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# Experimental Investigation of Vibration Analysis of Multi-Crack Rotor Shaft

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**Abstract**

In recent years, the dynamic behaviour and diagnostic of cracked rotor have been gained momentum. In literature, several studies are available for cracked rotor systems, however very few authors have addressed the issue of multi-cracked rotor system. This paper deals with the nonlinear dynamic behaviour of multi cracked rotor system, which is analyzed experimentally and analytically with the considerations of the effects of the crack depth, crack location and the shaft's rotational speed. A new extension of Lagrangian method is used for analyzing the dynamic behaviour of a multi-cracked rotor system through Umbra Lagrangian formalism. The effects of crack depth on the shaft's stiffness and natural frequencies are analyzed experimentally. Natural frequencies have been obtained through vibration analyzer using impact hammer test under static conditions. This analysis also includes the dynamic response of rotor with breathing crack by using data acquisition system called OROS. It has been noticed that the stress concentration on the first crack has increased due to the presence of the second crack. Another interesting phenomenon is the influence of one crack over the other crack for mode shapes and for threshold speed limits. All such analysis has been carried out on experimental test rig consist of a symmetrical system, which has mild steel shaft between a pair of identical self-aligned double groove high speed bearings. The experimental results can be further validated with the analytical results.


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*Keywords:* Multi cracked rotor; Crack depth; Impact hammer test

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**Nomenclature**

| | |
|---|---|
| $A_n$ | amplitude of rotor |
| $R_c$ | damping coefficient of dissipative coupling |
| $\Omega$ | excitation frequency |
| $\Omega_n$ | natural frequency of the rotor shaft |
| $\Delta k$ | change in stiffness due to crack |
| $L$ | length of the rotating shaft |
| $\mu_i$ | internal damping of the rotor |
| n | mode number and |
| $\lambda$ | position of the discrete damper |

## 1. Introduction

Heavy loaded rotating components of various machines (such as steam and gas turbine, pumps, generators and high speed compressor etc.) being diversely used in various fields such as aircraft, automobile and power generation. Rotor shaft is considered as one of the important part of various rotating machines. Due to manufacturing defect or cyclic loading, fatigue crack frequently appears in rotating shaft. Fatigue crack is considered as one of the main reasons for catastrophic failures in rotating shaft. In thermal machinery such as steam turbines, thermal stresses and thermal shocks are also accountable factors for high stress intensity which is also a cause of crack initiation and its propagation. This failure may cause human injury, great economic loss and equipment loss etc. Due to this reason, cracked rotors have attracted the attention of researchers since the last 50 years.

Important progress has been shown in the last 40 years in cracks detection. Since 1970, a large number of research papers on crack detection and its effects have been published. Various review papers have also been published periodically by various researchers like Wauer [1], Gasch [2], Dimarogonas [3], Sabnavis et al. [4] etc. Some recent papers were presented by Papadopoulos [5], Kumar et al. [6] and Sekhar [7]. Sekhar and Balaji [8] worked on dynamic analysis of a rotor system considering a slant crack in the shaft. They considered FEM analysis of the rotor bearing system for flexural vibrations. Reduction in the eigen frequencies of all the modes with an increased crack depth has been observed. For the same crack location, it is also observed that change in eigen frequencies is smaller in slant crack compared to transverse crack. Sekhar [9] has worked on vibration characteristics of a cracked rotor with two open cracks and applied finite element analysis on a rotor system having two transverse open cracks for flexural vibration. He noticed that in the case of two cracks of different depths, the larger crack has more significant effect on the eigen frequency.

In present study, the effect of crack depth on the shaft's stiffness and natural frequencies are investigated using experimentally. Natural frequencies have been obtained through vibration analyzer system (OROS system) system using impact hammer test under static conditions. This work also includes the dynamic response of rotor with breathing crack, which has been addressed in this paper by using the data acquisition system called OROS.

## 2. Methodology

The concept of extended Lagrangian Hamilton mechanics has been used to diagnose the dynamic of cracked rotor system by Mukherjee et al. [13, 14 and 15] and Rastogi et al. [16]. Umbra–Lagrangian density function has been used to describe the motion of such continuous rotor system instead of Umbra–Lagrangian by Mukherjee et al [17].

As a case study, one dimensional rotor shaft with internal and external damping along with transverse crack in middle span has discussed. The rotor is symmetrical, which is to be driven by a constant speed source through a dissipative coupling by Kumar [18]. Further, another crack has been generated and an amplitude equation has been established considering extension of Noether's theorem over manifolds.

The given amplitude may be written as:

$$A_n = \sqrt{\frac{2R_c\left[\Omega - \min\Omega_n\left\{1+\frac{2\Delta kL^3}{\mu_i n^4\pi^4}Sin^2\frac{n\pi}{L}\lambda\right\}\right]}{\xi_n\Omega_n}} \tag{1}$$

Where $\xi_n = \frac{\mu_i n^4\pi^4}{L^4}$

## 3. Experimental Study

### *3.1. Experimental setup*

Experiment study has been conducted to analyze the multi-crack system. The schematic diagram of experimental setup for stiffness measurement is shown in Fig. 1. Test rig shown in Fig. 2 has been used for experimentation and analysis is carried out using OROS-36. Various components of the Test rig are shown in Fig. 2. All the components are connected to each other as per requirement so that they can share output signals generated during the experiment. In experimental testing, one may use symmetrical system which has a mild steel rotor shaft between a pair of identical self aligning bearing. The shaft has constant circular cross section which is fitted with a DC motor by using a flexible coupling to reduce the vibration transfer from a motor to the rotor. The maximum rotation speed of the motor is 6000 rpm. Impact hammer test is also carried out to find the natural frequency in static condition as shown in Fig. 2 (b).

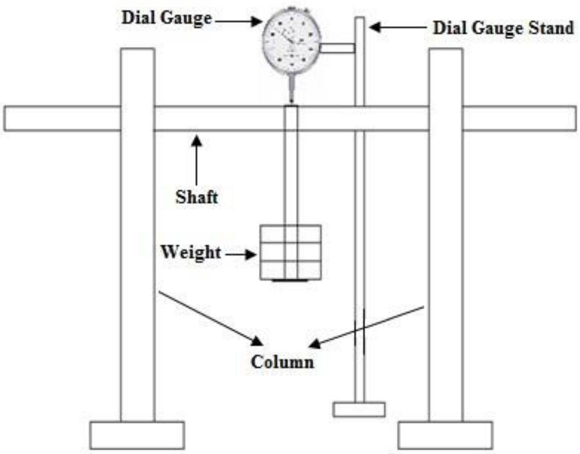


Fig. 1. Schematic diagram of experimental setup for stiffness measurement.

### *3.2. Test specimen*

The material of shaft was chosen based on the criteria that it is referred by American Society for Testing Material Engineering (ASTM) as standard for the shaft material (0.3-0.6%). Test specimens used for analysis are made from mild steel. The schematic diagrams of single crack and multi crack specimens are shown in Fig. 3 and Fig. 4 respectively.

### 3.3. Experimental procedure

Creating the artificial transverse crack is one of the most challenging tasks in the present study. It is made in shaft by using very fine jewel saw. Initially the dimension of crack was considered 1mm depth and width 0.2mm. Therefore, the single crack has a range of depth between 0 to 23% (5mm) of diameter. Moreover, the following crack depths were considered for analysis: 4.54% (1mm), 9.09% (2mm), 13.64% (3mm), 18.18% (4mm) and 22.73% (5mm). First crack was formed at the centre of the two bearing support. Stiffness of the intact shaft and single cracked shaft has been determined on experimental set up as shown in Fig. 1. A known weight is applied on specimen. With the help of dial gauge indicator, deflection of shaft due to known load is measured. After this, using relation '$K=F/x$', stiffness has been determined for various crack depth. Shaft has been divided into 12 sections on the circumference having 30° angular interval simultaneously. The arrangement of shaft at first position is shown in Fig. 5.

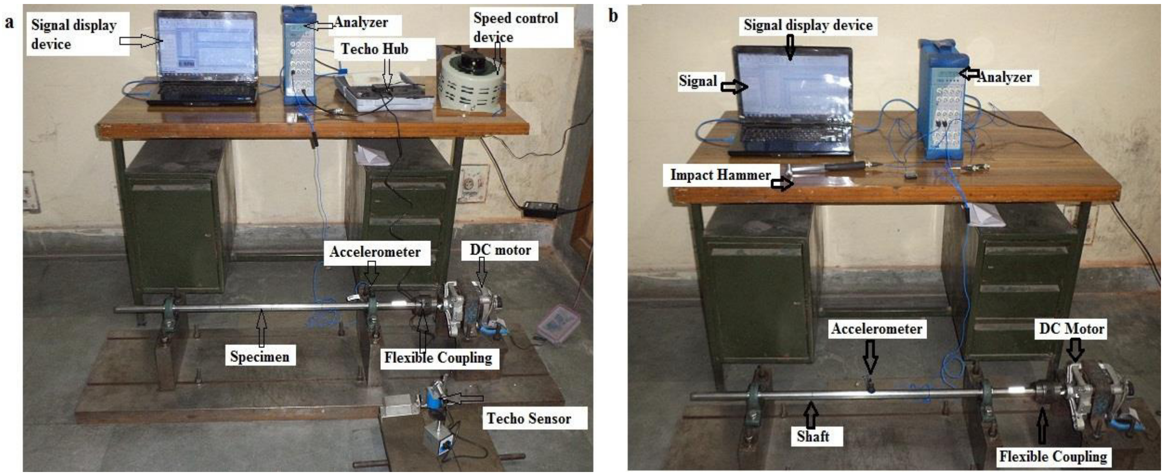


Fig. 2. (a) Experimental Setup of Dynamic Test; (b) Experimental Setup of Impact Hammer Test.

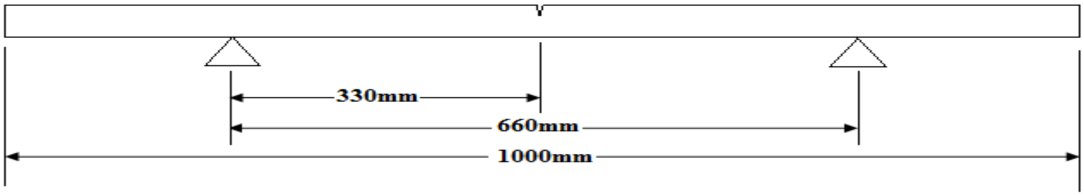


Fig. 3. Schematic diagram of single crack test specimen.

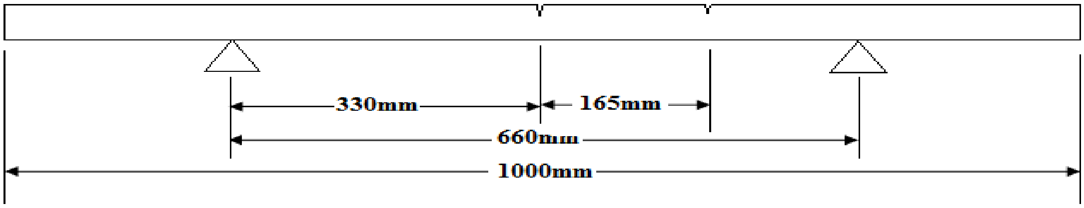


Fig. 4. Schematic diagram of multi crack test specimen.

The arrangement of second transverse crack is shown in Fig. 4. In this case, depth varies in range 0 to 13.64% (3mm) of diameter. Further, the following crack depths were considered for analysis: 4.54% (1mm), 9.09% (2mm), 13.64% (3mm). This crack is generated at the distance of 165 mm from the middle crack. Stiffness of the double cracked shaft has been evaluated through experiments and used in analytical formulation of amplitude equation.

Table 1. Details of test specimen.

| Shaft specification | |
|---|---|
| Shaft material | Steel |
| Effective length ( Bearing to bearing) | 660 mm |
| Overhang ( Each side) | 175 mm |
| Diameter of shaft | 22 mm |
| Weight of shaft | 1.83 Kg |
| Density | 7850 Kg/m$^3$ |
| Ixx and Iyy | 11503 mm$^4$ |
| Stiffness | 285 N/mm |

The shafts are rotated at various speeds in range 1200-5500 rpm, which is used for dynamic investigation. The signal is transformed into signal analyzer which is further analyzed through NVGate® software. It has been found from literature that probes are placed near the bearing or over the bearing for analysing shaft behaviour as per the Tong Zhou et al. [19]. The accelerometer is attached on the top of bearing for capturing a signal and transmitted to display monitor.

Furthermore, impact hammer test has been conducted to find out static natural frequencies. This test performed through equipment OR36 (maximum range 20 KHz) integrate with compact real time multi-analyzer. A single accelerometer is used which mounted at mid of the shaft. Experiment has been conducted on the un-cracked, single cracked and multi cracked shafts. Each specimen (shaft) is hit by the hammer (with ICP coupling) at the centre of the shaft to create a force and vibration which is captured by OROS hammer and accelerometer respectively. For each shaft the input force, the trigger levels and accelerometer response had to be calibrated in the NVGate® analyzer project for measurement and analysis.

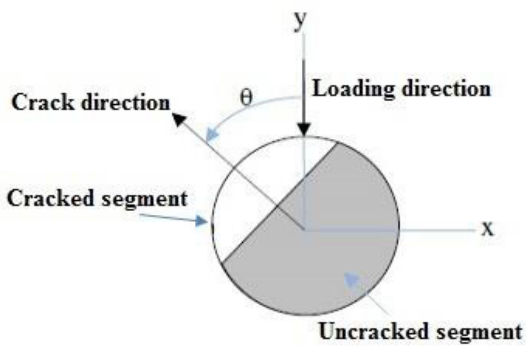


Fig. 5. Cross sectional view of rotating angle between crack and loading direction.

## 4. Results and Discussion

Experimental results are presented in Figs (6-9). Figure 6(a) shows the variation in stiffness with the single crack depth variation. It shows that the stiffness variation increases with increase in crack depth. Figure 6(b) shows the variation in stiffness due to second crack. Different positions of rotating cracked shaft are shown in Fig. 7. For different crack depths, the slope of the curve decreased as the shaft rotated from fully closed ($0^o$ and $360^o$) to partially closed ($90^o$ and $270^o$) to fully open ($180^o$), which is termed as breathing phenomenon.

The variation of stiffness of single cracked rotor and multi cracked (two crack) shafts at different angles are shown in Fig. 8. It is evident from the plots that the stiffness of the shaft decreases with increase in crack depth. In multi crack analysis, the change in stiffness is marginal as compared to single crack propagation. Therefore, stress concentration increases rapidly at one point in case of single crack but the effect of stress concentration at the same point is marginal due to creation of other crack. The peaks indicating the amplitude of acceleration are shown in Fig. 9. These are used to find natural frequencies for cracked and un-cracked shaft.

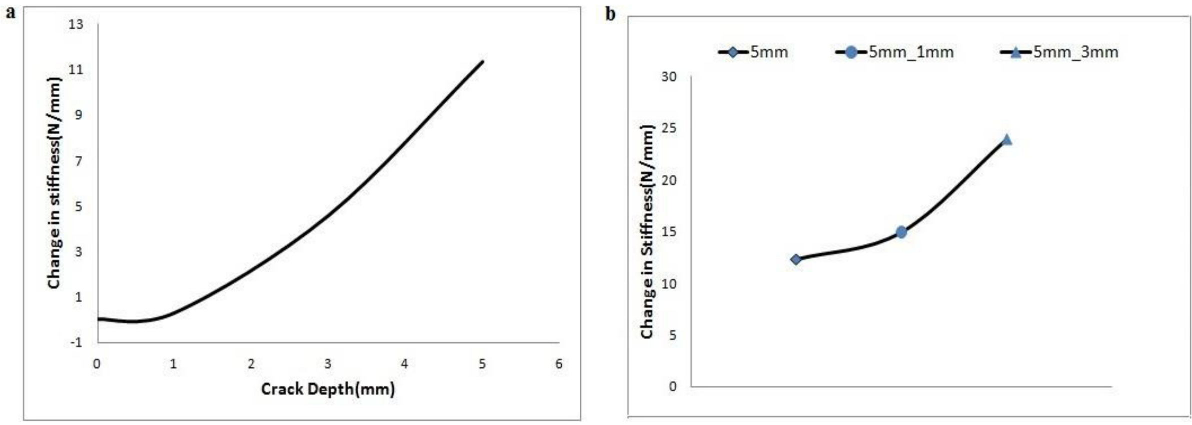


Fig. 6. (a) Variation in stiffness with the single crack depth; (b) Variation in stiffness with the Multi crack depth.

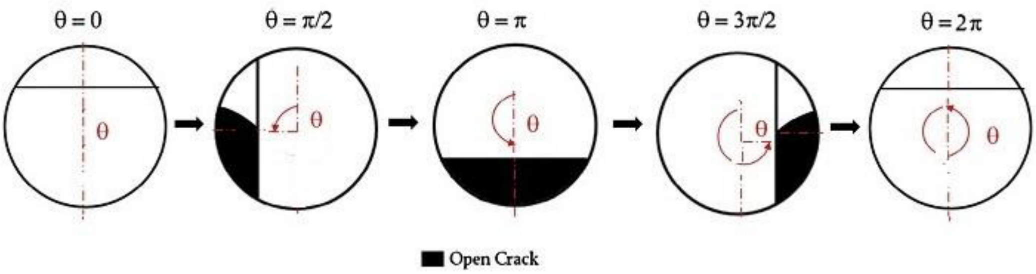


Fig. 7. Some of the angular positions of the crack during one rotation of the shaft.

In multi crack shaft, slope of the stiffness curve decreases due to second crack as compare to single crack shaft. Thus, increase in the stress concentration is marginal near the first (middle) crack as compare to the middle crack depth.

The natural frequencies at various crack depth have also been obtained from Fig. 9. For a single crack, natural frequency of the shaft decreases rapidly, which is clearly observed in Fig. 9(a). In case of multi-crack, the changes in natural frequency are marginal observed. In Fig. 9(b), first natural frequency comes out to be 114.5Hz for intact shaft and 90.5Hz for 1mm cracked shaft, 86Hz for 3 mm cracked shaft, 85Hz for 5 mm cracked shaft. In Fig. 9(b) natural frequency comes out to be 84.5Hz for multi-cracked shaft (5 mm and 1 mm crack) and 83.5 Hz for second case of multi-cracked shaft (5 mm and 3 mm crack).

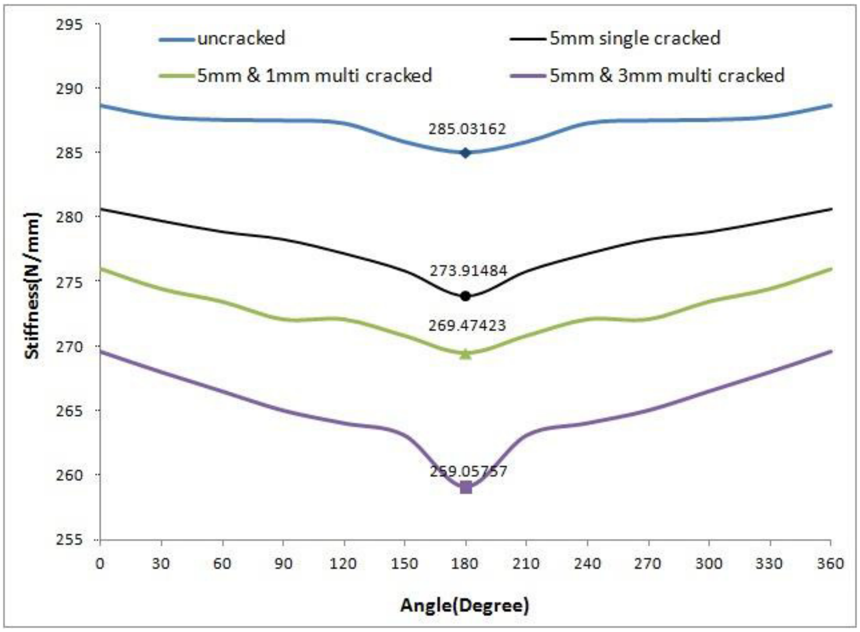


Fig. 8. Stiffness of cracked rotor at different angle.

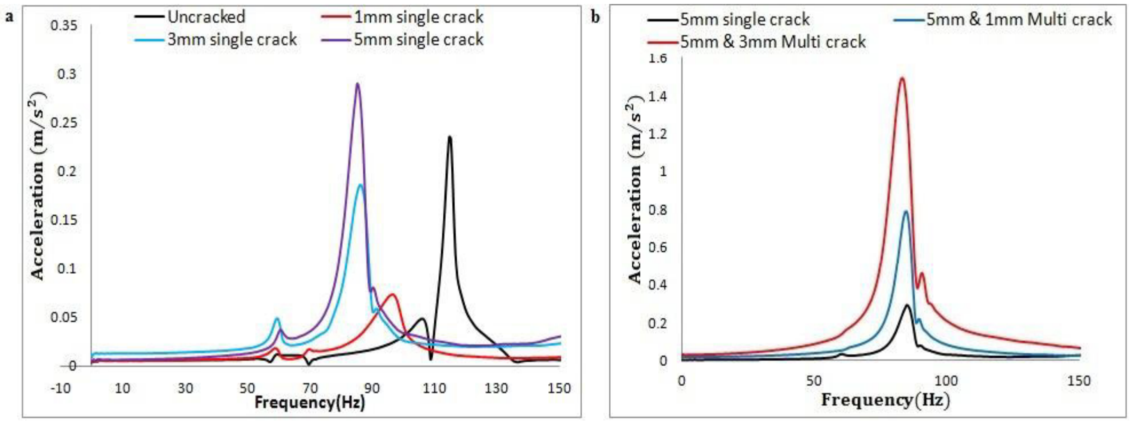


Fig. 9. (a) Frequency Vs acceleration for un-cracked shaft and single cracked shaft; (b) Frequency Vs acceleration for multi-cracked shaft.

## 5. Conclusion

Experimental investigations of vibration have been successfully conducted and validated through analytical equation of crack rotor system. The following conclusions have been drawn:

- Variation in stiffness has been observed clearly with the increase in crack depth whereas the stiffness of un-cracked 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. Stress concentration for the shaft is marginally affected at the middle crack.
- It has been observed that larger crack depth has the more significant effect on the shaft.

- Crack at mid of the rotor shaft has optimum effect on the stiffness as compared to the second crack, created at any other position of the shaft. Position of crack plays a significant parameter in multi-crack rotor detection.
- Since the damaged rotor shaft contains a component of nonlinearity, OROS system can even give better results when may be used for a real system.
- It has been clearly demonstrated that the natural frequency is reduced with the reduction in stiffness which may also cause resonance at very low excitation speed of the motor.
- The use of OROS system for crack detection can be conveniently applied with nonlinear models of multi-cracked rotor system.

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