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Handling Stability Simulation and Test Research on an 8×4 Type Construction Truck

Gu Yufeng [1] and Dong Fulong [2]

[1] *Key Laboratory for Highway Construction Technology and Equipment of Ministry of Education, Chang'an University*
E-mail: guyufeng@chd.edu.cn
[2] *College of Mechanical Engineering, Anhui Science and Technology University*
E-mail: dongfl@ahstu.edu.cn

Abstract

Full-vehicle model plays a key role in vehicle handling stability simulations, and the same vehicle model does not adapt to all kinds of vehicle simulations. In the thesis, the vehicle mechanisms of an 8×4 type construction truck were appropriately simplified and the full-vehicle parameterized multi-body dynamic model was developed according to the topology mechanical map of each part. The vehicle model suitability was researched through comparing the vehicle simulation results with the test results in lemniscate and returnability simulations. The comparison results show that the lemniscate simulation errors of the steering wheel angle, yaw angle velocity and lateral acceleration amplitude of the cargo mass centre are 3.9%, 4% and 4.4% respectively, and the returnability simulation errors of the residual steering wheel angle, residual yaw angle velocity and lateral acceleration of the cargo mass centre are 95%, 77.8% and 80% respectively. The simulation and test results all show that the lemniscate simulation errors are smaller than the returnability simulation errors because the model input was the steering wheel angular displacement in lemniscate simulation while the model input was the forces caused from the road and vehicle movement in returnability simulation. In brief, there exist significant error variations for the same vehicle model in different simulations.

Keywords*: 8×4 type construction truck, ADAMS full-vehicle modelling, Handling stability simulation, Vehicle test, Comparison*

1. Introduction

Full-vehicle multi-body dynamic model developing plays a more and more important role in vehicle handling stability simulations [1]. In the simulations, the interaction relationships between each moving components in full-vehicle model are more complex than that in ride comfort simulations, and this will place greater demands on the vehicle model precision. For example, steering wheel angular displacement is loaded in lemniscate simulation and then the model outputs corresponding responses, while in returnability simulation, the model outputs responses according to its' mechanical behavior. In other words, the simulation input, inner components behavior and relative motion are different for the same vehicle model in different kind of simulations, and these will lead to great difference in simulation precision. Andrew Hall, S Hegazy and Shaomin Lou [2, 3, 4] discussed the simplification and building of full-vehicle model, and Jianxing Huang, Daisheng Zhang and Zehao Huang [1, 5, 6] researched the vehicle handling simulations. However, the differences between different simulations using the same vehicle model were seldom researched in the available literatures, and these were very important in validating the full-vehicle model.

Steering and suspension system models are the most important component parts in vehicle model in handling stability simulations [7, 8]. Steering drive mechanism

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modelling are the crux in steering system modelling, and the crux in suspension modelling is the steering axle leaf spring modelling, equalizing suspension leaf spring in driving axle and guide mechanism modelling. Steering mechanism model and suspension guide mechanism model were commonly developed according to the vehicle Pro/E model, and the leaf spring models were usually built using SAE 3-Link method which only consists of several components [9]. Heavy duty truck models were difficult in parameterizations because they included more parts and parameters than car models in the chassis system [2]. Topology mechanical map as used in [4, 10] demonstrated the inner connection relationships of the system components clearly, and it had become the modelling method of complex dynamic system. In the thesis, the available research achievements were summarized, and the parameterized full-vehicle model of an 8×4 type construction truck was developed in ADAMS software based on the vehicle topology mechanical maps. The model consisted of the steering system, steering axle suspension, equalizing suspension, wheels, body and *etc*. The suitability of the vehicle model was analyzed through comparing the handling stability simulations with the test results.

2. ADAMS Vehicle Model Developing

In the thesis, an 8×4 type construction truck model was built in ADAMS software based on the vehicle topology mechanical maps. The model assumed that all the components in the model were rigid bodies except the elastic components and anti-roll stabilizer, and each joint clearance and drive train influences were all neglected. The model also assumed that the cab, cargo, engine, gearbox and clutch were all connected with frame using fixed joints. The coordinate system was specified as following. The positive direction of axis x is the vehicle forward direction. The positive direction of axis y is the vehicle left direction. The positive direction of axis z is the straight up direction. The origin of the coordinates o was located at the middle point of the first axle.

2.1. Steering Drive Mechanism Modeling

The steering drive mechanism consisted of the steering wheel, steering column, steering gear, the first axle swing arm, the first axle drag link, the first axle steering knuckle arm, the first axle left knuckle, the first axle steering trapezoidal mechanism, middle connecting rod, middle swing arm, the second axle swing arm, the second axle drag link, the second axle steering knuckle arm, the second axle left knuckle, and the second axle steering trapezoidal mechanism. According to the steering movement transfer path, the steering drive mechanism topology mechanical map could be developed, which showed the connection relationships clearly within the steering drive mechanism in a graphic form. Thus the ADAMS model of steering drive mechanism shown in Figure **1** was built based on its topology mechanical map.

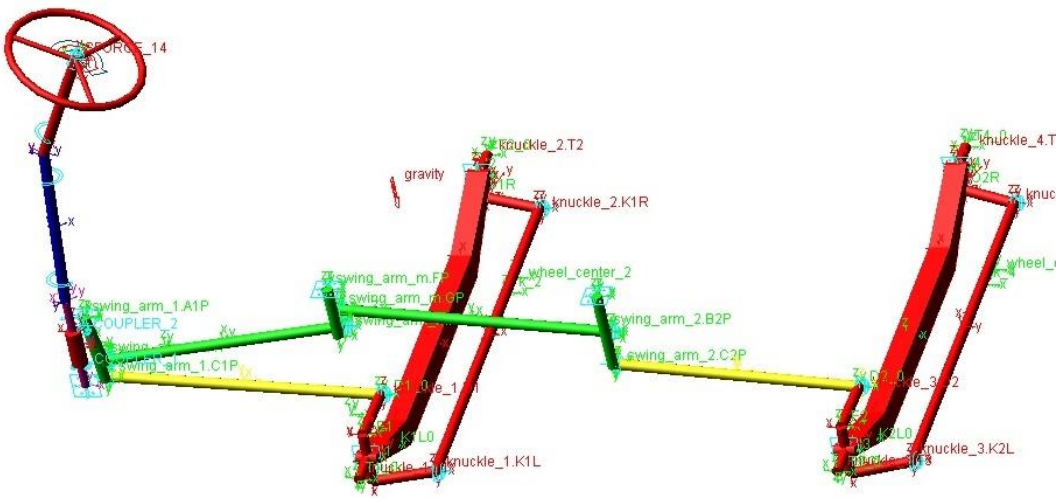

Figure 1. Steering Drive Mechanism ADAMS Model

2.2. Suspension System Modeling

The steering axle suspension system consisted of each steering axle leaf springs, back eyes, leaf spring seat, saddle clamp bolt, damper and *etc*. The equalizing suspension system in driving axles consisted of the equalizing suspension leaf springs, leaf spring seat, suspension mandrel, guide mechanism and *etc*. According to the SAE 3-Link leaf spring theory, the topology mechanical maps of the steering axle suspension and equalizing suspension in driving axles could be developed respectively, which showed the connection relationships clearly within the components of the leaf spring models. The ADAMS models of leaf springs shown in Figure **2** and Figure **3** were built respectively based on each mechanical map. The damper model was built using the translational spring-damper command in ADAMS software, and the guide mechanism model was built using the straight rods. The other components models in the suspensions were developed based on the vehicle Pro/E model.

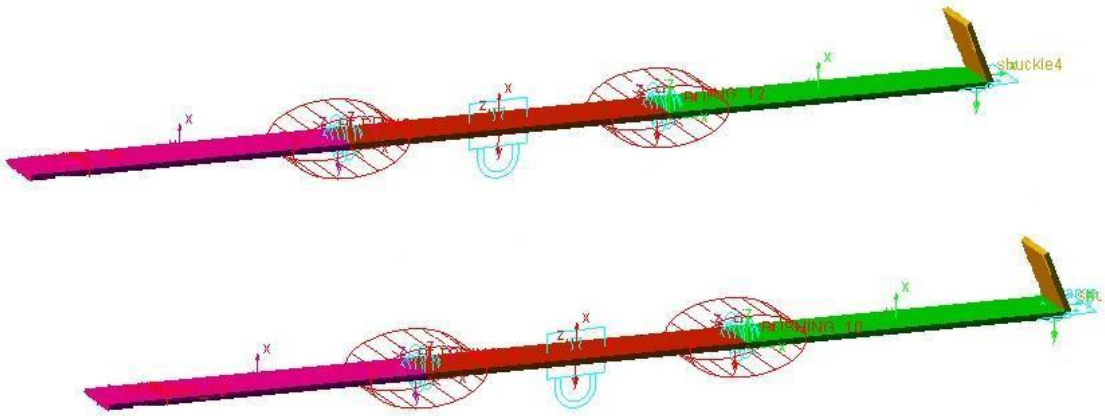

Figure 2. Steering Axle Suspension ADAMS Model

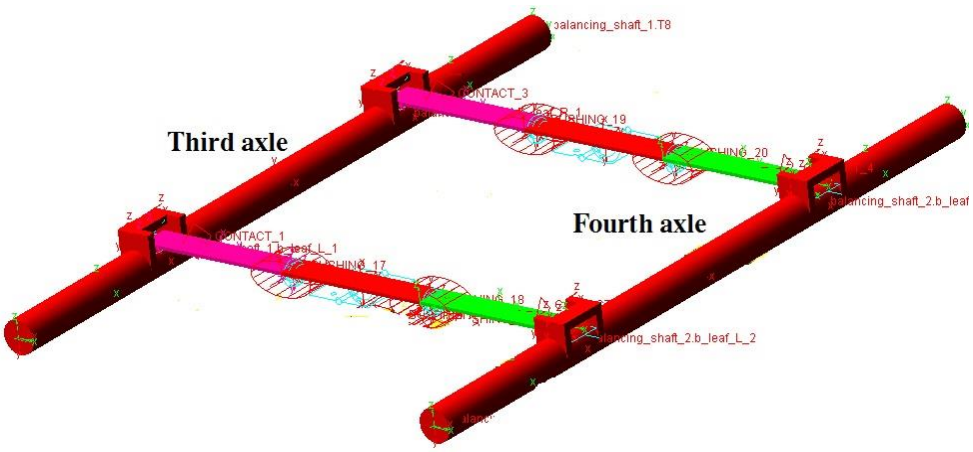

Figure 3. Equalizing Suspension ADAMS Model

2.3. Anti-Roll Stabilizer Modeling

The anti-roll stabilizer was split into two parts from the midpoint at which equivalent torsion spring was used to link the two parts. The left and right parts were linked with the steering axle using revolute joints respectively. The anti-roll stabilizer model was also developed according to its topology mechanical map, which showed the connection relationships clearly within the components of the anti-roll stabilizer model. The ADAMS model of the anti-roll stabilizer was shown in Figure **4**.

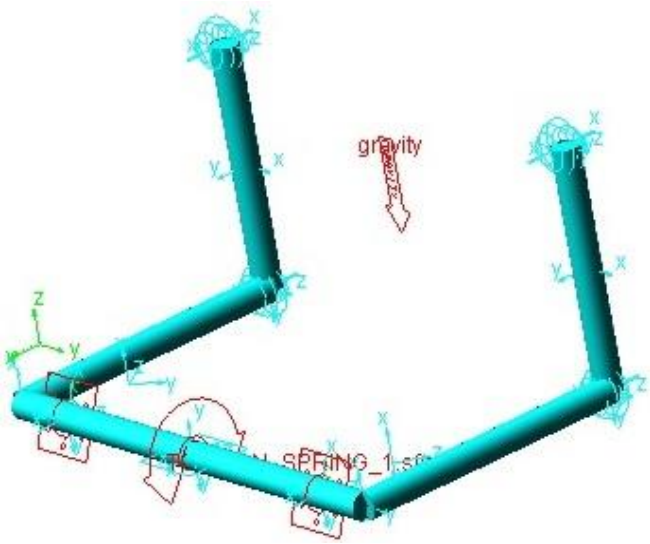

Figure 4. Anti-Roll Stabilizer ADAMS Model

2.4. Wheels, Body and Full-Vehicle Modeling

The wheel models adopted the pac2002 model. The frame, cab and cargo models were developed though importing the Pro/E model into ADAMS software. The engine and gearbox were replaced by quality points. Synthesized all the component models stated above, the full-vehicle model could be developed as shown in Figure **5**. Main structural parameters of the vehicle were listed in appendices.

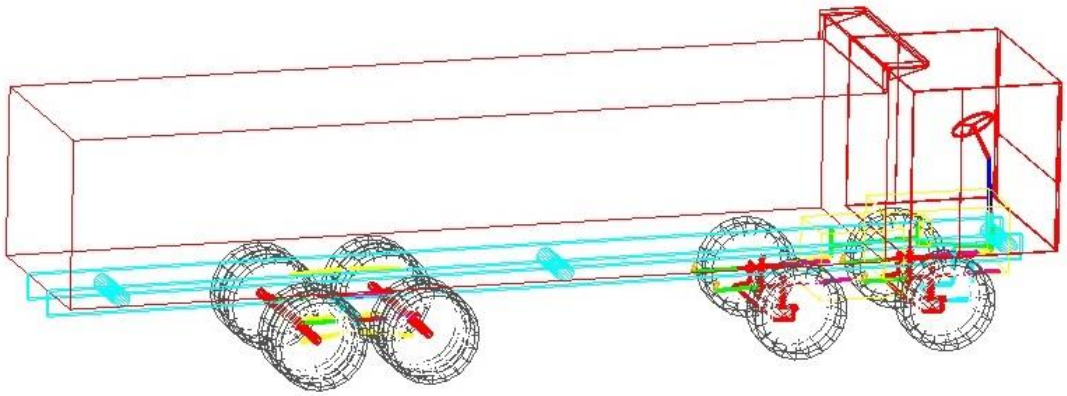

Figure 5. 8×4 Type Construction Truck Full-Vehicle ADAMS Model

3. Full-vehicle Handling Stability Tests

Filled the cargo with sand, and the vehicle curb mass was over than 30 tons, therefore protection supports shown in Figure **6** were designed for the handling stability tests. The test instruments were shown from Figure **7** to Figure **9**. Steering wheel parameters test instrument shown in Figure **7** was used to measure the angle and torque of the steering wheel, and the units were 'deg' and ' N·m ' respectively. GPS shown in Figure **8** was used to measure the vehicle velocity, and the unit was 'Km/h'. 6-freedom gyroscope embedded at the cargo mass centre shown in Figure **9** was used to measure the three axle accelerometers, rotational angular displacements and velocities, and the units were ' mm/s^2', 'deg', and 'deg/s' respectively. The vehicle tests were carried out according to GB/T 6323.3-1994.

Figure 6. Protection Supports for Handling Stability Tests

Figure 8. GPS Velocity Test Instrument

Figure 7. Steering Wheel Parameters Test Instrument

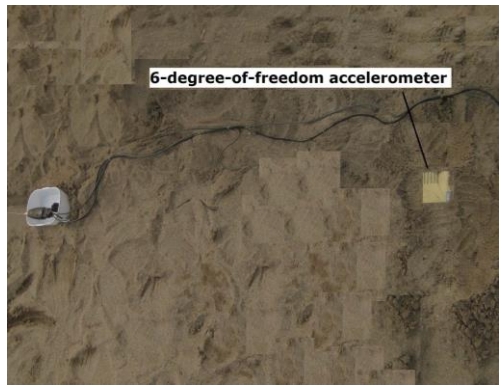

Figure 9. 6-Freedom Gyroscope

4. ADAMS Model Simulation and Vehicle Test Analyzing

Limited by the length of thesis, only the leniscate and returnability simulations were researched compared with the corresponding test results.

4.1. Lemniscate Simulation and Test

Set end time for 70s, and set steps for 7000 in the ADAMS model simulation. The lemniscate simulation was carried out on B-grade road with the vehicle velocity of 11Km/h. The simulation trajectory was shown in Figure **10**, where $R_{\min} = 13.6m$ and $D = 40.3m$. The steering wheel angle, yaw angle velocity and lateral acceleration of the cargo mass centre were simulated and tested. The simulation and test comparing results were shown from Figure **11** to Figure **13**.

The steering wheel angle input in lemniscate simulation was shown in Figure **11**, from which the maximum steering wheel angle in simulation was 685°, the average maximum angle in test was 658°, and the error was 3.9%. The yaw angle velocity responses in lemniscate simulation were shown in Figure **12**, from which the yaw angle velocity amplitude in simulation was 13.1°/s, the average amplitude in test was 12.6°/s,

and the error was 4%. The lateral acceleration responses in lemniscate simulation were shown in Figure **13**, from which the lateral acceleration amplitude in simulation was 632.7mm/s^2, the average amplitude in test was 606.3mm/s^2, and the error was 4.4%.

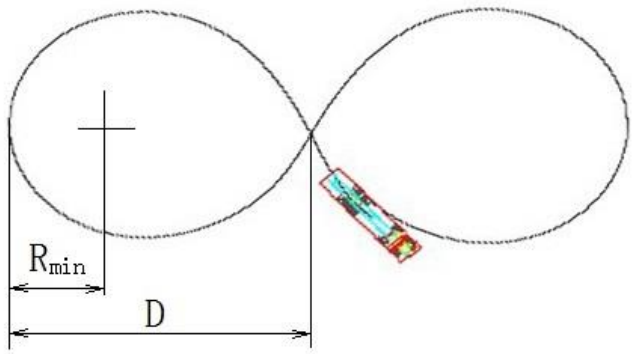

Figure 10. Lemniscate Simulation Trajectory

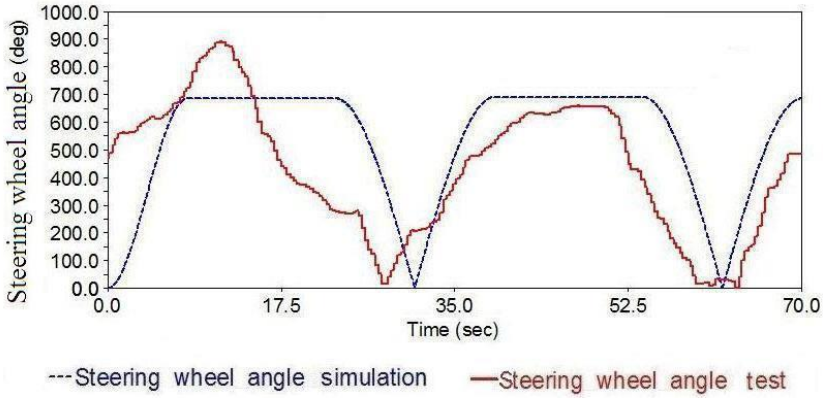

Figure 11. Steering Wheel Angle Input in Lemniscate Simulation

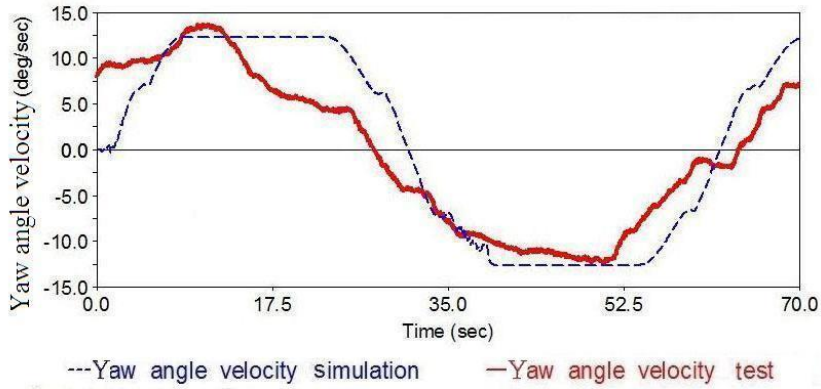

Figure 12. Yaw Angle Velocity Responses in Lemniscate Simulation

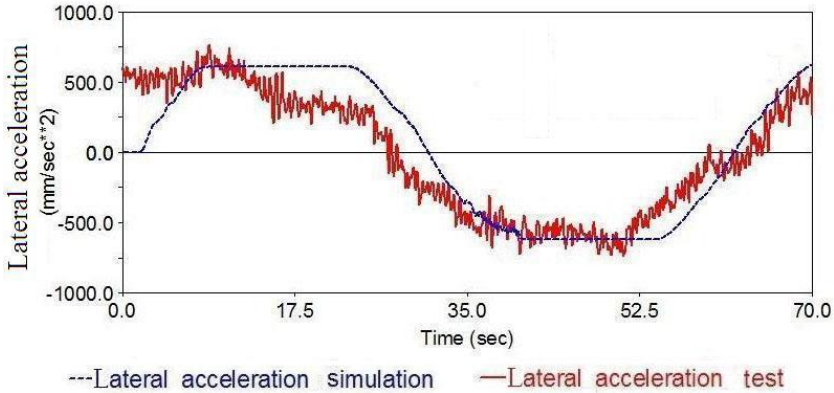

Figure 13. Lateral Acceleration Responses in Lemniscate Simulation

4.2. Returnability Simulation and Test

The returnability simulation and test were carried out with the same conditions in lemniscate simulation and test, except the vehicle velocity of 40Km/h. The simulation trajectory was shown in Figure **14**, where $R_r = 16m$. The steering wheel angle, yaw angle velocity and lateral acceleration of the cargo mass centre were also the considered parameters. The simulation and test comparing results were shown from Figure 15 to Figure 17.

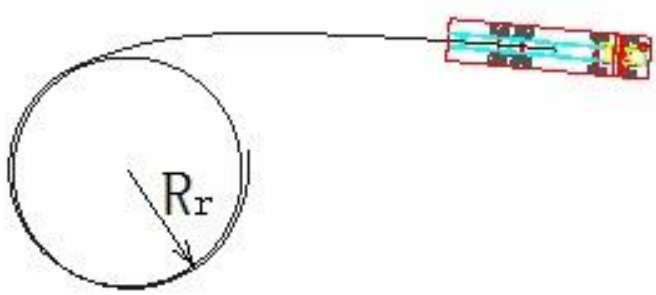

Figure 14. Returnability Simulation Trajectory

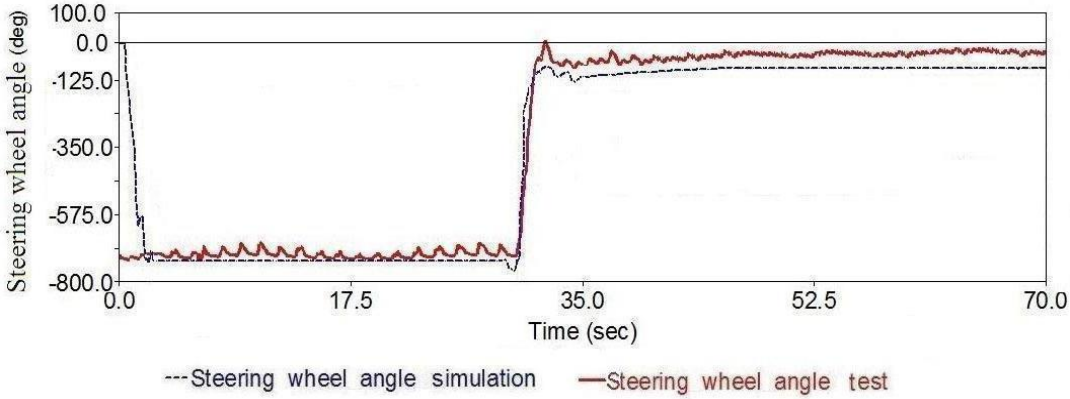

Figure 15. Steering Wheel Angle Input in Returnability Simulation

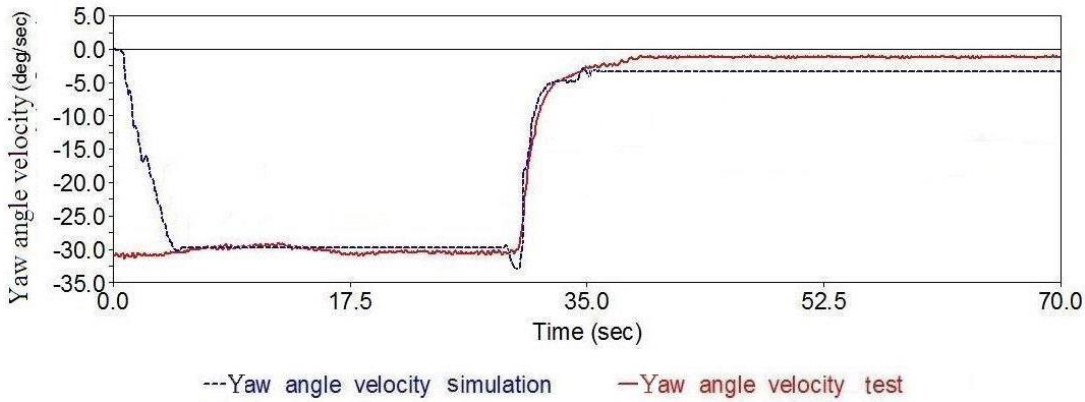

Figure 16. Yaw Angle Velocity Responses in Returnability Simulation

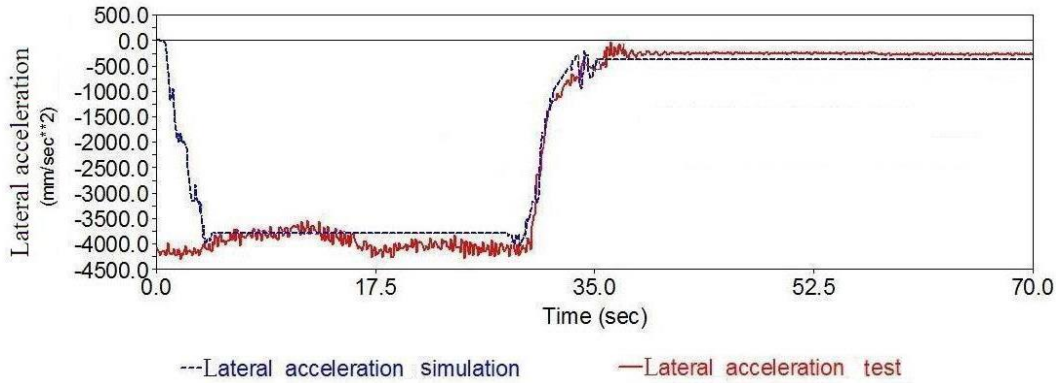

Figure 17. Lateral Acceleration Responses in Returnability Simulation

The steering wheel angle input in returnability simulation was shown in Figure 15, from which the residual steering wheel angle in simulation was -78°, the residual steering wheel angle in test was -40°, and the error was 95%. The yaw angle velocity responses in returnability simulation were shown in Figure 16, from which the residual yaw angle velocity in simulation was -3.2°/s, the residual yaw angle velocity in test was -1.8°/s, and the error was 77.8%. The lateral acceleration responses in returnability simulation were shown in Figure 17, from which the residual lateral acceleration in simulation was -225mm/s2, the residual lateral acceleration in test was -125mm/s2, and the error was 80%.

Through comparing the lemniscate simulation with the returnability simulation, the following conclusions could be acquired that the considerate parameters in simulations and tests had the same variation trends, and the returnability simulation results had bigger errors than the results in lemniscate simulation. One reason was that the values of the bushing joint parameters in the vehicle model had errors. These bushing joint included the bushings in steering drive mechanism, equalizing suspension guide mechanism, leaf spring eyes and *etc*. The other reason was that the calculated stiffness in x and y directions in SAE 3-Link leaf spring models also had errors. The vehicle model input in lemniscate simulation was the steering wheel angular displacement volume, and the vehicle structure could guarantee the output responses good precision. In contrast, the vehicle model input in returnability simulation was the equivalent

driving forces derived from the road and vehicle movement, and the simulation results had bigger errors because of some uncertain mechanics parameter values like bushing parameters.

5. Conclusions

According to the vehicle structure, the ADAMS models of the steering system, suspension system, anti-roll stabilizer and the full-vehicle model of the 8×4 type construction truck were developed based on the topology maps of each part through simplifying the vehicle structure.

The vehicle model suitability in different handling stability simulations was researched. The comparison results between the lemniscate simulation and test showed that the steering wheel angle error was 3.9%, for the cargo mass centre, the yaw angle velocity amplitude error was 4% and the lateral acceleration amplitude error was 4.4%. The comparison results between the returnability simulation and test showed that the residual steering wheel angle error was 95%, for the cargo mass centre, the residual yaw angle velocity error was 77.8% and the residual lateral acceleration error was 80%. On the whole, the results of the multi-body dynamic model simulations and vehicle tests had the same variation trends, and the errors between them were small. The errors of returnability simulation results were relatively larger than the errors of lemniscate results, because the values of some mechanics parameters in the full-vehicle model were not accurate. The simulation results of the full-vehicle model drive by displacement input had high precision, while the simulation results in the same conditions drive by force input had bigger errors.

Limited by the length of thesis, only part of the vehicle simulations and tests were carried out, and these were enough to support the conclusions in the thesis. Mechanics parameters in the full-vehicle model could be ascertained exactly using parameter identification and testing methods in future works and the full-vehicle model precision could be increased.

6. Appendices

Main structural parameters of the 8×4 type construction truck were shown in Table **1**.

Table 1. Main Structural Parameters of the 8×4 Type Construction Truck

Parameters	*Values*
Curb mass	30.5 tons
Vehicle size	11990 mm , 2470 mm , 3970 mm
Axle distance	1810mm+4500 mm +1365 mm
Leaf spring stiffness of steering axle suspension	413 N/mm
Leaf spring stiffness of equalizing suspension	3235N/mm
Wheel mass	109Kg×12
Frame mass	1530Kg
Coordinate of Frame centre of mass	(5010,0,-165)
Engine mass	880 Kg

Coordinate of Engine centre of mass	(-195,-9,-65)
Cargo mass	17550Kg
Coordinate of cargo centre of mass	(5698,0,245)
First axle mass	905 Kg
Coordinate of first axle centre of mass	(1350,-569)
Second axle mass	805 Kg
Coordinate of second axle centre of mass	(3175,0,-569)
Third axle mass	905 Kg
Coordinate of third axle centre of mass	(6310,0,-569)
Fourth axle mass	805 Kg
Coordinate of fourth axle centre of mass	(7660,0,-569)
Cab mass	730 Kg
Coordinate of cab centre of mass	(-230, 33.5,1445)
Gear box mass	260Kg
Coordinate of gear box centre of mass	(821,-31,-269)

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Authors

Gu Yufeng, he received his M.Sc. in Mechatronic Engineering (2006) and PhD in Vehicle Engineeing (2010) from Northwestern Polytechnical University. Now he is a lecturer at Mechanical Design and Manufacturing Department, Chang'an University. His current research interests include Vehicle chassis system designing and dynamic modeling.

Dong Fulong, he received his M.Sc. in Mechanical Design and Theory (2008) from Guizhou University. Now he is a lecturer at School of Mechanical Engineering, Anhui Science and Technology University University. His current research interests include Vehicle chassis system and lamp designing.

