Vibration and Acoustic Pre-Assessment Study for Free Piston Engine Structure

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Abstract

This paper presents modelling and simulation study of vibration and acoustic for a new free piston engine. The free piston engine is a new engine concept where its piston motion is not restricted by the crankshaft component. The free movement of the piston influenced by forces developed from the fuel combustion process and air compression in the engine. The piston movement has become an issue or a problem which consequently developed vibration to the engine structure because of the unbalance forces. Vibration analysis has been developed using finite element software which is MSC.PATRAN in order to determine the natural frequency and frequency response of the engine structure. Theoretical development of the engine balance motion and frequency response has also been conducted. From the simulation and finite element analysis, the force response pattern of the engine vibration can be determine and compare with its natural frequency. The vibration analysis will then be used as the input data for acoustic analysis of the engine. The acoustic analysis used boundary element method coupled with finite element method to determine the noise level produce by the engine structure. This integration determined the noise - frequency data that affected the engine structure towards the occurrence of engine noise especially when engine is in operation mode.

Keywords: Vibration, Acoustic, Free Piston Engine, Finite element analysis, Boundary element method

1. Introduction

Many investigations and studies have been motivated by the engineering applications of vibration, such as the design of machines, foundations, structures, engines, and control system. In all vibration situations, the machine component or structure could fail because of material fatigue resulting from the cyclic variation of the induced stress. Meanwhile, the transmission of vibration could create excessive noise and reducing the reliability of a machine or structure. Most prime movers would have vibration problem due to inherent unbalance of the structure itself or in the engine. The unbalance may be due to faulty design, poor manufacture or other factors.

Other factor that indirectly influences the unbalance is the natural frequency of the structure design itself. This is to avoid the appearance of resonance especially during working condition. It is important to determine the natural frequency of the engine structure, which will lead in the determination of frequency response and the frequency range for noise of the engine at the design stage. According to [1], the rotary engine noise for four-stroke engine is dominant in the middle to high frequency- range from 500 to 2500 Hz (air borne noise area). It also has been determined that this noise is mainly caused by the fundamental natural vibrations of the cylinder block. Mentioned in [2] research, the cylinder block vibration that causes engine air borne noise is mainly excited by main bearing force, combustion force and piston side force. Although it is found [3] that the natural frequency of a linear motion engine as in free piston engine (FPE) that was used as a prime mover operates at a frequency that is significantly lower than the frequency of rotary or conventional engines but the new FPE design in this study has yet to be determined.

Free piston engine (FPE) is an engine with no rotary movement involved such as crank and camshaft. In this engine, two horizontally opposed pistons are mounted on a common connecting rod, which is allowed to oscillate between the two end-mounted cylinders [4]. As the piston moves in either direction, one cylinder will undergo the expansion process while the other undergoes compression process.

Many of the past research {5, 6, 7] on the engine indicated that combustion process occurs in both cylinders, in order to get the linear movement of the piston. However, in this study, combustion only occurs at one cylinder. A compressed pressure at another end cylinder will force the piston back and forth in an alternating fashion linearly. This engine as shown in Figure 1 can be categorized as single FPE with rebound devices but in this study it is called as kickback chamber.

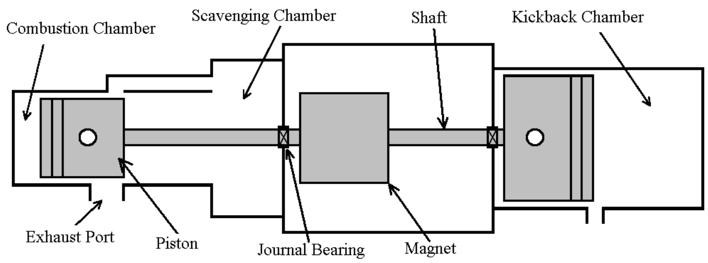


Figure 1 Free piston engine simple model.

Previous free piston engine study according to [8] has determined that a single free piston engine with rebound device has developed and unbalanced situation. This situation is considered in the design of the combustion chamber and kickback chamber of the new FPE structure. In order to determine the vibration and acoustic data of the engine structure, finite element method (FEM) was used in modelling and to predict the vibration characteristics such as natural frequencies, normal mode and frequency response of the engine at 3000 revolution per minute (rpm). Using FEM to predict the dynamic properties of any structure has become increasingly important in the industries especially in aerospace and automotive industry. One of the reasons this method was chosen is because complex structure can be configured and modelled using the finite element tools and the response at any desired point of the structure can be easily determined [9].

Furthermore, FEM models can be integrated with other methods such as boundary element methods (BEM). From reference [10], the BEM has been found to be a very promising tool in the investigation of acoustical problems. Data from vibration analysis will be the input of the acoustic data in order to determine the noise level of the engine structure where the point of air borne noise has been identified at the journal bearings of the engine.

2. Theoretical Modelling of Free Piston Engine

2.1 Dynamic System of Engine Structure

Controlling engine noise through structural modifications is a difficult task because it requires considerations of different engineering aspects including finite element modelling, modal analysis, damping and others [11]. However, to develop a solution required a study of the interactions between the engine structure and the internal forces. This will led to the need for understanding the dynamic behaviour of the engine structure of the engine structure. The statement agreed by [2], where the free piston engine modelling should begin with a dynamic analysis. These analysis begin by considering the case as an engine with two reciprocating pistons linked by a solid shaft, that oscillate back and forth in a left-to-right motion.

The system obeys Newton's second law where the right hand side of the equation (1) represents the summation of the forces that act in the plane of motion in x-direction,

$$m\frac{d^2x}{dt^2} = \sum_i F_{ix} \tag{1}$$

where, m is the mass, $\frac{d^2x}{dt^2}$ is the acceleration of the piston and $\sum F_{ix}$ is the summation of acting forces.

From the previous study [12], these acting forces are the combustion pressure (P_c), the scavenging pressure (P_s), the air kickback pressure (P_k), the piston ring-wall friction (F_f), and electromagnetic force (F_e). The forces balance on free piston is given by equation (2) and for after combustion and equation (3) indicate for after air kickback condition.

$$P_c A_c - P_k A_k - P_s A_s - F_f - F_e = m\frac{d^2x}{dt^2} \tag{2}$$

$$P_k A_k - P_c A_c + P_s A_s - F_f - F_e = m\frac{d^2x}{dt^2} \tag{3}$$

where, A is the area for combustion, kickback and scavenging with subscripts c, k and s respectively. The balance equations show that the acting forces that were generated during the combustion and kickback process in the engine. These forces reacted to the engine structures that create a vibration and noise to the engine.

Presently, in typical noise-structural engine systems, there are two basic structural elements that need to be analyse which are: 1) The internal load-carrying structure, piston-connecting rod or shaft, and 2) outer load carrying, cylinder block structure [13]. The motion of these internal structures which will contribute most of the noise to the outer structure. Theoretically, the unidirectional forces or acting forces will excites the top part of the engine cylinder head and the piston was excited by the combined gas forces plus the inertia forces.

The displacement of the solid shaft when the mass is subjected to the acting forces, F_{ix}, where the system can be illustrated similar to a simple linear spring mass system and its vibration can be represented by equation (4):

$$m\frac{d^2x}{dt^2}+c\frac{dx}{dt}+kx=F_{ix} \tag{4}$$

where, c is the damper constant, k is the stiffness constant, is the velocity, and x represents the displacement of the solid shaft. In order to determine the noise and vibration of the outer structure, the theoretical analysis of force balance for the internal structure was developed in normal mode and natural frequencies model.

2.2 Normal mode and natural frequencies

Normal mode study is importance in dynamic analysis since a system responds in its normal modes condition. From [14], the natural frequencies of the structures should be estimated at the design stage and, if required, the structure should be stiffened so that the natural frequencies will be increased above the frequency of the excitation forces or frequency response. This will avoid resonance from occur in the system and the importance of resonance consideration have been studied by [15]. In normal modes condition, in order to get its natural frequency, this acting forces as in equation (4) are assume to be zero and the dynamic model is based on the structural model used for a static analysis. Theoretical development of balance equation of the internal structure system is important as to determine the normal mode and natural frequencies of the structure. As illustrated in figure 2, k represents the 'spring' constant or stiffness of the shaft, m is the mass which subscripts *1*, *2* and *3* refer to piston combustion, magnet and piston kickback, respectively.

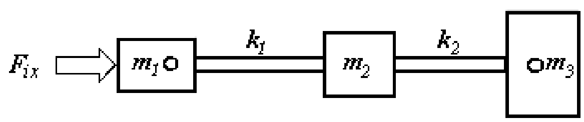


Figure 2. Model of internal structure.

The balance equation of the system will be:

$$\begin{aligned} m_1\ddot{x}_1+k_1x_1-k_1x_2&=F_1\\ m_2\ddot{x}_2-k_1x_2+k_2x_2-k_2x_3&=F_2\\ m_3\ddot{x}_3+k_2x_2-k_2x_3&=F_3 \end{aligned} \tag{5}$$

In normal mode analysis the acting forces, F_{ix} was set equal to zero. So the equation in matrix form will become:

$$\begin{bmatrix} m_1 & 0 & 0\\ 0 & m_2 & 0\\ 0 & 0 & m_3 \end{bmatrix}\begin{bmatrix}\ddot{x}_1\\ \ddot{x}_2\\ \ddot{x}_3\end{bmatrix}+\begin{bmatrix} k_1 & -k_2 & 0\\ -k_1 & k_2 & -k_2\\ 0 & k_2 & -k_2 \end{bmatrix}\begin{bmatrix}x_1\\ x_2\\ x_3\end{bmatrix}=\begin{bmatrix}0\\ 0\\ 0\end{bmatrix} \tag{6}$$

when a system vibrates in a natural modes, the solution is in the form,

$$x=Xe^{j\omega t} \tag{7}$$

therefore,

$$\begin{Bmatrix}\dot{x}_1\\ \dot{x}_2\\ \dot{x}_3\end{Bmatrix}=\begin{Bmatrix}X_1\\ X_2\\ X_3\end{Bmatrix}e^{j\omega t} and \begin{Bmatrix}\ddot{x}_1\\ \ddot{x}_2\\ \ddot{x}_3\end{Bmatrix}=-\omega^2\begin{Bmatrix}X_1\\ X_2\\ X_3\end{Bmatrix}e^{j\omega t}$$

where, ω and X are the natural frequency and the corresponding mode shape respectively and the frequency equations will be as equation (8) and by solving its determinant, the frequency of the system can be determined.

$$\begin{bmatrix} k-m_1\omega^2 & -k & 0 \\ -k & k-m_2\omega^2 & -k \\ 0 & -k & k-m_3\omega^2 \end{bmatrix} \begin{Bmatrix} X_1 \\ X_2 \\ X_3 \end{Bmatrix} e^{j\omega t} = 0 \tag{8}$$

According to [16], a study of and solution for eigenvalue and eigenvectors is importance because it's defined the undamped free vibrational modes and frequency of the system. In normal modes study, the system was considered as an undamped, free vibrations structure of a finite element assembly with *N* degrees of freedom,

$$M\ddot{x} + Kx = 0 \tag{9}$$

Assuming the acting forces is in periodic response, the equation arrived at the following *generalized eigenvalue problem* associated with the dynamic system in equation (1):

$$K\Phi = M\Phi\Omega^2 \tag{10}$$

where, Φ is the matrix containing the generalized eigenvectors along its columns and Ω^2 is a diagonal matrix containing the corresponding eigenvalue, i.e.,

$$\Omega^2 = \begin{bmatrix} \omega_1^2 & & \\ & \ddots & \\ & & \omega_n^2 \end{bmatrix}$$

The accurate estimation of the eigenpairs of a system is an important step in dynamic analysis since eigenvectors can often be used for *reduced order modelling* of the response of the system. That is, the response can often be accurately resolved by its projection along only a few of the generalized coordinates. Therefore, the accurate computation of the response of the structure is closely related to the accuracy of the first few eigenpairs. As in static, the accurate estimation of the eigenpairs depends on three key parameters: the choice of elements, choice of mesh and choice of solution procedure. In addition, factors such as using a lumped mass matrix instead of a consistent mass matrix can also be of significant importance.

3 Finite Element Modelling of Free Piston Engine Structure

The model of the engine structure was imported from a solid modelling software, CATIA to finite element software called MSC PATRAN. The imported 3-D solid model structures were assembled with other related components such as head, cylinder block, scavenging and related parts using Rigid Body Element (RBE) between nodes of adjacent parts. In linear elastic static analysis by the displacement method, stiffness and mass properties, used in the generation of gravity and inertia loads, are input either as properties of elements or as properties of grid points. In finite element analysis, constraint elements are very useful and convenient for specifying the special boundary condition. In the finite element model of the engine structure, there are several contact areas for example, cylinder head hole for bolt, etc. concerning multi-point constraints (MPCs).

Therefore, constraints are employed for the following purposes: (1) to specify the prescribed enforced displacements (2) to simulate the continuous behaviour of displacement in the interface area and (3) to enforce rest condition in the specified directions at grid points of reaction [1]. Because of the complexity of geometrical design and load path of the engine, it is not easy to model the complicated stiffness distribution of engine structure using simply analytical model. The engine needs to be modelled using finite element analysis tools, which can predict the load path that acting at the engine structure. Therefore, a 3-D solid model in figure 3 was developed for the FPE structure in order to predict the natural frequency of the engine in detail. A complete engine structure with specified boundary condition was used with 108062 four-sided solid elements with 4 to 10 grid points.

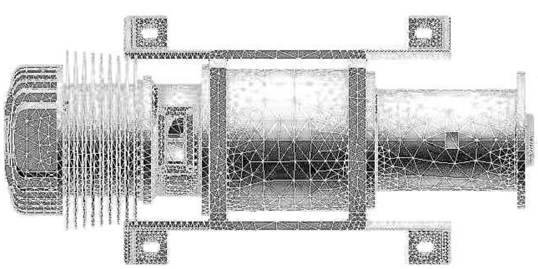


Figure 3 Finite element of FPE parasolid model

In normal mode analysis, there will be no loading configurations, which also known as a free-free situation. The MPCs were introduced to connect the parts thru the interface nodes between the adjacent parts. These MPCs were acting as an artificial bolt and nut that connect each parts of the engine structure. Each MPC used RBE elements that allow users to connect the rigid body to an arbitrary number of nodes. The independent degrees of freedom are the six components of motion at a single node and the dependent degrees of freedom are specified at an arbitrary number of nodes. From the viewpoint of machine design, the bolts were used to connect the cylinder head at its top end with the cylinder block structure at its bottom end. Therefore, to described this condition, the MPC was created at the bottom end of the cylinder head bolt hole. RBE element with six-degrees of freedom were assigned by creating an independent node from the head properties and dependent node created from cylinder block properties. Similar MPCs were also have been created in each bolt hole of the engine parts in order to simulate the displacement continuous response of corresponding node pairs for the normal modes analysis.

4. Engine Structure Vibration Analysis

The vibration analysis of the engine system was lead by the normal mode analysis using finite element software for mode shape and frequency solution. The output data from the normal mode analysis was then used for vibration and noise analysis of the engine structure using acoustic software. The data input for frequency and force response analysis were taken from previous study performed by [8] for combustion, kickback and friction force data. An additional forces data for bushing or bearing element is defined by two axis systems, one on each of the two bodies connected by this element. An axis system is created on each body in order to define the location and orientation of the force. Another additional input data is translational spring damping actuator (TSDA), which defines a spring-damper-actuator force element between two bodies. The combination of a value of stiffness, a value of damping, and a value of actuation torque define the TSDA force. The actuation force can be defined as a constant and/or as a function of simulation time.

These data were then used in motion analysis study to accommodate the inclusion of flexibility within bodies of the engine, with incorporation modal data obtained from the FE analysis. The technique is referred to as component mode synthesis or modal superposition. With this method, the flexibility of the body is represented by a set of flexible body modes. Normal modes are used to represent the natural vibration of the body and static modes are used to account for localized loading and deformation caused by coupling the body to other mechanism components. The input and output (IO) points of the engine were at the journal bearing. A load function set was created using fast fourier transformation (FFT) method to convert a force over frequency (0-5000Hz) results with higher tolerance (10^{-7}). The load function set manages the relationship between load data and input points. It consists of the load data (displacements, forces, etc) that need to be applied to the test models as well as the association of these data with the input points. The next step in acoustic analysis of the linear engine is to get its forced response function (FRF). Generally, the basic methodology uses an FRF model description of the structural or vibro-acoustical system, which relates a loading, or excitation vector $L(\omega)$ to the response vector $R(\omega)$ by an FRF matrix $[H(\omega)]$ is,

$$\{R(\omega)\} = [H(\omega)]\{L(\omega)\} \tag{11}$$

where, L (the load) can be force, displacement, velocity or acceleration.

There were two set of data imported in FRF analysis, the normal frequency (0-800Hz) from MSC/Nastran (SOL 103) using Modal-Based Forced Response Case and the load function from the previous motion analysis. The load function can be translated into frequency response using the FRF-Based Forced Response Case. In Modal-Based Forced Response Case, for a selected response degree of freedom (DOF) i, the equation for modal contribution analysis can be written as,

$$R_i(\omega) = \sum_{j=1}^{N} \psi_{ij} Q_j(\omega) \tag{12}$$

where, R_i is the structural displacement response at the i-th DOF, Q_j represents the modal participation factor of the j-th mode, ψ_{ij} is the mode shape value of the j-th mode at the i-th DOF and N stands for the number of modes. As for the FRF-Based Forced Response Case, for a selected response DOF i, this equation can be written as:

$$R_i(\omega) = \sum_{j=1}^{N} H_{ij}(\omega) L_j(\omega) \tag{13}$$

where, L represents the load at DOF j, and N the number of inputs, or paths. The result for the frequency response analysis is the combination between mode set and the load function set.

5 Pre-Acoustic and Acoustic Analysis

In this section, the results from FE dynamic response analysis are given as an input vibration displacement to acoustic analysis using the acoustic software. In order to perform an acoustic analysis, the acoustic meshes are needed, where this mesh can be derived from a structural mesh used for a vibration analysis. This will convert the finite element mesh to boundary element mesh of the engine structure as shown in figure 4.

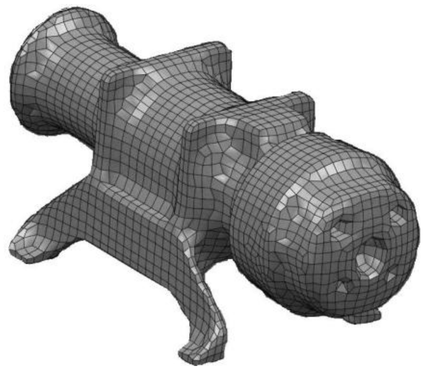


Figure 4 Boundary Element Mesh of the engine

Pre/Acoustics is enables to reduce the number of elements and remove all of the irrelevant features of a structural mesh, in order to perform acoustic analyses on a less-refined but smoother mesh. Creating an optimal mesh for acoustic radiation, starting from a structural mesh, must be performed in several steps. In boundary element acoustic calculation, there are several steps needed, in order to determine the overall effect of the engine structure to the noise as indicated below:

a. The component data were gathered and managed to identify data that characterize the discretized acoustic model. The model representation includes a model mesh (BE or FE mesh), acoustic properties and symmetry definitions. The possible properties are fluid properties (air), ISO field and symmetry planes.
b. Inserting an analysis case, which is the acoustic transfer vector (ATV) analysis case to compute the ATV of the engine. ATVs are input-output relation between the structural velocity on the boundary mesh and the pressure at some specific locations in the acoustic domain.
c. To post-process the defined data sets and analysis results. A mesh mapping set is used to transfer node based data from one mesh, which is the source mesh to another target mesh, such as a set of modes calculated on a fine structural mesh, that should be used on a coarser surface mesh in an acoustic calculation. Each node of the target acoustic mesh can be influenced by zero to N nodes of the source structural mesh. Each nodal value of an influencing node of the source mesh is weighted with a factor according to the chosen algorithm to obtain the corresponding nodal value on the target mesh.
d. The data transfer analysis cases performs the actual mapping of data (response, modal or transfer functions). From the source mesh onto the target mesh using the mapping data computed in a mesh-mapping object. The transfer vector set is designed to set up and manage transfer data vectors defining a model. Its basic parameters are the physical type of structural inputs, structural outputs and acoustic outputs. As a transfer vector defines a relationship between a physical quantity at an output location (i.e. the response) and a physical quantity at an input location (i.e. the load), transfer vectors are identified by a frequency value and a location that was identified by number. This location can be an input location or an output location. It is assumed that the frequency values of the transfer vectors are the same for all these locations.
e. Finally, the results of the acoustic response can be converted into modal ATV using the modal ATV response analysis case.

6 Results and Discussions

From the normal mode analysis, it was found that the output data from the journal bearing at combustion chamber and kickback chamber as illustrated in figure 5 and 6 respectively. It was indicated that at the combustion chamber bearing, the maximum eigenvector or amplitude occurred at frequency range of 300 to 800 Hz or the exact frequency value was at 558.47.

However, for the kickback chamber the maximum eigenvector occurred at frequency range of 950 to 1100 Hz or the exact frequency value was at 1056.62 Hz. These maximum eigenvector occurred at mode number 11 and 22 for combustion and kickback chamber of the engine as in figure 7(a) and (b) respectively.

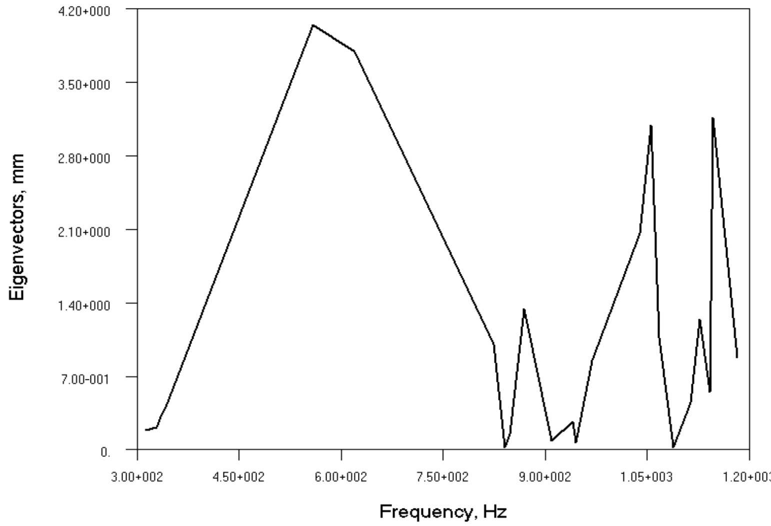


Figure 5 Graph Eigenvectors versus frequency for combustion chamber

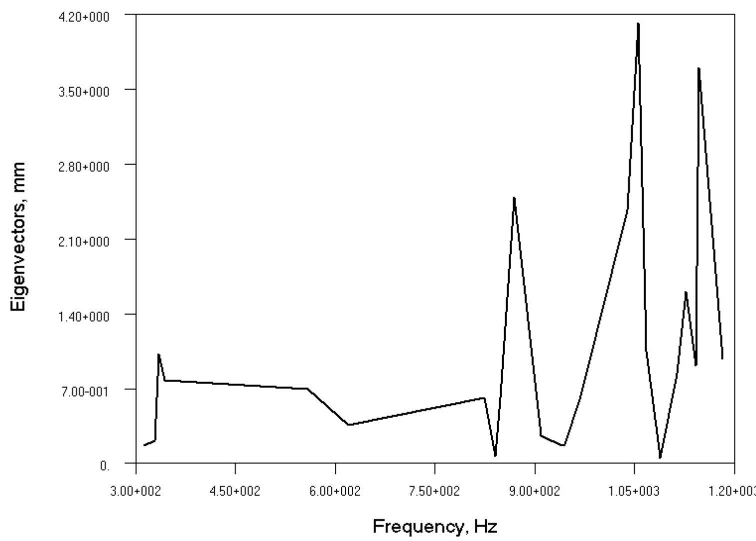
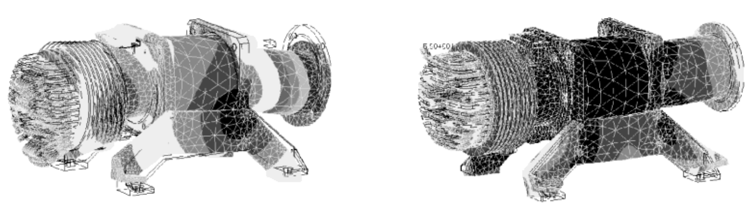


Figure 6 Graph Eigenvectors versus frequency for kickback chamber



(a) Combustion (b) Kickback

Figure 7 Engine mode shape deformation.

From the pre acoustic analysis, the FE meshes were converted to BE meshes in order to analyse the noise radiation from the engine structure. Typically, the noise radiation from an acoustic source is solved in a two-step process: first, the boundary problem is solved to obtained the so-called potentials, second the potentials are used to compute the noise radiated from the boundary in the acoustic domain. The mesh on which the boundary problem is solved is referred to as model mesh whereas mesh on which the acoustic radiation is computed is often referred to as Field Point Mesh or Visualization Mesh. In this analysis, ISO Power Field Point Mesh is used as shown in figure 8.

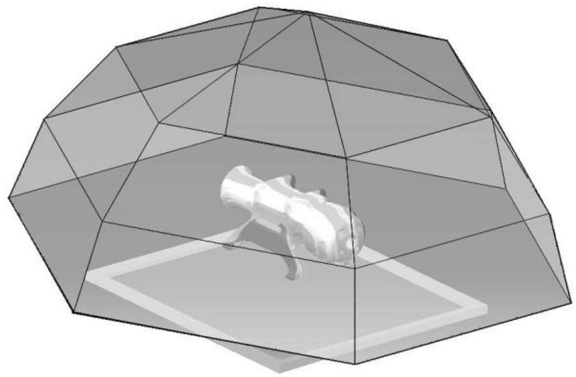


Figure 8 ISO Power Field Point Mesh

Results from the acoustic analysis power field consists the output graph for force response frequency was illustrated in figure 9 and for noise result which, were measured in pressure (dB) versus the frequency in hertz was illustrated in figure 10.

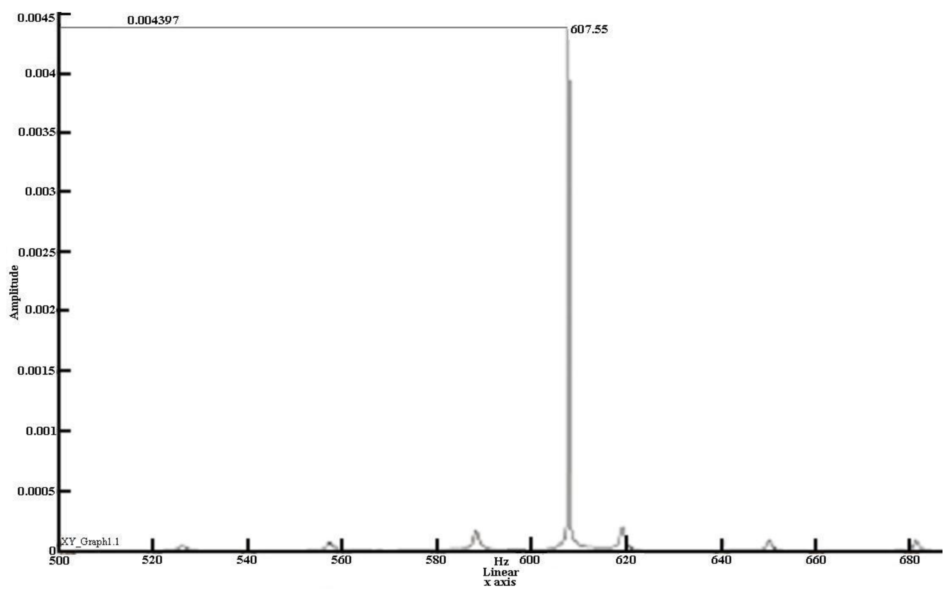


Figure 9 Force response frequency graph.

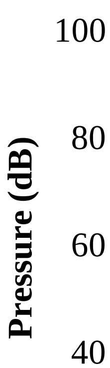


Figure 10. Graph Pressure (dB) versus Frequency (Hz).

From the frequency response analysis, the maximum amplitude for bearing occurred at frequency equal to 607.55 Hz. From the integration of normal mode and frequency response analysis using acoustic software, the maximum pressure of 98.775 dB occurred at working frequency equal to 62 Hz or at 3100 cycle per minute (cpm). For the engine to run on 1800 cpm or at 30 Hz, the engine noise was at 62.34 dB.

It is also found that the combustion bearing natural frequency has slightly lower results compared to the FRF graph in figure 9. This result was different for kickback chamber, where its natural frequency was larger than the FRF. The kickback chamber has additional force response frequency of 449.07 Hz from its natural frequency. This will create an unbalanced condition for the structure. The kickback chamber structure vibrate more because the chamber was only jointed with the mount and there is no other structure that can hold it at the end site of the chamber. Compared with the combustion chamber, even though the pressure in the chamber was higher than the kickback chamber, there was other structure, which is the cylinder head that can absorb some of the energy to reduce vibration. From frequency response graph in figure 9, it is indicate that the higher frequency of the engine was in the range of 500 – 2500 Hz, which was in the air-borne noise area. The pressure (dB) against frequency graph shows that the free piston linear generator engine structure has a fluctuation range of frequency where the noise range is from 45.75 dB until the highest value at 98.775 dB that occurred at 3100 cpm as shown in figure 10. This condition happen due to the unbalance condition of the structure which need to be redesign with better mounting and structure that can stand the excitation forces. Better boundary condition with additional fluid interaction might improve the simulation data of the system.

7 Conclusions

This study have indicated that for the new free piston engine design has a frequency response range from 500 – 680 Hz with the highest frequency at 607.55 Hz. The operational frequencies of the engine were at range 10 – 124 Hz with the maximum pressure (dB) for noise is at 98.775 at speed 3100cpm. This data will be used as the initial data for design improvement of the engine.

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