

# Performance Analysis of Active Independent Front Steering (AIFS) for Commercial Vehicles with Greater Lateral Load Shift Propensity

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## ABSTRACT

An Active Independent Front Steering (AIFS) offers attractive potential for realizing improved directional control performance compared to the conventional Active Front Steering (AFS) system, particularly under more severe steering maneuvers. The AIFS control strategy adjusts the wheel steer angles in an independent manner so as to utilize the maximum available adhesion at each wheel/road contact and thereby compensate for cornering loss caused by the lateral load transfer. In this study, the performance potentials of AIFS are explored for vehicles experiencing greater lateral load transfers during steering maneuvers such as partly-filled tank trucks. A nonlinear yaw plane model of a two-axle truck with limited roll degree-of-freedom is developed to study the performance potentials of AIFS under different cargo fill conditions. The lateral movement of the cargo within a partly-filled cylindrical tank is described by the resulting lateral load shift, lateral force and the roll moment using the quasi-static fluid motion. The AIFS control strategy based on a simple PI controller is subsequently integrated and simulations are performed under different loading conditions for a steady- turning maneuver. Simulation results in general show that while both the AFS and AIFS can achieve the target response to a steering command, the AFS control causes the inner wheel to saturate under lateral accelerations well below the rollover threshold limit. The AIFS strategy, on the other hand, realizes the target response at far greater speeds, while the steered wheels do not encounter cornering force saturation until the vehicle approaches rollover condition. The results further show superior tire work-load performance of the vehicle with AIFS, irrespective of the load.

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## INTRODUCTION

Active Front Steering Systems (AFS) have been designed to vary steer angles in an active manner to realize a target directional response [1]. The steer angles of both the inner and outer wheels, however, are altered simultaneously, while maintaining Ackerman geometry [2]. Furthermore, the ability of a tire to generate a lateral force is not adequately considered in the AFS synthesis. Alternatively, an Active Independent Front Steering (AIFS) concept that can realize steer angle corrections of the right- and left-steered wheels in an independent manner has been shown to outperform the conventional Active Front Steering (AFS) system in the context of handling performance of automobiles [3]. The AIFS concept permits maximum utilization of the tire-road adhesion available at each wheel/road contact without

approaching cornering force saturation to realize the target response. In doing so, the AIFS can maximize the performance limit of a road vehicle not only on a homogeneous surface but also on a split- $\mu$  road condition. The lateral load transfers encountered during a steering maneuver could lead to loss of cornering ability of the steered wheels and thereby limited handling performance under high speed maneuvers. The control strategy of AIFS system is based on compensation of the tire cornering force loss attributed to lateral load transfer across the steered wheels apart from utilization of the available tire-road adhesion so as to achieve improved handling performance, particularly under more severe high-g steering maneuvers.

The results obtained through parametric sensitivity analysis of the AIFS concept suggested greater effectiveness of AIFS for vehicles with higher mass center and thus greater

load transfer [3]. The AIFS is thus expected to yield better performance for commercial vehicles with their inherent high center of mass and roll motions that lead to large lateral load shifts. This phenomenon is more pronounced in liquid cargo vehicles particularly under partial fill conditions, where the lateral load transfers could be far more significant under steering and braking maneuvers [4]. The main focus of the present study is thus to examine the effectiveness of an AIFS in enhancing the handling performance of tank trucks with full and partial loads. It is well known that partially filled tank trucks exhibit substantially lower directional stability and control limits compared to rigid cargo trucks, which is primarily attributed to their high mass center and excessive cargo movement under steering maneuvers [5]. Such vehicles are thus more frequently involved in road accidents [6]. The implications of accidents involving such vehicles could be catastrophic, particularly when hazardous materials are involved. Numerous studies have illustrated the adverse influences of the liquid cargo movement within tank trucks on their directional stability and rollover propensity, which are further dependent upon the tank cross-section and the fill volume [7, 8, 9]. A partial fill condition yields lower center of mass (CG) height, and therefore superior stability limit and handling performance may be expected. The additional roll moment caused by cargo movement, however, may not only counter the gain due to lower CG height but also lead to lower directional control performance of the vehicle. This trend is evident from the reported studies on wider cross-section tanks such as oval and modified oval tank vehicles [7]. For cylindrical cross-section tank, it has been shown that the additional roll moment due to cargo movement nearly nullifies the low CG gain in the context of rollover threshold limit. The cylindrical tank trucks thus exhibit lower sensitivity of the rollover threshold to variations in the fill height, when cargo load is constant as observed in general purpose tank truck fleets transporting products of varying weight density [9]. The handling performance of partly-filled tank trucks, which exhibit excessive load transfers irrespective of the tank cross-section, may be enhanced through AIFS.

From the review of studies reporting the performance potentials of AFS, it is apparent that AFS control strategy has been mostly explored with an objective to reduce the rollover hazard in an emergency maneuver [10]. However, the results suggest that enhancing the vehicle rollover threshold may lead to lateral instability during an evasive maneuver. An AFS system may thus provide enhanced vehicle rollover immunity at the expense of poor handling performance through relatively greater path deviation [11]. Some studies have proposed integration of Direct Yaw-moment Control (DYC) together with Active Steering in order to compensate for the path deviations [12]. Such an approach would lead to significant system and control complexities. Furthermore, it has been reported that systems with DYC frequently lead to a level of disturbance and annoyance to the driver [13-14]. The application of AFS control to an articulated tractor-

semitrailer combination has also been studied for prevention of jackknife instabilities during severe driving maneuvers [15].

The AFS control that can apply differential steering to right- and left-steered wheels in an independent manner with appropriate considerations of instantaneous load transfer and possible cornering force saturation of the tires offers considerable advantages for enhanced handling performance over a wider speed range. The potential handling performance gains of an AIFS have been illustrated for a passenger car [3]. The study also showed far greater performance potentials of an AIFS for high CG vehicles subject to high-g maneuvers, the situations involving greater load transfers. For passenger cars with only moderate load shift across the steered wheels, the performance gain was realized by primarily increasing the steer angle of the outer wheel with larger normal load, while the controller strategy was based on predetermined slip angle limit for tire saturation. Alternatively, an improved control strategy may be realized through online monitoring of impending tire force saturation in terms of tire work-load. A control strategy that attempts to equalize tire work-loads of inner and outer wheels would not only permit maximum possible cornering force but also retain sufficient road adhesion for developing essential longitudinal forces.

In this study, the performance characteristics of an AIFS are investigated for applications in commercial vehicles with high CG and lateral load transfers. A two-axle tank truck is considered as a case example. A nonlinear yaw-plane model of the tank truck integrating roll dynamics of the sprung mass is developed to study the directional performance potential of the AIFS. The load shift associated with liquid cargo motion in the roll plane of a cylindrical tank is evaluated using the quasi-static approach and incorporated in the vehicle model in terms of resultant lateral force and roll moment attributed to quasi-static cargo shift. The AIFS strategy based on a yaw rate reference model is synthesized considering delays associated with tire lag and steering response. The controller strategy permits independent control of each wheel so as enhance handling performance limit through maximum utilization of the available tire-road adhesion prior to approaching the saturation limit. Simulation results are obtained for both the full and partial load conditions under a steady-turning maneuver at different forward speeds to demonstrate the effectiveness of an AIFS system for applications in rigid as well as partly-filled liquid cargo trucks.

## DIRECTIONAL DYNAMIC MODEL OF THE TRUCK

A simple yaw plane model of a two-axle truck with roll-DOF is formulated, as shown in Fig. 1. The model comprises right- and left- wheels with independent steering angles in order to study the performance potentials of the AIFS. The model also considers nonlinear tire model in order to predict

tire forces under severe maneuvers and possible impending cornering force saturation. The model with independent steering of the right- and left-wheels is formulated considering roll motion of the sprung mass, as well as lateral and yaw motions of the vehicle, as shown in Figs. 1(a) and (b), respectively. In the figure, 'RC' refers to roll center of the sprung mass and  $\phi$  denotes the sprung mass roll angle. The equations of motions obtained considering large steer angles are summarized below:

$$mV(\dot{\beta} + r) - m_s h_s \ddot{\phi} = F_{YFr} \cos \delta_r + F_{YFl} \cos \delta_l + 2(F_{YRr} + F_{YRl}) \quad (1)$$

$$(I_{xs} + m_s h_s^2) \ddot{\phi} - I_{xzs} \dot{r} - m_s h_s V(\dot{\beta} + r) = (m_s g h_s - K_{\phi F} - K_{\phi R}) \phi - (C_{\phi F} + C_{\phi R}) \dot{\phi} \quad (2)$$

$$(I_{zs} + I_{zu}) \dot{r} - I_{zxs} \ddot{\phi} = -\sum_{i=F}^R \sum_{j=l}^r M_{ij} + F_{YFr}(b \cos \delta_r - T_F \sin \delta_r) + F_{YFl}(b \cos \delta_l + T_F \sin \delta_l) - 2c(F_{YRr} + F_{YRl}) \quad (3)$$

Where  $V$ ,  $r$  and  $\beta$  are the speed, yaw rate and side-slip angle of the vehicle, respectively. The distances of the front and rear axle from the CG are denoted by  $b$  and  $c$ .  $I_{xs}$ ,  $I_{xzs}$  and  $I_{zs}$  are sprung mass moments of inertia about the  $x$ ,  $xz$  and  $z$  axis at the mass center, respectively, and  $I_{zu}$  is the moment of inertia of the unsprung mass.  $K_{\phi i}$  and  $C_{\phi i}$  ( $i = F, R$ ) are the total front- and rear-axle suspension roll stiffness and roll damping, respectively.

In the above equations, tire lateral force  $F_{Yij}$  and self-aligning moment  $M_{ij}$  ( $i = F, R$ ;  $j = r, l$ ) are derived using the Magic Formula [16] considering the normal load  $F_{zij}$  and slip angle  $\alpha_{ij}$  of each wheel, where subscripts  $F$  and  $R$  refer to front- and rear-tires, and  $r$  and  $l$  denote the right- and left-wheels, respectively. The normal load on each wheel can be derived as a function of the longitudinal ( $a_x$ ) and lateral ( $a_y$ ) accelerations as well as the suspension roll stiffness ( $K_{\phi i}$ ) and damping ( $C_{\phi i}$ ), in the following manner:

$$F_{ZFr} = \frac{W_F}{2} - \frac{a_x}{2L} (m_s h_{cgs} + m_{uF} h_{uF} + m_{uR} h_{uR}) + \frac{a_y}{2T_F} (m_{sF} h_{Froll} + m_{uF} h_{uF}) + \frac{1}{2T_F} (K_{\phi F} \phi + C_{\phi F} \dot{\phi}) \quad (4)$$

$$F_{ZFl} = \frac{W_F}{2} - \frac{a_x}{2L} (m_s h_{cgs} + m_{uF} h_{uF} + m_{uR} h_{uR}) - \frac{a_y}{2T_F} (m_{sF} h_{Froll} + m_{uF} h_{uF}) - \frac{1}{2T_F} (K_{\phi F} \phi + C_{\phi F} \dot{\phi}) \quad (5)$$

$$F_{ZRr} = \frac{W_R}{4} + \frac{a_x}{4L} (m_s h_{cgs} + m_{uF} h_{uF} + m_{uR} h_{uR}) + \frac{a_y}{4T_R} (m_{sR} h_{Rroll} + m_{uR} h_{uR}) + \frac{1}{4T_R} (K_{\phi R} \phi + C_{\phi R} \dot{\phi}) \quad (6)$$

$$F_{ZRl} = \frac{W_R}{4} + \frac{a_x}{4L} (m_s h_{cgs} + m_{uF} h_{uF} + m_{uR} h_{uR}) - \frac{a_y}{4T_R} (m_{sR} h_{Rroll} + m_{uR} h_{uR}) - \frac{1}{4T_R} (K_{\phi R} \phi + C_{\phi R} \dot{\phi}) \quad (7)$$

Where  $T_F$  and  $T_R$  are the half tire track widths of the front and rear axles, respectively,  $L$  is the wheel base,  $m_{si}$  and  $m_{ui}$  ( $i = F, R$ ) refer to the front and rear sprung and unsprung masses, respectively, and  $m_s$  is the total sprung mass.

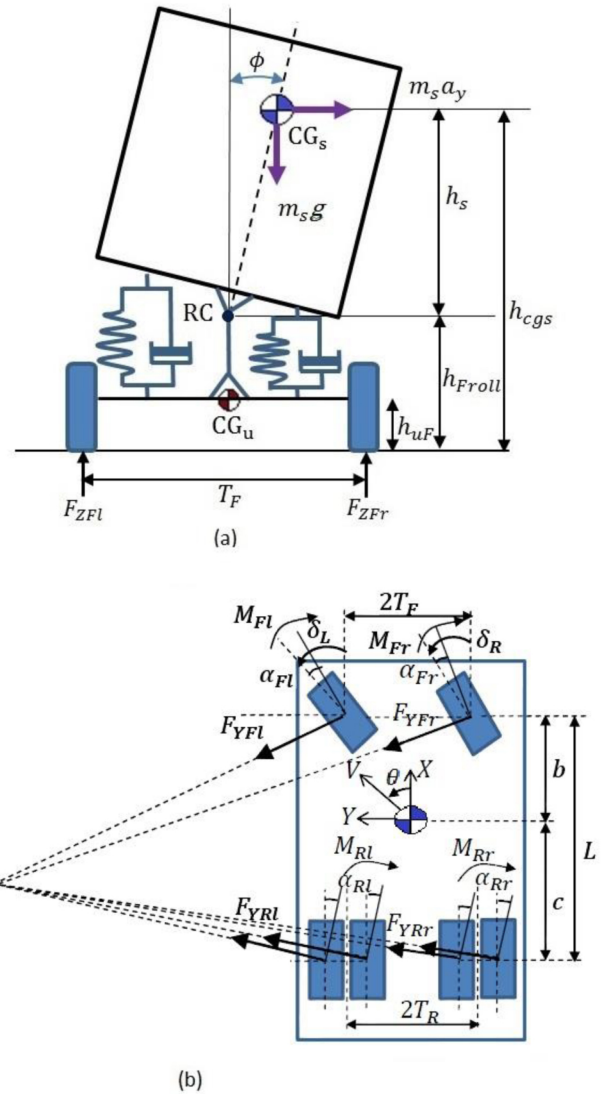


Figure 1. (a). Three-DoF directional dynamic model of a two axle truck: (a) roll-plane; and (b) yaw plane.

## Load Shift in a Partially-Filled Tank

Equations (1), (2), (3), (4), (5), (6), (7) describe the steering dynamic responses of the vehicle with only rigid cargo. The vehicle however is subjected to additional lateral load shift, lateral force and the roll moment due to lateral movement of liquid cargo within the partly filled tank. Figure

2 illustrates the roll plane of the vehicle equipped with a partly-filled cylindrical tank. The center of mass of the liquid cargo experiences a lateral shift ( $y_c$ ) due to roll motion of the sprung mass and the maneuver- induced lateral acceleration  $a_{yc}$ .

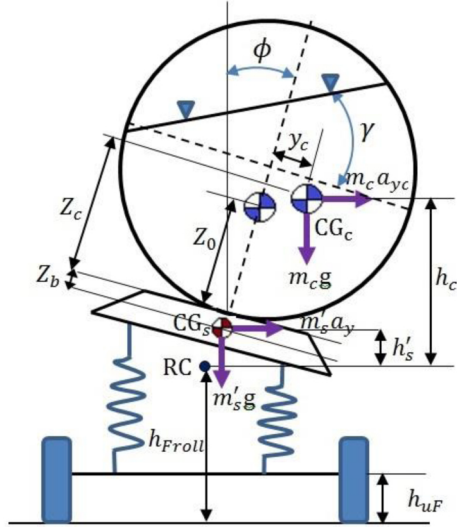


Figure 2. Partially filled tank truck model.

The magnitude of  $y_c$  depends on the tank geometry, fill height and liquid free surface gradient ( $\gamma$ ). Assuming quasi-static motion and inviscid fluid cargo, the liquid free surface gradient is related to  $a_{yc}$  and  $\phi$ , such that [7]:

$$\gamma = \frac{a_{yc} + \phi}{1 - a_{yc}\phi} \quad (8)$$

For a cylindrical tank of radius  $R$ , the lateral ( $y_c$ ) and vertical ( $Z_c$ ) movements of the liquid cargo mass center can thus be expressed as [7]:

$$y_c = (R - Z_0)\sin\gamma \quad (9)$$

$$Z_c = (R - Z_0)\cos\gamma + Z_b \quad (10)$$

Where  $Z_0$  is static CG height of the liquid cargo, which depends on the fill level.  $Z_b$  is tank base height from the chassis CG, as shown in Fig. 2. Assuming relatively small sprung mass roll angle compared to the lateral acceleration encountered during a steering maneuver yields,  $\gamma \approx a_{yc}$ . Furthermore, assuming  $\sin\gamma \approx \gamma$  yields:

$$y_c = (R - Z_0)a_{yc} \quad (11)$$

The lateral acceleration at the cargo mass center may be expressed in terms of the vehicle forward speed, vehicle sideslip angle, roll acceleration and the yaw rate, as:

$$a_{yc} = V(\dot{\beta} + r) - h_c\ddot{\phi} \quad (12)$$

From Fig. 2, it is evident that lateral movement of the cargo imposes an additional roll moment ( $OM$ ) about the roll center, as:

$$OM = m_c g y_c \quad (13)$$

Substituting for  $y_c$  from Eqns (11) and (12) yields following expression for  $OM$ :

$$OM = m_c(R - Z_0)(V(\dot{\beta} + r) - h_c\ddot{\phi}) \quad (14)$$

Subsequently, considering independent steering of the right- and the left-wheels, the equations of motion for the partly- filled tank truck model incorporating forces and moments due to cargo motion are formulated as:

$$mV(\dot{\beta} + r) - (m'_s h'_s + m_c h_c)\ddot{\phi} = F_{YFr} \cos \delta_r + F_{YFl} \cos \delta_l + 2(F_{YRr} + F_{YRl}) \quad (15)$$

$$I_{xsc}\ddot{\phi} - I_{xzsc}\dot{r} - (m'_s h'_s + m_c(h_c + R - Z_0))V(\dot{\beta} + r) + m_c h_c(R - Z_0)\ddot{\phi} = ((m'_s h'_s + m_c h_c)g - K_{\phi F} - K_{\phi R})\phi - (C_{\phi F} + C_{\phi R})\dot{\phi} \quad (16)$$

$$(I_{zsc} + I_{zu})\dot{r} - I_{zxsc}\ddot{\phi} = -\sum_{i=F}^R \sum_{j=R}^L M_{ij} + F_{YFr}(b\cos\delta_r - T_F \sin\delta_r) + F_{YFl}(b\cos\delta_l + T_F \sin\delta_l) - 2c(F_{YRr} + F_{YRl}) \quad (17)$$

Where  $m'_s$  is the mass due to chassis and the tare tank, and  $m_c$  is mass of the liquid cargo.  $I_{xsc}$ ,  $I_{xzsc}$  and  $I_{zsc}$  are the mass moments of inertia of the chassis and the cargo mass about the  $x$ ,  $xz$  and  $z$  axis, respectively, at the roll center.  $I_{zu}$  is mass moment of inertia of the unsprung mass.

## AIFS REFERENCE MODEL AND CONTROL STRATEGY

The AIFS control structure has been formulated to track the steering response of an idealized reference model based upon maximum utilization of the available adhesion limits of both the steered wheels, especially in a high lateral acceleration maneuver, as described in [3]. A simple PI controller is synthesized in this investigation to explore the performance potential of the AIFS concept applied to the two-axle truck. The controller is designed to provide a corrective steering action to achieve the reference yaw rate corresponding to a neutral steer condition, which is expressed as a function of forward velocity and instantaneous driver steering command  $\delta_{st}$ , such that: