Heavy vehicle model to study ride performance

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Abstract: The objective of the computer-aided method is to provide engineers in industry with a realistic and practical tool for parametric analysis of off-road wheeled vehicle performance and design. Experience in the applications of vehicle modelling software's to performance and Design evaluation of off-road wheeled vehicles has demonstrated that it is a practical and use full tool for engineering practitioners. This paper presents the modeling and validation of a 7- degree of freedom (DOF) full vehicle model to study ride performance of a heavy vehicle. To improve suspension control system that can reduce roll over effect and improve ride comfort dynamic modeling of passive heavy vehicle model was constructed such simulation model was developed in MATLAB Simulink software. Several assumptions related to 7-degree of freedom modeling were made and stated in this paper. This heavy vehicle model was validated using vehicle dynamics simulation software known as TruckSim done on solid works. The validation was done by comparing the simulation results. A ride test was conducted at two different speeds and the simulation results consist of roll angle, pitch angle, and vehicle body displacement are analyzed.
[M.A.A. Mohamdeen, K.A. Abd El-Gawwad, M, M. M Salem and A.T. Osman. **Heavy vehicle model to study ride performance.** *J Am Sci* 2018;14(11):42-48]. ISSN 1545-1003 (print); ISSN 2375-7264 (online). http://www.jofamericanscience.org. 6. doi:10.7537/marsjas141118.06.

Keywords— Heavy vehicle, MATLAB/Simulink, Ride Model, Validation Model

1. Introduction

While A. W. Burton, A. J. Truscott and P. E. Wellstead, analyzed an advanced suspension system which is self-leveling. Y. Chen, J. He, W. Zhang, investigated how to optimize suspension system to improve ride comfort, work done by T.R.M. Rao, G.V. Rao, K.S. Rao, and A. Purushottam, to analyze performance of suspension system when vehicle passes over bump with speed. S.A. Pazooki, D. Cao, modeled off road vehicle ride dynamics. K. Hudha, H. Jamaluddin, P.M. Samin, make a research on how to improve suspension performance of light armored vehicles in order to make it reject disturbance when moving over un prepared road. D.J.M Sampson, G. McKevitt, D. Cebon, improved system to control active roll for the heavy vehicle to make it stable on the road. S.A.A Bakar, R. Masuda, H. Hashimoto, T. Inaba, H. Jamaluddin, R. Rahman, P.M. Samin, studied suspension of electric vehicles trying to improve ride comfort of them they used Magnetorheological Semi Active Suspension System. P.M. Samin, H. Jamaluddin, R.A. Rahman, S.A.A. Bakar, K. Hudha Modeled and Validated a 7-DOF Full Car for Ride Quality CADME07. While J.D. Setiawan, M. Safarudin, A. Singh, modeled and validated a 14 DOF Full Vehicle Model. Analyzed vehicle ride characteristics of tractor- trailer. Work of A. Forsén, focused on Heavy Vehicle Ride and Endurance–Modelling and in validating model to define its reliability.

In this research a heavy military heavy vehicle that transports troops needs high vehicle stability, ride comfort and road friendliness. This heavy vehicle is regularly driven on different terrains, and thus the stability of the vehicle needs to be studied to improve the vehicle ride performance. This simulation model was validated with vehicle simulation software to represent the vehicle’s ride behavior. in order to improve ride behaviour of these heavy military vehicles.

2. M Athematical Modeling.

The heavy vehicle ride model in this study is based on a four wheels vehicle. The ride model consists of 7-degrees of freedom which involves vehicle body bounce, pitch, roll, and four wheels vertical motions. Fig. 1 shows the vehicle ride model.

There are some assumptions made in this study. The vehicle aerodynamic effect is neglected and the road is assumed to be level except for road disturbance. The vehicle is also assumed rigid where the load transfer from one point to another is hundred percent effective. Parameters of the vehicle are also assumed to be constant throughout the simulation process such as tire stiffness, spring stiffness, and damper coefficient Based on the 7-degree of freedom model in Fig. 1, the displacement of sprung mass is defined by

$$m_B \hat{Z}_B = - F_{SFL} - F_{DFL} - F_{SFR} - F_{DFR} - F_{SRL} - F_{DRL} - F_{SRR} - F_{DRR} \quad (1)$$

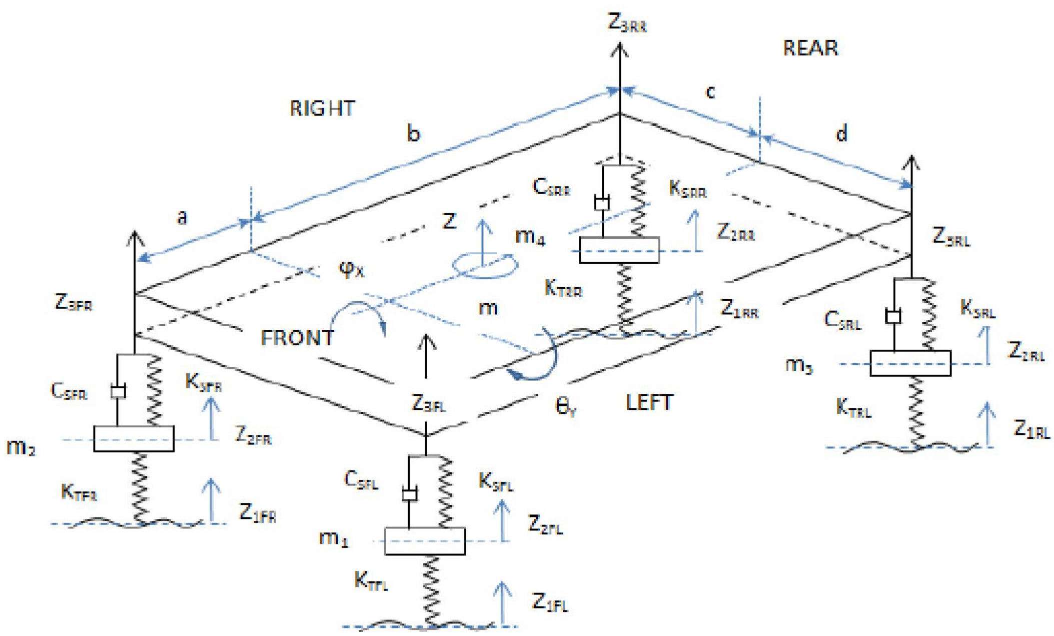

Fig. 1 Seven degree of freedom of vehicle ride model

Where m_B is the mass of the vehicle $\hat{Z}_B$ is the body acceleration and F is the force acting on vehicle model (S for spring, D for damper, FL for front left, FR for front right, RL for rear left, RR for rear right. The spring forces F_{sij} (i for front or rear and j for left or right) that act on the suspensions are given by

$$F_{sij} = k_{sij} (Z_{Bij} – Z_{Uij}) \tag{2}$$

Where Z_{Bij} is the sprung vertical displacement Z_{Uij} is the unsprung mass vertical displacement and K_{Sij} is the suspension spring stiffness. Then the damper forces F_{dij} of the suspensions are given by

$$F_{Dij} = C_{sij} (\dot{Z}_{Bij} – \dot{Z}_{Uij}) \tag{3}$$

Where $\dot{Z}_{Bij}$ is the sprung vertical velocity $\dot{Z}_{Uij}$ is the unsprung mass vertical velocity and C_{sij} is the suspension damper coefficient. Acceleration at unsprung mass is given by:

$$M_{uij}\ \hat{Z}_{uij} = F_{Sij} + F_{Dij} + F_{Tij} \tag{4}$$

Where M_{uij} is the unsprung vertical $\hat{Z}uij$ is the vertical acceleration at unsprung mass and F_{Ty} is the dynamic tyre forces. F_{Ty} is defined as:

$$F_{Ty} = k_{Tij} (Z_{Uij} – Z_{Rij}) \tag{5}$$

Where k_{Tij} is the tyre stiffness Z_{Rij} is the road profile where the disturbance on the road act. The pitch effect of the vehicle is given by:

$$J_y\ \ddot{\Theta} = – (F_{SFL} + F_{DFL} + F_{SFR} + F_{DFR})\ a + (F_{SRL} + F_{DRL} + F_{SRR} + F_{DRR})\ b \tag{6}$$

Where: J_y is the moment of inertia about x-axis and $\ddot{\Theta}$ is the pitch acceleration, while a is the length of vehicle from the center of gravity to the front end and b is the length of vehicle. From the center of gravity to the rear end of the vehicle. The roll effect of the vehicle can is given as follows:

$$J_X\ \ddot{\Theta} = – (F_{SFL} + F_{DFL} + F_{SFR} + F_{DFR})\ c + (F_{SFR} + F_{DFR} + F_{SRR} + F_{DRR})\ d \tag{7}$$

Where JX is the moment of inertia about x-axis and $\ddot{\Theta}$ is the pitch acceleration, while c is the length of the vehicle from the center of gravity to the right end and d is the length of vehicle from the center of gravity to the left end of the vehicle.

3 – Modeling And Validation Of 7-DOF Ride Model For Heavy Vehicle

Simulation of the heavy vehicle model was conducted by using Simulink. The model was then validated with solid works. A ride test was conducted for both simulations. The road profile as the disturbance was applied on the left tires and followed by the right tires for both simulations. The height and length of the bumps is 0.1 m (incremental elevation) and 5 m (station) respectively for both sides, as shown in Fig.2.

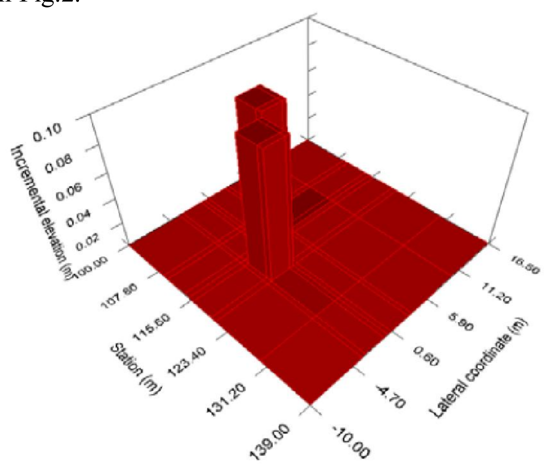

Fig.2 Road disturbance profile

Fig. 3 shows graphically the arrangement of the bumps and the vehicle movement when it hits the bumps. The left and the right bumps were arrange such that the change in the direction of disturbance occurs instantaneously.

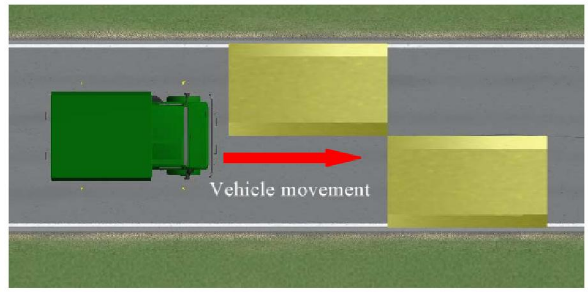

Fig. 3 Ride test road profile

Fig. 4 shows the Simulink block diagram of the ride model. The road profile disturbance acts on the unsprung mass system. The signal from unsprung mass block diagram namely suspension tire forces are transmitted to the sprung mass pitch, and roll block diagram to compute the output variable Then the output from sprung mass, pitch, and roll are fed back to the unsprung system.

The output variable namely pitch, roll, and sprung mass displacement are recorded and compared with TruckSim simulation results. All parameters of the vehicle are assume constant throughout simulation.

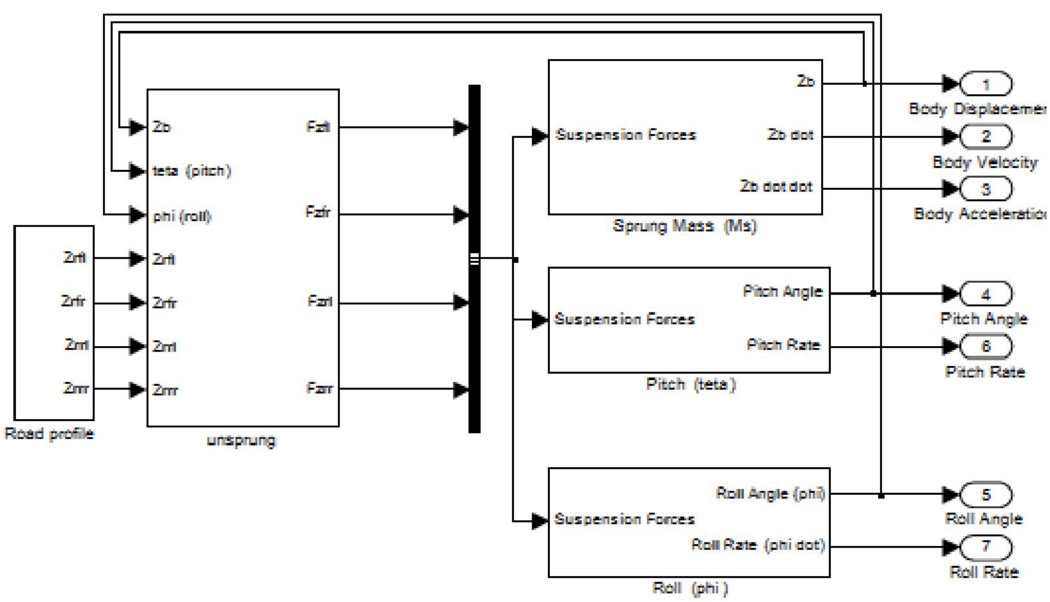

Fig. 4 Simulink block diagram

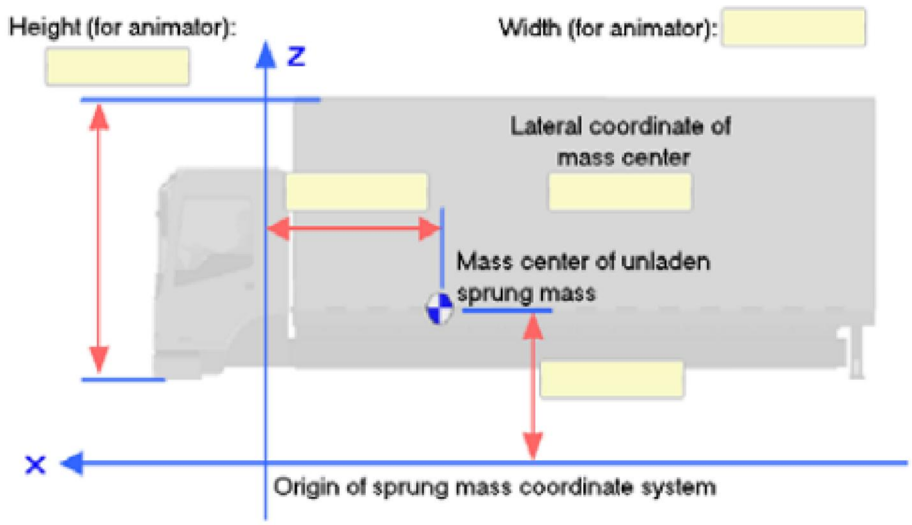

Fig. 5 User interface for heavy vehicle

4.2.2 TruckSim Model

The same source of road disturbance applied in Simulink simulation is used by the TruckSim. The heavy vehicle parameters are defined using the solid works user interface These are, spring, damper, tire, track width, vehicle length unsprung and sprung masses, moment of inertia at x and y axes. Fig 5 shows the user interface for the input parameters to be defined for the heavy vehicle that are used in the simulation in TruckSim.

5. Results and discussion

5.1 Simulation: 36 km/h

The performances of the simulation models studied are in3- Study of vehicle parameters using rvpm. Terms of pitch, roll, and vertical displacement responses are compared between Simulink and TruckSim simulation models. The road profile as shown in Fig. 5.2 was used as the road disturbance. The speed of the vehicle model is kept constant throughout the simulation that is 36 km/h Figs.6 to 8 show the simulation results of both Simulink and sloid works performances when passing through the external disturbance. The vehicle hits the first bump at 5 second on the left side and hit the second bump at 6 second on the right side. Figs. 7 and 8 show the Simulink and solid works simulation have similar trend but slightly different in magnitude. Fig.8 shows the simulation result of Simulink body displacement, which has the same trend as TruckSim simulation but a slight different in magnitude. This error maybe due to simplified model used in Simulink, while TruckSim model that is based on the actual tested vehicle simulation process thus becomes more precise.

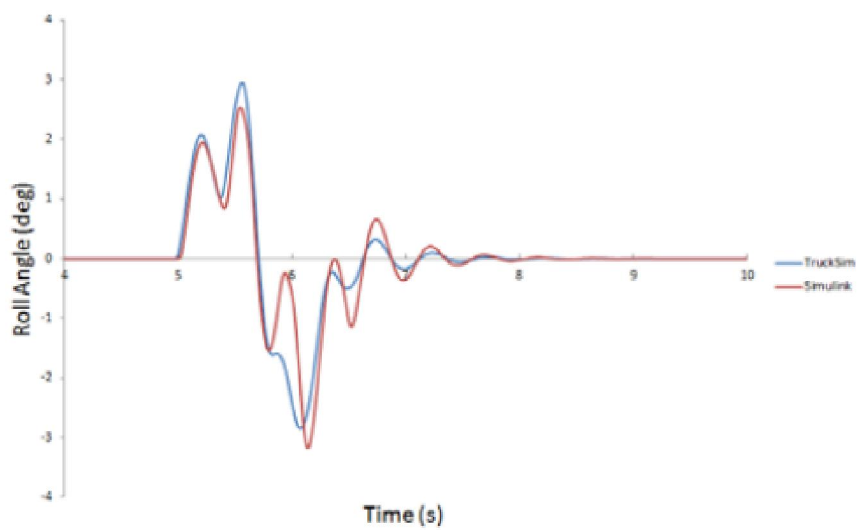

Fig. 6 Roll angle response at 36 km/h

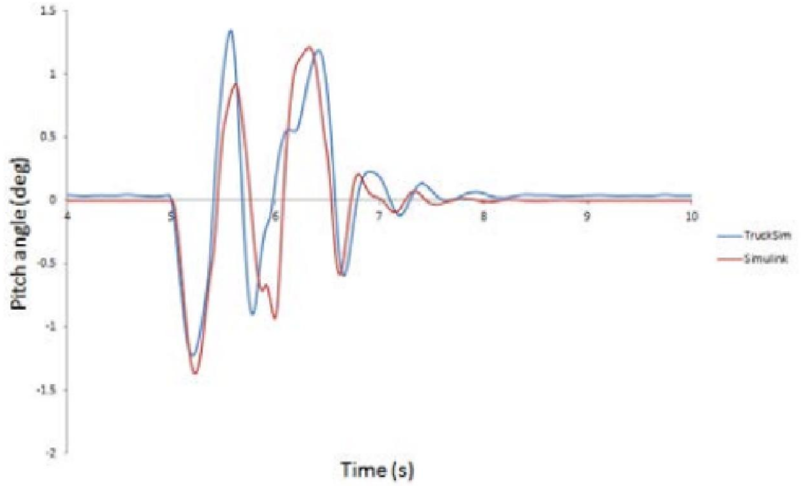

Fig. 7 Pitch angle response at 36 km/h

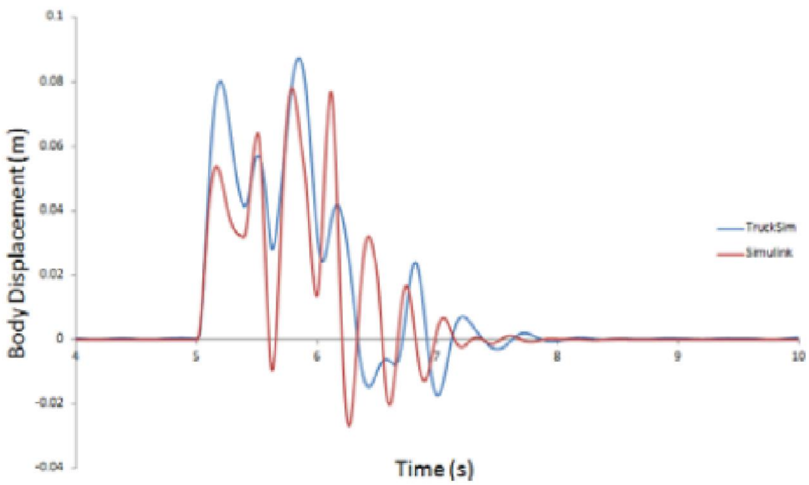

Fig. 8 Body displacement response at 36 km/h

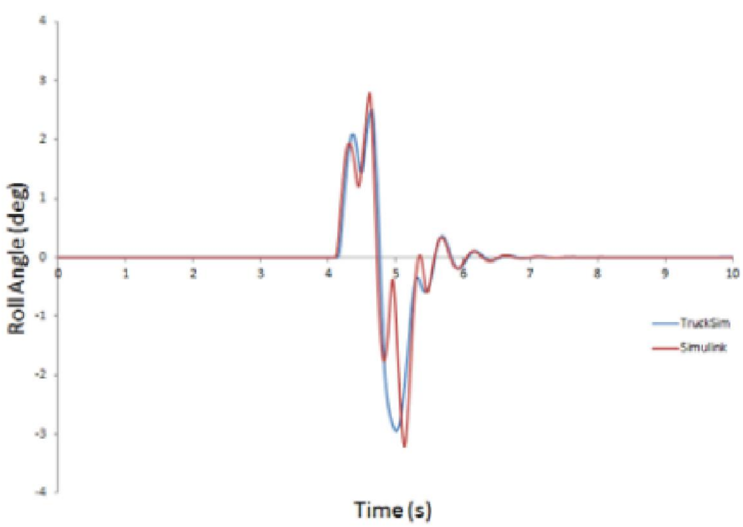

Fig. 9 Roll angle response at 43km/h

5.2 Simulation: 43 km/h

The result of the simulation at 43 km/h show similar trend of roll angle, pitch angle, and body displacement between Simulink and TruckSim. These are shown in Figs 9 to 11, the time taken to hit the first bump is a bit faster compared with the first simulation, because the speed of the Vehicle is faster. The time taken to hit the first bump for 43 km/h is about one second faster than 36 km/h the different vehicle speed used in the simulations is to show that the trend of the output of the Simulink model is consistent with independent speed. Figs. 9 to 11 show similar trends of the speed at 36 km/h are observed.

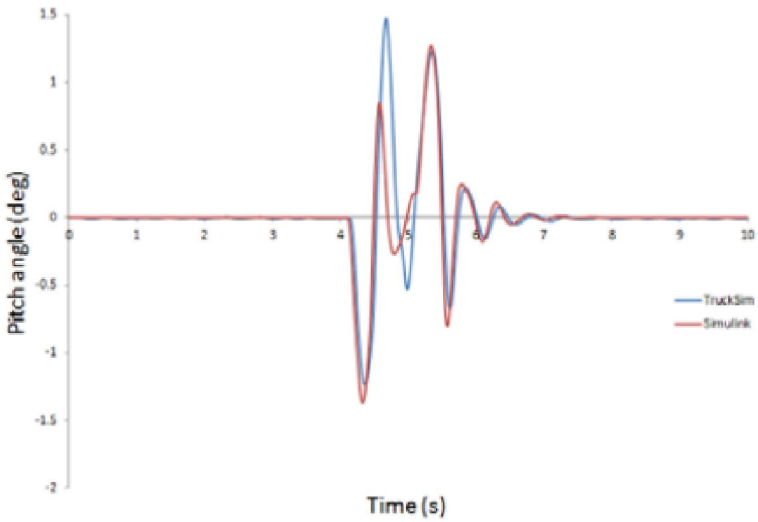

Fig. 10 Pitch angle response at 43km/h

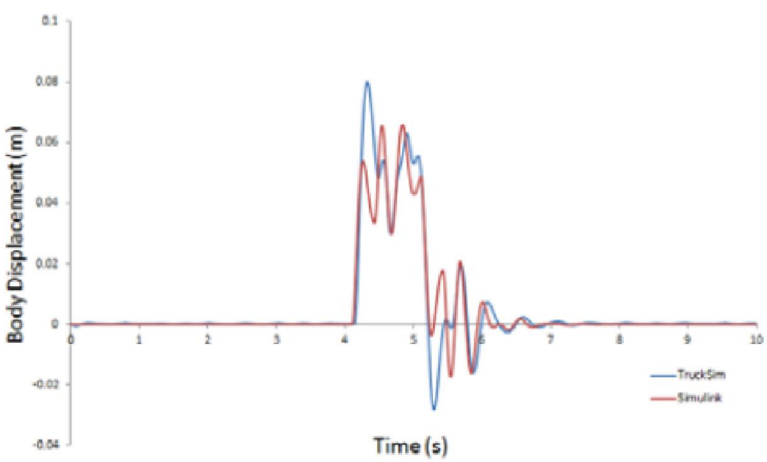

Fig. 11 Body displacement response at 43km/h

6. Conclusion

In comparison with the simplified methods for modeling vehicle terrain interaction described the computer-aided methods NWVPM and solid works. etc. can provide a realistic and quantitative assessment of the interaction of wheeled vehicles and tracked vehicles over unprepared terrains. Thus it demonstrates that computer aided methods are useful tools for adequately addressing the issue concerning design of heavy duty vehicles like:-

1. comparison of wheeled vehicles vs tracked vehicles from the traction perspective,
2. measurement and characterization of the mechanical properties of terrain pertinent to vehicle mobility,
3. The mechanics of vehicle–terrain interaction.

4. the study of vehicle–terrain interaction from the traction perspective,

5. Prediction of off-road vehicle performance. Through examples,

6. Applications of terra-mechanics to parametric analyses of terrain–vehicle systems and to the rational development and design of off-road vehicles from the traction perspective.

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11/18/2018