Theoretical AndExperimental Investigation Of A Shaft Disc System With A Crack

A THESIS SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF

Bachelor of Technology

(Mechanical Engineering)

by

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CERTIFICATE

This is to certify that the thesis entitled "**Theoretical and Experimental Investigation of a Shaft Disc System with a Crack"**submitted by **Sitesh Kumar Goel (Roll Number: 108ME021)**in partial fulfilllment of the requirements for the award of ***Bachelor of Technology*** in the department of Mechanical Engineering, National Institute of Technology, Rourkela is an authentic work carried out under my supervision and guidance.

To the best of my knowledge, the matter embodied in the thesis has not been submitted elsewhere for the award of any degree.

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A C K N O W L E D G E M E N T

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Sitesh Kumar Goel
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Place: NIT Rourkela

ABSTRACT

This work proposes the vibration analysis for the periodically time-varying rotor system with Transverse crack. Identification of a crack in a rotor is major issue in industries. Sudden failure of the rotor system may lead to huge maintenance cost and production hault.Therefore it is necessary to diagnose the problem while operation is carried on so that at appropriate intervals maintenance can be done.This work uses Vibration Signature Analysis to analyse the difference between responses of sound shaft and a cracked shaft. Based on the inference of the two responses it can be predicted at what time the shaft should be changed before it leads to catastrophic failure. The details of experimentation and analysis are given in the following context.

CONTENTS

LIST OF FIGURES

Chapter 1

INTRODUCTION

Machine fault problems are major concern for industries as it includes high maintenance cost and unwanted downtime. Machine fault analysis can be done with different methodologies as vibration signature analysis, noise signature analysis, lubrication signature analysis and temperature monitoring.

Vibration signature analysis is the most popularly used methodology which is based on the principle that all the system produces vibration. When a machine is operating properly the vibration is small and constant, however when fault develops and some of the dynamic processes in the machine changes, the vibration spectrum also changes.

In this project we will be dealing with one of the issues in concern nowadays that is identification of cracks in rotor systems. When a transverse crack occurs in the shaft the stiffness of the shaft is deteriorated so that continuous operation may change the vibration behavior of the rotor system into failure. To prevent this catastrophic failure, a rotor condition is diagnosed through continuous vibration monitoring from which some types of crack information may be detected.

For a rotor system with a cracked shaft ,which is in status of varying stiffness in accordance with the rotor revolution ,the harmonic input and output responses can well identify a stiffness asymmetry that features crack conditions.

1.1 Vibration Analysis

Modern condition monitoring techniques encompass many different themes; one of the most important and informative is the vibration analysis of rotating machinery. Using vibration analysis, the state of a machine can be constantly monitored and detailed analysis may be made concerning the health of the machine and any faults which may arise or have already arisen. Machinery distress very often manifests itself in vibration or a change in vibration pattern. Vibration analysis is therefore, a powerful diagnostic and troubleshooting

tool of major process machinery. On-load monitoring can be performed mainly in the following
three ways.

(i)Periodic field measurements with portable instruments; this method provides information about long-term changes in the condition of plant. The portable instruments are employed with a high load factor and can often be placed in the care of only one man. Use of life curves and the LEO approach assist the decision making.

(ii) Continuous monitoring with permanently installed instruments; it is employed when machine failures are known to occur rapidly and when the results of such failure are totally unacceptable as in the case of turbine generator units.

(iii) Signature analysis: scientific collection of information, signals or signatures, diagnosis and detection of the faults by a thorough analysis of these signatures based on the knowledge hitherto acquired in the field, and judging the severity of faults for decision making, all put together, is called signature analysis. The technique involves the use of electronic instrumentation especially
designed for the purpose of varied capacities, modes of application and design features. Vibration signals are the most versatile parameters in machine condition monitoring techniques. Periodic vibration checks reveal whether troubles are present or impending. Vibration
signature analysis reveals which part of the machineries defective and why. Although a number of vibration analysis techniques have been developed for this purpose, still a lot of scope is there to reach a stage of expertise.

1.2 Present aim of work

For conducting the experiment, first of all we will be preparing the machine setup. This machine is already available in the market, but our aim is to prepare this machine using some conventional machining methods, so that we can have a machine at a cheap rate. . The machine will be measuring the vibration response of the ceramic solid shaft. This machine will be consisting of a high-speed D.C motor connected to a Variable Frequency Drive. The motor will rotate a stainless steel cylindrical shaft with a disc fitted at the middle of the shaft. The shaft will be aligned in a straight line with the help of bearings. The shaft will be coupled to the high speed motor. Vibration response will be taken through bearing with the help of Oscilloscope.

Chapter 2

LITERATURE SURVEY

The effect of cracks on the dynamic behavior of structural elements such as shafts,beamsetc has been the subject of several investigations. Anyaccidental(such as cracks) or intentional modification in a structure will affect its dynamical behavior and change its stiffness and damping properties.

Long ago, a finite element for the cracked shaft for structural analysis is developed by H.D Nelson and C.Nataraj [1]and showed considerable changes in the Resonant frequencies and vibration amplitudes by the existence of moderate cracks, which are developed using compliance

Matrix based on strain-energy expressions.

G.Meng,W.Garsh[2] have analyzed the stability and stability degree of a cracked flexible rotor supported on journal bearing and proposed the stability of the shaft decreases with the presence of crack.

O.S Jun and H.J Eun have done the modeling and vibration analysis of a simple rotor with a breathing crack.

A.K Darpe has experimentally investigated the response of a cracked rotor to periodic axial excitation and thus proposed that the response of the rotor to axial impulse excitation could be used for reliable diagnosis of rotor cracks.

Dong Ju Han did the vibration analysis of periodically time-varying rotor system with transverse crack. From the feasibility of the crack modeling by harmonically varying stiffness, for which the dynamic behaviors of breathing crack was investigated and directional frequency response functions was carried out which showed that the presence of crack in the shaft reduces the frequency of vibration of shaft.

Chapter 3

THEORETICAL ANALYSIS

3.1 Finite element formulation

A uniform shaft of length L, subjected to an axial compressive load $P(t)$ and rotating at a constant speed Ω is illustrated in Fig. 3.1. A typical finite rotating shaft element is shown in figure 3.2. The element consists of two nodes and each node has four degrees of freedom: two translations and two rotations. With the axial motion neglect, a typical cross-section of the element, located at a distance 's' from the left node, in a deformed state is described by the translations $V(s,t)$ and $W(s,t)$ in the Y and Z directions and small rotations $B(s,t)$ and $\Gamma(s,t)$ about Y and Z. The two translations (V,W) consist of a contribution (V_b,W_b) due to bending and a contribution(V_s,W_s) due to transverse shear deformation; the rotations (B,Γ) are only related to the bending deformations(V_b,W_b). These relationships can be expressed as follows

$$\left.\begin{aligned} V(s,t) &= V_b(s,t) + V_s(s,t) \\ W(s,t) &= W_b(s,t) + W_s(s,t) \end{aligned}\right\} \tag{3.1}$$

$$\left.\begin{aligned} \mathrm{B}(\mathrm{s},\mathrm{t}) &= -\,\partial \mathrm{W_b}(\mathrm{s},\mathrm{t})/\,\partial \mathrm{s} \\ \Gamma(\mathrm{s},\mathrm{t}) &= \partial \mathrm{V_b}(\mathrm{s},\mathrm{t})/\,\partial \mathrm{s} \end{aligned}\right\} \tag{3.2}$$

The translations and rotations of a typical point within the element can be related to the nodal displacement vector $[q^e]$ and the translational and rotational shape function matrices $[N_t(s)]$and $[N_r(s)]$, respectively, as

$$\begin{pmatrix} V(s,t) \\ W(s,t) \end{pmatrix} = \begin{bmatrix} \theta_1 & 0 & 0 & \theta_2 & \theta_3 & 0 & 0 & \theta_4 \\ 0 & \theta_1 & -\theta_2 & 0 & 0 & \theta_3 & -\theta_4 & 0 \end{bmatrix} \begin{Bmatrix} V_1 \\ W_1 \\ B_1 \\ \Gamma_1 \\ V_2 \\ W_2 \\ B_2 \\ \Gamma_2 \end{Bmatrix}$$

$$\begin{pmatrix} V(s,t) \\ W(s,t) \end{pmatrix} = [N_t(s)][q^e(t)]$$

(3.3)

$$\begin{pmatrix} B(s,t) \\ \Gamma(s,t) \end{pmatrix} = \begin{bmatrix} 0 & -\phi_1 & \phi_2\, 0 & 0 & -\phi_3\phi_4 & 0 \\ \phi_1 & 0 & 0\, \phi_2 & \phi_3 & 0 \quad 0 & \phi_4 \end{bmatrix} \begin{Bmatrix} V_1 \\ W_1 \\ B_1 \\ \Gamma_1 \\ V_2 \\ W_2 \\ B_2 \\ \Gamma_2 \end{Bmatrix}$$

$\begin{pmatrix} B(s,t) \\ \Gamma(s,t) \end{pmatrix} = [N_r(s)][q^e(t)]$
(3.4)

Where

$$[q^e(t)] = \{V_1, W_1, B_1, \Gamma_1, V_2, W_2, B_2, \Gamma_2\}^T$$

From equations (3.1)-(3.4), the two transverse shear strains(V_s', W_s') can be related to the nodal displacement vector as

$$\begin{pmatrix} V_s' \\ W_s' \end{pmatrix} = \left([N_t'] - \begin{bmatrix} 0 & 1 \\ -1 & 0 \end{bmatrix} [N_r]\right) [q^e(t)]$$

$$\begin{pmatrix} V_s' \\ W_s' \end{pmatrix} = \begin{bmatrix} \eta_1 & 0 & 0\, \eta_2 & \eta_3 & 0 \quad 0 & \eta_4 \\ 0 & -\eta_1 & \eta_2\, 0 & 0 & -\eta_3\eta_4 & 0 \end{bmatrix} \begin{Bmatrix} V_1 \\ W_1 \\ B_1 \\ \Gamma_1 \\ V_2 \\ W_2 \\ B_2 \\ \Gamma_2 \end{Bmatrix}$$

$\begin{pmatrix} V_s' \\ W_s' \end{pmatrix} = [B_s][q^e(t)]$(3.5)

Where the symbol "′" indicates differentiation with respect to axial distance's'. The representation of the shape functions can be derived by using the expression of the static deflection of a Timoshenko beam, and their detailed forms as well as the shape function matrices are listed below

$$\theta_1 = [1 - 3\xi^2 + 2\xi^3 + (1 - \xi)\phi]/(1 + \phi)$$

$$\theta_2 = l[\xi - 2\xi^2 + \xi^3 + (\xi - \xi^2)\phi/2]/(1 + \phi)$$

$$\theta_3 = (3\xi^2 - 2\xi^3 + \xi\phi)/(1 + \phi)$$

$$\theta_4 = l[-\xi^2 + \xi^3 - (\xi - \xi^2)\phi/2]/(1+\phi)$$

$$\phi_1 = 6(-\xi + \xi^2)/[l(1+\phi)]$$

$$\phi_2 = [1 - 4\xi + 3\xi^2 + (1-\xi)\phi]/(1+\phi)$$

$$\phi_3 = 6(\xi - \xi^2)/[l(1+\phi)]$$

$$\phi_4 = [-2\xi + 3\xi^2 + \xi\phi]/(1+\phi)$$

$$\eta_1 = \phi/[l(1+\phi)]$$

$$\eta_2 = \phi/[2(1+\phi)]$$

$$\eta_3 = -\phi/[l(1+\phi)]$$

$$\eta_4 = \phi/[2(1+\phi)]$$

$$\xi = s/l$$

$$\phi = 12EI/(kGAl^2)$$

3.2 The governing equations of motion

The potential energy U^e of the uniform shaft element of length *l*, including the contributions of elastic bending and shear energy and the energy due to a spatial independent, axial compressive load *P* is given by

$$U^e = \frac{1}{2}\int_0^l EI\left[\left(V_b''\right)^2 + \left(W_b''\right)^2\right] ds + \frac{1}{2}\int_0^l kGA\,[(V_s')^2 + (W_s')^2]\,ds \quad (3.6)$$

Where *E* is the young modulus, *I* is the second moment of area, *k* is the shear coefficient, *G* is the shear modulus and *A* is the cross-sectional area of the shaft.

Under the assumption that the shaft rotates at a constant speed Ω, the kinetic energy T^e of the shaft element including both the translational and rotational form is given by

$$T^e = \frac{1}{2}\int_0^l \rho A\left[(\dot{V})^2 + (\dot{W})^2\right] ds + \frac{1}{2}\int_0^l I_d\left[(\dot{B})^2 + (\dot{\Gamma})^2\right] ds - \frac{1}{2}\,\Omega \int_0^l I_p\left[\dot{\Gamma}B - \dot{B}\Gamma\right] ds$$

$$+ \frac{1}{2}\Omega^2 \int_0^l I_p ds \qquad (3.7)$$

Where the symbol "." denotes differentiation with respect to time t, ρ is the mass density of the shaft material, I_d and I_p are the diametral and polar mass moment of inertia of the shaft per unit length.

Upon substituting Eqs.(3.3) – (3.5) into Eqs. (3.6) and (3.7), respectively, the potential energy U^e and kinetic energy T^e can be rewritten in terms of the nodal displacement vector as,

$$U^e = \frac{1}{2}[q^e]^T[K_b^e][q^e] + \frac{1}{2}[q^e]^T[K_s^e][q^e] - \frac{1}{2}\,P\,[q^e]^T\left[K_g^e\right][q^e] \qquad (3.8)$$

$$T^e = \frac{1}{2}[\dot{q}^e]^T[M_t^e][\dot{q}^e] + \frac{1}{2}[\dot{q}^e]^T[M_r^e][\dot{q}^e] - \frac{1}{2}\,\Omega\,[\dot{q}^e]^T[H^e][q^e] + \frac{1}{2}\Omega^2 I_p l \qquad (3.9)$$

Where

$$[K_b^e] = \int_0^l [N_r']^T\ EI\ [N_r']\, ds$$

$$[K_s^e] = \int_0^l [B_s]^T\ kGA\,[B_s]\, ds$$

$$\left[K_g^e\right] = \int_0^l [N_t']^T\ [N_t']\, ds$$

$$[M_t^e] = \int_0^l [N_t]^T\ \rho A\ [N_t]\, ds$$

$$[M_r^e] = \int_0^l [N_r]^T\ I_d[N_r]\, ds$$

$$[H^e] = \int_0^l I_p[N_r]^T \begin{bmatrix} 0 & 0 \\ 1 & 0 \end{bmatrix} [N_r]\, ds$$

$[K_b^e], [K_s^e]$and$\left[K_g^e\right]$areelement bending stiffness , element shear stiffness and element geometric stiffness matrix respectively.

$[M_t^e]$and$[M_r^e]$areelement translational mass matrix and element rotary inertia mass matrix respectively.

The equation of motion for the rotating shaft is obtained by using the Lagrangian, L=T-U in the Lagrange's equation.

$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{q}_k}\right) - \left(\frac{\partial L}{\partial q_k}\right) = 0$,For k=1 to n, n is the total number of coordinates.

The equation of motion for the finite rotating shaft element is given as

$$([M_t^e] + [M_r^e])[\ddot{q}^e] - \Omega[G^e][\dot{q}^e] + ([K^e])[q^e] = 0 \qquad (3.10)$$

With $[G^e] = [H^e] - [H^e]^T$

3.3 System Equation of motion

The equation of motion in matrix form for axially loaded discrete system is,

$$[M][\ddot{q}] - \Omega[G][\dot{q}] + ([K])[q] = 0 \qquad (3.11)$$

Where

$$[M] = ([M_t^e] + [M_r^e])$$

$$[G] = [G^e]$$

$$[K] = ([K_b^e] + [K_s^e])$$

$$[q] = [q^e]$$

Where $[K], [M], [S], [G]$and$[q]$are the global elastic stiffness matrix, global mass matrix, global geometric stiffness matrix,global gyroscopic matrix and global displacement vectorrespectively.

3.4 The Stiffness matrix K_c of a cracked beam element

To study the effect of a crack upon the stability of an elastic structure, one has to establish the local stiffness or flexibility matrix of the cracked member under the loading condition. To this end, a prismatic bar is considered with a crack of depth a along the y-axis with a uniform depth along the z-axis. The cracked beam element is loaded with axial force P_1,P_4,shear forceP_2,P_5 and bending moments P_3,P_6.Under general loading,the additional displacement u_i along the direction of force P_i due to the presence of the crack will be computed using Castiglione's theorem and by generalization of the Paris equation.

To this end, if U_T is the main strain energy due to crack ,Castiglione's theorem demands that the additional displacement is u $=\partial U_T/\partial P_i$ along the force P.The strain energy will have the form

$$U_T=\int_0^{\alpha}\frac{\partial U_T}{\partial\alpha}d\alpha=\int_0^{\alpha}da \qquad (3.12)$$

Where J=$\frac{\partial U_T}{\partial\alpha}$ the strain energy density function.Therefore,

$$u_i=\frac{\partial}{\partial P_i}[\int_0^{\alpha}J(\alpha)d\alpha] \qquad (3.13)$$

The flexibility influence coefficient c_{ij} will be,by definition

$$c_{ij}=\frac{\partial u_i}{\partial P_j}=\frac{\partial^2}{\partial P_i\partial P_j}\int_0^{\alpha}J(\alpha)d\alpha \qquad (3.14)$$

The strain energy density function J has the general form

$$J=\frac{1}{E'}[(\sum_{i=1}^{3}K_{\Pi})^2+(\sum_{i=1}^{3}K_{\Pi})^2+k(\sum_{i=1}^{3}K_{\Pi})^2 \qquad (3.15)$$

Where E'=E for plane stress,E'=E/(1-ν^2) for plane strain,k=1+ν,E and ν the Young's Modulus and the Poisson ratio respectively. Then integrating along the cut(axis y),

$$c_{ij}=\frac{1}{E'b^2}\int_0^{\alpha}[\frac{\partial^2}{\partial P_i\partial P_j}\sum_m\int_0^{B}(e_m\sum_n K_{mn})^2\,dy]dx \qquad (3.16)$$

Where e_m=k for m=III and e_m=I,II.Furthermore,$K_{mn}$is the stress intensity factor of mode m(m=I,II,III)due to the loadP_n(n=1,2,3)since the strain energy is additive.Then,

K_{11}=$\frac{P_1}{BW}\sqrt{\pi\alpha}F_1$(α/W) (3.17)

K_{13}=$\frac{6P_3}{BW^2}\sqrt{\pi\alpha}F_2$(α/W) (3.18)

K_{12}=K_{113}=0

K_{112}=$\frac{\boldsymbol{P_2}}{\boldsymbol{BW}}\sqrt{\boldsymbol{\pi\alpha}}$F(α/W) (3.19)

Where

$$F_1(\alpha/W)=\sqrt{\frac{2W}{\pi\alpha}\tan\frac{\pi\alpha}{2W}}\frac{0.752+2.02\left(\frac{\alpha}{W}\right)+0.37(1-\sin\frac{\pi\alpha}{2W})}{\cos\frac{\pi\alpha}{2W}}$$

$$F_2(\alpha/W)=\sqrt{\frac{2W}{\pi\alpha}\tan\frac{\pi\alpha}{2W}}\frac{0.923+0.199(1-\sin\frac{\pi\alpha}{2W})}{\cos\frac{\pi\alpha}{2W}}$$

$$F_{II}(\alpha/W)=\frac{1.30-0.65\left(\frac{\alpha}{W}\right)+.37\left(\frac{\alpha}{W}\right)+0.28\left(\frac{\alpha}{W}\right)}{\sqrt{1-\frac{\alpha}{W}}}$$

Using expressions we get

$$C_{11}=\frac{\emptyset_{11}}{E'B^3W^3}\ C_{13}=\frac{6\emptyset_{12}}{E'B^3W^3}\ C_{33}=\frac{6\emptyset_{22}}{E'B^3W^3} \tag{3.20}$$

$$C_{22}=\int_0^{\alpha}\frac{4F_{II}^2\left(\frac{\alpha}{W}\right)}{E'B^2W^2\pi\left(\frac{\alpha}{W}\right)}d\left(\frac{\alpha}{W}\right) \tag{3.21}$$

$$\emptyset_{ij}=\int_0^{\alpha}\frac{\pi\alpha}{W}F_i\left(\frac{\alpha}{W}\right)F_i\left(\frac{\alpha}{W}\right)d\alpha$$

The overall flexibility matrix for the crack will have the form

$$C_{ovi}=\begin{bmatrix}c_{11} & 0 & c_{13}\\ 0 & c_{22} & 0\\ c_{31} & 0 & c_{33}\end{bmatrix} \tag{3.22}$$

Total flexibility of the cracked element is

$$C_{total}=C_{intact}+C_{ovi}=\begin{bmatrix}c_{11}+\frac{l}{EA} & 0 & c_{13}\\ 0 & c_{22}+\frac{l^3}{3EI} & 0\\ c_{31} & 0 & c_{33}+\frac{l}{EI}\end{bmatrix} \tag{3.23}$$

The transformation matrix is $\pi = \begin{bmatrix} -1 & 0 & 0 \\ 0 & -1 & 0 \\ 0 & -l & -1 \\ 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix}$ (3.24)

And hence the stiffness matrix K_c of a cracked beam element is

$K_c = \pi C_{total}^{-1} \pi^T$

(3.25)

Chapter 4

FABRICATION AND EXPERIMENTAL PROCEDURE

4.1 Design Of The Test Rig

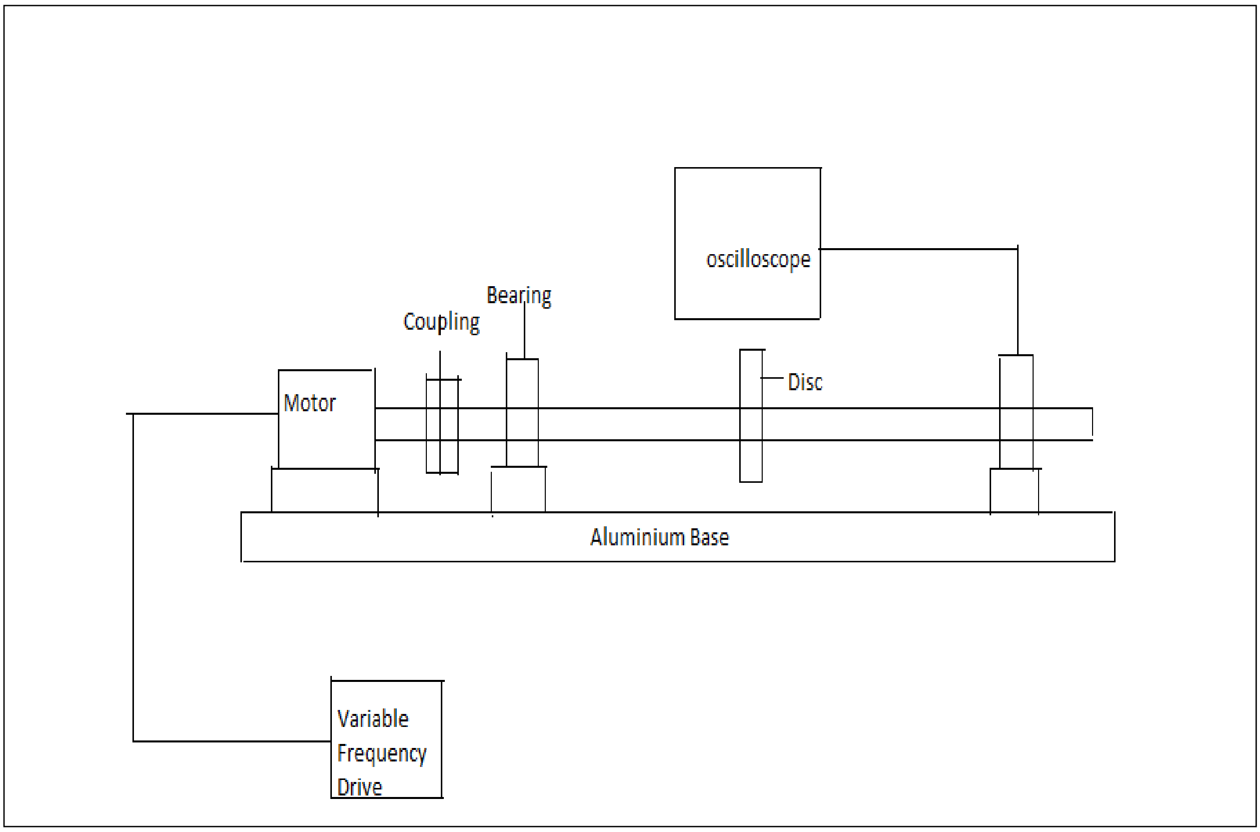

Fig 4.1 Schematic Diagram OfExperimental Setup

Schematic Diagram of the experimental setup was dawn as shown above.

4.2 FABRICATION OF THE TEST RIG Specification of the various equipmentsused in fabricating the test rig are:-

1)Motor-Siemens 1.1KW,215V,2.3A,50Hz,8250rpm

Fig 4.2 Motor2

2)Oscilloscope-Textronics 250MHz,2.5GS/s

Fig 4.3 Oscilloscope

3.Bearing Material –Aluminium, Height 11.5cm,Length-10cm,Width-2cm

4.Solid Shaft Material-Stainless Steel Length-1.1m,diameter-15.8mm

Fig 4.4 Solid Shaft held in between two Bearings

5)DiscMaterial-Mild Steel Thickness-15mm,Diameter-125mm,
Hole Diameter-15.8mm

Fig 4.5 Disc

4.3Experimental Setup

A 10 mm thick and 25.4 cm width aluminium strip was used as a base to set up all the equipments on it.Motor was fixed on it with the help of 8mm nuts and bolts.Motor was connected to the Variable Frequency Drive to impart desired rotational speed to the shaft. Two bearings were similarly fixed to the base with the help of nuts and bolts. The shaft was coupled to the motor and it was aligned in a straight line with the help of bearings. The disc was located in the middle of the shaft. The vibration response was taken by the help of Oscilloscope in which a magnetic sensor was attached to the bearing.

4.4 Procedure

The motor was given speed of 500rpm with the help of Variable Frequency Drive which was connected to the electric supply. The magnetic sensor was attached to the bearing through an iron plate. The vibration response of sound shaft first was recorded in the oscilloscope. Then a 5mm crack was introduced in the shaft with the help of hacksaw and then the vibration response was recorded .Both the responses were compared and inference was drawn from them.

Chapter 5

RESULTS AND DISCUSSION

	Without Crack	With crack
Frequency	83.3MHz	31.3MHz
Amplitude	6mv	15.6mv
Frequency	9.62MHz	8.06MHz
Amplitude	6mv	16.4mv

Table 5.1 Comparison of Vibration Response With And Without Crack

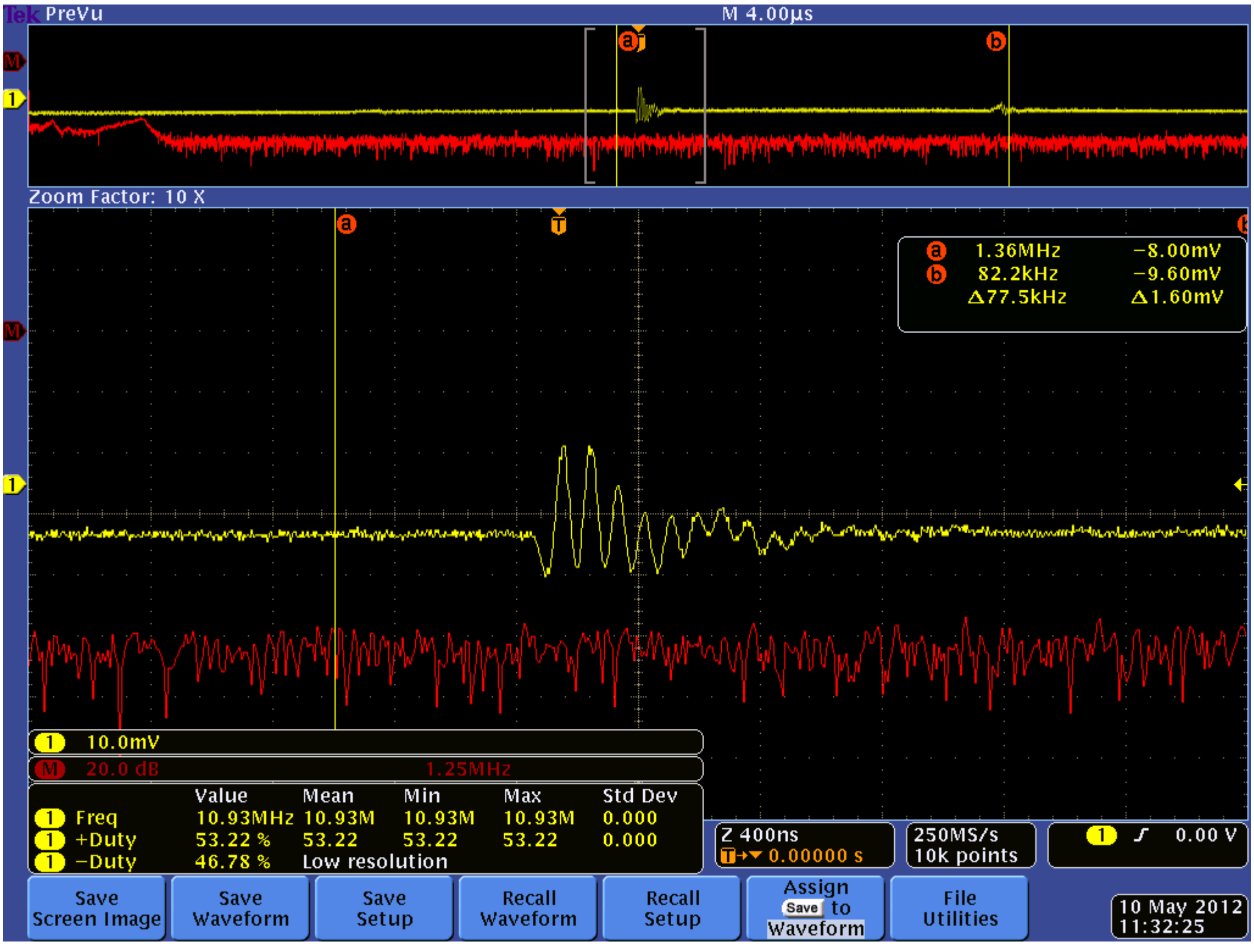

Fig 5.1Vibration Response Without Crack

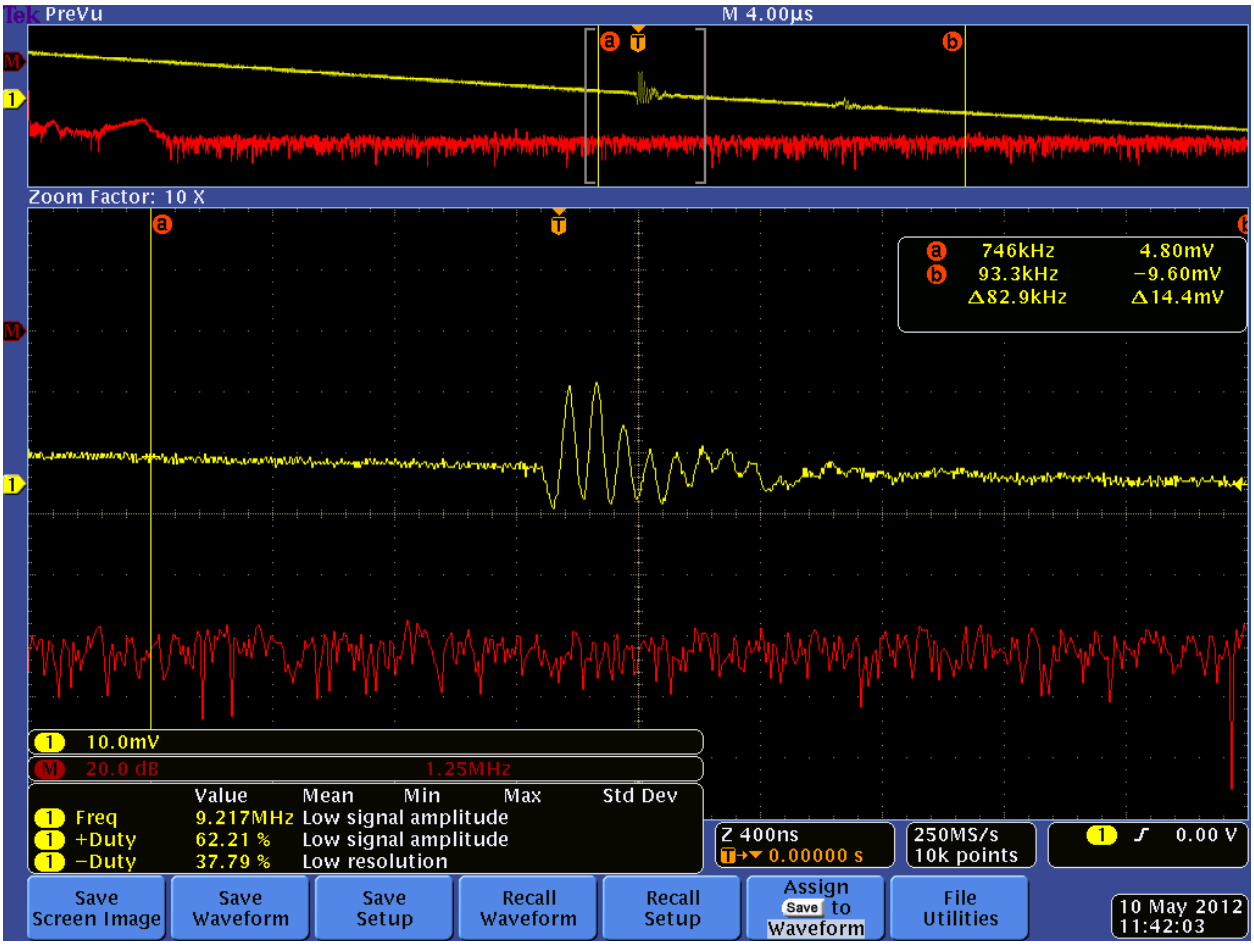

Fig 5.2Vibration Response With Crack

Chapter 6

CONCLUSIONS

The following conclusions were drawn:

1.With the presence of Crack in the shaft the frequency of Vibration decreases.

2.Crack produces more oscillation of the shaft hence the amplitude of Vibration increases.

3.The above information can be used to predict the failure of a shaft in a rotor system and preventive steps can be taken

4.Experimental observation proves the theoretical analysis

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