THESIS FOR THE DEGREE OF DOCTOR OF PHILOSOPHY

IN

SOLID AND STRUCTURAL MECHANICS

Advances in Heavy Vehicle Dynamics with Focus on Engine Mounts and Individual Front Suspension

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Göteborg, Sweden 2012

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ISBN 978-91-7385-760-4

Doktorsavhandlingar vid Chalmers Tekniska Högskola
Ny serie nr 3441
ISSN 0346-718X

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Cover:
Volvo FH16 equipped with Individual Front Suspension (IFS), property of Volvo Truck Corporation

Chalmers Reproservice
Göteborg, Sweden 2012

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Abstract

The main objective of the research presented in this thesis is to enhance driver comfort and handling of a heavy truck that in turn leads to better vehicle safety and stability. The focus has been put on studying two suspension systems of the truck, engine and front axle suspensions, due to their significant impact on the dynamic performance of the vehicle.

At first, a computational model that evaluates the nonlinear behavior of dynamic stiffness and damping of the elastomeric engine mounts is developed and successfully verified and validated against measurement data. This model is then utilized to examine adaptronic mounting systems in which actuators, sensors and controller are incorporated in the passive engine suspension. Widely used conventional engine mounts in commercial vehicles cannot fulfill the conflicting requirements for the best isolation concerning both road and engine induced excitations. Hence, to improve the noise and vibration suppression it is necessary to go beyond passive isolators and use semi-active/active suspension. The results of simulations of engine vibration dynamics and transmitted forces to the vehicle structure have shown good potential for adaptronic engine mount setup.

Subsequently, the effects of front axle suspension design, i.e. Individual Front Suspension (IFS) as well as semi-active damper, on the comfort and handling characteristics of the vehicle are discussed. Employing the model of the tractor semitrailer combination, the study presents the results of comparison between the trucks with IFS and rigid front axle by assessing the responses of the vehicles to various road excitations and steering input. The obtained results show enhanced comfort and steering feeling for the truck with IFS. Moreover, the capabilities of a semi-active front axle suspension are investigated through control strategies such as skyhook, Linear Quadratic (LQ) and Model Predictive Control (MPC). The developed controllers are verified by simulation with respect to the identified road inputs and maneuvers. The outcome of the semi-active damper compared to that of the passive suspension that are presented quantitatively and qualitatively clearly show the great influence of the semi-active dampers on the vehicle dynamic performance. With continuous semi-active dampers accelerations in the cab are decreased particularly up to wheel hop frequency. Also, tire forces are either unchanged or improved. Hence, unlike passive suspension that is a compromise between ride comfort and handling, semi-active suspension facilitates enhancing both target criteria.

Keywords: heavy vehicle, individual front suspension, comfort, handling, kinematics, semi-active damper, engine mount, nonlinear dynamic stiffness and damping, vibration isolation, adaptronic mounting system

Preface

The research presented in this thesis was performed between August 2006 and November 2012 partly at the department of Applied Mechanics, Chalmers University of Technology and partly at Chassis and Vehicle Dynamics Engineering, Volvo Group Trucks Technology. The project was financed by VINNOVA, the Swedish Agency for Innovation Systems, and Volvo Truck Corporation that is gratefully acknowledged.

During the course of my PhD studies I have had the privilege of meeting many fantastic people whose support has made this work possible and whom I would like to acknowledge. Besides my time at the university doing research, taking courses, teaching and writing I worked at Volvo Group Trucks Technology. Spending more than half of my time at Volvo I have had the chance to collaborate with engineers dealing with real challenges and also to work on other subjects than my PhD project, which helped me gain a lot of experience.

First I would like to express my deep gratitude to my supervisor Professor Viktor Berbyuk for his guidance and always having time for discussions. I am very grateful to Inge Johansson, Erik Wikenhed, Peter Nilsson and Stefan Preijert in my steering committee and also Niklas Fröjd, Fredrik Öijer and Nicolas Dela from Volvo Group Trucks Technology for their valuable advice, ideas and fruitful discussions.

I am thankful to Anders Carlson, Mats Fagergren and my colleagues at Volvo Group Trucks Technology, especially Farid Zareiyan, for the inspiring discussions and their understanding. Furthermore, I acknowledge the support of my former colleagues Rickard Nilsson and Andreas Gustavsson when I first started working at Volvo. I would also like to thank everyone at the Division of Dynamics at Chalmers University of Technology for creating a friendly and pleasant working environment.

Last but not least, I would like to express my deepest gratitude to my family, Andreas, for his love and always being there for me, my parents and brothers, for their unconditional support and encouragements as well as, Anita and Leif, for their kindness.

Göteborg, November 2012
Hoda Yarmohamadi

List of Appended Papers

Paper A H. Yarmohamadi, V. Berbyuk, "Computational model of conventional engine mounts for commercial vehicles: validation and application", Vehicle System Dynamics, 49:5, 761-787 (2011).

Paper B H. Yarmohamadi, V. Berbyuk, "Vibration dynamics of a commercial vehicle engine suspended on adaptronic mounting system", Proceedings of the 9^{th} International Conference on Motion and Vibration Control (MOVIC2008), Munich, Germany, 15-18 September (2008).

Paper C H. Yarmohamadi, V. Berbyuk, "Kinematic analysis of a heavy truck with individual front suspension", Proceedings of the 22^{nd} International Symposium on Dynamics of Vehicles on Roads and Tracks (IAVSD2011), Manchester Metropolitan University, Manchester, UK, 14-19 August (2011).

Paper D H. Yarmohamadi, V. Berbyuk, "Comfort and handling of a commercial vehicle with individual front suspension", Proceedings of the of the ASME 2011 International Design Engineering Technical Conferences & Computer Information in Engineering Conference (IDETC/CIE 2011), Washington, DC, USA, 28-31 August (2011).

Paper E H. Yarmohamadi, V. Berbyuk, "Kinematic and dynamic analysis of a commercial vehicle with individual front suspension", Submitted for international publication (2012).

Paper F H. Yarmohamadi, V. Berbyuk, "Effect of semi-active front axle suspension design on vehicle comfort and road holding for a heavy truck", SAE Technical Paper 2012-01-1931 (2012).

Paper G H. Yarmohamadi, N. Landin, V. Berbyuk, "Improving comfort and handling of heavy vehicles with individual front suspension using semi-active dampers - an approach based on clipped LQ and model predictive control", Submitted for international publication (2012).

The appended papers have been planned and prepared in collaboration with the co-authors. The author of this thesis has performed model implementations, numerical simulations, analysis and writing of Papers A-F. Also, in Paper G about 60% of the progress including model implementations, simulations, analysis and writing has been conducted by the first author.

Contents

1 Introduction

Engine, chassis and cab of a heavy truck are susceptible to undesirable vibrations due to two sources of excitation: unbalanced forces in the reciprocating engine and disturbances from the road transmitted through the suspension system. Engine induced vibrations generally have frequency range of 30-250 Hz with amplitudes less than 0.3 mm. To isolate the transmitted forces from the engine to the vehicle structure, it is desirable to reduce the stiffness and damping of the engine mounts. Nevertheless, this reduction is restrained by the limitations on the relative motion of the engine to satisfy mechanical constraints and the fact that the engine mounting system should stand the weight of the engine. Furthermore, for road induced vibrations or vibrations from the engine at idle that typically have frequencies below 30 Hz and amplitudes greater than 0.3 mm, engine mounts should have high dynamic stiffness and damping.

As explained above, engine and road induced vibrations require conflicting engine mounts characteristics for their best isolation performance. These contradictory characteristic requirements of stiffness and damping, however, cannot be fulfilled by widely used conventional elastomeric engine mounts in commercial vehicles. Therefore, there is a need for performance improvement of engine mounting systems in heavy trucks that can be achieved by developing active or semi-active engine suspensions capable of addressing the conflicting requirements in different road and engine conditions. To improve existing and develop new advanced engine mounts, validated and verified computational models of mounts are well desirable. These models can especially be useful in analyzing the effects of frequency and amplitude of external excitation on dynamic stiffness and damping of a mount.

The other topic of the thesis concerns front axle design and its suspension properties. Since the front axle is located under the cab and it is integrated with the steering system, its design and suspension characteristics are very important for ride comfort as well as handling of the vehicle. In this work a commercial vehicle with Individual Front Suspension (IFS) is considered in which the left and right wheels are suspended independently using the double wishbone concept. This technology is adopted in heavy trucks because of the desire for improved handling and driver comfort that in turn leads to better vehicle safety and stability.

Moreover, front axle suspension damping in a heavy truck that is employed to control chassis motion and wheel load variations is in general provided by passive hydraulic dampers due to their low cost and simplicity. However, these dampers cannot fulfill the contradictory requirements for different driving scenarios. This results in a final damper setting that is a compromise among optimal settings for different load cases. To overcome this problem semi-active and active damping have been studied over the years, although semi-active dampers have attracted more interest compared to active ones due to lower cost and energy consumption.

1.1 Objective

The main goal of the research presented in this thesis is to enhance the performance of a heavy truck, in terms of comfort and handling, with focus on active engine suspension, front axle design and semi-active front axle damping. Therefore, the work comprises two main parts. The purpose of the first part is to develop mathematical and computational models of elastomeric engine mounts for heavy trucks, investigate the nonlinear behavior of dynamic stiffness and damping of the mounts and validate the models against experimental data. Subsequently, use the developed models to design an adaptronic engine suspension, which refers to a mounting system where sensor(s), force generating actuator(s) and controller are added to to the passive suspension, and at last analyze it compared to the conventional passive engine mounting system. The adaptronic suspension can be either semi-active or active depending on the applied constraints on the actuator force. Here the only restriction on the force is on its magnitude implying that the considered adaptronic engine suspension is active. In the second part, the intent is to study the effects of IFS as well as semi-active front axle damper on ride comfort and handling characteristics of a truck by simulations of a complete vehicle model of a 4×2 tractor semitrailer combination using realistic road inputs and steering maneuvers. Furthermore, for the semi-active damper hybrid skyhook/groundhook, clipped Linear Quadratic (LQ) and Model Predictive Control (MPC) algorithms are developed and compared to the passive suspension system.

1.2 Limitations of Scope

In this thesis active engine suspension has not been evaluated together with IFS or semi-active front axle damper in the complete vehicle model. In addition, the locations of the actuators have not been optimized in the active engine suspension and semi-active front axle damper dynamics has not been modeled. Finally, driver model has not been considered and the driver role is restricted to steering input and braking moment in the performed simulations.

1.3 Thesis Outline

In addition to the appended papers, Papers A-G, an extended summary of the research work is included. Chapter 2 presents the background on heavy vehicle engine and front axle suspensions together with the relevant existing literature. In Chapter 3 modeling of engine mounts, model parameter identification, model validation and finally adaptronic engine mounting system are described. Chapter 4 demonstrates a summary of IFS modeling, kinematics and dynamics by means of simulation. Subsequently, semi-active front axle damping together with the developed control strategies are discussed in Chapter 5, and their effectiveness are analyzed. Lastly, Chapters 6 and 7 summarize the scientific contributions and ideas for future work, respectively.

2 Background

As stated in the previous chapter engine, chassis and cab of the vehicle are subject to undesirable vibrations from road disturbances and engine excitations. Thus a number of suspension systems are used to attenuate vibrations, as depicted in Figure 1.

In this chapter, a review of the existing literature on modeling and analysis of passive, semi-active and active engine mounting systems for heavy trucks, IFS, semi-active front axle damping and its controller design are presented.

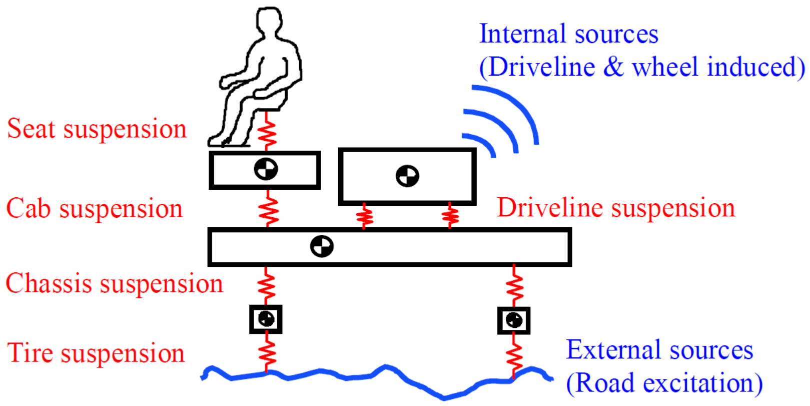


Figure 1. Sketch of the main suspension systems of a heavy truck [1].

2.1 Engine Suspension

The vehicle engine mounting system is a vibration control system comprising several mounts that connects the engine to the vehicle structure. This vibration control system can be passive (elastomeric or hydraulic), semi-active or active as has been described in review papers [2–4], which summarize features and objectives of different mounting systems.

2.1.1 Model of the Passive Engine Mounts

Kelvin-Voigt model that consists of a spring coupled in parallel with a viscous damper, illustrated in Figure 2, has a wide usage in modeling rubber isolators and components as engine mounts. This model is the most encountered model when considering rubber components as stated by [5]. Adopted by [6–9], Kelvin-Voigt model represents the engine mounts while optimizing the layout and parameters of the passive elastomeric engine mounting system. However, this model does not reflect the nonlinear behavior of the mount, for example due to friction phenomenon. With constant coefficients of stiffness and damping for the spring and damper elements, not only it does not show the amplitude dependency of stiffness and damping of the mount but also overestimates both of them for higher frequencies [10]. In vehicle modeling, Kelvin-Voigt model with a lookup table for the stiffness and damping coefficients can be used to model the mounts. Nevertheless, the frequency dependency of the mount characteristics cannot be taken care of in transient response.

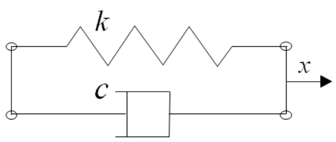


Figure 2. Presentation of the Kelvin-Voigt model.

Modeling of amplitude dependencies of rubber behavior is as important as modeling the frequency dependencies. Earlier, nonlinear models for rubber isolators that consider changes of stiffness and damping based on both frequency and amplitude of excitation have been studied. Kelvin-Voigt model, for which the spring and damper coefficients are updated in every cycle has been studied by [11]. In another study it is suggested to use the Kelvin-Voigt model with nonlinear expressions for stiffness and damping coefficients [12], where stiffness and damping are calculated for harmonic input excitations.

Furthermore, [13, 14] propose a nonlinear model by adding frequency dependencies to stick-slip friction components using integer derivatives. A smooth rubber friction behavior model as a function of excitation amplitude is introduced by [15,16] in a nonlinear model of rubber springs that is used in dynamic analysis of rail vehicles. This friction model shows better similarities to rubber friction behavior. The latter friction model is adopted by [10,17] to present a nonlinear model of rubber isolators by adding the friction model to a fractional Kelvin-Voigt model. In another study [18] a coupled mount model based on Bouc-Wen hysteresis model is proposed and analyzed in software package Adams. Also, a general methodology for the characterization of the bushing joints used in the models of road and rail vehicles is developed by [19] using Abaqus.

In light of the studies on engine mount modeling, a nonlinear model of the mount is developed to analyze the dynamics of the engine suspension in the presented research. The developed model expresses the nonlinear behavior of the stiffness and damping of a commercial vehicle engine mount as a function of both frequency and amplitude of excitation while keeping the number of model parameters low.

2.1.2 Semi-Active and Active Mounting Systems

Schematic models of passive and semi-active/active engine mounts are shown in Figure 3. Several studies have addressed optimization of passive engine suspension characteristics to achieve a better performance, for instance [20–23]. Nonetheless, [24] optimizes the parameters of a passive mount in frequency and time domain and shows that no passive mount is adequate to perfectly deal with all applications and isolation criteria. Thus, to enhance the behavior of the mounts further there is a necessity to go beyond the conventional mounting systems and consider semi-active and active engine mounts. Most of the conducted researches concern passenger cars and actually not so many studies related to commercial vehicles exist to the author's knowledge. This is partly due to the fact that hydraulic engine mounts are mostly the subject of the semi-active and active engine mounting design, since stiffness and damping of the mount can be controlled by the movement of the fluid; however these mounts require a significantly larger space for heavy and large truck engines. A hydraulic active engine mount for commercial vehicles is studied by [25], using which the transmitted force is reduced for low frequencies of 20-30 Hz. This mount consists of main rubber, two fluid chambers, inertia track, decoupler, an electromagnetic actuator and an adaptive controller. Also, an adaptive hydraulic engine mount that is tuned to road and engine conditions by changing the length of the inertia track and effective decoupler area is proposed in [26]. An adaptronic hydraulic engine mount for a variable displacement engine is proposed and analyzed in [27]. In this design a magnetic actuator is used to create mechanical pulses. Moreover, [28] presents a new active control engine mount that uses adaptive control to improve the quietness of diesel engine vehicles. This active mount has been constructed by incorporating an electromagnetic actuator and a load sensor in a fluid-filled engine mount. An active engine mounting system comprising a pair of electromagnetic actuators and hydraulic mounts is studied in [29]. More active hydraulic engine mounting systems are discussed in [30–32].

In addition to the above mentioned engine mounting systems, smart material based engine suspensions have been the aim of the research for passenger cars. Performance of a Magneto-Rheological (MR) mount has been evaluated in [33] where the mount incorporates MR fluid in a conventional hydraulic mount. In another study by [34] a semi-active Electro-Rheological (ER) engine mounting system is developed for passenger

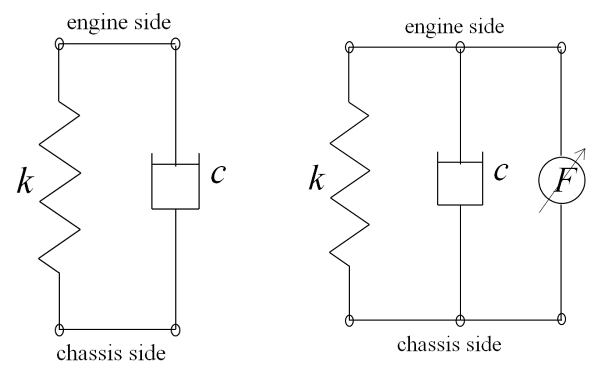


Figure 3. Schematic models of passive and semi-active/active mounts, respectively.

cars with a skyhook controller that is implemented through a Hardware-in-the-Loop (HIL) simulation. Also, [35] proposes a ER active engine mount.

2.2 Front Axle Suspension

Front axle is located under the cab and includes the steering system of the vehicle, therefore, its geometry and suspension property have a great influence on the ride comfort and handling of the vehicle through reducing the transmitted road disturbances to chassis and dynamic tire force variations, respectively.

Concise surveys of the available research on road vehicle suspension design and dynamics are described in [36–39], for instance. Passive suspension system designs and their effects on road vehicle dynamics and stability are presented in terms of in-plane and full-vehicle arrangements, in the latter.

2.2.1 Independent Front Suspension

Independent suspension is a well-proven concept for passenger cars unlike heavy trucks. Truck manufacturers have investigated IFS because of the demands for better ride comfort and vehicle handling that in turn leads to better vehicle safety and stability. However, there is a lack of published literature in this area. Independent suspension designs for heavy trucks have been reviewed in [40, 41]. The latter studies four different concepts and concludes double wishbone configuration has the best performance and possibilities. Additionally, [42] investigates IFS design as well as ride comfort analysis by means of a multi-body model in Matlab/SimMechanics. Ride comfort and dynamic wheel load sensitivity to front axle spring stiffness, damping and unsprung mass are also conducted.

2.2.2 Semi-Active Axle Suspension

Semi-active damper characteristics can be theoretically shown in Figure 4 compared to passive and active dampers. Here the damper force is given as a function of the velocity over the damper.

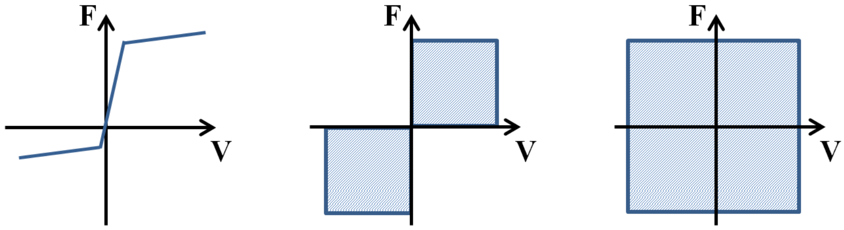


Figure 4. Schematic of passive, theoretical semi-active and theoretical active dampers, from left to right.

Foregoing literature regarding semi-active dampers on heavy trucks has been studied, in which various types of models with different complexities, objectives and control strategies have been discussed. For instance quarter vehicle, half vehicle and three Dimensional (3D) full vehicle models have been employed in the existing studies as ride comfort, road holding and handling of the vehicle have been the main design objectives. Moreover, for the controller design skyhook, fuzzy, LQ, robust H_2/H_∞ and MPC methods have been adopted. Various principles of variable dampers are discussed in [43].

Skyhook concept introduced by [44] has been studied in semi-active suspension design for ground vehicles in many years. [45] reviews studies in which skyhook control policy, the most widely used control law for semi-active dampers in vehicles, and some of its variations are described. The important factor according to [38] is that skyhook is a model free algorithm. Performance of five different skyhook control methods have been studied by [46] experimentally in a quarter car rig. More work on skyhook semi-active dampers can be found for example in [47–49].

In addition, [50] examines the outcome of clipped optimal and skyhook control strategies for semi-active vehicle suspensions based on a genetic algorithm and a quarter car model. Also, [51–53] evaluate semi-active LQ dampers using a quarter car model.

Controllers via MPC technique using a half vehicle model are developed by [54,55] in order to improve the suspension behavior. The results from this control method shows a better performance compared to skyhook and clipped LQ methods. Also, a MPC controller is designed by [56] and compared to clipped LQ control strategy through simulating a quarter car model with stochastic random road disturbances and shock inputs.

Furthermore, semi-active damping by means of fuzzy control approach have been studied by [57–59] for heavy truck applications. Finally, in studies by [60, 61] semi-active H_2/H_∞ control laws have been considered. In the latter, a seven Degree of Freedom (DoF) 3D model has been applied to compare the response of H_2/H_∞ controller to both passive and semi-active skyhook suspensions. The article has concluded that the proposed controller enhances sprung body accelerations and tire forces.

2.2.3 Performance Evaluations

To examine the primary suspension system of a heavy truck, first of all an analysis of the kinematic characteristics is performed. This is followed by an assessment of the dynamics of the vehicle, i.e. ride comfort and handling, employing a complete vehicle model. Although measurements of the vehicle under real driving conditions might be the most accurate way of determining these objectives, it is very time consuming and expensive. Thus, computer simulations are carried out in order to give an estimate of the performance in the development phase.

Suspension Kinematics

Several properties such as vertical stiffness, roll stiffness, toe variation, camber variation, track width change, roll steer and brake steer can be investigated in a kinematic analysis for the axle assembly. For example, [62–64] have analyzed some of these properties in various ways. Vertical stiffness, roll stiffness and roll steer of heavy truck suspensions have been estimated experimentally by [62] through applying vertical forces at axle ends. Vertical load and translational displacements at the axle end have been measured to establish the suspension kinematics. In the work by [63] kinematic behavior of a McPherson-type suspension including camber, caster and steer angles that influence the handling of the vehicle has been inspected using a three dimensional model with two inputs: travel of the strut and steering input. Furthermore, [64] proposes a procedure for the optimal dimensional synthesis of double wishbone suspension systems by monitoring features as camber, toe, kingpin and caster angles using fourteen design parameters including link dimensions, ground and moving joint positions and axle orientation.

Comfort and Handling

Road roughness is classified as the primary input of interest when designing suspension systems for ride performance in [36]. Also, ride jumps have been examined in [37] using mathematical models. Effects of suspension friction, road roughness and tire/wheel unevenness on vehicle ride are studied by [65, 66].

International Organization of Standardization (ISO) offers road roughness classification (Classes A-H) in [67] from spectral densities by assuming a constant velocity Power Spectral Density (PSD). Moreover, in a research by [68] statistical modeling of measured road signals is addressed. To estimate ride comfort for various road roughnesses another standard [69], which concerns human exposure to whole-body vibration with respect to the comfort and annoyance of the occupants, can be applied. It specifies frequency based weighting for accelerations in the cab with regard to human sensitivity to different frequency ranges.

Handling in terms of road holding characteristics examined through analyzing dynamic tire forces, is widely used by the researchers. This is a result of utilizing two dimensional (2D) quarter car or pitch plane models. However, there are studies in which a 3D vehicle model is developed to study handling based on the steering behavior. For example, [70] employs a full vehicle model of a heavy truck to analyze vehicle handling in terms of understeering and roll over threshold. Understeering is investigated by simulating the vehicle in a curve and monitoring the steering angle and lateral acceleration.

3 Engine Mounting System

As already mentioned, conventional rubber-metal mounts are commonly used in heavy vehicle engine mounting systems. In vehicle models, these engine mounts are usually represented by Kelvin-Voigt model where a spring is coupled in parallel with a viscous damper. This model does not take friction into account and evaluates the stiffness and damping of the mounts with respect to the frequency of excitation input but not its amplitude. To express the stiffness and damping of the mount as a function of both frequency and amplitude of excitation a computational model is developed. In this chapter, conventional mount modeling and validation and adaptronic engine suspension are covered.

3.1 Modeling

To mathematically model a conventional mount, functional components representing elastic, viscous and friction behaviors are put together in parallel as shown in Figure 5. The elastic component is a linear spring, the viscous component consists of two parallel sets of series linear spring and damper and lastly the friction component is represented by a nonlinear Coloumb type friction model grasping the essential behavior of hysteresis loops in force-displacement curves [15, 16]. This friction behavior is defined by two parameters: maximum developed friction force F_{fmax} and the displacement needed to develop half the maximum friction force x_2.

The plots of elastic, viscous and friction forces for different amplitudes and frequencies of harmonic excitation are shown in Figure 6. Elastic and friction forces are not dependent on the frequency of excitation.

The developed model has seven parameters, five parameters for the elastic and viscous functional components as illustrated in Figure 5 and two parameters for the friction functional component as mentioned above. Model parameters can be retrieved from measurement data in different ways. Here, the parameters have been identified through optimization as will be described later on.

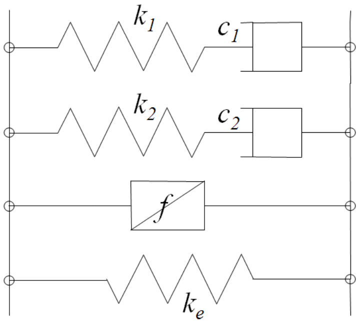
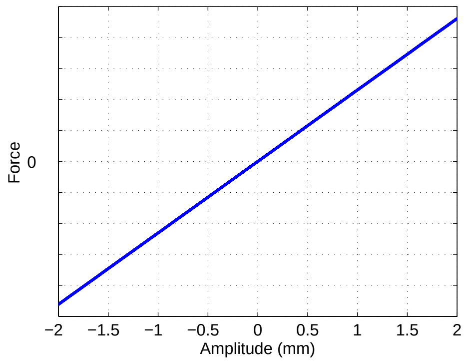
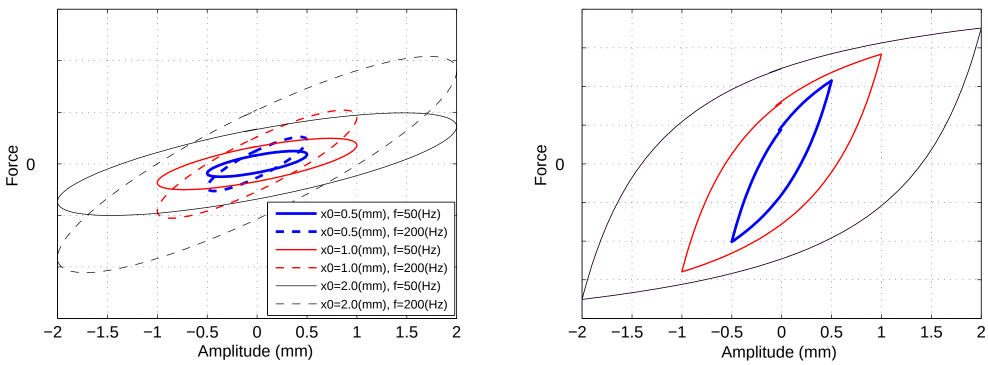


Figure 5. Sketch of the conventional mount model.



(a) Elastic component



(b) Viscous component (c) Friction component

Figure 6. Force displacement plots for different frequencies of harmonic excitation.

The model is one dimensional where the relation between force and displacement is based on the superposition of elastic, friction and viscous forces (F_e, F_f and F_v, respectively):

$$F = F_e + F_f + F_v. \tag{1}$$

For a harmonic excitation input the measures of stiffness, S, and damping, D, which have been used through the work are given by

$$S = \frac{F_0}{x_0}, \qquad D = \frac{E}{F_0 \ x_0}, \tag{2}$$

where, F_0 is the the steady state total force amplitude, x_0 is the amplitude of the harmonic excitation and E is the total energy loss per cycle. Damping measure is non-dimensional and shows how much energy has been dissipated. The higher the value D is, the higher is the energy loss in the mount. In Equation (2), elastic, viscous and friction functional components have uncoupled contributions to the results. Elastic component, only affects the stiffness value as an ordinary elastic stiffness. Viscous part takes care of frequency dependency of the stiffness and damping that is independent of the amplitude and the amplitude dependency is taken care of by the friction part that is independent of frequency.

3.1.1 Model Parameter Identification

Model parameters are attained according to measurement data through optimization with Least Mean Square (LMS) algorithm (Paper A). The available measurement data is the dynamic force that is measured directly with sensors for various amplitudes and frequencies of harmonic excitation in the range of 0.025-2 mm and 5-100 Hz, respectively. The energy loss per cycle is evaluated from the dynamic force measurements and the values of the measured stiffness and damping are calculated from Equation (2).

To conduct this optimization Matlab subroutine $fmincon$ is used. The constraints are lower and upper bounds of the model parameters and the cost function used for the optimization is the summation of the squared errors between the measured and estimated values of the stiffness and damping as written in Equation (3). C_S and C_D are weighting parameters. N_S and N_D are the number of the stiffness and damping values available from measurements for different frequencies and amplitudes of excitation, respectively.

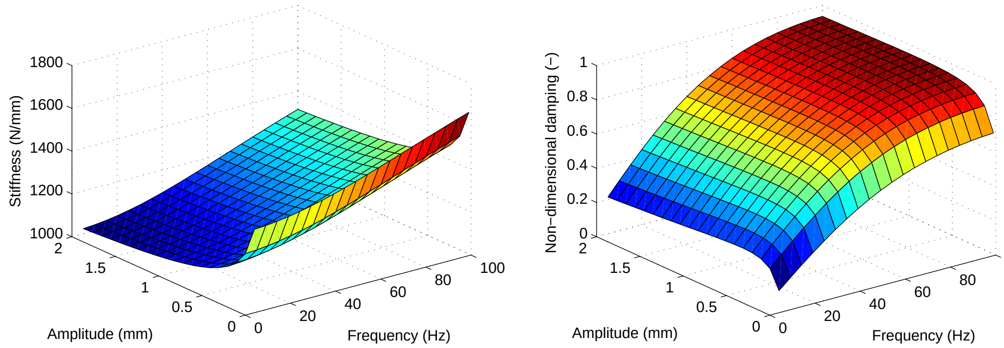
$$\text{func} = \sum_{i=1}^{N_S} C_S (S_{measured(i)} - S_{simulated(i)})^2 + \sum_{j=1}^{N_D} C_D (D_{measured(j)} - D_{simulated(j)})^2. \qquad (3)$$

Since values of the damping are very small compared to stiffness, weighting parameters are needed to compensate for damping optimization. Otherwise, model parameters are identified only for optimized stiffness values. Weighting parameter C_S is set to 1 and C_D is set to 10^7. Smaller or bigger weighting parameter for damping can be used according to the desired error tolerances for stiffness and damping.

3.1.2 Model Validation

The developed computational model is first implemented in Matlab. Measurement data for three different front and rear engine mounts are used to achieve seven model parameters. Since the model is one dimensional, three sets of parameters are obtained for each mount defining the nonlinear behavior of the mount in longitudinal, x, lateral, y, and vertical, z, directions.

Figure 7 shows 3D plots of the stiffness and damping of the mounts with respect to amplitude and frequency of harmonic excitation. The different mounts are referred to with Mount 1, Mount 2 and Mount 3 and also the directions in which the stiffness and damping are presented for are named with Direction 1, Direction 2 and Direction 3 due to confidentiality reasons[1].

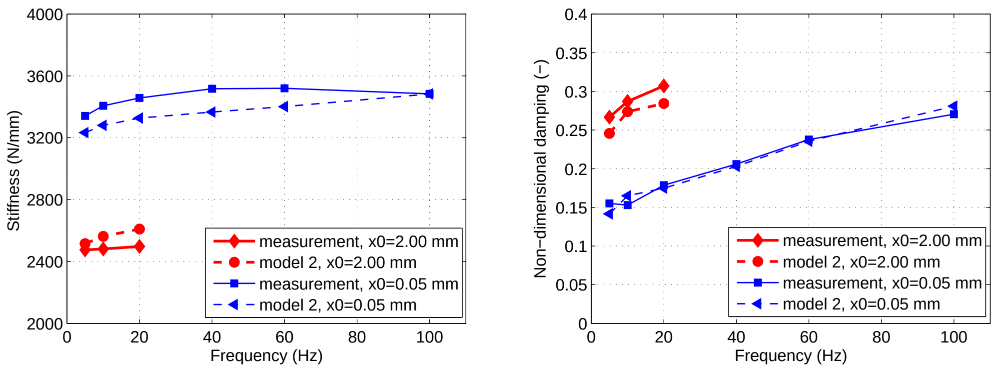


(a) Stiffness (b) Non-dimensional damping

Figure 7. 3D plots of stiffness and damping of Mount 2 along Direction 3.

Analysis of the plots in Figure 7 shows that for this mount, stiffness decreases up to 6% and increases up to 28% with increase in amplitude and frequency, respectively. Non-dimensional damping rises up to 79% and 489% with amplitude and frequency increase. If we consider all three engine mounts, stiffness and damping show similar behavior with changes in frequency and amplitude of harmonic excitation. Comprehensive qualitative and quantitative results are shown in Paper A.

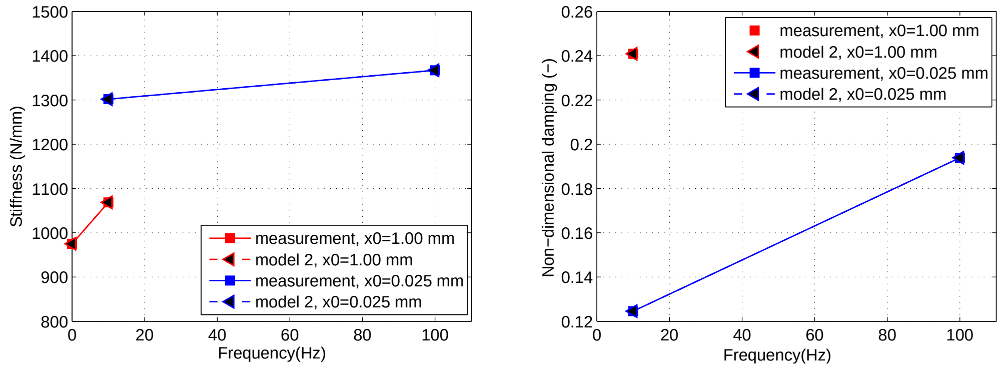
In order to validate the computational model, the stiffness and damping output from the model is compared to the measurement data. Here some plots of comparison are presented in which the solid lines represent measurement data and dashed lines are simulation results obtained by the computational model (see Figures 8 and 9).



(a) Stiffness (b) Non-dimensional damping

Figure 8. Stiffness and damping of Mount 1 in Direction 1 for two different amplitudes.

[1]For more information about the investigated mounts contact Volvo Group Trucks Technology, Göteborg.



(a) Stiffness (b) Non-dimensional damping

Figure 9. Stiffness and damping of Mount 3 in Direction 1 for two different amplitudes.

According to Figures 8 and 9, the simulated stiffness and damping have similar behavior with amplitude and frequency changes. In Figure 8 the model gives higher stiffness for higher frequencies; however, the measurements show a decrease after some frequency. Maximum error between measured and simulated data for each available data point is 8% and 20% for stiffness and damping, respectively. To reduce the error tolerance for damping, frequency based decomposition of the model has been studied. This means finding two sets of model parameters in each direction for the mount: one for low frequencies, 5-25 Hz, and one for higher frequencies, 30-100 Hz. The results of the frequency based decomposition reduces the error tolerance to 11%.

3.1.3 Implementation in Nastran

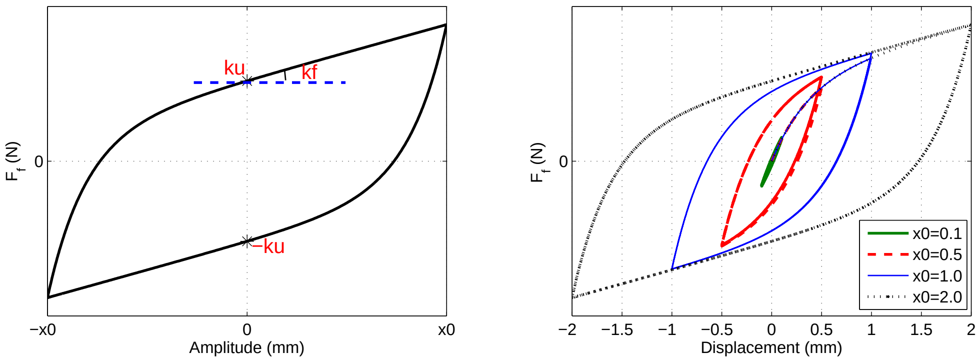
It is desirable to implement the developed computational model in complete vehicle model in Nastran. Elastic and viscous behavior models can be implemented in Nastran using the available spring and damper elements. Nevertheless, the friction model used in Matlab cannot be implemented in Nastran since the future values of the amplitude of excitation is not accessible. Therefore, the used friction component is replaced by another friction functional component named the leaf spring friction model [71]. For a harmonic excitation of $x = x_0\ sin(\omega t)$, the leaf spring friction model is defined as

$$\begin{aligned} F_1(i) &= k_f x(i) + k_u \qquad \text{if} \quad x(i) \geq x(i-1), \\ F_1(i) &= k_f x(i) - k_u \qquad \text{if} \quad x(i) < x(i-1), \end{aligned} \tag{4}$$

$$F_f(i) = F_1(i) + (F_f(i-1) - F_1(i))\ e^{\frac{-|x(i)-x(i-1)|}{\beta}}, \tag{5}$$

where F_f is the friction force, x is the displacement and k_f and k_u are stiffness coefficients illustrated in Figure 10(a). Additionally, Figure 10(b) shows how the friction force changes with amplitude of harmonic excitation input. Similar to the previously used friction functional component, the leaf spring friction model is independent of the frequency of excitations.

The leaf spring friction model has three parameters: k_f, k_u and β. Therefore, the



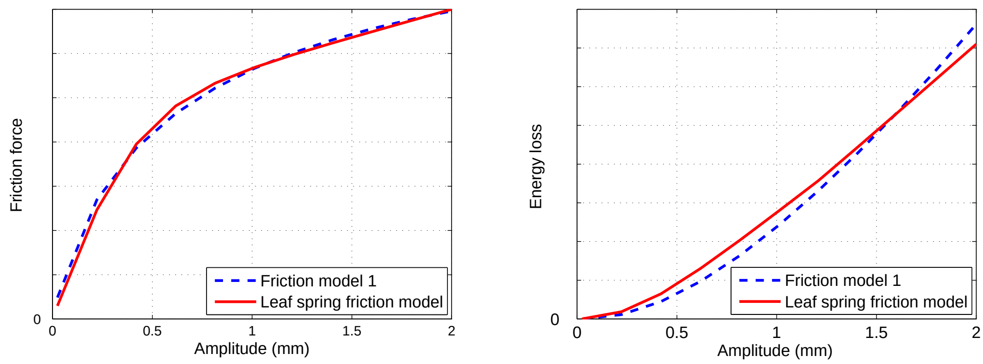
(a) Stiffness parameter illustration

(b) Leaf spring friction force for different amplitudes

Figure 10. Leaf spring friction model.

computational model comprising elastic, viscous and leaf spring functional components has eight parameters which are identified through optimization with LMS algorithm according to the measurement data in Matlab. First an auxiliary optimization problem is defined to determine the three parameters of the leaf spring friction model to get the same response as the previously used friction model. The output of this auxiliary optimization problem together with the known parameters of the elastic and viscous parts represent the initial guess for the optimization process of the computational model with leaf spring friction model.

Figure 11 depicts the outcome of the auxiliary optimization problem in terms of force magnitude and energy loss per cycle.



(a) Friction force magnitude

(b) Energy loss per cycle

Figure 11. Leaf spring friction model.

3.1.4 Kelvin-Voigt Model versus the Developed Model

Characteristics of the engine mounts calculated from Kelvin-Voigt model, with constant spring and damper coefficients, and the developed computational model are compared.

For a mount represented with Kelvin-Voigt model with a spring coefficient k and damper coefficient c, the stiffness and damping are calculated for harmonic input $x = x_0\ sin(\omega t)$. The system is one dimensional and the relation between force and displacement is

$$F = k\ x + c\ \dot{x}. \tag{6}$$

The steady state force amplitude can be written as [72]

$$F_0 = x_0\ \sqrt{k^2 + c^2\omega^2}, \tag{7}$$

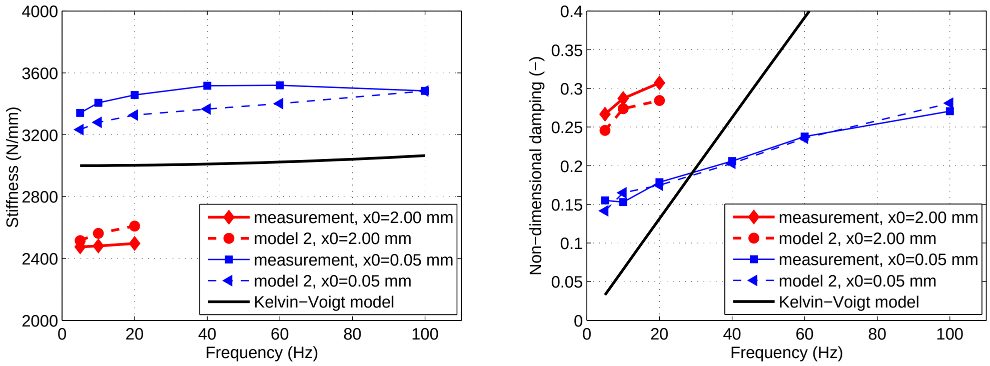
and the energy loss per cycle can be given by

$$E = \pi {x_0}^2 c\ \omega. \tag{8}$$

From Equations (2), (7) and (8), the stiffness and damping are

$$S = \sqrt{k^2 + c^2\omega^2}, \qquad D = \frac{\pi c\ \omega}{\sqrt{k^2 + c^2\omega^2}}. \tag{9}$$

Stiffness and damping are only functions of frequency of excitation as can be seen in Equation (9). To visualize the difference between the computational model of the mount and the Kelvin-Voigt model, plots of stiffness and damping for mount 1 in direction 1 calculated with Kelvin-Voigt model is added to Figure 8, see Figure 12. For the Kelvin-Voigt model, $k = 3000$ N/mm and $c = 1$ Ns/mm are used. Higher values of c will make the damping grow even faster with the increase in frequency.



(a) Stiffness (b) Non-dimensional damping

Figure 12. Comparison between Kelvin-Voigt model and the developed model.

3.2 Adaptronic Engine Mounting System

Semi-active and active systems are solutions for the need of performance improvement of engine suspension since they are capable of addressing the conflicting requirements in different road and engine conditions.

Two approaches have been considered for active engine mounting design. First approach is mount with active structure and smart materials that replaces the passive system. Second approach that has been chosen and studied, is adaptronic mounting system comprising conventional elastomeric mounts as well as additional actuators incorporated into the mounting system. The general topology of the adaptronic mounting system is shown in Figure 13 in which the plant represents the passive engine suspension. The controller sends commands to the force generating actuator based on the inputs from the sensor(s).

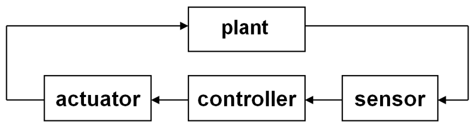


Figure 13. General topology of the adaptronic mounting system.

The advantages of adaptronic mounting system is improving the characteristics of the mounting system while keeping the original design of the conventional mounts unchanged. The disadvantages are increase in cost and space requirement.

To create an adaptronic engine suspension, two actuators are added to the conventional engine mounting system comprising three mounts (one front mount in the middle and two rear mounts on the sides). Since no special kind of actuator is chosen prior to the analysis of the results, a general nonlinear model with expressions of elastic and viscous components (see Equation (10)) is used for actuator dynamics. The maximum exerted force by the actuators is set to be 500 N, which approximately corresponds to 2.7% of the engine weight.

$$F_a = ax + bx^2 + c\dot{x} + d\dot{x}^2, \tag{10}$$

where, F_a is the actuator force, x and $\dot{x}$ are the displacement and velocity over the actuator. a, b, c and d are parameters that are evaluated during engine vibration dynamics analysis to minimize the cost function. The cost function for the engine vibration dynamics is the transmitted forces to chassis.

The actuator force can be implemented in both Matlab/Simulink and Nastran to analyze engine vibrations suspended on an adaptronic mounting system.

3.3 Simulation and Analysis

Vibration dynamics of engine suspended on conventional and adaptronic mounting systems is analyzed. Conventional engine mounts are modeled with both traditional Kelvin-Voigt model and the developed computational model. In addition, adaptronic engine mounting system is modeled with the developed computational model and general dynamics of actuator force. The intent is to investigate the transmitted forces to the vehicle structure under different engine excitations and realistic road inputs for both conventional and adaptronic mounting systems.

A 3D multi-body model of the engine suspended by three mounts on the vehicle chassis is created in Matlab/Simulink. The engine is modeled with a rigid body of mass $M_e = 1900$ kg, see Figure 14. $O_1x_1y_1z_1$ represents the global reference frame and $Oxyz$ is the body fixed reference frame with its origin at engine center of gravity. The x-axis is taken parallel to the crank shaft and z-axis is in the vertical direction when in static position. The engine equations of motion are written using the Newton-Euler approach [73], which are given in more detail in Paper B.

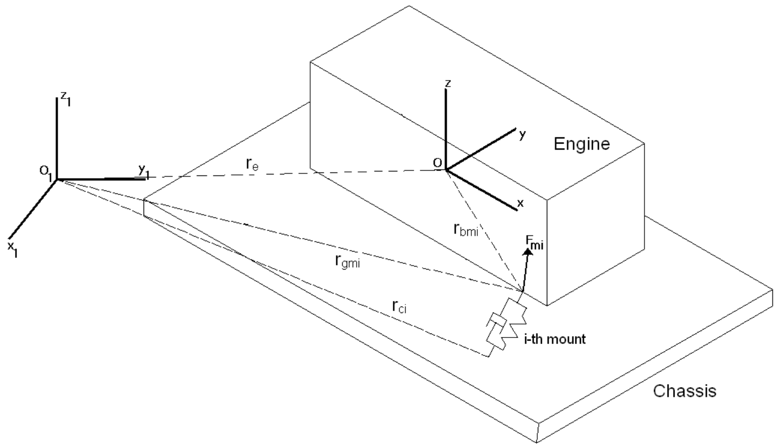


Figure 14. Sketch of the engine suspended on mounting system [74].

Transmitted force to chassis in each direction (x, y, z) is calculated as the sum of the forces transmitted to the frame through engine mounts in that direction. The total transmitted force, F_T, is the magnitude of the resulting force at each instant of time as stated in Equation (11).

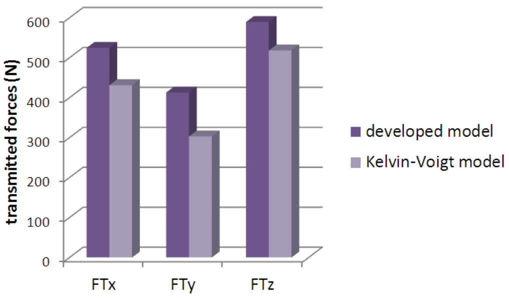
$$F_T = \sqrt{{F_{Tx}}^2 + {F_{Ty}}^2 + {F_{Tz}}^2}. \tag{11}$$

The model is subjected to inputs from road surface irregularities as well as engine internal processes simultaneously. The latter is represented by small amplitude, high frequency excitation while road roughness is modeled with realistic road data, smooth and rough roads, from Volvo Group Trucks Technology, Göteborg. The resultant displacement of the chassis at each mount connection point to chassis, is applied to the mounts in the considered 3D multi-body model.

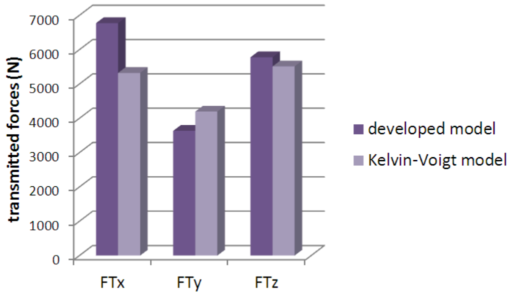
The outputs of the model are translational and rotational accelerations, velocities and displacements of the engine, as well as the transmitted forces to chassis, engine mount deflections and forces.

3.4 Comparison of the Results

First, the results of engine vibration dynamics for conventional mounting system using Kelvin-Voigt model and the developed computational model are studied. Considering the results from smooth road excitation input, both models of the mount show almost the same Root Mean Square (RMS) and Max value of the displacement of the center of mass (CM) of the engine; however, the acceleration is always higher when the developed model is used. The engine CM acceleration is higher up to 35% and 3.5% in RMS and Max value for the developed model, respectively. Additionally, the developed model shows higher estimation of the transmitted forces to the vehicle structure than the Kelvin-Voigt model by 20% in RMS and 3.6% in Max value when studying the total transmitted forces. As an example, the transmitted forces to chassis evaluated with the developed model are compared to the ones from the Kelvin-Voigt model in Figure 15 in each direction. Lastly, the results of engine vibration dynamics for conventional and adaptronic mounting



(a) RMS value on smooth road



(b) Max value on rough road

Figure 15. Effect of the used model on transmitted forces to chassis.

system are compared together where the mounts are represented with the developed computational model. Engine displacement and acceleration is not affected much with the adaptronic mounting system. Most probably this is due to the fact that the actuator forces are small compared to the weight of the engine. Nevertheless, the maximum value of the total transmitted forces to the chassis has decreased by 24% and 2% while the RMS value of the transmitted forces has decreased by 14% and 6% for good and bad road inputs, respectively. Figure 16 depicts the effect of adaptronic suspension on the total transmitted forces to chassis compared to the passive suspension system. In this plot the values are normalized separately in each case with regard to the passive mounting system values. Papers A and B offer more results and information in this context.

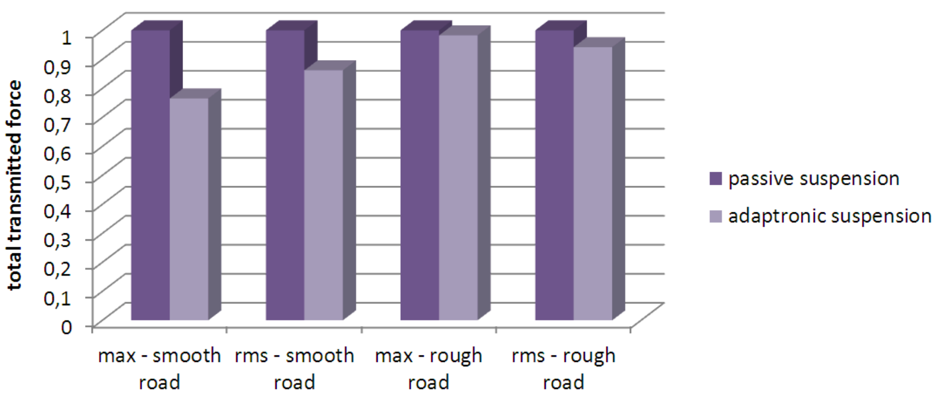


Figure 16. Effect of adaptronic suspension on total transmitted forces to chassis.

4 Individual Front Suspension

IFS is developed and modeled according to the double wishbone concept illustrated in Figure 17. It should be noted the steering system is based on the rack and pinion arrangement as in passenger cars instead of the commonly used steering linkage in heavy trucks.

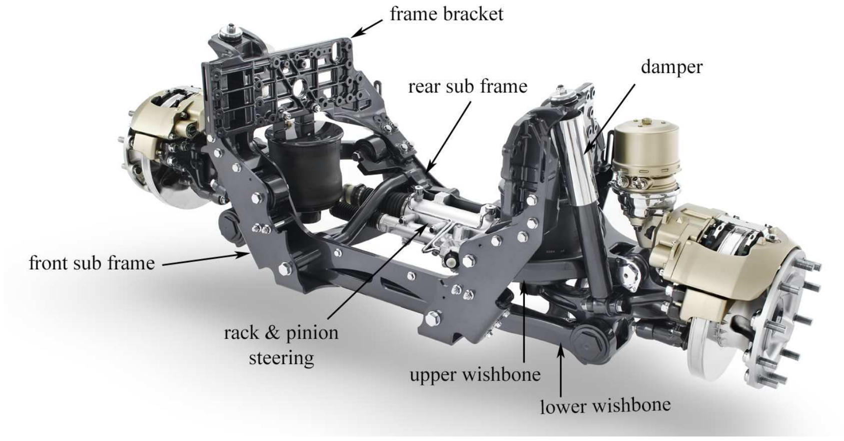


Figure 17. Double wishbone concept of the IFS (a property of Volvo Truck Corporation).

The model of the IFS is created using shell and beam elements that connect to each other with different applicable joints and constraints. Lower and upper control arms are connected to two sub frames, front and rear that are placed between the left and right frame rails to increase the stiffness of the IFS especially in the lateral direction, with linear bushings created via spring elements. Finally, the sub frames attach to a bracket, which connects to the chassis through several bolts.

4.1 IFS Kinematics

To investigate the kinematic aspects of the IFS in heavy vehicles the developed IFS, explained above, is considered with the front part of the chassis and the steering system.

The utilized length of the flexible truck chassis, represented by shell elements, is almost 3 m since only the front axle is examined in the study. Boundary conditions are applied on the truck frame so that the front part is fixed in vertical direction and the rear part is fixed in all translational directions, as shown in Figure 18.

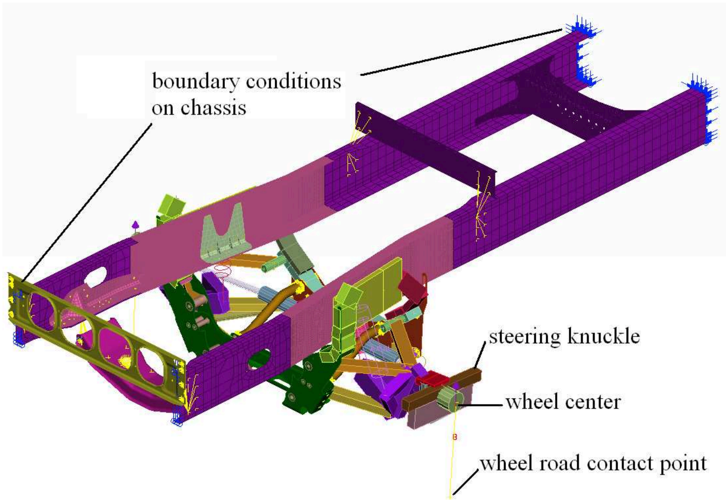


Figure 18. Kinematic analysis model arrangement.

Furthermore, Figure 19 depicts the steering system of the vehicle comprising rack and pinion, tie rods and steering arms.

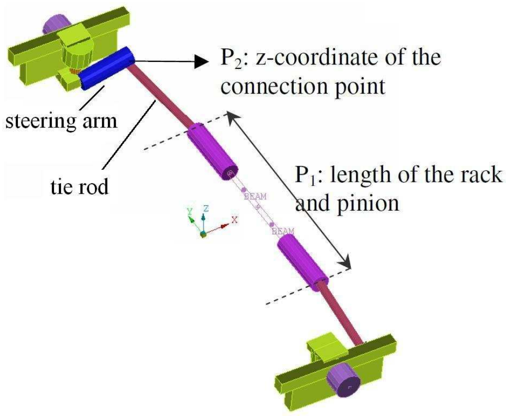
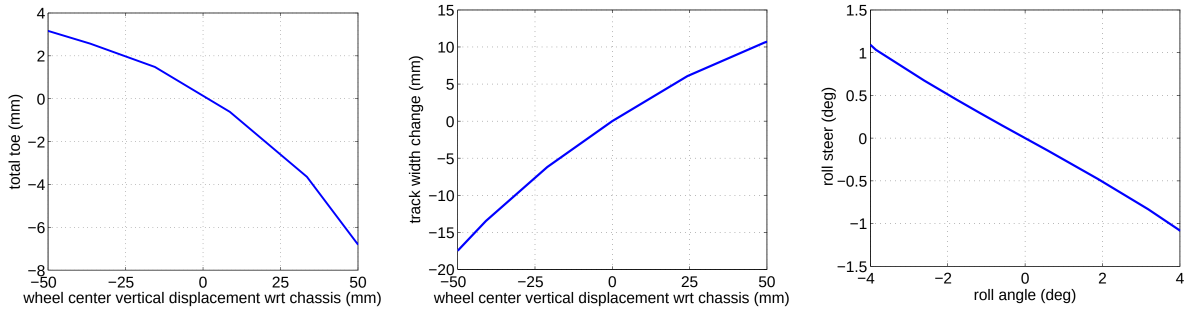


Figure 19. Model of the steering system using beam and connector elements.

Front suspension kinematic properties coupled to vehicle understeering and tire wear, i.e. roll steer, toe variation and track width change, are investigated in this thesis. Bump analysis is performed to establish measures of the total toe variation (both wheels) and track width change while roll steer is evaluated in rolling scenario and presented with respect to the roll angle of the vehicle chassis relative to the front axle. These analyses are carried out by applying corresponding vertical forces on the wheel centers of a fully loaded truck at driving height. Figure 20 illustrates the achieved total toe, track width and roll steer curves for the IFS.



(a) Total toe (b) Track width change (c) Roll steer

Figure 20. Kinematics of the IFS within the considered concept.

Furthermore, a sensitivity analysis is done in which he effects of the steering system parameters on the kinematics of the truck are investigated. Design parameters P_1, length of the rack and pinion, and P_2, z-coordinate of the steering arm connection point to the tie rod with respect to the road surface, are varied in realistic evenly distributed admissible sets for this purpose.

The outcome is presented in several Pareto fronts, see Paper C, that imply parameter P_1 should be at its possible minimum while P_2 can be chosen based on a compromise between roll steer and toe objective functions.

4.2 IFS Dynamics

The work has been continued by studying the dynamic behavior of the IFS. Therefore, the developed IFS model is incorporated into the Finite Element (FE) model of the 4×2 tractor and semitrailer combination, which is used for dynamic analysis in time and frequency domains in MSC.Nastran, demonstrated in Figure 21.

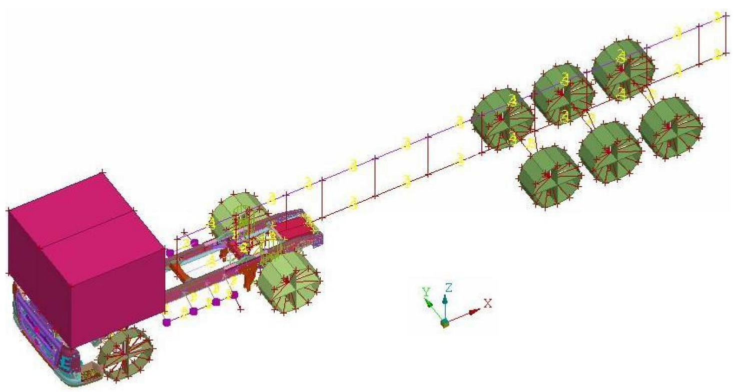


Figure 21. Model of the tractor semitrailer combination.

It is worth mentioning, the global coordinate system is positioned so that the x-axis is parallel to the vehicle body facing backwards, y-axis is along the front axle facing the right hand side of the vehicle and the z-axis corresponds to the vertical direction. All the results throughout this work are presented in the global coordinate system.

At first a sensitivity analysis of the comfort and handling characteristics of the vehicle to five chosen most influential damping parameters is performed. The five considered damping parameters belong to: cab front vertical, cab rear vertical, cab rear lateral, front axle and rear axle shock absorbers.

For the sensitivity analysis, two simulations in frequency domain with vehicle velocity of 80 km/h are conducted: road input and steering input frequency responses. In investigation of the dynamic performance capabilities of the vehicle in response to road excitations, bounce and roll load cases are considered. Here vertical force , F_z, is exerted on the wheel road contact points, the magnitude of which corresponds to the forces needed to make 1 deg roll on each axle and is given in Table 1. The forces are exerted on the left and right wheels in the same and opposite directions for bounce and roll, respectively.

Table 1. Vertical forces used in the bounce and roll analysis.

Axle	F_z (kN)
Front axle	18.3
Rear axle	32.5
Semitrailer axle	19.7

On the other hand, for the steering input analysis a torque that corresponds to turning the front wheels with 1 deg is applied on the steering gear. The steering gear is modeled with a torsional spring and is connected to the rack and pinion. This is to calculate the so-called yaw rate time delay, which shows the understeering of the vehicle.

Ride comfort is estimated using iso-filtered acceleration spectra, i.e. the spectra of acceleration data filtered by the ISO 2631-1 standard [69], on the cab points shown in Figure 22. Also, to approximate handling yaw rate time delay and the knuckle induced steering are taken into account. Yaw rate time delay is obtained using the phase angle of the cab at the frequency of 0.2 Hz and knuckle induced steering is measured as the maximum steering angle on the knuckle over the frequency range of 0-1 Hz.

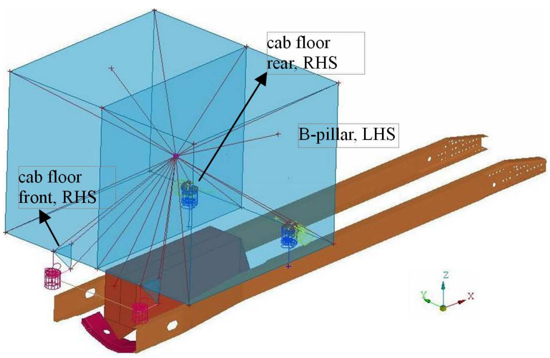
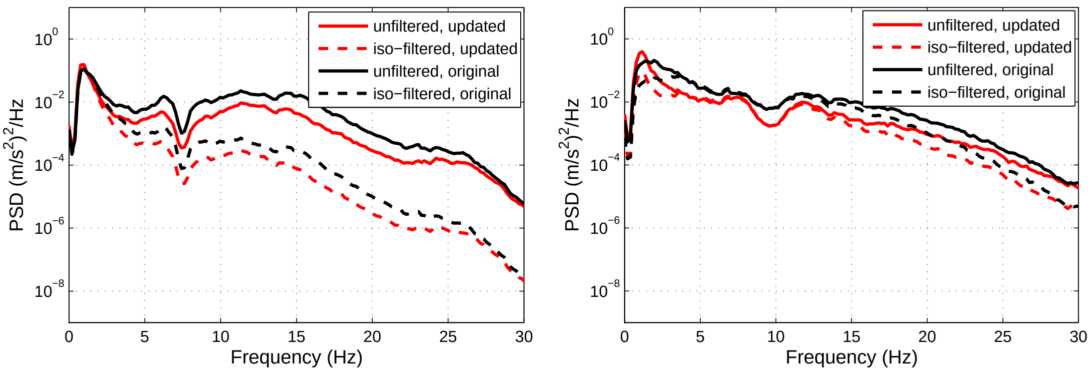


Figure 22. Cab model and the comfort points.

The results of the sensitivity analysis that are demonstrated in Paper D clearly show the influence of the studied parameters, particularly the front axle and cab lateral dampers, on the vehicle comfort and handling for the performed simulations. Consequently, the vehicle model has been updated with the chassis and cab lateral damper coefficients to run simulations on random roads in time domain for more detailed comfort examinations. For instance, Figure 23 shows the accelerations on the B-pillar in lateral and vertical directions for the model with updated parameters compared to the model with the original setting simulated on random rough road input.



(a) Lateral direction

(b) Vertical direction

Figure 23. Unfiltered and filtered acceleration spectra on B-pillar, on rough road.

4.2.1 Comparison to Rigid Front Axle

In order to identify the influences of the IFS on the vehicle dynamic performance a similar vehicle, 4×2 tractor semitrailer combination with rigid front axle and leaf spring suspension, is also considered. The differences between the complete vehicle models of the truck with IFS and rigid front axle are the front axle assembly (geometry, stiffness and damping) and steering modules. Figure 24 illustrates the model of the rigid front axle and its steering system.

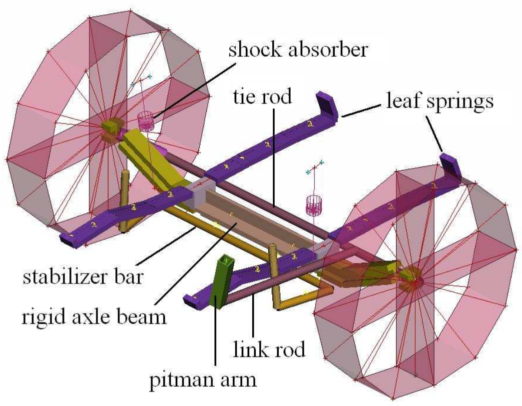


Figure 24. Model of the rigid axle beam with leaf spring suspension.

Driver comfort is investigated by analyzing cab accelerations from simulations on random roads, smooth and rough, and bump excitation. Moreover, vehicle handling is determined by assessing the results of the frequency response to steering inputs.

The outcome of the analysis on random road disturbances in the time domain are listed in Table 2. The RMS of the iso-filtered accelerations of the vehicle with IFS compared to the ones from the vehicle with rigid front axle are estimated by percentage difference expressed in Equation (12).

$$\Delta = \frac{X_{ifs} - X_{rigid}}{0.5\ (X_{ifs} + X_{rigid})}\ \times 100\ \%, \tag{12}$$

where X_{rigid} and X_{ifs} are in general the considered signal of the vehicles with rigid front axle and IFS, respectively. A negative Δ implies that X_{ifs} is lower than X_{rigid}. Thus, in Table 2 a negative value shows lower acceleration levels for the truck equipped with IFS.

Table 2. Percentage difference of RMS of iso-filtered accelerations, IFS compared to rigid front axle.

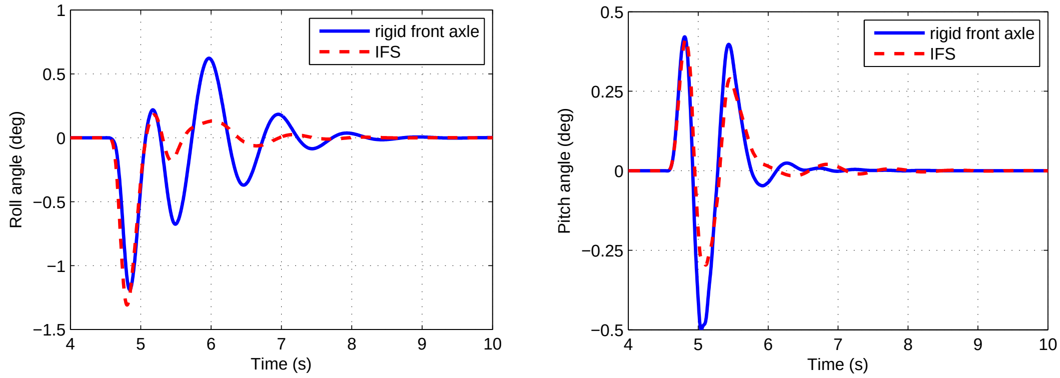
Signal	Smooth road (%)	Smooth road (%) freq > 4 Hz	Rough road (%)	Rough road (%) freq > 4 Hz
B-pillar, longitudinal	-2.1	-4.1	0.0	-2.7
B-pillar, lateral	3.4	-13.2	4.1	-12.2
B-pillar, vertical	-18.8	-21.2	-16.7	-20.3
Cab floor front, lateral	-0.3	-5.0	-0.4	-3.3
Cab floor rear, lateral	2.8	-47.2	3.0	-45.7

The utmost vibration reduction in RMS term is 47.2% occurring on the smooth road excitation inputs on the cab floor rear lateral direction for frequencies above 4 Hz. Additionally, the largest improvement on the B-pillar point can be seen in the vertical direction. However, the RMS values deteriorate in the lateral direction on B-pillar and cab floor rear point if the whole frequency range is taken into account since the peak at around 1 Hz, that has a higher order, increases using IFS.

Considering transient single and double sided road bumps, roll and pitch angles of the cab center of gravity (CG) are damped faster for the truck with IFS as can be seen in Figure 25, for instance. In addition, Figure 26 depicts the acceleration signals on the cab B-pillar in longitudinal and vertical directions for the double sided road bump (see Paper E for more information).

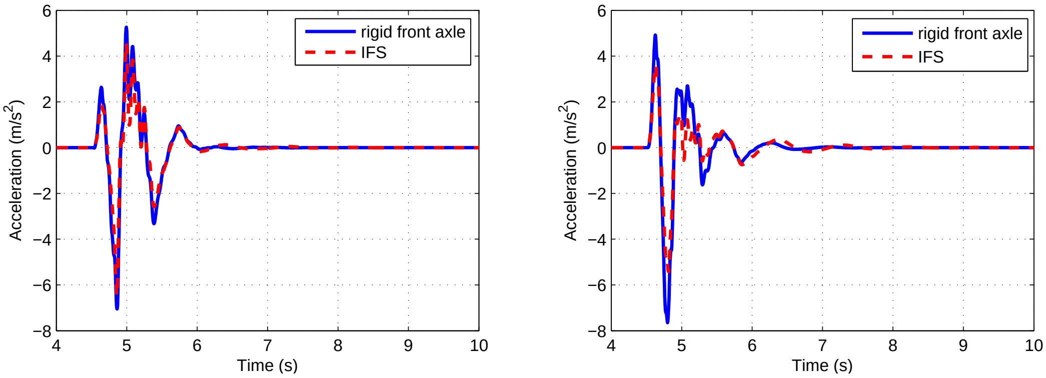
In addition to analysis of ride comfort, results of the steering input frequency response that are associated with the handling behavior of the vehicle are investigated. Figure 27 demonstrates the yaw rate response in terms of the transfer function between cab yaw rate and steering wheel angle as well as yaw rate time delay. It is desired to avoid peaks and flatten both curves in order to improve lateral stability for the entire frequency range of the steering angle. Shorter time delay indicates faster vehicle response.

It can be seen that the truck with rigid front axle has a smaller time delay up to 0.5 Hz, however, the maximum value is higher and the delay varies in the frequency range. Also, the curve is more even for IFS giving better stability feeling. The yaw rate transfer function to steering angle input almost has the same behavior showing lower value for rigid front axle in the low frequency region and a higher peak value.



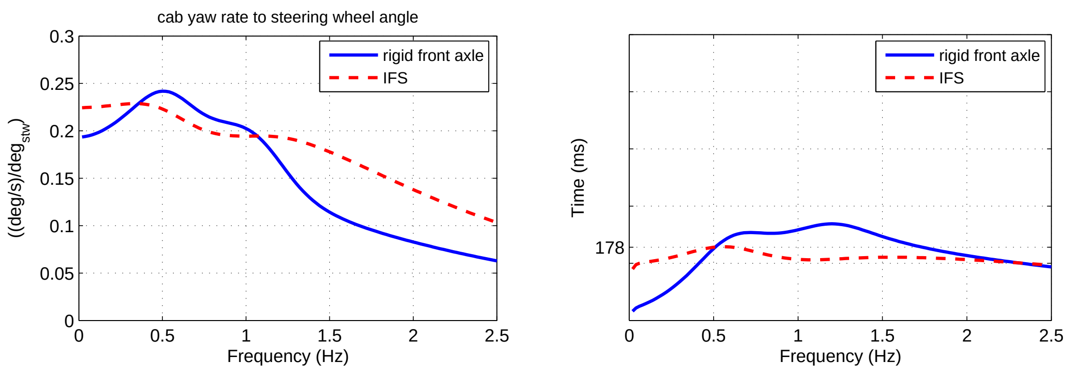
(a) Roll (b) Pitch

Figure 25. Cab CG rotations during single sided bump.



(a) Longitudinal (b) Vertical

Figure 26. Accelerations on the cab B-pillar during double sided bump.



(a) Yaw rate (b) Yaw rate time delay

Figure 27. Yaw rate response for steering inputs.

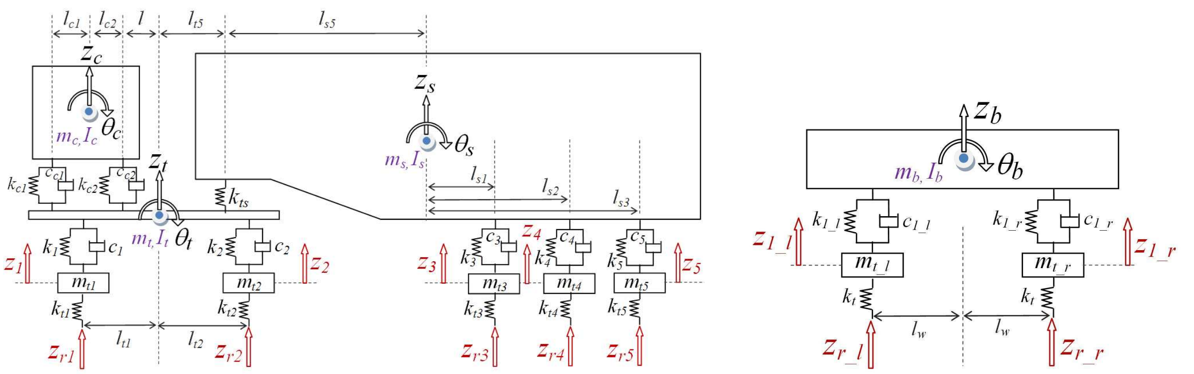
Paper E presents extensive results on ride comfort and handling and their analysis for the trucks with IFS and rigid front axle, the summary of which is enhanced comfort and steering feeling for the truck with IFS.

5 Semi-Active Front Axle Suspension

Semi-active suspension systems for ground vehicles have been the focus of research for several years as they offer improvements in vehicle comfort and handling. At the start, there has been a necessity of model development suitable for controller design and analysis since the FE model in MSC.Nastran is not appropriate for this objective. Models with various complexities have been developed and used during the course of this work as explained in the next section.

5.1 Vehicle Model

Vehicle models are divided by [75] into three classes; quarter, half and full vehicle models. To begin with, a half truck model of a 4×2 tractor and semitrailer combination with 11 DoFs together with a 4 DoF roll plane model are developed in Matlab/Simulink, shown in Figure 28. The roll plane model is considered to capture the roll motion of the vehicle body mass while pitch and vertical motions are obtained from the half truck model. Considering the above mentioned models and realistic road disturbances, namely



(a) 11 DoF half truck model

(b) 4 DoF roll plane model

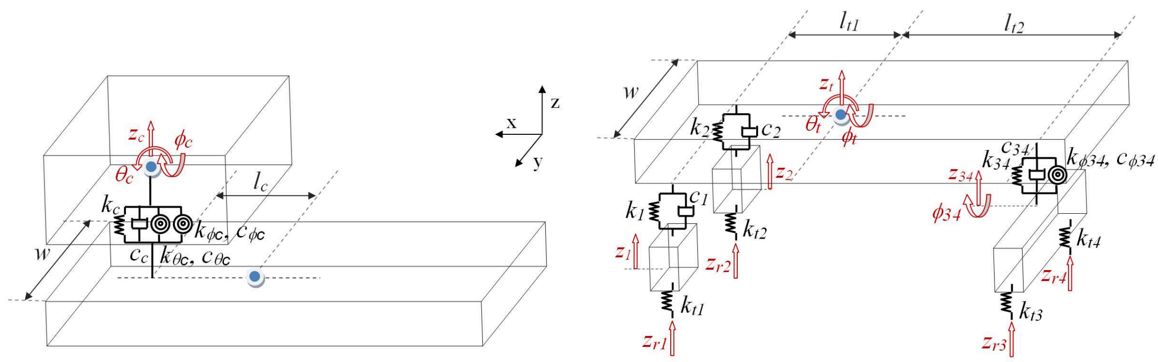
Figure 28. Vehicle models developed in Matlab/Simulink.

random road and single/double-sided bump inputs, results from skyhook/groundhook on-off and continuous variable semi-active damping systems are compared to the ones from the passive suspension system according to comfort and handling. Handling in terms of road holding characteristics is examined by means of analyzing dynamic tire forces while monitoring rattle space simultaneously.

On-off groundhook dampers do not provide any significant progress for driver comfort although they can improve dynamic loading of the tire. However, on-off skyhook dampers can offer possibilities in enhancing comfort by considerably reducing acceleration levels up to wheel hop resonance frequency. The drawback is the increased peak in the tire load curve at wheel hop frequency.

System responses enhance by adopting continuous semi-active damping, however; suspension deflection is increased and should be applied as a constraint (Paper F).

The work on semi-active suspension has been followed by using a 3D multi-body model of the truck with IFS in Matlab/Simulink for controller design. This model comprises chassis, cab, front and rear tractor axles as illustrated in Figure 29. The front axle, which is an IFS includes two bodies with vertical DoF connected to chassis via linear spring and damper elements. Moreover, rear axle is a rigid beam with 2 DoFs, vertical and roll, attached to the vehicle body by means of both linear translational and rotational spring and dampers. The model is defined in state space form in Paper G.



(a) Tractor cab and chassis (b) Tractor chassis and axles

Figure 29. 3D model of the tractor.

This model is employed to design control algorithms for the semi-active front axle damper with hybrid skyhook/groundhook, clipped LQ and MPC algorithms. However, in order to evaluate and verify the developed controllers, a more advanced vehicle model with sufficient accuracy for different driving scenarios is required. Therefore, a nonlinear 3D model of the vehicle in Mtlab/SimMechanics is used to study the outcome of the semi-active dampers compared to the original passive front axle damper. The existing model of the 4×2 tractor semitrailer combination in Matlab/SimMechanics [76] is used as a basis for the tractor model with an independent front suspension. The model of the IFS is developed using two rigid bodies that are connected to the vehicle chassis by means of spring and damper elements and then exchanged with the original rigid front axle in the full vehicle model. Furthermore, since in this vehicle the nonlinear tire behavior is modeled with Magic Formula according to [77], it is possible to investigate the steering characteristics of the vehicle as damper tuning may have significant influence on handling manoeuvres as lane change [1].

5.2 Passivity Constraint

The passivity constraint defines the region in which the typical semi-active damper force is realizable. This region that is shown in Figure 30 is a part of the theoretical semi-active region shown before in Figure 4. This passivity constraint is implemented in Matlab and used to calculate and obtain the semi-active damping force in the developed controllers.

In the following, the velocity over the damper i is denoted by v_i while the semi-active force in damper i is denoted by u_i. The shape of the passivity region is of course device dependent but typical characteristics include a linear lower limit and a saturating high-damping behavior. These characteristics are illustrated in [54, 78, 79]. Inspired by the lemniscate of Gerono we find that inequality (13) serves our purposes well.

$$0 \leq \left(\frac{u_i}{u_{max}}\right)^2 - \left(\frac{u_i}{u_{max}}\right)^4 - \left(\frac{v_i}{v_{width}} - \frac{u_i \cdot v_{\max}}{v_{width} \cdot u_{max}}\right)^2 + p_t^2. \tag{13}$$

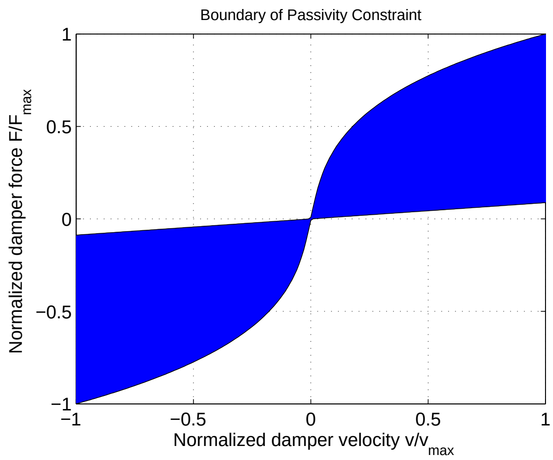


Figure 30. Illustration of inequality (13) with parameters $u_{max} = 1.9, v_{width} = 29.9, v_{max} = 30, p_t = 0$.

5.3 Control Strategy

Dynamic performance of the vehicle with semi-active suspension has been studied considering the 3D model of the 4×2 tractor semitrailer with IFS in Matlab/SimMechanics. For the semi-active damper hybrid skyhook/groundhook, clipped LQ and MPC algorithms are developed and evaluated with respect to the passive suspension system. The developed controllers are verified by simulation with respect to the identified road inputs and maneuvers including stochastic and transient road irregularities and emergency braking.

5.3.1 Hybrid Skyhook/Groundhook

This method is a non model-based control design where the controller seeks to emulate a damper connected between the sprung mass and an inertial reference, skyhook, unsprung mass and an inertial reference, groundhook, or a combination of these two cases, hybrid. Hybrid control policy adopted for continuous semi-active damping is summarized in Equations (14) and (15). The value of coefficient $\alpha \in [0, 1]$ determines if the controller is skyhook, groundhook or hybrid. The controller is skyhook if $\alpha = 1$, groundhook if $\alpha = 0$ and hybrid otherwise.

$$F_{sa} = K[\alpha F_{sky} + (1 - \alpha)F_{ground}]. \tag{14}$$

$$\begin{cases} v_1 \cdot v_{12} > 0, & F_{sky} = v_1, \\ v_1 \cdot v_{12} \leq 0, & F_{sky} = 0, \end{cases} \quad \& \quad \begin{cases} -v_2 \cdot v_{12} > 0, & F_{ground} = v_2, \\ -v_2 \cdot v_{12} \leq 0, & F_{ground} = 0. \end{cases} \tag{15}$$

Here F_{sa} is the established semi-active damper force, K is a positive constant, F_{sky} is skyhook force and F_{ground} is the groundhook forces. Also, v_1 and v_2 are velocities of the sprung and unsprung masses, respectively, while $v_{12} = v_1 - v_2$ stands for the relative velocity over the damper.

5.3.2 Clipped Linear Quadratic Regulator

In this method the damper force is obtained by computing the active LQ control law and clipping it to enforce the passivity constraints. The unconstrained LQ control law is obtained through minimizing the quadratic cost function

$$\mathfrak{J}_{LQ} = \int_0^\infty (y^T \cdot Q \cdot y + u^T \cdot R \cdot u + 2y^T \cdot N \cdot u)dt. \tag{16}$$

The optimal solution can be expressed in state feedback form as

$$u^* = -K \cdot x = \underset{u}{\operatorname{argmin}}\ \mathfrak{J}_{LQ}. \tag{17}$$

5.3.3 Model Predictive Control

The MPC strategy consists of, at each sampling time, t_i, solving an open-loop optimal control problem over a finite horizon, starting at the current state. At the next time step the optimal control problem is solved starting from the new state and over a shifted horizon, leading to a moving horizon policy. In contrast to the LQ infinite horizon control law of the previous section a finite horizon control law with general constraints lacks exact closed form solutions. A by now well-established method, employed by the Multi-Parametric Toolbox (MPT), consists of partitioning the state space into a number of polytopes and designing a linear feedback controller for each polytope. Thus, for each polytope the optimization problem is reduced to a simple evaluation of a matrix-vector product. While this approach possesses many enviable properties it still suffers from the curse of dimensionality: the time required to even offline compute the linear feedback gains for an exponential number of polytopes and the online problem of detecting what polytope the present state belongs to and consequently choosing what linear feedback to use both become prohibitive even when dealing with systems of moderate dimension.

A somewhat different approach is adopted in [54], where the optimal non-linear control law is approximated using a set-membership approach. Since the online control law still is computed via a polytope selection, it is not different from the algorithms of the MPT in this respect. The original formulation of MPC algorithms was discarded early on for fast applications requiring high sampling rates due to its computational complexity. However, with faster hardware and specialized algorithms, MPC has experienced a resurgence for fast applications [80]. Performing the optimization online offers several advantages: it enables adaptation since no lookup table needs to be recomputed for new design parameters and it can be applied to higher dimensional systems as long as the number of inputs remains reasonably small.

The MPC objective function can be formulated by

$$\mathfrak{J}_{MPC} = \sum_{k=i+1}^{i+N_{p,y}} \Big({y_k}^T \cdot Q_y \cdot y_k \Big) + \sum_{k=i}^{i+N_{p,u}-1} \Big({u_k}^T \cdot Q_u \cdot u_k \Big). \tag{18}$$

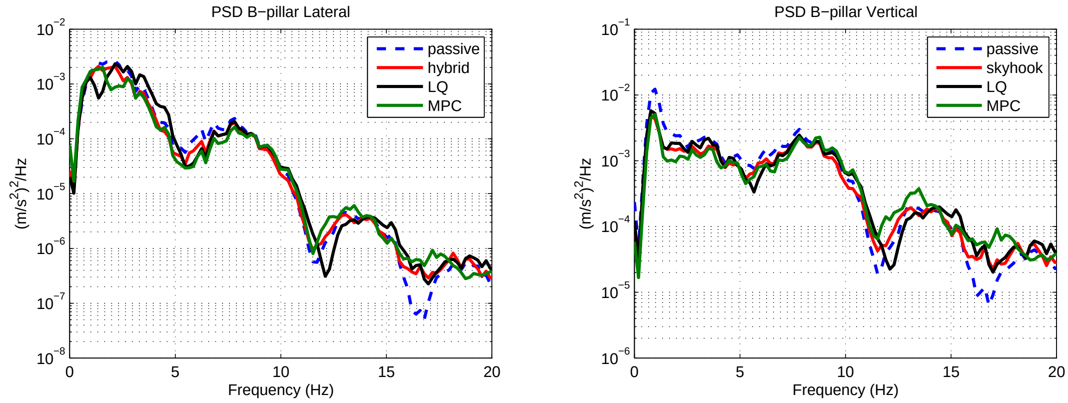
5.4 Analysis and Results

Employing the nonlinear tractor semitrailer combination, the performance of the candidate control algorithms are studied through comparing the responses of the vehicle with semi-active dampers to the ones from the original truck with passive dampers considering road irregularities and emergency braking maneuver.

Figure 31 shows the iso-filtered acceleration PSD of the cab B-pillar. Furthermore, PSD of the left and right tire forces during analysis on random road are depicted in Figure 32. Two peaks can be recognized in the plot, the first one is reduced by all three considered semi-active dampers, the second one however, is a matter of compromising to some extent using hybrid and LQ control techniques. Nonetheless, the MPC control has managed to improve both peaks simultaneously.

Figure 33 presents the desired, u_{des}, and realized, u_{real}, semi-active damper forces from the MPC satisfying the passivity constraint. As shown, the desired damping forces are only approximately fulfilling the passivity constraint since the computation is based on one sample old data.

The quantitative analysis of the results shows that Max and RMS of the vertical accelerations on the B-pillar are reduced up to 17.4% and 16.5% using the MPC, respectively. Moreover, suspension deflection have been improved using all control designs up to 48.1% and 33.6% in terms of RMS and Max values, respectively. Among



(a) Lateral (b) Vertical

Figure 31. PSD of iso-filtered B-pillar acceleration during simulation on random road.

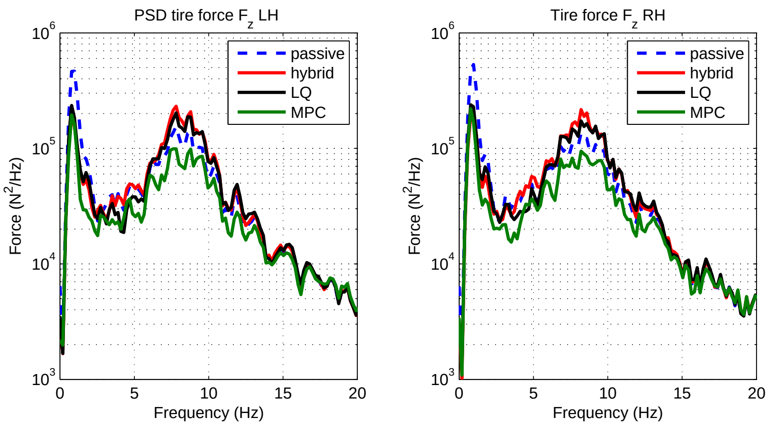


Figure 32. PSD of vertical tire forces during simulation on random road.

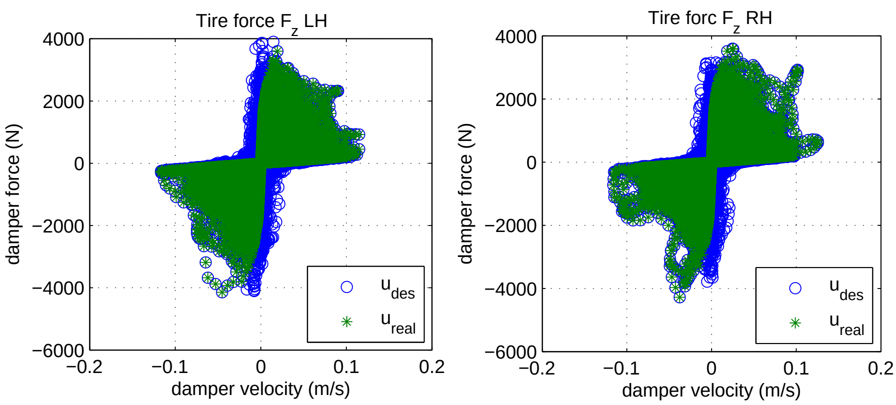


Figure 33. MPC based semi-active damper characteristics from simulations on random road.

the developed controller, MPC has the best effect on the tire forces. Hence, unlike passive suspension that is a compromise between ride comfort and handling, semi-active suspension facilitates enhancing both target criteria.

6 Conclusions

The general goal of this work has been to identify and study the functions and performance of two of the main suspension systems in a heavy truck, namely engine and front axle suspension.

In order to investigate the nonlinear behavior of the dynamic stiffness and damping of engine mounts, mathematical and computational models of passive rubber-metal mounts consisting of elastic, viscous and friction functional components have been developed. An optimization approach is used to determine model parameters according to measurement data available for front and rear engine mounts, which are in use for commercial vehicle engine mounting systems. Furthermore, the model has been validated with reference to measurement data for different inputs of amplitude and frequency, against which the model shows admissible agreement. The computational model of the mount is then implemented in Matlab/Simulink in a 3D engine model with which vibration dynamic analysis under engine excitations and realistic road inputs are conducted for passive and adaptronic engine suspensions. In this model the engine is a rigid body that is suspended on the chassis with three elastomeric mounts. The outcome of the engine vibration dynamics differs for the passive system based on the employed model. For instance, the developed model shows higher estimation of the transmitted forces to the vehicle structure than the Kelvin-Voigt model by 20% in RMS and 3.6% in Max value when studying the total transmitted forces. Additionally, the advantages of using active engine mounts have been studied by incorporating two actuators into the 3D engine model with conventional mounting system for heavy trucks; one actuator is put in the lateral direction in front and the other in vertical direction in rear, making the engine mounting system an adaptronic one. Simulations of the engine vibration dynamics under similar engine and road excitations have also been done for the adaptronic mounting system and compared to the results from the conventional engine mounting system. Using the adaptronic engine suspension the total transmitted forces to chassis has decreased by 24% and 14% in RMS and Max values, respectively.

Moreover, the influences of IFS, based on the double wishbone concept, and semi-active front axle damping in a heavy truck on ride comfort and handling characteristics are addressed. The model of the IFS, released recently by Volvo Truck Corporation, is developed and incorporated into the FE model of the 4×2 tractor semitrailer combination that is used for simulations on realistic road inputs and steering maneuvers. Ride comfort and handling of the truck equipped with IFS are then evaluated and compared to the reference vehicle with rigid axle and leaf spring suspension. The outcome shows a better comfort in terms of vibration reduction up to 47.2% as well as better steering feeling for the vehicle with IFS.

The work is followed by studying the effects of semi-active front axle damping on the dynamic performance of the vehicle. Various control strategies as hybrid sky-hook/groundhook, clipped LQ and MPC are developed and evaluated with regard to the passive suspension system employing the 3D multi-body model of the tractor semitrailer in Matlab/SimMechanics, which is suitable for controller implementation. The developed controllers are verified by simulation with respect to the identified road inputs and maneuvers.

The outcome of the research described in this thesis is published in the appended scientific papers, Papers A-G, the scientific contributions of which are summarized here.

Paper A - Computational Model of Conventional Engine Mounts for Commercial Vehicles: Validation and Application

The focus in this article is to develop a fairly simple and accurate computational model for conventional engine mounts of commercial vehicles in order to express the nonlinear behavior of the dynamic stiffness and damping of mounts as functions of both frequency and amplitude of excitation. The model puts functional components of elastic, viscous and friction behaviors together. Elastic and viscous parts comprise linear springs and dampers and the friction component is a nonlinear Coulomb-type friction model grasping the essential behavior of the hysteresis loops in force-displacement curves. Model parameters are identified through optimization with LMS algorithm using measurement data. The developed model has been validated against measurement data for harmonic excitations with a frequency range of 5-100 Hz and an amplitude range of 0.025-2 mm employing three different engine mounts used in heavy trucks. The model shows admissible agreement with measurement data keeping the error of estimation below 11%. Subsequently, simulation results of engine vibrations dynamics are given with both proposed model and commonly applied Kelvin-Voigt model of the mounts under different engine excitations and realistic road inputs. The comparison analysis shows significant differences in estimation of engine mount displacements, transmitted forces to chassis as well as engine displacements and accelerations with regard to the used mount model.

Paper B - Vibration Dynamics of a Commercial Vehicle Engine Suspended on Adaptronic Mounting System

In this paper the proposed mount model in Paper A has been used to examine the engine vibration dynamics of an adaptronic engine suspension compared to the passive one. The adaptronic mounting system is developed by adding two actuators, with maximum capacity of 500 N, to the conventional mounting system and its mathematical model is presented. A parameter optimization of the proposed adaptronic suspension has also been done. Finally, simulations of engine vibration and transmitted forces to the vehicle structure for conventional and adaptronic mounting systems subject to the same road and engine excitation inputs are conducted. Comparison of the obtained results demonstrates improvement in commercial vehicle engine vibration isolation and up to 24% reduction in the magnitude of the transmitted forces to the vehicle structure achieved with adaptronic engine mounting system.

Paper C - Kinematic Analysis of a Heavy Truck with Individual Front Suspension

This paper presents the kinematic aspects of a heavy truck suspension equipped with IFS, which are coupled to vehicle dynamics and tire wear, in terms of roll steer and toe variation. These characteristics are assessed through cornering and bump simulations, respectively. Furthermore, the impacts of the steering system parameters on these properties are studied in a sensitivity analysis and the results are provided in Pareto fronts, which not only show the sensitivity of the kinematics on the design parameters but also the contradictory requirements on them.

Paper D - Comfort and Handling of a Commercial Vehicle with Individual Front Suspension

The objective of this article is to get an insight into the dynamic performance of a heavy truck with IFS with respect to ride comfort and handling. Firstly, a model of the IFS is created and incorporated into the FE model of the 4×2 tractor and semitrailer combination. This model is then used to examine the influence of cab, front and rear axle shock absorbers on the vehicle dynamic properties by means of a sensitivity analysis. Two comfort and two handling objective functions are defined to evaluate the results of the frequency responses to road disturbances and steering inputs. The results that are provided in Pareto fronts clearly show the great influence of the studied parameters, particularly the front axle and cab lateral dampers, on the vehicle comfort and handling for the performed simulations. Lastly, the model is updated with the chassis and cab lateral damper coefficients to run a simulation on random roads for more detailed comfort examinations. This analysis confirms the obtained improvements in the outcome of the sensitivity study showing reductions in cab accelerations up to 62% in RMS value compared to the original settings.

Paper E - Kinematic and Dynamic Analysis of a Commercial Vehicle with Individual Front Suspension

This article addresses the kinematic and dynamic performance of a heavy truck with IFS. Employing the FE models of the 4×2 tractor semitrailer arrangement, this study evaluates and compares the dynamic characteristics of heavy trucks with IFS and reference rigid front axle according to comfort and handing properties. This is done by analyzing the responses of the vehicle to various road excitations (random roads, single and double sided bumps) as well as steering inputs. The obtained results that are demonstrated qualitatively by substantial plots and also summarized quantitatively in tables show enhanced comfort, by reducing the comfort objective by up to 47.2%, and steering feeling, by reducing maximum yaw rate time delay by 13.1%, for the truck with IFS setup.

Paper F - Effect of Semi-Active Front Axle Suspension Design on Vehicle Comfort and Road Holding for a Heavy Truck

In this paper the capabilities of semi-active front axle suspensions are investigated for a commercial vehicle with regard to comfort and handling in terms of road holding characteristics, examined by means of analyzing dynamic tire forces. A half truck model of a 4×2 tractor semitrailer combination together with a 4 DoF roll plane model are created to run analysis in the frequency and time domains. The results of the frequency analysis illustrate the impact of passive front axle damping coefficient on comfort and road holding of the vehicle. Moreover, for analysis in time domain stochastic and transient road irregularities are considered to analyze semi-active suspension compared to a passive one. The control strategies of the semi-active damping are based on the skyhook theory, in which the damping coefficient is either switchable or continuously controlled. Skyhook, groundhook and hybrid control laws have been implemented and evaluated in the vehicle models. With continuous semi-active dampers accelerations in the cab are drastically decreased particularly up to front axle resonance frequency. With the considered parameter settings, RMS and Max values of the iso-filtered acceleration has been reduced up to 39% and 45%, respectively. Unlike passive suspension that is a compromise between ride comfort and handling, semi-active suspension facilitates enhancing both target criteria.

Paper G - Improving Comfort and Handling of Heavy Vehicles with Individual Front Suspension Using Semi-Active Dampers - An Approach Based on Clipped LQ and Model Predictive Control

In this work the continuous variable semi-active damping is analyzed compared to passive damping in the front axle suspension of a heavy truck with IFS according to ride comfort and handling. This is done through simulating the nonlinear 3D multi-body truck model of a tractor semitrailer combination in Matlab/SimMechanics that is suitable for controller evaluation and verification with various road excitations and steering inputs. The candidate control algorithms for the semi-active damping are hybrid skyhook/groundhook, clipped LQ and MPC. Furthermore, driver comfort is investigated by analyzing cab accelerations and rotations from simulations on random roads and bump excitation while vehicle handling is determined by assessing the tire forces. Finally, simulation results of the semi-active front axle dampers are compared to the passive ones. The comparison analysis shows the great improvements in the trade-off between comfort and handling with semi-active dampers in particular by means of MPC that enhances the RMS of tire forces up to 23.4% on random roads while reducing the vertical cab acceleration RMS by 16.5%.

7 Future Work

In the future research in the area of semi-active/active engine suspension a few studies can be performed. Investigations on optimized number and location of the actuators, actuator type as well as more advanced modeling of the actuator dynamics can be done. Another aspect is addressing various controller laws suitable for semi-active/active engine mounting systems. Different types of controllers can be evaluated to find the most effective one together with the actuator dynamics.

Moreover, semi-active/active engine suspension can be assessed along with IFS and semi-active front axle suspension in a complete vehicle model to determine the effects on vibration attenuation.

Considering only semi-active suspension design, the presented work can be continued by implementation of the semi-active damper and its controller into the real vehicle for experimental evaluation of semi-active suspension and validation of the simulation results. Furthermore, benefits of semi-active suspension on the rear axle of the truck as well as trailer axles can be investigated. Lastly, semi-active suspension by means of controlling both damping and stiffness of the front axle suspension can be assessed. The performance of the vehicle can be affected merely to a certain limit by means of semi-active front axle damper. There are more possibilities of improvement if both stiffness and damping can be controlled.

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Nomenclature

A	state matrix in the linear state space model
a_i^{iso}	iso-filtered acceleration along the i direction
a_i^s	acceleration spectrum along the i direction
B_u	input matrix in the linear state space model according to control input
B_w	input matrix in the linear state space model according to disturbance
C	output matrix in the linear state space model
c_i	damping coefficient of the i^{th} damper
D	damping
D_u	feedthrough matrix in the linear state space model according to control input
D_w	feedthrough matrix in the linear state space model according to disturbance
E	energy loss
F_a	actuator force
F_e, F_f, F_v	elastic, friction and viscous forces
F_{sa}	semi-active damping force
F_{Tx}, F_{Ty}, F_{Tz}	Transmitted force to chassis in x, y and z directions, respectively
F_x, F_y, F_z	Force in x, y and z directions, respectively
$I_{\phi i}$	roll inertia of the i^{th} body
$I_{\theta i}$	pitch inertia of the i^{th} body
k_i	stiffness coefficient of the i^{th} spring
m_i	mass of the i^{th} body
S	stiffness
u	input vector in the linear state space model
v_i	velocity of the i^{th} body
x	state vector in the linear state space model

y	output vector in the linear state space model
z_i	vertical displacement of the i^{th} body
z_{ri}	road input on the i^{th} wheel center
ϕ_i	roll angle of the i^{th} body
θ_i	pitch angle of the i^{th} body

Abbreviations

2D	two dimensional
3D	three dimensional
4×2	four by two, notation for a two axle truck with driven rear axle
AMS	adaptronic mounting system
CG/cog	center of gravity
CM	center of mass
CMS	conventional mounting system
DoF	degree of freedom
ER	electro-rheological
FFT	fast fourier transform
FE	finite element
HIL	hardware-in-the-loop
IFS	individual front suspension
ISO	international organization for standardization
LMS	least mean square
LQ	linear quadratic
MPC	model predictive control
MPT	multi parametric toolbox
MR	magneto-rheological
PSD	power spectral density
RMS	root mean square