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Study on Effects of Road Conditions on the Lateral Instability of Tractor Semitrailer Vehicle during Turning Maneuver

TA Tuan Hung[1], DUONG Ngoc Khanh[2*], VO Van Huong[1]
[1] University of Transport Technology, Hanoi, Vietnam
[2] Hanoi University of Science and Technology, Hanoi, Vietnam
[*]Email: khanh.duongngoc@hust.edu.vn

Abstract

Instability of vehicle can be defined as an unexpected response maneuver that induces disturbance, occurring in the ground plane. This can include the longitudinal, lateral, pitch, yaw, roll direction, or their combinations. Many tractor semitrailer vehicle accidents can be caused by lateral instabilities which may be classified into two types: rollover and yaw instability. Rollover occurs when centrifugal forces imposed on the vehicle during a maneuver exceed the rollover threshold of the vehicle. Yaw instability often occurs in tractor semitrailer vehicles during turning maneuver on the road with low friction coefficient. The yaw instability is shown by the loss of motion trajectory or Jack-knife. This paper establishes a dynamic model of tractor semitrailer vehicle, based on Multi-Body System Method analysis and Newton-Euler equations with Burckhardt's tire model. This model is applied to evaluate the effect of road conditions on lateral instability of the tractor semitrailer vehicle during turning maneuver. The results can serve as the basis for determining the early warning and controlling the lateral instability of tractor semitrailer vehicle with the dynamic model.

Keywords: Yaw instability, rollover, tractor semitrailer vehicle, Jack-knife, Burckhardt's tire model, road conditions.

1. Introduction

In recent years, transportation by articulated vehicles has developed robustly to improve transportation productivity and reduce traffic jams, emissions, and environmental pollution. In Vietnam, the maximum allowable weight for a 6-axle tractor semi-trailer vehicle is 48000 kg. However, the development of such vehicles could cause problems such as increased pressure on roads, reduced road lifetime, and more traffic accidents. Accidents involving tractor semitrailer vehicle have serious consequences for road users, and incidents induce major congestion or damage to the environment or the infrastructure at disproportionate economic costs. The risk of having deaths in accidents involving heavy vehicles is 2.4 times higher than that in accidents involving only light vehicles. This is mainly due to the important gross mass difference between light vehicles and trucks.

Lateral instability of heavy vehicle can be defined as an unexpected response maneuver inducing disturbance, occurring in the ground plane. This can include the longitudinal, lateral, pitch, yaw, roll direction, or their combinations.

Nowadays, tractor semitrailer vehicles often pose serious highway safety risks due to their excessive weights, larger dimensions, and directional and roll stability limits. Lateral instability of tractor semitrailer vehicles can be classified into two types: yaw instability and roll instability or rollover [1]. The yaw instability is defined as swing trailer, oscillation trailer and Jack-knife. The yaw instability can be caused by either braking or combined braking and steering maneuvers on the low adhesion coefficient of roads (Fig. 1). Jack-knife is characterized by rapid and uncontrollable relative angular yaw motion between the tractor and the semitrailer [2].

The rollover occurs when centrifugal forces imposed on the tractor semi-trailer vehicle during a maneuver exceed the rollover threshold of the latter. The rollover of the vehicle can be further classified into two main categories: tripped rollover and maneuver rollover. Tripped rollover can occur when there is a collision with another vehicle or with any obstacle. Rollover maneuvers can occur during lane changes or turning maneuvers on roads with high adhesion coefficients. The rollover condition of tractor semitrailer vehicle is determined when tires on axles lose road contact (wheel lift-off) [3].

Specifically, this paper focuses on the effect of road conditions on lateral instability of tractor semitrailer vehicle during turning maneuver. A dynamic model of tractor semitrailer vehicle is established on the basis of Multi-Body System Method analysis and Newton-Euler equations with Burckhardt's tire model. These results can serve as the basis for determining the early warning and controlling the lateral instability of tractor semitrailer vehicle with the dynamic model.

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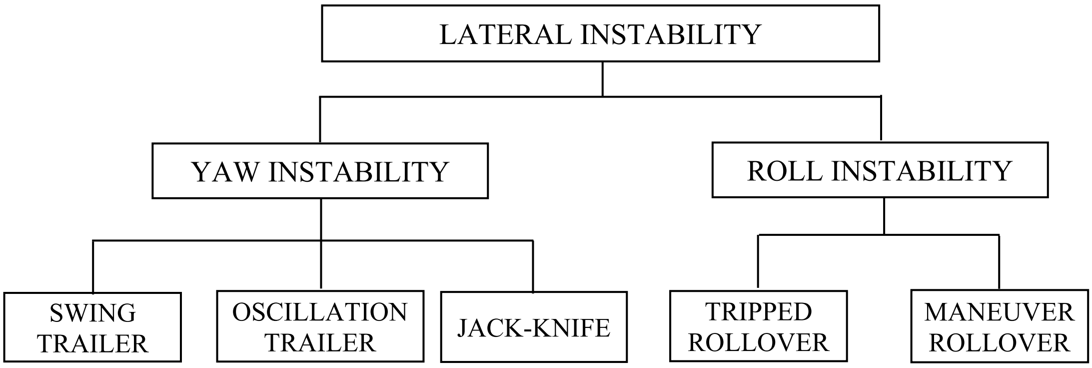

Fig. 1. Lateral instability categorization of tractor semitrailer vehicle

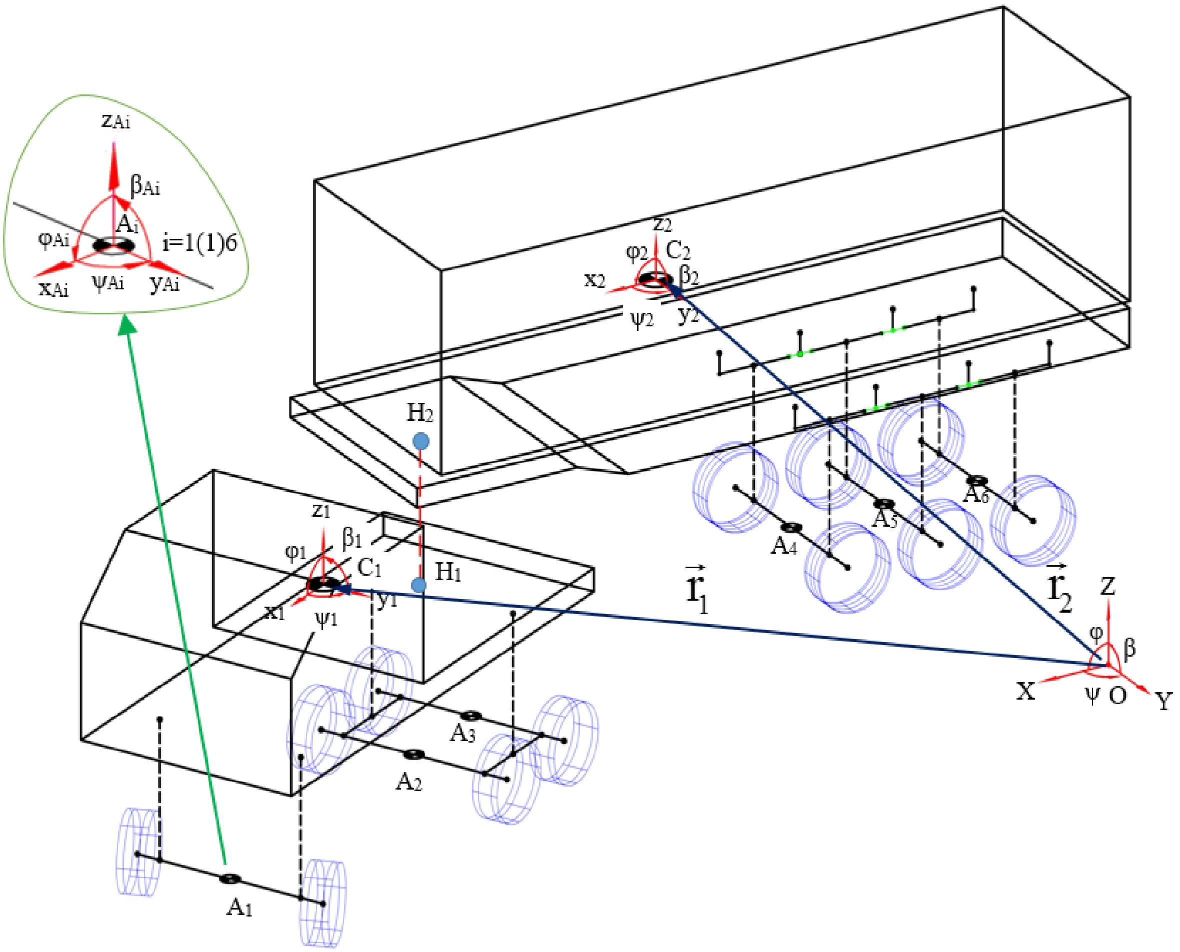

Fig. 2. Tractor semitrailer vehicle coordinate systems

2. Dynamic Model of Tractor Semitrailer Vehicle

2.1. Equations of Motions

This study is framed to focus on the tractor semitrailer vehicle which is composed of a 3-axle tractor vehicle and a 3-axle semitrailer vehicle. The tractor consists of a sprung mass and axles and tires. The semitrailer vehicle has a sprung mass, axles and tires. The tractor and the semitrailer vehicle are connected at the fifth wheel hitch as shown in Fig. 2.

The motion of two sprung masses in the coordinate system model is assessed. *OXYZ* is the earth-fixed coordinate system. $C_1x_1y_1z_1$ and $C_2x_2y_2z_2$ are sprung masses coordinate systems of the tractor and semitrailer, which are fixed at the center of gravity, respectively. The relative motion of $C_1x_1y_1z_1$ and $C_2x_2y_2z_2$ with the fixed coordinate system *OXYZ* are the rotation matrices. These matrices are based on a set of body (*X-Y-Z*) rotations (Roll-Pitch-Yaw) with β_k-φ_k-ψ_k angles [4] as follows:

$$
{}^{O}_{C_k}R = \begin{bmatrix} c\psi_k c\varphi_k & c\psi_k s\varphi_k s\beta_k - s\psi_k c\beta_k & c\psi_k s\varphi_k c\beta_k + s\psi_k s\beta_k \\ s\psi_k c\varphi_k & s\psi_k s\varphi_k s\beta_k + c\psi_k c\beta_k & s\psi_k s\varphi_k c\beta_k - c\psi_k s\beta_k \\ -s\varphi_k & c\varphi_k s\beta_k & c\varphi_k c\beta_k \end{bmatrix}
$$

$c = \cos \qquad s = \sin$

(1)

From these coordinate systems, the six motions of the sprung mass *k* are established with Newton's and Euler's equations [5] of motion in the sprung mass coordinate systems as follows:

$$
\begin{cases} m_k(\dot{v}_{xk} - v_{yk}\omega_{zk} + v_{zk}\omega_{yk}) = F_{xk} \\ m_k(\dot{v}_{yk} - v_{zk}\omega_{xk} + v_{xk}\omega_{zk}) = F_{yk} \\ m_k(\dot{v}_{zk} - v_{xk}\omega_{yk} + v_{yk}\omega_{xk}) = F_{zk} \\ I_{xk}\dot{\omega}_{xk} + (I_{zk} - I_{yk})\omega_{zk}\omega_{yk} = M_{xk} \\ I_{yk}\dot{\omega}_{yk} + (I_{xk} - I_{zk})\omega_{xk}\omega_{zk} = M_{yk} \\ I_{zk}\dot{\omega}_{zk} + (I_{yk} - I_{xk})\omega_{yk}\omega_{xk} = M_{zk} \end{cases} \tag{2}
$$

where: *k*=1: sprung mass of the tractor; *k*=2: sprung mass of the semitrailer; v_{xk}, v_{yk}, v_{zk}: the translational velocities of sprung mass *k*; ω_{xk}, ω_{yk}, ω_{zk}: the rotational velocities of sprung mass *k*; m_k: the mass of the sprung mass *k*; I_{xk}, I_{yk}, I_{zk}: moments of inertia of the sprung mass *k*; F_{xk}, F_{yk}, F_{zk}: the total applied forces acting on the sprung mass *k* resolved parallel to $C_kx_ky_kz_k$; M_{xk}, M_{yk}, M_{zk}: the total applied moments acting on the sprung mass *k* resolved parallel to $C_kx_ky_kz_k$.

Each of the axles is thus characterized as a rigid beam with 2 DOFs (vertical z_{Ai} and roll motion β_{Ai}) (Fig. 3).

Vertical and lateral forces and roll moment balance on the axles lead to the following equations:

$$
\begin{cases} m_{Ai}\dot{v}_{zAi} - m_{Ai}(\omega_{yAi}v_{xAi} - \omega_{xAi}v_{yAi}) = F_{AZi} \\ J_{Axi}\dot{\omega}_{xAi} - (J_{yAi} - J_{zAi})\omega_{yAi}\omega_{zAi} = M_{AXi} \end{cases} \tag{3}
$$

Lateral forces between the sprung masses and the axles, denoted by F_{Ri}, are assumed to be transmitted through the respective roll centers.

Total applied forces and moments acting on sprung mass *k* are calculated from the suspension systems forces, aerodynamic forces [6], and fifth wheel hitch forces and moments. The spring and damper forces of the steering axle are calculated from the vertical displacement between sprung mass of tractor vehicle and steering axle. 'Walking-beam' model with 2 degrees of freedom of the combined beam joins the two axles is used to calculate the spring and damper forces of the rear suspension of the tractor vehicle [7]; The total applied forces and moments acting on the axle are calculated from the suspension systems and tire-road interaction. The tire forces are longitudinal, lateral and vertical. Tire forces are dependent on tire-road deformation, road adhesion coefficient of friction, steering wheel angles, etc. These forces are determined by the Burckhardt tire model [8,9].

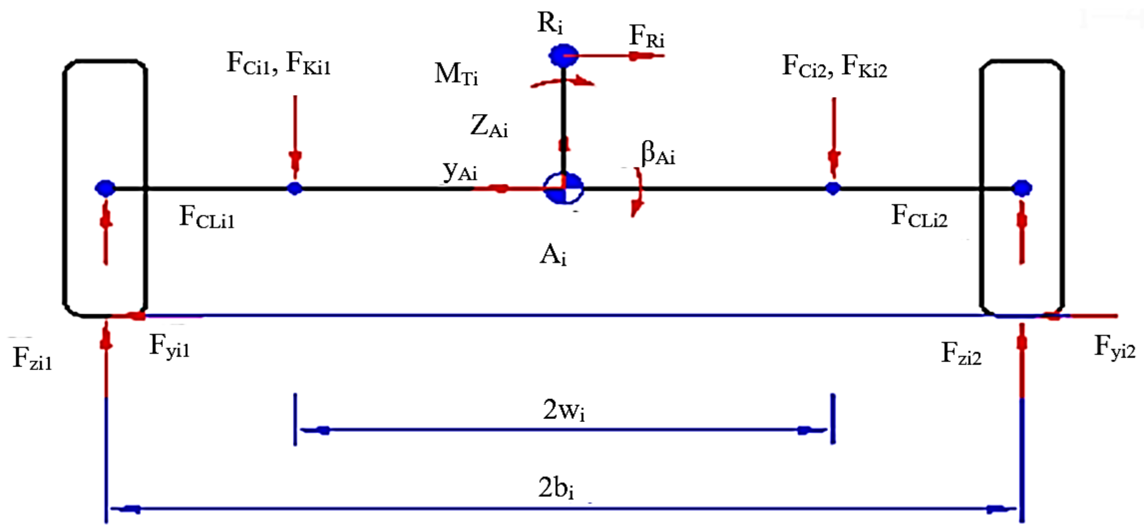

Fig. 3. Model of unsprung masses

2.2. Equations of Motion of the Wheel

This paper assumes that wheels are described as elastic on rigid roads. The torque transmitted to the wheels T_{Wij}, the longitudinal tire forces F_{xij} and the effective radius of wheels r_{dij} are the inputs of wheel dynamics models (Fig. 4).

The rotational velocity of the wheels ω_{Wij} is the output of these models. The dynamic equations for the wheel rotational dynamics are:

$$I_{Wij}\dot{\omega}_{Wij} = T_{Wij} - F_{xij} r_{dij} \qquad (i = 1 \div 6;\, j = 1 \div 2) \quad (4)$$

where: T_{Wij}>0 for the driving wheels (rear wheels of the tractor vehicle); T_{Wij}=0 for the non-driven wheels.

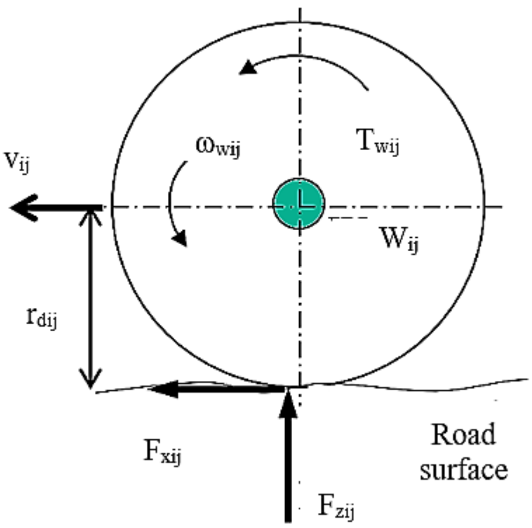

Fig. 4. Schematic of wheel dynamics

Table 1. Values of the Burchkhardt tire model coefficients [9]

Road Surface	C_1	C_2	C_3
Asphalt, dry	1,281	23,99	0,52
Cobblestone, wet	0,4004	33,7080	0,1204
Snow	0,1946	94,129	0,0646
Ice	0,05	306,39	0

2.3. Tire Modelling

Vehicle motions are primarily caused by forces and moments developed at the tire-road interface. This paper assumes that the overturning moment and other moments are negligible. Pacejka tire models and Burckhardt tire models mostly exhibit similar behavior in different road conditions [8]. The longitudinal and lateral forces are computed based on Burckhardt Tire Model as follows:

$$\begin{cases} F_{xij} = \dfrac{s_{xij}}{\sqrt{s_{xij}^2 + s_{yij}^2}}\left(C_1\left(1 - e^{-C_2\sqrt{s_{xij}^2+s_{yij}^2}}\right)\right. \\ \qquad \left. - C_3\sqrt{s_{xij}^2 + s_{yij}^2}\right)F_{zij} \\ F_{yij} = \dfrac{s_{yij}}{\sqrt{s_{xij}^2 + s_{yij}^2}}\left(C_1\left(1 - e^{-C_2\sqrt{s_{xij}^2+s_{yij}^2}}\right)\right. \\ \qquad \left. - C_3\sqrt{s_{xij}^2 + s_{yij}^2}\right)F_{zij} \end{cases} \quad (5)$$

The inputs are tire vertical loads F_{zij}, lateral slip angles s_{ij}, and longitudinal slip ratios s_{ij} etc. The values of the Burckhardt tire model coefficients C_1, C_2, and C_3 are shown in Table 1.

2.4. Modelling of Fifth Wheel Hitch

The modelling of fifth wheel hitch is presented in Fig. 5. Assume that the coupling mechanisms are related to rigid in translation. The forces transmitted through the coupling are determined from kinematic constraints such as:

$$\vec{R}_{H1} - \vec{R}_{H2} = 0 \quad (6)$$

This means that the acceleration at a coupling point is the same for both the tractor and the semitrailer of the vehicle.

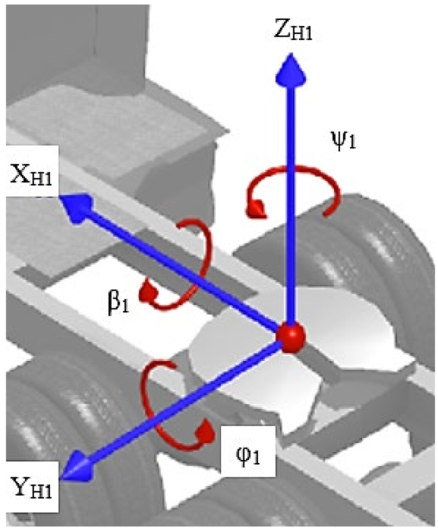

Fig. 5. Model of fifth wheel hitch

The roll moment M_{Hx1} acting through the fifth wheel may be computed as:

$$\begin{cases} M_{Hx1} = C_{mHx}(\beta'_1 - \beta_1) \\ M_{Hx2} = -\cos(\psi_2 - \psi_1)M_{Hx1} \end{cases} \quad (7)$$

where C_{mHx} is the roll angle stiffness of the fifth wheel hitch.

β'_1 is calculated as:

$$\beta_1^{'} = \text{atan}\frac{\sin\beta_2\cos(\psi_2-\psi_1)-\varphi_2\cos\beta_2\sin(\psi_2-\psi_1)}{\varphi_1\sin\beta_2\sin(\psi_2-\psi_1)+\cos\beta_2} \quad (8)$$

2.5. Assessment Criteria

The rollover signal is based on the load transfer ratio. Roll Safety Factor (*RSF*) is the load transfer ratio between the left and the right sides of all tires without the tires of the 1st axle [2]. The formula for the 6-axle tractor semitrailer vehicle is as follows:

$$RSF=\frac{\left|\sum_{i=2}^{6}(F_{zi2}\text{-}F_{zi1})\right|}{\sum_{i=2}^{6}(F_{zi2}+F_{zi1})} \quad (9)$$

where the vertical tire force F_{zij} (i=1÷6; j=1: left wheels, j=2: right wheels) at each wheel is calculated from the vertical deflection of tire.

In this paper, the articulated angle is used to determine the Jack-knife of tractor semitrailer vehicle. This angle is calculated from the yaw angle of sprung masses ψ_k as follows:

$$\psi_H = \psi_1 - \psi_2 \quad (10)$$

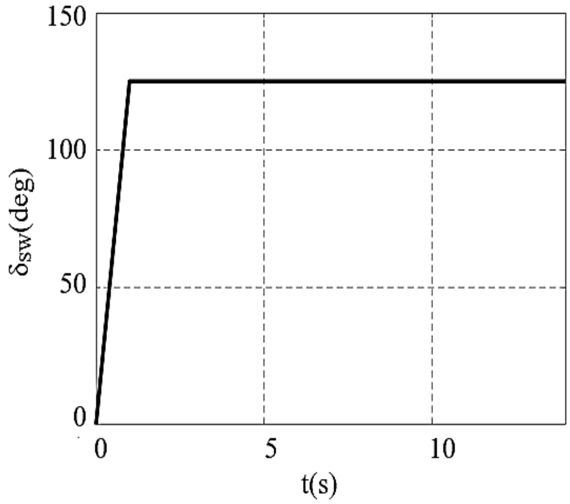

Fig. 6. Steering wheel angle

Table 2. Simulation parameters of a 6-axle tractor semitrailer vehicle.

Parameters	Symbol (Unit)	Value
Sprung mass of the tractor	m_1(kg)	7620
Sprung mass of the semi-trailer	m_2(kg)	34715
Unsprung masses of the axles	m_{A1}; $m_{A2,3}$ $m_{A4,5,6}$(kg)	640;1150;780
Wheelbase of the tractor	L_1+c(m)	3.24+1.34
Wheelbase of the semi-trailer	L_2+d+d(m)	6.945+1.31+1.31
Half-track width of the axles	b_1; $b_{2,3}$; $b_{4,5,6}$ (m)	1.025; 0.93; 0.925
Half spring spacing of the axles	w_1; $w_{2,3}$; $w_{4,5,6}$ (m)	0.6; 0.5; 0.5
Height of the fifth wheel hitch	h_H (m)	1.33
Height of tractor's CoG	h_1 (m)	1.22
Height of semi-trailer's CoG	h_2 (m)	2.2
Tire size		11.00R20
Roll moment of inertia of tractor's sprung mass	I_{x1}(kgm^2)	11494.3
Roll moment of inertia of semi-trailer's sprung mass	I_{x2}(kgm^2)	52828.7
Pitch moment of inertia of tractor's sprung mass	I_{y1}(kgm^2)	38399.2
Pitch moment of inertia of semi-trailer's sprung mass	I_{y2}(kgm^2)	484022.2
Yaw moment of inertia of tractor's sprung mass	I_{z1}(kgm^2)	34969.9
Yaw moment of inertia of semi-trailer's sprung mass	I_{z2}(kgm^2)	467066.4
Suspension stiffness of the axles	C_{1j}, C_{23j}, $C_{4,5,6j}$(kN/m)	250; 1400; 2500
Suspension damping ratio of the axles	K_{1j}, $K_{2,3j}$, $K_{4,5,6j}$(kNs/m)	15; 30; 30
Tire vertical stiffness of the single wheel	C_L (kN/m)	980
Maximum adhesion coefficient of friction	$\mu_{max,0}$	0.8
Air resistance coefficient	$C_{x,y}$	0.9

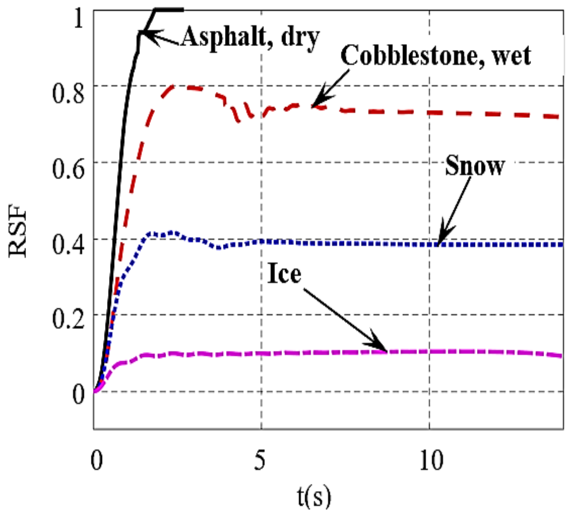

Fig. 7. Roll Safety Factor

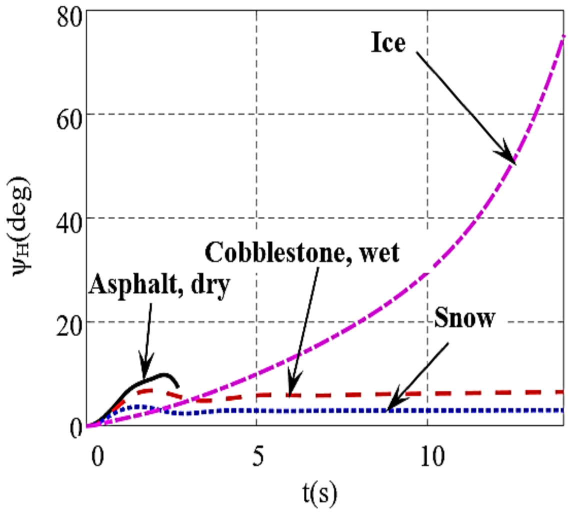

Fig. 8. Articulated angle

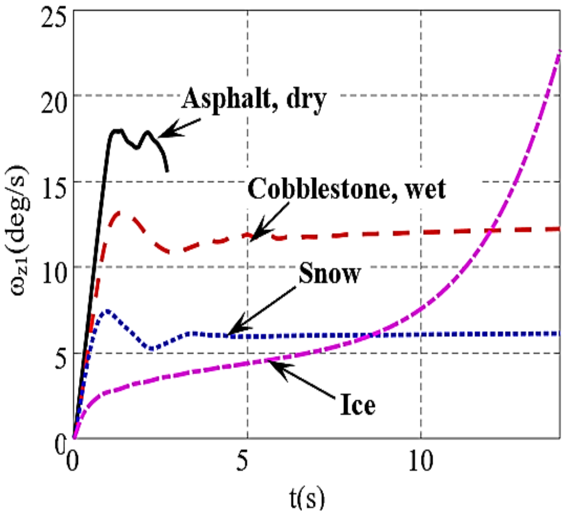

Fig. 9. Yaw rate of sprung mass of tractor

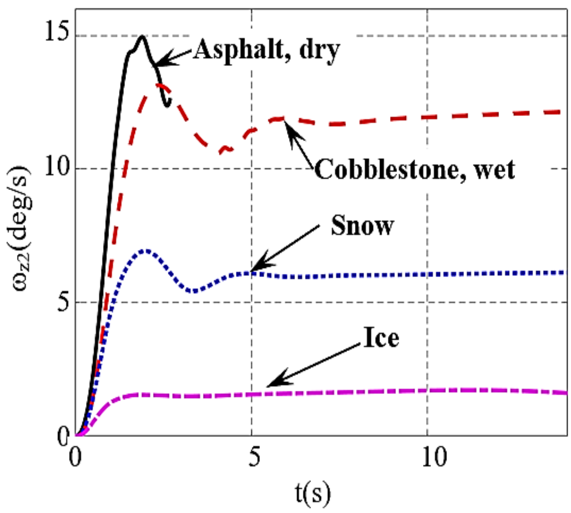

Fig. 10. Yaw rate of sprung mass of semitrailer

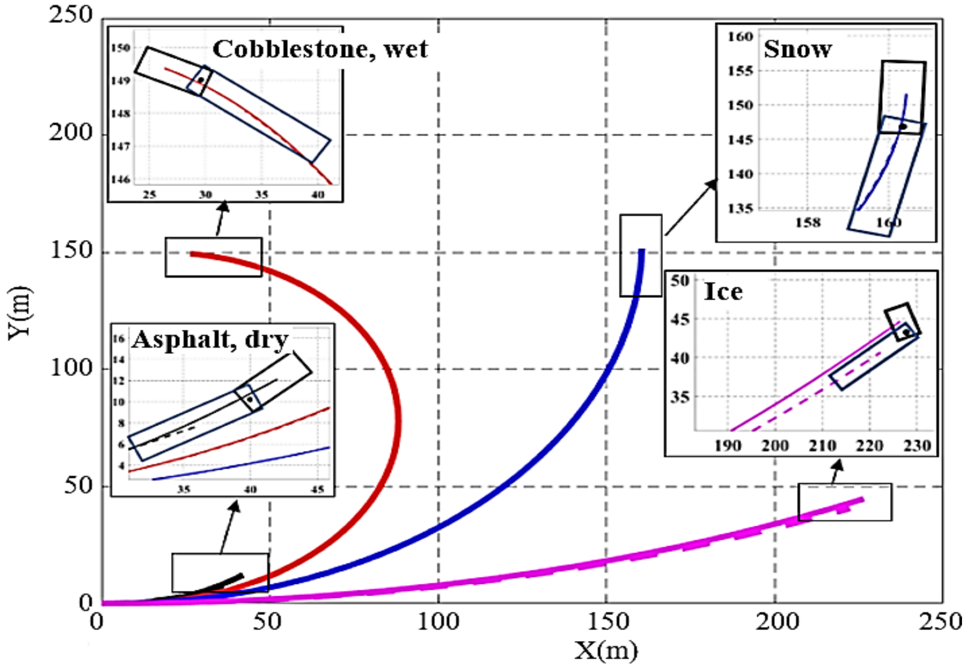

Fig. 11. Trajectory of motion

3. Simulation Results and Discussions

The tractor semitrailer vehicle model is simulated. All parameters of the 6-axle tractor semitrailer vehicle are defined in Table 2 [10]. The model is simulated in certain road conditions by the Burckhardt model with parameters in Table 1. The turning maneuver in an open-loop mode is characterized by a Ramp Steer Maneuver (*RSM*) [11]. The definition of the *RSM* is shown graphically in Fig. 6 which shows the steering wheel angle profile. The *RSM* is based on the steering wheel angle input at a constant rate until the peak steering magnitude is achieved. The magnitude of the steering wheel angle δ_{SWmag} is equal to 125 (deg). The initial of longitudinal velocity is 60 (km/h). This velocity is high for the heavy vehicle in turning maneuvers. The results of the *RSF*, yaw rate of bodies, articulated angle and trajectory of motion of tractor semitrailer vehicle are shown below from Fig. 7 to Fig. 11.

Fig. 7 illustrates the roll safety factor (*RSF*) of the 6-axle tractor semi-trailer vehicle in the time domain. When the vehicle is turning to maneuver on the Asphalt and dry of road, the *RSF* is toward 1 quickly (at the 2,1s). This is a signal of rollover conditions of tractor semitrailer vehicle. For the other road, the tractor semitrailer vehicle is not rollover (*RSF*<1). However, the trajectory of the tractor semitrailer vehicle increases. In addition to other parameters such as the yaw rate of the bodies which reach lower values depending on the road. Especially, when the tractor semitrailer vehicle is turning on an ice road, yaw rate of semitrailer increase slowly and yaw rate of tractor increase rapidly (Fig. 9 and Fig. 10). That shows a faster increase in the yaw angle of the tractor than that of the semitrailer. The articulated angle increases very quickly and reaches 76 (deg) at the simulation time of about 14(s) (Fig. 8). This is the early signal of the Jack-knife of the tractor semitrailer vehicle. This is shown clearly in the trajectory of motion (Fig. 11).

4. Conclusion

The instability of tractor semitrailer vehicle is often demonstrated in two types: rollover and yaw instability. In this paper, a dynamic model of a 6-axle tractor semitrailer vehicle is established based on Multibody System analysis with the Burckhardt tire model. This model is applied to evaluate the effects of road conditions on the instability of the tractor semitrailer vehicle during turning maneuver. The results of this paper show that, when the tractor semitrailer vehicle turns at a velocity of 60 (km/h), the vehicle will be lost in the trajectory of motion on a snow and ice road. The articulated angle can be the early signal of a Jack-knife. As evaluated in this paper, the rollover of tractor semitrailer vehicle might occur during turning on the Asphalt and dry of road with the reach to 1 of *RSF*. Arguably, these results can serve as the basis for determining the early warning and controlling the lateral instability of tractor semitrailer vehicle with the dynamic model.

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