

Yaw Stability Enhancement of Articulated Commercial Vehicles via Gain-Scheduling Optimal Control Approach

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ABSTRACT

In this paper, a gain-scheduling optimal control approach is proposed to enhance yaw stability of articulated commercial vehicles through active braking of the proper wheel(s). For this purpose, an optimal feedback control is used to design a family of yaw moment controllers considering a broad range of vehicle velocities. The yaw moment controller is designed such that the instantaneous tractor yaw rate and articulation angle responses are forced to track the target values at each specific vehicle velocity. A gain scheduling mechanism is subsequently constructed via interpolations among the controllers. Furthermore, yaw moments derived from the proposed controller are realized by braking torque distribution among the appropriate wheels. The effectiveness of the proposed yaw stability control scheme is evaluated through software-in-the-loop (SIL) co-simulations involving Matlab/Simulink and TruckSim under lane change maneuvers. Simulation results demonstrate that the proposed gain scheduling optimal controllers can yield enhanced yaw stability of articulated commercial vehicles by the updating of control gains with the change of vehicle velocity.

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INTRODUCTION

Commercial vehicles continue to be the dominant mode of freight transportation via roads (about 70%) in North America [1]. Owing to their large sizes and inertia, and high mass center, heavy commercial vehicles exhibit relatively lower roll and yaw stability limits compared with other road vehicles, and thereby pose greater risks of property damages and fatalities due to their disproportionally higher inertia. What's more, the freight carried by commercial vehicle is forecast to increase by around 40% from 2012 to 2040 according to U.S Department of Transportation [2]. Therefore, there is an increasing need for further improving the stability and safety of commercial vehicles and reducing the traffic crashes caused by loss of control.

A typical articulated commercial vehicle consists of two units: tractor and semitrailer. The tractor unit is controlled directly by the driver while the semitrailer, connected to tractor unit through the fifth wheel, is used to transport goods and materials. Handling dynamics and directional control properties of articulated commercial vehicles differ significantly from those of other road vehicles such as passenger cars and light trucks. These vehicles may experience two types of instability behaviors [3]: rollover and loss of control (LOC). Rollover is known to be the most dangerous type of instability for heavy vehicles, which is not the focus of this paper. Loss of control, also referred to the loss of directional/ yaw stability, jackknifing and trailer swing, is another significant safety concern. Jackknifing accident is defined as a loss of stability in the yaw motion of the articulated commercial vehicle, which is one of the main causes of accidents resulting from loss of control.

Recently, some active control strategies have been studied to improve the lateral/yaw stability of articulated commercial vehicles, such as PID (Proportional integration derivative) control [4, 5], LQR (Linear quadratic regulator) [6, 7, 8, 9], SMC (Slide mode control) [10], LMIs (Linear matrix inequalities) [11] or MPC (Model predictive control) [12]. For example, Palkovics and El-Gindy [6] developed a LQR controller based on differential braking to improve the directional behavior of articulated commercial vehicle. Zong et al. [7] proposed an optimal yaw controller based on a single-track linear vehicle model to reduce the risks of rollover and jackknifing. Yang [10] designed a two-level control structure consisting of sliding-mode yaw moment controller and optimal braking force distributor to stabilize the yaw dynamics of the tractor-semitrailer vehicle. Of all the proposed control algorithms in literature, LQR method based on the linear vehicle model is the widely-implemented control algorithm with fixed control gains.

However, vehicle lateral/yaw dynamics varies with respect to vehicle/ tire parameters and road conditions, such as vehicle mass, center of mass, cornering stiffness, and so on. On the other hand, vehicle velocity may vary within a wide range and thereby lateral/yaw dynamics is changed accordingly. So, it is not realistic to employ a fixed-gain linear controller in real time because the characteristics of vehicle dynamics are nonlinear in nature. Gain scheduling method is a common used controller design approach for nonlinear systems. Inspired by the work [11, 13] which proposed gain-scheduling control methods for roll control of articulated vehicles, a gain-scheduling optimal nonlinear controller is proposed to enhance the yaw stability of articulated vehicles in this paper. Gain scheduling is realized using

a linear optimal approach for the simplified linear vehicle model with respect to vehicle velocity. The effectiveness of the proposed controller is verified through software-in-the-loop co-simulations in Matlab/Simulink and TruckSim under the lane change maneuver.

The rest of the paper is organized as follows. Vehicle models for controller design and its verification are described in Section 2. The gain-scheduling optimal controller design is presented in Section 3. Section 4 present the simulation results. Finally, the concluding marks are drawn in Section 5.

VEHICLE MODELS

In this section, two vehicle models are presented including a single-track linear vehicle model for controller design and a nonlinear vehicle model used for validating the proposed control system.

Single-Track Vehicle Model

For designing a gain-scheduling optimal yaw moment controller, a single-track linear vehicle model shown in Figure 1 is derived by linearizing the nonlinear model for lateral dynamics with constant forward velocity [5, 9]. This model is particularly well-suited to generate desired vehicle states for the control system design in a highly efficient manner. To derive this model, some assumptions must be made. The articulation angle between tractor and semitrailer units and the tire slip angles are assumed to be small. In addition, the lateral tire forces are modeled as a linear function of tire slip angles. Furthermore, the roll dynamics, load transfer and the effects of lateral winds and road camber are not considered.

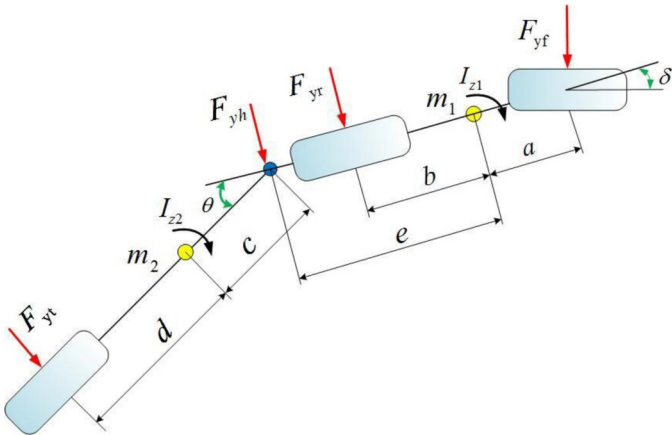


Figure 1. Single-track linear articulated vehicle model

Four motion states are considered in this model : tractor lateral velocity, tractor yaw rate, articulation angle between the tractor and semitrailer and articulation rate. The equations of motion for the linear model of the tractor-semitrailer combination are expressed as:

$$m_1(\dot{v}_{y1} + v_x\gamma_1) = F_{yf} + F_{yr} - F_{yh} \quad (1)$$

$$I_{z1}\dot{\gamma}_1 = aF_{yf} - bF_{yr} + F_{yh}e + \Delta M_1 \quad (2)$$

$$m_2a_{y2} = F_{yh} + F_{yt} \quad (3)$$

$$I_{z2}\dot{\gamma}_2 = F_{yh}c - F_{yt}d + \Delta M_2 \quad (4)$$

Considering the following kinematic relationship,

$$\gamma_2 = \gamma_1 + q \quad (5)$$

$$a_{y2} = \dot{v}_{y1} + v_x\gamma_1 - e\dot{\gamma}_1 - c(\dot{\gamma}_1 + \dot{q}) \quad (6)$$

The lateral forces developed at the tractor front and rear axle, and semitrailer axle wheels are assumed to be proportional to the tire slip angle, such that:

$$F_{yf} = -C_{af}\alpha_f = -C_{af}\left(\frac{v_{y1} + a\gamma_1}{v_x} - \delta\right) \quad (7)$$

$$F_{yr} = -C_{ar}\alpha_r = -C_{ar}\left(\frac{v_{y1} - b\gamma_1}{v_x}\right) \quad (8)$$

$$F_{yt} = -C_{at}\alpha_t = -C_{at}\left(\frac{v_{y1} - (c+d+e)\gamma_1 - (c+d)q}{v_x} - \theta\right) \quad (9)$$

Where F_{yf} , F_{yr} and F_{yt} are the lateral forces developed at tractor front and rear, and trailer axle tires, respectively, which are evaluated assuming constant cornering stiffness (C_{af} , C_{ar} , C_{at}) and α_f , α_r and α_t are the respective side-slip angles.

The description for the main parameters of tractor semitrailer system is given in Table 1 below.

Rewrite the motion equations in matrix form as below:

$$M\dot{x} = Kx + B_1\delta + B_2u \quad (10)$$

where M is the mass matrix; K is the stiffness matrix; B_1 and B_2 are the input matrices. δ is the front steering angle. The control input u , and the system state vector x are defined by

$$x = [v_y \quad \gamma \quad q \quad \theta]^T, \quad u = [\Delta M_1 \quad \Delta M_2]^T \quad (11)$$

To observe the dynamics of the model, the equation above can be converted into state space format, shown as

$$\dot{x} = Ax + Bu + E\delta \quad (12)$$

with

$$A = M^{-1}K$$

$$B = M^{-1}B_2$$

$$E = M^{-1}B_1$$

The matrices M , K and B_1 are given in the Appendix. It should be noticed that matrix K is a function of vehicle velocity v_x . And the validity of single-track linear vehicle models can be found in Reference [5].

Table 1. Primary parameters of tractor semitrailer

Symbol	Parameter
m_1	Mass of tractor, kg
m_2	Mass of semitrailer (empty), kg
I_{z1}	Moment of inertia of tractor, $\text{kg}\cdot\text{m}^2$
I_{z2}	Moment of inertia of semitrailer, $\text{kg}\cdot\text{m}^2$
a	Distance from tractor CG to front axle, m
b	Distance from tractor CG to rear axle, m
c	Distance from trailer CG to fifth wheel, m
d	Distance from trailer CG to trailer axle, m
e	Distance from tractor CG to fifth wheel, m
γ_1, γ_2	Yaw rate for tractor and semitrailer, rad/s
v_x	Vehicle velocity, m/s
v_{y1}	Tractor's lateral velocity
a_{y2}	Semitrailer's lateral velocity, m^2/s
θ	Articulation angle, rad
q	Articulation rate, rad/s
F_{yf}, F_{yr}, F_{yt}	Lateral forces at tractor front, rear and trailer axles, N
F_{yh}	Lateral force action on the fifth wheel, N
C_{af}, C_{ar}, C_{at}	Cornering stiffness at tractor front, rear and trailer axles
δ	Front steering angle, rad
$\Delta M_1, \Delta M_2$	Corrective yaw moments acting on tractor and semitrailer

Nonlinear Vehicle Model

TruckSim is a multi-body dynamics software used widely for simulating and analyzing the dynamic characteristics of medium to heavy trucks [14]. It can model multiple bodies such as a tractor, trailer and dolly. The tractor-semitrailer nonlinear vehicle model is formulated in TruckSim platform to verify the effectiveness of the proposed controller under specified maneuvers considering the model uncertainties, as shown in Figure 2.

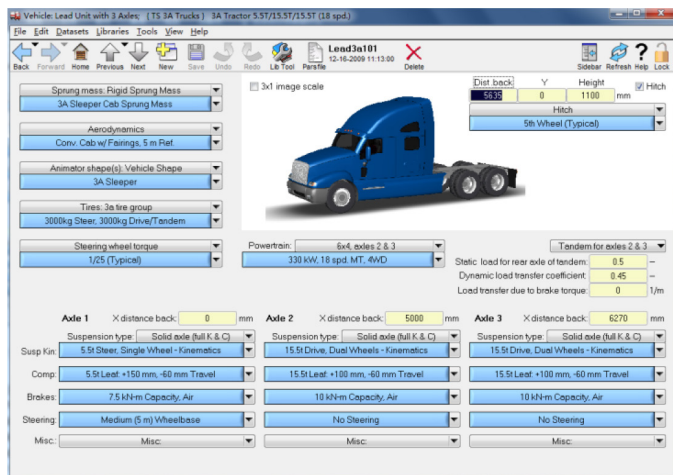


Figure 2. TruckSim vehicle model snapshot

CONTROL SYSTEM DESIGN

The block diagram of the proposed control system is shown in Figure 3. This control system structure includes reference model, gain-scheduling optimal yaw moment controller and braking torque distribution. x_d and x are the desired vehicle state and actual vehicle states, respectively. δ is the front steering wheel angle. e is the tracking error between the actual states and the desired states. ΔM_1 and ΔM_2 are the corrective yaw moments for tractor and semitrailer, respectively. T_{bLi} and T_{bRi} are the braking torques for left and right wheels. The main goal of the proposed control system is to make the actual tractor yaw rate and articulation angle to follow the desired values obtained from reference model. The reference model is developed to generate the desired vehicle states that keep the vehicle within the linear region. The gain-scheduling optimal yaw moment controller responds to the inputs: vehicle state errors for optimal controller design and vehicle velocity as a scheduling variable, and the outputs are the corrective yaw moments for tractor and semitrailer. Vehicle states such as vehicle velocity, tractor yaw rate, articulation angle, articulation rate and lateral velocity are assumed to be available in this study. In the following sections, each block is presented in details.

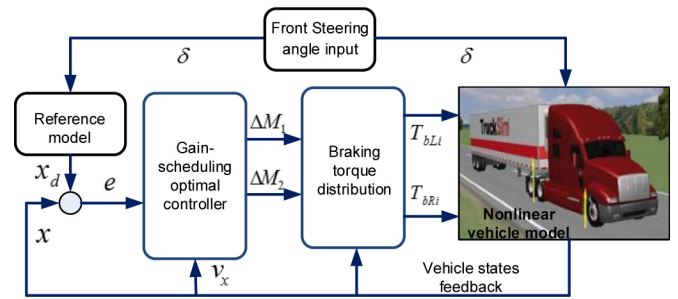


Figure 3. Block diagram of proposed control system

Reference Model

Vehicle active control systems are synthesized to track the desired responses generated by a reference model. In this section, the reference vehicle model is presented in details to produce four desired vehicle states: tractor yaw rate, tractor lateral velocity, articulation angle and articulation rate.

The desired tractor yaw rate is the steady-state yaw rate response of a linear single-track vehicle model to a steering command at steady state cornering, which is given by

$$\gamma_{1ss} = \frac{v_x}{l_1 + k_t v_x^2} \delta \quad (13)$$

k_t is understeer coefficient for tractor and defined as blow

$$k_t = \frac{m_1 b l_2 + m_2 d (b - e)}{l_1 l_2 C_{af}} - \frac{m_1 a l_2 + m_2 d (a + e)}{l_1 l_2 C_{ar}} \quad (14)$$

where $l_1 = a + b$, $l_2 = c + d$.