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Sergey Y. Yurish

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Study of Vibration Characteristics of a Multi Cracked Rotating Shaft Using Piezoelectric Sensor

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Abstract: Dynamic behavior of a structure changes due to presence of crack. Scientific analysis of such phenomena can be utilized for fault diagnosis and detection of cracks in structures. In this paper attempts have been made to detect the cracks of a mild steel rotating shaft. Theoretical expressions have been developed for determining natural frequencies and mode shapes for rotating shaft using flexibility influence coefficients and local stiffness matrix. The numerical results for the beams having 'no' crack, 'single' crack and 'two' cracks are compared. It is observed from the numerical results that, there are appreciable changes in vibration characteristics of the rotating shaft with and without cracks which can be utilized for multi crack identification of structures. *Copyright © 2012 IFSA.*

Keywords: Rotor, Natural frequency, Multiple cracks, Vibrations, PicoScope.

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1. Introduction

In vibration-based identification, a common non-destructive evaluation procedure for detecting a crack location and size is to measure the natural frequency response of the structure with cracks. Since the dynamic behavior of structure changes due to presence of crack. The crack specification such as, crack depth, location of crack, orientation of crack and number of cracks highly influence the dynamic response of structure. The frequencies of natural vibrations, amplitude of forced vibrations and areas of dynamic stability change due to existence of such cracks. Many papers on this topic have published. Irwin [1], has studied a crack on an elastic structural element introduces considerable local flexibility due to the strain energy concentration at the crack tip under load. Due to the presence of crack, the torsional natural frequencies decrease due to the added flexibility [2]. Silva and Gomez [3] have performed experimental dynamic analysis for the location and the depth of cracks in straight beams.

Ruotolo et al. [4] has investigated forced response of a cantilever beam with a crack that fully opens or closes, to determine depth and location of the crack. Tsai and Wang [5] have investigated diagnostic method of determining the position and size of a transverse open crack on a stationary shaft without disengaging it from the machine system. Nahvi and Jabbari [6] have developed an analytical, as well as experimental approach to the crack detection in cantilever beams by vibration analysis. An experimental setup is designed in which a cracked cantilever beam is excited by a hammer and the response is obtained using an accelerometer attached to the beam. Patil and Maiti [7] have utilized a method for prediction of location and size of multiple cracks based on measurement of natural frequencies has been verified experimentally for slender cantilever beams with two and three normal edge cracks. Sekhar [8, 9] applied the theory of model based identification in a rotor system with two cracks. In this work the crack induced change of the rotor system is taken into account by equivalent loads in the mathematical model. Zheng and Kessissoglou [10] have studied the natural frequencies and mode shapes of a cracked beam are obtained using the finite element method. Ertuğrul et al. [11] have studied to obtain information about the location and depth of cracks in cracked beams. For this purpose, the vibrations as a result of impact shocks were analyzed. The signals obtained in defect-free and cracked beams were compared in the frequency domain.

The goal of this paper is to show that the comparison between FFT diagrams found by different specification of rotating shafts and identify the number of cracks present in the shaft.

2. Mathematical Modeling

The presence of transverse surface cracks of depth ‘a’ and at a distance of ‘b_1’and ‘b_2’ from the fixed end on a shaft of diameter ‘D’ introduces a local flexibility which can be defined in a matrix form as equation(1) and (2) respectively.. The geometry of the cracked section is shown in Fig. 1.

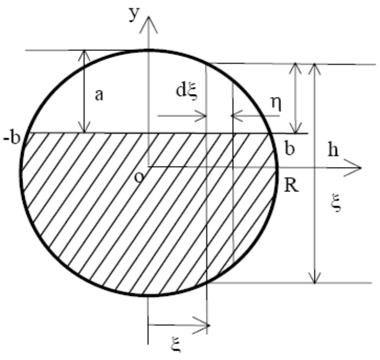

Fig. 1. The Cross-Section of the Cracked Shaft.

Castigliano’s theorem and strain energy release rate the compliance matrix can be obtained. And the local stiffness matrix can be obtained by taking the inversion of compliance matrix. The stiffness matrices for relative crack position b_1 and b_2 are obtained as:

$$K' = \begin{bmatrix} k'_{11} & k'_{12} \\ k'_{21} & k'_{22} \end{bmatrix} = \begin{bmatrix} C'_{11} & C'_{12} \\ C'_{21} & C'_{22} \end{bmatrix}^{-1} \text{ and } K'' = \begin{bmatrix} k''_{11} & k''_{12} \\ k''_{21} & k''_{22} \end{bmatrix} = \begin{bmatrix} C''_{11} & C''_{12} \\ C''_{21} & C''_{22} \end{bmatrix}^{-1}$$

Now, the equation of motion in stationary coordinate for the rotor system is

$$M\ddot{z} + C\dot{z} + Kz = F, \qquad (1)$$

where, M, C, K and F are the total mass, damping, stiffness and external exciting force matrices of rotor system respectively. Z is the displacement of the element node.

For the rotor system considered here, each beam element has two nodes and each node has two degrees of freedom representing transverse and deflecting displacements in the corresponding cross-section. Here, only the mode shape in the ξ-axis direction is discussed by assuming the rotor system is rigid supported at the bearing position. The mode shape can be obtained by solving the homogeneous part of (1) without considering the effect of the damping.

$$M\ddot{z} + Kz = 0 \qquad (2)$$

Substituting $x_i=A^i\sin(\omega_i t+\varphi_i)$ we get

$$(-\omega_i^2 M+K)A^i=0, \qquad (3)$$

where ω_i and A^i is the i^{th} nature frequency and eigenvector (mode shape).

3. Experimental Investigation

An experimental set up as shown Fig. 2 is arranged where a motor through coupling rotates the shaft. The instruments used for experimental analysis are Piezoelectric Accelerometer, Signal Conditioner, FFT analyzer and related accessories as shown in Fig. 4, Fig. 6 and Fig. 7. The specimen beams used are mild steel with length of 200 mm as shown in Fig. 3.The calculation of frequency was taken using a portable vibration analyzer to investigate the vibration spectrum of rotating shaft with or without cracks. The vibration signal was received from the Piezoelectric Accelerometer that put contact to the bearing of the rotating shaft as shown in Fig. 3. Then the signal is transformed to the vibration analyzer for analysis by using FFT relation, and the output shown by the digital monitor screen. The outputs are shown in the Fig. 8, Fig. 9 and Fig. 10.

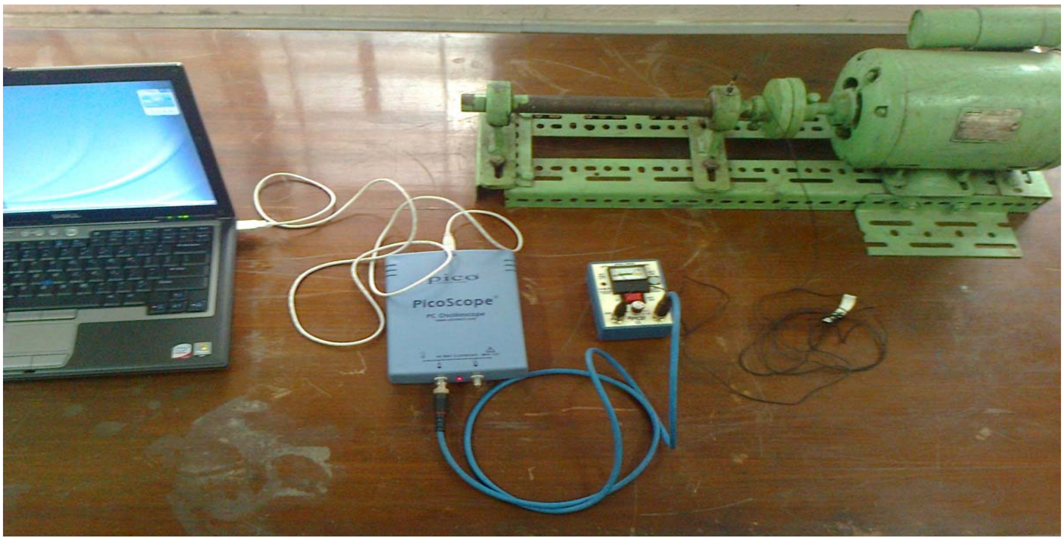

Fig. 2. Experimental Set-up.

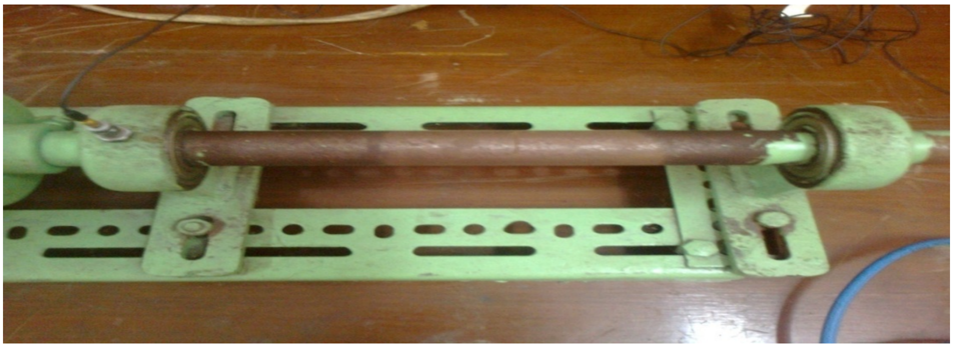

Fig. 3. Cracked Shaft.

3.1. Piezoelectric Accelerometer

An accelerometer is an instrument that measures the acceleration of a vibrating body. The accelerometer can be used for measuring acceleration, velocity, and amplitude of an oscillating body. The most commonly used type of sensor for vibration measurements is the piezoelectric acceleration sensor. Piezoelectric accelerometer as shown in Fig. 4, operate on the principle that a mass moves will stresses a piezoelectric crystal Fig. 5. Acceleration forces cause the amount of applied force on the crystal to vary cyclically and this, by the varying nature of the piezoelectric effect, produces an electric output signal proportion to compression. Since this stress is it directly related to acceleration, such a signal is proportional to the vibratory acceleration of the surface on which it is mounted. Since the electric output is charges, a charge amplifier has to be used to convert the signal to voltage signal.

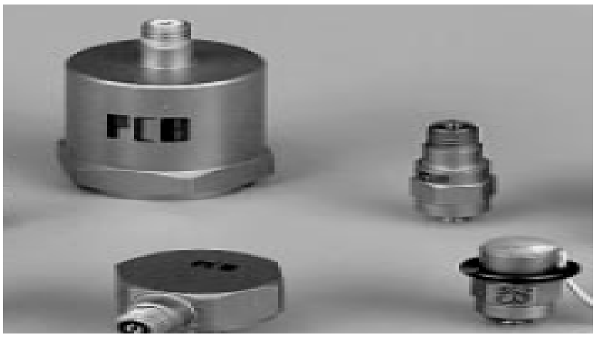

Fig. 4. Piezoelectric Accelerometer.

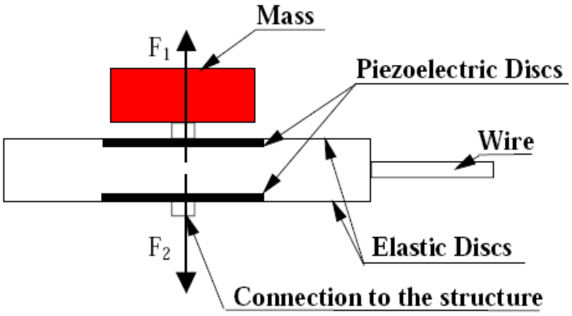

Fig. 5. Schematic Diagram of Piezoelectric Sensor.

3.2. Signal Conditioner

Signal conditioning means manipulating an analogue signal in such a way that it meets the requirements of the next stage for further processing. More generally, signal conditioning can include amplification, converting, and any other processes required to make sensors output suitable for conversion to digital format. It is primarily utilized for data acquisition Fig. 6.

3.3. Analyzer (Picoscope-2202)

An analyzer, Fig. 7 is used to perform the tasks of signal processing and modal analysis using suitable software. The response signal, after conditioning, is sent to an analyzer for signal processing. A commonly used analyzer is called a Fast Fourier Transform (FFT) analyzer.

Fig. 6. Signal Conditioner.

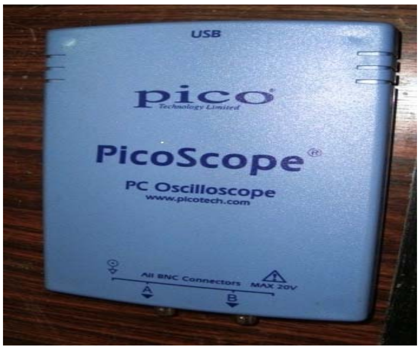

Fig. 7. PicoScope 2202.

Such an analyzer receives analog voltage signals (representing displacement, velocity, acceleration, strain, or force) from a signal conditioning amplifier, filter, and digitizer for computations. The analyzed signals can be used to find the natural frequencies, damping ratios, and mode shapes either in numerical or graphical form. The analyzer converts the analog time-domain signals, x(t), into digital frequency-domain data using Fourier to facilitate digital computation. Thus the analyzer accepts the analog output signals of Piezoelectric Accelerometers or force transducers, x (t), and computes the spectral coefficients of these signals a_0, an and b_n using equations 2, 3 and 4 in the frequency domain. This process is performed by an analog-to-digital (A/D) converter, which is part of a digital analyzer. If x (t) is a periodic function with period τ, its Fourier's Series representation is given by

$$\begin{aligned} x(t) &= \frac{a_o}{2} + a_1 coswt + a_2 cos2wt + \cdots + b_1 sinwt + b_2 sin2wt + \cdots \\ &= \frac{a_0}{2} + \sum_{n=1}^{\infty}(a_n \cos wt + b_n \sin nwt), \end{aligned} \tag{4}$$

where the digital spectral coefficients a_0, a_n and b_n are given by

$$a_0 = \frac{2}{N}\sum_{i=1}^{N} x_i \ ;$$

$$a_n = \frac{2}{N}\sum_{i=1}^{N} x_i \cos\frac{2n\pi t_i}{\tau} \text{ and } b_n = \frac{2}{N}\sum_{i=1}^{N} x_i \sin\frac{2n\pi t_i}{\tau}$$

It is possible to calculate the natural frequencies of the shaft using analysis of data gathered by piezoelectric sensor (PicoScope) connected to the bearing as shown in the Fig. 3. The Fig. 8, Fig. 9 and Fig. 10 show the spectral response of the PicoScope when connected to the bearing of the rotating shaft with un-cracked shaft, shaft having single crack and shaft having double crack respectively.

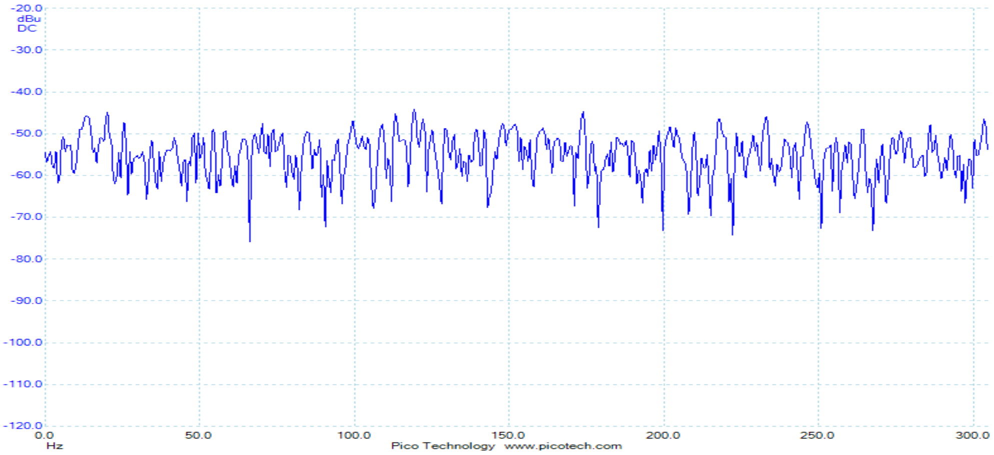

Fig. 8. FFT Spectrum of shaft without crack.

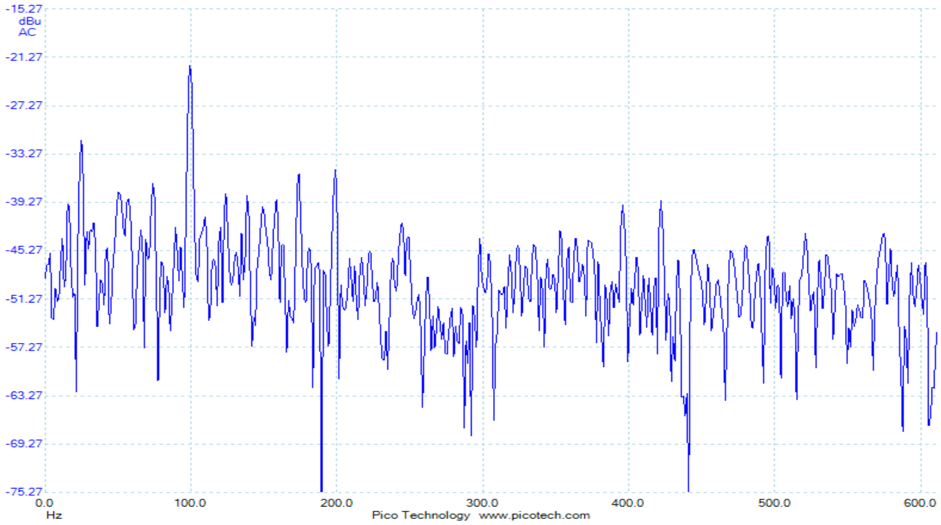

Fig. 9. FFT Spectrum of Cracked Shaft with Single Crack.

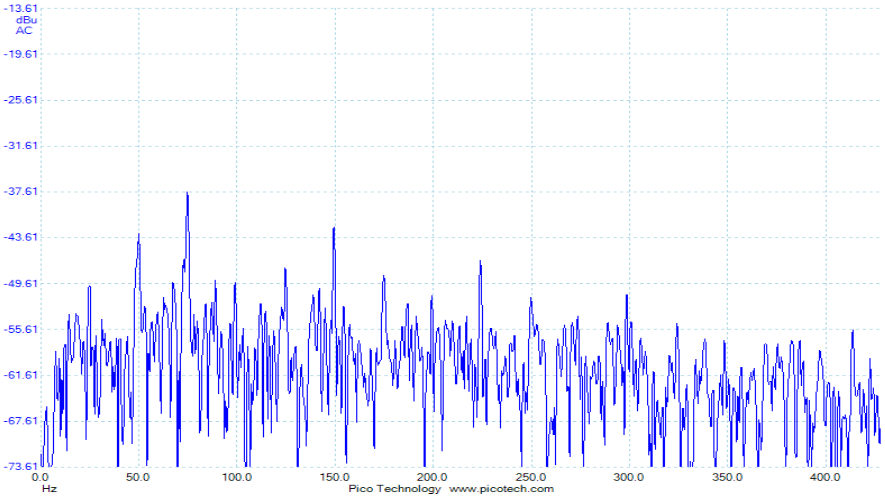

Fig. 10. FFT Spectrum of Cracked Shaft with Double Crack.

4. Result and Discussion

The natural frequency of a rotating shaft found to be considerably influenced by the presence of a transverse crack. The quantitative evaluation of this effect based on the derivation of an equation of motion to derive the formula of calculating the natural frequency of the rotating shaft. Also it is depending on the strain energy function to get the integral relation between the local flexibility and the stress intensity factor. By the present method, it is noticed from the graphs mentioned above that the natural frequency of the rotating shaft will be decreased by increasing the number of cracks. It is interesting to note that frequency at the peak decreases with increasing number of cracks. In case of no crack the frequency is 135 Hz, for single crack the frequency is 100 Hz and in case of double it is 74.74 Hz. Also increasing the crack depth rapidly decreases the values of frequencies. This is done by using a portable vibration analyzer with piezoelectric acceleration which support to the near point of rotating shaft. The crack on the rotating shaft will change in some property like the local flexibility. So the local flexibility of a shaft in bending due to the crack is evaluated from the theoretical and experimental results relating to the derivation of the strain energy release function to the crack depth. These methods can have many practical applications because there is a wealth of analytical results for strain energy release function. In present paper it is noticed that local flexibility increased by increasing the number of cracks and crack depth and this is observed by calculating the theoretical values of local flexibility from the equation derived above. And for the experimental results the calculate the values of natural frequencies of the rotating shaft with and without crack then used the expression which depend on finding the flexibility from the change of frequencies to get the local flexibility.

5. Conclusions

Finally, this paper will represent a technique for non-destructive testing methods depending on, use the vibration analysis and the spectrum of vibration and monitoring it on a screen. So it can be used for identification of the number, location and the magnitude of the crack on a rotating shaft, without direct inspection, even at running conditions. It allows also for continuous monitoring in shaft in service, especially for machine which has welded rotors and frequent inspections are impractical. The analysis of the cracked rotating shaft with longitudinal cracks can be used for further research on the subject. Also the study on cracked rotating shaft under the influence of external forces will be used for future work.

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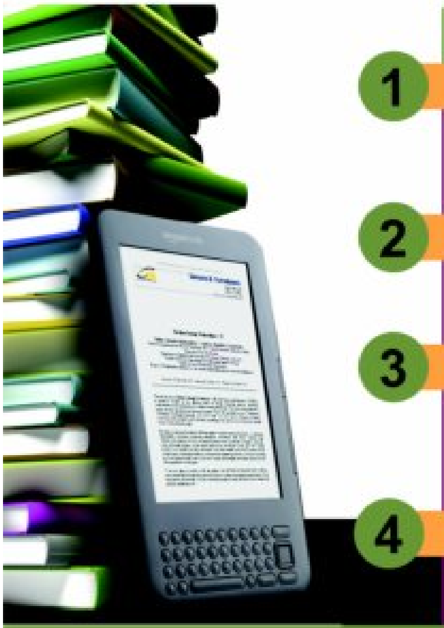

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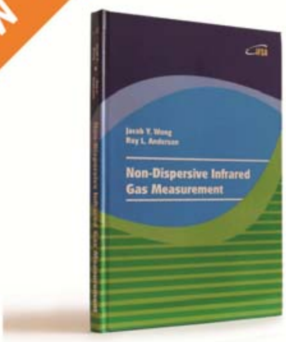

Jacob Y. Wong, Roy L. Anderson

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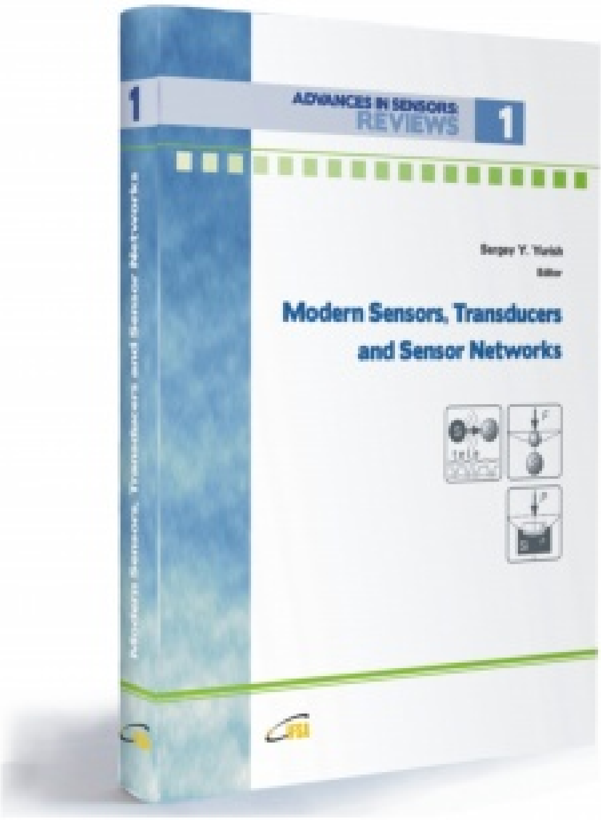

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