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Citation for the original published paper (version of record):
Yarmohamadi, H., Berbyuk, V. (2011). Kinematic Analysis of a Heavy Truck with Individual Front Suspension. In Proceedings of the 22nd International Symposium on Dynamics of Vehicles on Roads and Tracks, IAVSD2011, 14-19 August 2011, Manchester Metropolitan University, Manchester, UK

22nd International Symposium on Dynamics of Vehicles on Roads and Tracks
14-19 August, 2011, Manchester, IAVSD2011, paper64.

KINEMATIC ANALYSIS OF A HEAVY TRUCK WITH INDIVIDUAL FRONT SUSPENSION

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Abstract

This paper presents the suspension kinematic characteristics of a commercial vehicle with Individual Front Suspension (IFS) that are important to the truck dynamics and tire wear. The model used here consists of the front axle assembly, truck chassis, and the steering system. This model is employed to perform a sensitivity analysis to investigate the influence of the steering system parameters, length of the rack and pinion and height of the steering arm connection point to the tie rod, on the kinematics of the vehicle. The results, which are provided in Pareto fronts and Pareto sets not only show the sensitivity of the kinematics on the design parameters but also the contradictory requirements on them.

1 INTRODUCTION

To the authors knowledge, all the heavy trucks produced up to now have a rigid front axle connecting the left and right wheels. In the presented work a commercial vehicle with IFS is considered in which the left and right wheels are suspended individually using the double wishbone concept. This technology is adopted in heavy trucks due to the desire for improved handling and driver comfort that in turn leads to better vehicle safety and stability.

To evaluate a heavy truck suspension system, both kinematic and dynamic characteristics are considered. Suspension kinematic analysis of road vehicles has been a topic for research studied, for instance, in [1, 2, 3]. 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.

This paper mainly focuses on the suspension kinematic aspects of the heavy truck that are coupled to the vehicle understeering and tire wear, i.e. 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, the results of which are provided in Pareto fronts and Pareto sets.

2 MODELING

Kinematic aspects of heavy vehicles with IFS are investigated through analytical expressions as well as simulations with a vehicle model. The model developed for this purpose consists of the front axle assembly, front part of the chassis and the steering system. The considered length of the flexible truck chassis, represented by shell elements, is almost 3 (m) since only the front axle is examined in the study. The steering system of the vehicle comprises rack and pinion, tie rods and steering arms (depicted in Figure 1).

This model that is developed for nonlinear kinematic analysis in Abaqus, is made with multi bodies and shell elements that connect with each other by various applicable joints and constraints. Moreover, lower and upper wishbones are attached to the vehicle frame with bushings. 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, shown in Figure 2.

The global coordinate system is located on the road surface 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 in this work are presented in the global coordinate system.

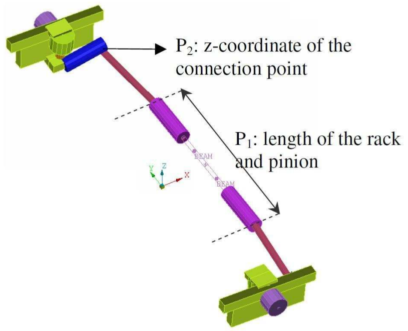


Figure 1: Multi body model of the steering system

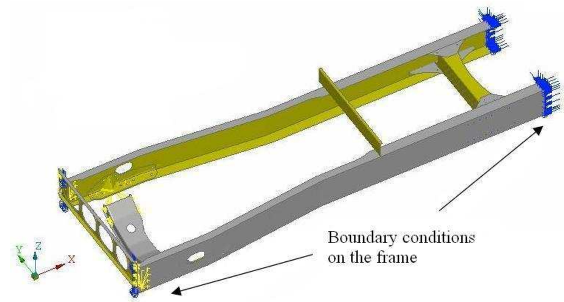


Figure 2: Boundary conditions for the kinematic analysis

2.1 Inputs and Outputs

In investigation of the kinematic behavior of the vehicle, the inputs to the model are forces that are applied on the front axle wheel centers to simulate different driving scenarios. The output addresses the truck suspension kinematics- namely toe variations, roll steer, track width change, vertical stiffness and roll stiffness- that are important to the vehicle handling and tire wear.

3 KINEMATICS OF THE FRONT AXLE

In order to analyze the kinematics of the front axle within the considered IFS concept and developed model, we take a set of load cases and scenarios into account. The two load cases used in this study are bumping and cornering analysis configured by vertical forces acting on the wheel centers.

To establish a measure of the vertical stiffness, track width change and total toe variation (both wheels) the bump analysis is performed. However, for roll steer and roll stiffness estimations, cornering analysis is considered. Roll steer of the vehicle is presented with respect to the roll angle of the vehicle chassis relative to the front axle. Table 1 lists the vertical loads that are exerted on the left and right wheel centers during the above mentioned analyses.

Table 1: Loads applied on the wheel centers for the considered load cases.

Load case	F_z^{left} (N)	F_z^{right} (N)
Bumping	7.16 E4	7.16 E4
Cornering	7.16 E4	1.00

Total toe angle and roll steer can be written as below according to Figure 3:

$$total\ toe = b - a, \quad roll\ steer = \frac{\alpha_R - \alpha_L}{2} \tag{1}$$

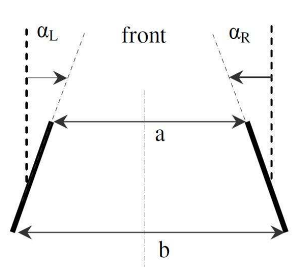
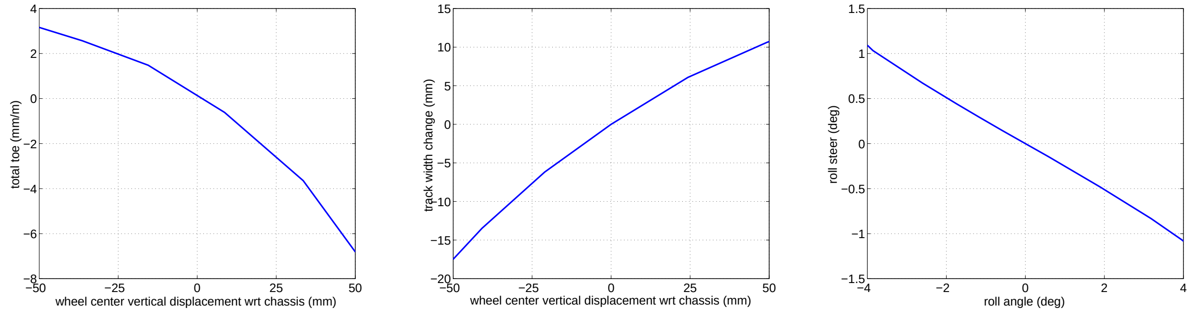


Figure 3: Top view of the left and right wheels

Figure 4 shows the total toe and track width change during bump analysis and also roll steer while rolling, respectively. It can be seen that bumping of the vehicle results in a toe out (negative total toe) and a larger track width.



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

Figure 4: Kinematics of the IFS within the considered concept

4 SENSITIVITY ANALYSIS OF THE IFS KINEMATICS

Furthermore, the model is employed to perform a sensitivity analysis where the effects of the steering system parameters, length of the rack and pinion (P_1) and z-coordinate of the steering arm connection point to the tie rod (P_2), have been investigated on the kinematics of the truck.

The focus of the performed study is on the toe variation and roll steer behaviors relating to the vehicle's tire wear and understeering. As stated previously, roll steer is assessed in a cornering analysis while toe variation is estimated in bumping simulation. In the sensitivity analysis two additional scenarios to investigate toe variation as functions of the ride height and applied vertical load are also performed. Consequently, total toe is evaluated in three different scenarios listed below:

- Scenario 1: increasing the vertical force from full load to bump load (2×full load).
- Scenario 2: changing the ride height with means of air bellow pressure while keeping the vertical force at kerb load.
- Scenario 3: increasing the vertical force from kerb load to full load (5.5-8 tonnes) while preserving the ride height.

4.1 Simulation Setup

Design parameters P_1 and P_2 are varied in realistic evenly distributed admissible sets that can be written as:

$$P_1 \in [P_1^{min}, P_1^{max}], \quad P_2 \in [P_2^{min}, P_2^{max}] \tag{2}$$

Where the percentage difference of the upper and lower bounds are,

$$200 \cdot \frac{P_1^{max} - P_1^{min}}{P_1^{max} + P_1^{min}} \approx 7\%, \quad 200 \cdot \frac{P_2^{max} - P_2^{min}}{P_2^{max} + P_2^{min}} \approx 0.6\% \tag{3}$$

Matlab is used to update the parameters in the input file, run iterations in Abaqus with the developed model and postprocess the outputs.

4.2 Analysis Results

Figure 5-8 show the results of the sensitivity analysis for toe variations gained form the above-mentioned scenarios as well as roll steer. The effect of our design parameters, P_1 and P_2, on the outcome is depicted separately in Figure 9-16 for further examinations. It is desired to minimize toe variations with respect to wheel movements and applied loads to decrease tire wear. Also, to increase understeering a roll steer curve with larger slope is preferred.

It is visible that both parameters significantly affect the kinematic features of the truck. Both ends of the toe curves go toward toe out when P_1 is increased. However, P_2 has a different influence. The increase of P_2 gives more toe in when rebounding and more toe out when bumping in scenario 1 and 2. Looking at the roll steer plots, parameters have contradictory impacts so that increasing P_1 reduces the slope of the roll steer curve while increasing P_2 increases it. Therefore, a combination of minimum P_1 and maximum P_2 results in the highest understeering behavior.

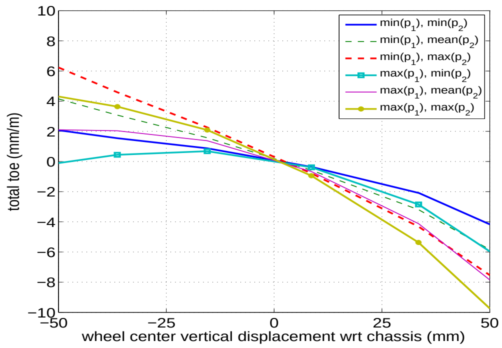


Figure 5: Total toe obtained from scenario 1

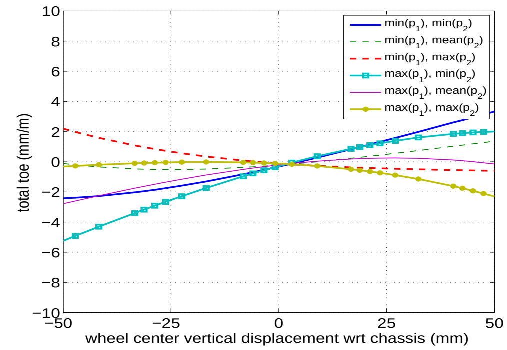


Figure 6: Total toe obtained from scenario 2

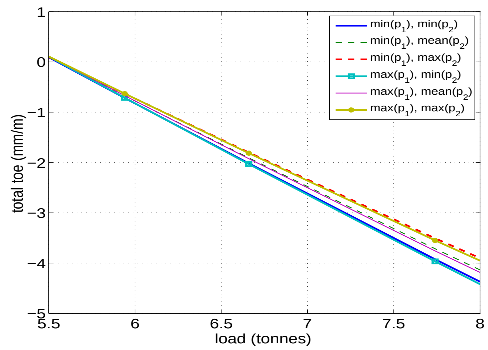


Figure 7: Total toe gained from scenario 3

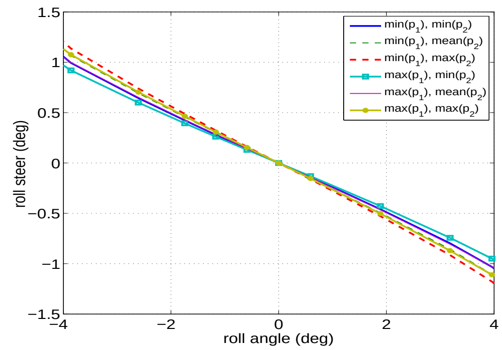


Figure 8: Roll steer vs. roll angle

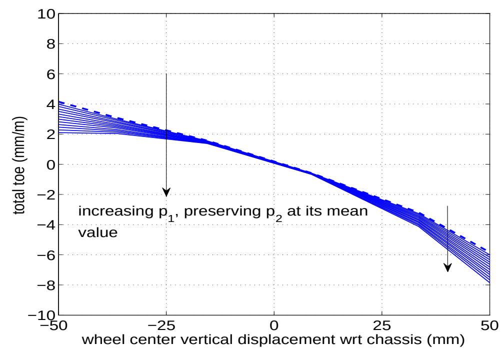


Figure 9: Influence of P_1 on total toe from scenario 1

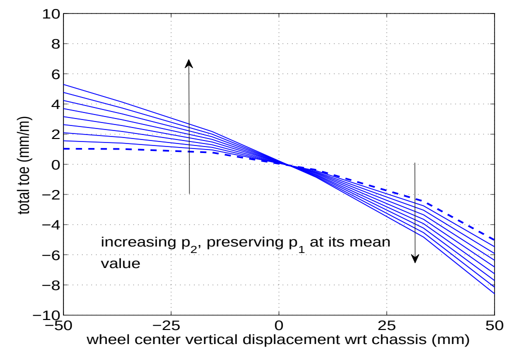


Figure 10: Influence of P_2 on total toe from scenario 1

5 PARETO FRONT AND PARETO SET

In order to tune the two considered parameters, it is beneficial to summarize the outcome of the study in Pareto fronts. For this reason objective functions are defined that convert each toe and roll steer curve into one scalar number. To identify toe variations, the toe objective function is the RMS (Root Mean Square) value of the total toe calculated over the range of ± 50 (mm) of wheel center vertical displacement with respect to chassis. The objective function corresponding to the roll steer is defined as the inverse of the RMS of the roll steer for the range of ± 4 (deg) of roll angle. This choice is explained by the objective of increasing roll steer for obtaining a more understeered vehicle.

Figure 17-19 provide the Pareto front and Pareto set plots of the total toe from three different scenarios against the roll steer. The results clearly show the great effect of the studied parameters on the kinematic characteristics of the vehicle. Quantitative values of toe objective functions demonstrate a percentage difference of 81.5% from scenario 1, 190.2% from scenario 2 and 12.3% from scenario 3 considering the minimum and maximum values. Moreover, a percentage difference of 21.6% is achieved from roll steer objective function.

It is also shown that the demands on the parameters with respect to toe variations and roll steer are contradictory. To achieve an optimum behavior, based on the Pareto curves, P_1 and P_2 should be selected close to their lower and upper limits, respectively.

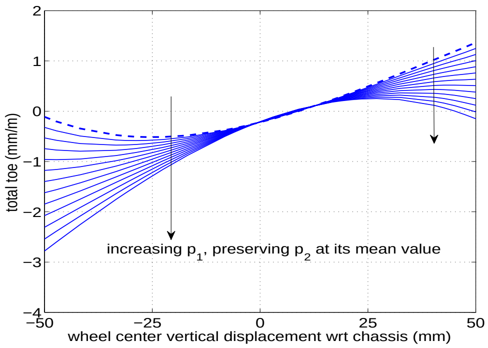


Figure 11: Influence of P_1 on total toe from scenario 2

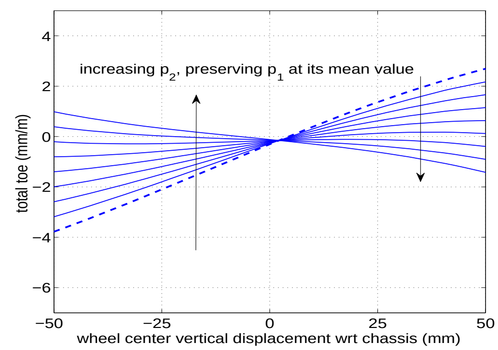


Figure 12: Influence of P_2 on total toe from scenario 2

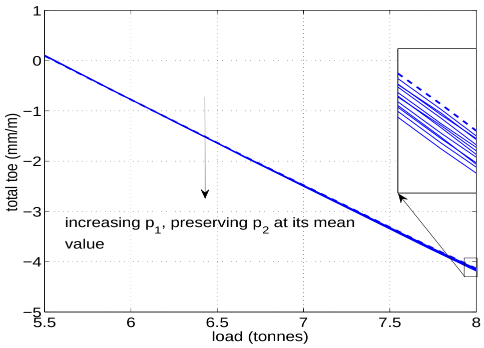


Figure 13: Influence of P_1 on total toe from scenario 3

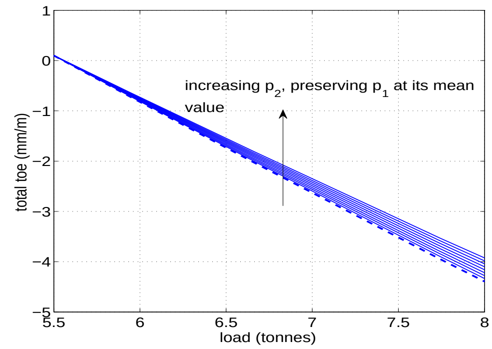


Figure 14: Influence of P_2 on total toe from scenario 3

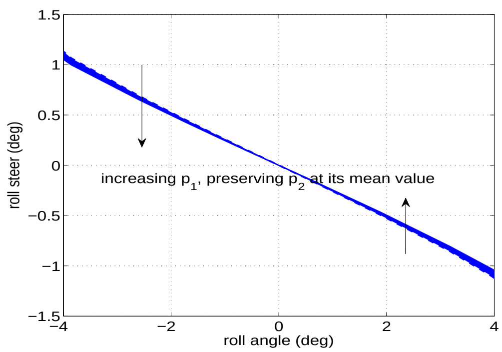


Figure 15: Influence of P_1 on roll steer

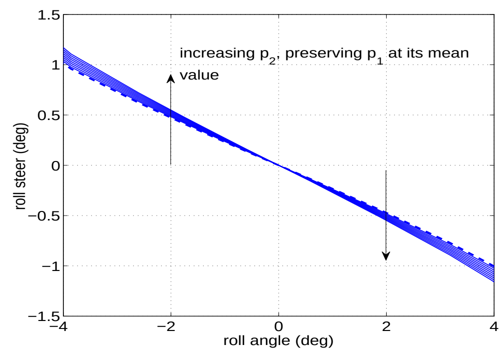
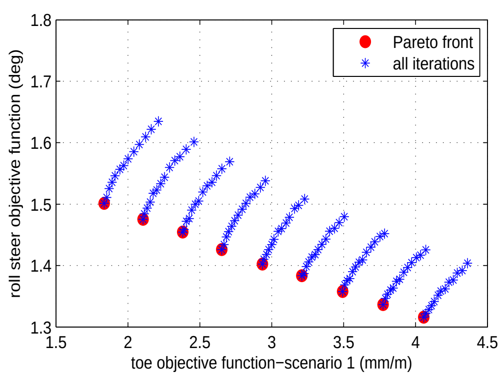
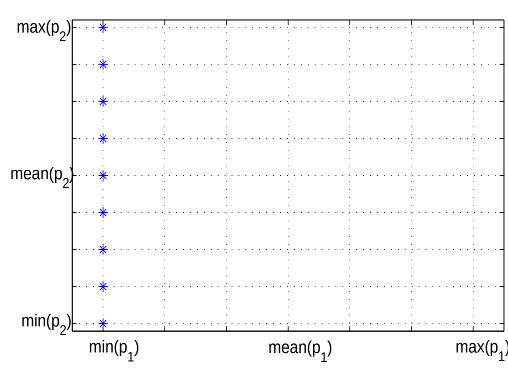


Figure 16: Influence of P_2 on roll steer

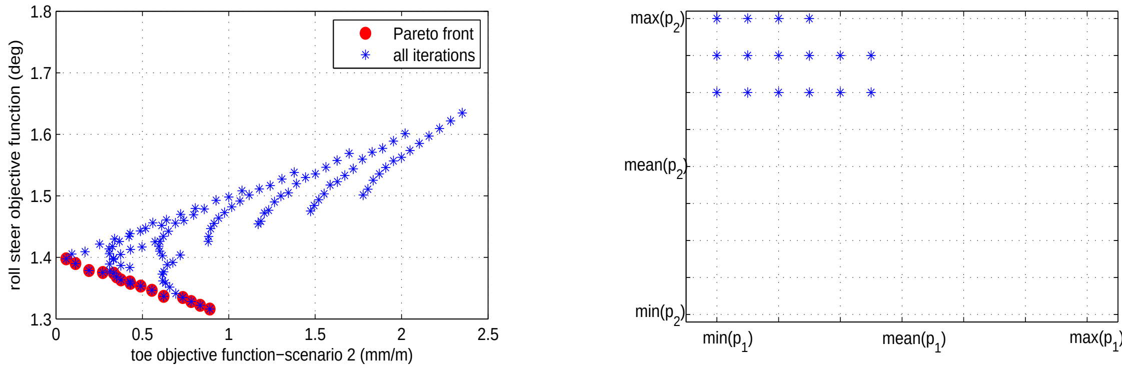


(a) *Pareto front*



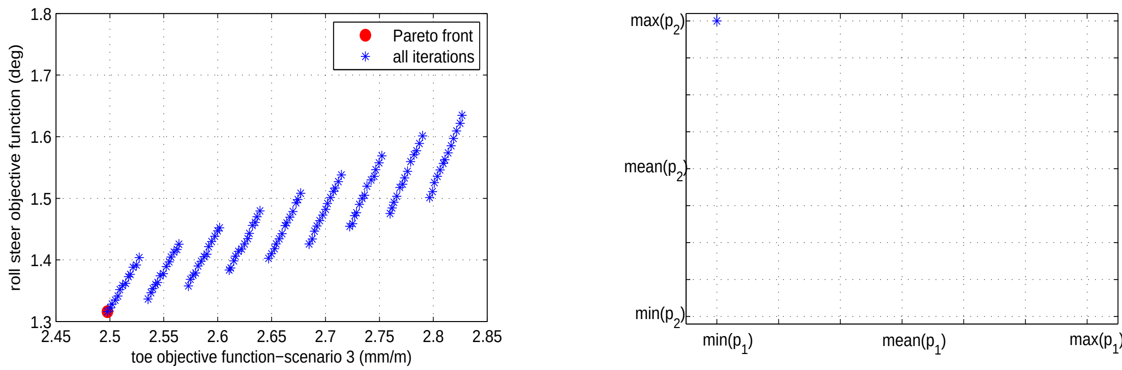
(b) *Pareto sets of the design parameters P_1 and P_2*

Figure 17: Pareto front and Pareto set for total toe (scenario 1) and roll steer objective functions



(a) *Pareto front* (b) *Pareto sets of the design parameters P_1 and P_2*

Figure 18: Pareto front and Pareto set for total toe (scenario 2) and roll steer objective functions



(a) *Pareto front* (b) *Pareto sets of the design parameters P_1 and P_2*

Figure 19: Pareto front and Pareto set for total toe (scenario 3) and roll steer objective functions

6 CONCLUSION

In this article, the kinematics of a heavy truck suspension equipped with IFS are presented. Characteristics such as roll steer, toe variation and track width change that are coupled to vehicle understeering and tire wear, respectively have been assessed through bump and roll simulations. The influences of two selected parameters on roll steer and toe properties are examined in a sensitivity analysis and the results are provided in Pareto fronts and Pareto sets, which not only show the sensitivity of kinematics on design parameters but also the contradictory requirements on them. It is concluded that for optimum kinematic properties, it is advantageous to keep the length of the rack and pinion as short as possible while choosing the z-coordinate of the connecting joint between the steering arm and tie rod toward its upper limit.

Acknowledgements

The Swedish agency for innovation systems, VINNOVA, and Volvo Truck Corporation are gratefully acknowledged for financing the project.

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