EFFECT ON VIBRATION AMPLITUDE IN ALUMINIUM RECTANGULAR PLATE USING SIMULATION APPROACH

NUR NAJAA MASTURA BINTI MOHAMAD JAMIL

UNIVERSITI TEKNIKAL MALAYSIA MELAKA

EFFECT ON VIBRATION AMPLITUDE IN ALUMINIUM RECTANGULAR PLATE USING SIMULATION APPROACH

NUR NAJAA MASTURA BINTI MOHAMAD JAMIL

This report is submitted in fulfillment of the requirement for the degree of Bachelor of Mechanical Engineering (Design and Innovation)

Faculty of Mechanical Engineering

UNIVERSITI TEKNIKAL MALAYSIA MELAKA

2017

DECLARATION

I declare that this project report entitled "Effect On Vibration Amplitude In Aluminium Rectangular Plate Using Simulation Approach" is the result of my own work except as cited in the references.

Signature : ....

Name : ....

Date : ....

APPROVAL

I hereby declare that I have read this project report and in my opinion this report is sufficient in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering (Design & Innovation).

Signature	: ....
Name of Supervisor	: ....
Date	: ....

DEDICATION

To my beloved mother and father.

ABSTRACT

This study is about the effect on vibration amplitude in aluminium rectangular plate using simulation approach. The purpose of this study is to study the vibration effect of aluminium plate using simulation technique. From the analysis that has been made, the frequency value is obtained from CATIA software. The result shows that the plate have the same frequency response when load is being applied at each point on the plate which means the model is in homogeneous condition. However, this study is conducted with three conditions of model. The analysis shows that the third condition which consist of the plates with springs, rubber between the second condition's plates and base have the best result. This is proven as the percentage error is calculated from the frequency value obtained.

ABSTRAK

Kajian ini adalah mengenai kesan ke atas getaran amplitud pada plat aluminium segi empat menggunakan pendekatan secara simulasi. Tujuan kajian ini adalah untuk mengkaji kesan getaran plat aluminium menggunakan teknik simulasi. Berdasarkan analisis yang dilakukan, nilai frekuensi telah diperoleh daripada perisian CATIA. Hasilnya menunjukkan bahawa plat mempunyai tindak balas frekuensi yang sama apabila beban dikenakan pada setiap titik di atas plat yang bermaksud model berada dalam keadaan sekata. Walau bagaimanapun, kajian ini dijalankan berdasarkan model yang terbahagi kepada tiga keadaan. Analisis menunjukkan bahawa keadaan ketiga yang terdiri daripada plat berserta spring, getah antara plat dan tapak mempunyai hasil yang terbaik. Ini terbukti apabila ralat peratusan dikira dari nilai frekuensi yang diperoleh.

ACKNOWLEDGEMENTS

It is such a great pleasure and gratitude to acknowledge people whose guidance and encouragement contribute tremendously towards the completion of my final year project.

First and foremost, a huge thank you to my supervisor Dr. Mohd Azli Bin Salim for giving me this opportunity to do final year project with him, being supportive in coaching and sharing knowledge with me and guiding me throughout the entire period of this final year project.

Secondly, I want to convey my appreciation to my family and friends who were never stop to support and motivated me. Finally, I wish to thank you all of the people who gave contributed directly or indirectly in helping me to complete this final year project.

TABLE OF CONTENT

CHAPTER	CONTENT	PAGE

LIST OF FIGURES

LIST OF TABLES

LIST OF SYMBOL

f	=	Frequency
A	=	Area
CL	=	Velocity of propogation
D	=	Longitudinal stiffness
dx	=	Displacement
E	=	Modulus of elasticity
K	=	Wave number
P	=	Pressure
ε	=	Strain
μ	=	Wave length
ρ	=	Density
σ	=	Stress
ω	=	Operating frequency

CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION OF STUDY

The application of plate plays a big role which is the most commonly used structures in the industrial world. In the structured design, plate is stated only to withstand applied static loads which are inadequate for the application. Possibility of large cyclic displacement and stresses should be considered due to periodic or random time changing the forces acting on the plate's lateral surface.

Some examples for the issue are the aircraft components or stationary structures exposed to high wind velocities which random forces are expected on the plate surface, tangential fluid force on plate surface used in ship hulls, submarines or offshore structures and regular or periodic excitation forces on plates used as structural housing on rotating or reciprocating machinery like compressors and reciprocating engines.

There exist a large number of isolated frequencies at which rectangular plate will go through a large-amplitude vibration by constant time changing the forces of matching frequencies as shown in Figure 1.1 below. The sensitivity of the plate vibration to the variation of weight limitations can be moderately different from the case of a single traveling load (A. Nikkhoo et al., 2014). The related elements with each plate's natural frequency is a characteristic or mode shape.

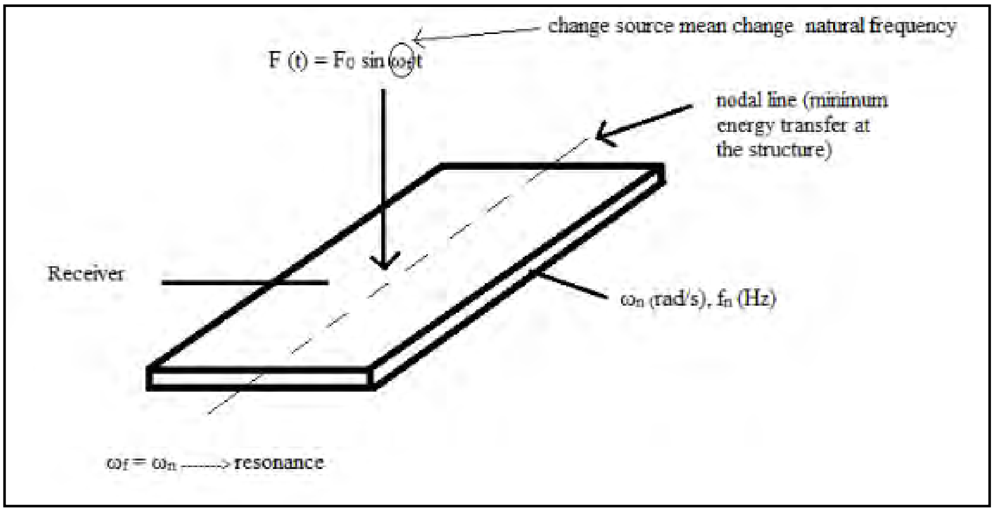


Figure 1.1: The schematic diagram of rectangular plate.

There are several steps to analyze the forced vibration. Before solving the problem, the value of natural frequency and mode shape need to be determined. Then, it can assist to analyze free vibration or forced vibration where the focus are known as amplitude of vibration.

From the previous study, a mixed method is used for the solution of free and forced vibration of rectangular plate which are Ritz method and differential quadrature method. The formulation of the problem is stated in matrix form and therefore programming is not hard. A good rate of convergence is demonstrated and the results are in good agreement with the available results even with a few numbers of Ritz terms and DQ sample points in the numerical examples presented (S.A. Eftekhari, 2011).

1.2 PROBLEM STATEMENT

The vibration analysis is a significant step for obtaining the result of frequency on the aluminium rectangular plate. The problem of the structured design is when there is matching between driving forces and the plate's natural frequencies which is known as resonance. The analysis is needed to avoid the possible resonance from occur in the application.

1.3 OBJECTIVE

The objective of this study is to study the vibration effect of aluminium plate using simulation technique.

1.4 SCOPE OF PROJECT

The scopes of this study are:

1. Conduct the simulation by using CATIA software.
2. Analyze the vibration occur on the aluminium plate.

CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

Literature review is the information gathering which already exists in the previous research or project. The purpose of literature review is to define the research topic and it consists of several information of the previous research. The information can be obtained from referring to journal, article or information from website. Relevant sources from online articles and websites are compiled and cited to complete this literature review. Hence, by review the related information, it will help to understand the objective of the research.

2.2 THEORY OF PLATE

As stated in the background of study, the application of plate is widely used in the industrial world. The plate consists of two basic theories which are Kirchhoff plate theory and Mindlin plate theory. However, the plate properties such as the Young's Modulus, density and poisson's ratio is important for the different used of plate in any application. The previous research have found that different plate properties provide a different bending stiffness (Lu et al., 2006). This means that the plate properties lead the bending wave of the plate.

2.3 FREE VIBRATION OF PLATE

Basically there are three boundary conditions of plate that will affect the amplitude vibration occur on it which are simply supported, clamped and free. The boundary condition can be mixed with each other or just the same condition. It is important to prevent from the resonance which will happen when there is matching between driving forces and the plate natural frequencies. Free vibration analysis is significant to avoid resonance between the plate and driving force system.

From the previous research, the study is about free vibration analysis with three different boundary conditions by using Bessel functions. The conditions chosen are fully simply supported, fully clamped and the last one is the mixed condition which is two opposite boundaries with simply supported and another two boundary with clamped. The result of the analysis is appeared to be the natural frequency obtained is different for each boundary condition (Wu et al., 2007).

Another research had been made about the analysis of free vibration by using characteristic orthogonal polynomials in assumed mode method. The analysis is done by comparing the different thickness of the plate. As a result, higher thickness increased the natural frequency of the plate as shown in the graph analysis in Figure 2.1 below (Kim et al., 2012).

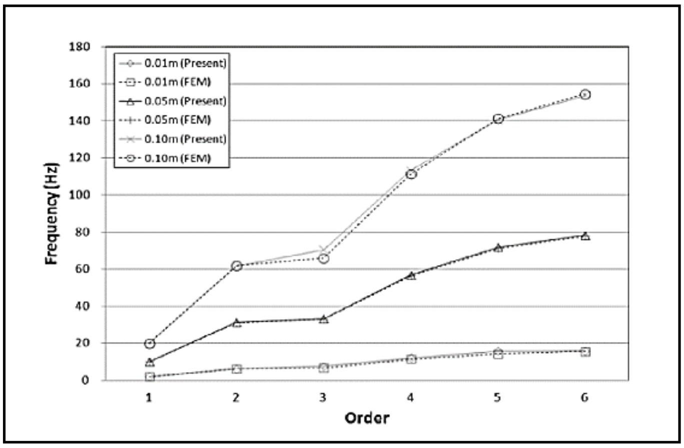


Figure 2.1: Natural frequencies in Hz of 4m × 3m rectangular plate with various thickness (Kim et al., 2012).

2.4 FLEXURAL WAVE

The previous research had done the flexural vibration analysis of a rectangular plate for the lower normal modes. The results show that different mode number produce different frequency. The frequency increased as the mode number is bigger as shown in Figure 2.2.

Table 5
Frequencies of a rectangular plate of 203 mm × 355 mm and thickness h=6.12 mm.

Mode number N	Classification	Experimental frequency f_N (Hz)	Theoretical frequency f_N (Hz)
1	(2, 0)	255.5	253.3
2	(1, 1)	266.8	272.4
3	(2, 1)	602.9	613.2
4	(3, 0)	705.3	701.0
5	(0, 2)	815.9	803.2
6	(1, 2)	989.3	988.5
7	(3, 1)	1083.6	1098.9
8	A	1350.8	1361.6
9	B	1470.4	1471.5
10	(4, 1)	1758.0	1778.8
11	(3, 2)	1981.0	2022.1

The pairs (n,m) represent the number of nodal lines in the long (n) and short (m) sides, respectively. For modes labelled "A" and "B" this classification does not apply.

Figure 2.2: The frequency of different mode number. (Manzanares-Martnez et al., 2010)

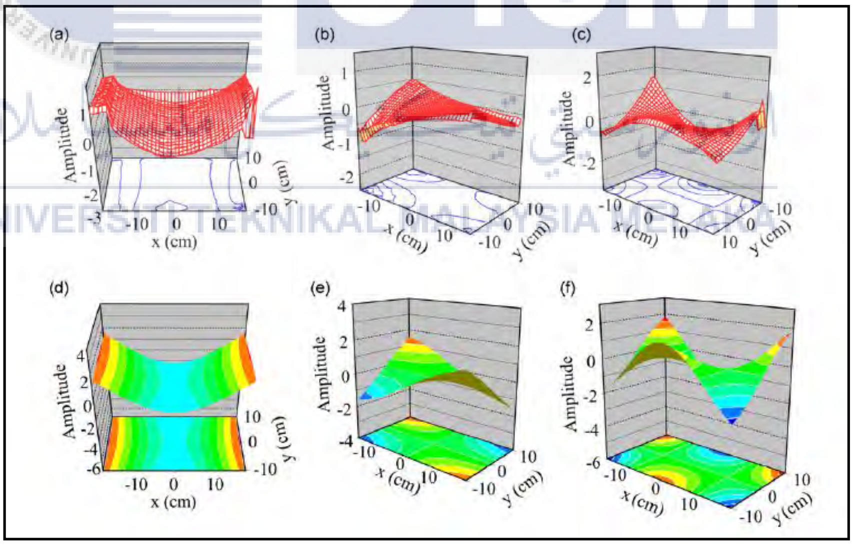


Figure 2.3: Normal-mode amplitudes for flexural waves of the rectangular plate. (Manzanares-Martnez et al., 2010)

The Figure 2.3 represent the result which the upper parts, (a), (b), and (c), are experimental results. The lower part, (d), (e), and (f), corresponds to theoretical amplitudes. The modes from left to right correspond to the (2, 0), (1, 1) and (2, 1) modes of Table 5.

2.5 SEISMIC ISOLATION SYSTEM

Currently, the further research from Korea Atomic Energy Research Institute found that the response of seismic isolation system is affected on the variability of the mechanical properties of lead rubber bearings, the response depends on the movement for different ground motions. The researcher states that the variation in the mechanical properties of base isolator will result significant influence on the shear resistance and the acceleration response of the structure.

It was discovered that the response of the isolation system is increased by the lower limit in variations, while the response of the superstructure is increased by the upper limit in variations. The upper limit in the stiffness variation of the isolators can reduce the decoupling performance of the isolation system (Choun, Park, & Choi, 2014). In the application of seismic isolation system, the building structures need to be related with safety, the different type in the materials and mechanical properties of the isolation system must be carefully controlled, furthermore to minimize the accidental torsion caused by the different similarity stiffness of variation isolators.

The factor of aging, environmental effects, and temperature in mechanical properties will contributes the stiffness and characteristic of isolation rubber metal, those the stiffness and damping values of the metal plate used for construction should be contrasts from other values used for design. Basically, the stiffness distribution of based isolated system and the mass distribution of structures at the ground level can be unbalance. A low natural frequency response of structure, such as seismically isolated structures is highly sensitive to the frequency input ground motion.

2.6 LAMINATED RUBBER-METAL SPRING (LR-MS) MODEL

Laminated rubber-metal spring is the multiple layer of rubber layer and metal plates were arranged alternating. The number of layer is determine based on the frequency vibration condition. A research had been done by the Centre for advanced Research on Energy, Universiti Teknikal Malaysia Melaka. By using dynamic analysis of finite element method, the performance of vibration transmissibility of LRMS can be determined. The laminated rubber-metal spring model is shown in Figure 2.4 below.

Figure 2.4: LR-MS Model

The FEA method has good accuracy for analyzing cylindrical solid rubber and LR-MS isolators and more interlayer metal plate inside LR-MS has better transmissibility performance at higher frequency, (M. A. Salim et al., 2013). There are many applications that use the LR-MS as their vibration absorption on superstructure to avoid for collapsing and disaster because of resonance frequency itself onto the superstructure. Penang second bridge is one of the examples which are designed to transfer the resonance frequency of the structure away from ground earthquake vibration frequency. Figure 2.5 shows the Penang Second Bridge and the rubber bearing used under the bridge is shown in Figure 2.6.

Figure 2.5: Penang Second Bridge (M. Salim et al., 2016)

Figure 2.6: Rubber bearing under Penang Second Bridge (M. Salim et al., 2016)

The performance of LR-MS is very well-known due to the capability in handling a large amount of the applied force and absorbing a vibration frequency by a natural resonance of an earthquake. Penang second bridge, Malaysia is an achievement of using LR-MS as the isolator system to the bridge and LR-MS were concentrated at preventing the stemming effects from the natural environment problem, such as ground motion by an earthquake (M. Salim et al., 2016).

CHAPTER 3

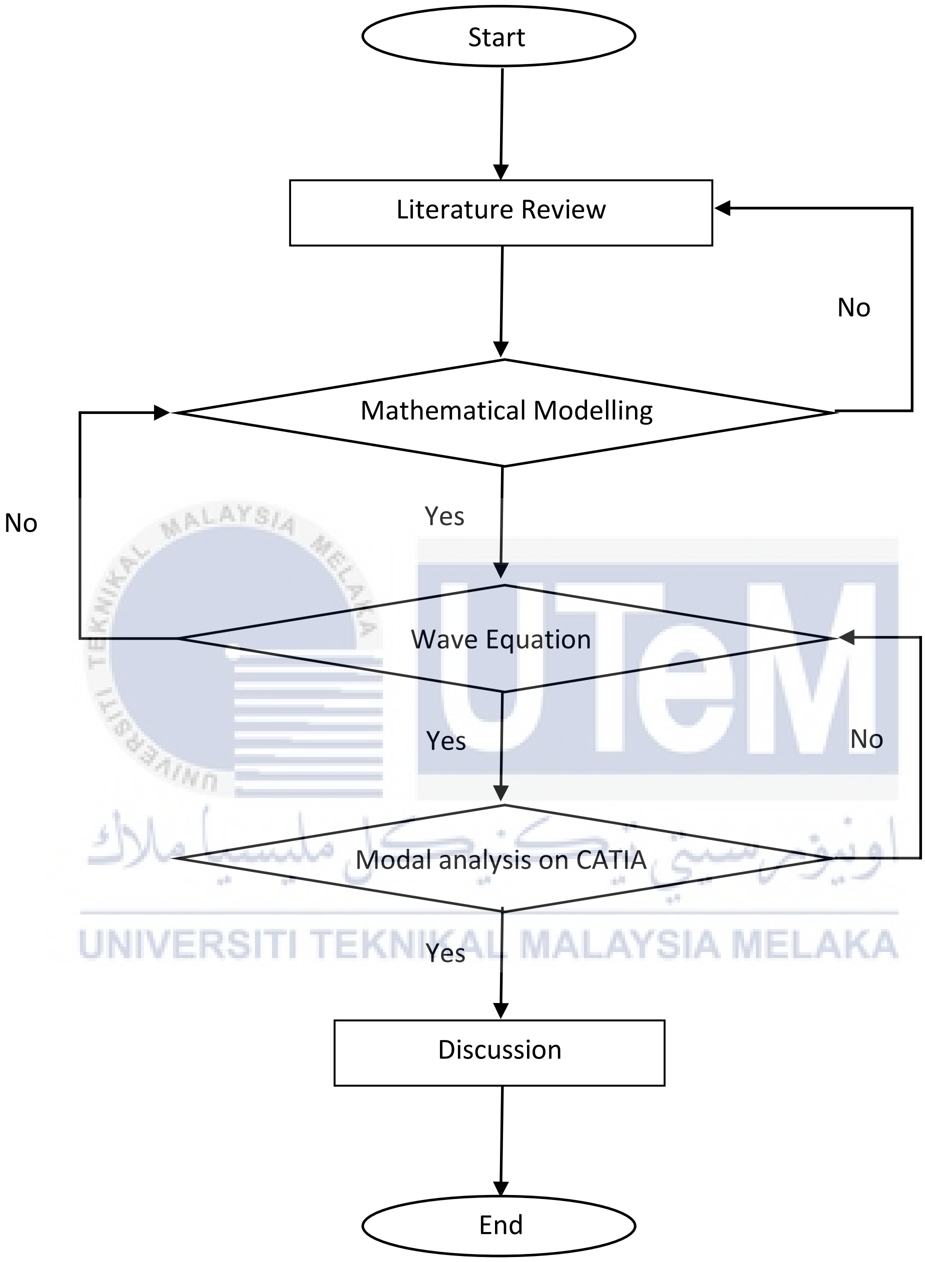
METHODOLOGY

3.1 INTRODUCTION

This chapter explains the methodology involves in this study to obtain the vibration effect of aluminium plate using simulation technique. Methodology is the strategies flow which explain the step to conduct the study. The process of methodology is to gather information which to interpret data within the scope of study. The step of methodology is represent in the flow chart to summarize the flow of entire study as it is easier to be referred.

For this study, the method includes the process of developing a mathematical model by using the method of separation of variable which has been chosen to complete the study of the vibration effect of aluminium plate using simulation technique. Then, from the mathematical model development, the wave equation can be obtained to relate with the simulation. The simulation of modal analysis will be made by using CATIA software and the discussion will be done based on the result of the simulation.

The methodology of this study can be summarized in the flow chart as below:



3.2 MATHEMATICAL MODELLING

The mathematical modelling of this study is developed by using the method of separation variables.

3.2.1 Wave Equation

Longitudinal Vibration of Rods/Beams.

Given the figure of beam:

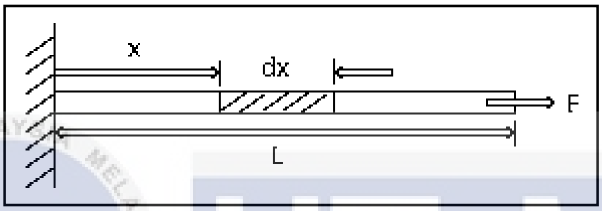


Figure 3.1: Schematic diagram of beam.

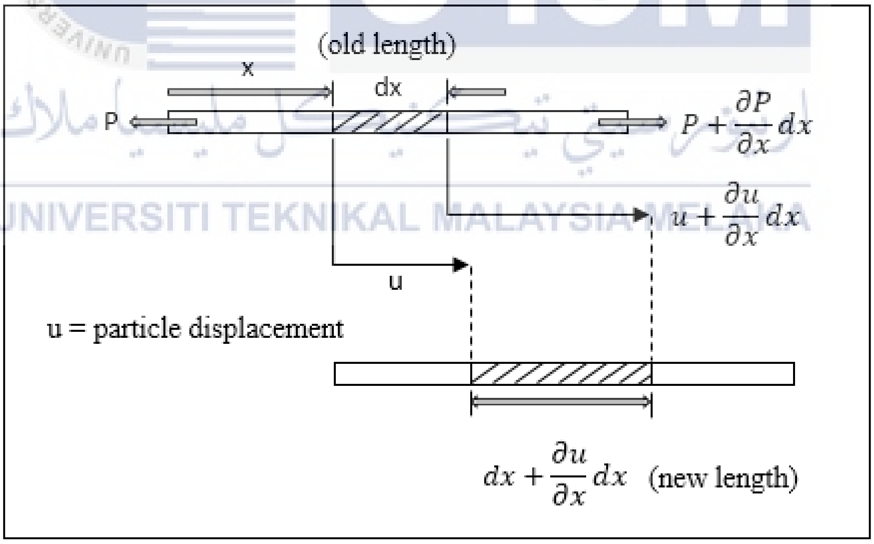


Figure 3.2: The change of length of element.

The length changed of element = new length – old length

$$= \left(dx + \frac{\partial u}{\partial x} dx \right) - dx$$

$$= \frac{\partial u}{\partial x} dx \qquad (1.1)$$

The unit of strain, $\varepsilon = \dfrac{new\ length - old\ length}{old\ length}$

$$= \frac{\frac{\partial u}{\partial x} dx}{dx} = \frac{\partial u}{\partial x} \tag{1.2}$$

Obtained the equation from the Hooke's Law, $\sigma = D\varepsilon$

Where,

$$D = E = \frac{\sigma}{\varepsilon}\ ,\ \sigma = \frac{P}{A}$$

D = Longitudinal stiffness

E = Modulus of elasticity

P = Pressure

A = Area

σ = Stress

Therefore, the derivation of the equation are :

$$\sigma = D\varepsilon$$

$$\frac{P}{A} = D\left(\frac{\partial u}{\partial x}\right)$$

$$P = AD\left(\frac{\partial u}{\partial x}\right)$$

Second derivative, $$\frac{\partial P}{\partial x} = AD\left(\frac{\partial^2 u}{\partial x^2}\right) \tag{1.3}$$

Depending on the material parameters is the longitudinal stiffness. From the relation, E and μ can be derive as below :

$$D = \frac{E}{1 - \left(\frac{2\mu^2}{1-\mu}\right)}$$

$$= \frac{E}{\frac{(1-\mu)-(2-\mu)}{1-\mu}}$$

$$= \frac{E(1-\mu)}{-2\mu^2 - \mu + 1}$$

$$= \frac{E(1-\mu)}{(1+\mu)(1-2\mu)} \qquad (1.4)$$

Equation from the Newton's Second Law ; $\sum F = ma$

Derive formula from the figures using the equation above.

$$\rho A dx\left(\frac{\partial^2 u}{\partial f^2}\right) = \left(P + \frac{\partial P}{\partial x}dx\right) - P$$

Where,

ρAdx : mass = density x volume

ρ = density (kg/m^3)

A = area (m^2)

dx = displacement (m)

$\frac{\partial^2 u}{\partial f^2}$: acceleration, $\frac{\partial u}{\partial f} = \frac{\partial^2 u}{\partial f^2}$

$P + \frac{\partial P}{\partial x}dx$: the force change with small elongation of dx

P : reaction force of P

Simplify the derivation formula :

$$\rho A dx\left(\frac{\partial^2 u}{\partial f^2}\right) = \left[P = AD\left(\frac{\partial^2 u}{\partial f^2}dx\right)\right] - P$$

$$\rho dx\left(\frac{\partial^2 u}{\partial f^2}\right) = D\left(\frac{\partial^2 u}{\partial x^2}dx\right)$$

$$\left(\frac{\partial^2 u}{\partial f^2}\right) = \frac{D}{\rho}\left(\frac{\partial^2 u}{\partial x^2}\right) \tag{1.5}$$

The displacement or stress wave in the rod can express the velocity of propogation as shown below.

$C_L = \sqrt{\frac{D}{\rho}}$ (Depending on the stiffness of the structure and density)

$$C_L{}^2 = \frac{D}{\rho} \tag{1.6}$$

Substitute (1.6) into (1.5),

$$\frac{\partial^2 u}{\partial f^2} = C_L{}^2\left(\frac{\partial^2 u}{\partial x^2}\right) \tag{1.7}$$

Therefore, the pure longitudinal vibration of rod can be included. The wave equation is :

$$\frac{\partial^2 u}{\partial x^2} = \frac{1}{C_L{}^2} \cdot \frac{\partial^2 u}{\partial f^2} \tag{1.8}$$

Quasi-longitudinal wave equation :

$$\frac{\partial^2 u}{\partial x^2} = \left(\frac{\rho}{D}\right)\frac{\partial^2 u}{\partial f^2} = \left(\frac{\rho}{E}\right)\frac{\partial^2 u}{\partial f^2}$$

Let $\mu = 0.5$, both D and C_L can be describe as infinity. From that, the value of μ indicates that the large area of solid rubber mats cannot be used as vibration isolators.

Wave number : $K_L = \frac{\omega}{C_L} = \omega\sqrt{\frac{\rho}{D}}$ (1.9)

Wave length : $\mu_L = \frac{C_L}{f}$ (1.10)

Quasi-longitudinal waves in solid :

Wave propogation speed : $C_L{}' = \sqrt{\frac{E}{\rho}}$ (1.11)

Wave number : $K = \omega\sqrt{\frac{\rho}{E}}$ (1.12)

3.3 CATIA SOFTWARE ANALYSIS

CATIA software is used to get the results in three dimensional (3D) form and to obtain data from the modal analysis. The analysis is done with the frequency case. There are several step before conducting the analysis by using the software which begin with the sketching. Each part of the model in CATIA are known as body.

3.3.1 Step before conduct the modal analysis by using CATIA.

Step 1: Open the CATIA software. Select the "Mechanical design" from the "Start" menu and choose the "Part Design" option.

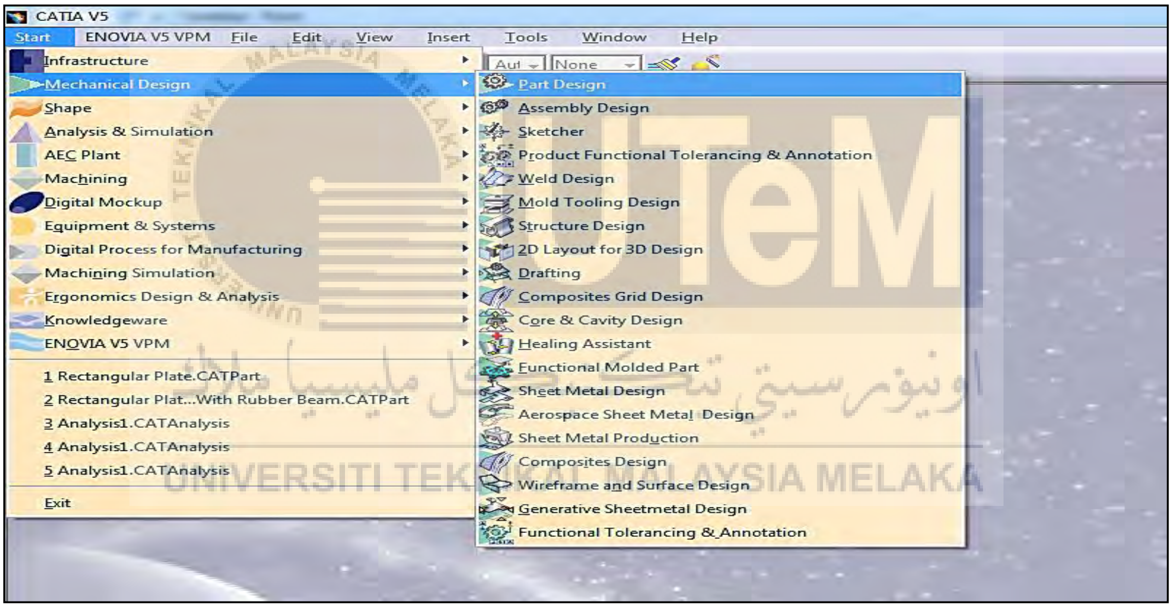


Figure 3.3: Part design option.

Step 2: Select "xy plane" and choose "Sketch" option to start sketching.

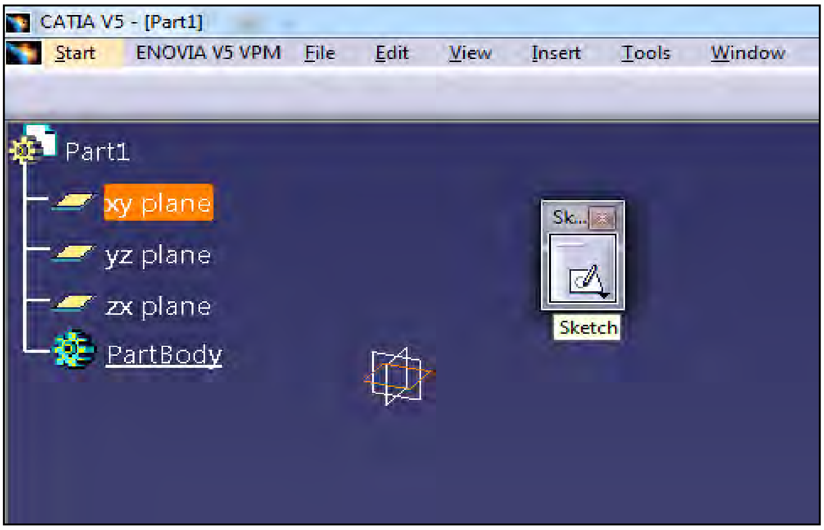


Figure 3.4: Plane selection.

Step 3: Sketch the body using the "Profile" option and specify the constraint. Then, select "Exit workbench" after completing the constraint option.

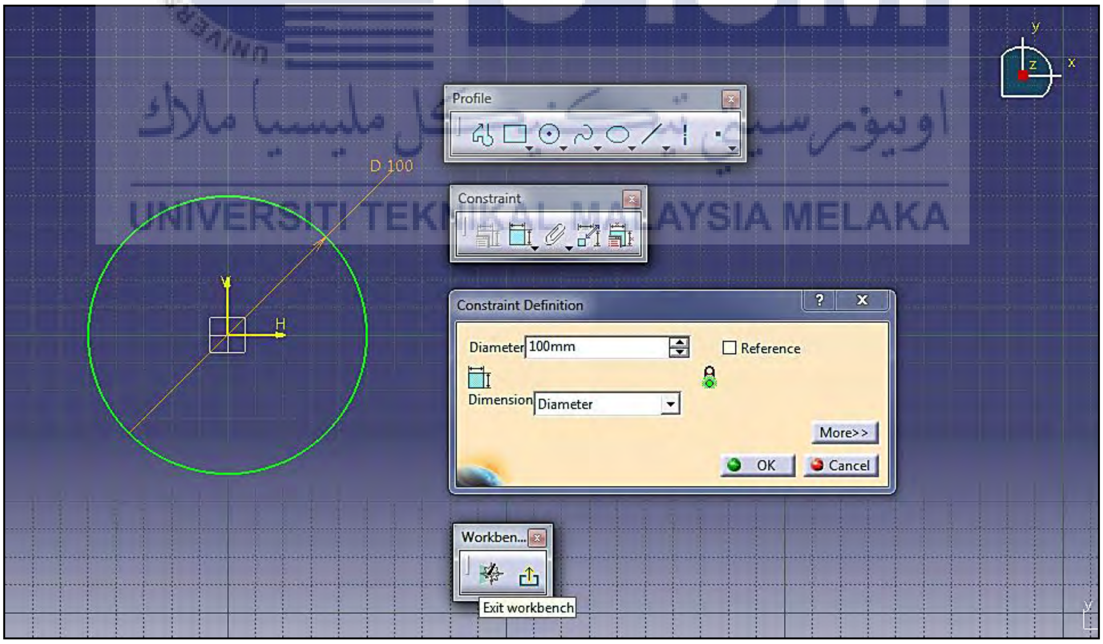


Figure 3.5: Sketching the part.

Step 4: Select the “Pad” option and specify the length for thickness of the body and click “OK”.

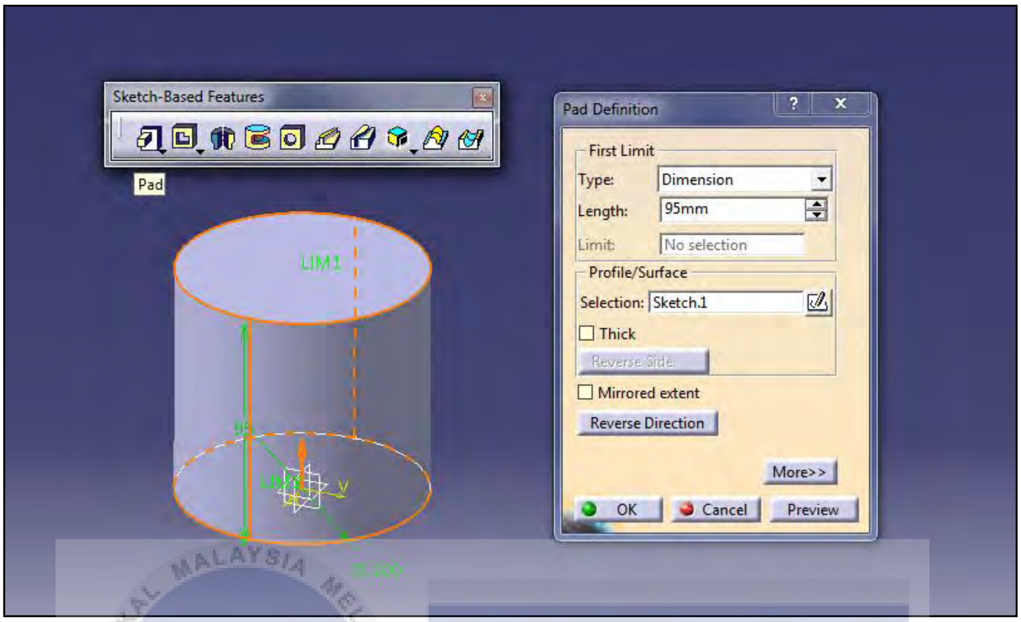


Figure 3.6: Pad definition.

Step 5: Choose the material to apply on the body.

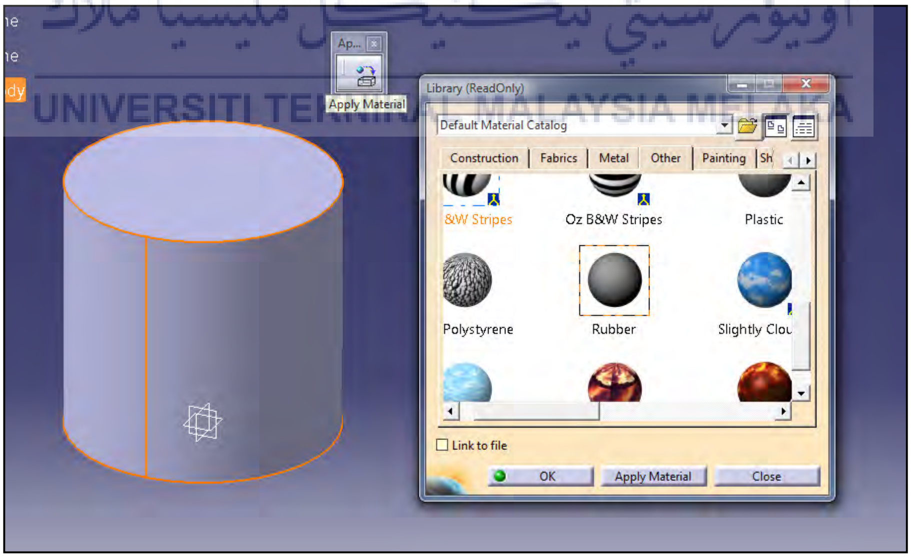


Figure 3.7: Material selection.

Step 6: Select "Insert" and choose "Body" to insert another body with different type of material. Then, repeat Step 2 until Step 5 for each body needed.

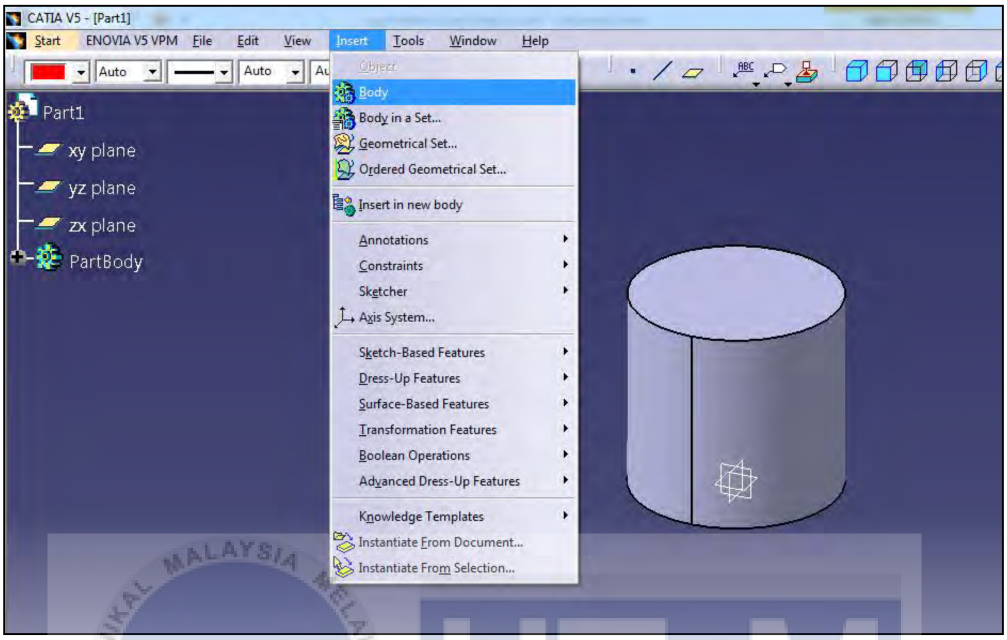
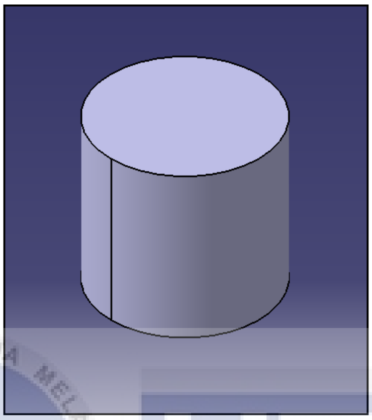
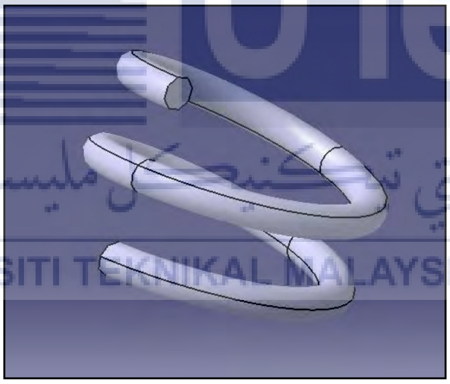
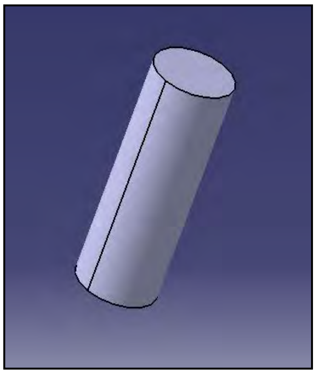


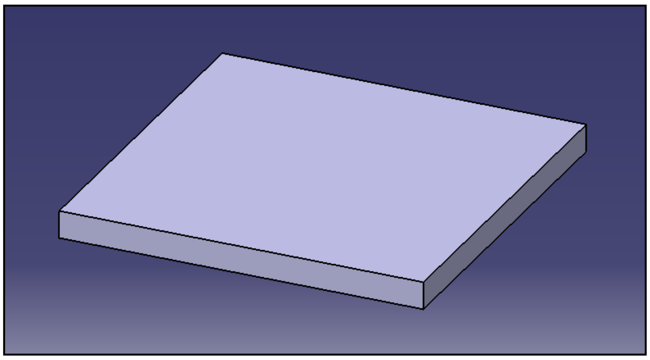
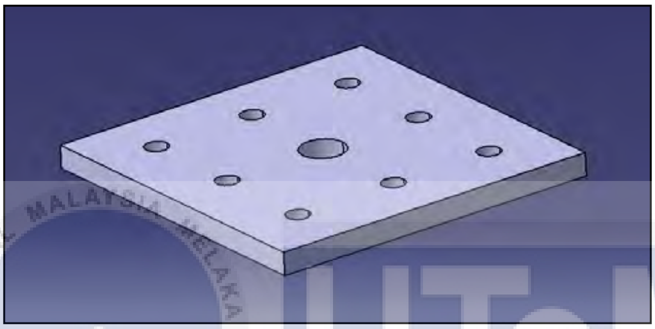
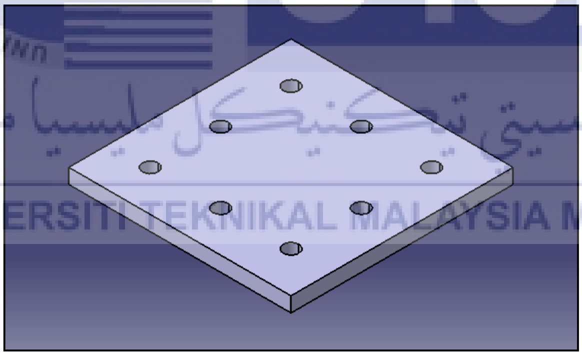
Figure 3.8: Insert body.

3.3.2 The component of the body.

Table 3.1 shows the details of component that is used for the analysis.

Table 3.1: Component of the body.

Name	Part	Specifications
Rubber		Material: Rubber Quantity: 1 Used in third condition.
Spring		Material: Steel Quantity: 8 Used in third condition.
Spring Holder		Material: Aluminium Quantity: 8 Used in second and third condition.

Base		Material: Aluminium Quantity: 1 Used in third condition.
Main plate		Material: Aluminium Quantity: 1 Used in all conditions.
Lower Spring Holder Plate		Material: Aluminium Quantity: 1 Used in second condition.

3.3.3 Condition used for the modal analysis.

The modal analysis have three conditions which are single plate, plates with spring holders and plates with springs, rubber and base. The main plate have nine holes with one hole is bigger than others placed at the center of the plate and the holes is labeled from 1 until 9.

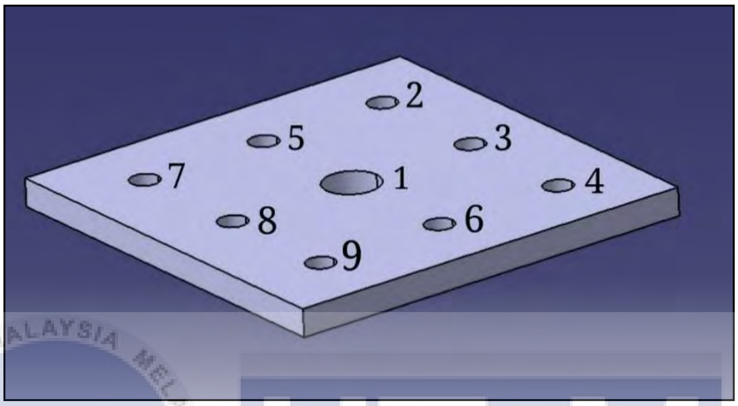


Figure 3.9: The labeled holes.

Figure 3.11 shows the holes with label. The label is used to represent the data which the simulation is done by applying load on each hole. The three conditions are used to compare the result of the modal analysis on the plates when the load is applied.

3.3.3.1 First Condition

The first condition is only the single plate with the nine holes. Figure 3.12 shows the single plate or the main plate used.

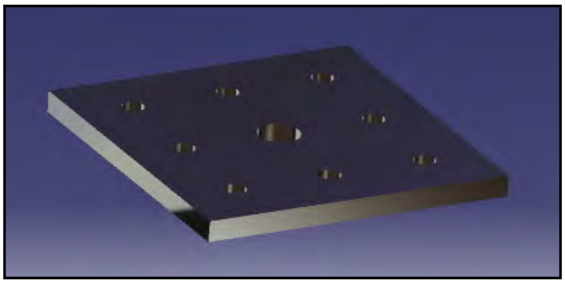


Figure 3.10: Single Plate

3.3.3.2 Second Condition

The second condition is the plates with the spring holders at the eight holes except at the center hole. Figure 3.13 shows the plates with spring holders.

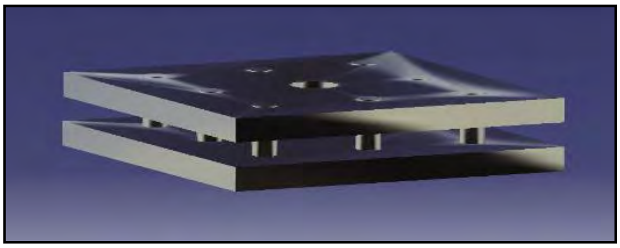


Figure 3.11: Plates with spring holders.

3.3.3.3 Third Condition

The third condition is the plates with springs, rubber between the second condition's plates and base. Figure 3.14 shows the third condition and Figure 3.15 shows the springs on each spring holders.

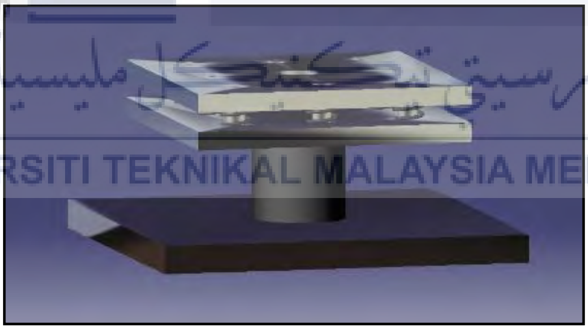


Figure 3.12: Plates with springs, rubber and base.

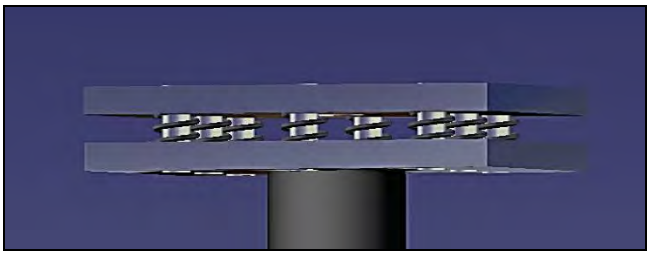


Figure 3.13: Springs on spring holders.

3.3.4 Step for the modal analysis of frequency case.

Step 1: After done the sketching and selecting material, select “Analysis & Simulation” from the “Start” menu and choose “Generative Structural Analysis” option.

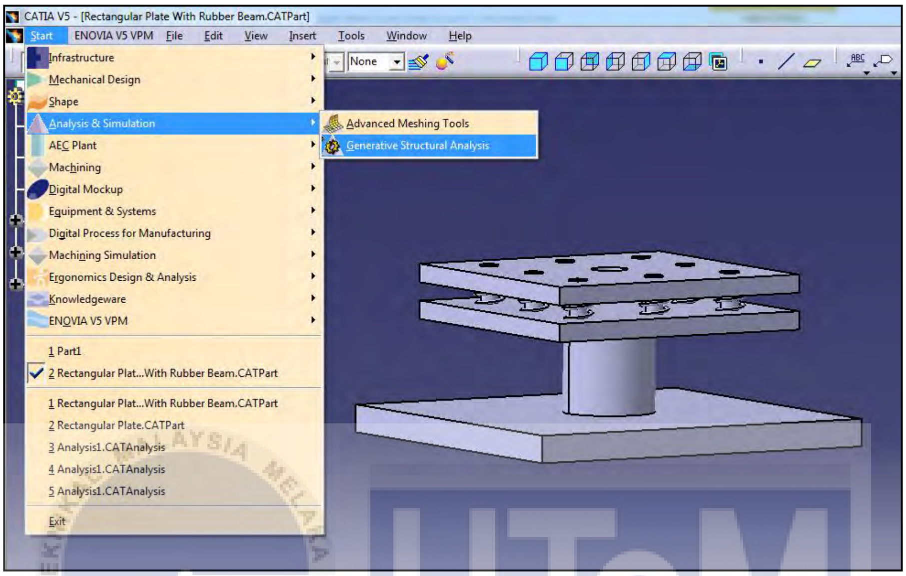


Figure 3.14: Analysis & Simulation option.

Step 2: Select the “Insert” menu and choose ‘Frequency Case”.

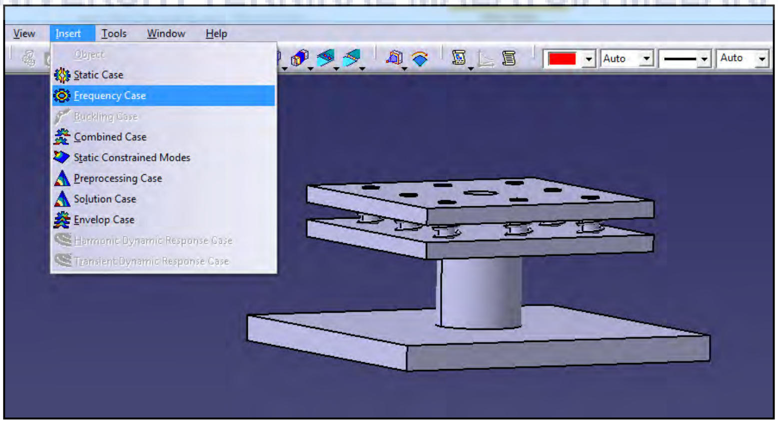


Figure 3.15: Frequency case option.

Step 3: Choose “Clamp” from the “Restraints” box and select the edges that need to clamp to hold the body. The red object shows the clamp applied at the edges.

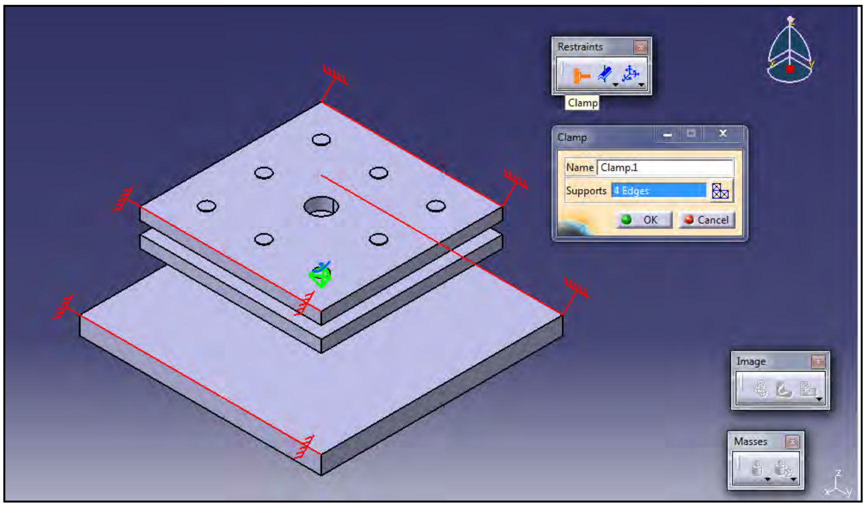


Figure 3.16: Clamp option.

Step 4: Select “Distributed Force” from the “Loads” box and choose the point to apply the load. Then, fill in the value of “Force Vector” for Z direction and click “OK”.

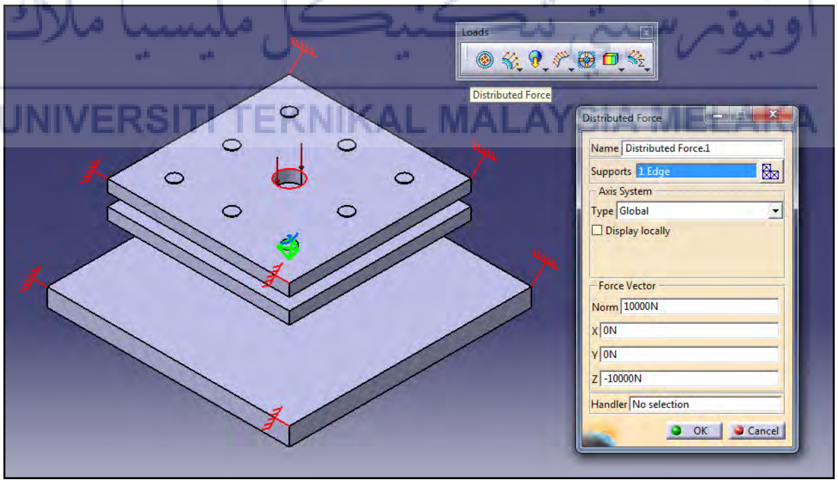


Figure 3.17: Applying the load.

Step 5: After that, compute the analysis of the body to calculate the result.

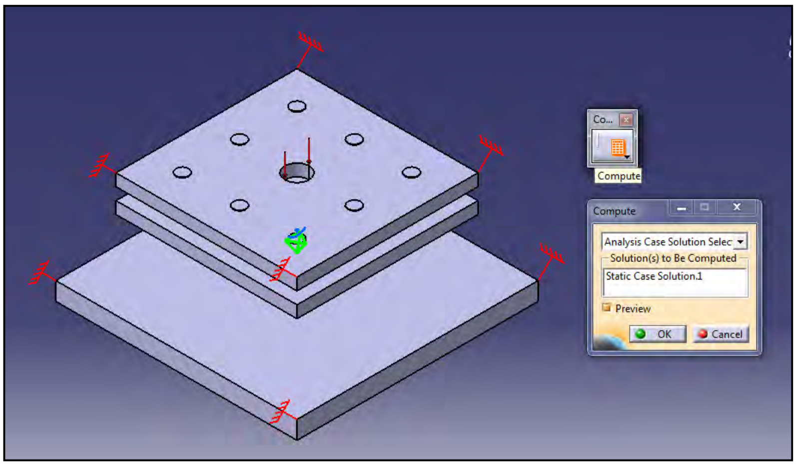


Figure 3.18: Compute the analysis.

CHAPTER 4

RESULTS AND DISCUSSION

4.1 INTRODUCTION

This chapter shows the result obtained from the modal analysis of the study by using CATIA software. For this study, the analysis is done based on the laminated rubber metal spring model dimension.

All the data obtained is tabulated and the graph is constructed based on the data. The results is based on the three conditions which gives different results in frequency value. From the analysis, the results in three dimension can be seen which shows the effect on the models after being applied by load.

The analysis is done with the applied load of 10kN and the value of frequency is obtained. From the results, the percentage error is calculated, then the data for all condition are compared.

4.2 CATIA SOFTWARE ANALYSIS

After done the computation, select the "Von Mises Stress" from the "Image" box to obtain the result of the analysis. The different colour represent the different values of the Von Mises stress occur on the body and the frequency value is obtained. Figure 4.1 shows the example of the result obtained from the simulation.

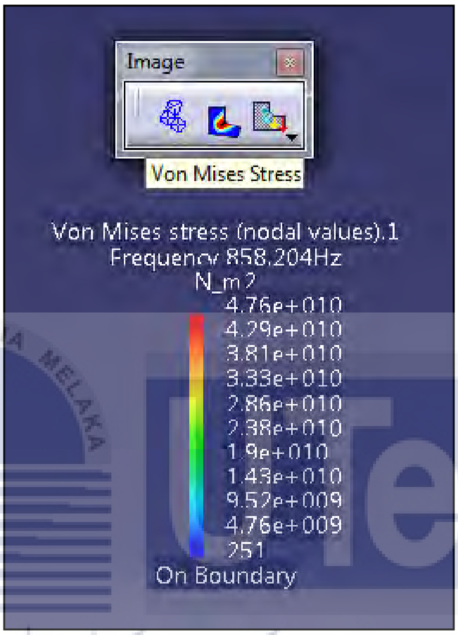
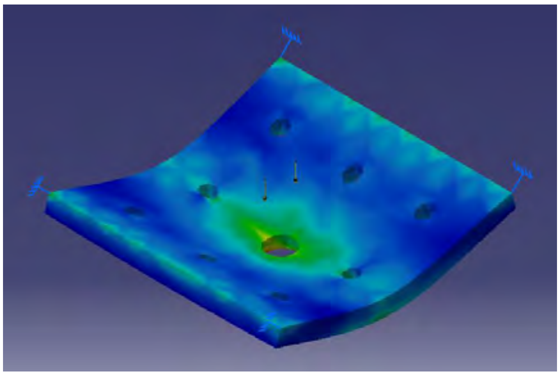
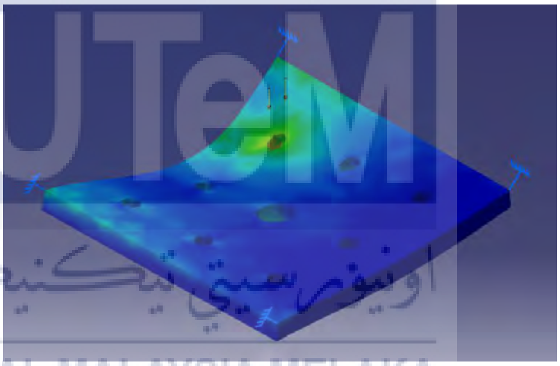
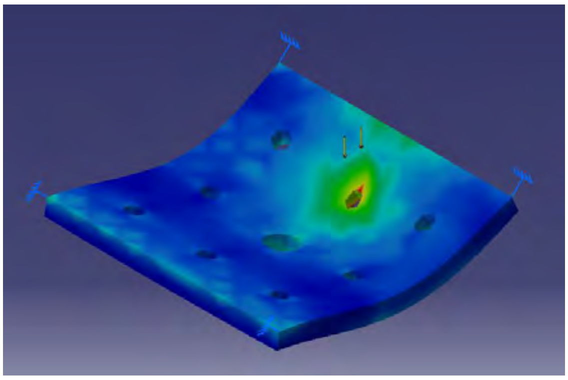


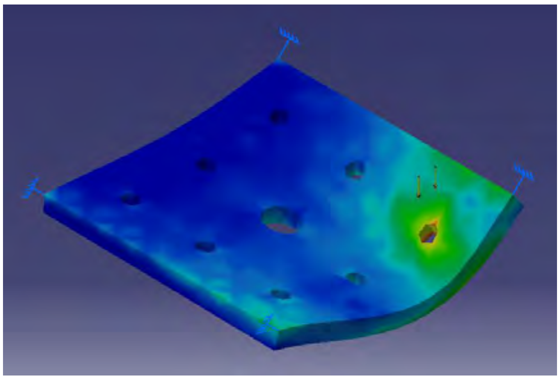
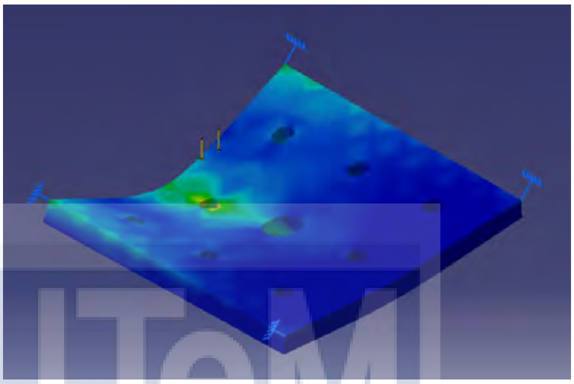
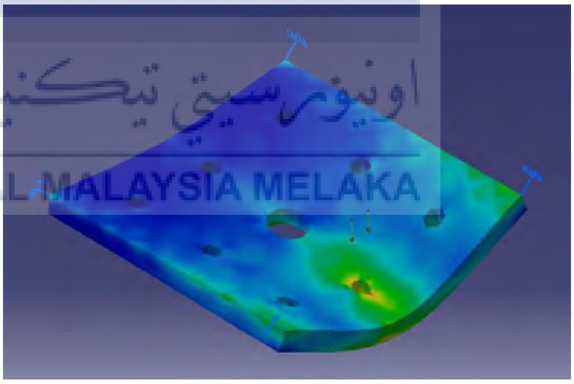
Figure 4.1: Result of simulation.

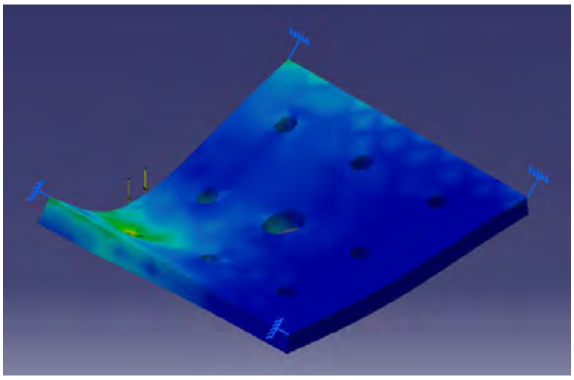
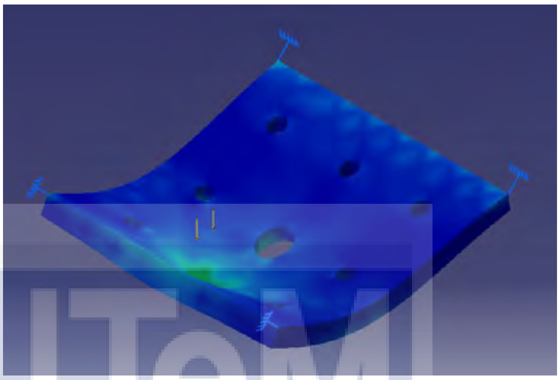
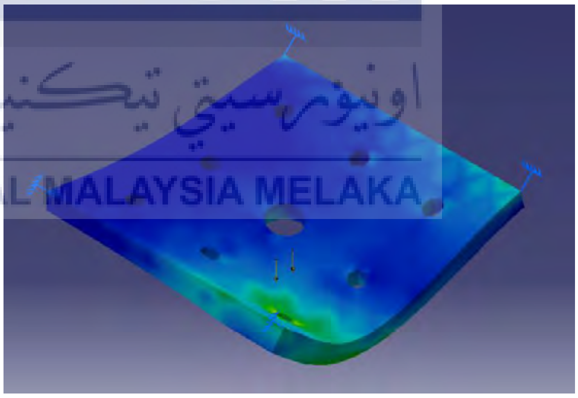
Table 4.1 to 4.3 shows the result from CATIA analysis of frequency case for all conditions. From the result, it shows that the Von Mises stress is occurred around the point where the load is applied. It is the stress which is allowed to be applied on the structural material based on the given force on the structure.

4.2.1 First Condition

Table 4.1: Result of frequency analysis for first condition.

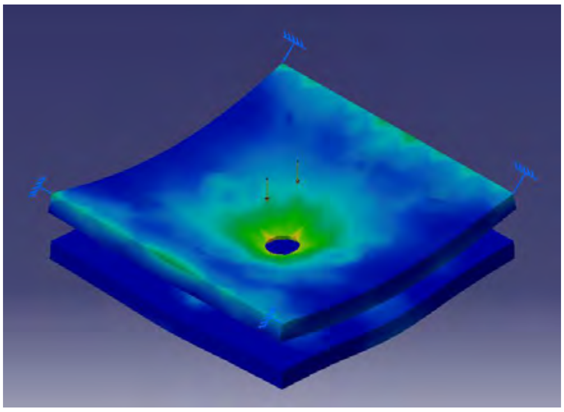
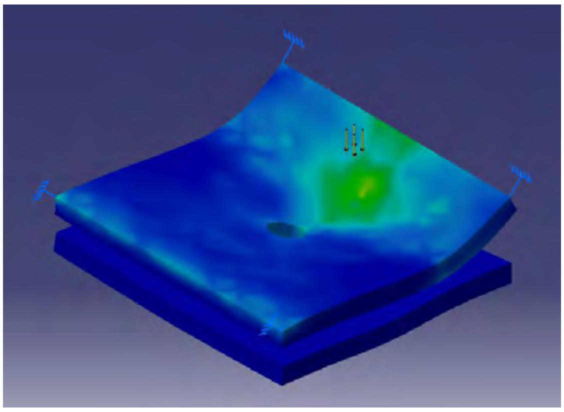
Point	Frequency (Hz)	Visual
1	1002.65	
2	1001.86	
3	1001.81	

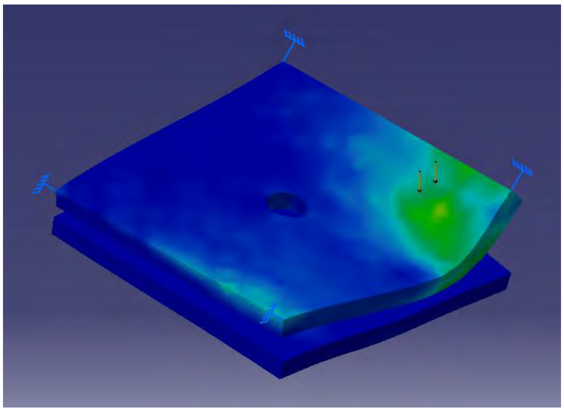
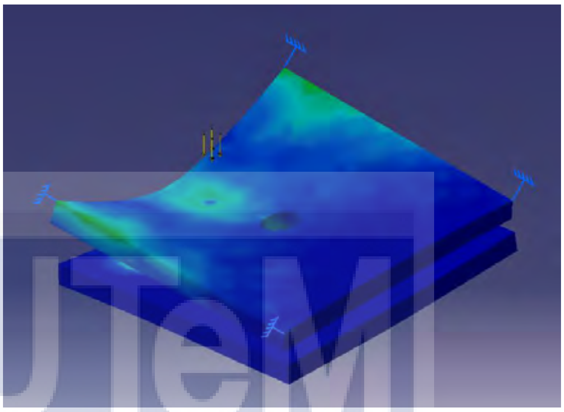
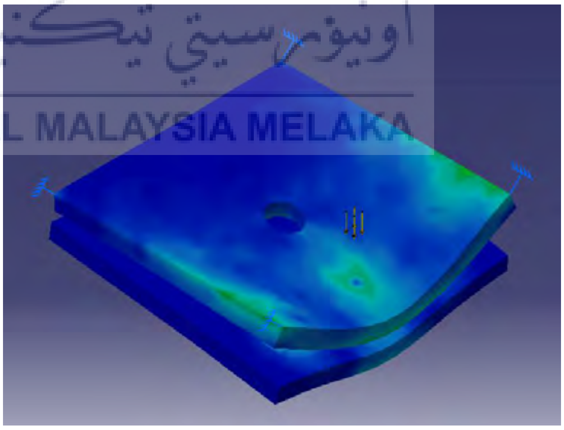
4	1001.81	
5	1002.72	
6	1002.62	

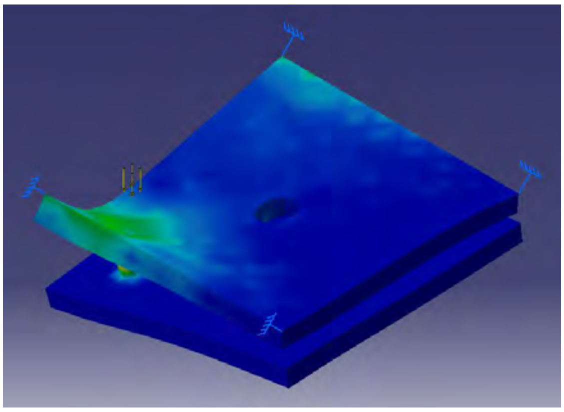
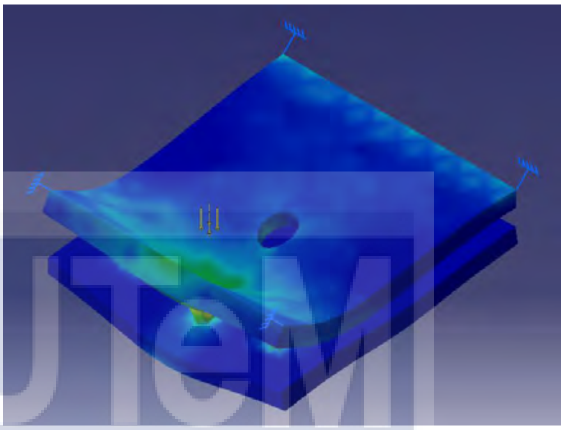
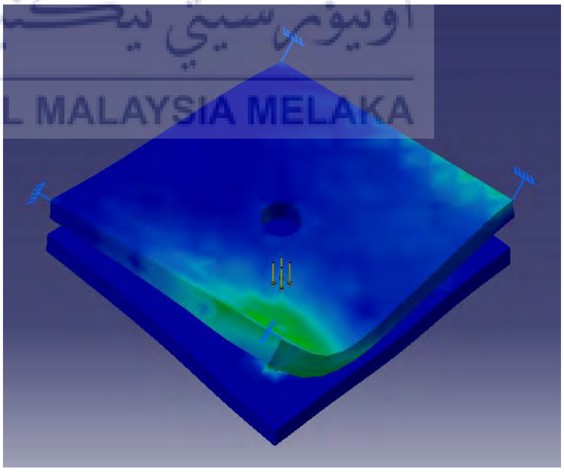
7	1001.88	
8	1001.88	
9	1001.83	

4.2.2 Second Condition

Table 4.2: Result of frequency analysis for second condition.

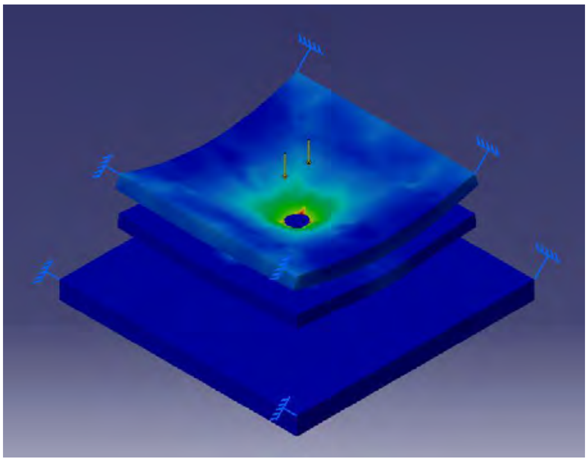
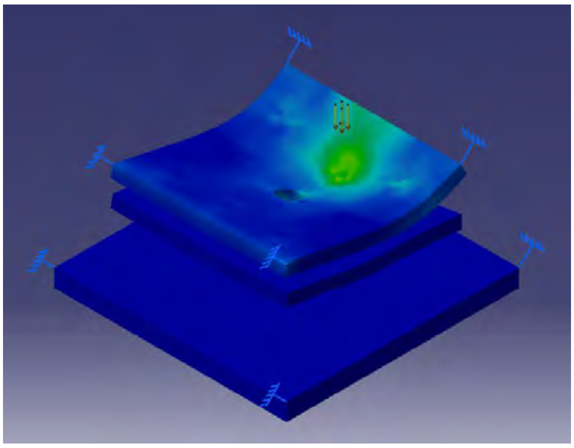
1	1181.60	
2	1181.74	
3	1181.71	

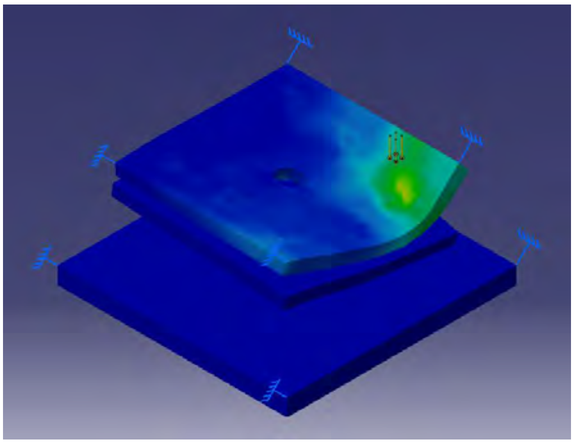
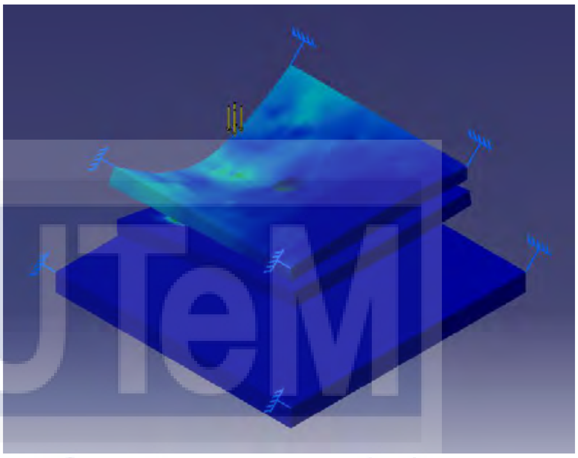
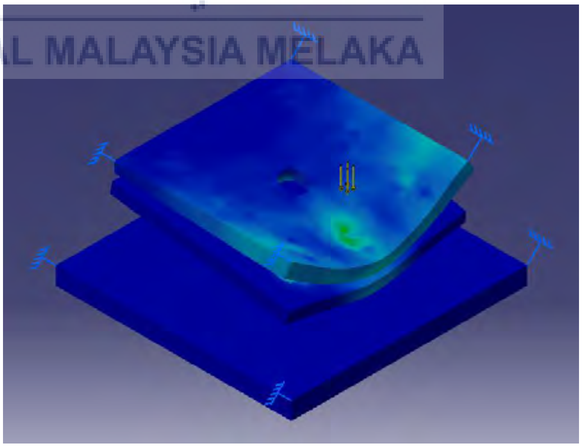
4	1181.71	
5	1181.88	
6	1181.84	

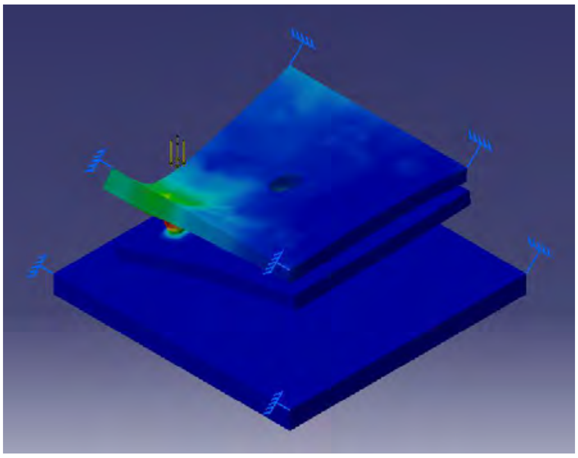
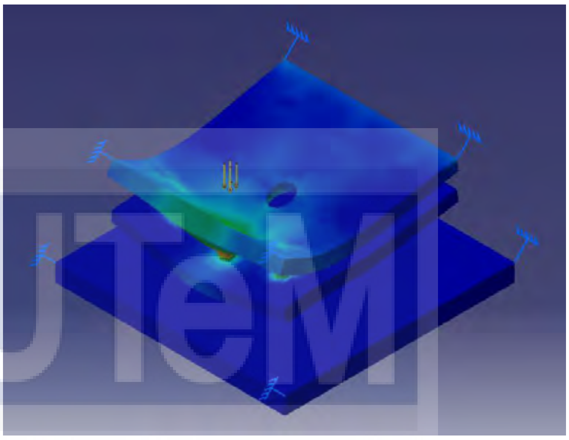
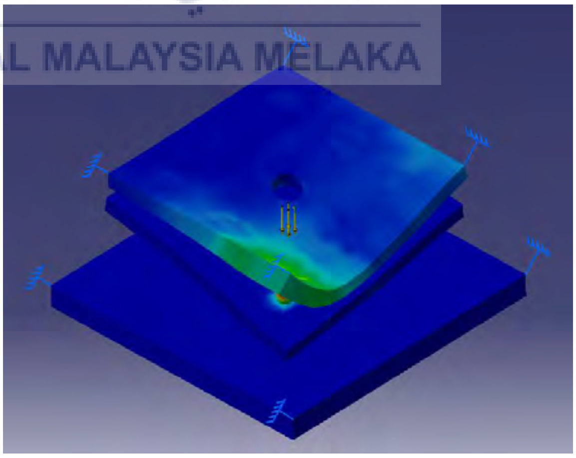
7	1181.74	
8	1181.71	
9	1181.71	

4.2.3 Third Condition

Table 4.3: Result of frequency analysis for third condition.

1	858.204	
2	858.204	
3	858.204	

4	858.204	
5	858.204	
6	858.204	

7	858.204	
8	858.204	
9	858.204	

4.2.4 Discussion

Table 4.1 to Table 4.3 shows the result of modal analysis on for all condition. Basically, modal analysis is the study of the dynamic properties of the systems in frequency domain. A typical example would be testing structures under vibrational excitation. Modal analysis is the field of measuring or calculating and analyzing the dynamic response of structures during excitation. From the analysis, the result obtained shows that the frequency value for all point has not much different when the load is being applied. It shows that the model is in homogeneous condition.

From the diagram, it can be seen that when the load is applied on each point, there are different color around the point that appeared on the model. The colors represent the stress or wave occurred on the model which is shown in Figure 4.1. However, it also can be seen that the side that have been clamped is similarly have slightly changed in color. It is because stress is also occurred on that area.

Quasi-longitudinal wave equation :

$$\frac{\partial^2 u}{\partial x^2} = \left(\frac{\rho}{D}\right)\frac{\partial^2 u}{\partial f^2} = \left(\frac{\rho}{E}\right)\frac{\partial^2 u}{\partial f^2}$$

where,

ρ	: Density
D	: Longitudinal stiffness
E	: Modulus of elasticity
$\frac{\partial^2 u}{\partial f^2}$	: Acceleration

The equation above which has been derived in chapter three, represent the particular result from the analysis. It can be related which the equation is the wave occurred on the model when being analyzed with CATIA. The vibration occur on the model is higher at the point where the load is being applied, so the wave is also high.

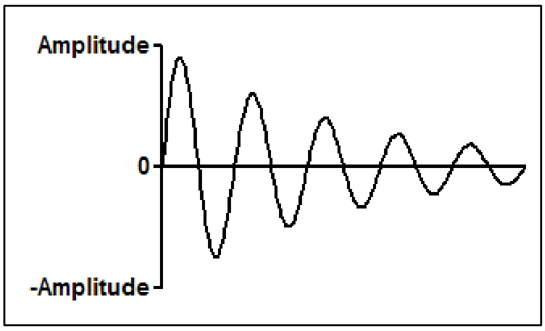


Figure 4.2: The vibration amplitude occurred on the plate.

Figure 4.2 shows the amplitude of vibration when force is being applied on the model. The figure indicates that the amplitude becomes smaller as the distance from the point which being applied by load is farther. So, it can be conclude that vibration occur on the plate is higher at the point which being applied by load.

4.3 DATA AND RESULT

Table 4.4 until Table 4.6 shows the result obtained from the simulation for the three conditions. The analysis is done separately on each hole with the applied load of 10kN. The result of analysis on Point 1 acts as the theoretical value for each condition. The error value is calculated based on the frequency value and multiply by 100% to obtain the percentage error.

Formula of percentage error calculation:

$$[(\text{Theoretical value} - \text{Experimental value})/\text{Theoretical value}] \times 100\% = \text{Percentage Error}$$

The graph of the percentage error for the analysis of the conditions is shown in Figure 4.3, Figure 4.5 and Figure 4.7.

4.3.1 First Condition

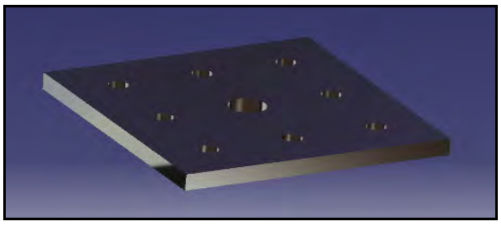


Figure 4.3: The single plate with the nine holes.

Table 4.4: Result of analysis for first condition.

POINT	FREQUENCY (Hz)	PERCENTAGE ERROR
1	1002.65	-
2	1001.86	7.87×10^{-2}
3	1001.81	8.38×10^{-2}
4	1001.81	8.38×10^{-2}
5	1002.72	0.70×10^{-2}
6	1002.62	0.30×10^{-2}
7	1001.88	7.68×10^{-2}
8	1001.88	7.68×10^{-2}
9	1001.83	8.18×10^{-2}

Example of calculation:

$$\left|\frac{1002.65 - 1001.86}{1002.65}\right| x\ 100\% = 7.87\ x\ 10^{-2}$$

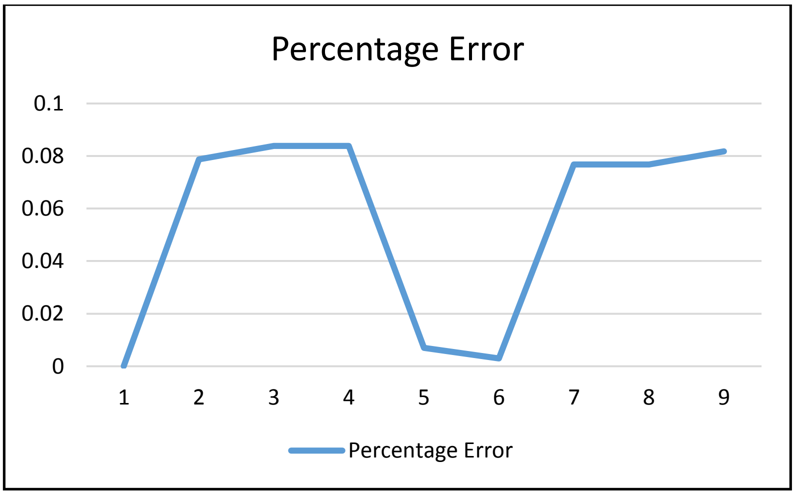


Figure 4.4: Graph of percentage error for analysis of first condition.

4.3.2 Second Condition

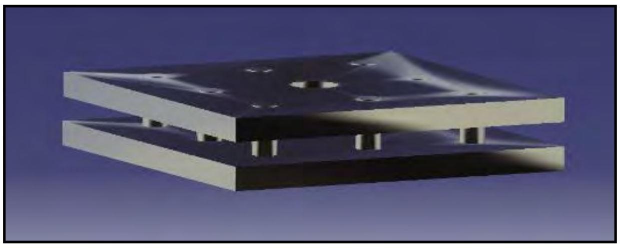


Figure 4.5: The plates with spring holders.

Table 4.2: Result of analysis for second condition.

POINT	FREQUENCY (Hz)	PERCENTAGE ERROR
1	1181.60	-
2	1181.74	1.18×10^{-2}
3	1181.71	0.93×10^{-2}
4	1181.71	0.93×10^{-2}
5	1181.88	2.37×10^{-2}
6	1181.84	2.03×10^{-2}
7	1181.74	1.18×10^{-2}
8	1181.71	0.93×10^{-2}
9	1181.71	0.93×10^{-2}

Example of calculation:

$$\left|\frac{1181.60 - 1181.74}{1181.60}\right| x\ 100\% = 1.18\ x\ 10^{-2}$$

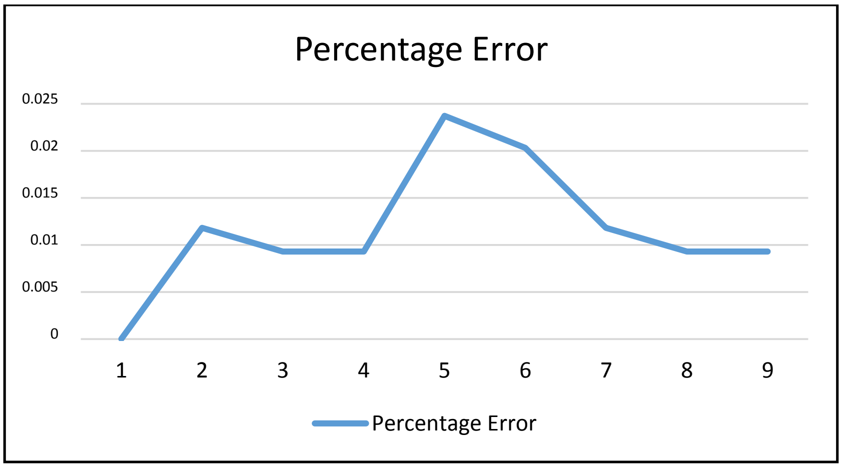


Figure 4.6: Graph of percentage error for analysis of second condition.

4.3.3 Third Condition

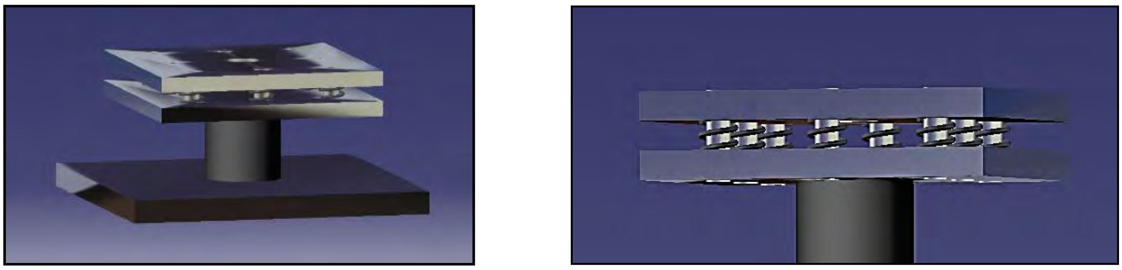


Figure 4.7: (a) The plates with springs, rubber between the second condition's plates and base and (b) springs at spring holders.

Table 4.3: Result of analysis for third condition.

POINT	FREQUENCY (Hz)	PERCENTAGE ERROR
1	858.204	-
2	858.204	0
3	858.204	0
4	858.204	0
5	858.204	0
6	858.204	0
7	858.204	0
8	858.204	0
9	858.204	0

Example of calculation:

$$\left|\frac{858.204 - 858.204}{858.204}\right| x\ 100\% = 0$$

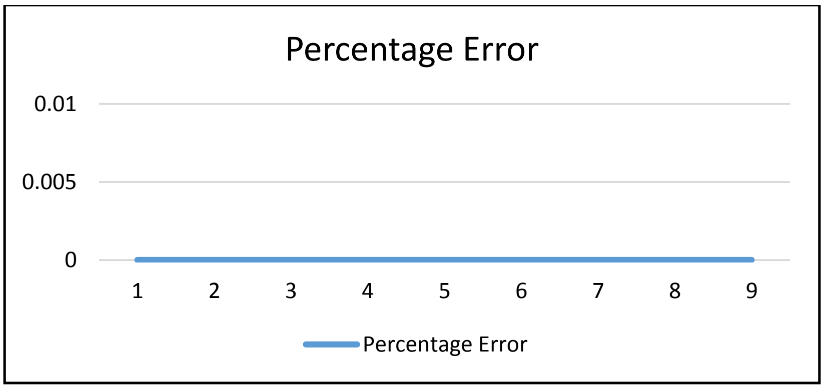


Figure 4.8: Graph of percentage error for analysis of third condition.

4.4 SUMMARY

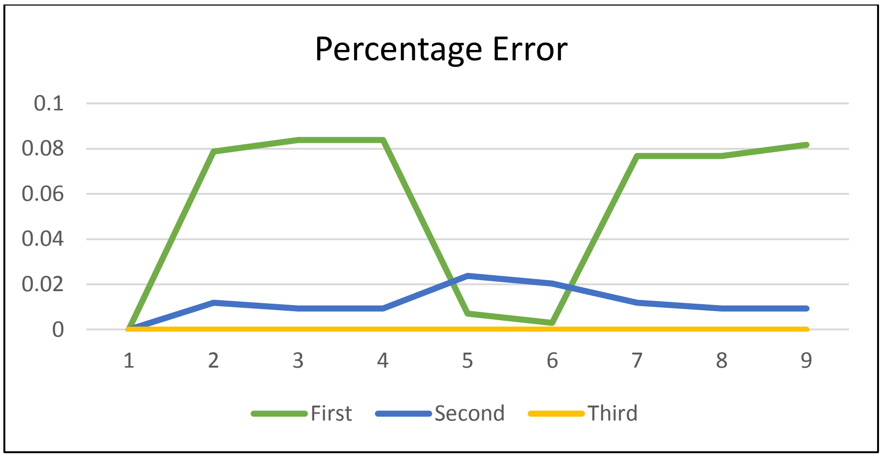


Figure 4.9: Graph of percentage error for all condition.

From the analysis, the result obtained is different for all condition. The graph shows that all condition have different pattern. The result of first condition shows that it has the largest value of percentage error compared to the other two conditions. However, the smaller value of percentage error at point 5 and 6 is due to the condition which has nothing that supporting at the center of the plate. The second condition has lower value of percentage error than the first condition but higher than the third condition.

From the analysis, it shows that the third condition is the most effective condition which absorb the vibration equally when load is applied at each point as the frequency value obtained is the same at all point. This is proven from the percentage error obtained from the result. The result is affected by the structure of the model and the materials of the components. The rubber applied on the model is the isolators which absorb vibration and reduce the flexural wave occur on the model. It can be conclude that the real model of LR-MS is the good isolator and safe to be used.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 CONCLUSION

For this study, the information from the previous research was gathered in the literature review in order to understand the objective of the study. The information obtained is about the theory of plate, free vibration of plate, flexural wave, seismic isolation system and the laminated rubber-metal spring model. All information from the research is important as to be the reference for this study of the effect on vibration amplitude in aluminium rectangular plate using simulation approach.

After the information is gathered, the methodology was design as a process to interpret data within the scope of study. For this study, the methodology includes the process of developing a mathematical model by using the method of separation of variable for rectangular plate. Then, from the mathematical model development, wave equation is obtained. The study is continued with the modal analysis for frequency case. The analysis of the study is done by using CATIA software and the discussion is done based on the result of the analysis.

For the modal analysis of the frequency case, there were three type of condition for the model. The first condition is only the single plate with the nine holes. The second condition is the plates with the spring holders at the eight holes except at the center hole. The third condition is the plates with springs, rubber between the second condition's plates and base. All the condition can be referred in chapter three.

The analysis is done by applying the load of 10kN on each hole for every condition. From the analysis, the frequency value is obtained and then the percentage is calculated. The result shows that the third condition is the most effective condition which absorb the vibration equally when load is applied at each point as the frequency value obtained is the same at all point. It can be conclude that the outcome is due to the structure and the possible resonance can be avoid from occur in the application.

5.2 RECOMMENDATION

From the study that has been done, there are some recommendations that can be made for a better understanding on this study. For example, the analysis can be made with several value of loads and also various size of the model. Moreover, different material and shape of the plates can also be used to compare the results with this study. Finally, another recommendation for the future study that can be made is by using different method to obtain the result such as using different software to do the analysis to obtain the result.

REFERENCES

A. Nikkhoo, M.E. Hassanabadi, S.E. Azam, J. V. A. (2014). Vibration of a thin rectangular plate subjected to series of moving inertial loads. *Mechanics Research Communications*, *55*, 105–113.

Apuzzo, A., Barretta, R., & Luciano, R. (2015). Some analytical solutions of functionally graded Kirchhoff plates. *Composites Part B: Engineering*, *68*, 266–269.

Batista, M. (2010). Analytical solution for free vibrations of simply supported transversally inextensible homogeneous rectangular plate, (July). Retrieved from http://arxiv.org/abs/1007.2539

Benachour, A., Tahar, H. D., Atmane, H. A., Tounsi, A., & Ahmed, M. S. (2011). A four variable refined plate theory for free vibrations of functionally graded plates with arbitrary gradient. *Composites Part B: Engineering*, *42*(6), 1386–1394.

C. Vemula, A. . N. (1997). Flexural Wave Propagation and Scattering on Thin Plate Using Mindlin Theory. *Wave Motion*, *26*, 1–12.

Chen, W. Q., & Ding, H. J. (2002). On free vibration of a functionally graded piezoelectric rectangular plate. *Acta Mechanica*, *153*(3–4), 207–216.

Cho, D. S., Vladimir, N., & Choi, T. M. (2013). Approximate natural vibration analysis of rectangular plates with openings using assumed mode method. *International Journal of Naval Architecture and Ocean Engineering*, *5*(3), 478–491.

Choun, Y. S., Park, J., & Choi, I. K. (2014). Effects of mechanical property variability in lead rubber bearings on the response of seismic isolation system for different ground motions. *Nuclear Engineering and Technology*, *46*(5), 605–618.

Hosseini-Hashemi, S., Fadaee, M., & Atashipour, S. R. (2011). A new exact analytical approach for free vibration of ReissnerMindlin functionally graded rectangular plates. *International Journal of Mechanical Sciences*, *53*(1), 11–22.

Kim, K.-H., Kim, B.-H., Choi, T.-M., & Cho, D.-S. (2012). Free vibration analysis of rectangular plate with arbitrary edge constraints using characteristic orthogonal polynomials in assumed mode method. *International Journal of Naval Architecture and Ocean Engineering*, *4*(3), 267–280.

Lu, P., He, L. H., Lee, H. P., & Lu, C. (2006). Thin plate theory including surface effects. *International Journal of Solids and Structures*, *43*(16), 4631–4647.

Manzanares-Martnez, B., Flores, J., L. Gutiérrez, L., Méndez-Sánchez, R. A., Monsivais, G., Morales, A., & Ramos-Mendieta, F. (2010). Flexural vibrations of a rectangular plate for the lower normal modes. *Journal of Sound and Vibration*, *329*(24), 5105–5115.

Méndez-Sánchez, R. A., Morales, A., Gutiérrez, L., & Al, E. (2009). Flexural Vibrations of Plates: Theory and Experiment. In *Set.Eesc.Usp.Br* (pp. 4–8). Retrieved from

S.A. Eftekhari, A. A. J. (2011). A mixed method for free and forced vibration of rectangular plates. *Applied Mathematical Modelling*, *36*(6), 2814–2831.

Salathe, E. P., Arangio, G. A., & Salathe, E. P. (1989). An application of beam theory to determine the stress and deformation of long bones. *Journal of Biomechanics*, *22*(3), 189–199.

Salim, M. A., Putra, A., Thompson, D., Ahmad, N., & Abdullah, M. A. (2013). Transmissibility of a Laminated Rubber-Metal Spring: A Preliminary Study. *Applied Mechanics and Materials*, *393*, 661–665.

Salim, M., Putra, A., Mansor, M., & Musthafah, M. (2016). Sustainable of Laminated Rubber-Metal Spring in Transverse Vibration. *Procedia*, *19*, 203–210.

Sen Gupta, G. (1970). Natural flexural waves and the normal modes of periodically-supported beams and plates. *Journal of Sound and Vibration*, *13*(1), 89–101.

Sharma, J. N., Sharma, P. K., & Rana, S. K. (2009). Flexural and transversal wave motion in homogenous isotropic thermoelastic plate by using asymptotic method. *Journal of Sound and Vibration*, *329*(1), 804–818.

Soldatos, K. P., & Hadjigeorgiou, V. P. (1990). Three-dimensional solution of the free vibration problem of homogeneous isotropic cylindrical shells and panels. *Journal of Sound and Vibration*, *137*(3), 369–384.

Wen, J., Wang, G., Yu, D., Zhao, H., & Liu, Y. (2005). Theoretical and experimental investigation of flexural wave propagation in straight beams with periodic structures: Application to a vibration isolation structure. *Journal of Applied Physics*, *97*(11).

Wu, J. H., Liu, A. Q., & Chen, H. L. (2007). Exact Solutions for Free-Vibration Analysis of Rectangular Plates Using Bessel Functions. *Journal of Applied Mechanics*, *74*(6), 1247.

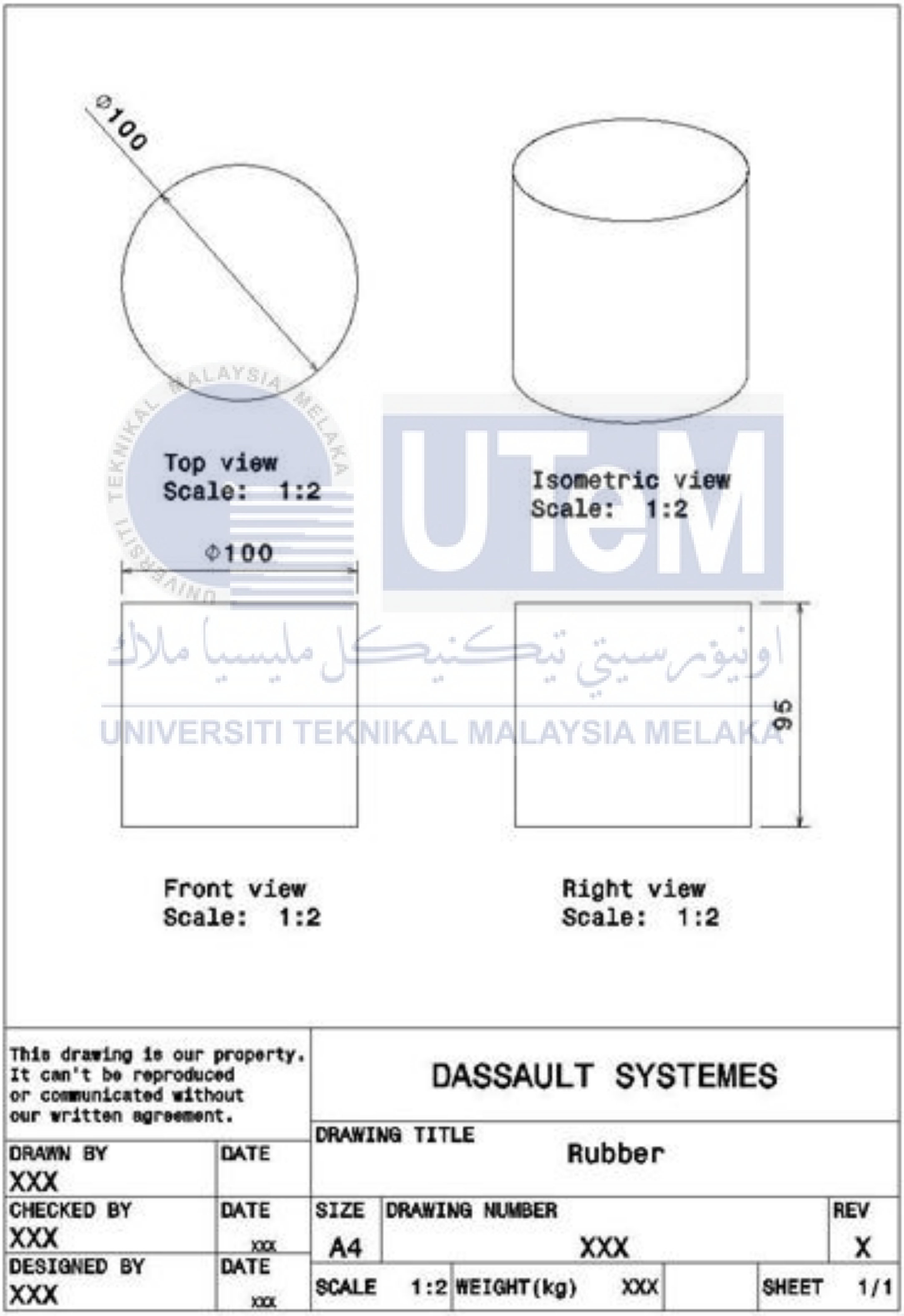
Xing, Y., & Liu, B. (2009). *New exact solutions for free vibrations of rectangular thin plates by symplectic dual method*. *Acta Mechanica Sinica/Lixue Xuebao*.

Zhao, Y. F., Liu, D., Liao, S. Q., Liao, X. X., Lin, S. B., Universitatis, F., … M. A. Salim, A. Putra, M. R. Mansor, M.T. Musthafah, M.Z. Akop, M.A. Abdullah, M.N. Abdul Rahman, M.N. Sudin, M. A. S. (2016). Sustainable of Laminated Rubber-Metal Spring in Transverse Vibration. *Procedia*, *10*(1), 203–210.

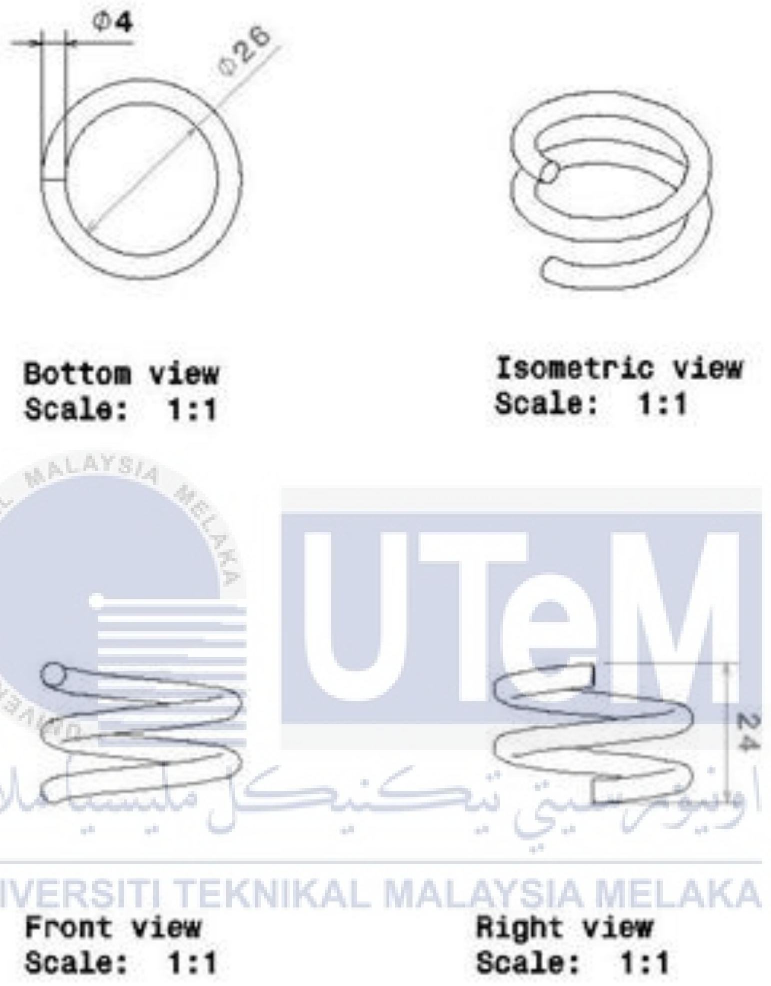
Zhili Hao, Ahmet Erbil, F. A. (2003). An analytical model for support loss in micromachined beam resonators with in-plane flexural vibrations. *Sensors and Actuators, A: Physical*, *109*(1–2), 156–164.

APPENDICES

APPENDIX A

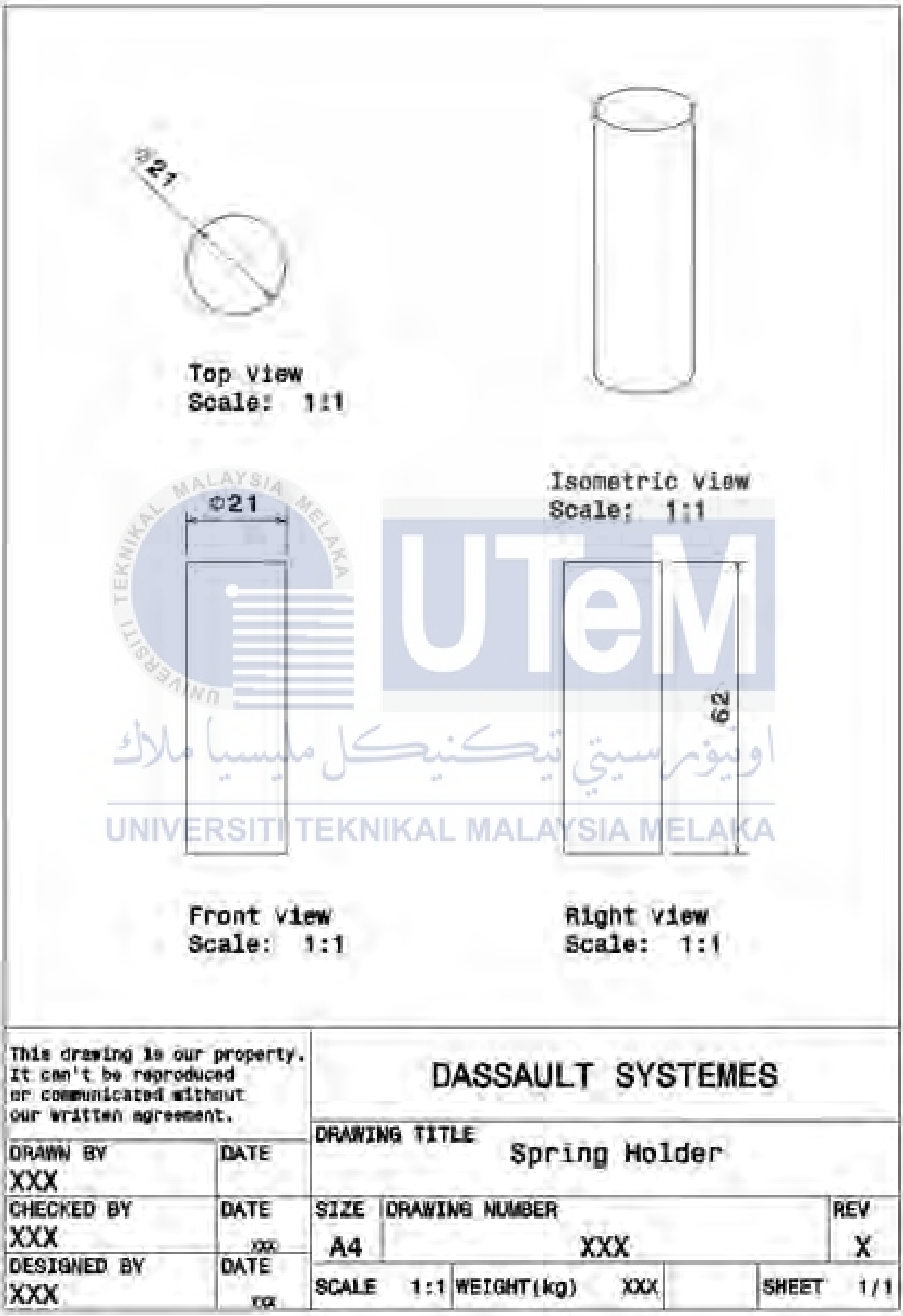


APPENDIX B

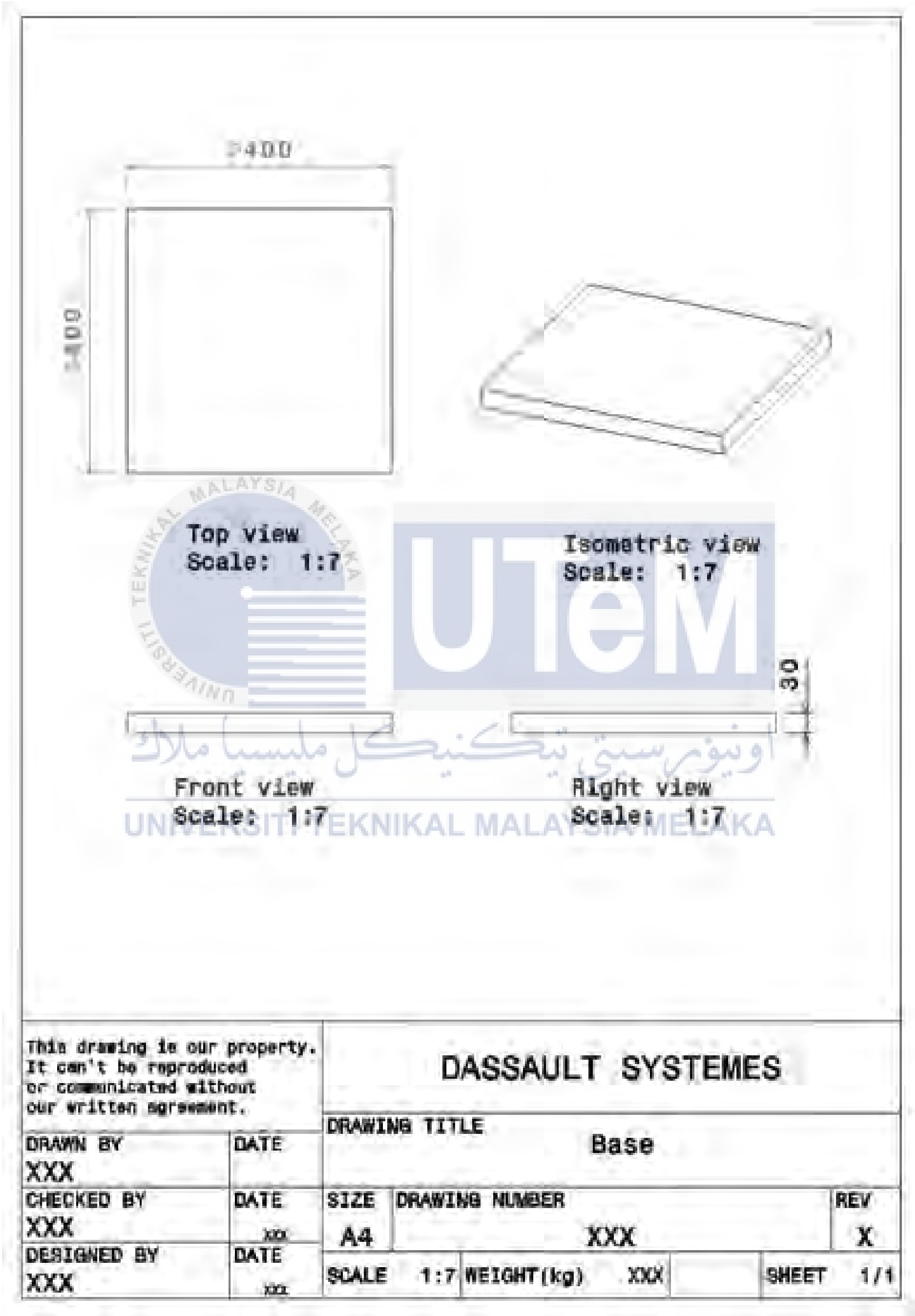


This drawing is our property. It can't be reproduced or communicated without our written agreement.		DASSAULT SYSTEMES					
DRAWN BY XXX	DATE	DRAWING TITLE Spring					
CHECKED BY XXX	DATE XXX	SIZE A4	DRAWING NUMBER XXX				REV X
DESIGNED BY XXX	DATE XXX	SCALE 1:1	WEIGHT(kg) XXX			SHEET 1/1	

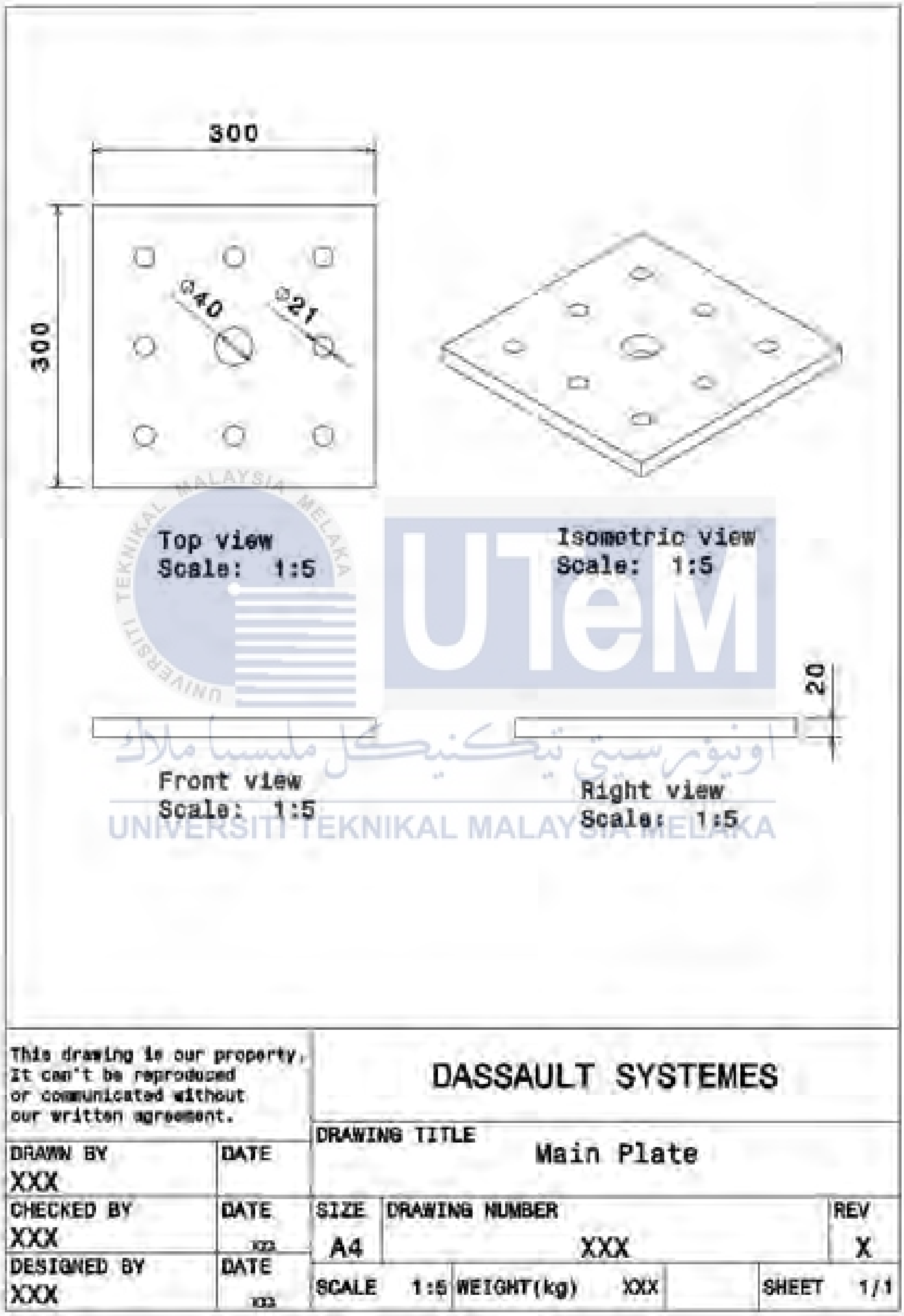
APPENDIX C


Ø21
Top View
Scale: 1:1
Isometric View
Scale: 1:1
Ø21
62
Front View
Scale: 1:1
Right View
Scale: 1:1
This drawing is our property.
It can't be reproduced
or communicated without
our written agreement.
DASSAULT SYSTEMES
DRAWN BY
XXX
DATE
DRAWING TITLE
Spring Holder
CHECKED BY
XXX
DATE
XX
SIZE
A4
DRAWING NUMBER
XXX
REV
X
DESIGNED BY
XXX
DATE
XX
SCALE 1:1
WEIGHT(kg) XXX
SHEET 1/1

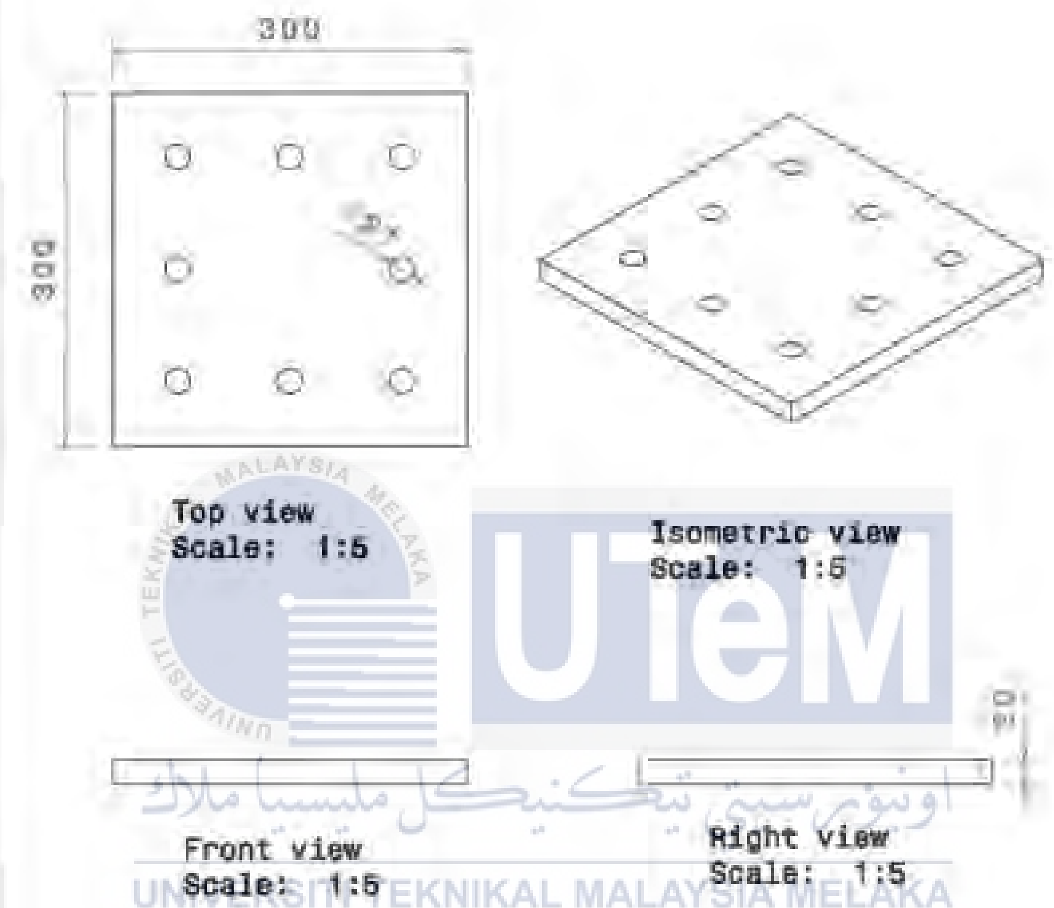
APPENDIX D


Top view
Scale: 1:7
Isometric view
Scale: 1:7
30
Front view
Scale: 1:7
Right view
Scale: 1:7
This drawing is our property.
It can't be reproduced
or communicated without
our written agreement.
DASSAULT SYSTEMES
DRAWING TITLE
Base
DRAWN BY
XXX
DATE
CHECKED BY
XXX
DATE
XXX
DESIGNED BY
XXX
DATE
XXX
SIZE
A4
DRAWING NUMBER
XXX
REV
X
SCALE 1:7
WEIGHT(kg) XXX
SHEET 1/1

APPENDIX E


300
300
Ø40
Ø21
Top view
Scale: 1:5
Isometric view
Scale: 1:5
20
Front view
Scale: 1:5
Right view
Scale: 1:5
This drawing is our property, It can't be reproduced or communicated without our written agreement.
DASSAULT SYSTEMES
DRAWING TITLE
Main Plate
DRAWN BY
XXX
DATE
CHECKED BY
XXX
DATE
XXX
DESIGNED BY
XXX
DATE
XXX
SIZE
A4
DRAWING NUMBER
XXX
REV
X
SCALE 1:5
WEIGHT(kg) XXX
SHEET 1/1

APPENDIX F



This drawing is our property. It can't be reproduced or communicated without our written agreement.		DASSAULT SYSTEMES			
DRAWN BY XXX	DATE	DRAWING TITLE	Lower Spring Holder		
CHECKED BY XXX	DATE XXX	SIZE A4	DRAWING NUMBER XXX		REV X
DESIGNED BY XXX	DATE XXX	SCALE 1:5	WEIGHT(kg) XXX		SHEET 1/1