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A PERFORMANCE STUDY ON STRUCTURAL PARAMETERS OF CENTRE-AXLE-TRAILER COMBINATIONS

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Abstract
Compared with rigid-trucks, centre-axle-trailer (CAT) combinations significantly improve fuel economy and reduce greenhouse-gas emissions. However, with respect to rigid-trucks, CAT combinations exhibit lower lateral stability at high speeds, and display poorer path-following off-tracking (*PFOT*) at low speeds. This study intends to address these problems. To this end, eigenvalue analysis and simulation were conducted to evaluate the directional performance of CAT combinations considering the variations of typical structure parameters. To coordinate the trace-off between the lateral stability in terms of rearward amplification (*RWA*) and *PFOT* of CAT combinations, a CAT design with a variable-length drawbar was proposed. The drawbar length may be altered under different operating conditions, e.g., low-speed curved-path negotiations and high-speed evasive manoeuvres. The proposed variable-length drawbar is feasible in design and cost-effective in implementation. The insightful results derived from this study provide useful guidelines for the design CAT combinations to improved directional performance.
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Key Words: Rearward Amplification (*RWA*), Dynamic Stability, Path-Following Off-Tracking (*PFOT*), Manoeuvrability, Centre Axle Trailer, Numerical Simulations

1. INTRODUCTION

Centre-axle-trailers are popular for freight transportation in Australia, Europe, and North America. Compared to rigid-track, CAT combinations can significantly improve fuel economy ad greatly reduce greenhouse-gas emissions [1, 2]. In reality, rigid-trucks may be used for goods delivery in urban areas and even travel on city streets; for highway freight transportation, a rigid-truck can be easily coupled with a CAT to form a vehicle combination to improve fuel economy and transportation efficiency.

In 2016, the national standard, GB 1589-2016 [3], was issued in China. The standard specifies the configuration of various CATs: a trailer with an axle or axles longitudinally positioned close to the vehicle unit's centre of gravity. With uniformly distributed load on CAT, the extra vertical load exerted to the towing vehicle is very low, and can be neglected. It is well known that for an articulated vehicle (e.g., a truck-CAT combination), its static (or divergent) instability was determined by the steady-state handling characteristic of the towing vehicle with the extra vertical load from the trailer [4]. For an articulated vehicle with steady-state handling characteristic of oversteer, the static instability referred to an aperiodic divergence of yaw motion at a forward speed above its static critical speed [5]. The unstable mode of jack-knifing might result from the aperiodic divergence of towing-vehicle-swing [4]. For a truck-CAT combination, the truck's steady-state handline characteristic is not affected by the CAT due to the neglectable extra vertical load from the trailer. For a rigid-truck with well-designed steady-state handling performance, the resulting truck-CAT combination could ensure its static stability under various operating conditions. The above steady-state handling performance is the theoretical basis for the popularity of truck-CAT combinations.

However, similar to typical articulated heavy vehicles, e.g., tractor/semitrailer, truck-CAT combinations exhibit worse high-speed dynamic stability and poorer low-speed manoeuvrability due to multiple-unit configuration and enlengthened dimension with respect to rigid-trucks [6]. Under low-speed curved path negotiations, a truck-CAT shows poor path-following off-tracking (*PFOT*) performance, and widen roads and larger spaces are required for safe operations. In high-speed evasive manoeuvres, the truck-CAT may experience an oscillatory trailer yaw response with increasing amplitude known as trailer sway, and this unstable motion mode frequently caused severe highway accidents. The past four decades have witnessed extensive studies to improve the dynamic stability and enhance the *PFOT* performance of articulated vehicle vehicles [7, 8]. To improve the dynamic stability of tractor/semitrailer combinations, trailer axles are generally positioned close to the vehicle unit's end. For enhancing the low-speed *PFOT* performance of tractor/semitrailer combinations, passive trailer steering systems, e.g., self-steering axles, had been developed [9]. It is commonly accepted that there is a trade-off between the high-speed stability and the low-speed manoeuvrability [7]. However, very few cost-effective design solutions have been found to address the trade-off relation for articulated heavy vehicles and, in particular, truck-CAT combinations.

To address the trade-off relation for truck-CAT combinations, this study proposed a variable-length drawbar for CATs. The proposed drawbar could be adjusted in length to accommodate different operating conditions. To assess the directional performance of the truck-CAT with the proposed variable-length drawbar, linear dynamic stability based on eigenvalue analysis and numerical simulation using TruckSim software were conducted.

The remainder of this paper is organized as follows. Section 2 briefly reviews the state-of-the-art studies on improving the directional performance of articulated heavy vehicles. Section 3 introduces a linear truck-CAT model with 3 degrees of freedom (DOF), which is used for linear stability analysis. The linear model is validated using the corresponding nonlinear vehicle model developed with TruckSim software and experimental data. Section 4 presents the simulation results in terms of the sensitivity analysis of truck-CAT directional performance considering the variations of relevant structural parameters. Finally, the conclusions are drawn in Section 5.

2. STATE-OF-THE-ART

The high-speed dynamic stability is an important performance indicator for articulated vehicles. The effects of parameters variations on dynamic stability were investigated. Yang et al. [10] conducted a performance optimization for dynamic stability of a truck-CAT combination, and Huang et al. [11] performed a multi-objective optimization for improving the handling stability of a CAT train. However, these studies only focused on experimental simulation, lacking theoretical analysis in root causes for unstable motions of truck-CAT combinations. Islam et al. [12] evaluated the lateral dynamics of long combination vehicles based on a single-track linear model, but they did not conduct adequate linear stability analysis. An articulated vehicle will experience the dynamic instability if the vehicle travels at a forward speed above the 'dynamic' critical speed, and the damping ratio of the least damped motion mode becomes negative [13]; generally, the damping ratio of the least damped motion mode drops as vehicle forward speed increases [14]; eigenvalue analysis is an effective tool for the linear stability analysis of articulated vehicles.

Various mathematical models, including nonlinear models [15-17], are used to simulate the lateral dynamics of articulated vehicles and to optimize the system parameters for improving the dynamic stability. Barbieri et al. [18] compared the dynamic behaviours of articulated heavy vehicles with different configurations under different operating conditions. Unfortunately, truck-CAT combinations were not considered. A number of structural parameters impose significant effects on the dynamic stability of articulated vehicles. The dominant parameters included longitudinal position of trailer centre of gravity, hitch point, and trailer axles [19].

On the other hand, these dominant structural parameters also impact the *PFOT* performance of articulated vehicles. However, most studies only focused on the specified aspect(s) of directional performance of these large vehicles. Zhang et al. [20] analysed the relationships among the motions of yaw and roll, as well as longitudinal and lateral motions to improve the dynamic stability of CAT train, but the low-speed *PFOT* was not considered. In contrast, Juraj Jagelčák et al. [21] examined the manoeuvrability of selected tractor/semitrailer combinations in low-speed curved path negotiations, whereas the dynamic stability was not investigated. Considering the trade-off between the high-speed dynamic stability and the low-speed *PFOT* performance of articulated vehicles, past studies explored some potential solutions to the trade-off problem. He and Islam [22] presented a design method for addressing the trade-off issue using the active trailer steering technique. However, to implement the active trailer steering technique, sophisticated control units should be designed [23], and expensive steering actuators and independent power packs need to be installed on the trailer(s). Hence, the cost-effectiveness of this technique is questionable.

To address the above issues identified from the literature review, this study proposed a variable-length drawbar for CATs, which may be a feasible and cost-effective solution to the trade-off problem of truck-CAT combinations.

3. METHODOLOGY

3.1 Vehicle modelling and validation

Vehicle models: To evaluate the linear stability of the truck-CAT combination using eigenvalue analysis, the linear yaw-plane model with 3 degrees of freedom (DOF) is shown in Fig. 1. The towing and trailing units are connected at the articulation joint.

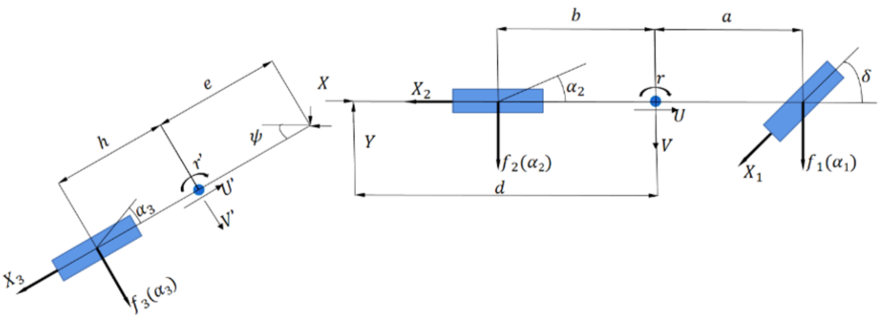

Figure 1: Schematic representation of the 3-DOF linear truck-CAT model.

The equations of motions of the towing vehicle (i.e., truck) can be expressed as:

$$m_1(\dot{U}-Vr)=-X_1\cos\delta-X_2+X \tag{1}$$

$$m_1(\dot{V}+Ur)=f_1(\alpha_1)+f_2(\alpha_2)+X_1\sin\delta-Y \tag{2}$$

$$I_1\dot{r}=af_1(\alpha_1)-bf_2(\alpha_2)+aX_1\sin\delta+dY \tag{3}$$

and the equations of motions of the CAT are written as:

$$m_2(\dot{U}'-V'r')=-X_3-Y\sin\psi-X\cos\psi \tag{4}$$

$$m_2(\dot{V}'+U'r')=f_3(\alpha_3)+Y\cos\psi-X\sin\psi \tag{5}$$

$$I_2\dot{r}'=hf_3(\alpha_3)-e(-Y\cos\psi+X\sin\psi) \tag{6}$$

where m_1 and m_2 are the total mass of truck and trailer, respectively, *U* and *U*' the longitudinal velocity of truck and trailer, accordingly, *V* and *V*' the respective lateral velocity of truck and trailer, *r* and *r*' the corresponding yaw rate of truck and trailer, *X* and *Y* the reaction forces at the hitch defined in the truck-fixed coordination system, X_1, X_2 and X_3 denote tire longitudinal force of each axle, f_1, f_2 and f_3 tire lateral force of each axle, I_1 and I_2 the yaw moment of inertia of truck and trailer, δ is the steering angle of truck front axle wheel, the articulation angle between truck and trailer, *a*, *b*, *d*, *e* and *h* represent the geometric parameters of the vehicle structure shown in Fig. 1.

The tire cornering force per axle is assumed to be proportional to the tire slip angles:

$$f_1 = -c_1\alpha_1 = -c_1\left(\frac{v_1 + ar_1}{u_1} - \delta\right) \tag{7}$$

$$f_2 = -c_2\alpha_2 = -c_2\left(\frac{v_1 - br_1}{u_1}\right) \tag{8}$$

$$f_3 = -c_3\alpha_3 = -c_3\left(\frac{v_2 - hr_2}{u_2}\right) = -c_3\left[\frac{u_1\sin\psi + (v_1 - dr_1)\cos\psi - (h+e)r_2}{u_1\cos\psi - (v_1 - dr_1)\sin\psi}\right] \tag{9}$$

At the articulation point, the velocities and accelerations expressed in the coordinate systems fixed with the truck and trailer must be equal, respectively. This allows the trailer equations to be described in the truck fixed coordination system. Based on the following assumptions, the equations are linearized: 1) the forward speed *U* is constant, and Eqs. (1) and (4), governing the forward motions of truck and trailer are ignored; 2) small angle approximations are assumed: cos $\Psi = 1$, sin $\Psi = \Psi$; 3) all products of variables are neglected; and 4) linear tire model is used.

For zero initial conditions:

$$\dot{\psi} = r - r' \tag{10}$$

The linearized equations of motion can be written in the state-space form as:

$$M\{\dot{x}\} + D\{x\} + F\delta = 0 \tag{11}$$

where the state variable is defined as:

$$\{x\} = \{V, r, \dot{\psi}, \psi\} \tag{12}$$

The matrices *M*, *D*, and *F*, are shown below:

$$M = \begin{bmatrix} m_1 + m_2 & -m_2 d & -m_2 e & 0 \\ -m_2 d & I_1 + m_2 d^2 & m_2 ed & 0 \\ -m_2 e & m_2 ed & I_2 + m_2 e^2 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix},\ F = \begin{bmatrix} c_1 \\ c_1 a \\ 0 \\ 0 \end{bmatrix} \tag{13}$$

$$D = \frac{1}{u} \times \begin{bmatrix} -c_1 - c_2 - c_3 & -c_1 a + c_2 b + c_3 d + (m_1 + m_2)u^2 & c_3(h+e) & -c_3 u \\ -c_1 a + c_2 b + c_3 d & -c_1 a^2 - c_2 b^2 - c_3 d^2 - m_2 du^2 & -c_3 d(h+e) & c_3 du \\ c_3(h+e) & -c_3 d(h+e) - m_2 eu^2 & -c_3(h+e)^2 & c_3(h+e)u \\ 0 & -u & u & 0 \end{bmatrix} \tag{14}$$

The 3-DOF linear truck-CAT model was implemented using Matlab/Simulink software. To evaluate the performance measures of the dynamic stability and *PFOT*, a 3 dimensional (3-D) nonlinear truck-CAT model was generated using TruckSim software. Fig. 2 a shows the 3-D nonlinear TruckSim model.

Model verification: To validate the 3-DOF linear model and the 3-D nonlinear TruckSim model, the simulation results based on the two models were compared with the experimental data achieved under a high-speed single lane-change testing manoeuvre. The testing truck-CAT combination was manufactured by an Automobile Co., China, as shown in Fig. 2 b.

a) b)

Figure 2: a) 3-D TruckSim model, b) testing truck-CAT combination.

The system parameters of the truck-CAT combination [11] are listed in Table I.

Table I: The baseline vehicle system parameters.

Description	**Notation**	**Value**
Truck total mass (kg)	m_1	17000
Truck yaw moment of inertia (kg·m^2)	I_1	50960
Truck dimension (m)	a	2
Truck dimension (m)	b	3.6
Tractor dimension (m)	d	5.25
Trailer total mass (kg)	m_2	18000
Trailer yaw moment of inertia (kg·m^2)	I_2	29767.9
Trailer dimension (m)	e	6.11
Trailer dimension (m)	h	0
Front tire cornering stiffness (N/rad)	c_1	-125400
Rear tire cornering stiffness (N/rad)	c_2	-235290
Trailer tire cornering stiffness (N/rad)	c_3	-237110

In the proving ground test of the truck-CAT combination, experimental devices were used to collect the running data of the vehicle. A VBOX data acquisition system and two RT-3002 gyroscopes were utilized to measure vehicle dynamic responses, e.g., yaw rate and lateral acceleration of the truck and trailer, etc. The test instruments also included a steering force gauge, a meteorological meter, a tire air gauge, a portable weighing meter, etc. The proving ground test and main devices are shown in Fig. 3.

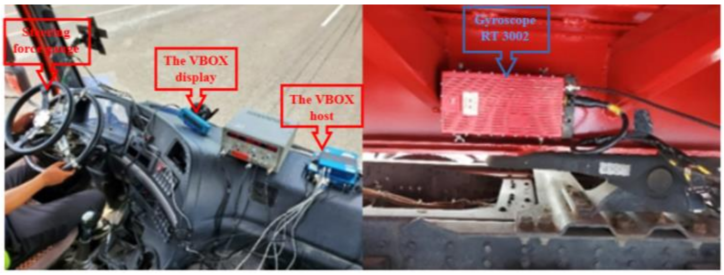

Figure 3: Proving ground test and main devices.

To validate the linear and nonlinear truck-CAT model, a single lane-change test recommended by ISO-14791-2000 [24] was conducted on the proving ground shown in Fig. 3. Over the testing manoeuvre, the towing unit tracked the predefined testing course at the constant speed of 80 km/h. Fig. 4 compares the testing data and the corresponding simulation results of the two vehicle models in terms of the time-histories of yaw rate and lateral acceleration of the truck and trailer, as well as the time-history of the articulated angle between the two vehicle units. It is observed that the simulation result of the 3-D TruckSim model matches the testing data better compared against that of the 3-DOF linear vehicle model (implemented using

Simulink). Overall, the simulation results of the two vehicle models are in good agreement with the experimental data.

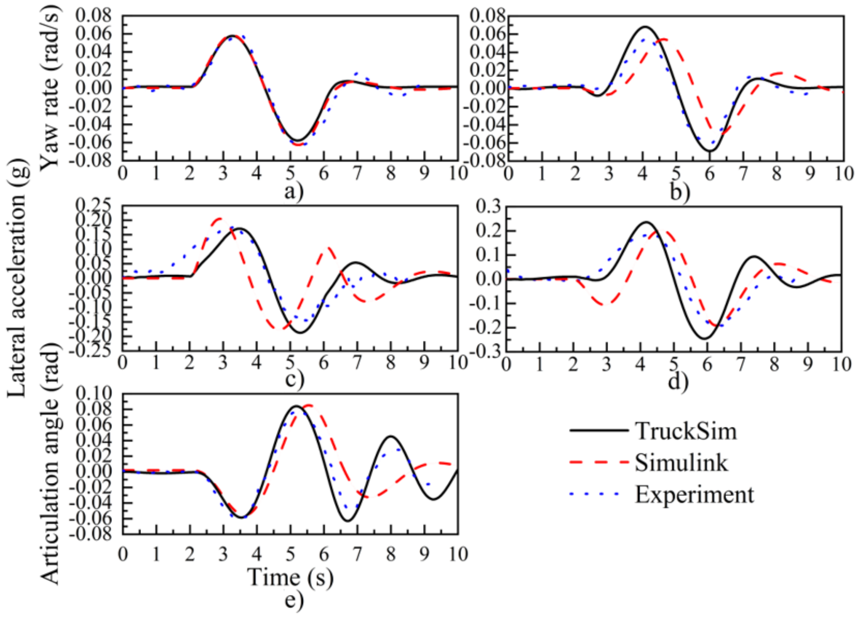

Figure 4: Dynamic response comparison: a) yaw rate of tractor; b) yaw rate of trailer; c) lateral acceleration of tractor; d) lateral acceleration of trailer; e) articulation angle between truck and trailer.

3.2 Evaluation of truck-CAT directional performance

To ensure the safe operations of articulated vehicles, various performance-based standards have been issued to specify the requirements on directional performance, which is generally evaluated in terms of the high-speed dynamic stability and low-speed manoeuvrability [7]. To effectively evaluate the low-speed manoeuvrability, the measure of path-following off-tracking (*PFOT*) was adapted in this study, which is defined as the maximum radial offset between the path of the truck's front axle centre and that of the rearmost trailer's rear axle centre during curved path negotiations. On the other hand, to assess the high-speed dynamic stability, the rearward amplification (*RWA*) ratio is commonly accepted, which is defined as the ratio of the maximum lateral motion of the rearmost trailer to the maximum of the lateral motion of the truck [22]. An articulated vehicle with high dynamic stability should have a low *RWA* ratio. The yaw damping ratio poses significant impacts on the trailer swing and the articulation angle between the leading and trailing units. This yaw damping ratio is related to the *RWA* ratio. The effects of system parameters on the damping ratio of articulated vehicles can be assessed using eigenvalue analysis. The evaluation methods for the directional performance of the truck-CAT combination are briefly introduced as follows.

Eigenvalue analysis: For the truck-CAT combination, the damping ratio of a motion mode can be determined by eigenvalue analysis of the 3-DOF linear model. Based on Eq. (11), the system matrix is formulated as $-M^{-1}D$. Given a vehicle motion mode, the corresponding complex eigenvalue of the system matrix can be expressed as:

$$s_{1,2} = R_e \pm j\omega_d \tag{15}$$

The damping ratio, ζ, can be determined by:

$$\zeta = \frac{-R_e}{\sqrt{R_e^{\ 2} + \omega_d^{\ 2}}} \tag{16}$$

The complex eigenvalue governs the features of the motion mode as follows:

(1) If $\omega_d = 0$, the eigenvalue is a real number, and the vehicle experiences a steady-state motion. When $R_e < 0$, the damping ratio is 1, $\zeta = 1$, the system is in monotonic convergence. When $R_e > 0$, the damping ratio is -1, $\zeta = -1$, the system is in a monotonous divergence state, and the vehicle is static instable. The vehicle may experience jack-knifing.

(2) If $\omega_d \neq 0$, the eigenvalue is a complex number. When $R_e < 0$, the damping ratio is limited in the range of $0 < \zeta < 1$, and the system is in a state of oscillation convergence. While $R_e > 0$, the damping ratio is bounded by $-1 < \zeta < 0$, and the system is in a state of oscillation and dynamic instability. Hence, the trailer will swing with increasing amplitude, and the truck-CAT will lose yaw stability.

With the system matrix of the 3-DOF linear truck-CAT model, an eigenvalue analysis was conducted. Fig. 5 shows the result of the two damped motion modes in terms of damping ratio versus vehicle forward speed. As shown in the figure, the damping ratios of the two motion modes decreases with the increase of vehicle forward speed. For the least damped motion mode (red dashed curve), the damping ratio takes the value of 0 at the speed of 20 m/s, which refers to the ‘dynamic’ critical speed, above which the truck-CAT will lose yaw stability.

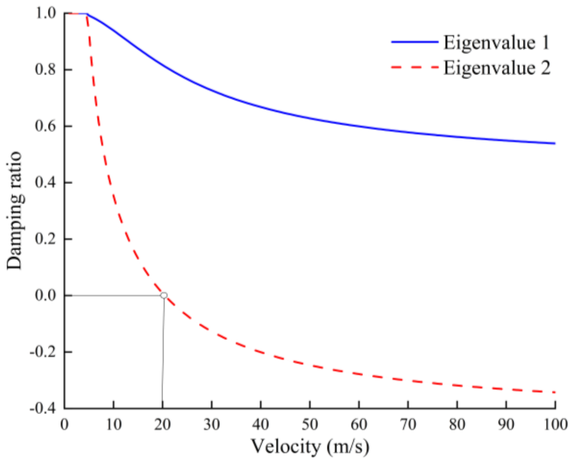

Figure 5: Damping ratios versus vehicle forward speed.

The system matrix, $-M^{-1}D$, is dependent on vehicle forward speed, structural parameters, inertia parameters, and tire cornering stiffnesses. Thus, using the system matrix, we may conduct eigenvalue analysis to evaluate the sensitivity of vehicle structural parameters to the dynamic stability of the truck-CAT combination.

Evaluation of *RWA* ratio: The *RWA* measure was adopted in terms of the ratio of the maximum yaw rate of the trailer to that of the truck, that is, $RWA = r_2 / r_1$. For the *RWA* measure assessment, two single lane-change testing manoeuvres were commonly accepted: 1) an open-loop procedure with a single sinewave steering input; and 2) a closed-loop manoeuvre with a single sinewave lateral acceleration input. To effectively evaluate the sensitivity of dynamic stability to structural parameters, the first testing method was adopted.

Evaluation of *PFOT* measure: The *PFOT* was measured under low-speed curved path negotiations only considering the kinematics of the truck-CAT combination. Fig. 6 shows the plan views while the truck-CAT negotiates the 360º roundabout with the outer and inner circle diameters of 25,000 and 10,600 mm, respectively. Based on the definition of *PFOT*, this measure is calculated as follows:

$$PFOT = R_1 - \sqrt{R_1^2 - L^2 + KO^2 - KA^2} \tag{17}$$

where R_1, R_2, R_3 and R_4 are the turning circle radius of truck front axle centre, truck rear axle centre, trailer axle centre, and articulation point, respectively, L is truck wheelbase, KO the distance between truck rear axle centre and articulation point, and KA the distance between trailer axle centre and articulation point. Eq. (17) indicates that the *PFOT* measure is only determined by vehicle structural parameters and geometric parameters of the manoeuvring.

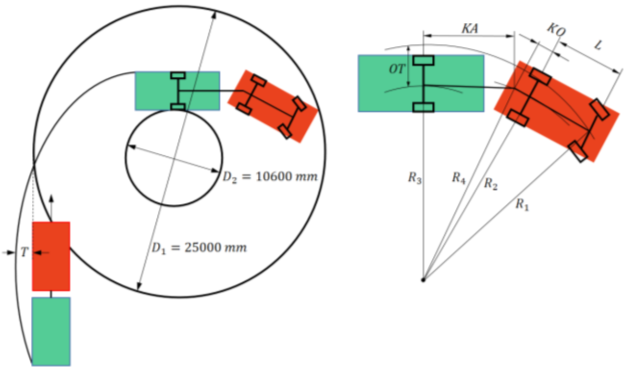

Figure 6: Roundabout manoeuvre showing the geometric parameters governing the *PFOT* measure.

4. RESULT ANALYSIS AND DISCUSSION

With the nominal values listed in Table I, the 3-DOF linear model or the 3-D nonlinear TruckSim was simulated to represent the lateral dynamics of the baseline truck-CAT combination. Based on the 3-DOF linear model, eigenvalue analyses were conducted to evaluate the sensitivity of the vehicle dynamic stability to its structural parameters. From the sensitivity analyses, the conceptual design of the variable-length drawbar for CATs was established. Then, the proposed variable-length drawbar was validated using the simulation of the 3-D nonlinear TruckSim model.

4.1 Sensitivity of the dynamic stability of truck-CAT to structural parameters

For articulated vehicles, structural parameters significantly affect the directional performance [7, 8]. In this study, the eigenvalue analysis approach was conducted to evaluate the sensitivity of the dynamic stability of the truck-CAT combination to the selected structural parameters listed in Table II. In the sensitivity analysis, one parameter was varied at a time. Each parameter was permitted to vary by ±2 % from its nominal value. The optimal value of each parameter is listed in Table II. Note that in the table, U_{1c} denotes the 'dynamic' critical speed identified using the eigenvalue analysis.

Table II: Effects of structural parameters on the 'dynamic' critical speed of the CAT combination.

Structural parameters	Nominal values	Upper bounds	Lower bounds	Optimal values	Nominal U_{1c} (m/s)	Optimized U_{1c} (m/s)	Increase of U_{1c} (%)
a (m)	2	2.04	1.96	2.04 ↑	20.451	20.456	0.024
b (m)	3.6	3.67	3.53	3.67 ↑	20.451	20.681	1.124
d (m)	5.25	5.36	5.15	5.15 ↓	20.451	20.672	1.081
e (m)	6.11	6.23	5.99	6.23 ↑	20.451	20.910	2.244
h (m)	0	0.2	-0.2	0.2 ↑	20.451	24.450	19.554

As indicated in Table II, among the five parameters, the parameter *h* (distance from the trailer axle to its centre of gravity) has the most significant impact on the increase of critical speed. However, considering the configuration constraint of the centre-axle-trailer, the trailer axle should be positioned exactly below the trailer centre of gravity, that is, *h* should take the value of zero. It is found that the parameter *e* poses the next most significant impact on the critical speed. Increasing the value of *e* led to the increase of the critical speed. Under the constraint of $h = 0.0$, the parameter *e* represents the 'equivalent' trailer drawbar length denoted as '*KA*' as shown in Fig. 6. Eq. (17) implies that increasing '*KA*', i.e., the parameter *e*, resulted in the increase of the *PFOT* measure. Thus, increasing *e* alone improves the dynamic stability (i.e., increasing the critical speed), but degrades the manoeuvrability (i.e., increasing *PFOT* measure). It is concluded that varying the parameter *e* leads to the trade-off between the high-speed dynamic stability and the low-speed manoeuvrability. This conclusion inspires the conceptual design of the varying-length drawbar for the CAT trailer: to improve the high-speed dynamic stability, the drawbar takes longer length, whereas to enhance the low-speed manoeuvrability, the drawbar takes shorter length. In the following subsections, the variable-length drawbar was to be introduced, which would then be validated using simulation based on the 3-D nonlinear TruckSim model.

4.2 Variable-length drawbar for CAT

Fig. 7 shows the dimensions of the truck-CAT combination. GB 1589-2016 specifies the variation ranges of the relevant structural parameters, i.e., $L_{max} = 20$ m, $L_{0\max} = 12$ m, $L_{2\max} = 8$ m, $A < 1.4$ m, $A+T+L_2 < 12$ m, and $-6° < \alpha < 6°$. Based on the permitted variation of the parameter *S* (the minimum clearance between adjacent truck corner and the trailer) in Fig. 7 b, the national standard of China, GB/T 37245-2018 [25], specifies three sets of structural parameter values denoted by S_1, S_2, and S_3, as listed in Table III, according to different length of *A*.

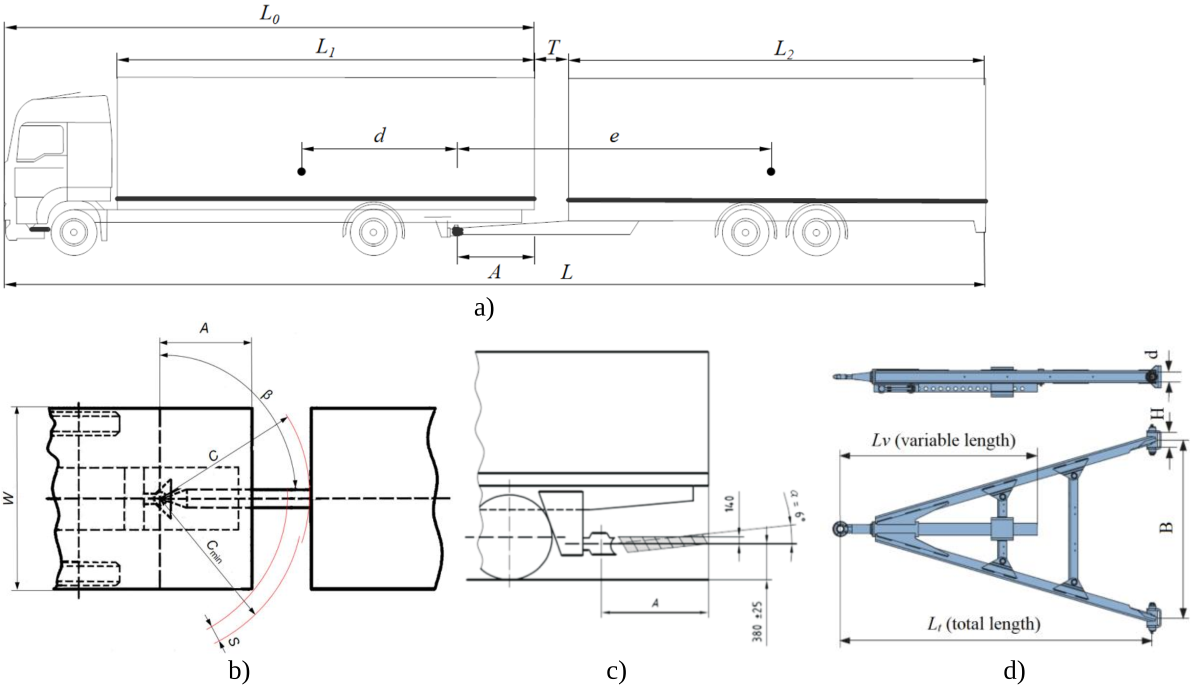

Figure 7: a) vehicle dimensions, b) parameters determining the drawbar length (top view), c) parameters determining the drawbar length (side view), d) configuration of variable-length drawbar.

In Fig. 7 b, *C* denotes the horizontal distance between the pintle hitch of the drawbar and the front end of the trailer. The minimum value of *C*, i.e., $C_{\min}$, is determined by:

$$C_{\min} = \sqrt{\left(\frac{W}{2}\right)^2 + A^2} + S \tag{18}$$

$$T_{\min} = C_{\min} - A \tag{19}$$

where *A* is the longitudinal distance between the drawbar hitch and the truck rear end, *W* the truck width, and $T_{\min}$ the minimum gap between the truck and trailer. For safe vehicle operations, *S* shall not be less than 250 mm.

Table III: Three sets of parameter values and performance measures based on TruckSim simulation.

Parameter value sets	***A*** (mm)	C_{min} (mm)	T_{min} (mm)	***e*** (mm)	***d*** (mm)	***PFOT*** (mm)	***RWA***
S_1	1400	2150	750	6060	5365	3599	1.68
S_2	1600	2300	700	6210	5215	3714	1.64
S_3	1900	2,540	640	6450	4975	3902	1.62

Thus, with given values of *A*, *S* and *W*, the values of C_{min} and T_{min} can be calculated using Eqs. (18) and (19), respectively, while the values of *e* and *d* can be determined based on the specifications of GB 1589-2016. To accommodate the variation of the parameter *S*, the variable-length drawbar shown in Fig. 7 d is recommended. The length of L_v can be adjusted according to the relevant specifications and operating conditions, e.g., low-speed curved path negotiations or high-speed evasive manoeuvres.

4.3 Assessment of variable-length drawbar

To examine the effect of the variable-length drawbar, the roundabout manoeuvre at the forward speed of 10 km/h and the single lane-change manoeuvre at the forward speed of 80 km/h were simulated based on the three sets of parameters values, i.e., S_1, S_2, and S_3, listed in Table III. Fig. 8 shows the simulation results derived from the simulated high-speed evasive manoeuvre and low-speed roundabout.

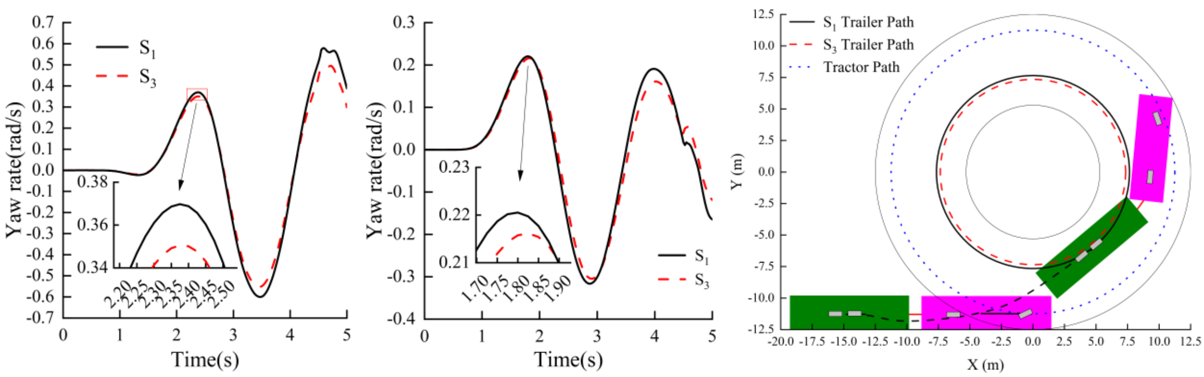

Figure 8: Simulation results using TruckSim model: a) yaw rate of truck in high-speed manoeuvre, b) yaw rate of trailer in high-speed manoeuvre, c) trajectory of truck and trailer in low-speed roundabout.

Fig. 8 a and 8 b demonstrates that compared with the S_1 case, the design of S_3 improves the high-speed dynamic stability, while Fig. 8 c indicates that with respect to the S_1 case, the design of S_3 degrades the low-speed manoeuvrability. As indicated in Table III, among the three sets of parameters values, i.e., S_1, S_2, and S_3, the respective values of *e* are 6060, 6210, and 6450 mm, respectively. As the value of *e* increases, the *RWA* measure decreases from 1.68, via 1.64, to 1.62, thereby increasingly improving the dynamic stability of the vehicle. The simulation results derived from the high-speed evasive manoeuvre based on the TruckSim model are consistent with the eigenvalue analysis results listed in Table II. On the other hand, as the value of *e* increases, the *PFOT* measure also increases from 3599, via 3714, to 3902 mm, thereby increasingly degrading the low-speed manoeuvrability. Once again, with the increase of *e*, the trade-off relationship the low-speed manoeuvrability and high-speed dynamic stability is clearly disclosed.

The above simulation results can be used to implement the variable-length drawbar design and operation: under low-speed curved path negotiations, the length of L_v (see Fig. 7 d) should be shortened, resulting in a smaller *e* value and improving the vehicle manoeuvrability, whereas under high-speed evasive manoeuvre, the length of L_v should be lengthened, leading to a larger *e* value and increasing the vehicle dynamic stability.

5. CONCLUSIONS

The effects of structural parameters on the directional performance was explored in a truck-CAT combination in terms of low-speed manoeuvrability and high-speed dynamic stability. A 3-DOF linear yaw-plane vehicle model was generated to conduct linear stability evaluation based the eigenvalue analysis method, a 3-D nonlinear TruckSim model was developed for high fidelity numerical simulation. This study disclosed that for the truck-CAT combination, among the selected structural parameters, the distance between the pintle hitch to the trailer axle showed the most significant impact on both the high-speed dynamic stability and the low-speed manoeuvrability. This disclosure inspired the conceptual design of the variable-length drawbar for central-axle-trailers. With the rigid constraints on the dimensions of truck-CAT combinations specified by the standards of GB 1589-2016 and GB/T 37245-2018, a feasible and cost-effective scheme of the variable-length drawbar was proposed and evaluated by numerical simulation. It is demonstrated that under different operating conditions, varying the length of the drawbar can improve either the low-speed manoeuvrability or enhance the high-speed dynamic stability. The length adjustment of the variable-length drawbar can only be implemented manually prior to the vehicle operation under a specified operating condition. The active control of the variable-length drawbar may be explored in the near future so that this drawbar can be controlled and manipulated in real-time for improving the directional performance of truck-CAT combinations.

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