

Design and Evaluation of Active Front Wheel Steering System Model and Controller

Jeonghoon Song
 Tongmyong University

ABSTRACT

In this study, a model of Active Front wheel Steer (AFS) system are developed and tested. In addition, an Integrated Dynamics Control with Front steer (IDCF) controller is also designed to investigate the performance of AFS system when it is integrated with a brake system. The IDCF system composed of an AFS system and a DYC (Direct Yaw moment Control) system of rear wheels.

The AFS controller and IDCF controller are compared under several driving and road conditions with and without braking input and steering input. A 8 degree of freedom vehicle model is also employed to test the controllers. The results show that the model of AFS system shows good kinematic steering assistance function. Steering ratio varies depends on vehicle velocity between 12 and 24. Kinematic stabilization function also shows good performance because yaw rate of AFS vehicle tracks the reference yaw rate.

IDCF shows improved responses compared to AFS because body side slip angle is also reduced regardless of road condition, steer input and brake input. These results represent that IDCF enhances lateral stability and steerability. Results also prove that AFS system shows satisfactory result when it is integrated with another chassis system. On a split- μ road, two controllers forced the vehicle to proceed straight ahead.

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INTRODUCTION

The steering system acts a significant role of making car convenient to handle and enhance the vehicle stability. In the past one hundred years, the development of steering system has experienced many stages, and the Steer-by-Wire system (SBW) is the newest technology of steering system for passenger cars. However, the Steer-by-Wire system has not yet accepted by public consumers and permitted by state regulations, in consideration of the reliability and safety of the system.

Active Front Steering (AFS) is a newly technology for passenger cars developed by BMW, that implements an electronically controlled superposition of an angle to the hand steering wheel angle that is prescribed by the driver. However, the permanent mechanical connection between steering wheel and road wheels remains [1]. AFS could adjust the vehicle performance by means of intervening the road wheel angle in condition of the driver have top priority, which avoid the people's concerns about the Steer-by-Wire system.

Additionally to the enhanced dynamic behavior of the steering system and vehicle stabilization, the AFS system should also provide an improved steering comfort by reducing steering effort [2]. Therefore, the torque assistance is required to limit manual forces to a reasonable level.

Here, we propose to use an AFS model based on the assumption that the AFS system is composed of four sub-systems, and the dynamics of each sub-system can be described using a set of differential equations. The AFS model simulates the steering assistance function, which is activated via a variable steering ratio (VSR). A sliding-mode AFS controller is developed using the AFS model, which realises the kinematic stabilisation function and modifies the corrective steering angle to allow the yaw rate to track the reference yaw rate.

We also describe an integrated dynamics control with front wheel steering (IDCF) systems, which are similar to the IVDC (Integrated Vehicle Dynamics Control) [3] and use the AFS system. This is developed to investigate the performance of the

AFS system when integrated with a braking or a steering system. It is composed of an AFS system and a DYC system for the rear wheels.

The goal of this paper is to develop an AFS model and controller, which not only helps ensure the vehicle's response operability matching the driver's sense, but also helps prevent the vehicle from falling into an unstable state.

SYSTEM MODEL

Vehicle Model

The creation of a model-based automotive simulation requires the availability of empirical and analytical mathematical descriptions of the system components. The components are the wheels, brakes, steering, suspension, tire/road interface, driver, environment and in some instances, the powertrain dynamics [4].

Before developing a vehicle model, it was assumed that the following sensor information is available: the speed of each wheel ω_i , the longitudinal acceleration a_x , the lateral acceleration a_y , and the yaw rate at the vehicle centre of gravity $\dot{\gamma}$. The vehicle body slip angle β_p is defined as the angle between the longitudinal axis of the vehicle, x , and the orientation of the vehicle velocity vector, v_{COG} can be observed by a nonlinear observer [5].

The vehicle body was represented by a 8-degrees-of-freedom model, as shown in Figure 1. Although the suspension was not included, the model used a simplified description of body roll assuming a fixed roll axis defined by the height of the roll centres of the front and rear axles of the vehicle.

Using Newton's second law of motion and Figure 1, the equations of motion for the vehicle are as follows [5,6]:

$$m_{total} \left(\dot{v}_x - v_y \dot{\gamma} \right) = \sum_{i=1}^4 F_{X_i} - F_D \cos \beta \quad (1)$$

$$m_{total} \left(\dot{v}_y + v_x \dot{\gamma} \right) = \sum_{i=1}^4 F_{Y_i} - F_D \sin \beta \quad (2)$$

$$\begin{aligned} \Gamma &= I_z \frac{d^2 \gamma}{dt^2} \\ &= aFY_1 + \frac{t_f}{2}FX_1 + aFY_2 - \frac{t_f}{2}FX_2 - bFY_3 + \frac{t_r}{2}FX_3 - bFY_4 \\ &\quad - \frac{t_r}{2}FX_4 + M_{z1} + M_{z2} + M_{z3} + M_{z4} + M_{zw} \end{aligned} \quad (3)$$

$$\begin{aligned} I_{roll} \ddot{\phi} + B_{roll} \dot{\phi} + K_{roll} \phi \\ = m_s g h_s \sin \phi - m_s \left(\dot{v}_y + v_x \dot{\gamma} \right) h_s \cos \phi \end{aligned} \quad (4)$$

where m_{total} and m_s are the total mass and the sprung mass of the vehicle, F_D is the aerodynamic drag force, Γ is the total torque acting on the vehicle in the yaw plane around the z -axis, I_z and I_{roll} are the vehicle moment of inertial around z -axis and roll axis, t_f and t_r are the front and rear wheel distance, M_z is the tire aligning moments, M_{zw} is the aerodynamic yaw moment, ϕ is the roll angle, B_{roll} and K_{roll} are the roll axis torsional damping and stiffness, h_s is the distance from sprung mass cg (center of gravity) to roll axis and F_x and F_y are obtained from the tire model.

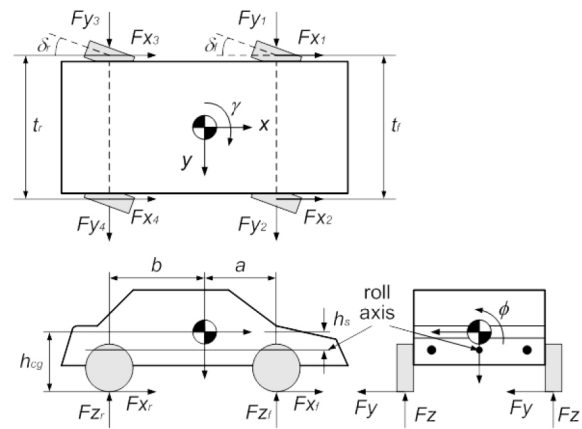


Figure 1. Vehicle Model ([5]).

The wheel acceleration is [5]

$$\begin{aligned} I_{wi} \dot{\omega}_i &= -T_{bi} - T_{ti} - T_{rolli} + T_i \\ &= -AR_b P_{bi} - F_{xi} R_w - F_{zi} f_{ri} R_w + T_i \end{aligned} \quad (5)$$

where I_{wi} is the mass moment inertia of the wheel about the axis of rotation, T_{bi} is the brake torque, T_{ti} is the tire tractive torque, T_{rolli} is the rolling resistance torque, T_i is the torque from the driveshaft, A is the area of master cylinder, R_b is the distance from center of wheel to brake path, P_{bi} is the brake fluid pressure, F_{xi} is the longitudinal force, F_{zi} is the normal force, and f_{ri} is the rolling resistant coefficient.

The forces represented in Eqs. (1), (2) and (3) are

$$F_{X_i} = F_{x_i} \cos \delta_i - F_{y_i} \sin \delta_i \quad (6)$$

$$F_{Y_i} = F_{y_i} \sin \delta_i + F_{x_i} \cos \delta_i, \quad (7)$$

AFS System Model

In the mechanism and the structure of AFS, the planetary gear set is inserted between steering column and rack-pinion system [1,2]. The entire steering system with AFS includes hand steering wheel, the planetary gear set, rack-pinion system and wheel. The dynamic characteristics of steering system are determined by four sub systems and the related physical parameters such as inertial moments and rotational stiffness of shafts.

As shown in Figure 2, the dynamic equations can be drawn from the diagram. When considering the equivalent torsional stiffness and damping between hand steering wheel and the input of planetary gear set, the dynamic equation can be depicted by

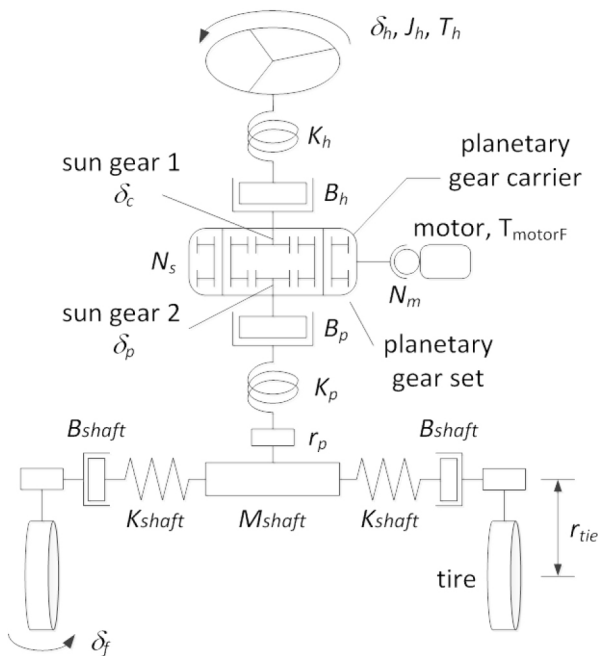


Figure 2. A schematic of AFS system.

$$J_h \ddot{\delta}_c + B_h \left(\dot{\delta}_c - \dot{\delta}_h \right) + K_h (\delta_c - \delta_h) = T_h \quad (8)$$

where, J_h denotes the rotary inertial moment of hand steering wheel assemblies. B_h and K_h are the equivalent torsional damping and stiffness coefficients, respectively. δ_h and δ_c are the angles of handing steering wheel input and sun gear I in planetary gear set. T_h is the input torque from driver's steering action.

When considering the equivalent torsional stiffness and damping between sun gear II and the pinion, the dynamic equation can be depicted by

$$J_g \ddot{\delta}_p + B_g \left(\dot{\delta}_p - \frac{\dot{\delta}_c}{N_s} \right) + K_g \left(\delta_p - \frac{\delta_c}{N_s} \right) = T_g \quad (9)$$

where J_g denotes the equivalent rotary inertial moment of output shaft, B_g and K_g are the equivalent torsional damping and stiffness coefficients, respectively. δ_p is the angle of the pinion in rack-pinion set. N_s is the ratio of steering wheel angle (δ_h) to pinion angle (δ_p). T_g is the output torque from planetary gear set to pinion and consist of torques from driver and from assistant motor to generate the corrective rotation.

$$T_g = N_s T_h + N_m T_{motorF} \quad (10)$$

where N_m is the gear ratio between the steering wheel and the assistant motor.

In rack-pinion section of the steering system, the rotational movement of the column shaft is converted to the rectilinear movement of the rack and tie rod. The movement and simulation of this section is as follows:

$$M_{shaftF} \ddot{x}_{shaftF} + B_{shaftF} \left(\dot{x}_{shaftF} - \dot{\delta}_p \times r_{pF} \right) + K_{shaftF} (x_{shaftF} - \delta_p \times r_{pF}) = F_{rF} \quad (11)$$

where, M_{shaftF} denotes the equivalent mass of rack and tie road, B_{shaftF} and K_{shaftF} are the equivalent damping and stiffness coefficients, respectively. x_{shaftF} is the movement of the rack and tie road. F_{rF} denotes the force exerted at the engaged rack-pinion, and r_{pF} is the radius of the pinion.

The tire movement about the z-axis can be described by

$$J_{tire} \ddot{\delta}_{tire} + B_{tire} \left(\dot{\delta}_{tire} - \frac{\dot{x}_{shaftF}}{r_{tie}} \right) + K_{tire} \left(\delta_{tire} - \frac{x_{shaftF}}{r_{tie}} \right) = -T_{ext} = -F_{ext} r_{tie} \quad (12)$$

The force F_{ext} delivered to the rack from the longitudinal and lateral forces of the tires in Eq. (12) is [5]

$$F_{ext} = F_t \times \sin \delta_f + F_s \times \cos \delta_f \quad (13)$$

F_t and F_s are the longitudinal and lateral forces and can be calculated from Dugoff's tire model [4].