Experimental Study on Spur Gear Vibration Characteristics under Various Loading Conditions

by

Mohammad Izzatul Akmal Bin Abdillah

Dissertation submitted in partial fulfillment of

the requirements for the

Bachelor of Engineering (Hons)

Mechanical Engineering

MAY 2012

Universiti Teknologi PETRONAS

Bandar Seri Iskandar

31750 Tronoh

Perak Darul Ridzuan

CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
Mechanical Engineering Programme
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BACHELOR OF ENGINEERING (Hons)
(MECHANICAL ENGINEERING)

Approved by,

____

(Ir. Idris bin Ibrahim)

UNIVERSITI TEKNOLOGI PETRONAS
TRONOH, PERAK
May 2012

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

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MOHAMMAD IZZATUL AKMAL BIN ABDILLAH

ABSTRACT

Gear faults start as miniscule at the early stage, which may propagate, leading to excessive vibration of the system, thus worsening the system's overall performance. This will result in unplanned downtime and expenses as to mend the damages. Thus, it is vital to mitigate the effects of gear damages by early detection of the phenomena. The project studies the vibration characteristics of spur gears under faulty conditions. The outcome of this research is to establish a correlation between gear faults and vibration signal and hence, enable to assist in alerting the imminent failures of the gears. An experimental approach has been adopted in this research. A pair of spur gear is designed and fabricated from an aluminum plate. The loading condition refers to a number of different experimental setups employed, including misalignment and bending moment. The experiment is run continuously until noticeable fault developed. The analysis is based on the frequency spectrum, RMS and Peak to Peak vibration signatures. Results show different vibration signatures and gear defects exhibit different vibration characteristics.

ACKNOWLEDGEMENT

First and foremost, I would like to express my most heartfelt gratitude to Allah S.W.T the Most Benevolent for guidance and blessings throughout my entire two semesters of Final Year Project period.

To Ir Idris bin Ibrahim, I am very thankful for his endless support in ensuring the success of this project and guidance in going through the detours I have stumbled upon along the way.

My humblest gratitude is also extended to my Mr Rizal Lias, for sharing his knowledge on the mechanics of gear faults, basics of gear design and also the standard operating procedure for the gear test rig.

I am also highly appreciative of National Instrument Corporation, for providing me with the Labview SignalExpress 2011 software, which is an essential part in the analysis of my project results.

To all of the technicians of Universiti Teknologi PETRONAS, I thank them for unwaveringly assisting me in the fabrication of the gear and protective cover, usage of the laboratory and also the gear test rig.

Finally, I would like to express my utmost appreciation to those who have directly or indirectly supported me since the beginning of project. It has been a beneficial and enjoyable experience for me and would surely be perpetuated in my memory always.

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CHAPTER 1

INTRODUCTION

1.1 Project Background

A gear is a type of rotating machinery that provides torque transmission by meshing with another gear. Smaller gears will have faster rotation but smaller torque while bigger gears will have otherwise. Two main functions of a gear include changing the rate of rotation and changing the direction of rotation, which are done by spur gears (Fig 1.1) and bevel gears(Fig 1.2), respectively.

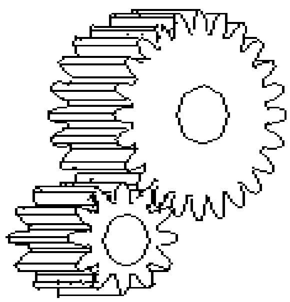


Fig. 1.1 A pair of spur gears (*Retrieved from* *http://www.cs.cmu.edu*)

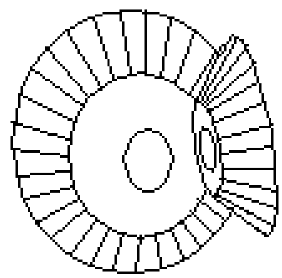


Fig. 1.2 A pair of bevel gears (*Retrieved from* *http://www.cs.cmu.edu*)

There are several types of gear fault that occur in the machine environment, which include pitting and bending fatigue breakage (cracking). Pitting, as shown in Fig 1.3, is characterized by a small cavity forming on the tooth of a gear. This type of fault usually occurs in localized parts of a gear tooth and is caused by over stressing. Pitting is sometimes known as corrective pitting due to the load redistribution done by the fault itself by removal of high contact spots.

Bending fatigue failure, or cracking, as shown in Fig 1.4, occurs in the root section of a gear tooth. This fault will keep on propagate until breakage of the tooth. Cracking is caused by excessive tooth loads that causes repeated root stresses which will ultimately exceed the endurance limit.

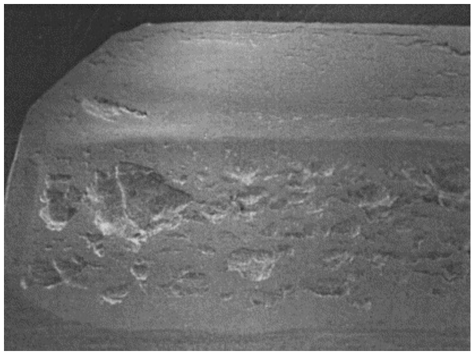


Fig. 1.3 Pitting formation on a gear tooth (*obtained from S. Glodez et al. (2003)*)

Fig. 1.4 Bending fatigue failure or cracking (*Retrieved from www.machinerylubrication.com*)

In a vast area of gear applications, these faults are ubiquitous. These faults may start as miniscule in the earlier stages, but after long hours of operation of the gears, the faults will start to propagate, consequently leading to excessive vibration of the system, thus worsening the system's overall performance. For example, excessive vibration in a gear train will introduce wear on flanks, and these flanks will result in unanticipated additional load (G. D. Mehta, 2011). Expenses will be incurred as to replace the broken gears, and equipment downtime or shutdown is also a potential setback to the system. As evident here, it is vital to mitigate the effects of gear damages by early detection of the phenomena. Various techniques are available in the condition monitoring of a gear system, which include acoustic emission, spectrometric oil samples and also the standard vibration analysis (C. K. Tan, 2007).

In investigating the correlation between the mechanics of gear faults and the accompanying vibration characteristics, a gear test rig is usually utilized. A gear test rig is an apparatus that is used to conduct tests on a set of gears in order to determine vibration characteristics of the gear set at different period of time or frequencies. From these data, the mode of fault propagation on the gears can be analyzed and appropriate actions can be taken in the future in mitigating the damage.

The important components in a gear test rig usually consist of an electric motor as a means of providing power, a shaft located at the motor side and another shaft located at the driven side, a load applier to provide varying load conditions upon the gears, and accelerometers to measure gear vibrations. The gear vibrations' signals are sent to a personal computer via data acquisition hardware connects the gear test rig to the personal computer.

1.2 Problem Statement

Gear faults will result in unplanned downtime and expenses as to mend the damages. Thus, it is vital to mitigate the effects of gear damages by early detection of the phenomena. This research strives to study the vibration characteristics of gears under faulty conditions. By establishing a correlation between gear faults and vibration signal, the outcome of this research would be able to assist in alerting the imminent failures of the gears.

1.3 Objectives

The main objective of this research is to study gear vibration characteristics under various loading conditions and establish a correlation between vibration signals and gear faults. A number of key activities will be done in realizing the objective, which includes:

1. To develop a manual of the experimental research.
2. To design and fabricate a new pair of gears.
3. To study the mechanics of gear faults.
4. To perform an experiment without loading condition (as to acquire the baseline data).
5. To perform an experiment with various loading conditions, which are misalignment, bending moment and continuous run until noticeable deterioration.
6. To establish a correlation between fault and vibration data.

1.4 Scope Of Study

Throughout the research the scope of study will only be limited to spur gears. Thus, the succeeding analysis and result will be based on the characteristics of a spur gear only. During the analysis of the result, the outcomes will be presented in the forms of residual signal based parameters which are Root Mean Square (RMS) and Peak to Peak, and frequency spectrum.

CHAPTER 2
LITERATURE REVIEW

2.1 Fracture Mechanics

A number of studies and research have been done in view of the topic discussed. S. Glodez et al observed that the occurrence of pitting at flanks is characterized by the formation of small pits on the contact region. They stated that towards pitting, there would be an initial formation of surface-breaking cracks that forms under repeated contact leading. These cracks will consequently become large causing unstable growth, which will result in a portion of the material layer breaking away (S. Glodez, 2000).

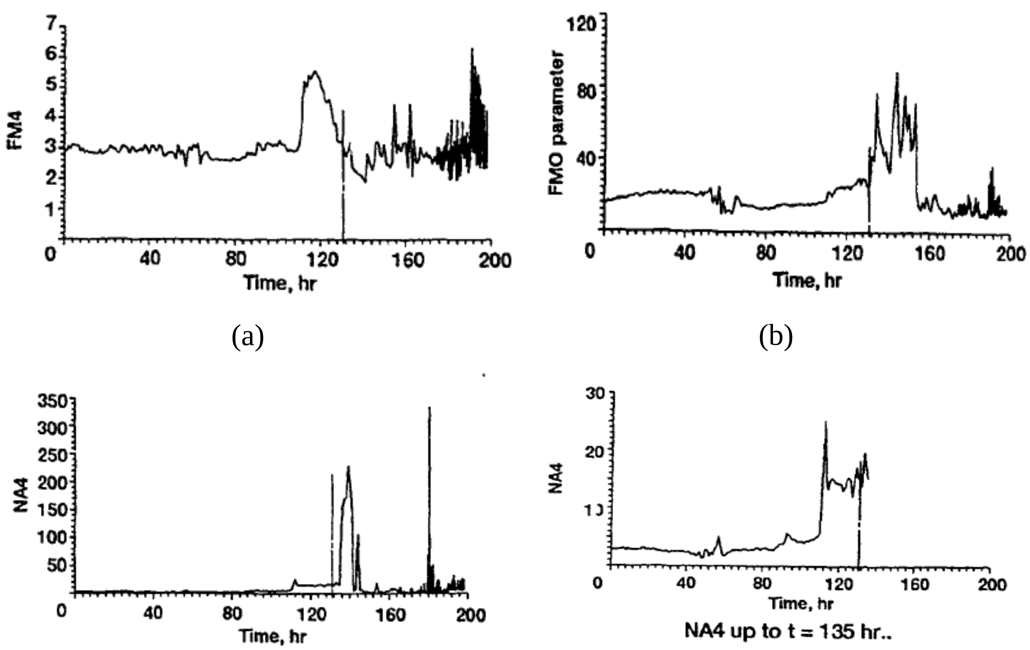
S. Suresh et al listed the three stages of pit formation, which starts from the initiation of crack, followed by growth of the crack and finally the breaking away of surface material layer (S. Suresh, 1984).

Also commenting on failure mechanics is Wilson Q.W. classified pitting and tooth breakage as localized gear faults. He further explained that pitting occurs due to contact stress exceeding the fatigue endurance limit of a certain material. When the material undergoes repeated cycle of loading, metal particulates on the material surface experience fatigue and will be removed. Once happens, this will propagate at an increasing rate due to the need of the unpitted areas to cater for the extra load removed. He had also discussed that cracking or tooth breakage occurs because of bending fatigue resulting from repetition of loading on gear teeth. Tooth breakage starts with a small crack, which will then spread to the point of the gear tooth to break off (Q.W., 2002).

In a study by Yesilyurt et al, it is stated that when damaged, a gear tooth would experience reduction in gear tooth stiffness. However, results showed that the reduction in stiffness only applied for localized damage in a gear tooth. The main culprit of surface damage on gear tooth was actually a deviation in involute profile of the gear. (Isa Yesilyurt, 2002) In view of machinery failures, K.L Johnson et al commented that the failures of gears are mostly contributed by surface pitting. In the study, it is also stated that surface pitting is the major mode of failure for machines that are subjected to rolling contacts (Johnson, 1989).

2.2 Past Research

In a study by Choy F.K. et al, a combination of techniques in vibration signature analysis was incorporated. It was found that the techniques would produce better and more reliable fault detection when used together, instead of employing them separately (F.K. Choy, 1994). A similar approach was done by Zakrajsek et al, where a combination of previously published failure prediction techniques and a newly developed technique was put to test. It was observed that the already established method FMO and the newly developed technique NA4 yielded convincing results in vibration detection. However, they proceeded with commenting that parallel utilization of the techniques would give a better means of detecting faults (J.J. Zakrajsek, 1993). The following diagrams depict the results obtained by Zakrajsek:



(a) (b) (c) (d)

Fig. 2.1 Zakrajsek's observation of different parameters and results *(obtained from Zakrajsek et al (1993))*

In Fig. 2.1(a), the parameter detected the tooth damage at t=110 hours, 21 hours before the rig was stopped and damage was recorded (t=131h). The FM4 peaked at a value of 5.4, and then dropped to the nominal value of 3. The parameter in Fig.

2.1(b), which is FMO, detected the damage at the time the rig was stopped for damage recording. During this time, the FMO increased three times its nominal value. In Fig. 2.1(c), the parameter detected the damage at the same time as FM0. In the expanded scale in of up until 135 hours in Fig. 2.1(d), the NA4 showed robust reaction to damage, increasing to 25 at the start of damage, and remained steady at 15 after that, despite other parameters dropping down to their nominal values.

For Yesilyurt's experimental part of their research, they have analyzed the results by monitoring vibration signals at 3 hour operating intervals. (Isa Yesilyurt, 2002). The results are presented in the forms of time-based waveform, vibration spectra and residual signal based parameters, as shown below:

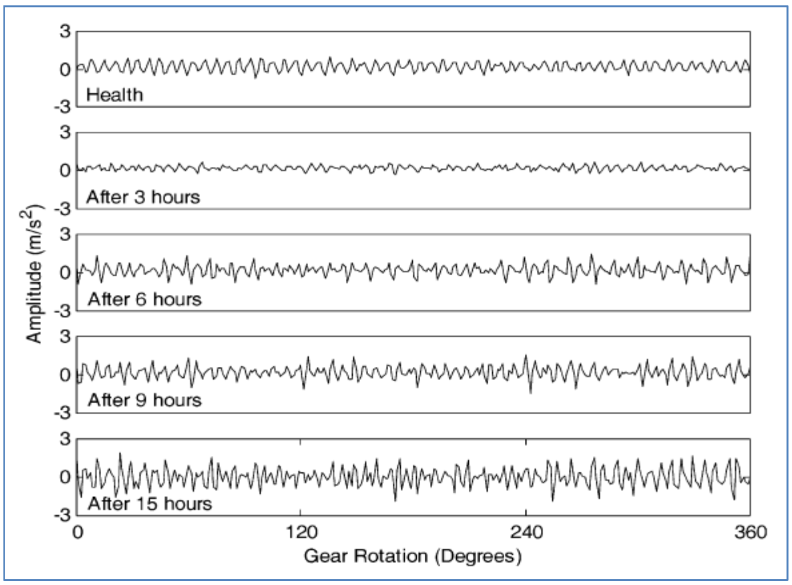


Fig. 2.2 Time-degree representation (*obtained from Yesilyurt et al. (2003)*)

From Fig. 2.2, it is indicated that the healthy gear vibration yields a uniform sinusoidal waveform in the time-degree representation. However, after 3 hours, the amplitude of the time-degree representation was reduced. At hours 6, 9 and 15, the appearance of the time-degree representation becomes random and the amplitude of vibration also increased.

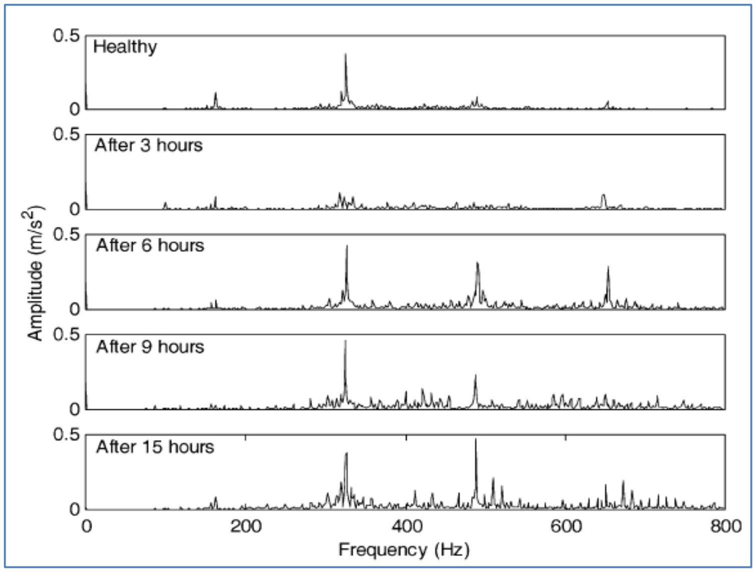


Fig. 2.3 Vibration spectra representation (*obtained from Yesilyurt et al. (2003)*)

In Fig. 2.3, it can be seen that the second tooth-meshing harmonic located at 325Hz was the strongest component of a healthy gear vibration. However, after 3 hours, the amplitudes of the first three tooth-meshing harmonics were reduced, but the fourth toothmeshing harmonic was increased slightly. Yesilyurt commented that this is due to the small deviations in the involute tooth form. At hours 6, 9 and 15, the second, third and less significantly, fourth tooth-meshing harmonic yielded larger amplitudes.

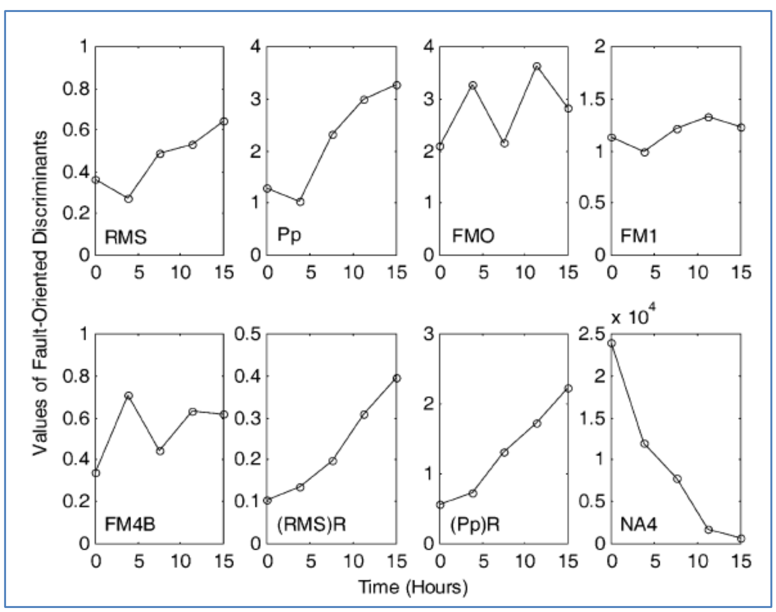


Fig. 2.4 Residual signal based parameters' reaction to vibration damage (*obtained from Yesilyurt et al. (2003)*)

In the representation of the residual signal based parameters’ reaction to vibration damage as shown in Fig. 2.4, it is observed that only two of the parameters: RMS and Peak to Peak reflected the progression of damage, where the two parameters decreased slightly at the beginning, but increased with the progression of damage. The other parameters did not exhibit a clear representation of vibration damage progression. NA4 also showed a trend in the progression of damage, but with a decreasing trend.

A similar research was conducted by Drosjack et al, in view of the correlation between gear mesh stiffness and damage. Drosjack et al studied on the effects of simulated pits upon pitch-line of the gear, and it was found that stiffness of the gear mesh changes with pitting ((M.J. Drosjack, 1977).

CHAPTER 3

METHODOLOGY

3.1 Project Activites

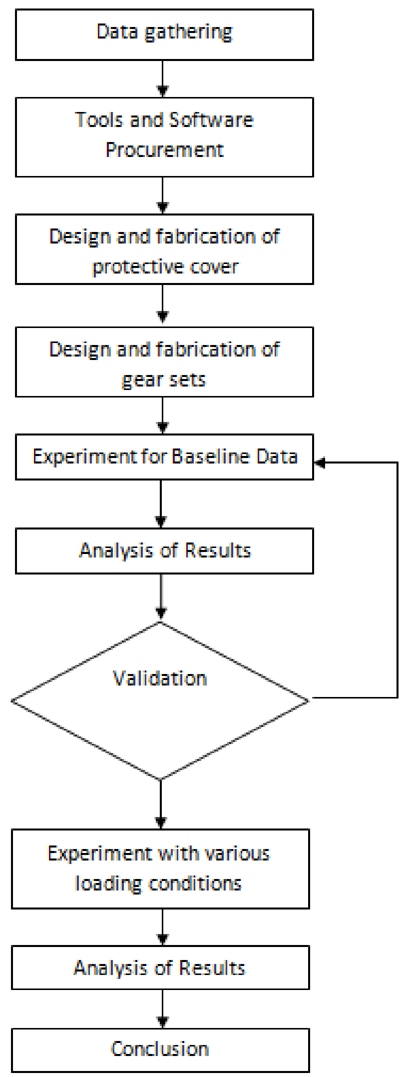


Fig. 3.1 Flowchart of the project

The project commenced with data gathering, where the author did an extensive literature review on available articles, journals or any references that are related to the project. Next, the author proceeded with the procurement of required tools and software for the project to run. These include confirming the availability of the gear test rig, and legally obtaining the vibration signal analysis software LabVIEW SignalExpress 2011 from the supplier. The gear test rig is located at Block 18 Academic Block. As the gear test rig does not have a protective cover, the author will design and fabricate a cover, as a safety measure. The cover will be made from Perspex. The test gears for the experiment were fabricated with the assistance of technicians at the Block 16 Academic Block. Two sets of spur gears were fabricated from a readily available S6061 Aluminum. In order for the technicians to fabricate the gears, the author has provided them with an AutoCAD drawing of the gears.

The baseline data for the experiment refers to the vibration data of a non-faulty gear set. To get this baseline data, the gear sets were mounted on the gear test rig without any deviation done to the shafts. The vibration signal will be picked up by accelerometers which will transmit the signals to a DAQ (Data Acquisition) system which sends the data to a personal computer, which will be analyzed in vibration data analysis software, LabVIEW Express. Analysis will be done by using a number of residual signal based parameters, which are Root Mean Square (RMS) and Peak to Peak. The corresponding FS and time-domain graphs will also be analyzed.

The second set of the experiment involved misalignments or deviations to occur on the shafts. This deviation caused an extra loading upon the shafts. The faulty gears were tested similar to the method done during the baseline data acquisition. Data analysis was also done the same way, making use of the same residual signal based parameters. The data that has been obtained were used in establishing a correlation between the types of fault that have been generated and the vibration signals that the gears give. Additionally, the vibration signal was also used to identify the relationship between the vibration characteristics and the severity of fault occurring on the gears.

3.2 Gear Design and Parameters

The material that has been chosen for the fabrication of the gear as statrd before is Aluminum S6061. This particular material has been chosen due to its low hardness, thus formation of defects upon the gears would be accelerated. Table 3.1 shows the parameters for the design of the gear and the pinion:

Table 3.1 Parameters of gear and pinion

Parameter	Gear	Pinion
Number of teeth	100	174
Pressure angle (^{o})	20	20
Module (mm)	1.0	1.0
Outside diameter (mm)	102	176
Pitch diameter (mm)	100	174
Root circle (mm)	97.5	171
Base circle (mm)	94	162.5
Normal pitch (mm)	2.95	2.95
Circular pitch (mm)	3.142	3.142
Angular pitch (^{o})	3.6	2.1
Thickness (mm)	10	10
Tooth height (mm)	2.25	2.25

3.3 Experimental Procedure

3.3.1 Arrangement of the Gear Test Rig

1. The gear test rig consists of two shafts, of which one of them is connected to a 3-phase electric motor. This is the driving side of the test rig. The other shaft is the driven shaft, where a hooking appendage is located at the other end for the loadings to be applied. Available loadings are 1kg, 2kg, 3kg and 5kg. Figure 3.2 shows the experimental setup.
2. The platform that supports the shafts can be adjusted vertically or horizontally to cater for different sizes of gears.
3. On each shaft, there are two bearing casings that house the bearings. On top of each casing an accelerometer can be attached in order to measure the system's vibration. These accelerometers will be connected to a DAQ (Data Acquisition System) hardware, which is connected to a computer or laptop, where the data will be recorded and analyzed.
4. It should be noted that in order to fit gears onto the shafts, the center of the gears might have to be fabricated in order to have the same size as the shafts.

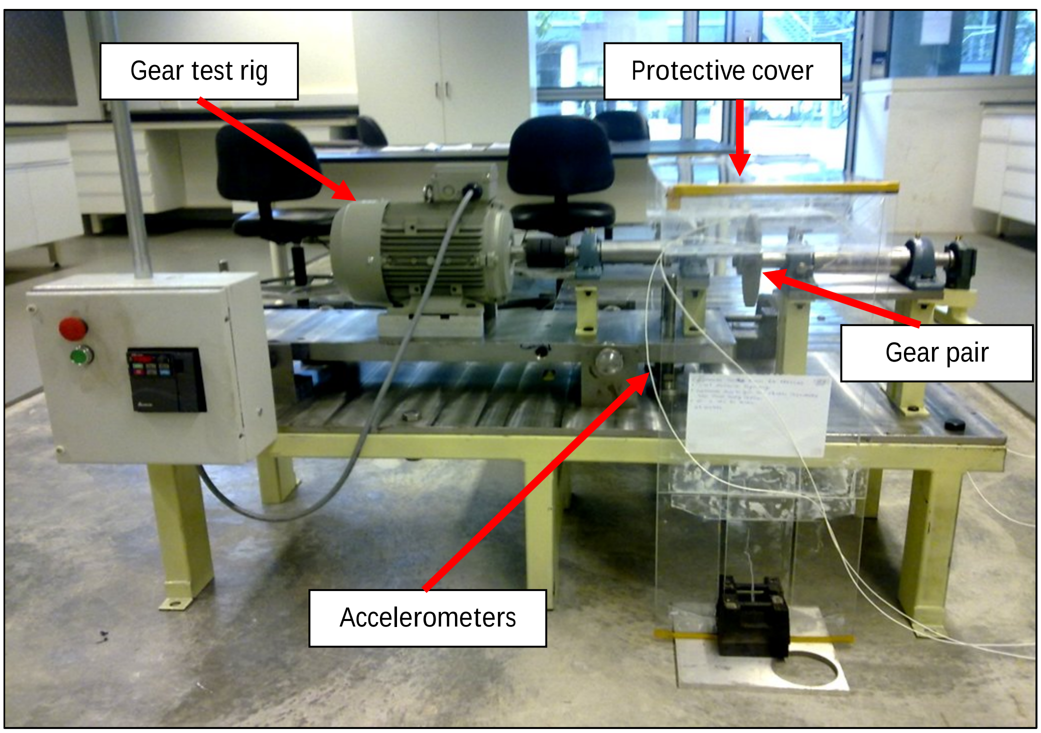


Fig. 3.2 Experimental setup

Table 3.2 shows the tools required for the experiment.

Table 3.2 Tools required for the experiment

Tool/Software	Usage
Aluminum S6061 Plates	Material for gear fabrication.
LabVIEW SignalExpress	To analyze vibration signal sent by the DAQ hardware.
Gear Test Rig	To perform experimental procedures upon the gear sets.
Accelerometers	To detect and measure gear vibrations.
DAQ hardware	To transfer vibration signals from accelerometers to the personal computer.
Miniature laser and protractor	To set the datum point for the shaft and measure shaft deviation.
Tachometer	To read/determine the current motor speed.

3.3.2 Experiment for Baseline Data

1. A gear set is installed on the test rig.
2. Both of the shafts are ensured to be parallel. This is set to be the datum.
3. Four accelerometers are mounted onto four points on the test rig (refer to Fig. 3.3):
   I. Pinion – Vertical
   II. Pinion – Horizontal
   III. Gear – Vertical
   IV. Gear - Horizontal
4. The test rig is switched on and run initially at 500rpm for 10minutes. The acceleration graph is recorded and observed.
5. The procedure is repeated at a speed of 1250rpm.

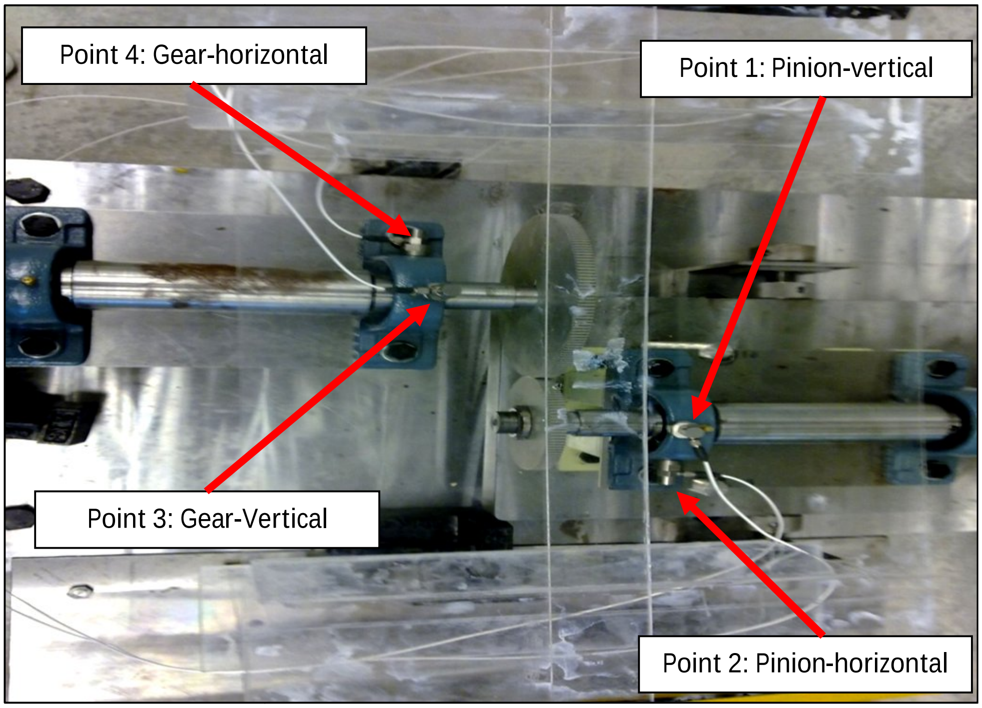


Fig. 3.3 Placement of accelerometers

3.3.3 Experiment with Loading Conditions

1. The setup in (A) is retained.
2. The pinion shaft is tilted 3^{o} from the datum point, which would cause the gears to tightly mesh.
3. The test rig is run at 1250rpm for 5 minutes. The acceleration graph is recorded and observed.
4. The pinion shaft is returned to the datum point. A 2kg weight is loaded at the end of the gear shaft and the test rig is run for 5 minutes. The acceleration graph is recorded and observed. This step is repeated with a 5kg weight.

3.3.4 Experiment with Continuous Run

1. The setup in (A) is retained.
2. The test rig is run at 1250rpm for 15 minutes. The acceleration graph is recorded and observed. At the end of 15 minutes, any damage on the gears is observed.
3. Repeat step 2 until significant amount of deterioration is achieved.

CHAPTER 4

RESULTS AND DISCUSSIONS

4.1 Gear Design and Fabrication

The gears were first designed by using AutoCAD. Figure 4.1 shows the complete design of the gears.

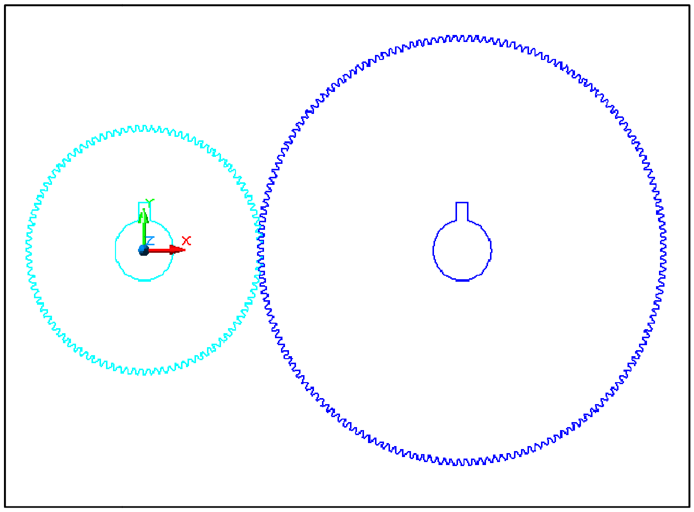


Fig. 4.1 AutoCAD drawing of the gear and pinion

The gears were fabricated by emloying an EDM Wire Cut machine. Figure 4.2 shows the fabricated gears.

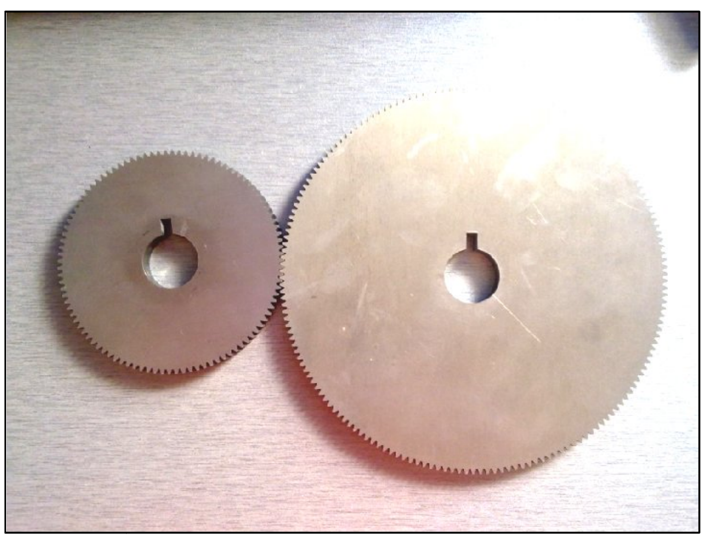


Fig. 4.2 Fabricated pinion (left) and gear (right)

4.2 Visual Observation of Spur Gear

The experiment was conducted for a total of 3 hours and 30 minutes when severe deterioration was noticed at the gear and pinion's teeth. During the run, the test rig was stopped at 15-20 minutes interval to allow for visual observation of the gears' condition. The following are the findings for every recorded interval:

- 15 minutes: Some darkened spots at the edge of gear teeth, while no change in condition for pinion.

- 45 minutes: Increasing intensity of the darkened spots. Scuffing exhibited on both gears, along with some obvious scratches on some of the teeth. These observations are shown in Figure 4.3 below.

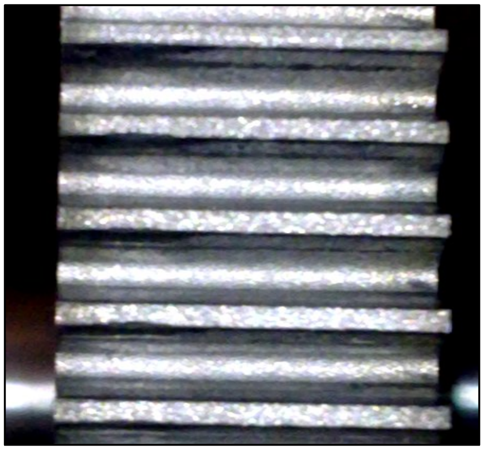


Fig. 4.3 Gear condition after 45 minutes

- 1 hour 30 minutes: Dark coloured particles begin to accumulate.

- 2 hours: As shown in Figure 4.4, small pit-like formations can be seen on some of the tooth. These formations were increasing in size. The addendum of the gears exhibited severe scratch marks and wear.

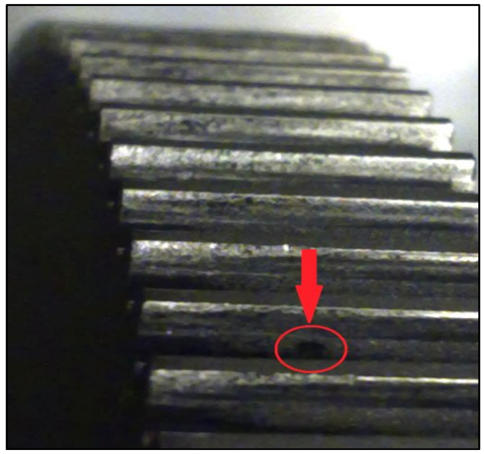


Fig. 4.4 Gear condition after 2 hours

- 3 hours 30 minutes: All of the teeth for both gears were completely covered with scratch marks and darkened spots on the addendum. Dark lines begin to form at the bottom of the gear teeth. These observations are shown in Figure 4.5.

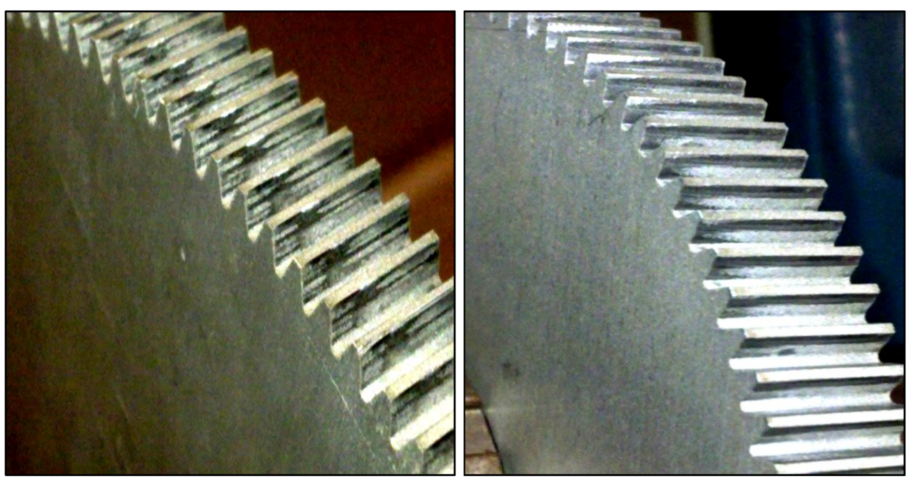


Fig. 4.5 The conditions of the gear (left) and pinion (right) after 3 hours of run

4.3 Vibration Analysis

Three separate analyses were done during the experiment, corresponding to the four different experimental setups, which are baseline acquisition, deviation of the pinion shaft, bending moment acting on the gear shaft and continuous run for 3 hours and 30 minutes.

Before proceeding with the analyses, it would be best to give brief information on the types of residual signal based parameters and the frequency spectrum that are used extensively in the analyses:

- Root-Mean-Square (RMS) indicates the amount of vibration energy or vibration content for particular machine. The parameter is good in investigating the health of a machine. It is a good indication of a machine's overall noise level. RMS is calculated as below:

  $$rms = \sqrt{\frac{1}{N}\sum_{i=1}^{N}(x_i - \mu)^2}$$

  Where, N is the number of samples, x_i is the amplitude of a sample and μ is the sample's mean value.
- Peak to peak value is the difference between the positive and negative amplitudes of a vibration signal. Higher values indicate increasing vibration severity.
- Frequency spectrum is a graphical display of difference frequencies of a vibrating system along with the corresponding amplitude level for each frequency. A system will possess different number of frequencies as there will be more than just one simple vibratory motion taking in place simultaneously. By studying these different frequencies, the causes of vibration can be inferred from these data. (How Is Machine Vibration Described)

For the analyses, a number of essential information needs to be known. The gear to pinion ratio for the experiment is 1:1.74, with a rotating speed of 1250rpm. This rotating speed gives a fundamental tooth-meshing frequency of 2083 Hz. The fundamental tooth-meshing frequency is obtained from the below relationship:

$$Fundamental\ tooth - meshing\ frequency = rpm \times no.of\ teeth$$
$$= 1250rpm\ \times 100$$
$$= 125{,}000\ rpm$$
$$= \mathbf{2083\ Hz}$$

For the analysis, the raw vibration data was sampled at 10,000 Hz, which would sufficiently reveal the frequency content of the vibration up to the fifth tooth-meshing harmonic. The vibration data was averaged using RMS averaging with 10 averages. The results were mainly analyzed from the resulting frequency spectrum for each of the experimental setup.

The initial setup was to run the gears at speeds 500rpm and 1250rpm without any loading condition. This was done to acquire the baseline data as a basis for comparison for the subsequent setups. Below is the resulting frequency spectrum for the initial setup:

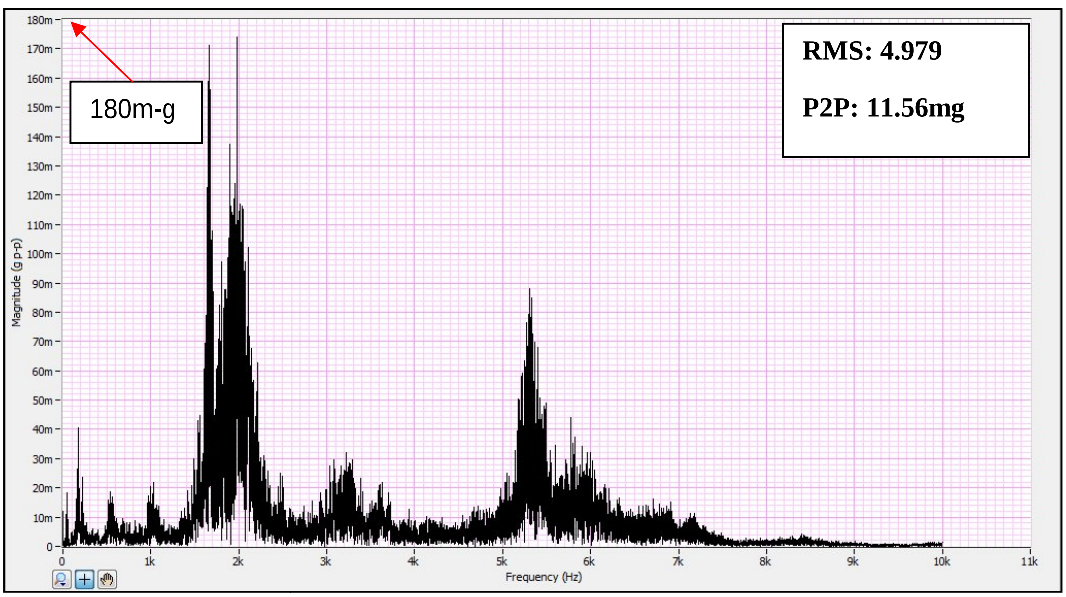


Figure 4.6 Frequency spectrum for baseline data of 500rpm

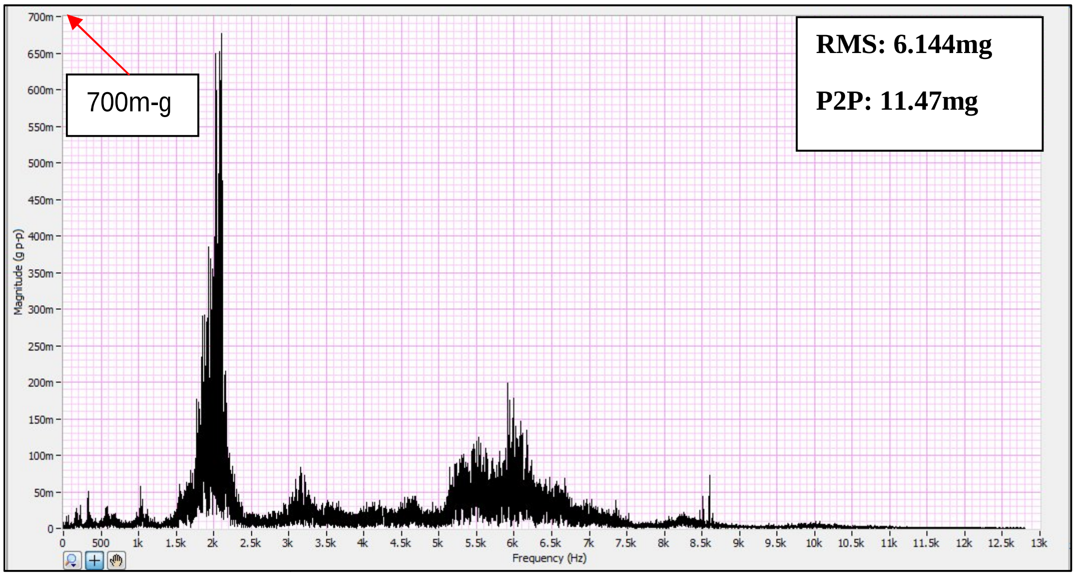


Fig. 4.7 Frequency spectrum for baseline data of 1250rpm

Figures 4.6 and 4.7 show the FS for speeds 500rpm and 1250rpm without any loading conditions. As evidenced by the peak value of each of the spectrum, the setup with the higher speed exhibited a higher peak value compared to the lower speed. This result infers that at the same time interval, in this case 10 minutes, a higher speed would cause more vibration. This inference is supported by the values of RMS and P2P, which are 4.979mg and 11.56mg for 500rpm and 6.144mg and 11.47mg for 1250rpm respectively. The increase in the RMS and P2P are not significant as at the start of the experiment, the test rig is still in good condition as it was newly purchased. It should be noted that the fundamental tooth-meshing frequency – in this case, 2083Hz, possesses the highest peak due to the inherent vibration by the gear itself. (Shreve, 1994)

The second setup was to run the test rig with the pinion shaft tilted 3 degrees from its original position, resulting in tight meshing between the two gears. The resulting FS is as below:

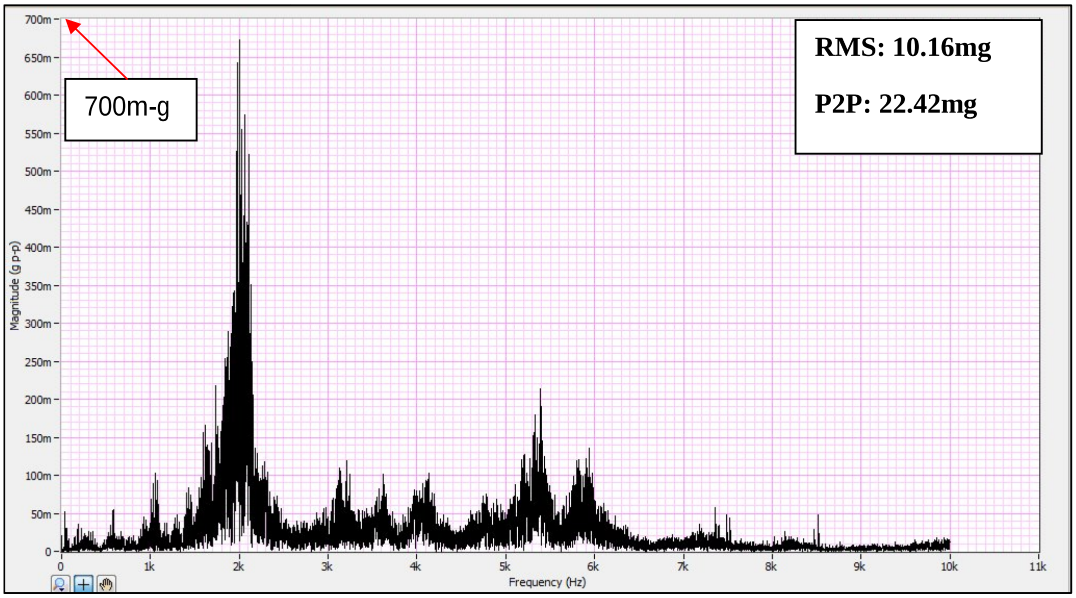


Fig. 4.8 Frequency spectrum for deviated shaft

When the resulting frequency spectrum in Figure 4.7 is compared with that of Figure 4.8, it can be seen that higher amplitudes are reached at the first and especially nearing the second harmonics (6249Hz). This is attributed to the misalignment between the shafts, which caused complete contact between the gears without any clearances. This inference is supported by an excerpt from the vibration troubleshooting guideline chart as shown in Table 4.1.

Table 4.1 Vibration guideline chart (courtesy www.reliability.com)

VIBRATION GUIDELINE CHART		
Frequency	**Probable Cause**	**Other Possible Causes**
Running Speed (1X)	Imbalance	Misalignment Bent Shaft Resonance Eccentric journals, gears or pulleys Reciprocating forces Electrical
Two Times or First Harmonic (2X)	Mechanical Looseness	Misalignment; high in the axial direction Reciprocating forces Resonance Loose bearing or part
Second Harmonic (3X)	Misalignment (shaft to shaft)	Excessive axial clearance Electrical; air gap Internal misalignment Machine part

Additionally, the RMS and P2P values for this setup are 10.16mg and 22.42mg respectively, indicating an almost two-fold increase from the baseline data, further implying that misalignment causes severe vibration within a vibrating body.

The third setup involved hooking a weight at the end of the gear shaft. Two magnitudes of weight were used, which are 2kg and 5kg. These weight represented bending moment within the system, specifically at the shaft that holds the gear. Figure 4.9 and Figure 4.10 shows the resulting frequency spectrums due to the setup.

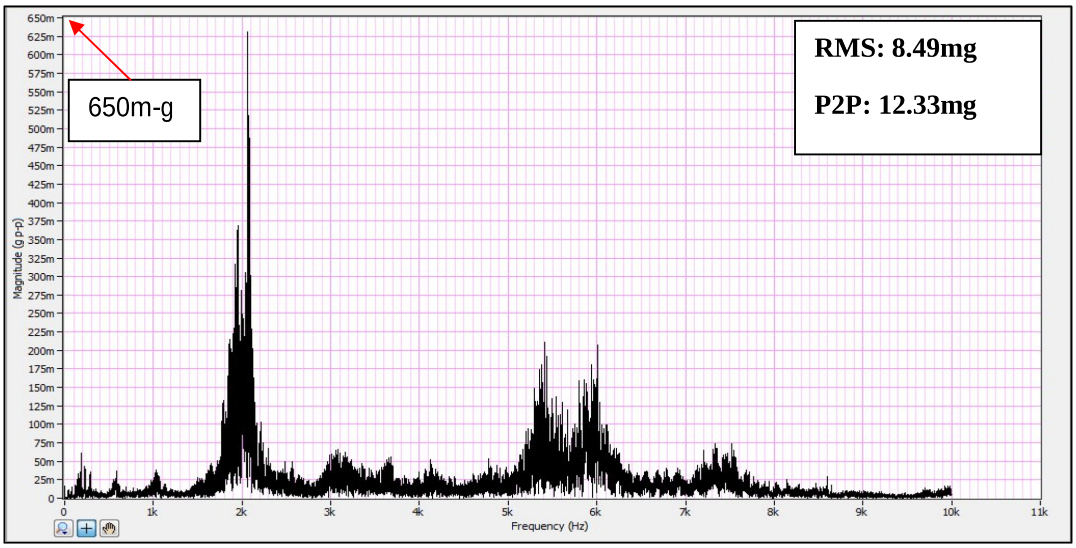


Fig 4.9 Frequency spectrum for 2kg loading

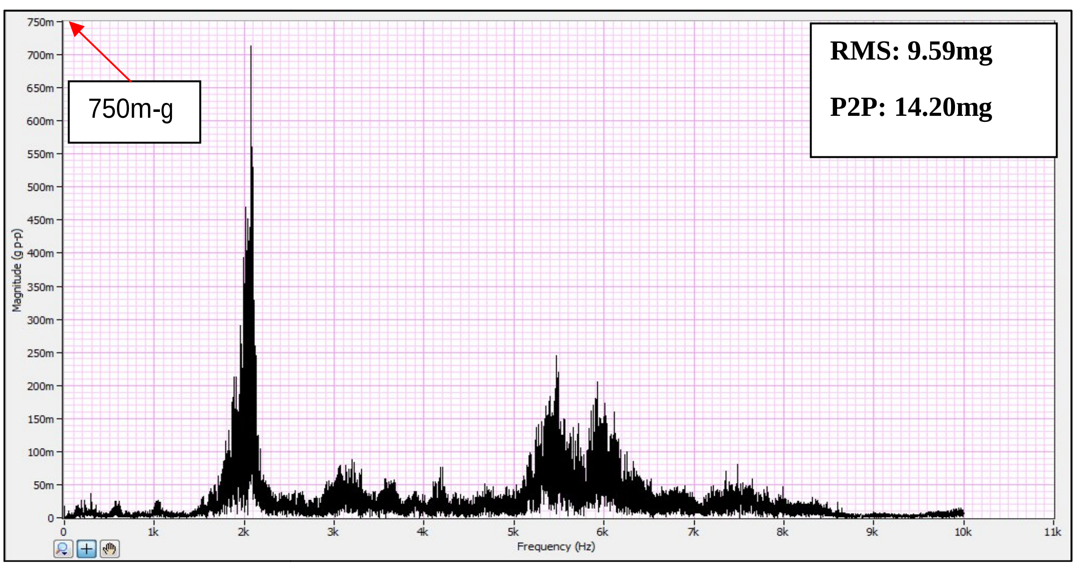


Fig. 4.10 Frequency Spectrum for 5kg loading

In comparison with the second setup, where the misalignment was introduced to the system, the resulting frequency spectrum for the bending moment setup do not differ as much as one would expect, where only the peak values are different, albeit a small amount. However, a significant difference between the two setups is noticeable in the values of the RMS and P2P. For the 2kg weight, the RMS and P2P values are 8.49mg and 12.33mg, while for the 5kg weight, the RMS and P2P values are 9.59mg and 14.20mg. The values are much lower – almost half than that of the second setup. From here, it can be inferred that both setups cause misalignment, as evidenced by the similar amplitude increment in the second harmonic. However, a bending moment in a system would result in a lower vibration energy compared to a deviated shaft.

The final setup was to run the test rig continuously until severe deterioration was observed. Figure 4.11 shows the frequency spectrum after 2 hours of run.

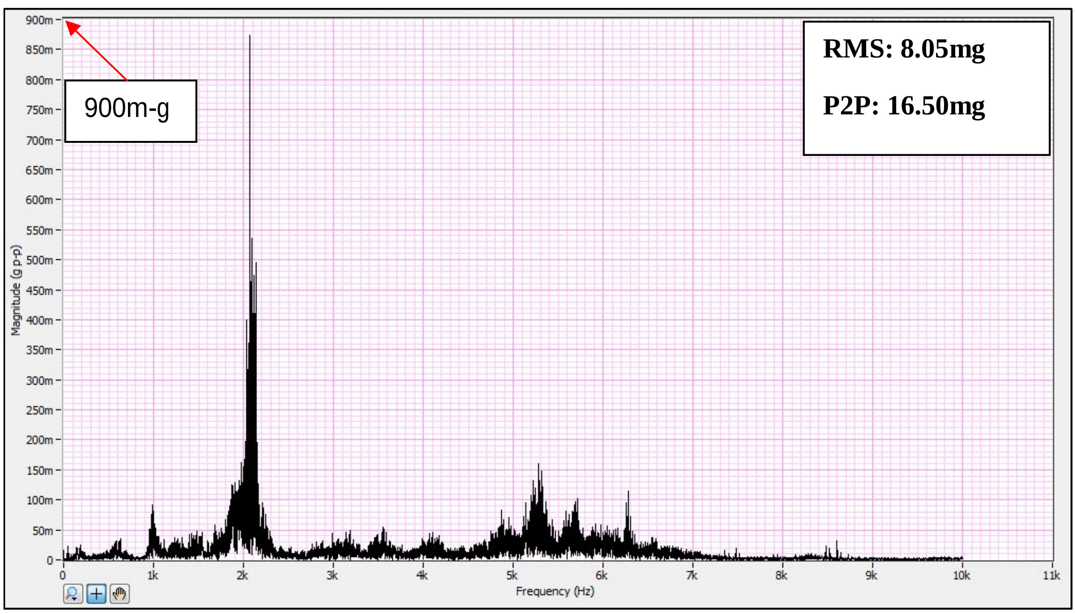


Fig. 4.11 Frequency spectrum after 2 hours of continuous run

The result was taken after 2 hours of run, as it was after this time interval that severe deterioration started to manifest. From the frequency spectrum in Figure 4.11, it is shown that the peak value is higher than the previous setups (900m-g.) As no loading condition was applied for this setup, it is best to compare with the baseline data of 1250rpm. There is a large difference between the RMS and P2P values for both of the setups. For this setup, the RMS and P2P values are 8.05mg and 16.50mg respectively, both almost twice the baseline value. From this result, it can be inferred that when subjected to long operation periods, a system would exhibit high vibration energy, and the increase in P2P value shows increasing radial displacement of the shafts. It was at this period that pit-like formations are observed. Also, as no loading conditions are applied in this setup, no significant increase in amplitude is observed at the second harmonic.

The results that were taken after time intervals 15 minute, 30 minutes, 45 minutes, 1 hour, 1 hour 15 minutes, 1 hour 30 minutes and 1 hour 50 minutes do not show any significant vibration signatures nor there were any faults detected, thus the vibration signatures are only included in APPENDIX III (Appendices 3.1-3.7).

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 Conclusion

The type of gear defects and causes of vibration can be determined by analyzing residual signal based parameters such as Root-mean-Square and Peak-to-Peak levels and the frequency spectrum.

A misalignment in a shaft is represented by a significant amplitude increase in the 2nd harmonic in a frequency spectrum (3 times the fundamental tooth meshing frequency) at higher RMS and P2P levels compared to the effect bending moment and continuous run. The peak in the FS of a bending moment-induced vibration is lower compared to an FS of a misalignment-induced vibration. Thus, a deviation in a shaft that causes tight meshing between the gears will result in severe vibration.

As a machine is continued to run, the RMS and P2P levels will rise, along with the amplitude in the frequency spectrum. This is accompanied by a formation of pits at the gear teeth. However, no increase in amplitude will be observed in the second tooth-meshing harmonics.

5.2 Recommendations

For the improvement of future research along the lines of this study, the author would like to suggest a number of recommendations. Firstly, the gear test rig could be run at a faster speed, as to simulate a real environment of a standard gear operation. This would also allow for an accelerated fault occurrence, as opposed to the progress of the fault in this research. Secondly, the experiment would greatly benefit from more number of accelerometer points, as this would allow for a more detailed result in the end. For example, a pair of accelerometers can be placed at the face of the bearing houses, as to obtain axial vibration data. Thirdly, during the analysis and processing of the raw vibration data, various manipulations of the data – which include the different degrees of filtering, type of windows, number of averaging, can be varied as to acquire a larger spectrum of result that would open to more in-depth conclusions on the study. Lastly, the material of the gear can be varied as the type of material that is used to fabricate the gears will give different vibration characteristics based on the materials' properties.

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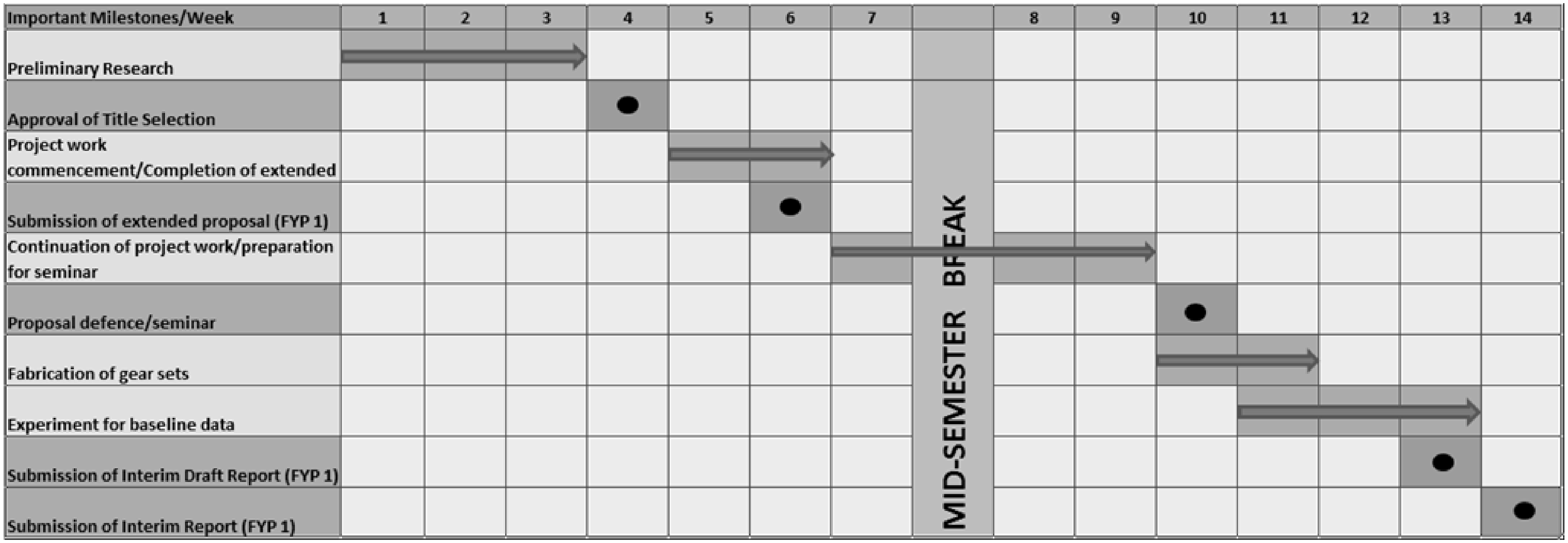
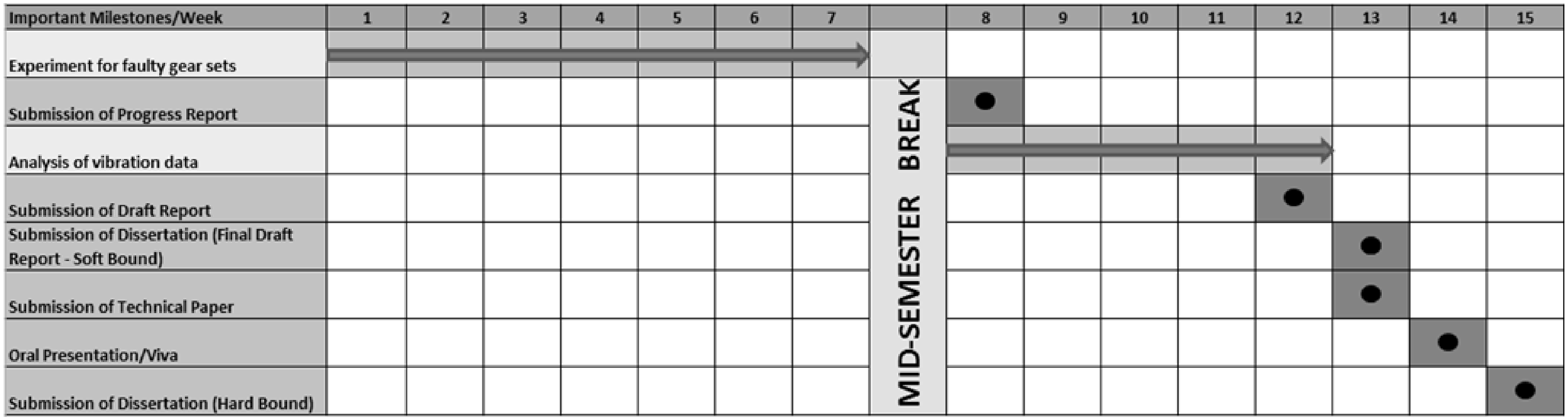
APPENDIX I



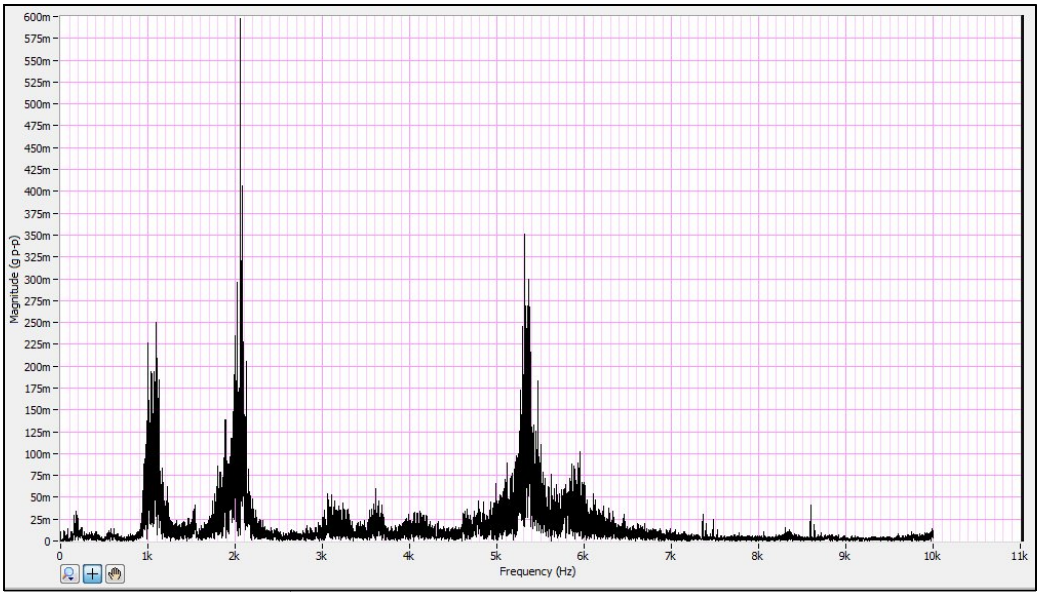
Appendix 1.1 Gantt Chart for FYP1

APPENDIX II

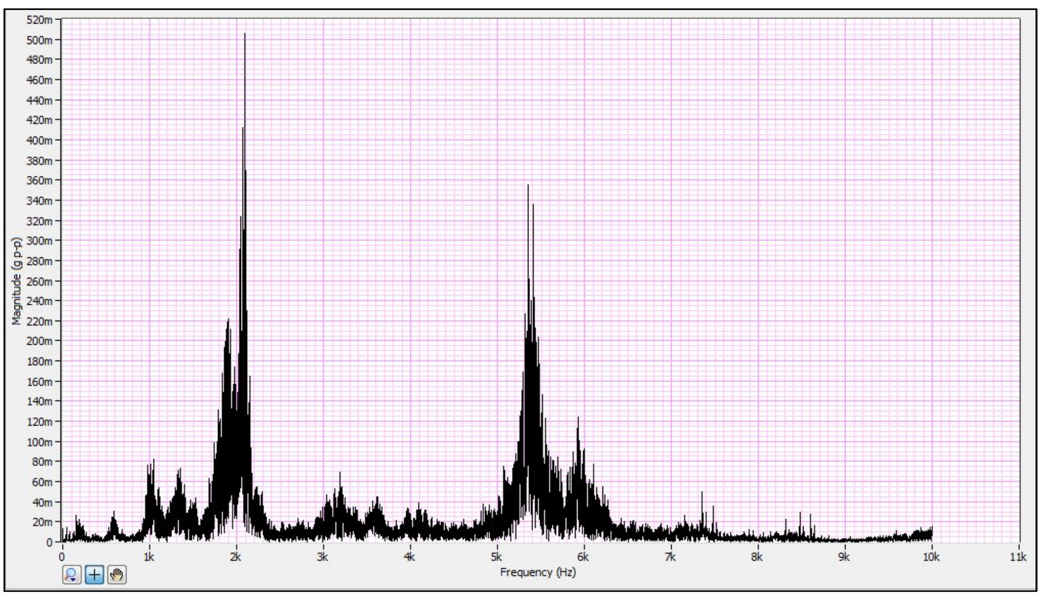


Appendix 2.1 Gantt Chart for FYP2

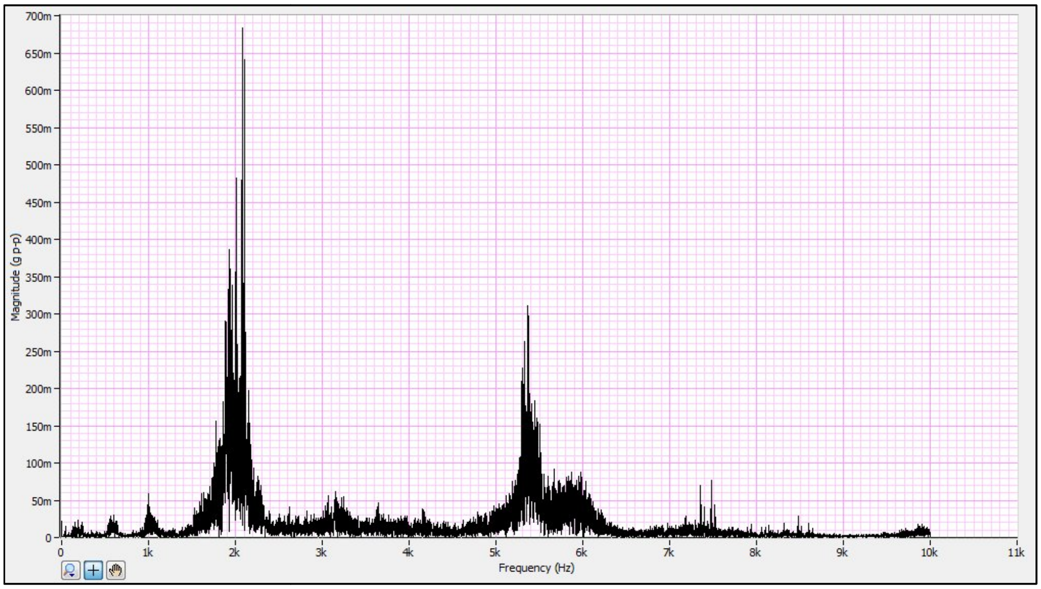
APPENDIX III



Appendix 3.1 Frequency spectrum after 15 minute run

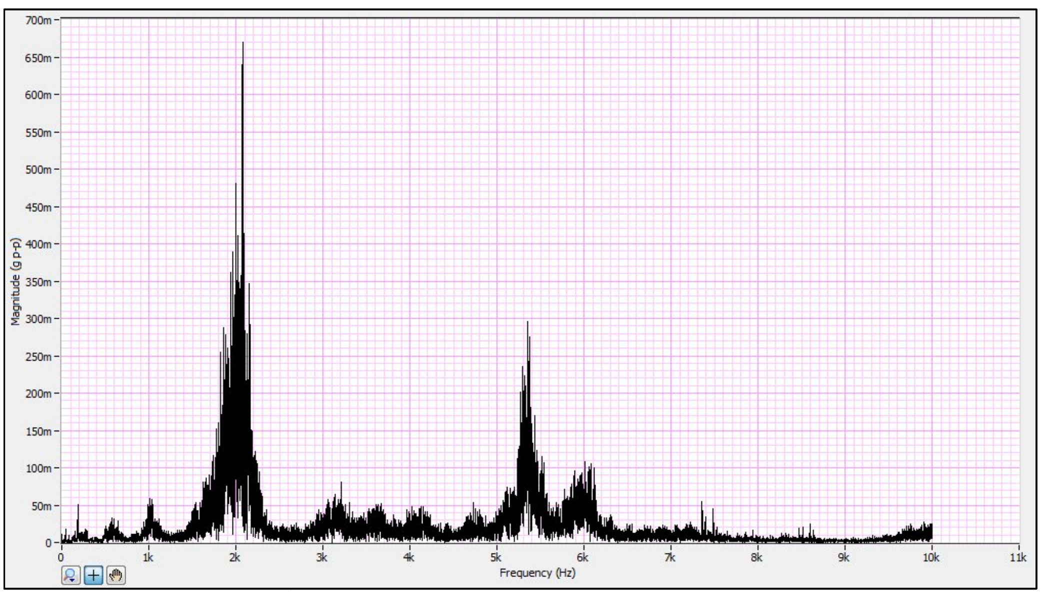


Appendix 3.2 Frequency spectrum after 30 minute run

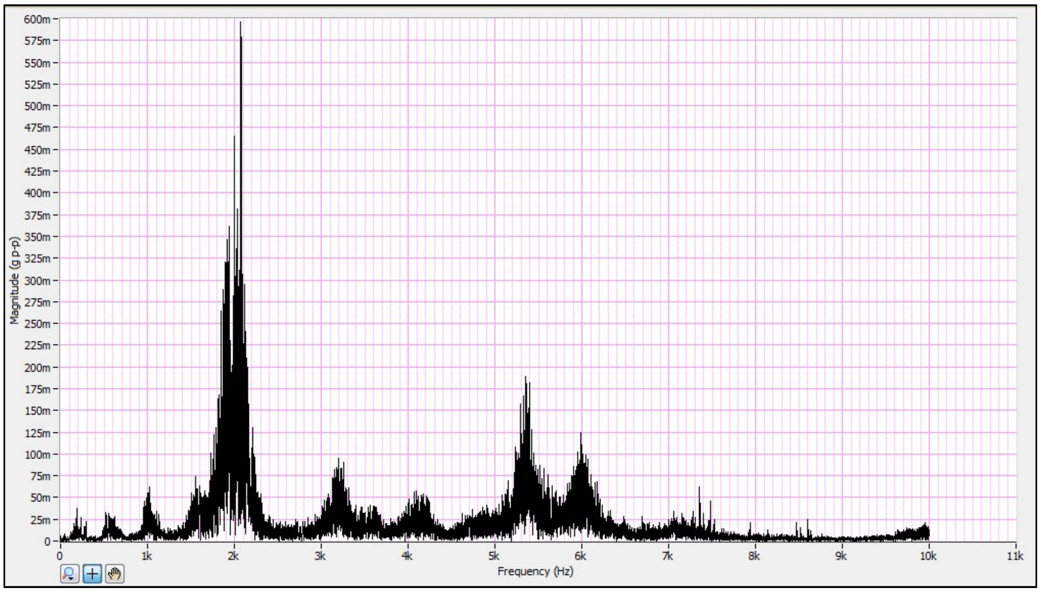


Appendix 3.3 Frequency spectrum after 45 minute run

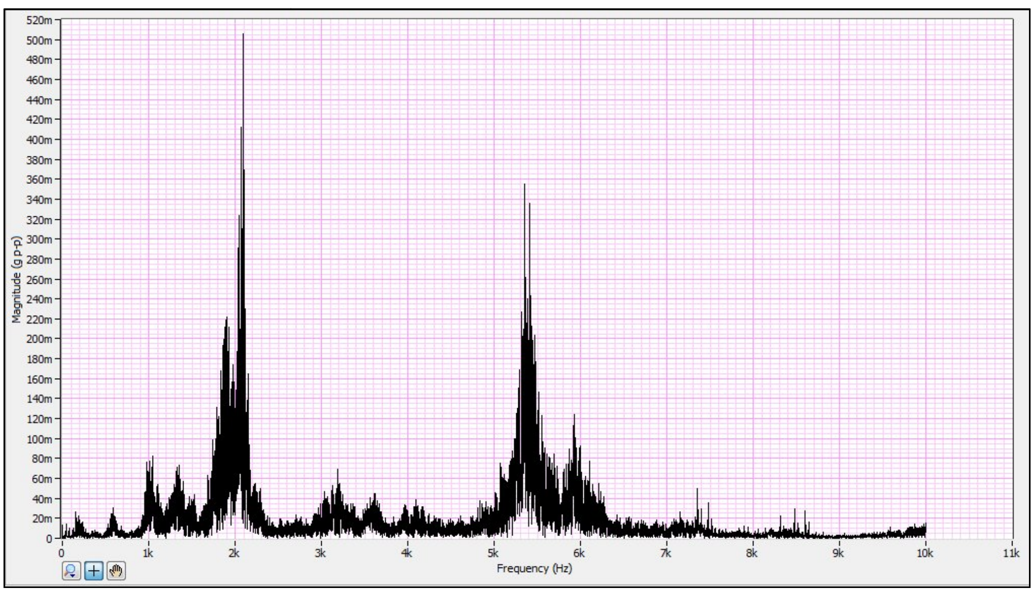
Frequency spectrum after 1 hour run



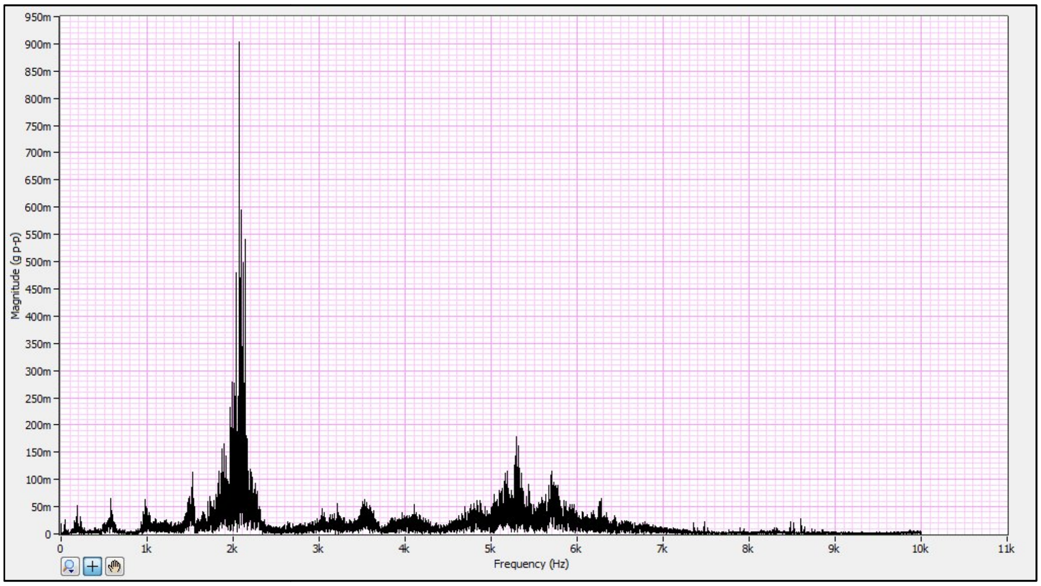
Appendix 3.4 Frequency spectrum after 1 hour run



Appendix 3.5 Frequency spectrum after 1 hour 15 minute run



Appendix 3.6 Frequency spectrum after 1 hour 30 minute run



Appendix 3.7 Frequency spectrum after 1 hour 50 minute run