DYNAMIC ANALYSIS OF A RECIPROCATING COMPRESSOR

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Approved by,

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UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

August 2013

CERTIFICATION OF ORIGINALITY

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

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CLARE MARIE SIA AN AN

ABSTRACT

Reciprocating compressors are widely used in the industry and plays an important role in maximising productivity as it increases the pressure and reduces the volume of a gas. Compressors are sensitive equipments and maintenance has to be performed when the equipment fails. Reciprocating compressor has both rotary and reciprocating motion involved. Unbalance forces are bound to occur on the dynamic components and this causes a lot of vibration in which might spoil the compressor. Analyzing these forces might help to overcome the unbalance forces and thus reduce the vibration and faultiness in the compressor.

Using softwares that are made available, analyzing of these forces will be possible with less hassle. CATIA is used to produce the model of the compressor and with the completed model, it is then imported in ADAMS for simulating and analyzing. Results are shown through graphs and discussions were made to analyze the results.

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CHAPTER 1: INTRODUCTION

1.1 Background Study

1.1.1 Reciprocating Compressor

Reciprocating compressors are classified under positive displacement compressors. Volumes of gas are confined within a limited space and increased to a higher pressure.

Reciprocating compressors are the mostly used compressors in the class of positive displacement compressors. The mechanism is simply by the means of a pushing force of a piston in a cylinder. The piston moves up and down inside the cylinder and this force the gas into a smaller space, and thus increases the pressure. The basic reciprocating compression element is a single cylinder compressing on one side of the piston. The use of both sides will be the two basic single-acting elements operating in parallel in one set of up-down action.

Rotary motion from the motor or any other external driver to the compressor is translated to linear motion by using the crankshaft, and the piston rod. The end of the piston rod is secured by the crankpin to the crankshaft, and the other by the piston, as the crankshaft turns, reciprocates in a linear motion. The suction and discharge valves are usually located at the top and bottom of the cylinder and are simply, check valves that allows the one way flow of the gas. When the piston moves up, a partial vacuum in the lower end of the cylinder will occur; the pressure differential makes the valves to open and allowing gas to flow into the cylinder. Whereas for the downward stroke, pressure in the cylinder exceeds the pressure in the discharge line will cause the valve will open and allow the gas to flow from the cylinder to the discharge. If this occurs on one side of the piston, it is called ‘single-acting’ compression whilst both side; it is called ‘double-acting’ compression.

The drawing of a reciprocating compressor and its components can be referred to Appendix I

1.2 Problem Statement

Following through the repair of the compressor during the industrial internship period, it was brought to attention that the cylinder was of an oval cross-section due to the rubbing of the piston against the cylinder wall.

A specialist from the original equipment manufacturer was called in to conduct the repair, still the compressor experience the same problem after a period of time. Thus this led to conducting a dynamic analysis on the reciprocating compressor in which the balancing of the compressor will be studied.

Figure 1 Friction Marks on the Piston

Figure 2 Friction Marks on the Cylinder

1.3 Objectives

The project is set out to achieve a few objectives which are listed as follows:

- To identify the unbalanced forces of the reciprocating compressor.
- To model and simulate out the reciprocating compressor's dynamics.
- Suggest in any possible modifications and recommendations.

1.4 Scope of Study

The project will study in the range of the balancing and unbalancing of the reciprocating compressor

- Balancing of reciprocating compressor.
- Four cylinder, vertical reciprocating compressor.
- Analysis of the crankshaft, crosshead and piston.

CHAPTER 2: LITERATURE REVIEW

2.1 Slider-Crank Mechanism

The slider-crank mechanism is mainly used to translate the rotary motion to a reciprocating motion or vice versa.

Equation 1:

$$\boldsymbol{a_p = Acceleration\ of\ Piston}$$

$$= r\omega^2\left[\cos\theta + \frac{\cos\ 2\theta}{n}\right] \quad ; \quad n = \frac{1}{r}$$

Equation 2:

F_i = Force required to accelerate the mass 'm'

$$= mr\omega^2\left[\cos\theta + \frac{\cos 2\theta}{n}\right]$$

$$= mr\omega^2\cos\theta + \ mr\omega^2\ \frac{\cos\ 2\theta}{n}$$

In the second equation, $\boldsymbol{mr\omega^2\cos\theta}$ is called the primary accelerating force whilst $\boldsymbol{mr\omega^2\ \frac{\cos 2\theta}{n}}$ is known as the secondary accelerating force. The maximum value of primary accelerating force is $\boldsymbol{mr\omega^2}$. Maximum value for secondary accelerating force is $\boldsymbol{mr\omega^2/_n}$. The primary force is big compared to the secondary force and thus the secondary force can be safely neglected in slow speed engines.

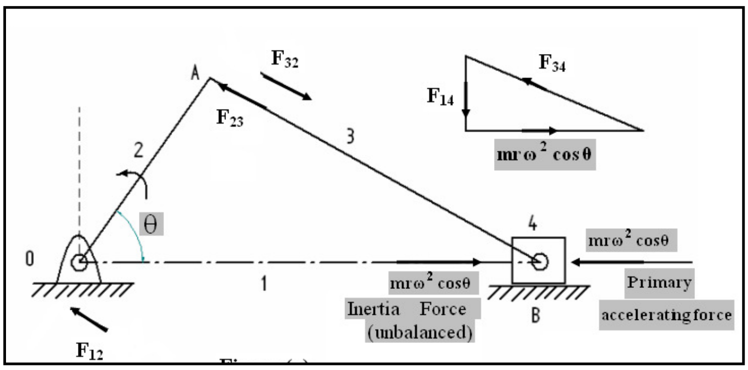


Figure 3 Slider-Crank Mechanism

Figure 3 shows the inertia force caused by the accelerating force. Whereas Figure 4 shows the inertia forces; at'0' the force exerted by the crankshaft on the main bearings has two components, the horizontal and vertical marked by F_{21}^{h} and F_{21}^{v} respectively. F_{21}^{h} is an horizontal force, which is an unbalanced shaking force. F_{41}^{v} and F_{21}^{v} balance each other but form an unbalanced shaking couple. (Bongale)

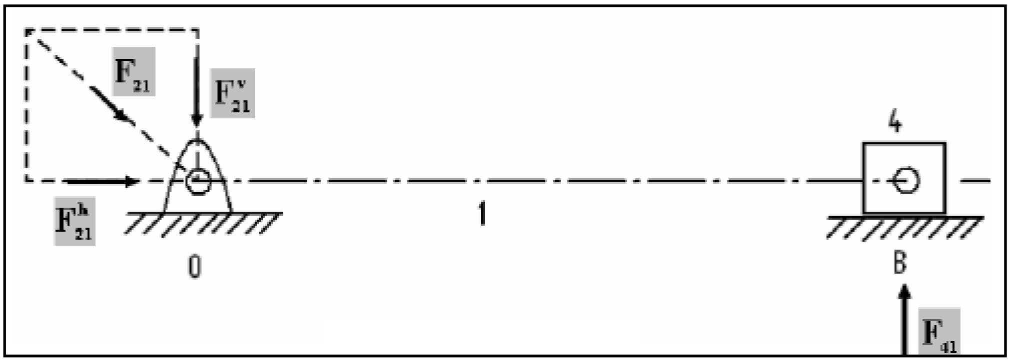


Figure 4 Forces Acting in the Engine Frame

2.2 Unbalance Forces

From the point of view in design, the unbalanced forces are produced by rotating and reciprocating masses. Reciprocating forces occur in all compressors from acceleration and deceleration of the reciprocating weights. Rotating forces result from the centrifugal force produced from the unbalanced weights of the crank-throw and part of the connecting rod. (Bloch, 2006)

Since the shaking force and a shaking couple differ in magnitude and direction during the engine cycle, therefore they cause very objectionable vibrations. In most of the mechanisms, the shaking force and a shaking couple can be reduced by adding appropriate balancing mass, but it is usually not practical to eliminate them completely. This means that the reciprocating masses are only partially balanced. In reference of Figure 5, the primary force acts from 'O' to 'P' along the line of stroke. Consequently, balancing of primary force is considered as equivalent to the balancing of mass 'm' rotating at the crank radius r. This is balanced by having a mass B at a radius b, placed diametrically opposite to the crank pin C. (Khumi, 2010)

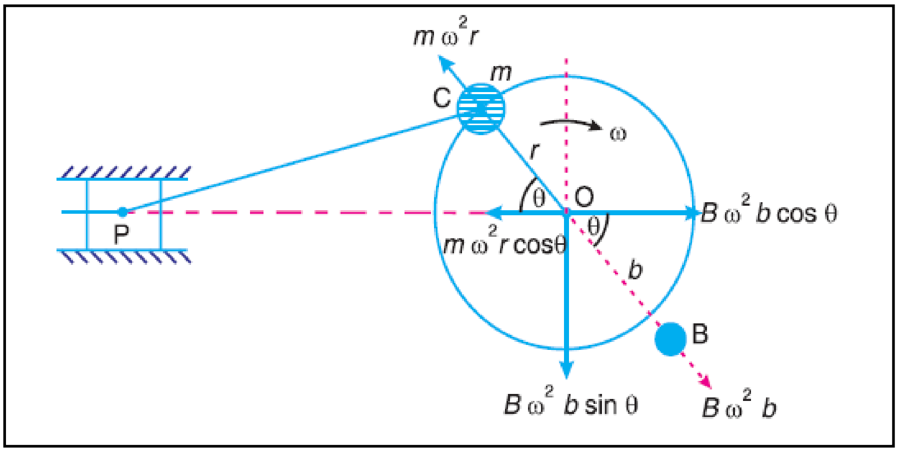


Figure 5 Partial Balancing of Unbalanced Primary Force in a Reciprocating Engine

Referring again to Figure 4, the primary force will be completely balanced if ***Bb* = *mr***. However the centrifugal force by the revolving mass B, also has a vertical component in which it is perpendicular to the line of stroke, of magnitude ***Bω2b sin θ*** and this force remains unbalanced. The maximum value of this force is equivalent to ***Bω2b*** when θ is at 90° and 270°, which is same as the maximum value of the primary force ***mω2r***.

The primary unbalanced force acts along the line of stroke whereas in the second case, the unbalanced force acts along the perpendicular to the line of stroke. The maximum value of the force remains same in both the cases. Thus, the effect of this balancing changes the direction of the maximum unbalanced force from along the line of stroke to the perpendicular of line of stroke. A fraction 'c' will be used to compromise for the reciprocating masses balanced, giving: ***cmr* = *Bb***

Unbalanced force along the line of stroke:

$$= (1 - c)m\omega^2\, r \cos\theta$$

Unbalanced force along the perpendicular to the line of stroke:

$$= m\omega^2 r\sqrt{(1-c)^2\cos^2\theta + c^2 sin^2\theta}$$

2.3 Rotating and Reciprocating Unbalance

Torsional oscillation in the crankshaft and in the shafting of driven machinery is an important group of vibration phenomena of practical importance in reciprocating machines. A combination of periodic accelerations of moving parts (piston, connecting rod, crank) and periodic variations in cylinder steam or gas pressure seems to be the cause.

For a single cylinder vertical reciprocating machine, the piston executes an alternating motion, and in the process experiences alternating vertical accelerations. To have a downwards motion on the piston, there must exist a downward force acting on it and this force must have a reaction pushing upward against the stationary parts of the engine. Therefore, the alternating acceleration of the piston is coupled with an alternating force on the cylinder frame, making a feel of vibration in the engine and its supports. Whereas in the lateral direction perpendicular to both the crankshaft and the piston rod, moving parts are also being accelerated which are the crank pin and the part of the connecting rod. The forces that cause these accelerations must have equal and opposite reactions on the frame of the machine. This last effect is the horizontal unbalance. In the crankshaft main axis, no inertia forces appear, since all moving parts remain in planes perpendicular to the crankshaft.

These various inertia forces can cause moments can cause moments. When two vertical cylinders are considered with a crank set 180^o apart, when one piston is accelerated downward the other piston is accelerated upward, the two inertia forces from a couple tend to rock the machine about a lateral axis. Consequently, the horizontal or lateral inertia forces of the two cracks are equal and opposite, forming a couple tending to rock the machine about a vertical axis. Rocking about the crankshaft axis can happen even in a single cylinder machine. If the piston is accelerated downwards by a pull in the connecting rod, this pull exercises a torque about the crankshaft axis. Since the piston acceleration is alternating, this inertia torque is also alternating. (Rangwala, 2006)

If the whole machine mounted on weak springs is considered as the mechanical system, the external torque is zero and with any increase in clockwise angular momentum of the moving parts must be neutralized by an increase in counter-clockwise angular momentum of the stationery parts of the machine. Considering merely the dynamic parts of the machine as the mechanical system, an increase in clockwise angular momentum of the moving parts must be caused by a clockwise torque on these parts, which has a counter-clockwise reaction torque is transmitted to the foundation and may cause problems. However, if instead it is mounted on soft springs, reaction to the foundation cannot penetrate through these springs and the counter torque is absorbed as an inertia torque of the frame and cylinder block, and the machine block must vibrate. (Rangwala, 2006)

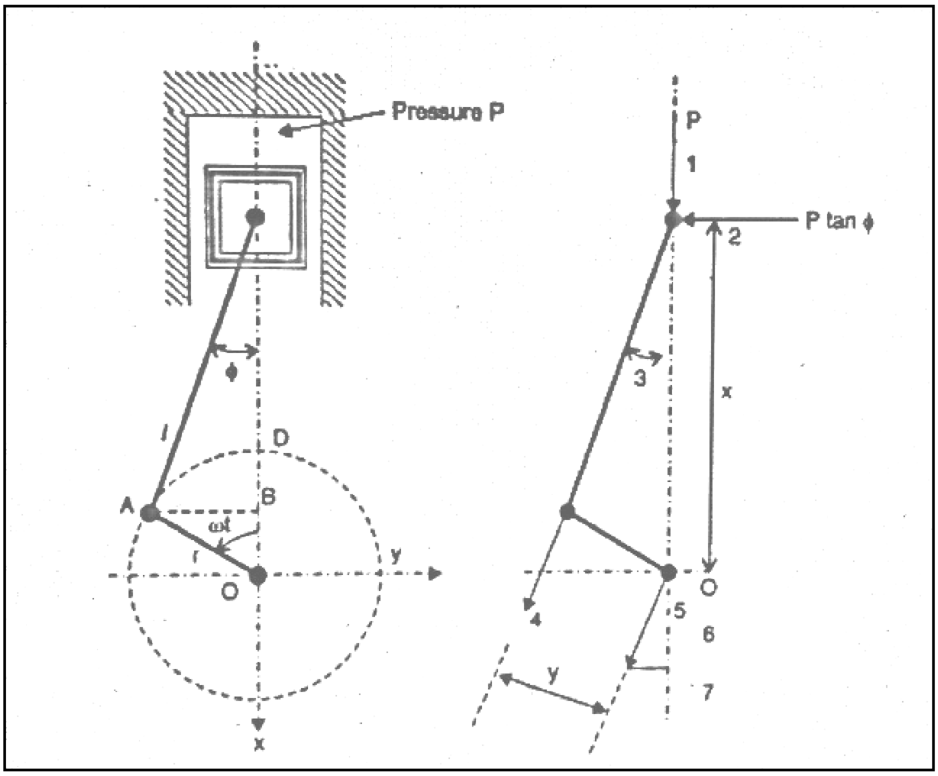


Figure 6 Gas Pressure Load on a Single Cylinder Machine

In reference to Figure 6, inertia forces are excluded by assuming the machine to be rotating at a slow and constant speed ω, or the moving parts have negligible mass. Apart from acting on the piston, the gas pressure also presses upward against the cylinder head. Pressure force on the piston is transmitted through the piston rod to the crosshead. If friction is neglected, it is held there in equilibrium by forces 2 and 3 in Figure 6. Of the forces acting on the crosshead, force 3 is a compression in the connecting rod, and 2 is a reaction pressure on the frame to the right, of magnitude $\boldsymbol{P_v \tan(\phi)}$. Force 3 of magnitude $\boldsymbol{P/\cos(\phi)}$ is transmitted through the connecting rod to the crank pin. Force 5 is taken up by the main bearings at O and can be resolved into a vertical component 6 and a horizontal component 7. Triangle 1, 2, 3 and 5, 6, 7 are alike so the magnitude of 6 is $\boldsymbol{P}$ and that of 7 is $\boldsymbol{P \tan(\phi)}$.

No forces occur along the longitudinal crankshaft axis of a machine, while in the lateral and vertical directions, only inertia forces appear. On the vertical and lateral axes, only inertia torques were found, and in the longitudinal direction both an inertia torque and a cylinder gas pressure torque occur. If assumptions are made that the machine is built of an elastically non-deformable members, the problem is one of static balance only. The frame and other stationary parts generally fulfil this condition, but the crankshaft can be twisted significantly, making torsional vibrations possible. The subject may be divided into three categories:

2.3.1 Inertia Balance

Refers to the balancing of the machine against vertical and lateral forces, and against moments about the vertical and lateral axes.

2.3.2 Torque Reaction

Under this heading the effects of torque due to inertia and gas pressure acting on the stationary parts about the longitudinal axis are analyzed.

2.3.3 Torsional Vibrations of Crankshaft

Consequences of the longitudinal torque on the moving parts of the reciprocating machine are dealt with. (Rangwala, 2006)

2.4 Balancing of Reciprocating Engines

Thus, from the equations of the unbalance forces, the balancing mass required in balancing the rotating and reciprocating masses can be obtained by the following equation:

$$Bb = (m_1 + cm)r$$

Where: $m_1 = Magnitude\ of\ rotating\ masses, kg$

$m = Magnitude\ of\ reciprocating\ masses, kg$

$B = Mass\ required\ to\ balance, kg$

$b = Rotational\ radius\ of\ B,\ m$

2.5 Balancing of Single-Cylinder Cranktrain

Actions that compensate for the outside effect of mass forces are the balancing of reciprocating masses. Stress still remains in the engine as a result of the mass forces. For a single cylinder cranktrain, a complete compensation of the mass forces is possible when two pairs of rotating masses are added. The compensating masses are arranged in a way that the resulting alternating harmonic forces act in the centre of the cylinder and are opposite to the masses forces F_1 and F_2. The arrangement of the balancing masses can be shown in Figure 7.

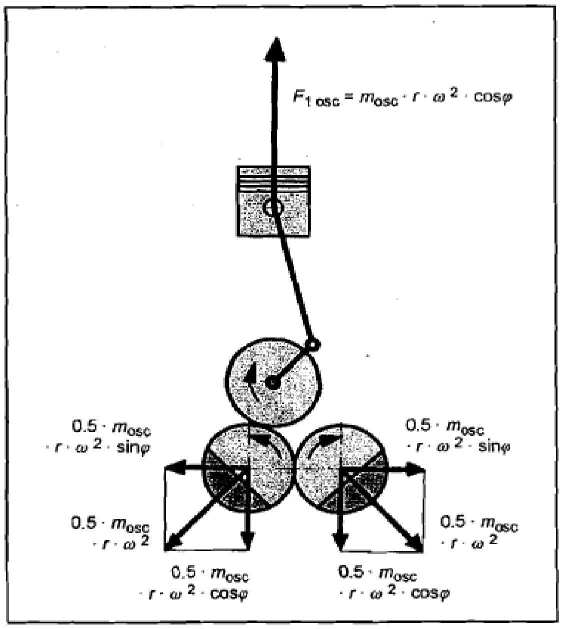


Figure 7 Complete Balance of the Inertial Forces of the First Order

2.6 Balancing of a Multi-Cylinder Cranktrain

The mass force of a multi-cylinder cranktrain is the sum of the mass forces of the individual cylinders. Since the reciprocating mass forces of the individual cylinders act in the individual cylinder centre lines, mass couples can result. For in-line engines, a schematic calculation is used to determine the mass forces due to a parallel cylinder arrangement. It is sufficient to consider the vectors of the individual cylinders ***F+1k***, ***F+2k (k** is the number of cylinders that rotate in the same direction).* For each order, these vectors can be added directly to resulting rotating vectors ***F+I*** or ***F+2***. (Baharom, 2013)

In a multi-cylinder, it is possible to balance some or all of the inertia forces and torques by proper arrangement of the cranks. Figure 8 shows the general arrangement of six cylinder compressor. To balance the force, the total inertia force in the *x* and *y* directions must be zero. (Singiresu S. Rao, 2011).

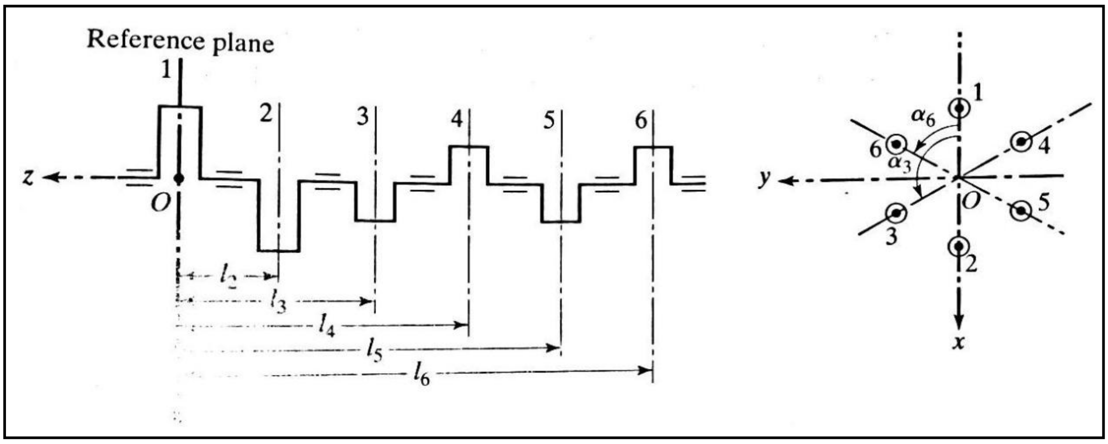


Figure 8 General Arrangement of a 6 Cylinder Compressor

Multi-cylinder engines have mutual counteractions between the various components in the crankshaft assembly which is one of the essential factors in determining the selection of the crankshaft's configuration. The inertial forces will be balanced when the common centre of gravity for all moving crankshaft-assembly components lies at the crankshaft's midpoint. In other words, the crankshaft is symmetrical when viewed from the front. The crankshaft's symmetry level is defined using geometrical representations of 1^{st} and 2^{nd} order forces or the star diagrams. The 2^{nd} order star diagram for the four-cylinder in-line engine is asymmetrical, denoting that this order is characterized by substantial free inertial forces. These forces can be balanced using two countershafts rotating in opposite directions at double the rate of the crankshaft as also shown in Figure 7. (Robert Bosch Gmbh, 2002)

Table 1 Forces and Moments Applied to the Piston, Connecting Rod and Crankshaft on an Engine

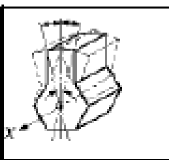
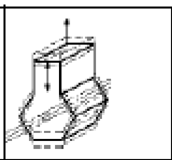
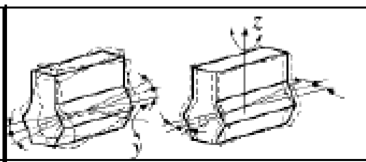
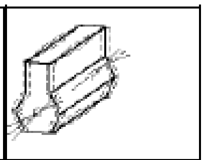
Forces and moments at the engine				
Designation	Oscillating torque, transverse tilting moment, reaction torque	Free inertial forces	Free inertial moment, longitudinal tilting moment about the y-axis (transverse axis) ("pitching" moment) about the z-axis (vertical axis) ("rolling" moment)	Internal flex forces
Cause	Tangential gas forces as well as tangential inertial forces for ordinals 1, 2, 3 and 4	Unbalanced oscillating inertial forces 1st order in 1 and 2 cylinders; 2nd order in 1, 2, and 4 cylinders	Unbalanced oscillating inertial forces as a composite of 1st and 2nd order forces	Rotating and oscillating inertial forces
Design factors	Number of cylinders, ignition intervals, displacement, pi, ε, pz, m0, r, ω, λ	Number of cylinders, crank configuration m0, r, ω, λ	Number of cylinders, crank configuration,cylinder spacing, counterweight size influences inertial torque components about the y- and z-axes m0, r, ω, λ, a	Number of throws, crank configuration, engine length, engine-block rigidity
Remedy	Can only be compensated for in exceptional cases	Free mass effects can be eliminated with rotating balancing systems, however this process is complex and therefore rare; crank sequences with limited or no mass effects are preferable		Counterweights, rigid engine block
	Shielding of the environment through flexible engine mounts (in particular for orders ≥ 2)			

Table 2 Star diagram of the 1st and 2nd order for three- to six-cylinder, in-line engines

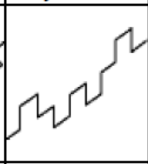
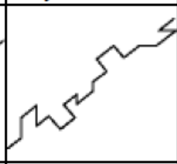
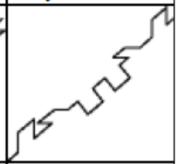
	3-cylinder	4-cylinder	5-cylinder	6-cylinder
Crank sequence				
Star diagram 1st Order	1 2 3	1,4 2,3	1 4 5 3 2	1,6 3,4 2,5
Star diagram 2nd Order	1 3 2	1,2,3,4	1 2 3 4 5	1,6 2,5 3,4

Table 3 Resultant Mass Forces and Mass Couples for Reciprocating Engines

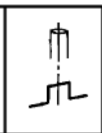
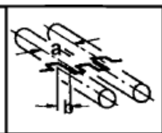
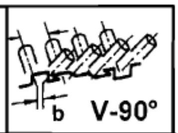
	Type							
Free forces	1. Order	F_{01}						
Free forces	2. Order	F_{02}	$2 \cdot F_{02}$		$4 \cdot F_{02}$			
Free moments	1. Order		$F_{01} \cdot a$	$\sqrt{3} \cdot F_{01} \cdot a$				$\sqrt{10} \cdot F_{01} \cdot a$ *
Free moments	2. Order			$\sqrt{3} \cdot F_{02} \cdot a$			$2 \cdot F_{02} \cdot b$	

Table 3 show the analyzed resultant mass forces and mass couples of the reciprocating engines for different type of configurations. Through this table, it is shown that the best configuration that will eliminate both the free moments and free forces will be a 6-cylinder in-line engine configuration.

CHAPTER 3: METHODOLOGY

This project is separated into a few phases where different methods are carried out. In the beginning, identifying the problem and objective of project was performed. Next the project background study was conducted. Then the project will proceed with the research phase where ample studies and literature review is being done. The project would then be continued with modelling stage using relevant software. The methodology involved for all project phases is explained at the few methodology breaks down below:

3.1 Research

Preliminary research was done to explore the topic further. Sufficient reading was done on relevant books from the Information Resource Centre as well as conference proceedings and journals that are available in the internet. The research will be carried out throughout the study to better understand the project. As dynamic analysis has a very wide scope, it will be narrowed down to the unbalance forces of the reciprocating compressor.

Literature reviews on research papers, journals and articles previously done by researchers were conducted. The centre of the literature reviews includes balancing of reciprocating compressors and engines, vibration analysis of the reciprocating compressor and unbalanced forces in single and multi cylinder engines. Nevertheless, it is not limited to the few focuses stated as it changes may develop through the project as it goes on. Literature review would be done constantly throughout the project to gather more information or as data validation process.

3.2 Modelling and Simulation

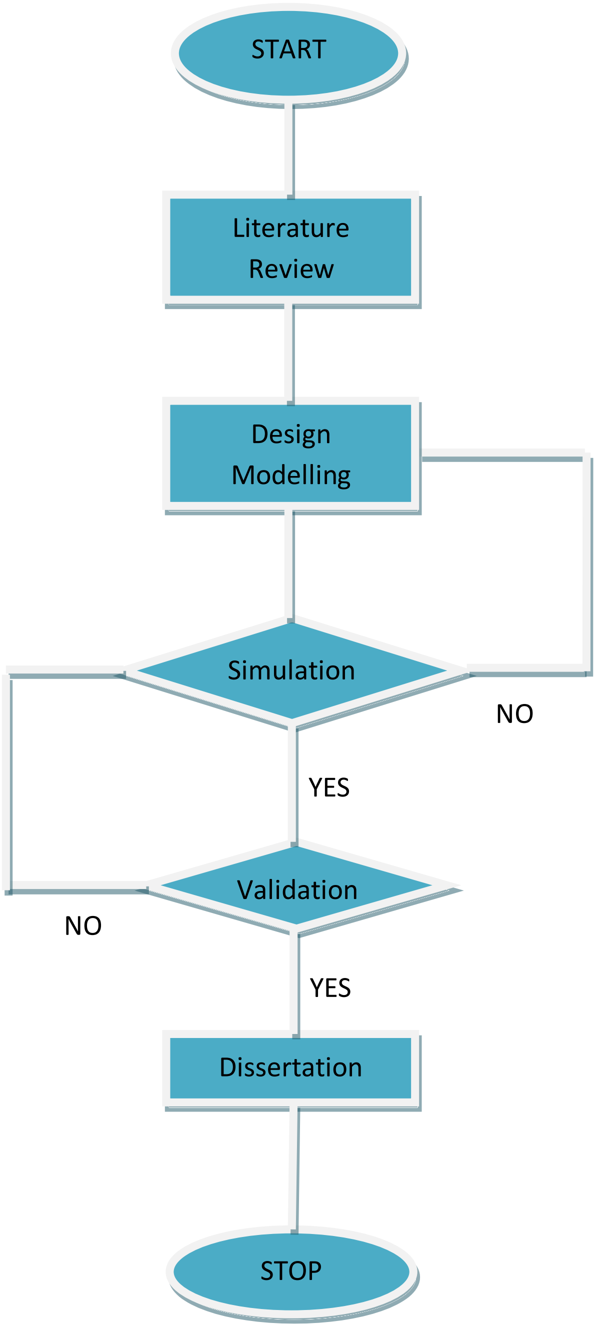
ADAMS software is used directly to simulate the forces in X, Y and Z direction for a single crank-crosshead-piston assembly. Links in ADAMS were set up and joined in order to simulate. The dimensions were not to scale with the actual reciprocating compressor mentioned in the problem statement. Weight was also added into the simulation to obtain a more accurate simulation and result.

Modelling out the primary internal components of the reciprocating compressor using software called CATIA. Firstly, the crankshaft was modelled out, since the dimensions, tolerances and other relevant measurements of the compressor is kept confidential to the original engineering manufacturer (OEM), the compressor modelled is similar to the compressor mentioned in the problem statement in terms of component arrangement and not to scale or to the exact dimension. Modelling in CATIA consist of drawing out using the software platform and then extrusion to produce a 3D model. When each component is completed, the components are added in together through the CATIA assembly platform. After the modelling is completed, the centre of gravity markers for each component connection is taken down. The next step is to save each component in the assembly drawing as 'igs' format so it can be imported into ADAMS to be simulated. Through the ADAMS software, the whole operation can be simulated and the forces in X,Y and Z direction can be analyzed for the three main components which is the crankshaft, crosshead and the piston. Vibration analysis can be obtained after simulating and through plotting of the graphs.

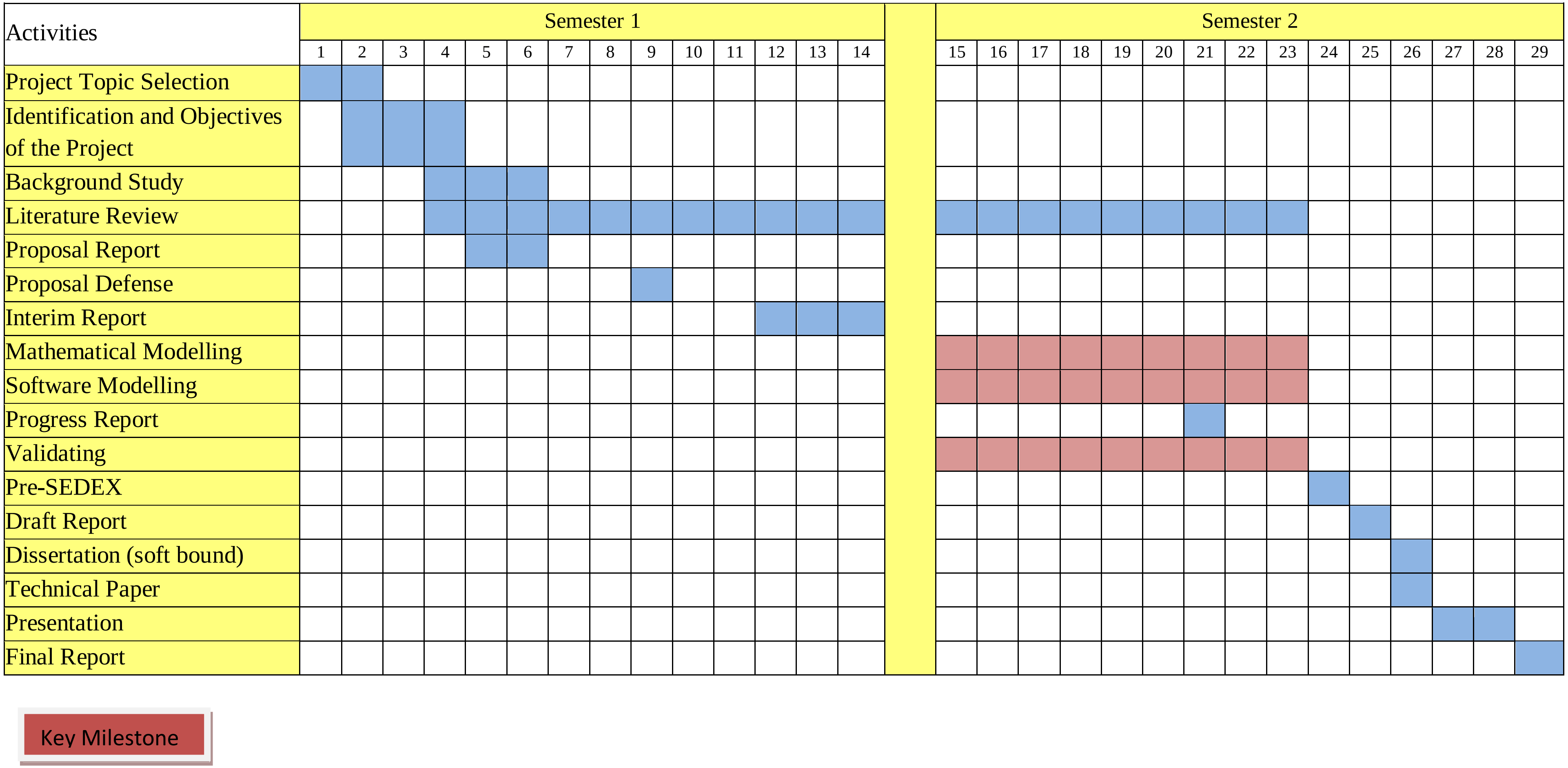
3.3 Data Analysis

After simulating is done, the software, ADAMS, will be able to plot out graphs of forces in X,Y and Z direction for the primary components of the compressor. Another set of graphs will be plotted for vibration analysis; this graph is obtained through the difference in forces during static simulation and dynamic operation. By obtaining this, analyzing the compressor will be possible and more accurate.

3.4 Project Flow Chart



3.6 Gantt Chart



CHAPTER 4: RESULTS & DISCUSSIONS

The reciprocating system is to be modelled out before it can be simulated and analyzed. The exact design parameters of the reciprocating compressor that is to be modelled cannot be obtained since it is restricted to the original equipment manufacturer, thus a reciprocating mechanism is modelled out that is similar to the equipment. This is to prove that the system, if given the exact parameters, can be analyzed through modelling and simulation, thus able to magnify into the factors causing the problem.

4.1 Modelling and Simulation

Figure 9 shows the model of a single crank-crosshead-piston configuration mechanism modelled in ADAMS to analyze a single cylinder composition and to depict the forces acting on the piston and the crosshead.

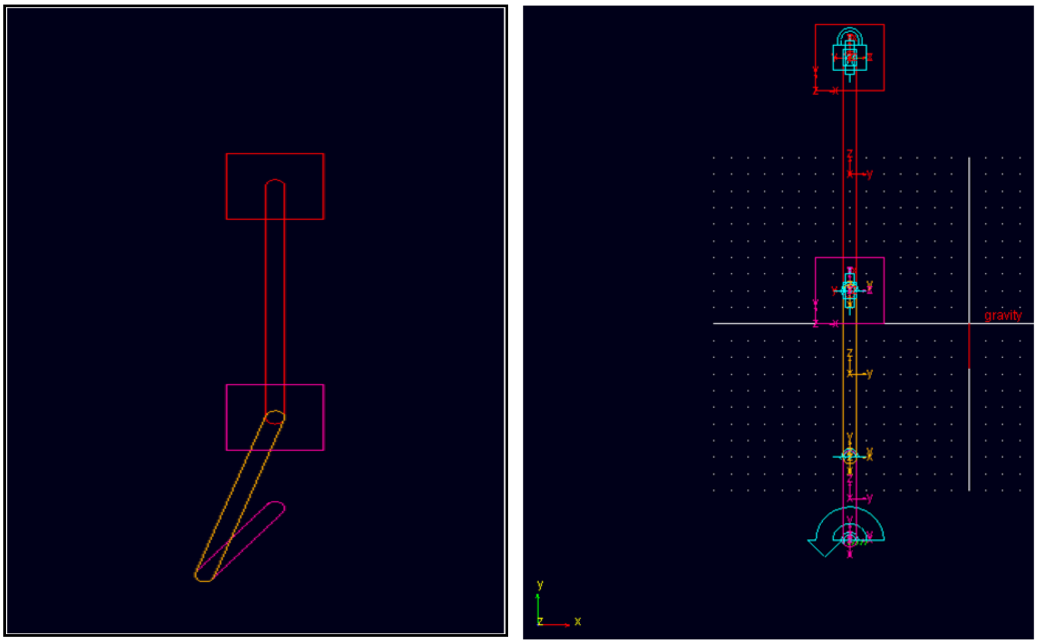


Figure 9 ADAMS Analysis on Single Crank-Crosshead-Piston Configuration Mechanism

Figure 10 and Figure 11 below shows the model of the reciprocating mechanism created in CATIA after assembly the parts has been assembled together. Figure 12 shows the model after it has been imported into ADAMS. After simulation, the graphs to analyze can be plotted.

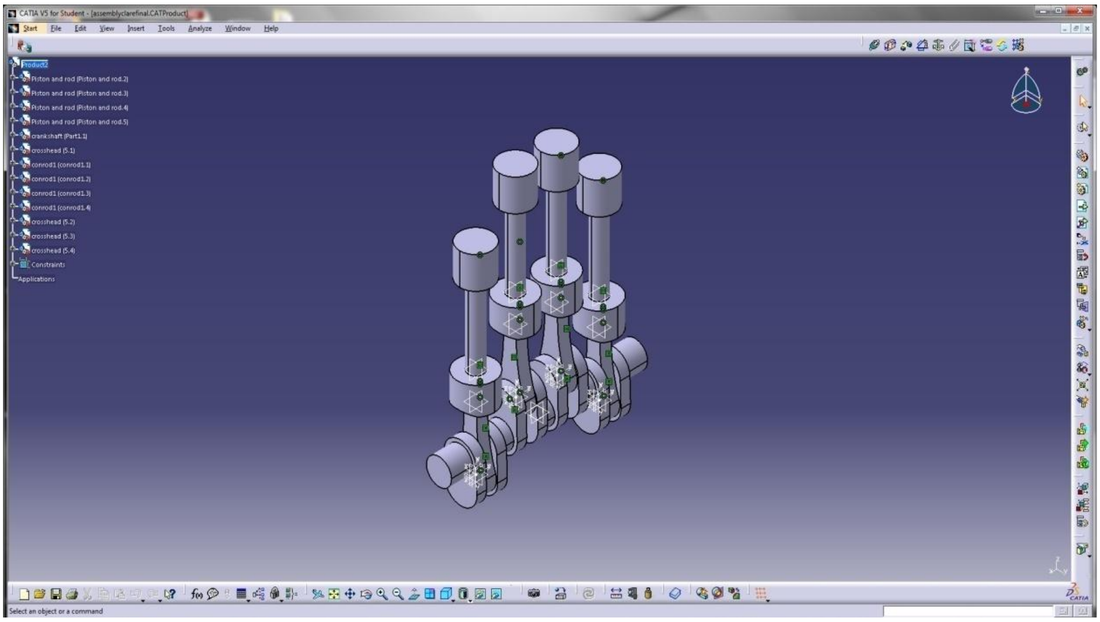


Figure 10 Completed Model in 3-dimensional View

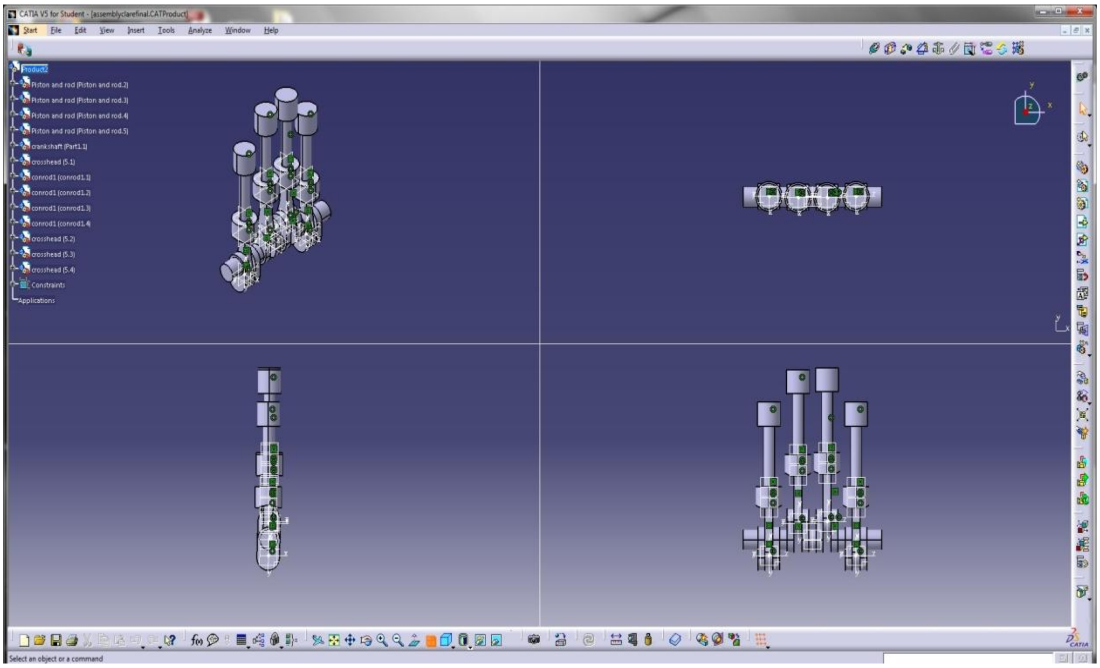


Figure 11 Completed Model in Multiple Views

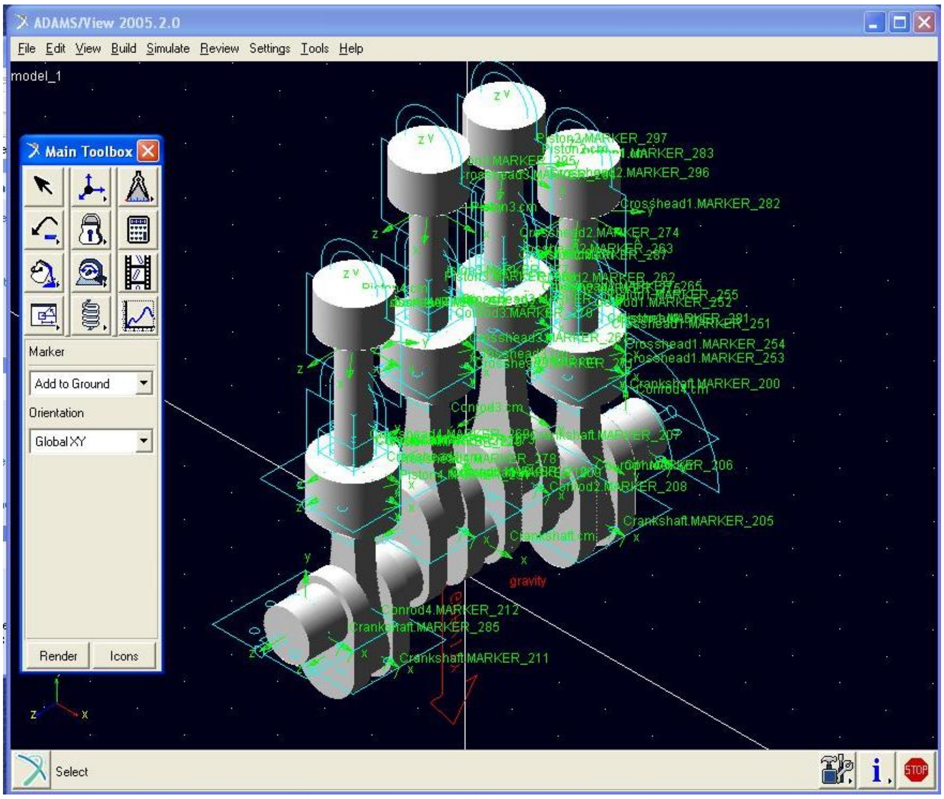
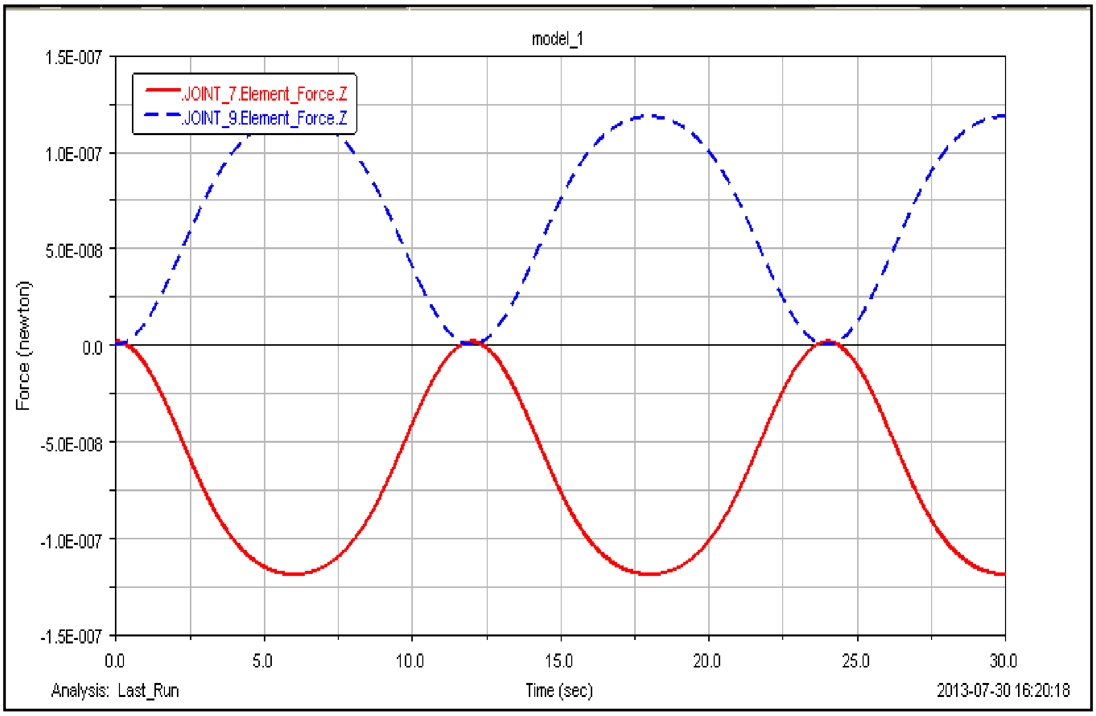


Figure 12 CATIA model Imported into ADAMS

4.2 Graph Analysis

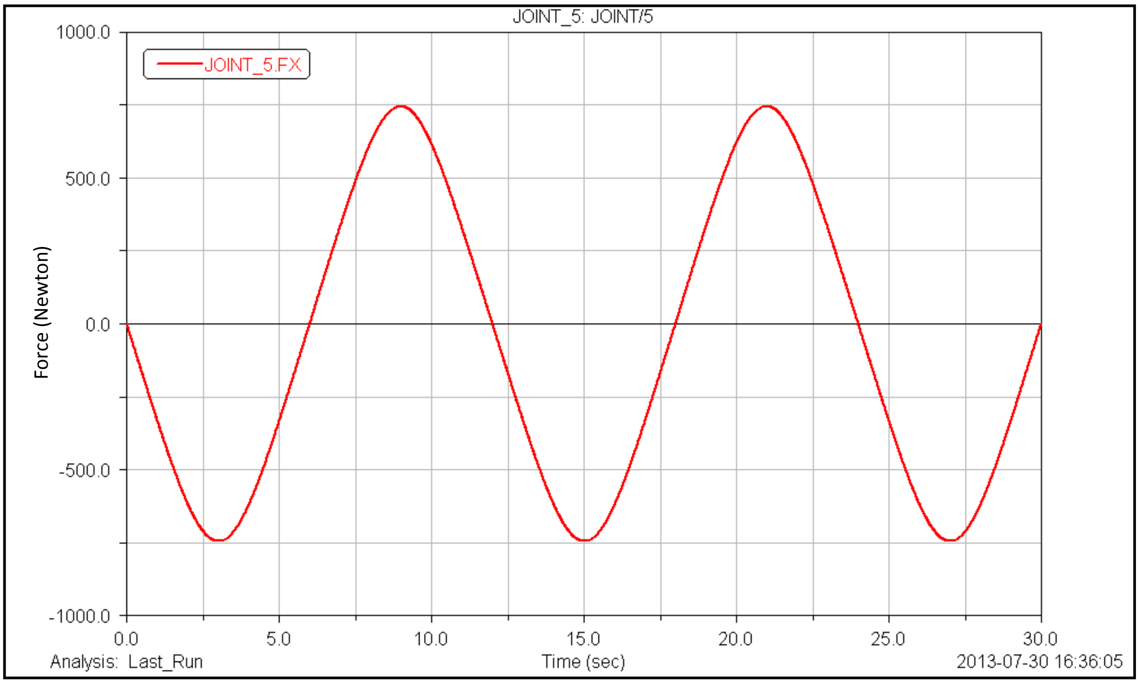
The elements analyzed were the forces in X, Y and Z direction given Y is pointing upwards in the global axis. The components that were analyzed were the piston, crosshead and crankshaft. Since the joint at the crosshead is has both revolute and translational, thus both movement is analyzed in the graph.

4.2.1 Element Force Analysis for Single Cylinder Configuration



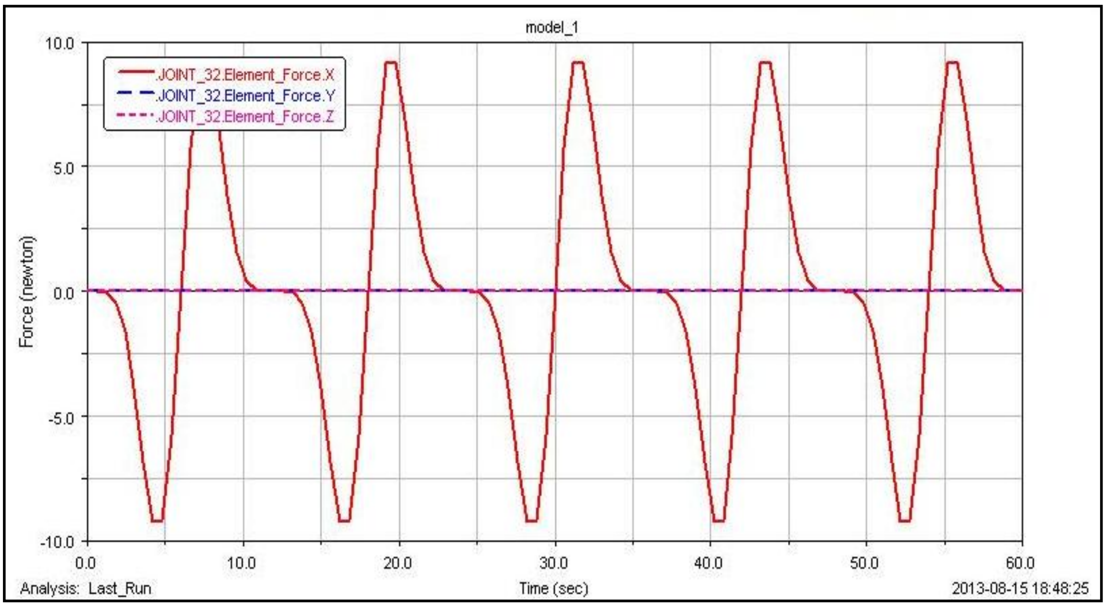
Graph 1 Piston Force Analysis

From Graph 1, it can be seen that there is very minimal element force on the piston and can be negligible. Whereas the element force at the crosshead is very significant, this is shown in Graph 2. This shows that the crankshaft-crosshead-piston configuration does indeed eliminate most of the side forces that is experienced by the piston when translating the rotational movement of the crankshaft to a transverse movement, then transferring it to the crosshead.

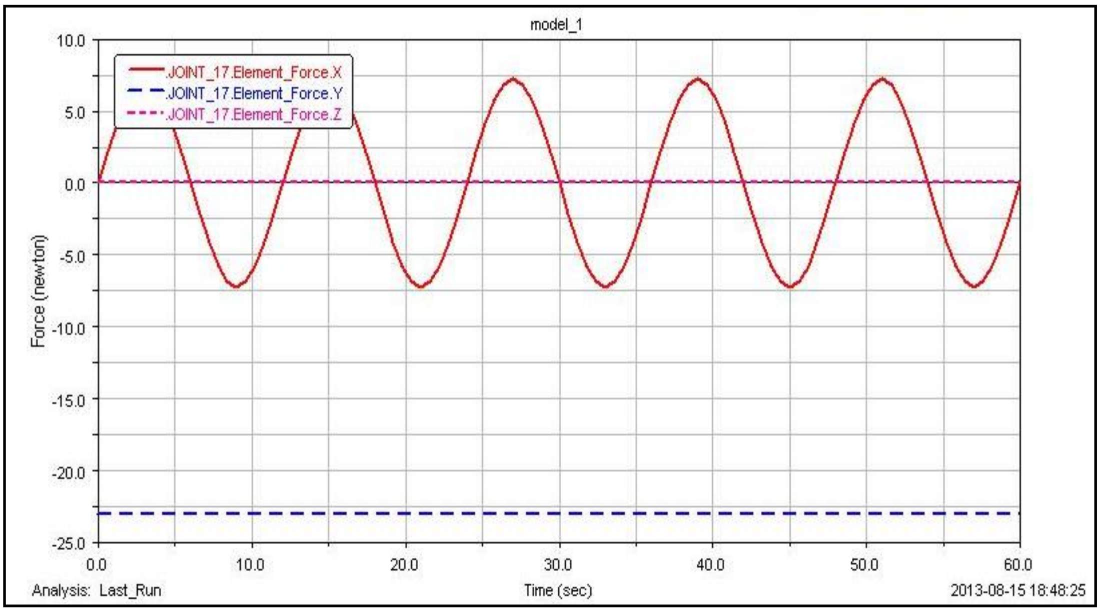


Graph 2 Crosshead Force Analysis

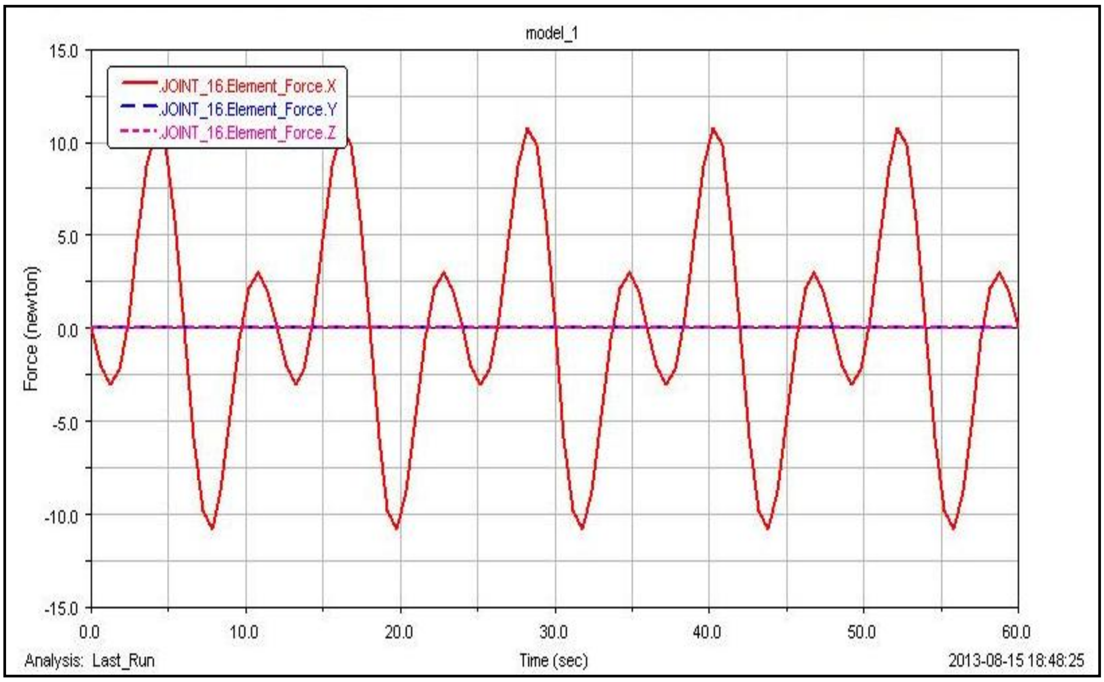
4.2.2 Element Force Analysis at Cylinder 1



Graph 3 Piston Force Analysis

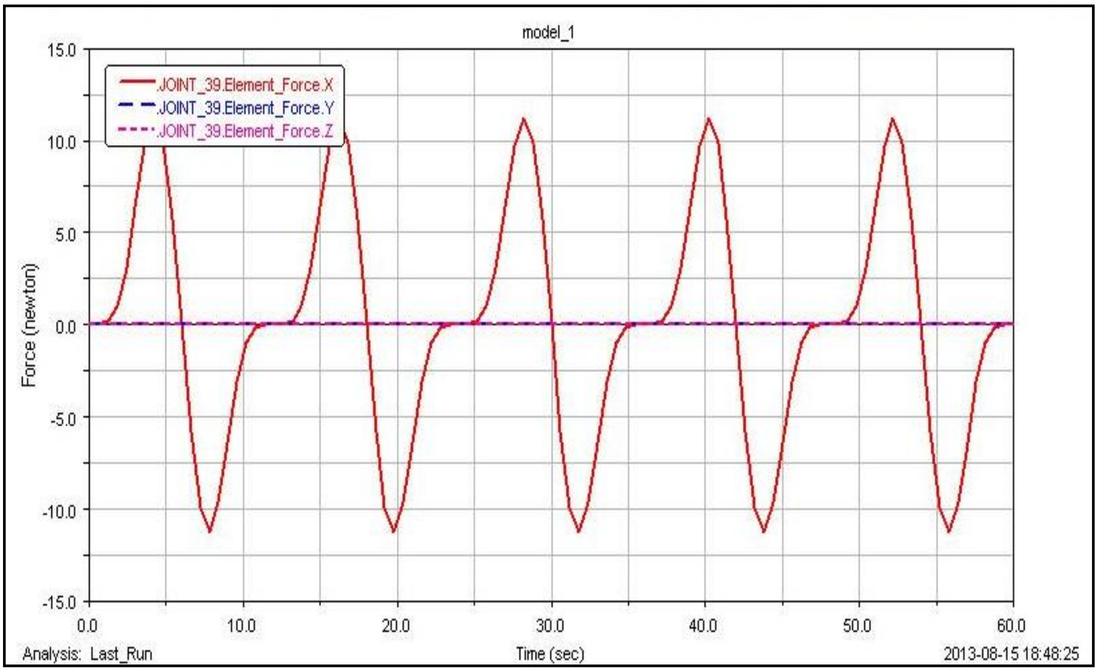


Graph 4 Crosshead (Force Analysis at Revolute Joint)

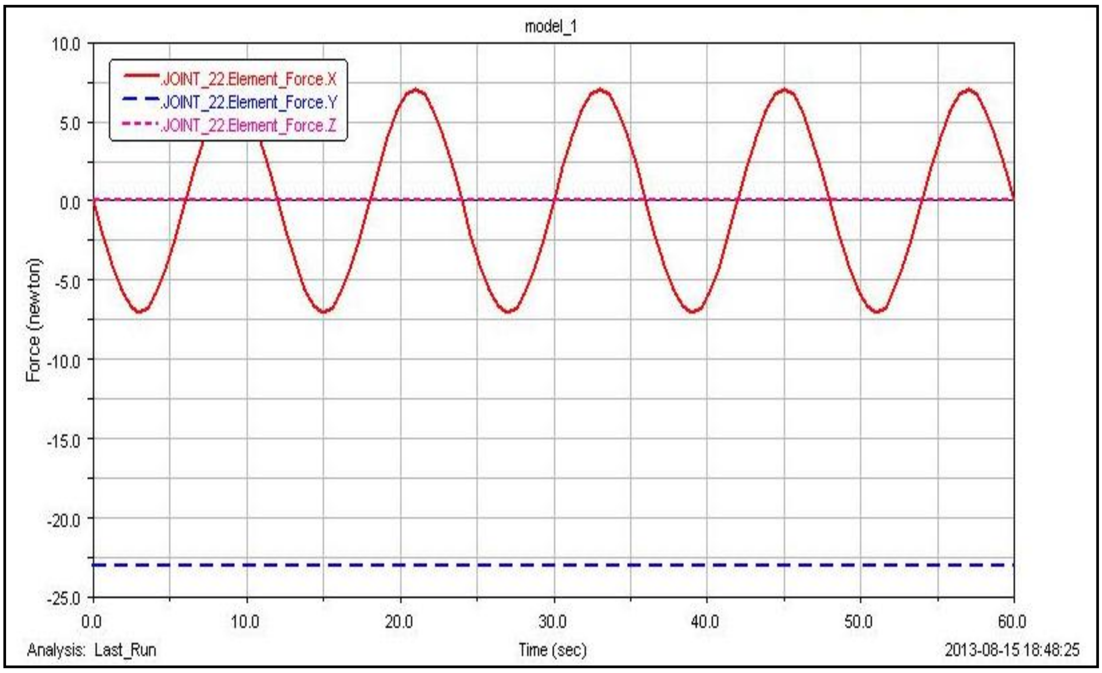


Graph 3 Crosshead (Force Analysis at Translational Joint)

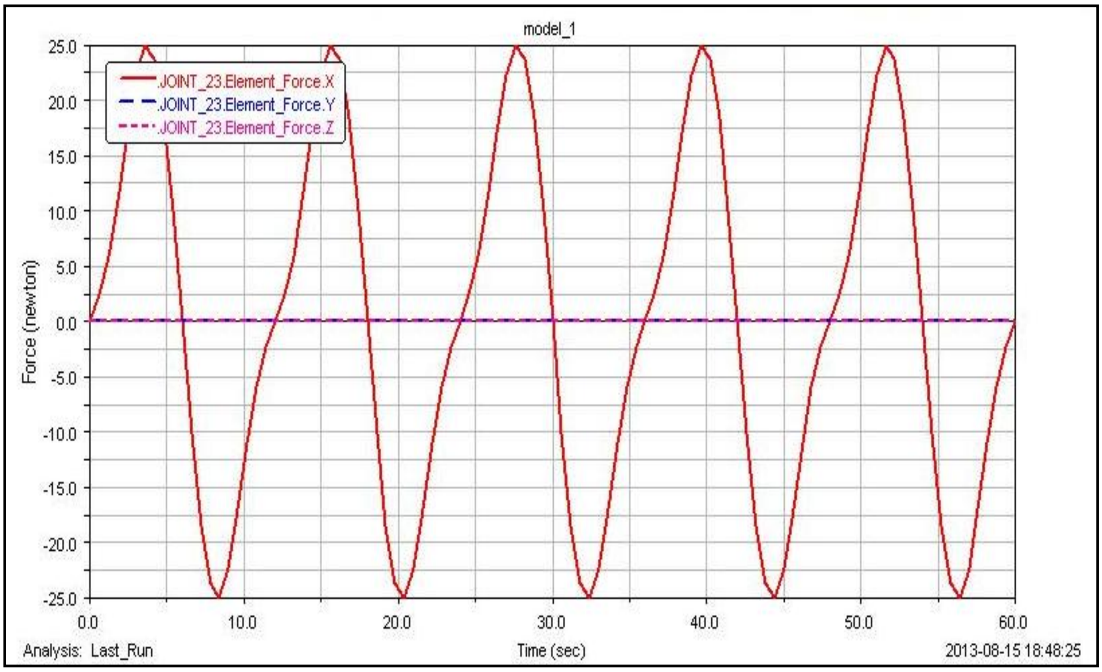
4.2.3 Element Force Analysis at Cylinder 2



Graph 4 Piston Force Analysis

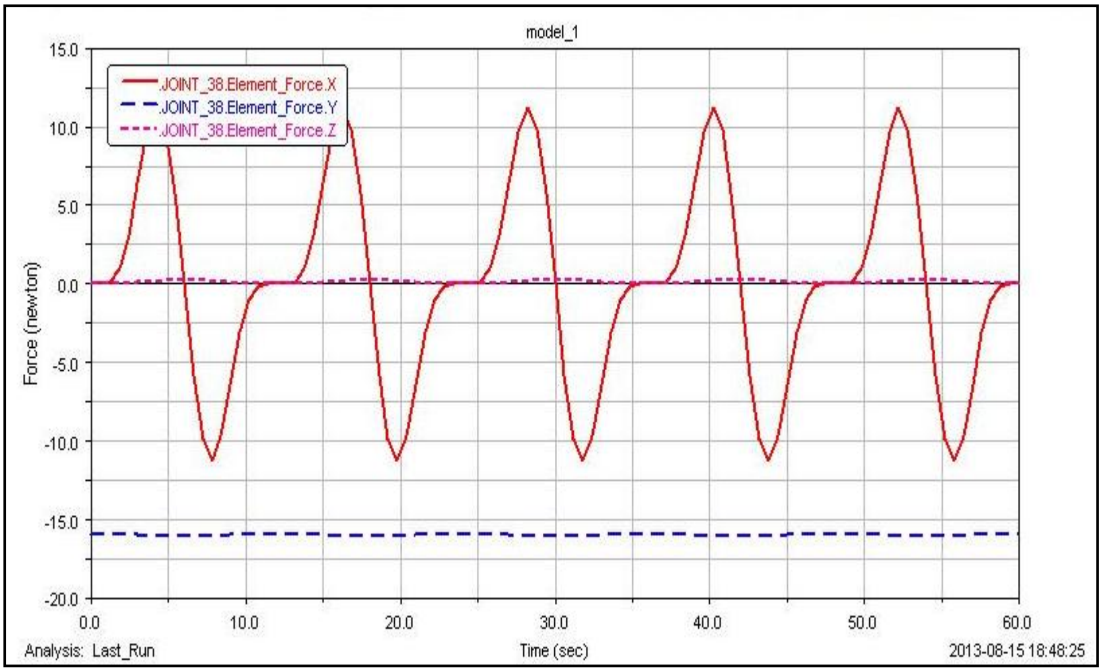


Graph 5 Crosshead (Force Analysis at Revolute Joint)

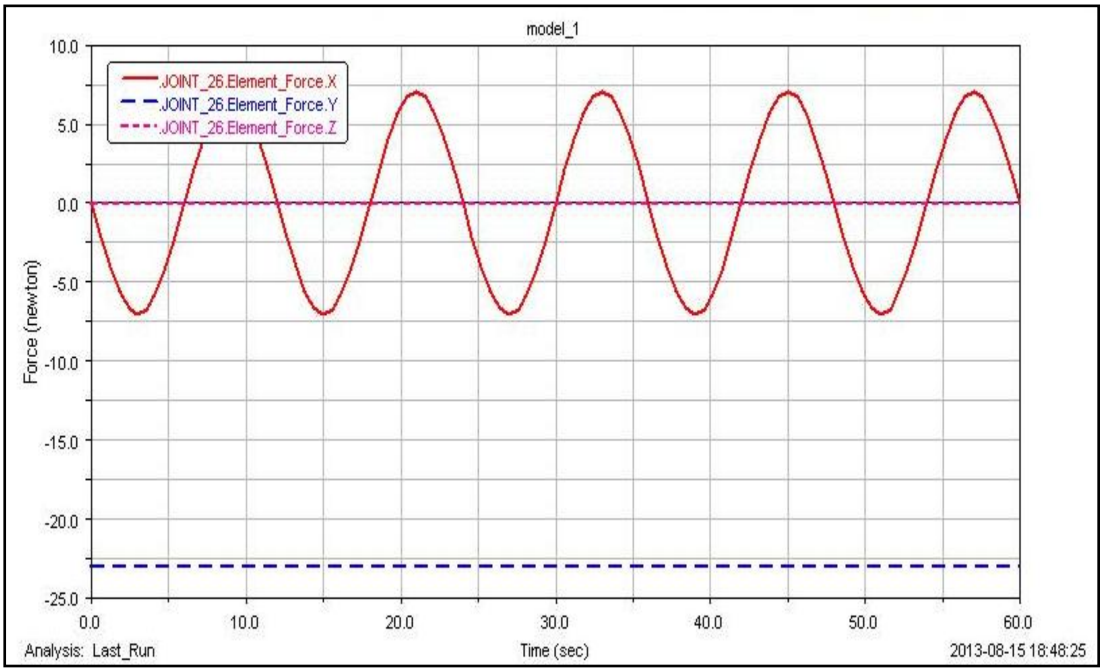


Graph 6 Crosshead (Force Analysis at Translational Joint)

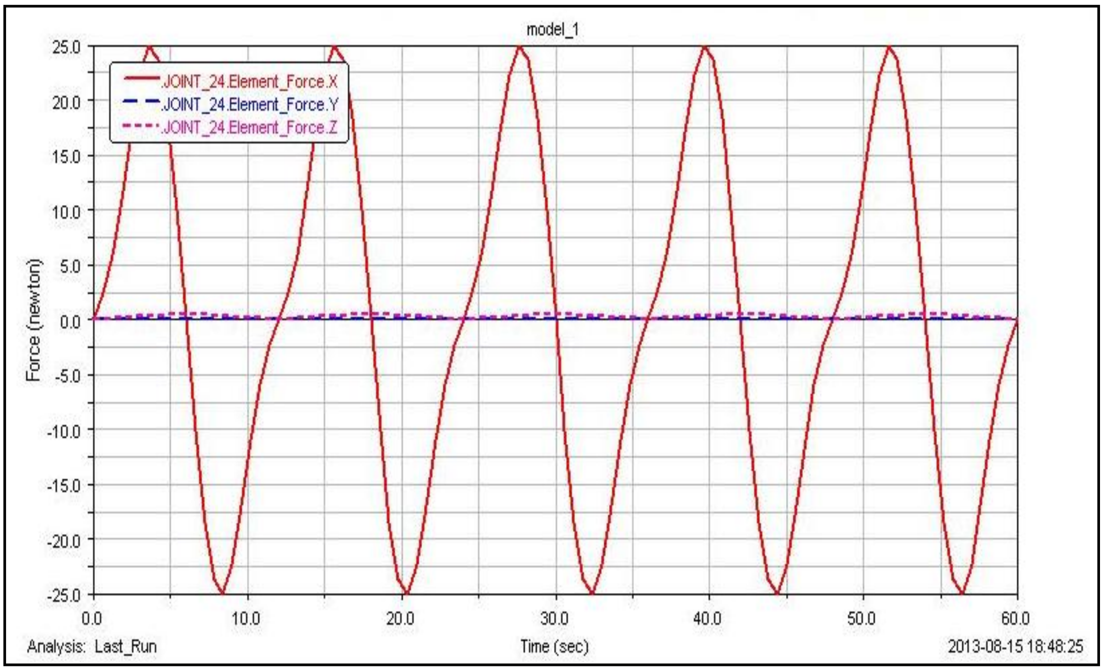
4.2.4 Element Force Analysis at Cylinder 3



Graph 9 Piston Force Analysis

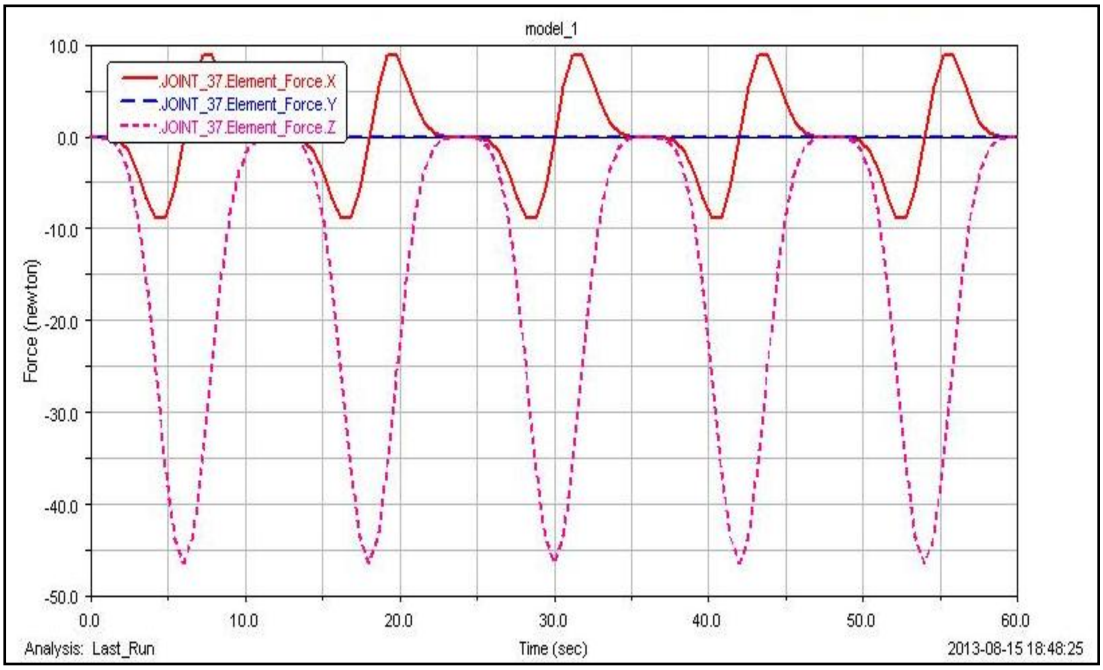


Graph 10 Crosshead (Force Analysis at Revolute Joint)

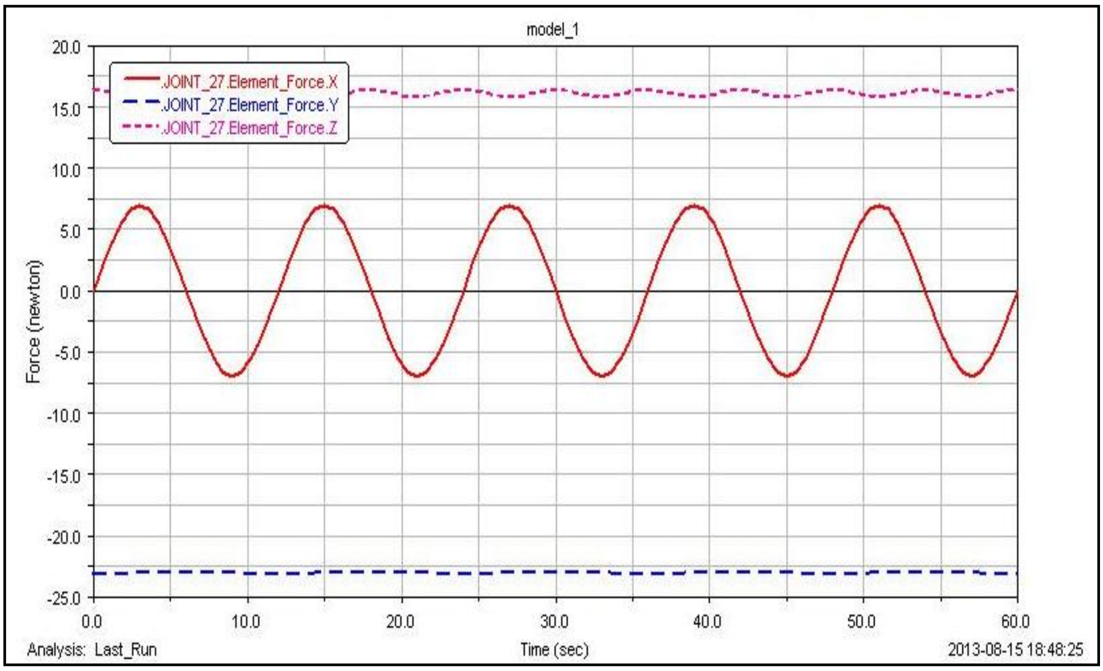


Graph 11 Crosshead (Force Analysis at Translational Joint)

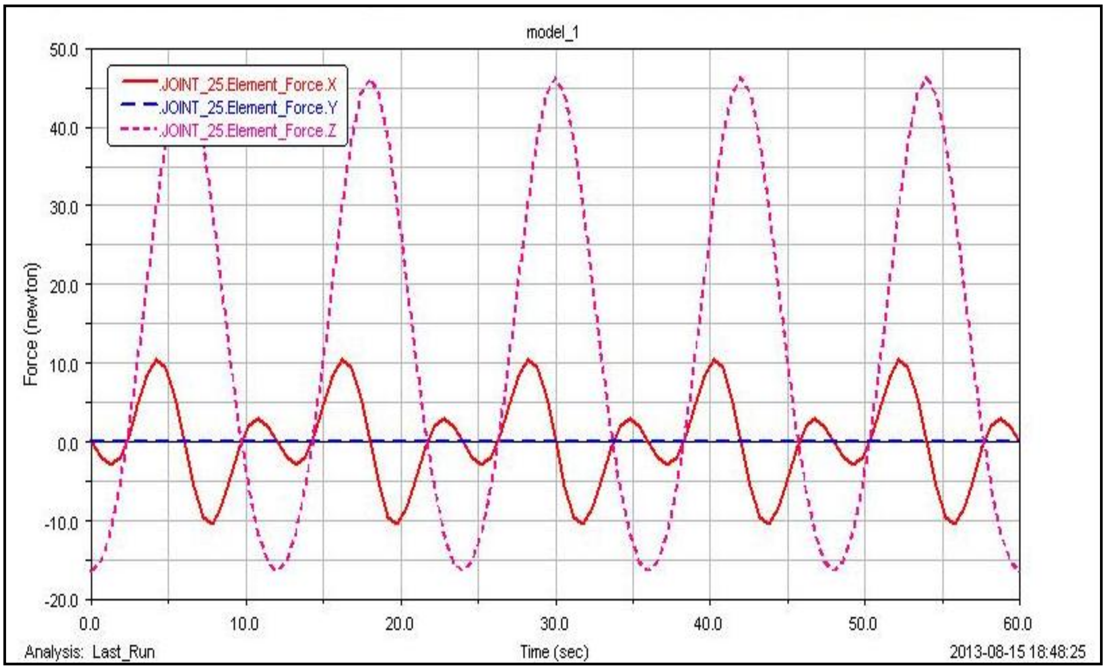
4.2.5 Element Force Analysis at Cylinder 4



Graph 7 Piston (Force Analysis)



Graph 8 Crosshead (Force Analysis at Revolute Joint)



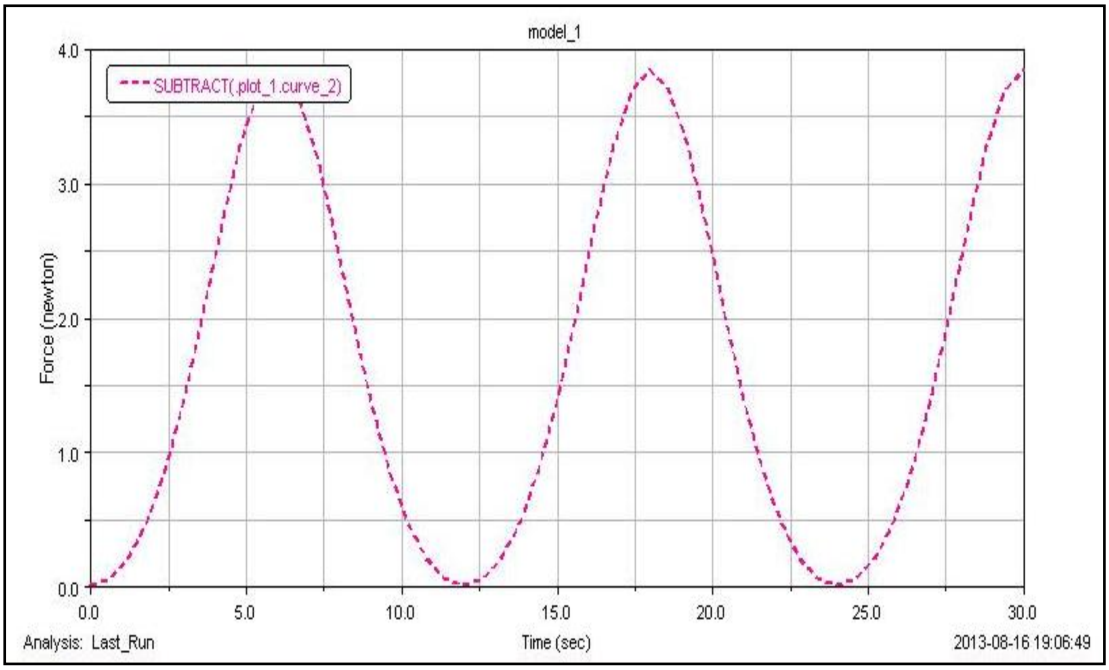
Graph 9 Crosshead (Force Analysis at Translational Joint)

Analyzing the graphs (Graph 3,6,9 and 12) of the force analysis of the piston, it is shown that in the X-direction the middle pistons of cylinder 2 and 3 have higher element for compared to the pistons of cylinder 1 and 4. The element force in the Y-direction is only present at the pistons of cylinder 2 and 3, while for element force of Z-direction is only present at the pistons of cylinder 3 and 4.

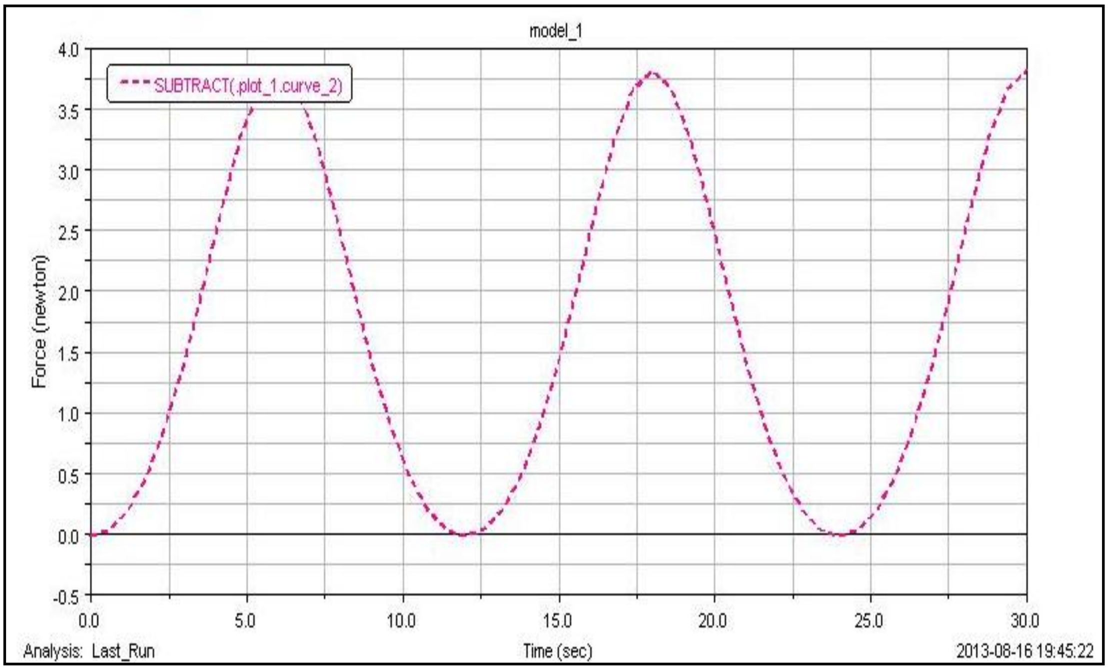
Comparing Graphs 4, 7, 10 and 13 the crosshead's revolute joint can be analyzed. In the X-direction, the readings for all the crossheads are considered constant and only a minimal increase of 0.5 at the crosshead of cylinder 1. As for the Y-direction, only two crossheads experience the element force which is the crossheads at piston 2 and 4. Look at the Z-direction, only two crossheads experience the element force which is the crossheads at cylinder 3 and 4.

As for Graphs 5, 8, 11 and 14, it depicts the element force for the crosshead translational joint. Analyzing the element force in X-direction, it is seen that the crossheads at cylinder 2 and 3 experience higher force compared to the force experience by the crossheads at cylinder 1 and 4. In the Y-direction, no element force is detected for all the crossheads. Analyzing the Z-direction, only crossheads at cylinder 3 and 4 experience the element force.

4.2.6 Vibration Analysis at Crankshaft



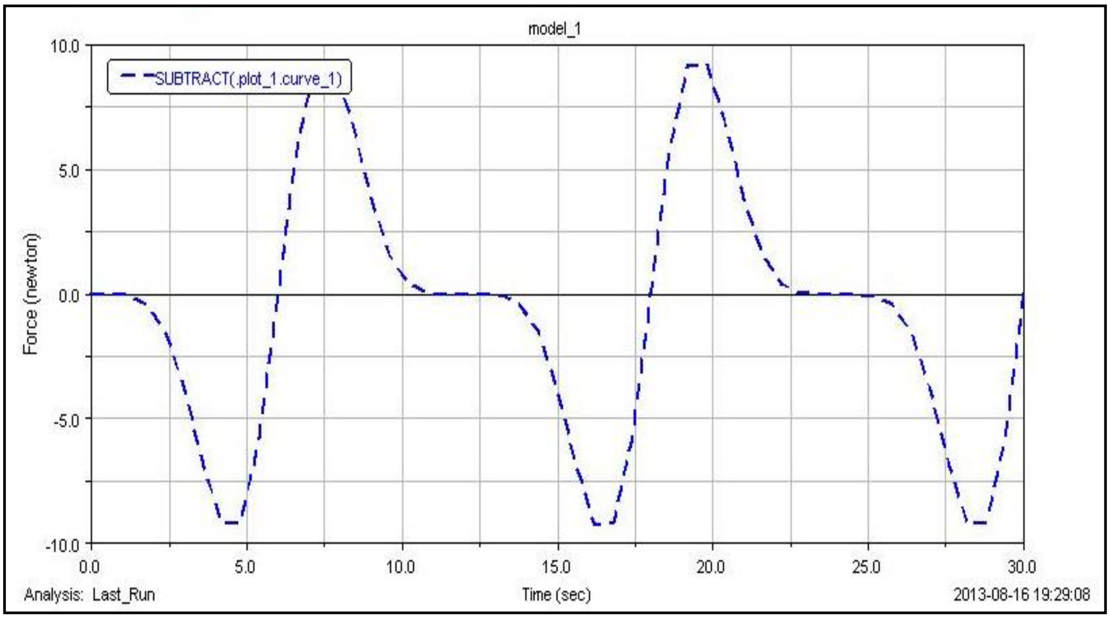
Graph 10 Vibration Analysis of Crankshaft at the End of Cylinder 1



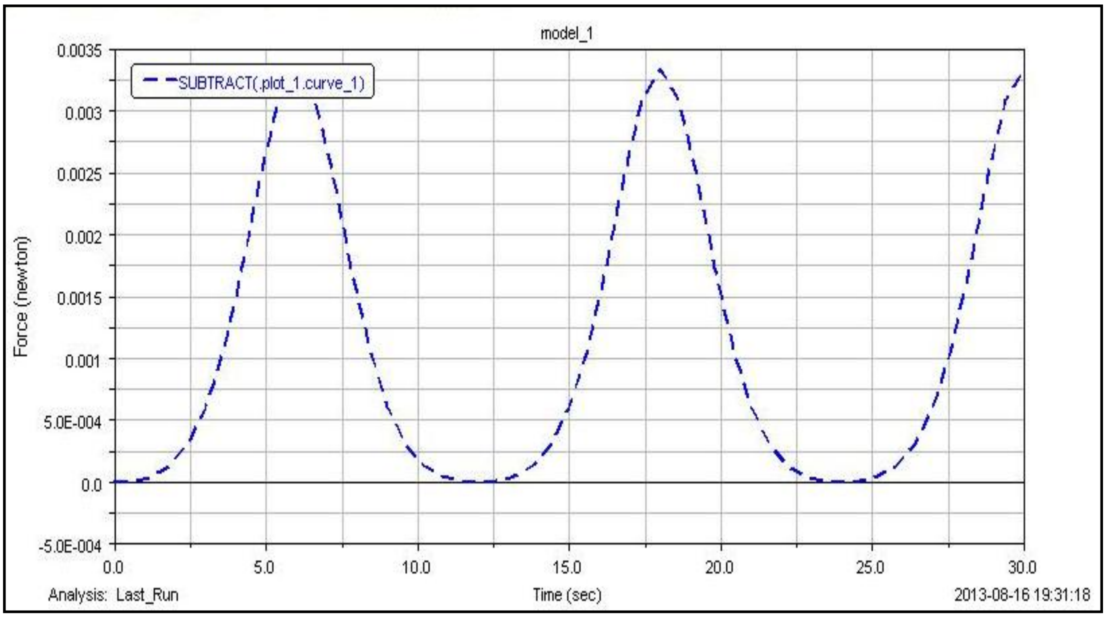
Graph 11 Vibration Analysis of Crankshaft at the End of Cylinder 4

The graph plotted out for vibration analysis on the crankshaft shows that the vibration of the crankshaft is equivalent to at both ends of the crankshaft.

4.2.7 Vibration Analysis at Piston 1

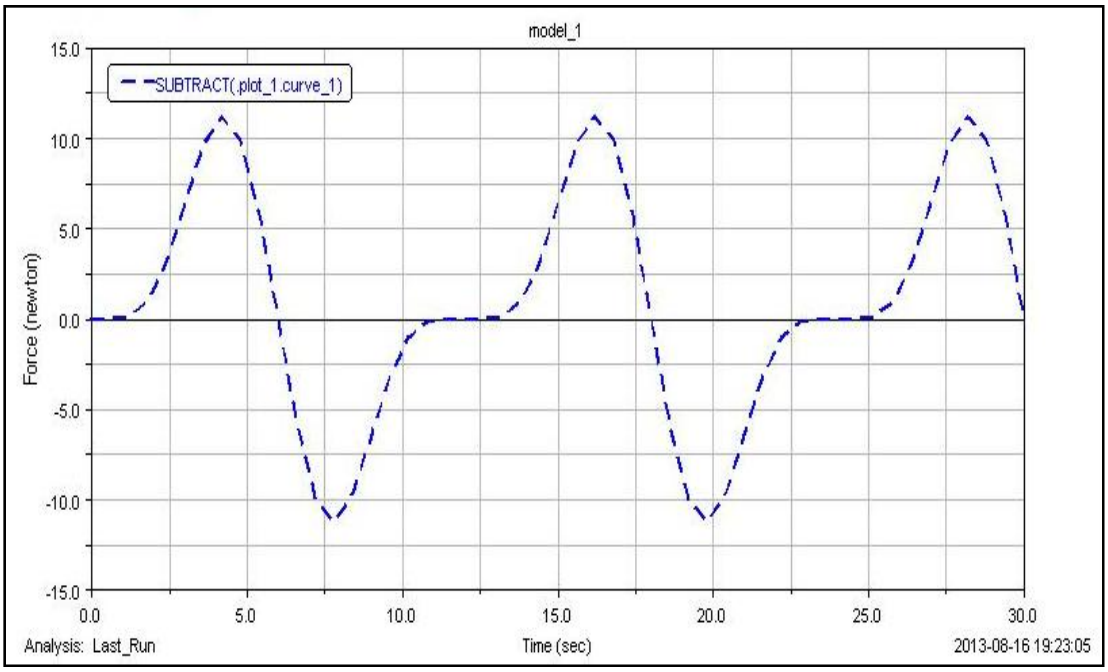


Graph 12 Vibration Analysis at Piston 1 in X-direction

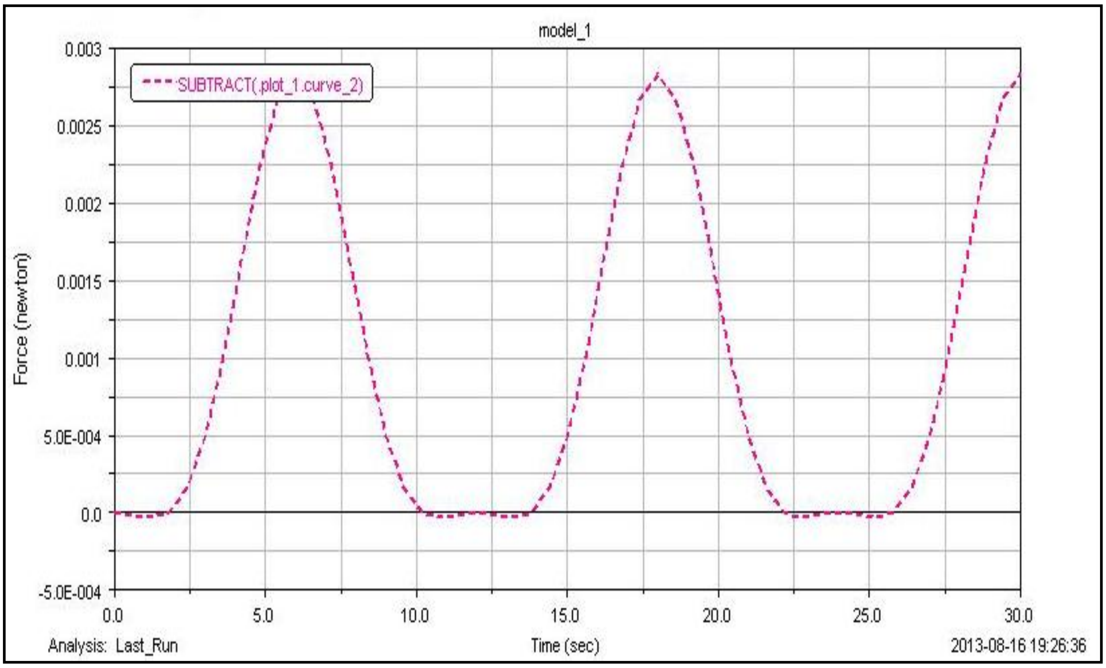


Graph 13 Vibration Analysis at Piston 1 in Z-direction

4.2.8 Vibration Analysis at Piston 2

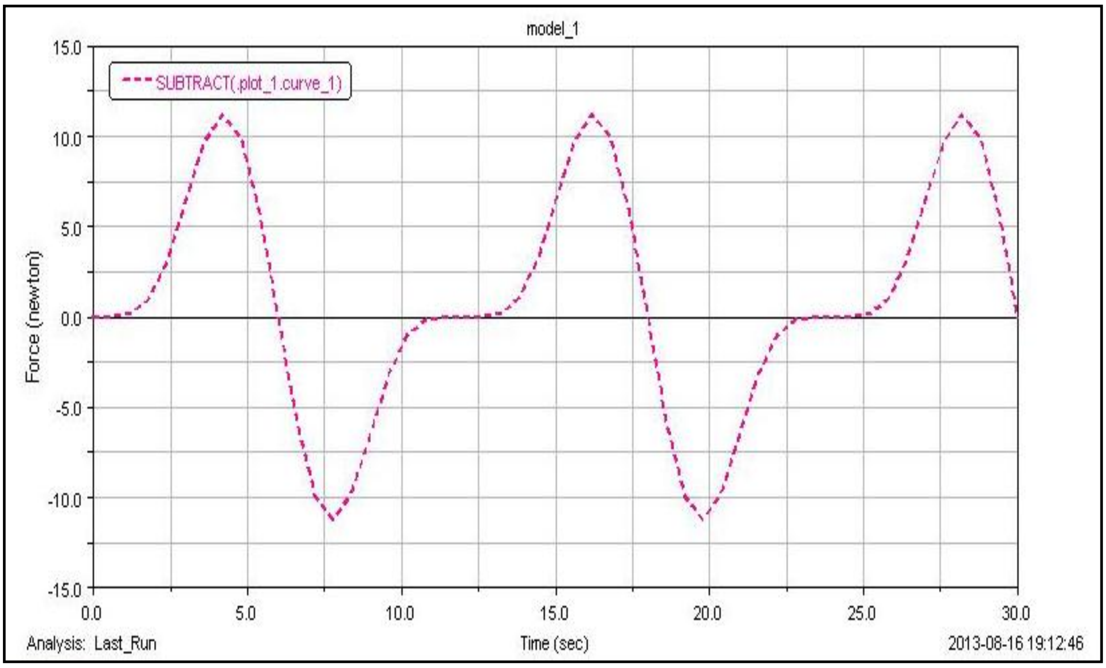


Graph 14 Vibration Analysis at Piston 2 in X-direction

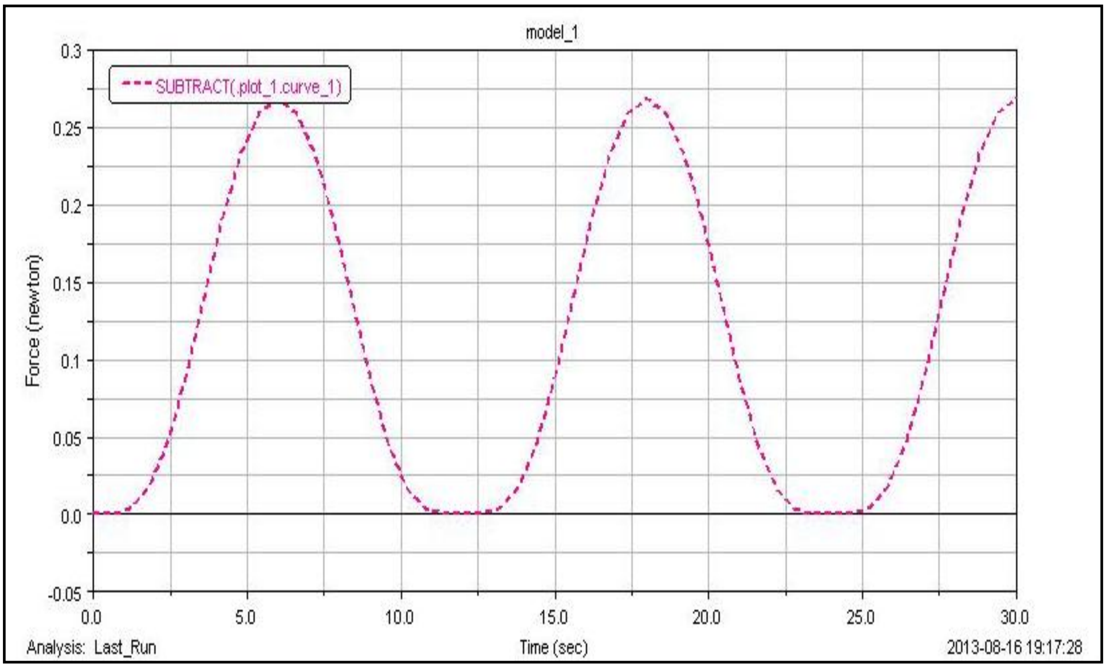


Graph 15 Vibration Analysis at Piston 2 in Z-direction

4.2.9 Vibration Analysis at Piston 3

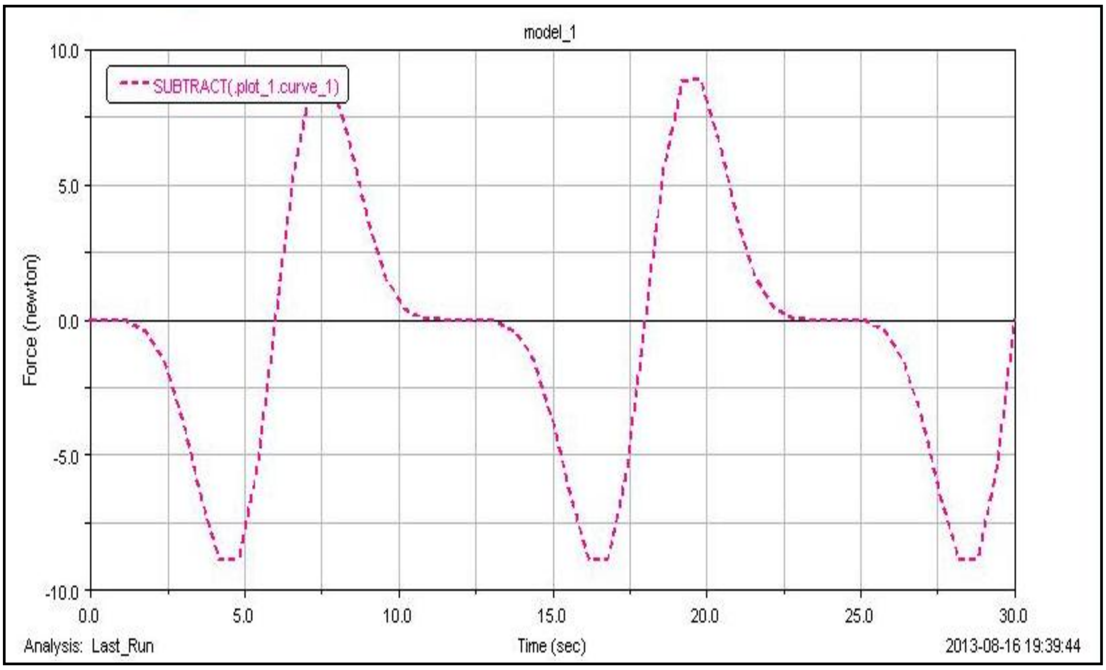


Graph 16 Vibration Analysis at Piston 3 in X-direction

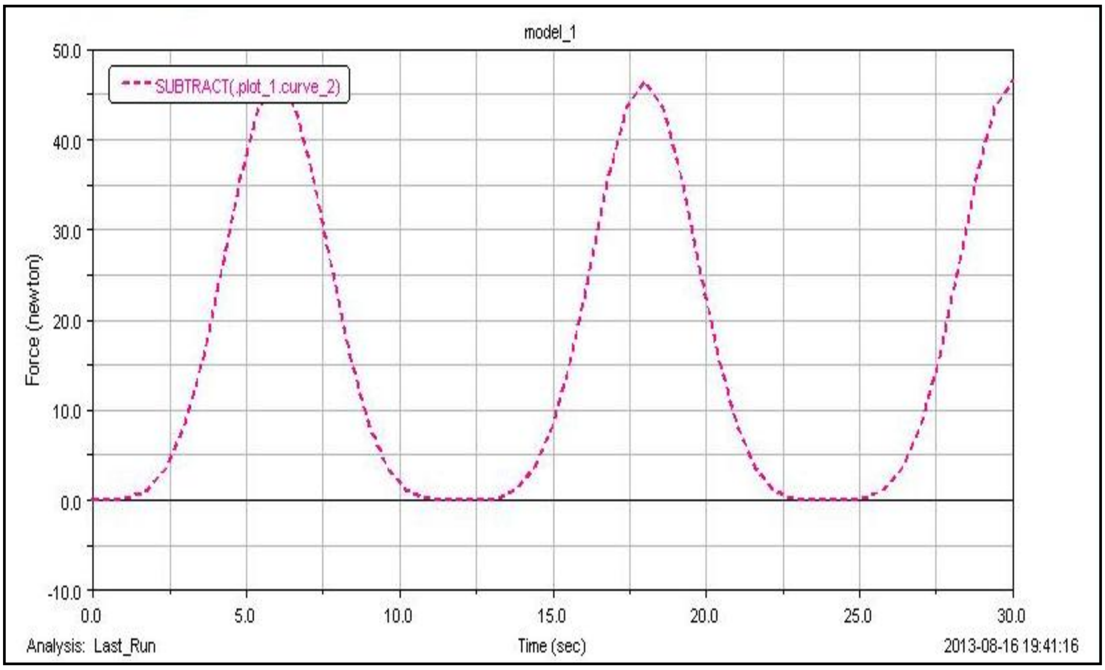


Graph 17 Vibration Analysis at Piston 3 in Z-direction

4.2.10 Vibration Analysis at Piston 4



Graph 18 Vibration Analysis at Piston 4 in X-direction



Graph 19 Vibration Analysis at Piston 4 in Z-direction

From the vibration analysis graphs, it is shown that there is vibration in the piston. Comparing the X-direction and the Z-direction, it can be seen that the vibration is larger at the X-direction compared to the Z-direction. However, at piston 4, the vibration in the Z-direction is the highest compared to all the vibration data. This shows that the compressor modelled on a whole experiences vibration in the piston and this can cause the piston to knock onto the wall of the cylinder and to rub against it. This may pose a problem to the compressor.

CHAPTER 5: RECOMMENDATIONS

Future recommendations to further improve the results of the project would be obtaining the actual dimensions, tolerances and clearances from the OEM to achieve the actual results. Modification towards the model can be done to attain a more complete and similar model for example modelling a more likely shape for the crosshead as shown in Figure 13.

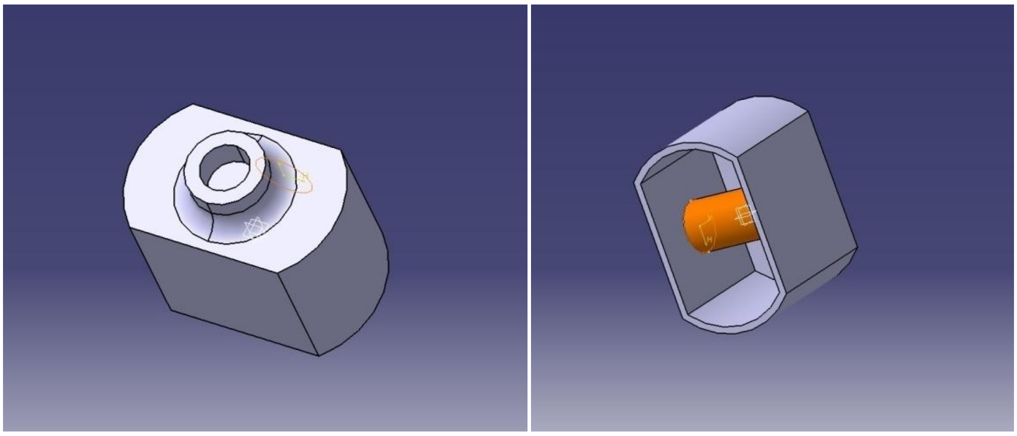


Figure 13 Crosshead Top and Bottom View

Apart from that, the crankshaft design is important to reduce the overall system vibration. Thus it is important to model the crankshaft according to the recommended star-diagram in Table 2 so that minimum vibration is produced as shown in Table 3. Counter mass balancing can also be placed as shown in Figure 6 to balance the compressor.

Nonetheless, time is an important factor to play in the completion of the project. Due to time constraint, a more similar compressor cannot be modelled out as it takes more time to complete a more complicated model.

CHAPTER 6: CONCLUSION

The project has reached its end with the results and analysis that was done and tabulated. The objective in identifying the unbalanced forces of the reciprocating compressor is achieved by modelling the compressor and analyzing the forces and vibration acting on its components. By using CATIA and ADAMS, the modelling and simulating of the compressor to obtain its dynamics was also achieved. Through literature review, recommendations could be made to improve the model and also the design of the compressor. Apart from that, suggestions on how to improve the project were given as well. On a whole, the project has been completed and has achieved its objectives.

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APPENDIX

APPENDIX I