Vibration Assessment of Diesel Engine Genset Mounts

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ABSTRACT

With the expanding use of diesel engine generator (DG) sets for emergency standby power, peak saving and demand response, there has come an increased focus on monitoring the vibration levels & thus the noise radiated by these generators. Vibration monitoring, testing, and experimentation are important in the design, implementation, maintenance, and repair of generator sets. Before even designing or controlling DG Set for good vibratory performance, it is important to understand, represent (model), and analyse the vibratory characteristics of the system. This has been accomplished here through computer analysis of Genset analytical models, testing and analysis of test data.

In this work, numerical simulation of Diesel Engine Generator (DG) set was carried out using finite element method (FEM). Engine & Alternator Mountings were analysed with the help of Frequency Response Functions (FRF) approach called Point Mobility (PM) Analysis. Local dynamic stiffness at mountings was studied to understand how the excitation forces from engine & alternator mounts propagates to rest of the generator structure. In addition to this, Normal Modes Analysis (Modal Analysis) was also carried to locate the potential modes & natural frequencies of DG set which are closer to the engine harmonics as these may get combined with excitation frequency of engine which may further lead to failure of the DG set structure. Finally, generator structure was validated using experimental data.

Commercial software codes used for simulation were NASTRAN from MSC Software and FEA modeling was done using Altair's Hyperworks.

Keywords***: Vibration, Frequency Response Functions (FRF), Point Mobility (PM), Normal Modes Analysis, Modal Analysis, Engine Mounts, DG Set, Genset.***

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1. INTRODUCTION

Mechanical vibrations and shock are present, to varying degrees, in virtually all locations where equipment and people function. Vibrations can naturally occur in an engineering system and may be representative of its free and natural dynamic behaviour. Also, vibrations may be forced onto a system through some form of excitation. The excitation forces may be either generated internally within the dynamic system, or transmitted to the system through an external source. Some vibrations are desirable, while others are not. When the frequency of the forcing excitation coincides with that of the natural motion, the system will respond more vigorously with increased amplitude. This condition is known as resonance, and the associated frequency is called the resonant frequency. Excessive vibrations in equipment can not only damage the equipment itself but also decrease functionality [1].

Engine driven generator sets produce vibrations, just as most machinery with moving and rotating parts. Vibrating generator components induce pressure waves as sound into the environment. Also, anything that is attached to the generator set can cause vibrations to be transmitted into the building structure or foundation. These attachment points include skid anchors, radiator discharge air ducts, exhaust piping, coolant piping, fuel lines, electrical conduit and basically all these comes from reciprocating engine and its attachments (Mounts) [3].

In the case of generator sets, excessive vibration can cause disruption of power delivery hence proper design and control are crucial in maintaining a high-performance level and production efficiency, and prolonging the useful life of diesel engine generator set. Before even designing or controlling generator for good vibratory performance, it is important to understand and analyse the vibratory characteristics of the system. The transmission of these vibrations should be minimized to avoid nuisance from noise and vibration, as well as physical damage to the generator set itself and the structure supporting the generator set. More importantly, reducing vibration helps the generator set to achieve its main purpose, which is to produce reliable electric power [8].

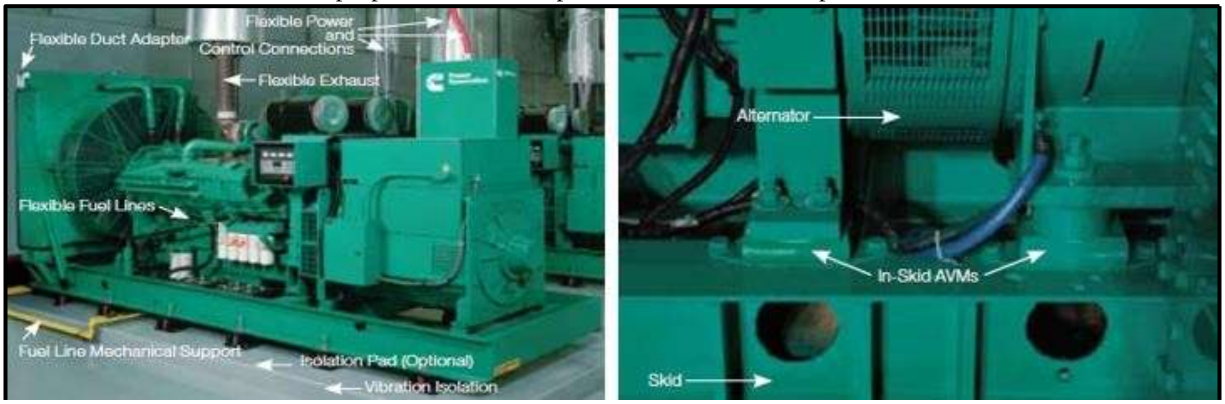

Fig-1: A Typical Installation of DG Set with Vibration Control Measures

2. THEORITCAL BACKGROUND

2.1 Normal Modes Analysis

The usual first step in performing a dynamic analysis is determining the natural frequencies and mode shapes of the structure with damping neglected. These results characterize the basic dynamic behaviour of the structure and are an indication of how the structure will respond to dynamic loading. The natural frequencies of a structure are the frequencies at which the structure naturally tends to vibrate if it is subjected to a disturbance. The deformed shape of the structure at a specific natural frequency of vibration is termed its normal mode of vibration. Each mode shape is associated with a specific natural frequency. Natural frequencies and mode shapes are functions of the structural properties and boundary conditions. The solution of the equation of motion for natural frequencies and normal modes requires a special reduced form of the equation of motion. If there is no damping and no applied loading, the equation of motion in matrix form reduces to

$$[M]\{\ddot{u}\} + [K]\{u\} = 0 \quad (1)$$

Where, [M] = Mass Matrix, [K] = Stiffness matrix, [u] = Displacement, [ü] = Acceleration
This is the equation of motion for undamped free vibration. To solve Eq. (1) assume a harmonic solution of the form

$$\{u\} = \{\varphi\} \sin \omega t \quad (2)$$

Where, {u} = Eigenvector or Mode shape, {φ} = Circular Natural Frequency

Aside from this harmonic form being the key to the numerical solution of the problem, this form also has a physical importance. The harmonic form of the solution means that all the degrees-of-freedom of the vibrating structure move in a synchronous manner. The structural configuration does not change its basic shape during motion; only its amplitude changes. If differentiation of the assumed harmonic solution is performed and substituted into the equation of motion, the following is obtained:

$$-\omega^2[M]\{\varphi\} \sin \omega t + [K]\{\varphi\} \sin \omega t = 0 \quad (3)$$

which after simplifying becomes

$$([K] - \omega^2[M])\{\varphi\} = 0 \quad (4)$$

This equation is called the eigen equation, which is a set of homogeneous algebraic equations for the components of the eigenvector and forms the basis for the eigenvalue problem [6].

2.2 Point Mobility (PM) Analysis

Frequency Response Function (FRF) is a frequency dependent function that indicates the response of a mechanical system to dynamic excitations. Implementing this method facilitates the tracking of load paths or energy flows through a structure, commonly known as Transfer Path Analysis (TPA). The equation of motion for frequency response analysis can be written as equation (5).

$$(-\omega^2[M] + i\omega\,[C] + [K])\{u\} = [Z]\,\{u\} = \{F\} \qquad (5)$$

Where, [Z] is the dynamic stiffness matrix of the system. Solving the equation (5) for the response displacements, {u}, results in relation (6) as follows.

$$\{u\} = [Z]^{-1}\{F\} = [H]\{F\} \qquad (6)$$

Matrix [H] in equation (6) is the receptance matrix of the system. In general, it is identified as the FRF of system response. It is noteworthy to mention that FRF response of a system is a function of the applied force frequency, ω. Thus, different FRF values are obtained for a system in response to forces with different frequencies. Rows of an FRF matrix illustrate response degrees of freedom (DOFs), while its columns represent excitation dimensions. Term Hij of the FRF matrix indicates the displacement in the i-th DOF due to a unit load imposed to the j-th DOF. Therefore, the rows in the j-th column of the FRF, demonstrates the displacements at all response DOFs due to a unit load applied to the j-th DOF. Correspondingly, the columns in the i-th row of the FRF represent the displacements at the i-th DOF due to unit loads applied to all excitation DOFs. In practice, the FRF matrix is computed by solving equation (6) for a series of unit loads. Equation (7) represents equation (6) for a series of unit loads on the right-hand side [5].

$$[U] = [Z]^{-1}[I] \qquad (7)$$

The solution of the equation (7) for the matrix of displacements gives the preferred FRF matrix, [H]. Matrix of displacements, [U], which is resulted from equation (7) is equivalent to the FRF matrix or receptance represented in equation (6). Assuming that excitation is introduced in all DOFs and the responses are calculated for all DOFs as well, it is inferred that the FRF matrix is a square. However, in most practical cases, considering the raised problem specific points are candidate for applying the force, while the response values are calculated for a defined set of locations, usually of specific interest. Hence, the practical FRF matrix does not form a square generally and is considered as a subset of the matrix [U] resulted from equation (7) in the form of a rectangle that can be regarded as an (r × c) matrix where r stands for the number of actual response DOFs and c represents the number of actual excitation DOFs [5].

Point Mobility (PM) is nothing but the type of Frequency Response Function (FRF) only. It is the complex ratio of velocity and force taken at the same point in a mechanical system during simple harmonic motion. Here point means both a location and direction. Rather in simple words if the excitation is given at a point and response is also taken at the same point then resulting transfer function is called as Drive Point Mobility or Point Mobility. The Mobility is Frequency dependent and it gives the magnitude and phase relationships which exist between the excitation and vibration response. Point Mobility (PM) is different from FRF in the sense that, all the resonances in PM functions are separated by anti-resonances [7].

3. POINT MOBILITY (PM) MEASUREMENTS

Measurement of FRFs and analysis of it in various ways produces results that give Point Mobility (PM) and many other dynamic properties of structure. These measurements are popularly called as Experimental Modal Testing. In all the measurement methods, modes are efficiently excited and the response is measured at several points on the structure. The time histories so obtained are then Fourier transformed to obtain frequency domain representation. Since the phase relation is also involved hence both cross and auto spectrums are obtained. These are used to estimate frequency response functions. The numerator in below equation is a cross-spectrum and the denominator is an auto-spectrum. For estimating FRFs one needs knowledge of Fourier transformed response and the forcing function [2].

$$H(\omega) = \frac{U(\omega)F(\omega)^*}{F(\omega)F(\omega)^*}$$

The measurements of responses and forces are in time domain. The time domain signals are obtained using outputs from appropriate transducers. For this purpose, generally a dedicated FFT analyser (older concept) is used or a data acquisition system with software that performs the FFT is used. The signal from transducer is in an analog form. These analog signals pass through anti-aliasing filters to remove high frequency components. Then the signals are digitized using analog to digital converters (ADC). The digitization process decides the resolution and dynamic range. Next step is the data conversion to frequency domain by Fast Fourier Transform (FFT).

If sample data is not periodic, there can be a leakage that is some spurious components occur at some or all frequencies. The Weighting reduces the amplitude of FT coefficients, but corrections can be applied so that amplitudes revert back to original values. Some popular weighting functions are hanning, exponentials etc. Averaging is the performed to remove any random noise from the measured signals. Now the cross & auto spectrums required for the analysis can be estimated and ultimately the frequency response functions (FRF). The measurements hardware consists of four components: i) The mounting system, ii) Excitation mechanism, iii) Measurement transducers for force & response and iv) data acquisition system. Also, there is a requirement of analysis tools in the form of specialized software [2].

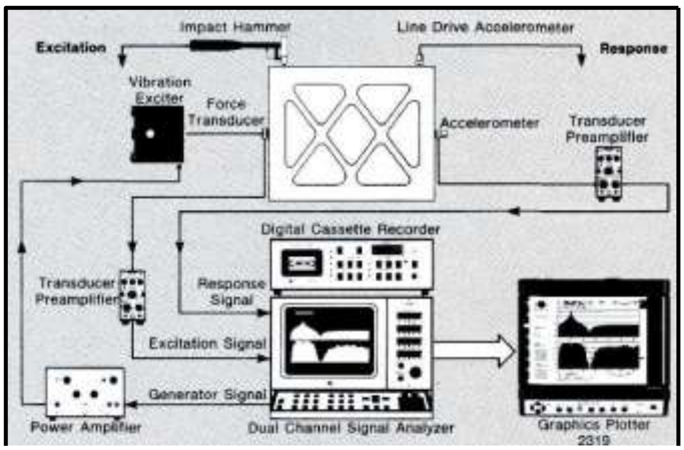

Fig-2: A Typical Setup for Point Mobility (PM) Measurements

4. FINITE ELEMENT MODELING

Any continuous object has infinite degrees of freedom and it is not possible to solve the problem in this format. In such case Finite Element Method (FEM) can be effectively used to address the problem. The Finite Element Method reduces the degrees of freedom from infinite to finite with the help of discretization or meshing (nodes and elements). The Finite Element Method only makes calculations at a limited (Finite) number of points and then interpolates the results for the entire domain (surface or volume). Solving at limited number of points makes FEM method much faster and the accuracy of the results can be increased by increasing element or Node count [2].

Altair's Hypermesh was used for the pre-processing as well as post-processing of the generator set CAD model. First, the geometric clean-up was performed. Genset components were mostly modelled with shell elements and attachments (Mounts) were modelled by 3D elements as they have significant thickness. Components which do not add significant stiffness to generator structure were modelled with point masses such as Engine, Alternator, radiator etc. Material properties were assumed to be isotropic and standard mesh quality criteria was followed for meshing as the accuracy of the results is directly proportional to mesh quality. Then this same model was used for both modal as well as FRF analysis. Modal Analysis (SOL 103) and Frequency Response Analysis (SOL 111) was carried out in MSC Nastran commercial software. Experimental Modal Testing was done by third party testing laboratory. All the results have been discussed here along with comparisons.

5. RESULT & DISCUSSION

Results for various dynamic analysis have been presented here with appropriate reasoning.

5.1 Normal Modes Analysis

Normal Modes Analysis was performed till 300 Hz as it covers more than 10 harmonics of engine. Lanczo's method was used to compute Normal Modes as it is fairly accurate & have performance advantage over other methods.

	Engine Operating Frequency = (Operating Speed/60) ***Engine Operating Speed = 1500 RPM**									
Harmonics	1^{st}	2^{nd}	3^{rd}	4^{th}	5^{th}	6^{th}	7^{th}	8^{th}	9^{th}	10^{th}
Frequency(Hz)	25	50	75	100	125	150	175	200	225	250

Table-1: Engine Harmonics

Following (refer table 2) are some the modes which were observed to be closer to Engine Harmonics. These may coincide with operating frequency of engine and amplify the response of structure. Although other modes will also have the contribution to the response but that will be lower compare to below modes. Hence these modes should be moved away from the engine harmonics in order to avoid any failure to the generator structure.

Mode No.	70	196	332	479	612	811	971	1150
Frequency(Hz)	25.0	50.80	75.64	100.6	125.1	150.0	174.9	200.4

Table-2: Modes Close to Engine Harmonics

5.2 Point Mobility (PM) Analysis

Engine Mount & Alternator Mounts were given frequency dependent excitation (Unit Load) in Vertical axis & response in terms of acceleration was taken at same location for same DOF. Frequency for the analysis was kept up to 250Hz. It was observed that PMs for the engine & alternator mount 1 were almost same and similar was the case for engine & alternator mount 2, hence only two plots are discussed here. It was also seen that Point Mobilities for Engine & Alternator Mount 2 were on higher side compare to Mounts 1 in few frequency bands. That also means that mounts 2 have lower local dynamic stiffness and can transfer more forces to main structure.

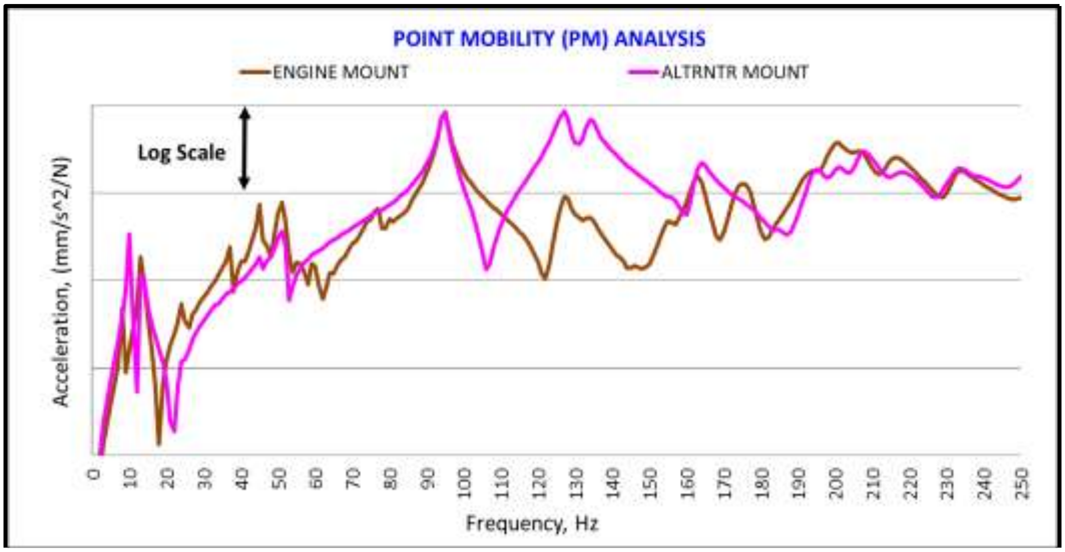

Fig-3: Engine & Alternator Mount Point Mobility (PM)

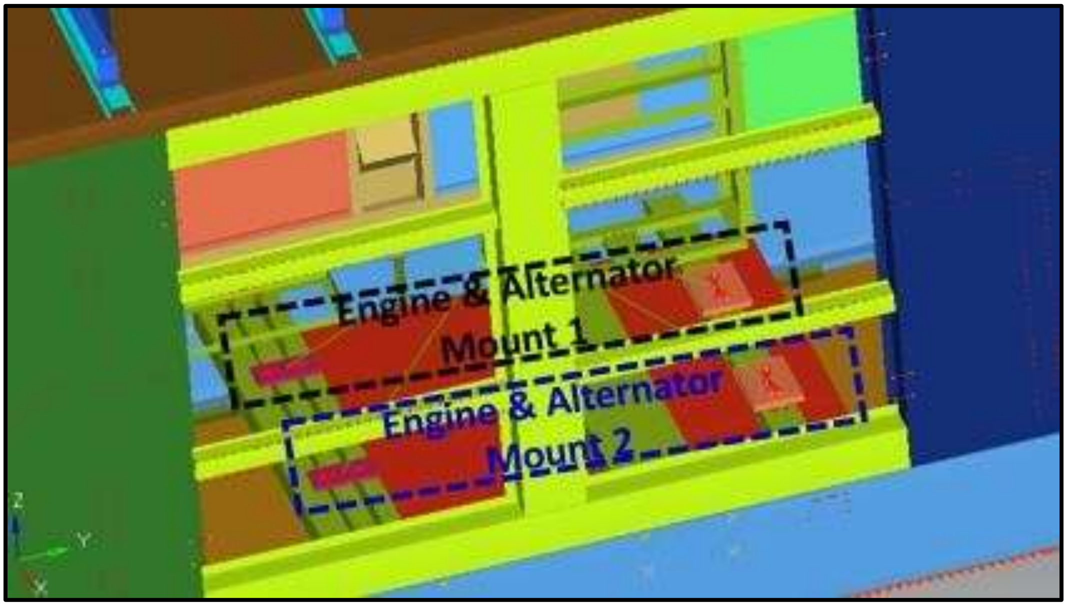

Fig-4: Engine & Alternator Mount Nomenclature

Following is the Drive Point Mobility (PM) comparison of Measured results Verses Predicted for various engine & alternator mountings. Simulation (predicted) results showed good agreement with experimental results with less than 10% of error in most of the frequency range.

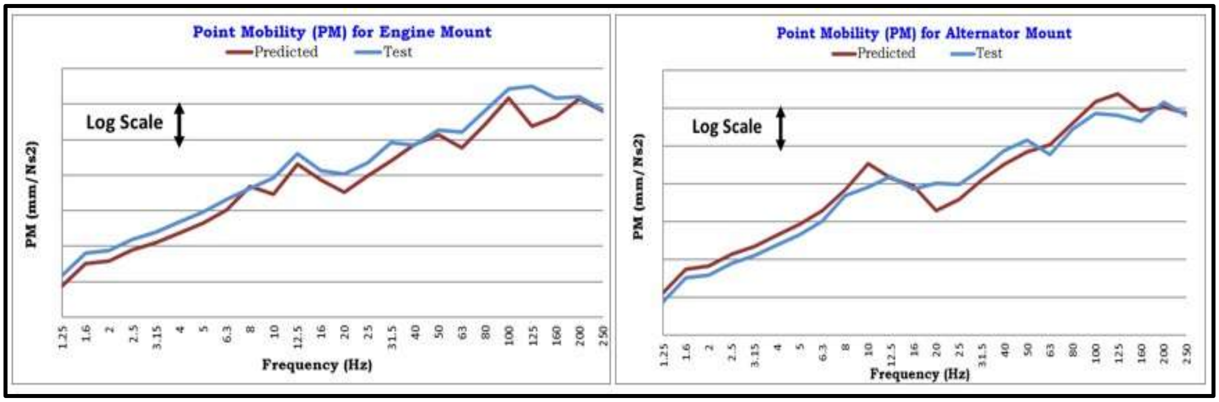

Fig-5: Point Mobility (PM) Validation

6. CONCLUSION

In this paper, a Diesel Engine Generator set is successfully analysed with the help of Finite Element numerical method (FEM). Generator set is assessed from vibration perspective using the Point Mobility (PM)/ Frequency Response Function (FRF) techniques at Engine & Alternator Mounts. It is observed that PMs or Local dynamic stiffness for engine & alternator mounts 1 is almost same and similar is the case with mounts 2. It is also seen that the PMs for mounts 2 are higher in few frequency bands compare to mounts 1 which implies that mount 2 will transfer more dynamic forces to generator structure compare to mounts 1 in those particular frequency bands when excited with same forcing function. In rest of the frequencies response is comparable.

Few modes are observed to be closer to the engine harmonics and these may amplify the vibration response of generator set. Hence these are potential modes and should be moved away from operating frequencies of engine in order to avoid any failure to generator structure.

ACKNOWLEDGEMENT

The authors are thankful to the SCOE and SKNCOE, Pune for giving the opportunity to present the paper and for providing continues support throughout the work.

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