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Vibration Signature of Gear Pump of Missing One and Two Teeth

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ABSTRACT

An unexpected failure of the gear pump may cause significant effect on its performance and structure, consequently there will be economic losses. For this reason, fault diagnosis in meshed gears has been the subject of serious research. Vibration analysis is one of the pumps on condition monitoring techniques that determines a suitable maintenance action. Generally, an increased vibration level can be considered as a warning form before failure or breakdown. By measuring, processing, and analyzing the frequencies of vibration signal, it is possible to determine both the type and severity of the defect, and hence predict the machine's failure. The vibration signal of a gear pump contains the signature of the defects in the gears, and early fault detection of the gear pump is possible by analyzing the vibration signal using different signal processing techniques (Frequency Spectrum, Envelop and Power Spectrum Density). This research experimentally, shows the external gear pump signature for normal and faulty gear at different rotational speeds (1080, 1200 and 1439 RPM). The considered faults herein are one missing tooth and two missing teeth in a single gear of the pump. The Faulty gear signature due to missing one tooth and two missing teeth and at different rotational speeds are identified. The amplitude of vibration increases by increasing the rotational speed. When the two missing teeth are facing each other results in decreasing the vibration amplitude than one missing teeth size.

Abbreviations

AC	Alternative Current
A/D	Analog to Digital
AE	Acoustic Emission
Amp.	Amplitude
DAS	Data Acquisition System
DC	Direct Current
FFT	Fast Fourier Transform
Hz	Hertz
Max.	Maximum
PC	Personal Computer
PSD	Power Spectrum Density
RMS	Root Mean Square
RPM	Revolution Per Minute
SA	Synchronous Average

1. Introduction

Gear pump is a fixed positive displacement machine where it uses the meshing of two gears to pressurize fluid by displacement. It is commonly used in fluid power and chemical applications for highly viscosity fluids. There are two main types; external gear pumps which use two external meshing gears and internal gear pumps which use an internal meshing.

Spalling, cracking, pitting, scuffing, wear, and

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excessive clearance are the famous problems that may cause operational performance degradation. The pump encounters many physical problems such as. These problems may cause rotor imbalance and consequently high vibration level, this resulting in a performance deterioration or structure damage or both. Then the vibration analysis (nondestructive test) is considered as an effective tool for the gear pump on condition monitoring.

Kaveh Mollazade [1] studied the fault diagnosis of pumps due to the vibration. The vibration signal was taken for the following conditions: journal bearing with inner face wear, normal pump, gear with tooth face wear, and journal bearing with inner face wear in addition to gear with tooth face wear of three working levels for pump speed (1000, 1500, and 2000 RPM). Then Power Spectral Density for vibration spectra was then calculated. Results proved that the peak value of PSD occurred at the frequency between 70-120 Hz at all conditions. Comparison for numerical data produced by using the calculation area under PSD.

Renata Klein [2] introduced different methods to separate natural frequencies of the structure and excitations resulting of the rotating components (signal pre-whitening). Several methods were proposed to achieve the above separation tasks. The present study compared between some of these methods. For pre-whitening, the study compared between liftering from the high frequencies and adaptive clutter separation. The method of adaptive clutter separation was suggested in this paper for the first time. For separating between the asynchronous and the cyclo-stationary signals the study compared between two methods: liftering in the frequency-domain and dephase. The methods were compared using both simulated signals and real data.

Zhipeng Feng, et. al. [3] presented Torsional vibration signals that were theoretically free of the amplitude modulation effect caused by time variant vibration transfer paths because of the rotation from planet carrier, sun gear and therefore their spectral structure were simpler than transverse vibration signals. Thus, it was potentially easy and effective to diagnose planetary gearbox faults via torsional vibration signal analysis. We gave explicit equations in order to model torsional vibration signals, considering both distributed gear faults (such as manufacturing or assembly errors) and local gear faults (such as pitting, crack or breakage for one tooth), and concluded the characteristics for both the traditional Fourier spectrum and the suggested demodulated spectra for amplitude envelope and instantaneous frequency. These derivations were not only effective to diagnose single gear fault for planetary gearboxes but can also be propagated to detect and determine multiple gear faults. We validated experimentally the signal models, in addition to the Fourier spectral and demodulation analysis methods.

Rishi Kumar Sharma, et. al. [4] applied Fast Fourier Transformation “FFT” spectrum for both healthy and faulty gears. It was observed that the effect of gear tooth breakage appears at frequency domain vibration signal as Sidebands of fundamental frequency. as well, it can be showed that the amplitudes from harmonics in case of gearbox with breakage fault were more than the amplitude of harmonics of healthy gearbox. The presence of tooth breakage fault in any gearbox given rise to peaks and generates sidebands. From FFT technique it can be predicted that the gear box had some fault but, the severity cannot be determined.

Shen Guoji, et. al. [5] focused the Condition monitoring and fault diagnosis. this paper also illustrated an important issue of gearbox maintenance and safety. The critical process concerned in such activities was to extract reliable features representative for the condition of gears or gearbox. A framework was presented of the application of spectrum to the analysis of gearbox vibration. The spectrum for a composite signal consisting of multiple periodic components had peaks at frequencies that correspond to closely related components that can be generated by any nonlinearity. As a result, phase verification was necessary to decrease false alarming of any spectrum-based method. A model based on modulated signals was adopted to reveal the spectrum characteristics of the vibration of a faulty gear, the corresponding amplitude and phase for the spectrum expression were deduced for a helicopter gearbox.

The author of a dissertation [6] developed a numerical procedure to simulate the dynamics of gear transmission system with the effect of gear tooth damage because of profile deviation. The numerical results were validated with data acquired from the experimental test rig. The study examined the vibration signal from both simulations in addition to experiment using joint time-frequency method through Continuous Wavelet Transform. The numerically simulated model can provide an accurate dynamic representation for the results from the experimental gear test rig without the complications because of the various operational environments

Dadon et. al. [7] introduced an approach for measuring the vibration pattern. This study based on empirical observations derived from actual measurements from vibration patterns mined. A simple spur gear transmission system was modelled and analyzed. In the first stage, a non-linear dynamic model for a spur gear transmission was developed. In the second stage, an analytical method was proposed to estimate the gear mesh stiffness as a function of mesh angle in the case of both healthy gear and faulty gears. The dynamic response of the system was computed by using the gear mesh stiffness derived by the analytical model. The model simulates the theoretical vibration response in healthy and faulty tooth gearwheels. The dynamic responses derived of the model were compared with experimental vibration data, to verify that model. The experimental phase included changing the load and speed settings, allowing the examination from the fault effect on the vibration signature under different test conditions.

S. N. Gawali, [8] used the vibration analysis technique to detect faults in gearbox system. In vibration analysis technique gear faults were detected based on Time Frequency analysis by the help of “MATLAB” software. Various types from gear defects can be artificially due to gear tooth such, missing tooth, one corner defect, two corner defect and inadequate lubrication. When comparing the data from healthy condition gear with faulty condition through the FFT analyzer. Analysis was carried out with graphs of high-frequency vibrations. “MATLAB” software is successfully used to validate the data. This paper had examined the gear fault detection by using feature extraction parameters and time frequency domain parameter.

Ajanalkar Sagar Shivputra, et. al. [9] prepared an experimental setup to measure the vibration signals from both healthy and faulty gears at different speeds. The three fault conditions were single tooth damaged pinion, double teeth damaged pinion and worn pinion. The vibration signal behaviors were analyzed by using time-waveform and FFT spectrum. Furthermore, some of the numerical techniques as RMS value and variance were used for comparing the behavior of faulty pinion with respect to healthy pinion.

Yu Guo, et. al. [10] presented combining the synchronous average and the resonant demodulation analysis. An envelope SA scheme had been proposed for the early detection of multi-axis gear faults. Three advantages of the proposed approach can be drawn as follows: Firstly, the envelope extraction by the resonant demodulation makes the localized fault related weak impulsive feature component simplified in the demodulation band, and the disturbances beyond the band attenuated heavily. Secondly, the envelope SA was employed to further eliminate the interferences in the demodulation band, and only the components with integer multiple periods of the reference rotating shaft were most reserved. Thirdly, the gear faults in a multi-axis gearbox can be exposed clearly by performing the envelope SA to different shafts in a gearbox one by one. The proposed approach is positively supported by simulations and experiments.

Chenyi Jin, [11] presented Vibration-based and Acoustic Emission tests were performed on a set of low speed gearboxes that were part of the strip metal forming line. The vibration signals collected of on site and offline tests were analyzed in both the time-frequency domains. Hit-based, time driven and frequency domain analysis were performed on the AE signals collected from offline tests.

Laxmikant S, et. al. [12] 2018, illustrated small incipient fault in gear and bearing causes multiple faults in gear-bearing system leading to catastrophic failure. The purpose of study was to explore more complex situation of compound gear and bearing fault. The compound faults such as a fault in the inner and outer race of bearing along with two teeth of gear having corner damage or three teeth of gear having corner damage were investigated using experimentation. To improve the effectiveness of diagnosis, vibration measurement was done at different speed and load condition. This paper proposed new compound fault features, extracted from continuous and discrete wavelet transform of vibration signal. The methodology consisted of proposing the features in time–frequency domain and comparison of its diagnostic potential with respect to the features extracted from time-frequency domain for compound fault identification using three different classifiers. The fault classification accuracy of these features was found to be better than the conventional time-frequency domain parameters.

Shengli Zhang, et. al. [13] presented a Gear fault diagnosis relied heavily on the scrutiny for vibration responses measured. The data is processed in the angle frequency domain to solve the issue for phase shifts between signal segments because of uncertainties caused by clearances, input disturbances, and sampling errors. The enhanced results were then analyzed out of feature extraction algorithms to recognize the most distinct features of fault classification and identification. Experimental and numerical simulation were performed to prove the effectiveness of angle frequency domain synchronous averaging on enabling feature extraction and classification.

Xingxing Jiang, et. al. [14] presented the determination of an index to balance the impulsiveness and cyclostationarity of an expected component was an interesting research topic in mechanical health monitoring. A no dominated solution set based on time and frequency infographics was then proposed to preferably synthesize the impulsiveness and cyclostationarity of detecting the local damage of rotating machines. Specifically, the proposed method was conducted through four steps: firstly, decompose the measured signal into certain levels. Secondly, construct the time and frequency infographics. Several case studies were conducted to validate the proposed method. Analysis and comparison results in indices in identifying fault features from rotating machines.

Zaigang Chen, et. al. [15] presented prompt development for modern railway transportation towards high speed and high load capacity, the high-power locomotive was urgently required. Once gear failures were presented, such gear tooth crack or breakage, it was likely to threaten the operation safety for the locomotive. So, deep insight into the fault features for the locomotive gear transmission was urgently necessary of prevention for the induced disastrous consequences. This paper concentrated on the fault feature extraction for a locomotive in presence of a crack in gear tooth root under the sophisticated dynamic excitations. Thereafter, angular synchronous average technique was proposed to enhance the fault vibration features, and the statistical indicators extracted in frequency-domain were developed to reveal the evolution law of the crack propagation scenarios. The analyzed results showed that the angular synchronous average technique is suitable to the tooth crack propagation in frequency-domain. The main aim of this study is to investigate the signature of two faults of the gears while running in a positive displacement pump. The faults are artificially applied to the gears. In each option, the following methodology has been used:

- Running under different rotational speeds (1080, 1200 and 1439 RPM)
- The data is acquired through a data acquisition through a tailored software in TESTPOINT programming language.
- The software processes the data and convert the time domain into frequency domain.
- The data from the frequency domain is analyzed through MATLAB and finally it is plotted using Grapher software.

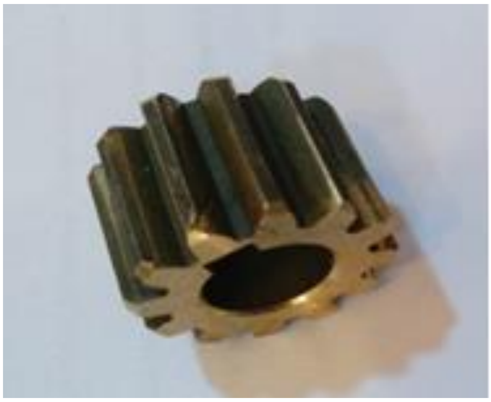

Fig. 1 Healthy Gear

Fig. 2 Missing one tooth

Fig. 3 Missing two teeth facing each other

2. Experimental Test Rig

Fig. 4 show the real test rig.

Fig. 4 The real test rig.

2.1 Test rig description

2.1.1 Pump Description

The studied healthy model is an external gear pump. It consists of one standard pair of spur gears. Each gear has a module of 2.5mm, number of teeth of 12 and width of about 19 mm. The transmission does not reduce the speed of the "Out" shaft containing the driven gear and the loading device. Both shafts are supported by two journal bearings. There is one meshing tooth. Two deteriorated of faulty models were examined; missing one and two teeth, Figs. (1) through (3) illustrate healthy, missing one and missing two teeth, respectively.

2.1.2 Hydraulic System

Fig. 5 shows the layout of the experimental hydraulic test rig which is used in this research. The system consists of 36x11.5x17 cm oil tank (1), oil filter (2), positive displacement gear pump (3), 0.5 HP electric motor running at 1500 RPM (4), pressure control valve (5), 4/3 directional control valve (6), double acting hydraulic cylinder (7). The experiment was carried out at a neutral position of a closed center directional control valve (no flow to/from the actuator cylinder).

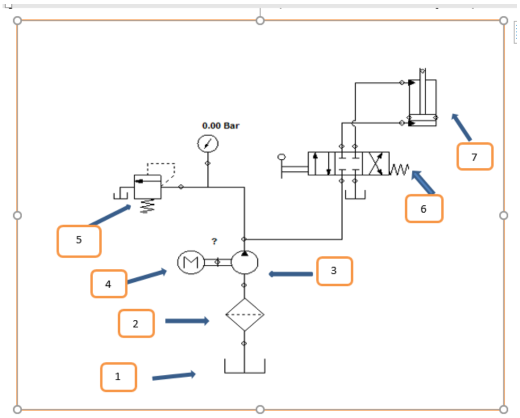

Fig. 5 Layout of the hydraulic circuit

2.1.3 **Sensors**

A sensor model "Dytran 3093B" measures the vibration in the three directions. Table (1) identifies the main specifications of accelerometer. The motor speed is changed by using variable frequency drive. The default speed of the motor is 1500 RPM.

Table 1 Vibration Sensor Specifications

Model	3093B
Accelerometer Type	Triaxial
Sensitivity	10 mV/m/s²
Frequency Response	0.6 to 5000 Hz
Frequency Response Gain	±10 dB
Temperature Range	-51 to +121 ºC

2.1.4 Data Acquisition System (DAS)

A typical DAS consists of the following components: [16]:

1. Signal Conditioning to convert and amplify AC signal from the vibration sensor to DC signal.
2. Analog to Digital Card from Data Translation model DT3001.
3. Terminal connector to connect wires.
4. PC computer
5. Software (Test point program)

Fig. 6 illustrates the component of data Acquisition System

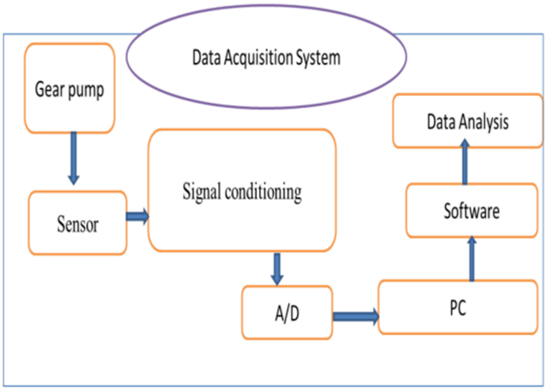

Fig. 6 Layout of the data acquisition system

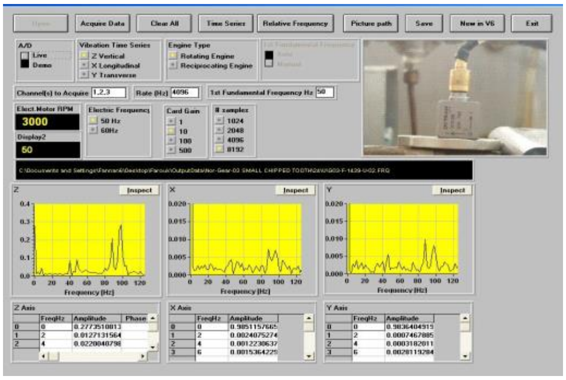

Fig. 7 application software to acquire and analyze the vibration signal

2.1.5 Test Procedure

Physical parameter to be measured is the vibration data on gear pump.

Vibration sensor measures the vibration in 3 axes.

Switch on the electric motor to run the system.

Adjust the running speed of the motor from the variable frequency drive (1080, 1200 and 1439 RPM) at different cases like (healthy, missing one and two teeth).

Press live then acquire data to see the spectrum of the vibration signals in the frequency domain and the use of computer technology in the measurement and control operations.

Press saves to record the data to analyze it.

Software program (Fig. 7) is used to acquire and record data through A/D card to PC.

The "Test point" program is activated to record the test data.

After saving the data, the MATLAB program is used to process and plot it.

2.1.6 Software

The user interface of the application software is shown in Fig. 7. The user selects the total number of data points and speed of acquisition. The data is acquired in the three directions (X,Y,Z). The time domain data is converted into the frequency domain and displayed. The user can save data and see the time domain.

2.1.7 Gear Pump Performance

When purchasing a pump, it is important to look at the pump performance curve. The performance curve represents the performance based on testing conducted by the manufacturer. Fig. 8 illustrates the gear pump performance. The pump performance curve indicates how a pump will perform regarding pressure and flow. An experiment show that an increasing of tooth notch affects the operating gear pump pressure which as the tooth notch increase as the operating pressure decrease.

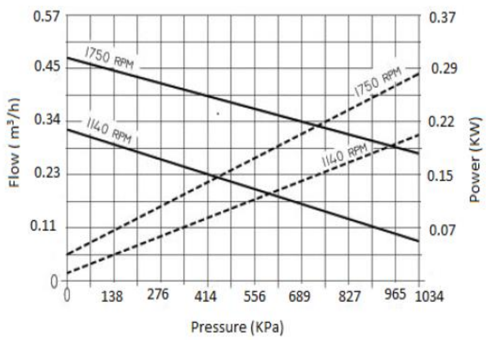

Fig. **8** Gear pump performance curve

3. Experimental results

The time domain vibration signals of both healthy gear and gear with missed one tooth and missed two teeth are converted into frequency domain using Fast Fourier Transform FFT. The vibration is measured at different pump speeds of 1080, 1200, and 1439 RPM by using the accelerometer. The motor speed is adjusted by using the variable frequency drive. Table (2) defines the equivalent frequency for each running speed. A computer program, based on MATLAB is programmed to process and analyse the data according to three different methods; Frequency Spectrum, Envelop, and Power Density [17].

Table 2 The Equivalent Modulated Frequency

Rotating speed, RPM	Frequency, HZ
1080	36.5
1200	40.5
1439	48.5

4. Results and Discussion

4.1 Frequency Spectrum

This methodology directly uses the frequency generated from the FFT function. All frequencies are divided by the 1st fundamental frequency according to each running speed. This division is a relative frequency.

Fig. 9 shows a comparison between the healthy gears of the pump when running at speeds 1080, 1200 and 1439 RPM. The curve contains 1900 data point for each speed. Each speed is divided by its 1st frequency (18, 20, and 24 respectively). Increasing the speed results in increasing the amplitude. The low frequency zone is dominant with higher amplitudes. Increasing the speed up to the maximum spend of the motor (1500 RPM) results in increased the amplitudes in another zone where the relative frequency >≈ 16. The meshing frequencies appears at 2,3,4 and so on. i.e. multipliers of the 1st fundamental frequency.

Fig. 10 shows a comparison between the signature of the gears of the pump when there is a missing one tooth at speeds 1080, 1200 and 1439 RPM.

The amplitudes of the higher zone are the predominant frequencies. The amplitudes in the higher zone are greater than the 1st fundamental frequencies. The shape of the curve looks like a normal distribution curve and the top of the bell shape increases as the speed increases.

Fig. 11 shows a comparison between the signature of the gears of the pump when there are two missing teeth at speeds 1080, 1200 and 1439 RPM.
It was supposed to have higher amplitudes than the case of missing one tooth but, the curve shows lower amplitudes than the expected. This can be justified because of the dynamic balance as the missed teeth are facing each other as shown in Fig. 3.
Due to the dynamic balance, the frequency in the lower zone is the dominant frequencies than in the higher zone.

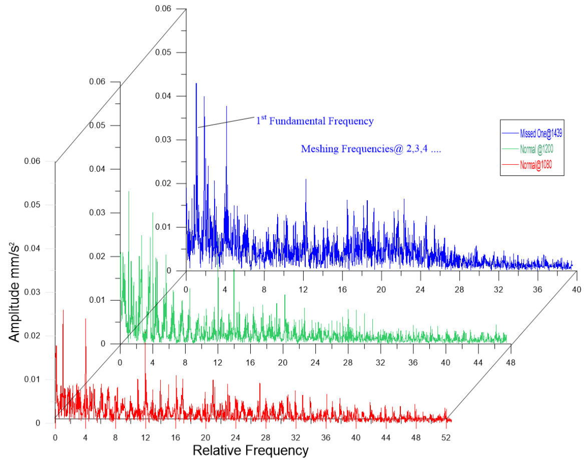

Fig. 9 comparison between healthy gears signature at different RPM

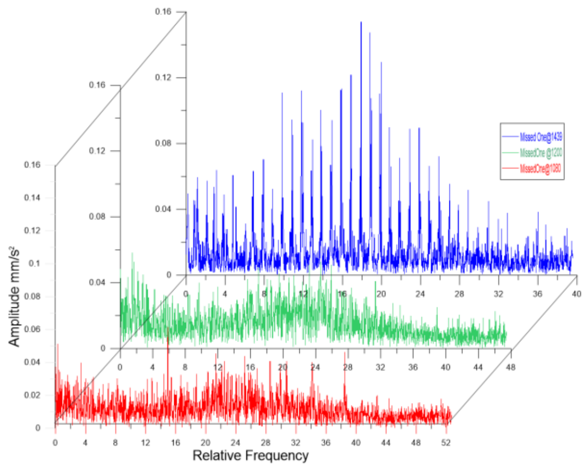

Fig. 10 Comparision between missed one tooth signature at different RPM

Fig. 12 shows a comparison between healthy, missing one tooth and missing two teeth at 1080 RPM. The amplitudes in the lower zone "for the missed two teeth" are higher than that of the healthy case but smaller than that of the missed one tooth. When testing the one missed tooth, it is found that the maximum amplitudes moves towards the higher zones (i.e. relative frequencies >16).

Figs. 13 shows a comparison between Healthy, Missing One Tooth and Missing Two Teeth at 1200 RPM. As the speed increased to 1200, then the amplitudes rise for the three cases; healthy, missed one tooth and missed two teeth in the low zone relative frequency (< 16). On the other hand, it rises

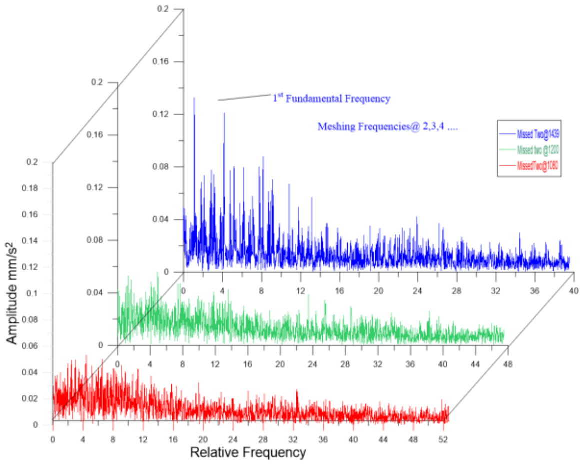

Fig. 11 Comparision between missed two teeth signature at different RPM

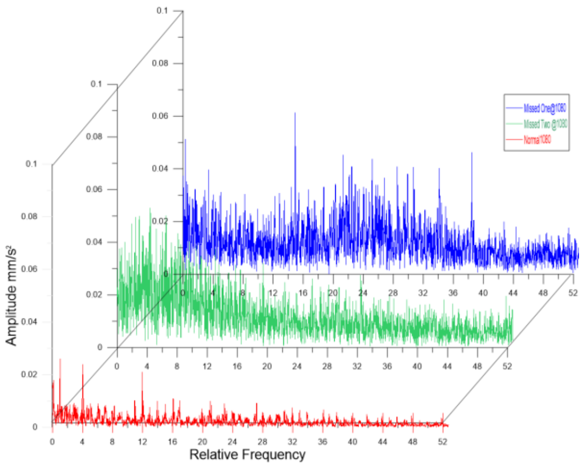

Fig. 12 Comparision between healthy, missing one tooth and missing two teeth @1080 RPM

on high frequency zone in the case of missing one tooth. The bell shape is clearly confirmed on this figure. Fig. 14 shows the same comparison between healthy, missing one tooth and missing two teeth at 1439 RPM. As the speed increased to 1439, then the amplitudes rise for the three cases; healthy, missed one tooth and missed two teeth in the low zone relative frequency (< 16). On the other hand, it rises on high frequency zone in the case of missing one tooth. The bell shape is clearly confirmed on this figure

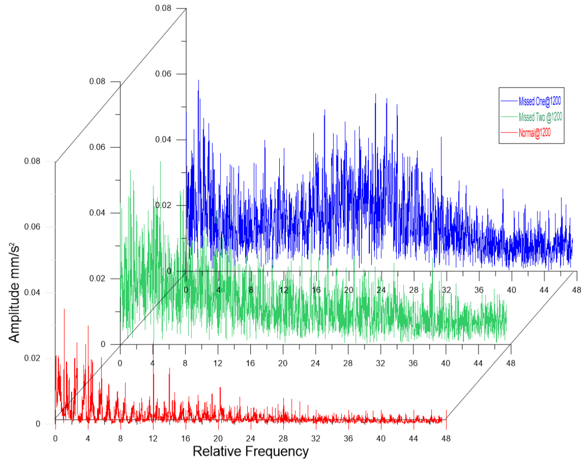

Fig. 13 Comparison between healthy, missing one tooth and missing two teeth @ 1200 RPM

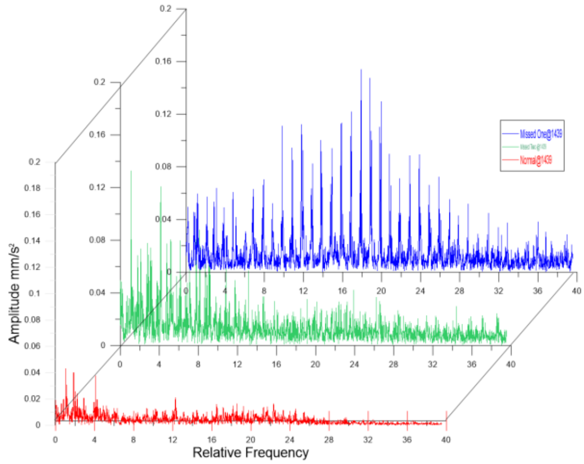

Fig. 14 Comparison between healthy, missing one tooth and missing two teeth at 1439 RPM

Due to the nature of this problem (gear meshing, interference of frequencies and side band frequencies), it will be so difficult to exactly diagnosis the problem, but we can identify it. Figs. 10 through 14 show the trend of the phenomenon without quantifying it. Fig. 15 shows the signature of the missed one tooth compared with the normal case @1080. There is a big difference between the two cases. Missing one tooth results in increasing the amplitude of the frequency. The amplitude reaches 0.061@14.95 compared with the healthy case which reaches 0.026@1 and 0.021@12. So, we can say, it is about 3 times the normal amplitude.

Fig. 16 shows the signature of the missed one tooth compared with the normal case @1439. There is a big difference between the two cases. Missing one tooth results in increasing the amplitude of the frequency. The amplitude reaches 0.155@18 compared with the healthy case which reaches 0.043@1.04. So, we can say, it is about 3.6 times the normal amplitude. Table 3 summaries the data of the processed data using this methodology.

Table 3 Comparison Between Healthy and Faulty Gears Different RPM

Fault Type	RPM	Max. Amp. mm/s^2	@Relative Freq.
Healthy Gear Fig. 1	1080	0.026	01.05
Missing one Tooth		0.061	14.95
Missing Two Teeth		0.053	04.50
Healthy Gear	1200	0.035	01.05
Missing One Tooth		0.075	17.00
Missing Two Teeth		0.580	04.52
Healthy Gear	1439	0.040	04.05
Missing One Tooth		0.155	18.00
Missing Two Teeth		0.122	04.55

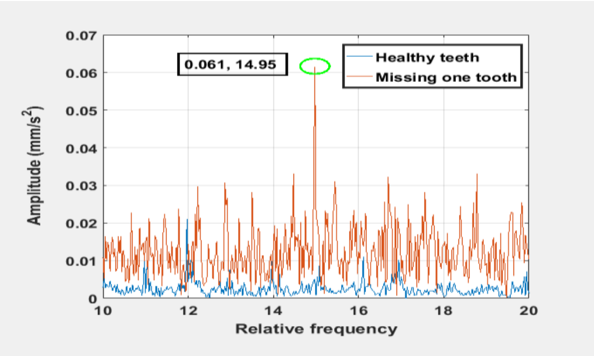

Fig. 15 Zoom on the maximum amplitude @1080 RPM of missing one tooth

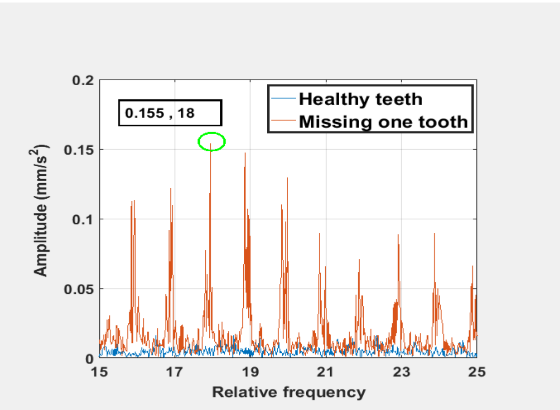

Fig. 16 Zoom on the maximum amplitude @1439 RPM of missing one tooth

4.2 Envelope Method

This method computes and plot the upper and lower RMS envelopes of the signal of the FFT Spectrum for vibration level with a length of 100 samples. From “MATLAB SOFTWARE”. Figure 17 compares the envelope for different speeds (1080, 1200 and 1439 RPM) for a healthy gear. The vertical axis is drawn with the same length to keep the proportion of the visuality. Figure 18 compares the envelope for different speeds (1080, 1200 and 1439 RPM) for a missed one tooth gear. The figure shows the envelope as a bell shape. It has it maximum amplitudes at relative frequency (High frequency zone) 16-20. Figure 19 compares the envelope for different speeds (1080, 1200 and 1439 RPM) for two missed teeth gear. The figure shows the maximum amplitudes resides in the low frequency zone. Again, this method indicates the trend without actual quantification for the problem. We can say that if you get high amplitude than healthy in the low zone frequency, then there is a problem in the gear but there is still dynamic balance. If you get a bell shape in the high frequency zone, then there is unbalance in the rotating gears.

Table (4) demonstrate that; the amplitude of vibration at running speed 1439 RPM is the highest amplitude comparison with speeds 1080 and 1200 RPM and the missing one tooth is the highest amplitude of vibration comparison with healthy teeth, and missing two teeth.

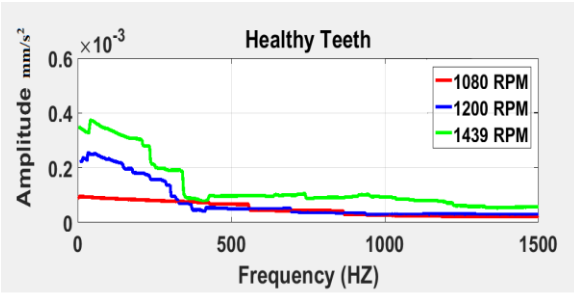

Fig. 17 Envelope representation at speeds 1080, 1200 & 1439 for healthy gear

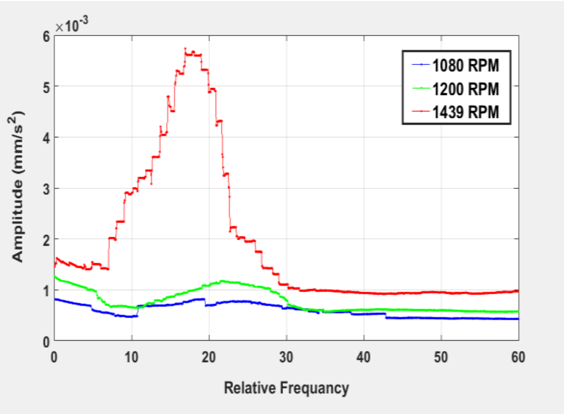

Fig. 18 Envelope representation at speeds 1080, 1200 and 1439 for missing one tooth

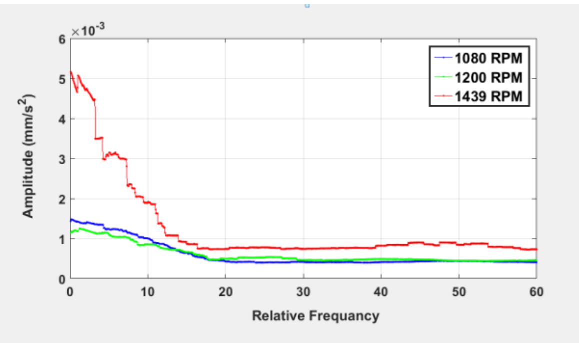

Fig. 19 Envelope epresentation at speeds 1080, 1200 and 1439 for missing two teeth

Table 4 Summary of comparison Between Healthy and Faulty Gears at Different RPM (Envelop)

Fault Type	Fig.	RPM 1080	RPM 1200	RPM 1439
		Amplitude mm/s^2		
Healthy Gear Fig. 1	16	0.19x10^{-3}	0.25x10^{-3}	0.38x10^{-3}
Missing one Tooth Fig. 2	17	1.20x10^{-3}	1.60x10^{-3}	5.75x10^{-3}
Missing Two Teeth Fig. 3	18	1.10x10^{-3}	1.20x10^{-3}	5.20x10^{-3}

4.3 Power Spectrum

$$PS(dB) = \frac{1}{2\pi}\int_{-\infty}^{+\infty} |F(\omega)|^2 d\omega \qquad (1)$$

- Power Spectrum is the integration of square of Fast Fourier Transform. Power of the spectrum Density (PSD) is measured in decibel (dB) and F(ω) is the Fourier Transform of vibration signal. Area under the FFT Spectrum curve gives an indication to the kinetic energy losses due to vibration, resulting from both healthy and faulty models. Fig. 20 shows these areas as function of rotational speed at different model faults. As the rotational speed increases, the energy index increases. If the index of the power spectrum density is in the range of 0 to 50, then the gears are healthy.
- If the index of the power spectrum density is in the range of 50 to 250, then the gears have a problem, but they are dynamically balanced.
- If the index of the power spectrum density is greater than 250, then the gears are dynamically unbalanced due to wear in the gears or missed one tooth.

Although there is a dynamic balance when running under two missed teeth, but the power spectrum density identifies the problem as the index increased to 10-23 times the healthy gear (see Table 6). In case of missing one tooth, the percentage increased from 15 to 31 times the healthy gear. As shown on Table 5, there is an interference area, e.g 23 times the healthy gear. You must look for the frequency zone. If you have a bell shape in the high frequency zone, then this increase will be due to dynamic unbalance.

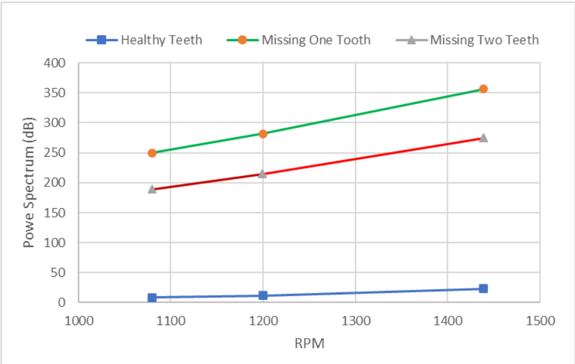

Fig. 20 Power spectrum comparison between healthy, missing one and missing two teeth @ different speeds

Table 5 illustrates that the energy index at all faults and all speeds and each point mean curve of FFT spectrum and these points take from "MATLAB SOFTWARE".

Table 5 Shows a comparison Between Healthy and Faulty Gears Different RPM (Power Spectrum)

RPM	Healthy Teeth	Missing One Tooth	Missing Two Teeth
1080	8.2	256.1	189.3
1200	11.8	281.5	208.1
1439	23.5	356.4	245.1

Table 6 Relative Increase of Amplitude at Different RPM

RPM	Healthy Gear	Missing Two Teeth		Missing One Tooth	
	dB	Absolute dB	Relative increase in Amp.	Absolute dB	Relative increase in Amp.
1080	8.2	189.3	23.1	256.1	31.2
1200	11.8	208.1	17.6	281.5	23.9
1439	23.5	245.1	10.4	356.4	15.2

5. Conclusions

An Experimental test rig that investigates the vibration response of gear pump when the it has missing one tooth and missing two teeth. The results are compared with healthy gear pump. Three techniques were used to explain the effect of these faults on the vibration level at different pump rotational speeds (1080, 1200 and 1439 RPM). These techniques are: Frequency Spectrum, Envelope and Power Spectrum.

- As the speed increases, the amplitude of vibration levels increases (for all conditions healthy and faulty).
- The vibration level of faulty models is higher than that of healthy model for all speeds.
- The amplitude of vibration of missing two teeth is slightly lower than that of missing one tooth, because of the nearly dynamic balance. The relative increase in amplitude of missing two teeth and one missing tooth is 23.1 and 31.2 @ 1080 RPM, 17.6 and 23.9@ 1200 RPM and 10.4 and 15.2@1439 RPM sequentially.
- The maximum amplitude occurs at the case of missing one tooth with relative amplitude 31.2, 17.6 and 15.2)
- The highest amplitude (missing one tooth) occurs at 1439 RPM with relative amplitude 15.2, where its equivalent absolute value is 356.4.
- The Power Spectrum is the best method for comparison between healthy and faulty condition
- The highest Power Spectrum occurs at the case of missing one tooth and at 1439 RPM.
- If the index of the power spectrum density is in the range of 50, then the gears are healthy.
- If the index of the power spectrum density is in the range of 50 to 250, then the gears have a problem, but they are dynamically balanced.
- If the index of the power spectrum density is greater than 250, then the gears are dynamically unbalanced due to wear in the gears or missed one tooth.

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