

Integrated Chassis Control for Enhancement of High Speed Cornering Performance

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ABSTRACT

This paper describes an Integrated Chassis Control (ICC) strategy for improving high speed cornering performance by integration of Electronics Stability Control (ESC), Four Wheel Drive (4WD), and Active Roll Control System (ARS). In this study, an analysis of various chassis modules was conducted to prove the control strategies at the limits of handling. The analysis is focused to maximize the longitudinal velocity for minimum lap time and ensure the vehicle lateral stability in cornering. The proposed Integrated Chassis Control algorithm consists of a supervisor, vehicle motion control algorithms, and a coordinator. The supervisor monitors the vehicle status and determines desired vehicle motions such as a desired yaw rate, longitudinal acceleration and desired roll motion. The target longitudinal acceleration is determined based on the driver's intention and vehicle current state to ensure the vehicle lateral stability in high speed maneuvering. The vehicle motion control algorithm calculates a desired longitudinal force, yaw and roll moment for the generation of the desired vehicle motions. In the coordinator, actuator control inputs are coordinated to optimize the driving performance based on proposed strategies. Closed loop simulations of a driver-vehicle-controller system were conducted to evaluate the performance of the proposed control algorithm. The performance of the Integrated Chassis Control has been compared to those of individual chassis control systems such as ESC, 4WD, ARS, and Electronic Control Suspension (ECS). Simulation results show that the proposed ICC algorithm reduces lap time compared to the individual chassis control systems.

CITATION: Heo, H., Joa, E., Yi, K., and Kim, K., "Integrated Chassis Control for Enhancement of High Speed Cornering Performance," *SAE Int. J. Commer. Veh.* 8(1):2015, doi:10.4271/2015-01-1568.

INTRODUCTION

In order to cope with the complicated operation conditions and to improve vehicle stability, maneuverability, ride comfort and/or vehicle safety, various chassis control systems, such as Anti-skid Brake Systems (ABS), Four Wheel Steering (4WS), Electronic Stability Control (ESC), Active Roll Control Systems (ARS) and Electronic Control Suspensions (ECS), have been developed and equipped in vehicles one after another since the late 1970s. The ESC and 4WS have been mainly used to control the lateral dynamics of vehicles. The ECS has been used for the improvement of ride comfort, road holding and roll dynamics. The ARS has been used to stabilize the roll motions and also to control yaw dynamics.

An integration of individual modular chassis control systems such as the ESC, AFS, 4WD, ECS and ARS is a more efficient method to provide crucial benefits for vehicle dynamics, such as agility, maneuverability or additional stability improvement [1]. The integration of these chassis control systems is loosely referred to as the Unified Chassis Control (UCC) system, and has been an area of intensive development by many companies in recent years [2]. Coordination of steering and individual wheel braking actuation to achieve better vehicle yaw stability has been reported [3]. Intelligent

vehicle motion control that interfaces a theoretical controller with existing braking and steering control subsystems has been proposed [4]. Linear quadratic (LQ) control theory has been applied to the design of the integrated direct yaw moment control (DYC) and Active Front Steering control (AFS) [5]. A unified chassis control (UCC) strategy has been proposed for improving agility, maneuverability, and vehicle lateral stability through the integration of active front steering (AFS) and electronic stability control (ESC) [6]. The proposed UCC system consists of a supervisor and a coordinator to integrate the active front steering angle brake. In order to obtain optimal control inputs, the control modes are determined by the supervisor, and the Karush-Kuhn-Tucker (KKT) condition is used to compute the optimized coordination of tire forces, considering tire friction circle constraints [7, 8]. An improved Inverse Nyquist Array (INA) based feedback ICC controller is designed for AFS and ESC integration to decouple the plant dynamics under different frequency bands [9]. Smakman proposed an integrated control algorithm for active suspension and braking using a similar strategy to that used in coordination of AFS and ESC, i.e. giving priority to active suspension and the active braking is activated when vertical load control cannot adequately generate the stability yaw moment [10].

Crolla added a central differential controller for the integrated control algorithm of ARS and 4WD to allocate more traction moment to the wheels [11].

The proposed integrated chassis control (ICC) algorithm has been designed to maximize driving velocity in cornering situation by integrating three individual chassis control modules: 4WD, ESC and ARS. The proposed strategy coordinates longitudinal and vertical forces optimally to use the maximum lateral tire forces at the limits of handling. The proposed integrated chassis control algorithm consists of three modular algorithms, i.e., a supervisory controller, vehicle motion control algorithm (upper-level controller), and a coordinator (lower-level controller). The supervisory controller monitors the vehicle status and determines the desired vehicle motions: the desired longitudinal acceleration, yaw rate and roll motion. The upper-level controller determines the desired control inputs necessary to track the target vehicle motions such as roll and yaw moment. The coordinator determines the actual control inputs of individual chassis modules, such as front/rear drive torques, 4-wheel brake torques, and active roll moment based on the optimized control allocation method. The performance of the ICC system was investigated via closed-loop driver-vehicle-controller computer simulations with commercial software MATLAB/SIMULINK and CARSIM. The high-speed optimized driver model was implemented to evaluate the performance of ICC at the limits of handling. From the simulation results, ICC reduced lap time by using higher lateral forces than other individual control algorithms.

ANALYSIS OF VEHICLE AND TIRE DYNAMICS

The vehicle cornering dynamics on steady-state is described based on the planar model and tire characteristics. The maximum velocity in cornering is determined by vehicle and tire force characteristics. The ideal ratio of the front and rear lateral tire forces is determined by the location of the mass center point. The tire force limitation is calculated based on load distribution with nonlinear tire characteristics.

Vehicle Cornering Dynamics

The ideal ratio of front and rear lateral tire forces is described based on the planar model. Fig. 1 shows the planar model [25].

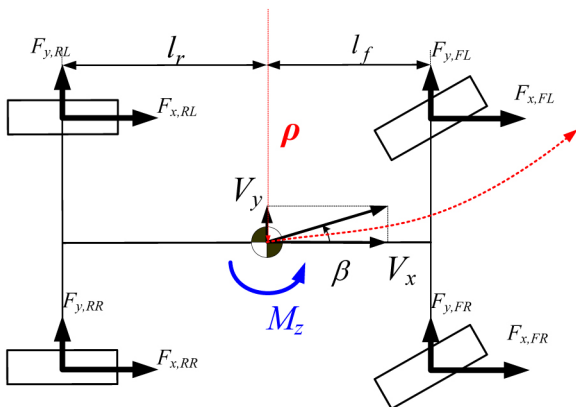


Fig. 1. 2-D Planar vehicle model

The lateral acceleration for steady state cornering is described as follows:

$$a_y = \dot{V}_y + \gamma V_x \approx \gamma V_x = \frac{V_x^2}{\rho} \quad (1)$$

The force and moment equilibrium equations are described as follows:

$$\begin{cases} F_y = \underbrace{(F_{y,FL} + F_{y,FR})}_{F_{yf}} + \underbrace{(F_{y,RL} + F_{y,RR})}_{F_{yr}} = m \cdot a_y \\ M_z = l_f \cdot F_{yf} - l_r \cdot F_{yr} = 0 \end{cases} \quad (2)$$

From these equations, the ratio of front and rear lateral tire forces is determined as follows:

$$\frac{F_{yr}}{F_{yf}} = \frac{l_f}{l_r} \quad (3)$$

The maximum lateral tire forces of each axle can be represented as follows:

$$F_y = c_1 (F_z + c_2 F_z^2) \quad (4)$$

where, c_1 is a positive constant coefficient of tire characteristics; $-c_2$ is a negative constant coefficient of tire characteristics. By these nonlinear characteristics of tire, the maximum lateral acceleration is determined as follows:

$$a_{y,\max} = \begin{cases} (1 + l_f / l_r) F_{yf,\max} & \text{if } F_{yf,\max} > F_{yr,\max} \\ (1 + l_r / l_f) F_{yr,\max} & \text{else} \end{cases} \quad (5)$$

where $F_{yf,\max}$ and $F_{yr,\max}$ are maximum lateral force of the rear and front axle, respectively. In accelerating and decelerating situation, the correlation between the longitudinal and lateral tire forces should be considered to determine the limitation. To analyze the nonlinearities, the maximum lateral tire forces are described as follows:

$$F_{yi,\max} = G_{y\lambda} G_W F_{yi,\max,0} \quad (6)$$

where, $F_{yi,\max,0}$ is the nominal value of the maximum lateral tire force. Two compensation coefficient used in this paper can be determined as follows:

$$G_{y\lambda,f} = \frac{\sqrt{\mu^2 F_{z,f} - (F_{x,f,4WD} - F_{x,FL,ESC} - F_{x,FR,ESC})^2}}{\mu F_{z,f}}$$

$$G_W = \left\{ 1 - \frac{\partial F_y}{\partial F_z} \left(\frac{\Delta F_{z,a_y} + \Delta F_{z,a_x} + \Delta F_{z,ARS}}{F_z} \right)^2 \right\}$$
(7)

where $\partial F_y / \partial F_z$ denotes the tire characteristics, $\Delta F_{z,a_x}$, $\Delta F_{z,a_y}$, and $\Delta F_{z,ARS}$ denotes the load transfer of accelerations and ARS control force, respectively. From the maximum lateral acceleration, the maximum velocity, $V_{x,max}$, in steady-state cornering is described as follows:

$$V_{x,max} = \sqrt{\rho \cdot a_{y,max}}$$
(8)

CHASSIS CONTROL PERFORMANCE ANALYSIS

Target Performance

From the analysis of vehicle cornering dynamics in the previous section, the maximum lateral acceleration in cornering can be increased by using maximum front and rear lateral tire forces simultaneously, as shown in Fig. 2. To use the maximum tire forces, ARS enlarges the saturated tire limitation and 4WD/ESC distributes the front/rear longitudinal forces optimally.

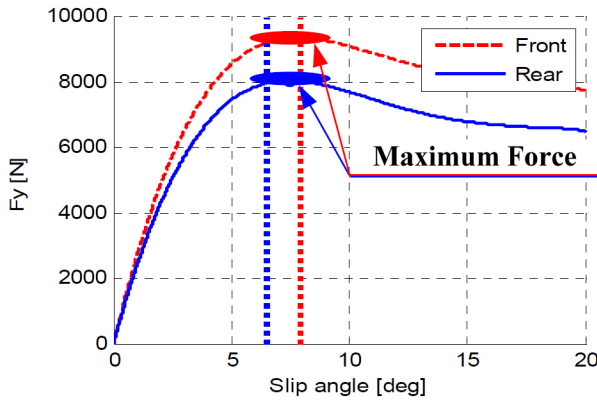


Fig. 2. Optimal tire forces to obtain maximum lateral acceleration

Architecture of Integrated Chassis Control Algorithm

A schematic diagram of the integrated chassis control algorithm proposed in this paper is shown in Fig. 3. As shown in the figure, the proposed algorithm consists of three parts: a supervisory controller, an upper-level controller, and a lower-level controller. The supervisory controller monitors the vehicle status and determines target dynamics such as the target yaw rate considering agility for neutral steer and the target longitudinal acceleration based on the human driver's inputs, sensor signals, and the vehicle limitations. The vehicle limitations denote the relationship between the vehicle speed and maximum curvature of a vehicle considering the steering and wheel slip. The upper-level controller computes the desired yaw

moment, roll moment, and longitudinal force to track the desired vehicle motions. The lower-level controller is designed to obtain the actuator commands, which are the front and rear drive engine torques, independent brake torques, and active front/rear roll moments. The algorithm is designed based on an optimization-based control allocation strategy to apply the upper-level control inputs to the test platform, taking into account the actuator constraints. In the actuator-level controller, the wheel slip control ensures that the wheel slip ratio does not exceed the slip ratio limit.

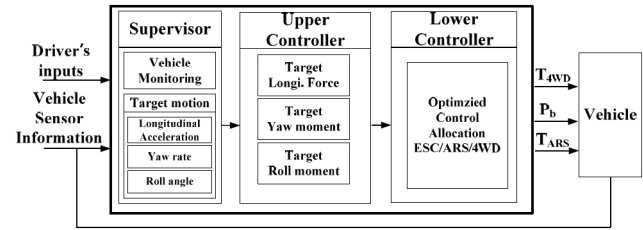


Fig. 3. Architecture of Integrated Chassis Control Algorithm

Supervisory Controller

Target Lateral Motions

In yaw rate control, the target yaw rate was determined using a 2-D bicycle model. Fig. 4 shows the 2-D bicycle model, including a direct yaw moment.

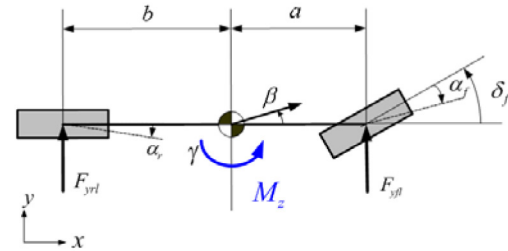


Fig. 4. 2-D bicycle model including direct yaw moment

The corresponding dynamic equations are as follows:

$$\begin{bmatrix} \dot{\beta} \\ \dot{\gamma} \end{bmatrix} = \begin{bmatrix} \frac{-2(C_f + C_r)}{mV_x} & \frac{2(-l_f C_f + l_r C_r)}{mV_x^2} - 1 \\ \frac{2(-l_f C_f + l_r C_r)}{I_z} & \frac{-2(l_f^2 C_f + l_r^2 C_r)}{I_z V_x} \end{bmatrix} \begin{bmatrix} \beta \\ \gamma \end{bmatrix} + \begin{bmatrix} \frac{2C_f}{mV_x} \\ \frac{2l_f C_f}{I_z} \end{bmatrix} \delta_f + \begin{bmatrix} 0 \\ \frac{1}{I_z} \end{bmatrix} M_z$$
(9)

$$a_y = V_x (\dot{\beta} + \gamma)$$
(10)

From (1), the cornering stiffness of the front tire for the case of neutral steer can be expressed as follows: