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HEAVY DUTY ENGINE MOUNT ADDING WITH SENSITIVITY ANALYSIS APPROACH

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Heavy-duty engine mount applications are very common in automotive industry. Different mount configuration and quantities are important for engineers to prevent sound or vibration problem. Heavy-duty engines generally have four-mount system, which are located on engine only. There are few example of the 5th and 6th mount system which are placed on transmission case. However, additional mount can be added onto transmission case to prevent durability issues. In this paper, different kinds of mount designs and effects are investigated with the vibration characteristics. Accordingly, design parameters related with different mount number and bracket designs. Moreover, parameters are optimized according to package space. Optimum design solution is acquired with the help of test solutions. Parameters are optimized and accomplised with the test results. Test results helped to find the optimum solution, which is accurate with the finite element analysis (FEA) results. Test and FEA results compared and determined for these FEA mount compliance and mount vibration parametric analysis.

Keywords: Heavy Duty, Design Criteria, Active Side Mount Vibration and Compliance

1. Introduction

In today’s world, heavy-duty vehicles have crucial part in construction and transportation. Most of the heavy-duty vehicles deals with harsh environment while driving. To be able to handle harsh environment, robust design is needed on heavy-duty engines and accessories. Engineers developed various solutions in years. One of the many and popular solution is to design vigorous engine mounts but sometimes additional mounts are required to solve durability issues. Additional mounts are also needed to be study in aspect of Noise, Vibration and Harshness (NVH). [1]

Every sound or vibration starts with excitation, which is called source. Source is transmitted through the receiver by paths. In most cases, multiple paths carry energy from source to receiver. Mounts are the paths that carry noise and vibration to driver or passenger. [2] Therefore, it is important to understand the ranking or relative importance of the various paths. New transfer paths are opened with the additional mounts. Moreover, it is necessary to assess the effects of extra mounts in NVH perspective.

The main structures that connects engine to vehicle are mounts. Mounts have enormous effects on both durability and NVH persp. Number of mounts and location depends on engine type and configuration. North-south configured passenger cars usually have three-mount system, which are located two mounts on engine and one mount on transmission case. East-west configured passenger cars mostly have same amount of mounts like north-south engines but in different position. Heavy-duty engines generally have four-mount system, which are located on the engine block itself. Sometimes rear mounts can be shifted onto the transmissions case, which covers extra mounts on transmission case [1]. However, there are few examples of additional 5th & 6th mounts because of packaging, cost and NVH issues.

Various analyses are used to predict the NVH behavior of 5th & 6th mounts and compared with test results. Modal analyses were performed to see the mode shapes of mount brackets. Frequency Response Function Analyses (FRF) were performed to see the displacement and natural frequencies of all mounts. Wide-open throttle active side mount bracket vibration (Mount Vibration) analyses were performed to obtain displacement results under the actual loads, which are combustion, crank-train and valve-train, loads. As a result of mount vibration analysis, loudness levels are calculated.

In this study, main investigation is to see structural vibration effects of various designs of 5th and 6th mounts on power pack system and correlate the computer aided engineering (CAE) results with test results. Base design and engine mounts are compared with test results. Four different designs of 5th and 6th mounts are investigated with each other and baseline. With the help of trustworthy predictions, the best design option determined in aspect of NVH.

2. Correlation and Design Study

2.1 Mass Moment of Inertia (MMoI)

2.1.1 FRF Based Determination of Structural Inertia Properties by Testing

Finding the center of gravity and the moment of inertia of structure is essential to correlate FEM with test. Mass Moment of Inertia (MMoI) has crucial importance in our case. For a simple object, inertial properties can be found with simple mathematical equations. However, complicated structural object like in heavy-duty engine and transmission system, such calculation is not applicable. Trifilar pendulum is used to calculate MMoI of system with a large experimental error. Consequently, trifilar pendulum is dangerous and time consuming. [3] New test methods have been developed to find the rigid body properties of complex objects, which are categorized into two groups. First, one is modal property method. Other is mass line method. Rigid body characteristics are important steps for the reduce computation time and simulation. These methods are using vibration data as input and FRF functions. [4] Different set of frequency response functions (FRF) are used for calculation between excitation degrees of freedom and response degrees of freedom. In the FRF graphs, there is a place, which represents the mass line, is selected between the rigid body modes and flexible modes. Mass line is showed in the figure 1. In this test, mass line method is used for determination of MMoI.

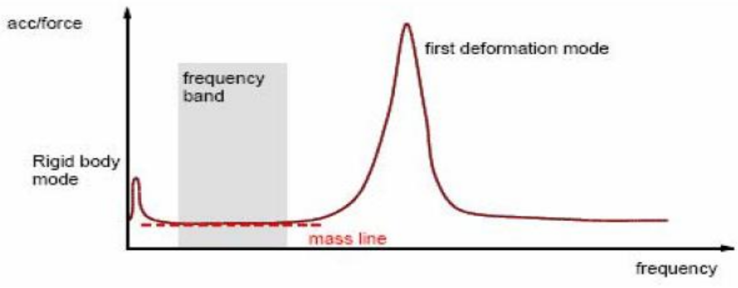

Figure 1: Frequency band of the FRF's represents mass line, which is located between rigid body and flexible modes

Advantages of using FRF based mass line method these are;

- Results are obtained from the real structure
- Require basic suspension setup

- Standard FRF measurement process and parameters
- There are a lot of different techniques while determining the optimum mass line
- Fluid's mass effect can be calculated
- Alternative hardware configuration can be easily access
- Deformation modes can be easily observed

Only disadvantage is this method require real physical prototype.

While accomplishing the test LMS Test.Lab Rigid Body Calculator is used, which program is working some steps these are;

- Determination of the system mass
- Test setup while in the free boundary condition
    - Low frequency suspension (<1 Hz)
- Setting up the models wire-frame geometry
    - Observatory center of gravity axis must be away from the center of gravity and moment of inertia
    - Deformation mode's points must be observatory area
- Excite the structure while using hammer and measure the FRF's.
- For CoG/MMoI analysis FRF sets must be choose.
- Frequency band of the FRF's represents mass line, which is located between rigid body, and flexible modes have to be in observatory area.
- Determine the suitable Mass line methods
    - Unchanged FRF method
        - If there is enough distance between first flexible mode and rigid body modes. (>%10)
        - Calculations are made from measured FRF results.
    - Corrected FRF method
        - If there is not enough distance between first flexible mode and rigid body modes.
        - Mode decomposition is apply.
        - It can be fixed with detached the FRF's flexible modes.
    - Lower Residual method
        - If there is no distance between first flexible mode and rigid body modes.
        - While using calculating FRF, deformation modes decompose with lower residuals
        - Deformation modes using lower residual terms rigid body modes between mass line interval recalculate
- Calculating the inertia [6]

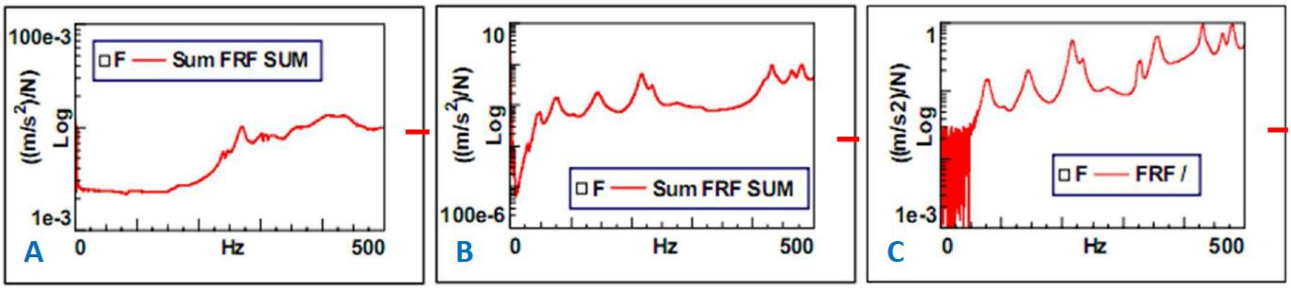

Figure 2: Mass-Line Methods A-Unchanged B-Corrected C-Lower Residual Method examples

The mass line values are used for calculating rigid body properties according to the selected mass line method. All the spectral lines of the frequency band are used as input for a global (least square) solution. [5]

2.1.2 Engine and Transmission Mass Moment of Inertia (MMoI) Correlation

MMoI test is accomplished only engine itself. For the transmission correlation computer added design (CAD) model is used and difference in physical properties are applied to FEM. Engine model is checked for same values after getting the ten structural properties. FEA model is prepared in Hyper-Mesh program. FEM results and test results are different from each other. For this reason, in-house excel tool is used for correlation of mass and moment of inertia values. Using the excel tool to calculate the CoG difference and apply the difference between these inertia values to get same physical properties. Correlation of MMoI between FEA model and test results gives confidence for further analyses.

Figure 3: Engine Mass Moment of Inertia Testing

2.2 Mount Bracket Compliance Analysis

Mount bracket compliance analysis is vital to understand dynamic behavior of components. Displacements on mount brackets due to power pack excitations are critical to NVH. Stiffer mount brackets have lower displacement level. Transmission of dynamic displacement of power pack through the mounts can cause vibration at seat track, steering wheel and the floor pan. These results can lead to NVH problems, mostly structure-borne noise.

2.2.1 Comparison of 5th &6th mount design and engine mounts with test and CAE Results

Mount bracket compliance is computed through forced frequency response analysis of the power pack excited by unit loads in all three direction. CAE results are read from where the accelerometers are in test conditions. Only z-direction compliance results are showed in this study because of the main excitations are on z-axis, which are combustion, gear train and valve-train. Compliance results are compared between 100-350Hz because of heavy-duty engine working limits.

Test data of all four engine mounts and current design of 5th & 6th mounts mount bracket compliance results are shown in figure 4 and figure 5.

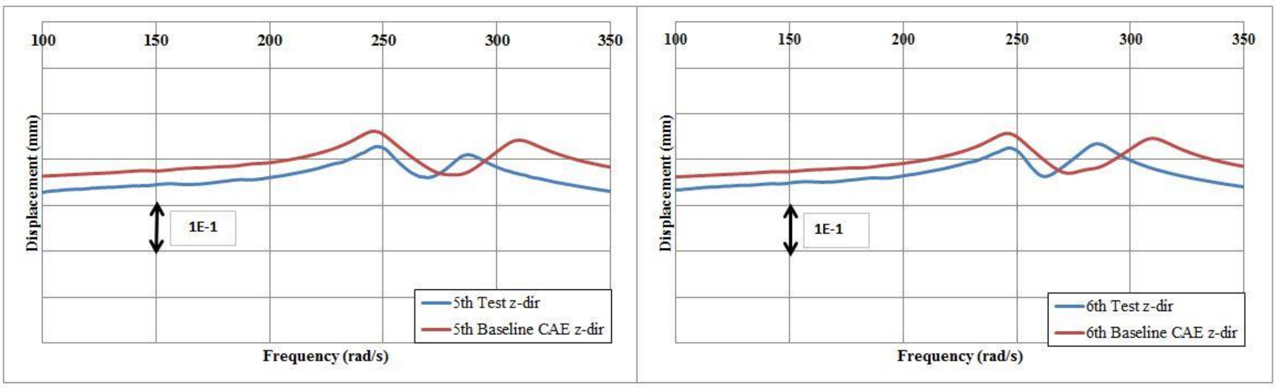

Figure 4: Transmission Mount Bracket Compliance Results

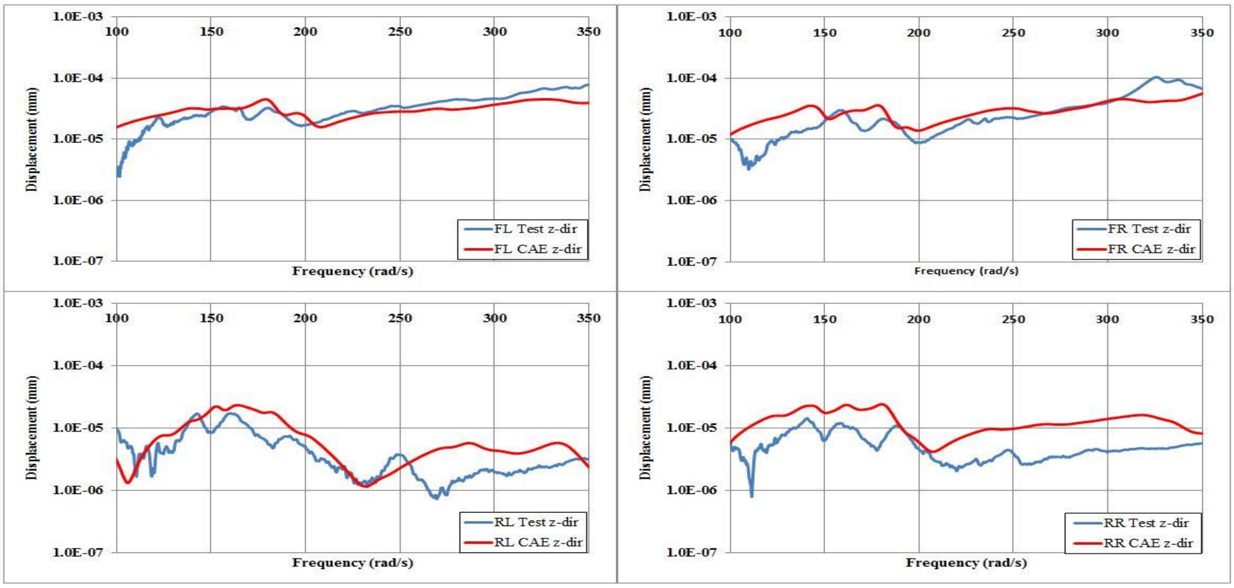

Figure 5: Engine Mount Bracket Compliance Results. Front Left (FL), Front Right (FR), Rear Left (RL) and Rear Right (RR)

Four different mounts, which are Front Left (FL), Front Right (FR), Rear Left (RL), Rear Right (RR), and 5th & 6th mounts compliance results are compared with test data. As shown in compliance results trend line matches with test data. Further investigation is needed on structural damping for amplitude correlation.

2.2.2 Design Iterations of 5th & 6th Mounts

Further design iterations are analyzed on 5th & 6th mount. Iterations are proposed with considering package space. Results are showed in figure 6 and figure 7.

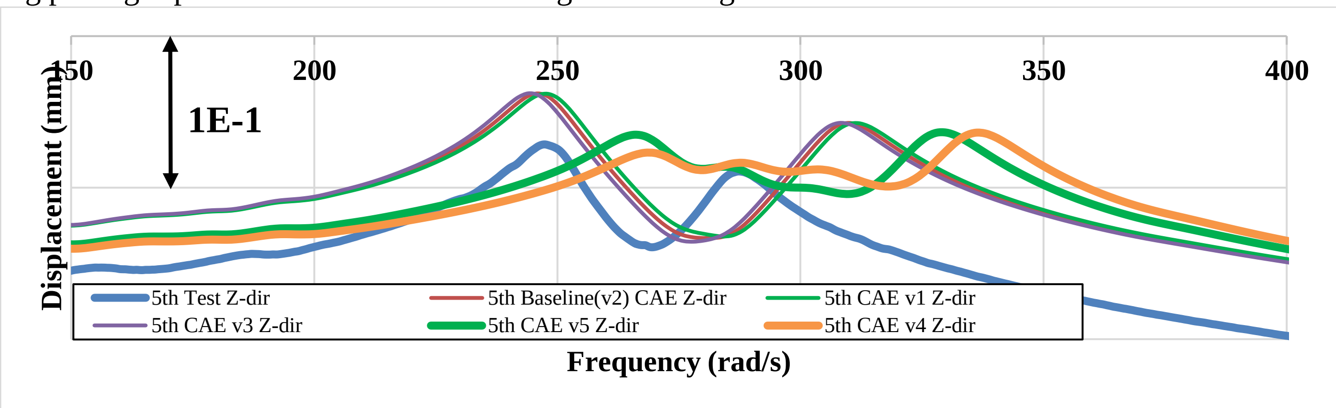

Figure 6: 5th Mount Bracket Compliance Iterations

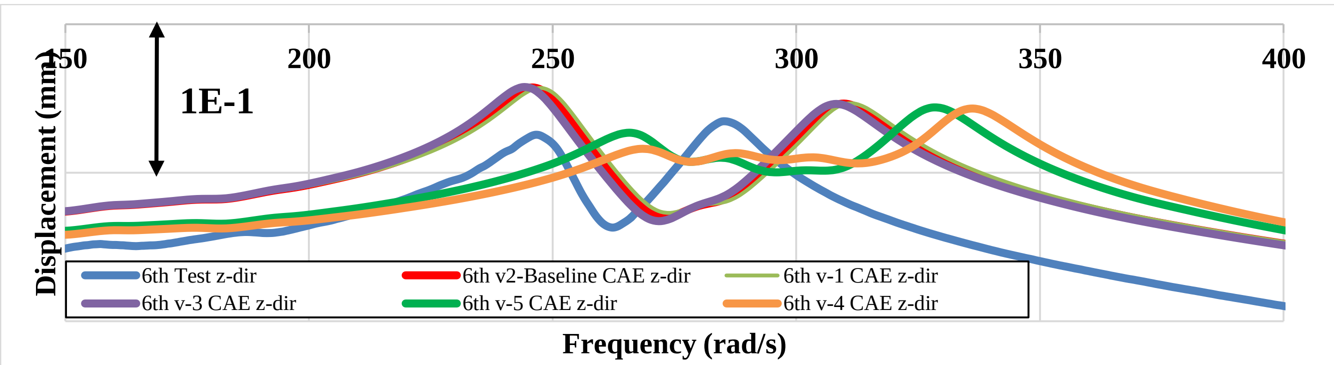

Figure 7: 6th Mount Bracket Compliance Iterations

As a result, v4 iteration is better than current design (v2). Displacements are lower than baseline and the first bracket mode is higher than baseline.

2.3 Mount Vibration Analysis

2.3.1 Wide Open Throttle Maneuver (WOT) Test

WOT Maneuver is the vehicle with hot engine condition and selected gear by cruising at minimum possible engine speed on selected surface which is to be clear of snow, dirt, gravel and other debris. Furthermore, open throttle fully and accelerate up to cut-off speed. The initial throttle opening process should be smooth taking about 0.5 sec. Record data from minimum possible engine speed up to cut-off speed. The test should be repeated at least minimum three times for solid results. Tests must be performed in all gears indicated unless restricted gear requirements have been set in a program specific. After the test analysis to be performed on the recorded NVH signals and post-processing available for this purpose.

2.3.2 Correlation with current FEA model (V2)

Using mount vibration analysis to obtain acceleration results under the actual loads, which are combustion, loads, crank-train loads, and valve-train loads. Loudness can also determine which is the noise in term of sones for mount vibration analysis. All loads are generated with different programs such as combustion loads are generated with a multi-body dynamic solver, which is an in-house program. Gear-train and Valve-train loads are generated with the AVL-Excite Timing Drive.

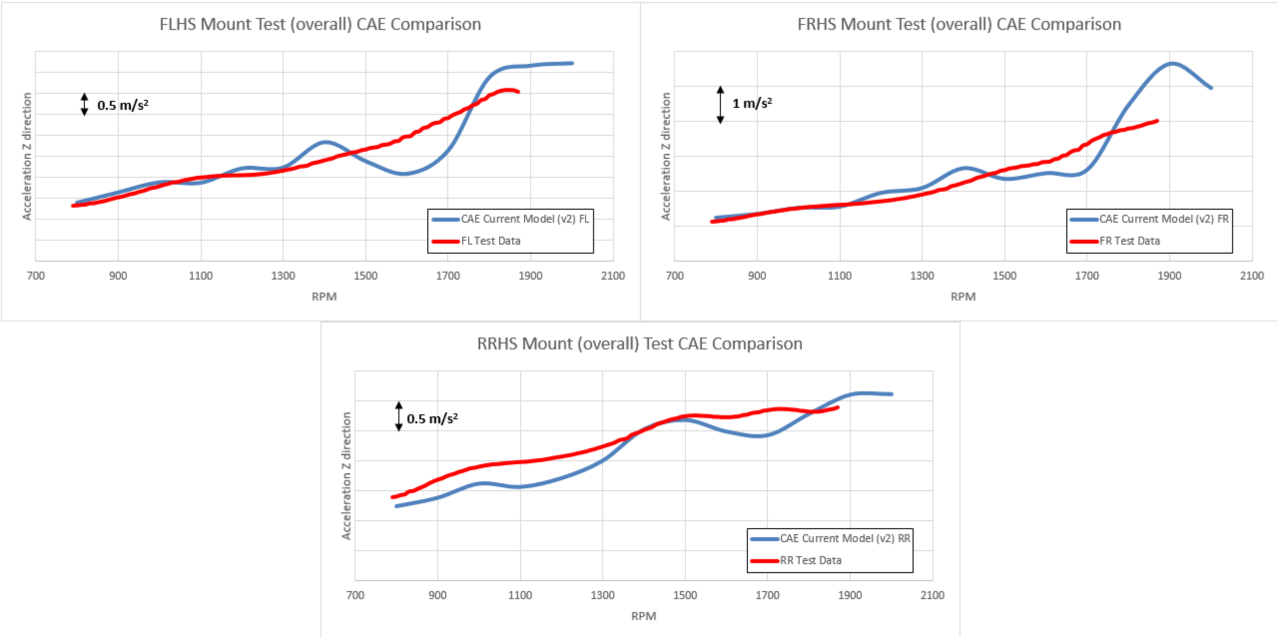

Figure 8: Engine Mounts Vibration Results (Front Left Hand Side FLHS, Front Right Hand Side FRHS, Rear Right Hand Side RRHS)

Only z-direction mount vibration results are showed in this study because of the main excitations are on z-axis, which are combustion, gear train and valve-train. Besides, there are no rear left hand side (RLHS) results because there are no test data. As shown in figure 8 CAE results are represented with a blue lines, test results are represented in a red lines. Engine mounts are highly correlated with analysis. Different mounts affect seen easily

2.3.3 Different Design Results with Mount Vibration

Four different bracket (5^{th} & 6^{th}) and four mount system proposal is investigated. Iterations have been designed according to package space. Correlation in v2 and test results are shown in the figure 9 and figure 10. As shown in these figures 1200-rpm, 1400-rpm and over the 1800-rpm results had slightly better than the current model (V2). V4 has the best option for both compliance and mount vibration results.

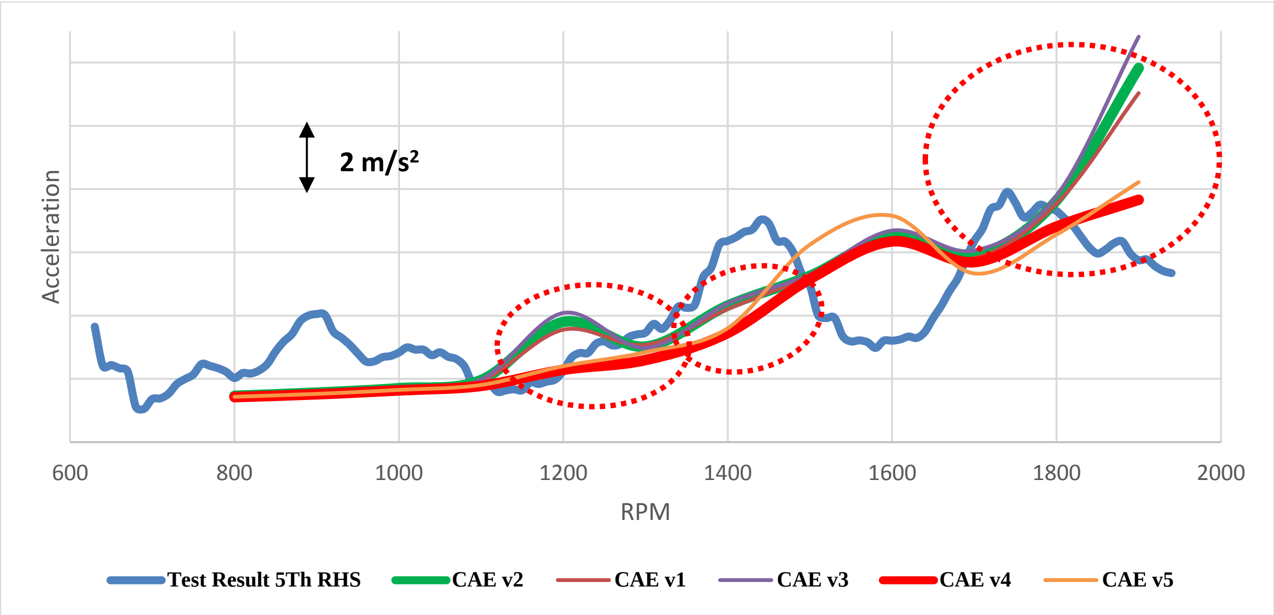

Figure 9: 5th Right Hand Side Mount (6th) Vibration Results

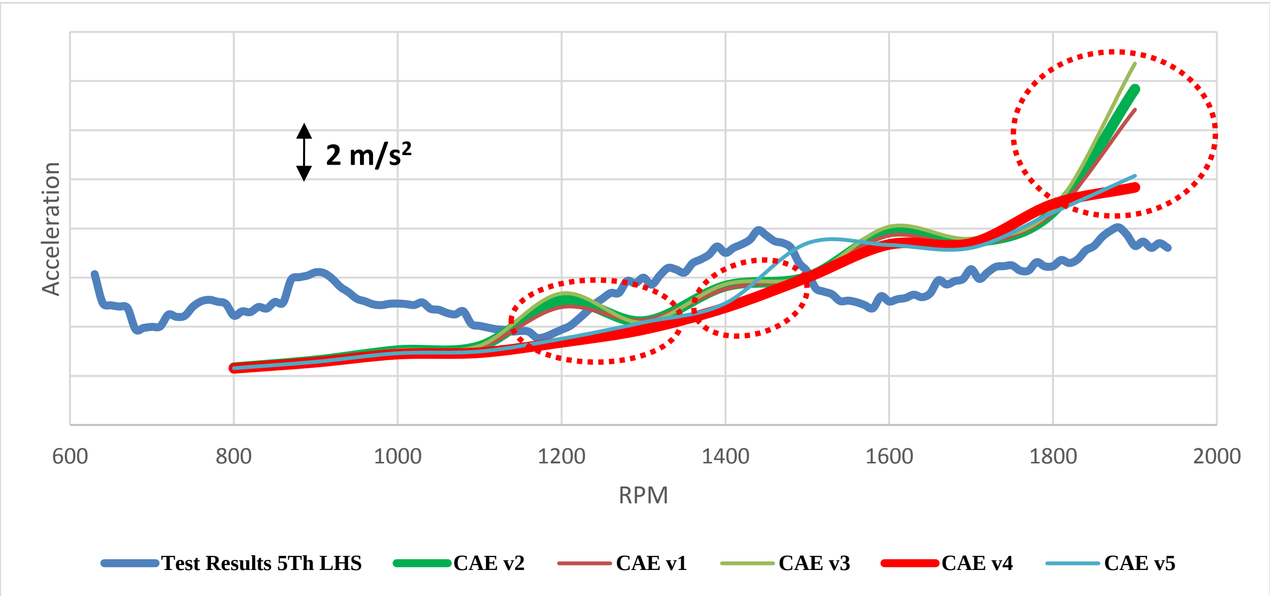

Figure 10: 5th Left Hand Side Mount Vibration Results

Loudness level is mesasured by listener studies in which participants were asked to match the certain pure tone noise at certain frequencies. Loudness levels shows a function of frequency for several values of noise intensity. Measured unit is called as ‘’phons’’. Accordingly, loudnees is the numerical funtion of loudness level. Empirical formula is created with test to suggest proportional metric. Unit of loudness is ‘’sones’’ and it’s not linear. For example, 50 sones is not twice as loud as 25 sones.

Equalevent structural borne loudness level comparison between design iterations and four mount system results (V6) are showed in the figure 11. As shown in figure 11, structural borne loudness (sones) outputs are better and as mentioned before, 1200-rpm and over the 1500-rpm results slightly better than the current model (V2). There is no effect on engine mounts with the change of 5th and 6th mount bracket designs.

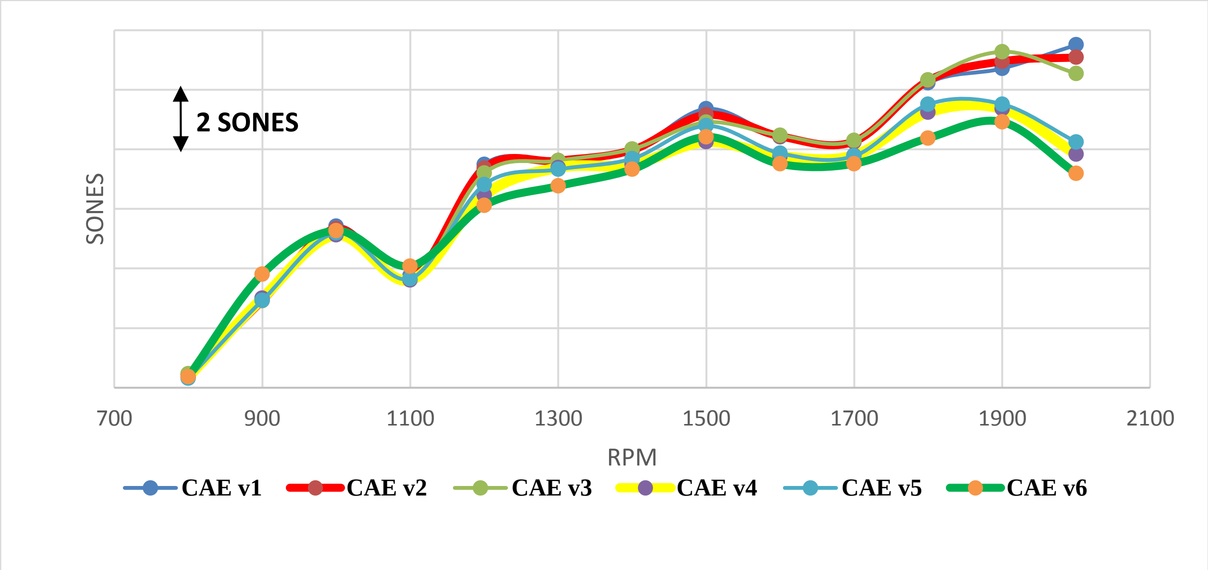

Figure 11: Overall Sones Results

3. Conclusion

A practical computational modeling and testing was performed to understand the 5th and 6th mount dynamics of the power pack. The experimental studies that are MMoI, mount bracket compliance and WOT Maneuver are accomplished for the accuracy of FEM. Correlated model is important to see design changes without any testing. Additional mounts iterations proposed beside, the current 5th and 6th mount for the powertrain mounting system to maximize the isolation of engine vibrations. Overall, mount bracket compliance and mount vibration outcomes indicate that adding extra mounts have significant effect on NVH characteristics. Optimizing the mount bracket is affected the results positively. Different designs are chanced the characteristic of the power pack vibration characteristic.

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