

**ONLINE MONITORING AND DETECTION OF ROTATING
CRACKED-SHAFT**

BY

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A Thesis Presented to the
DEANSHIP OF GRADUATE STUDIES

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS

DHAHRAN, SAUDI ARABIA

In Partial Fulfillment of the
Requirements for the Degree of

MASTER OF SCIENCE

In

MECHANICAL ENGINEERING

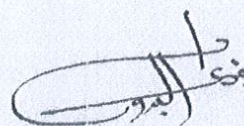
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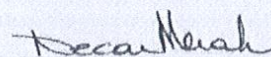
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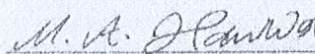
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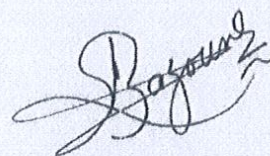
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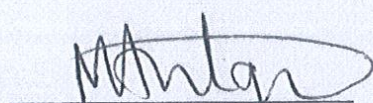
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To my family for their patience, understanding and unwavering support during the long hours I spent working on this thesis. I am grateful to have you by my side.

To my advisor Dr Fadi, who has guided me, challenged me, and inspired me to push the boundaries of my knowledge and abilities. I am forever grateful for your wisdom and support.

ACKNOWLEDGMENTS

| First and foremost, I would like to express my deepest gratitude to my advisor, Dr. Fadi, for his invaluable guidance, expertise, and support throughout the course of this research. Your patience, encouragement, and insightful feedback have been instrumental in shaping my work, and I am truly grateful for the opportunity to learn from your wealth of knowledge and experience.

I would also like to extend my heartfelt thanks to my thesis committee, professors and faculty members who have contributed to my academic journey. I am grateful for their invaluable input, constructive criticism, and encouragement, which have played a crucial role in the development of this thesis.

To my family, words cannot express how grateful I am for your unwavering love, support, and encouragement.

Lastly, I would like to acknowledge my friends and colleagues who have been an integral part of this journey. Your camaraderie, shared experiences, and support have not only made this thesis possible but have also made the journey enjoyable and unforgettable one.

This thesis would not have been possible without the support and encouragement of all the people mentioned above, and I am truly grateful for each and every one of you. |

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LIST OF ABBREVIATIONS

FFT	Fast Fourier Transform
STFT	Short time Fourier Transform
CWT	Continuous wavelet transform.
DWT	Discrete wavelet transform.

ABSTRACT

Full Name : [Obadah Ghazi Sharief]

Thesis Title : [ONLINE MONITORING AND DETECTION OF ROTATING
CRACKED-SHAFT]

Major Field : [Mechanical Engineering]

Date of Degree : [December 2022]

Rotating machinery are the beating hart for the industries. Cracks in a rotating shaft have the potential to cause a catastrophic failure, not only to the machine itself, as this may result in fatality for workers within its area. Not to mention the financial losses due to damage to assets or production stoppage. In order to prevent/ minimize similar incidents, critical machines are continuously monitored through a variety of sensors to ensure adequate of their operation. Cracks are one of the common fatigue related problems, which must be detected early before the failure/fracture occurs. Techniques for crack detection have gained some momentum in the past few years. In this study, the effect of a slant crack size on a rotating shaft lateral and torsional vibrations were investigated to identify the most sensitive measurement method for crack signature extraction and size quantification. During lateral vibration analysis, the overall signal vibration was observed to have a notable increase on the side containing a 40% depth crack. This crack signature was characterized by a rise in both RMS and peak-to-peak values of the signal. As a result, proximity probes could successfully detect a 40% crack depth, but were unable to do so at shallower depths.

In contrast, when examining torsional vibration, conventional vibration techniques failed to reveal any significant correlation between vibration response and crack presence,

regardless of the tested speeds and crack depths. By employing advanced signal methodologies, such as scalograms, short-time Fourier transforms, and wavelet transforms, the signal was accurately captured, including the pulse generated by rubbing. However, these methods also did not provide any substantial evidence of crack existence. Consequently, with the current setup for slant cracked shafts, there is no discernible response up to 40% crack depth that can be directly linked to the presence of a crack. A few sources of error were discussed in this work including the shape of the base of the crack that was flat unlike normal cracks which should have sharp edges. This, among other sources of error, might have prevented the study from reaching a definitive indicator of the existence of the crack. |

ملخص الرسالة

الاسم الكامل: عباده غازي شريف

عنوان الرسالة: المراقبة الفورية والكشف عن التصدع في الاعمدة الدوارة

التخصص: هندسة ميكانيكية

تاريخ الدرجة العلمية: ديسمبر 2022

تعتبر الماكينات الدوارة قلب الصناعات النابض. تمتلك الشقوق في اعمدة الدوران إمكانية تسبب فشل كارثي، ليس فقط للآلة نفسها، حيث قد يؤدي ذلك إلى وفاة العمال المتواجدين في منطقتها. عدا عن الخسائر المالية الناجمة عن تلف الأصول أو توقف الإنتاج. لمنع أو تقليل حوادث مماثلة، يتم مراقبة الآلات الحرجة بشكل مستمر من خلال مجموعة متنوعة من المستشعرات لضمان كفاءة تشغيلها. تعد الشقوق واحدة من مشكلات التعب المشتركة، والتي يجب اكتشافها مبكرًا قبل حدوث الفشل أو الكسر. لقد اكتسبت تقنيات اكتشاف الشقوق بعض الزخم في السنوات القليلة الماضية. في هذه الدراسة، تمت دراسة تأثير حجم الشق المائل على الاهتزاز الجانبي والتواء عمود دوار لتحديد أكثر طرق القياس حساسية لاستخراج توقيع الشق وتحديد حجمه. خلال تحليل الاهتزاز الجانبي، لوحظ أن الاهتزاز الإشارة الكلي يميل إلى الارتفاع الملحوظ على الجانب الذي يحتوي على شق بعمق 40%. تميز توقيع هذا الشق بزيادة كل من قيم RMS وقيم الذروة إلى الذروة للإشارة. ونتيجة لذلك، تمكنت المشابك القريبة من اكتشاف عمق الشق بنسبة 40٪، لكنها لم تتمكن من القيام بذلك على أعماق أقل. بالمقابل، عند فحص الاهتزاز الشامل، فشلت تقنيات الاهتزاز التقليدية في الكشف عن أي ارتباط كبير بين استجابة الاهتزاز ووجود الشق، بغض النظر عن سرعات الاختبار وأعماق الشق.

من خلال استخدام منهجيات إشارة متقدمة، مثل المقياس الزمني، وتحويلات فورييه القصيرة الأمد، وتحويلات الموجة المتلاشية، تم التقاط الإشارة بدقة، بما في ذلك النبضة التي أنشأها الاحتكاك. ومع ذلك، لم توفر هذه الطرق أي دليل جوهري على وجود الشق. وبالتالي، مع الإعداد الحالي لعمود مشقوق مائل، لا يوجد استجابة ملحوظة تصل إلى عمق الشق بنسبة 40% يمكن ربطها مباشرة بوجود الشق. تم مناقشة بعض مصادر الخطأ في هذه الورقة بما في ذلك شكل قاعدة الشق الذي كان مسطحًا على عكس الشقوق العادية التي يجب أن تكون حوافها حادة. قد يكون هذا بين مصادر الخطأ الأخرى قد منع الدراسة من الوصول إلى مؤشر حاسم لوجود الشق. |

CHAPTER 1

INTRODUCTION

1.1 Background

A crack in a rotating shaft would initiate due to cyclic loading around stress risers. If went undetected, the crack would propagate until a critical size where material is no longer capable of handling the stresses, and brittle fracture may occur. Rotating machines run at high speeds and thus contain a huge amount of kinetic energy. When the shaft fractures, the debris would project with all this energy, which sometimes is sufficient to penetrate the casings of the machine and even penetrate the surrounding walls. This could easily cause fatality to any personnel in the vicinity not to mention the financial impact such a failure could have due to stopped production and the burden of restoration. Cracks usually go through three stages that happen in a consecutive manner which starts with crack initiation then followed by crack propagation and ends with the final fracture [1]. General classifications of the cracks include the crack shape, depth, and orientation. However, there are several crack detection methods that can be divided into two main categories. The first one is the offline detection methods which include radiography testing, ultrasonic, dye penetrate test, magnetic particle, and eddy current testing. The major factor of these tests is that they are done offline meaning the machine should be taken off service and stripped down to do the test. The second category of testing is online detection methods. Which is

mainly using sensors, and signal acquisition and analysis tools. Vibration analysis techniques are one of the commonly used methods. Additionally, several signal processing techniques; including the ones discussed in this study, such as fast Fourier transform, short time Fourier transform and even going through advanced techniques such as wavelet transforms, are utilized for fault signature identification and quantification.

1.2 Problem Statement

Crack detection and diagnosis are vital for keeping machines running safely. In the open literature, there are many studies that investigated the crack diagnosis from different aspects. Experiments were conducted with the sole purpose of identifying the existence of cracks in rotating shafts. Most of the experimental works in this area were directed on transverse or radial cracks, and their effect on cracked shaft lateral vibration, using displacement or acceleration measurement. Some analytical and numerical models were also carried out to understand the cracked shaft dynamic behavior.

1.3 Research Objectives

The main objective of this investigation is to study the effect of slant crack size on the shaft lateral and torsional vibrations and identify a crack signature that could be made as a general indication of existence of slant cracks in rotating shafts. The additional value to the literature is using torsional vibration as literature is lacking in this regard. The objectives of this research work are:

- Identifying the significance of utilizing shaft torsional vibration measurement in detecting shaft and/or quantifying (size) over shaft radial displacement.
- Design and development of the experimental setup; including test rig, operating conditions (i.e., rotational speeds, startup coast down), identification of crack dimensions and vibration measurement sensors and data collection (sampling frequency).
- Perform the experiment and collect vibration measurement data for healthy shaft (as a reference) and cracked shafts of varies crack depth.
- Develop advanced signal processing techniques for detecting slant cracks in a rotating shaft based on time-frequency and wavelet transforms.
- Study superiority of developed techniques with respect to convention time and frequency signal analysis in terms of groove detection capabilities.

1.4 Methodology

Before the beginning of the study a literature review was conducted to know the state of the art in crack identification methods.

- Manufacturing of the disc and shafts with different crack sizes. In addition, setting up the test rig and connecting the required sensors and connections. Then taking the measurements.
- The experiment was conducted, and data were acquired and then analyzed to compare the significance of using the shaft torsional vibration over the radial displacement.

- Collecting the data in order to study it and to compare the original shaft with the cracked ones and also study the significance of using the torsional vibration over the radial displacement.
- Develop advanced signal processing techniques for detecting slant crack in a rotating shaft based on time-frequency and wavelet transforms.
- Study superiority of developed techniques with respect to convention time and frequency signal analysis.
- Develop finite element model to validate the result from the experiment.

1.5 Thesis Outline

Literature review was carried out to start the study where types of cracks, detection methods, and data gathering were discussed extensively in chapter 2 in addition to some background about the digital signal processing techniques that were used in the study such as FFT and short time Fourier transform. Experimental setup and design of experiment are discussed in chapter 3 with elaboration on the details of the experiment starting from the test rigs and their specification, sensors, data acquisition, crack specification, rotating speed, and ending with shaft specification among the details of the experiments. The beating heart of the study is chapter 4 where the results are presented, analyzed, and discussed. Lastly, the wavelet analysis techniques were used to further analyze the signal. In chapter 4 also, section 4.5 where a finite element model was developed to validate the experimental result. At the end of this thesis comes chapter 5 which summarizes the result and draws the conclusion of this study. |

CHAPTER 2

LITERATURE REVIEW

This literature review is comprised of four parts. The first part gives some necessary background about the field of vibration including cracks classifications and detections. The second part of the literature review focuses on the experimental works available in open literature. Part three focusses on analytical and numerical modeling of cracked shafts. This section is further divided into two sections where the first one is for the analytical works and the second one is for finite element models. Part four shades some light on the signal processing techniques incorporated in this study. Finally, section five gives some theoretical background about the issue of cracks in vibrations.

2.1 Vibration analysis

2.1.1 Types of cracks in rotating shafts

There is no formal definition of a crack in a shaft. However, a crack could be thought of as discontinuity or fracture on the material of the shaft. Cracks go through three stages which are initiation, propagation and then fracture. The crack initiates due to high local stresses concentration due to design or manufacturing defects. Then, the crack will propagate under cyclic loading. Until the crack reaches the critical size at which the shaft cannot handle the rising stresses. At this point the shaft will fracture. [1].

Cracks in rotating shaft could be classified based on their direction and shape. The cracks that are perpendicular to the axis of rotation of the shaft would normally be called

transverse cracks. Whereas cracks that are parallel to the axis of rotation are classified as longitudinal cracks. Finally, any crack that has an angle to the shaft axis is called a slant crack. Figure 2.5 illustrates the three cracks' orientations. One more crack that is not very common in the literature is the helical crack which has an angle and a curvature along the shaft very similar to a spring [2].

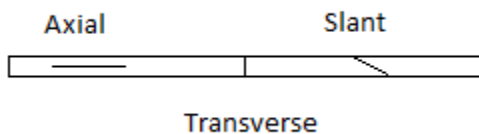


Figure 2.1: Crack orientations

Another way to classify cracks would be their behavior to alternating stresses. A crack that remains open when an alternating stress is applied to it is called an open, notch or gaping crack. Whereas the crack that opens and closes when encountered with alternating stresses is called a breathing crack. Another way for classification would be whether the crack is a surface crack, which appears on the surface, or a sub-surface one, which is not visible [3]. Cracks happen at stress raisers such as keyways and shaft diameter step changes. They normally occur at the coupling and discs, such as impellers and gears. While these effects cannot be always avoided, they can be mitigated by using proper design practices and procedures obtained from internationally credible standards.

2.1.2 Crack detection methods:

Offline detection methods:

There are many methods used for crack detection for static, non-rotating structures. Firstly, Radiography is a method where measurement of the differences in radiation absorption on the shaft material to locate defects. It is used mainly for subsurface cracks detection. Secondly, Ultrasonic, where high frequency sonar is used to send and receive signals and based on the speed and direction of the signal, defects on the shaft could be detected. It can detect both surface and subsurface cracks. However, it is limited if the surface is not smooth or contains some irregularity and coarse-grained materials such as carbon steel creates some challenges because of its low sound transmissibility and high signal noise. Thirdly, in dye penetrate test, where a dye is used to penetrate the surface of the shaft into the open defects and it will color it therefore, the defect could be noticed with the naked eye. However, it is only viable for surface cracks. Fourthly, magnetic particle which is a method where we use iron powder and a magnetic field to locate defects in the material of the shaft. Lastly, eddy current test which is based on magnetic induction. It is mainly used for surface crack detection. [4]

Offline testing and detection methods can give very accurate and detailed understanding of the crack and its characteristics, including size, orientation and location. However, the point of diagnosis and prognosis of equipment is to turn the repair job from a sudden failure to a planned event where the maintenance crew can be fully prepared in terms of spare parts, manpower and services. Offline detection methods require that the machine to be taken down and dismantled for close inspection and testing. While it gives superior information to online detection methods, to do it the machine will be already offline and the operation of the machine or the plant has already stopped to accommodate the inspection.

Online detection methods:

Based on available reported studies in the open literature, online monitoring of cracked shaft is conducted through studying the dynamic behavior of the shaft. Vibration monitoring is the preferred way for detecting the majority of machine defects due to the fact that any dynamical disturbance to the machine will manifest itself in the vibration signal of the machine. To monitor the vibration signal for any machine, firstly sensors are selected based on the expected dynamic response and installed in the proper locations where vibration transmission path is sufficient. Then those sensors should be connected to a data acquisition device to convert analog signal generated by the sensor to a digital format, therefore the data acquisition sampling capability is crucial. After that a computer should be used to store and show the signal. The signal measured could be manipulated using digital signal processing techniques to make sense of the data.

2.1.3 Data gathering and interpretation.

The mechanical parameter that is the subject of analysis should be measured and interpreted to give a clear understanding to the current health or condition of the machine. Vibration is the most used parameter to test and examine the condition of mechanical equipment. Vibration can manifest itself as displacement, velocity or acceleration. Displacement measurement is preferable for the low frequency signals where the acceleration is more suitable for the high frequency signals. Moreover, the velocity is best in the mid frequency range. To understand the vibration, signal a transducer should be installed to measure the mechanical input and convert it to an electrical signal. This signal then is transferred to a data acquisition device which collects the data and send to a

computer for storage and signal processing. Depending on the signal being measured there is a wide range of transducers that could be selected. If the signal being measured is a displacement signal, proximity probes, capacitive probes and position potentiometers could be used to name a few. An example of a transducer that measures velocity would be self-generating low-impedance vibration velocity transducer. Moreover, to measure the acceleration of a signal and accelerometer could be used. Accelerometers have different technologies such as piezoelectric transducers. These devices are conventional contact devices that are being used everywhere in industry to measure axial and radial vibrations. When the contact devices are not suitable for the purpose of measurement laser-based transducers could be used. Such devices would be the laser doppler vibrometer which is a high-tech device that operates under the doppler effect. Another measurement for vibration would be measuring the torsional vibration which is a high-tech method. There are some international standards one can refer to for vibration related issues such as ISO 2954:2012, ISO 7919-3:2009, ISO 10817-1:1998, ISO 13373-1:2002 and ISO 13373-2:2005 to name a few. [4]

2.2 Experimental Work

Hu et al. studied the instantaneous frequency (IF) attributes of crack and rub faults [5]. They also provided the relation between the intra-wave frequency modulation and fast Fourier transform (FFT) spectrum. The intra-wave frequency modulation attributes of the failures are provided by the Hilbert–Huang transform (HHT) analysis of the experimental vibration signal. In the case of rub only, the intra-wave frequency modulation is presented by the instantaneous frequency of the local rub rotor, at its rotational speed. The rotor's

motion period is co-related to the main modulation frequency. Whereas in the case of Crack only, the intra-wave frequency modulations (FM) are seen in both the crack and rub experiments. The faults of crack and rub of the rotor can be differentiated through the composition of the modulation frequencies. In the case of combined Crack and Rub, modulation frequency harmonics in the crack-rub fault are not as much as only rub or only crack fault. It is also seen that the modulation frequency is suppressed for higher frequency harmonics. Guo et al. verified the theoretical model that exhibit the dynamic behavior and the empirical mode decomposition (EMD) of cracked shafts using experimentation. [6] The real crack was modeled by inducing a slit crack and let it propagate in a similar fashion as the real fatigue crack. The $1/2$, $1/3$, and $1/4$ subcritical speeds were studied extensively using the whirl orbits. These subcritical speeds were decomposed into subcomponents using the EMD to study the high frequency components. They also carried out FFT analysis and found that the FFT, EDM from the dynamic analysis and the experiments agree with each other. In figures 2a through 2d the evolution of the orbit of the shaft around the $1/3$ subcritical speed is presented. Two loops show up inside the orbit plot of figure 2 (a) and it gets bigger with the increase of the shaft rotational speed closer to the $1/3$ subcritical speed. These two inner loops start getting smaller after that and keep getting smaller with the increase in shaft rotation speed until they disappear finally in figure d. The Orbit revolves and changes its orientation by about 90 degrees through this process. Similar behavior was also noticed while passing through the $1/2$ subcritical speed with the difference that only one loop appears inside the orbit and the orbit changes its direction by 180 degrees. Additionally, the vibration response of the rotor shows peaks at the $2x$ and $3x$

components as the speed increases and around the 1/2 and 1/3 of the subcritical speed. However, the 1x component does not show this peak.

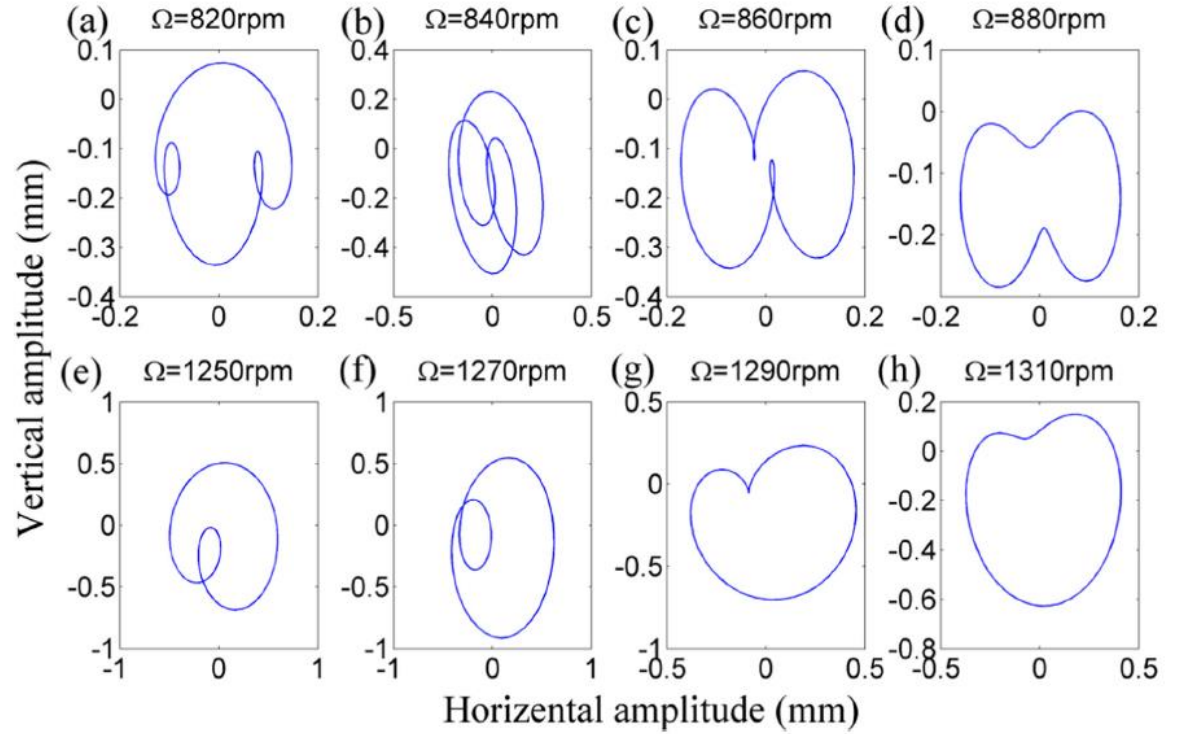


Figure 2.2: The evolution of the orbit of the shaft around 1/3 subcritical speed. [6]

Lebold et al. carried out experimental investigation to demonstrate the viability of using the torsional vibration in detecting cracks in shafts [7]. They collected data from their test set-up and analyzed it utilizing FFT to measure the fundamental torsional frequency. A real fatigue crack was achieved by initiating a slit and let it propagate to simulate the real fatigue crack. Their analysis was based on measuring and comparing the shaft stiffness and the torsional rigidity at different stages as the crack propagates. MTS 319 axial-torsional test machine was utilized to measure the torsional rigidity of the shaft, compared to the non-cracked shaft, after each increment of fatigue cycling. At a rate of 20 lb.in/s a triangular torque waveform was applied twice with fully reversed cycles. Measurement of both the angle and the torque of twist was done during both cycles. Torsional rigidity was calculated

based on the relationship between torque and angle which was determined by using a linear regression method. The experiments result show that as the crack propagates and increases in depth, the torsional rigidity and the fundamental torsional frequency of the shaft will decrease.

Stoisser and Audebert have run different iteration of experiments on rotating machinery in the aim to predict the behavior of the rotor using a lumped crack beam model based on the general problem of elasticity incorporating a crack [8]. They ran different analyses in the model to see the behavior of the model both statically and dynamically. Then By comparing the 2x and 3x components and static deflection, it was found that the experimental results were consistent with the derived model in both the static and dynamic tests. Tlaisi et al. did numerical and experimental study to examine the effect of a crack on a shaft torsional and bending vibrations [3]. They modeled the rotor with both cases of cracked and non-cracked shaft. Their method of analysis was based on observing the change in bending and torsional frequencies and mode shapes in relation to crack depth. They found that monitoring the third mode shape and the first torsional frequency at 1x can be a good method in detecting a crack in a rotor. The engineering innovation software package (LMS test lab) with two measurement channels was used. The first channel was connected to a modal hammer to do modal testing on the stationary shaft during the experiment. The second channel was connected to an accelerometer for measuring the vibration signal. Using these methods, they could detect a crack when the depth reaches 20% of the shaft diameter. Lin and Chu investigated the effect of a crack in a rotor has on the bending vibration of the rotor numerically and experimentally [9]. Two different crack types were studied: a slant crack

and a transverse crack. A torsional excitation was incorporated in the model. The rotation frequency and the excitation frequency were found to be very significant for the slant crack but cannot be identified for the transverse crack. An experiment was carried out to verify the numerical analysis and it was consistent with it. They concluded that investigating the existence of different vibration components in other directions could be used to identify whether the crack is a transverse or a slant crack.

Chasalevris and Papadopoulos studied the coupled bending vibration in a static cracked rotor [10]. They have used Euler Bernoulli beam theory to model the shaft. They observed a horizontal plane response in relation to the crack orientation. Due to the coupling effect all peaks would split into two and will show up in the vibration spectrum representing the crack presence. They carried out experimental analysis to verify the model they have used. Both the model and the experimental result almost agree with others. A small modification to the model had to be made in order for the result to match.

Lopez et. al. studied the effect of different faults in rotors on the vibration response numerically and experimentally [11]. They modeled the rotor based on Timoshenko beam theory using ANSYS. Different types of faults were studied including cracks and some combinations of imbalance, misalignment and bent shaft. In the experimental set-up various excitation were induced to see the response of them with the faults. Their results show marginal change in the natural frequency and mode shape due to cracks when using normal vibration measured by normal sensors and not active sensors. So, they suggest using active sensing, on torsional and axial accelerometers, sensors that require external power source, which they tested in their experiment and found it to be very sensitive to cracks in

terms of detecting them by observing changes in natural frequency and mode shapes while doing modal analysis. Xiang et al. studied the use of laser Doppler vibrometer to measure the torsional vibration of rotors [12]. They conducted experimental and numerical analysis. More than one model was suggested. Some experiments were conducted to verify the numerical analysis. The natural frequencies of different modes were excited and measured experimentally and numerically. Both analyses agree with each other and show consistent results.

Kumar et al. investigated the dynamic response of a cracked shaft experimentally and numerically using finite element modeling (FEM) [13]. They used a Jeffcott rotor and introduced different crack depths on the rotor in order to study their effect on the vibrations. At a depth of 12% and beyond, the change of vibration amplitude when compared to non-cracked rotor was significant. The observed different vibration response between the inboard and outboard bearings were attributed to the coupling effect and the crack position. To further their study an FE model of the set-up was introduced. Using the FE model, a substantial increase in deflection occurred, measured by the difference in vibration amplitudes between different depth ratios, due to the crack at depth of 4mm and beyond compared to non-cracked case. Darpe et al. investigated the vibration response of cracked shafts and their effect on the coupling of longitudinal and transverse vibrations [14]. The rotor was excited, and measurements were made. For non-cracked shaft there is no difference in the vibration response when the axial excitation is present in comparison to the absence of excitation. However, for the cracked rotor the vibration response observed varies significantly when the shaft is axially excited in comparison to no excitation. They concluded that a crack in a rotor could be detected successfully using the axial excitation

as an indication of the presence of a crack in the shaft due to presence of axial excitation frequency on the lateral vibration response.

Mohamed et al. studied the vibrations characteristics of a rotor with two crack types with aim of finding a correlation between the crack and the vibration response [15]. The types of cracks used were a notch cut and a crack growth of fatigue crack. As the experiment commenced different depths of the crack was examined. The propagation of the crack will leave beach marks which were examined at the end of the experiment to match the depth with vibration signal. A shift of peaks to the left was observed in the vibration frequency spectrum with some changes in the amplitudes. They concluded that the changes in amplitude and the shift of frequency are a good indication of the presence of cracks in the rotor. Jain et al. have carried out analytical and experimental study on the nonlinear dynamic's behavior of a shaft with multiple cracks [16]. Their method of analysis was mainly concerned with vibration response of the shaft under different cracks, namely crack location, crack depth the rotating speed of the shaft. A model based on the Lagrangian method was built with an extension. They have examined the response of the shaft under breathing cracks. They focused on the change of stiffness and natural frequency of the shaft as the cracks were introduced and propagated. Sinou has carried out experimental analysis to determine the effect a crack has on rotor vibration [17]. His analysis was mainly concerned with $2x$ and orbital pattern at half the critical speed of the shaft. An increase in the n_x harmonic components was observed in addition to a decrease in the first critical speed. A small increase in the inside loop of the orbital patterns when the shaft speed is around $1/2$ of the first critical speed is a key indicator for the detection of a transverse crack in a rotor. Additionally, the difference between the horizontal and vertical, vertical being

the higher, 2x vibration amplitudes at one half of the first critical speed seems to be an excellent indicator of the existence of a crack in shafts.

Raut et. al. carried out experimental investigation to study the effect of a slant crack on a bare shaft supported by two bearings [18]. They have fixed the crack depth at 4.2 mm and varied the rotational speed and the location of the crack along the shaft. By analyzing the frequency spectrum extracted using FFT, they have found that the amplitude of the vibration peaks will increase as the rotational speed increases. A correlation between the crack location and the vibration response was noticed. The further the crack from the point of measurement the higher the vibrational peak response will be. The shaft with transverse crack has higher vibrational response in comparison with the slant crack. Patil et. al conducted experimental iterations to investigate the effect of a longitudinal crack on a bare shaft supported by two bearings [19]. They placed the crack at a distance of 370 mm from the first bearing and kept changing the depth and rotational speed to examine the effect. By analyzing the FFT they have found that the amplitude of the highest peak of vibration will increase as the rotational speed increases. A correlation between the crack depth and the vibration response was noticed. Increasing the crack depth will also increase the vibration amplitude. Raut et. al. and Patil et. al have found comparable results in which the highest peak of the vibration response will increase with the increase in the rotational speed [18, 19].

Gómez et. al carried out experimental investigation for crack detection in rotating shaft and utilized wavelet packet transforms to analyze the measured vibrations [20]. Wavelet packet transforms (WPT) is generally used to do local analysis on non-stationary signals. In their

work, WPT was used to extract the 3x component from the signal. The aim is to study the detection capability of 3x energy method studied previously in their earlier paper. A shaft having 9 different cracks at different locations and depths ranging from 4% to 50% was tested. Based on their method they conclude that 3x is a robust detection method that can detect cracks beyond 12% depth. In another study, Gómez et. al. have used a combination of the Wavelet Packets transform and Artificial Neural Networks with Radial Basis Function architecture (RBF-ANN) on vibration signals for the use in cracks detection in a rotor system [21]. The system is a rotor consisting of a bare shaft that under its own weight. Data was acquired while the rotor is at steady state condition with varying crack depths. Nine different cracks were induced in the system with depths varying from 4% up to 50% of the diameter of the shaft. An optimization technique was used for optimization of the success rates of the result of the Wavelet Packets transform and RBF-ANN. Running some statistical calculations, they were able to have a probability of detection of almost 100% for 4% crack size with a false alarm of 1.77%. In another study, Gómez et al. investigated the effect of cracks on a rotating shaft vibration signal using experiment and validated using wavelet transform [22]. An experiment was carried out on a rotating machinery with 9 cracks with varying depths between 4% and 50% of the shaft diameter. The cracks were lateral and breathing ones on a Jeffcott rotor. A validation using wavelet packet transform energy analysis has been carried out to the signals generated by the analytical model employed for all test conditions. With the presence of the crack, packets showing bigger differences were observed. These were related to the first three harmonics of the shaft rotation speed 1x, 2x and 3x by shown significant energy increments in their related

wavelet packets. After the analysis they found that only 3x gives a clear indication of the existence of cracks in the shaft with higher energy for deeper cracks.

Tamrakar et al. have done an experimental work and finite element method to investigate the effect of different cracks on the vibration response of an overhung rotor system [23].

The 2x component of the frequency spectrum was found to be increasing due to the crack size. Natural frequency was also studied to see how it is affected by the existence of the crack. Moreover, as the crack depth increases, the natural frequency decreases due to the decrease in the stiffness of the rotating shaft. However, the amplitude of the vibration was found to increase by 40% when the crack depth reaches 50% and beyond. In another study, Tamrakar et al reported that 1x of the overhung rotor changes slightly due to the effect of the crack when compared to the healthy shaft. However, when an unbalance was introduced the 1x changed significantly for the cracked rotor [24].

The aim of doing experimental work on crack diagnosis is to find an indication of the crack or what could be called crack signature. Based on the experimental works reviewed above, there has been a common reference to the 2x component of vibration as a good indicator of crack presence [5, 6, 8, 17, 22], 3x was also reported [6, 20, 22]. Changes of the overall frequency spectrum in terms of frequency and amplitude is a common feature of the existence of cracks [9, 10, 11, 13, 15, 18, 19]. Some researchers [10, 14] reported the coupling effect is a good indicator where the different direction of excitation frequencies would appear in other directions' response, like the axial excitation frequency appearing lateral vibration response. Reduction in stiffness of the rotor and changes on the natural frequency were also observed [7, 16]. Table (1) provide a summary of the above reviewed

experimental work with a focus on experimental setup and diagnostics tools and fault signature.

Table 2.1: Experimental works crack signature

Crack type	Crack size	Vibration measured direction and type (lateral axial torsional) (displacement velocity acceleration)	Analysis tool (Time wave, FFT, orbit cascade...etc.	Findings (signature)	reference
45-degree slant crack	No info	Lateral Displacement	Time waveform, FFT, Orbit, Hilbert spectrum and Marginal spectrum.	2X is the indication of crack	[5]
Elliptical crack	41%	Lateral Displacement	Time wave	2x gives indication	[8]
Slit and propagation	No info	Lateral Displacement	Time wave and Orbit	Super-harmonic in 1/3 and 1/2 subcritical regions	[6]
No info	4-50%	Lateral Acceleration	Waveform and WPT signal decomposition	1x, 2x, 3x energy level rise significantly	[22]
Transverse crack	20-65%	Angular displacement	Torsional rigidity and torsional natural frequency	Natural frequency shifts	[7]
Transverse crack	0-60%	Lateral acceleration	FFT	Changes in amplitude and frequency	[15]

Transverse and open	0-70%	Lateral and torsional displacement	Natural frequency	Effective crack length/effective bending length for the mode) of 1/12 to 1/16	[3]
Transverse and 45 degrees	42%	Lateral displacement	FFT	Changes in frequency	[9]
Transverse and open	0-40%	Lateral displacement and acceleration	FFT	Double peak in the FFT indicating coupling between vertical and horizontal vibrations	[10]
No info	4-24%	Lateral displacement and velocity	FFT	Increase in vibration amplitude	[13]
Transverse crack	0-14%	Lateral acceleration	Stiffness and FFT	Reduction in stiffness	[16]
Transverse	No info	Lateral displacement and velocity	FFT	Presence of axial and bending excitation frequencies in lateral response	[14]
Transverse	10-50%	Lateral displacement	FFT and Orbits	2X frequency indicates	[17]
Transverse and open	12.5%	Lateral acceleration	FFT and power spectral densities	Changes in frequency	[11]

Slant crack	4.2 mm	Lateral displacement	FFT	Increase in Vibrational response	[18]
Longitudinal crack	4.2-12.4 mm	Lateral displacement	FFT	Increase in vibrational response	[19]
Slit	4-50%	Lateral displacement	Wavelet packet transform	Train artificial neural network	[21]
Saw cut	4-50%	Lateral Acceleration	Wavelet packet transform	3x is the best indicator	[20]

2.3 Modeling of cracked shaft and correlation with the dynamic behavior of shaft

2.3.1 Analytical

Saeed and Eissa have applied the Bifurcation theory of periodic motion on a transversely cracked nonlinear Jeffcott rotor [25]. The disc eccentricity, disc weight, nonlinear spring characteristics, the angle between imbalance directions and the crack, breathing of the shaft crack, were all incorporated in the model. The rotor system exhibits a backward and forward whirling motion at certain ranges of spinning speeds. However, as the crack propagates, the backward whirling orbit ceases to exist, and only forward whirling motion remains. Due to the existence of the crack the rotor stiffness coefficient will reduce. The vibrational response amplitude of the rotor system will start oscillating as the crack reaches about 11% depth and beyond with zero-disc eccentricity. Additionally, as the stiffness coefficient continues to decrease with propagation of the crack, two stable periodic solution attractors show up in addition to the trivial solution in the analytical solution when the stiffness drops below 20%. They have conducted analytical and numerical studies, and both agree with each other. Sinou and Lees have constructed a model to study the effects a transverse crack has on a rotor system [26]. They have observed the modal properties and the dynamic response of the rotor in addition to the orbit. The alternate frequency/time domain approach was used because it captures the nonlinearity of breathing crack accurately. Changes in the nonlinearity of the crack at half the resonance speed seem to be the prevalent indication to a breathing crack. Moreover, the changes in the orbit and the appearance of a double loop and an inside loop is a good indicator to the presence of a transverse crack. However, the amplitude of the orbit increases as the crack depth

increases. Also, the shaft speed would decrease and the 2x harmonic component would be maximum. This finding is in agreement with Sinou's finding [17].

Darpe et. al. have studied the coupling of torsional, bending, and longitudinal vibrations of a cracked rotor system [27]. They have used a modified Timoshenko model to take the crack into consideration. Two coupled models were developed: one for the torsional and bending vibrations, the second for the torsional and longitudinal vibrations. Since all the coupling effects have been considered in the model, any changes in a particular mode of vibration would affect the other modes. In addition to the rotational effect of the cracked shaft with the nonlinearities due to the breathing of the crack. All this would create some difference in frequencies. The presence of frequencies from other modes in the frequency spectrum of a certain mode in addition to the existence of the sum and difference of frequencies in the vicinity of the excitation frequencies and their harmonics would indicate the presence of a crack. Figure 2.2 shows the presence of the torsional excitation frequency ω_o in the vertical vibration response of the rotor.

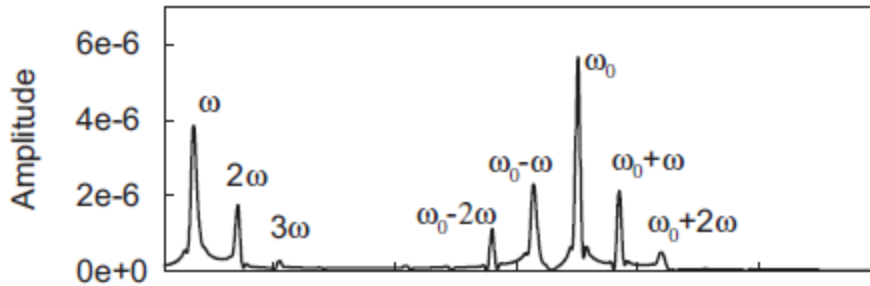


Figure 2.3: Presence of the torsional excitation frequency ω_o in the vertical vibration response of the rotor [27]

Khorrami et. al. have developed a model to simulate the effect single and double breathing cracks have on the vibrational response of a rotor using a modified harmonic balance

method [28]. They have investigated the effect of crack angle, depth and location on the lateral response, critical speed and shaft center orbits of the rotor. Taking into account the location and depth of the first crack, the vibration behavior of a two-crack shaft in a rotor disc-bearing system the existence of a second crack showed an increase of the impact of the first crack on the first critical speed. Moreover, the effect the second crack has is reduced for deeper first crack when compared to shallower first crack.

Darpe has studied the effect of a slant crack on the dynamics of a Jeffcott rotor [29]. Based on flexibility matrix from fracture mechanics, a model for the Jeffcott rotor was constructed. A comparison between the effect of the slant crack and a transverse crack was also carried out. Torsional and imbalance excitations were incorporated into the model. The crack was assumed to be breathing synchronously with the torsional excitation. Frequency component $2x$ shows good indication for crack detection and monitoring. Monitoring the $3x$ frequency component could give an indication of the existence of a propagating fatigue crack and would show whether the crack is slant or transverse. The stiffness of the rotor in both lateral and longitudinal directions is greater for a slant crack compared to a transverse crack. When the crack depth ratio exceeds 0.3, the transverse crack rotor exhibits a decreasing trend in the $3x$ amplitude. In contrast, the slant crack rotor demonstrates a slight departure from the sharp upward trend in both vertical and horizontal directions.

Bovsunovsky has developed a methodology to examine how sensitive or efficient a certain vibrational detection method is in comparison to other vibrational detection methods in

detecting damage in structures [30]. The aim is to find the best method for determining the existence of damage in shafts because different methods are appropriate for different damage types. The methodology was developed by utilizing linear fracture mechanics and relative compliance of the cracked section. The method was applied to a transversely cracked shaft. He found that studying the bending vibrational response is more efficient than torsional vibration response for a transverse crack. There is an inverse proportionality between the slenderness ratio of a rotor system and how sensitive the vibrational detection method to damage is. Moreover, the sensitivity of the vibrational detection method is proportional to the stress level in the crack cross section. Szolc, et. al. have proposed a stochastic approach for detecting and identifying faults in rotating machine shafts, using Monte Carlo simulations to analyze lateral, torsional, and longitudinal vibrations caused by transverse cracks of varying locations and depths. A highly reliable and computationally efficient structural hybrid model is utilized, enabling the creation of extensive databases for crack identification through an inverse problem solution. By assuming arbitrary probability densities of crack parameters, the method ensures a wide range of dynamic responses for the faulty mechanical system. With a sufficiently large database, the technique can estimate crack depth and axial position with identification errors under 9% and 5% respectively, in less than a second. [31]. Gounaris and Papadopoulos have used fracture mechanics techniques to model the crack in a rotor system [32]. They investigated the vibrational effect of the crack on the rotor system. The model was based on Timoshenko beam. The axial vibration and the gyroscopic effect were taken into account because of the coupling effect between the bending and axial vibrations. The excitation in the bending direction will affect the axial response. They studied the contours

of the vibration response and compared the non-cracked shaft response with the cracked one. The result Gounaris and Papadopoulos reached is consistent with Darpe et al [27, 32].

Lin and Chu have studied the dynamics of a shaft with a slant crack by modeling using strain energy approach, it was found that there is a coupling in the stiffness between bending and torsion, bending and axial and torsion and axial [33], the coupling between bending and torsion was due to eccentricity. They used an open crack model because, based on their analysis, there is no significant difference in the frequency response between an open crack and a breathing crack. Although a breathing crack behaves more non-linearly. Analyzing the responses shows that the torsional excitation frequency and the rotating speed in the transverse response in addition to the torsional excitation frequency in the longitudinal direction could give an indication to the presence of slant crack in the shaft. The conclusion that the presence of the excitation of one direction in the frequency spectrum in other direction reached by Lina and chu is consistent with the result Gounaris and Papadopoulos and Darpe et al [27, 32, 33].

Vashish and Peng have studied the effects of nonlinearities by including flexible support and ball bearing, on a rotor dynamics response at steady state and transient [34]. They have concluded that the transient response gives superior clarity and indication. They combined short time Fourier transform and active vibration control into a switching control strategy and used it to study their rotor system. Their proposed nonlinear dynamic model was able to detect the cracks quite accurately. Hu et. al. used the holospectrum technique to detect a crack on a shaft [35]. They claim their method detect three crack parameters which are

location, type and angle between cracks. Using the 3D holospectrum crack location can be detected while 2D holospectrum is used to detect crack type and angle between cracks. Figure 2.3 shows an example of 3D holospectrum. The amplitude of 3D Holospectrum identifies the crack location and the inclination angle of the major axis of 1x ellipse on could be used to identify the crack type while IPV angle could be used to identify the cracked element, and the angle between cracks. Where IPV is the vector pointing from the shaft center to the vector in the orbit pointing at position at $t=0$.

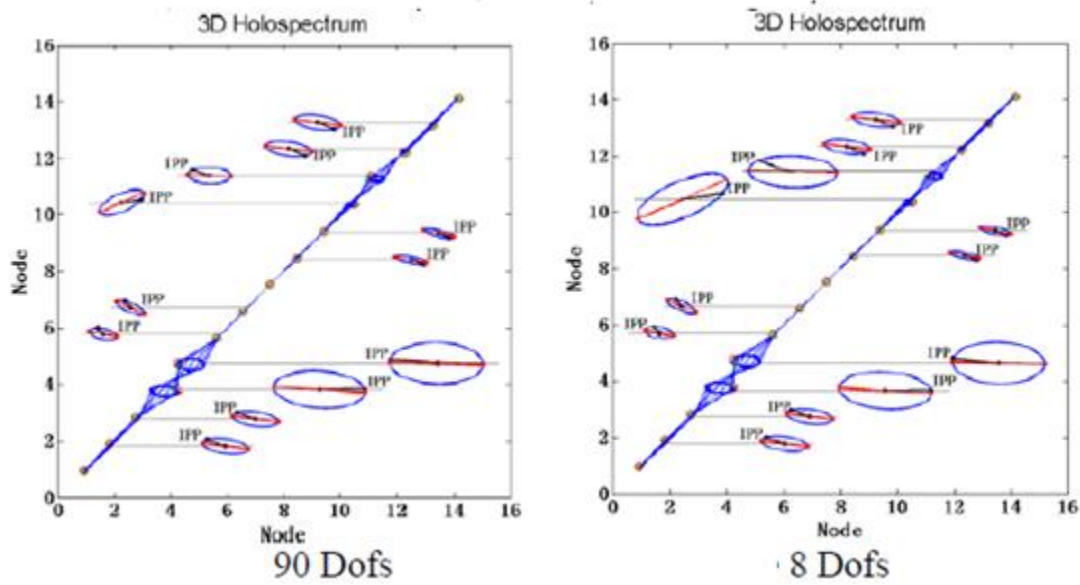


Figure 2.4: 1x HOLOSPECTRUM OF A TRANSVERSE CRACK AND A SLANT CRACK [35]

Wang et al studied the nonlinear and stability response of a rotor system that has a transverse breathing crack, and an initial bending was introduced to the system [36]. One of the findings of this study is that the as the crack size increases the natural frequency would decrease and the vibration amplitudes would increase. Additionally, the increase in the initial bending would increase the range and amplitude of the resonance area. Moreover, several other frequency components appear in the spectrum as the size of the

crack increase such as natural frequency and combined frequency. Khorrami, et al investigated the dynamic effect a crack has on a Jeffcott rotor analytically [37]. The study considered the vertical transient response. The crack is of transverse type that is also a breathing crack. Using empirical mode decomposition, the crack was possible to detect with analyzing the spectrum where the crack would increase the vibrational response. Additionally, the first critical speed would decrease as the crack depth increases due to the changes in stiffness in the rotor. This method was able to detect cracks that are 20% in depth of the shaft and beyond. Xiong et al have studied the coupling effect multi crack rotor system has on the dynamic behavior of the rotor including the crack orientation [38]. One of the findings is that the coupling effect changes the stiffness and would create a nonlinear behavior of the rotor system on the dynamic response. Moreover, the orbits and the harmonics amplitudes are found to be larger for the coupled cracked shaft when compared to the healthy shaft. Mohemmed et. al. have done experimental work to study the effect of multiple cracks on a rotor as seen in the previous section. [15].

2.3.2 Finite element modeling

Modeling using finite element method is a way for solving dynamics problem when the analytical solution is not feasible or possible due to the complexity of the problem. Wang et. al. have developed a finite element model (FEM) for cracked shaft and studied the impact the parameters of the crack have on the super-harmonic components of the rotor [39]. They have used a combination of Kriging Surrogate Model and non-dominated sorting genetic algorithm-III (NSGA-III) [40, 41] as an optimization method to relate the

parameters of the cracks with the dynamic response of the shaft. The method is summarized in figure 2.4. Using their method of analysis, they have been able to identify the parameters

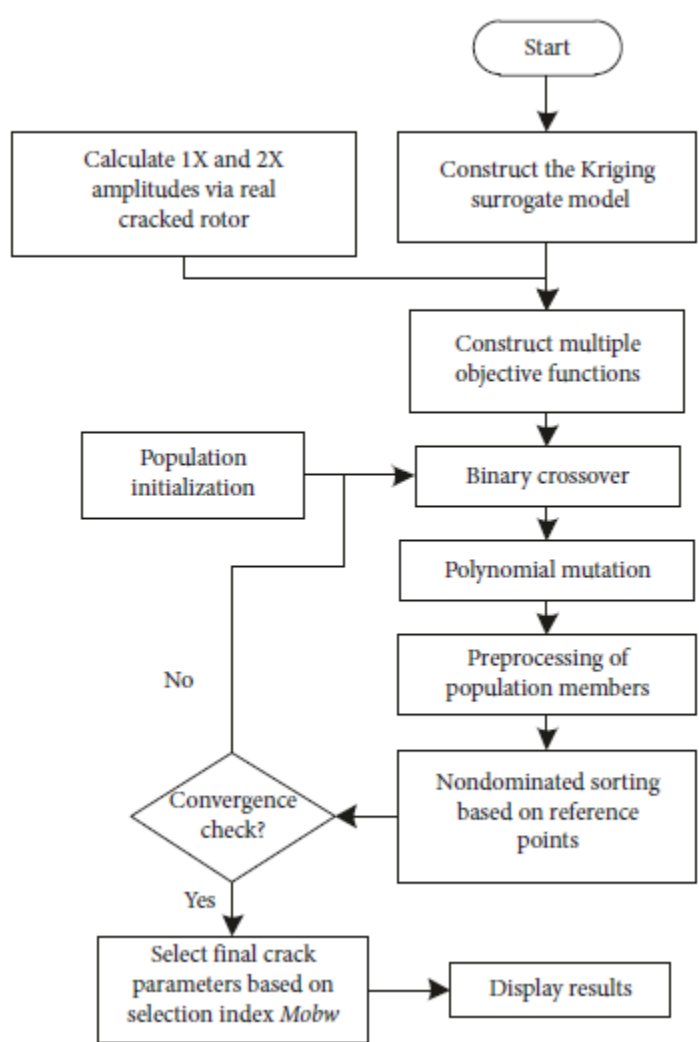


Figure 2.5: Flowchart of the method for crack parameters [39]

of the crack fairly accurately. Singh and Tiwari have developed a methodology for crack detection that could give the crack location and depth [42]. Their methodology consists of two parts. The first being modeling the rotor by using finite element method based on Timoshenko theory. Using the result of this model they could detect the cracks number and approximate locations. As the shaft vibrates because of a harmonic force the elastic line will

show a slope discontinuity due to the presence of a crack in the shaft. Although not easy to detect, especially with high noise. Nevertheless, as soon as detected the location and number of cracks can be identified. The second part is using an optimization technique based on genetic algorithm which enabled them to identify the exact location of the cracks in addition to their depths. This methodology has been tested numerically on a shaft with two open cracks and the cracks have been detected successfully.

Gayen et. al. have studied the dynamics of functionally graded shaft with a transverse crack using finite element method [43]. The functionally graded shaft is composed of stainless steel and zirconia. By using their model, they were able to find the critical speeds in addition to the backward and forward whirling frequencies. The main focus was to study the effect of the temperature gradient, crack size, slenderness ratio and power-law gradient index on the dynamic response of the shaft. Their results indicate that the power-law gradient index, which can take any value between zero and infinity, highly influences the critical speed and whirling frequencies in both cases of cracked or non-cracked functionally graded shaft and should be considered during designing a functionally graded shaft.

Silani et. al. modeled a rotor system with a crack in order to study the ability to detect the crack using vibrational response [44]. Incorporating a breathing function to simulate the opening and closing of the crack under rotation. A finite element analysis was also used. The open crack model induces nonlinearity in the system which restricts the usage of normal analysis. Hence, transient response was used in the analysis. They have used a short

time Fourier transform (STFT) as a signal processing technique to manipulate the signal. Start-up response has a better detectability of the crack when compared with steady state response. STFT does not respond very effectively to small cracks, nonetheless, it could be used for crack detection. Patel and Darpe have studied the dynamics response of shaft with a transverse crack, unbalance, and rotor-stator rub [45]. They have used finite element method to model the shaft under torsional vibration, taking into consideration the coupling of stiffness caused by the crack. Hilbert Huang transform was used to study the effect of the stiffness changes due to breathing of the crack. They have found that high torsional vibrational response at 1x and 2x due to the existence of crack. While 1x and torsional natural frequency have a high response for the rub failure in the rotor system. Using these frequencies in addition to the modulation of the natural frequency gives an indication to the existence of both crack and rub failures.

Han et. al. have studied the vibrational effect of a breathing slant crack on a geared rotor system [46]. They have derived a finite element model and took into consideration the stress intensifier factor using fracture mechanics. Steady-state response analysis, whirling analysis, and parametric instability analysis were used to study the response of the geared rotor system. The whirling frequency of a slant crack reduced more in comparison to a transverse crack. The slant crack showed more skewedness to the right for the instability regions when compared to the transverse crack. Due to the breathing crack, there are many sub-critical resonances. The peaks of the slant crack shifted slightly to the right compared with the resonant peaks of the transverse crack. The sub-critical resonances are more significant if the crack appears at the center point of the shaft. Satpute et. al. studied the

first three natural frequencies of transverse mode crack using finite element method [47]. They observed a decrease in the natural frequency for the first three transverse modes of vibrations, as the crack propagates and deepens the crack introduces local flexibilities in the structural member. Utilizing this method eliminates the need for shaft removal during inspection. By simply comparing the original natural frequency with the current one, both the location and size of the crack can be accurately determined, significantly streamlining the process. The analytical analysis is in agreement with the behavior of the existing cracked shaft. Hence, they conclude that the transverse vibration modes could be used to indicate the location and the size of a crack in the shaft as the natural frequency depends on size and location of the crack along the shaft.

Chondro and Labeas have investigated the torsional vibrational response of a cracked shaft analytically and numerically by using finite element method [48]. Hu–Washizu–Barr variational formulation was used to derive the differential equation and the equations of motion were then solved. Then Rayleigh’s quotient was used to approximate the natural frequency. A finite element method model was derived to verify the analytical result and they found good agreement between both methods. Bachschmid et. al. have studied the effect of a helical crack on the vibrational response of a static shaft using a 3D finite element model [2]. The torsional loading emphasizes the small differences between the dynamic response of a helical crack and a transverse one. Another simpler model was derived to capture and further study the dynamic behavior of the crack. They concluded that small cracks show only small amplitudes in the frequency spectrum of the vibrational response. Bending vibrations generated by bending loads and the breathing of the crack are

the most significant indicator of a helical crack. Which can be easily measured at the run down passing through resonance frequencies.

Darpe modeled a rotor based on finite element method with a slant crack to study the vibrational coupling [49]. Based on flexibility matrix from fracture mechanics a model for the slant crack was developed. A comparison between the effect of the slant crack and a transverse crack was carried out. The influence of the orientation angle of the slant crack was also investigated. The slant crack has more cross-coupling when compared to the transverse crack case. This results in more cross-coupling in the bending-longitudinal-torsional vibrations of the slant crack in comparison to the transverse one. The unbalance lateral vibrational response due to the additional torsional excitation could be used to detect the existence of a crack in the rotor. A potential crack detection parameter would be the torsional excitation frequency and the sidebands around it as they show up in the lateral vibration spectrum. The coupling effect between different directions of vibrations i.e., between lateral-torsional, lateral-longitudinal and torsional-longitudinal was found by Lina and chu is consistent with the result Gounaris and Papadopoulos, and Darpe et al [27, 32, 33, 49]

Naik et. al. carried out theoretical and finite element analyses to investigate the coupled torsion-bending vibration of Euler-Bernoulli and Timoshenko with a randomly oriented open crack [50]. Their analysis was based on the compliance approach and considering the local flexibility in the shaft due to the crack. The natural frequencies of both cases were calculated theoretically and using finite element method at different crack sizes, locations

and orientations. Applying the full compliance matrix makes the natural frequencies independent of crack orientation. However, representing the crack with a single torsional spring results in dependencies of the transverse vibration on the crack orientation. Darpe has devised a methodology to detect transverse surface crack in a rotor system by modeling the rotor using finite element [51]. Taking into consideration the non-linearity of the breathing of the crack and the cross coupling between the bending and the torsional vibrations. Wavelet transforms were used in this study to capture the transient vibrational signal. He found a correlation between the vibrational signal and the transverse crack behavior. The feature of this response is special and do not exhibit the behavior of other types of cracks hence, it is unique and will not be confused with other failures.

Rao et. al. have studied crack detection in a rotating shaft using the wavelet transform and artificial intelligence techniques [52]. By training the artificial neural networks, it was able to locate the crack and its depth successfully. Daubechies mother wavelet was used which increases by the increase in the crack depth. One of the findings of this study is that the rotational modes of the rotor have higher sensitivity to cracks when compared to translation modes. Additionally, the first detail of the discrete wavelet transform can be correlated to the crack depth. However, this study concluded that the second rotational mode shape of the wavelet transform can be used to identify crack location in a rotor system. Sathujoda et. al. did a finite element analysis on a rotor bearing system with a slant crack [53]. The type of crack studied was a breathing crack and the study was based on steady-state and transient responses. The flexural vibration was generated using finite element analysis where the transient responses, steady state responses, and the natural frequencies were calculated accordingly. It was found that with the increase in depth of the crack, the natural

frequency would decrease. This was attributed to the fact that increase in crack depth would also decrease the stiffness of the rotor system which will result in the reduction in the natural frequencies. Based on this study, it was also found that the location of the crack along the shaft will result in different vibrational response. As the crack which is close to the nodes such as bearing will have less significance. Additionally, the amplitude of the harmonics would increase with the increase in the crack depth. Moreover, the subharmonics frequencies would be a good indicator to the severity of the crack.

There is a lack in literature to studies that have examined the behavior of cracks other than transverse ones. Additionally, the torsional vibration ability as an identification method is lacking in the literature. This work experimentally investigated the effect of slant crack, i.e., 45 degrees, size on the torsional and lateral vibrations of a rotating shaft.

2.4 Digital signal processing

Digital signal processing (DSP) is the field of study that is concerned with manipulating an array of measured data points to extract meaning or to make sense of the physical meaning of it. This is carried out by using certain mathematical transformations with digital techniques. The use of DSP allows us to turn a physical phenomenon into measured data points. The purpose of this is to transform the data into more meaningful form which will enable us to understand the behavior and even predict the behavior if the same measurement was carried out under different conditions. This physical phenomenon can be a range of things, from movement of vehicles to electronic devices to name a few. Transforming the data into meaning form is done using physical laws or some models that

can be explained mathematically. One of the famous mathematical transforms used in DSP is known as Fast Fourier transform. This transform enables us to go from a convolution expression of the system into simple multiplication which makes the calculation a lot easier. This transform converts from time domain to frequency domain.

The array of the measured data points, of the physical phenomenon, is called a signal in the language of DSP. A signal could be temperature measurements inside a room under different times during the day or the changes in the lateral response of the velocity at a certain point along the shaft measured by vibration transducer to name a few. After measuring the physical phenomenon by using a sensor the measured data points are displayed. In DSP we normally deal with digital signals which are discrete in nature i.e., their measurements are equally spaced data points. Digital signals are compared to their counterpart that is analog signals which are continuous in nature.

Signals are generally categorized as deterministic or random. This classification determines the methodology by which these signals are analyzed. Usually, the signals are of mixed type. Which leads to difficulty to analyze the signal based on this classification so, the knowledge about the physical nature of acquired data points is necessary to determine the analysis method. [54]

Generally, the aim of signal processing is to extract physical meaning from an observed phenomenon. In particular, the ones that cannot be concluded from direct observation. The process usually goes from the data acquisition passing through processing to finally, interpretation. Deterministic signals are classified into periodic and non-periodic signals. Periodic signals are the signal that repeat a certain waveform over a specific period of time.

Non-periodic signals are further classified into almost periodic and transient signals. Almost periodic signals are, as the name implies, very close to the periodic signal but if observed carefully they are not exactly periodic. However, transient signals are the one which will go to zero as the time goes to infinity. A close example would be the impact test done in vibration to determine the natural frequencies.

One of the main methods to analyze periodic signals is the Fourier series. The base of the Fourier analysis of such signals is to represent them in terms of sine and cosine functions that are added up together. DFT is the discrete Fourier transform which is a mathematical process to transform the signal from time domain to frequency domain. The DFT requires a lot of computation. So, the FFT was invented which is a faster method to achieve the same conversion done by DFT but with less computation. [54]

Short time Fourier transform is basically a Fourier transform of different windows of a signal stacked beside each other. In a given signal, if we take 10 windows and do an FFT for each window as a separate signal and then stack them together and see a different response say in the third FFT we will know that this change happened in the third window of the original signal. Using the normal FFT, it would not be possible to get the same result as the FFT gives the average frequency along the signal STFT gives more information about the location of certain frequencies along the signal. We are trading off frequency resolution for time resolution in STFT. This trade off depends on the size of the window we create. The larger the window gives better resolution in frequency domain but gives worse resolution in time domain.

Wavelets are mathematical functions that start at zero and then goes to an oscillation and returns back to zero. Compared to the FFT where signal modeled as stacks of sines and cosine's function, here the signal is modeled as stacks of wavelets. In comparison to STFT, it gives high resolution in the time domain at high frequencies and high resolution in the frequency domain at low frequency. Figure 2.6 shows a visual representation with the resolution explained. [54]

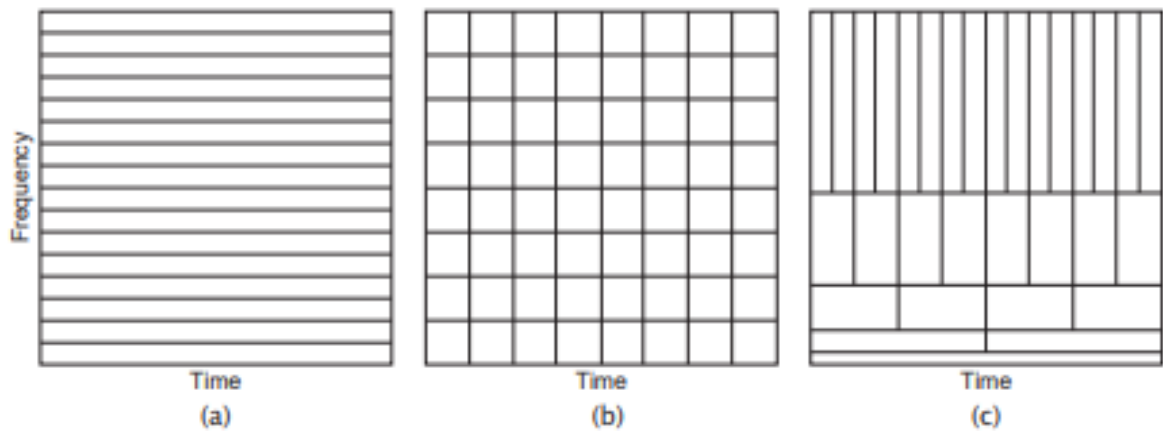


Figure 2.6: (a) FFT, (b) STFT, (c) WT [54]

2.5 Theoretical background

This section will shed some light on the theoretical background of the dynamics of the rotor systems with the existence of cracks. First of all, Newtonian equations of motion for all three axial directions and torsional direction are derived. With the assumption that there is no axial excitation. Bovsunovsky have derived the following four equations of for a Jeffcott rotor with a slant crack as seen in figure 2.7. [30].

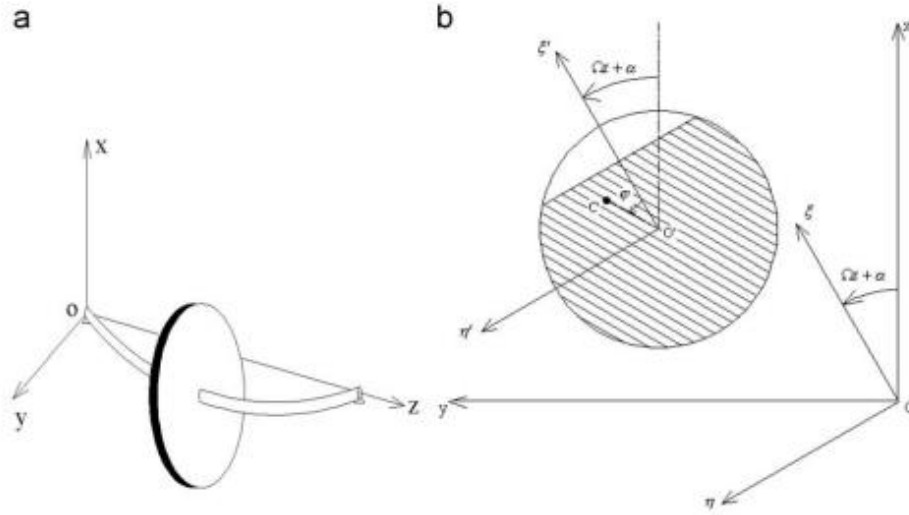


Figure 2.7 a) Jeffcott rotor model. b) projection of the crack [30]

$$m\ddot{x} + c\dot{x} + k_x x + k_{xy}y + k_{xT}\alpha + k_{xu}u$$

$$= -mg + me(\Omega + \dot{\alpha})^2 \cos(\Omega t + \alpha + \varphi) + me\ddot{\alpha} \sin(\Omega t + \alpha + \varphi)$$

$$m\ddot{y} + c\dot{y} + k_{xy}x + k_y y + k_{yT}\alpha + k_{yu}u$$

$$= me(\Omega + \dot{\alpha})^2 \sin(\Omega t + \alpha + \varphi) - me\ddot{\alpha} \cos(\Omega t + \alpha + \varphi)$$

$$J_p \ddot{\alpha} + c_T(\Omega + \dot{\alpha}) + k_{xT}x + k_{yT}y + k_T\alpha + k_{Tu}u$$

$$= M(t) + mge \sin(\Omega t + \alpha + \varphi) + m\ddot{x}e \sin(\Omega t + \alpha + \varphi) - m\ddot{y}e \cos(\Omega t + \alpha + \varphi)$$

$$m\ddot{u} + c_u \dot{u} + k_{xu}x + k_{yu}y + k_{Tu}\alpha + k_u u = 0$$

The mass center is denoted by \$C\$, while \$\varphi\$ represents the phase angle of the disk's mass center. \$\Omega\$ signifies the rotating speed, and \$\alpha\$ refers to the angle produced by the torsional vibration. Time is represented by \$t\$. The disk's mass is given by \$m\$, and \$J_p\$ stands for the

moment of inertia. The damping coefficients for transversal, torsional, and longitudinal vibrations are denoted by c , c_T , and c_u , respectively. The displacements of the two transversal vibrations, the angle of torsional vibration, and the displacement of longitudinal vibration are represented by x , y , α , and u , respectively.

The effect the crack has on the dynamics of this system is on the stiffness of the rotor. So the stiffness of the rotor is written as follows.

$$[k]_g = \begin{bmatrix} k_x & k_{xy} & k_{xT} & k_{xu} \\ k_{xy} & k_y & k_{yT} & k_{yu} \\ k_{xT} & k_{yT} & k_T & k_{Tu} \\ k_{xu} & k_{yu} & k_{Tu} & k_u \end{bmatrix}$$

Each of the term is calculated separately. The existence of the crack will cause a local flexibility which is factored in the shaft flexibility. The local flexibility is calculated according to the following equations.

$$[c]_l = [G_1][\Delta c][G_2] + [c_s]$$

$$[c_s] = \text{diag} \left[\frac{l^3}{48EI} \quad \frac{l^3}{48EI} \quad \frac{l}{2EA} \quad \frac{l}{2GI_p} \right]$$

$$[G_1] = \text{diag} \left[\frac{1}{2} \quad \frac{1}{2} \quad 1 \quad 1 \right]$$

$$[G_2] = \text{diag} \left[\frac{1}{4} \quad \frac{1}{4} \quad 1 \quad 1 \right]$$

$$[\Delta c_{ij}] = \frac{\partial^2 W}{\partial q_i \partial q_j}$$

$$[\Delta c_{ij}] = \begin{bmatrix} c_{11} & 0 & 0 & c_{14} & c_{15} & 0 \\ 0 & c_{22} & 0 & 0 & 0 & c_{26} \\ 0 & 0 & c_{33} & 0 & 0 & c_{36} \\ c_{41} & 0 & 0 & c_{44} & c_{45} & 0 \\ c_{51} & 0 & 0 & c_{54} & c_{55} & 0 \\ 0 & c_{62} & c_{63} & 0 & 0 & c_{66} \end{bmatrix}$$

The stiffness of the shaft is related to the local flexibility according to the following equations.

$$[k]_l = [C]_l^{-1}$$

$$[k]_g = [H]^{-1}[k]_l[H]$$

Where H is the transforming matrix that can be written as follows.

$$\begin{bmatrix} \cos(\phi) & \sin(\phi) & 0 & 0 \\ -\sin(\phi) & \cos(\phi) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

$$\phi = \Omega t + \alpha$$

These equations of motion could be solved with some numerical methods as reported in [30]. Nevertheless, it can be seen that the local flexibility matrix and the stiffness matrix have some non-diagonal terms. These non-diagonal terms are due to the coupling effect that is taking place because of the existence of the crack in the shaft. This coupling has been reported in several papers [10, 13, 14].

CHAPTER 3

Experimental Setup and Design of Experiment

This chapter discusses the experimental setup, which includes the test rig used and sensors. Additionally, it provides the design of the experiment and functional requirements such as speed of the rotor and the number of runs.

3.1 Test rig

This study was done by using a rotor kit for fault simulation (Spectra Quest-USA); shown in Figure 3.1. The rotor kit consists of a long shaft which has the diameter of 0.625 in (15.875mm) and a length of 30 inches (76.2 mm). Coupled with a shorter shaft having the same diameter with a length of 15 inches (38.1 mm). A disk is located in the middle of the long shaft, between two ball bearings with an approximate weight of 0.5 kg, as shown in Figure 3.2. The rotor is connected by a means of flexible coupling to a 3-phase induction motor of 220 Volt with 2.24 kW and 60 Hz from one side. The other side has a disc with 12 blades and a 2-kW generator connected also with a flexible coupling. The generator is adding only static load to the system while the disc with blades is used to generate the torsional rub in the experiment by a rubbing device that can rub against the tip of the blades.

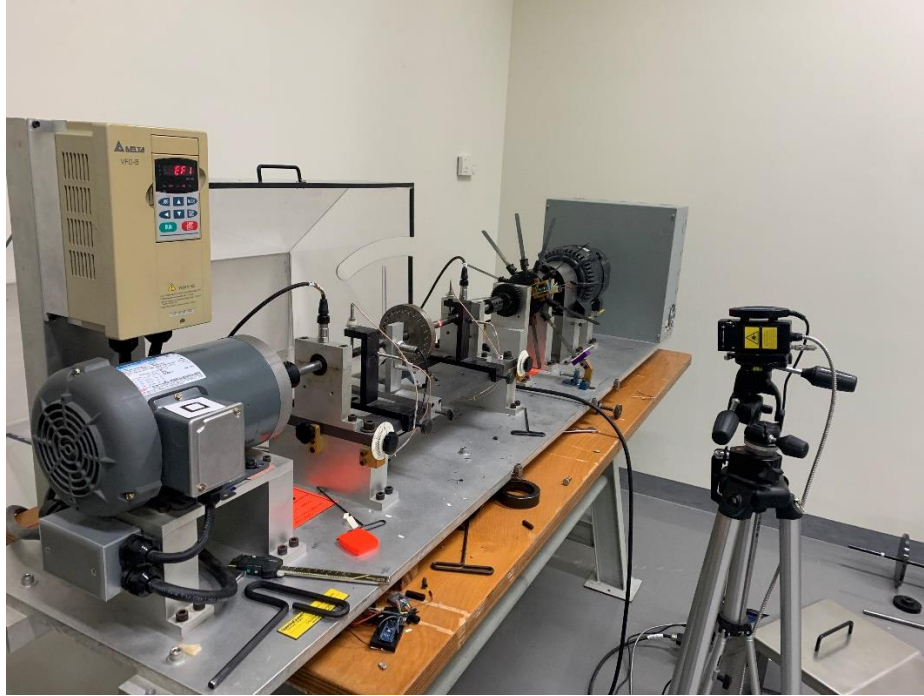


Figure 3.1: Experimental setup.

3.2 Sensors and data acquisition system

Part of the experiment was to set up the test rig and attach all vibration transducers. For shaft lateral vibration, four proximity probes were installed near around 1 in from the bearings to measure the radial displacement in two directions: vertical (V) and horizontal (H). The 5 mm proximity probes (Metrix instrument co. model of 10001-00-30-05-02) have sensitivity of 193.33 mV/mils (7611.42 mV/mm). Laser doppler vibrometer (Polytec RLV-5000A – Germany) was used to measure the shaft torsional vibration. For data gathering, a 16-channel commercial data acquisition system specialized for rotating machinery (Viblow7000- NADA- Korea) with user software for data recording and analysis (VibLow TSI-NADA- version 1.2) was used for data acquisition and analysis in time and frequency domains.

A sampling frequency of 2048 Hz was utilized with a maximum frequency of 400 Hz for the proximity probes and 800 Hz for the laser doppler vibrometer. The number of line of

resolution used for the proximity probes is 3200 lines and the number of lines of resolution used for the laser doppler vibrometer is 6400 lines.

3.3 Design of experiment

The crack is located in the middle of the rotor just outside the disc towards the inboard bearing as seen in Figure 3.2. Additionally, the shape of the crack is a 45-degree slant crack that was made by milling machine with a cutting edge of 2 mm width and the depth ranges on the four shafts from as shaft for reference and comparison, to 10% crack depth on the first shaft, 25% depth on the second shaft, and 40% depth for the deepest studied crack on this experiment Figure 3.3 shows the three cracks..

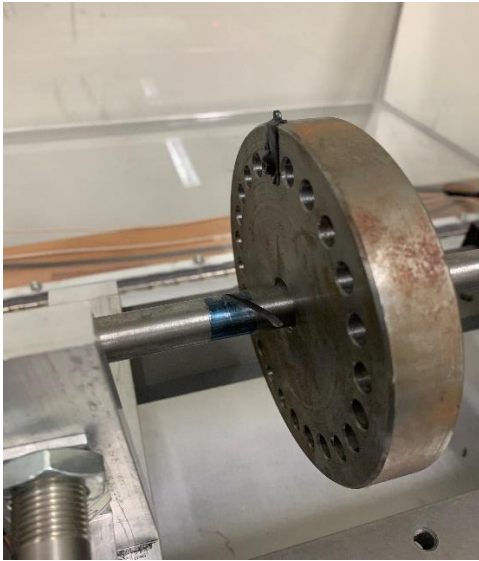


Figure 3.02: Disk and crack location



Figure 3.3: Crack sizes, 10%, 25%, 40%

This variation on the crack depth is for studying the sensitivity of the response to the crack depth. The rotational speeds for the experiments were selected based on the common speeds found in the industry of rotating machinery. These speeds are 1200RPM for the lowest considered speed, increasing to 1800RPM for the middle speed and topping at 2400RPM for the highest studied speed. The first critical speed is 92.361Hz which is equivalent to 5541,66 RPM that was estimated using finite element analysis; details in section 4.5 of this study. Accordingly, the experiment is run below the shaft first critical speed. The aim of having various rotational speeds is to study the cracked system at different conditions to see the effect on the vibrational response. The experiments were logically coded according to Figure 3.4 and Table 3.1. For example, OH-18-25 stands for outboard horizontal (OH) proximity probe with 1800 RPM (18) and 25% (25) crack depth.

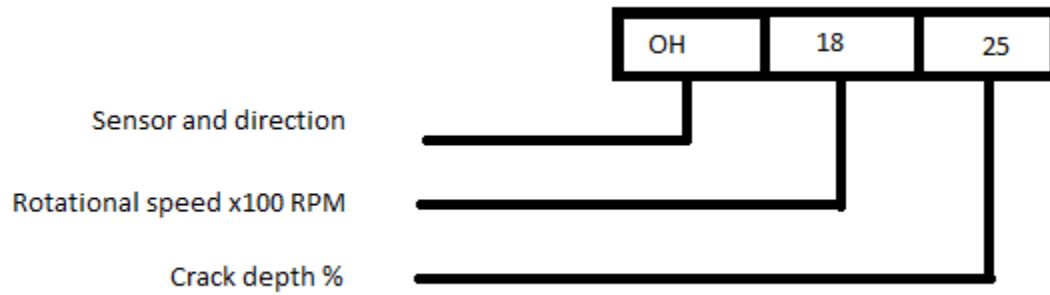


Figure 3.04: Experimental code

Table 3.1: Experimental code meaning.

Code	Meaning
IV	Inboard vertical Proximity probe
IH	Inboard horizontal Proximity probe
OV	Outboard Vertical Proximity probe
OH	Outboard horizontal Proximity probe
TV	Torsional laser-doppler vibrometer

Code	Meaning
12	1200RPM
18	1800RPM
24	2400RPM

Code	Meaning
00	0%
10	10%
25	25%
40	40%

3.4 Signal processing techniques

The first signal processing technique used in this study is the time wave analysis. Specifically, waveform analysis. By observing and examining the signal for different shapes and forms. For example, beating pulses would appear in the waveform of the signal. Then the analysis would start to find the physical phenomenon behind this shape of the waveform.

Frequency spectrum analysis was also used in this study using FFT; which is basically a faster way to perform DFT [55]. In FFT, the waveform is transformed from time domain to frequency domain. One of the techniques used to improve the quality of the FFT is windowing which is a technique applied to the time-domain signal before computing its FFT. The purpose of windowing is to mitigate spectral leakage, which occurs when a non-integer number of cycles of a periodic signal is present within the analysis interval. Spectral leakage results in the smearing of energy across multiple frequency bins, making it difficult to accurately identify the true frequency content of the signal. A window function is a mathematical function that smoothly tapers the time-domain signal to zero at the beginning and end of the analysis interval. Another technique commonly used is averaging which is a technique to improve the signal-to-noise ratio of the frequency-domain data. Noise can originate from various sources, such as measurement errors, electrical interference, or background vibrations, and can obscure the true frequency content of the signal. Root mean square (RMS) averaging of the FFT was applied in this study. The signal is divided into multiple overlapping segments and applied windowing to it. Then computing the FFT and power spectra for each segment. After that, performing RMS averaging to the signal by averaging the power spectra.

STFT is a powerful time-frequency analysis tool used to analyze non-stationary signals [56]. It provides a way to study how the spectral content of a signal evolves over time, which can be essential for identifying and analyzing transient events or varying operation conditions in rotating machinery. To construct a STFT we start by dividing the signal into equal segments with a certain overlap. The amount of overlap between the adjacent windows affects the time resolution of the STFT and the smoothness of the resulting spectrogram. A higher overlap results in a smoother spectrogram and better time resolution, but it also increases the computation complexity of the analysis. Typical overlaps for STFT range from 25% to 75% of the window length. In this experiment an overlap of 75% was chosen. However, the divided segments of the signal need to have a certain window. The window length is often chosen as a power of 2 to take advantage of the computation efficiency of the FFT. In this study a window size of 256 data points was chosen. After that, these segments will go through the FFT to come up with their frequency spectrum. Then the computed FFTs will be arranged into a two-dimensional matrix.

The discrete wavelet transform (DWT) is a widely used time frequency analysis technique that decomposes a discrete time signal into a set of wavelet coefficients, which represent the signals frequency content at different scales and time points [57]. The DWT is particularly useful for analyzing non-stationary signals where the frequency content changes over time. The DWT works by iteratively applying a pair of complementary filters, the low-pass filter (approximation filter) and the high-pass filter (detail filter), to the input signal or its approximation coefficients. The filtering process results in a hierarchical, multi-resolution decomposition of the signal, where the approximation coefficients represent the lower frequency content, and the detail coefficients represent the higher

frequency content. The result of the DWT is a set of wavelet coefficients, which can be organized as a hierarchical tree structure with multiple levels. In this study 7 decomposition levels were used. Each level represents a different time frequency resolution, with the top level providing the lowest frequencies and the bottom level providing the highest frequencies. And Daubechies mother wavelet was used as it is commonly used by other researchers such as Prabhakar et.al. [58].

The Continuous Wavelet Transform (CWT) is a time-frequency analysis technique that decomposes a continuous-time signal into a set of wavelet coefficients, which represent the signal's frequency content at different scales and time points. The CWT is particularly useful for analyzing non-stationary signals with transient or time-varying characteristics. The CWT works by correlating the input signal with a family of wavelet functions, which are derived from a single mother wavelet by scaling and translating it in time. The wavelet coefficients in the CWT are computed by integrating the product of the input signal and the scaled and translated wavelet functions over the entire time domain. Daubechies mother wavelet was used as it is commonly used by other researchers such as Prabhakar et.al. [58]. Scale of 2 to the power of 8 was used in this experiment. The result of the CWT is a continuous, two-dimensional representation of the input signal's time-frequency content, known as a scalogram or a time-scale representation. [57]

CHAPTER 4

Result and Discussion

This chapter presents the experimental results and discussion, as this study aims to understand and find an indicator to the existence of the crack in a rotating Shaft. The analysis is done through traditional signal processing techniques and advanced ones. The advanced signal processing started with the STFT and scalogram then went through the wavelet transform.

4.1 Traditional vibration Analysis techniques.

4.1.1 Time wave analysis

The classical vibration analysis for fault detection in rotating equipment starts with looking at the timewave of the acquired signal. The trained vibration analyst would normally start looking at the shape of the waveform in order to examine it for any apparent faults. Some waveform shapes such as beats, harmonics, truncated, pulses and modulated signals have well known causes that have been studied extensively and standardized in the vibration analysis field. Moreover, after checking for these shapes the trained analyst would normally relate the shape of the waveform to physics and faults of the machines behind this shape [59]. Figure 4.1 through 4.4 show the time waveform of 4 different samples of different runs of the experiment. Those signals are 1 second long. Figure 4.1 shows the shaft relative displacement near inboard bearing measured at steady state rotational speed

of 2400 RPM with no crack which is compared to figure 4.2 that is at the same location with the same speed and 40% crack depth. Figure 4.3 shows the outboard bearing's vibration response at 2400 RPM and with no crack which is compared to figure 4.4 that is at the same location with the same speed and 40% crack depth. Examining these waveforms shows that they are of a sinusoidal nature. The outboard horizontal bearing shows higher amplitude for the 40% crack compared to the reference shaft. Both have a slight amplitude modulation. This signal was extracted from the experimental run where there was a rubbing effect of the rotor. The inboard vertical bearing shows no clear difference for the case of 40% crack depth when compared to the reference case.

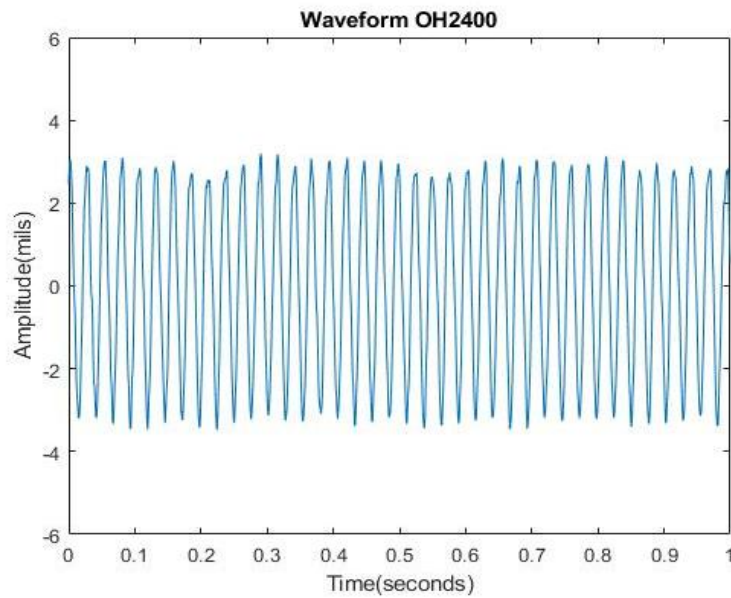


Figure 4.01: Waveform OH-24-00

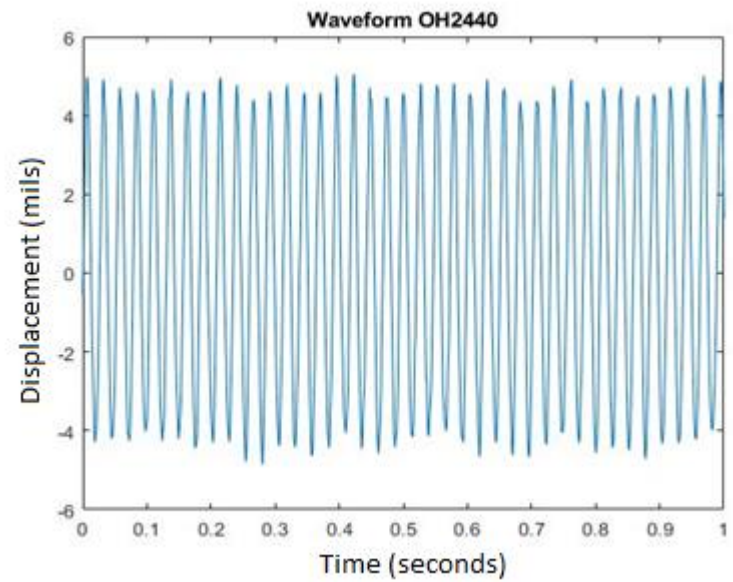


Figure 4.02: Waveform OH-24-40

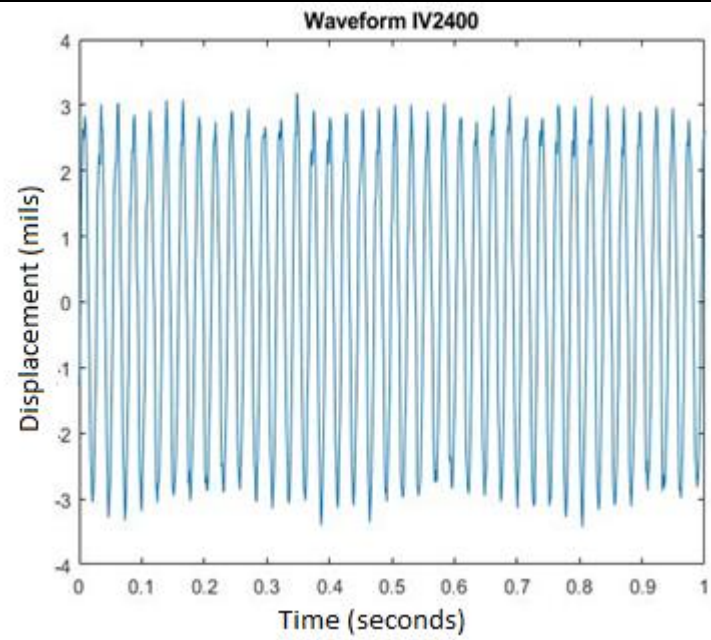


Figure 4.03: Waveform IV-24-00

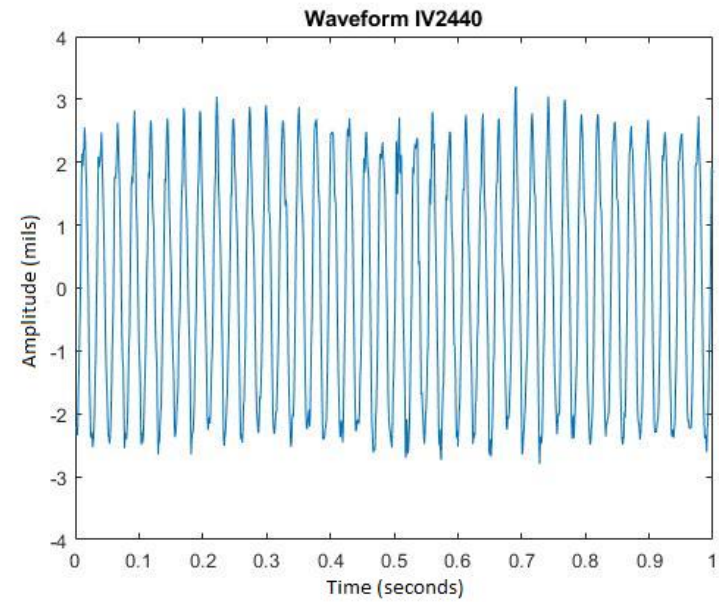


Figure 4.4: Waveform IV-24-40

To further the analysis of the waveform signals, root mean square (RMS), peak to peak and phase of (1x) were calculated and the results are indicated in figure 4.5 through figure 4.12. The first measurement is the root mean square of the signal. Increasing the speed is resulting in increasing the RMS value of the signal which is consistent for both directions of the inboard and the outboard bearings at different crack sizes. This means that the existence of the crack is increasing the amplitude of the vibration. [16, 17] have similar result where increasing the speed has resulted in increasing the overall amplitude of the vibration. This could be attributed to the unbalanced forces and stiffness decrease as seen in [9] which is clear for the case of 10% and 40% but not for the case of 25% crack depth. One phenomenon that could be considered here is the stiffening of the shaft due to rotation which is basically the increase in stiffness due to rotation. Several researchers are not including this phenomenon in their consideration when studying open grooves such as [30] because it is only significant at high speed. Moreover, the RMS value is also increasing with the increase in crack size at 40% which is consistent for both directions of the inboard and the outboard bearings at different speeds. This result is consistent with [7] where larger crack size is more prone to detection. Additionally, similar behavior for the peak-to-peak (P-P) values of the signals has been observed. However, the values for RMS and P-P have increased significantly at 40% depth crack for the onboard bearing when compared with 25% and 10% depth cracks. This same behavior was not observed at the inboard bearing this result was observed previously in the waveform analysis. This behavior could be attributed to the fact that the location of crack is outside of the disc slightly towards the inboard bearing. Where this phenomenon is observed due to the bowing of the Jeffcott Rotor with the existence of the crack which was not observed at lower than 40% depth

crack [6]. Nevertheless, the third measurement taken was the phase difference which did not show any significant changes between the cracked and the uncracked shafts as indicated in figure 4.5 to 4.12.

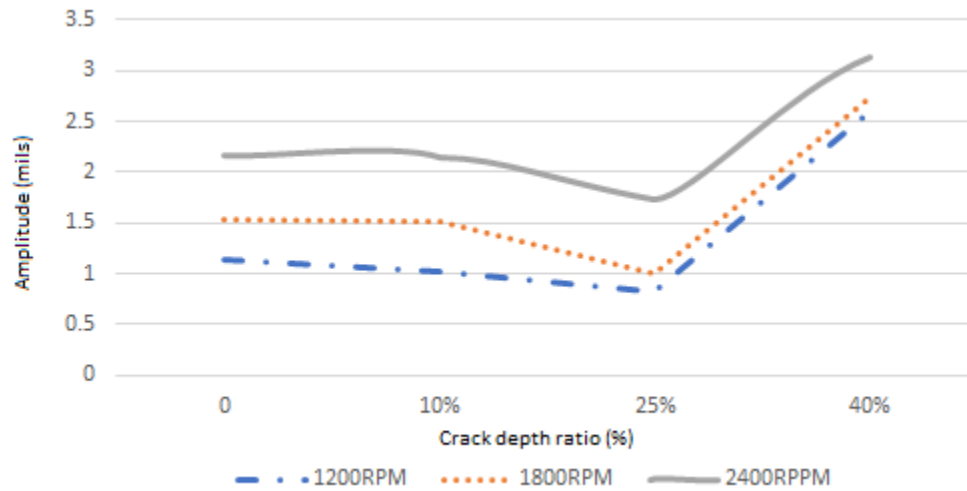


Figure 4.5: RMS values OH Bearing

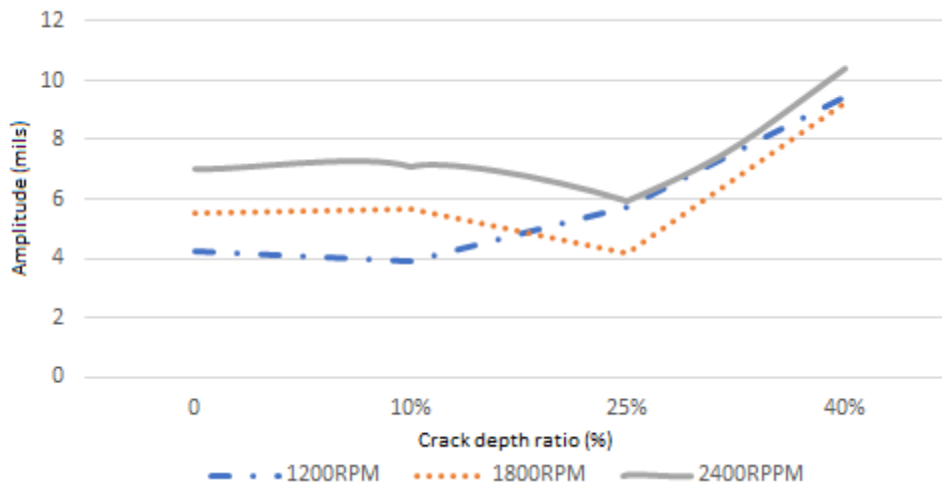


Figure 4.6: P2P Values OH Bearing

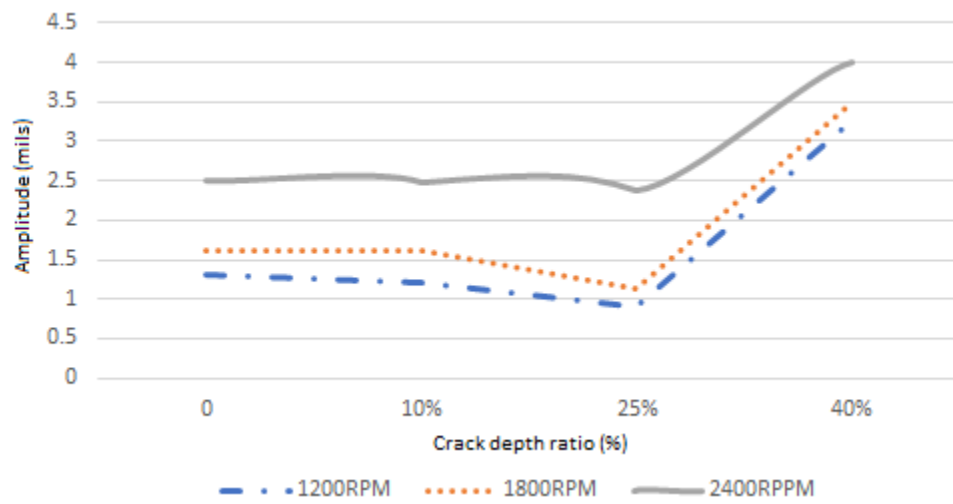


Figure 4.5: RMS values OV Bearing

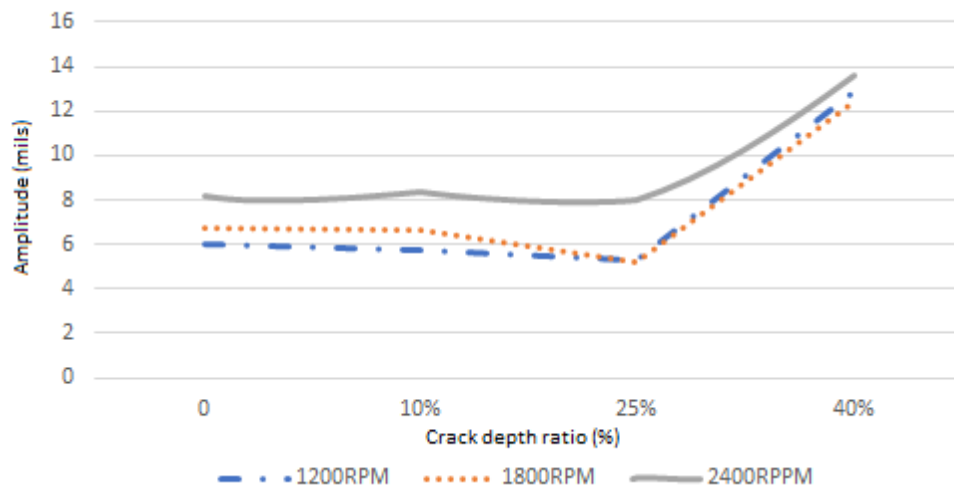


Figure 4.6: P2P values OV Bearing

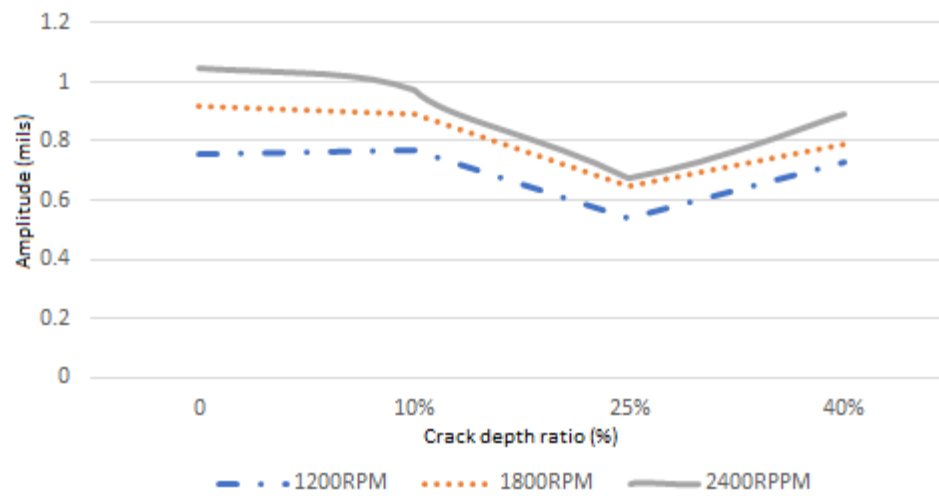


Figure 4.07: RMS values IH Bearing

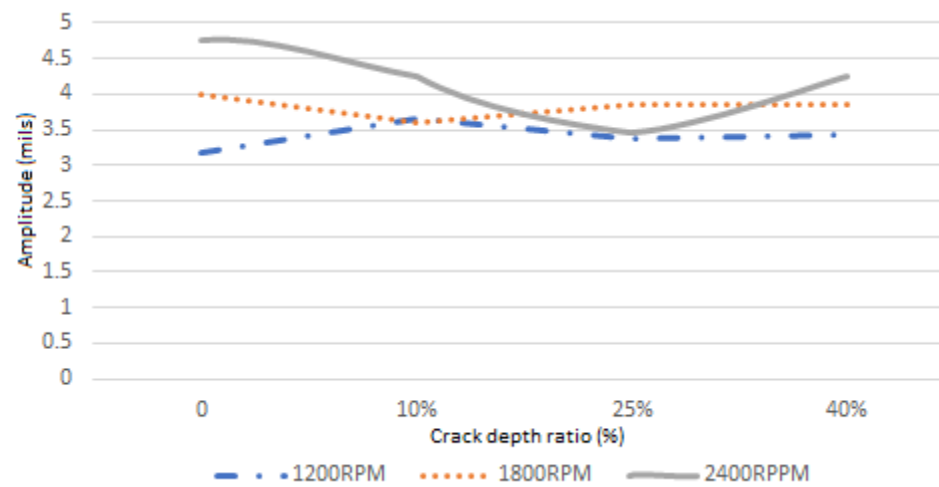


Figure 4.8: P2P values IH Bearing

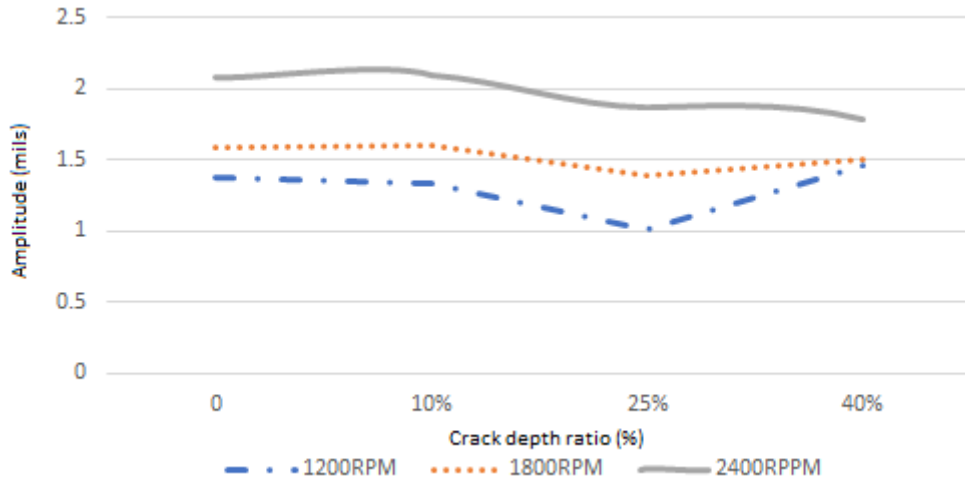


Figure 04.9: RMS values IV Bearing

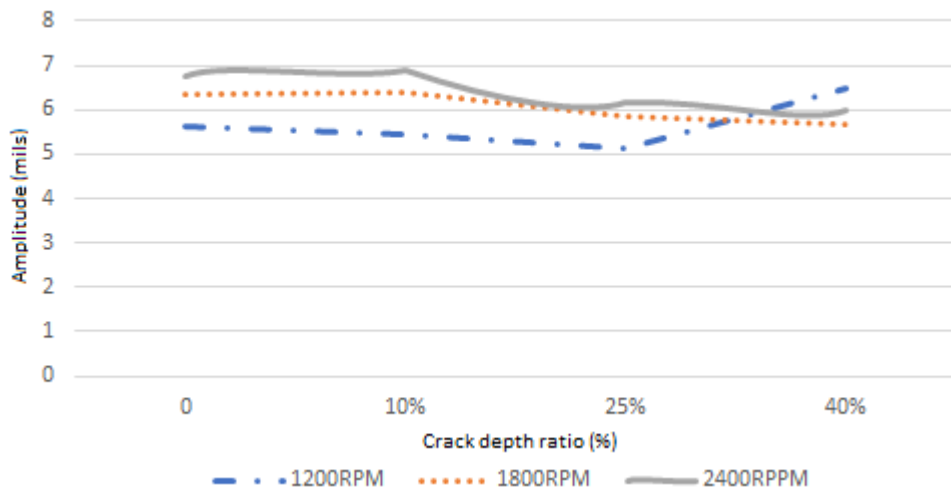


Figure 4.10: P2P values IV Bearing

4.1.2 frequency spectrum analysis

After analyzing the waveform comes the frequency spectrum analysis techniques using the classical Fast Fourier Transform (FFT). Frequency analysis techniques include different methods such as analyzing the sidebands, direct frequency, spectrum shape, harmonics, orders, and demodulated signal. Sidebands usually form around a certain frequency such as the natural frequency or a forcing frequency which is typically generated by frequency

modulation or amplitude modulation [59]. Another form of frequency analysis is the direct frequency identification which can be done to see the forcing frequency of the machine fault. These forces are not easy to measure directly and are difficult to be seen on the waveform. However, if the system is linear the forcing frequency will be the same as the response frequency. In these experiments there has not been seen any forcing frequency that is consistent along all different crack depths and rotor speeds as seen in the representative sample from Figure 4.13 to Figure 4.16 apart from the increase in 1x which is attributed to the system imbalance [59]. Beats are when two sources of frequency that are slightly different are affecting the same area, a phenomenon happens which is called Beats which is also considered one of the frequency spectrum analysis techniques. The beating phenomenon was not observed in our experiments. However, there are sidebands around the harmonics which can be seen in figure 4.13 through 4.16 these are attributed to the motor slip frequency of 4.1667%. Last but not least is a commonly used spectrum analysis technique which is the harmonics and orders technique. Harmonics and orders are the one, two, ... times the rotating speeds which are commonly referenced with the symbols 1x, 2x and so on. Many researchers [10, 11, 13, 14, 23, and 26] have shown that 2x and 3x are the most harmonics that could indicate the existence of a crack in a rotating system. Small side bands are seen around 1x, 2x, and 3x, and these can be attributed to the motor slip frequency. These small side bands are in frequency range of 1.25 Hz. Based on the motor has a line voltage frequency of 60 Hz and synchronous speed of 3600 since it has only 2 poles. Moreover, the rated speed of the motor is 3450 RPM. Using these pieces of information, the motor slip can be calculated, and it turns out to be 4.17% which is 2.5 Hz. Nevertheless, 1x is seen to be increasing at 40% crack depth at the outboard bearing at

2400RPM. The system has a small imbalance force that was put to make sure that the system is excited to examine the crack effect. A classical response of imbalance load is increase in 1x frequency. However, which is using the proximity probes to measure the lateral vibration did not give any consistent response on 2x or 3x in the vibration spectrum of the experiments as summarized by the sample of frequency spectrums in Figure 4.13, 4.14, 4.15, 4.16. Hence, there is no conclusive evidence indicating the existence of cracks on this experimental setup using the FFT analysis.

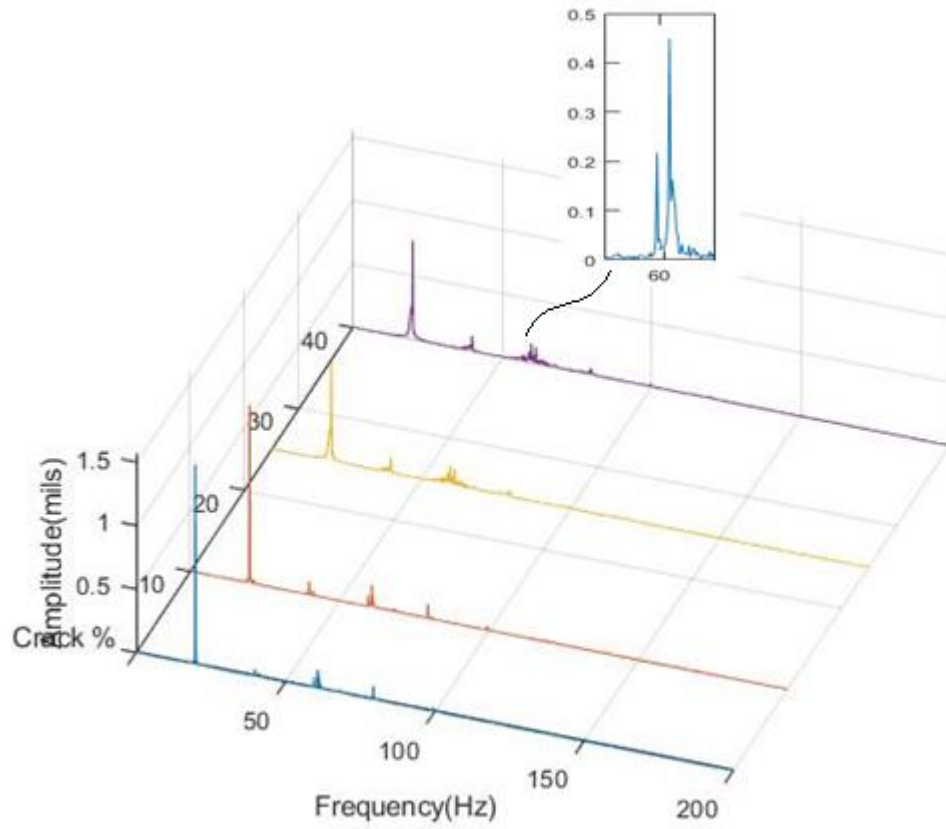


Figure 4.11: 3D FFT OH12

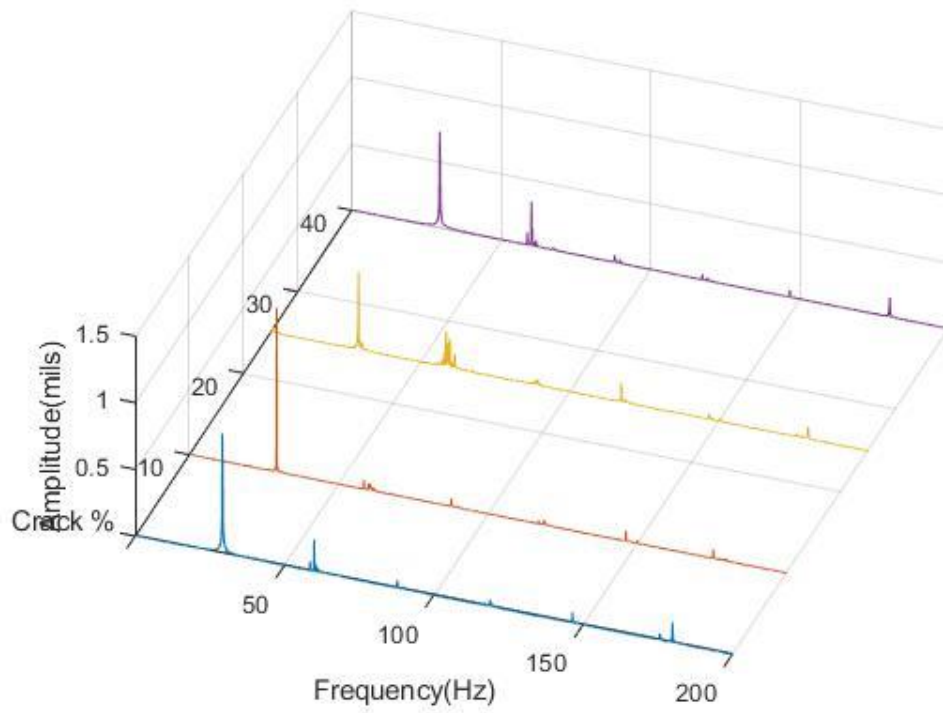


Figure 4.012: 3D FFT IH18

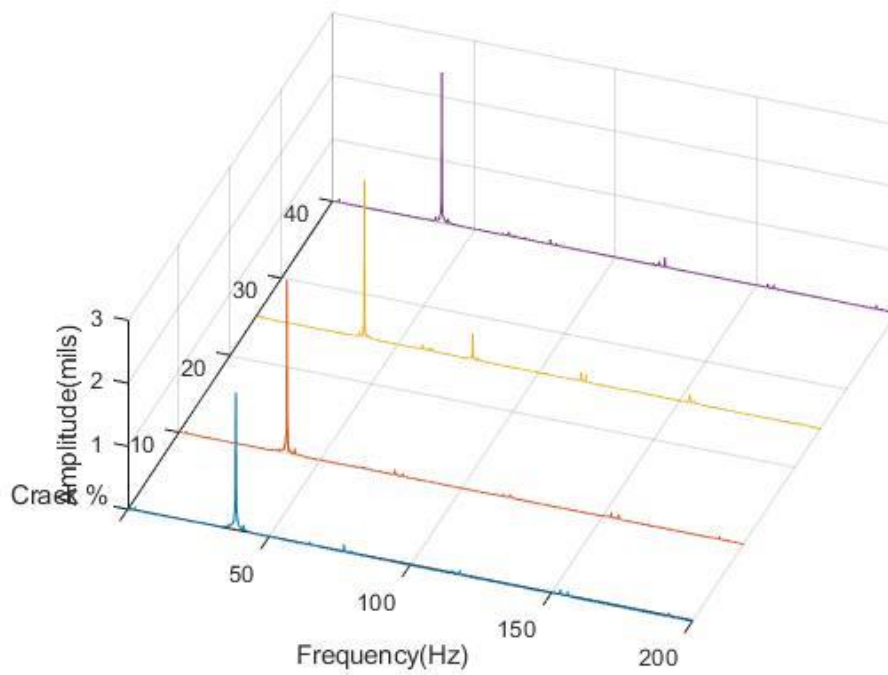


Figure 4.13: 3D FFT OV24

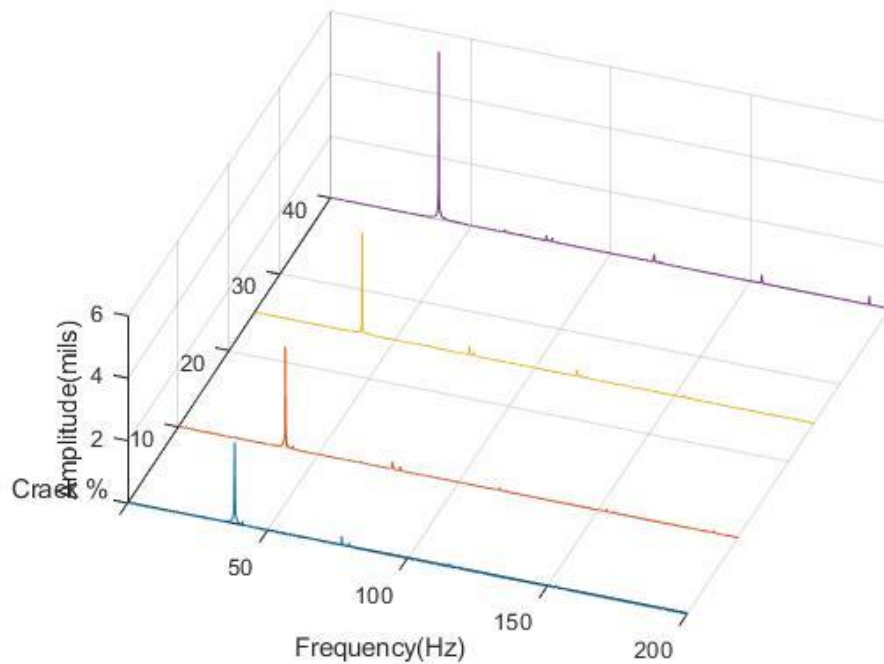


Figure 4.14: 3D FFT IV24

4.1.3 Orbit analysis

Orbits are examined for their shape which gives the relative motion of the shaft close to the bearings. These shapes can give some insight into the type of failures that exist in the rotor system. One of the common faults that can be detected by orbits are the mass unbalance and misalignment.[59]. Orbits shape normally is circular can have an oval shape during for example unbalance. It also can have an 8 shaped orbit when it has excessive misalignment. However, in [6] there were two loops inside the orbit which were attributed to the existence of the transverse breathing crack when passing through $1/3$ of the critical speed and 1 loop when passing through $1/2$ the critical speed. This can be seen due to the imbalance forces and the breathing crack which dominate the $2x$ and $3x$ speeds [6]. Figures 4.17 and 4.18 show samples of the orbits in this experiment. Their orbits are filtered to show only the first 7 harmonics. Additionally, it can be seen that the orbits have a circular

shape without any loops. However, they do not exhibit any signs of abnormality when comparing the cracked and uncracked cases.

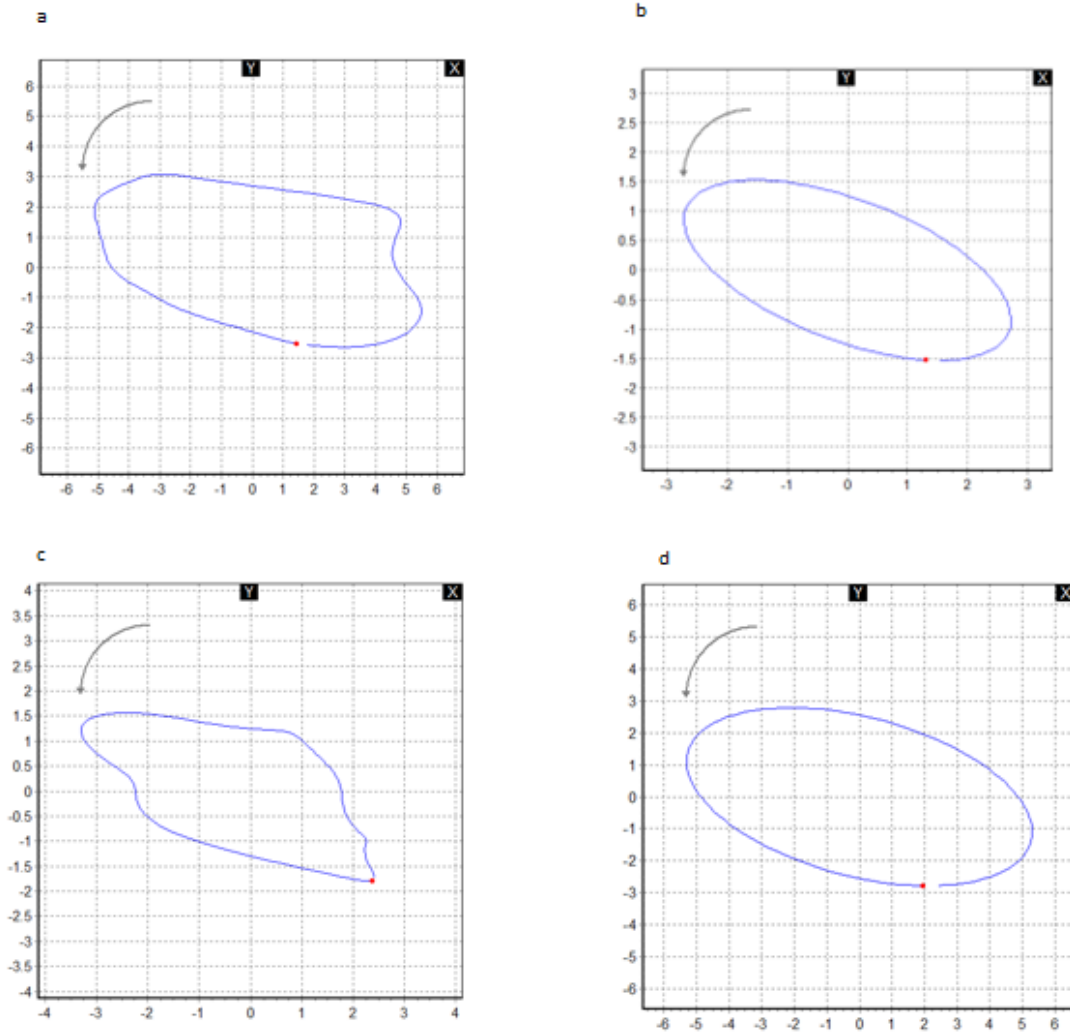


Figure 4.15: Orbit plots for OV & OH at 1800RPPM. a) 1x through 7x at 0% crack. b) 1x at 0% crack. c) 1x through 7x at 40% crack. d) 1x at 40% crack.

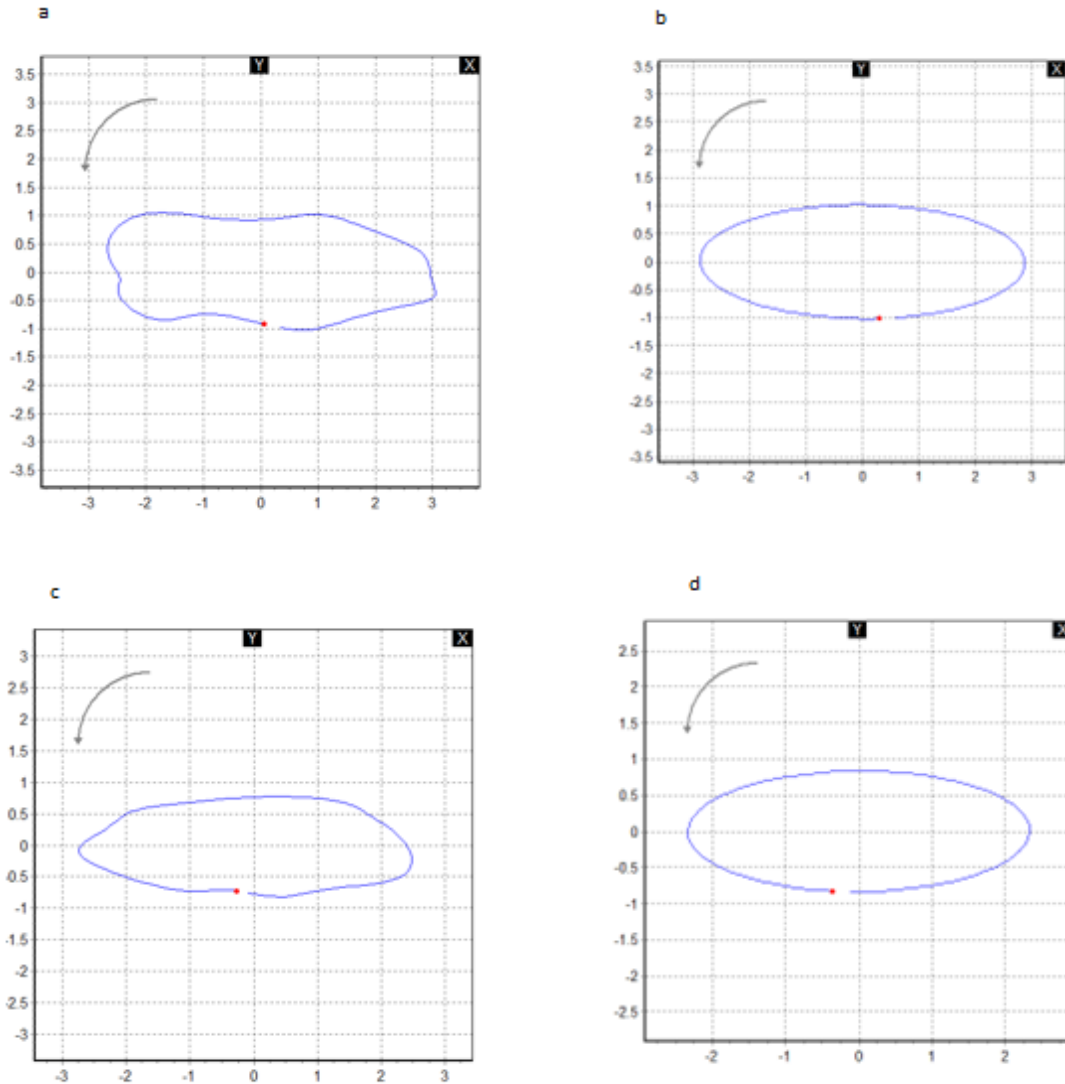


Figure 4.16: Orbit plots for IV & IH at 2400RPPM. a) 1x through 7x at 0% crack. b) 1x at 0% crack. c) 1x through 7x at 40% crack. d) 1x at 40% crack.

4.2 Torsional Vibration Analysis techniques.

One of the main goals of this study is to look at torsional vibration analysis as a tool to detect the existence of cracks in rotating shafts. Figure 4.19 shows the torsional vibration waveform of the torsional laser doppler vibrometer sensor for the uncracked shaft at 2400 RPM. An excitation to the rotor system was induced at the blades by rubbing against the

tip of the blades that caused a pulse like response as shown in figure 4.19. The purpose of this induced rubbing is to apply a torsional impulsive force (friction between the rubbing pad and blade) and excite the system torsional response of the vibration due to the crack. The aim is to look at the waveform and see if this pulse has caused any difference in the vibration between before and after the pulse. There is no significant difference for the vibrational response depicted by figure 4.20 when looking at the response before the pulse and comparing it with the one after the pulse. Looking at both figure 4.19 and figure 4.20 side by side, there is no significant difference between the vibration response between the 40% crack depth rotor and the no crack condition. Figures 4.21 and 4.22 show the RMS and peak to peak values for the waveform of the torsional vibration. Unlike the lateral vibration where there was a clear indication at the 40% crack depth, the torsional vibration for the RMS and peak to peak values do not show consistent relation that could be considered as an attribution of the existence of the crack at different speeds with all crack depths examined for both: the RMS and the peak-to-peak values. Figures 4.23 & 4.24 show the 3D representation of FFT for the laser doppler vibrometer sensor for all crack depths at the speed of 2400RPM with rubbing and without rubbing respectively. Looking at figure 4.23 the 3D FFT for the crack depths side by side there is noise at very low frequencies (ski-slope) that is consistent for all crack depths but is inconclusive for the existence of the crack as it did not change in terms of frequency or amplitude. Additionally, 1x, 2x, and 3x all have some amplitudes changes without showing a certain trend by increasing or decreasing the amplitude to indicate any difference in the vibration. Moreover, there seems to be no change or shift of frequency for 1x, 2x, and 3x. There appears to be small sidebands around the harmonics frequencies that are due to the slip frequency of the induction motor.

At low frequencies in the 40% FFT there is a high peak that corresponds to the motor slip frequency. Figure 4.24 shows the same information for the non-rubbing case. It has similar behavior for the case with rubbing effect shown in figure 4.23.

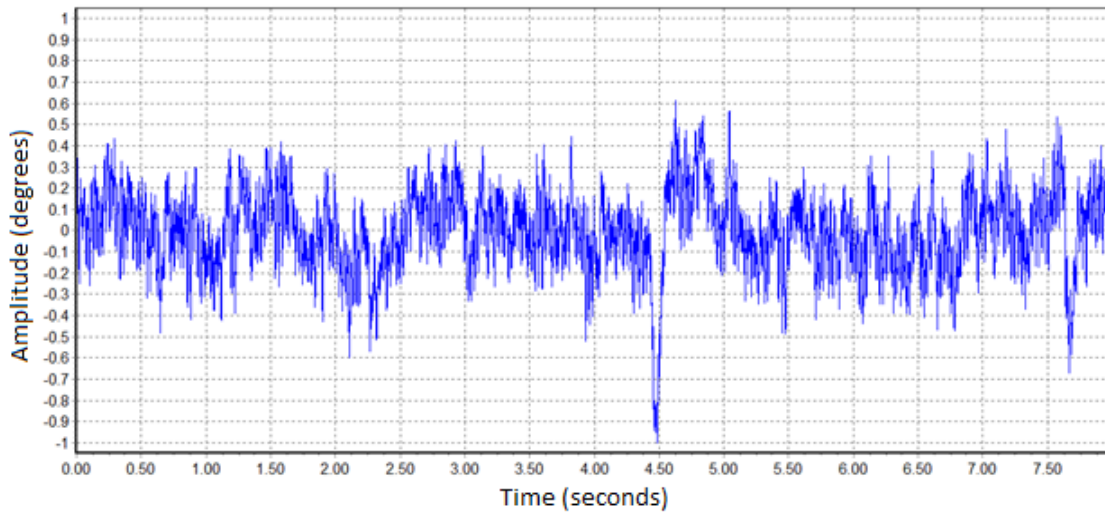


Figure 4.17: Waveform for TV-24-00

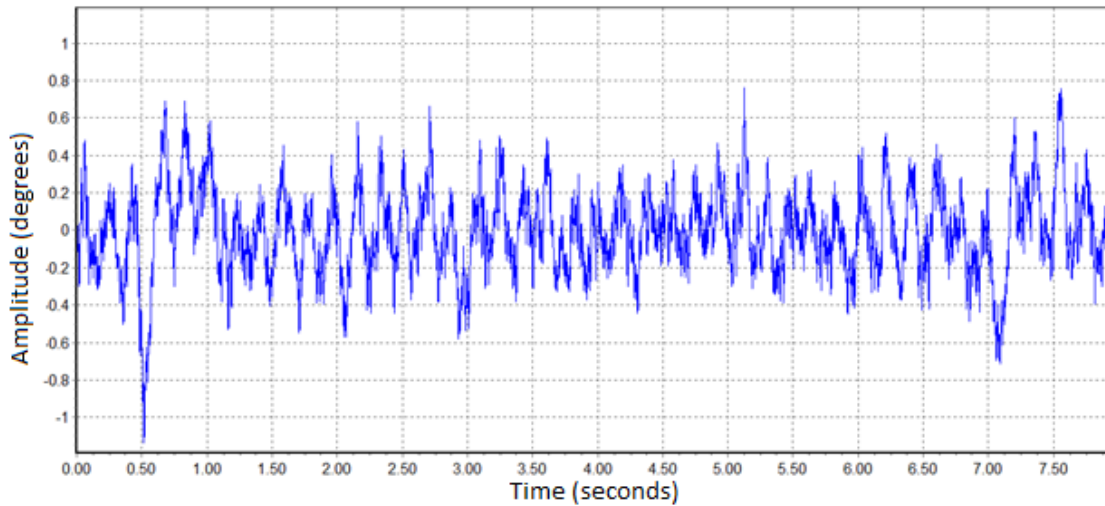


Figure 4.18: Waveform for TV-24-40

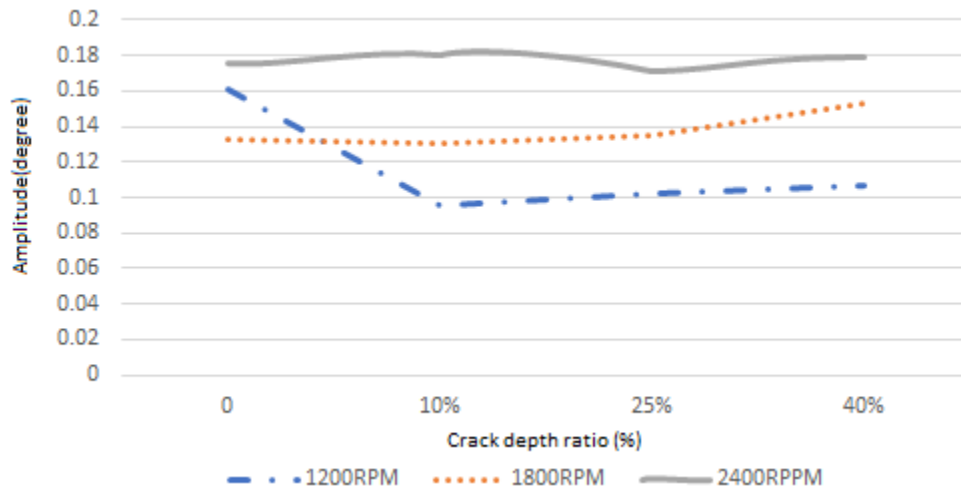


Figure 4.19: RMS values for TV

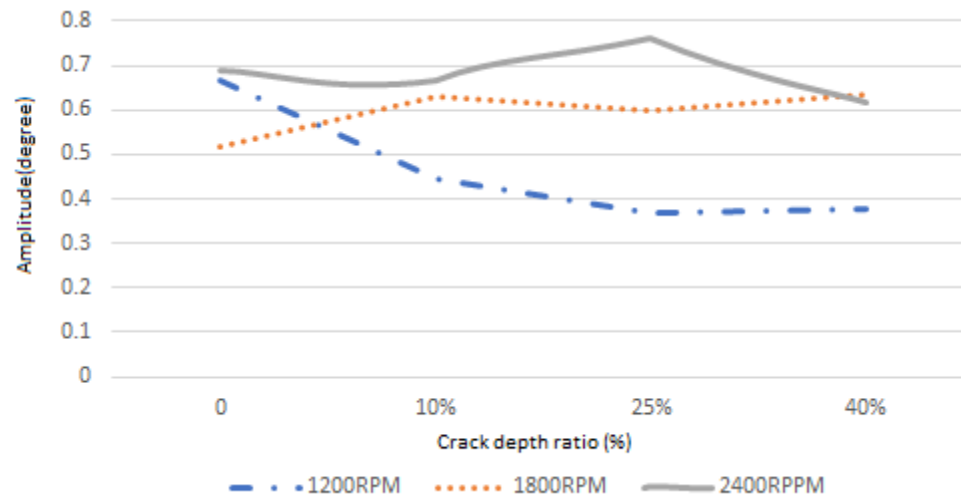


Figure 4.20: Peak values for TV

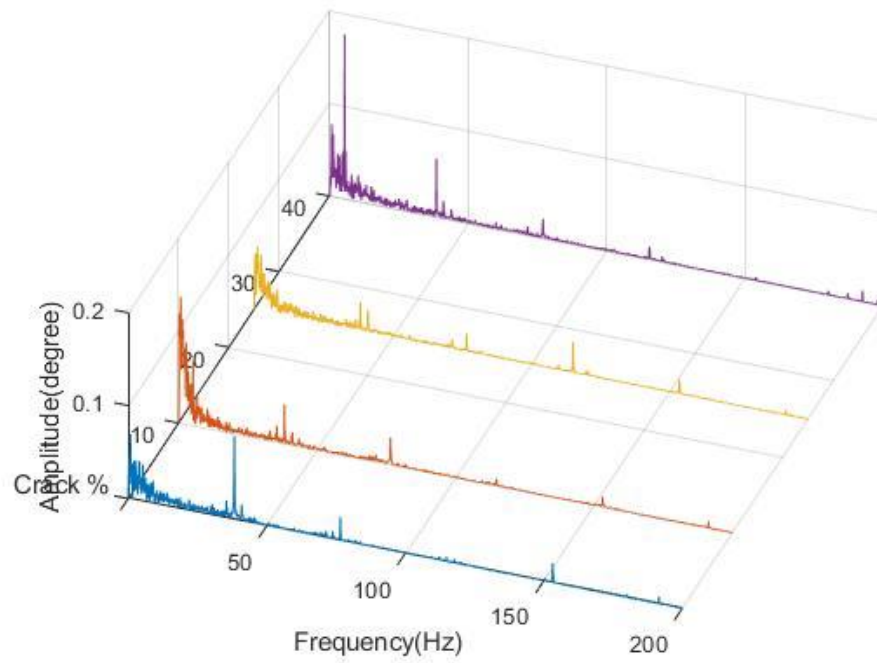


Figure 4.21: 3D FFT CH7 2440 RPM with rubbing

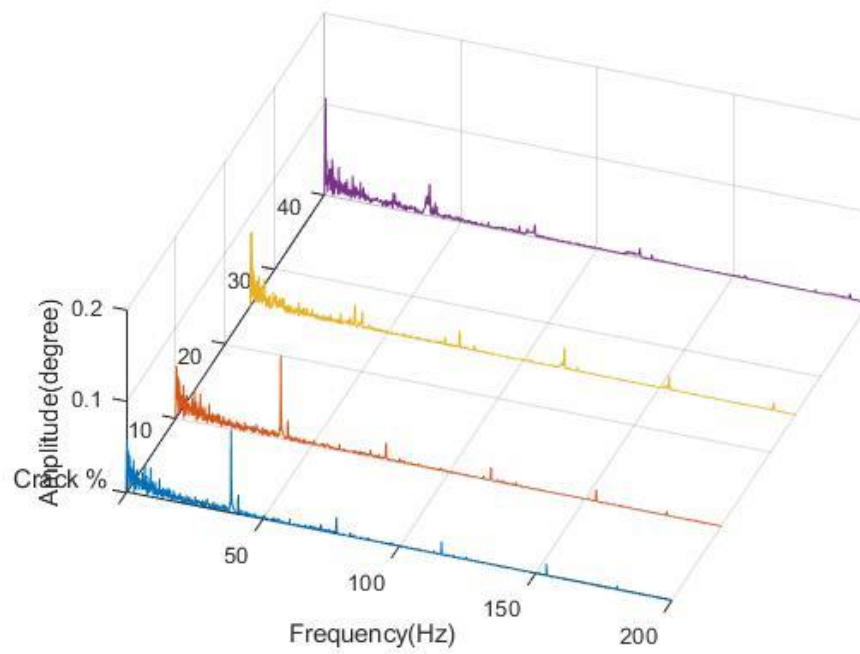


Figure 4.24: 3D FFT CH7 2440 RPM without rubbing.

4.3 Advanced Signal Processing

4.3.1 Time frequency analysis

4.3.1.1 Scalogram

The scalogram is a powerful signal analysis technique that is based on the continuous wavelet coefficients. The scalogram is basically a visual representation of the percentage of the energy for each coefficient of the matrix of the continuous wavelet coefficients that represent the signal. Complex Gaussian wavelet was used as the preferred wavelet for scalogram [58]. Figure 4.25 shows the scalogram of TV-12-00. All Speeds with 0% crack are considered as a reference. In the figure, it can be seen that the pulse which was caused by the rubbing at around 1.6 seconds on the signal, shows a high energy content in the scalogram. Additionally, it can be seen that there is a negligible energy content on the scalogram before and after the rub energy. Figure 4.26 shows the scalogram of the laser doppler vibrometer at 40% crack depth with 1200 RPM. Two high energy spikes are seen at around 3.5 seconds and 6 seconds that are corresponding to the high amplitude caused by the rubbing effect. Similar to the no crack case, there is not much difference between the scalogram energy response between before and after both rubs.

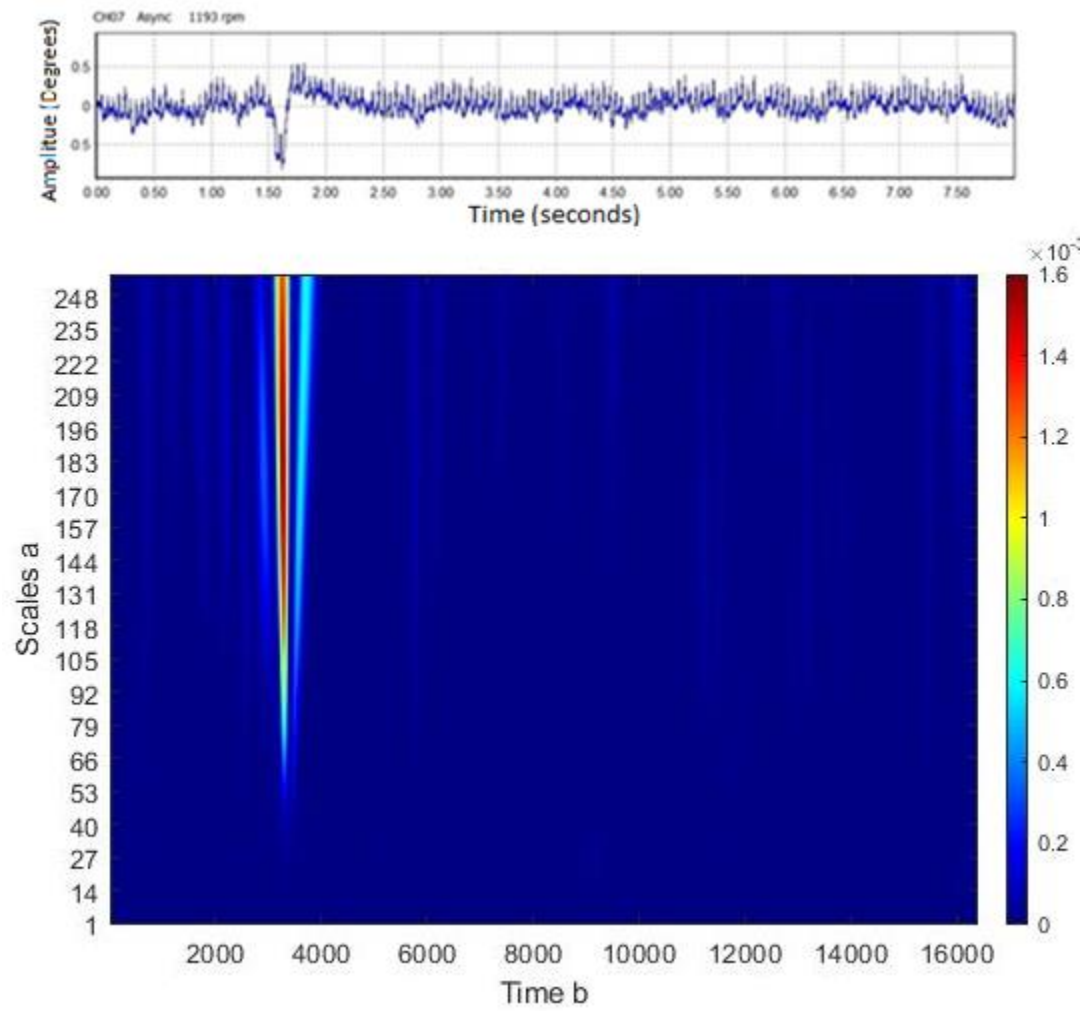


Figure 04.25: Scalogram of TV1200

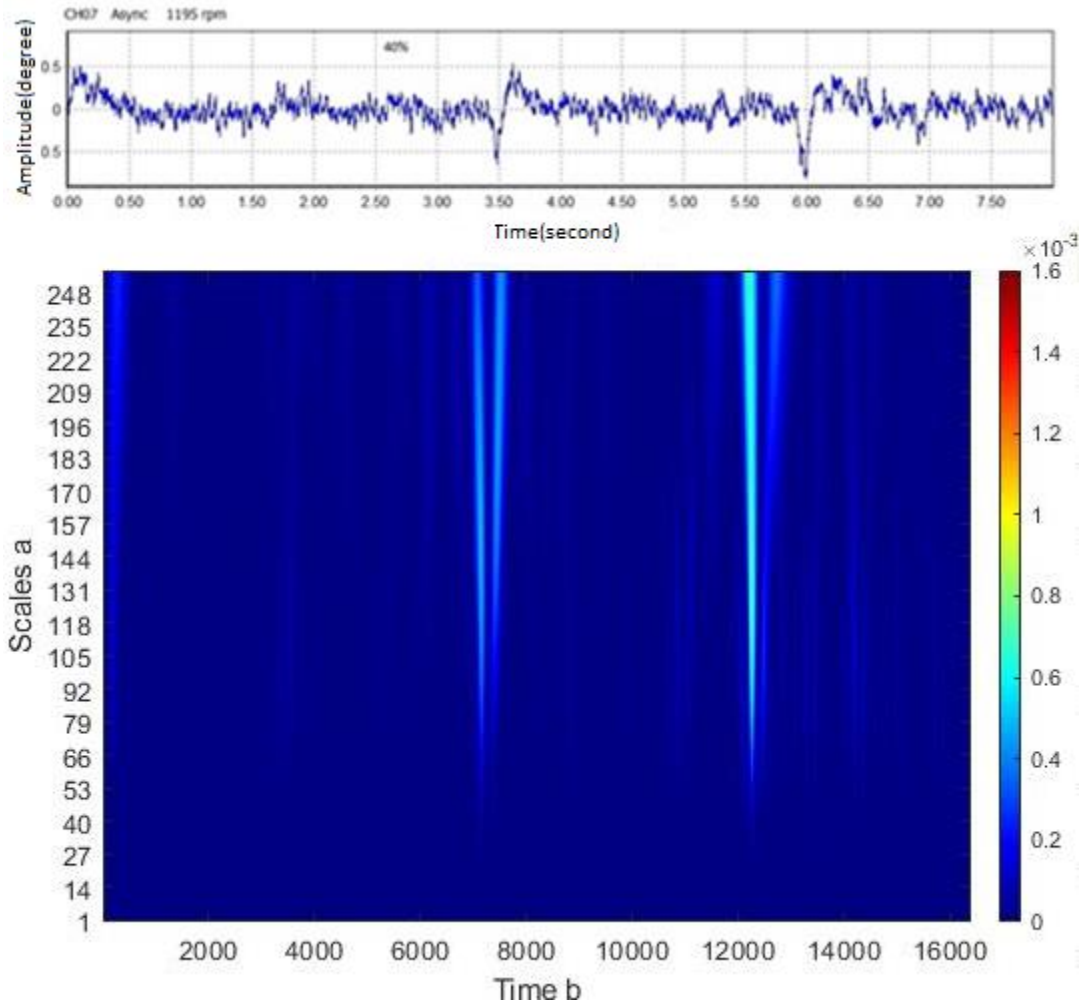


Figure 4.26: Scalogram of TV1240

Looking further into higher rotational speed cases to see the effect of crack with higher speed on the vibrational response through the scalogram. Figure 4.27 shows the scalogram of the torsional vibration at 0 crack depth with a rotational speed of 2400 RPM. A pulse that was caused by rotational rubbing against the blades is showing at a time of 4.5 seconds and a smaller one at a time slightly higher than 7.5. These pulses show a visible higher energy on the scalogram with intensity relative to their amplitude. The response on the scalogram is relatively similar after the rubbing when compared to before. Figure 4.28 shows the scalogram of the torsional vibration at 40% crack depth

with a rotational speed of 2400RPM. The scalogram does not show any visible or significant differences when compared to the case without the crack.

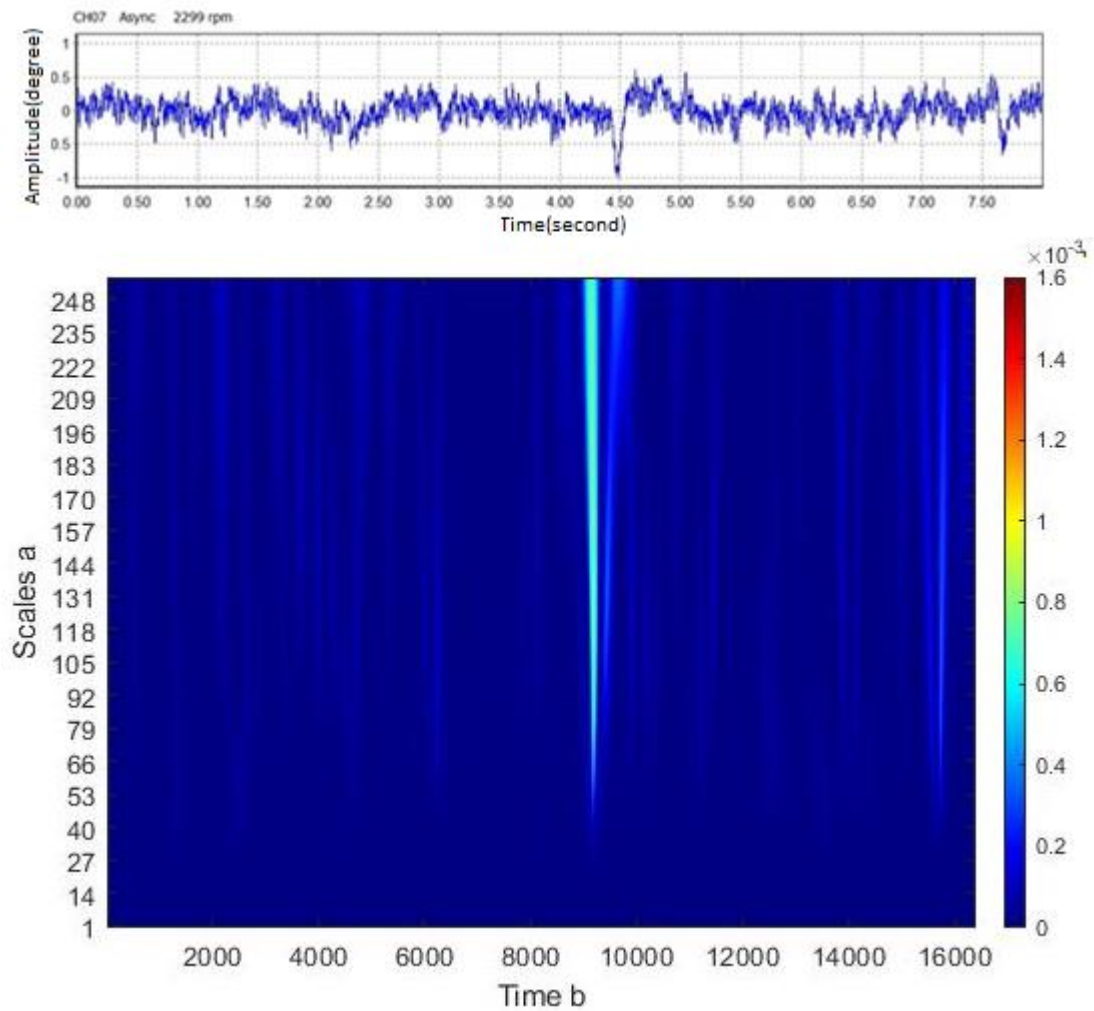


Figure 4.27: Scalogram of TV2400

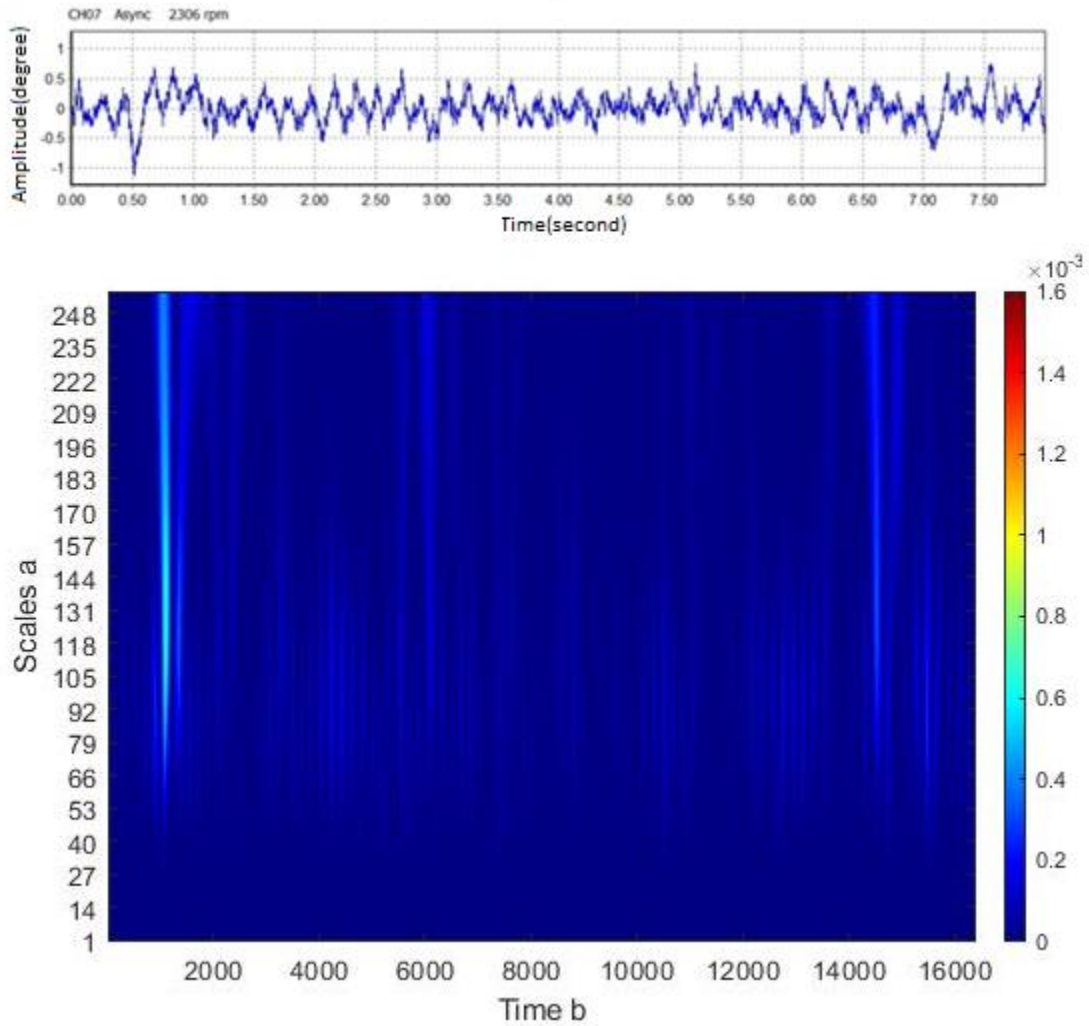


Figure 4.28: Scalogram of TV2440

4.3.1.2 Short time Fourier transform STFT.

Short time Fourier transform (STFT) is a time frequency signal processing technique. STFT has an advantage over the regular FFT of giving an indication about the relative location of the frequency along the signal. Figures 4.29 and 4.30 show the STFT with a window size of 128 and an overlap of 75%. It can be seen that the STFT was capable of detecting the high energy of the rubbing. However, there is no significant difference between the energy levels before and after the spike of the rub.

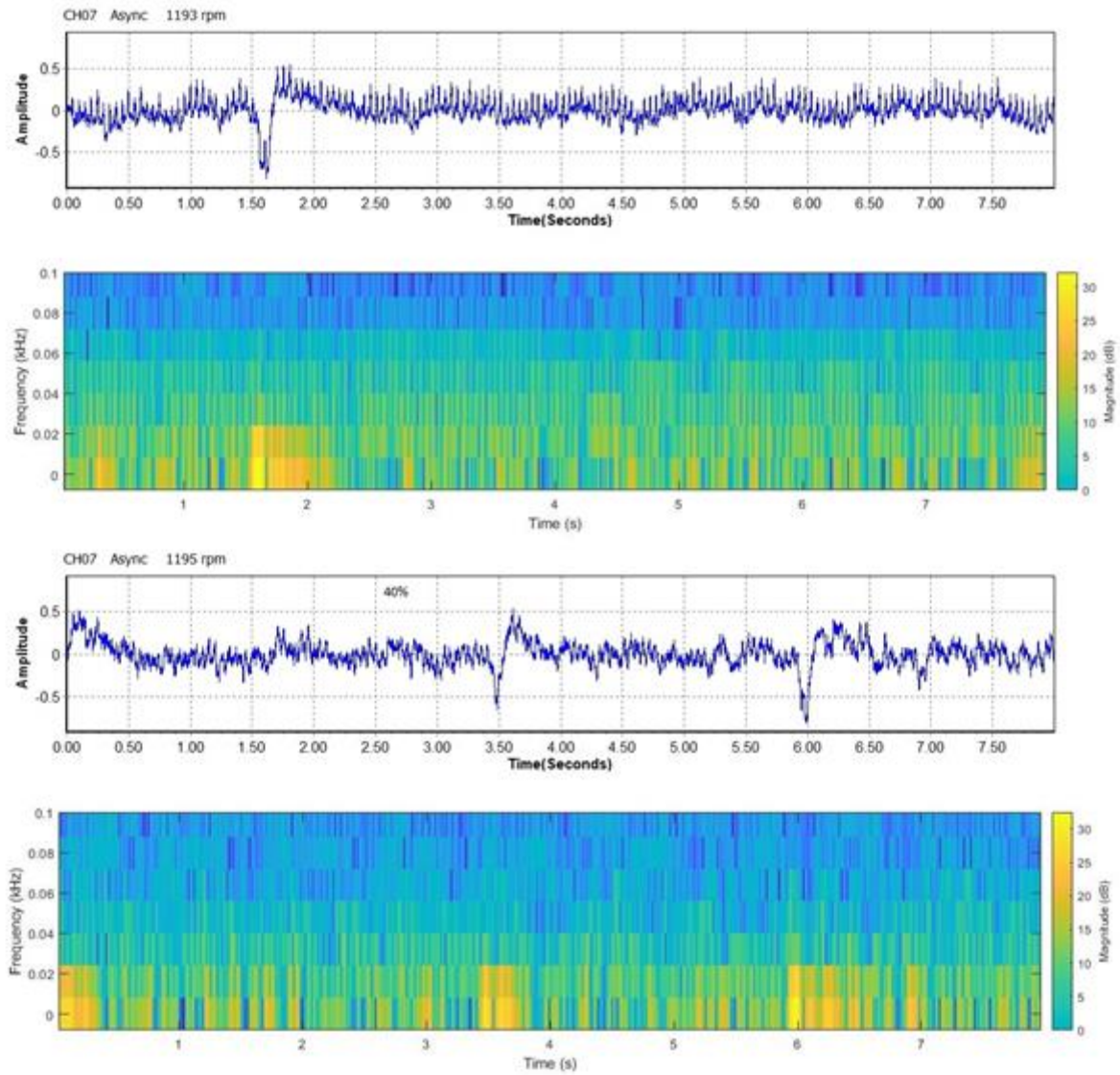


Figure 04.29: STFT for TV1200 & TV1240

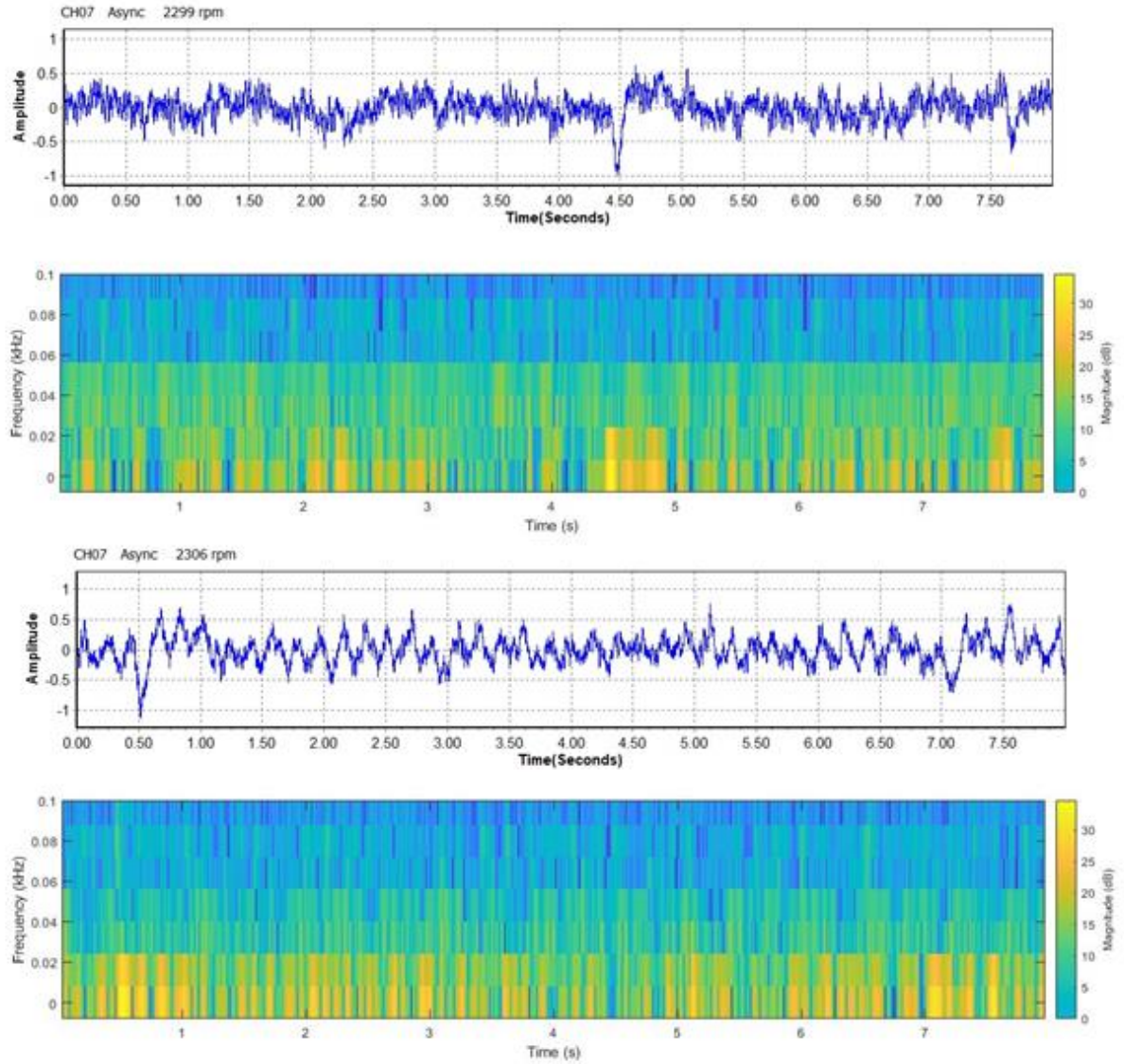


Figure 4.30: STFT for TV2400 & TV2440

4.3.2 Wavelet Analysis.

Wavelet analysis is an advanced signal processing technique that has powerful applications to vibration analysis. Moreover, the wavelet analysis has an advantage over the short time Fourier transform where it provides higher resolution in the time domain at high frequencies and higher resolution in the frequency domain at low frequencies [54]. This

way it has the capacity to detect even the slightest changes in frequencies at high frequency signals.

4.3.2.1 Discrete wavelet transform DWT.

Discrete wavelet transform was done on the signal of the experiment of laser doppler vibrometer 0% at 2400 RPM with a wavelet of Daubechies with 7 decompositions levels. This mother wavelet was used because it has the sinusoidal shape that best captures the signal in this study. Several other researchers are using it such as Prabhakar et.al. [58]. At the lower levels, as seen in Figure 4.31, the DWT did not capture any significant differences between the response before and after the pulse. Basically, the decomposition at lower levels acted like a filter to filter out the noise from the main signal. However, at a higher level the decomposition started showing the sinusoidal shape of the signal and the main signal was seen with the pulse like response induced by the rubbing effect on the shaft. Figure 4.32 shows the DWT of the torsional vibration signal for 40% crack at 2400 RPM, likewise, the signal was decomposed with 7 levels of decomposition and a Daubechies wavelet was used to build up the transform. The DWT was able to capture the complete signal including the pulse shape of the rubbing effect. Nevertheless, the DWT did not show any indication of a significant difference when comparing the case with the 40% crack with the case without crack.

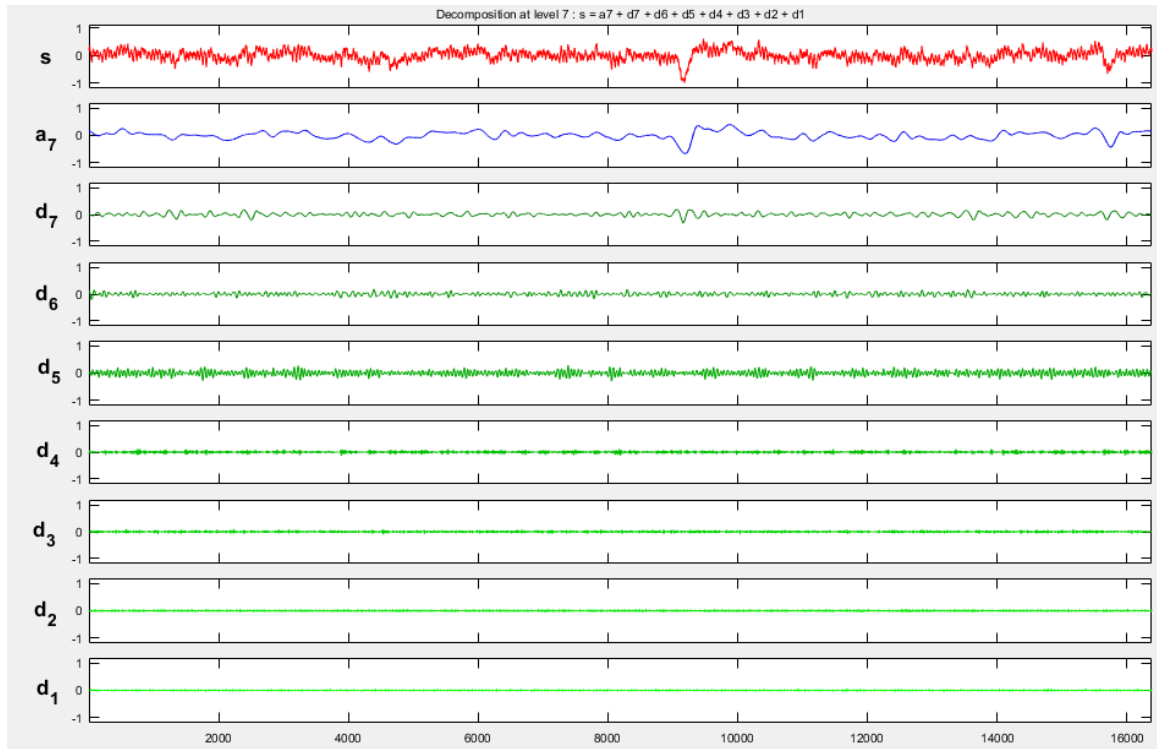


Figure 4.031: DWT for TV2400

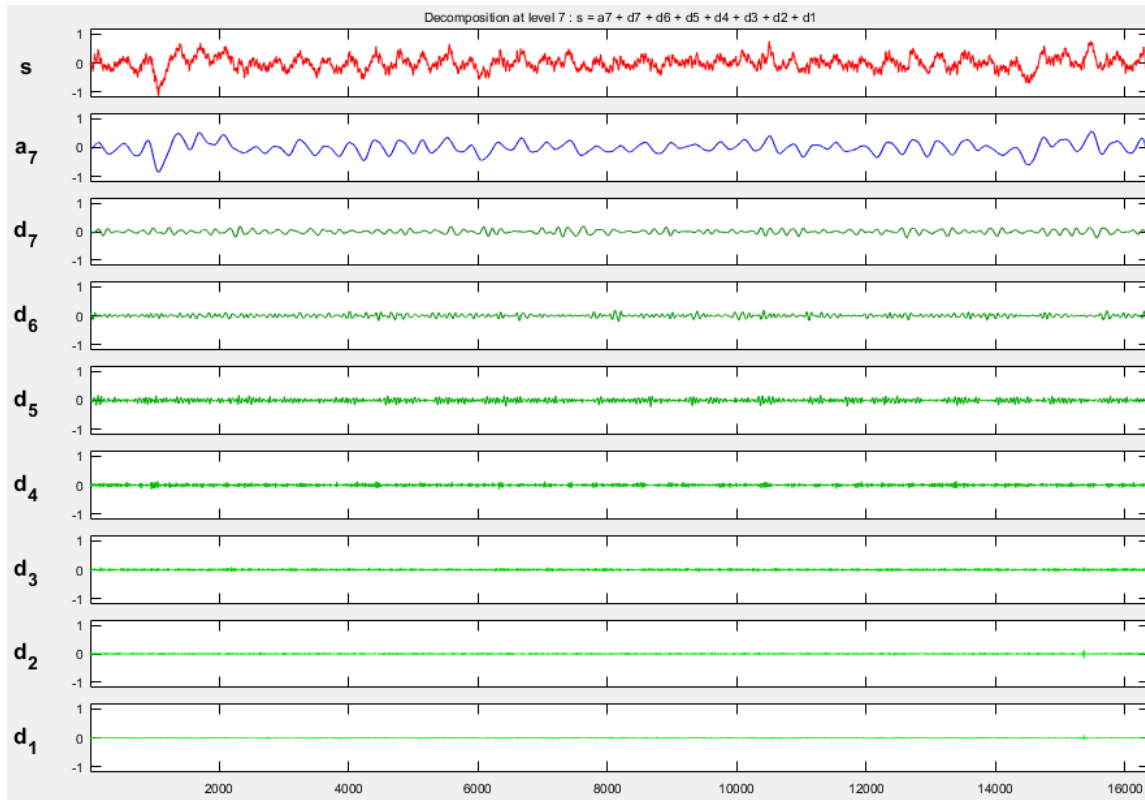


Figure 4.32: DWT for TV2440

4.3.2.2 Continuous wavelet transform CWT.

Continuous wavelet transform is one of the advanced techniques of the wavelet transform that is being used to analyze signals in the vibration analysis field [54]. Figure 4.33 shows the CWT that was used to transform the signal TV-24-00. In the figure, the top diagram shows the original signal without any modification and the lower part shows the coefficients of CWT calculated using Daubechies mother wavelet. This mother wavelet has been used by other researchers as reported in literature such as Prabhakar et.al. [58] due to its properties and capabilities. It has a smooth and regular shape which makes it suitable for low and high frequencies especially for higher orders wavelets. The CWT provides the energy of the signal at different locations along the signal. The higher the

energy the brighter the color and the lower the energy the darker the color. So, the pulse-like shape of the rubbing effect is seen at the CWT with a very bright color. Figure 4.34 shows the CWT of the signal TV-24-40 in order to compare it with the no crack case. The CWT in this figure shows the high energy of the pulse effect but no indication of a significant degree to the crack in terms of energy between before and after the pulse when compared to the case without crack.

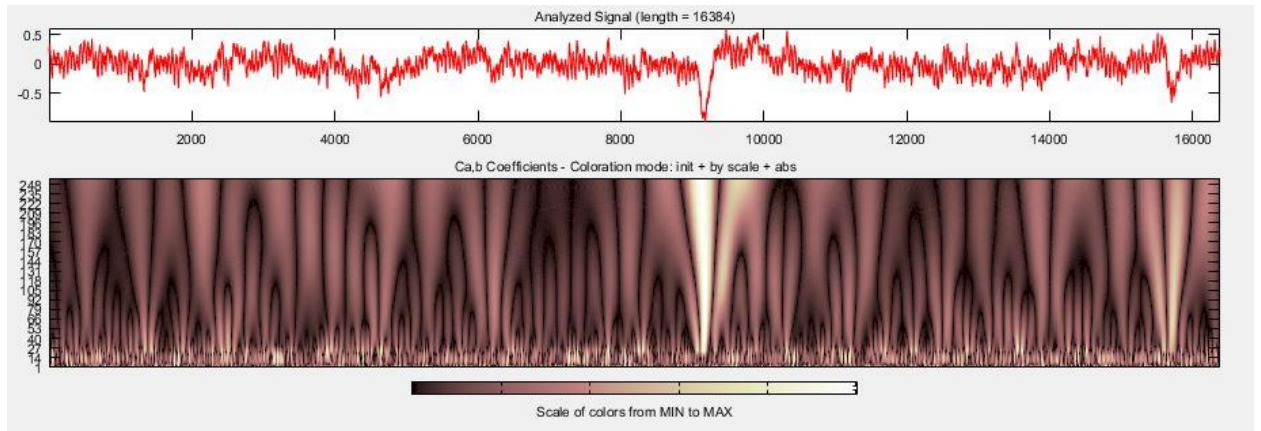


Figure 4.33: CWT for TV2400

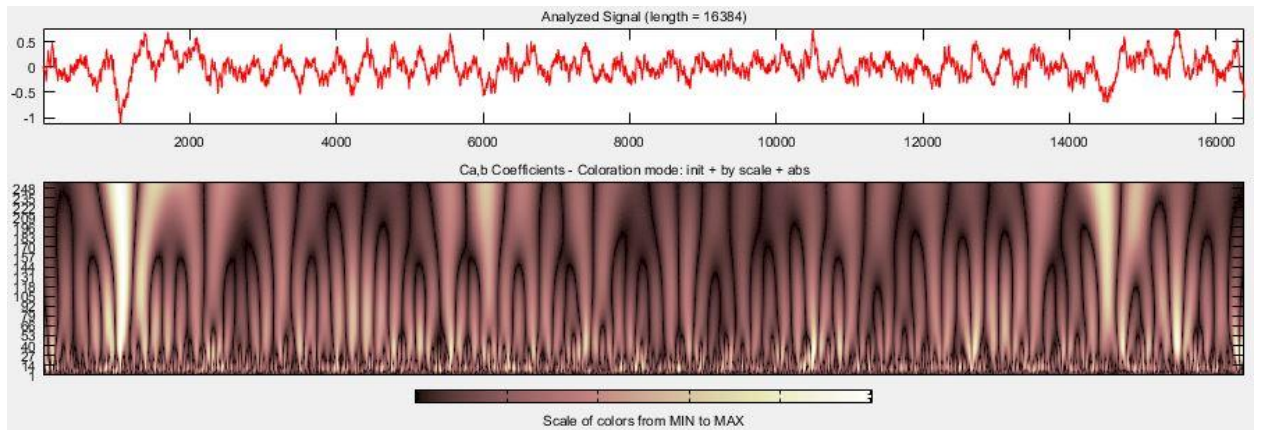


Figure 4.34: CWT for TV2440

4.4 Validation through finite element modeling.

In order to validate the result of the experiment shown in the previous sections, the following finite element model was developed and solved using Abaqus environment. The details of the model itself are explained and then the outcome of the model is laid out for the validation against the experimental results.

Steel was used as a material for both the disk and the shaft during the simulation. Both the disk and the shaft were meshed as (C3D8R) 8 noded-linear brick (Hex element) with reduced integration at size of 1/4in. The boundary condition at the bearing with all translational movements were fixed and rotational movements were free. The interaction between the disk and the shaft was tied in all degrees of freedom. The loading was applied at the motor side by applying a constant angular velocity while a pulse was exerted at the other side of the shaft as a torque. The pulse was applied at a single increment of time during the simulation. The readings were taken at a time increment of 0.00048828125 seconds to take into account the sampling and maximum frequency of 2048 and 400 Hz and to avoid aliasing. In the case of the shaft without the crack, there were 4478 elements with 6613 nodes in the model. Where in the case of the shaft with the 10% crack, there were 4534 elements with 6709 nodes. The disk was modeled without the holes to have less computations and to make the runs faster as seen in figure 4.35.

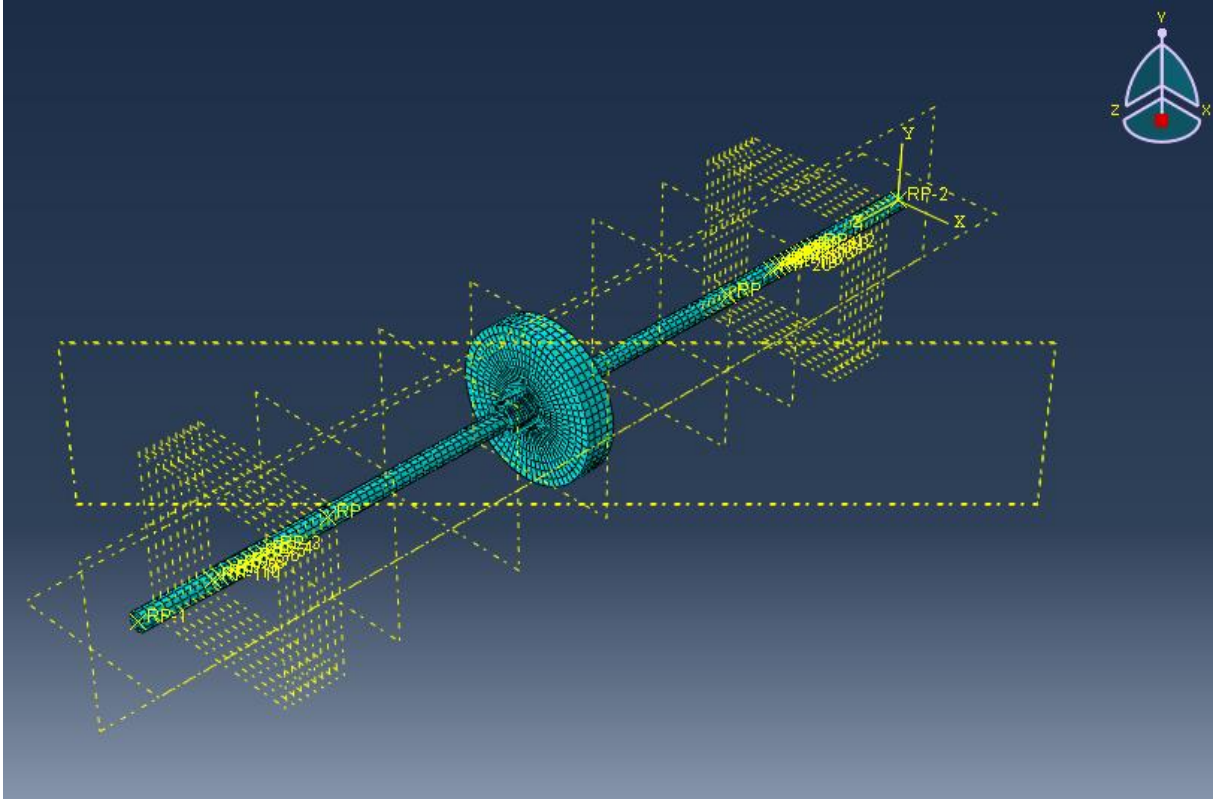


Figure 4.35: Finite element model for the rotor with planes

The finite element model was used to validate the result of the experiment at 0% crack and 10% crack depth at 1200 RPM. However, at higher speeds and higher crack depths, the model would cease to run to the steady state condition because of the deformation happening in the model.

Figure 4.36 shows the waveform of the 0% crack at 1200 RPM. The waveform is sinusoidal with the amplitude reaching up to 2 mils. This is similar to the behavior seen earlier in the experimental work as seen in figure 4.37. Moreover, FFT was also taken to see the frequency content of this signal. Figure 4.38 shows the FFT of the run with 0% crack at 1200RPM. 1x (20Hz) which is the running speed of 1200 RPM) is showing an

amplitude of 1.7 mils which is expected to happen due to the imbalance in the model as seen previously in the experimental run in figure 4.39.

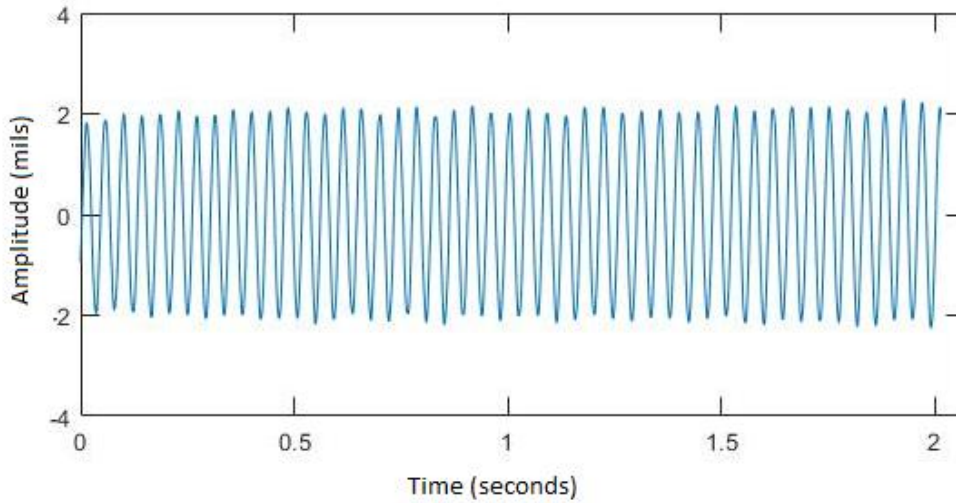


Figure 4.036: waveform of FEM result for 1200RPM at 0% crack

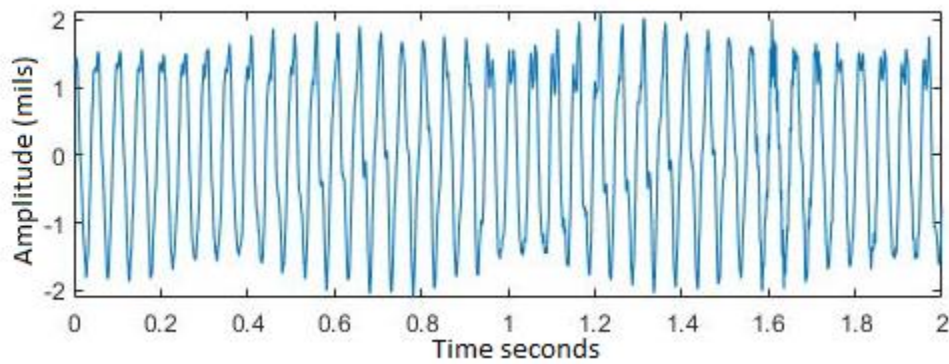


Figure 4.37: Waveform for experiment with 1200 RPM and 0% crack.

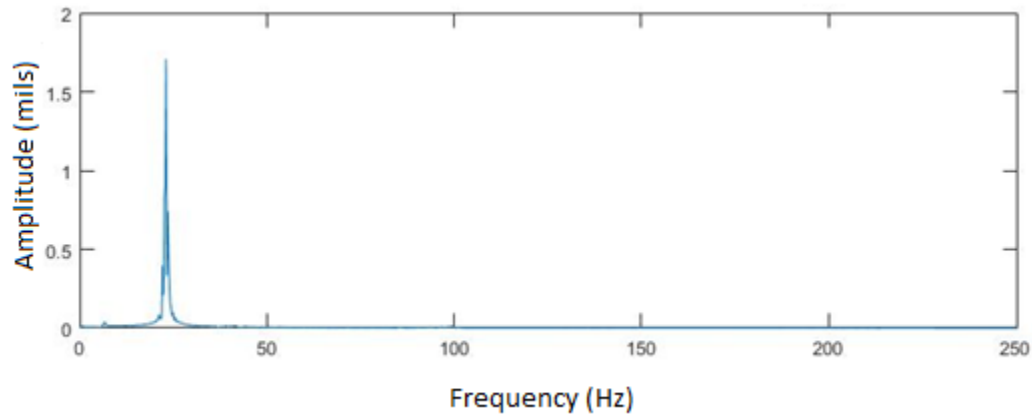


Figure 4.38: FFT for FEM with 1200 RPM and 0% crack.

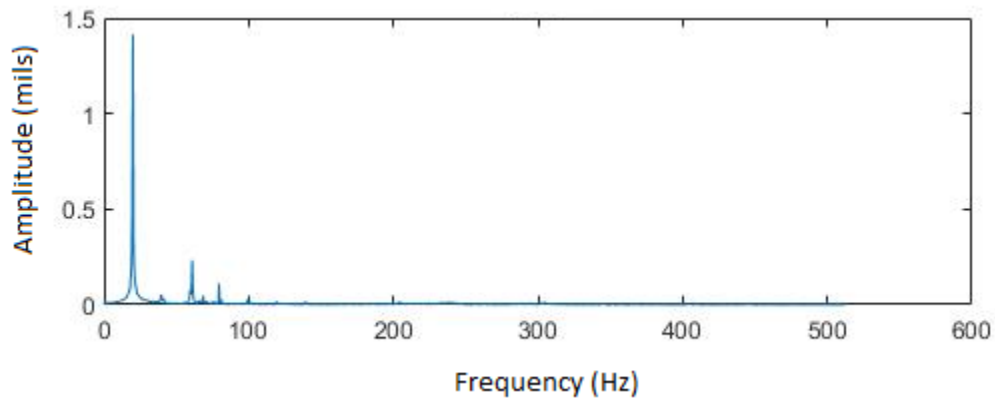


Figure 4.39: FFT for experiment with 1200 RPM and 0% crack.

The second run for 10% crack depth at 1200 RPM. The waveform shows that the signal is a sinusoidal wave with amplitude encapsulated between 2.5 and -2.5 mils as seen in figure 4.40 and in comparison, with the experimental run in figure 4.41. FFT was also taken as seen in the figure 4.42, the FFT has a peak at 1x, which is expected due to the imbalance in the system which is exciting the 1x. The experimental counterpart is in figure 4.43.

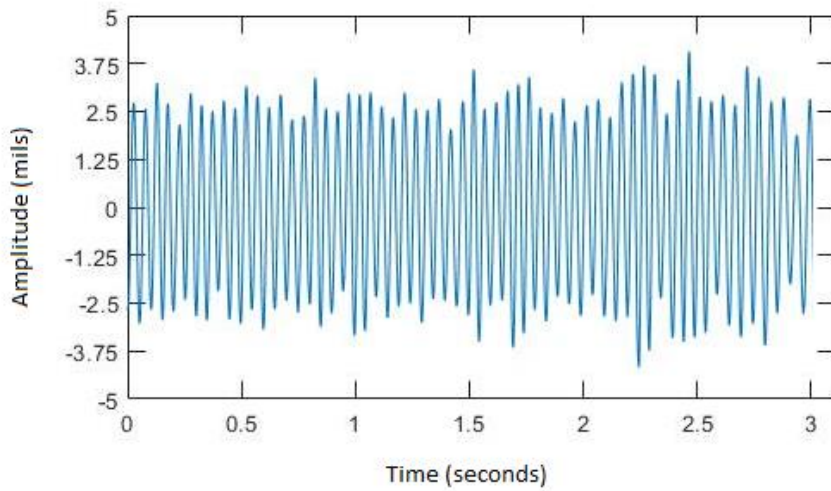


Figure 4.40: Waveform for FEM with 1200 RPM and 10% crack.

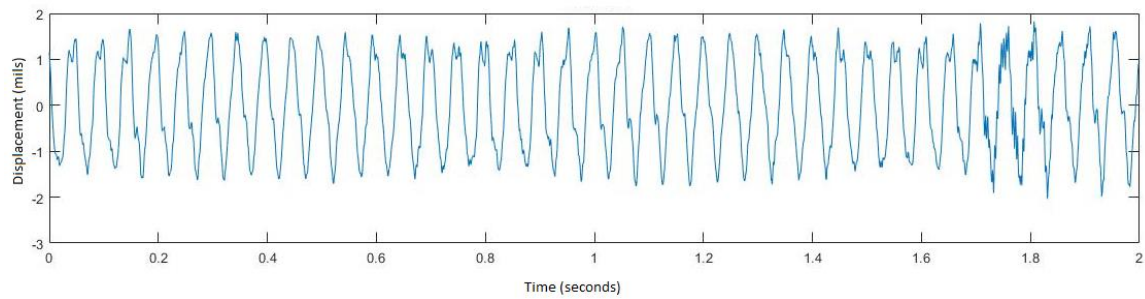


Figure 4.41: Waveform for experiment with 1200 RPM and 10% crack.

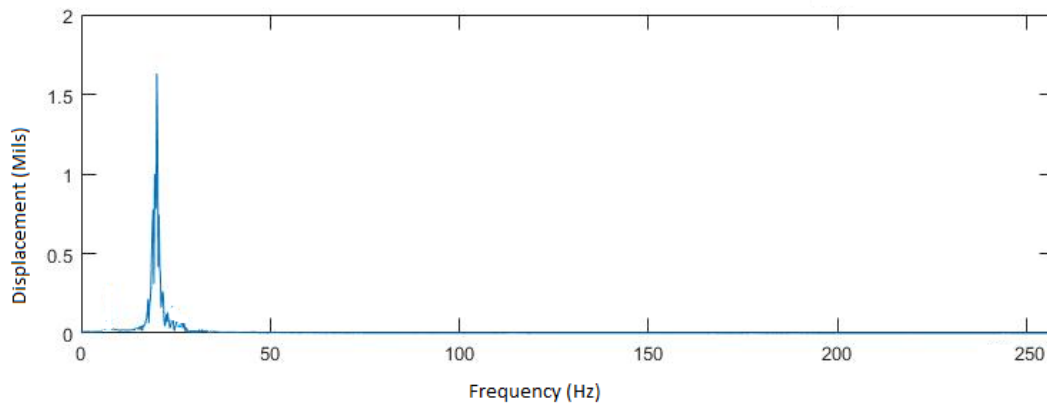


Figure 4.42: FFT for FEM with 1200 RPM and 10% crack.

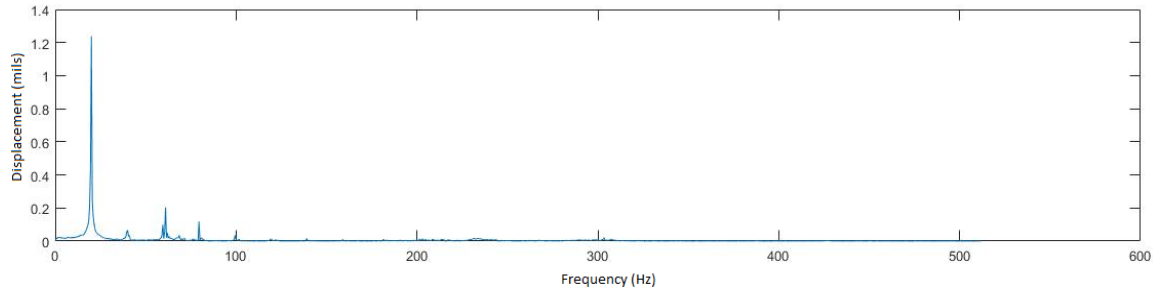


Figure 04.43: FFT for experiment with 1200 RPM and 10% crack.

Figure 4.44 shows the modal analysis of the finite element model of the first mode of the shaft with zero crack.

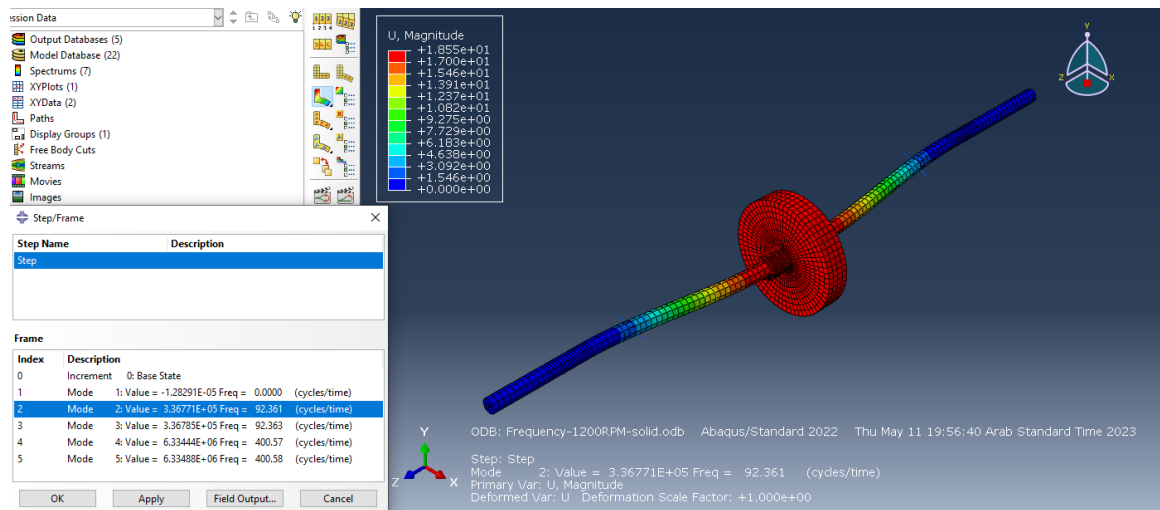


Figure 4.44: First mode of the reference shaft.

Figure 4.45 shows the modal analysis of the finite element model of the first mode of the shaft with 10% crack depth.

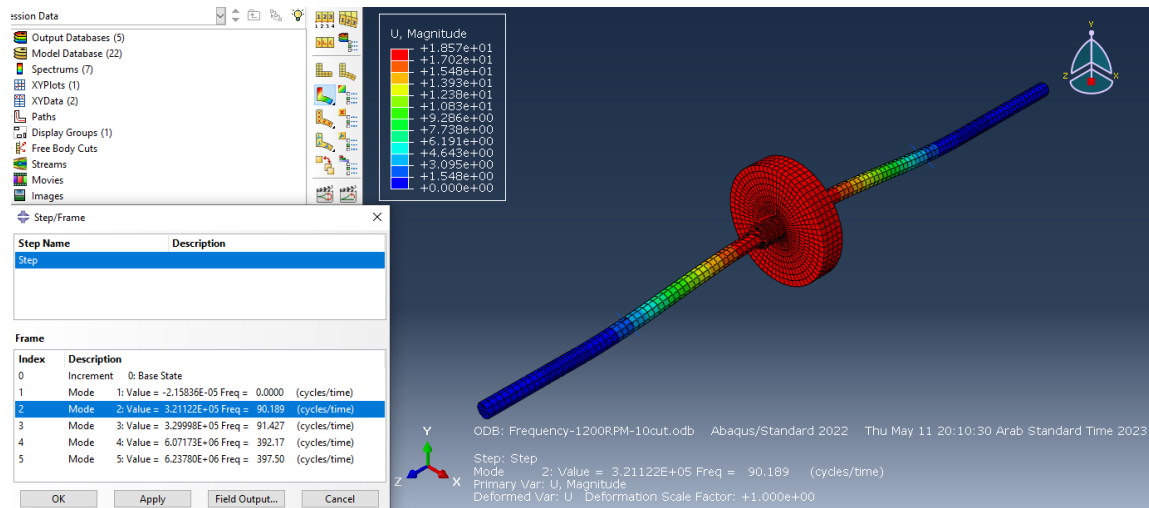


Figure 4.45: First mode of the shaft with 10% crack.

The results from the finite element model as seen in this section is very similar in nature to the results coming from the experimental runs which were seen in the previous sections. However, there is a difference of 28% between the experimental and the finite element model. The natural frequencies are reducing with the existence of the crack as seen in other papers [7, 16].

4.5 Sources of error.

There are some possible sources of error in this study that might have prevented the study from reaching the crack identification using the vibration analysis. Firstly, the crack was made using a milling machine which made the root of the crack have a flat surface instead of a sharp edge. This might have made what was called a crack in this study act more like a groove. Some other researchers have done a slit on the shaft surface and then simulated a real crack by making the crack propagate as done in [6]. Secondly, the running speeds used in this study were below the critical speed and do not coincide with 1/2 or 1/3 of the critical speed of. Thirdly, the shafts that were used in these experiments had large tolerances at the bearing side which might have resulted in slight size variation that effected

the bearing pre-load this could explain the lower values in amplitude found at the 25% crack depth in the lateral vibration runs compared to the other depths. Fourthly, the motor used in this experiment was a variable speed drive induction motor. That means the motor has a certain slip that varies with the rotational speed. These faults were apparent in the lateral and torsional vibrations result where they caused the small sidebands in the frequency spectrum as seen in figure 4.17 through 4.20 and figure 4.27. Fifthly, the imbalance in the system that was induced to excite the system might have caused some error in the result. Lastly, some error due to digitalization might have happened where the maximum frequency was set to be 400 Hz for the proximity probes for the lateral vibrations and 800 Hz for the laser doppler vibrometer for the torsional vibrations. These maximum frequencies mean any frequency with higher magnitude will not be visible in the spectrum.

CHAPTER 5

Summary and Conclusion

In this thesis, an experimental study was conducted to investigate the relationship between crack size, rotational speed, and vibration response. A finite element model was used to validate the experimental results. Experimental setup was prepared where four shafts with crack sizes of 0%, 10%, 25%, and 40% of the shaft diameter were tested. Each shaft was tested at 3 different rotational speeds which are 1200, 1800, and 2400 RPM. During the experiment a rubbing effect was induced to excite the system. Two different types of sensors were used to acquire the data from the experiment; a proximity probe to measure the shaft lateral vibration and a laser doppler vibrometer to measure torsional vibration. After capturing the raw data from the experimental setup, different signal processing techniques were implemented to study the data. Firstly, traditional vibration analysis techniques to analyze the information coming from the proximity probe such as timewave analysis, FFT analysis with averaging and overlapping, and orbit analysis. Secondly, the torsional vibration analysis using the same techniques for the proximity probe but this time for the signal coming from the laser doppler vibrometer. These techniques are the timewave analysis and the FFT analysis. Finally, advanced signal processing methods were used to further analyze the data coming from laser doppler vibrometer. These techniques are divided into two different categories. The first one is the time frequency analysis which include the scalogram and the short time Fourier transform (STFT). Moreover, the second one is the wavelet analysis which include the discrete wavelet transform (DWT) and the

continuous wavelet transform (CWT). Then a finite element model was constructed to test and validate the experimental work. Using FEM, the 1200 RPM at 10% crack and 0% crack were tested and found to agree with the experimental results. The conclusion drawn from this experimental study are listed as follows:

- In the lateral vibration, the overall vibration of the signal was measured with significant increase on the side that has the crack at 40% crack depth. The crack signature was the increase in the RMS and peak-to-peak values for the signal. So, the proximity probes were able to detect at 40% crack depth but not at a lower depth.
- In the torsional vibration, the traditional vibration techniques did not show significant indication in terms of vibration response to the existence of the crack at all tested speeds and crack depths.
- Using advanced signal techniques, namely, the scalogram, the short time Fourier transform, and the wavelet transforms, the signal was captured properly including the pulse that was created by the rub. However, it did not show any significant indication for the existence of the crack. Hence, with the current setup up to 40% crack depth for a slant cracked shaft there is no significant response that could be attributed to the existence of the crack.
- One of the major sources of error in this study is the shape of the base of the crack. In this experiment it was flat where in real crack it should have a sharp edge. Another major source of error is that the experiments were run below the first critical speed. These are among the possible sources of error that were discussed in section 4.6.

Appendix

MATLAB code for plotting waveform.

```
close
clear
clc
% fft and time wave
wave=load('CH1 0%-2299RPM-With Rub-Short.txt')
x=wave;
L=length(x);
t=0:1/1023:1;

plot(t,x);
xlabel('Time(seconds)')
ylabel('Amplitude(mils)')
title('Waveform OH2400')
```

MATLAB code for plotting FFT.

```
close
clear
clc
% fft and time wave
wave=load('CH1 40%-1195RPM-With Rub.txt')
Fs=2048;
x=wave;
L=length(x);
t=0:1/Fs:(length(x)-1)/Fs;;

%c = cceps(x);

%plot(t,c)
%xlabel('Time (s)')
%title('Complex cepstrum')

wave=load('CH1 40%-1195RPM-With Rub.txt')

mean0=mean(x);
x=wave(:,1);
Fs=1024;
x=x-mean0;
L=length(x);
t=0:1/Fs:(length(x)-1)/Fs;;
%w = hann(L);
%x=x*w';
subplot(2,1,1)
plot(t,x)
title('Waveform')
```

```

Y = fft(x);
%N = length(x_hann);
%%Y = fft(x_hann, N);

%f_axis = (0:N/2-1)*(Fs/N);

%Y_mag = abs(Y/N);
%Y_mag = Y_mag(1:N/2);

P2 = abs(Y/L);
P1 = P2(1:L/2+1);
P1(2:end-1) = 2*P1(2:end-1);
f = Fs*(0:(L/2))/L;

subplot(2,1,2)
plot(f,P1)
title('FFT')

```

MATLAB code for getting the peak-to-peak and RMS values.

```

close
clear
clc
%1
wave=load('CH7 0%-1193RPM-Without rub.txt');
x=wave-mean(wave);
root1=rms(x)
pp1=peak2peak(x)
x1=max(x)
n1=min(x)

```

MATLAB code for plotting the 3D FFT

```

close
clear
clc

f = load('f PPch7.txt');
P11 = load('P1 CH7 0%-2299RPM-With Rub.txt');
P12 = load('P1 CH7 10%-2296RPM-With Rub.txt');
P13 = load('P1 CH7 25%-2304RPM-With Rub.txt');
P14 = load('P1 CH7 40%-2306RPM-With Rub.txt');
fr = [0 10 25 40];

plot3(f, fr(1)*ones(size(f)), P11);
grid on
hold on
plot3(f, fr(2)*ones(size(f)), P12);
plot3(f, fr(3)*ones(size(f)), P13);
plot3(f, fr(4)*ones(size(f)), P14);

```

```

%box on;
hAxis = gca;
hAxis.XRuler.FirstCrossoverValue = 0; % X crossover with Y axis
hAxis.XRuler.SecondCrossoverValue = 0; % X crossover with Z axis
hAxis.YRuler.FirstCrossoverValue = 0; % Y crossover with X axis
hAxis.YRuler.SecondCrossoverValue = 0; % Y crossover with Z axis
hAxis.ZRuler.FirstCrossoverValue = 0; % Z crossover with X axis
hAxis.ZRuler.SecondCrossoverValue = 0; % Z crossover with Y axis
grid on

xlabel('Frequency(Hz)')
ylabel('Crack %')
zlabel('Amplitude(degree)')
view(20, 60)

```

MATLAB code for plotting scalogram

```

close
clear
clc
wave=load('CH7 40%-2306RPM-With Rub.txt');
%tfrrscalogram(wave,1:16384,7,0.0001,0.5,10,1);
COEFS = cwt(wave,1:256,'mexh');
SC = wscalogram('image',COEFS);

colormap(jet);
caxis([0,0.0016]);
%h= colorbar;
%set(h, 'Ticks', 0:0.0002:0.0016);
title('Scalogram TV2440')

```

MATLAB code for plotting STFT

```

clear
clc
close all

Fs = 2048; % sampling frequency
Winsize = 256 ; % window size
Overlap = floor(.75*Winsize); % overlap as percentage integer

% 1 STFT
wave=load('CH7 0%-2299RPM-With Rub.txt');
x=wave-mean(wave);
t=0:1/Fs:(length(x)-1)/Fs;
subplot(2,1,1)
stft(wave,Fs,'Window',hann(Winsize),'OverlapLength',Overlap,'FFTLenght',
,Winsize,'FrequencyRange','onesided');
title('CH7 0%-1193RPM-With Rub')

```

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