"Analysis Of Anti Vibration Mounts For Vibration Isolation In Diesel Engine Generator Set"

V.S.CHAVAN[1], R.ASKHEDKAR[2],S.B.SANAP[3]
[1]P.G student, Mechanical Engineering Department.Sinhgad College of Engineering,Pune, University of Pune,India.
[2]General Manager, Kirloskar Oil Engines Ltd Pune, India.
[3]Assistant Professor, Mechanical Engineering Department.Sinhgad College of Engineering, Pune, University of Pune, India.

Abstract

Engine vibrations have been one of the major problems for the engine manufacturers in the world. The engine excitation forces, arising from the gas pressure and unbalance forces are the sources of vibrations.Anti Vibration Mounts(AVMs) are the structures used to absorb the vibrations and dampen the harm causing forces. Mount result in a system which is modeled as mass/spring/damper. The elastomer is under vertical and shear load at the same time. The engine is bolted freely on mount which dampens the vibrations going to the base frame/canopy. The validation of their characteristics is implemented in a single degree of freedom system with response to frequency domain. The engine is mounted freely on the AVM's which helps to dampen the vibrations to base frame. Frequency Response Function(FRF) is derived for analysis, analytically and Computer Aided Engineering(CAE). Comparative is also made to suggest which material is to be selected for AVMs. A single AVM is analyzed for modal and frequency response with the upper plate as a part of engine structure and the bottom plate as the base frame. Comparative study of the two conditions using materials and also without mount and with mount is analyzed.

Keywords*:* Anti-Vibration Mounts (AVMs), CAE,Elastomers,Engine,FRF,Vibrations.

I. Introduction

Engine vibrations have been one of the major problems for the engine manufacturers in the world. In a competitive world market noise and vibration reduction ranks high where the customers perception is to have overall quality of the system influenced by how quiet and smooth running the system operates[1]. The situation aggravates if the operational frequencies of the engine lies near the natural frequencies of the engine [11]. AVMs play a major role to reduce the overall vibrations from the engine to base frame structure.To reduce the overall vibrations of the engine towards base and the canopy structure of the power generating set, Anti-Vibration Mounts (AVM's) are used.AVM consist of the rubber/elastomer sandwiched between the two metal cover plates.The elastomer used leads to development of non-linear behavior of the mount[4][7].The ability of an elastomer to convert energy of motion allows it to absorb vibration. Its viscouscomponent is most useful for absorbing low frequency loadssuch as a single large impact[8]. Alternatively, the elastic component can return the elastomericcomponent quickly to its original state, ready for the next cycle of deformation, and thus it can copewith low to higher frequency loads [9].

II. Methodology

The stages involved for the analysis of the condition starts with the calculation of natural frequencies of the system and the mounts associated to those. The mounts are selected dependent on the results and the load deflection curves. Whether mounts are capable to attenuated the vibrations is based on the percentage isolation which is obtained by mount material selected.The analytical results are obtained for selected mounts with the Voigt model which consists of the spring and damper system.Nonlinear static analysis is done in ANSYS 12.0 to get the modal analysis, stress and displacements values.The FRF and dynamic stiffness graphs are obtained to analyse for material selection. The pre-processing work is done in HYPERMESH to get the desired mesh pattern. The mounts consist of elastomer so the processingfrequency analysis of the mount goes in nonlinear phase carried out in NASTRAN.The FRF analysis of the mount is done which is compared with that of the simply bolted engine structure and base frame and the advantages are stated. The CAE result validation is done with the analytical, experimental testing for the vibrational testing using sensors and the results are stated.

III. Analytical Method

The mounting system effectiveness is commonlymeasured as the Transmissibility. The inertia forcedeveloped in a reciprocating engine or unbalancedforces produced in any other rotating machineryshould be isolated from the foundationso that the adjoining structure is not set into heavyvibrations. Transmissibility is the amount of enginevibration force which is transmitted through themounting system to the vehicle structure as apercentage. A transmissibility of 0.4 or less ofengine idle speed is necessary for a good mountingsystem. In the region of attention rather thanreferring to the transmissibility, we use theisolation efficiency as a measure reduction ofvibration input usually as a percentage valueoccurring for a particular disturbing frequency. Table 1 shows the percentage isolation values for different damping ratio and frequency ratio for selection of mount materials.

Table 1. Percentage Isolation for different damping ratio and frequency ratio.

Frequency ratio	0	1.5	2	2.5	3	3.5	4	4.5	5
Damping Ratio	% Isolation								
0	0	20	66.66667	80.95238	87.5	91.11111	93.33333	94.80519	95.83333
0.05	0	19.68123	66.08183	80.38839	86.95878	90.58695	92.82233	94.30501	95.34254
0.1	0	18.78383	64.41383	78.80004	85.46345	89.17067	91.47462	93.01873	94.11255
0.15	0	17.45891	61.88188	76.42977	83.28839	87.16689	89.6195	91.29396	92.50306
0.2	0	15.8922	58.75385	73.53835	80.6904	84.82404	87.49231	89.3497	90.71523
0.3	0	12.65538	51.65577	66.98263	74.88871	79.67603	82.88436	85.18792	86.92559

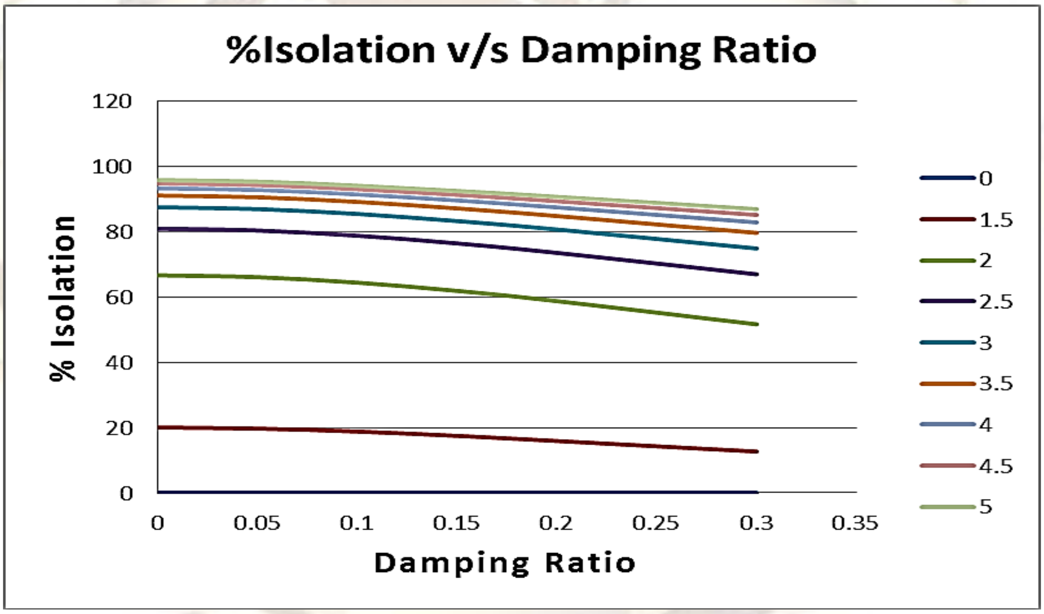

Fig.1 Isolation against the Damping ratio.

The analytical calculations are done for the rubber and neoprene is done and the values obtained for those are as follows:

Table.2 Comparison of Rubber and Neoprene parameters.

Material	Stiffness(N/mm)	Damping Ratio	Natural Frequency(Hz)	Frequency Ratio	%Isolation(graph)
Rubber	693.13	0.03	18	2.77	84.813
Neoprene	722.022	0.05	10.08	4.63	95.316

The above shown Fig.1 is plotted from the values derived in the Table no.1.The transmissibility graph shows the desired isolation zone for the frequency ration higher than $\sqrt{2}$.From the above analytical calculations and the graphs plot from them the stiffness, damping ratio, frequency ratio and percentage isolation delivered by the rubber material is less than neoprene material which is used in the mounts.

IV. Frequency Response Analysis

The frequency response function and dynamic stiffness of engine are calculated analytically for a range of frequency (0 Hz to 100 Hz) and shown in Fig.2 and Fig.3. In case of admittance, accelerancethe rubber and neoprene where the graph are compared.

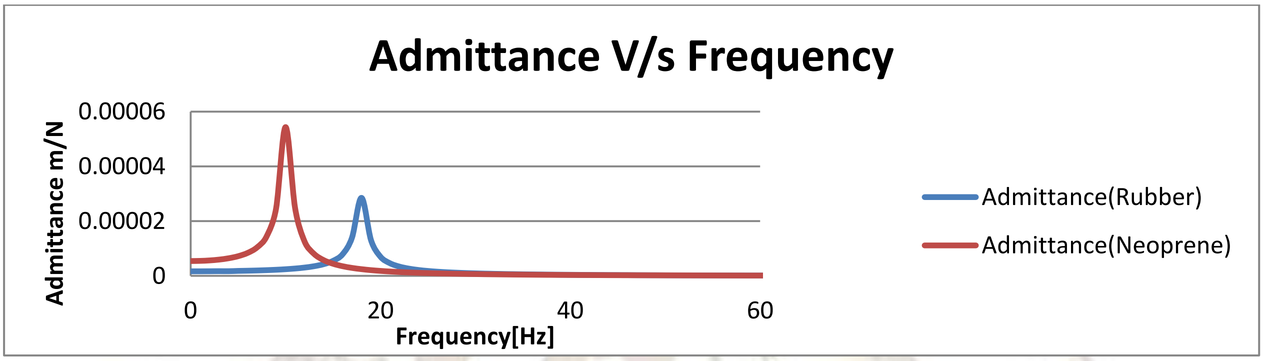

Fig.2 Calculated admittance for both mounts.

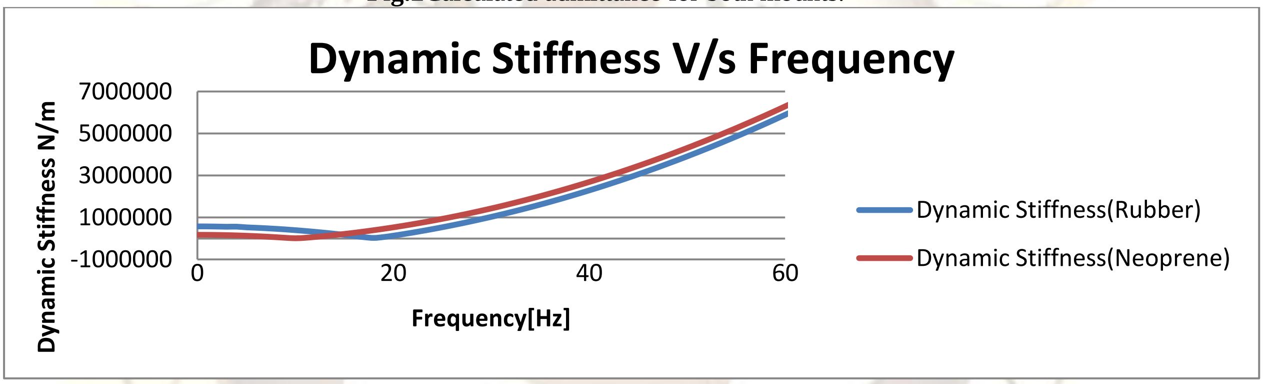

Fig.3 Calculated Dynamic Stiffness for both mounts.

The figure shows that the dynamic stiffness is frequency dependent. The dynamic stiffness is increasing when the frequency increased after the resonance of 40 Hz. For the excitation frequency before resonance, the dynamic stiffness decreased while the excitation frequency increased.

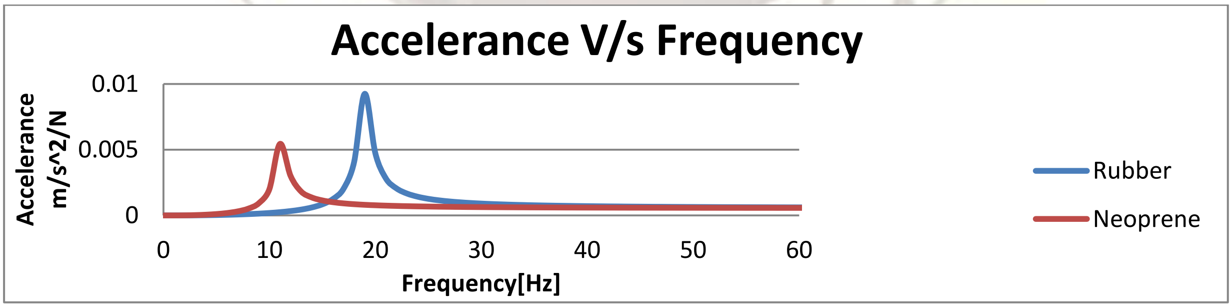

Fig.4 Calculated Accelerance for both mounts.

The minimum dynamic stiffness occurred at natural frequency of the system where small applied force resulted in large deformation. Similar trend was found where the measurement on an isolator is done by resonant method. The analytical calculation is done up to 60 Hz. Minimum dynamic stiffness is found at resonant frequency and the values of dynamic stiffness increased when the frequency increases above the natural frequency. The response of the mount in case of the accelerence is calculated in Fig.4 with range of 0Hz to 60 Hz for rubber and neoprene mount. The neoprene shows the higher response at 10Hz approximate and rubber at 18Hz, comparative rubber shows the higher acceleration response than neoprene.

V. CAE Results

The CAE results are compared for frequency response where the unit load applied on the upper surface of the mount and the response obtained on the bottom plate as shown in the Fig.5 .The FRF in terms of accelerance and admittance for the mount with rubber, neoprene and without mount conditions are compared in Fig.6 and Fig.7.

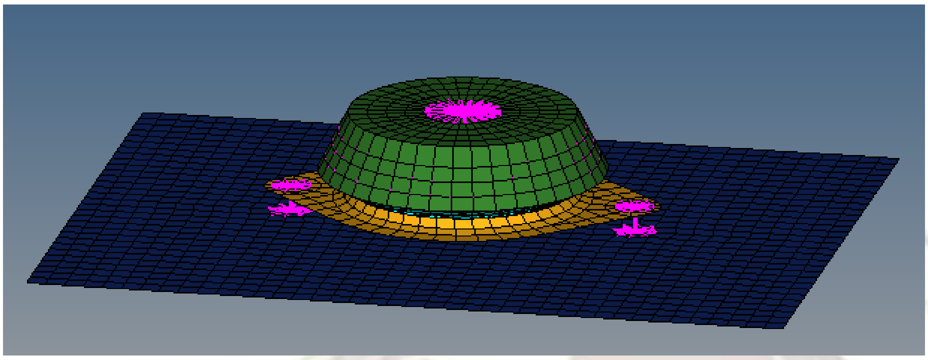

Fig.5 Meshed mount with bolt simulated on plate**.**

The peaks are basically observed where the frequency matches the natural frequency of these mounts. Fig.6 shows peak in the without mount and the rubber mount are observed near the operating condition of engine whereas that of the neoprene mount is far from it.

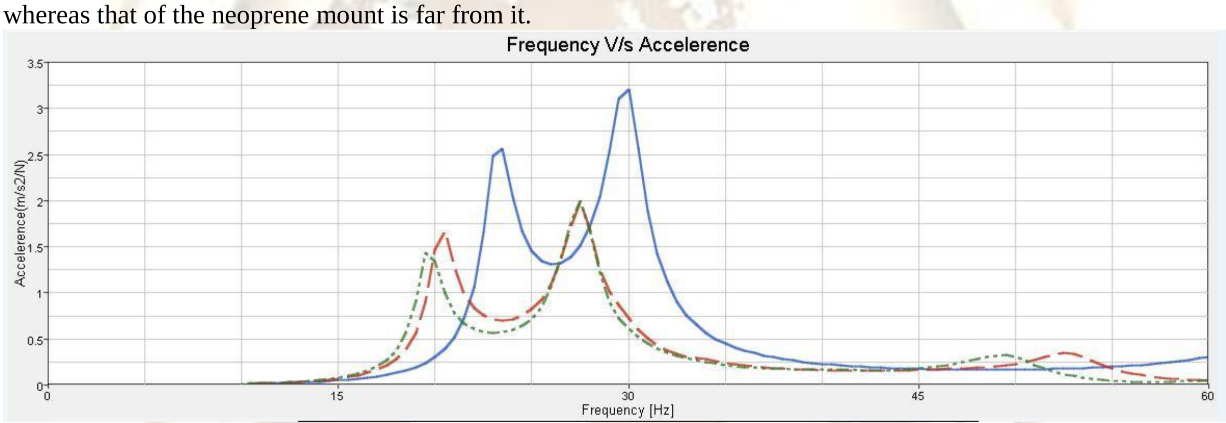

Fig.6 Simulation results ofaccelerance (-without mount, - rubber, -neoprene).

There the aceelerance at the bottom plate are reported with the peak for without mount conditions within the operation conditions which is basically high vibration transfer whereas for the rubber mount the peaks gets shifted to avoid the system resonance. The neoprene mount have the least level of accelerance in the operating range and also have achieved in shifting the peaks.

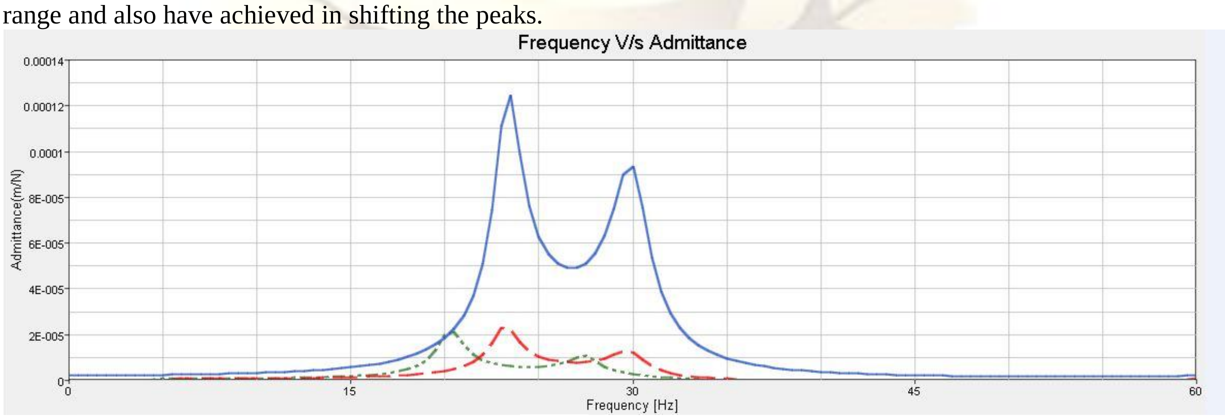

Fig.7 Simulation results of admittance (-without mount, - rubber, -neoprene).

The response in terms of displacement (admittance) is also shown in fig. where the plate without mount shows a higher displacement in the range 22Hz to 30Hz, compared to those with neoprene and rubber mounts. The shows comparatively the least displacement at those particular frequencies ranges.

VI. Experimental Results

The analytical and CAE results have shown that the neoprene material mounts comparatively shows the best suitable characteristics for the vibration absorbing system. Here the selected Rubber and Neoprene material mounts are tested on the actual engine and the results are shown with actual mount position shown in Fig.8where measure peak was observed at operating frequency of 25 Hz.

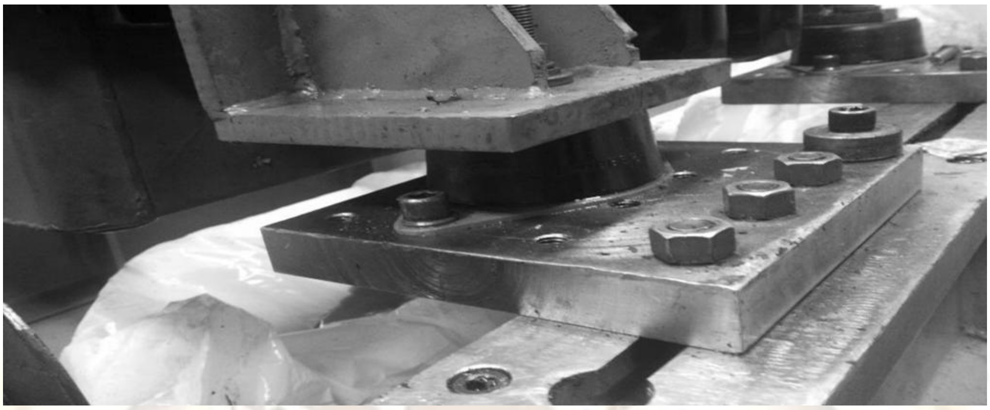

Fig.8 Actual location of mount at engine base.

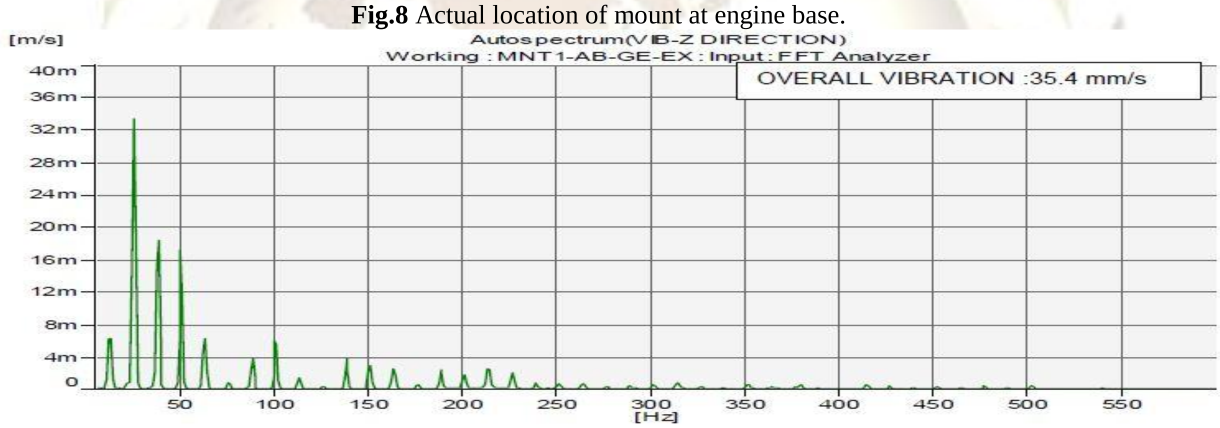

Fig.9 Experimental autospectrum results at base of engine without mount condition.

Fig.9 shows the autospectrum results of the engine base where the mounts are not used and the vibration levels are measured with the overall level of 35.4 mm/s which is much high as per the industrial standards.

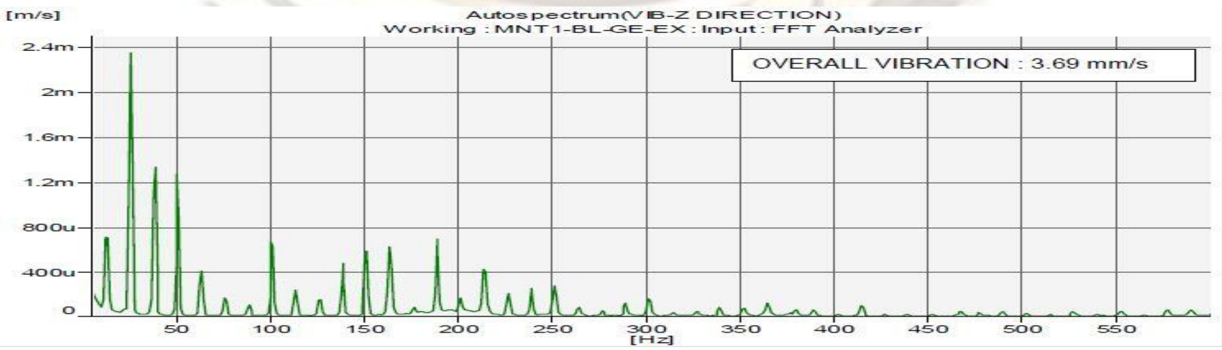

Fig.10 Experimental autospectrum results at base of engine below mounts (rubber mounts).

Fig.10 shows the vibration levels with the rubber mount used and the overall vibrations reduced to 3.69 mm/s which is about 89.72% which fall well within the industrial standards.

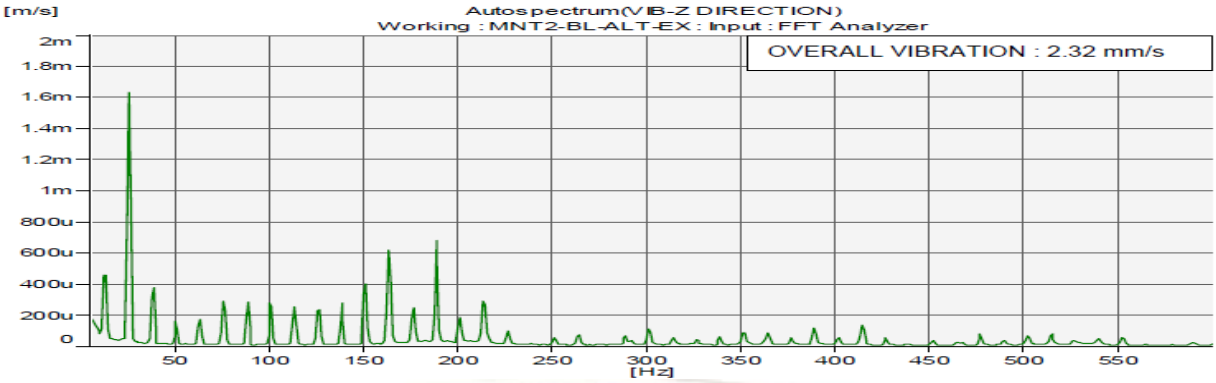

Fig.11 Experimental autospectrum results at base of engine below mounts (neoprene mounts).

The Fig.11 shows the experimental results with the neoprene material mount used and the overall vibration level measured were 2.69 mm/s which is 93.44% which very well matches with the analytical calculations made in the section 3.

VII. Conclusion

Analytical calculations are made with the mount as single degree of freedom mass spring damper system and the frequency responses are obtained. A comparative analysis for the selection of material is done where rubber and neoprene were suitable with the higher percentage isolation obtained. The results ofFRF from analytical and CAE match well and the neoprene shows that the neoprene material gives out the higher vibration isolation. The dynamic stiffness is frequency dependent where it is increasing when the frequency increased after the resonance. For the excitation frequency before resonance, the dynamic stiffness decreased while the excitation frequency increased. The minimum dynamic stiffness occurred at natural frequency of the system where small applied force resulted in large deformation.The dynamic conditions stated at the bottom of engine without mount and with mount compared shows that mounts plays a vital role in absorption of engine vibrations towards the base frame. The deviation in total isolation values in analytical and experimental is of 1.8%.

References

[1] Schmitt.R.V. andLeingang.C.J ,Design of Elastomeric Vibration Isolation Mounting Systems for Internal Combustion Engines, Society of Automotive Engineers,Technical Paper ,760431.

[2] Raymond L.Straw,The Development of Isolation Mounts, Society of Automotive Engineers, Technical Paper, 840781.

[3] Ramachandran. T, Padamanabhan K. P, Nesamani. P,Modelling and Analysis of IC Engine Rubber Mounts using FEM and RSM, Journal ofProceedia Engineering 38 (2012),1683-1692.

[4] CarellaA,Non-linear identifications using Transmissibility: Dynamic Characterization of Anti Vibration Mounts with standard approaches and Nonlinear Analysis, Journal of Mechanical Science 63(2012),74-85.

[5] Chandra PrakashMishra,Vinay Kumar Singh,AbhishekPyasi,,A Comparative Study of Engine Mounting System for NVH Imporovement,Society of Automotive Engineers, Technical Paper, 870797.

[6] Hamid Mir, “Focused 4-Mount Concept Evaluation”, Technical report, NVH Developmentand Engineering, DaimlerChrysler Corporation, 2001.

[7] Andrea DallAsta, Laura Ragni, Dynamic Systems With High Damping Rubber: Nonlinear Behavior And Linear Approximation,10th World Conference on Seismic Isolation, Energy Dissipation and Active VibrationsControl of Structures, Istanbul, Turkey, May 28-31, 2007

[8] LuEanOoi, ZaidiMohdRipin,Dynamic Stiffness and Loss Factor Measurement of Engine Rubber Mount by Impact Test, Journal of Materials and Design 32 (2011), 1880–1887.

[9] Tian Ran Lin, Nabil H. Farag, Jie Pan, Evaluation of frequency dependent rubber mount stiffness and damping by impact test, Journal of Applied Acoustics 66 (2005), 829–844.

[10] J. S. Tao, G. R. Liu, K. Y. Lam, Design Optimization of Marine Engine–MountSystem, Journal of Sound and Vibration 235(3) (2000), 477-494.

[11] L. Barelli, G. Bidini, C. Buratti, R. Mariani, Diagnosis of Internal Combustion Engine Through Vibration And Acoustic Pressure Non - Intrusive Measurements, Journal Applied Thermal Engineering (2008), 8-9 .

[12] C.Natarajan, P.K.Jayadev, Study Of Diesel Engine Idle Vibration Variation Analysis, International Journal of Communications and Engineering Volume 05– No.5, 136-140.