

Investigation of Trailer Yaw Motion Control Using Active Front Steer and Differential Brake

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ABSTRACT

This paper presents a control system development for a yaw motion control of a vehicle-trailer combination using the integrated control of active front steer (AFS) and differential brake (DB). A 21 degree of freedom (dof) vehicle-trailer combination model that represents a large SUV and a medium one-axle trailer has been developed for this study. A model reference sliding mode controller (MRSMC) has been developed to generate the desired yaw moment. Based on the understanding of advantages and limitations of AFS and DB, a new integrated control algorithm was proposed. The simulation result shows that integrated control of AFS and DB can restrain the trailer's oscillation effectively and shows less longitudinal speed drop and higher stable margin compared to the DB activated only case while maintaining the yaw stability.

1. INTRODUCTION

Stability of a vehicle-trailer combined system at high speed is strongly affected by the dynamics of a trailer. With a disturbance due to driver steer input, side wind, or uneven road condition, a trailer has a tendency of lateral swing, also known as a snaking oscillation. This oscillation is strongly related to a vehicle operation speed. When a vehicle speed is below the critical vehicle speed, the lateral swing will be damped out with time [1]. However, when an operating vehicle speed is higher than the critical speed, the lateral swing grows with time and it is difficult for a non-expert driver to stabilize this situation. Without a correct prediction of a vehicle-trailer behavior and proper reaction from a

driver, the vehicle-trailer system will ultimately become unstable [2].

During the past decades, many studies have been proposed to help stabilize the vehicle-trailer combination system during this situation. In most cases, active braking and steering systems on the vehicle or trailer have been used for this situation to generate a desired yaw moment that can decrease trailer swing and stabilize a combination system. Plöchl et al [1] proposed a pure trailer based control system in which individual braking force can be applied to the trailer wheels to stabilize the lateral motion of a trailer. A similar trailer based individual braking control system also can be found in [2, 3]. Deng et al [4, 5] designed all wheel steer systems for stabilization of a vehicle trailer in which a state-feedback control method was used to determine desired steer angles at each wheel that can track a reference model from decoupled vehicle and trailer models. Kageyama et al [6] presented a four wheel steering of a towing vehicle to stabilize the trailer lateral swing at high speed. There have been researches that use existing brake based active safety systems in the towing vehicle such as ESC to stabilize a lateral swing of a trailer by updating its control algorithms [7, 8]. Although Electronic Stability Control (ESC) systems are effective ways to stabilize the vehicle lateral dynamics, a vehicle speed drops significantly with application of brake actuation. Since the braking intervention is not driver's intention, it is considered to be intrusive to the driver and often it becomes a source of complaints [9]. In order to minimize the activation of braking and to further increase the safety margin, various control efforts have been made to integrate active front steering and active differential braking. In general, these integration control methods can be classified as two types. One is a

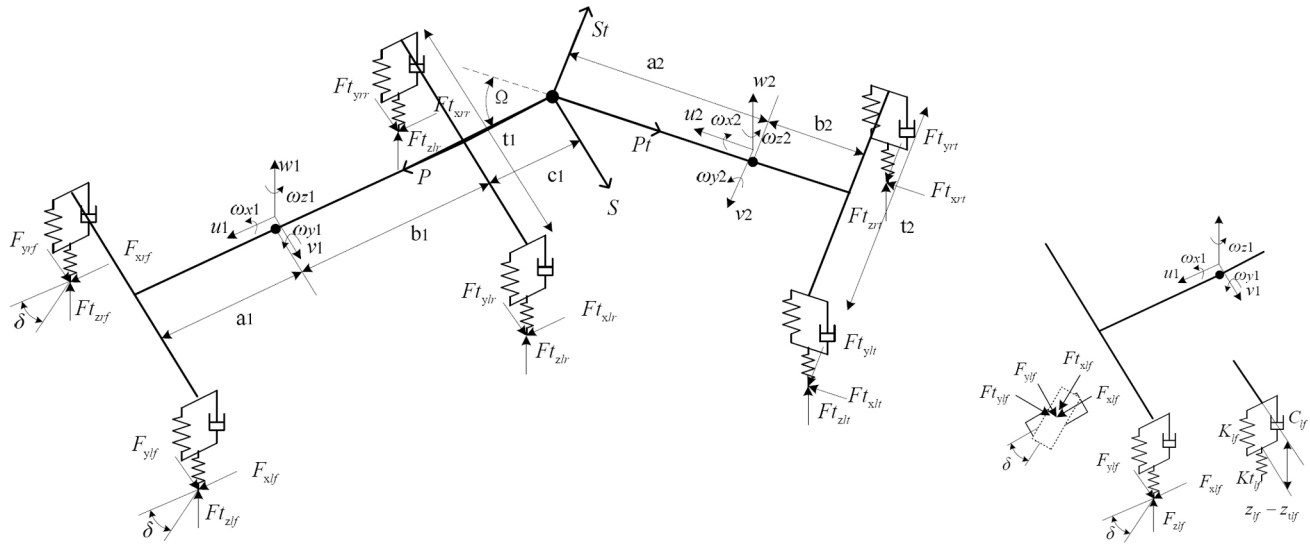


Figure 1. Diagram of vehicle-trailer dynamic model

supervisory control that allocates the desired control needs to separately designed steering and brake control systems and the other is an integrated controller that determines the controls needed between steering and braking based on MIMO control theory. A supervisory controller typically has rule sets using vehicle states, such as a stable region in terms of side-slip angle or the desired yaw angle. Based on these rules, the desired yaw moment needed to improve vehicle lateral motion is allocated between the steering and braking systems [9, 10, 11, 12]. The control system based on multi-inputs and multi-outputs has a multiple objectives. For instance, in references [9, 13, 14], control systems have been developed for a path following application of a vehicle trailer combination with two control objectives (yaw rate and side slip angle). These control systems are effective and have a larger safety margin while decreasing the longitudinal dynamic effect.

Among the published literatures, there are few reported works in the integration of active front steering and active braking only using a towing vehicle. This paper propose a new integration method that combines an active front steer and differential brake on towing vehicle to enhance the yaw motion stability of a vehicle-trailer combination system. The proposed controller was developed using a sliding mode control theory and will make the yaw rate of a vehicle with trailer to follow the reference yaw rate determined from a vehicle without a trailer by using an active front steer and differential brake. This control system consists of two closed-loops: inner-loop and outer-loop. The inner-loop controls active front steering and the outer-loop modulate differential brake forces. The inner loop will be always applied first and the outer-loop will be activated only when the yaw rate tracking error becomes larger than the predefined value. For this study, a 21-dof nonlinear vehicle-trailer model including roll, pitch, and yaw motions is developed and validated with

commercially available software (Carsim). The simulation results show that the proposed controller effectively controls an oscillation of a trailer.

The rest of the paper is organized as follows. Section 2 introduces the development of a nonlinear vehicle trailer model and a control system development is presented in section 3. In section 4, the effectiveness of the proposed controller is demonstrated through simulations.

2. DEVELOPMENT OF A NONLINEAR VEHICLE-TRAILER MODEL

Figure 1 shows a schematic of a vehicle-trailer combination model that consists of a two-axle vehicle and a one axle trailer. The vehicle-trailer is modeled as a rigid body. The coordinates $x_1y_1z_1$ and $x_2y_2z_2$ indicate the body fixed coordinates attached at vehicle center of gravity (CG) and trailer CG, respectively. u_i , v_i , and w_i , denote the vehicle forward, lateral, and vertical velocities, respectively. ϕ_i , θ_i , and ψ_i indicate the vehicle roll, pitch and yaw angles respectively. There are yaw angular velocity ω_{zi} , roll angular velocity ω_{xi} and pitch angular velocity ω_{yi} . Ω denotes a hitch angle. This model has a total 21 dof including 6 dof of a vehicle motion and 3 dof of a trailer motion. In addition, this model has 2 dof at each of the 6 wheels including vertical suspension travel and wheel spin. The longitudinal, lateral, and vertical motions at trailer CG. are constrained by the hitch point articulation.

The equations of motion for a vehicle and trailer can be derived from direct application of Newton's laws.

Equations (1), (2), (3), (4), (5), (6) show equations of motion for a vehicle:

$$M_1 \cdot (\dot{u}_1 + \omega_{y1} \cdot w_1 - \omega_{z1} \cdot v_1) = F_{xlf} + F_{xrf} + F_{xlr} + F_{xrr} + P + m_{1s} \cdot g \cdot \sin \theta_1 \quad (1)$$

$$M_1 \cdot (\dot{v}_1 + \omega_{z1} \cdot u_1 - \omega_{x1} \cdot w_1) = F_{ylf} + F_{ylr} + F_{yrf} + F_{yrr} + S - m_{1s} \cdot g \cdot \sin \phi_1 \cdot \cos \theta_1 \quad (2)$$

$$m_{1s} \cdot (\dot{w}_1 + \omega_{x1} \cdot v_1 - \omega_{y1} \cdot u_1) = F_{zlf} + F_{zrf} + F_{zlr} + F_{zrr} - N_H - m_{1s} \cdot g \cdot \cos \phi_1 \cdot \cos \theta_1 \quad (3)$$

$$J_{x1} \cdot \dot{\omega}_{x1} = (F_{zlf} + F_{zlr} - F_{zrf} - F_{zrr}) \cdot t_1 / 2 + m_{1s} \cdot a_{y1} \cdot h_{z1} + m_{1s} \cdot g \cdot h_{z1} \cdot \sin \phi_1 \cdot \cos \theta_1 - S \cdot h_{h1} - k_b \cdot \phi_1 \quad (4)$$

$$J_{y1} \cdot \dot{\omega}_{y1} = (F_{zlr} + F_{zrr}) \cdot b_{1s} - (F_{zlf} + F_{zrf}) \cdot a_{1s} + m_{1s} \cdot a_{x1} \cdot h_{p1} + m_{1s} \cdot g \cdot h_{p1} \cdot \sin \theta_1 - N_H \cdot c_{1s} + P \cdot h_{ph1} \quad (5)$$

$$J_{z1} \cdot \dot{\omega}_{z1} = (F_{ylf} + F_{yrf}) \cdot a_1 - (F_{ylr} + F_{yrr}) \cdot b_1 + (-F_{xlf} + F_{xrf} - F_{xlr} + F_{xrr}) \cdot t_1 / 2 - S \cdot c_1 \quad (6)$$

Equations (7), (8), (9), (10), (11), (12) show equations of motion for a trailer:

$$M_2 \cdot (\dot{u}_2 + \omega_{y2} \cdot w_2 - \omega_{z2} \cdot v_2) = F_{xlt} + F_{xrt} - P_t + m_{2s} \cdot g \cdot \sin \theta_2 \quad (7)$$

$$M_2 \cdot (\dot{v}_2 + \omega_{z2} \cdot u_2 - \omega_{x2} \cdot w_2) = F_{ylt} + F_{yrt} - S_t - m_{2s} \cdot g \cdot \sin \phi_2 \cdot \cos \theta_2 \quad (8)$$

$$m_{2s} \cdot (\dot{w}_2 + \omega_{x2} \cdot v_2 - \omega_{y2} \cdot u_2) = F_{zlt} + F_{zrt} + N_H - m_{2s} \cdot g \cdot \cos \phi_1 \cdot \cos \theta_1 \quad (9)$$

$$J_{x2} \cdot \dot{\omega}_{x2} = (F_{zlt} - F_{zrt}) \cdot t_2 / 2 + m_{2s} \cdot a_{y2} \cdot h_{z2} + m_{2s} \cdot g \cdot h_{z2} \cdot \sin \phi_2 \cdot \cos \theta_2 + S_t \cdot h_{h2} \quad (10)$$

$$J_{y2} \cdot \dot{\omega}_{y2} = (F_{zlt} + F_{zrt}) \cdot b_{2s} + m_{2s} \cdot a_{x2} \cdot h_{p2} + m_{2s} \cdot g \cdot h_{p2} \cdot \sin \theta_2 - N_H \cdot a_{2s} - P_t \cdot h_{ph2} \quad (11)$$

$$J_{z2} \cdot \dot{\omega}_{z2} = -S_t \cdot a_2 - (F_{ylt} + F_{yrt}) \cdot b_2 + (-F_{xlt} + F_{xrt}) \cdot t_2 / 2 \quad (12)$$

In the above equations, N_H represents the normal force on the hitch point, S and P (S_t and P_t) are lateral and longitudinal forces on the hitch point as shown in [Figure 1](#). F_{txij} and F_{tyij} are the longitudinal and lateral tire forces described in the tire coordinate. The longitudinal and lateral tire forces in the vehicle coordinate, F_{xij} and F_{yij} , can be determined from the following transformation,

$$\begin{cases} F_{xij} = F_{txij} \cdot \cos \delta_j - F_{tyij} \cdot \sin \delta_j \\ F_{yij} = F_{txij} \cdot \sin \delta_j + F_{tyij} \cdot \cos \delta_j \end{cases} \quad (13)$$

where the subscript ij denotes l (left), r (right) and f (front), r (rear) respectively. For a front wheel steervehicle case, $\delta_f = \delta$; $\delta_r = 0$.

Magic formula tire model [15] has been used to a development of tire forces F_{txij} and F_{tyij} . The longitudinal slip and lateral slip angle used in the tire model are calculated as follows:

$$\lambda_{ij} = \frac{R_{wij} \cdot \omega_{ij} - (V_{xij} \cdot \cos \delta_j + V_{yij} \cdot \sin \delta_j)}{(V_{xij} \cdot \cos \delta_j + V_{yij} \cdot \sin \delta_j)} \quad (14)$$

$$\alpha_{ij} = \frac{V_{yij}}{V_{xij}} - \delta_j \quad (15)$$

where, V_{xij} and V_{yij} are the lateral and longitudinal velocity at the tire contact patch.

This model includes suspension units consisting of springs and dampers shown in [Figure 1](#). The suspension force can be derived as:

$$F_{zij} = K_{ij} \cdot (z_{ij} - z_{tij}) + C_{ij} \cdot (\dot{z}_{ij} - \dot{z}_{tij}) \quad (16)$$

It is assumed that the equivalent suspension force is always normal to the ground.

For the wheel rotational dynamics, the following equations have been used,

$$J_{twij} \cdot \dot{\omega}_{ij} = T_{bij} - F_{txij} \cdot R_{wij} \quad (17)$$

where, J_{tw} is wheel inertia, R_w is tire radius, T_{bij} is the input torque applied to the wheel, and F_{txij} represents the longitudinal tire force.

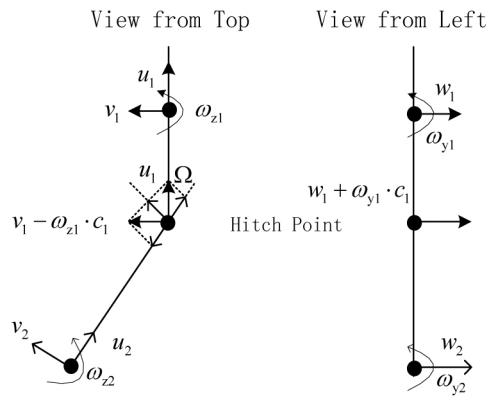


Figure 2. Articulation conditions

Figure 2 shows the velocity components at the hitch point. Since this paper is focused on the study of yaw motions of vehicle and trailer, yaw plane motion is fully considered in the articulation condition. It is assumed that the relative angle in pitch and roll plane between vehicle and trailer are small enough to be neglected. In addition, the hitch point lateral and longitudinal velocity changes caused by vehicle or trailer roll and pitch motions are also ignored. On the hitch point, the following articulation conditions have been imposed from vehicle and trailer velocities,

$$\begin{cases} u_2 = u_1 \cdot \cos \Omega - (v_1 - c_1 \cdot \omega_{z1}) \cdot \sin \Omega \\ v_2 = u_1 \cdot \sin \Omega + (v_1 - c_1 \cdot \omega_{z1}) \cdot \cos \Omega - a_2 \cdot \omega_{z2} \\ \dot{\Omega} = \omega_{z1} - \omega_{z2} \\ w_2 = w_1 + \omega_{y1} \cdot c_1 + \omega_{y2} \cdot a_2 \end{cases} \quad (18)$$

Model Validation

The developed nonlinear vehicle-trailer model was implemented in MATLAB/SIMULINK environment and has been compared and validated with a commercially available vehicle-trailer model in CARSIM. A step steering wheel angle of 100° is applied to both models with a vehicle speed of 72 km/h. Figure 3 compares the responses of yaw rate, roll angle and normal force between the developed model and CARSIM model. As shown in Figure 3, the responses of a developed model are closely matched with those of a CARSIM model. The existing discrepancies shown in Figure 3 come from the differences in the dynamic complexity between two models. The vehicle-trailer model in CARSIM has more dof compared with the 21-dof model used in this paper.

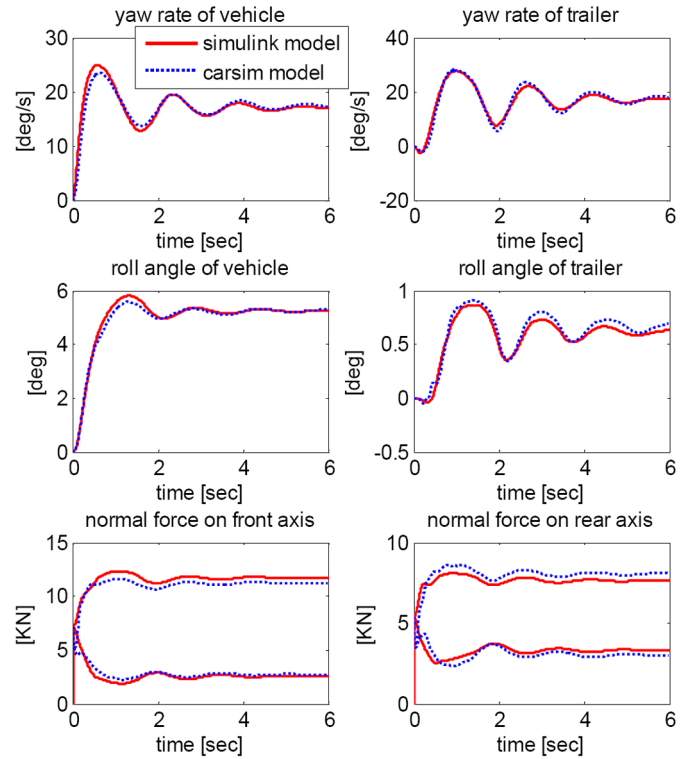


Figure 3. The responses compare between the developed model and CARSIM

DEVELOPMENT OF A CONTROL SYSTEM

Figure 4 shows a schematic diagram of the vehicle-trailer lateral stability control system based on integrated active front steer and differential brake. The proposed control system aims to decrease and to damp out the trailer lateral swing in order to preserve the stability of an overall vehicle-trailer combination system. This system will achieve the desired goal by regulating the yaw rate of the vehicle. From a given drivers' steering inputs, the desired yaw rate response was determined from a reference model. In here, a vehicle model without a trailer is used as the reference model. A nonlinear sliding mode controller is developed to calculate the desired yaw moment needed to track the desired yaw rate. Then, this desired yaw moment is distributed to two control loops. The inner loop uses an active front steer and the outer loop uses a differential brake to generate the yaw moment that can reduce the yaw rate error in the vehicle. In the proposed control system, the active front steer is always applied first in order to minimize the longitudinal vehicle speed drop while achieving desired yaw response. When the tracking error (yaw rate error) is smaller than a predetermined threshold value, only the inner loop (active front steer) will be executed. The outer loop (differential brake) will be executed only when the tracking error becomes larger than the threshold value for the case that active front steer alone cannot reduce the tracking error. Two first order delay terms