MODAL ANALYSIS FOR ENGINE CRANKSHAFT

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We hereby declare that we have checked this project and in our opinion this project is satisfactory in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering with Automotive Engineering

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MODAL ANALYSIS FOR ENGINE CRANKSHAFT

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A thesis submitted in fulfillment of the
requirements for the award of the degree of
Bachelor of Mechanical Engineering with Automotive Engineering

Faculty of Mechanical Engineering
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Dedicated to my beloved

parents and friends

for their support and motivation that they gave

while working on this thesis

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ABSTRACT

Crankshaft is a fundamental and a very crucial part in internal combustion engine. Its role as the main translational-rotational converter have been used and perfected as early as 1226 by Al-Jazari in his water pump machines. This paper consists of finding the mode shape and natural frequency of a 3 cylinder 4 stroke engine crankshaft. The test is done in both simulation and also experimental using a simple test rig. The crankshaft is modeled using Solidworks computer aided design (CAD) software and simulation analysis is done in ALGOR computational aided engineering (CAE) software. Experimental is done by using impact hammer to excite the crankshaft and data recorded using data acquisition system (DAS) connected to sensor located on the crankshaft. The post processing software used after experimental is done is Me'ScopeVES software. The results for both simulation and experimental is compared. The mode shapes is simulated using ALGOR. The differences in the results between simulation and experimental is discussed. The final selected natural frequency for simulation is based on mesh aspect ratio of 80%. Simulation natural frequency in 1st mode is 688.494 Hz (bending), 2nd mode is 707.661 Hz (bending), 3rd mode is 1098.9 Hz (bending), 4th mode is 1273.63 Hz (torsion) and 5th mode is 1664.23 Hz (bending). Meanwhile, the experimental natural frequency (x-axis) in 1st mode is 668 Hz, 2nd mode is 722 Hz, 3rd mode is 1300 Hz, 4th mode is 1480 Hz and 5th mode is 1580 Hz. Experimental natural frequency (y-axis) in 1st mode is 724 Hz, 2nd mode is 742 Hz, 3rd mode is 850 Hz, 4th mode is 1130 Hz and 5th mode is 1300 Hz. Experimental natural frequency (z-axis) in 1st mode is 475 Hz, 2nd mode is 724 Hz, 3rd mode is 775 Hz, 4th mode is 1120 Hz and 5th mode is 1320 Hz. The discrepancy errors recorded between simulation and experimental is ranging from 2 – 23.11%.

ABSTRAK

Aci engkol adalah salah satu alat terpenting didalam enjin pembakaran dalaman. Ia bertindak sebagai alat pengubah gerakan searah kepada putaran yang telah disempurnakan oleh Al-Jazari pada seawal 1226 dan telah digunakan didalam mesin pam airnya. Kertas ini terdiri dalam mencari bentuk mod dan frekuensi semulajadi aci engkol sebuah enjin 3 silinder 4 lejang. Ujian ini dijalankan dalam dua keadaan iaitu eksperimen dan simulasi komputer. Aci engkol akan dimodelkan menggunakan perisian CAD SOLIDWORKS dan analisis simulasi akan dilakukan didalam perisian CAE ALGOR. Eksperimen pula akan dilakukan menggunakan tukul impak untuk menggetarkan aci engkol tersebut. Data akan direkod menggunakan alat pengumpulan data dimana ia disambungkan pada sensor yang terletak pada permukaan aci engkol tersebut. Kemudian, data eksperimen akan diproses menggunakan perisian Me’ScopeVES. Keputusan yang diperolehi dari kedua-dua eksperimen dan simulasi akan dibandingkan. Bentuk mod pula akan dihasilkan oleh simulasi melalui perisian ALGOR. Perbezaan kepada kedua-dua data telah dibincangkan. Nilai frekuensi semulajadi yang muktamad dipilih dari nisbah mesh 80%. Cadangan untuk mempertingkatkan lagi kualiti eksperimen dan simulasi di masa hadapan juga telah disenaraikan. Frekuensi semulajadi secara simulasi pada mod 1 ialah 688.494 Hz, mod 2 ialah 707.661 Hz, mod 3 ialah 1098.9 Hz, mod 4 ialah 1273.63 Hz dan mod 5 ialah 1664.23 Hz. Frekuensi semulajadi secara eksperimen (paksi-x) pada mod 1 ialah 668 Hz, mod 2 ialah 722 Hz, mod 3 ialah 1300 Hz, mod 4 ialah 1480 Hz dan mod 5 ialah 1580 Hz. Frekuensi semulajadi secara eksperimen (paksi-y) pada mod 1 ialah 724 Hz, mod 2 ialah 742 Hz, mod 3 ialah 850 Hz, mod 4 ialah 1130 Hz dan mod 5 ialah 1300 Hz. Frekuensi semulajadi secara eksperimen (paksi-z) pada mod 1 ialah 475 Hz, mod 2 ialah 724 Hz, mod 3 ialah 775 Hz, mod 4 ialah 1120 Hz dan mod 5 ialah 1320 Hz. Peratus ralat yang telah direkodkan berada dalam lingkungan 2 – 23.11%.

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LIST OF ABBREVIATIONS

CAD	Computer Aided Design
CAE	Computer Aided Education
DAS	Data Acquisition System
DOF	Degree Of Freedom
EMA	Experimental Modal Analysis
ERA	Eigensystem Realisation Algorithm
FEA	Finite Element Analysis
FRF	Frequency Response Feedback
FYP	Final Year Project
ICE	Internal Combustion Engine
MIMO	Multiple Input Multiple Output
SIMO	Single Input Multiple Output

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CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION

Since 1335, the theory of automobile had been recorded, from the first to be driven by wind to the latest state of the art electric vehicle. In 1876, Nicolaus August Otto invented the first internal combustion engine known as The Otto Cycle Engine. Then in 1886, Gottlieb Daimler improved the idea of internal combustion engine that is practical to be used as an automotive engine. Since then, the idea of internal combustion engine had improved exponentially but, the very basic concept of Otto cycle still remains.

In Otto cycle engine or Internal Combustion Engine (ICE), a piston, or series of it will reciprocate in a linear manner, for example, ups and downs. Such linear motion need to be translated into rotational motion in order for the power generated from the movement of the piston into any mechanism that can move the car. The mechanical part that translates all the piston's linear motions is called crankshaft.

In modern ICE, the crankshaft can rotates up to 20000 rpm. At 20000 rpm, with forces exceeds 3000 N push down on the crankshaft, careful consideration of material and calculations is needed to create a crankshaft that can take not only directional forces, but also rotational motion. As the crankshaft rotates, vibration will occurs. The crankshaft working cycles will be shortened dramatically if the vibration is not contained. Frequency of vibration will often occur when a system is vibrating, in this case, the crankshaft.

In ICE, every part has its own natural frequency. Natural frequency is the frequency at which a system naturally vibrates once it has been set into motion. In other words, natural frequency is the number of times a system will oscillate (move back and forth) between its original position and its displaced position, if there is no outside interference. If the crankshaft vibrating frequency is near or same as its natural frequency, resonance will happened and the effect on the crankshaft or even the ICE itself is catastrophic.

1.1.1 Project Background

Crankshaft is a fundamental and a very crucial part in ICE. Its role as the main translational-rotational converter have been used and perfected as early as 1226 by Al-Jazari in his water pump machines. Typically in an ICE, the crankshaft would be fitted with flywheel in order to reduce the pulsating of sudden force exerted during combustion. Without the flywheel, the crankshaft will experience even higher pulsating and thus, increase the crankshaft wear and tear in the crankshaft structure.

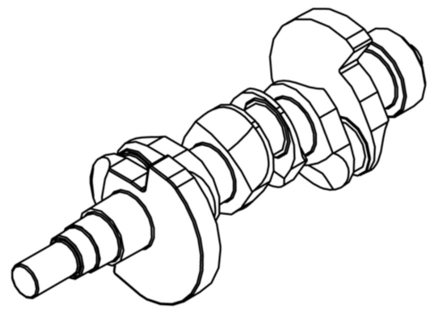

Figure 1.1: Engine crankshaft

Source: Solidworks 2008

Another major problem when dealing with crankshaft is its out-of-balance characteristics. From the figure above, we can see that the crank pins is off from the center, will surely will act as an out-of-balance mass. With the crankshaft is out of balance, it will vibrate when the crankshaft rotate. Frequency will be generated when the crankshaft rotates and if the crankshaft rotating frequency is equal to its natural frequency, resonance will happened and the crankshaft will break in the long term.

Crankshaft in modern ICE normally had been fitted with a counter-balance mass just opposite to compensate with the mass of crank pins. Even though this works amazingly well, it is really impossible to contain all the vibration on the crankshaft, and thus resonance can still happens.

1.2 PROBLEM STATEMENTS

In early era which is the usage of a crankshaft is limited, wood is the most favorable material to be used. Later, when crankshaft became one of the most important parts of machinery, steel have been used. The most commonly used is cast ductile iron, as it is easy to manufacture and easily available at a low price compared to the more advanced forged steel.

Material selection is important as it affects the crankshaft performance, for example, a normal road car engine typically used a cast iron crankshaft, with an optimal torque to be at around 3000-4000 rpm, the crankshaft need to be fabricates so that resonance would not occur during those working conditions. On the other hand, a higher rpm engine with optimal torque output at around 6000-8000 rpm or even higher, material such as forged steel have been seen used. Furthermore, extreme applications such as F1 engine crankshaft which normally operates exceeds 18000 rpm, material such as 4340 VAR Electric Arc Vacuum Re-melt Chrome-Moly Alloy steel have been used.

The selection of material will affect the natural frequency of the crankshaft. Proper selection is needed to avoid resonance in the crankshaft, which in this case, whirling. Whirling can affect the crankshaft’s life and engine performance. Whirling causes the crankshaft to deflect, and when crankshaft is deflected, even in micron meters, not only shorten its life but also reduced the engine power output.

In ICE, the crankshaft will likely to fracture when experienced excessive vibration in a long term period. Possible types of failure that normally occurs in crankshaft including shearing and bending and possible breaking. This vibration normally gets bad during resonance which the crankshaft vibration amplitude will vibrate at its peak limit. Resonance will happen when the crankshaft vibration oscillates at its own natural frequency.

1.3 PROJECT OBJECTIVES

These are the main objective for this project:

a) To find the crankshaft natural frequency, ω_n and the mode shapes

b) To compare the above results with Finite Element Analysis & Experimental Modal Analysis

1.4 PROJECT SCOPES

To achieve the aforementioned objectives, a crankshaft from a Daihatsu Mira EJDE, 3 cylinders 4 stroke engine will be used. This project consists both experimental and simulation, which are:

Experimental

a) Building the test rigs

b) Mounting the crankshaft and sensors to the test rigs

c) Analyzing using PulseLab and ME Scope

Simulation

a) Modeling using Solidworks CAD Software

b) Importing into ALGOR for modal analysis testing

1.5 CHAPTER OUTLINE

Chapter 1 described the purpose of finding the natural frequency and the mode shapes on the engine crankshaft using finite element analysis and experimental modal analysis. This chapter also defines the problem associated with crankshaft during operating and how to overcome these problems.

Chapter 2, the literature reviews give an insight on how to perform such analysis, whether experimentally or simulation, using various method on various things such as car chassis, fun kart chassis and also on tires. The study on basic and fundamental of experimental modal analysis and also finite element modeling by previous researchers is important and it is the key to successfully complete this research.

Chapter 3, the methodology, will explains all the procedures and methods used before, during and after test have been done for both experimental and simulation. All the apparatus and method used in experimental and simulation, respectively, are shown.

Chapter 4 provides all the result from experimental and simulation. Comparison of results between experimental and simulation are shown. All the differences are discussed thoroughly in this chapter.

Lastly, in chapter 5, the results on both experimental and simulation are summarized. Recommendations on future research for engine crankshaft are included.

1.6 GANTT CHART

The purpose of a gantt chart is to display the duration together with work implementation. Gantt chart for final year project 1 and 2 can be referred in Appendix A and B.

CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

In real life application and testing, experimental modal analysis is a crucial and accurate method to find the natural frequency and mode shape, but it is a very time consuming method especially on a large object such as a bridge or a steel tower like The Eiffel Tower in France. Finding these objects mode shapes and natural frequency can take weeks or even months to complete. This is where Finite Element Analysis comes in handy.

Finite Element Analysis or better known as FEA has been developed in 1943 by Richard Courant, who used the Ritz method of numerical analysis and minimization of variational calculus to obtain approximate solutions to vibration systems. From the engineering side, the finite element analysis originated as the displacement method of the matrix structural analysis, which emerged over the course of several decades mainly in British aerospace research as a variant suitable for computers. By late 1950s, the key concepts of stiffness matrix and element assembly existed essentially in the form used today and NASA issued request for proposals for the development of the finite element software NASTRAN in 1965.

2.2 EXPERIMENTAL MODAL ANALYSIS (EMA)

Modal Analysis, or more accurately Experimental Modal Analysis, is the field of measuring and analysing the dynamic response of structures and or fluids when excited by an input. Examples would include measuring the vibration of a car's body when it is attached to an electromagnetic shaker, or the noise pattern in a room when excited by a loudspeaker. Classically this was done with a SIMO approach, Single Input, Multiple Output, that is, one excitation point, and then the response is measured at many other points. However in recent years MIMO has become more practical.

Typical excitation signals can be classed as impulse, broadband, swept sine, chirp, and possibly others. Each has their own advantages and disadvantages. The analysis of the signals typically relies on Fourier analysis. The resulting transfer function will show one or more resonances, whose characteristic mass, frequency and damping can be estimated from the measurements.

2.2.1 Related Technical Paper/Journal on EMA

2.2.1.1 Modal Analysis of Tire In-Plane Vibration

This paper [1] presents experimental modal analysis on experimental and theoretical analysis of non-rotating tire for two boundary conditions. The tire is excited using impact hammer and the responses are measure in tangential and radial direction with accelerometers.

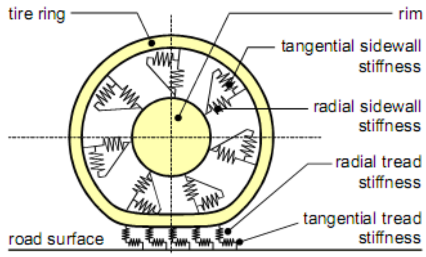

Figure 2.1: Model of a tire used

Source: Peter W.A. Zegelaar 1997

2.2.1.2 Experimental Modal Analysis

This paper [2] presents the fundamental of experimental modal analysis. The investigation are been done using impact hammer and stroke hammer. The objective is to investigate the relation of modes in modal analysis.

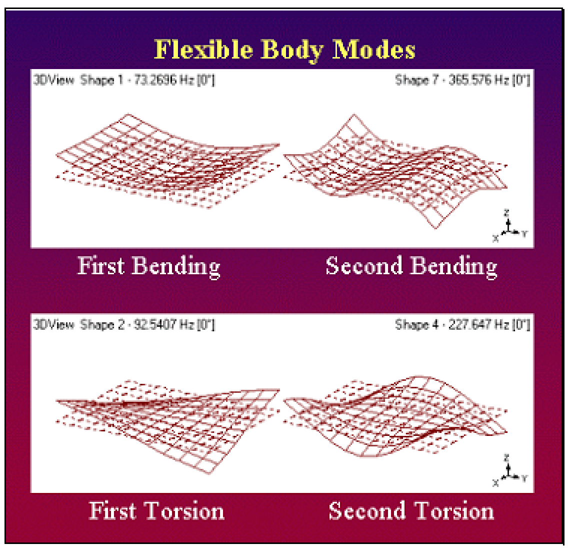

Figure 2.2: Flexible body modes

Source: Brian J. Schwarz & Mark H. Richardson 1999

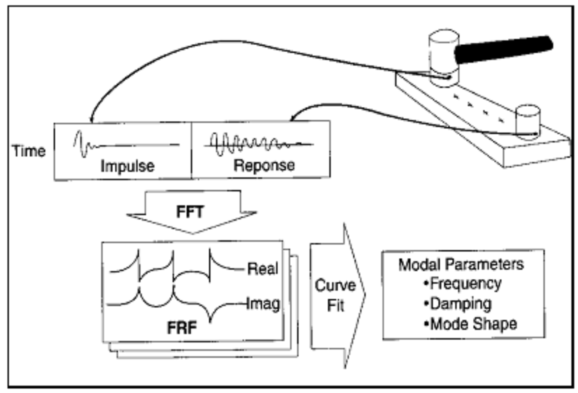

Figure 2.3: Impact testing

Source: Brian J. Schwarz & Mark H. Richardson 1999

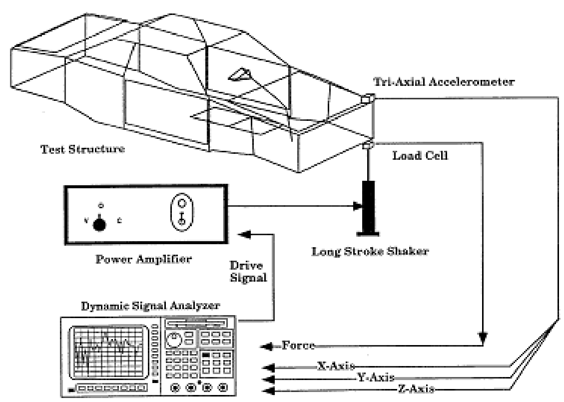

Figure 2.4: Shaker test setup

Source: Brian J. Schwarz & Mark H. Richardson 1999

2.2.1.3 New Solutions in Experimental Modal Analysis of Mechanical Structures

This paper [3] purposes new types of solution of problems which are essential for modal analysis test engineers. The investigation is related to automated modal analysis which is under development in many laboratories worldwide.

2.2.1.4 Modified ERA Method for Operational Modal Analysis in The Presence of Harmonic Excitations

This paper [4] is more of a backup method if operational or experimental modal analysis failed to identify the modal parameters properly when harmonic inputs are close to an eigenfrequency of the system. Modified ERA on the other hand was able to compute modal parameters of a structure even when the harmonic excitation frequencies are close to the natural frequency.

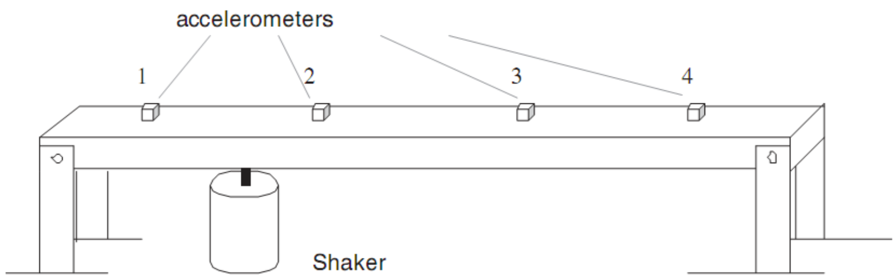

Figure 2.5: Experimental set-up for a beam

Source: Prasenjit Mohanty, Daniel J. Rixen 2004

2.2.1.5 Experimental and Numerical Power Unit Vibration Analysis

Internal Combustion Engine is known to generate lots of vibration during operation. This paper [5] investigate all types of vibration occurred using experimental and also numerical method.

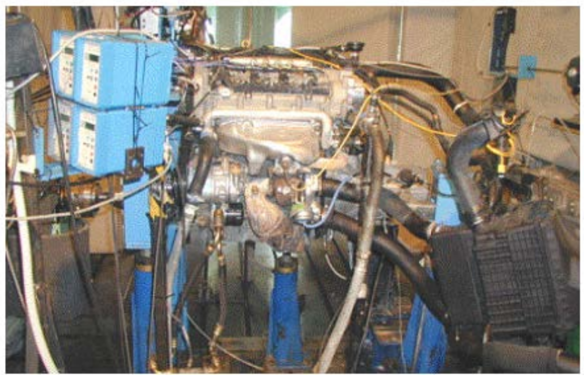

Figure 2.6: Engine on the test rig

Source: Borislav et al 2005

2.2.1.6 Experimental Modal Analysis of Three Small-Scale Vibration Isolator Models

Vibration isolators are the effective and a very cost effective way to reduce vibration. Investigation is done on some vibration isolators to determine their effectiveness in isolating vibration exerted. Typical configuration of vibration isolators are single-stage and two-stage. The author stated [6] that this investigation consists of two single-stage isolators and one two-stage isolator. The first comprises of two steel plates and one rubber element, the second one consists of two steel plates and four rubber element while the third one consists of three steel plates and eight rubber element.

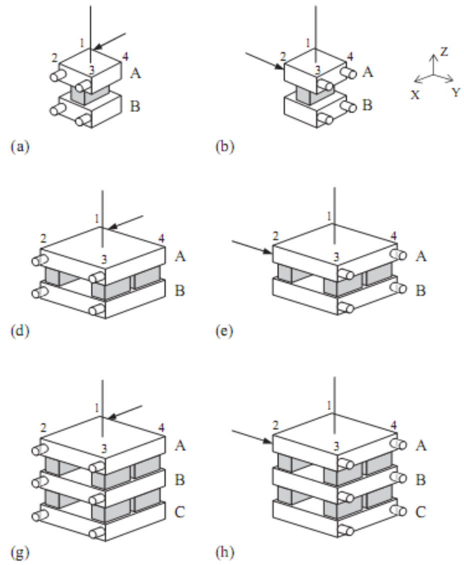

Figure 2.7: Vibration isolator’s configurations

Source: J.A. Forrest 2005

2.2.1.7 Dynamics Modal Testing of Fun Kart Chassis

This paper [7] investigates more on dynamic modal analysis testing on a fun kart chassis. Their using impact hammer and electromagnetic shaker in determining the mode shapes for the fun kart chassis.

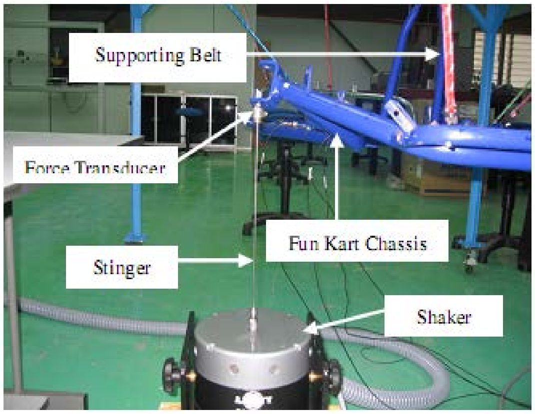

Figure 2.8: Experimental set-up

Source: M.S.M.Sani and N.Nurba 2007

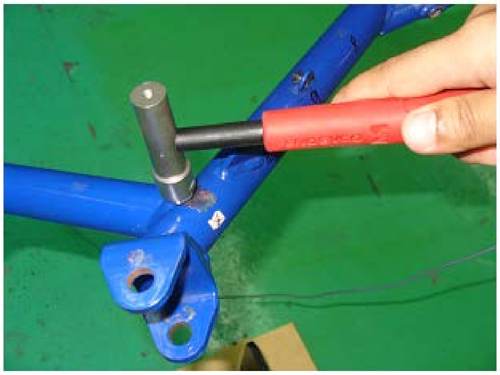

Figure 2.9: Excitations using impact hammer

Source: M.S.M.Sani and N.Nurba 2007

2.3 FINITE ELEMENT ANALYSIS (FEA)

Finite element analysis (FEA) is a computer simulation technique used in engineering analysis. It uses a numerical technique called the finite element method (FEM). Development of the finite element method in structural mechanics is usually based on an energy principle such as the virtual work principle or the minimum total potential energy principle.

The finite element analysis from the mathematical side was first developed in 1943 by Richard Courant, who used the Ritz method of numerical analysis and minimization of variational calculus to obtain approximate solutions to vibration systems. From the engineering side, the finite element analysis originated as the displacement method of the matrix structural analysis, which emerged over the course of several decades mainly in British aerospace research as a variant suitable for computers. By late 1950s, the key concepts of stiffness matrix and element assembly existed essentially in the form used today and NASA issued request for proposals for the development of the finite element software NASTRAN in 1965.

2.3.1 Related Technical Paper/Journal on FEA

2.3.1.1 Finite Element Modeling, Correlation and Model Updating of Stiffened Plate

This paper [8] looks into modal parameter extraction of aluminum plate using finite element analysis and then, it will be correlated with experimental results. A finite element model of an aluminum plate with and without ribs were developed and analyzed.

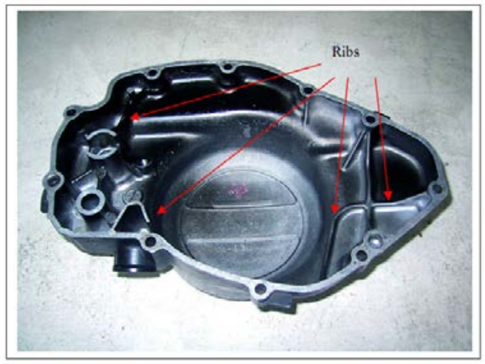

Figure 2.10: Sample of aluminum plate used

Source: Mazlan Zubair, Norsham Amin and Roslan Abd Rahman 2003

2.3.1.2 Finite Element Modeling of Silo Based On Experimental Modal Analysis

Silo which normally being seen built scorching high normally tends to ovaling especially during heavy wind, storm or earthquakes. This paper [9] presents a finite element model of a silo and it is validated with experimental modal analysis. The model will be reduced to two-dimension, so it can be coupled to a two-dimensional wind flow. The objective is to create fluid structure interaction calculations that aim to predict the onset flow velocity.

Figure 2.11: Silo tower

Source: D. Dooms, G. Degrande, G. De Roeck, E. Reynders 2007

2.4 PAPER OF INTERESTS

2.4.1 Modal Analysis and Experiments for Engine Crankshaft

This paper [10] is about experiment and simulation on a four and six cylinder engine. Using impact hammer in the experiment. The crankshaft is modeled and analyzed as a beam element in simulation. Up to 8 mode shapes is observed.

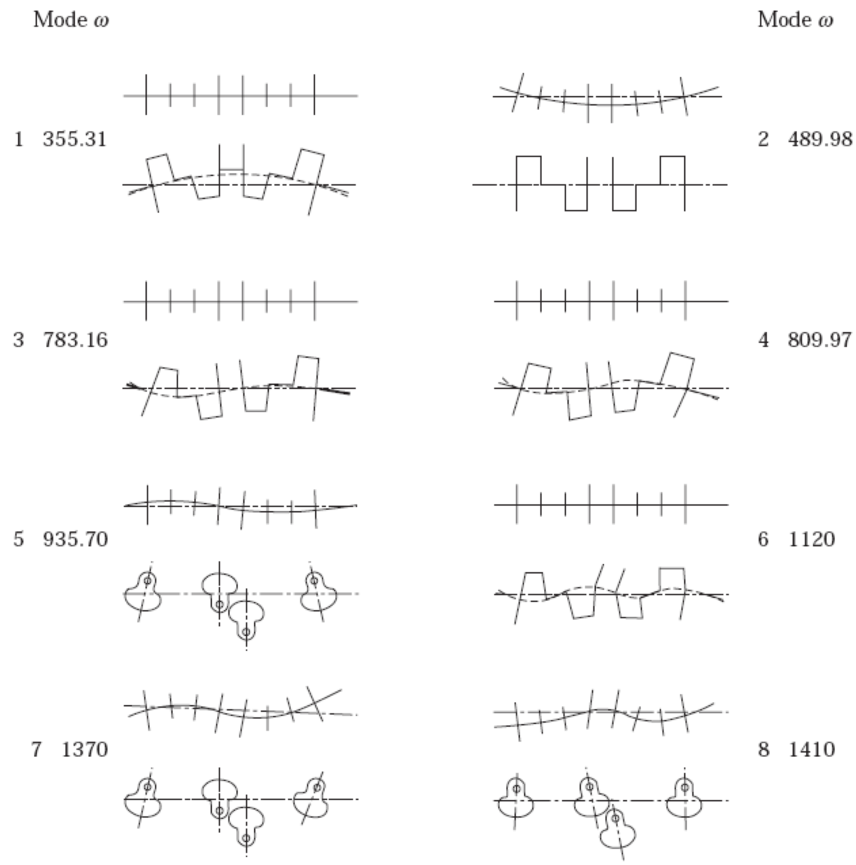

Figure 2.12: Mode shapes and frequency of a four cylinder crankshaft

Source: Y.Kang, G. J. Sheen, M. H. Tseng 1997

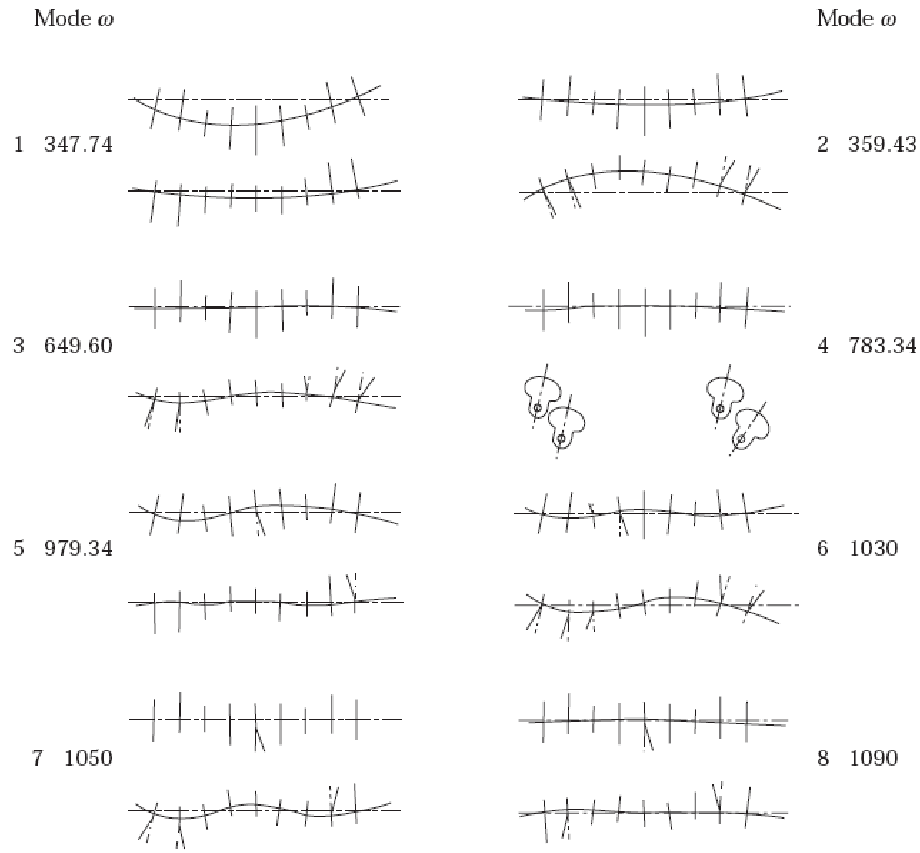

Figure 2.13: Mode shapes and frequency of a six cylinder crankshaft

Source: Y.Kang, G. J. Sheen, M. H. Tseng 1997

CHAPTER 3

METHODOLOGY

3.1 INTRODUCTION

This chapter will mostly describes all the procedures used before, during and after for both experimental and simulation method. For experimental method, PulseLab software is used to determine and captured all the signals detected by the sensors. Then, Me'scopeVes software is used to translate all the captured signals and further determine the natural frequency. Solidworks and Algor software is used for simulation methods. Solidworks are used mostly to develop the crankshaft model in CAD and then Algor is used to simulate and analyzed the crankshaft modal analysis and mode shapes. In simulations, precautions for mesh generated and the location for boundary conditions need to be taken into absolute consideration to obtain accurate results.

3.2 PROJECT FLOW CHART

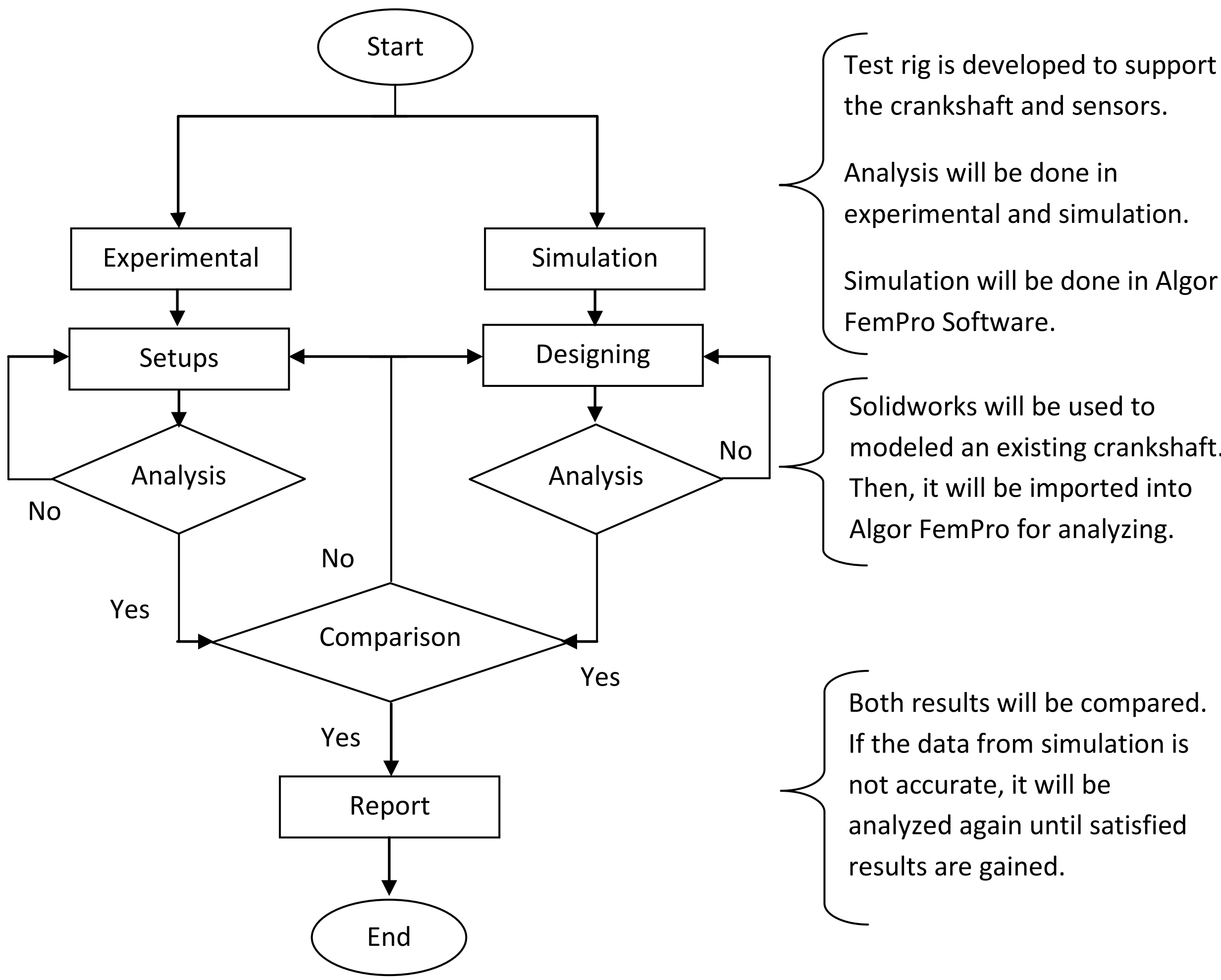

Figure 3.1: Project flow chart

3.3 PROJECT METHODOLOGY

Rule of thumb, before beginning any project, extensive research regarding of the title needed to be done. In order to finish this project in the given time, flow chart as in figure 3.1 is used as guidelines.

This project is done on both simulation and experimental. In order to perform the simulation analysis, the crankshaft is drawn using CAD program such as Solidworks by Dassault Systems. The dimension to modeled the crankshaft in CAD program is taken directly from the one used in experimental. The dimension is taken as accurate as possible to assure that the result from simulation is correlated with experimental data.

After the crankshaft has been modeled (figure 3.2, 3.4), it is imported into Algor Fempro (figure 3.3) to further analyze the crankshaft modal analysis and mode shapes. Careful consideration need to be taken especially on mesh density, as a coarse mesh will generally give poor results. All the data generated by Algor is recorded and stored for further comparison.

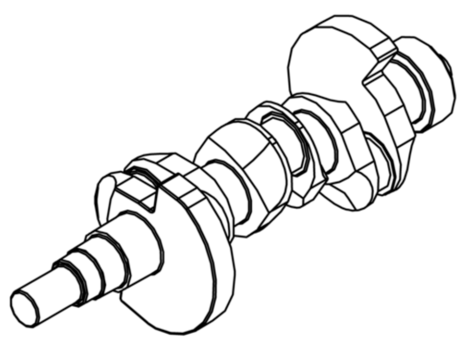

Figure 3.2: Rendered modeled of a crankshaft

After the crankshaft has been modeled, it is imported into Algor Fempro to further analyze the crankshaft modal analysis and mode shapes. Careful consideration need to be taken especially on mesh density, as a coarse mesh will generally give poor results. All the data generated by Algor is recorded and stored for further comparison.

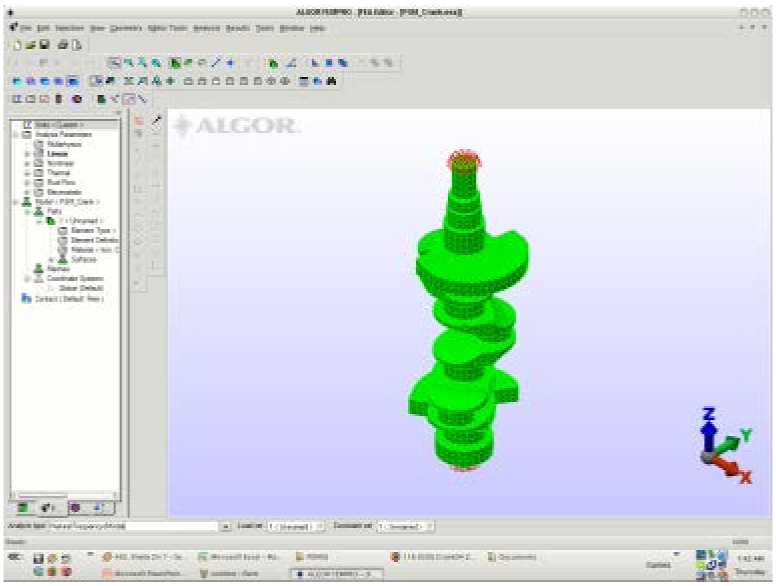

Figure 3.3: Algor Fempro

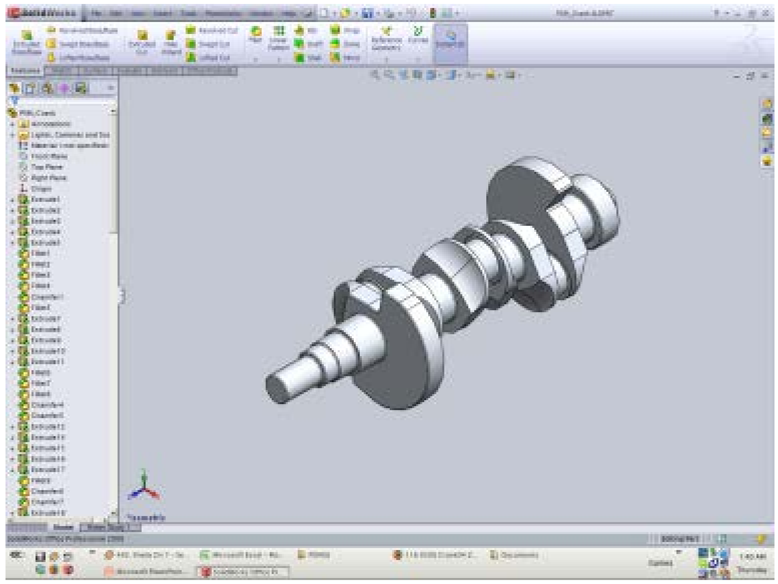

Figure 3.4: Solidworks

Y. Kang [10] stated that if an engine crankshaft is modeled by beam element, the result of analysis are somewhat questionable compared to a solid modeled. Since the crankshaft is modeled in Solidworks with fine accuracy, Algor FEMPRO will mesh the modeled with solid element.

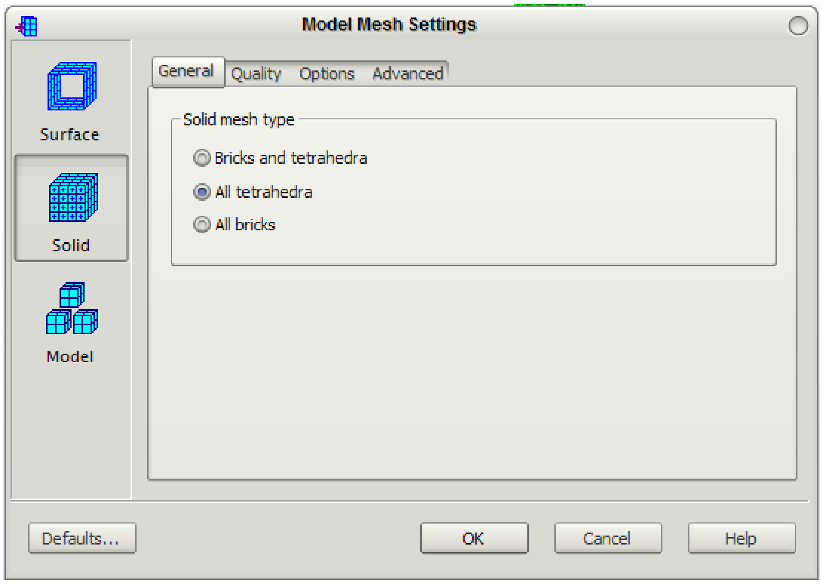

Figure 3.5: Algor mesh solid setups

In experimental, a crankshaft, a test rig and a sensor are needed. Data acquisition system (DAS) is also used in order to obtain the result from the sensors. The DAS used for this experimental test is 8 channels FFT Analyzer with PulseLab software. Then, it will be paired to a computer with Me'scopeVes software (figure 3.6) to further analyze it into graph and any other output that can be used for comparison with simulation.

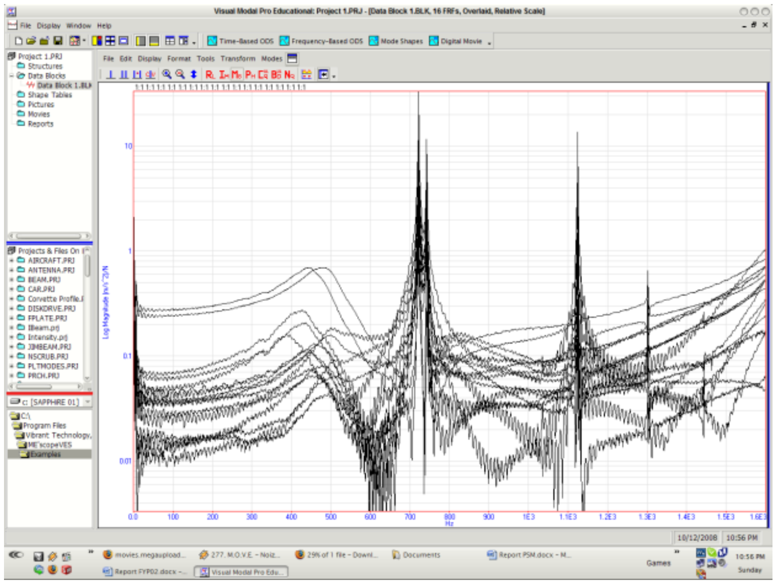

Figure 3.6: Me’scopeVes software

Figure 3.7: Experimental test rig

Figure 3.8: Crankshaft mounted on the test rig

Figure 3.9: Tri-axial accelerometer mounted on the crankshaft

Figure 3.10: Impact hammer used in experimental

Figure 3.11: The data acquisition system

The experimental setup is shown on figure 3.7. The crankshaft is mounted using ropes. The tri-axial accelerometer sensor, which sense any signal or input in X, Y and Z-direction is mounted on the crankshaft at a designated place. The crankshaft will be hit using an impact hammer at 16 different points. The vibration occurs after the impact will be recorded using data acquisition system with Pulselab software. Data from pulselab is compiled and then it will be exported into Me'scopeVES software. From here, the signal taken from the sensor will be calculated and the natural frequency from mode 1 until mode 5 can be determined.

3.4 LIST OF APPARATUS

Below are the list of apparatus used for this experimental and simulation.

Table 3.1: List of apparatus

No	Apparatus	Function
1	8 Channel FFT Analyzer Model: PULSE Type 3560 C	Used to collect time data and convert it to the FRF measurement. Then, responses will be displayed in the computer.
2	Computer with PULSE-Lite software version 10.2, ME' Scope version 4.0, Solidworks 2008 and Algor FemPro V21	PULSE-Lite is used to display the collected data, ME' Scope is used to simulate or analyze the data converted from the analyzer. Solidworks is used to create a CAD modeled of the crankshaft and Algor is used in simulation for modal analysis.
3	Tri-Axial Accelerometer Model: Bruel & Kjaer Type 4506B	Used to measure the response signal from each DOF from impact hammer test. Measurement includes 3-axis (X,Y,Z).
4	Impact Hammer Model: ENDEVCO Type 2302-10	Used to impact all the DOF's point on the crankshaft. The tip on the impact hammer is plastic.

CHAPTER 4

RESULT AND DISCUSSION

4.1 INTRODUCTION

This chapter provides the result of all the analysis. Comparison between simulation methods of modal analysis is displayed and discussed. The differences and their way to overcome it are also been discussed.

4.2 RESULT COMPARISON

The results, both in natural frequency and mode shapes are fully shown in this section.

4.2.1 Experimental Results

The experimental result is taken from the FRF data in Me'ScopeVES software. Figure 4.1 shows the frequency on each points measured during experiments. The data on each 16 points measured in being overlap in one single graph on its respective axis.

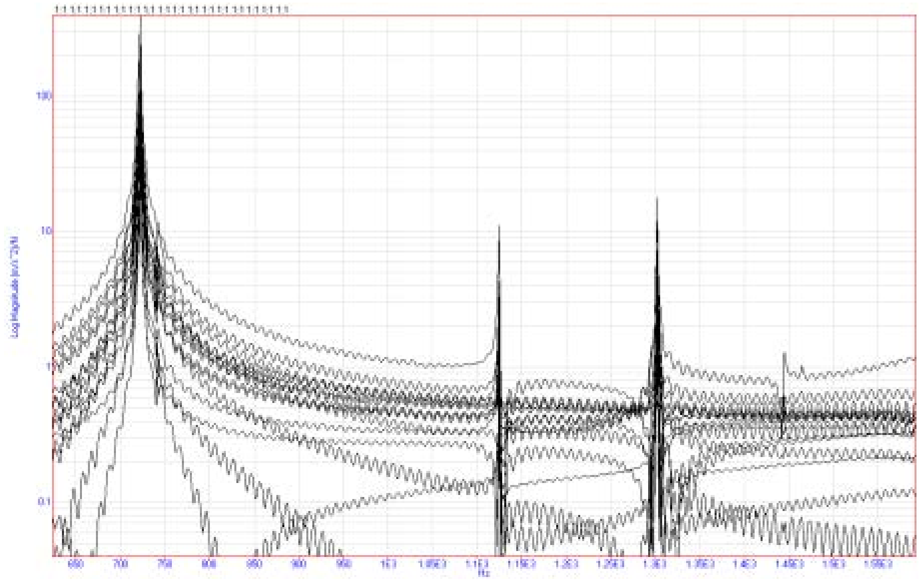

Figure 4.1a: Frequency measured on X-axis

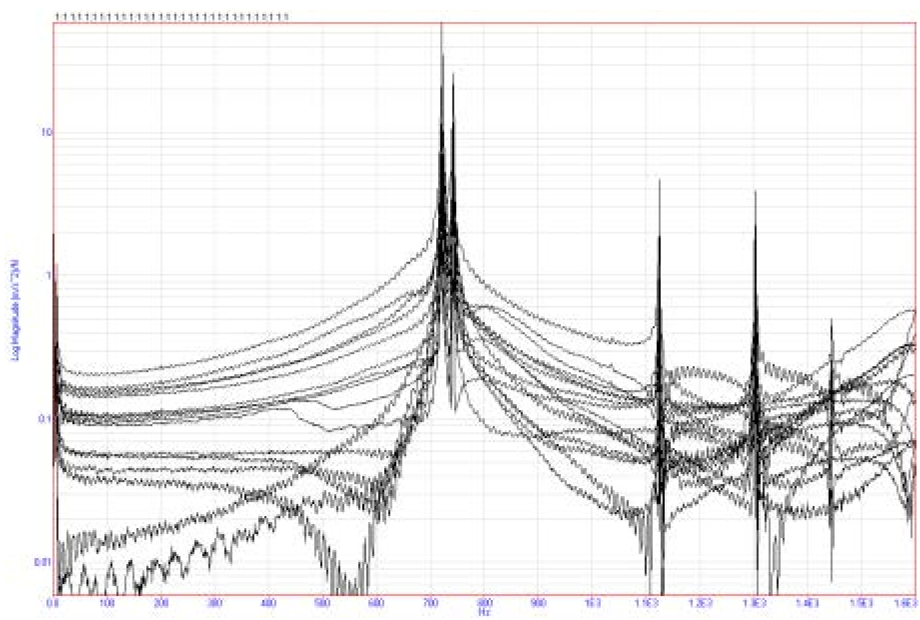

Figure 4.1b: Frequency measured on Y-axis

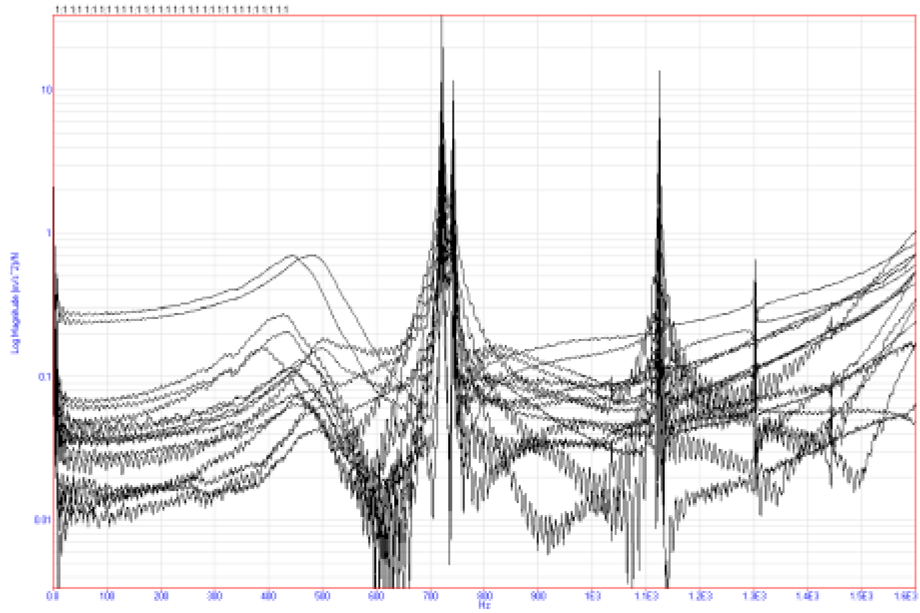

Figure 4.1c: Frequency measured on Z-axis

Table 4.1: Experimental natural frequency

Mode	Natural Frequency, Hz		
	X-axis	**Y-axis**	**Z-axis**
1	668	724	475
2	722	742	724
3	1300	850	775
4	1480	1130	1120
5	1580	1300	1320

Table 4.1 shows the experimental natural frequency obtained using Me'ScopeVES software. The natural frequency obtained is on their respective axis.

4.2.2 Simulation Results

4.2.2.1 1st Simulation

Mesh Aspect Ratio : 100%

Final Mesh Size : 6.24383 mm

Element Type : Tetrahedral

Material Type : Iron, Gray Cast ASTM A-48 Grade 60

Table 4.2: Natural frequency and deflection types for 1st simulation

Modes	Natural Frequency, Hz	Deflection Types
1	708.682	Bending
2	727.661	Bending
3	1098.9	Bending
4	1273.63	Torsion
5	1664.23	Bending

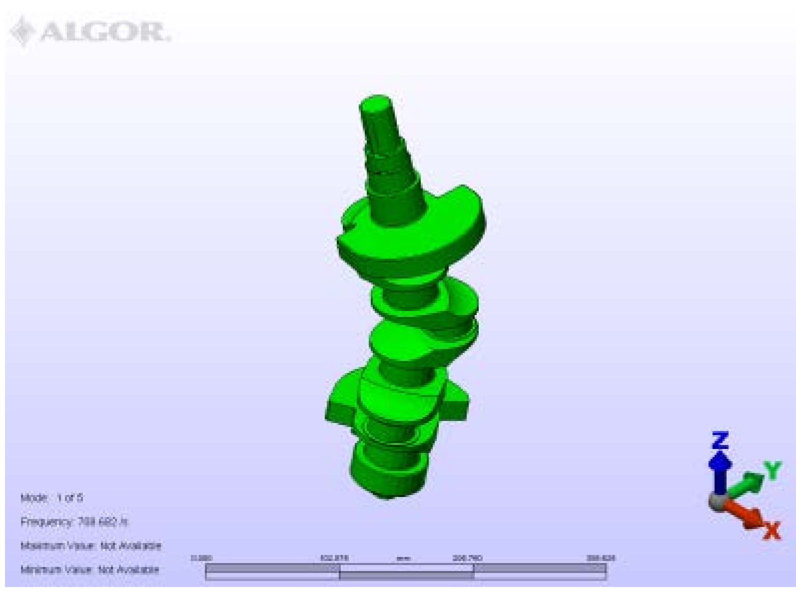

Figure 4.2: 1st Mode, Bending

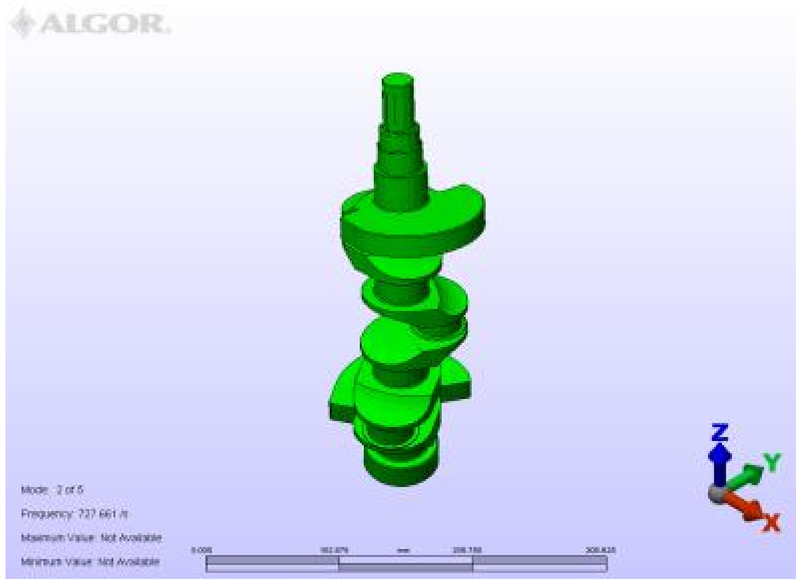

Figure 4.3: 2nd Mode, Bending

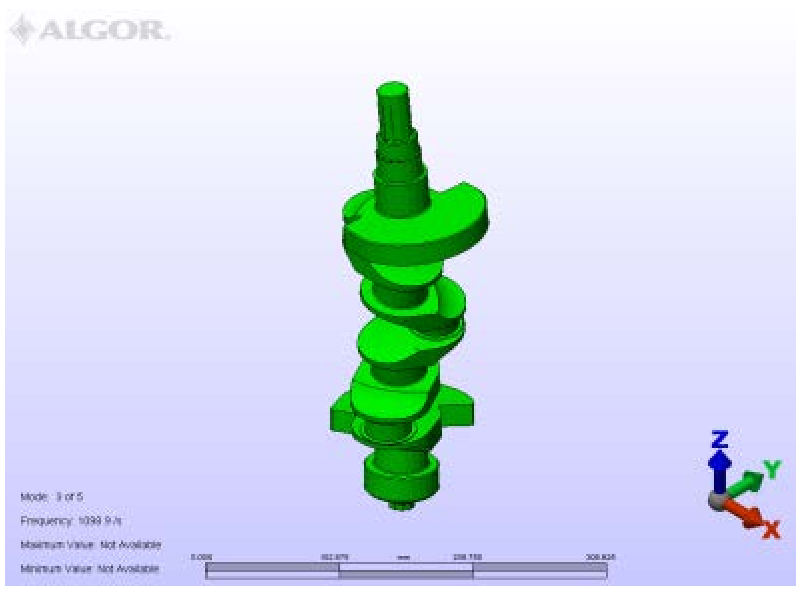

Figure 4.4: 3rd Mode, Bending

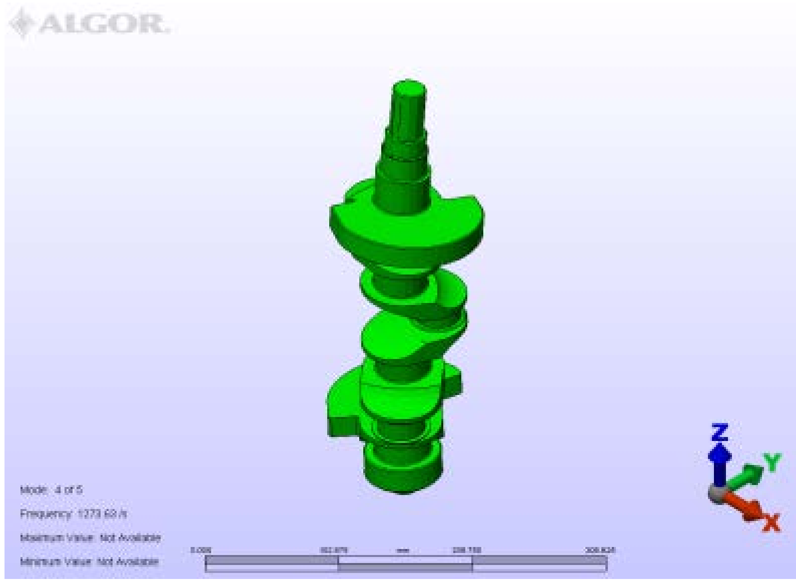

Figure 4.5: 4th Mode, Torsion

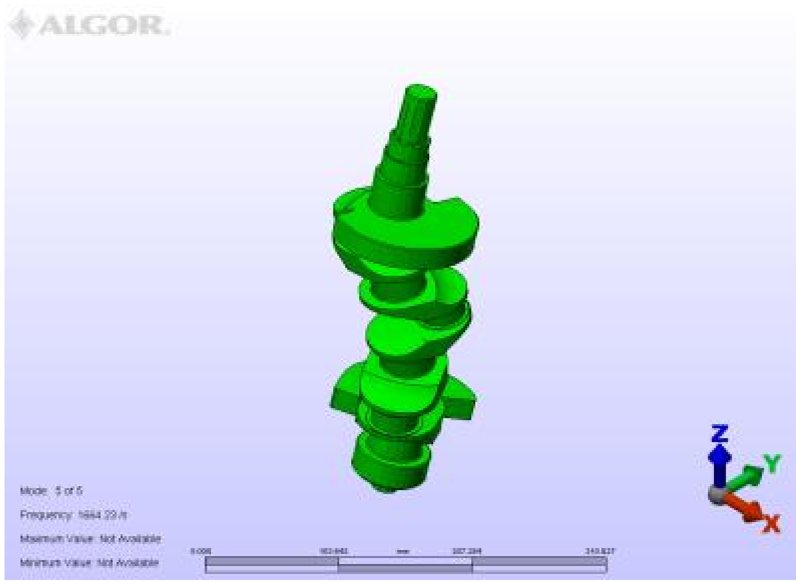

Figure 4.6: 5th Mode, Bending

The mode shapes and deflection types for the 2nd simulation until the 4th is the same as figures on previous testing, except the natural frequency for each mode is differs. Value of natural frequency for each mode is shown in the table under their designated simulation parameters. The graphical comparison of non-deflected and deflected crankshaft can be seen on appendix F.

4.2.2.2 2nd Simulation

Mesh Aspect Ratio : 80%
Final Mesh Size : 4.99507 mm
Element Type : Tetrahedral
Material Type : Iron, Gray Cast ASTM A-48 Grade 60

Table 4.3: Natural frequency for 2nd simulation

Modes	Natural Frequency, Hz	Deflection Types
1	688.494	Bending
2	707.661	Bending
3	1098.9	Bending
4	1273.63	Torsion
5	1664.23	Bending

4.2.2.3 3rd Simulation

Mesh Aspect Ratio : 60%
Final Mesh Size : 3.7463 mm
Element Type : Tetrahedral
Material Type : Iron, Gray Cast ASTM A-48 Grade 60

Table 4.4: Natural frequency for 3rd simulation

Modes	Natural Frequency, Hz	Deflection Types
1	670.5	Bending
2	689.55	Bending
3	1042.2	Bending
4	1228.8	Torsion
5	1583.9	Bending

4.2.2.4 4th Simulation

Mesh Aspect Ratio : 40%

Final Mesh Size : 2.49753 mm

Element Type : Tetrahedral

Material Type : Iron, Gray Cast ASTM A-48 Grade 60

Table 4.5: Natural frequency for 4th simulation

Modes	Natural Frequency, Hz	Deflection Types
1	653.56	Bending
2	672.08	Bending
3	1015.8	Bending
4	1210	Torsion
5	1534.9	Bending

4.3 SUMMARY

The natural frequency on each axis is compared between experimental and simulations. Detailed comparison including errors calculated is shown in table 4.7 until 4.9 in their respective axis and mesh aspect ratio. Table 4.6 shows the overall simulation natural frequency with their respective mesh aspect ratio.

Table 4.6: Summary of simulated natural frequency

Mode	Natural Frequency, Hz			
	100%	**80%**	**60%**	**40%**
1	708.682	688.494	670.5	653.56
2	727.661	707.619	689.55	672.08
3	1098.9	1069.48	1042.2	1015.8
4	1273.63	1248.92	1228.8	1210
5	1664.23	1625.1	1583.9	1534.9

Table 4.7a: Comparison on experimental and simulation data, X-axis with 100% aspect mesh ratio

Mode	Experimental Natural Frequency, Hz	Simulation Natural Frequency, Hz	Errors, %
1	668	708.682	6.09
2	722	727.661	0.78
3	1300	1098.9	15.47
4	1480	1273.63	13.94
5	1580	1664.23	5.33

Table 4.7b: Comparison on experimental and simulation data, X-axis with 80% aspect mesh ratio

Mode	Experimental Natural Frequency, Hz	Simulation Natural Frequency, Hz	Errors, %
1	668	688.494	3.07
2	722	707.619	2
3	1300	1069.48	17.81
4	1480	1248.92	15.61
5	1580	1625.1	2.85

Table 4.7c: Comparison on experimental and simulation data, X-axis with 60% aspect mesh ratio

Mode	Experimental Natural Frequency, Hz	Simulation Natural Frequency, Hz	Errors, %
1	668	670.5	0.37
2	722	689.55	4.49
3	1300	1042.2	19.83
4	1480	1228.8	16.97
5	1580	1583.9	0.24

Table 4.7d: Comparison on experimental and simulation data, X-axis with 40% aspect mesh ratio

Mode	Experimental Natural Frequency, Hz	Simulation Natural Frequency, Hz	Errors, %
1	668	653.56	2.16
2	722	672.08	6.91
3	1300	1015.8	21.86
4	1480	1210	18.24
5	1580	1534.9	2.28

Table 4.8a: Comparison on experimental and simulation data, Y-axis with 100% aspect mesh ratio

Mode	Experimental Natural Frequency, Hz	Simulation Natural Frequency, Hz	Errors, %
1	724	708.682	6.09
2	742	727.661	0.78
3	850	1098.9	15.47
4	1130	1273.63	13.94
5	1300	1664.23	28.02

Table 4.8b: Comparison on experimental and simulation data, Y-axis with 80% aspect mesh ratio

Mode	Experimental Natural Frequency, Hz	Simulation Natural Frequency, Hz	Errors, %
1	724	688.494	4.9
2	742	707.619	2
3	850	1069.48	17.81
4	1130	1248.92	15.61
5	1300	1625.1	25.01

Table 4.8c: Comparison on experimental and simulation data, Y-axis with 60% aspect mesh ratio

Mode	Experimental Natural Frequency, Hz	Simulation Natural Frequency, Hz	Errors, %
1	724	670.5	7.39
2	742	689.55	4.49
3	850	1042.2	19.83
4	1130	1228.8	16.97
5	1300	1583.9	21.84

Table 4.8d: Comparison on experimental and simulation data, Y-axis with 40% aspect mesh ratio

Mode	Experimental Natural Frequency, Hz	Simulation Natural Frequency, Hz	Errors, %
1	724	653.56	9.73
2	742	672.08	6.91
3	850	1015.8	21.86
4	1130	1210	18.24
5	1300	1534.9	18.07

Table 4.9a: Comparison on experimental and simulation data, Z-axis with 100% aspect mesh ratio

Mode	Experimental Natural Frequency, Hz	Simulation Natural Frequency, Hz	Errors, %
1	475	708.682	6.09
2	724	727.661	0.78
3	775	1098.9	15.47
4	1120	1273.63	13.94
5	1320	1664.23	26.08

Table 4.9b: Comparison on experimental and simulation data, Z-axis with 80% aspect mesh ratio

Mode	Experimental Natural Frequency, Hz	Simulation Natural Frequency, Hz	Errors, %
1	475	688.494	3.07
2	724	707.619	2
3	775	1069.48	17.81
4	1120	1248.92	15.61
5	1320	1625.1	23.11

Table 4.9c: Comparison on experimental and simulation data, Z-axis with 60% aspect mesh ratio

Mode	Experimental Natural Frequency, Hz	Simulation Natural Frequency, Hz	Errors, %
1	475	670.5	0.37
2	724	689.55	4.49
3	775	1042.2	19.83
4	1120	1228.8	16.97
5	1320	1583.9	20

Table 4.9d: Comparison on experimental and simulation data, Z-axis with 40% aspect mesh ratio

Mode	Experimental Natural Frequency, Hz	Simulation Natural Frequency, Hz	Errors, %
1	475	653.56	2.16
2	724	672.08	6.91
3	775	1015.8	21.86
4	1120	1210	18.24
5	1320	1534.9	16.28

4.4 DISCUSSION

From table 4.7 until 4.9, the comparison between simulation and experimental data for X, Y and Z axis are shown. The percentage of errors between these two data is calculated. Different mesh sizes generate different value of natural frequency, thus different percentage of errors calculated from the data.

The experiment is run on some assumptions:

a) Static working condition

b) Free suspended on each end using elastic rope

c) 16 degree of freedoms (DOF) is chosen

Simulation assumptions:

a) Free-Free boundary condition

b) Solid modeling

c) The crankshaft model is accurate and near precise with the real crankshaft

d) The material chosen in the simulation is the same as the real crankshaft

Table 4.10: Average errors between simulation and experimental natural frequency based on X-axis

Mesh Aspect Ratio, %	Range Of Errors, %	Average Errors, %
100	0.78 – 15.47	8.322
80	2 – 17.81	8.268
60	0.37 – 19.83	8.38
40	2.16 – 21.86	10.29

Table 4.11: Average errors between simulation and experimental natural frequency based on Y-axis

Mesh Aspect Ratio, %	Range Of Errors, %	Average Errors, %
100	0.78 – 28.02	12.86
80	2 – 25.01	13.066
60	4.49 – 21.84	14.104
40	6.91 – 21.86	14.962

Table 4.12: Average errors between simulation and experimental natural frequency based on Z-axis

Mesh Aspect Ratio, %	Range Of Errors, %	Average Errors, %
100	0.78 – 26.08	12.472
80	2 – 23.11	12.32
60	0.37 - 20	12.332
40	2.16 – 21.86	13.09

From table 4.10 until 4.12, the lowest average error shown is when the mesh aspect ratio is 80% (except for Y-axis). Since the mesh aspect ratio of 80% is the most frequent lowest average errors, the final simulation natural frequency will be based on mesh aspect ratio of 80%.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 INTRODUCTION

In this chapter, the results on both experimental and simulation are summarized. Recommendations on future research for engine crankshaft are included further in this chapter.

5.2 SUMMARY

At the beginning of this project, crankshaft is modeled in CAD software. In this project, the real challenge is to model the crankshaft in CAD software for computer analysis. The crankshaft need to be modeled as precisely as possible since the result for both simulation and experiment will be compared. Even a single mistake in modeling can result in significance error margin when comparing the results. Then, when computer analysis is started, some assumptions need to be taken into consideration.

In the experiment, the real crankshaft is supported by an elastic rope to simulate a Free-Free boundary conditions. 16 degree of freedoms (DOF) is selected as appropriate points for impact. The crankshaft is then hit with an impact hammer at all the DOF. The data detected tri-axial accelerometer is send to the DAS for further analyzed its frequency response. Data from will comes in three different axes, X, Y and Z. Table 4.1 shows the comparison for all three axis. After both result for simulation and experimental have been obtained, comparison and analyzing can be done.

5.3 CONCLUSIONS

Both experiment and simulation is performed to determine the crankshaft natural frequency and mode shapes. The errors calculated from table 4.6 shows a lowest error margin in coarse mesh but a higher error margin in finer mesh. Since the lowest average errors is in 80% mesh aspect ratio, the data will be used the final simulation results.

Mode shape deflections (shown in Appendix F) are varies from modes to modes. This mode shapes can happened in real life situation when the working frequencies of the crankshaft is equal to the natural frequency on each mode. For example, in 1st mode, the deflection type is bending with the natural frequency is at 653.555 /s with maximum deflection range at about 39.2891 mm.

Thorough research on working frequency and natural frequency of the crankshaft is crucial to avoid resonance during operation.

5.4 RECOMMENDATIONS

The errors margin shown in table 4.10, table 4.11 and table 4.12 clearly shows that the error varies from 2% to 25.01% for mesh aspect ratio 80%. This is because, lots of assumption is considered. Below are some recommendation that can further reduce the errors between simulation and experimental.

Simulation:

a) Using data from manufacturer such as modulus elasticity and density of material can further enhance the simulation analysis.

Experimental:

a) Using more than one tri-axial accelerometer to increase the tri-axial sensitivity on different points.

b) Adjust the sensitivity of the tri-axial accelerometer.

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APPENDIXES

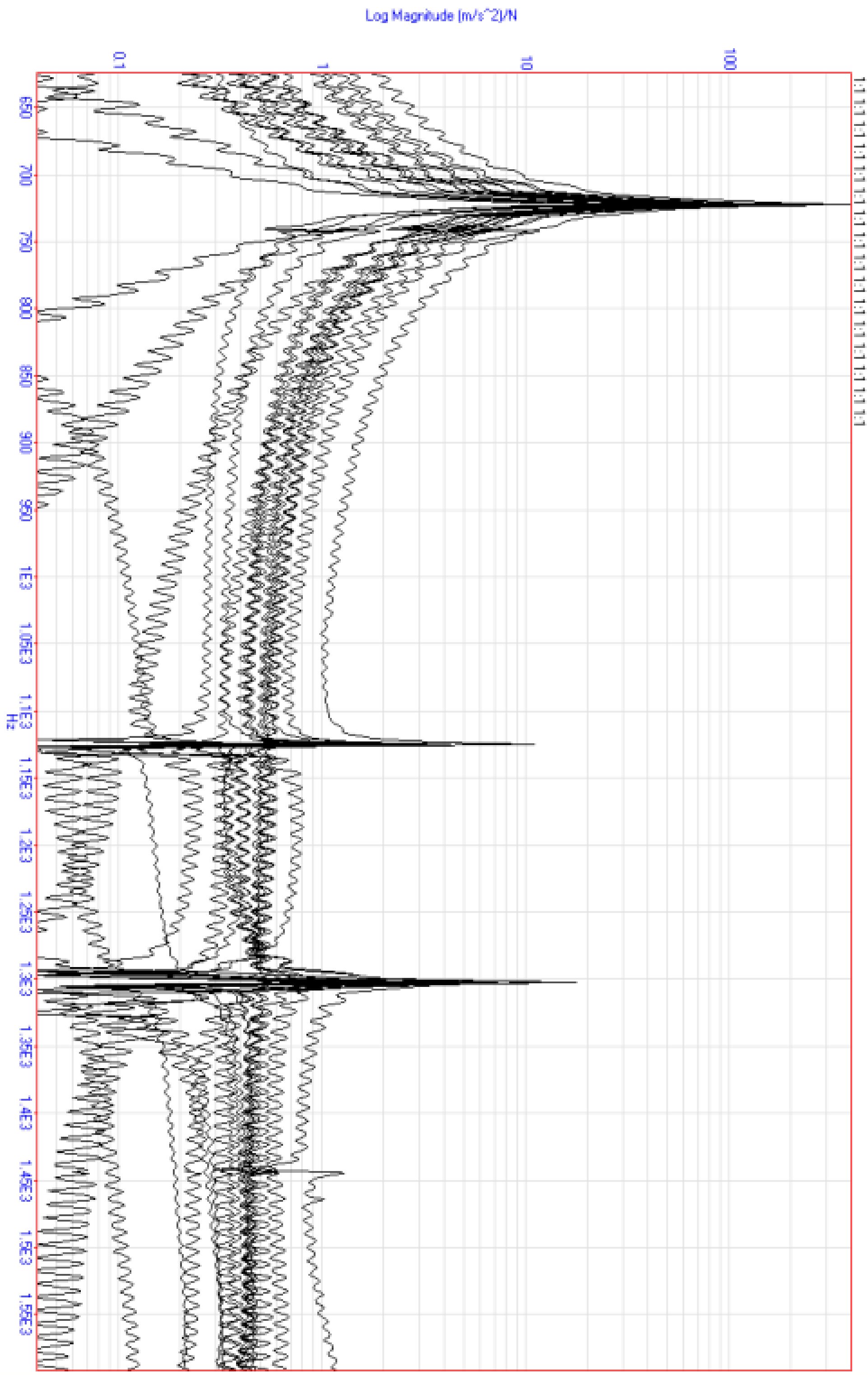
Appendix A: FYP01 Gantt Chart

Week / Activity	01	02	03	04	05	06	07	08	09	10	11	12	13	14
Project briefing by supervisor														
Identify scopes and objective														
Literature review														
Slide presentation preparation														
Proposal and report preparation														
Presentation FYP01 preparation														
FYP01 presentation														

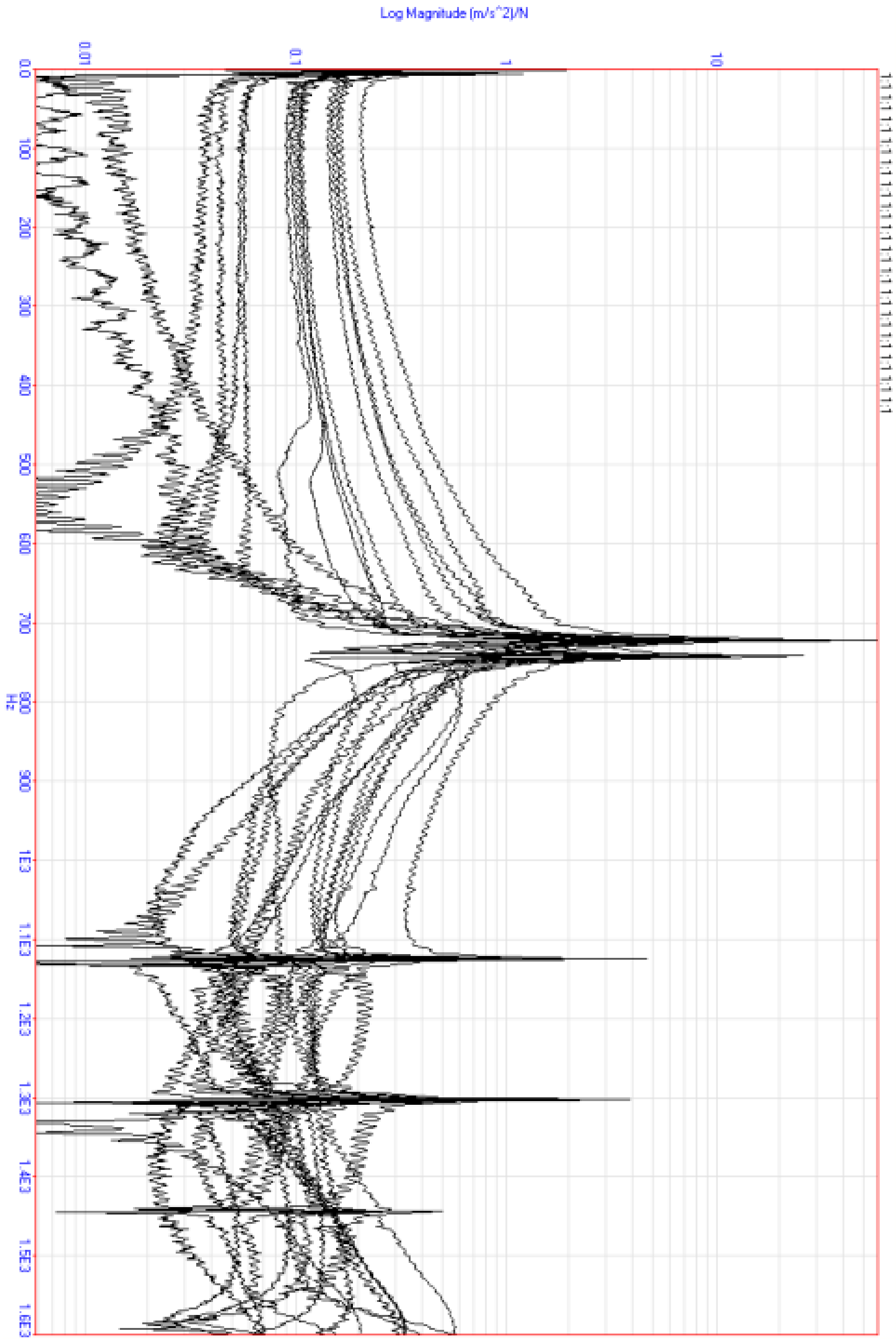
Appendix B: FYP02 Gantt Chart

Week / Activity	01	02	03	04	05	06	07	08	09	10	11	12	13	14
Experimental Analysis														
Crankshaft modeling														
Analysis (FEMPRO)														
Chapter 4: Result and Discussion														
Chapter 5: Conclusion														
Final report preparation														
Final report approval & submission														
FYP02 presentation														

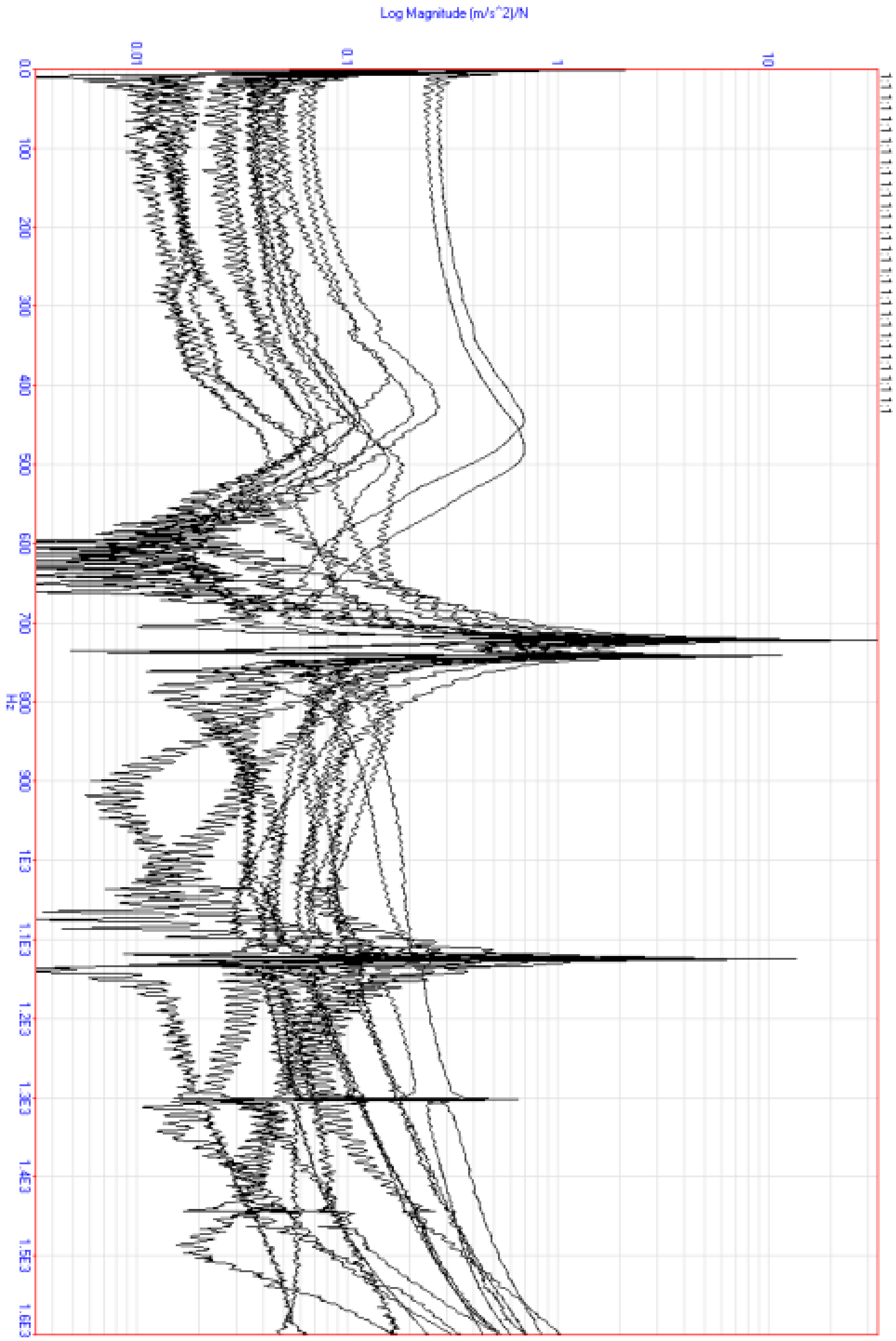
Appendix C: X-Axis Frequency Response Data



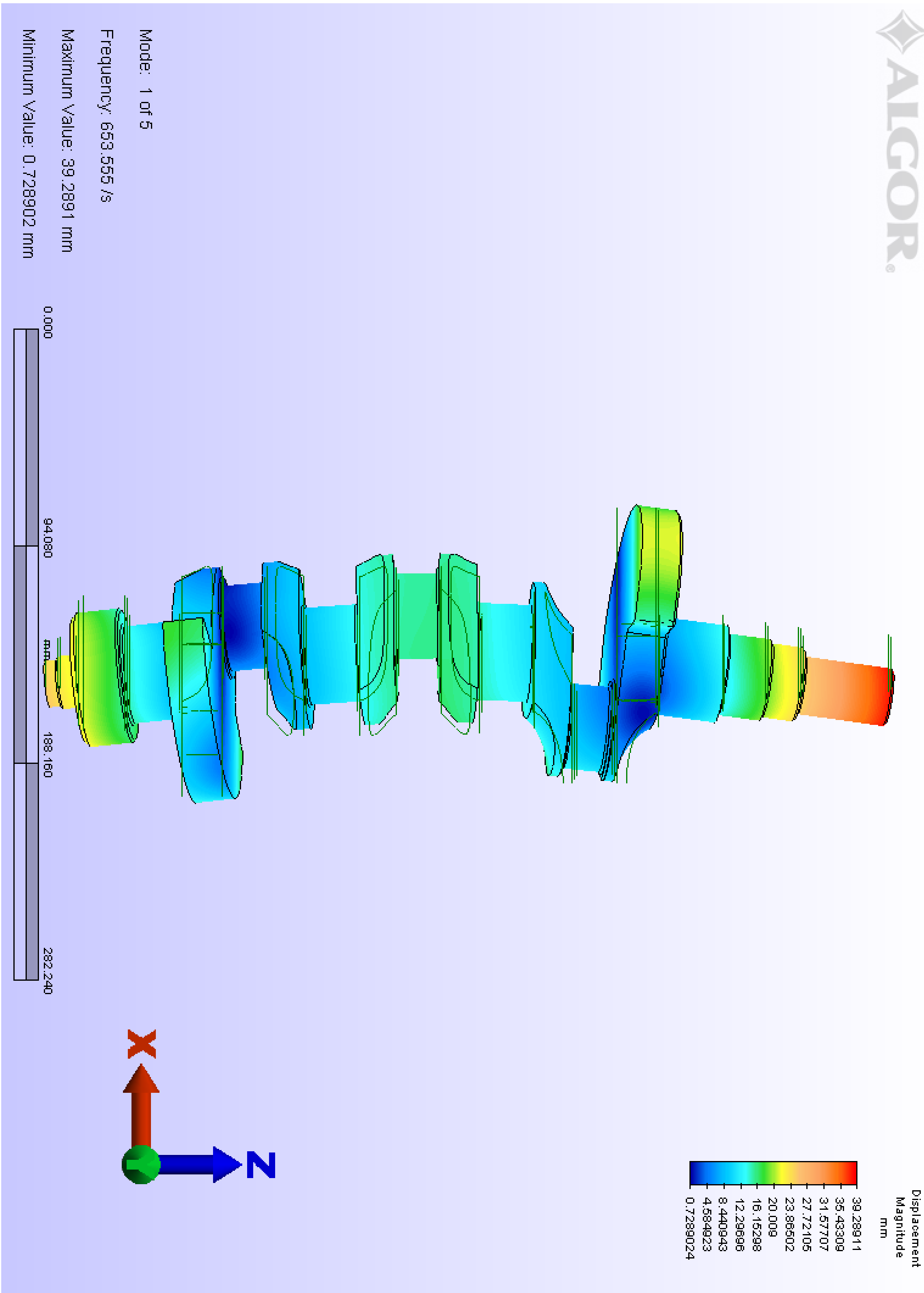
Appendix D: Y-Axis Frequency Response Data

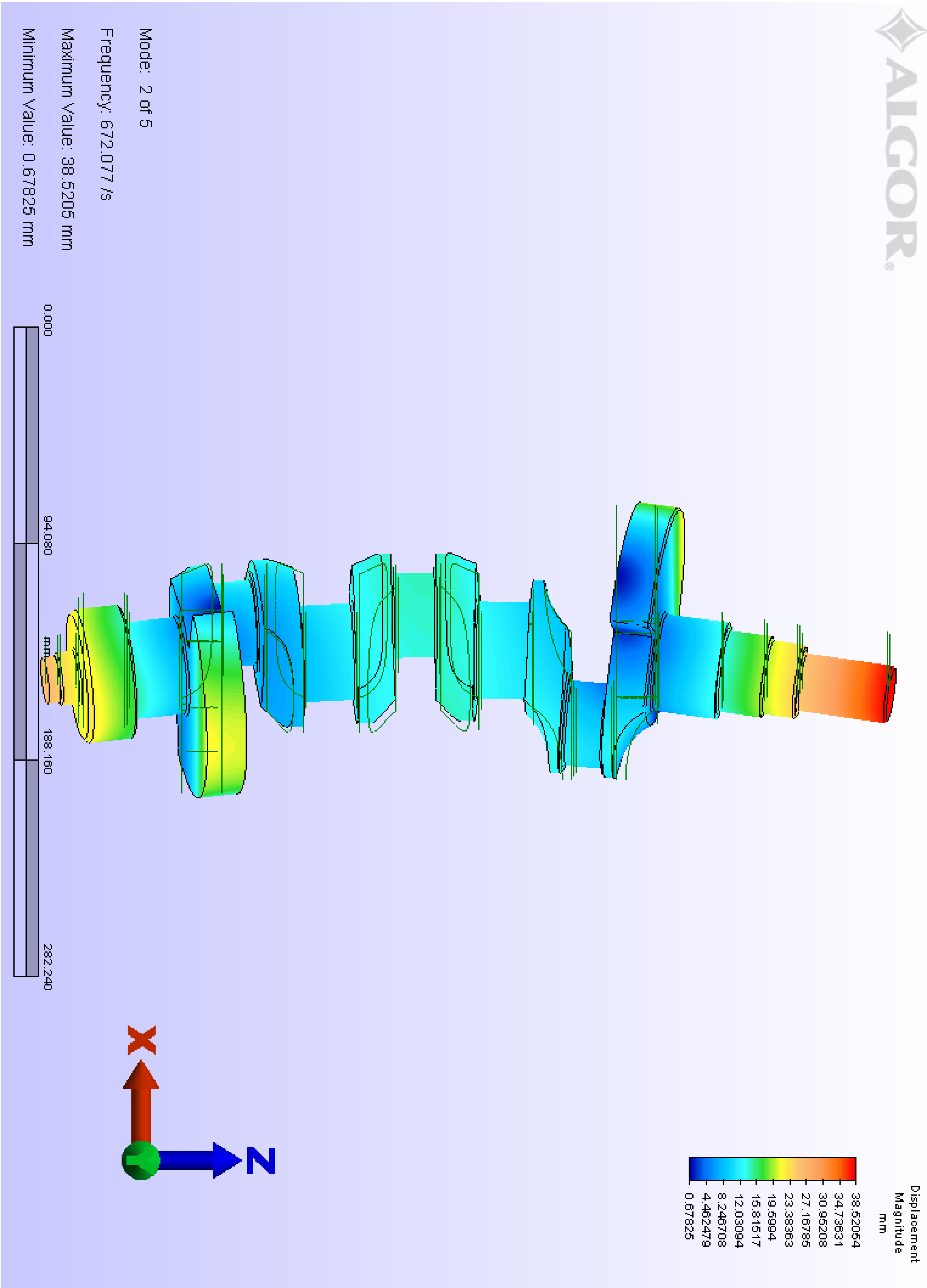


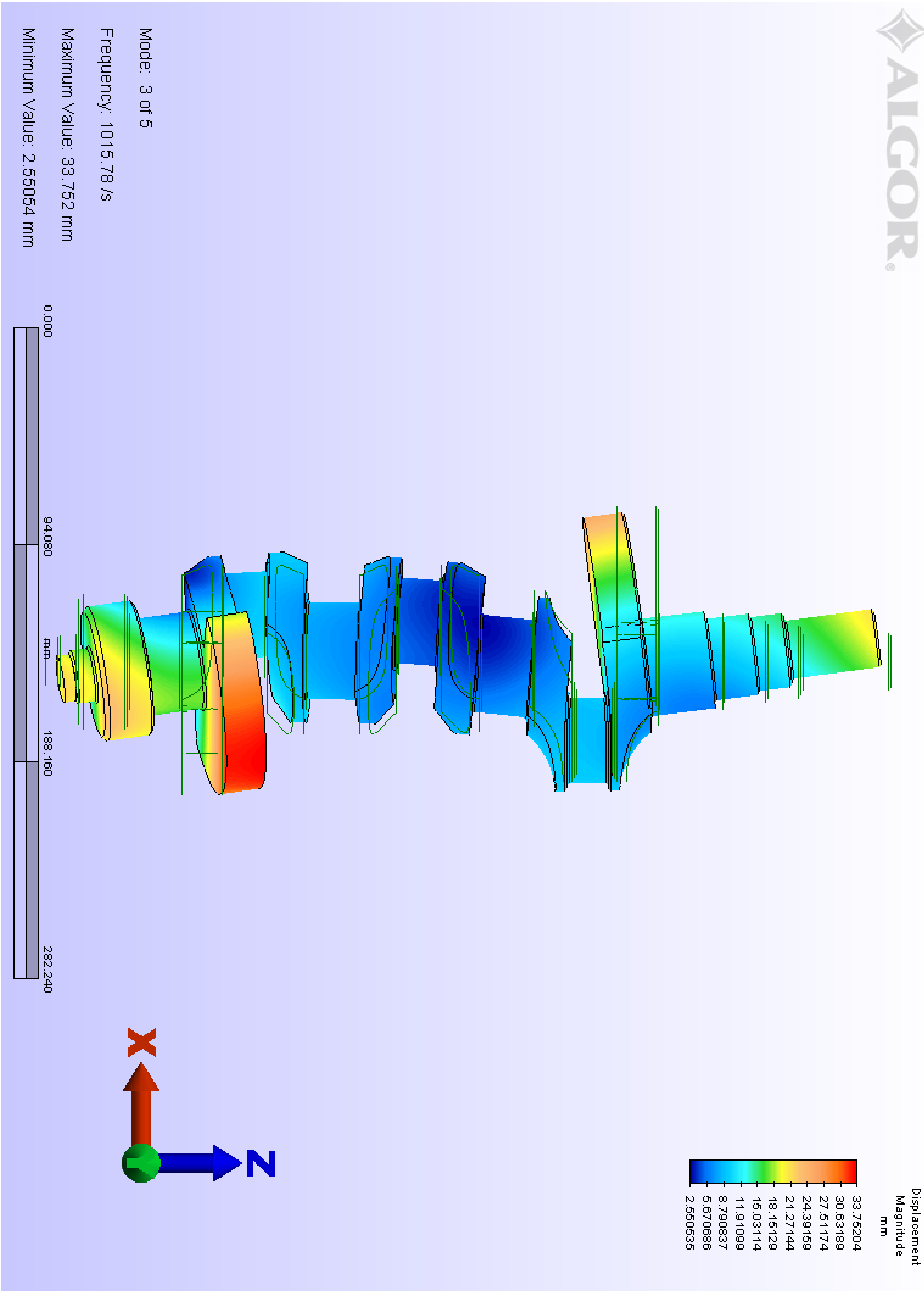
Appendix E: Z-Axis Frequency Response Data

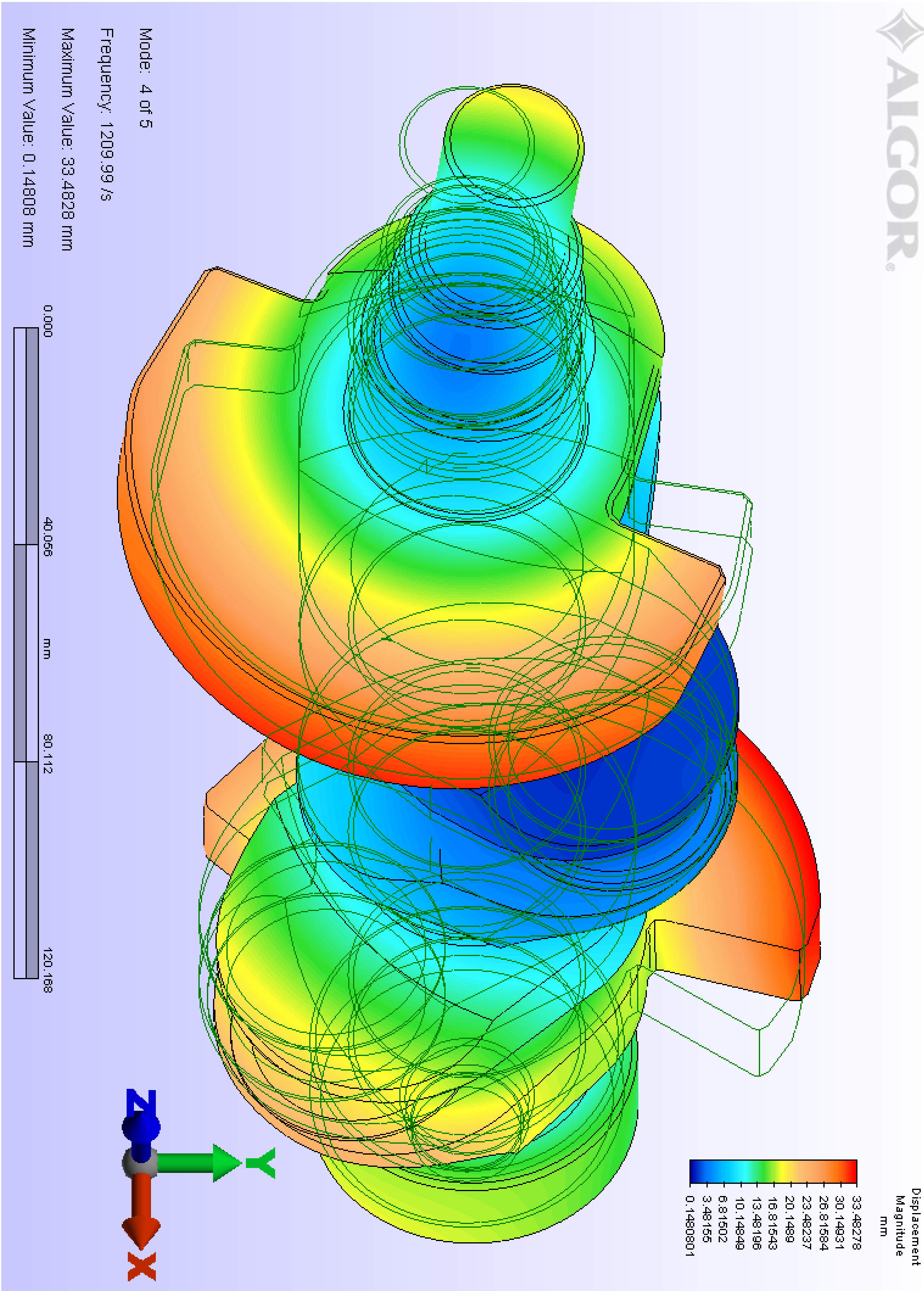


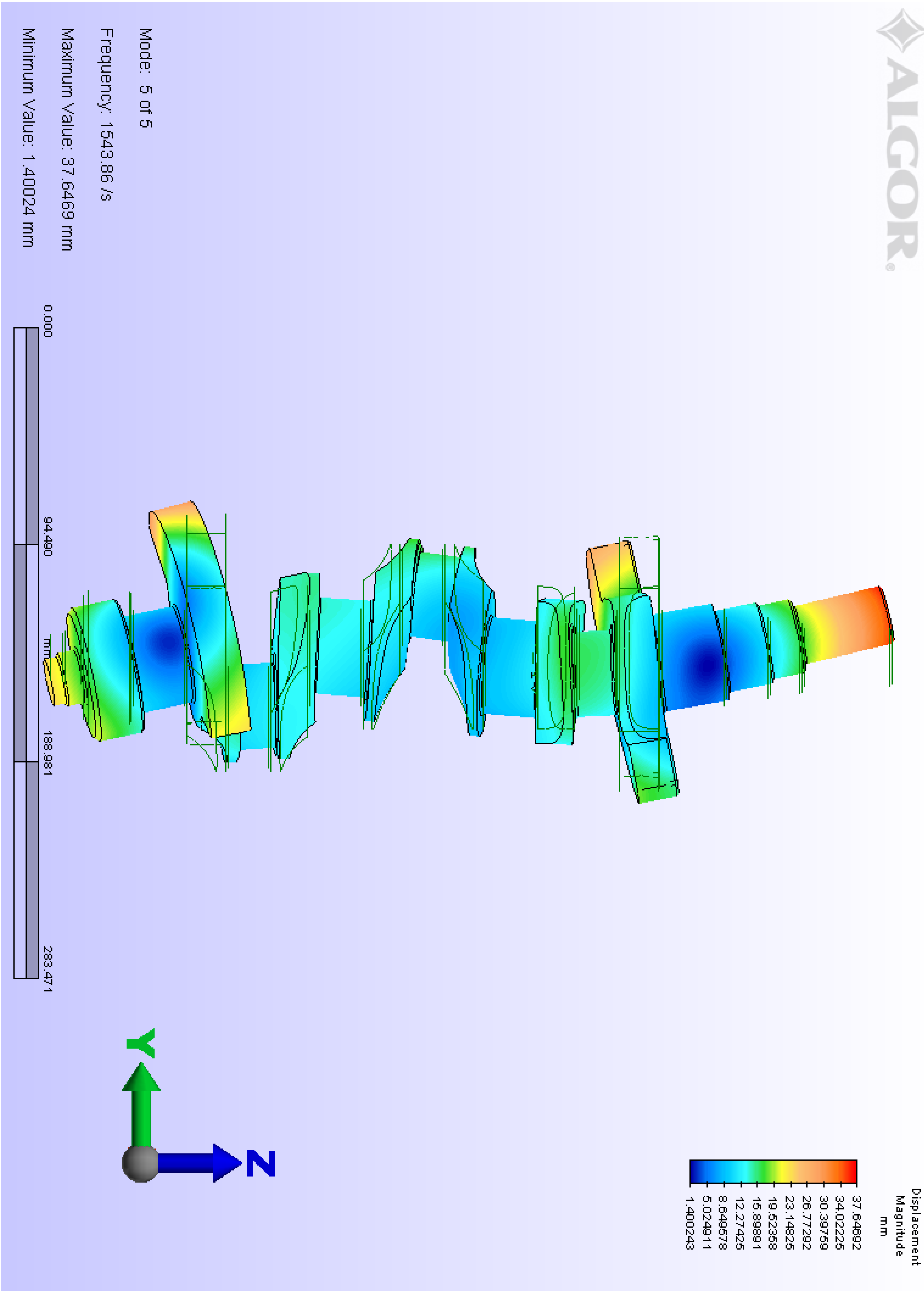
Appendix F: Graphical Comparison on Simulation Analysis




ALGOR
Mode: 2 of 5
Frequency: 672.077 /s
Maximum Value: 38.5205 mm
Minimum Value: 0.67825 mm
0.000
94.080
188.160
282.240
Displacement
Magnitude
mm
38.52054
34.73631
30.95208
27.16785
23.38363
19.5994
15.81517
12.03094
8.246708
4.462479
0.67825
X
Z


Mode: 3 of 5
Frequency: 1015.78 /s
Maximum Value: 33.752 mm
Minimum Value: 2.55054 mm
0.000
94.080
mm
188.160
282.240
Displacement
Magnitude
mm
33.75204
30.63189
27.51174
24.39159
21.27144
18.15129
15.03114
11.91099
8.790837
5.670686
2.550535
X
Z


ALGOR
Mode: 4 of 5
Frequency: 1209.99 /s
Maximum Value: 33.4828 mm
Minimum Value: 0.14808 mm
0.000
40.056
mm
80.112
120.168
Displacement Magnitude mm
33.48278
30.14931
26.81584
23.48237
20.1489
16.81543
13.48196
10.14849
6.81502
3.48155
0.1480801
Z
Y
X


ALGOR
Mode: 5 of 5
Frequency: 1543.86 /s
Maximum Value: 37.6469 mm
Minimum Value: 1.40024 mm
0.000
94.490
mm
188.981
283.471
Displacement
Magnitude
mm
37.64692
34.02225
30.39759
26.77292
23.14825
19.52358
15.89891
12.27425
8.649578
5.024911
1.400243
Y
Z