Simulation Research on the Driving Stability of Articulated Vehicle Based on ADAMS

Wei-Ping Fang\textsuperscript{1,a}, Xiao-Jun Yang\textsuperscript{2,b}, Yong Li\textsuperscript{3,c}

\textsuperscript{1}Department of Vehicle Engineering Shandong Transport Vocational College Weifang 261206, China
\textsuperscript{2}Department of Vehicle Engineering Shandong Transport Vocational College Weifang 261206, China
\textsuperscript{3}Department of Vehicle Engineering Shandong Transport Vocational College Weifang 261206, China
\textsuperscript{a}fang0537@163.com, \textsuperscript{b}hillbreeze@163.com, \textsuperscript{c}117830859@qq.com

Abstract—The vehicle dynamical model was established by taking the articulated vehicle driven electrically as the research object, and the vehicle dynamical simulation was carried out by using ADAMS. The effects of different center of mass position, steering parameters, mass, lateral stiffness of tire, suspension stiffness and damping on the driving stability of articulated vehicle were analyzed. The simulation results showed that the driving stability of the articulated vehicle was improved by factors such as the position of the center of mass closing to the hinged point, increasing the steering stiffness and damping, reducing the lateral stiffness of the tire, increasing the suspension stiffness and damping, appropriately increasing the rear body mass, etc. This study provided a reference for the design and improvement of the vehicle.

1. INTRODUCTION
Compared with rigid vehicles, articulated vehicles have good mobility and wide adaptive capacity [1-3]. The articulated body structure makes the articulated vehicle have good pass ability and small turning radius, but the driving stability is poor [4-8]. This research took the 60T electrically driven articulated vehicle as the research object. The 3D dynamical simulation model of the vehicle was built based on ADAMS software. And the influence of different center of mass, steering parameters, mass, tire lateral stiffness, suspension stiffness and damping on the driving stability of articulated vehicle were analyzed.

2. MULTI-BODY DYNAMICS MODEL OF THE VEHICLE
The 3D software SolidWorks was used to build the 3D model of the articulated vehicle, which was imported into ADAMS/Views for simulation analysis. The articulated vehicle model was referred with Figure 1.
A 12-DOF six-wheel articulated model driven electrically was established in ADAMS. The front and rear body were connected by a hinged point. The front and rear frame were connected by a rotating pair. The front frame pivoted on the hinged point for integral steering. The rear axle adopted rigid connection. The articulated vehicle used springs and dampers instead of cylinders at the hinged point.

2.1. Suspension model

An oil-gas suspension was used for cushioning and shock absorption in the articulated vehicle. Then the dynamic stiffness of the oil-gas suspension was as follows:

\[ K(x) = \frac{r p_0 V_0 (A_1 - A_2)^2}{[V_0 - (A_1 - A_2)x]^3} \]  

(1)

Where,  
- \( r \) - gas polytropic index;  
- \( p_0 \) - air pressure;  
- \( V_0 \) - air volume;  
- \( A_1, A_2 \) - effective cross-sectional area of the first cavity and second cavity respectively;  
- \( x \) - displacement of the piston;  
- \( K(x) \) - the dynamic stiffness of the suspension.

2.2. Selection of the tire and establishment of the road spectrum

1) selection of the tire: In this research, UA tires were used to build the solid model of the articulated vehicle. The formulas of tire longitudinal force, lateral force and vertical force were as follows respectively:

\[ F_x = \text{sign}(s)[k_x S_x l_n^2 + \mu_x F_z (1 - 3 l_n^2 + 2 l_n^3)] \]  

(2)

\[ F_y = -\text{sign}(\alpha)[k_y S_y l_n^2 + \mu_y^{(m)} F_z (1 - 3 l_n^2 + 2 l_n^3) + k_y S_y] \]  

(3)

\[ F_z = k_z \delta + c_z \dot{\delta} \]  

(4)

Where,  
- \( s \) - the tire slip rate;  
- \( k_x \) - longitudinal stiffness of the tire;  
- \( S_x \) - longitudinal displacement of the tire;  
- \( l_n \) - ground mark length of the dimensionless tire;  
- \( \mu_x \) - the speed of the front body in the \( x \) direction;  
- \( \alpha \) - slip angle of the tire;  
- \( k_y \) - lateral stiffness of the tire;  
- \( S_y \) - lateral displacement of the tire;  
- \( \mu_y^{(m)} \) - modified friction coefficient;  
- \( k_y \) - outward rigidity of the tire;  
- \( S_y \) - vertical displacement of the tire;
$k_z$ - radial equivalent linear stiffness of the tire;
$\delta_z$ - radial deformation of the tire;
$c_z$ - damping coefficient of the tire;
$\dot{\delta}$ - radial deformation speed of the tire.

2) selection of the pavement spectrum: The model adopted a flat pavement spectrum in ADAMS software. ADAMS/Tire treated the road as a series of concrete triangular units and tires as a series of cylinders. This model could simulate the movement of vehicles on irregular roads [9-11].

3. SIMULATION MODEL OF THE ARTICULATED VEHICLE

The vehicle was set to run at a stable speed of 36km/h. When studying the stability of articulated vehicle running at high speed, the main consideration was whether the phenomenon of "snaking" instability would occur. The yaw velocities of the center of mass of the front body and rear body were analyzed in the model.

The vehicle accelerated in the first 5 seconds and then reached a constant speed. At this time, if an angular step was given to the articulated vehicle, the vehicle would appear obvious left and right lateral pendulum phenomenon, which was known as the "snake" instability. The yaw velocity of the front and rear vehicle centroid was referred with Figure 2. The solid curve was the yaw velocity of the front body, and the broken curve was the yaw velocity of the rear body. It could be seen from the Figure 2 that the rear body had a higher yaw velocity than the front body, and the swing of the rear body lagged behind the front body.

![Figure 2. Yaw velocity of the front body and the rear body](image)

3.1. Effects of the position of the center of mass

When the distance between the center of mass of the front body and the hinged point was 2m, 1.6m and 1.4m respectively, the yaw velocity curves of the rear body were referred with Figure 3. When the center of mass of the front body was close to the hinged point, the yaw velocity of the rear body lagged behind and the amplitude of the yaw velocity decreased.

![Figure 3. Yaw velocity of the rear body (change in the center of mass of the front body)](image)
When the distance between the center of mass of the rear body and the hinge point was 4m, 3.6m and 4.5m respectively, the yaw velocity curves of the rear body were referred with Figure 4. When the center of mass of the rear body was close to the hinge point, the yaw velocity response time shortened and amplitude of the rear body decreased. When the center of mass of the rear body was far from the hinge point, the yaw velocity increased.

![Figure 4](image1.png)

**Figure 4.** Yaw velocity of the rear body (change in the center of mass of the rear body)

### 3.2. Effects of the steering parameters

When the steering stiffness changed from 122803 N·m/rad to 200000N·m/rad and 100000 N·m/rad, the yaw velocity curves of the rear body were referred with Figure 5. The yaw velocity response time of the rear body shortened, the frequency increased and the maximum amplitude decreased with the increase of equivalent torsional stiffness. It was beneficial to the driving stability of the articulated vehicle.

![Figure 5](image2.png)

**Figure 5.** Yaw velocity of the rear body (change of steering stiffness)

When the stiffness was unchanged but the steering damping changed to 1000N. M /rad and 2000N. M /rad, the yaw velocity curves of the rear body were referred with Figure 6. The amplitude of the yaw velocity decayed with the extension of time. When the steering damping increased, the frequency of yaw velocity and lateral acceleration didn't change almost, but the amplitude obviously reduced. Therefore, increased steering damping was also beneficial to the driving stability of the articulated vehicle.

![Figure 6](image3.png)

**Figure 6.** Yaw velocity of the rear body (change of steering damping)
3.3. Effects of the mass of the rear body
When the mass of the rear body changed from 24010kg to 35000kg and 40000kg, the yaw velocity curves of the rear body were referred with Figure 7. When the rear body mass increased, the response time of the rear body yaw velocity extended, the frequency dropped and the yaw velocity decreased.

![Figure 7. Yaw velocity of the rear body (change of mass of the rear body)](image)

3.4. Effects of the lateral stiffness of the tire
Lateral stiffness was an important characteristic of the tire. When the lateral stiffness of the tire was set as 40,000 N/rad, 60,000 N/rad and 80,000 N/rad respectively, the yaw velocity curves of the rear body were referred with Figure 8. When the lateral stiffness increased, the yaw velocity of the rear body increased. Therefore, appropriately reducing the lateral stiffness of the tire was conducive to improving the driving stability of the articulated vehicle.

![Figure 8. Yaw velocity of the rear body (change of the lateral stiffness of the tire)](image)

3.5. Effects of the suspension stiffness and damping
When the suspension stiffness was changed, the yaw velocity curves of the front body were referred with Figure 9. When the suspension damping was changed to 10 times and 0 times of the original respectively, the yaw velocity curves of the front body were referred with Figure 10. As shown in the Figure 9, the greater the suspension stiffness, the greater the yaw velocity of the front body. As shown in the Figure 10, when the damping of the suspension increased, the yaw velocity of the front body became smaller and the response time increased. Therefore, the increase of suspension stiffness and damping was beneficial to the driving stability of articulated vehicle.
Figure 9. Yaw velocity of the front body (change of the suspension stiffness)

Figure 10. Yaw velocity of the front body (change of the suspension damping)

4. CONCLUSIONS
In this research, the vehicle dynamics model of 60T electric articulated vehicle was established by using ADAMS software. The effects of nonlinear parameters were fully considered in the modeling establishment. The model of the articulated vehicle was simulated and verified in ADAMS. The curves of the yaw velocity of the front and rear frame with time were simulated with a flat pavement spectrum. The effects of different center of mass, steering parameters, mass, lateral stiffness of tire and suspension parameters on the stability of articulated vehicle were analyzed. The results showed that the driving stability of the articulated vehicle was improved by factors such as the position of the center of mass closing to the hinged point, increasing the steering stiffness and damping, reducing the lateral stiffness of the tire, increasing the suspension stiffness and damping, appropriately increasing the rear body mass, etc.

ACKNOWLEDGMENT
The research was financially supported by the Applied Basic Research Program of MOT (No.2013319284130).

REFERENCES
[1] R. Adam, D. Lars, and S. T. Annika. Snaking stability of articulated frame steer vehicles with axle suspension. Int. J. Heavy Vehicle Systems, Vol.17, 2001, pp35-42.
[2] X. F. Wang, and X. H. Ren. Discussion on influencing factors of vehicle handling stability based on ADAMS/Car. Light Vehicle Technology, Vol.5, 2008, pp225-226.
[3] Y. He, A. Khajepour, and J. McPhee, etc. Dynamic modeling and stability analysis of articulated frame steer vehicles. International Journal of Heavy Vehicle Systems, Vol.12, 2005, pp 28—59.
[4] Z. J. Yang, and Q. H. He. Simulation of dynamic characteristics of hydraulic power steering system of articulated vehicles. Journal of Central South University of technology: Natural Science, Vol.35, 2004, pp 80–85.

[5] Y. Zhang, and X. H. Liu. Yaw stability for steering motion of articulated vehicle[J]. Journal of Jilin University: Engineering and Technology Edition, Vol.42, 2012, pp 266-271.

[6] G. Liu, Z. D. Zhang, etc. Dynamic modeling of articulated truck stability. Engineering Machinery, Vol.8, 1996, pp 5-8

[7] I. Heikki. Study of Driveline functionality during off-road driving of an articulated hauler. 15th European ADAMS User’s Conference, Vol.11, 2000, pp231-246.

[8] J. C. Wang, and W. H. Yang. Analysis on the straight line running stability of articulated loader. Mining Machinery, Vol.2, 1996, pp 2-6.

[9] C. W. Zhang, B. Chen, and W. Guo. Research on loader multi-body Dynamics based on ADAMS. Road building machinery and Construction Mechanization, Vol.11, 2006, pp 47-49.

[10] Z. F. Li, and G. Q. Wu. Research on vehicle system dynamics based on virtual prototype. Shanghai Automobile. Vol.2, 2000, pp 8-10.

[11] L. Q. Jin, and Q. N. Wang. Simulation of dynamic control system of four-wheel independent drive electric vehicle. Journal of Jilin University, Vol.10, 2004, pp 547-553.