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A New Method for Directional Control of a Tractor Semi-trailer

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Abstract: In order to improve the stability and maneuverability of the articulated vehicle, a reference model for directional control is proposed. First, both linear and nonlinear models of the articulated heavy vehicle are developed. The linear model is used to design the controller and generating the reference response while the nonlinear model is used to evaluate the system responses. The yaw rate and lateral velocity of the towing unit and the articulation angle between the towing and trailer unit are considered as the primary motions associated with the directional behavior of the articulated vehicle. The desired values of these states are defined in such way to improve the maneuverability and the stability of the articulated vehicle. A linear controller based on the optimal control theory has been designed to make the actual vehicle model follow the ideal responses. Finally, this paper investigates the effectiveness of different methods on vehicle and driver behaviors during high speed lane change and low speed turning maneuvers. Computer simulation results confirm the significant effects of the proposed method on enhancing the handling behavior of the articulated vehicle.

Key words: Reference Model, Directional Control, Articulated Vehicle, Control Steering Angle, Off-tracking.

INTRODUCTION

The goal of the active chassis control systems has been focused on the handling and stability enhancement. In the last three decades, automotive manufactures have made great progress in the field of active chassis control systems. Active Front Steering (AFS), Four Wheel Steering (4WS), Vehicle Stability Control system (VSC), etc can improve the handling and/or the stability performance of the passenger vehicles. However, in articulated vehicles, the dynamics control systems are still in the beginning stages of development.

In passenger vehicles, the yaw rate and/or the side slip angle are often regarded as the control variables, so that the desired yaw rate tracking and the side slip bounding may lead to the handling and stability improvement, respectively (Yang *et al*., 2009). While in articulated vehicles, as considered in the literature, there is not a unified pattern in the selection of control variables and the definition of their reference values.

An articulated heavy vehicle in typical form consists of two units including tractor and semi-trailer. The tractor unit is controlled by the driver and the trailer unit used to carry heavy freight is connected to the tractor unit through the mechanical coupling point called fifth wheel. The researches which have been conducted in the field of the active chassis control of articulated heavy vehicle can be categorized in two main groups which are studied as follows:

In the first group, some researches limited their scope to the improvement of the semi-trailer unit behaviour so that some state variables of the semi-trailer unit may be regarded as the control objects. One of the main concerns of the trailer manoeuvrability is the off-tracking which refers to the phenomenon in which the rear end of the trailer unit does not closely follow the path of the tractor unit. Chen and Shie, 2011; Fancher *et al*., 1998; Odhams *et al*., 2011; and Tabatabaei *et al*., 2012 mainly concentrated on the control of the articulation angle to enhance the manoeuvrability through off-tracking mitigation. In these researches, a reference articulation angle has been derived so that the reference value tracking makes the trailer unit closely follow the trajectory of the tractor unit. Furthermore, Cheng and Cebon, 2008 and Cheng *et al*., 2011 proposed a model-based method which minimizes both of the off-tracking and the lateral acceleration of the trailer centre of gravity by the trailer steering at the same time.

On the other hand, it should be borne in mind that the driver's steering input is governed mainly by his/her reaction to the behaviour of the tractor unit (Palkovics and El-Gindy, 1996). Consequently, the behaviour of the tractor unit should be controlled to provide the driver with an appropriate situation to make him more at ease decisions. Therefore, in the second group, several research efforts have been made on the simultaneous active control of both tractor and trailer units. Palkovics and El-Gindy, 1996 proposed a Linear Quadratic Regulator (LQR) controller to improve the high speed directional behaviour of the articulated vehicle. The improvement is supplied through minimizing the state variables including tractor yaw rate, tractor side slip angle, articulation angle and the articulation rate. The idea of minimization using LQR controller was also followed by El Gindy *et al*., 2001 and Hac *et al*., 2008. However, it seems that the idea of minimization reflects more the stability improvement than the manoeuvrability enhancement. Instead of using the idea of minimization, there is a vital

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need for devising a control system which can make the actual articulated vehicle behaviour track a desired behaviour.

Unfortunately, very little research has been conducted in this area. Wang *et al*., 2008 proposed an adaptive steering controller for improving the handling and stability of driver-combined-vehicles system. In this research, a desired value for the articulation angle has been derived based on a geometric analysis. However, it seems that the proposed reference value is applicable only in the case of a constant driver steering angle.

Based on the foregoing studies, a reference model for the directional control of the articulated vehicle has been proposed. The proposed reference model includes the desired value of the primary motions associated with the directional behaviour of the articulated vehicle. The desired values may be selected in such a way to provide an improvement of the directional stability and the manoeuvrability of the articulated vehicles.

The rest of this paper is organized as follows: a nonlinear and linear model of the articulated heavy vehicle is developed in section 2. In section 3, a reference model for the directional control of the articulated vehicle is presented. A linear regulator controller is designed in section 4 to make the lateral state variables of the articulated vehicle track the desired values generated by the reference model. Computer simulations are carried out in section 5 to investigate the advantages of the proposed approach to enhance the handling behaviour of the articulated vehicles. Finally, conclusions are given in section 6.

Dynamic Modelling:

In this section, two models including nonlinear planar model which will be used for the evaluation of the control system and the linear planar model for designing the control system and the reference model have been developed.

Nonlinear Vehicle Model:

The fourteen degrees of freedom model reflecting the directional characteristics of the articulated vehicle is used to test the controller. The articulated vehicle was modeled using two rigid bodies including the tractor and semi-trailer units with steering on all axles. As shown in Figure 1, the degrees of freedom are longitudinal, lateral and yawing motion of tractor, the articulation angle between the tractor and the semitrailer and the rotational motion of each wheel.

Three coordinate systems for the model are considered. The first is the inertial coordinate system (X_nY_n) fixed to the ground, the second is tractor coordinate system (X_tY_t) fixed to the tractor's center of gravity and the third is semitrailer coordinate system (X_sY_s) mounted to the semitrailer CG. The X_tY_t and X_sY_s coordinates moving with the tractor and the semitrailer units have yaw velocities of r_t and r_s, respectively. So the rate of the articulation angle can be represented by:

$$\dot{\gamma} = r_s - r_t \tag{1}$$

The motion equations representing longitudinal, lateral and yaw dynamics of the tractor and semitrailer units can be simply derived. Eliminating the coupling forces from these equations leads to the following four relations represent the planar motions of the articulated vehicle in X_tY_t coordinate.

$$(m_t + m_s)(\dot{V}_{xt} - r_tV_{yt}) + m_s(L_{wt}r_t^2 + L_{fs}r_s{}^2\cos\gamma + L_{fs}\dot{r}_s\sin\gamma) = F_{xtt} + F_{xts} \tag{2}$$

$$(m_t + m_s)(\dot{V}_{yt} + r_tV_{xt}) + m_s(L_{fs}r_s{}^2\sin\gamma - L_{wt}\dot{r}_t - L_{fs}\dot{r}_s\cos\gamma) = F_{ytt} + F_{yts} \tag{3}$$

$$I_{zt}\dot{r}_t + m_t(\dot{V}_{yt} + r_tV_{xt})L_{Wt} = M_{zt} + F_{ytt}L_{wt} \tag{4}$$

$$I_{zs}\dot{r}_s - m_t(\dot{V}_{xt} - r_tV_{yt})L_{fs}\sin\gamma + m_t(\dot{V}_{yt} + r_tV_{xt})L_{fs}\cos\gamma = M_{zs} - F_{xtt}\sin\gamma L_{fs} + F_{ytt}\cos\gamma L_{fs} \tag{5}$$

Where F_{xtt} , F_{ytt} and M_{zt} are the total longitudinal force, lateral force and yaw moment acting on the tractor unit in tractor coordinated system (X_tY_t); F_{xts}, F_{yts} and M_{zs} are the total longitudinal force, lateral force and yaw moment acting on the semitrailer unit in tractor coordinated system (X_tY_t). The wheel rotational dynamics can be expressed as:

$$I_w \dot{\omega}_i = -R_w F_{ti} + T_i \tag{6}$$

Where T_i is the applied torque to the wheel, F_{ti} is the tractive force, ω_i is the angular velocity of each wheel. The tractive and side forces acting on the articulated vehicle are generated at the contact path between tyre and road. In this paper, the Dugoff *et al*., 1970 model is used to simulate these forces. The inputs of the model are the side slip angle, wheel slip ratio and normal tyre load. Finally, a SIMULINK program has been developed to simulate the motion of the articulated vehicle.

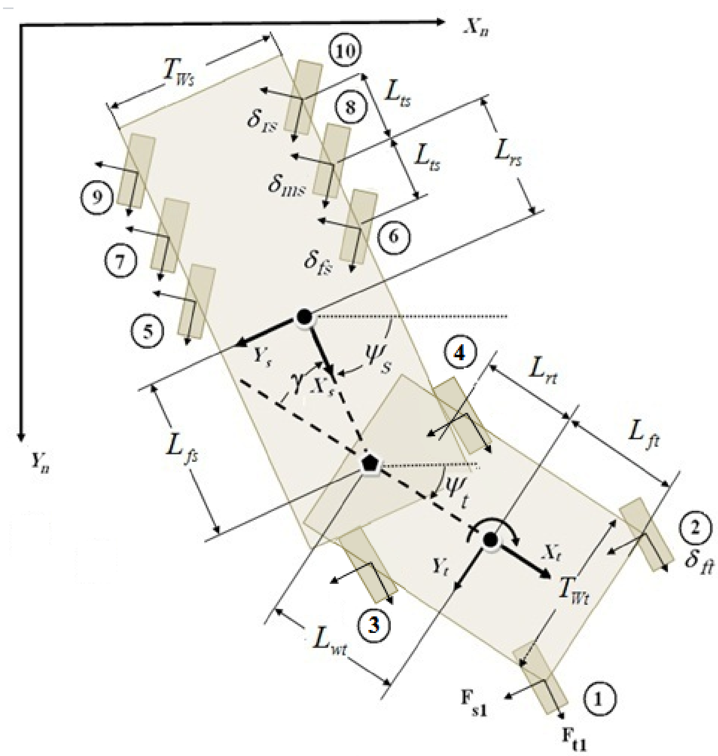

Fig. 1: Fourteen degrees of freedom articulated vehicle model.

Linear vehicle model

The linear model which is used to design the control system and generate the desired states, is a single track 4-th order model shown in Figure 2. As schematically shown, it is assumed that one tyre is located at the center of the tractor front axle, tractor rear axle and semitrailer rear axles. Furthermore, the steering angle of each wheel, the articulation angle, and the tyre slip angles are assumed to be small. In addition, each tyre lateral force is modeled as a linear function of the tyre slip angle. The above assumptions lead to the linear motion equations of the articulated vehicle can be represented as:

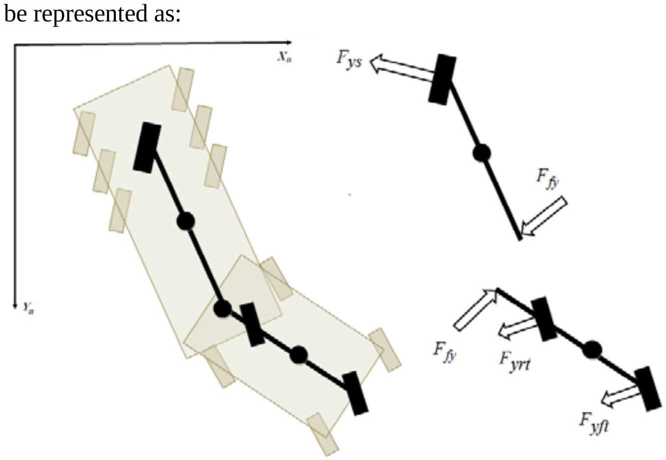

Fig. 2: Linear model of the articulated vehicle.

$$\underbrace{\begin{bmatrix} m_s+m_t & -m_s(L_{wt}+L_{fs}) & -m_s L_{fs} & 0 \\ m_t L_{wt} & I_{zt} & 0 & 0 \\ -m_s L_{fs} & I_{zs}+m_s L_{fs}(L_{fs}+L_{wt}) & m_s L_{fs}^2+I_{zs} & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}}_{M} \begin{Bmatrix} \dot{V}_{yt} \\ \dot{r}_t \\ \ddot{\gamma} \\ \dot{\gamma} \end{Bmatrix} = \underbrace{\begin{bmatrix} a_{11} & a_{12} & a_{13} & a_{14} \\ a_{21} & a_{22} & 0 & 0 \\ a_{31} & a_{32} & a_{33} & a_{34} \\ 0 & 0 & 1 & 0 \end{bmatrix}}_{A_1} \begin{Bmatrix} V_{yt} \\ r_t \\ \dot{\gamma} \\ \gamma \end{Bmatrix} +$$

$$\underbrace{\begin{bmatrix} C_{ft} & C_{rt} & C_s \\ C_{ft}(L_{ft}+L_{wt}) & C_{rt}(L_{wt}-L_{rt}) & 0 \\ 0 & 0 & C_s(L_{fs}+L_{rs}) \\ 0 & 0 & 0 \end{bmatrix}}_{C} \underbrace{\begin{Bmatrix} \delta_{ft} \\ \delta_{rt} \\ \delta_s \end{Bmatrix}}_{u} \tag{7}$$

Where

$$a_{11} = -\frac{C_{ft}+C_{rt}+C_s}{V_{xt}}$$
$$a_{12} = \frac{C_{rt}L_{rt}-C_{ft}L_{ft}+C_s d_5-(m_t+m_s)V_x^2}{V_x}$$
$$a_{13} = \frac{C_s d_6}{V_{xt}}$$
$$a_{14} = C_s$$
$$a_{21} = -\frac{C_{ft}(L_{ft}+L_{wt})+C_{rt}(L_{wt}-L_{rt})}{V_{xt}}$$
$$a_{22} = \frac{-C_{ft}(L_{ft}^2+L_{ft}L_{wt})+C_{rt}(L_{wt}-L_{rt})L_{rt}-m_t L_{wt}V_{xt}^2}{V_x}$$
$$a_{31} = \frac{C_s L_5}{V_{xt}}$$
$$d_5 = L_{wt}+L_{fs}+L_{rs}$$
$$d_6 = L_{fs}+L_{rs} \tag{8}$$

Equation (7) can be expressed in a matrix form as follows:

$$A_1 x + Cu = M\dot{x} \tag{9}$$

Where the state vector, x is defined as:

$$x = [V_{yt}, r_t, \dot{\gamma}, \gamma]^T \tag{10}$$

The equation (9) can be represented in the state space form by premultiplying on both side by M^{-1}:

$$\dot{x} = Ax + Bu \tag{11}$$

Where

$$A = M^{-1}A_1, \quad B = M^{-1}C \tag{12}$$

Reference Model:

The main objectives of the control system designed in this paper are the maneuverability and stability improvement of the articulated vehicle. As inferred from the linear model developed in section 2, the primary motions associated with the directional behavior of the articulated vehicle are three state variables including yaw rate (r_t) and lateral velocity (V_{yt}) of the tractor unit and the articulation angle (γ) between tractor and semitrailer units. In the following paragraphs, a reference model producing the desired state responses is presented.

In passenger vehicles, the yaw rate is made to follow the steady state yaw rate which is deduced based on a linear 2-DOF vehicle model in order to provide a familiar situation for drivers (Ghaffari *et al*., 2011). Similarly, in this paper, the steady state yaw rate response of the linear model developed in section 2 is defined as the reference signal for the yaw rate of the tractor unit.

The lateral velocity reflects more the stability performance of the vehicle. In passenger vehicles, a zero value has been selected as the desired value in some references (Nagai *et al*., 2002; Boada *et al*., 2005). However, the steady state side slip angle may be defined as the desired response (Rajamani, 2006). In this paper, zero lateral velocity is considered as the reference signal for the controller.

Finally, the articulation angle between the towing and trailer units should be considered as the new control variable for the articulated vehicles. This state may affect the lateral stability and the maneuverability of the trailer unit. Jack-knifing and snaking are treated as the two types of yaw instability of the trailer unit (Hac *et al*., 2008). In jack-knifing, the articulation angle exceeds a critical threshold without undergoing oscillations, while in the snaking mode, the articulation angle exhibits an oscillatory response with an increasing amplitude. On the other hand, the maneuverability performances of the trailer unit can be assessed through the off-tracking evaluation. It may be possible to eliminate the off-tracking through adjusting the articulation angle.

The steady state articulation angle of the linear model of the single tracked articulated vehicle may be selected as the reference value for this state (Changfu *et al*., 2012). Using equation (11), the steady state value of the yaw rate (r_{sd}), lateral velocity (V_{sd}), articulation angle rate $(\dot{\gamma}_{sd})$ and the articulation angle (γ_{sd}) can be simply defined as:

$$\begin{Bmatrix} V_{sd} \\ r_{sd} \\ \dot{\gamma}_{sd} \\ \gamma_{sd} \end{Bmatrix} = -A^{-1} B \delta_f \tag{13}$$

Where δ_f is the steering angle applied by the driver. Although the tracking of the steady state value of the articulation angle may be effective for the prevention of jack-knifing, the off-tracking mitigation through tracking the value is questioned. Therefore, a new desired articulation angle through kinematic and geometric analysis is defined. The semi-trailer rear end is made to follow the fifth wheel path when the proposed desired articulation is tracked. Figure 3, schematically shows the planar motion of the vehicle. The main goal is to eliminate the position deviation between the point f at the present time (t) and the point S at the time of $(t + t_d)$. The delay time, t_d, is the time required for the articulated vehicle to travel the distance between the rear end of the trailer (point S) and the fifth wheel (point f) which can be approximated as : $t_d = L_s / V_{xt}$. Referring to Figure 3, the longitudinal and lateral velocity of the fifth wheel can be expressed in the global coordinate as follows:

$$V_{xf}(t) = V_{xt}(t)\cos\Psi_t(t) + r_t(t)L_{wt}\sin\Psi_t(t) - V_{yt}(t)\sin\Psi_t(t) \tag{14}$$

$$V_{yf}(t) = V_{xt}(t)\sin\Psi_t(t) - r_t(t)L_{wt}\cos\Psi_t(t) + V_{yt}(t)\cos\Psi_t(t) \tag{15}$$

By dividing the delay time , t_d, into equal j parts calculated as $j = t_d / \Delta T$ and using geometric and kinematic analysis, the desired articulation angle can be defined as

$$\gamma_d = -\Psi_t(t) + \arctan\left(\frac{\sum_{i=-j}^{-1} \left[V_{yf}(t + i\Delta t)\Delta t \right]}{\sum_{i=-j}^{-1} \left[V_{xf}(t + i\Delta t)\Delta t \right]} \right) \tag{16}$$

Where $\Psi_t(t)$ is the yaw angle of the tractor unit and ΔT is the time increment which is sufficiently small.

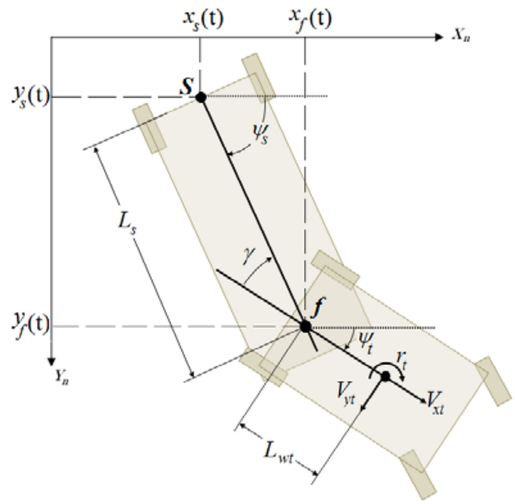

Fig. 3: Schematic of the planar motion of the articulated vehicle.

Control System Design:

As stated above, three output variables including tractor yaw rate, tractor lateral velocity and articulation angle should be controlled for the reference responses. Obviously, three control inputs are required to achieve an acceptable responsiveness. In this paper, the steering angle of the tractor front axle (δ_{ft}), tractor rear axle (δ_{rt}) and trailer axles (δ_s) are used to make the state variables follow the desired responses presented in section 3. The control structure depicted in Figure 4 consists of two compensators including the feed-forward controller using the reference states and feedback controller using the error signal.

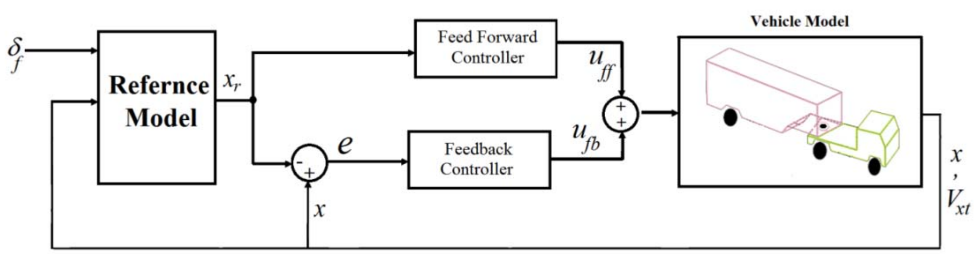

Fig. 4: Block dyagram of the control system.

In order to design the feed-forward and feedback controllers, the problem can be formulated as follows. Tracking error between the actual and reference states (x_r) is defined as follows:

$$e = \begin{Bmatrix} V_{yt} \\ r_t \\ \dot{\gamma} \\ \gamma \end{Bmatrix} - \begin{Bmatrix} V_r \\ r_r \\ \dot{\gamma}_r \\ \gamma_r \end{Bmatrix} = x - x_r \tag{17}$$

By differentiating the above equation and Substituting the terms at the right side of equations (11) into the derivative expression, the following equation can be obtained:

$$\begin{aligned} \dot{e} &= \dot{x} - \dot{x}_r = Ax - Ax_r + Bu - \dot{x}_r + Ax_r \\ &= A(x - x_r) + Bu - \dot{x}_r + Ax_r \\ &= Ae + Bu - \dot{x}_r + Ax_r \end{aligned} \tag{18}$$

Here, let the control input be defined as follows:

$$u = u_{ff} + u_{fb} \tag{19}$$

Where u_{ff} is the feed forward control input and u_{fb} is the feedback control input. Substituting equation (19) in equation (18), the following relation can be obtained:

$$\dot{e} = Ae + Bu_{fb} + Bu_{ff} - \dot{x}_d + Ax_d \tag{20}$$

Consequently, the feed forward control input can be defined as follows:

$$u_{ff} = B^{-1}(\dot{x}_d - Ax_d) \tag{21}$$

It is important to note that the Matrix B is not square and pseudo inverse of the matrix is used to calculate the feed forward control low. Now, combining equations (20) and (21), the tracking error equation can be represented in typical form as follows:

$$\dot{e} = Ae + Bu_{fb} \tag{22}$$

The feedback control input u_{fb} is determined by using the optimal control theory. The optimal feedback law can be obtained through minimizing the following performance index:

$$J = \int_0^{\infty} (e^T Qe + u_{fb}^T Ru_{fb})dt \tag{23}$$

Where Q and R are the weighting matrices of state variables and control inputs, respectively. An acceptable handling behavior which satisfies the limitations of the control actuators can be achieved by the proper selections of the weighting matrices. Finally, the control input (u) can be obtained using equation (19) where $u = [\delta_{ft} \quad \delta_{rt} \quad \delta_s]^T$. In this paper, the steer angle of the trailer's middle axle (δ_{ms}) sets to be equal to control output of δ_s. To adjust the steer angles of the other two axles, a method proposed by Cheng *et al*., 2011 is used. A key point of this method is to equalize the lateral forces of the semi-trailer's wheels. Consequently, all tyre slip angles should be equivalent. Therefore, the steering angles of the other two axles are:

$$\delta_{fs} = \arctan\left(\tan(\delta_s) + \frac{L_{ts}}{V_{xt}} r_t\right) \tag{24}$$

$$\delta_{rs} = \arctan\left(\tan(\delta_s) - \frac{L_{ts}}{V_{xt}} r_t\right) \tag{25}$$

Simulation Results:

In order to investigate the effectiveness of the proposed reference model, computer simulations are carried out on the nonlinear vehicle model presented in section 2. The vehicle parameters are listed in Table 1.

High Speed Lane Change Maneuverer:

In this subsection, a driver model is considered for performing the high speed lane change manoeuvre. It seems that the preview tracking model of the driver (Mokhiamar and Abe, 2004; Renski, 2001) is more similar to what happens in reality. Therefore, this model is used to perform the lane change maneuver. A single lane change maneuver with 3.5m lateral displacement is nominated as the driver's desired path. The vehicle runs with the speed of 75 kilometer per hour on a slippery road ($\mu = 0.3$). It is necessary to make a comparison to better understand the effect of the different methods on the system performance. Hence, the responses of the following controllers are examined in this maneuver.

Table 1: Vehicle parameters.

Parameters	Symbol	Value
Cornering stiffness of tractor front tyre	C_{ft}	160 KN/unit slip
Cornering stiffness of tractor rear tyre	C_{rt}	160 KN/unit slip
Cornering stiffness of trailer rear tyre	C_s	90 KN/unit slip
Tractor inertia moment about the yaw axis	I_{zt}	20679 kg m²
Semitrailer inertia moment about the yaw axis	I_{zs}	238898 kg m²
Distance between the trailer centre of gravity and coupling point	L_{fs}	5.65 m
Distance between the tractor centre of gravity and tractor front axle	L_{ft}	1.11 m
Distance between the tractor centre of gravity and tractor rear axle	L_{rt}	2.58 m
Distance between the trailer centre of gravity and trailer middle axle	L_{rs}	2.047 m
Distance between the fifth wheel and the rear end of the semitrailer	L_S	10 m
Distance between adjacent trailer axles	L_{ts}	1.31 m
Distance between the tractor centre of gravity and coupling point	L_{wt}	1.96 m
Total mass of the tractor unit	m_t	6525 kg
Total mass of the trailer unit	m_s	33221 kg
Wheel radius	R	0.5 m
Track width of the tractor unit	T_{wt}	2.04 m
Track width of the trailer unit	T_{ws}	2 m

Two different manoeuvres are considered here. The first one is a high speed lane change manoeuvre which is conducted on a slippery road and the second one is a low speed turning manoeuvre.

LQR Controller, Reference Model 1(LQR1):

The proposed LQR controller in section 4 is used to adjust the articulated vehicle handling behaviour. The steady state yaw rate, zero value and proposed value in equation (16) are selected as the desired response of the tractor yaw rate, the tractor lateral velocity and the articulation angle, respectively $(r_r = r_{sd},\ V_r = 0,\ \gamma_r = \gamma_d)$.

LQR Controller, Reference Model 2(LQR2):

For this method, the steady state articulation angle is selected as the desired value. This is the only difference between the two LQR control methods $(r_r = r_{sd},\ V_r = 0,\ \gamma_r = \gamma_{sd})$. However, the preliminary results indicated that the steady state articulation angle cannot provide a proper response. The main reason is that the length of the semi-trailer is not taken into consideration. Therefore, the desired articulation angle and driver steering angle have the same phase in terms of time. This could cause the control steering angle to be harsh and consequently it may deteriorate the performance of the system. To cope with this problem, the steady state articulation angle (γ_{sd}) derived in equation (13) is modified as follows:

$$\gamma'_{sd} = \frac{1}{1+T_\gamma \Delta}\gamma_{sd} \tag{26}$$

Where T_γ is a time lag and Δ is Laplace operator. The T_γ is a design parameter which depends on the dimensions and the vehicle speed. For this manoeuvre, the time lag is set at $T_\gamma = 0.55$.

PD Controller:

The controller only adjusts the articulation angle through the control of the steering angles of the trailer wheels. The proposed value in equation (16) has been selected as the reference articulation angle $(\gamma_r = \gamma_d)$.

Figure (5) shows the time history of the yaw rate response for all the four cases. It is clear that both LQR1 and LQR2 methods provide a satisfactory level of tracking performance according to the desired values. In spite of the fact that there is no control on yaw rate for the uncontrolled vehicle and vehicle equipped with PD controller, the yaw rate remains in an acceptable range in both cases. The lateral velocity responses are also shown in Figure (6). It can be seen that the lateral velocity is limited in a narrow band by both LQR controllers.

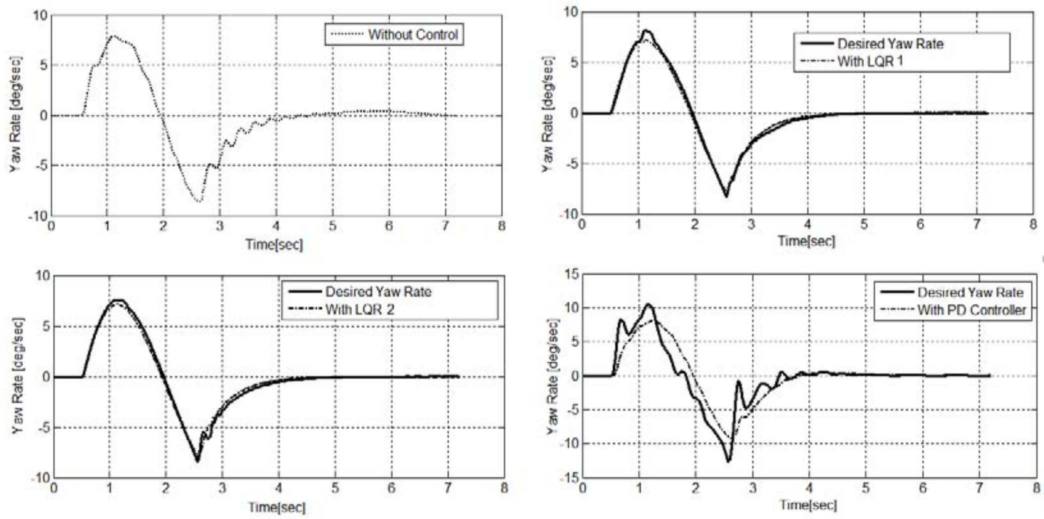

Fig. 5: Tractor yaw rate responses in lane change maneuver.

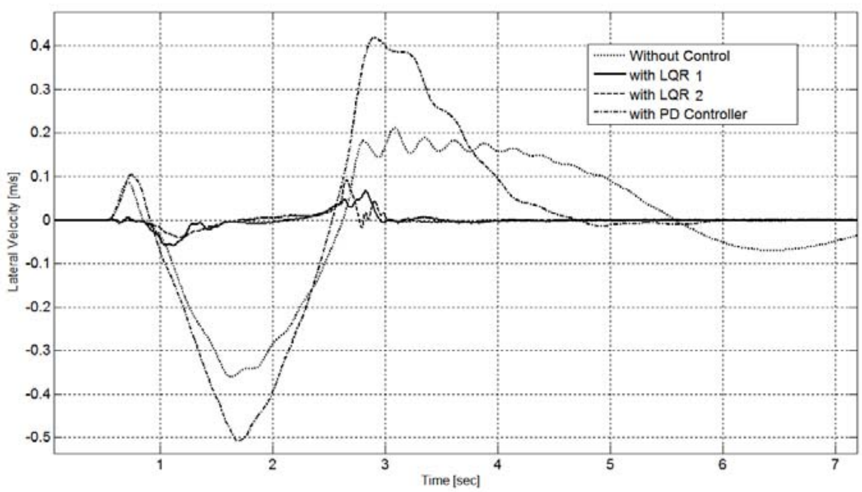

Fig. 6: Tractor lateral velocity responses in lane change maneuver.

The articulation angles in the cases without control and with the presented methods are shown in Figure 7. The desired articulation angles are shown by the solid line. It should be noted that the desired articulation angle defined in equation (16) is considered as the reference value for both LQR1 and PD controllers. On the other hand, the steady state articulation angle is defined as the reference signal for LQR2. As it can be seen, the controllers demonstrate the acceptable tracking performance. The effect of the following of the different reference articulation angles on off-tracking mitigation can be evaluated in Figure 8. The desired path of the lane change maneuver is shown with the solid green line in Figure 8. It is clear that the driver can successfully guide the tractor unit in the desired path for all the four cases. However, this task is performed for both LQR1 and LQR2 controllers without overshoot for the fifth wheel path. The direct control of the tractor unit handling behavior is the main reason behind what is found above. On the other hand, it is easy to find that the proposed reference articulation angle is very effective at reducing the off-tracking compared to the passive vehicle and the vehicle equipped with LQR2 controller which uses the modified steady state articulation angle as the reference signal.

The effect of the direct control of the tractor handling behavior on driver's activities is reflected in Figure 9. It is clearly observed that the driver applies a smaller and more moderated steering angle when LQR1 and LQR2 controllers are used.

The steer angles for the three mentioned control methods are shown in Figure 10. However, it should be noted that for the PD controller, there is no control on the steering of the tractor front and rear axles.

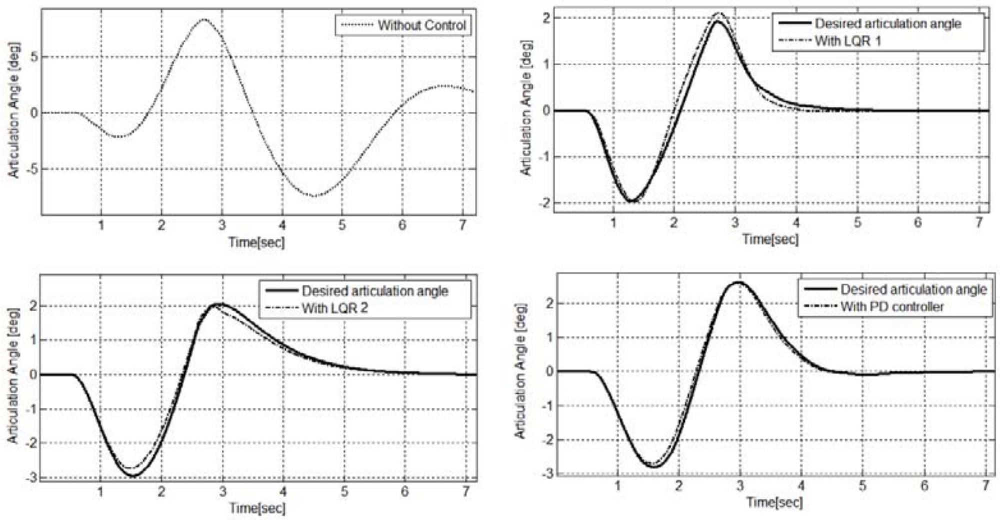

Fig. 7: Articulation angle responses in lane change maneuver.

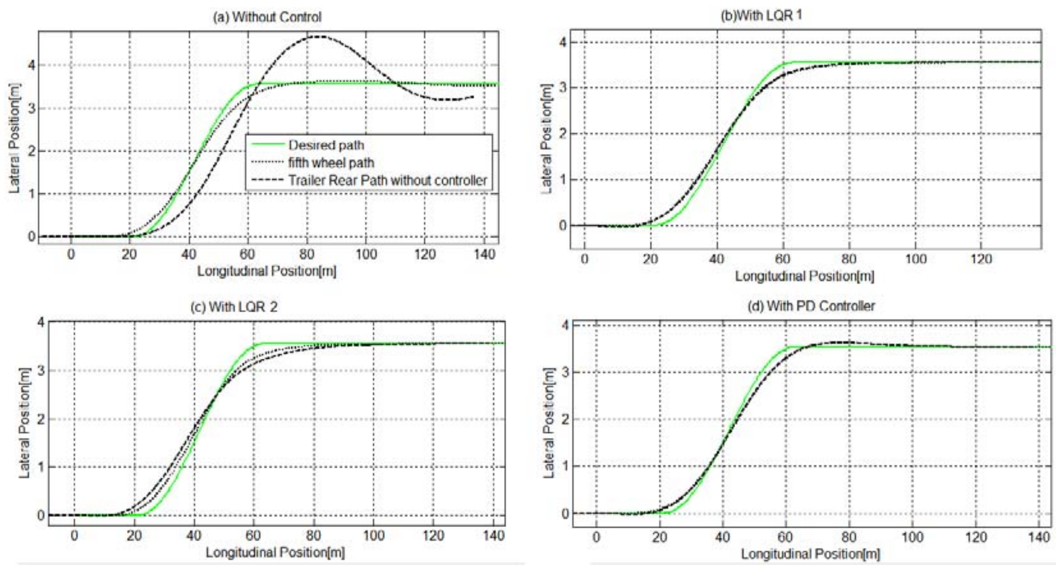

Fig. 8: The vehicle path in the four cases.

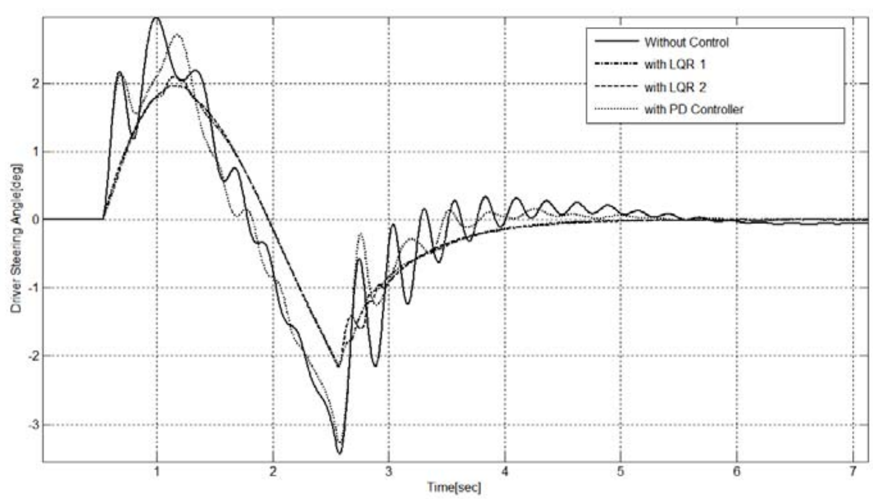

Fig. 9: The driver steering angles in four cases.

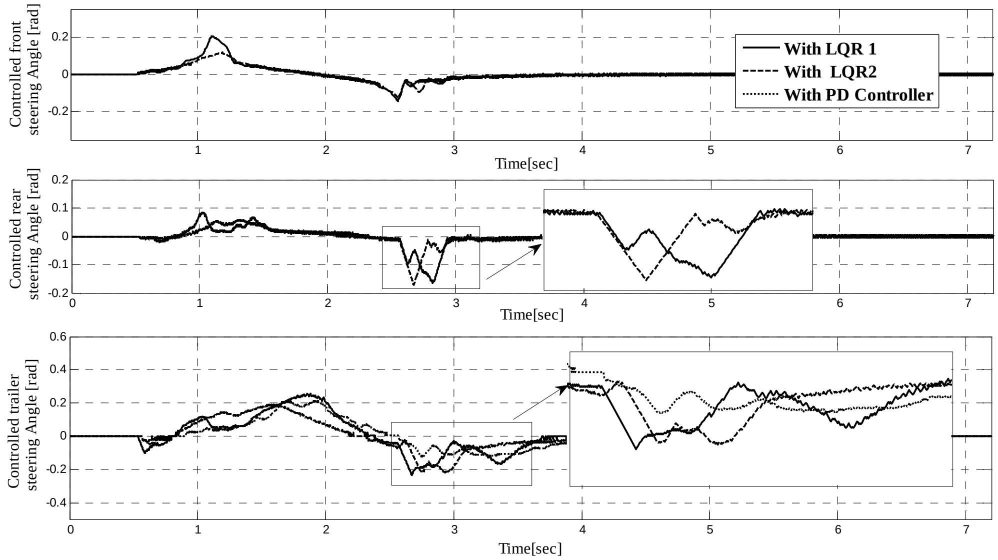

Fig. 10: The controlled steering angles of the articulated vehicle axles.

5.2 Low Speed Turning Maneuver:

At low speeds, the vehicle handling behaviour can be accurately predicted by a linear model. In other words, the directional state variables of the articulated vehicle including yaw rate, lateral velocity and the articulation angle follow the linear model response. Therefore, at low speeds, the direct control of the yaw rate and lateral velocity of the tractor unit seems unnecessary. However, the case will be different if the articulation angle is the control object. At low speeds, some examination should be done to see that the tracking of the proposed reference articulation angle (Eq. 16) is necessary or not. Therefore, in this subsection, the vehicle response is studied in the cases without control and with previous PD controller which made the articulation angle follow the proposed reference value. In this maneuver, the vehicle runs at the initial speed of 15Km/h on a dry road with the steering input shown in Figure 11(a). From Figures 11(b) and 11(c), it is obvious that the proposed desired articulation angle tracking has the significant effect on off-tracking elimination.

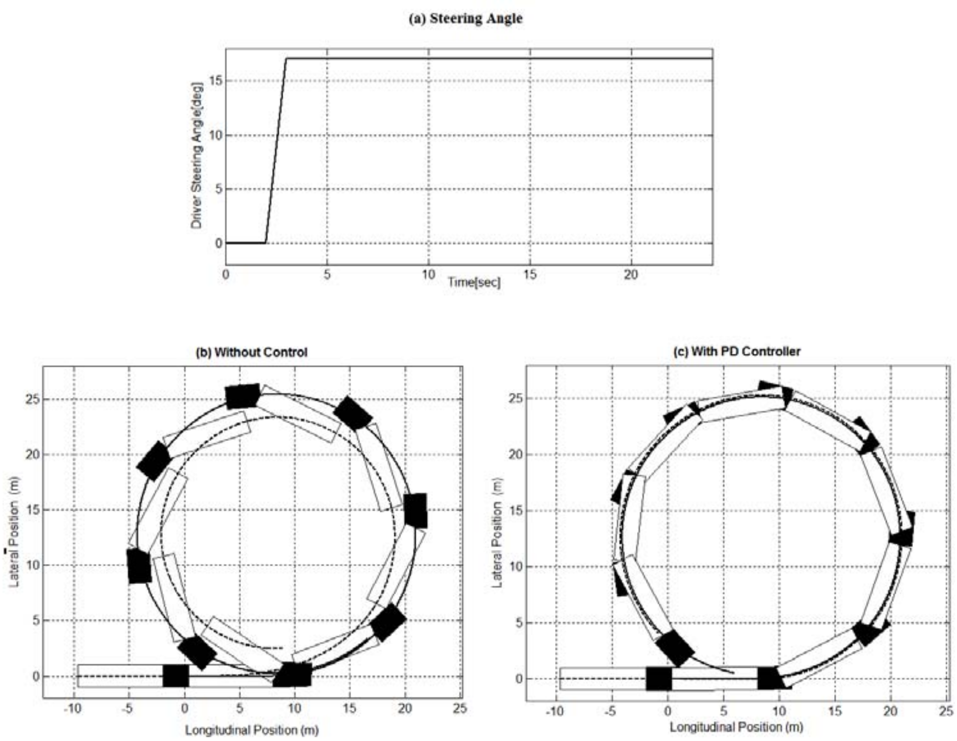

Fig. 11: (a) Steering angle in low speed turning manoeuvre, (b) Paths of the vehicle without control, (C) Paths of the vehicle with proposed method

The articulation angles in the cases without control, with the PD controller, the proposed desired value and steady state values are shown in Figure 12. It is clear that the PD controller is able to exactly track the proposed reference value. On the other hand, the steady state articulation angle is almost similar to that obtained from the non-controlled vehicle. Therefore, it can be concluded that the steady state articulation angle can not be selected as the reference value at low speeds.

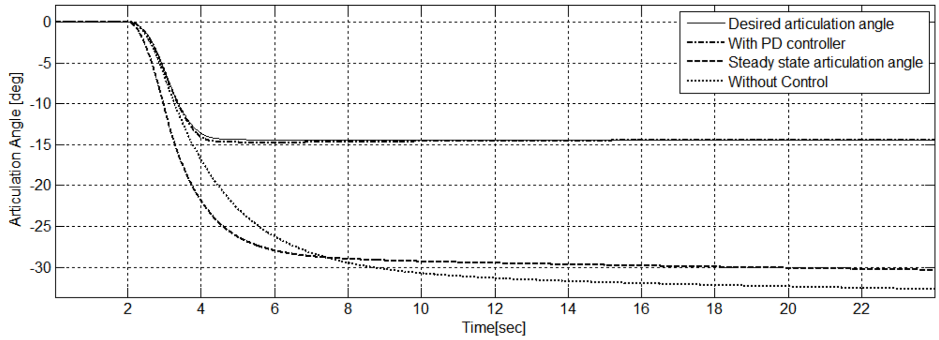

Fig. 12: Articulation angle response in low speed turning manoeuvre.

The steer angle of the trailer axle for the PD controller is shown in Figure 13. In Figure 14, the tractor yaw rate and lateral velocity are compared against the steady state values. As expected, the mentioned two states are exactly similar to the steady state values derived from Equation (13).

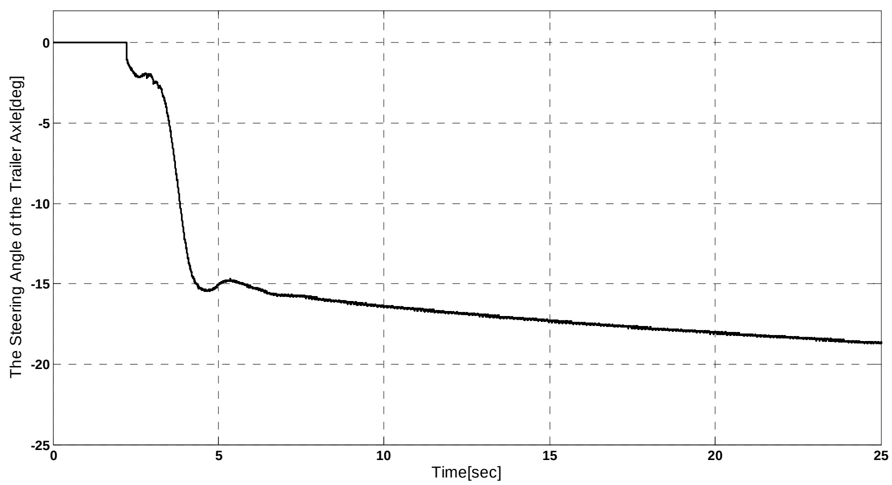

Fig. 13: The controlled steering angle of the trailer middle axle.

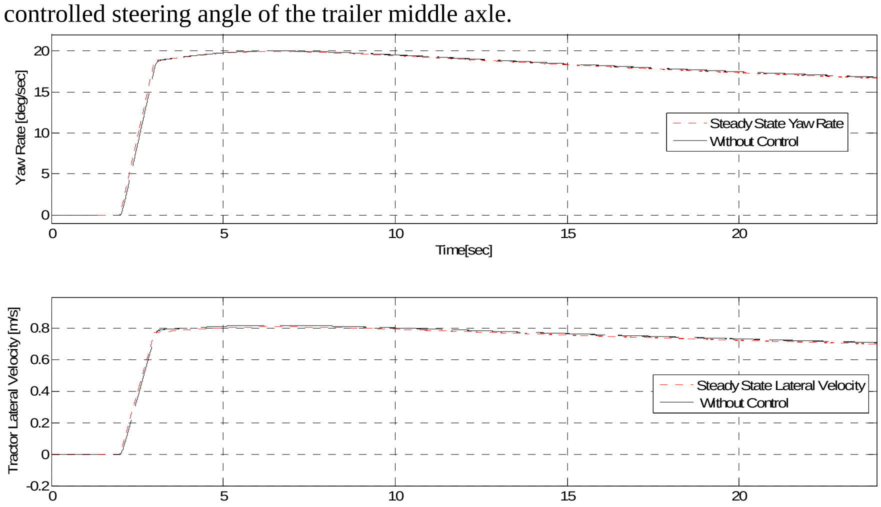

Fig. 14: Yaw rate and lateral velocity responses in low speed turning manoeuvre.

Conclusion:

1. The linear and nonlinear models of the articulated heavy vehicle have been developed. The linear model has been used to design the controller and generating the reference response while the nonlinear model has been used to evaluate the system responses.
2. The reference model for generating the desired response of the tractor yaw rate, tractor lateral velocity and the articulation angle has been proposed.
3. The simulation results confirm that the control of the lateral velocity and the yaw rate should be performed at high speed. The control of later states, not only improve the vehicle behavior but also reduce the driver's activities. However, the control of both mentioned states at low speed seems unnecessary.
4. The simulation results showed that the articulation angle should be controlled in all speeds. Furthermore, in different driving situations, the proposed reference articulation angle in equation (16) exhibits superior performance in terms of reducing off-tracking compared with the steady state articulation angle.

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APPENDIX 1:

Notation

C_{ft} cornering stiffness of tractor front tyres

C_{rt} cornering stiffness of tractor rear tyres

C_s cornering stiffness of trailer tyres

F_{xts}	total longitudinal force acting on semi-trailer unit
F_{yts}	total longitudinal force acting on semi-trailer unit
F_{xtt}	total longitudinal force acting on tractor unit
F_{ytt}	total longitudinal force acting on tractor unit
I_{zt}	tractor inertia moment about the yaw axis
I_{zs}	semi-trailer inertia moment about the yaw axis
L_{fs}	distance between the trailer centre of gravity and coupling point
L_{ft}	distance between the tractor centre of gravity and tractor front axle
L_{rt}	distance between the tractor centre of gravity and tractor rear axle
L_{rs}	distance between the trailer centre of gravity and trailer middle axle
L_S	distance between the fifth wheel and the rear end of the semi-trailer
L_{ts}	distance between adjacent trailer axles
L_{wt}	distance between the tractor centre of gravity and coupling point
M_{zs}	total yaw moment acting on semi-trailer unit
M_{zt}	total yaw moment acting on tractor unit
m_t	total mass of the tractor unit
m_s	total mass of the trailer unit
r_t	tractor yaw velocity
r_s	semi-trailer yaw velocity
V_{xt}	longitudinal velocity of the tractor CG
V_{yt}	lateral velocity of the tractor CG
γ	articulation angle
ω	wheel rotational speed
Δ	laplace operator