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Modelling and verification of tractor ride model

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Abstract. This project focuses on the modelling and verification of a tractor ride model as a part of tractor semi-trailer combination ride model development. The tractor ride model is established based on the three-axle tractor. The tractor ride model equations of motion, which consist of three degrees of freedom (DOF) of body motions and six DOF of unsprung mass motions are derived. The developed tractor ride model is simulated in Matlab-Simulink software. TruckSim software is used to verify the developed ride model by conducting the simulation on flat road condition. The vehicle model is simulated to travel at 90 km/h and hits 10 cm bump. The results of the study show that the dynamic behavior of the developed tractor ride model is closely followed the TruckSim tractor ride dynamic behavior. The percentage differences of RMS values between the developed and TruckSim model for all investigated parameters are less than 5%. It provides the confidence of utilizing the developed model established in this study for the purpose of developing the tractor semi-trailer combination model.

1. Introduction

Commercial vehicle, especially tractor semi-trailers have already become an essential mode of transporting goods especially in logistics and transportation industries. This is due to the tractor semi-trailer's capability to carry large quantity of goods and can access wider range of transportation area. However, in terms of maneuvering and dynamic stability, tractor semi-trailers possess low stability due to the vehicle high center of gravity [1]. This characteristic leads to the rollover condition of the tractor semi-trailer, especially during rapid lane change and excessive cornering at highway exit ramp. In addition, most of the road accidents due to rollover cause severe injuries to the truck driver. From statistical data of tractor semi-trailer rollover crashes from 1992 to 1996 indicated that this type of accident contributed up to 58% of total road fatalities of truck drivers [2]. Therefore, the dynamic analysis on the tractor semi-trailer combination needs to be conducted to identify the cause of the rollover.

In dynamic analysis, engineers are alternatively utilizing the computer simulation approach to evaluate and optimize the design concept [3]. In the simulation process, there are two approaches that can be used to model the vehicle. It consists of the usage of multi-body method to generate the equations of motion [4] and simplified modelling [5]. For a multi-body method, the vehicle is represented as rigid bodies and subjected to internal and external forces. Vehicle dynamic analysis software such as ADAMS [6] is used to generate the equations automatically. Meanwhile, in simplified modelling, there are three main types of simplified vehicle model that are commonly used in vehicle dynamic analysis, namely the quarter car, half car, and full car models. The analysis of vehicle model responses due to ride and handling are investigated separately. For the ride model, the input is assumed to be initiated from road bump. On the other hand, the handling model assumes that a vehicle is traveling on a flat road and encountering cornering and braking conditions. The developed vehicle model is then validated by using vehicle dynamic software such as TruckSim [7, 8] or real vehicle in order to produce an appropriate vehicle model, which can represent the real vehicle system.

In this study, the tractor model is developed based on 9 DOF ride model and begins with the derivation of sprung mass equations of motion. All the derived equations are modeled in Matlab-Simulink software. TruckSim software is used to verify the tractor model by conducting ride simulation test. According to Hafiz Harun, 2013 [9], the development and vehicle dynamics simulation of the tractor ride model by using Matlab-Simulink software are beneficial in terms of time and cost saving. The method of solving the problem by assembling physical data is the main concern by many researchers rather than writing the equations of motion [10]. Furthermore, by performing simulation, experimental costs such as the test circuit, test beds, devices, and drivers are not required [11].

This paper is organized as follows: The first section contains the introduction and review of some relevant preliminary works. It is followed by modelling and simulation of the tractor in second section. The third section introduces the tractor model verification by using TruckSim software and the results of the study are discussed in the fourth section. The final section contains the conclusion of the study.

2. Modelling of tractor ride model

Generally, tractor semi-trailer combination as shown in Figure 1 consists of a towing or lead unit, hitch, and one or more towed units [12]. In this study, the towing unit is introduced as a tractor while the towed unit is known as a semi-trailer. The hitch or coupling mechanism transmits the constrained forces and moments between tractor and semi-trailer. The tractor model utilized in this study consists of three axles with six tires and used to obtain the tire vertical force as illustrated in Figure 2. The sprung mass of the tractor generates the vertical, roll and pitch motions while the unsprung mass generates the vertical motion.

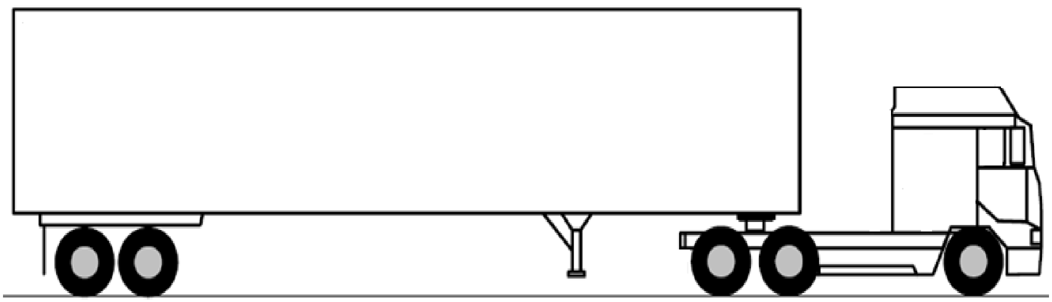

Figure 1. Tractor semi-trailer combination

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Figure 2. Tractor model

2.1 Modelling assumptions

Some of the modeling assumptions considered in this study are as follow: The tractor is modelled with 3 DOF body motion consists of vertical, roll and pitch motions while 6 DOF presents the vertical motion of each tire. The sprung and unsprung masses of the tractor model are connected to suspension system model that is represented as spring elements and viscous dampers. Furthermore, the tractor aerodynamic effect is ignored and the road is assumed to be a flat surface except for the road bumps. [13]. It is also assumed that during maneuvering, the tractor remains grounded at all times and all tires are contacted with the ground. Meanwhile, the tire vertical characteristic is established as a linear spring without a damper [14].

2.2 Equations of motion of tractor ride model

The derivation of equations of motion for tractor ride model is based on schematic diagram of a tractor model equipped with suspension system as shown in Figure 3 [15, 16].

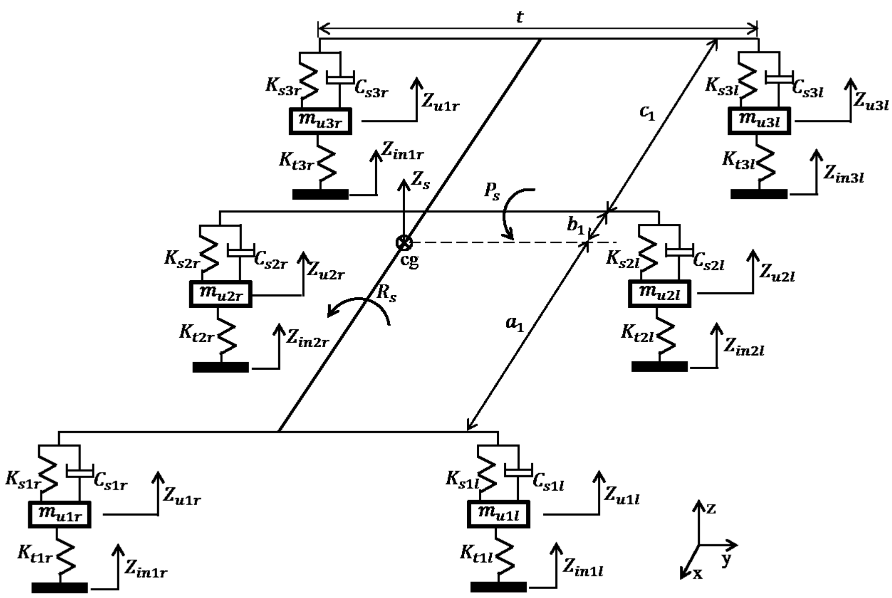

Figure 3. Schematic diagram of a tractor model equipped with suspension system

Schematic diagram of a tractor ride model equipped with suspension system as presented in Figure 3 contains the sprung mass, unsprung masses, springs, dampers and tires. Thus, the equations of motion are derived based on equations listed below:

$$\begin{aligned} m_s \ddot{Z}_s = {} & K_{s1l}(Z_{u1l} - Z_{s1l}) + C_{s1l}(\dot{Z}_{u1l} - \dot{Z}_{s1l}) + K_{s1r}(Z_{u1r} - Z_{s1r}) + C_{s1r}(\dot{Z}_{u1r} - \dot{Z}_{s1r}) + \\ & K_{s2l}(Z_{u2l} - Z_{s2l}) + C_{s2l}(\dot{Z}_{u2l} - \dot{Z}_{s2l}) + K_{s2r}(Z_{u2r} - Z_{s2r}) + C_{s2r}(\dot{Z}_{u2r} - \dot{Z}_{s2r}) + \\ & K_{s3l}(Z_{u3l} - Z_{s3l}) + C_{s3l}(\dot{Z}_{u3l} - \dot{Z}_{s3l}) + K_{s3r}(Z_{u3r} - Z_{s3r}) + C_{s3r}(\dot{Z}_{u3r} - \dot{Z}_{s3r}) \end{aligned} \tag{1}$$

where, m_s is the sprung mass;
$\ddot{Z}_s$ is the body vertical acceleration;
Z_{u1l} is the left unsprung mass motion on axle 1;
Z_{u1r} is the right unsprung mass motion on axle 1;
Z_{u2l} is the left unsprung mass motion on axle 2;
Z_{u2r} is the right unsprung mass motion on axle 2;
Z_{u3l} is the left unsprung mass motion on axle 3;
Z_{u3r} is the right unsprung mass motion on axle 3;
Z_{s1l} is the left body motion on axle 1;
Z_{s1r} is the right body motion on axle 1;
Z_{s2l} is the left body motion on axle 2;
Z_{s2r} is the right body motion on axle 2;
Z_{s3l} is the left body motion on axle 3;
Z_{s3r} is the right body motion on axle 3;
$\dot{Z}_{u1l}$ is the left unsprung mass vertical velocity on axle 1;
$\dot{Z}_{u1r}$ is the right unsprung mass vertical velocity on axle 1;
$\dot{Z}_{u2l}$ is the left unsprung mass vertical velocity on axle 2;
$\dot{Z}_{u2r}$ is the right unsprung mass vertical velocity on axle 2;
$\dot{Z}_{u3l}$ is the left unsprung mass vertical velocity on axle 3;
$\dot{Z}_{u3r}$ is the right unsprung mass vertical velocity on axle 3;
$\dot{Z}_{s1l}$ is the left body vertical velocity on axle 1;
$\dot{Z}_{s1r}$ is the right body vertical velocity on axle 1;
$\dot{Z}_{s2l}$ is the left body vertical velocity on axle 2;
$\dot{Z}_{s2r}$ is the right body vertical velocity on axle 2;
$\dot{Z}_{s3l}$ is the left body vertical velocity on axle 3;
$\dot{Z}_{s3r}$ is the right body vertical velocity on axle 3;
K_{s1l} is the left spring stiffness on axle 1;
K_{s1r} is the right spring stiffness on axle 1;
K_{s2l} is the left spring stiffness on axle 2;
K_{s2r} is the right spring stiffness on axle 2;
K_{s3l} is the left spring stiffness on axle 3;
K_{s3r} is the right spring stiffness on axle 3;
C_{s1l} is the left damping coefficient on axle 1;
C_{s1r} is the right damping coefficient on axle 1;
C_{s2l} is the left damping coefficient on axle 2;
C_{s2r} is the right damping coefficient on axle 2;
C_{s3l} is the left damping coefficient on axle 3;
C_{s3r} is the right damping coefficient on axle 3.

Similarly, moment balance equations are derived for pitch P_s and roll R_s, and are given as

$$\begin{aligned} I_{yy}\ddot{P}_s = -\Big[K_{s1l}(Z_{s1l}-Z_{u1l})+C_{s1l}(\dot{Z}_{s1l}-\dot{Z}_{u1l})+K_{s1r}(Z_{s1r}-Z_{u1r})+C_{s1r}(\dot{Z}_{s1r}-\dot{Z}_{u1r})\Big]a_1 + \\ \Big[K_{s2l}(Z_{s2l}-Z_{u2l})+C_{s2l}(\dot{Z}_{s2l}-\dot{Z}_{u2l})+K_{s2r}(Z_{s2r}-Z_{u2r})+C_{s2r}(\dot{Z}_{s2r}-\dot{Z}_{u2r})\Big]b_1 + \\ \Big[K_{s3l}(Z_{s3l}-Z_{u3l})+C_{s3l}(\dot{Z}_{s3l}-\dot{Z}_{u3l})+K_{s3r}(Z_{s3r}-Z_{u3r})+C_{s3r}(\dot{Z}_{s3r}-\dot{Z}_{u3r})\Big](b_1+c_1) \end{aligned} \tag{2}$$

$$\begin{aligned} I_{xx}\ddot{R}_s = \begin{bmatrix} K_{s1l}(Z_{s1l}-Z_{u1l})+C_{s1l}(\dot{Z}_{s1l}-\dot{Z}_{u1l})+K_{s2l}(Z_{s2l}-Z_{u2l})+C_{s2l}(\dot{Z}_{s2l}-\dot{Z}_{u2l})+ \\ K_{s3l}(Z_{s3l}-Z_{u3l})+C_{s3l}(\dot{Z}_{s3l}-\dot{Z}_{u3l}) \end{bmatrix}\frac{t}{2} - \\ \begin{bmatrix} K_{s1l}(Z_{s1l}-Z_{u1l})+C_{s1l}(\dot{Z}_{s1l}-\dot{Z}_{u1l})+K_{s2l}(Z_{s2l}-Z_{u2l})+C_{s2l}(\dot{Z}_{s2l}-\dot{Z}_{u2l})+ \\ K_{s3l}(Z_{s3l}-Z_{u3l})+C_{s3l}(\dot{Z}_{s3l}-\dot{Z}_{u3l}) \end{bmatrix}\frac{t}{2} \end{aligned} \tag{3}$$

where, I_{yy} is the pitch mass moment of inertia;
I_{xx} is the roll mass moment of inertia;
$\ddot{P}_s$ is the body pitch acceleration;
$\ddot{R}_s$ is the body roll acceleration;
a_1 is the distance from center of gravity to the axle 1;
b_1 is the distance from center of gravity to the axle 2;
c_1 is the distance from center of gravity to the axle 3;
t is the track width.

By performing force balance analysis at all six wheels, the following equations are obtained

$$K_{s1l}(Z_{s1l}-Z_{u1l})+C_{s1l}(\dot{Z}_{s1l}-\dot{Z}_{u1l})-K_{t1l}(Z_{u1l}-Z_{in1l})=m_{u1l}\ddot{Z}_{u1l} \tag{4}$$

$$K_{s1r}(Z_{s1r}-Z_{u1r})+C_{s1r}(\dot{Z}_{s1r}-\dot{Z}_{u1r})-K_{t1r}(Z_{u1r}-Z_{in1r})=m_{u1r}\ddot{Z}_{u1r} \tag{5}$$

$$K_{s2l}(Z_{s2l}-Z_{u2l})+C_{s2l}(\dot{Z}_{s2l}-\dot{Z}_{u2l})-K_{t2l}(Z_{u2l}-Z_{in2l})=m_{u2l}\ddot{Z}_{u2l} \tag{6}$$

$$K_{s2r}(Z_{s2r}-Z_{u2r})+C_{s2r}(\dot{Z}_{s2r}-\dot{Z}_{u2r})-K_{t2r}(Z_{u2r}-Z_{in2r})=m_{u2r}\ddot{Z}_{u2r} \tag{7}$$

$$K_{s3l}(Z_{s3l}-Z_{u3l})+C_{s3l}(\dot{Z}_{s3l}-\dot{Z}_{u3l})-K_{t3l}(Z_{u3l}-Z_{in3l})=m_{u3l}\ddot{Z}_{u3l} \tag{8}$$

$$K_{s3r}(Z_{s3r}-Z_{u3r})+C_{s3r}(\dot{Z}_{s3r}-\dot{Z}_{u3r})-K_{t3r}(Z_{u3r}-Z_{in3r})=m_{u3r}\ddot{Z}_{u3r} \tag{9}$$

where, K_{t1l} is the left tire stiffness on axle 1;

K_{t1r} is the right tire stiffness on axle 1;
K_{t2l} is the left tire stiffness on axle 2;
K_{t2r} is the right tire stiffness on axle 2;
K_{t3l} is the left tire stiffness on axle 3;
K_{t3r} is the right tire stiffness on axle 3;
m_{u1l} is the left unsprung mass on axle 1;
m_{u1r} is the right unsprung mass on axle 1;
m_{u2l} is the left unsprung mass on axle 2;
m_{u2r} is the right unsprung mass on axle 2;
m_{u3l} is the left unsprung mass on axle 3;
m_{u3r} is the right unsprung mass on axle 3;
$\ddot{Z}_{u1l}$ is the left unsprung mass vertical acceleration on axle 1;
$\ddot{Z}_{u1r}$ is the right unsprung mass vertical acceleration on axle 1;
$\ddot{Z}_{u2l}$ is the left unsprung mass vertical acceleration on axle 2;
$\ddot{Z}_{u2r}$ is the right unsprung mass vertical acceleration on axle 2;
$\ddot{Z}_{u3l}$ is the left unsprung mass vertical acceleration on axle 3;
$\ddot{Z}_{u3r}$ is the right unsprung mass vertical acceleration on axle 3.

2.3 Modelling of tractor ride model

The derived tractor model equations are modeled in Matlab-Simulink software as depicted in Figure 4. The derived equations involve the sprung mass, unsprung masses, tires, springs and dampers. The input data for the tractor ride test is established when the right side tires hit the 10 cm height bump at the speed of 90 km/h. This process starts with the parameters identification and the parameters are established based on Liu [12]. All the tractor parameters used in the simulation are shown in Table 1. Heun solver is used to simulate the tractor model with fixed step size of 0.01 seconds. The outputs of the simulation are the body vertical acceleration, body roll angle and body pitch angle.

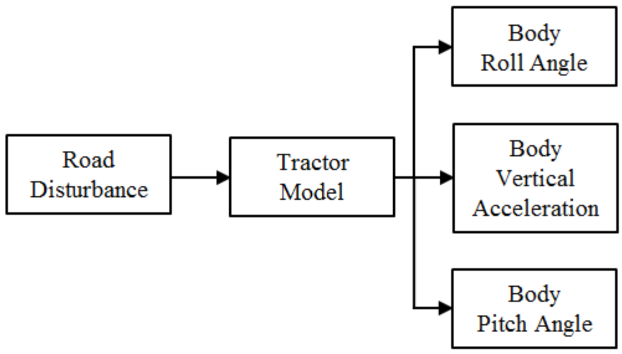

Figure 4. Tractor model in Matlab-Simulink

Table 1. The tractor parameters

Parameter	Values
m_s (kg)	4404
m_{u1} (kg)	624.5
$m_{u2,3}$ (kg)	1068
I_{xx} (kg.m^2)	6967.79
I_{yy} (kg.m^2)	19943.9
K_{s1} (N/m)	250000
$K_{s2,3}$ (N/m)	700000
C_{s1} (Ns/m)	15000
$C_{s2,3}$ (Ns/m)	30000
$K_{t1,2,3}$ (N/m)	875000
t (m)	2.032
a_1 (m)	1.385
b_1 (m)	2.115
c_1 (m)	3.385

3. Verification of tractor ride model

In this section, the tractor ride model is verified by using the TruckSim software. The tractor simulation ride test is conducted by using the three-axle cab over tractor. The truck travels at constant speed of 90 km/h on flat road and the tractor right side tires hit the 10 cm height bump at 1.6 seconds to generate the tractor motion responses. The TruckSim feature of the tractor ride test is shown in Figure 5.

Figure 5. Tractor ride test in TruckSim

4. Results and Discussion

This section discusses the simulation results of the proposed tractor ride model in comparison with the obtained verification results of using TruckSim software. Figure 6 presents the tractor roll angle responses of the ride simulation test at the speed of 90 km/h. The root mean square (RMS) values for proposed and TruckSim models are established in order to evaluate the performance of the tractor ride model [17]. The proposed tractor model roll angle response is observed to closely follow the Trucksim response with 3.35% error. Henceforth, it can be seen that the maximum error range is less than 5% [17]. The percentage of RMS values differences for all investigated parameters between proposed model and TruckSim model are presented in Table 2.

Table 2. Percentage values of RMS differences

Tractor Responses	RMS		Percentage Difference (%)
	Proposed Model	TruckSim Model	
Roll Angle	0.0231	0.0239	3.35
Pitch Angle	0.1125	0.1176	4.34
Vertical Acceleration	0.1176	0.1227	4.16

The maximum roll angle generated by the tractor is 3.4 degree and significantly stabilizes in 3.2 seconds. It can be observed that the suspension system is able to reduce the roll motion and retreat the vehicle back to normal condition and avoids the rollover condition. Hence, when hitting the 10 cm height bump at the speed of 90 km/h, the tractor suspension system is capable to absorb the external force and stabilizes the vehicle with a quick response. From Figure 6, it justifies the maximum speed limits within 80 km/h to 90 km/h enforced by the Malaysia Department of Transportation for a commercial vehicle is adequate to ensure the safety of commercial vehicle due to rolling condition. [18].

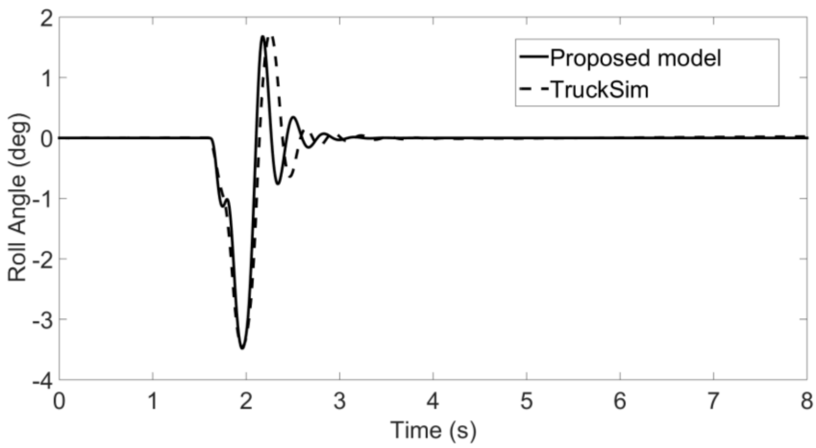

Figure 6. Tractor roll angle

The tractor pitch angle response of the ride simulation test at the speed of 90 km/h is shown in Figure 7. From this figure, it can be observed that the tractor pitch angle response is initially generated at 1.6 seconds when the tractor front tire starts to hit the bumps. In terms of tractor model verification, it can be noticed that the proposed tractor model response is closely followed the TruckSim model response with 4.34% error. The maximum pitch angle generated by the TruckSim model is 1.03 degree as compared to the proposed model of 0.88 degree. The difference is due to the effect of center of gravity distance between the front and rear axle. It significantly influences the moment of inertia of the pitch moment.

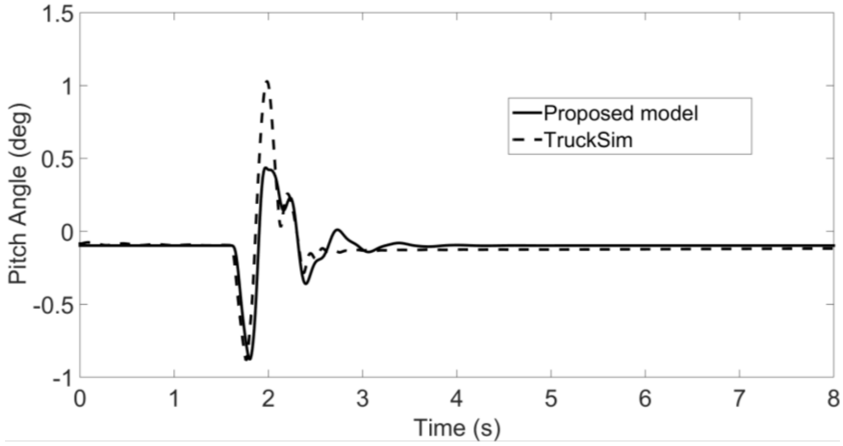

Figure 7. Tractor pitch angle

Figure 8 illustrates the tractor vertical acceleration of the simulation ride test at the speed of 90 km/h. It can be observed from Figure 8 that the proposed tractor model response is significantly in close agreement with the TruckSim model with 4.16% error. The maximum vertical acceleration produced by the TruckSim model is 14.3 m/s^2, which is slightly higher than the proposed model of 11.5 m/s^2. It is due to the effect of the tractor weight and the difference of parameters used for front and rear suspension system. Thus, it influences the tractor vertical motion.

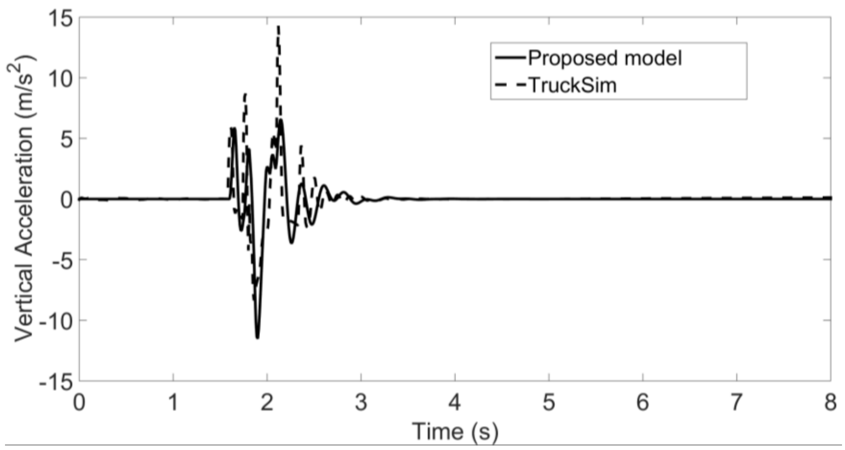

Figure 8. Tractor vertical acceleration

5. Conclusions

In conclusion, the modelling of tractor ride model equations of motion had been successfully established. The proposed tractor ride model equations of motion were effectively modelled and simulated in Matlab-Simulink software. The dynamic responses of the developed model were successfully verified by using TruckSim software. From the results, it demonstrated significant closeness between the proposed and TruckSim model responses. The percentage differences of RMS values between the proposed and TruckSim model for all investigated parameters were less than 5%. It provides the confidence of utilizing the proposed model established in this study for the purpose of developing the tractor semi-trailer combination model.

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