

## Research Article

### High Frequency Acceleration Envelope Power Spectrum for Fault Diagnosis on Journal Bearing using DEWESOFT

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**Abstract:** The aim of study is to apply the condition monitoring technique in the journal bearing to detect the faults at an early stage and to prevent the occurrence of catastrophic failures. This study presents fault diagnosis on journal bearing through the experimental investigation at high rotational speed. Journal bearings are widely used to support the shaft of industrial machinery with heavy loads, such as compressors, turbines and centrifugal pumps. The major problem in journal bearing is catastrophic failure due to corrosion and erosion, results in economic loss and creates high safety risks. So, it is necessary to provide condition monitoring technique to detect and diagnose failures, to achieve cost benefits to industry. High frequency acceleration enveloping facilitates the extraction of low amplitude, high frequency signals associated with repetitive impacts in journal bearings, providing a key tool for early detection in the onset of bearing damage and similar machinery health problems when coupled with standard FFT analysis. The DEWESOFT software-based methods for implementing and interpreting high frequency acceleration enveloping are presented and compared. In this study the application of STFT (Short Time Fourier Transform) and Autocorrelation through FFT are used for processing vibration signal to detect faults in journal bearing is presented. A bearing testing apparatus is used for experimental studies to obtain vibration signal from a healthy bearing and fault bearing.

**Keywords:** Condition monitoring, journal bearing, power spectrum, vibration

## INTRODUCTION

Vibration monitoring is a procedure for detecting the presence of vibrations and identifying their characteristics in mechanical machinery provided by Rajendran (2009). This study is concerned with the vibration monitoring of high speed rotating machines such as shafts and (hydrodynamic) journal bearings. Bearings are essential components in machines that operate at high speed. They have high load carrying capacity and are low in cost and they are designed to allow relative positioning and rotational freedom while transferring the speed provided by Ugural (2003). The failures in the journal bearings can lead to large losses and major safety problems, hence the need to monitor them and diagnose any problems as early as possible suggested by Raharjo *et al.* (2010). Journal bearings are widely used in pumps, turbines, compressors and gearboxes. The vibrations occur either axially or radially and their magnitude can be affected by alignment and balancing elements in the rotating parts as well as due to other faults and deficiencies. Vibration monitoring technique is used to identify problems early at the developing stage before they could otherwise

become serious and cause unscheduled downtime provided by Girdhar and Scheffer (2004). Wang and Lee (1999) examined vibration monitoring can detect mechanical looseness, eccentricity and defective bearings. By means of the vibration levels, it is also possible to detect poor maintenance such as an improper installation of a bearing or an inaccurate alignment of the shaft. In fact, misalignment and imbalance along with thermal growth are the most common problems in mechanical systems. De Castro *et al.* (2008) investigated the oil instability in Journal bearings occurs when the bearings are loaded lightly and the shaft whirls at a frequency that is close to half of the rotor angular speed. Taylor and Scatchard (2008) identified the Vibration monitoring is an effective tool and it has the advantage of being non-intrusive. It is often in combination with other non-intrusive condition monitoring techniques such as acoustic emission. Bannister and Findlay (1999) suggested that the vibration monitoring is considered to be the suitable tool, for low-speed rotating machinery because of the typically low energy loss rates and difficulty in measuring their fundamental frequency of operation. Kuboyama (1997) investigated the low speed journal

bearings are an exception as they have been shown to be capable of being subjected to vibration monitoring using band pass filtering to eliminate noise. The two components of the vibration signals are of importance for monitoring, namely frequency and amplitude. The standard methods for measuring vibration include the overall vibration, phase, acceleration enveloping, high frequency detection and other sensor resonant technologies identified in SKF (2000). DEWESOFT is currently used in automotive industries for vibration and condition monitoring of rotating machineries.

The main objective of this paper is to study the non-stationary signals of the journal bearing is produced due to fault vibration signals and the bearing fault detection method using autocorrelation through FFT and STFT is presented. In this study, first the original time domain signals are collected and the signals are transformed into frequency domain. Then, the signals are denoised through autocorrelation and STFT technique to identify the faults. The main objectives are as follows:

- To design the comprehensive experimental test system including mechanical, electrical and measuring systems for testing a journal bearing under different operating conditions
- To investigate surface vibration signal generation models for understanding the dynamic behaviors of the journal bearing
- To develop a condition monitoring measurement and analysis for the journal bearing
- To examine the non-stationary characteristics of surface vibration signal Response of the journal bearing under various conditions

- To study the characteristic of surface vibration due to mechanical looseness
- To study the characteristic of surface vibration due to improper lubrication
- To examine the characteristic of surface vibration due to different bearing surface faults caused by scratching
- To investigate autocorrelation through FFT and STFT for processing and analyzing the datasets from the accelerometer sensor in the time and frequency domains
- To develop the surface vibration characteristic of journal bearing for early fault detection

Also, the application of advanced signal processing was established to extract useful diagnostic features from the acquired vibration signals in both time and frequency domain. Finally, it was also concluded that the use of different signal processing methods is often to achieve meaningful diagnostic information from the signals. Hence, the outcome would largely contribute to the development of effective intelligent condition monitoring systems which can significantly reduce the cost of maintenance in the company.

## EXPERIMENTAL SETUP

The experimental set up consists of motor of 0.5 HP, 3700 RPM connected to the rotor, through a coupling supported with one ball bearing and journal bearing at the end of pulley and attached to a loading system is shown in Fig. 1. The vibration data is collected from the journal bearing through the



Fig. 1: Experimental setup



Fig. 2: No fault on the journal bearing



Fig. 3: Bore fault on the bearing (fault F1)



Fig. 4: Bore fault with hole on the bearing (fault F2)



Fig. 5: Material removal and bore fault with hole on the bearing (fault 3)

accelerometer sensor 621B40 and the vibration signals are processed with DWESOFT software. Peaks of low amplitude when bearing is normal, but amplitudes will go high when there are defects in journal bearing.

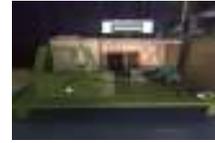


Fig. 6: Data acquisition system

Amplitude of peaks will be prominent about frequency when the bearing damage is severe.

In the experimental investigation, the vibration signal was collected at an ideal condition without any crack or wear that is healthy surface at a shaft speed of 3600 RPM and at constant load condition as shown in Fig. 2. The readings were recorded at a sampling rate of 5000 Hz. An eccentric bore fault at stage F1 was produced on inner surface of bearing and the corresponding vibration readings were taken shown in Fig. 3. Readings were also taken when the crack of 3mm on inner surface of the bearing and the material was broken partially and completely at stage F2 and F3 shown in Fig. 4 and 5. Figure 6 shows the data acquisition system. The Bearing Frequency (BF) is calculated to be  $= \text{RPM}/60 = 3600/60 = 60 \text{ Hz}$ . The accelerometer type 621B40, 1 M I sensor has sensitivity  $1.02 \text{ mv/m/sec}^2$  and frequency range up to 10 KHz, is used in vertical position on the bearing housing with magnetic base to collect vibration data. The other end of the accelerometer is connected to the vibration data collector with DWESOFT software. DWESOFT software is used to develop STFT and autocorrelation through FFT envelop power spectrum. Thus, the various types of faults are induced on the bearing surface and the corresponding signals for each fault are recorded and analyzed to check its severity.

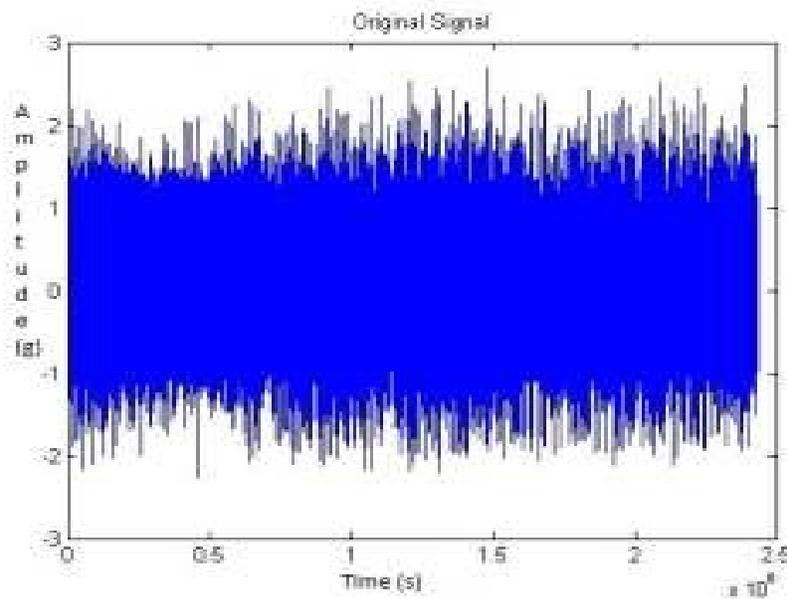


Fig. 7: Time domain signal for no fault

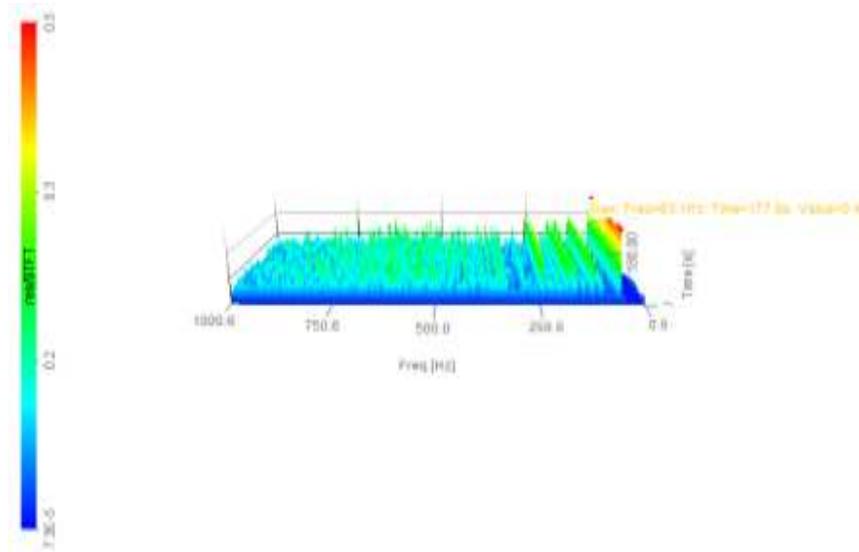


Fig. 8: STFT signal for no fault

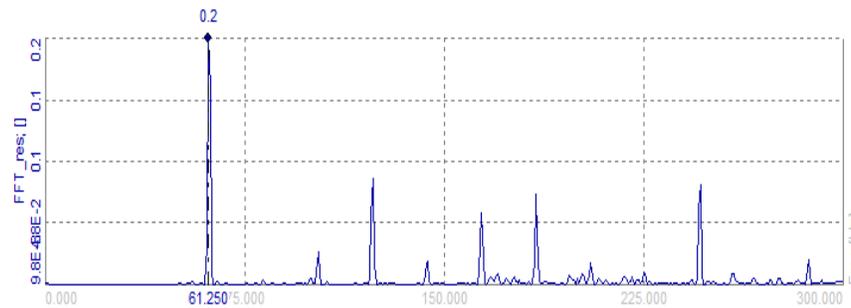


Fig. 9: Frequency amplitude domain response for no fault

## RESULTS AND DISCUSSION

A typical time domain signal obtained from the experimental setup without any defect, using accelerometer is given in Fig. 6. This is further processed using various signal processing techniques based on STFT and principle of Autocorrelation enveloped Power Spectrum Density (PSD) is presented. The PSD plot shows that the frequency in Hz along X-axis and the amplitude  $g$  in  $m/sec^2$  along Y-axis. The Fig. 6 to 30 explains the plots under various bearing conditions.

**Autocorrelation through FFT:** The PSD plot shows that the frequency in Hz along X-axis and the amplitude ' $g$ ' in  $m/sec^2$  along Y-axis. The Fig. 7 and 9, shows the time domain signal and FFT signal through autocorrelation technique, for no fault i.e., healthy bearing. From this power spectrum density plot, it is observed that through STFT the bearing frequency is 63.1 Hz with peak amplitude of 0.4 for the shaft speed of 3600 rpm. After applying the autocorrelation through FFT, the PSD plot shows the frequency of 61.25 Hz with the peak amplitude of 0.25. From the results it was

found that the autocorrelation through FFT technique is used to reduce the additional noise generated due to electrical interference during running conditions. Also it is clearly indicated that there is no fault. Results shows the important features of frequency content including the shaft frequency and harmonics of the shaft frequency of the vibration signal at 3600 rpm (61 Hz). There is no indication of the fault in the frequency spectrum.

In the journal bearings, half looseness may only develop at operating speed and temperature. First, the half looseness is provided on the bearing casing and the signals are recorded through the experimental set up. The Fig. 10 and 12 shows the time domain signal and autocorrelation through FFT signal, for half looseness of the bearing. After applying the STFT, the PSD plot shows the frequency of 123.8 Hz shown in Fig. 11. From this Power Spectrum Density (PSD) plot, it is observed that the bearing frequency for full mechanical looseness is 123.125 Hz shown in Fig. 12. Thus, after applying autocorrelation, considerable peak amplitude is getting reduced due to electrical interference of motor. This increased frequency is due to combination

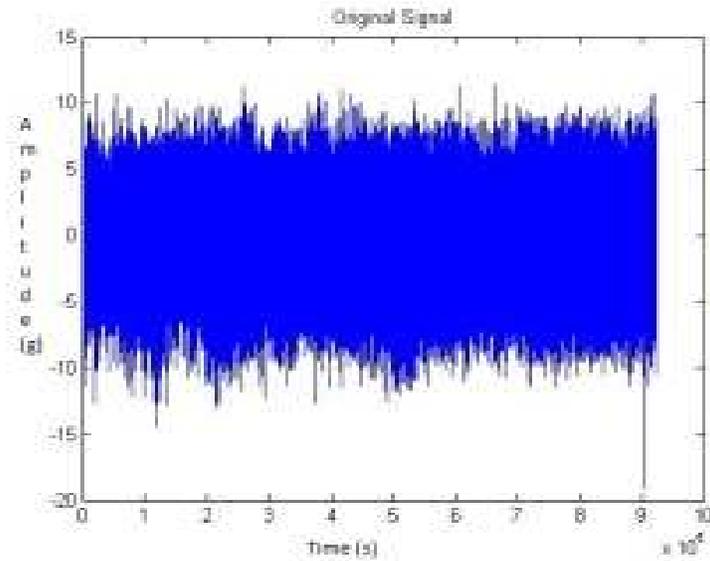


Fig. 10: Time domain signal for half looseness

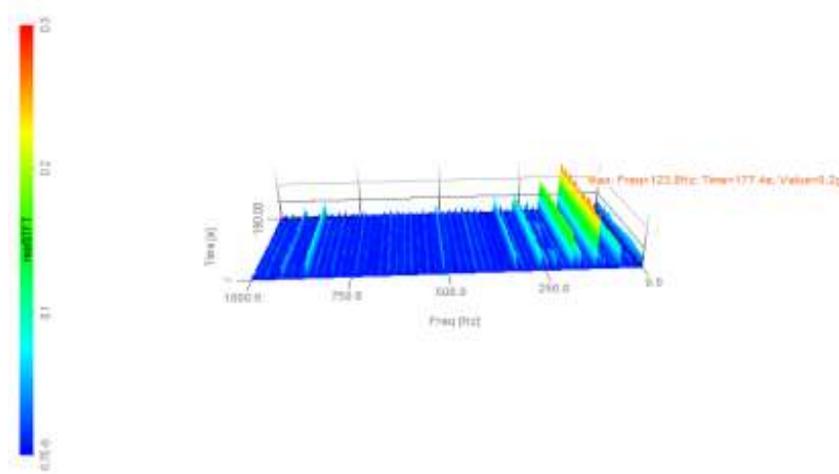


Fig. 11: STFT signal for half looseness

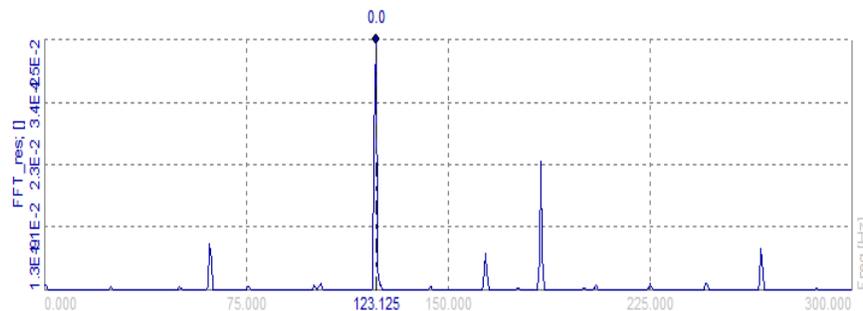


Fig. 12: Frequency amplitude domain response for half looseness

of sub harmonics and inter-harmonics for mechanical looseness.

In the journal bearings, mechanical looseness may only develop at high operating speed and temperature.

Next, the full looseness is provided on the bearing casing and the signals are recorded through the experimental set up. Figure 13 and 15 shows the time domain signal and autocorrelation through FFT signal,

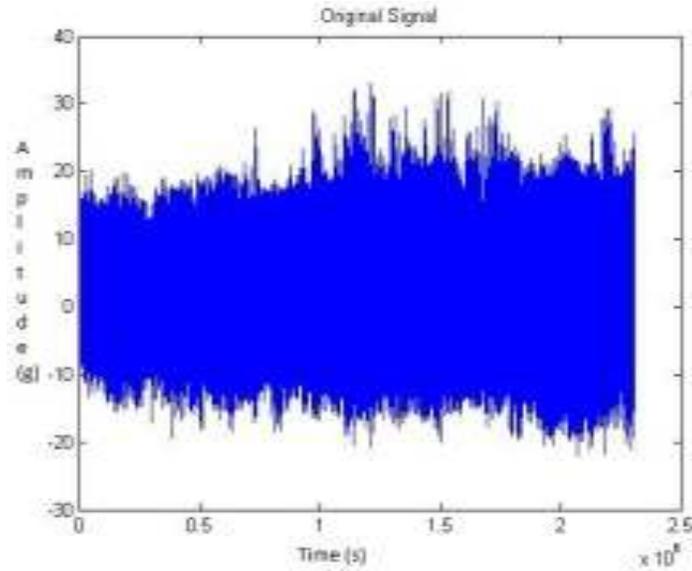


Fig. 13: Time domain signal for full looseness

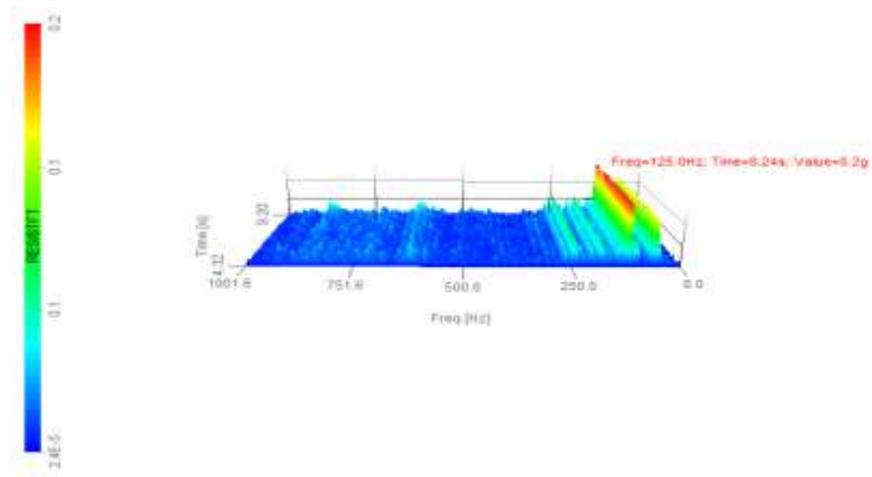


Fig. 14: STFT signal for full looseness

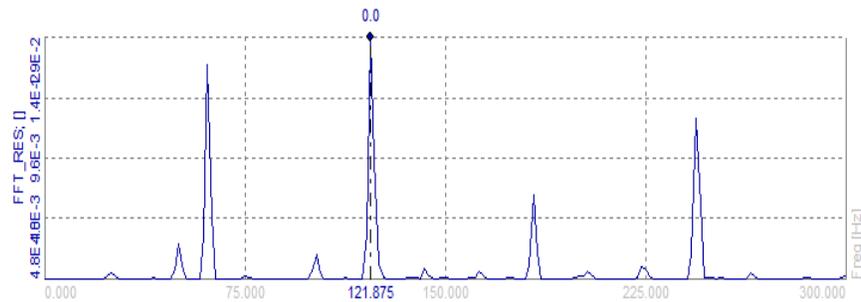


Fig. 15: Frequency amplitude domain response for full looseness

for full looseness of the bearing. After applying the STFT, the PSD plot shows the frequency of 125 Hz shown in Fig. 14. From this Power Spectrum Density (PSD) plot, it is observed that the bearing frequency for

full mechanical looseness is 121.875 Hz shown in Fig. 15. Thus, after applying autocorrelation, considerable peak amplitude is getting reduced due to electrical interference of motor. This increased

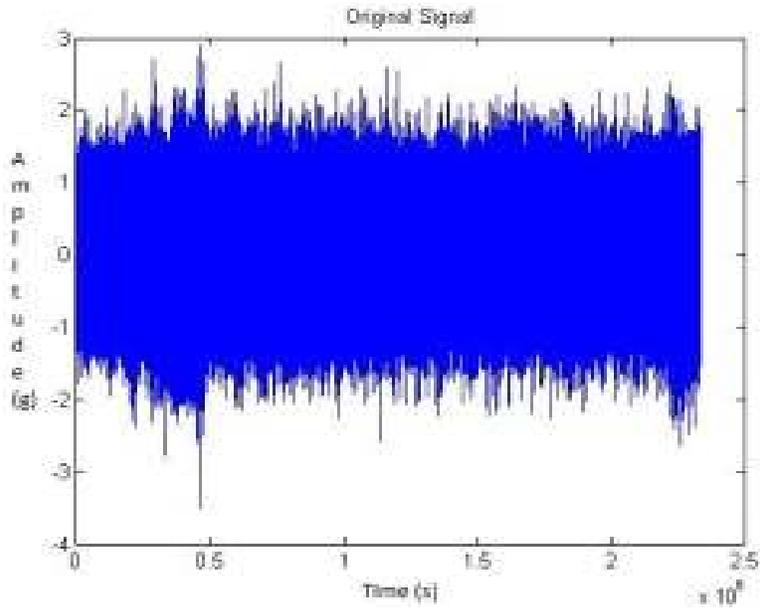


Fig. 16: Time domain signal for half lubrication

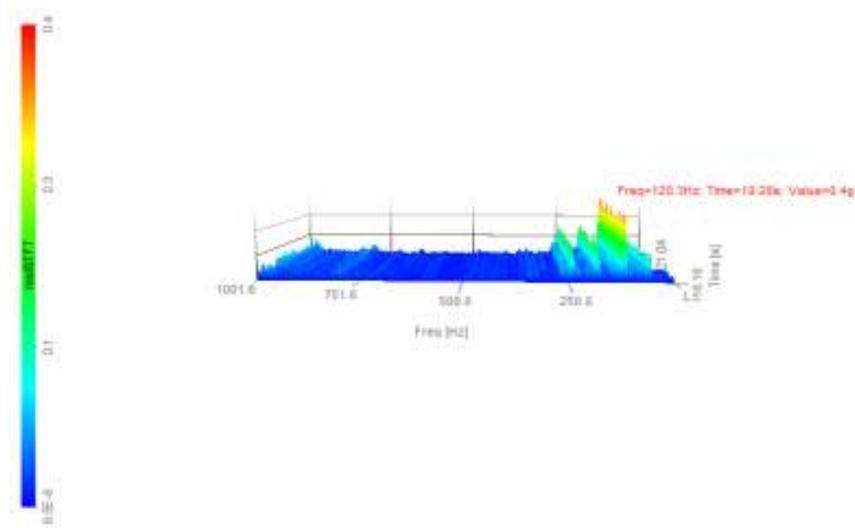


Fig. 17: STFT signal for half lubrication

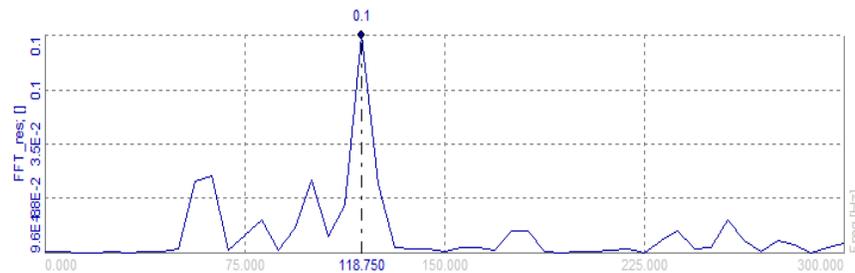


Fig. 18: Frequency amplitude domain response for half lubrication

frequency is due to combination of sub harmonics and inter-harmonics for mechanical looseness.

By comparing the time domain signals of Fig. 7, 10, 13, 16, 19, 22, 25 and 28, respectively shows a few

waveforms showing varying degrees of amplitude of the modulated waveform which proved to be a function of the intensity of the fault. These observations validate the results of indication of a fault contact within the journal. Thus, a vibration signature modulated at the rotational speed of the unit is illustrative of a continuous fault source. This continuous fault implies a sustained contact between the shaft and the bearing generating vibration levels above the operational background levels. Comparing the spectra in the Fig. 9, it can be found that the amplitudes of the friction frequency and its harmonics increase with the increasing misalignment conditions. This accelerates

the shaft causing it to rebound from the surface at 60 Hz vibrations in frequency spectrums.

Half lubrication behavior is due to side leakages of oil between shaft and the bearing. It can deteriorate and cause metal to metal contact in journal bearings. This will decrease eccentricity, increase oil temperature and reduce viscosity. In the journal bearings, oil instability may only develop at side leakages and poor mechanical maintenance. The half lubrication is poured through the oil hole provided on the bearing casing and the signals are recorded. Due to improper lubrication, the coefficient of friction gradually starts to increase between the shaft and the bearing. Figure 16 to 18

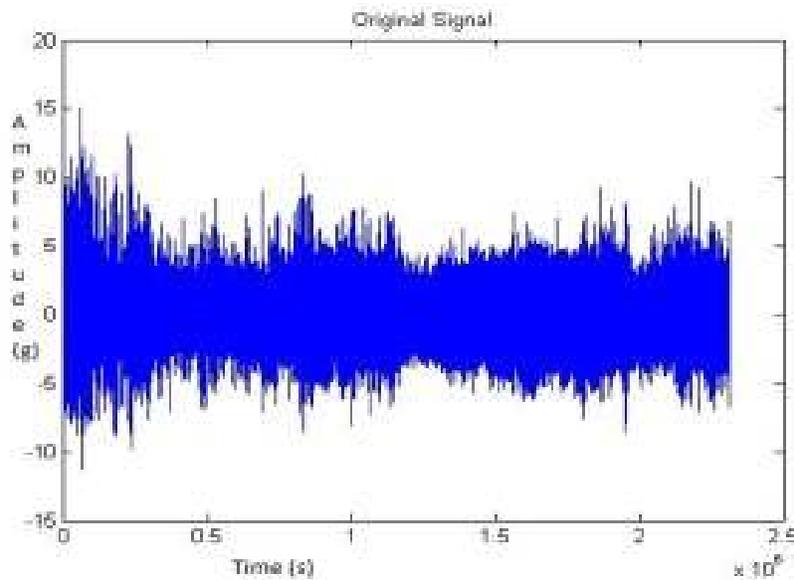


Fig. 19: Time domain signal for no lubrication

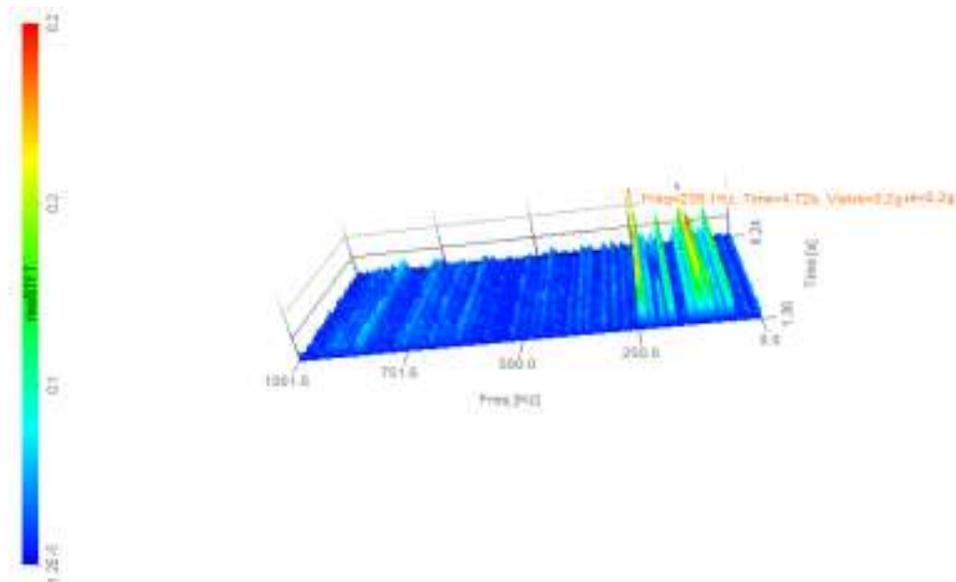


Fig. 20: STFT signal for no lubrication

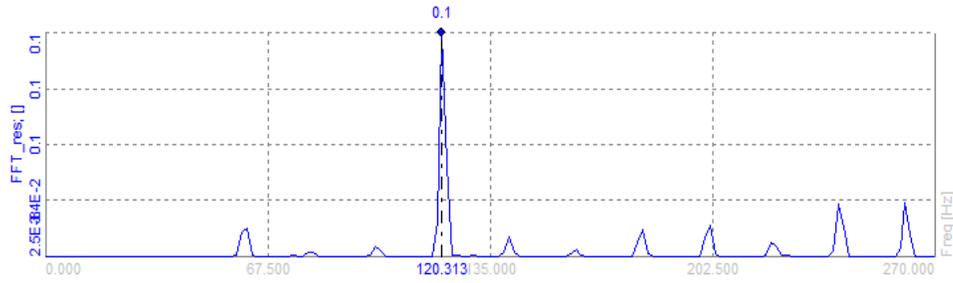


Fig. 21: Frequency amplitude domain response for no lubrication

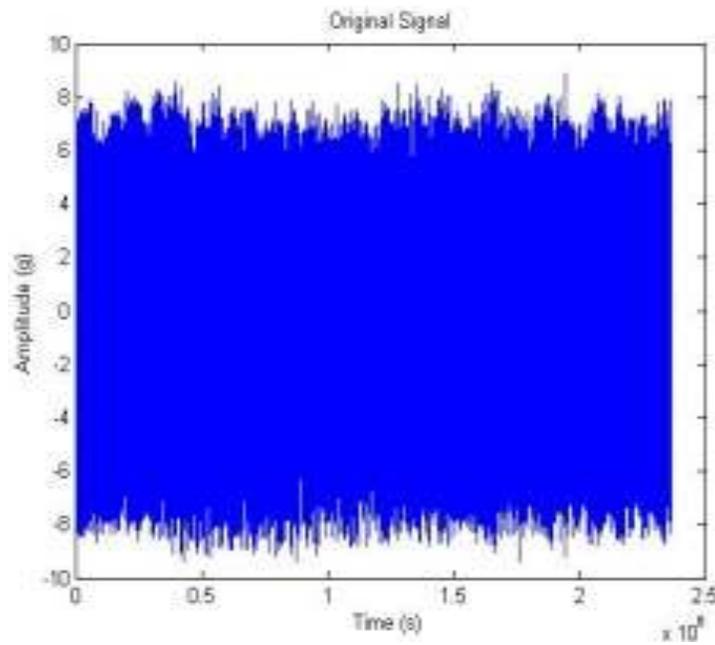


Fig. 22: Time domain signal for fault 1

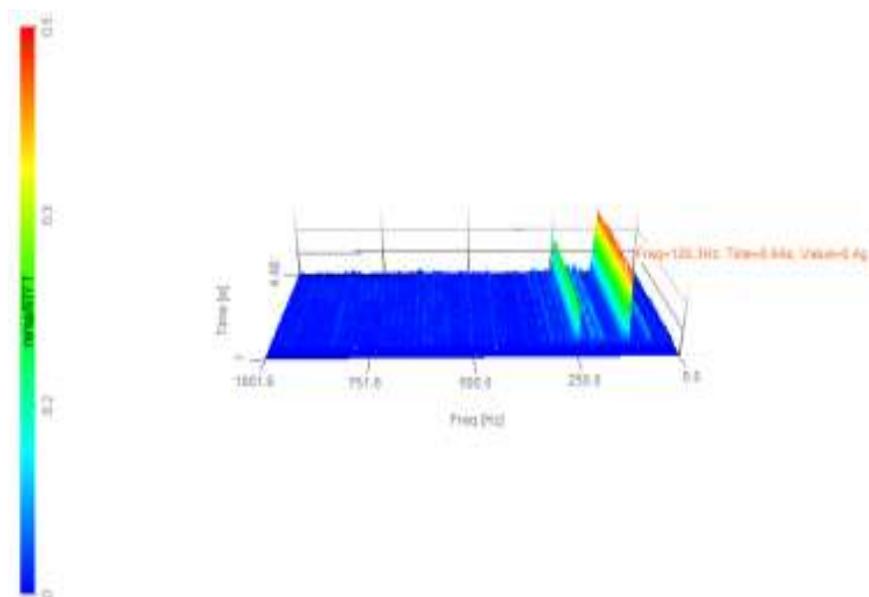


Fig. 23: STFT signal for fault 1

shows the time domain signal, STFT signal and autocorrelation through FFT signal, for half lubrication of the bearing. From this Power Spectrum Density (PSD) plot, it is observed that the bearing frequency for half lubrication is 118.75 Hz with the peak amplitude of 0.1 shown in Fig. 18 whereas through STFT, frequency is 120.5 Hz with the peak amplitude of 0.4. Hence, through autocorrelation considerable denoising is done, so that the peak amplitude is getting reduced.

No lubrication behavior is due to more side leakages of oil between shaft and the bearing. It can deteriorate and cause metal to metal contact in journal bearings. This will further decrease eccentricity, increase oil temperature and reduce viscosity. In the journal bearings, oil instability may only develop at side leakages and poor mechanical maintenance. The oil is completely drained and the signals are recorded. Due to absence of lubrication, the coefficient of friction increases between the shaft and the bearing. Figure 19 to 21 shows the time domain signal, STFT signal and autocorrelation through FFT signal, for half lubrication

of the bearing. From this Power Spectrum Density (PSD) plot, it is observed that the bearing frequency for no lubrication is 120.313 Hz with the peak amplitude of 0.1 shown in Fig. 21, whereas through STFT the frequency is 239.1 Hz with the peak amplitude of 0.2 shown in Fig. 20.

First the eccentric bore fault (Fault 1) was created shown in Fig. 4 and the readings were recorded with the shaft speed of 3600 rpm. Figure 22 to 24 shows the time domain signal, STFT signal and autocorrelation through FFT signal, for fault 1 of the bearing. From the Fig. 23 it is observed that the frequency is 120.3 Hz with the peak amplitude of 0.4. From this Power Spectrum Density (PSD) plot, it is observed that the bearing frequency for fault 1 is 120.313 Hz with the amplitude of 0.2 shown in Fig. 24 which is greater than the healthy bearing of 60 Hz. This shows that the fault is induced on the bearing. The result shows that autocorrelation technique is used to denoise signals. Hence through this technique considerable peak amplitudes are reduced.

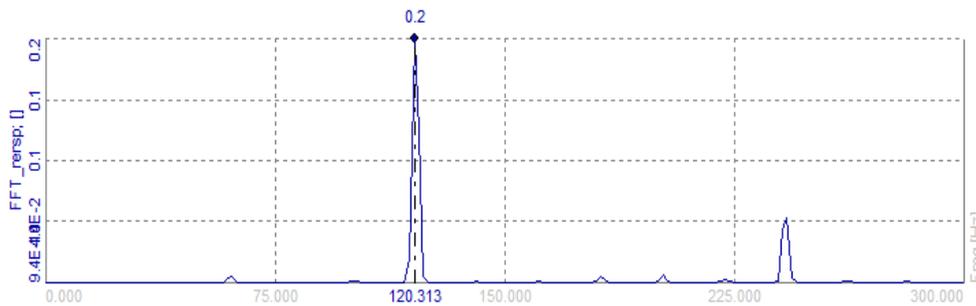


Fig. 24: Frequency amplitude domain response for fault 1

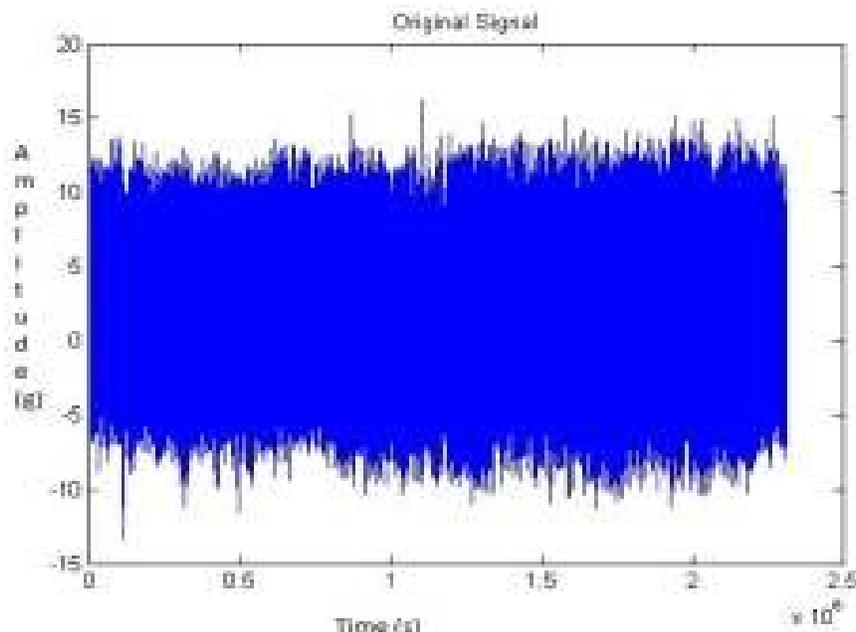


Fig. 25: Time domain signal for fault 2

Eccentric bore fault (fault 1) and fault with hole (fault 2) can cause strong harmonics of the shaft rotational speed in vibration analysis. In fault simulation once the shaft and bearing are in contact the stiffness of the test rig increases. This fault behavior tends to produce strong impulses between the journal and bearing which is used to excite system resonance and in addition create harmonics. From the Fig. 24 and 27, it was found that the dominant frequency of the vibration signals for Fault 1 and 2 conditions are registered to be approximately 120 and 122 Hz, respectively. The Non-synchronous peaks on the frequency spectrum are an indication of bearing fault. The increase in amplitude of these frequencies indicates the rate of fault. As the fault increases, the frequency of these peaks develops harmonics, indicative of impacting.

Next the eccentric bore fault with crack of 3 mm (Fault 2) shown in Fig. 4 was created and the readings were recorded with the shaft speed of 3600 rpm. The Fig. 25 to 27 shows the time domain signal, STFT signal and autocorrelation through FFT signal for fault 2 of the bearing. From this Power Spectrum Density (PSD) plot

shown in Fig. 27, it is observed that the bearing frequency for fault 2 is 121.875 Hz with the peak amplitude of 1.1, whereas through STFT shown in Fig. 26 is 121.9 Hz with the peak amplitudes of 1.3 which is greater than the healthy bearing of 60 Hz. The PSD plots shows more peaks and an increase in frequency are observed. From the result it shows that there is increase in the severity of the faults when compared to the fault 1. Also, considerable denoising is done through autocorrelation technique.

Finally, the material removal and bore fault with hole on the bearing (Fault 3) is shown in Fig. 5 was created and the readings were recorded with the shaft speed of 3600 rpm. Figure 28 to 30 shows the time domain signal, STFT signal, autocorrelation through FFT signal for fault 3 of the bearing. From this Power Spectrum Density (PSD) plot, through STFT shown in Fig. 29, it is observed that the bearing frequency for fault 3 is 260.91 Hz with the peak amplitude of 0.3. After applying the autocorrelation shown in Fig. 30, the PSD plot shows the frequency of 260.938 Hz with the peak amplitude of 0.01. Thus, after applying

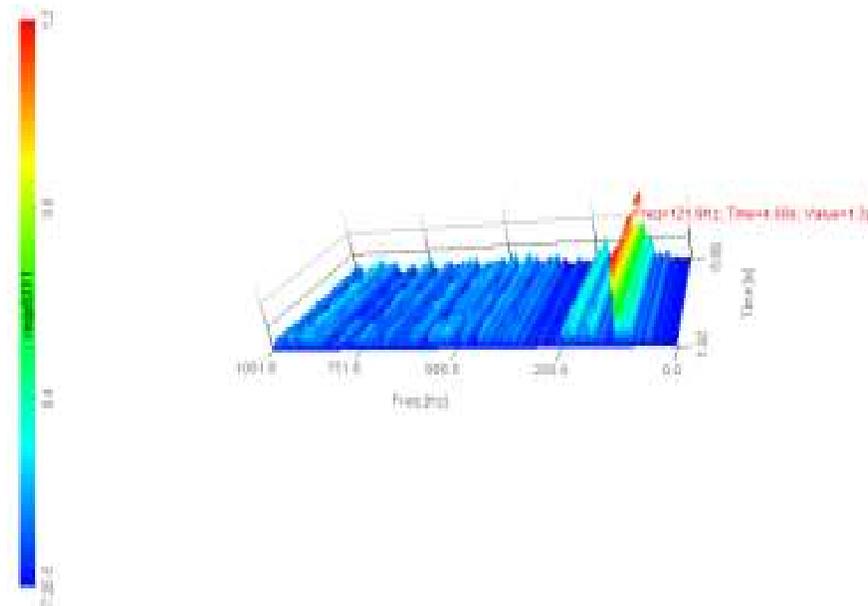


Fig. 26: STFT signal for fault 2

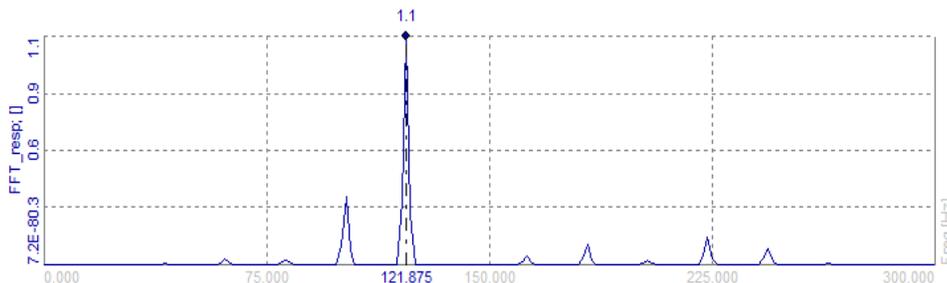


Fig. 27: Frequency amplitude domain response for fault 2

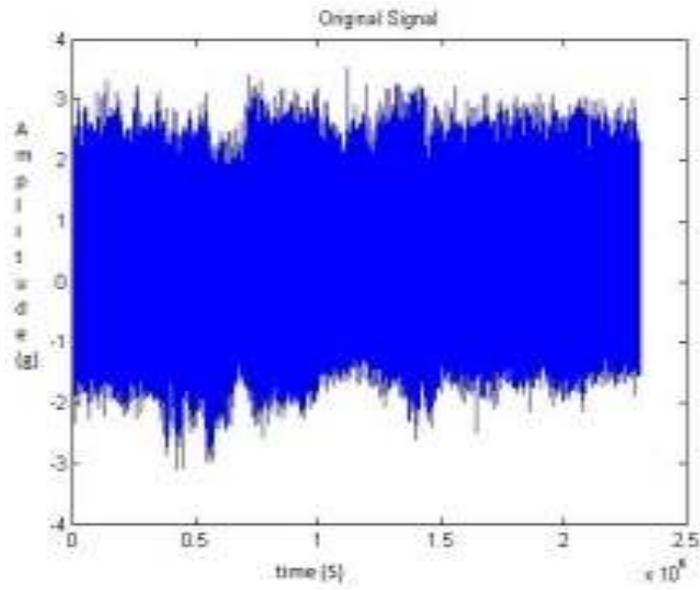


Fig. 28: Time domain signal for fault 3

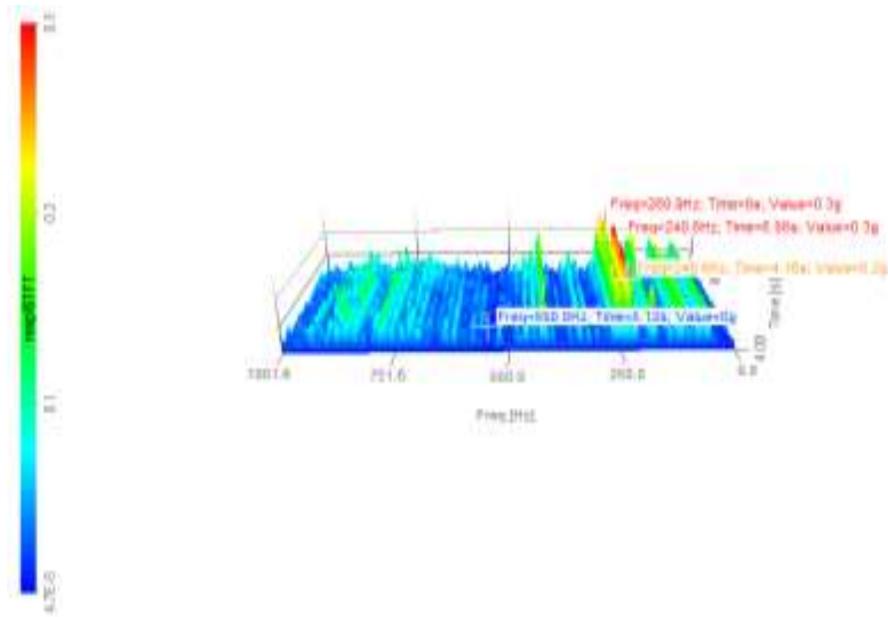


Fig. 29: STFT signal for fault 3

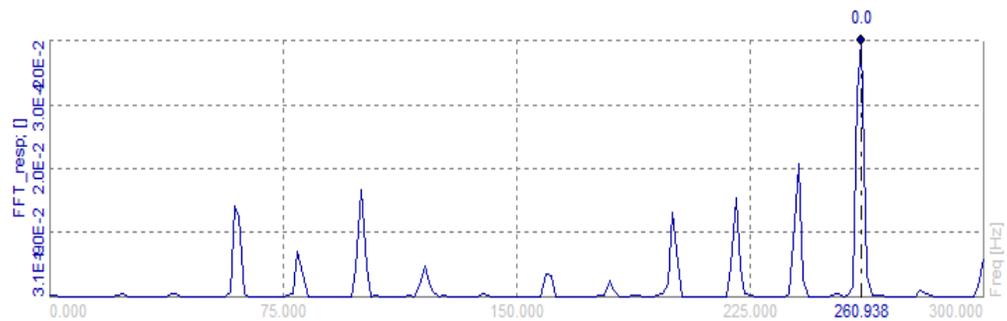


Fig. 30: Frequency amplitude domain response for fault 3

Table 1: Comparative analysis between autocorrelation and STFT

| Bearing conditions | Frequency (Hz) for autocorrelation | Amplitude (m/sec <sup>2</sup> ) for autocorrelation | Frequency (Hz) for STFT | Amplitude (m/sec <sup>2</sup> ) for STFT |
|--------------------|------------------------------------|---|-------------------------|--|
| Healthy            | 61.250                             | 0.20  | 63.10                   | 0.4                                      |
| Half looseness     | 123.125                            | 0.00  | 123.80                  | 0.2                                      |
| Full looseness     | 121.875                            | 0.00  | 125.00                  | 0.2                                      |
| Half oil           | 118.750                            | 0.10  | 120.50                  | 0.4                                      |
| No lubrication     | 120.313                            | 0.10  | 239.10                  | 0.2                                      |
| Fault 1            | 120.313                            | 0.20  | 120.30                  | 0.4                                      |
| Fault 2            | 121.875                            | 1.10  | 121.90                  | 1.3                                      |
| Fault 3            | 260.938                            | 0.01  | 260.91                  | 0.3                                      |

autocorrelation technique considerable peak amplitude is get reduced due to faulty running condition of the bearing and the electrical interference of motor. From the result it shows that there increase in the severity of the faults. When compared to the fault 2, more peaks and an increase in frequency are observed. Hence, the autocorrelation through FFT locates the exact fault on the bearing.

By increasing the fault (material removal fault 3) on the bearing, the frequency is increased, to the high frequency side of the spectrum. This shows that the system is hardening as a result of the stiffness produced by the contact of the shaft with bearing. From the Fig. 30 it is observed that the frequency increases as a result of product of the higher-energy content of the fault impacts. The enveloping analysis demonstrates that the dominant frequency of the vibration signals for fault 3 are registered to be approximately 260.938 Hz.

**STFT (Short Time Fourier Transform):** The different features can be extracted from time-frequency domain signals shown in Fig. 7, 10, 13, 16, 19, 22, 25 and 28, respectively. From the STFT results, it is observed that more peak amplitude is generated when the faults are increased. In the frequency domain section the spectrum of the vibration showed a broad bandwidth and is mostly concentrated between 100 and 300 Hz. However, as shown in the STFT spectrogram in greater detail, for fault the full waveforms are concentrated between 100 and 300 Hz with distinct features evident between the healthy and damaged bearing.

From the STFT Fig. 8, 11, 14, 17, 20, 23, 26 and 29, respectively shows the color coding is green for low amplitudes throughout and red for high amplitudes. The STFT spectrogram clearly indicates that the intermittent nature of the fault and in concordance with that the strong energy of the amplitudes is limited to frequencies between 100 to 300 Hz. This is attributed to metal wipe, or tearing and breaking of surface asperities during the fault simulation. This indicates a severe form of fault, as shown in Fig. 4 to 6. The time-frequency spectrogram showed again the wide spectral response and the differences between healthy and faulty condition of the journal bearing and location in time of the fault impacts.

It can be seen that as the experiment progresses the contact between shaft and bearing increases and higher frequency components appear. These components get their peak values near the middle of the time axis where the maximum contact is reached. They then begin to fade showing a trend for the higher frequencies to disappear as the contact decreases. These results confirm the wide spectral bandwidth of the fault phenomena. The time-frequency distribution of the signal obtained when the shaft was brought into contact with faulty bearing shows the wide frequency bandwidth in the varying degrees of amplitude response produced by the faulty behavior of the bearing.

It has been established in this study, that when the journal bearing is subject to a misaligned condition and faulty conditions; the metal wipe, tearing of the molecules and metal to metal contacts can occur between the bearing and the shaft producing high-energy vibration signatures.

Using DWESOFT software the signals is first de-noised in order to remove the unwanted noises, such as the 60 Hz AC supply electrical interference, which would have been picked up by the sensor during the experiment. Then these de-noised signals are used to study the frequency spectra using STFT and autocorrelation (Table 1). From this study, it was found that an increase in amplitude of peaks at Bearing Frequency BF with respect to various fault conditions.

## CONCLUSION

The test rig shows a complex structure, if there is any excitation, it will move around the structure. The most common sources of internal and external excitation identified in the experimental set up are; the coupling, the eccentricity of the shaft, the eccentricity of pulleys, the belt pitch variation causing angular acceleration and deceleration, the AC electronic motor, the ball bearing, etc. All these vibrations can be controlled by enveloping the autocorrelation through FFT with time domain signal where sources of internal noise and external noise is removed and it will give only vibration spectra of the journal bearing. Thus the condition monitoring provides more effective in early diagnosis of bearing faults. The results show that various fault vibration signals and the fault detection rate of the proposed method is more superior than that

of STFT. Hence this technique will produce considerable savings by reducing unwanted external and internal noises of machinery so a company can get the full benefit from condition monitoring.

Using High Frequency Enveloping algorithms outlined in this study, it is possible to properly construct an envelope of raw acceleration signals for effective use in early bearing fault detection of journal bearing. From the results, it is observed that as the fault increases the frequency and amplitude for the journal bearing is increased. This is a signal processing technology that must be utilized carefully in conjunction with traditional vibration analysis in order to obtain accurate results. This paper was intended to shed some light on the theory behind enveloping, provide a outline for the process and demonstrate two accepted ways for successfully accomplishing enveloping when analyzing vibration signals.

#### REFERENCES

- Bannister, R.H. and G.E. Findlay, 1999. Condition monitoring of low-speed rotating machinery using stress waves, part 1. *P. I. Mech. Eng. E-J. Pro.*, 213(3): 153-170.
- De Castro, H.F., K.L. Cavalca and N. Rainer, 2008. Whirl and whip instabilities in rotor-bearing system considering a nonlinear force model. *J. Sound Vib.*, 317(1-2): 273-293.
- Girdhar, P. and C. Scheffer, 2004. *Practical Machinery Vibration Analysis and Predictive Maintenance*. Elsevier, Burlington, MA, Amsterdam, Boston.
- Kuboyama, K., 1997. *Development of Low Speed Bearing Diagnosis Technique*. Fukuyama City, Japan.
- Raharjo, P., F. Al-Thobiani, F. Gu and B. Andrew, 2010. Early failure detection and diagnostics of high speed self aligning journal bearing. COMADEM, Nara, Japan.
- Rajendran, V., 2009. *Engineering Physics*. Tata Mcgraw-Hill Education Pvt. Ltd., ISBN-13: 978-0-07-107014-0, ISBN: 0-07-107014-1.
- SKF, 2000. *Vibration diagnostic guide*. SKF Condition Monitoring.
- Taylor, I. and D. Scatchard, 2008. More monitoring-less downtime. *World Pumps*, 507: 36-37.
- Ugural, A.C., 2003. *Mechanical design: An integrated approach*. McGraw-Hill, New York, pp: 864.
- Wang, B. and J. Lee, 1999. *Computer-aided Maintenance: Methodologies and Practices*. Kluwer Academic, Dordrecht, London.