

Directional Control of Articulated Heavy Vehicles

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ABSTRACT

In this paper, a method for directional control of articulated heavy vehicles is proposed. The tractor yaw rate, the tractor lateral velocity and the articulation angle are selected as the control variables. The desired values of these states are defined in such way to improve the maneuverability and the stability of the articulated vehicle. A linear quadratic regulator controller is designed based on the linear model of the articulated vehicle to make the control variables follow the desired responses. Furthermore, a nonlinear 14 Degrees of freedom (DoF) model is developed to evaluate the proposed control method. The significant effect of the proposed method on improving the directional behavior of the articulated vehicle is proved through the simulations of the high speed lane change maneuver on a slippery road.

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INTRODUCTION

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. 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.

In articulated heavy vehicle, 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. 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 that the rear end of the trailer unit does not closely follow the path of the tractor unit. Chen and Shie [1], Fancher et al. [2], Odhams et al. [3] and Tabatabaei et al. [4] mainly concentrated on the control of the articulation angle to enhance the manoeuvrability through off-tracking mitigation.

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 [5]. Therefore, several research efforts have been made on the simultaneous active

control of both tractor and trailer units. In 1996, Palkovics and El gindy [5] 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., in 2001 [6] and Hac et al, in 2008 [7]. However, it seems that the idea of minimization reflects more the stability improvement than the maneuverability enhancement.

Instead of using the idea of minimization, there is a vital need for devising a control system which can make the actual articulated vehicle behaviour track a desired behaviour. Therefore, in this paper, a reference model for directional control of articulated vehicle has been proposed. The reference model may be defined 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 behavior of the articulated vehicles. Finally, conclusions are given in section 6.

DYNAMIC MODELLING

In this section, two models including a nonlinear planar model which will be used for the evaluation of the control system and a 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 Fig. 1, the degrees of freedom are longitudinal, lateral and yawing motion of tractor, the articulation angle between the tractor and the semi-trailer and the rotational motion of each wheel. Three coordinate systems for the model are considered. The first is the inertial coordinate system ($X_n Y_n$) fixed to the ground, the second is tractor coordinate system ($X_t Y_t$) fixed to the tractor's Center of Gravity (CoG) and the third is semi-trailer coordinate system ($X_s Y_s$) mounted to the semi-trailer CoG. The $X_t Y_t$ and $X_s Y_s$ 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:

$$\dot{\gamma} = r_s - r_t \quad (1)$$

The motion equations representing longitudinal, lateral and yaw dynamics of the tractor and semi-trailer 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 $X_t Y_t$ coordinate.

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

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

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

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

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 ($X_t Y_t$); F_{xts} , F_{yts} and M_{zs} are the total longitudinal force, lateral force and yaw moment acting on the semi-trailer unit in tractor coordinated system ($X_t Y_t$). The wheel rotational dynamics can be expressed as:

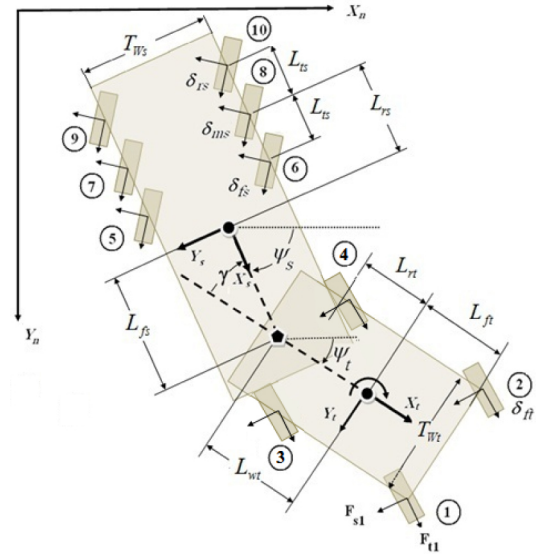


Figure 1. Fourteen degrees of freedom articulated vehicle model

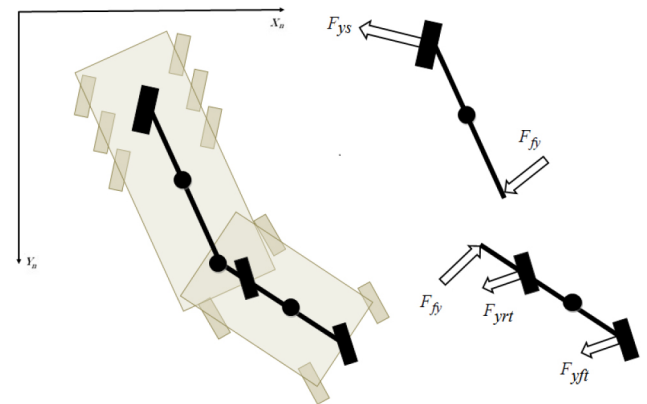


Figure 2. Linear model of articulated vehicle model

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

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 model is used to simulate these forces [8]. 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.

Linear Vehicle Model

The linear model which is used to design the control system and generate the desired states, is a single track fourth 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 semi-trailer 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 which can be represented as:

$$\underbrace{\begin{bmatrix} m_s + m_t & -m_s(L_{wt} + L_{fs}) & -m_s L_{fs} & 0 \\ m_t L_{wt} & I_{zt} & 0 & 0 \\ -m_s L_{fs} & I_{zs} + m_s L_{fs}(L_{fs} + L_{wt}) & m_s L_{fs}^2 + I_{zs} & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}}_M \underbrace{\begin{bmatrix} \dot{V}_{yt} \\ \dot{r}_t \\ \dot{\gamma} \\ \dot{\gamma} \end{bmatrix}}_u = \underbrace{\begin{bmatrix} a_{11} & a_{12} & a_{13} & a_{14} \\ a_{21} & a_{22} & 0 & 0 \\ a_{31} & a_{32} & a_{33} & a_{34} \\ 0 & 0 & 1 & 0 \end{bmatrix}}_{A_1} \underbrace{\begin{bmatrix} V_{yt} \\ r_t \\ \dot{\gamma} \\ \gamma \end{bmatrix}}_x + \underbrace{\begin{bmatrix} C_{ft} & C_{rt} & C_s \\ C_{ft}(L_{ft} + L_{wt}) & C_{rt}(L_{wt} - L_{rt}) & 0 \\ 0 & 0 & C_s(L_{fs} + L_{rs}) \\ 0 & 0 & 0 \end{bmatrix}}_C \underbrace{\begin{bmatrix} \delta_{ft} \\ \delta_{rt} \\ \delta_s \end{bmatrix}}_u \quad (7)$$

where

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

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

$$A_1 x + C u = M \dot{x} \quad (9)$$

Where the state vector, x is defined as:

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

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

$$\dot{x} = A x + B u \quad (11)$$

Where

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

REFERENCE MODEL

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 and lateral velocity of the tractor unit and the articulation angle between tractor and semi-trailer 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 [9]. 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 stability performance of the vehicle. In passenger vehicles, a zero value has been selected as the desired value in some references [10, 11]. However, the steady state side slip angle may be defined as the desired response [12]. 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 [7]. 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. 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. Fig. 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 Fig. 3, the longitudinal and lateral velocity