

Integrated Chassis Control for Improving On-Center Handling Behavior

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ABSTRACT

This paper proposes a new integrated chassis control (ICC) using a predictive model-based control (MPC) for optimal allocation of sub-chassis control systems where a predictive model has 6 Degree of Freedom (DoF) for rigid body dynamics. The 6 DoF predictive vehicle model consists of longitudinal, lateral, vertical, roll, pitch, and yaw motions while previous MPC research uses a 3 DoF maximally predictive model such as longitudinal, lateral and yaw motions. The sub-chassis control systems in this paper include four wheel individual braking torque control, four wheel individual driving torque control and four corner active suspension control. Intermediate control inputs for sub-chassis control systems are simplified as wheel slip ratio changes for driving and braking controls and vertical suspension force changes for an active suspension control. These control inputs are obtained by solving a quadratic optimization problem with constraints limiting vehicle motions and control inputs where a cost function represents tracking errors of vehicle target motions and control efficiency. Using simulation the control performance of the proposed ICC is evaluated in terms of on-center steering feel for a weave test where on-center steering feel is represented by lateral acceleration response to steering inputs. The proposed ICC shows the improvement of lateral response according to driver's intention, for example, driving torque control for acceleration, braking torque control for deceleration, or the combination of sub-chassis controls for aggressive driving.

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INTRODUCTION

As a driver steers a vehicle toward a desired direction vehicle motions provide feedback on the driver so that the driver adjusts steering inputs continuously to follow a desired path. At the moment of driving a vehicle steering feeling can be estimated by a driver based on direct perception of natural balance in vehicle response, steering angle and steering torque. Steering angle and steering torque are also appropriately realized by Motor Driven Power Steering (MDPS). Furthermore, additional chassis control systems will be useful to achieve desired vehicle dynamic responses which affect steering feel [1,2,3,4]. In order to enhance vehicle dynamics performance, a broad range of chassis control systems have been developed, for example, Electronic Stability Control (ESC), Active Front Steer (AFS), Active Suspension Control (ASC), Torque Vectoring Control (TVC) and other control systems. Moreover, integrated vehicle control has been extensively designed to achieve multiple-objective performance in terms of safety, comfort, reducing complexity of the control design, reducing system costs and providing design flexibility because of increasing numbers of individual control sub-systems [5,6].

In integrated chassis control, control plays a key role in vehicle control performance so that lots of research has been dedicated to an efficient design of control allocation, for

example, priority function based coordinator, artificial neural network based coordinator, fuzzy rule based coordinator, optimization control, and other controls. Particularly, control allocation using optimization control has been extensively developed to maximize control performance by adapting any constraints for vehicle states and chassis actuators [4,7,8].

This paper proposes a new integrated chassis control (ICC) using a predictive model-based control (MPC) for optimal allocation of sub-chassis control systems where the proposed ICC consists of three layers: (1) an upper layer for defining target vehicle motions, (2) an intermediate layer for the control allocation of sub-chassis controls, and (3) a lower layer for individual sub-chassis controls. In the intermediate layer, the predictive model has 6 Degree of Freedom (DoF) for rigid body dynamics. The 6 DoF predictive vehicle model consists of longitudinal, lateral, vertical, roll, pitch, and yaw motions while previous MPC research uses a 3 DoF maximally predictive model with longitudinal, lateral and yaw motions [9, 10]. The sub-chassis control systems in this paper include four wheel individual braking torque control, four wheel individual driving torque control and four corner active suspension control. Thus, the intermediate control inputs are simplified as wheel slip ratio change for driving and braking controls and vertical suspension force change for active suspension control. These control inputs are obtained by solving a quadratic optimization problem

with constraints limiting vehicle motions and control inputs where a cost function represents tracking errors of vehicle target motions and control efficiency.

Using simulation the control performance of the proposed ICC is evaluated in terms of on-center steering feel for a weave test (100kph, maximum steering input of 20 degrees at 0.3Hz on a dry road) where on-center steering feel is represented by lateral acceleration response to steering inputs. Several control strategies of the proposed ICC are investigated by adjusting constrain conditions in the MPC of the proposed ICC.

PROPOSED INTEGRATED CHASSIS CONTROL

The overall concept of proposed integrated chassis control (ICC) is briefly depicted in Figure 1 where the controller interacts with road conditions, vehicle dynamic response and driver's intention such as acceleration, deceleration and steering inputs. The proposed ICC comprises three layers: (1) an upper layer for defining target vehicle motions, (2) an intermediate layer for the control allocation of sub-chassis controls, and (3) a lower layer for individual sub-chassis controls.

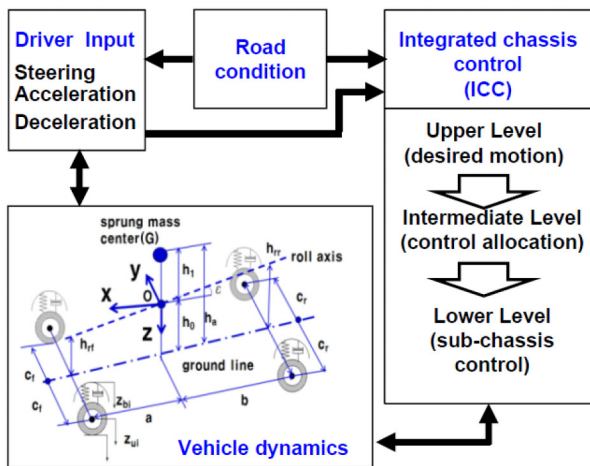


Figure 1. Concept of proposed integrated chassis control

The goal of the upper layer is to provide reference vehicle motions in order to maintain vehicle stability and to realize driver's intention with providing ride comfort. The target vehicle motions are represented by 6 DoF vehicle motions in terms of longitudinal speed for driver's acceleration and deceleration, side slip and roll angle for vehicle stability, yaw rate for steering input, vertical acceleration and pitch rate for ride comfort. These target vehicle motions are defined using conventional approaches such as a simple steady-state bicycle model to determine the target yaw motion, predefined thresholds to determine control activation, and other criteria.

In the intermediate layer, MPC determines intermediate control inputs of sub-chassis control systems to achieve target vehicle motions. The control inputs of sub-chassis control systems correspond to 3 directional forces at each wheel in the 6 DoF

predictive vehicle model, and are simplified as wheel slip ratio change for driving and braking controls and vertical suspension force change for active suspension control. These control inputs are obtained by solving a quadratic optimization problem with constraints limiting vehicle motions and control inputs where a cost function represents tracking errors of vehicle target motions and control efficiency.

Table 1. List of detail components of proposed ICC

(1) High Level (Desired motion)	
Longitudinal speed	$\Delta u_d = a_{xd} \cdot T_s$ $+ K_p \cdot (a_{xd} - a_x) \cdot T_s$
Yaw rate	$r_d = G_r \delta = \frac{V_x}{L + K_{us} V_x^2} \delta$
Lateral (side slip)	$\beta_d = 0$
Roll angle	$\phi_d = 0$
Pitch rate	$q_d = 0$
Vertical acceleration	$a_{zd} = 0$

(2) Intermediate Level (MPC)		
Input	Ref	$\Delta u_d \ r_d \ \beta_d \ \phi_d \ q_d \ a_{zd}$
	Measured or Estimated	*vehicle state : $x_0 : V_x V_y V_z \ p \ q \ r \ \phi \ \theta, z_{susp}$ *vehicle output : $y_0 : V_x \ r \ \beta \ q \ \phi \ a_z$
	Control input	$\Delta u_{k-1} : \Delta s x_i \ \Delta F_{zctr i}$
Predictive model	State	$x_k : V_x V_y V_z \ p \ q \ r \ \phi \ \theta, z_{susp}$
	Output	$y_k : V_x \ r \ \beta \ q \ \phi \ a_z$
Quadratic Programming (QP) problem	Cost function	$J = \frac{1}{2} (Y_d - Y)^T \overline{Q}_y (Y_d - Y)$ $+ \frac{1}{2} U^T \overline{Q}_u U + \frac{1}{2} \Delta U^T \overline{Q}_{\Delta u} \Delta U$
	Constraint condition	$braking : -0.2 < s_{x0} + \Delta s_x < 0$ $driving : 0 < s_{x0} + \Delta s_x < 0.2$ $F_{zctr} : F_{zctr MIN} < F_{zctr0} + \Delta F_{zctr}$ $F_{zctr0} + \Delta F_{zctr} < F_{zctr MAX}$
Output	Control input	$\Delta u_k : \Delta s x_i (Braking),$ $\Delta s x_i (driving), \Delta F_{zctr i}$
Prediction Horizon		$H_P = 5, H_U = 1$ (Prediction Time 0.25 sec)

(3) Low Level (Sub-chassis control)	
Input	Control input $\Delta s x_i (BT) \ \Delta s x_i (DT) \ \Delta F_{zctr i}$
Plant	1. Braking Torque (SMC) 2. Driving Torque (SMC) 3. Active suspension force (First order system)
Output	$BT_{ctr i} \ DT_{ctr i} \ F_{zctr i}$

where

$$r_d = \min \left(\left| r_d^* \right|, \left| \mu_H g / V_x \right| \right) \cdot \text{sign}(\delta), \quad a_{xd} = \min(a_{xd}, \mu_H g)$$

r_d : desired yaw rate

G_r : yaw rate gain

K_{us} : understeer gradient

u_d : desired vehicle speed

a_{xd} : desired longitudinal acceleration

T_s : sample time

K_p : Proportional Gain

a_x : longitudinal acceleration



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In the lower layer intermediate control inputs are achieved by individual sub chassis control systems such as four wheel individual braking torque control, four wheel individual driving torque control and four corner individual active suspension control. The braking torque control and driving torque control can adjust longitudinal tire forces at each wheel, and is realized using Sliding Mode Control (SMC) on a wheel slip control. The active suspension control can directly provide vertical reaction force on a rigid body against pitch and roll motions, and is considered as a first order system for simplicity. Moreover, an active steering control as one of sub chassis control systems can be easily implemented by considering the change of steering input in the intermediate layer. In this paper, a quadratic optimization problem is particularly formulated using Linear Time-Varying (LTV) MPC and can be solved using standard QP solvers such as MATLAB's optimization toolbox function, **quadprog** [12]. Furthermore, Table 1 shows the summary of control inputs and outputs of the proposed ICC, and more details of the MPC design are described more fully in reference [9], [10], and [11].

SIMULATION ANALYSIS

Vehicle Model

A simulation analysis is performed using a 14 DoF nonlinear vehicle model implemented in Matlab/Simulink where the nonlinear vehicle model consists of 6 DoF for rigid body dynamics, 4 DoF for wheel rotation and 4 DoF for vertical wheel movement. The vehicle rigid body has longitudinal, lateral, vertical, pitch, yaw and roll degrees freedom.

Table 2. Inputs, outputs, and states in vehicle dynamics

Input	*driver input : $\delta, a_x, \text{acceleration}, a_x, \text{deceleration}$ *tire forces : F_{xi}, F_{yi} *active suspension force : $F_{z_i} (i = fl, fr, rl, rr)$
Plant	*states : $V_x, V_y, V_z, p, q, r, \phi, \theta, z_{1i}, z_{2i}, z_{3i}$ *first order system : tire force
Output	*states : x *side slip : β *vehicle acceleration : a_x, a_y, a_z

where

δ : Hand wheel input angle

a_{xd} : acceleration / deceleration input

F_{xi} : longitudinal tire force

F_{yi} : lateral tire force

F_{z_i} : vertical tire force

$F_{zctr i}$: active suspension force

V_x, V_y, V_z : longitudinal / lateral / vertical velocity

p, q, r : roll rate / pitch rate / yaw rate

ϕ, θ : roll angle / pitch angle

z_{1i} : deflection of m_{us}

z_{2i} : suspension deflection

z_{3i} : vertical velocity of m_{us}

m_{us} : unsprung mass

The wheel dynamics includes brake torque, driving torque and a longitudinal tire force at the contact point on the ground. The longitudinal and lateral tire forces are represented by the nonlinear combined slip tire model based on Pacejka's Magic formula, a similarity model, as functions of tire longitudinal slip ratio, tire lateral slip ratio, a vertical load and surface friction coefficient [12, 13, 14]. Moreover, the vertical wheel movement responds to spring force, damping force, anti-roll bar reaction force and tire force. Table 2 shows inputs, outputs, and states in vehicle dynamics, particular vehicle parameters for simulation are based on a mid-size sedan.

Simulation Results

In this paper the enhancement of on-center steering feel is supposed to be proportional to vehicle lateral response so that the analysis of a control performance can focus on vehicle motions even though a driver may estimate steering feel through steering torque, steering angle, and lateral acceleration at the same time. In other words, vehicle response to steering angle inputs can be considered separately from a steering system because the response of steering torque to steering angle can be controlled independently by a steering system. As shown in Figure 2, a weave test for an on-center maneuver is used with an initial vehicle speed of 100kph and a steering input of 20 degrees at 0.3Hz on a dry road. During the weave test the maximum lateral acceleration is around 0.3g so that the proposed ICC may apply the yaw motion control of the upper layer to improve lateral response.

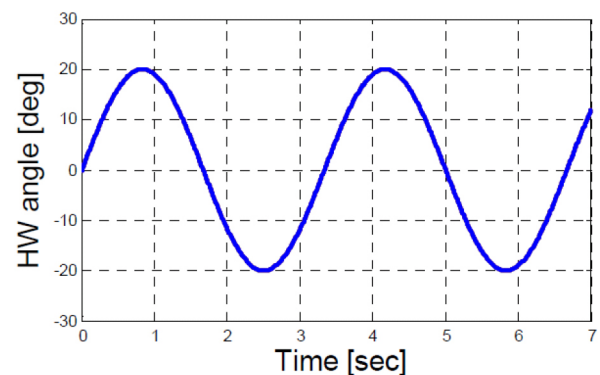


Figure 2. Hand wheel (HW) steering input of a weave test (HW angle 20 degrees at 0.3Hz)

The control strategies of the proposed ICC and driving scenarios of are as follows: 1) free rolling with four corner individual active suspension control (ASC), 2) acceleration with four wheel individual torque vectoring control (TVC), 3) deceleration with four wheel individual braking torque control (BTC), 4) acceleration with TVC and ASC, and 5) aggressive driving with TVC and BTC. These driving scenarios will be compared with free rolling response without control (No control). Each control mode is adapted in the proposed ICC by adjusting constrain conditions in MPC, for example, no change of negative slip ratio will deactivate BTC, while no change of positive slip ratio will limit TVC activation. Moreover, the change of active suspension force is engaged in ASC activation.