

Ride Optimization for a Heavy Commercial Vehicle

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Abstract

The ride comfort of the commercial vehicle is mainly affected by several vibration isolation systems such as the primary suspension system, engine mounting system and the cab mounting system. A rigid-flexible coupling model for the truck was built and analyzed in multi-body environment (ADAMS). The method applying the excitation on the wheels center and the engine mountings in time domain was presented. The variables' effects on the ride performance were studied by design of experiment (DOE). The optimal design was obtained by the co-simulation of the ADAMS/View, iSIGHT and Matlab. It was found that the vertical root mean square (RMS) acceleration and frequency-weighted RMS acceleration on the seat track were reduced about 17% and 11% respectively at different speeds relative to baseline according to ISO 2631-1.

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INTRODUCTION

Road roughness and the engine are considered as the major excitation source, in addition to the unbalance vibration from the tire/wheel and the driveline [1]. The vibration transmitted to the driver through the primary suspension system, the engine mounting system and the cab mounting system. As a result, the vehicle comfort behavior is closely associated with these vibration isolation systems [2].

Many studies described different ways to apply the excitation on the model. Karnopp, D. [3] analyzed the effect of a white noise road input in a simple car model. Mitschke et al. [4] introduced a road input to the vehicle considering the cross power spectrum between the different wheels. Tamboli et al. [5] tried to produce the real data from road profile and optimize the suspension system under the real road inputs.

Roebuck et al. [6] designed a control system with the "modified skyhook damping" and optimal full-state feedback strategies in semi-active suspension to reduce the dynamic body acceleration. Courteille et al. [7] improved the vehicle ride performance at an idle speed by tuning the stiffness of the elastic element to decrease the coupling of the powertrain and

chassis. Xu et al. [8] enhanced the cab vibration isolation performance by achieving the desired nature frequency range and minimizing the motion coupling at each mode of the cab.

This paper introduced an approach considering both the rough road and engine vibration in the ADAMS/View model. DOE method was used to identify the key parameters which were more sensitive to the objective functions. Furthermore, co-simulation among ADAMS/View, iSIGHT and Matlab was developed to obtain the optimal design of the vehicle.

EXPERIMENTS AND VEHICLE MODELING

This section described the experimental procedure and the full vehicle modeling. The experimental data from the vehicle's different measurements could be analyzed to define the vibration levels of the vehicle's different parts and some main natural frequencies of the subsystems. Then take the condition of 50 km/h for example, the experimental acceleration data on the axles and powertrain were used as the input in the ADAMS/View model. And the responses in the model were compared with the experimental data for validation.

Experimental Method

The experiments were carried out to measure the time-history acceleration of the truck. The experimental instruments included acceleration sensors, GPS velocity sensor (VBox), data acquisition system (DATARec 4) with the software (Head Recorder) and the power inverter. The acceleration sensors were used to measure the acceleration of the axles, the frame, the cab, the wheel, the seat track and the driver seat. The [figure 1](#) showed the measurements in the experiment.

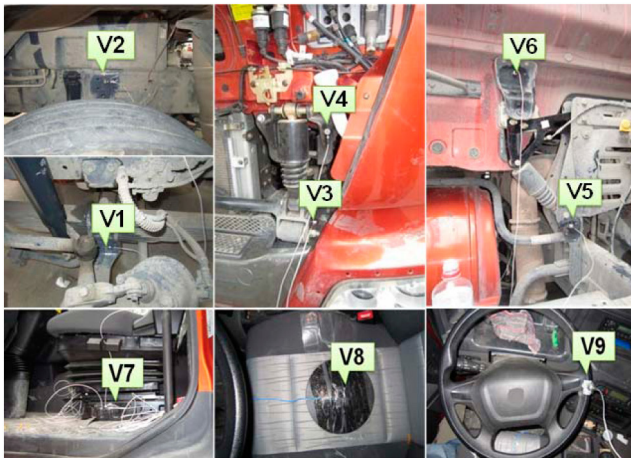


Figure 1. The measurements in the ride comfort test

The vehicle was driven on the B class road at constant speeds of 35, 40, 50, 60, 70, 80, 90 km/h. The road must be straight and the slope should not be too large. Then the vibration signals could be recorded successfully by the instruments.

Modeling and Validation

The no-load truck was built in a rigid-flexible model so that the dynamic properties of the vehicle could be reflected more accurate at the lower modal. The full vehicle and the subsystems were shown in [figure 2](#).

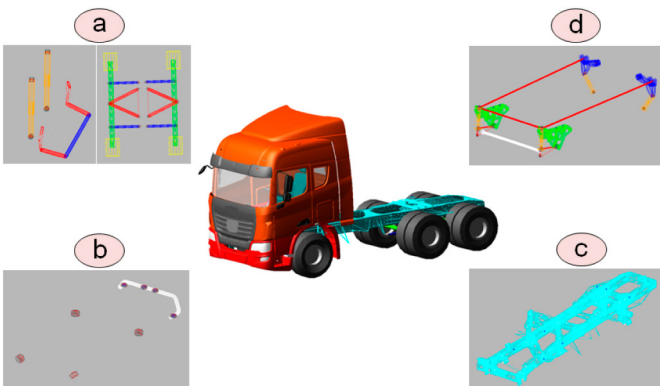


Figure 2. The vehicle model in ADAMS/View

- a: The vehicle's front and rear axles were built with multi-rigid-body models.
- b: The powertrain was supported by the elastomeric bushings, and the flexible torsion bar was built as the auxiliary mounting.
- c: The frame was built with a flexible body to ensure the natural frequency as much as possible.
- d: The antiroll bar and the brackets of the cab were built with flexible parts.

The full vehicle was supported on the ground by six elastomeric bushings instead of the tires, which were only given the vertical stiffness and damping. The four engine supports were modeled with flexible body to prevent interference when the excitation applied on the rigid engine, and there were two bushings connected with a torsion bar. The bushings' axes were parallel with the lateral direction of the full vehicle. The dynamic axial stiffness was 540 N/mm, and the dynamic radial stiffness was 1440 N/mm. The flexible torsion bar had great influence on not only the natural frequency but also the decoupling ratio of the powertrain.

The bench simulation was conducted by applying the displacement excitation to the actuators. It could be realized in frequency domain that the displacements were transformed from the acceleration data by the quadratic integral algorithm. And the acceleration data were measured at the speed of 50 km/h in the experiment above. The input positions and the directions were shown in [table 1](#) and [figure 3](#), respectively.

Table 1. Inputs description of [figure 3](#)

number	position	direction
1, 2	Front wheel center (left/right)	X,Y,Z/X,Z
3, 4	Middle wheel center (left/right)	X,Y,Z/X,Z
5, 6	Rear wheel center (left/right)	X,Y,Z/X,Z
7, 8	Front Engine support (left/right)	X,Y,Z
9, 10	Rear Engine support (left/right)	X,Y,Z
11	auxiliary Engine support	X,Y,Z

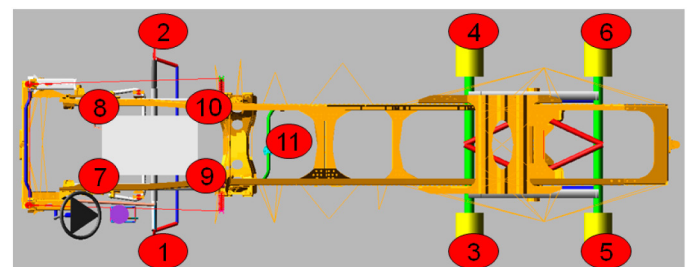


Figure 3. Input positions in the model

The model also defined several outputs to evaluate the acceleration at different parts. The locations of these outputs were the same as the experiment. On the frame there were a node up to the primary suspension, a node up to the rear suspension, four nodes below the engine, and two nodes at the both ends of the torsion bar which was connected

with the powertrain. In addition, there were eight nodes of the cab suspension system, a node on the seat track which was the most important place we focused on, a node on the seat and a node on the steering wheel. Figure 4 shows the positions.

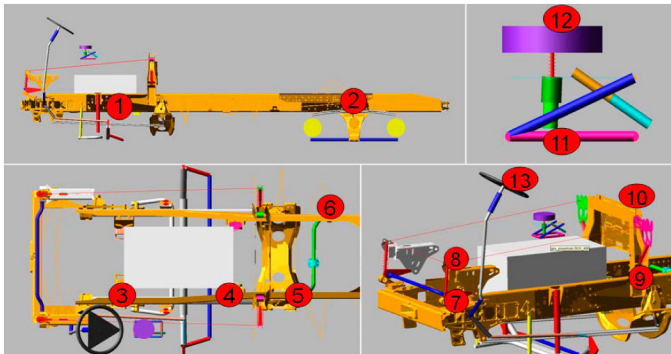


Figure 4. Output positions in the model

The results of the simulation and experiment were shown in table 2 and figure 5, 6, 7, 8. The model was verified in terms of the RMS and the Power Spectral Density (PSD) values, which reflected the vehicle system characteristics in both time domain and frequency domain.

Table 2 showed that the differences of the seat track's vertical RMS acceleration between the experiment and simulation was only 9%, and it was the most important measurement as it would be treated as the optimization objective later. Most measurements' RMS value differences were within 15%, and there were only three inessential ones up to 20%. Because the model used the rigid part to simplify the complicated structure of the vehicle, and even the error of the same experiment in different times might be up to 15%, the results in table 2 were acceptable.

From figure 5, figure 6, figure 7, figure 8, it was found that the main frequencies in simulation agreed well with that in experiment, especially the measurements on the seat track and the driver seat. In figure 5 and figure 6 there were some differences in amplitude of the left front cab's vertical acceleration curve at 1.8Hz and 2.2Hz, which was understandable because in the model the cab's pitch and vertical motion natural frequencies were just 1.8Hz and 2.2Hz respectively, and the input errors between the simulation and experiment would be amplified. In fact, it exerted very little influence on the seat track's result.

As a result, it could be concluded that the model was sufficiently accurate for further optimization, since the differences of vertical RMS acceleration were acceptable and the curves of PSD had a good agreement as well.

Table 2. The RMS values between the experiment and simulation

measurements positions	vertical RMS acceleration (g)		differences
	experiment	simulation	
left-front cab suspension on the frame	0.1859	0.1638	-12%
left-rear cab suspension on the frame	0.1139	0.0941	-17%
right-front cab suspension on the frame	0.1698	0.1688	-1%
right-rear cab suspension on the frame	0.1288	0.1101	-15%
left-front cab suspension on the cab	0.099	0.1077	9%
left-rear cab suspension on the cab	0.0869	0.0966	11%
right-front cab suspension on the cab	0.1117	0.091	-19%
right-rear cab suspension on the cab	0.1251	0.0985	-21%
seat track	0.0823	0.0893	9%
seat	0.093	0.1033	11%
steering wheel	0.1113	0.0994	-11%

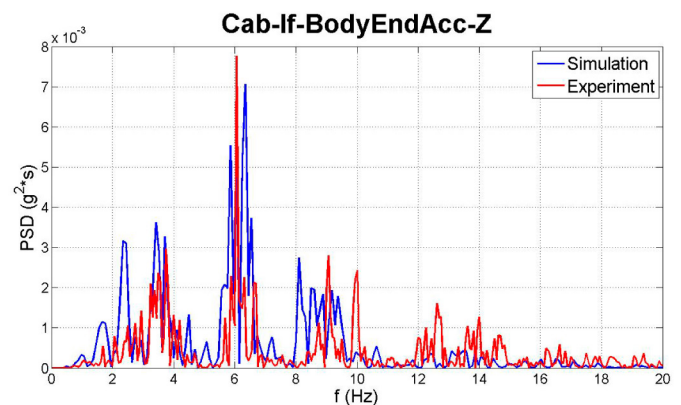


Figure 5. Vertical acceleration PSD of the left-front cab suspension on the frame (red line is the experiment result and the blue line is the CAE model result)

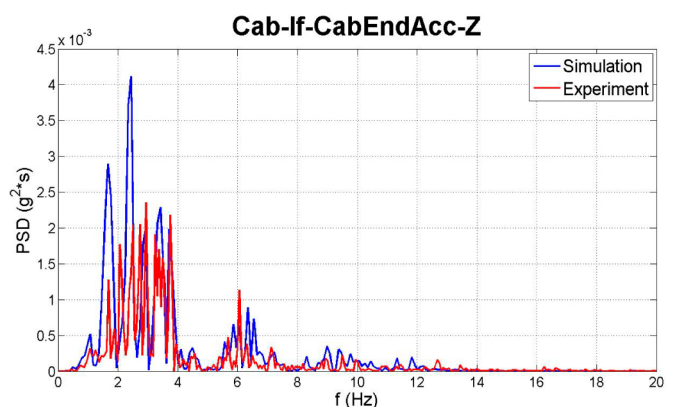


Figure 6. Vertical acceleration PSD of the left-front cab suspension on the cab (red line is the experiment result and the blue line is the CAE model result)