

Effect of Cab Suspension Configuration and Location on Tractor Semi-Trailer Driver Comfort

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ABSTRACT

It is well known that the ride quality of trucks is much harsher than that of automobiles. Additionally, truck drivers typically drive trucks for much longer duration than automobile drivers. These two factors contribute to the fatigue that a truck driver typically experiences during long haul deliveries. Fatigue reduces driver alertness and increases reaction times, increasing the possibility of an accident. One may conclude that better ride quality contributes to safer operation.

The secondary suspensions of a tractor have been an area of particular interest because of the considerable ride comfort improvements they provide. A gap exists in the current engineering domain of an easily configurable high fidelity low computational cost simulation tool to analyze the ride of a tractor semi-trailer. For a preliminary design study, a 15 d.o.f. model of the tractor semi-trailer was developed to simulate in the Matlab/Simulink environment. The developed model incorporates the effects of frame beaming. The contributions of the different modes of vibration to the ride comfort were analyzed. It is shown in this work that the ride at the driver seat can be significantly improved by relocating the cab mounts near the nodes of the first mode of bending of the tractor frame and by employing a full cab suspension. The developed simulation tool was used to quantify the improvements in the driver ride comfort.

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1. INTRODUCTION

It is well known that the ride quality of trucks is much harsher than that of automobiles. Additionally, truck drivers typically drive trucks for much longer duration (up to 16 hours per day) than automobile drivers. These two factors contribute to the fatigue that a truck driver typically experiences during long haul deliveries. A study conducted by the Federal Motor Carrier Safety Administration (FMCSA) [1] investigated 967 crashes involving heavy truck between 2001 and 2004. The study found that truck driver fatigue directly contributed to the accidents in at least 13% of the accidents. Fatigue reduces driver alertness and increases reaction times, increasing the possibility of an accident. Poor driver recognition of traffic and decision error which might be brought about partly by driver fatigue were the most cited critical reason (42%) for the accidents in the FMCSA study. Due to the size of trucks, accidents involving trucks cause much more damage than automobiles. One may conclude that better ride quality contributes to safer operation. The purpose of this study was to assess the benefits of optimizing the cab dampers and the cab mount location on driver's ride comfort at a preliminary design stage. Since 2012, the American Trucking Association's Technology and Maintenance Council are developing a set of standards and industry expectations for future truck designs through 2025 and beyond. It is hoped that the results of this study will help shape the future truck programs of various truck manufacturers.

Cab and seat suspensions have been an area of particular interest because of the considerable ride comfort improvements they provide. Use of cab suspension to improve truck ride has been studied since late 1960s. Foster [2] studied the effect of full-cab suspension for a cab-over-engine (COE) tractor using prototype vehicles. He concluded that the natural frequency of the cab suspensions should be lower than 3Hz to isolate the cab from the chassis modes. Flower [3] also assessed the ride quality of a COE tractor fitted with either a front or a rear cab suspension system analytically and subjectively. He used a linear, planar 54 d.o.f model to study the effects of cab isolation system on ride. The study found that the rear cab isolation was easier to install and provided some ride benefits due to decoupling of the cab motion from the frame bending. The front cab isolation system, where the rear of the cab was rigidly mounted to the frame, provided noticeably superior ride compared to the rear cab system or a cab without isolation system (which was standard configuration during 1978) though the installation of the front cab isolation system is more challenging. Tractors having rear cab isolation are now the industry standard in the USA.

Bending of the frame has a large bearing on the ride quality of the truck. Among other aspects, Gillespie [4] analytically studied the influence of frame stiffness on the transmission of road inputs to the cab of a COE tractor. He identified the first vertical bending mode or a beaming mode as occurring in the 6-9 Hz range. Spivey [5] investigated the effects of cab and seat suspension on the isolation of

the driver from the road inputs using a multi body ride model. His model included the effects of tractor frame beaming and the presence of a suspension at the fifth wheel joint. Among the parameters he investigated were the cab and seat suspensions, tractor axle suspensions, fifth wheel suspension, frame bending vibrations, and tractor wheelbase. The study concluded that inserting a fifth wheel vertical suspension system in a rear cab suspended truck was detrimental to ride comfort of the driver. Patricio [6] experimentally compared the effects of two different cab suspensions on ride comfort along with the effects of two different truck frames. The study observed lower heave accelerations at the B-post with the combined use of lightweight truck frames and a prototype rear cab suspension. Interestingly, though there was improvement in the vertical ride, the longitudinal ride comfort deteriorated with the use of prototype shocks.

Jain [7] studied the effect of various passive cab suspensions on ride comfort using an Adams model of the cab-over-engine tractor-trailer. The study compared the results of the ride dynamic simulations with the ISO-2631 [8] ride criteria. He found that for a conventional tractor setup, with the cab mounted to the truck frame at the front and suspended at the rear, the effect of tractor rear suspension stiffness on cabin ride was not significant. However, the simulation results showed a significant improvement in ride through the application of a low frequency (1.8 Hz) suspension system at the front of the cab. Gross, et al. [9] analyzed the effect of a four-point air cab suspension on driver ride comfort. A full cab suspension provided significant improvement in the ride comfort. However, due to the dominance of rear cab suspensions in the North American market, retrofitting full cab suspensions to existing trucks would be both time consuming and costly.

Most studies on truck ride agree that the presence of the front cab suspension or a full cab suspension significantly improves the ride. However installing a front cab suspension provides many design challenges. Also due to the dominance of rear cab suspensions in the North American market, retrofitting a full cab suspension to existing on-road trucks would be both time consuming and costly. The coupling of the frame beaming motion with the cab motion significantly affects the ride dynamics of a truck. Locating the cab suspension to minimize this coupling effect can be an effective way of improving the ride quality. There is no literature that evaluates the effects of cab mounting locations on the driver's ride quality. The succeeding sections report the results of a preliminary design study. The primary goal of the investigation was to explore the effects of a fully suspended cab and cab mount locations on the driver's ride comfort. A ride model incorporating the frame beaming modes, the road input model and the ride metrics are outlined. Improvements in the driver's comfort due to front and rear cab suspensions, and due to strategically located cab mounts are quantified using the developed simulation tool.

2. MODEL DESCRIPTION

A cab-over-frame tractor with a basic flatbed semi-trailer was modeled in this work. The deterministic 2-D vertical plane ride model of a tractor semi-trailer contains 15 degrees of freedom (d.o.f.). The tractor is modeled as a flexible frame (3 d.o.f.) suspended above 3 axles (1 d.o.f. each) with a rigid cab body (2 d.o.f.) and an engine (1 d.o.f.) mounted on the frame and a seat (1 d.o.f.) suspended above the

cab floor. The trailer (3 d.o.f.) is modeled as a flexible frame suspended above 2 axles (1 d.o.f. each) with the trailer load rigidly mounted onto the frame. The corresponding 2nd order ordinary differential equations (ODEs) for the rigid bodies are derived using the Newtonian approach. The degrees of freedom describing the model can be found in Table 1.

Table 1. Degrees of Freedom of Tractor Semi-Trailer

Sr. No	Description	Symbol
1	Vertical displacement of Driver's Seat	Z_s
2	Vertical displacement of Cab	Z_c
3	Pitch of Cab	θ_c
4	Vertical displacement of Engine	Z_e
5	Vertical displacement. of Tractor Frame	Z_t
6	Pitch of Tractor Frame	θ_t
7	Beaming of Tractor Frame	q_t
8	Vertical displacement of Trailer	Z_{tr}
9	Pitch of Trailer	θ_{tr}
10	Beaming of Trailer	q_{tr}
11	Vertical displacement of Axle #1	Z_1
12	Vertical displacement of Axle #2	Z_2
13	Vertical displacement of Axle #3	Z_3
14	Vertical displacement of Axle #4	Z_4
15	Vertical displacement of Axle #5	Z_5

A schematic of the model can be found in Figure 1. All displacements are measured from the static equilibrium. Data for the baseline model were obtained from a prototype class 8 model supplied by a major truck manufacturer. A detailed description of the geometric parameters, inertial properties and suspension parameters can be found in Appendix A. The following assumptions were made for the truck ride model:

1. Road inputs were assumed to be identical on the left and right sides of the vehicle.
2. No separation of the tires from the road surface.
3. Angular motions are small.
4. All suspension elements (except for the seat) in the model are represented as combinations of linear springs and damping elements.

In-line with the exploratory nature of this study, a linear ride model (except the seat) was chosen. The dynamics of the developed model are representative of that of an actual truck. The seat spring element is modeled with bump-stops. The seat spring behaves linearly in the normal operation regime of ± 7.5 cm from equilibrium. There is a smooth transition between the linear regime and the bump-stops. The cab was modeled with front and rear suspensions that permit both relative pitch and heave of the cab with respect to the tractor chassis. To simulate the effect of a rear suspended cab, the front cab suspension was made very stiff. Cab suspension parameters were tuned based on the resonance frequencies of cabs found in literature [2, 4, 10]. The engine is modeled as a lumped mass mounted on the tractor frame. The stiffness and damping of the modeled engine suspension is the summation of the stiffness and damping coefficients of all the engine mounts. The fifth wheel joint is modeled as a parallel spring and damper system. To approximate a conventional fifth wheel joint the values of the fifth wheel suspension stiffness and damping were set very high.

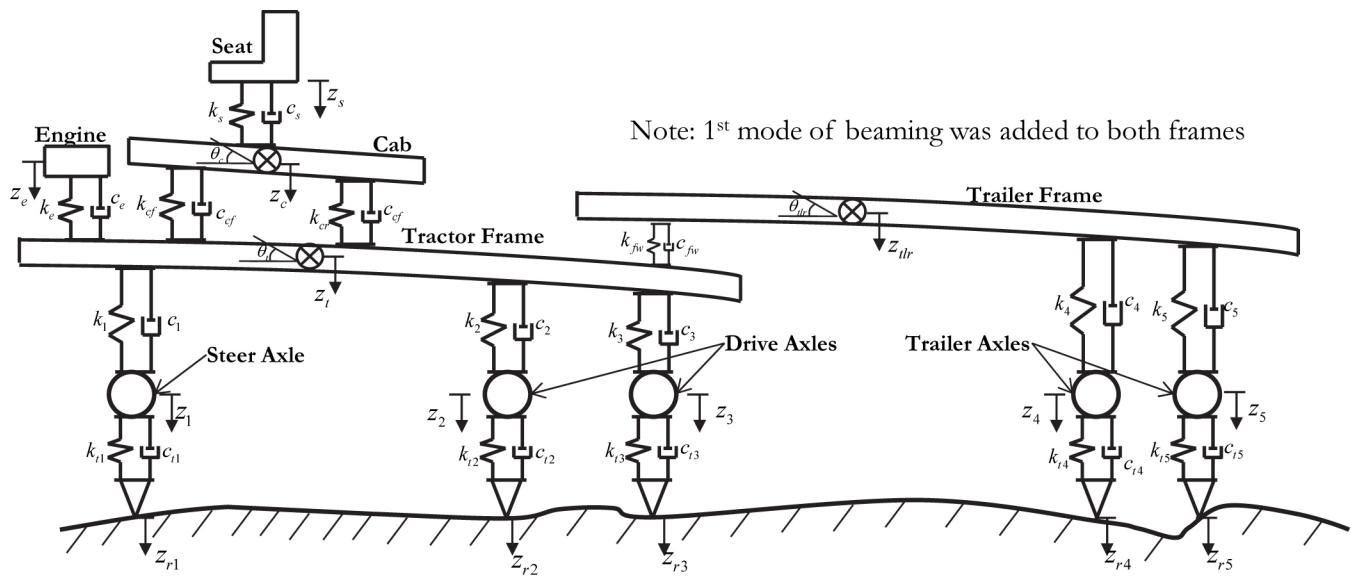


Figure 1. Side View Schematic of the 15 D.O.F. Tractor Semi-Trailer

The tractor and trailer frames consist of two parallel frame rails connected to each other with cross members. The tractor and trailer frames are modeled as Euler-Bernoulli beams. Applying the concept of separation of variables, the beam deflections were decomposed into temporal and spatial components. The resulting temporal equations of motion (EOMs) of the beams were merged with the other EOMs of the system using the assumed modes method. For this study it is assumed that the masses of the components bolted onto the tractor and trailer frames are uniformly distributed over the length of the frame. The set of ODEs that describe the dynamics of each frame are developed using the Lagrangian approach. The solutions for the spatial EOMs of the frames are dependent on the boundary constraints. Since the two frames are pinned at the fifth wheel, a pinned (simply supported) boundary condition is applied at that point. The other ends of the frames are free to flex. Therefore, the tractor frame is modeled as a free-pinned beam and the trailer frame is modeled as a pinned-free beam. The first bending or “beaming mode” normally occurs between 6-9 Hz [2,3]. Only the first bending mode is considered for this research since it is normally the most important mode. Bending of the tractor frame affects both the longitudinal and vertical acceleration at the driver’s seat. For this study the natural frequencies for the frame beaming modes were set to 7 Hz for both the tractor and trailer frames. A complete description of the model is too lengthy to be included in this paper and can be found in [11]. The tractor semi-trailer ride model was developed in the Simulink environment. Matlab scripts were used to prescribe the truck parameter values and the road profile, to initiate the time simulation and to process the time history results for ride analysis. The time domain response of the truck to random road irregularities was obtained and then processed to obtain power spectral densities of the ride environment.

No ride data were available from the truck manufacturer. The developed model was therefore compared to TruckSim, a commercially available truck dynamics simulation tool. TruckSim models the tractor and the trailer sprung masses as lumped rigid bodies with 6 d.o.f. for each. TruckSim represents each axle

suspension with 2 d.o.f. and each wheel with 1 spin d.o.f. The TruckSim program available for this work did not have the following d.o.f.: cab pitch and heave, seat heave, tractor and trailer frame bending, and engine heave. The developed 15 d.o.f. model has greater vertical ride dynamics fidelity compared to the TruckSim model due to the presence of tunable cab and seat suspensions as well as flexible tractor and trailer frames. To compare the two models, the tractor semi-trailer models were simulated traversing a small bump at 60 mph to mimic an impulse input to the systems. The comparison process was carried out by defining the same parameter values across the two models, and linearizing the TruckSim suspensions. Since TruckSim models the tractor sprung mass as a rigid lumped mass, the relative motion between the frame and the cab, and the cab and seat in the Matlab/Simulink model must be minimized. This was achieved by increasing the cab and seat stiffness in the Matlab/Simulink model by 5 orders of magnitude. Increasing the natural frequencies of the tractor and trailer frame beaming modes of the Matlab model to beyond 100 Hz effectively made the frames behave as rigid bodies in the frequency range of interest (<50 Hz). The developed 15 d.o.f. model and the TruckSim model were simulated over a small road bump at 60 mph. The study shows that the dynamic responses of the two models were indistinguishable. Detailed results of the comparative study can be found in [11].

3. ROAD PROFILE GENERATION

As described in [12], a road profile that is traversed at speed, V , can be characterized by its temporal power spectral density function,

$$S(\omega) = (2\pi \cdot V)^{N-1} \frac{C_{sp}}{\omega^N}, \quad (1)$$

where, $S(\omega)$ is the road PSD in $\text{m}^2/(\text{rad/s})$, V is the speed of the vehicle in m/s , ω is the temporal frequency in rad/s , and C_{sp} and N are constants. The PSDs of a few common road profiles as described by [12] are given in Table 2.

Table 2. Road Profiles from [12]

Road Type	C _{sp} (SI units)	N
Smooth Runway	4.3x10 ⁻¹¹	3.8
Rough Runway	8.1x10 ⁻⁶	2.1
Smooth Highway	4.8x10 ⁻⁷	2.1
Highway with Gravel	4.4x10 ⁻⁶	2.1

A program was developed in Matlab to generate a road profile that is used as an input for the deterministic truck ride model. The spatial characterization of the generated road profile yields the same power spectral density as the chosen road type [11,13]. For this study, two primary road profiles were created corresponding to the PSDs of the smooth highway and highway with gravel. These two road profiles were chosen to represent the extreme conditions observed during long hauls typical for trucks. The details of the procedure used to develop the road profiles are described in [11]. The PSDs of the two road profiles generated from the time histories are shown in Figure 2.

4. PERFORMANCE METRICS

The road profiles described in the previous section were used as inputs to the system. The time duration of each simulation run was 120 seconds. This duration allows sufficient road input power to be developed at low frequencies. The speed of the truck was maintained at 60 mph on smooth roads and 40 mph on gravel roads based on typical speed limits for US roads.

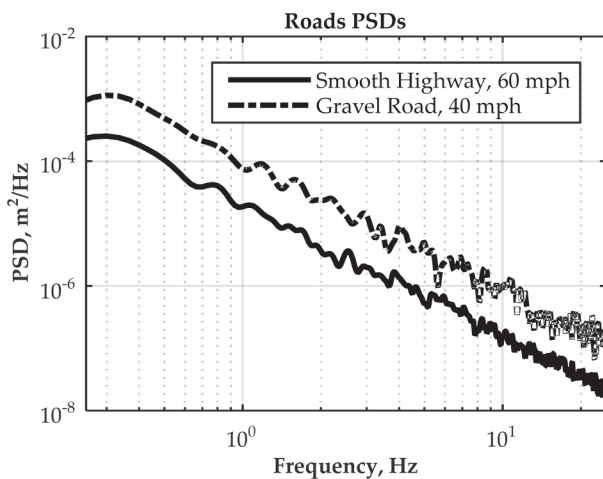


Figure 2. PSD of Generated Road Profiles

Frequency response analysis allows results to be easily compared to ride quality standards set forth by the International Standard Organization, ISO 2631. The ISO 2631 standard is still regarded as one of the leading standard for quantifying ride quality. The ride comfort boundaries are defined for root mean square (RMS) vertical and longitudinal accelerations measured at the driver's seat over the frequency range from 0.1 to 50 Hz. In ISO 2631:1974 [14], the boundaries in Figure 3 represent the amount of time the driver can sustain that particular acceleration before becoming uncomfortable. As one would expect, lower acceleration magnitudes can be tolerated as the driver operates the vehicle for longer periods of time.

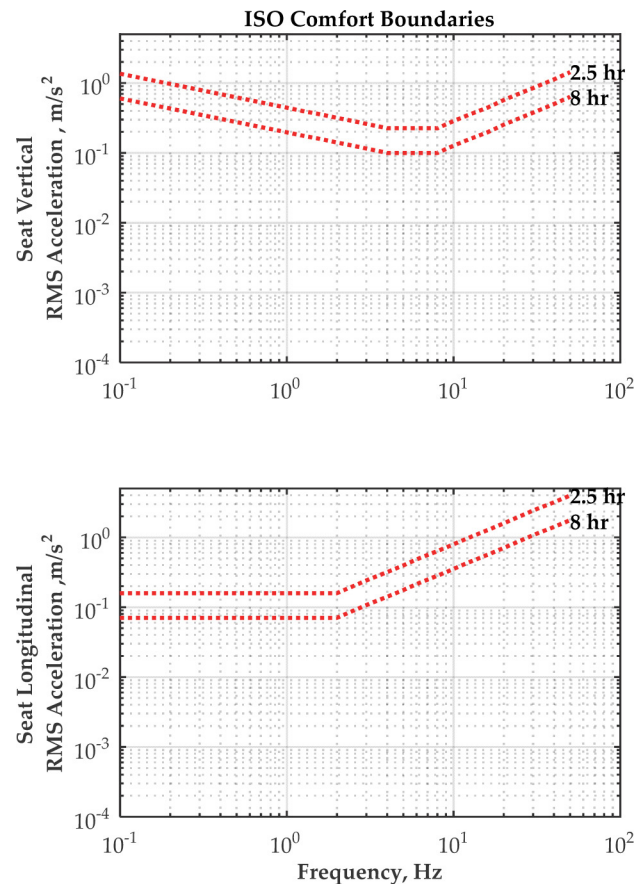


Figure 3. Vertical (top) and Longitudinal (bottom) ISO Comfort Boundaries [ISO 2631:1974]

The 1997 amended ISO 2631 dropped the dependency of the comfort criteria on time. ISO 2631:1997 [8] defines the calculation of a single weighted RMS acceleration number for measurement of ride comfort. For this study, the time dependent comfort boundaries are retained. Problem modes can be identified and appropriate remedies prescribed. As can be clearly seen from the ISO comfort boundaries, humans are most sensitive to vertical motion in the 4-8 Hz band and to longitudinal motion below 2 Hz. Eigenfrequencies of the modes of interest of the developed model are all substantially less than 50 Hz and the road excitation is very weak above 50 Hz. This ride analysis was accordingly limited to response of the vehicle to road excitations between 0.1 - 50 Hz.

The Matlab function *pwelch* was used to obtain the power spectral density of the responses from the time history results. ISO 2631 defines the low and high frequencies corresponding to the center frequency of each 1/3 octave band as,

$$\begin{aligned}\omega_{low} &= 0.89 * \omega_{center} \\ \omega_{high} &= 1.12 * \omega_{center}\end{aligned}\quad (2)$$

The center frequencies are defined by ISO 2631:1974. Based on ISO 2631 [8,14] the RMS of the vertical and longitudinal acceleration at the driver's seat are calculated from the respective PSDs ($S(\omega)$) of vertical and longitudinal acceleration for the various center frequencies using the following formula