OPTIMIZATION OF ENGINE MOUNTING BRACKET LOCATION

ILYAS BIN ISHAK

UNIVERSITI MALAYSIA PAHANG

OPTIMIZATION OF ENGINE MOUNTING BRACKET LOCATION

ILYAS BIN ISHAK

A thesis submitted in fulfillment of the
requirements for the award of the degree of
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Alamat Tetap:

No.33 Felda Bukit Sagu 2,
25730 Kuantan,
Pahang,

ZAMRI BIN MOHAMED

Tarikh: **10 NOVEMBER 2008**

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Signature
Name of Supervisor: MR. ZAMRI BIN MOHAMED
Position: LECTURER
Date: 10 NOVEMBER 2008

Signature
Name of Panel: MR. AZIZUDDIN BIN ABD AZIZ
Position: LECTURER
Date: 10 NOVEMBER 2008

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Name: ILYAS BIN ISHAK
ID Number: MH05036
Date: 10 NOVEMBER 2008

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ABSTRACT

One of the main causes of vibration that produce by a car is engine and transmission. Engine and transmission location determination is an important thing that must be considered for a car. Method of locating engine and transmission will be proposed to determine the right position. This method depends to the size and type of engines. The project objective is to find a location for the engine mounting bracket for a new car. It is important to reduce the vibration from the engine and transmission to the car's body or chassis. These is also important to fit the engine bay before consider the component of engine clearance with component of body. The stress analysis is done for the location test, different thickness of side member, and different cross section of side member. The FEA software Algor is use for this analysis and Solidwork to modeling the side member. Five different cross section and design are use to analyze the location of the engine mounting bracket. The range location of the mounting bracket is at the middle of the side member. This range depends to the packaging of the engine and the clearance between the components of engine with engine bay. Thickness of the side member also important to decrease the weight of the car and to minimize the stress value when load apply. For conclusion, not all the car has same location of mounting bracket location and many thing must consider before determine this location. The designs of side member influence the location of the mounting bracket.

ABSTRAK

Salah satu factor yang menyebabkan getaran oleh kereta ialah enjin dan system gear. Kedudukan enjin adalah perkara yang paling penting untuk sesebuah kereta. Cara untuk meletakkan enjin dicadangkan untuk mendapat kedudukan yang betul dan sesuai. Cara ini bergantung kepada saiz dan jenis sesebuah enjin. Objektif projek ini adalah untuk menentukan tempat dan kedudukan enjin *mounting* untuk kereta baru. Ini sangat penting untuk mengurangkan getaran daripada enjin kepada rangka kereta. Ini juga penting untuk menyesuaikan kawasan untuk meletakkan enjin sebelum memastikan jarak komponen enjin dengan rangka atau badan kereta. Analisis tekanan dibuat untuk menganalisis lokasi, perbezaan ketebalan, dan perbezaan keratin rentas untuk *side member.* perisian FEA, Algor digunakan untuk membuat analisis dan Solidwork digunakan untuk membuat model *side member.* Lima keratin rentas *side member* digunakan untuk analisis kedudukan untuk *mounting*. Kawasan untuk meletakkan *mounting* adalah ditengah-tengah *side member*. Kawasan ini bergantug kepada jarak komponen enjin dengan kawasan meletakkan enjin. Ketebalan *side member* adalah penting untuk mengurangkan berat sesebuah kereta dan untuk meminimumkan nilai tekanan apabila beban dikenakan. Kesimpulannya, tidak semua kereta mempunyai kedudukan *mounting* yang sama dan banyak perkara perlu diambil kira sebelum menentukan kedudukannya. Reka bentuk *side member* mempengaruhi kedudukan *mounting* sesebuah enjin.

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LIST OF ABBREVIATIONS

3D	Three Dimension
AISI	America Iron and Steel Institute
FEA	Finite Element Analysis
FYP	Final Year Project
KMC	Knowledge Management Centre
UHSS	Ultra High Strength Steel
UMP	Universiti Malaysia Pahang
UTS	Ultimate Tensile Strength
VPG	Virtual Proven Ground

CHAPTER 1

INTRODUCTION

1.1 ENGINES LAYOUT

There are two major characteristics of a drivetrain that impact the performance of a car. First is the engine placement and second is the driving wheels location. The engine placement is a big factor to determine the moment of inertia and the weight distribution of car because many other mechanical and electrical components of a car are usually located closed to the engine. Engines are placed in one of four locations on vehicles. The locations are rear mounted, mid engine, linear mount, and transverse mount.

A transverse engine is an engine in which the crankshaft is oriented side to side relative to the length of the vehicle. Most modern front wheel drive vehicles use this engine orientation while most rear wheel drive vehicle use a front to back longitudinal arrangement. The transverse engine is simply an engine that is mounted sideways so the output shaft from the transmission can connect to the front wheels in a front wheel drive car. The transverse engines were developed to improve fuel economy on four and six cylinder engines by pulling a car along the road and not pushing it. The other major reason for this was making the cars easier to manufacture on the assembly lines.

Basically it is taking the mid and rear mounted engine technologies and combining them.

1.2 ENGINE AND TRANSMISSION SIZE

Engine size was depending to the type of the engine, numbers of piston and valve, the arrangement of the piston and volume of the engine that will normally state in cubic centimeters (cc). Not all the engines were suitable for all types of cars. It depends to the chassis and design of the cars. The carburetor engines are the large size of the engine while an injection engine is compact engine and has a small size.

Transmission size was depending to the type of engine. Usually the same type of car has a same size of transmission. The engine and transmission size was important thing and this must consider to determine the mounting location and engine bay area.

1.3 PROJECT BACKGROUND

Engine and transmission location determination is an important step in car development. It can affect the vibration level of the car body. For a new car, the location of the engine mounting is an important factor and it must be analyses first. To analyze it, the center of gravity of the engine is proposed to be determined, so as the moment of inertia of the engine. These things can be very important to reduce the vibration through the chassis.

Engine vibration can also depend to the size and power of the engine. That explain why the proper engine location is substantial for a car. The method and calculation for the engine location can be determined to get the suitable engine location. All the type of the car can use this method to determine the engine location. A simulation test can be performed for the engine based on the method and numerical value.

1.4 PROBLEM STATEMENT

Method of locating engine and transmission will be proposed to determine the right position. This method depends to the size and type of engines. These could be many method to locate the engine and transmission but not widely known in literature.

All the cars are assembled at the factory and the method is usually kept as an engineering known how to the research and development department. Every company can have their own method for the different types of car that will be produced. The performance of the vibration and harshness characteristics can be depending to this method and procedure. That can explain some cars to have customer complaints on that aspect.

At the workshop, the mechanics usually locate the engine and transmission based on their commonsense and engine bay space. The trial and error method are commonly use by the mechanics. The things that are usually considered are the mounting location and the size of the engine.

Based on the method of locating engine and transmission we can analyze the engine location based on CAE method. This method will enhance automotive design and important for a repair process.

1.5 PROJECT OBJECTIVES

The project objective is to find a proper method to install a new engine and transmission for a new car. It is important to reduce the vibration from the engines and transmission to the car’s body. The scope for this project is to study on the vibration reduction from the engines system to the car’s body. The other scope is this analysis is for transverse engine mounting only. The stress analysis is done for side member of the car to find the suitable or proper location for engine mounting bracket. Solidwork is use to modeling the side member from the benchmark side member. The stress analysis is done for the model with the FEA software Algor. The analysis is considering the point location on the side member, the thickness of the side member, and the different cross section of side member.

CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

This chapter will provide detail description of literature review done regarding the project title of optimization of engine mounting bracket location.

2.2 EQUATIONS OF MOTION OF THE ENGINE

The general translational equations of motion of the engine are

$$Mr = f \tag{1}$$

where $f = [f_x, f_y, f_z]^T$ is the sum of all forces acting on the engine block and $r = [x, y, z]^T$ is the position of C.

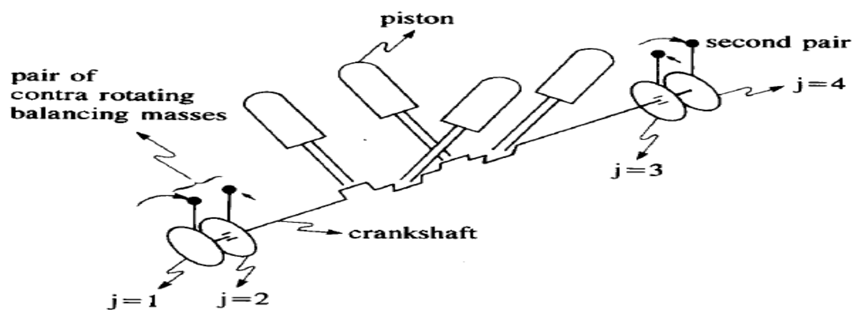

Figure 1: Schematic representation of balancing system

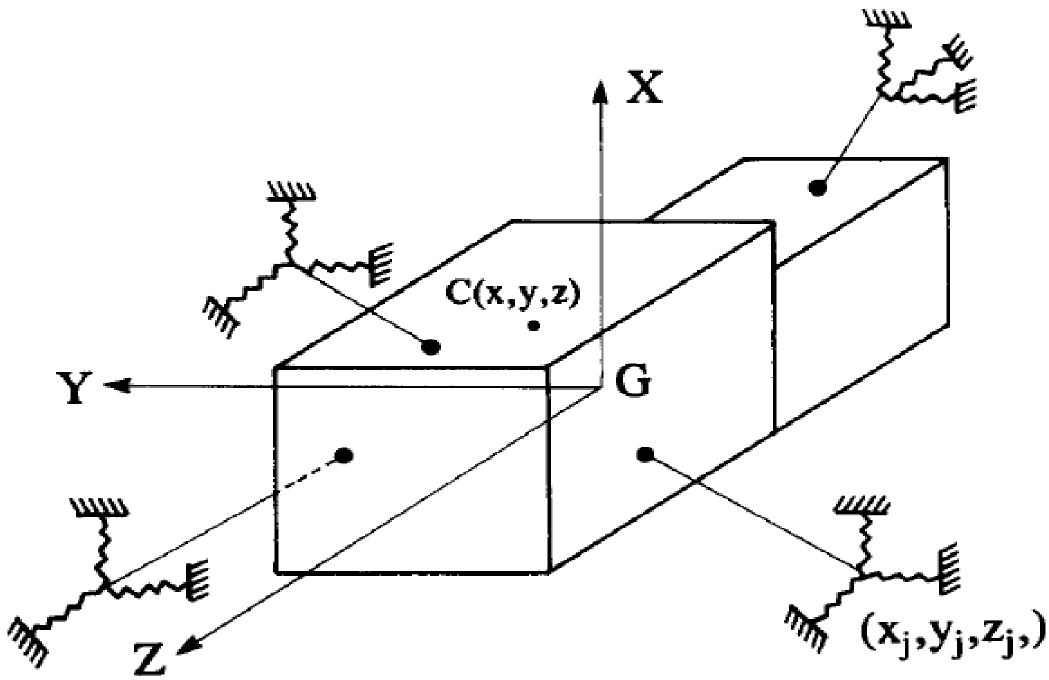

Figure 2: Rigid body engine model

The general rotational equations of motion are

$$I\acute{\omega} + \omega \times I\omega = \eta \tag{2}$$

where $\eta = [\eta_x, \eta_y, \eta_z]^T$, is the sum of the moments of the individual forces about C, $\omega = [\dot{\theta}_x, \dot{\theta}_y, \dot{\theta}_z]^T$ the angular velocity with θ_x, θ_y and θ_z the rotational angles about the x, y and z-axis respectively. I is the tensor of the moments and products of inertia given by

$$\boldsymbol{I} = \begin{bmatrix} I_{xx} & -I_{xy} & -I_{xz} \\ -I_{yx} & I_{yy} & -I_{yz} \\ -I_{zx} & -I_{zy} & I_{zz} \end{bmatrix} \tag{3}$$

If we assume that the coordinate axes of the engine are coincident with the principal axes of inertia then the products of inertia fall away and equation (2) may be written as

$$\begin{aligned} \eta_x &= I_{xx}\ddot{\theta}_x - (I_{yy} - I_{zz})\dot{\theta}_y\dot{\theta}_z \\ \eta_y &= I_{yy}\ddot{\theta}_y - (I_{zz} - I_{xx})\dot{\theta}_z\dot{\theta}_x \\ \eta_z &= I_{zz}\ddot{\theta}_z - (I_{xx} - I_{yy})\dot{\theta}_x\dot{\theta}_y \end{aligned} \tag{4}$$

which are known as Euler's equations.

The force elements acting on the engine body and giving rise to the resultant force f and moment η are of two kinds. Shaking forces (inertial forces) and moments are generated by the moving links of the slider-crank mechanisms and the rotating balancing masses. In addition reaction forces are applied to the engine frame at each of the supporting mounts. The forces generated at the supports are considered first.

2.2.1 Reactions at the engine mounts

Suppose the engine is supported by N_m mounts positioned at (X_{mi}, Y_{mi}, Z_{mi}), i = 1, 2 . . . . . N_m. It is assumed that the stiffnesses k_{xi}, k_{yi} and k_{zi} in the three coordinate directions are independent of each other. The forces f_{mi} on the engine exerted by the elastic mounts are then given by the components

$$
\begin{aligned}
f_{xmi} &= -k_{xi}(x + z_{mi}\theta_y - y_{mi}\theta_z),\\
f_{ymi} &= -k_{yi}(y + x_{mi}\theta_z - z_{mi}\theta_x)\\
f_{zmi} &= -k_{zi}(z + y_{mi}\theta_x - x_{mi}\theta_y), \quad i = 1, 2, \ldots, N_m
\end{aligned}
\tag{5}
$$

and the components of the sum of their moments η_{mi} about the coordinate axes are

$$
\begin{aligned}
\eta_{xmi} &= f_{zmi}y_{mi} - f_{ymi}z_{mi}\\
\eta_{ymi} &= f_{xmi}z_{mi} - f_{zmi}x_{mi}\\
\eta_{zmi} &= f_{ymi}x_{mi} - f_{xmi}y_{mi}
\end{aligned}
\tag{6}
$$

where it is assumed that the displacements at the supports are small compared to their distances from C. The resultant components f_{xm}, f_{ym}, f_{zm} and η_{xm}, η_{ym} and η_{zm} are obtained by summing over i. Next the inertial forces and moments due to the motion of the crank-conrod-piston assemblies are considered.

2.2.2 Inertial forces and moments exerted by the V-engine

Initially consider only the forces and moments which arise as a result of the action of a single piston moving in a vertical cylinder. The single cylinder engine model, depicted in Fig. 3 is adopted.

L = conrod length

R = crank length

m_1, m_2, m_3 = masses of crank, conrod and piston

JG = centroidal moment of inertia of the conrod

m_A, m_B = equivalent conrod tip masses

$J_{AB} = J_G - m_2ab$ = moment of inertia associated with equivalent conrod

θ = engine rotation angle

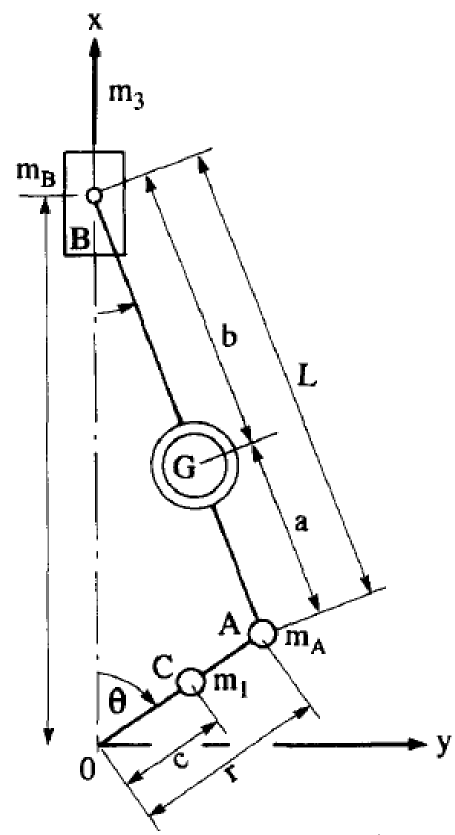

Figure 3: Single cylinder engine model

Of interest are the force components (X_f, Y_f) exerted by the single piston on the engine frame and the resultant inertial torque M_z of the moving parts about the crankshaft centre line O. In the case of the single piston engine the engine rotation angle is taken as the associated crank angle. Assuming the crank angular velocity ω to be approximately constant,

$$X_f = (m_{rot} + m_{rec})r\omega^2 \cos\theta + m_{rec} r\omega^2(A_2 \cos 2\theta - A_4 \cos 4\theta + A_6 \cos 6\theta - \cdots)$$

$$Y_f = m_{rot} r\omega^2 \sin\theta$$

$$M_z = M_f + M_r = -J_{AB}\lambda\omega^2(C_1 \sin\theta - C_3 \sin 3\theta + C_5 \sin 5\theta - \cdots) \tag{7}$$

where M_f is the torque exerted by the engine on the frame and M_r is the resisting moment on the crank, $m_{rec} = m_3 + m_B$, $m_{rot} = m_1\ c/r + m_A$, $\lambda = r/L$ and the As and Cs are appropriate Fourier coefficients.

The above is now extended and generalized to a four cylinder V-engine with V-angle equal to β. In this case all forces and moments must refer to a common global right hand coordinate system. Such a coordinate system is depicted in Fig. 4.

Quantities relating to the individual cylinders, respectively rotated through angles -β/2, β/2, -β/2 and β/2, are denoted by the subscript i, for i = 1-4, where necessary. It is assumed that the individual cylinder-piston assemblies are physically identical and that the axial distance between any two adjacent cylinders is a constant d. The central point on the crankshaft, halfway between the middle two cylinders, is denoted by O. The position of O with respect to the centre of mass of the engine C is given by $r_0 = [x_0, y_0, z_0]^T$ and the coordinates of the cylinders along the crankshaft, again relative to the centre mass C, are

$$x_{ci} = x_0$$

$$y_{ci} = y_O,\ i = 1\text{-}4$$

$$z_{c1} = z_o - 3d/2$$

$$z_{c2} = z_o - d/2$$

$$z_{c3} = z_o + d/2$$

$$z_{c4} = z_o + 3d/2 \quad (8)$$

The position vector for each cylinder is therefore given by $r_{ci} = [x_{ci}, y_{ci}, z_{ci}]^T$.

Since the individual cylinders are identical the expressions for the inertial forces and moments for each cylinder i, X_{fi}, Y_{fi} and M_{zi}, with reference to their respective local coordinate systems and given by subscripted equations (7), are identical, except that in each case the crank angle θ_i will be different and is given by

$$\theta_i = \theta + \delta_i \quad (9)$$

where δ_i denotes the crank angle of piston i when the engine rotation angle θ is equal to zero. In particular $\delta_1 = 0$. Note that the crankshaft angular velocity $\dot{\theta}_i = \omega_i = \omega$ is of course the same and assumed constant for all i. Transforming to the global coordinate system the components of the forces f_{ci} due to each individual cylinder i are given by

$$f_{xci} = X_{fi} \cos\frac{\beta}{2} + (-1)^{i+1} Y_{fi} \sin\frac{\beta}{2}$$

$$f_{yci} = (-1)^i X_{fi} \sin\frac{\beta}{2} + Y_{fi} \cos\frac{\beta}{2}$$

$$f_{zci} = 0 \quad \text{for } i = 1\text{–}4 \quad (10)$$

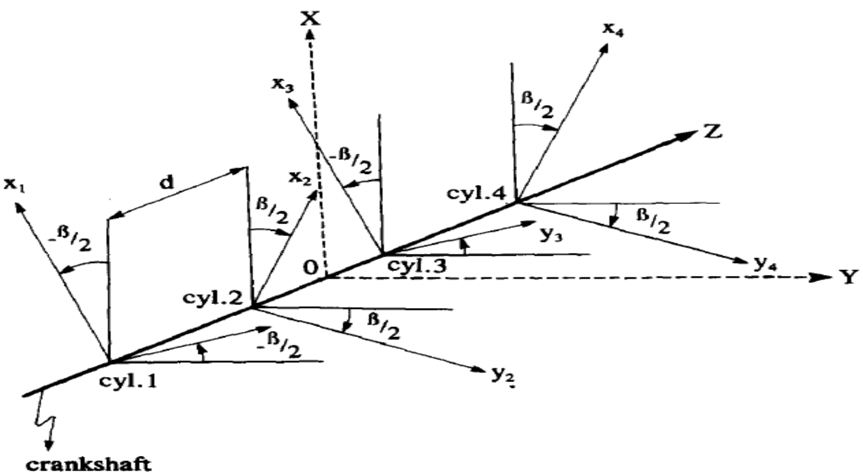

Figure 4: Global coordinate system XYZ

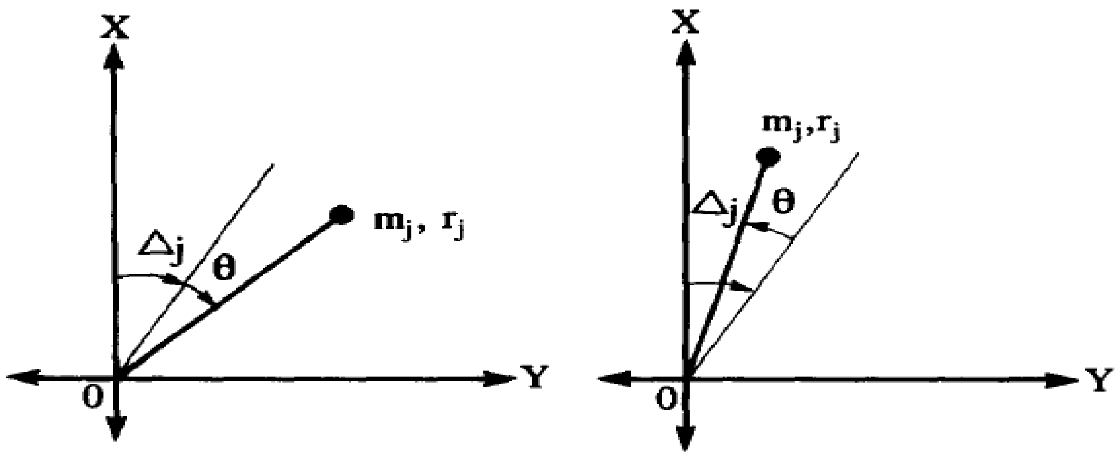

Figure 5: Clockwise (a) and anticlockwise (b) rotating balancing masses

The components of the resultant inertial force f_c are obtained by the summations

$$f_{xc} = \sum_i f_{xci}$$

$$f_{yc} = \sum_i f_{yci} \tag{11}$$

with $f_{zc} = 0$.

The resultant torque about the crankshaft, also with reference to equations (7), is given by

$$M_{z0} = \sum_i M_{zi} \tag{12}$$

With the inertial forces f_{ci} of the individual cylinders and the positions r_{ci} at which they act known, the resultant moment η_c about C due to the V-engine cylinder system may be computed by

$$\boldsymbol{\eta}_c = \sum_i \mathbf{r}_{ci} \times \mathbf{f}_{ci} + M_{z0}\mathbf{k} \tag{13}$$

where k is the unit vector along the z-axis.

2.3 TORQUE ROLL AXIS DECOUPLING CONCEPTS

Most dynamic design principles attempt to place the natural frequencies of the system below and above the excitation frequency range. However, if a system has several resonances within a narrow band, it becomes a rather difficult task to achieve. Therefore, a physically decoupled system has a better chance of producing fewer resonances over the operating range. For example, if a system is completely decoupled in physical modes, excitation along one physical co-ordinate should excite only one mode, as shown in Figure 6(c). Even when two or three different excitations are simultaneously applied to the powertrain, projected excitations along the physical co-ordinates may have only a few non-trivial values. Hence, physical decoupling of the powertrain is often necessary to ensure that vibro-acoustic performance goals will be achieved. Concurrently the mounting system must also satisfy the packaging needs, geometric constraints and static load-bearing requirements.

The elastic axis decoupling scheme is frequently a starting point in the industrial mounting system design practice. Elastic axes for an elastically supported rigid-body system are those axes along which only the displacement (or rotation), collinear with the direction of the applied static force (or moment), occurs as shown in Figure 7(a) and 7(b). If a physical co-ordinate system can be defined using elastic axes Γe and if it coincides with the principal inertial co-ordinate system Γp, the dynamic response consists of three decoupled translational and three rotational modes as shown in Figure 7(c). System mass and stiffness matrices are then diagonal either in Γp or Γe co-ordinates. However, maintains that the elastic center of mounts cannot always exist in a full three-dimensional (3-D) rigid-body system since only the six o!-diagonal terms are forced to vanish in the physical domain. Therefore, the complete decoupling of a practical powertrain by the elastic axis mounting concept is impossible to achieve when one considers the 3-D asymmetric shape of the inertial body and arbitrary placement of the mounts. Consequently, the focalization method, which is related to the elastic axis decoupling concept, has been implemented as it partially decouples a system. Direct design methods of the mounting system also have been tried, using optimization algorithms, regardless of the coupling of physical modes. Applied optimization algorithms to prevent the resonances of the engine natural rigid-body mode from being excited by engine excitation. Attempted to minimize the forces transmitted from the engine to the nacelle in an aircraft.

The true TRA mode decoupling strategy has also been sought by designers and researchers over the past two decades. Requirements for the existence of a decoupled TRA mode have been implemented in a numerical optimization scheme. But they could not obtain a more complete decoupling of the TRA mode. The TRA decoupling mechanisms therefore remain poorly understood and inadequately analyzed.

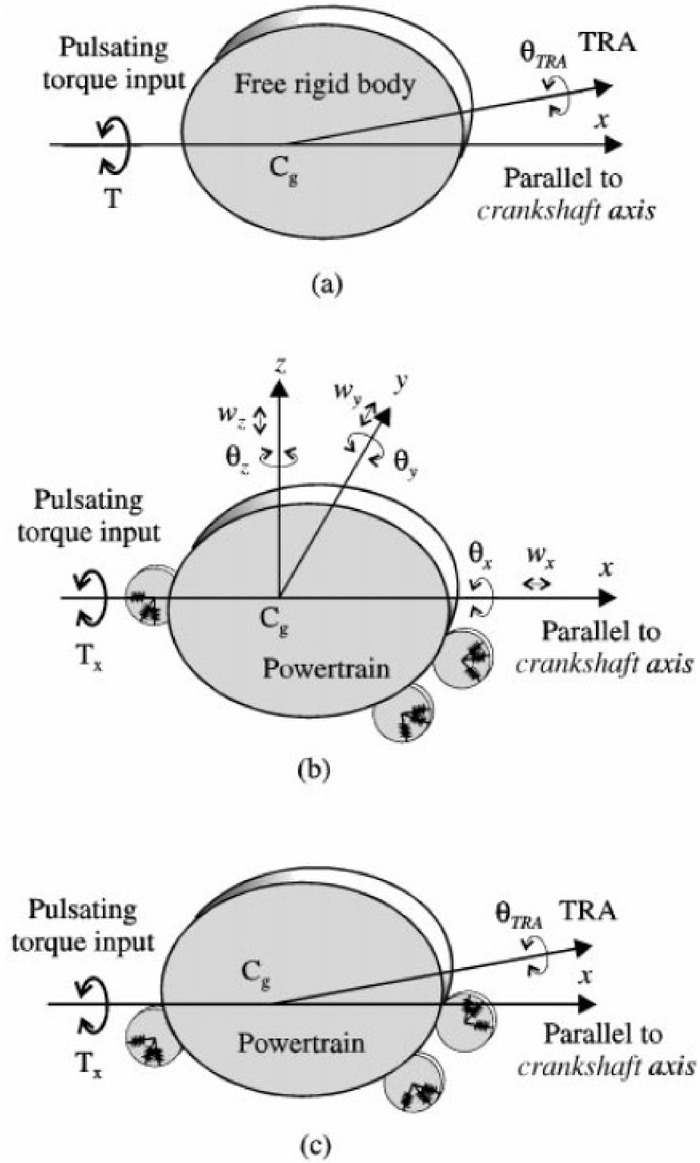

Figure 6: (a) TRA for a free rigid body, (b) coupled powertrain mount system, (c) TRA model decoupled system.

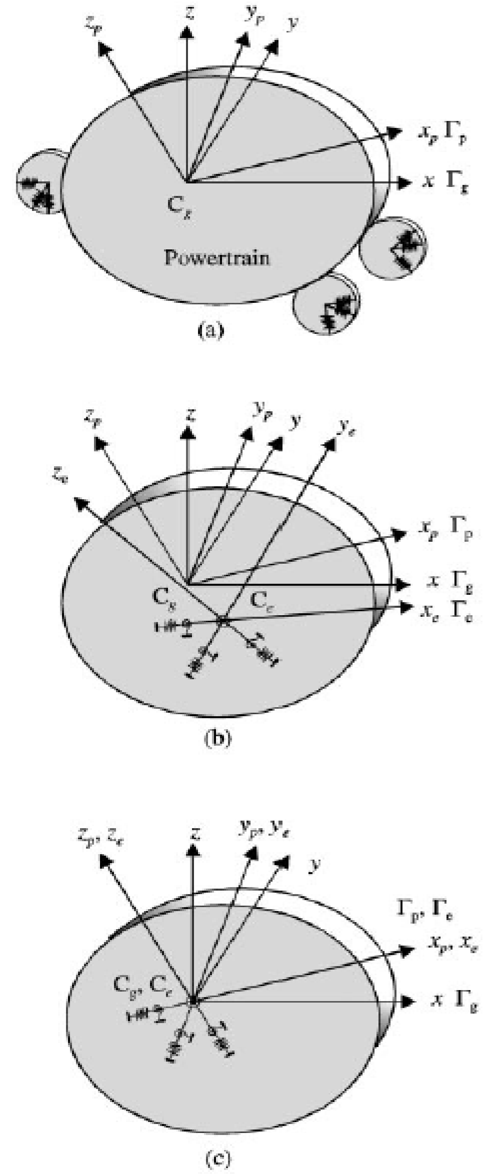

Figure 7: (a) A powertrain mount system, (b) equivalent system representation in the elastic axis, (c) elastic axis decoupled system.

CHAPTER 3

METHODOLOGY

3.1 INTRODUCTION

This chapter will provide detail explanation on the methodology of carrying out this project from the beginning to the end. Title was given by the supervisor in the beginning of this semester including objectives and scopes of the project. Since the title is "Optimization of engine mounting bracket location", a detailed related literature review was done and important information was acquired and explains in previous chapter. This project will start with benchmark. After that the model of side member was design by Solidwork and uses the FEA (Algor) to analysis the stress of the side member to find the suitable location for the engine mounting bracket. This analysis was to find where the point can minimize the stress when load apply at that point.

3.2 OVERALL METHODOLOGY

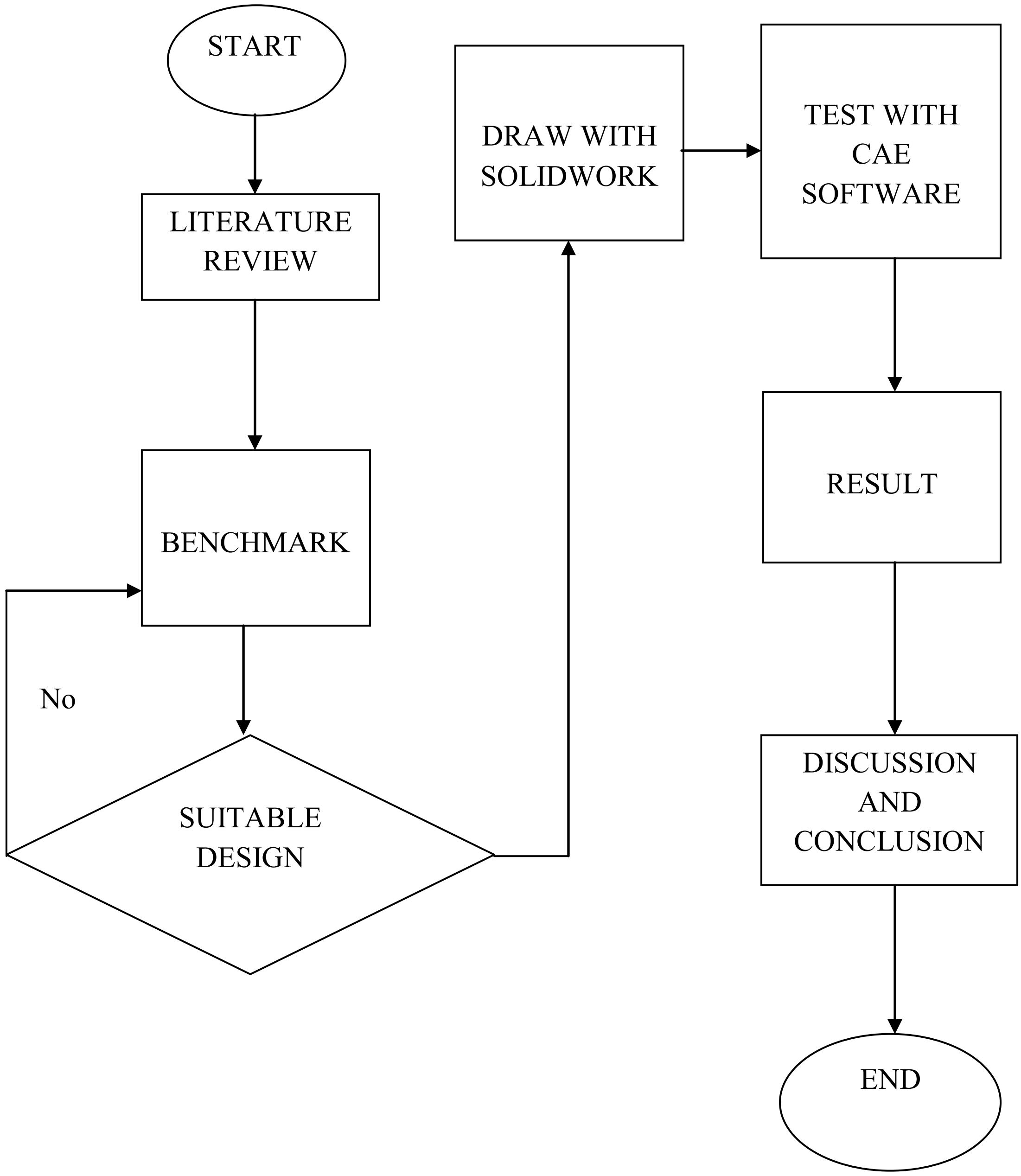

FIGURE 3.2.1: Flow chart

3.3 BENCHMARK

Check and compare engine room clearances based on several model like Toyota car, Honda car and Proton car. After checking, find and decide on suitable clearances for the engine and engine room.

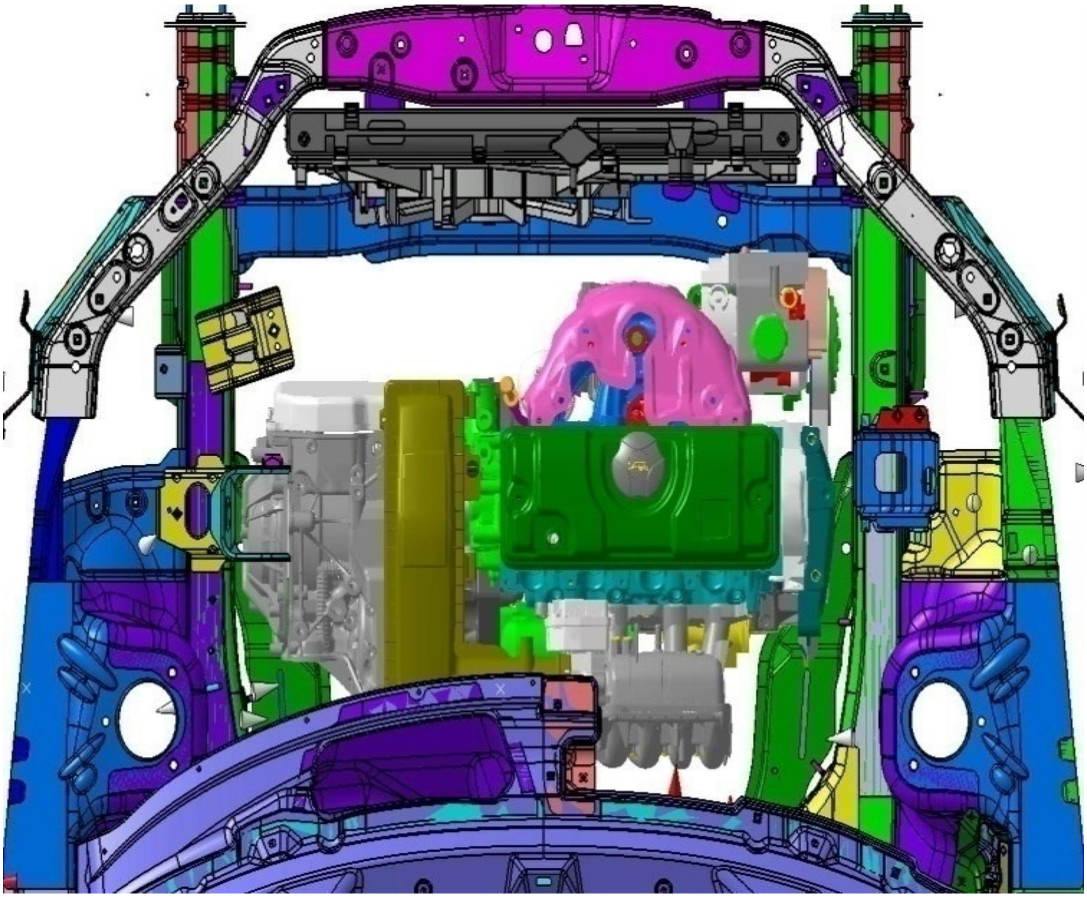

Figure 3.3.1: Drawing for the engine and engine room

Pos	Engine component	Body or chassis component	Clearance
1	FEAD belt	RH side member	30 mm
2	Transmission	LH side member	55 mm
3	Transmission	Dash panel	57 mm
4	Air cleaner top	Hood inner	60 mm
5	Oil cap	Hood inner	154 mm
6	Inlet manifold	Dash panel	108 mm
7	Engine + transmission	Exhaust front	5 mm
8	Oil pan	Subframe	17 mm

Table 3.3.1: Clearance between engine and chassis

3.4 SIDE MEMBER BENCHMARK AND MODELING

Side member is a component for the car to locate the engine bracket mounting. The location of side member is at the left and right side of the car's chassis. After benchmark of the side member design, the Proton Wira was selected. The side member was modeling by Solidwork with the correct dimension.

Figure 3.4.1: Cross section of the side member

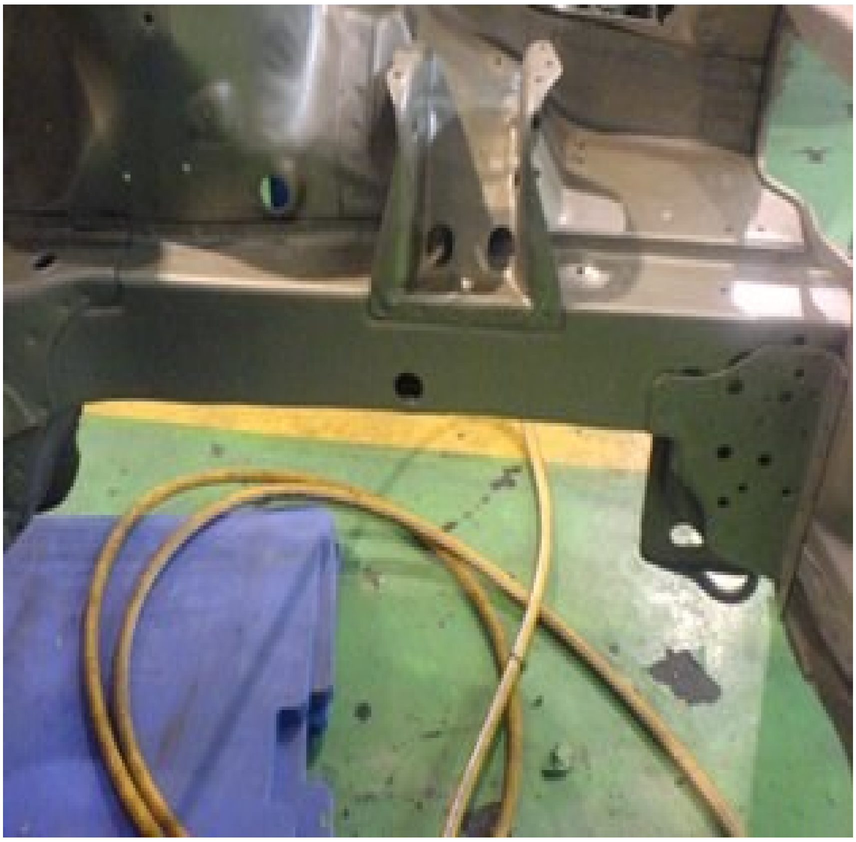

Figure 3.4.2: Side member

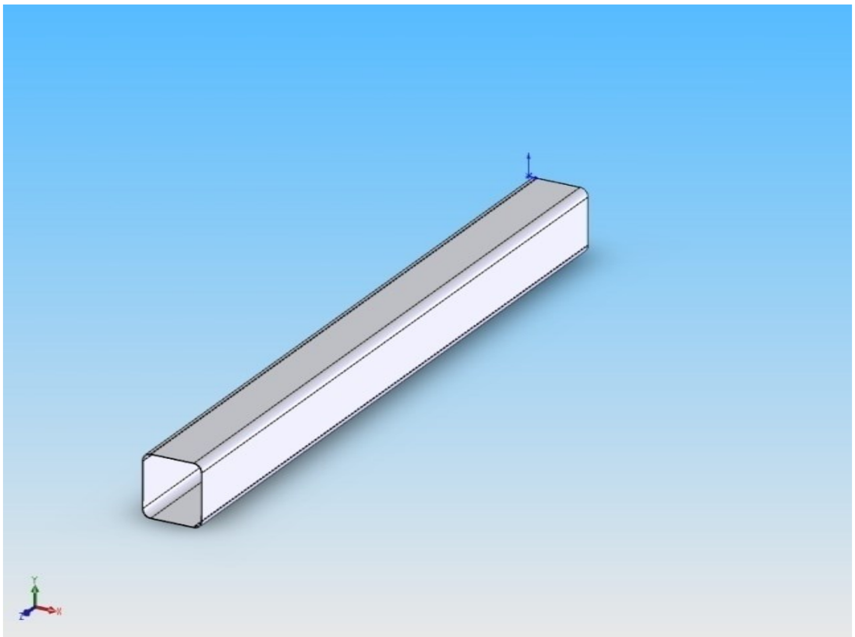

Figure 3.4.3: Model of side member 1

For the first modeling the dimension is same with the real side member. The dimension was 900mm for the length, 80mm for the high, 75 mm for the width, and 0.8mm for the thickness.

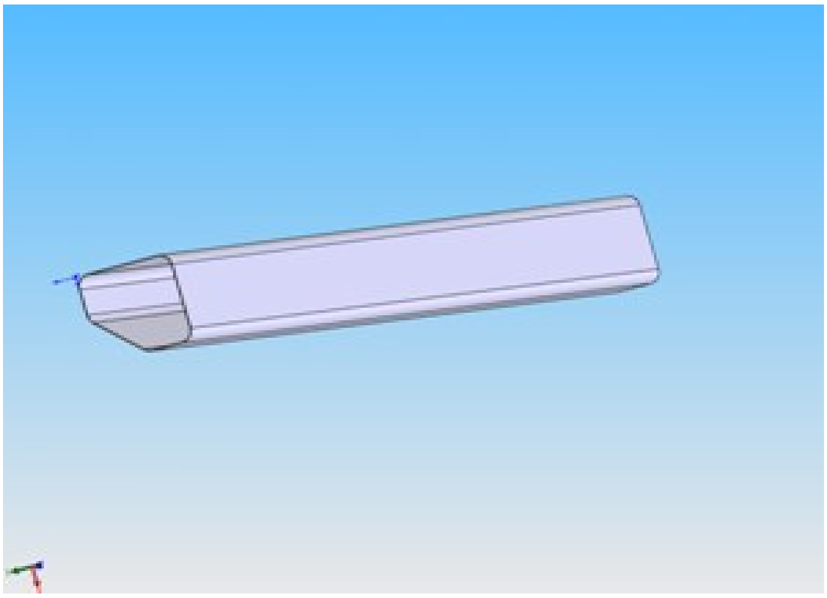

Figure 3.4.4: Model of side member 2

For figure 3.4.4, the cross section of the side member redesign back to get the different value of stress when apply the load. The thickness and length of this side member was same with the figure 3.4.3. The modification for this side member is at the bottom side.

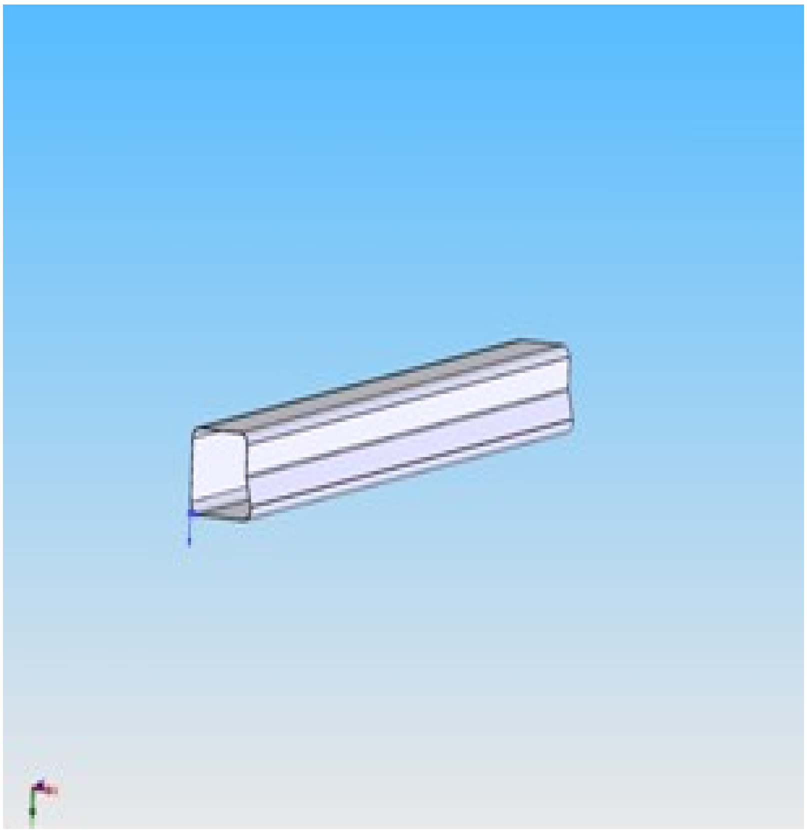

Figure 3.4.5: Model of side member 3

For this model, the modification is at the bottom of the side member. The cross section is wider and has a biggest base to support the load. The thickness and length of this model was also same with the previous one.

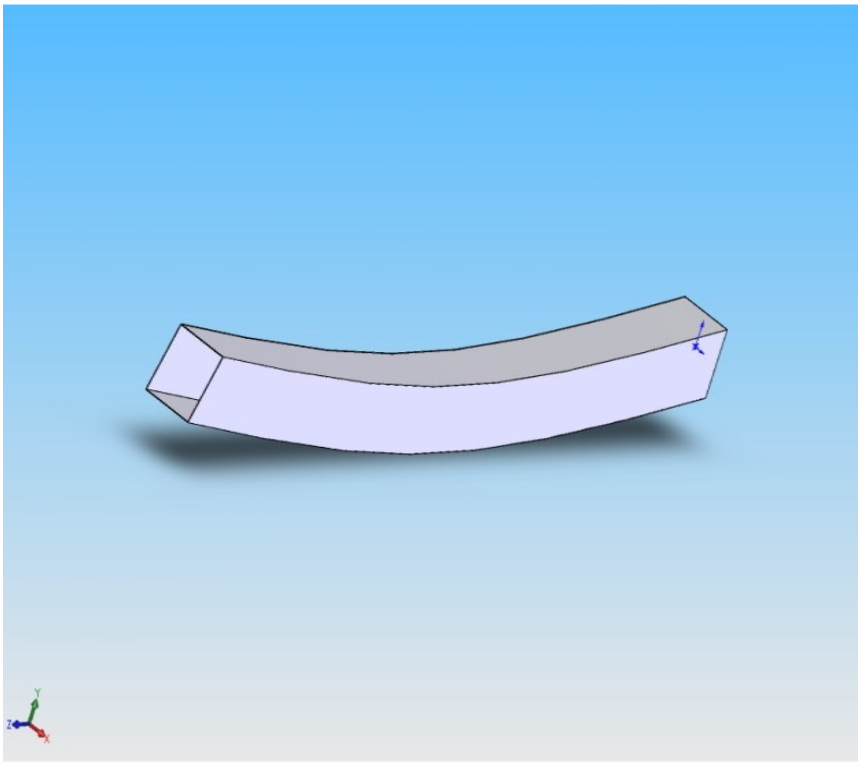

Figure 3.4.6: Model of side member 4

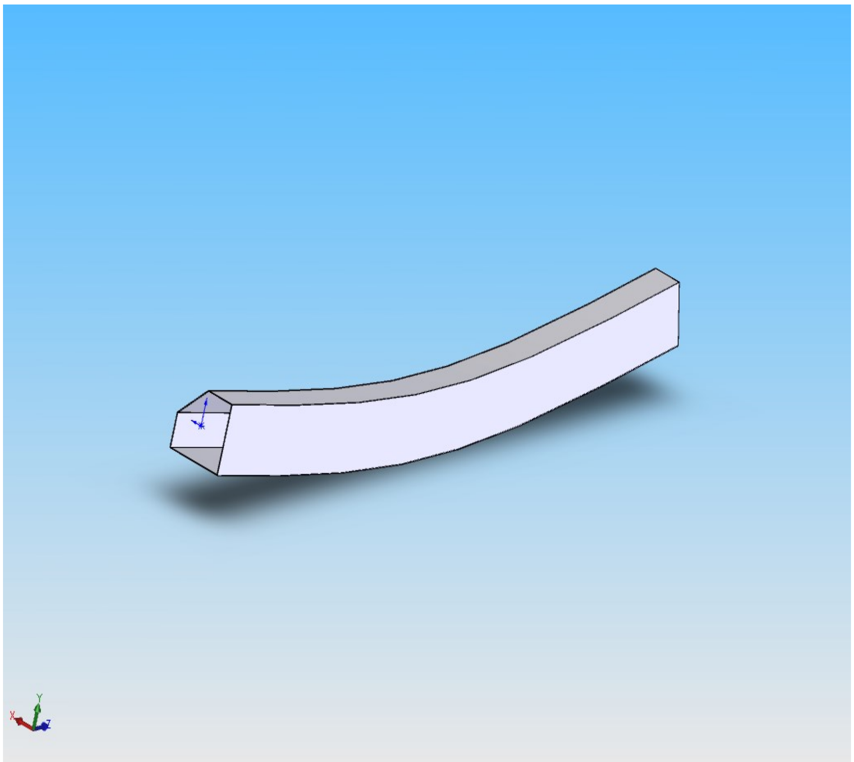

Figure 3.4.7: Model of side member 5

Model at figure 3.4.6 and figure 3.4.7 is a modification from the figure 3.4.3 and figure 3.4.4. The cross section for this model is same but the shape of side member is curved about 50 mm from the center.

3.5 STRESS ANALYSIS FOR THE SIDE MEMBER

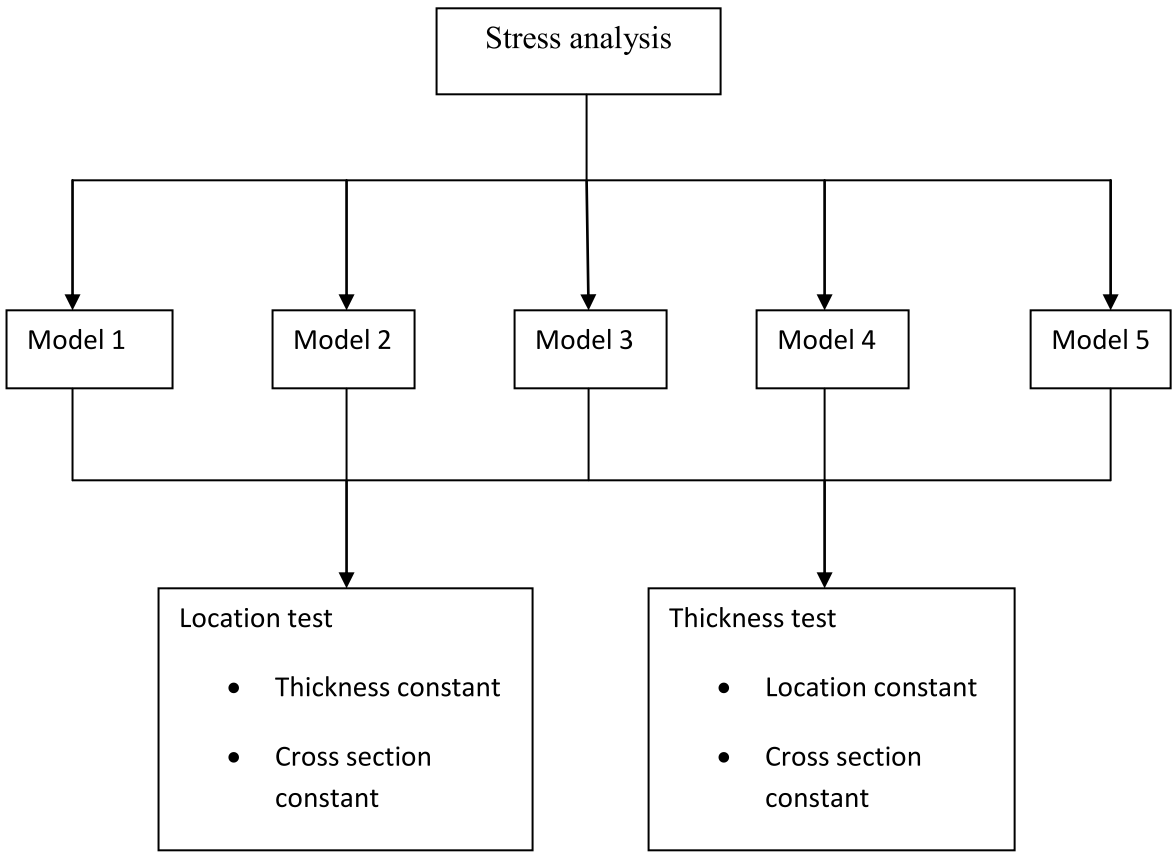

Figure 3.5.1: Flowchart of stress analysis

The stress analysis is to find the suitable location and thickness of the side member. The range of the location point is about 324mm < x < 540mm and the range of the side member thickness is 0.7mm < t < 1.0mm. The load that applies at the side member is 300N. The material that use for this side member is steel AISI 1050 annealed. The end of this side member and the other side is fixed. This boundary condition is to show the condition of the side member at the car's chassis.

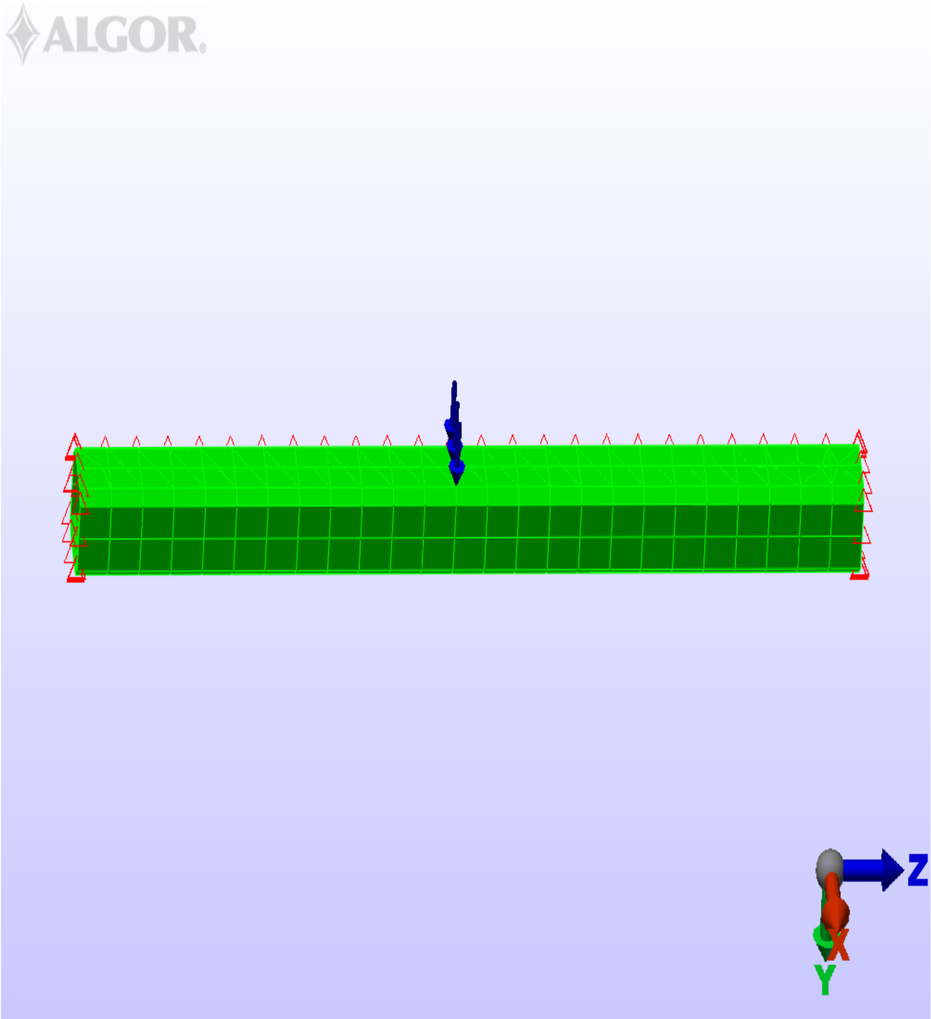

Figure 3.5.2: Boundary condition of the test

The model was import from the Solidwork data and then meshing about 80 percent from the solid. Figure 3.5.2 is to show the boundary condition that applies to all models and for all tests.

CHAPTER 4

RESULT AND DISCUSSION

4.1 INTRODUCTION

This chapter will discuss on the result and discussion that have been made based on the methodology on chapter 3. The discussion will focus on the behavior of the design after the load was applied, to study the effect of different thickness, different location, and different cross section of the side member. The stress analysis for side member is to determine the value of maximum stress at the different location, to determine the stress when the side member has a different thickness, and to determine the stress between the different cross section of side member.

4.2 LOAD APPLY FOR ANALYSIS

An engine may be regarded as having six degrees of freedom of vibration about orthogonal axes through its centre of gravity (COG). In practice, only three modes are usually of importance. The modes are vertical oscillations on the X-axis due to unbalanced vertical forces, rotation about the Y-axis due to cyclic variations in torque, and rotation about the Z-axis due to unbalanced vertical forces in different transverse planes (ENGINE TESTING THIRD EDITION, A.J.Martyr and M.A. Plint).

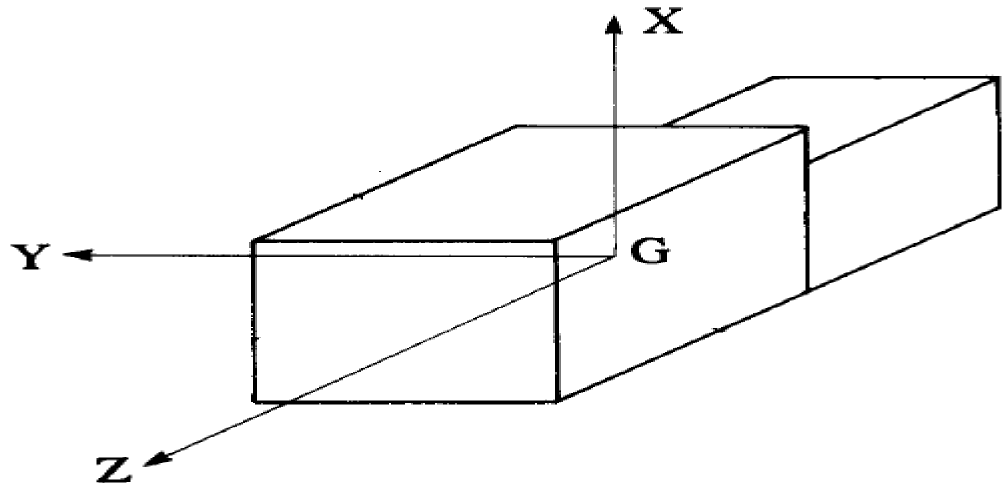

Figure 4.2.1: Rigid body engine model

For this analysis of stress, the first mode was considered. The mode was vertical oscillations on the X-axis due to unbalanced vertical forces. The mass of engine is the high force that attaches the side member through mounting bracket, that why this stress analysis just consider the vertical load. The other two load are the moment forces when the engine was running and this forces are lowest than mass of engine.

4.3 LOCATION TEST

Location test is to determine the suitable location point for the engine mounting bracket. This test was applied at all model of the side member with constant cross section and constant thickness. The result is to find which model or cross section that has a lowest value of stress when the load was apply and also to determine the point for the engine mounting bracket location.

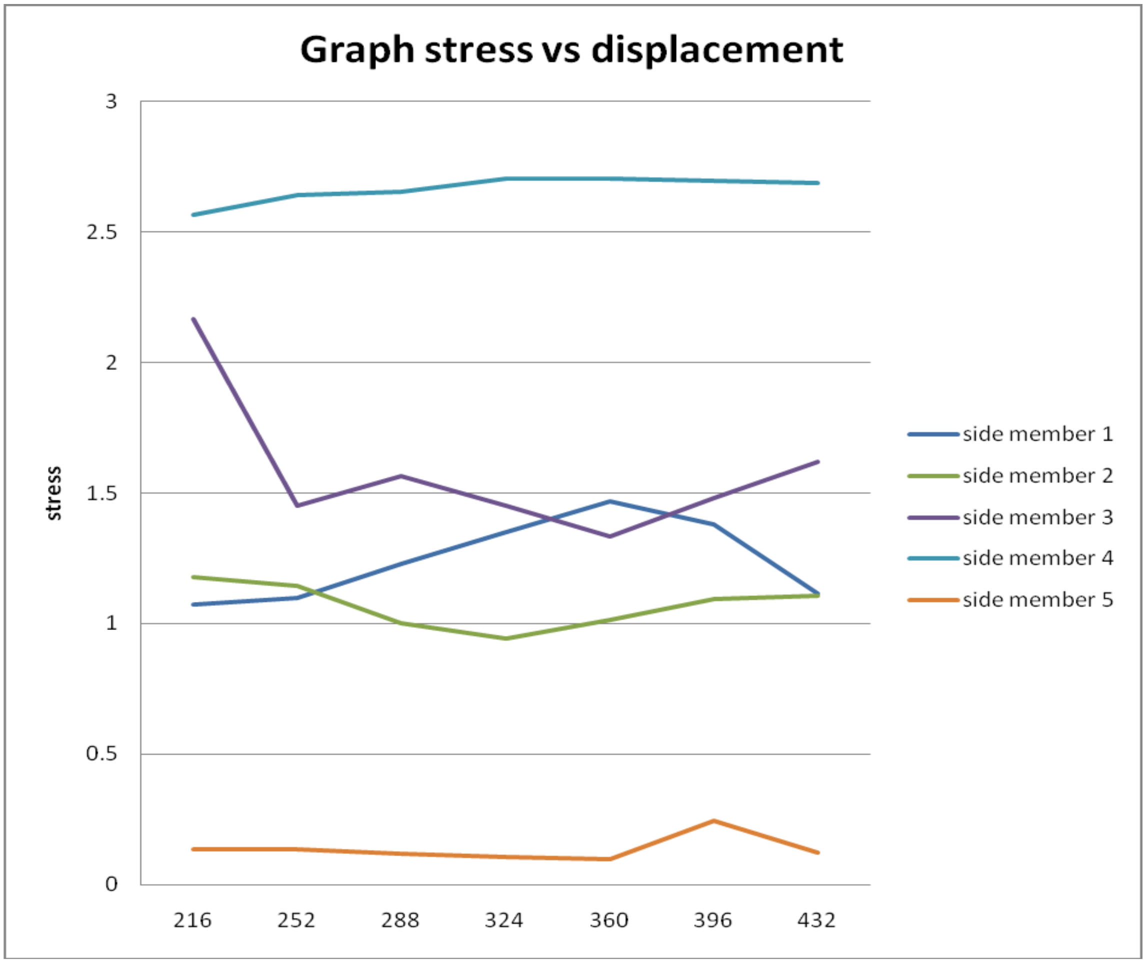

Figure 4.3.1: Graph stress vs displacement

Side member 1 has a lowest value of stress at the 216mm of the side member. This result was influence by the design of this model. The model have a rectangular cross section like a real side member cross section and straight bar. This model has a minimum stress value when the point of load apply is near to the fixed point. For the side member 4, the sross section is same with side member 1 but this side member is curved about 50mm from the center. This modification was made up to see the different stress value when the load is apply at the same point of the side member. From the graph, side member 1 have less value of stress then side member 4. This mean that the modification is not effective and must consider a new cross section. The point that have lowest value of stress is same with side member 1 about 216mm.

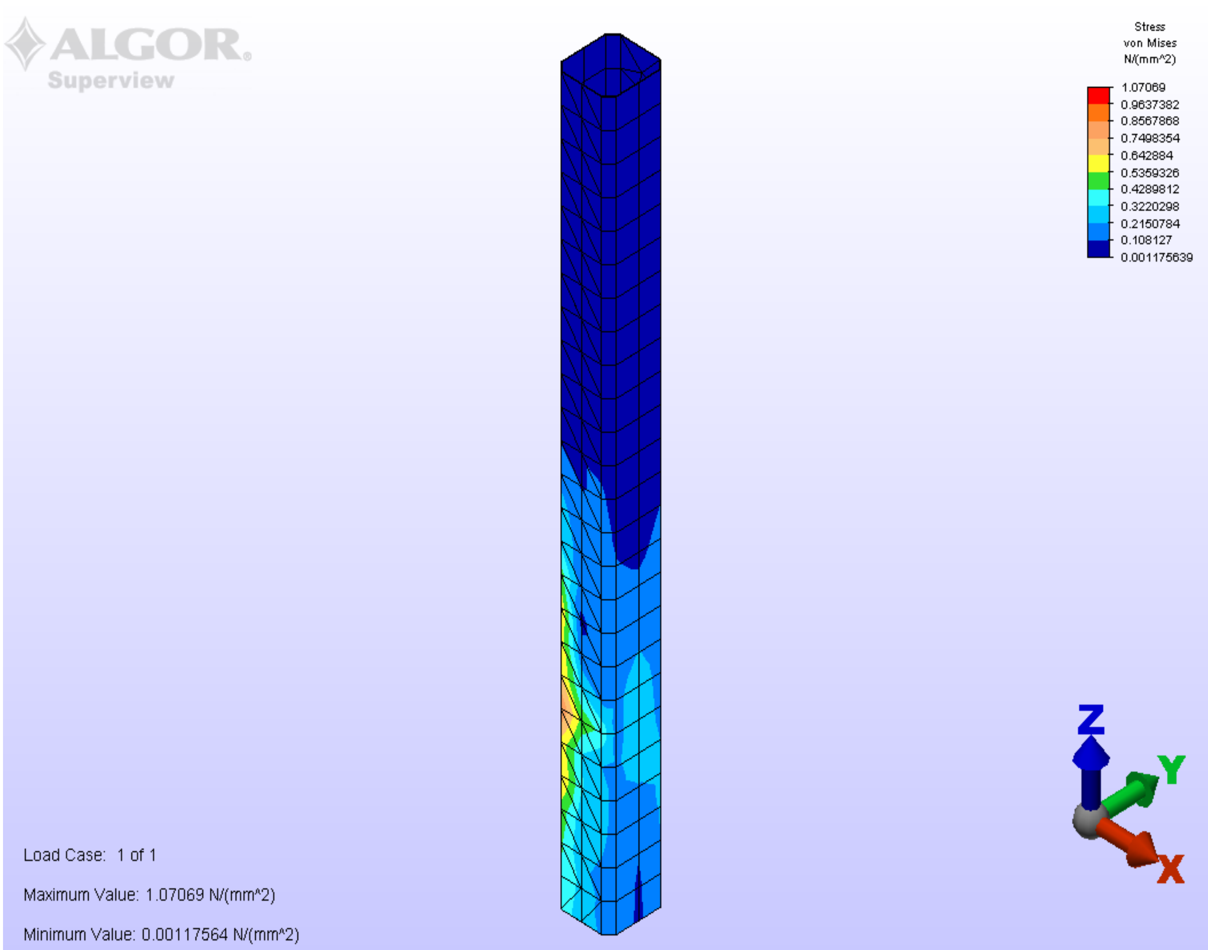

Figure 4.3.2: Result for side member 1

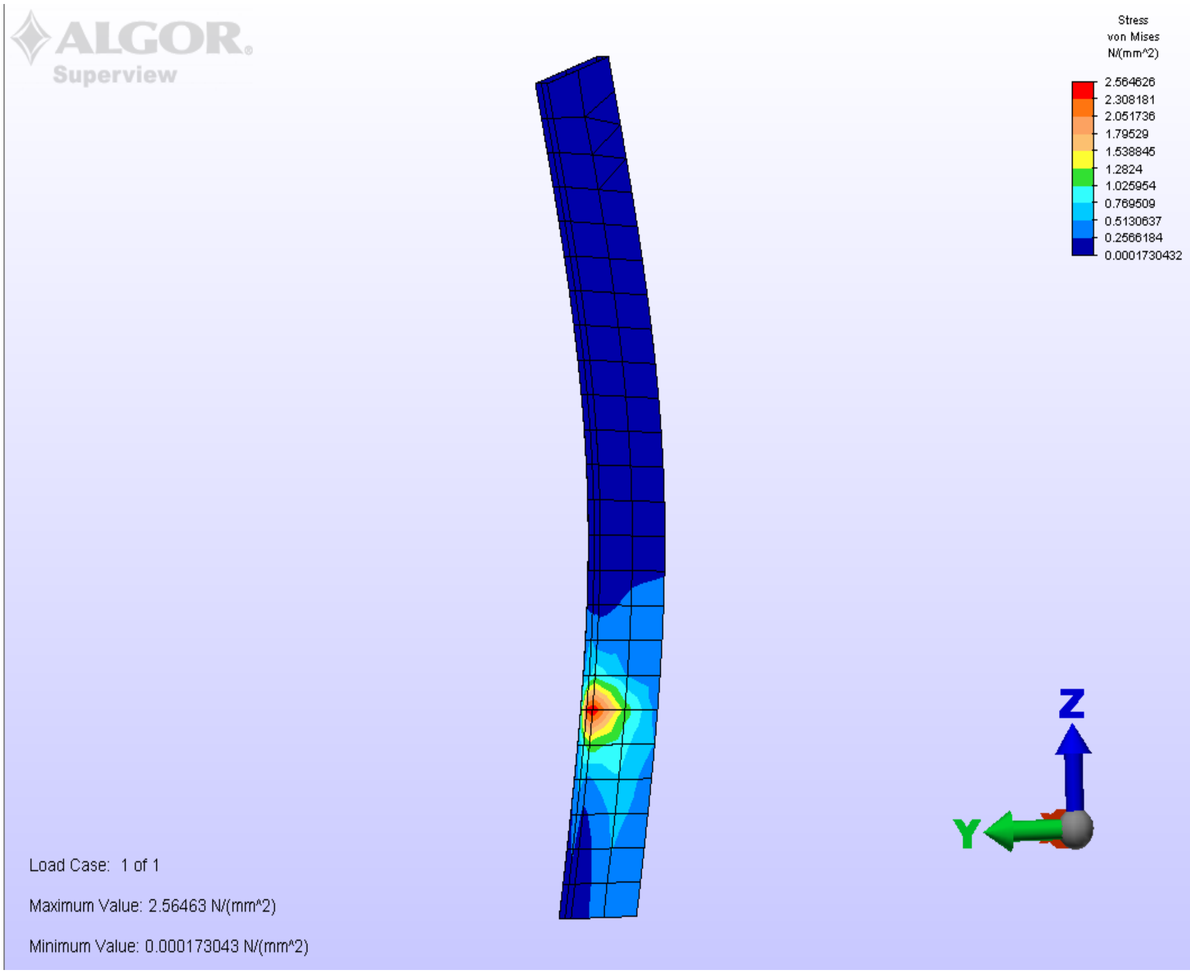

Figure 4.3.3: Result for side member 4

Side member 2 has some modification at the cross section. When the load applies, the value of stress is decrease from the side member 1. These mean that the different cross section of side member can affect the value of stress. The lowest value of stress for side member 2 is at point 324mm. The point is when the cross section change, the point for engine mounting bracket also change.

The side member 5 has done some modification from the side member 2. The cross section is same but this side member is curved about 50mm from the center. When the load was applies, the result is good from before. The value of stress is less from the others side member. The point that has a lowest value of stress is at 360mm. From the result, the curvy side member has a lowest value of stress.

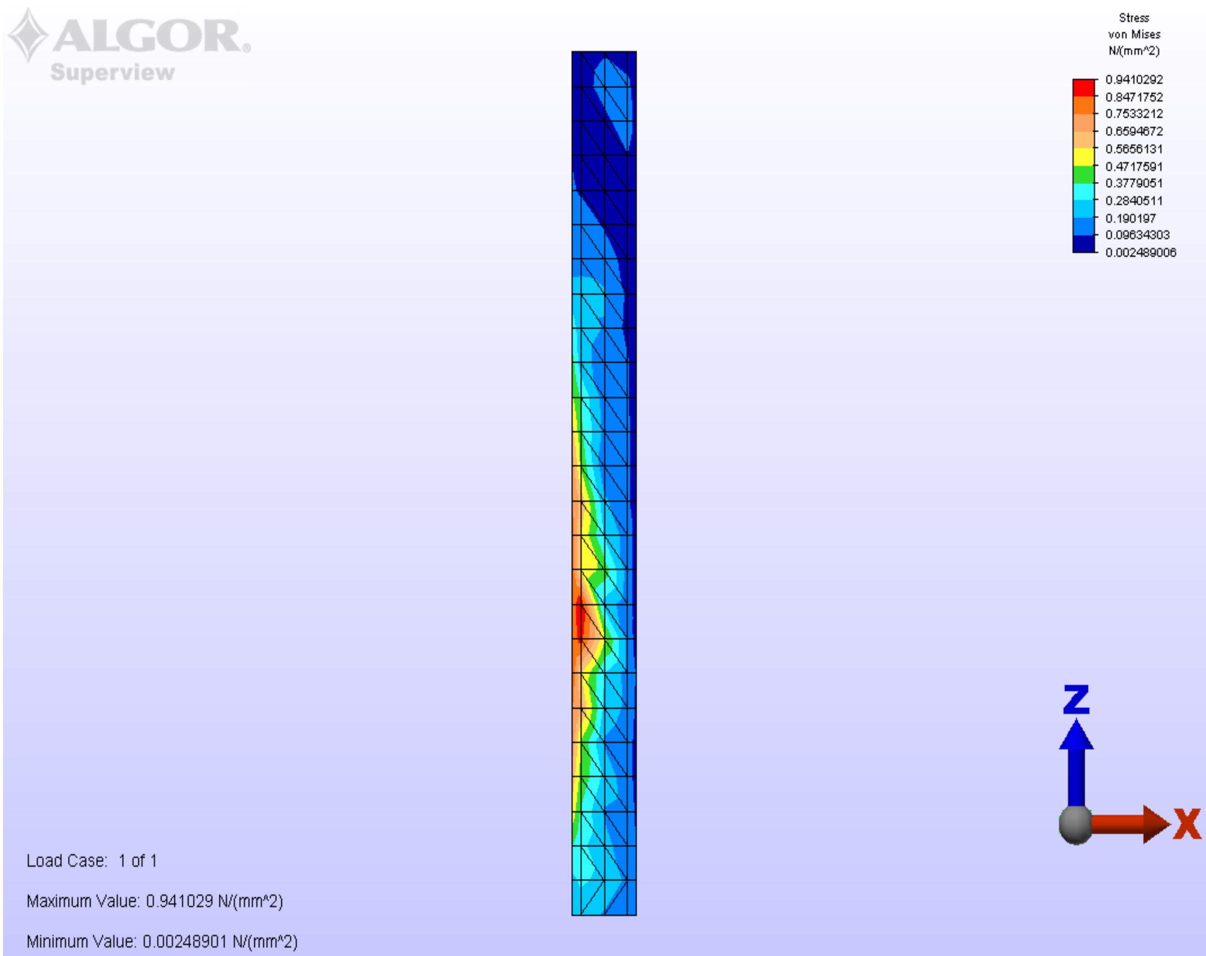

Figure 4.3.4: Result for side member 2

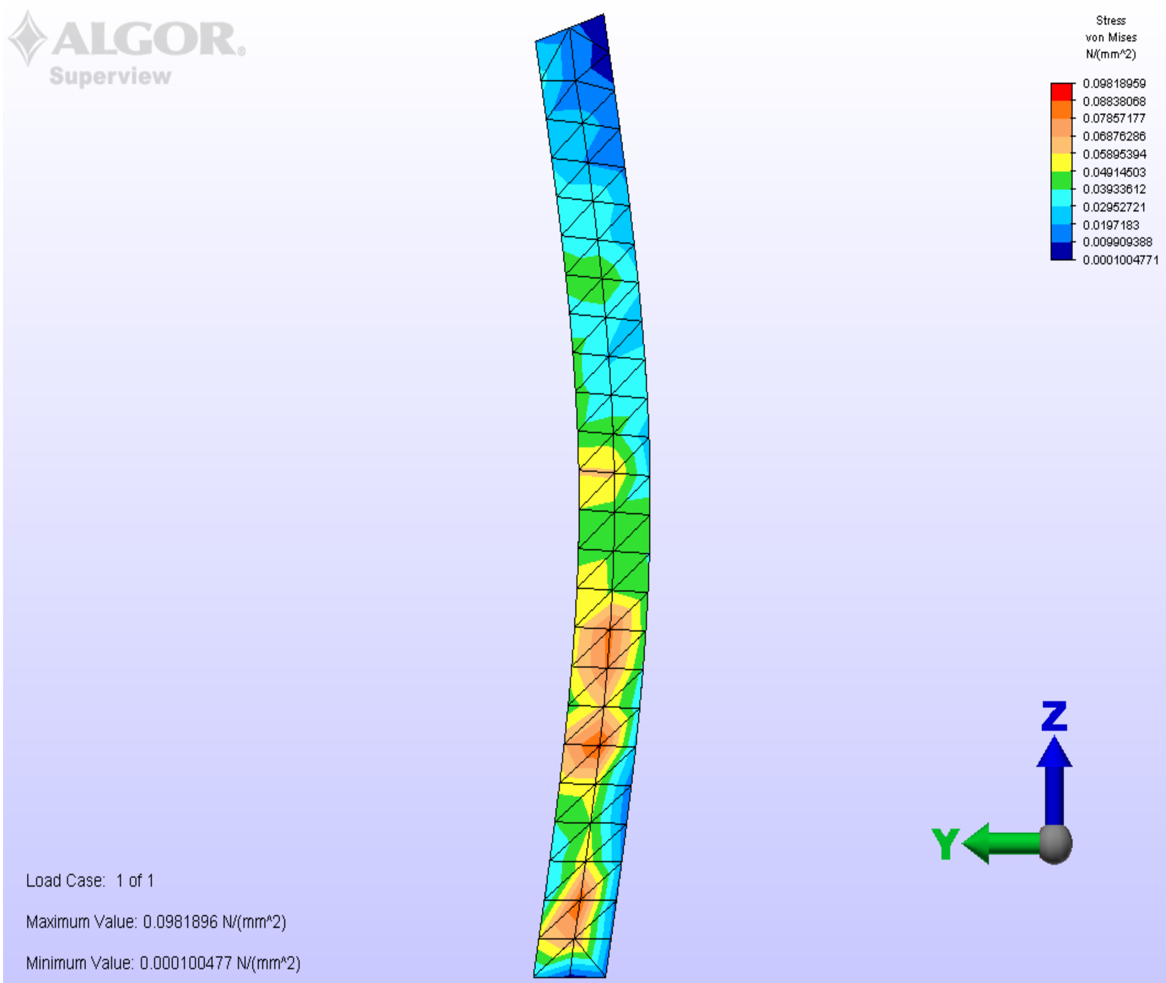

Figure 4.3.5: Result for side member 5

For side member 3, the point that has a lowest value of stress is at 360mm. this side member cannot have a modification because from the analysis, the stress value is higher than side member 1 and side member 3.

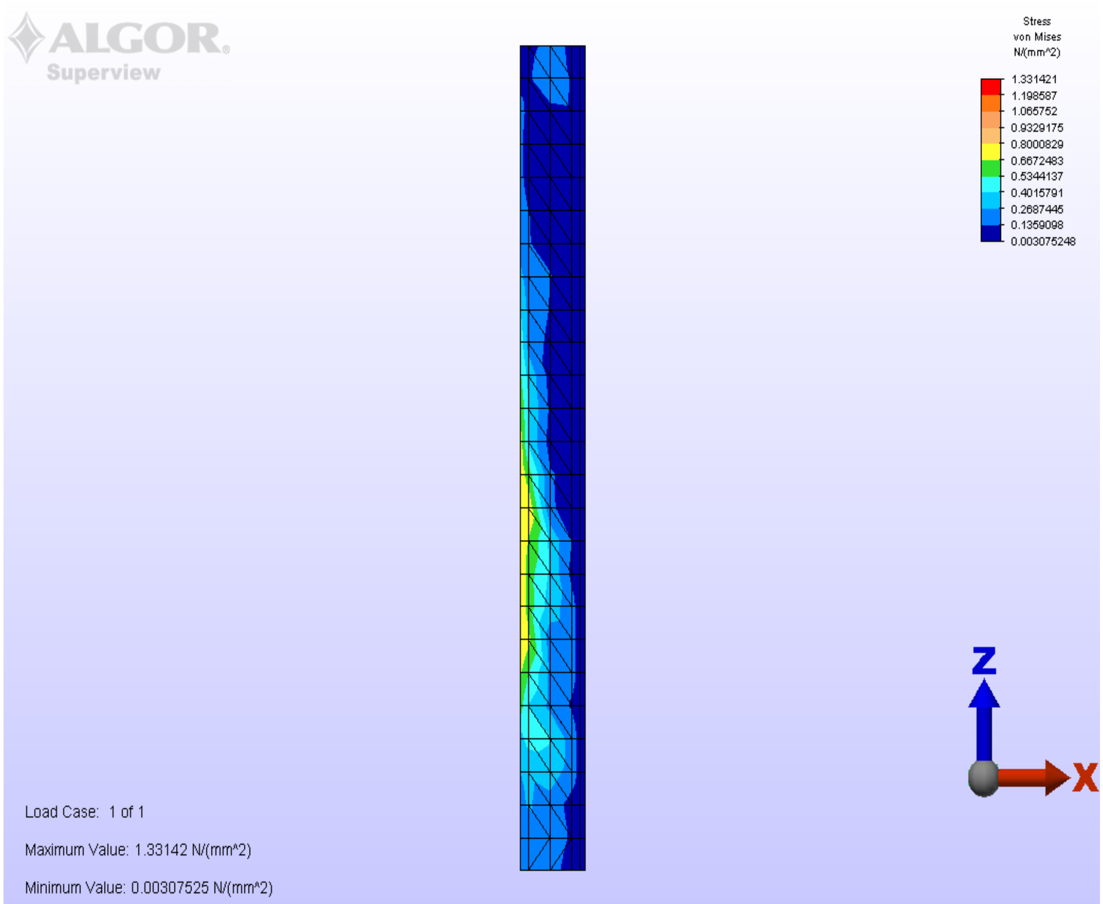

Figure 4.3.6: Result for side member 3

4.4 THICKNESS TEST

The objective for this test is to find the best thickness for the side member. This test was applied at all model of the side member with constant cross section and constant location of the side member. The location for all test is at 432mm. The range of thickness is from 0.7mm to 1.0mm.

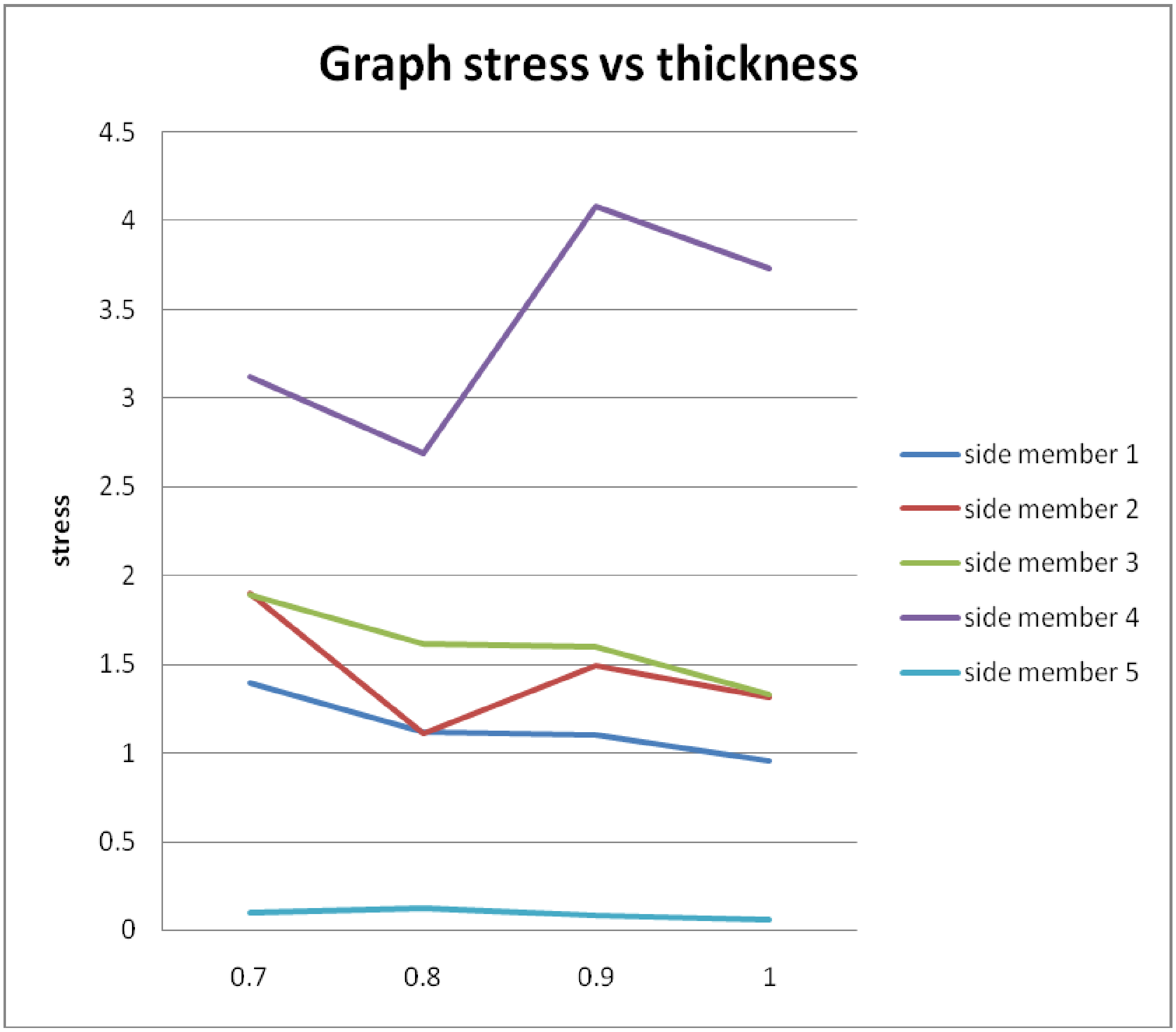

Figure 4.4.1: Graph stress vs thickness

The graph show the value of stress for different model of side member when the load apply and the thickness is changing from 0.7mm to 1.0mm. Side member 1 has a lowest stress when the thickness is decrease. For the side member 2 and side member 4, the lowest stress is when the thickness about 0.8mm. Side member 5 is a good design because it has lowest stress value for all thickness from other designs.

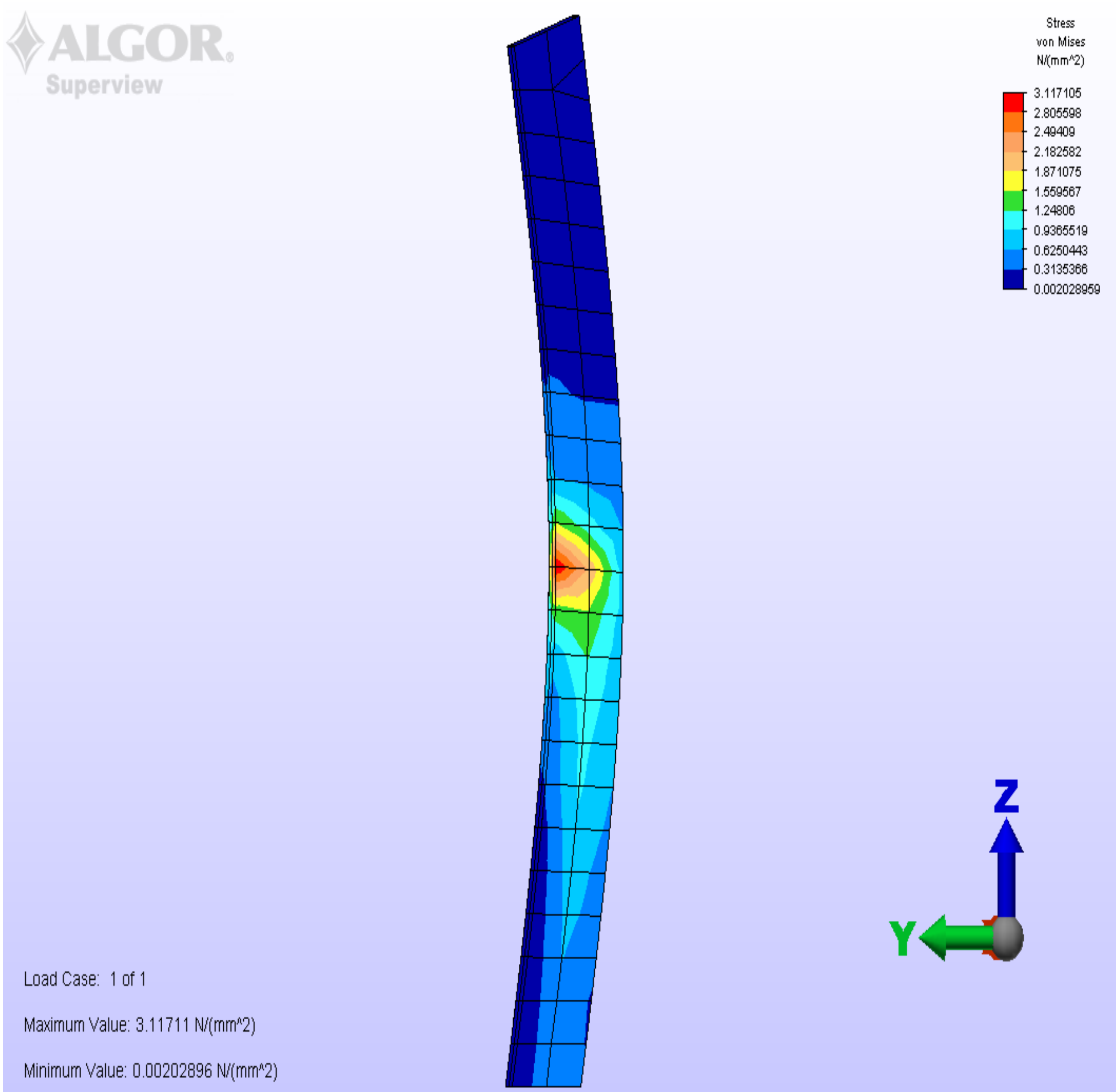

Figure 4.4.2: Result for side member 4

Side member 4 has a higher value of stress for all thickness. The maximum stress for side member when the thickness is 0.7mm about 3.11711 N/mm^2. From the figure 4.4.2, the maximum stress is show from the contour. The red colour is the point that has a maximum value of stress.

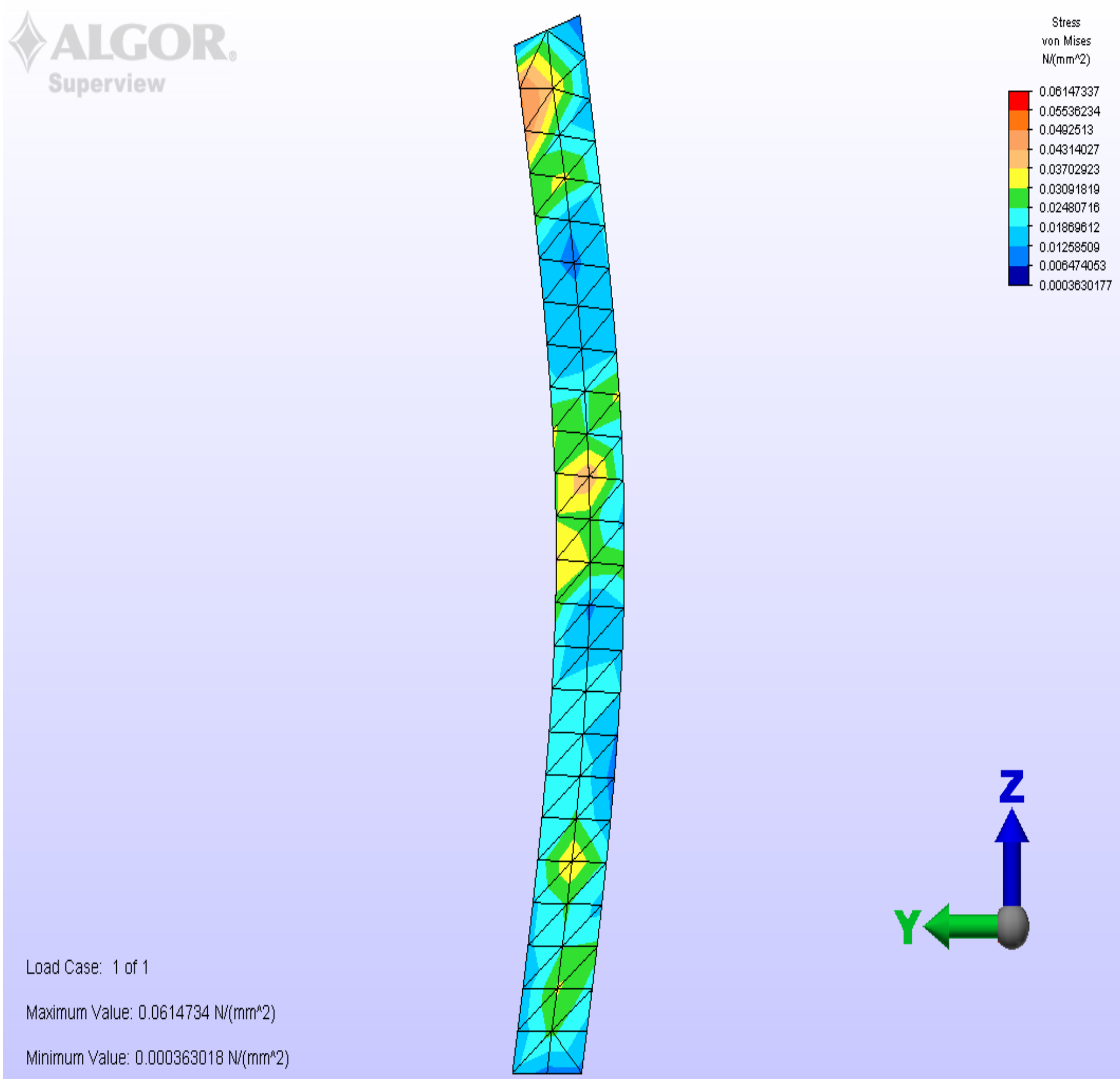

Figure 4.4.3: Result for side member 5

The maximum stress for side member 5 when the thickness is 1.0mm about 0.0614734 N/mm^2. The figure 4.4.3 show the contour of stress when the load apply at the side member. From this figure, the red colour cannot have at the model. This model for the 1.0mm thickness can support the high load from the other models.

4.5 DISCUSSION

From the packaging of engine in the industries, the dimension of engine bay must consider to determine the location of mounting bracket on the side member. The range of location mounting bracket was at the middle of side member. The mounting location is important to get a stability of engine and to reduce the vibration through the body of car.

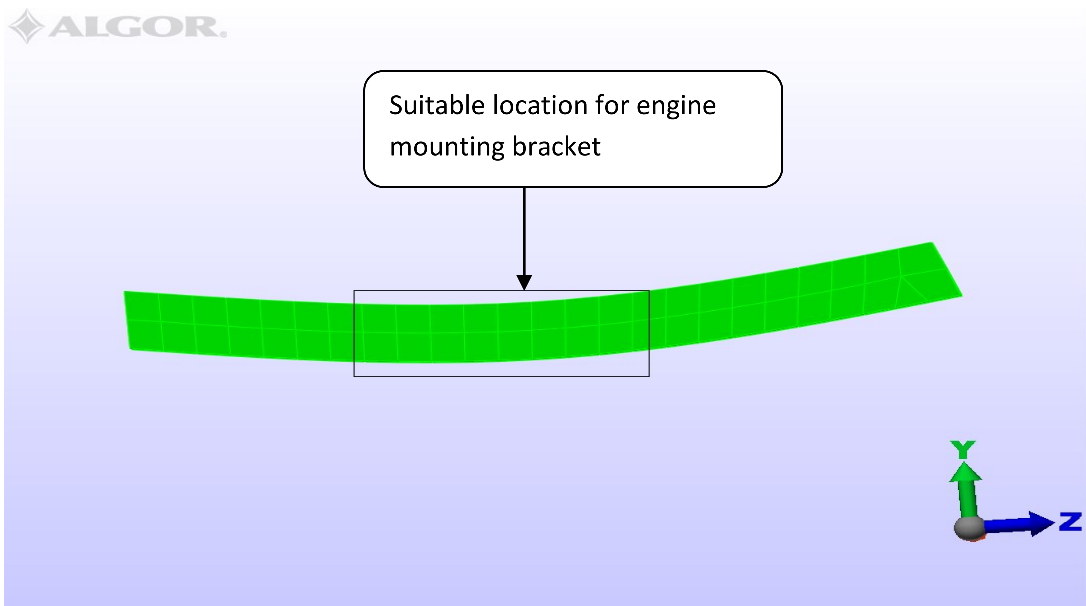

Figure 4.5.1: Range of mounting bracket location

The cross section of side member also affects the location of bracket mounting. The bracket mounting locations do not have a same location point at the side member. Many things must consider before define this location and not all the location was perfect to reduce the vibration, to stabilize the engine on the engine packaging, and to receive a high load in the different direction when the engine running.

The range point of location for engine mounting bracket is 324mm to 540mm. this is the suitable location because the dimensional of the engine bay. The clearance between engine component and body must consider before determine the mounting location. From the analysis, the best location for the engine mounting bracket is at point 360mm. The lowest stress at that point is 0.09819 N/mm^2.

At the industries, the range for thickness of side member is about 0.7mm to 1.0mm. From the result, the best thickness is 1.0mm. For this thickness, the value of stress is decrease for all model of side members. The suitable thickness for side member is 0.8mm because to reduce the weight of car. This thickness acceptable because all the side member for almost car use this thickness. From the analysis, the best thickness is 0.8mm. From the graph, mostly 0.8mm thickness has a lowest value of stress for all model side member.

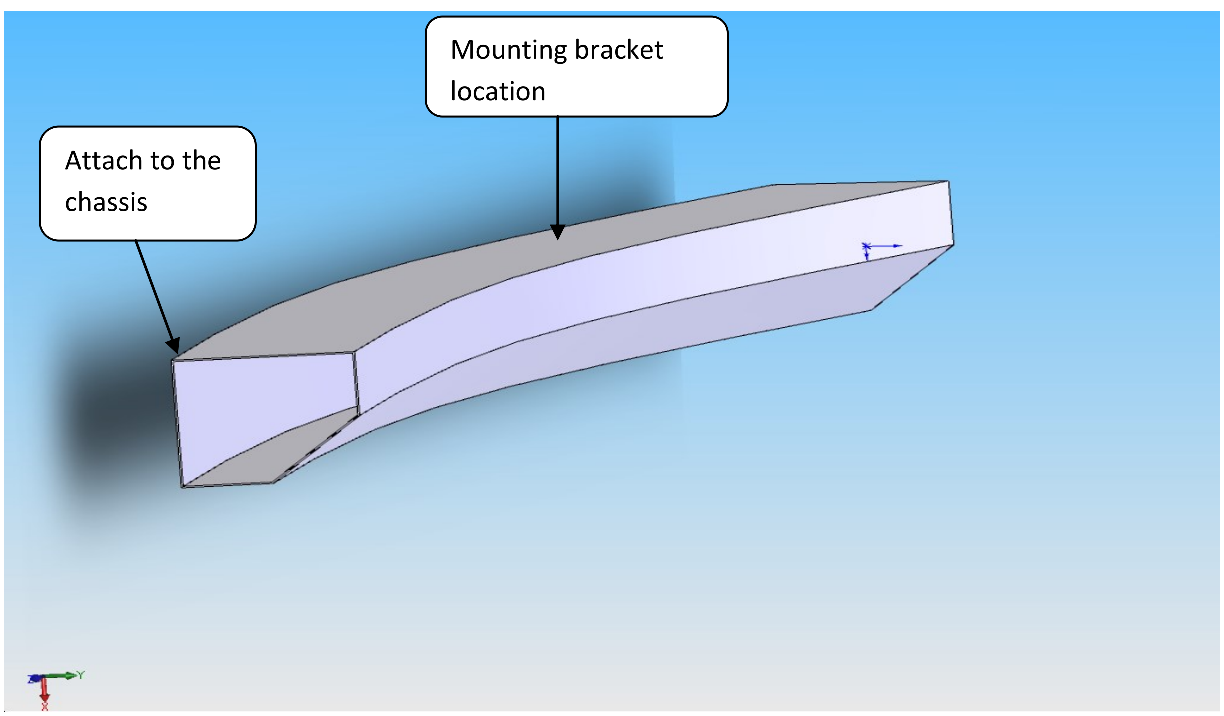

Figure 4.5.2: Design for side member

From this analysis, the suitable design for side member is side member 5. This side member has a lowest value of stress for all the analysis. The cross section of the side member can support the load that apply at the upper side of side member. This design can reduse the uses of engine bay space at the bottom side of engine bay.

4.6 MODAL ANALYSIS OF SIDE MEMBER

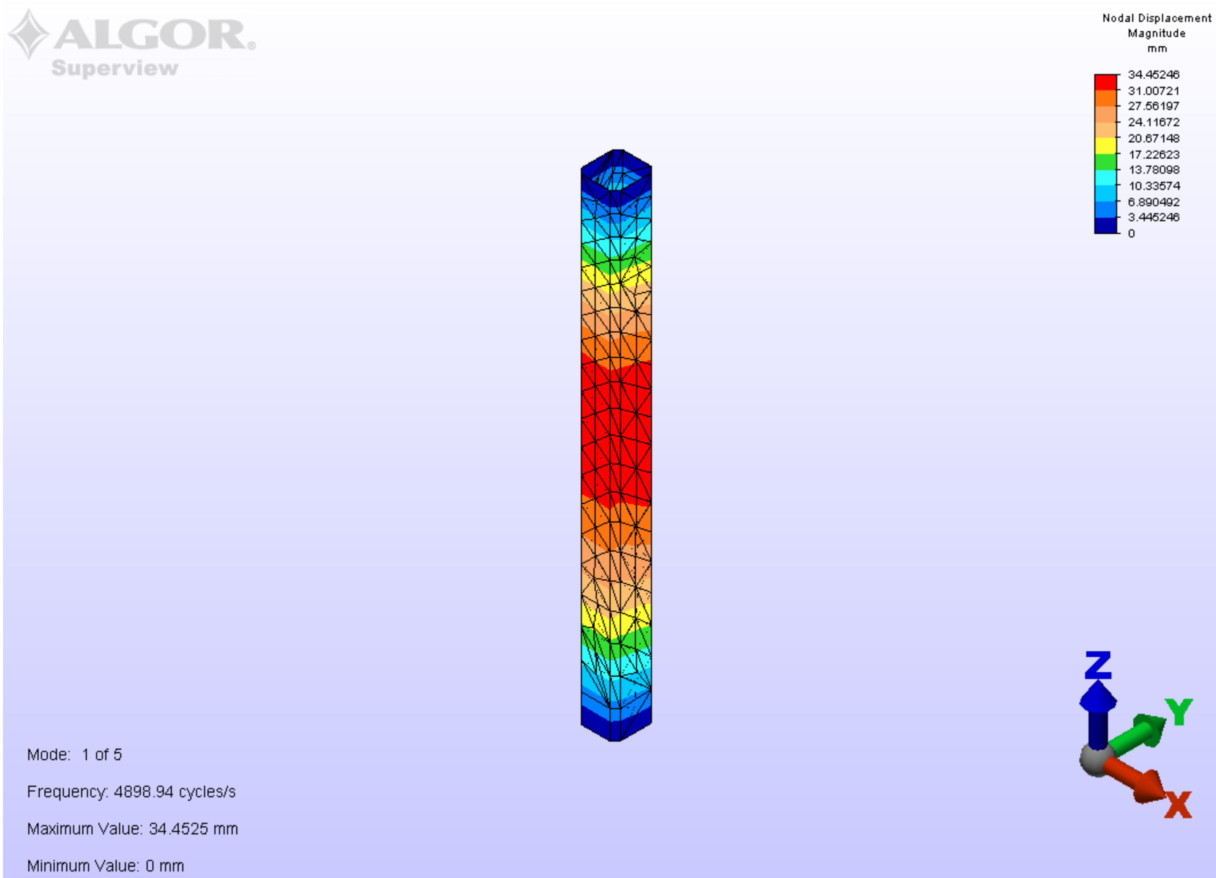

Figure 4.6.1: Mode shape 1

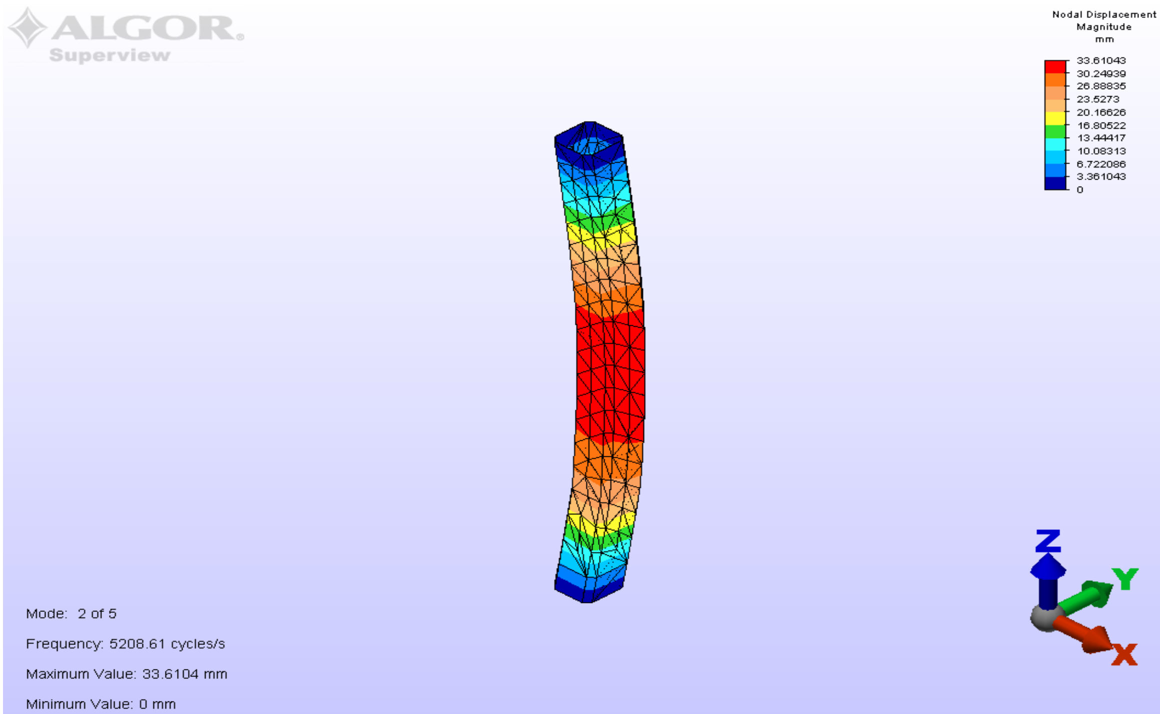

Figure 4.6.2: Mode shape 2

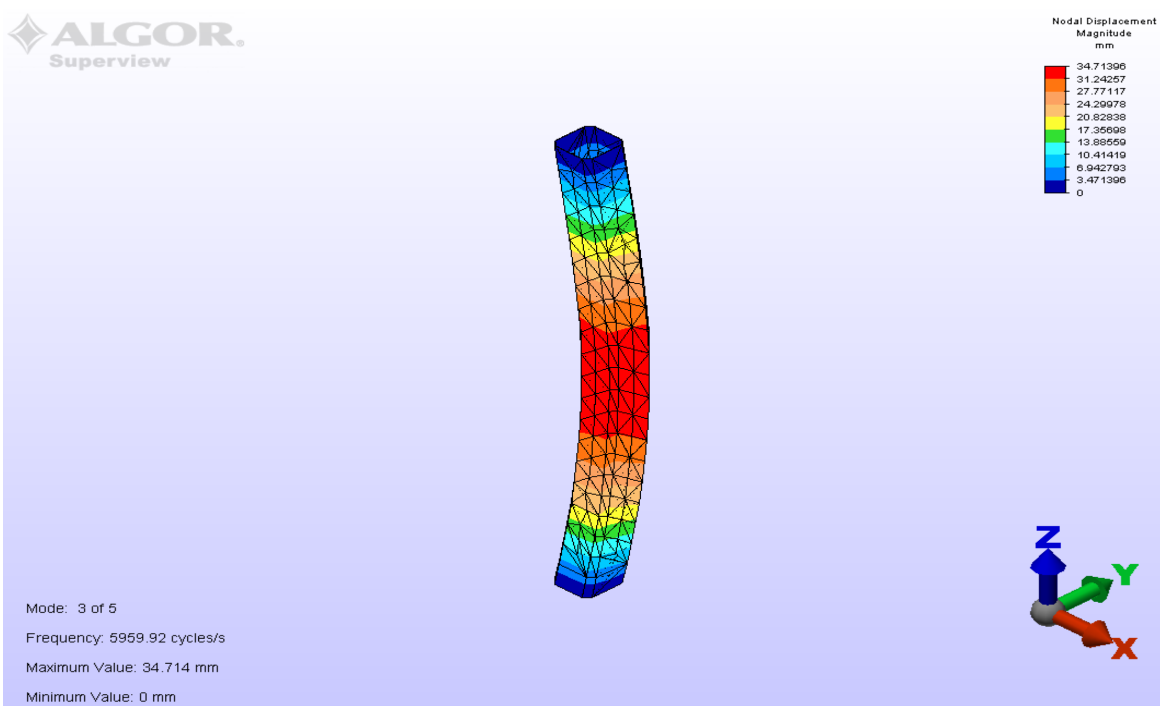

Figure 4.6.3: Mode shape 3

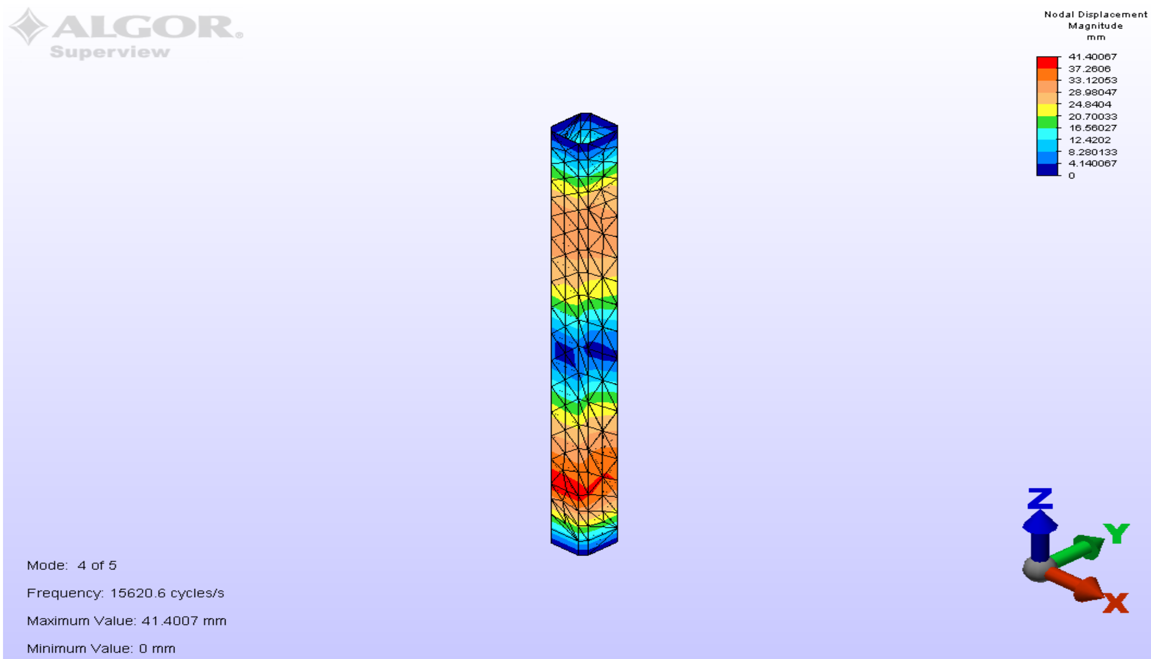

Figure 4.6.4: Mode shape 4

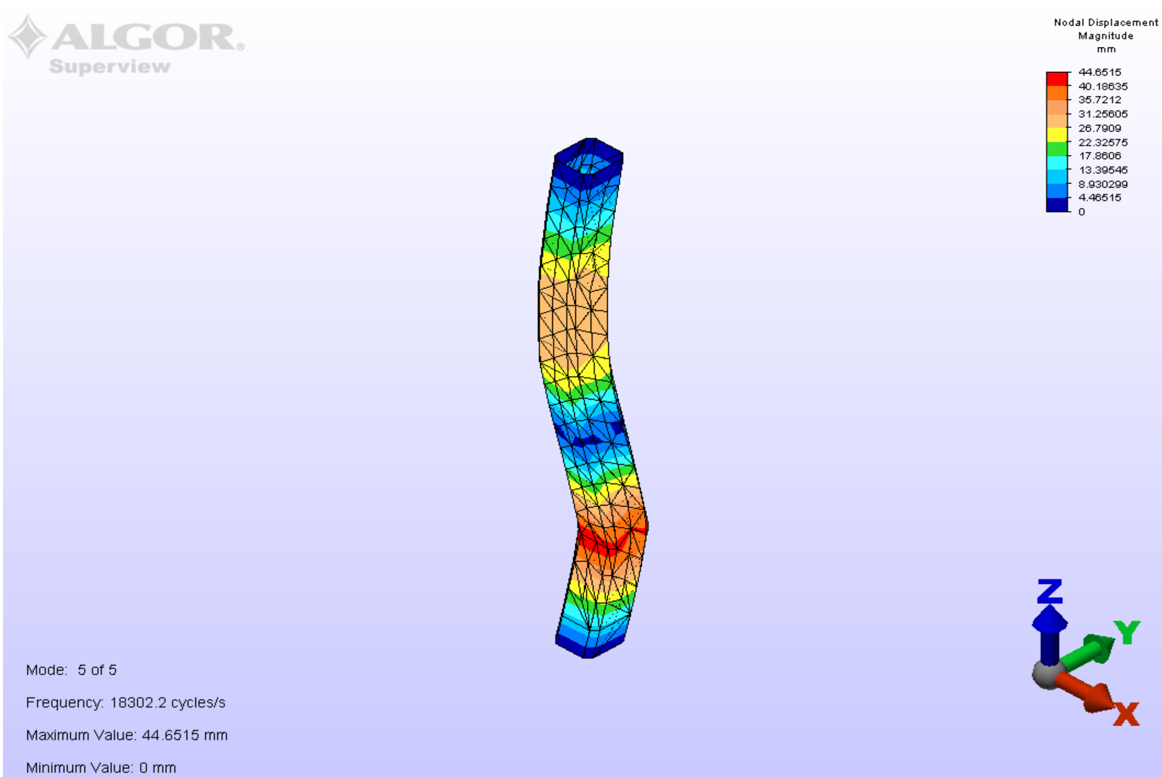

Figure 4.6.5: Mode shape 5

The goal of modal analysis in structural mechanics is to determine the natural mode shapes and frequencies of an object or structure during free vibration. It is common to use the finite element method (FEM) to perform this analysis because, like other calculations using the FEM, the object being analyzed can have arbitrary shape and the results of the calculations are acceptable. The types of equations which arise from modal analysis are those seen in eigen systems. The physical interpretation of the eigenvalues and eigenvectors which come from solving the system are that they represent the frequencies and corresponding mode shapes. Sometimes, the only desired modes are the lowest frequencies because it can be the most prominent modes at which the object will vibrate, dominating all the higher frequency modes.

It is also possible to test a physical object to determine it's natural frequencies and mode shapes. The results of the physical test can be used to calibrate a finite element model to determine if the underlying assumptions made were correct (correct material properties and boundary conditions were used). The modal analysis is done for side member is to determine the natural frequency for this model of side member. The highest natural frequency is better to avoid the model or part failure.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 CONCLUSION

Many things must consider to determine the location of engine mounting bracket. This is one of the methods to determine the engine mounting location. Not all the car has same location for engine mounting and the range location for engine mounting is at the middle of side member. This location is suitable because to make an engine in the stabilized position. The thicknesses of side member also important before locate a new engine. Usually, more thick the material uses more loads that it can support and the stress value is lowest. In the industries, the side member must have a proper thickness to reduce the weight of the car and to reduce the cost to produce this part.

The cross section of the side member also important to fit the engine bay dimension. The cross section also influences the value of stress when the load is applied. As the conclusion, many thing must consider before locate the engine mounting. The designs of side member also give the affect to the location of engine mounting.

5.2 RECOMMENDATION

This analysis also can be use for all type of mounting arrangement. To improve this project, the experimental analysis is better to prove the result from the software analysis. Side member must fabricate and run the test for the small engine. To get an accurate shape and dimension, the 3D scanner is the better tool to use.

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