Original Research Article

The Modelling and Correlation Procedure for Assessment of Vibration Performance of a Heavy Commercial Truck

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Abstract

Application of mathematical modelling and numerical methods is a key element of optimum vehicle development process. Upfront CAE (Computer Aided Engineering) simulations of vehicle attributes are required to select designs that meet vehicle attribute targets and legal requirements. CAE is an attractive approach to predict and optimize the performance of the vehicle, and to make decisions in terms of attribute trade-offs before the vehicle prototypes are available. Even though there are many commercial software packages available, the task of correlation of the CAE model to vehicle test conditions is not a trivial task. In this paper, a systematic correlation methodology starting from subsystem level to full vehicle level is presented. Full vehicle correlation is demonstrated on Power train Idle NVH evaluation. Correlation results show that simulation model is highly correlated to the subsystem and vehicle level test data and, therefore it can be reliably used in the product development cycle.

Key Words: Computer Aided Engineering (CAE), Finite Elements (FE), Multibody Dynamics, Noise, Vibration and Harshness (NVH), Model Correlation, Heavy Commercial Trucks

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1. Introduction

Simulations and optimization of the product development in an early stage of product development cycle is priority for all OEMs due to product complexity and competitive pressures [1]. These techniques have the secondary benefit, at least as important as reducing the development time and cost, of experimenting different design variations and answer what-if questions in the product development cycle. The accelerated development of new vehicles depends on the capability of predicting vehicle performance with mathematical modeling and numerical techniques. In order to predict the performance of the prototype vehicles, the CAE predictions should mimic "real-life" conditions. Confidence in CAE predictions for noise and vibration depends highly on the level of modeling details as well as testing techniques. Noise, Vibration and Harshness (NVH) has been an important attribute related to the reliability, quality, users and environmentalists [2, 3, 4]. NVH development has in recent years been supported by the development of Finite Element and Multi-Body System Modeling and software such as Nastran and MSC.ADAMS [5]. With the support of these tools, the development of the vehicle to improve structure-borne and air-borne characteristics is optimized.

Powertrain idle is a very common city operating driving condition and NVH responses associated with it has a very high level of customer perception [6]. Power train Idle NVH is dedicated to understanding vehicle-level responses to power train inputs across the range of customer usage under varying environmental conditions. There is little customer acceptance of vibration and noise especially for error states such as "boom" at engine idle speeds [7, 8]. It is a complex engineering problem with structure-borne and airborne contribution. For example, structure-borne contribution is related to many factors such as engine inertial forces, engine combustion forces, air conditioner loading, power train rigid body modes, suspension modes, steering column modes, steering wheel modes, flexible modes of the vehicle cabin, exhaust hanger forces and exhaust system modes. Therefore, it is very important to use a systematic methodology involving consistent evaluation/testing methods and CAE validation procedures in order to address power train idle NVH refinement [9]. The effect of vehicle static and dynamic stiffness on the correlation of ride, handling and NVH performance using experiments on the actual vehicles has been addressed in [10]. The effect of the body attachment stiffness on the idle noise and road noise for NVH performance improvement has been studied in [11]. The paper presented the correlation between experimental test and finite element based simulations as well as important design factors to improve idle noise and road. The problem of NVH CAE model correlation with respect to experimental testing has been addressed in [12]. An objective metric based on statistical hypothesis testing has been proposed and evaluated to estimate the prediction capability of the analytical model to that of reference test data. A guideline about the computer experiment selection based on complexity level of the system under consideration focusing Factorial Designs, Taguchi Orthogonal Arrays, Monte Carlo Simulation, and Latin Hypercube Sampling has been offered in [13]. In that study, possible techniques to simplify the NVH models are also discussed. Modal test results of various vehicle body structures at the fully trimmed and body-in-white configurations are presented in [14]. Global body mode correlation between the two configurations is established, and the scatter band of the resonance frequency ratio between these two configurations. Finite element based simulation results are compared with experimental measurements up to the limit of 800 Hz in order to increase the range of analysis capability on frequency response functions in [15]. A case study on low frequency NVH performance evaluation and refinement for heavy commercial vehicle truck using Hybrid Test-CAE

methodology has been presented in [16]. A vehicle concept finite-element model is experimentally assessed for predicting structural vibration to 50 Hz and comparisons are made between the predicted and 63 measured frequency-response-functions (FRFs) of nominally identical vehicles and modes of the body-in-white, the trimmed body, and the full vehicle in [17]. To achieve better NVH performance, it is important to set competitive overall vehicle level NVH targets and cascade it down to system and sub-system targets. Even though, there are studies showing correlation between analytical models and test data on system level, the overall vehicle level correlation of simulation models to test data systematically has not been addressed. This paper discusses a systematic methodology to correlate the power train idle NVH performance of a heavy commercial truck. System level modeling and correlation is first performed to correlate the rigid body and flexible modes of the engine, suspension, chassis and cabin. Then, vehicle level correlation is performed in order to correlate the customer experienced vibration metrics between the analytical model and vehicle level testing. The results show that high level of correlation can be achieved by systematically correlating models from subsystem level to full vehicle level. The methodology is demonstrated to correlate the simulation results on the steering wheel vibration while the truck is in idle conditions.

The paper is organized as follows: first, the details of subsystem and vehicle level truck models are presented along with the correlation methodology to correlate analytical model to physical testing is presented in Section 2. Section 3 explains the details of the system and vehicle level testing as well as correlation results between simulation model and test data. Finally, conclusions and future work is stated in Section 4.

2. Materials and Methods

In this section first modeling details of the subsystem and full vehicle is presented. Then, the correlation procedure in order to achieve good correlation levels on vehicle level is described in detail.

2.1. Description of System and Vehicle Level Models

Full vehicle model of a heavy commercial truck contains many subsystem models. The simulation tool allows users to use general finite element (FE) models of the structures and generate Euler beams, rigid body components, spring and bushing elements and mass elements with simple constraints. Each FE model is a modal model with its modal analysis data and that information is used to form global stiffness and mass matrix of the complete system. Details of each subsystem model is described in this section.

A trimmed cabin model, shown in Figure 1, consists of sheet metal of the body structure so called body in white structure, doors, instrument panel, seats, hood, mirrors, and exterior trim parts such as bumpers. Most of the structural components are represented by finite element models forming more than one million degrees of freedom system. While structural parts are FE models since the main objective of modelling of the system is to calculate vibration response of steering wheel, seat track and some other critical points, many of the trim parts are modelled as mass elements with representative mass and inertia properties to reduce degree of freedom of the system. Higher number of degree of freedom increases solution time, output data size and reduces post processing performance of the computers.
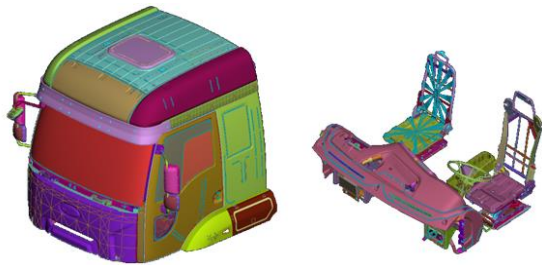
Figure 1. Trimmed Cabin (left), Instrument Panel and Seat Models (right)

Acoustic cavity model is generated in order to investigate the acoustic response of the vehicle such as sound pressure levels at driver ear position. Since the seat cushions are the sound absorber materials, they are modeled as FE models and connected to cavity model with coupling elements. Cavity model is connected to trimmed body structure model with coupling elements at the surface area of the cabin or cavity and called "wet" surface. Structural vibrations are transfered to cavity nodes as boundary conditions to calculate sound pressure levels in the cabin. Cavity model inside the cabin is shown in Figure 2.
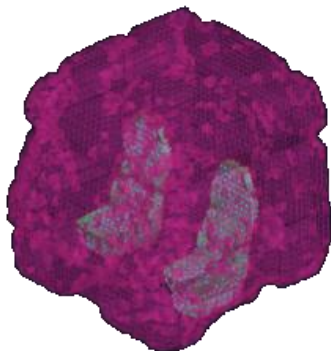

Figure 2. Cavity Model of the truck cabin

Trimmed chassis model consists of rails and cross members, which are the structural parts supplying the stiffness and where all the accessories such as muffler, fuel tank, fenders and bumpers are attached. Dynamic behavior of the chassis mainly depends on the mass and inertia of the subcomponents and modeling approach. Many of the structural components are modeled as finite element models since heavy components have low frequency resonances, which might affect the vehicle vibration and acoustic characterictic. As in the same case with trimmed cabin modeling approach the

number of degree of freedom depends on the mesh size, which needs to be correctly determined by means of structural tests to improve the solution performance, post processing time and effectiveness of the CAE models. Trimmed chassis model is shown in Figure 3.
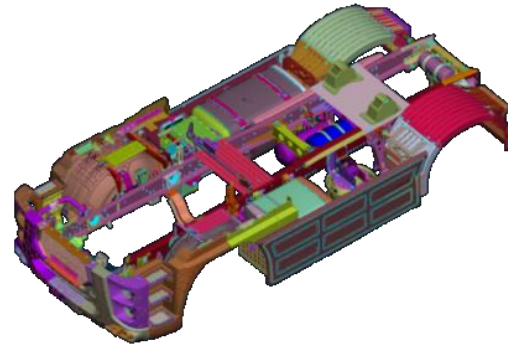

Figure 3. Finite Element Model of Trimmed Chassis

Suspension subsystem in the vehicle consists of front suspension, rear suspension and a cabin suspension between chassis and cabin structures. The critical point in modelling of structures is the resonance frequencies of the systems. If a system has a resonance frequency between the frequency ranges of interest such as engine excitation frequency flexible modelling of the system is required to correctly calculate the response of the whole structure. Rigid body modelling is usually preferred when there is no resonance frequency of the component in the frequency range of excitation sources to reduce the effort of finite element modelling and improving solution and post processing time.

Flexible modelling consists of finite element modelling for components with complex geometries and beam and spring element representation of components having simple geometrical properties such as tie-rods, struts, antiroll bars etc. Figure 4 shows finite element model of cabin anti roll bar and front suspension. Beam modelling of the antiroll bar is an example for this part but not considered here. Front suspension model consists of leaf springs, antiroll bar, knuckle and spindle system. Leaf spring has been modelled with shell element with different

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thickness to better represent variable cross section of each leaf.

Figure 4. FE Model of Cabin Anti Roll Bar and Front Suspension Model

Rear suspension system of this tractor (shown in Figure 5) consists of air springs which have been represented as spring and damping elements. Rear axle model consists of rear axle housing and rear spindles, represented with finite elements.
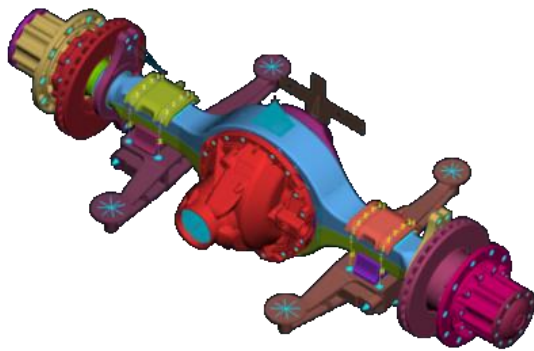

Figure 5. Rear Suspension Model

Vehicle development is a part of team work where system and system level targets are clearly determined and cascaded to each related engineering team. Critical part of NVH development is to separate resonance frequencies from the frequencies of excitation sources and resonance frequencies of other main components to prevent undesired vibrations and noise. A part of this power train should be designed in a way that the structural resonance frequencies need to be higher than that of engine excitation. Regarding this information a vehicle design engineer just needs the mass and inertia properties of the engine to investigate the effect of rigid body resonances of the power train and the effect of mass and inertia of the engine to vehicle response under engine excitation. Thus, power train system can easily be modelled as a rigid body with correct mass, inertia and engine mount stiffness information.

Most of the subcomponents in a vehicle system such as suspensions, antiroll bars, engines are connected to body or chassis structure with rubber bushings for better isolation of the body structure from excitation sources. NVH CAE model of the heavy commercial truck model has more than 60 bushings at different attachment points. Each of these bushings exhibits dynamic stiffness characteristics depending on the frequency as depicted in Table 1. K_x, K_y, K_z refer to the stiffness of the bushing in x, y, z directions and C is the damping to the bushing.

Table 1. Dynamic stiffness characteristics of a suspension bushing for displacement amplitude of 0.25 mm

Frequency	K_x (N/mm)	K_y (N/mm)	K_z (N/mm)	C (Ns/mm)
5	1519	1478	350	11.1
10	1574	1531	363	5.8
20	1638	1593	383	3.0
30	1678	1630	399	2.1
40	1709	1658	412	1.6
50	1734	1682	422	1.3

Figure 6 shows the assembly of a full vehicle model of heavy commercial tractor. All of the finite element models, rigid body model and Euler beams are presented as display models. Generic approach to the tire models is to create finite element models of the tires in deformed configuration due to the weight of the vehicle and run modal analysis at this configuration to obtain modal model of the tire. Another simple approach used in this case is to model the tires as spring and bushing elements with measured stiffness values. The modelling approach would depend on the engineering problem to be studied.
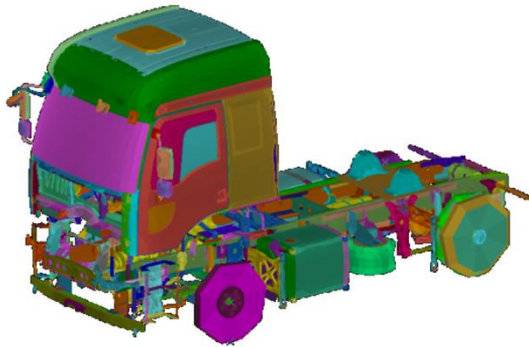
Figure 6. Full Vehicle NVH Model of the truck

Figure 7 shows the suspension, engine, driveline and steering system models of the truck. Driveline and steering system is modelled as mostly Euler beam elements. This approach allows engineers to run design iterations faster and generate response since it is easy to change section properties of the beams in the tool quickly and run simulations without solution of a million degree of freedom system again since the modal model of FE components are available. Engine and transmission models are rigid body models and shown as visual elements to investigate the performance under engine excitation.
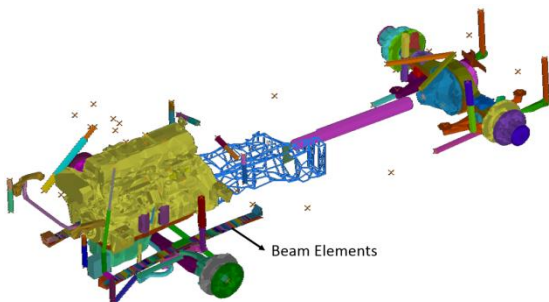

Figure 7. Beam & Rigid Body Models of Tractor

Internal combustion engine is one of the main excitation sources of the vehicles due to the oscillating and rotating masses of the crank train and combustion pressures. Because of the geometry of the crank train and the oscillating piston and connecting rod masses there exists a mass torque around crankshaft axis and shaking forces perpendicular to crankshaft axis of an internal combustion engine. The inertia forces can be expanded into Fourier series giving components of orders of engine speed. Each of the components is called engine orders. Based on the number and orientation of cylinders some of the forces corresponding to engine orders cancel each other.

Heavy commercial tractor vehicle, presented here has a six cylinder inline engine and crank train is shown in Figure 8. Calculation of the forces and moments show that the main inertia loads are the 3rd engine order inertia torque for a six cylinder engine. Although 6th engine order inertia torque has a considerable effect, remaining torques and shaking forces can be neglected.
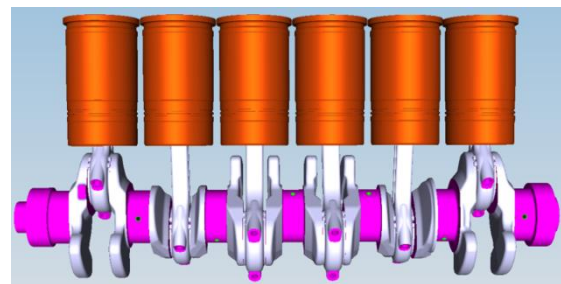

Figure 8. Crank train of a Six Cylinder Inline Engine

Another important excitation of the engine comes from the combustion pressure inside the cylinders. Combustion pressure in the cylinders, shown in Figure 9, depend on the engine speed and need to be considered separately while generating a load case to simulate wide open throttle conditions. Combustion torque of the engine is calculated by the multiplication of the piston area and crank train geometry. Combustion pressure is similarly expanded to Fourier analysis to see the dominant engine orders. For a six-cylinder engine there are two combustion events per revolution of the engine. 3rd engine order combustion torque is found to be dominant load acting on crank train and engine. Calculation of those forces has been performed by AVL Excite simulation tool. Engine load data is then calculated for each RPM of the engine for wide open throttle case and applied in full vehicle model to crank train and engine block to obtain the vehicle response.
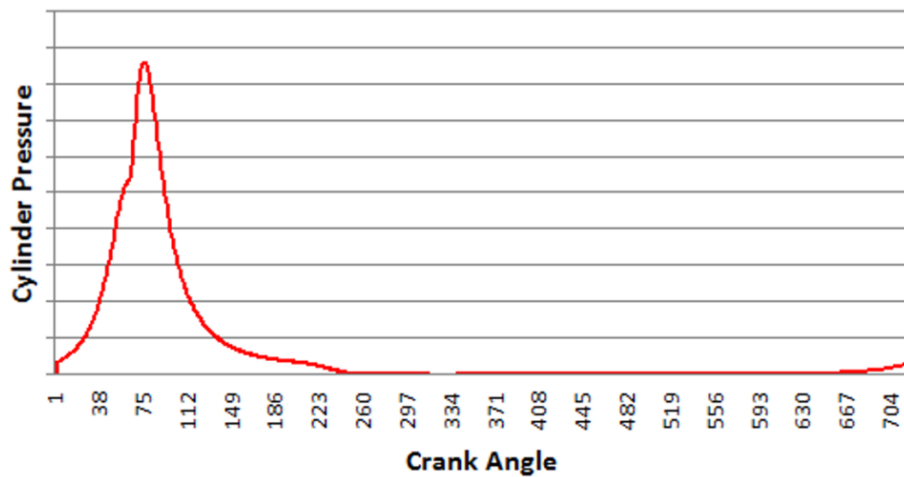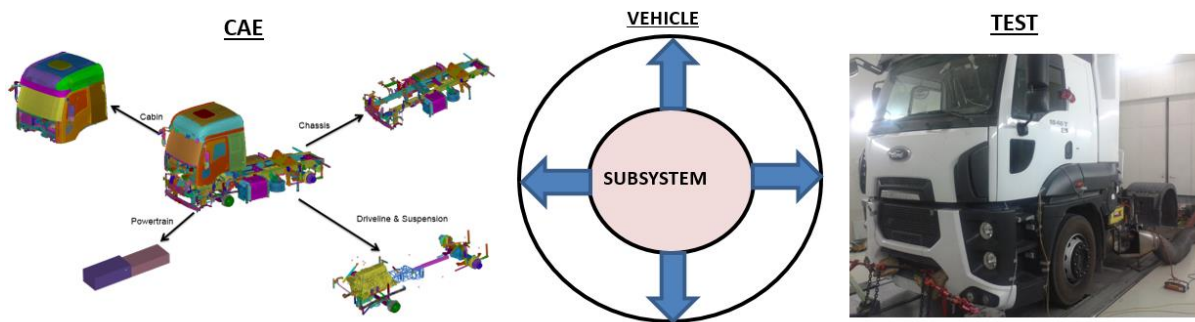
Figure 9. Combustion Pressure of a Cylinder

2.2. Description of the Model Correlation Procedure

Full Vehicle Correlation study is systematically carried out starting from subsystem level to full vehicle level correlation as shown in Figure 10.

1. Correlation on Rigid Body Modes
   - Rigid Body Modes (Engine)
   - Rigid Body Modes (Chassis)
   - Rigid Body Modes (Suspension)
   - Rigid Body Modes (Cabin)

2. Correlation on Rigid Body Modes
   - Flexible Modes (Chassis)
   - Flexible Modes (Cabin)

3. Vehicle Level Correlation

Figure 10. CAE – Test Model Correlation Procedure

Subsystem level and full vehicle level correlations consist of validation of the rigid and flexible modes of the subsystems in terms of the importance of the metrics on the overall NVH performance of the full vehicle as explained below:

2.2.1. Rigid Body Mode Correlation for Engine/Transmission

The engine mounts are designed in order to isolate the low frequency torsional vibrations from the engine. In normal operation of the vehicle, the lowest frequency vibration of concern is at idle engine speed. Decoupling of all rigid body engine modes is critical and to be made

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sure that power train and / or road excitations do not generate excessive motion of the engine in any direction [1]. A general rule of-thumb for what constitutes minimum frequency separation is 10% [3]. Due to significance on the NVH performance, correlation of the roll frequency of the engine is chosen as a subsystem metric.

2.2.2. Rigid Body Mode Correlation for Suspension Hop / Tramp Modes

Coincidence of the power train or road input excitation with engine bounce mode may affect the NVH and ride quality of the vehicle. So the desire is to separate the bounce mode of the engine from the hop modes of the suspension [1]. The second consideration is to ensure high correlation of the suspension vertical modes due to possible interaction between suspension vertical modes and engine bounce mode.

2.2.3. Chassis / Cabin Attachment Point Mobility

Achieving low driving point mobility (or high dynamic stiffness) at body attachment points allows the body to block the incoming vibration energy [4] and therefore considered important in the correlation of the NVH CAE model with respect to vehicle tests.

2.2.4. Cabin Flexible Modes

The resonances of a body panel will coincide with other resonances of the vehicle and they can coincide with excitation frequency coming to the vehicle body [4]. Excitation frequencies and body structural modes must be separated in order to ensure good NVH performance. There should be at least 1 Hz spacing between normal mode frequencies [8]. Therefore, cabin flexible modes are to be correlated to ensure high level of correlation for full vehicle.

2.2.5. Full Vehicle Correlation

Once the correlation of the subsystem correlation on subsystem level metrics, the full vehicle level correlation with integration of subsystem models as in prototype vehicle needs to be correlated under one of the customer usage conditions. Steering wheel velocity while truck is idling at idle speed is chosen as the full vehicle metric to be correlated between test and CAE.

3. Results

3.1. Measurement Hardware and Equipment

Correlation studies have been carried out in many facilities including semi-anechoic rooms and four-poster testing. The measurement hardware is summarized in this section. Technical specifications of the accelerometers and microphones used in the testing are tabulated in Table 2 and Table 3, respectively. These sensors are used to measure the acceleration of panels, chassis components, and steering wheel in order to correlate on subsystem level and vehicle level.

Table 2. ICP Accelerometer Technical Properties

Description	Triaxial
Sensitivity (pC/g)	3
Measurement Range (Hz)	1-8000
Operating Temperature (°C)	-55 to +175
Weight (gram)	13

Table 3. Free-Field Microphone Technical Properties

Description	Triaxial
Sensitivity (mV/Pa)	31,6
Measurement Range (Hz)	8-12500
Dynamic Range (dB)	15,8 to 146
Operating Temperature (°C)	-55 to +175
Weight (gram)	13

3.2. Rigid Body Mode Correlation

Rigid Body Mode correlation cover the rigid body of the power train/engine assembly (bounce, roll modes), and front and rear suspension hop and tramp modes. In order to calculate the rigid body mode of the power train/transmission and suspension modes, truck is excited on a four poster (by sweeping the frequency range where the rigid body mode is expected to be) as shown in Figure 11. Power Spectral Density (PSD) response of the accelerations (attached to engine block

and front/rear axles) gives the rigid body modes in test. For the calculation of the rigid body modes, normal modes analysis in Nastran has been performed and compared with the test data from four-poster testing. Correlation of the simulation model is defined as in Equation (1) showing how closely the simulation results can predict the actual test results. Comparison of the rigid body modes between test and simulation model is shown in Table 4.

Figure 11. Truck Testing on a four-poster

Correlation of the simulation model is defined as in Equation (1) showing how closely the simulation results can predict the actual test results. Test and simulation outputs are the frequencies of the rigid body modes such as the frequency of the engine bounce mode, engine roll mode, hop and tramp modes corresponding to front and rear suspensions. The second term in Equation (1) is the relative error between simulation and test data. It means if the simulation output is same as the test output, the error between the simulation model and test data is 0% and therefore correlation level is 100%. Comparison of the rigid body modes between test and simulation model is shown in Table 4. The correlation level of 90% is generally deemed sufficient [5].

$$Correlation(\%) = 100 - \left| \frac{\text{Test Output} - \text{Simulation Output}}{\text{Test Output}} \right| \tag{1}$$

Table 4. Rigid Mode Correlation

Description of Rigid Body Mode	% Correlation
Engine Bounce Mode	93
Engine Roll Mode	94
Front Suspension Hop Mode	95
Front Suspension Tramp Mode	99
Rear Suspension Hop Mode	97
Rear Suspension Tramp Mode	90

3.3. Flexible Mode Correlation

Flexible modes are important in terms of the vibration of the subsystem and sound generated in the cabin. Therefore, a significant effort has been spent in order to correlate the flexible modes of chassis, cabin and components such as steering wheel as being the interface to the customer. The modal testing of chassis, cabin and steering wheel subsystems are carried out with impact testing. Tests are carried out in a semi-anechoic room in order to obtain the Noise Transfer Functions as well. The instrumentation of the simulation model and vehicle in test is shown in Figure 12. LMS Test. Lab is used to obtain the flexible modes in testing, and Nastran SOL 103 solution is used to obtain the flexible modes in the simulation model.
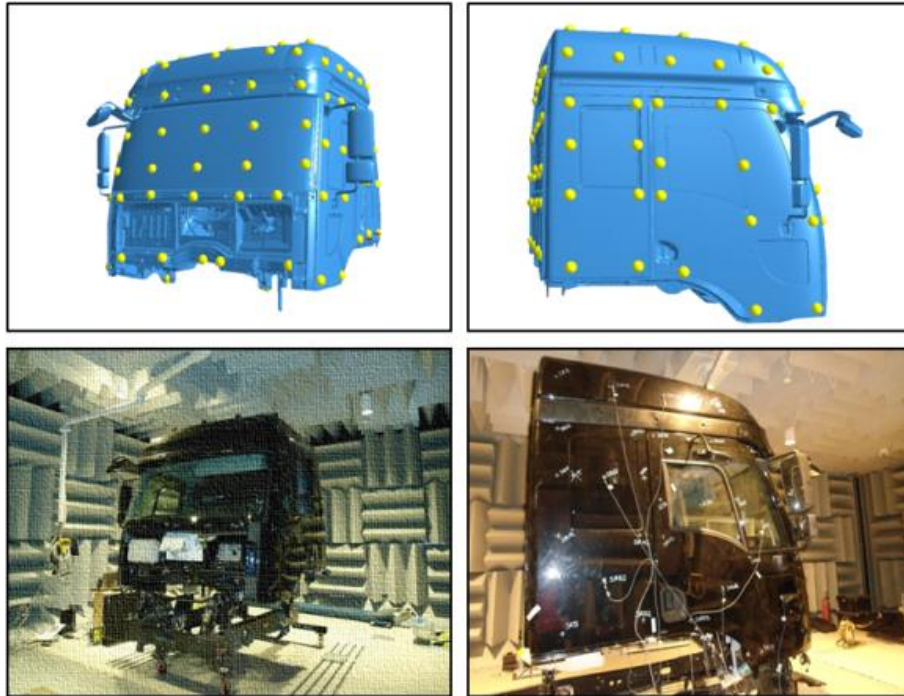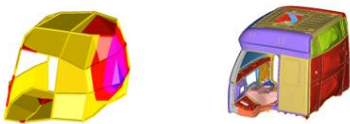
Figure 12. Instrumentation for determination of flexible body modes (top view : CAE, bottom view : physical testing in semi-anechoic room)

The correlation status on the flexible modes of the chassis, cabin and steering wheels are summarized in Table 5. The animations of the modal analysis results from the test and CAE facilitate the mode identification and similarity between test and CAE. The results show that there is a high correlation level between CAE and test which is promising for the correlation of vehicle level performance. Correlation is calculated according to Equation (1) and the correlation level of 90% is generally deemed sufficient [5].

Table 5. Correlation of flexible modes of chassis / cabin subsystems

Mode	Test / CAE		Correlation
Torsion Mode			92
Lateral Mode			98
Front Chassis Mode			94
Backpanel Mode			95
Roof Mode			96
Steering Wheel Mode			97

3.4. Direct Point Mobility Correlation

Direct mobility or driving point mobility is the complex ratio of the velocity and force taken at the same point in a mechanical system, which is a measure of the energy imparted to the system (ISO 7626-1, 1986). Correlation of the point mobility at the cabin attachment points is followed by the correlation of the Noise Transfer Functions, as a measure of the sound pressure levels felt by the driver. Point mobility is measured with impact testing and correlation results for the rear cabin attachment point are shown in Figure 13 with good correlation up to 200Hz. The correlation is performed by comparing the frequencies where the response has a peak in the amplitude calculation.
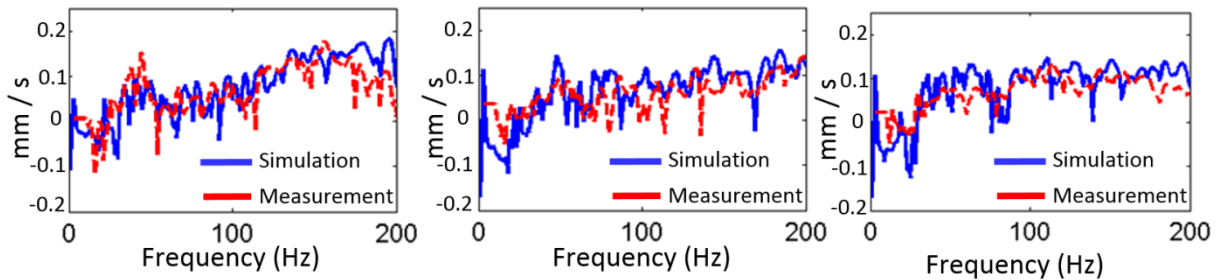

Figure 13. Direct Point Mobility Correlation for rear cabin attachment points (left : x-direction, middle : y-direction, right : z-direction)

3.5. Power train Idle Full Vehicle Correlation

Once the subsystem correlation is completed, full vehicle correlation is investigated while the truck is running in idle condition. Steering wheel vibration is considered in this section, as an important metric as the steering wheel is the interface to the customer. Root of sum of squares (RSS) of the steering wheel velocity is chosen as the full vehicle metric. Before correlating the steering wheel vibration, active side engine acceleration correlation is performed in order to make sure that acceleration to the vehicle replicates the real test conditions. Active side engine acceleration results are summarized in Figure 14 in x, y and z-directions. The results show good correlation levels between CAE & Test.
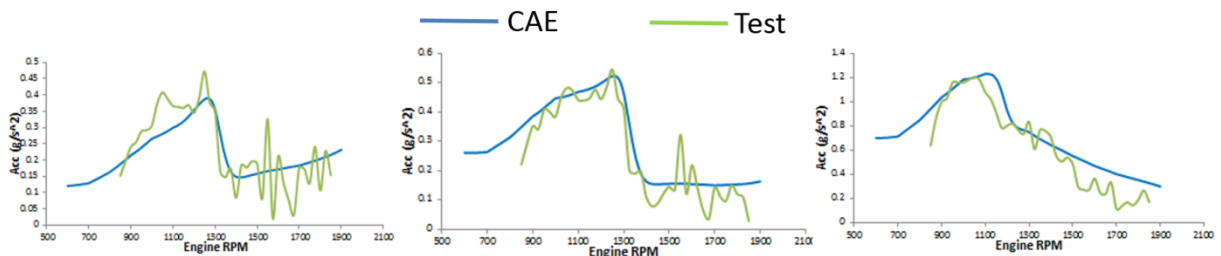

Figure 14. Active Side Engine Acceleration (left: x-direction, middle: y-direction, right: z-direction)

RSS of the steering wheel velocity between test and CAE is plotted in Figure 15. The results show good correlation in terms of the peak amplitude and the frequency where the peak amplitude occurs (550 rpm). The simulation model is capable of predicting steering wheel vibration within 1 mm/s of the vehicle test results. This correlation level is considered satisfactory in terms of representing the vibrations levels transmitted to the driver of the truck.
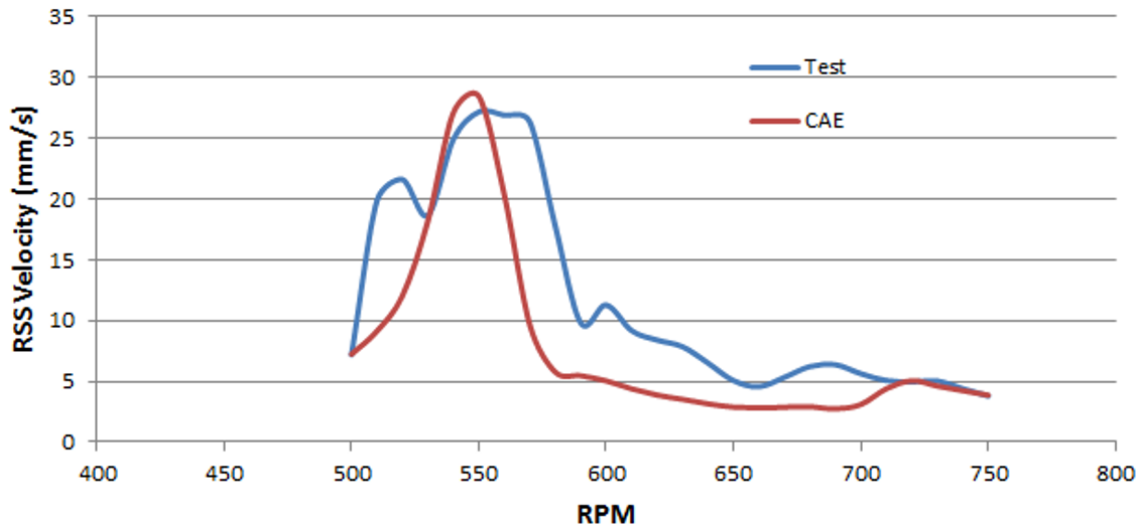
Figure 15. Steering Wheel Velocity RSS

4. Conclusions

In this work, a systematic methodology to correlate the vehicle level performance of Power train idle NVH of a heavy commercial truck is presented. High correlation in vehicle level is achieved starting from correlation of subsystem level to vehicle level. More specifically, correlation results on subsystem level metrics such the rigid body mode correlation for engine/transmission, rigid body mode correlation for suspension hop / tramp modes, chassis and cabin flexible modes are above 90%, which is generally deemed sufficient in the literature. In addition, good correlation on the chassis attachment point mobility, cabin attachment point mobility, and cabin vibration transfer functions up to 200 Hz to ensure high correlation in terms of full vehicle is achieved. Finally, it is shown with the correlation methodology that the simulation model is capable of predicting steering wheel vibration within 1 mm/s of the vehicle test results while the truck is idling. The high correlation on the vehicle level is attributed to the high correlation on the subsystem starting from power train to the steering wheel.

This work was limited to the correlation of the Power train Idle NVH response. Correlation of the CAE model in order to replicate real world scenarios such as power train acceleration NVH, road NVH (low and high frequency) is considered as future work. Sensitivity of the design parameters such as engine mount stiffness, cabin suspension bushing stiffness is important to address cross-attribute decisions and optimize the vehicle design from a broader perspective. More elaborate Design of Experiment and sensitivity of the model parameters on the subsystem and vehicle level metrics is needed in order to understand the overall design of the heavy commercial vehicle. Once the sensitivity of the design parameters identified, the desired vehicle level performance can be achieved. Design sensitivity and full vehicle optimization in terms of various attributes is also considered as future work.

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