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On the Use of Torque Measurements for Detection of Coupling Misalignment in Rotor Systems

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Abstract*— The detection of coupling misalignment in rotor bearing system has been modeled using rotor torque signals in the present paper. This has been accomplished by modeling the shaft and rotor as a flexible members and formulating the equation of motion of the rotor by adding the inertia torque and torsional shaft stiffness torque to the actual external torque acting on the rotor-bearing system. Generally there exists three kinds of misalignment namely parallel, angular and combined. In the present study, torque signals have been used for the detection of parallel coupling misalignment. To accomplish this finite element analysis has been performed on the commercial software package ANSYS 13.0 and the shaft has been modeled using the Solid 185 element. The unbalance forces has been applied on the rotor along horizontal and vertical directions. Further, the vibration acceleration and torque signals have been obtained and the chances of misalignment detection using the torque signals is investigated further. For the four cases of parallel misalignment the vibration and torque signals have been obtained. Signals at different speeds namely 25 Hz and 40 Hz have been used for diagonizing the fault. From results, it is concluded that the misalignment has been well detected from vibration acceleration and torque. Especially, such an identification can well be useful in misalignment detection in gas turbines rotor.

***Index Terms*—Misalignment, faultdiagnosis, Torqueresponse, FFTWavelet.**

I. INTRODUCTION

Misalignment can be classified into three categories namely: parallel, angular and a combination. Attaining perfect alignment between two shafts is very challenging in industrial environment. Even if an accurate alignment is secured, it cannot be continued for a long duration due to many external effects, like base foundation disturbance [1]. Shaft misalignment is a frequently encountered fault, and is a significant source of vibration. Precise detection of shaft misalignment faults can cut the operation cost and facilitate long and safe operation.

Dewell and Mitchell [2] experimentally proved that the vibration magnitude changes significantly at 2×and 4×harmonic components of the operating frequency as the misalignment level increases. It is reported that the frequencies of forces and moment from misaligned shafts are even multiples of the operating speed of the motor by Xu and Margoni [1]. Usually, it is expected that the existence of misalignment will lead to high vibrations with strong higher harmonics. But, it is reported by Protrowski [3] that the higher misalignment may lead to decrease of vibrations. In spite, of the above reported observation, a strong 2× vibration component is commonly accepted for misalignment fault detection.

Sekhar and Prabhu [4] analysed shaft misalignment in rotating components with a flexible coupling using higher-order finite element model by applying theoretically modelled forces and moments. Analysis of other defects with analytical models of rotating systems reported by Hamzaoui et al. [5] also indicated that misaligned shafts generates vibrations such that the magnitude of the 2× operating harmonic frequency component is sensitive to the fault. Lee and Lee [6] introduced a theoretical dynamic model for shaft misalignment in a rotating system with roller bearing connected using a flexible coupling. Sinha et al. [7] introduced a technique to diagnose the shaft misalignment from a shut-down data in which the shaft misalignment at couplings is assumed to develop steady forces and moments. A detailed review on different studies on coupling misalignment is presented by Sudhakar and Sekhar [8].

A brief literature review on wavelet based condition monitoring of rotating systems is presented as follows. Staszewski [9] reported a state of the art review on fault identification using wavelets transform. Tse and Peng [10] evaluated the use of the continuous wavelet transform based approach and the envelope identifications (EI) methods for roller element bearing defect monitoring. Rubbing, oil whirl and shaft misalignment are frequently observed in rotor bearing systems, Peng et al. [11] analysed these 3 categories of typical defects by wavelet based scalograms. A review on recent applications of the continuous wavelet transforms with focus on rotating machine fault detection is done by Yan et al. [12]. Misalignment effect on vibration amplitude of coupled rotors is investigated by Patel and Darpe [13]. Patel and Darpe [14] studied the misalignment type and its influence on the forcing characteristics of flexible coupling followed by experimental investigation. Verma et al. [15] investigated the misalignment and instability of a shaft mounted on bearings.

In some cases the rotor or the coupling assembly may not be directly available for alignment study. Also most of the rotating machines have restrictions on the disassembly. Thus, there is always a need to develop efficient alternative techniques for detecting the shaft misalignment of a rotor bearing system. The vibration response of the component generally provides useful data on working conditions that are closely related to the machinery fault features. Therefore, vibration based analysis has been the very important tool for diagnosing the defects of a rotating component, and thus has drawn attention of many researchers and engineers to understand.

Therefore, vibration based analysis has been the very important tool for diagnosing the defects of a rotating component, and thus has drawn attention of many researchers and engineers to understand the faults. However, due to the confusing spectral characteristics of vibration based misalignment lead to less reliability [3]. In addition there are other rotor faults such as shaft crack lead to similar characteristics (2× vibration component) of misalignment. Hence in the present study, a new strategy for misalignment fault detection is proposed. Here torque signals are used for misalignment detection.

To the best of authors' knowledge, torque responses have not been used for misalignment detection in literature. The authors (Chandra Sekhar reddy and Sekhar) used torque measurements on the experimental test rig for the misalignment detection in a rotor bearing system [16]. In the present study theoretical analysis and simulation for such monitoring are carried out.

The organization of subsequent section is as follows: section 2 describes the mathematical back ground behind the torque fluctuation and coupling misalignment model. In section 3 the details of misalignment theoretical analysis using finite element package ANSYS 13 software is presented, whereas in section 4 the simulated results using the Fast Fourier Transform approach and Continuous Wavelet Transform (CWT) based approach are presented. In the present study in addition to vibration, a technique based on torque modelling for detecting the shaft misalignment problem of rotating components is proposed. Finally, conclusions are reported in section 5.

II. MATHEMATICAL BACKGROUND

2.1 Torque fluctuation model

The theoretical facts lie in the detection of misalignment fault using torque signals is discussed elaborately here. Generally, a typical rotor consists of rigid disks (rotor) fixed on a shaft. The rotor is supported on rolling element bearings. By lumped parameter modeling, in performing the rotor dynamic analysis, it is assumed that the rotor is rigid, shaft is flexible and bearings are represented with discrete springs. Figure 1 represents the schematic diagram of the rotor set up used in the present analysis.

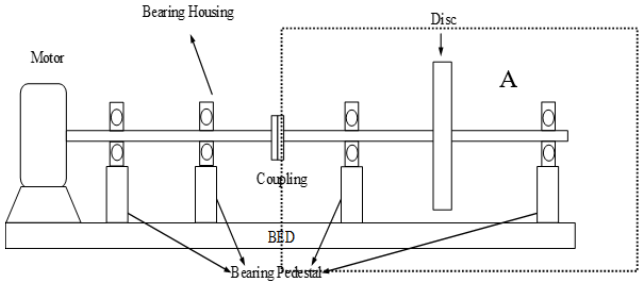

Figure 1: Schematic diagram of the model analyzed

The rotor disc is mounted on the shaft between two bearings. One end of the shaft is connected to motor through coupling as shown in Fig 1. The procedure to obtain the torque fluctuation in a rotor is summarized below.

The rotor is able to rotate freely about longitudinal axis with the externally applied torque from the motor. Hence, generally while representing the equations of motion, vertical and horizontal deformations are represented as vibration of the rotor system to the harmonic input. The portion of the rotor-bearing system in view 'A' in Fig.1 is used in the present analysis. The equations of motion of rotor-bearing system in view 'A' are as follows

$$m\ddot{x}+c\dot{x}+kx=me\Omega^2\cos(\Omega t)$$
$$m\ddot{y}+c\dot{y}+ky=me\Omega^2\sin(\Omega t)$$
$$T_{ext}=T_0\sin(\Omega t) \quad (1)$$

Where x and y represents horizontal and vertical vibration displacements, T_{ext} is the externally applied torque on the rotor. The externally applied torque varies with respect to shaft speed. The externally applied torque amplitude is evaluated at a given speed based on the motor torque speed characteristics. Generally with increase in speed the torque amplitude (T_0) decreases and vice versa.

From the linear displacement equation of motion (Eq.1), it can be concluded that the external applied force on the system (unbalance force which is a function of angular frequency of shaft) is balanced by the damping force, the inertial force and stiffness or internal resistive force of the dynamic system. The resultant or net torque is the difference between the driving torque and the sum of the counter moment and the resistive torque.

With proper modelling of the rotor dynamics, it can be concluded that internal resistive torque $K_T\theta$ and inertia torque $I\ddot{\theta}$ will try to oppose the net externally applied torque. Hence, the modified equation of motion in the rotational degrees of freedom can be represented as,

$$T_{net}=T_{ext}-I\ddot{\theta}-K_T\theta \quad (2)$$

Where T_{net} represents the net applied torque on the rotor bearing system. θ represents the rotational vibration displacement which can be captured in the present analysis by modeling the shaft as a flexible solid member. I and K_T represents the mass moment of inertia and torsional shaft stiffness respectively. Here internal resistive torque due to damping present in the system is neglected. The net or resultant torque as given in equation (2) though exists in rotor system, is usually very small. However, due to faults such as coupling misalignment, this torque can increase. This fact has been checked experimentally by the authors (Chandra Sekhar reddy srikanth and Sekhar). The present study is aimed at proving this fact theoretically.

A brief overview of coupling misalignment modeling is summarized from literature in the following subsection

A. Coupling misalignment modeling

An alignment fault is said to be present when the shafts of driving (motor shaft) and driven end are not on an identical centerline. Shaft misalignment can exist in two types namely; parallel, angular and a combination of parallel and angular cases is also very commonly observed. These possible types of misalignment cases are shown in Fig. 2

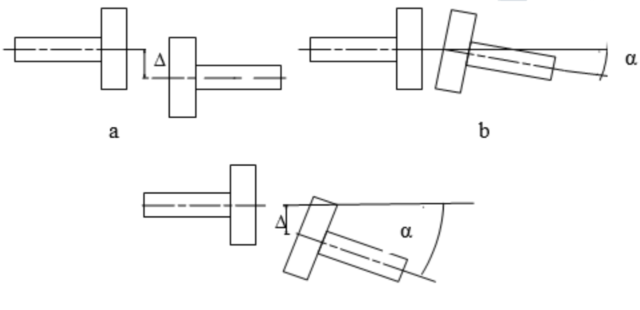

Fig. 2 Shaft misalignment conditions (a) Parallel, (b) Angular, (c) Combined

The most vulnerable parts of a rotor system are bearings and couplings because they are subject to axial and radial forces during operation. Shaft misalignment in rotating components generates both transverse and longitudinal vibrations, and frequently occurs with looseness, shaft bow, inertial unbalance. Due to the presence of misalignment the shaft and the bearing are subjected directional preloads. The coupling present in the system passes this directional load on to the rotating shaft [8]. The presence of measurable response features is dependent on the level of the preload, the shaft operating frequency, the machine type and the critical speeds. The nature of the defect generates a preloading effect on the rotor in transverse direction. This translational force is constant and is a steady state force which always tends to push the rotor sideways. This forces leads to rotation of the shaft in a bent configuration. The normal translational motion of a rotor shaft in a bearing is a circular or an ellipse but any increment in this radial preload results in an elliptical shaped orbits. Shape of the orbit represents the magnitude and direction of the preload.

The reaction forces and moments generated because of the coupling misalignment have been modeled in [4,16,17]. The modeling as given in Sekhar and Prabhu [4] is considered in the present study.

These moments and forces are static with respect to the stationary frame of reference. They act periodically on the operating shafts with a time period of π / Ω with a periodic function of half-sinusoids. In the current analysis, 1Ω, 2Ω harmonic components of the loads are considered and they are substituted into the nodal force vector {Q} as shown in Eqs. below at the corresponding nodes. Nodal force vectors at the corresponding nodes due to the coupling misalignment are given below.

$$\{Q_c^1\} = \begin{Bmatrix} F_{X1}\sin\Omega t + F_{X1}\sin 2\Omega t \\ F_{Y1}\cos\Omega t + F_{Y1}\cos 2\Omega t \\ 0 \\ 0 \end{Bmatrix} \tag{3}$$

$$\{Q_c^2\} = \begin{Bmatrix} F_{X2}\sin\Omega t + F_{X2}\sin 2\Omega t \\ F_{Y2}\cos\Omega t + F_{Y2}\cos 2\Omega t \\ 0 \\ 0 \end{Bmatrix} \tag{4}$$

Based on the above discussed mathematical background, to model the effect of misalignment on the torque response of the rotor bearing system, finite element approach is adopted in the present study. The details of which are discussed in detail in the next section.

III. FINITE ELEMENT MODELING

In the present misalignment model, solid continuum mechanics is used to capture the shear strain fluctuation about the rotational axis. From the dynamic equilibrium equation, the elastic shear strain is a function varying with time and dependent on displacement components. From equation (2) it is clear that the net torque depends upon internal resistive torque and inertia torque. At a given point the internal resistive torque depends upon time varying elastic shear strain. Hence the torque fluctuation at any point can be well captured by evaluating the elastic shear strain about the rotational or longitudinal axis. The rotor system is modeled as the equivalent structural model subjected to harmonic unbalance and misalignment forces.

Of all the components of rotor bearing system shaft is more flexible than bearing and rotor hence in the present analysis flexible solid shaft member is considered. The commercial software package ANSYS 13.0 is used for finite element modeling. The unbalance force is applied to the rotor as a harmonic input and since the shaft should be able to rotate freely along longitudinal axis; proper constraints are applied along the longitudinal axis. The linear Uz degrees of freedom are arrested at the bearing centre point only and across the bearing cross section Ux, Uy degrees of freedom are arrested besides Uz.

When compared with inertia of the rotor disc, the inertia of the shaft can be neglected, hence, the inertia of the entire rotor system is considered as the inertia of the rotor disc. The torsional shaft stiffness is calculated from the principle of pure torsion. The rotor always subjected to external torque and there exists resistance from inertia torque, and internal resistive torque.

In case of torsional loading for better capturing of rotational vibration displacement (elastic shear strain along longitudinal axis), it is recommended to model the shaft member with solid modelling. The finite element analysis is carried out in commercial analysis package ANSYS 13.0 software environment. To model the shaft, SOLID 185 is used. Figure 3 represent the schematic diagram of the finite element model (solid model) of the shaft.

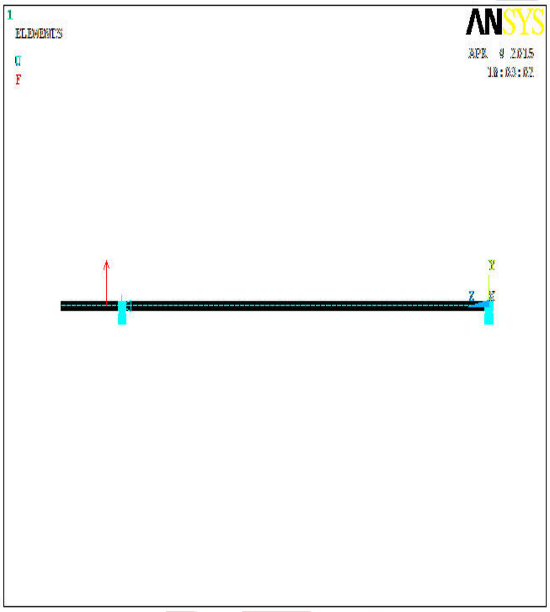

Figure 3: Finite element model of the flexible shaft with boundary conditions

In the present study, in the steady state condition, the torque fluctuation and vibration acceleration have been captured with and without misalignment at 25Hz and 40 Hz rotating frequencies of the rotor. This is achieved by exciting the shaft with unbalance and/or misalignment harmonic forces there by evaluating the rotational displacement fluctuation about longitudinal axis with respect to harmonic inputs. The torque fluctuation and vibration acceleration are captured in case of zero misalignment case at both 25Hz and 40Hz rotor frequencies by exciting the flexible shaft to harmonic inputs due to unbalance. The detailed modeling of parallel misalignment cases are studied at 25 Hz and 40 Hz for 150, 250, 400, 550 microns parallel misalignment. The procedure of misalignment modeling is summarized in the following section.

3.1 Zero misalignment vibration acceleration and torque signal

The finite element model is created in the commercial software package ANSYS 13 and for the zero misalignment case the vibration signal can be obtained as follows. The solid element used for the present analysis is SOLID 185. In zero misalignment condition the only excitation force to the dynamic system is the unbalance vector sum $me\omega^2$. Hence for steady state case, a rotor can be modeled by the standard method available in the rotor dynamics module in ANSYS 13 or it can be treated as a structure subjected to the harmonic excitations. However the effect of rotation can be directly captured through $me\omega^2$ term.

Hence, in the present analysis to get the acceleration signal or response, the flexible shaft member is subjected to harmonically excited force. The transient dynamic analysis is performed to get the acceleration signal of the system in steady state. The obtained time domain signal is converted to frequency domain by FFT through commercial MATLAB package. Later time-frequency technique, wavelets are also used to represent the misalignment results.

The same dynamic model discussed in the previous section is used to capture the fluctuation in the torque signal by evaluating torsional displacement (elastic shear strain along longitudinal axis) at each time step in the transient dynamic analysis. By subtracting the inertia torque and internal resistive torque from the externally applied torque, the net torque value at each time step can be obtained in the transient dynamic analysis of the flexible shaft. This time domain signal is converted to frequency domain to capture the torque variation.

3.2. Misalignment vibration acceleration and torque signal

The dynamic model discussed in section 3.1 is used to capture the misalignment effect on the rotor-bearing system. The harmonic forces and moments due to misalignment are calculated as per discussion in section 2.2. These forces and moments (1X and 2X components) are applied on the shaft end at center point and further the vibration acceleration and torque signals with misalignment are obtained. The vibration and torque signals with and without misalignment are analyzed and presented in detail in the following section.

IV. RESULTS AND DISCUSSION

The physical details of the rotor bearing system considered in the present study are given in Table 1. Parallel misalignment is assumed in vertical direction in the system.

For four cases of parallel misalignment (150, 250, 400 and 550 microns) the vibration and torque signals are obtained. Signals at different speeds 25 Hz and 40 Hz are used for diagnosis of misalignment fault

Table 1: Rotor bearing system details

Feature	Value (units)
Rotor Shaft (Steel)	
Length	0.9m
Diameter	0.016m
Density of shaft material	7800 kg m^{-3}
Young's Modulus (E)	200GPa
Disk	
Diameter	0.14m
Mass	1.5kg
Thickness	0.012m
Motor	
Speed range	0-5000 rpm
Motor Power rating	5.0 HP DC
Ball element bearing	
Stiffness	8×10^7 N/m

4.1 Parallel misalignment diagnosis

The flexible shaft member is excited with harmonic unbalance force and misalignment forces. The dynamic analysis is performed with implicit Newmark time integration algorithm. The vibration system is excited with harmonic forces and transient dynamic analysis is carried out up to 5 seconds with 0.001 time step interval. In the initial response, since the system has free vibration decay for misalignment analysis, the time domain signals are used from 2 to 5 seconds.

The time domain signal is converted into frequency domain with FFT to obtain the characteristic load frequencies of the dynamic system. From FFT it is observed that clear peaks are observed at 1X and 2X components.

For four parallel misalignment cases in addition to FFT analysis, the wavelet scalogram of the time data has also been analyzed for both vibration and torque responses. These are presented in figures (4-8), for the case of rotor speed of 25Hz. Here, a two-dimensional contour plot of the scalogram is used, together with the classical time and scale representations. The scalogram can capture the time varying features of the signal and separate the harmonic frequency components. Similarly for 40Hz operating speed of the rotor the Wavelet and Fourier spectra of signal are presented in Figs (9-13).

It is observed from the different FFT plots that the 2× component of (accelerometer) vibration signal increases with increase in misalignment level. Similarly in the torque spectrum, the 2× component increases with misalignment but not very significantly as compared to that of vibration signals.

The frequency resolution is adjusted using the center frequency of the Morlet mother wavelet to clearly distinguish the spectra. The chosen center frequency (F_C) is 1 Hz and is able to filter the harmonic components. Lower choice of center frequency will lead to spectrum overlap and less frequency resolution. A unique feature observed from wavelet representations is the continuity and clarity of the signal with respect to time. This signifies that the vibration and torque signals are steady and hence qualifies well for measurement, as seen from Figs. (4-13).

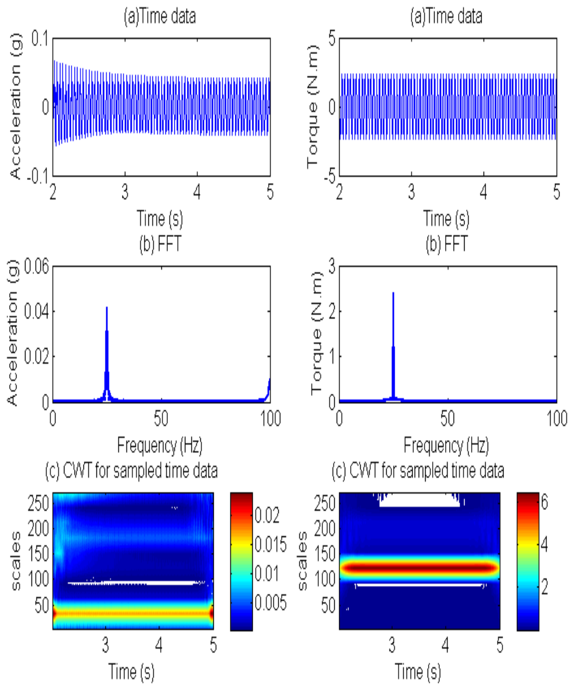

Figure 4: No misalignment case for rotor speed at sensor based diagnosis (i) Vibration and (ii) Torque

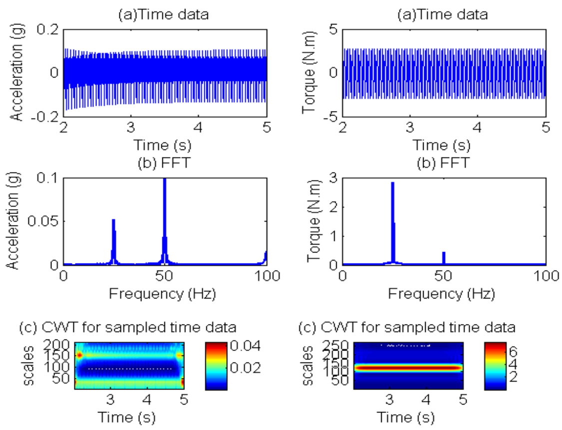

Figure 5: Comparison of accelometer and torque sensor based diagnosis: 150 um misalignment case for rotor speed of 25Hz (i) Vibration and (ii) Torque

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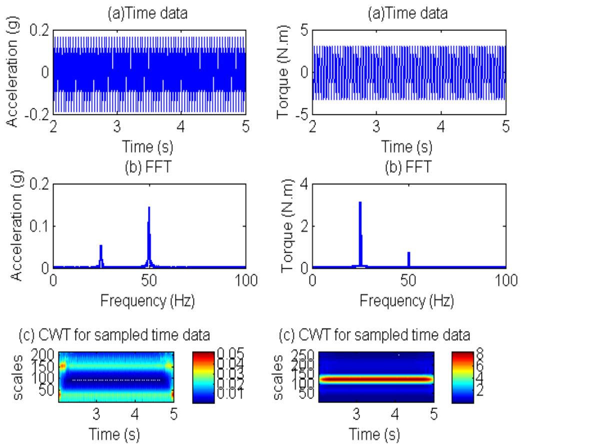

Figure 6: Comparison of accelerometer and torque sensor based diagnosis: 250 µm misalignment case for rotor speed of 25Hz (i) Vibration and (ii) Torque.

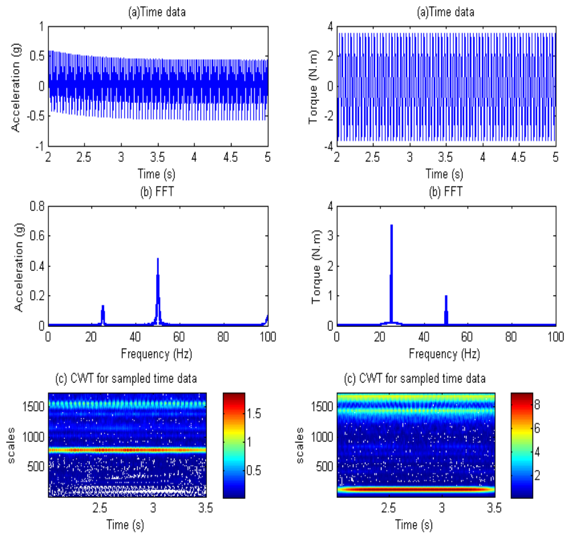

Figure 7: Comparison of accelerometer and torque sensor based diagnosis: 400 µm misalignment case for rotor speed of 25Hz (i) Vibration and (ii) Torque.

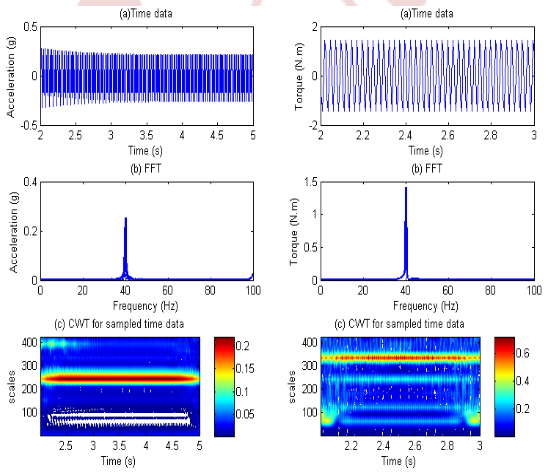

Figure 8: Comparison of accelerometer and torque sensor based diagnosis: 550 µm misalignment case for rotor speed of 25Hz (i) Vibration and (ii) Torque.

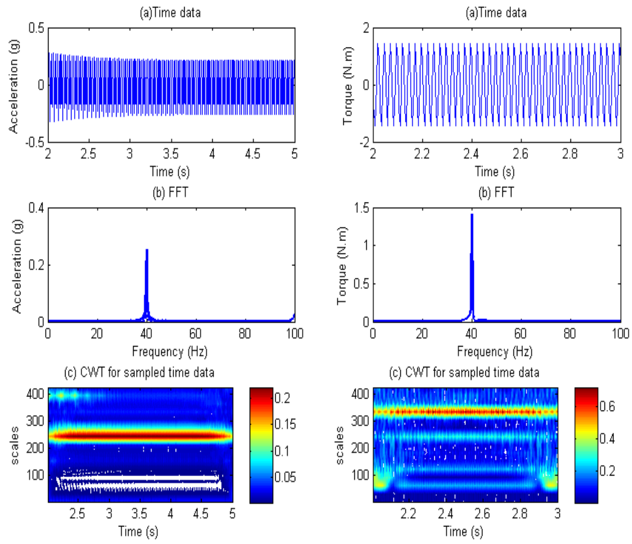

Figure 9: Comparison of accelerometer and torque sensor based diagnosis: No misalignment case for rotor speed of 40Hz (i) Vibration and (ii) Torque.

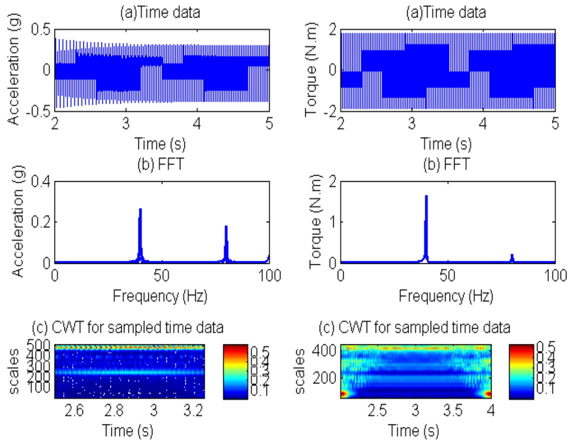

Figure 10: Comparison of accelerometer and torque sensor based diagnosis: 150 µm misalignment case for rotor speed of 40Hz (i) Vibration and (ii) Torque

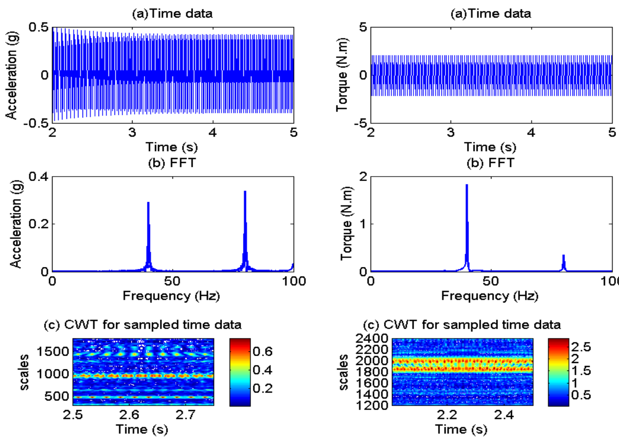

Figure 11: Comparison of accelerometer and torque sensor based diagnosis: 250 µm misalignment case for rotor speed of 40Hz (i) Vibration and (ii) Torque

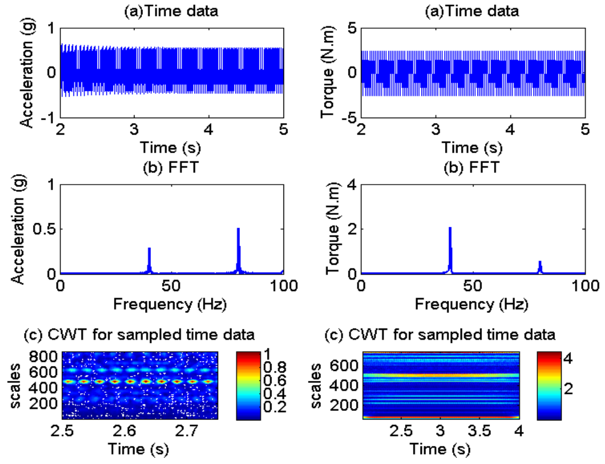

Figure 12: Comparison of accelerometer and torque sensor based diagnosis: 400 μm misalignment case for rotor speed of 40Hz (i) Vibration and (ii) Torque

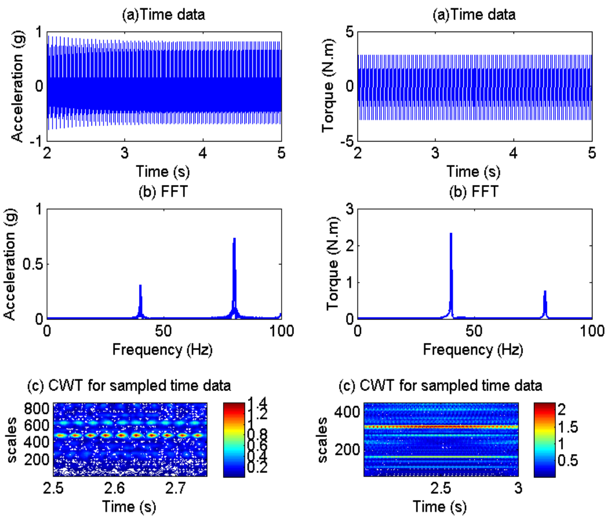

Figure 13: Comparison of accelerometer and torque sensor based diagnosis: 550 μm misalignment case for rotor speed of 40Hz (i) vibraton and (ii)Torque

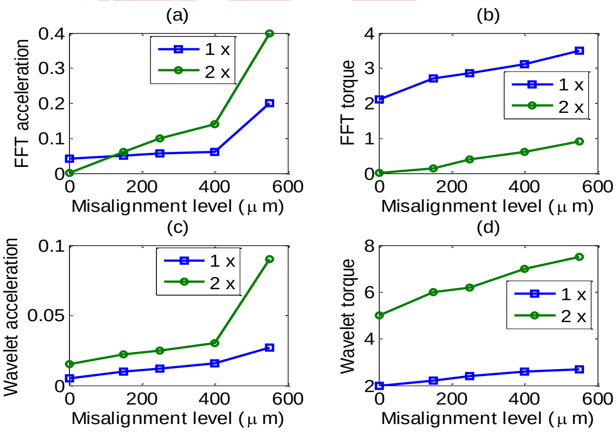

Figure 14: For 25Hz misalignment, the trend of 1× and 2× amplitudes of signals (a) FFT acceleration, (b) FFT Torque (c) Wavelet acceleration and (d) Wavelet Torque

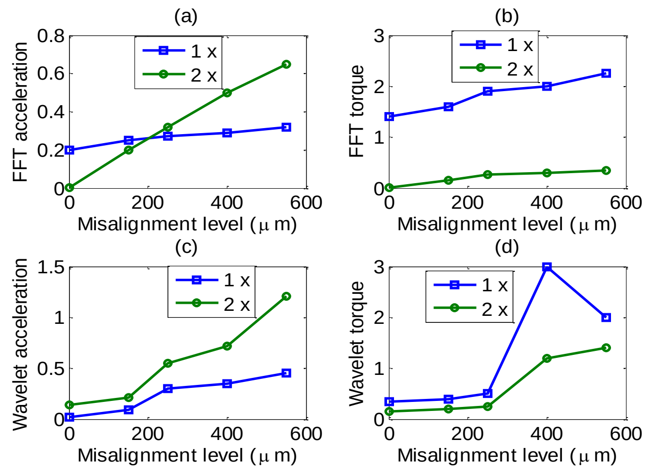

Figure 15: For 40Hz misalignment, the trend of 1× and 2× amplitudes of signal (a) FFT acceleration, (b) FFT Torque (c) Wavelet acceleration and (d) Wavelet Torque.

In the case of rotor speed of 25Hz and parallel misalignment, the trend of 1× and 2× amplitudes of acceleration and torque signals are presented in Figs. (15). It can be observed from the Figs. (15) that the 1 × component of torque can also be a significant indicator of misalignment fault. Also it is observed that the trend of wavelet amplitudes for different misalignment cases is steeper than the FFT trend, so the wavelets representation is more sensitive to the misalignment fault and is useful for monitoring the fault. In the case of rotor speed of 40Hz and parallel misalignment, the trends of amplitudes are shown in Figs. (15) and similar observations are noted. The acceleration and torque spectra indicate the misalignment characteristics. The comparison of both signals shows that torque signals are also quite useful for detecting the misalignment fault. This can be used as an additional and/or alternate technique for misalignment identification.

V. CONCLUSIONS

In this paper, an alternative method of using torque to detect misalignment is presented. The rotor dynamic analysis has been performed by accounting unbalance and torque loads. The shear strain and torsional acceleration have been obtained in time domain. Further, vibration acceleration and torque have been used for parallel misalignment detection. From results, it is concluded that the parallel misalignment can well be detected using both vibration and torque signals. It is also observed that with the existence of misalignment 2X higher harmonic is observed clearly in the frequency spectrum. Generally, the vibration sensor is much cheaper than torque sensor. Though this is a valid statement in lab scaled model, in the actual gas turbines models this can be compensated easily with product cost. Another advantage of torque signals is the torque is same on the rotating shaft at any place where as vibration will change along the gas turbines. Hence, by comparing the two signals torque signals is strongly recommended for misalignment detection in real case gas turbines

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