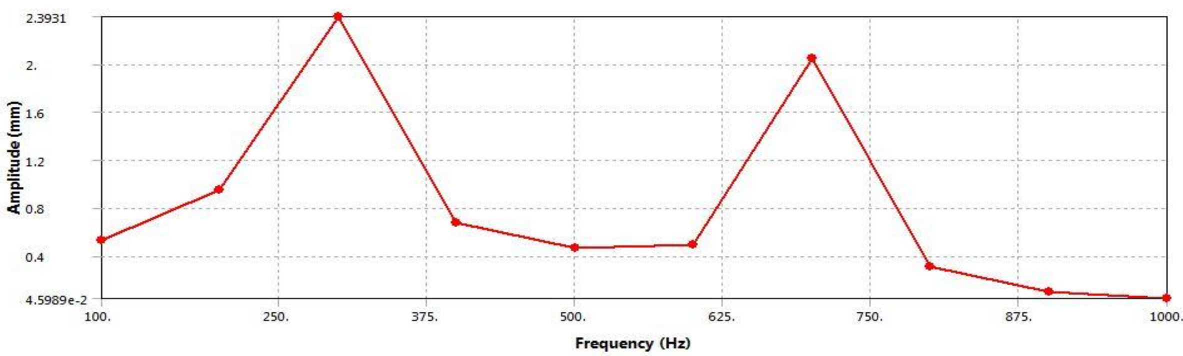


Figure 6: graphs generated by ANSYS at 1500rpm (d) crack at 275mm (e) crack at 367.5

Table 1: peak amplitude at different crack position and at different rotor speed

Crack position↓↓	Rotor speed→→	1500rpm	2000rpm	2500rpm	3000rpm	3500rpm
No crack		2.07	1.56	1.25	1.04	0.89
Crack at 125mm		2.65	2.00	2.00	1.33	1.14
Crack at 165mm		3.81	2.4	1.92	1.60	1.37
Crack at 275mm		4.36	3.2	2.63	2.19	1.87
Crack at 367.5mm		2.39	1.8	1.44	1.20	1.02

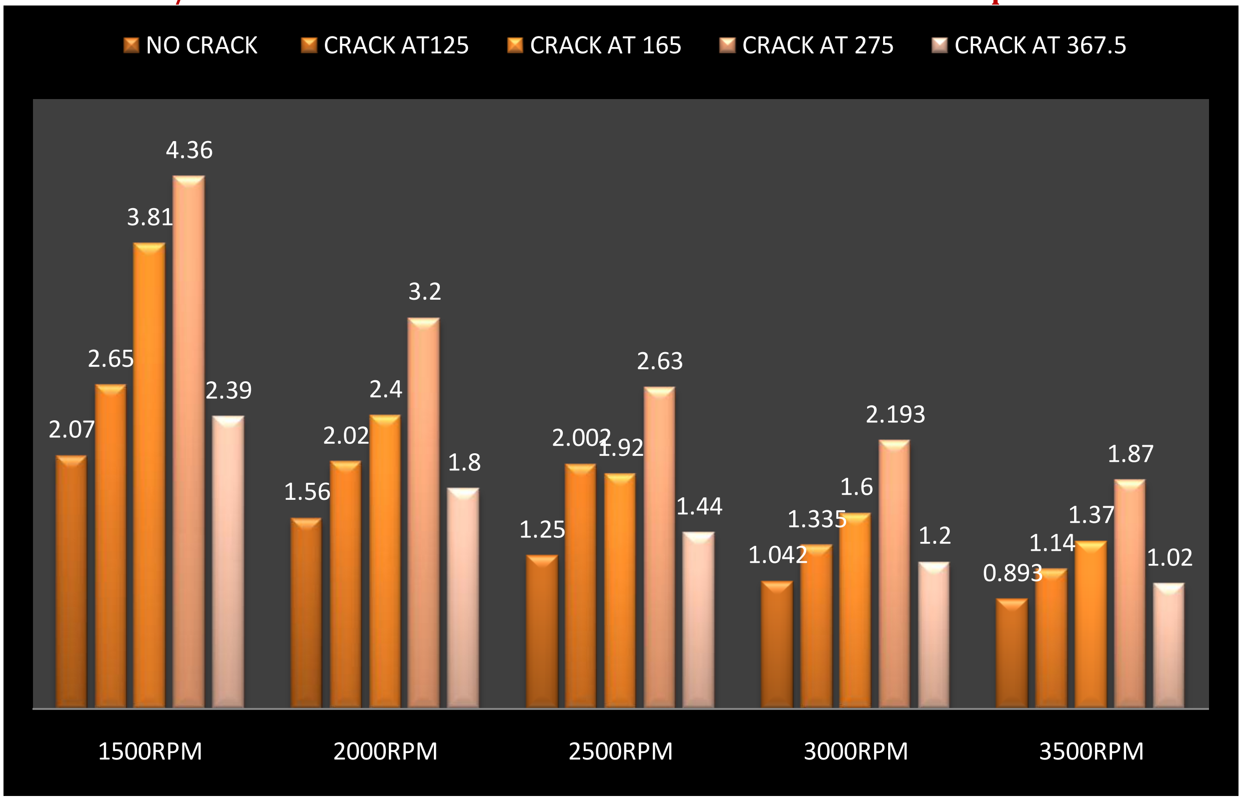


Figure 7: peak amplitude versus rotor speed at different location of crack in bar dig.

V. CONCLUSION

A major increase in displacement curve was observed after the crack was introduced. The results were well focused with the aid of displacement curves. These curves helped in better understanding the influence of crack location on displacement of rotor and also in the identification of crack.

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