NUMERICAL STUDY OF DISTURBANCE FORCE EFFECT ON HANDING STABILITY FOR ELECTRIC-DRIVEN ARTICULATED TRUCK IN A STRAIGHT-LINE

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The articulated truck has two separate parts that are linked by a hinge. The hinge will decrease the lateral stability of the truck while it travels at high speeds. In the actual operation of the vehicle, there are many factors that will bring in additional disturbance affecting the handling and stability such as uneven road obstacles, different wheel driving forces. In response to this situation, a kinematics model taking roll degree of freedom into consideration was built in this paper. The theoretic relationships between the yaw rate, roll angle steady-state value and disturbance force were established. Then the influence of disturbance force on the handling stability was studied. The results show that it has different influences on the articulated vehicle handling stability when the same disturbance force respectively acts on the front and rear body. And the relationship between the yaw rate and roll angle of the articulated vehicle is parabolic when the disturbance force is applied. This study can provide a theoretical reference for the independent control of electric-driven articulated vehicle wheels.

Keywords: articulated truck, disturbance force, yaw rate, roll angle.

1. Introduction

With the large-scale open-pit mine continuing to expand the scale, electric-driven articulated dump truck in the open-pit mine plays a crucial role. The front and rear body of the electric-driven articulated dump truck is connected by a hinge, and the original mechanical transmission structure is replaced by the series hybrid power system [1-5]. Azad N L [6] and Ge Shengqiang [7] analyzed the influence of structural parameters on the stability when the vehicle was traveling in straight by simplifying the articulated vehicles dynamic model. Azad N L et al. [8] analyzed the problems about the dynamic stability of the articulated dump truck and the research status and they pointed that the electronic control system can be used to control the dynamic stability of the articulated vehicle. Zhang Yang [9] summarized the influence of the combination of different steering types on the steady - state steering characteristics of the articulated vehicle by

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establishing the transfer function of the steering angle input and the absolute yaw rate output of the vehicle body. Wang Jianchun [10-11] established a simplified three degrees of freedom dynamic mathematical model of the articulated vehicle which moves along a straight line and discussed the steady response performances of the articulated vehicle to the step input of disturbing moment.

The articulated dump truck which consists of the front body, the rear body and the hinge is different from the rigid-frame vehicle. Due to the special structure, the stability of the articulated vehicle is worse than the rigid-frame vehicle. Mountains of articles of the handing stability were written about the rigid-frame vehicle, mountains of dynamic mathematical models of the vehicles handing stability were established only in the plane, and few models took the roll movement into consideration and fewer referred to the influence of the disturbance force on the yaw rate and roll angle.

In this paper, a kinematic model of a 60t six wheels electric-driven articulated dump truck is established while the roll movement of the sprung mass was taken into account. The relationships between the yaw rate, roll angle and disturbance force are studied during steady straight travel. Then the influence on the yaw rate and roll angle is discussed as the disturbance force acts on the front and rear body respectively during the vehicle straight travel.

2. Mathematical model of articulated vehicle during straight travel

The object of study in this paper is the 60t electric-driven articulated dump truck. The single longitudinal arm type hydro-pneumatic independent suspension is employed in the front body of the truck, so the roll center O_1 of the front body is located on the intersection of the longitudinal symmetry plane of the front body and the ground plane. The roll movement of the rear body is not taken into consideration because there is no suspension employed [12-14]. The plane motion model of the articulated vehicle is shown in Fig.1.

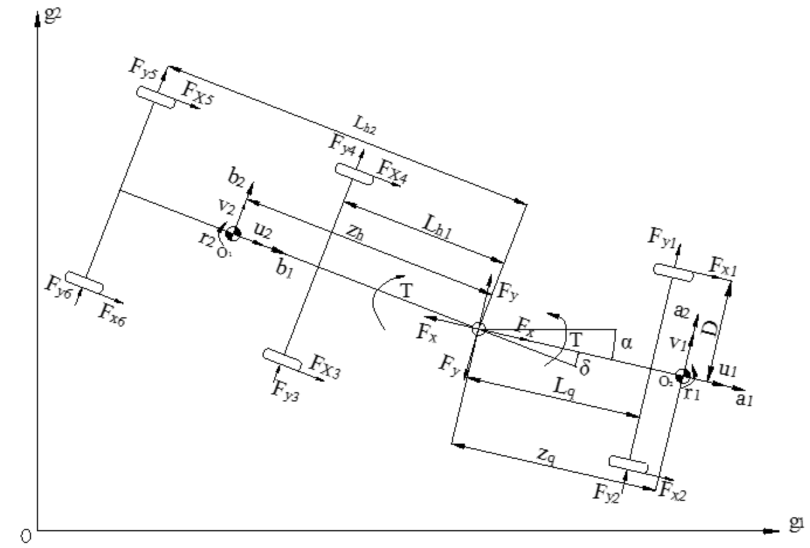

Fig.1.The plane motion model of the 60t articulated vehicle.

g_1Og_2— absolute coordinate system of vehicle movement; $a_1O_2a_2$— moving coordinate system on the centroid position of the front body; $b_1O_1b_2$— moving coordinate system on the centroid position of the rear body; u_1,v_1,r_1—longitudinal velocity, lateral velocity and yaw rate on the centroid position of the front body; u_2,v_2,r_2—longitudinal velocity, lateral velocity and yaw rate on the centroid position of the rear body; T— steering torque; F_x, F_y— the interaction force of the vehicle body at the hinged point; δ— turning angle between front body and rear body; F_{xi}, F_{yi}— the longitudinal force and cornering force of each wheel (i=1,2,······6); D—half of the wheel track ; Z_q— the distance between the front body centroid and the hinged point; Z_h— the distance between the rear body centroid and the hinged point; L_q— the distance between the front axle and the hinged point; L_{h1}—the distance between the middle axle and the hinged point; L_{h2}—the distance between the rear axle and the hinged point; α— heading angle of the articulated vehicle.

The roll model of the front body is shown in Fig.2.

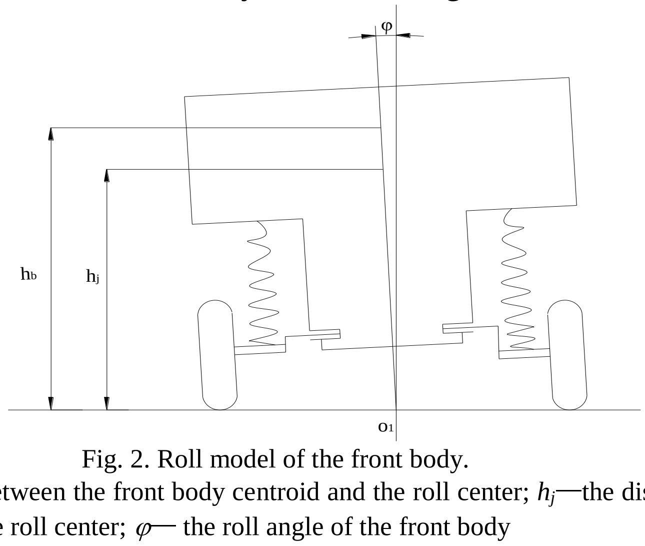

Fig. 2. Roll model of the front body.

h_b—the distance between the front body centroid and the roll center; h_j—the distance between the hinged point and the roll center; φ— the roll angle of the front body

2.1 Tire cornering force and vehicle steering torque

As the turning angle between front body and rear body is small, so $\sin\delta=\delta$, $\cos\delta=1$. Side slip angles of the tires can be expressed as follows.

$$\beta_1 = \beta_2 = \frac{v_1 - r_1(Z_q - L_q)}{u_1} \tag{1}$$

$$\beta_3 = \beta_4 = \frac{v_1 + u_1\delta - Z_q r_1 + L_{h1} r_2}{u_1} \tag{2}$$

$$\beta_5 = \beta_6 = \frac{v_1 + u_1\delta - Z_q r_1 + L_{h2} r_2}{u_1} \tag{3}$$

While the cornering forces of the tires can be expressed as follows.

$$F_{yi} = k_i \beta_i, (i = 1\sim6) \tag{4}$$

where

β_i - side slip angle of the tire;

k_i - cornering stiffness of the tire;

F_{yi} - cornering force of the tire;

i - tire number.

When the articulated vehicle travels straight or with a fixed radius, the steering valve closes, and the hydraulic steering system corresponds to a torsion spring acting on the hinge and connecting the front and rear body [15-16]. The hydraulic steering system that turns the front and rear bodies can be equivalent to a torsion spring acting on the hinge points and connecting the front and rear bodies. The equivalent torsion spring stiffness is k_j. According to the compressible fluid flow continuity equation, the steering moment of steady state steering is expressed as follows.

$$T = -k_j \delta \tag{5}$$

2.2 Motion equation of the vehicle

In order to simplify the analysis, the following assumptions are made for the articulated dump truck modeling: (1) Vehicle moves in the horizontal plane, regardless of the impact of uneven ground; (2) Ignore air resistance; (3) Ignore the friction that has little impact on the movement, such as the friction torque at the hinge point between the front and rear body; (4) When the body is rolling, the suspensions still have the linear characteristic. The roll angle stiffness K_φ and roll angle damping coefficient C_φ are constant.

The total kinetic energy and total potential energy of the system are expressed in the form of system variables, and they are substituted into the Lagrange equation. Then the motion equations can be obtained. Substitute equation (1)~(5) into the motion equations, seven differential equations about longitudinal velocity u, lateral velocity v, yaw rate r and roll angle φ are obtained as follows.

The front body:

$$m_1(\dot{u}_1 - v_1 r_1) + m_{1s} h_b (\dot{r}_1 \varphi + 2 r_1 \dot{\varphi}) = F_{x1} + F_{x2} - Fx \tag{6}$$

$$m_1(\dot{v}_1 + u_1 r_1) - m_{1s} h_b \ddot{\varphi} = k_q(\beta_1 + \beta_2) + Fy \tag{7}$$

$$I_{zzf}\dot{r}_1 - I_{xz}\ddot{\varphi} = (F_{x2} - F_{x1})D + T - k_q(\beta_1 + \beta_2)(Z_q - L_q) - Fy\, Z_q \tag{8}$$

$$I_{xx}\ddot{\varphi} + C_\phi \dot{\varphi} + (K_\phi - m_{1s} g h_b)\varphi - m_{1s} h_b (\dot{v}_1 + u_1 r_1) - I_{xz}\dot{r}_1 = -Fy h_j \tag{9}$$

The rear body:

$$m_2(\dot{u}_2 - v_2 r_2) = Fx\cos\delta + Fy\sin\delta + F_{x3} + F_{x4} + F_{x5} + F_{x6} \tag{10}$$

$$m_2(\dot{v}_2 + u_2 r_2) = k_h(\beta_4 + \beta_5 + \beta_3 + \beta_6) + Fx\sin\delta - Fy\cos\delta \tag{11}$$

$$\begin{aligned} I_{zzr}\dot{r}_2 &= k_h(\beta_5 + \beta_6)(L_{h2} - Z_h) - k_h(\beta_3 + \beta_4)(Z_h - L_{h1}) + \cdots \\ &\cdots + (Fy\cos\delta - Fx\sin\delta)Z_h + T + (F_{x4} + F_{x5} - F_{x3} - F_{x6})D \end{aligned} \tag{12}$$

where

m_1 - front body mass; m_{1s} - sprung mass of the front body;

m_2 - rear body mass; I_{xx} - roll inertia moment of the front body;

I_{xz} - inertial product of the front body roll and the yaw movement;

I_{zzf} - yaw inertia moment of front body;

I_{zzr} - yaw inertia moment of rear body.

When the truck runs at constant speed, the sum of the longitudinal forces is about zero. Ignoring the air resistance, the small product terms, and then the motion equations can be written as:

$$\boldsymbol{M\dot{X}} + \boldsymbol{JX} = \boldsymbol{H} \tag{13}$$

where

$\boldsymbol{H} = [D(F_{x2} - F_{x1}), D(F_{x4} + F_{x5} - F_{x3} - F_{x6}), 0, 0, 0, 0]^{\mathrm{T}}$

$\dot{\boldsymbol{X}} = [\dot{v}_1, \dot{r}_1, \dot{r}_2, \dot{\delta}, \dot{\varphi}, \ddot{\varphi}]^{\mathrm{T}}$

$\boldsymbol{X} = [v_1, r_1, r_2, \delta, \varphi, \dot{\varphi}]^{\mathrm{T}}$

When the articulated vehicle is in steady state, the forward speed and steering angle remain the same, so $\dot{X}$ =0. Then the equation (13) can be rewritten as:

$$\boldsymbol{JX} = \boldsymbol{H} \tag{14}$$

3 The variation of steady state yaw rate of electric-driven articulated vehicle

The drive control mode of the electric-driven articulated dump truck was equal torque. Ignoring the centroid bias and the difference between the front and rear tires, the driving forces of the left and right wheels in the horizontal direction are the same when the vehicle is running on a flat road. However, the wheels may contact with uneven road and there may be some other reasons that make the driving forces in the horizontal direction to be different. For example, when the wheel hits the road surface obstacle, the change of the driving force of the wheel in the horizontal direction is shown as in Fig. 3.

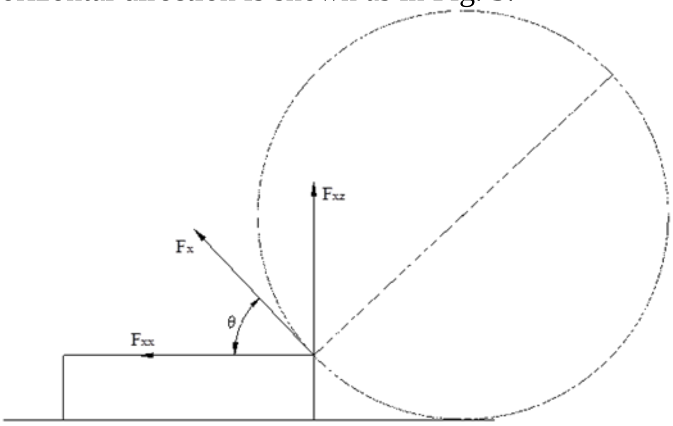

Fig.3. The driving force running on uneven road.

The driving force in the horizontal direction becomes F_{xx}=F_x $cos\theta$, so the driving force actually used to drive the wheel is reduced. This may result in unequal driving forces on the left and right wheels of the truck. When the effective driving forces of the left and right wheels are merged at the longitudinal axis of the vehicle body, there will be a resultant force and a torque [10-11]. In this paper, the additional torque is presented as a disturbance torque for the vehicle, and it will affect the yaw response characteristics of the vehicle. In the actual driving process of the vehicle, there are many factors which will draw up a similar effect, such as differences between the driving tire characteristics, different inflation pressure and unequal braking forces of the wheels.

3.1 The variation of yaw rate when the disturbance force is applied to the front body

When the electric-driven articulated truck travels straight in steady state, if one side wheel of the front body hits an obstacle of the road surface, then this will produce an additional disturbance torque in the front body. On this occasion, assume that there is no difference of the driving forces between the left and right rear wheels, then the amount of the driving force change is ΔF_q. Then:

$$\boldsymbol{H} = [D(F_{x2} - F_{x1}), 0, 0, 0, 0, 0]^{\mathrm{T}} \tag{15}$$

$$\Delta F_q = F_{x2} - F_{x1} = F_x(1 - \cos\theta) \tag{16}$$

From the equation (14) - (16), the relationship between the yaw rate of the articulated vehicle and the disturbance force acting on the front vehicle body can be obtained as:

$$\frac{r_{1q}}{F_x(1-\cos\theta)} = \frac{K_2 u_1}{K_1 + K_3 u_1^2} \tag{17}$$

where

$$K_1 = 2k_h^2 L_{23}^2 (k_j + 2k_q L_q) + 2k_q k_h k_j (L_{12}^2 + L_{13}^2) \tag{18}$$

$$K_2 = 4k_q k_h L_h D - k_j (k_q + 2k_h) D \tag{19}$$

$$K_3 = k_q k_j (m_2 Z_h - m_1 Z_q) + 2k_h k_j (m_2 Z_h - m_1 Z_q) + \cdots$$
$$\cdots + mk_j (k_q L_q - 2k_h L_h) + 4k_q k_h (m_1 L_h d_q + m_2 L_q d_h) \tag{20}$$

Similarly, the relationship between the vehicle roll angle and the disturbance force acting on the front vehicle body can be solved as:

$$\frac{\varphi_q}{F_x(1-\cos\theta)} = \frac{K_2}{K_1 + K_3 u_1^2} \cdot \frac{K_5}{K_4} \tag{21}$$

where

$$K_4 = (K_\phi - m_{1s} g h_b)(4k_h k_q L_h - 2k_j k_h - k_j k_q) \tag{22}$$

$$K_5 = 4u_1^2 h_b k_h k_q m_{1s} L_h - 4u_1^2 h_j k_h k_q m_2 d_h + u_1^2 k_j (2m_1 k_h h_j - \cdots$$
$$\cdots - m_2 k_q h_j - 2m_{1s} h_b k_h - m_{1s} h_b k_q) - 4h_j k_j k_h k_q L - 4h_j k_h^2 k_q L_{23}^2 \tag{23}$$

From the equation (17) and (21), the theoretical relationship between the yaw rate and the roll angle when the disturbance force is applied to the front body during straight travel can be described as:

$$r_{1q} = \frac{K_4 u_1}{K_5} \varphi_q \tag{24}$$

3.2 The variation of yaw rate when the disturbance force is applied to the rear body

As the same with front body, when an additional disturbance torque acts on the rear body, it is assumed that there is no difference of the driving forces between the left and right front wheels, the amount of the driving force change is ΔF_h, then:

$$\boldsymbol{H} = [0, D(F_{x4} + F_{x5} - F_{x3} - F_{x6}), 0, 0, 0, 0]^{\mathrm{T}} \tag{25}$$

$$\Delta F_h = F_{x4} + F_{x5} - F_{x3} - F_{x6} = F_x(1-\cos\theta) \tag{26}$$

The relationship between the yaw rate of the articulated vehicle and the disturbance force acting on the rear vehicle body can be obtained from equations (14), (25), and (26):

$$\frac{r_{1h}}{F_x(1-\cos\theta)} = \frac{K_{22}u_1}{K_1 + K_3u_1^2} \tag{27}$$

where

$$K_{22} = 4k_qk_hL_qD + k_j(k_q + 2k_h)D \tag{28}$$

Comparing equation (17) and (27), the different influence on the yaw rate can be got when the same disturbance force acts on the front body and the rear body respectively during vehicle running straight. The ratio of the two yaw rates can be described as:

$$\frac{r_{1q}}{r_{1h}} = \frac{K_2}{K_{22}} = \frac{4k_qk_hL_h - k_j(k_q + 2k_h)}{4k_qk_hL_q + k_j(k_q + 2k_h)} \tag{29}$$

From equation (29), we can observe that the ratio is mainly related to the structural parameters of the vehicle and the stiffness of the tire, the steering system stiffness, and it is not related to the speed.

Similarly, the relationship between the roll angle in steady state travel and the disturbance force acting on the rear vehicle body can be obtained as:

$$\frac{\varphi_h}{F_x(1-\cos\theta)} = \frac{K_{22}}{K_1 + K_3u_1^2} \cdot \frac{K_{55}}{K_{44}} \tag{30}$$

where

$$K_{44} = (K_\phi - m_{1s}gh_b)(4k_hk_qL_q + 2k_jk_h + k_jk_q) \tag{31}$$

$$K_{55} = 4u_1^2h_bk_hk_qm_{1s}L_q + 4u_1^2h_jk_hk_qm_1d_q + u_1^2k_j(-2m_1k_hh_j + \cdots$$
$$\cdots + m_2k_qh_j + 2m_{1s}h_bk_h + m_{1s}h_bk_q) + 4h_jk_jk_hk_qL \tag{32}$$

Comparing equation (21) and (30), the different influence on the roll angle can be got when the same disturbance force acts on the front body and the rear body respectively during vehicle running straight. The ratio of the two roll angles can be described as:

$$\frac{\varphi_q}{\varphi_h} = \frac{K_2}{K_{22}} \cdot \frac{K_5}{K_{55}} \cdot \frac{K_4}{K_{44}} \tag{33}$$

In the equation (33), the item of K_5/K_{55} contains u_1 which cannot be eliminated.

From the equations (27) and (30), the theoretical relationship between the yaw rate and the roll angle when an additional disturbance torque acts on the rear body during straight travel can be described as:

$$r_{1h} = \frac{K_{44}u_1}{K_{55}}\varphi_h \tag{34}$$

4. Influence of disturbance force on yaw and roll response of the truck

In this paper, it is assumed that the left wheels hit the obstacle of the road surface.

When the disturbance force acts on the front or rear body respectively, the yaw rate and roll angle of the articulated vehicle are defined as:

$$r_1 = \eta \times F_x(1-\cos\theta) = \eta \cdot F_r \qquad (35)$$

$$\varphi = \varepsilon \times F_x(1-\cos\theta) = \varepsilon \cdot F_r \qquad 36)$$

where

η - disturbance coefficient of yaw rate;

ε - disturbance coefficient of roll angle;

F_r - generalized disturbance force.

The structure parameters of the articulated truck are present in Table 1 for calculating the disturbance coefficient of yaw rate and roll angle. In order to reduce the influencing factors, the study will only calculate the situation when the vehicle is empty.

Table 1

Structure parameters of articulated truck

Names	Units	Values
Mass/(*kg*)	m_1	18766
	m_2	26494
	m_{1s}	8967
Distance/(*m*)	Z_q	2.090
	L_q	1.600
	Z_h	3.070
	L_{h1}	2.500
	L_{h2}	4.500
	h_b	1.632
	h_j	1.186
	D	1.892
Rotary inertia of front body/($kg \cdot m^2$)	I_{xx}	10016

	I_{xz}	681
	I_{zzf}	35231
Rotary inertia of rear body /($kg\cdot m^2$)	I_{zzr}	303001
Stiffness of tires/($N\cdot m^{-1}$)	k_q	-3×10^6
	k_h	-6×10^6
Suspension /($N\cdot m^{-1}$) /($N\cdot s\cdot m^{-1}$)	K_φ	60×10^5
	C_φ	17×10^5
Stiffness of steering system/($N\cdot m^{-1}$)	K_j	1.1837×10^7

4.1 Influence of disturbance force on yaw rate response

From equation (17) and (27), the yaw rate can be derived when the electric-driven articulated truck travels straight in different speeds. In this paper, the disturbance coefficient of yaw rate defined above is used to analyze the influence on the yaw rate when the disturbance forces act on the front and rear body respectively, for the disturbance force isn't easy to obtain in different driving conditions.

The influence on the yaw rate is shown in Fig.4, where curve 1 represents the disturbance coefficient of yaw rate changes with the change of vehicle speed when the disturbance force acts on the front body. Curve 2 represents the disturbance coefficient of yaw rate changes with the change of vehicle speed when the disturbance force acts on the rear body. Comparing curves 1 and 2 in this figure, the additional yaw rate generated by the disturbance force increases with the increase of the vehicle speed. When the same disturbance force acts on the front and rear body respectively, the yaw rates produced by the disturbance force have the same direction.

The ratio of curve 1 to curve 2 is shown in Fig.5. The influence on the yaw rate when the disturbance force acts on the front body is greater than when the disturbance force acts on the rear body. The ratio of curve 1 and curve 2 is equal to 2.677, and it is not related to the vehicle speed.

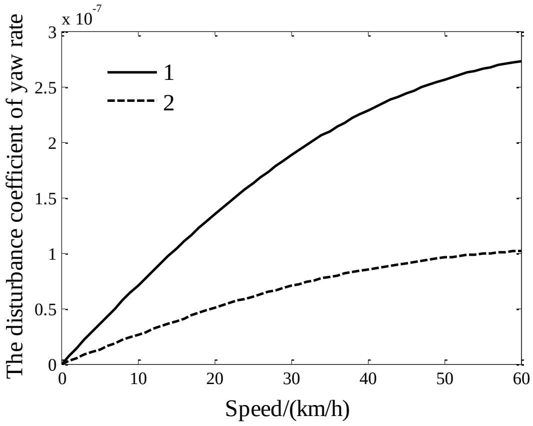

Fig.4. Influence of disturbance force on yaw rate response.

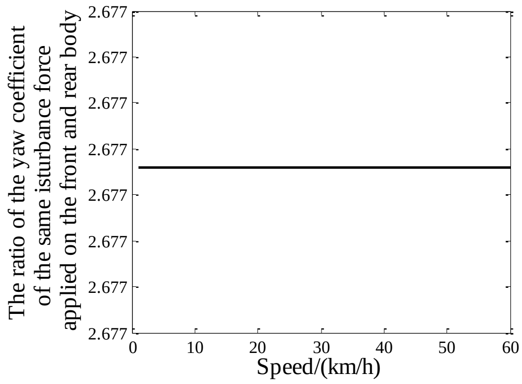

Fig. 5. The ratio of the yaw coefficients of the same disturbance force applied on the front and rear body

4.2 Influence of disturbance force on the roll angle response

The influence of speed on the roll angle is shown in Fig.6. Curve 1 in this figure represents the evolution of the disturbance coefficient of roll angle with the change of vehicle speed when the disturbance force acts on the front body. Curve 2 in the same figure represents the evolution of the disturbance coefficient of roll angle with the change of vehicle speed when the disturbance force acts on the rear body. Comparing curves 1 and 2 in Fig.6, the additional roll angle generated by the disturbance force decreases with the increase of the vehicle speed. When the disturbance force with the same value and the same direction acts on the front and rear body respectively, the direction of the roll angle is the same.

The ratio of curve 1 to curve 2 is shown in Fig.7. From Fig.7, the influence on the roll angle when the same disturbance force acts on the front body is less than when the disturbance force acts on the rear body, and with the increase of the vehicle speed, this trend is more obvious.

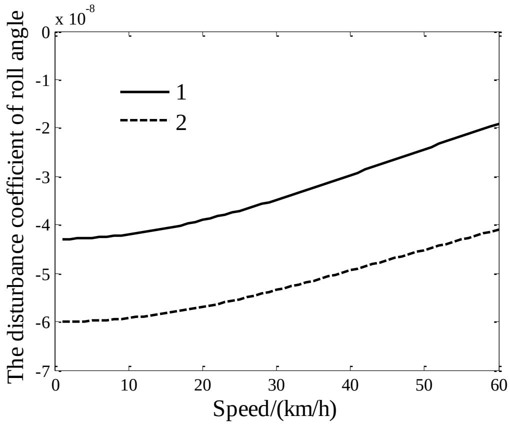

Fig. 6. The disturbance coefficient of roll angle.

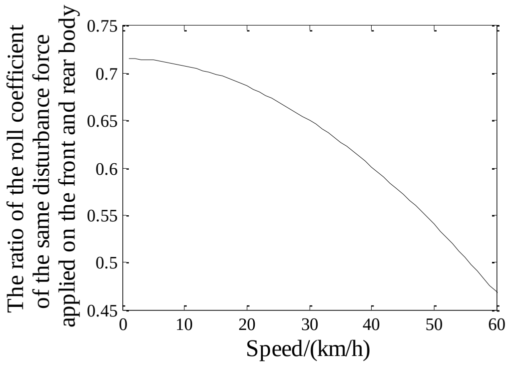

Fig. 7. The ratio of the roll coefficients of the same disturbance force applied on the front and rear body.

4.3 The relationship between the yaw rate and the roll angle of the articulated vehicle when the disturbance force is applied

According to equation (24) and (34), the relationship between the yaw rate and the roll angle when the articulated vehicle travels straight in steady state at different speeds can be obtained. The correlation between the yaw rate and the roll angle is shown in Fig.8. Curve 1 represents the variation of this parameter as the disturbance force acts on the front body, and curve 2 represents the variation of the same parameter as the disturbance force acts on the rear body.

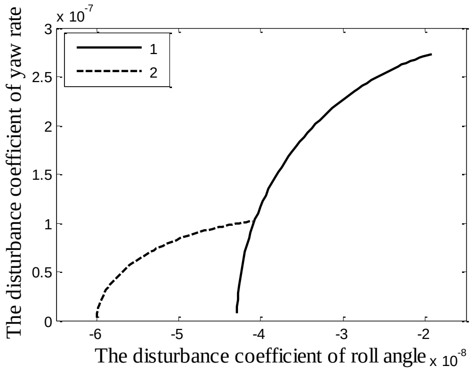

Fig. 8. The relationship of yaw rate and roll angle disturbance coefficient.

The results in Fig.8 show that the two curves are parabolic. The influence on the yaw rate when the same disturbance force acts on the front body is greater than when the disturbance force acts on the rear body, and the influence on the roll angle when the same disturbance force acts on the front body is less than when the disturbance force acts on the rear body. The results can provide a theoretical reference for the integrated control of steering stability and ride comfort when the articulated vehicle travels linearly.

5. Conclusions

(1) Considering the roll movement of the sprung mass, a kinematic model of a 60t six wheels electric-driven articulated dump truck is established. The relationship between the yaw rate, roll angle and a disturbance force is studied when the articulated vehicle travels straight in steady state.

(2) The influence of the disturbance force on the handling stability of the articulated vehicle is analyzed. The results show that when the disturbance forces with the same value and the same direction act on the front and rear body respectively, the yaw rates produced by the disturbance force have the same direction. And the influence on the yaw rate when the disturbance force acts on the front body is higher than when the disturbance force acts on the rear body. The ratio of two yaw rates is not related to the vehicle speed.

When the disturbance force with the same value and the same direction acts on the front and rear body respectively, the direction of the roll angle is the same. The influence on the roll angle when the same disturbance force acts on the front body is less than when the disturbance force acts on the rear body, and with the increase of the vehicle speed, this trend is more obvious. From the results above, it is possible to improve the handling stability of the articulated vehicle under different operating conditions by independently controlling the driving torque of the wheel.

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