

Coordination of Steer Angles, Tyre Inflation Pressure, Brake and Drive Torques for Vehicle Dynamics Control

Barys Shyrokau and Danwei Wang
Nanyang Technological University

ABSTRACT

During vehicle operation, the control objectives of stability, handling, energy consumption and comfort have different priorities, which are determined by road conditions and driver behavior. To achieve better operation characteristics of vehicle, coordinated control of vehicle subsystems is actively used. The fact of more active vehicle subsystems in a modern passenger car provides more flexibility for vehicle control and control algorithm development. Since the modern vehicle can be considered as over-actuated system, control allocation is an effective control technique to solve such kind of problem.

This paper describes coordination of frictional brake system, individual-wheel drive electric motors, active front and rear steering, active camber mechanisms and tyre pressure control system. To coordinate vehicle subsystems, optimization-based control allocation with dynamic weights is applied. The influence of different weights (subsystem restriction) on criteria of vehicle dynamics (RMSE of yaw rate, sideslip angle, dynamic tyre load factor) and energy consumption and losses (consumed/recuperated energy during maneuver, longitudinal velocity decline, tyre energy dissipation) were analyzed. Based on this analysis, the optimal solution was selected. The proposed control strategy is based on the switching between optimal criteria related to vehicle safety and energy efficiency during vehicle motion. Dynamic weights were utilized to achieve this switching.

The simulation-based analysis and evaluation of both variants was carried out using a nonlinear vehicle model with detailed models of actuators. The test maneuver is 'Sine with Dwell'. Both variants of control allocation guarantees vehicle stability of motion and good handling. Meanwhile, proposed variant demonstrates slightly higher longitudinal velocity at the end of maneuver and higher amount of recuperated energy up to 15%; however, tyre dissipation energy increased to 5% compared to optimal solution from simulation-based analysis.

CITATION: Shyrokau, B. and Wang, D., "Coordination of Steer Angles, Tyre Inflation Pressure, Brake and Drive Torques for Vehicle Dynamics Control," *SAE Int. J. Passeng. Cars - Mech. Syst.* 6(1):2013, doi:10.4271/2013-01-0712.

INTRODUCTION

Increasing requirements to vehicle safety, performance and environmental friendliness dictate a development of more complex vehicle subsystems to achieve better operation characteristics. Thus, the amount of active subsystems in modern vehicle is increased, and their integrated / coordinated control becomes more significant year by year.

Main stages of integrated/coordinated control are depicted in [Figure 1](#). The control demand with generalized vehicle forces and torques is obtained from high-level controller of vehicle dynamics control. Next stage is a distribution of control demand between four wheels, taking into account e.g. force capacity of each wheel, normal distribution, power minimization and other objectives. Some solutions for force allocation were proposed [1,2,3]. After force distribution, the stage of subsystem coordination is carried out. Control inputs

of vehicle actuators are shown in [Figure 1](#) into outer ring. Moreover, there are some hypothetical subsystems such as regulation of tyre contact temperature, changing of down force using aerodynamic spoilers, and others. Thereby, the control demand from high-level controller can be realized by many ways according to control tasks in current conditions of vehicle motion. One of most common control technique to distribute control demand for over-actuated system is control allocation. A detailed state-of-the-art of control allocation methods and their application for various fields is represented in [4].

For integrated vehicle control a type of allocation parameters depends on actuator control inputs, desired cost functions, inverse tyre model, number of active subsystems and other factors. Below there are some examples of configurations of allocation parameters:

- longitudinal and lateral tyre forces [5-6]. Only the problem of force allocation is being solved;
- kinematic parameters of the tyre (longitudinal slip and wheel slip angles). Based on tyre inversion model, reference tyre kinematic parameters can be obtained for force allocation [7,8];
- reference control inputs of actuators (brake pressure / torque, drive torque, steering angle) [9]. Both tasks, force allocation and subsystem coordination, can be solved jointly;
- mixed force/actuator output, for instance, longitudinal forces and steering angle [10].

Some of research issues related to control allocation are shown in Figure 2. There are representation of actuator dynamics, approximation of nonlinear allocation, adaptation of control allocation to uncertainties and disturbances, achievement of several objectives. On the right side, some solutions for abovementioned problems are shown.

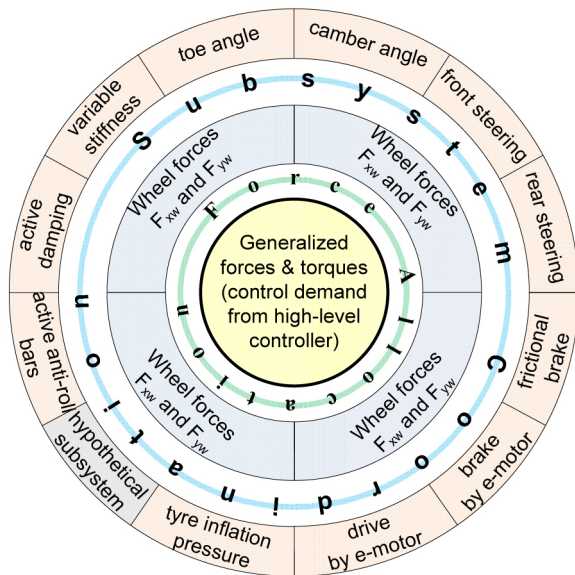


Figure 1. Control demand distribution

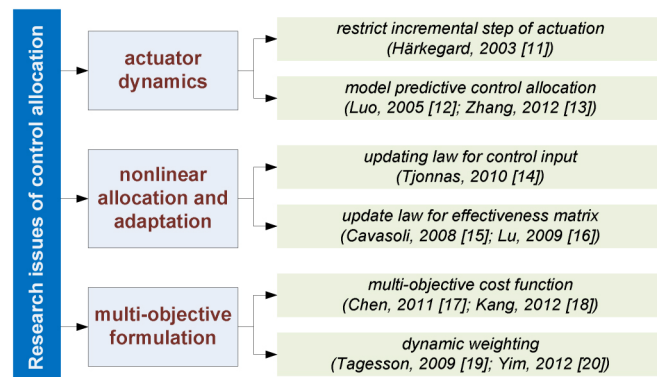


Figure 2. Research issues of control allocation

Multi-objective formulation is the most significant issue in the case of multi-actuators, because a set of control tasks can be considered. For instance, Chen and Wang [17] added power consumption (related to power consumption of all in-wheel motors in different modes) into the cost function, and called the method as “energy-efficient control allocation”. Kang et al. [18] proposed to add into the cost function not only energy consumption, but also energy loss (slip control). The drawbacks of the multi-objective cost function are that it could not be convex; the solution is optimal for all evaluation criteria with their pre-defined weighed coefficients; computational budget can increase as a result of smoothing of the function due to additional cost terms.

Another way to provide coordination between wheels/subsystems for several control tasks is dynamic weighting. Tagesson et al. [19] used a weighting matrix to coordinate brake pressures according to normal force distribution during heavy vehicle braking to obtain better vehicle stability. Yim et al. [20] proposed variable weighting to coordinate longitudinal and lateral forces developed by vehicle subsystems taking into account tyre slip and slip angles. Shyrokau and Wang [21] show that control allocation with dynamic weight scheduling demonstrates lower energy loss without significant impairment of stability of motion and vehicle handling compared to control allocation with fixed weight distribution.

The first aim of this paper is to investigate how vehicle subsystem coordination influences on vehicle operation characteristics. Based on this analysis, the optimal configuration of subsystem weights will be selected. The second aim is to investigate and demonstrate an effect of dynamic weighting compared with optimal solution from the first aim. The configuration of active vehicle subsystems includes active front steering, active rear steering, individual-wheel drive electric motors, electro-hydraulic brake system, active camber mechanisms and tyre inflation pressure system.

The paper is organized as follows: a brief description of vehicle model, which is used for analysis and evaluation, is presented in next section. The control system structure and control laws are discussed in Section 3; analysis of subsystem coordination is in Section 4; proposed strategy of dynamic weights is in Section 5. Thereupon simulation results are shown in Section 6. Conclusions and future work is discussed in Section 7.

VEHICLE MODEL

To analyze and evaluate a control system, original vehicle model for mid-sized passenger car was developed. The detailed mathematical description of vehicle model is described in [22]. The model has 14 DoF (schematically shown in Figure 3).

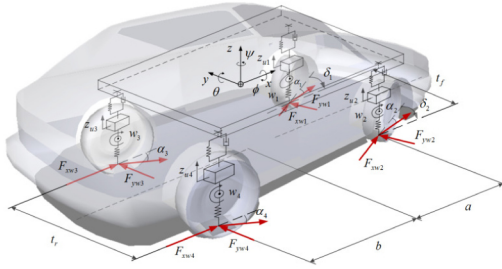


Figure 3. Vehicle model.

The dynamics of steering system is described by:

- front and rear steering mechanisms by the second-order transfer functions;
- electric motor of AFS by the first-order transfer function.

Steering kinematics is represented by Ackerman model. The dynamic effects related to roll and compliance steer are added. Active camber mechanism is described by the first-order transfer function.

The model of the electric driveline includes Thevenin battery model and four electric motors, which are represented by look-up tables with data from a real motor. The transient processes in the electric motors are described by the first-order plus time delay transfer function. The half-shaft torsion dynamics is neglected.

The brake system consists of a tandem master cylinder with a spring pedal travel simulator, a brake fluid reservoir, a hydraulic pump with a pump motor, a high-pressure hydraulic accumulator, block valves, compensating valves, control inlet/outlet valves for pressure build-up and decrease at each wheel, and brake mechanisms. EHB system is described by a system of differential equations and it includes valve dynamics and dynamic friction coefficient.

To characterize the steady-state tyre behavior, taking into account tyre inflation pressure, the extended Pacejka tyre model is used [23]. For transient tyre behavior, a relaxation length model is added.

Tyre inflation pressure system includes relay valves and pneumatic compressor. However, in this paper, system dynamics are described only by the first-order transfer function.

The developed vehicle model is realized in Matlab/Simulink. The verification of the developed model is carried out by the comparison of the developed vehicle behavior with the multibody vehicle model in the commercial software under vehicle motion without control. The simulation results of 'Sine with Dwell' test at initial velocity of 80 km/h and maximal amplitude of steering wheel angle of 100, 140 and 160 deg are shown in Figure 4. The developed model demonstrates close behavior to multibody model.

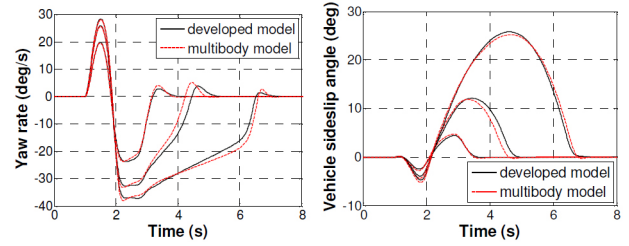


Figure 4. Vehicle model verification.

INTEGRATED CONTROL SYSTEM STRUCTURE

Control system, depicted in Figure 5, includes state observer, high-level controller for vehicle dynamics, middle-level block for optimization-based control allocation and lower-level individual controllers for each subsystem. The actuator control inputs are sent to full vehicle model with active subsystems, described in previous section.

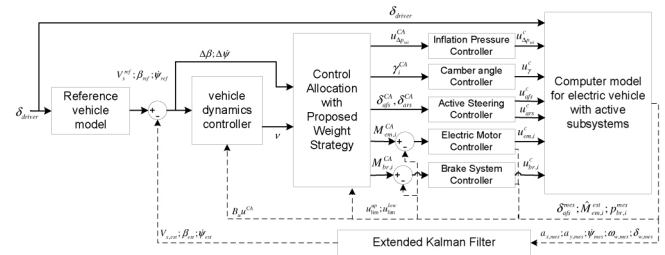


Figure 5. Vehicle model verification.

Vehicle Dynamics and Actuators Control

The high-level control consists of reference vehicle model and vehicle dynamics controller. Generalized longitudinal and lateral forces and yaw torque are determined according to control errors. The reference longitudinal acceleration is calculated based on the measured pedal force [10]. For lateral dynamics, the reference vehicle model is based on classical bicycle vehicle model [24]. The reference yaw rate is constrained according to adhesion limits [25]. The reference sideslip angle is equal to zero.

The control demand is founded using PI controller with anti-windup based on back calculation [9]:

$$\begin{cases} v_i = K_p e_i + \int \left(K_i e_i + \frac{1}{T_i} (B_u u^{CA} - v_{i-1}) \right) dt \\ e_i = \begin{cases} a_x^{ref} - a_x^{mes}, & |a_x^{ref} - a_x^{mes}| > 0.1 \text{ ms}^{-2} \\ -\beta_{mes}, & |\beta_{mes}| > 0.25^0 \\ \dot{\psi}_{ref} - \dot{\psi}_{mes}, & |\dot{\psi}_{ref} - \dot{\psi}_{mes}| > 0.5^0 \text{ s}^{-1} \\ 0, & \text{otherwise} \end{cases} \end{cases}$$

Proportional and integration gains are taken according to recommendations from [9]:

$$K_i = 20 \text{diag}([0.2m_a \quad 0.7m_a \quad 1.5I_{zz}])$$

$$K_p = \frac{2}{3}m_a \sqrt{K_i \text{diag}([16/m_a \quad 1/m_a \quad 1/I_{zz}])}$$
(2)

Actuator control should achieve the following targets:

- to guarantee precise tracking to reference control signals obtained from the middle level;
- to estimate boundary conditions for control allocation taking into account subsystem behavior and skid/slip control.

The control signals for front and rear steering, camber mechanisms and tyre inflation pressure system are additional steer angles, camber angles and tyre inflation pressures from middle-level control allocation accordingly.

Assuming that real torque of electric motor is precisely estimated, PWM signals are founded using PI motor control:

$$\begin{cases} u_{em,i}^c = K_p^{drive} e_{em,i} + K_i^{drive} \int e_{em,i} dt \\ e_{em,i} = M_{em,i}^{CA} - \hat{M}_{em,i}^{est} \end{cases}$$
(3)

Control signals for inlet and outlet valves of frictional EHB system are formulated as:

$$\begin{cases} u_{br,i}^c = K_p^{brake} e_{br,i} + K_i^{brake} \int e_{br,i} dt + K_d^{brake} \frac{de_{br,i}}{dt}; \\ e_{br,i} = p_{br,i}^{ref} - p_{br,i}^{mes} \end{cases}$$

$$u_{i,ideal}^{inlet} = \begin{cases} 1, & e_{br,i} \geq 0 \& |u_{br,i}^c| \geq 1 \\ |u_{br,i}^c|, & e_{br,i} \geq 0 \& |u_{br,i}^c| < 1 \\ 0, & e_{br,i} < 0 \end{cases}$$

$$u_{i,ideal}^{outlet} = \begin{cases} 0, & e_{br,i} \geq 0 \\ |u_{br,i}^c|, & e_{br,i} < 0 \& |u_{br,i}^c| < 1 \\ 1, & e_{br,i} < 0 \& |u_{br,i}^c| \geq 1 \end{cases}$$
(4)

The position limits of electric motors and brake system are varied to reach wheel skid/slip minimization and to correct limits according to subsystem behavior. For instance, torque limit of electric motor is corrected by weighed coefficients, which are related to electric motor temperature, state-of-the-charge, longitudinal vehicle velocity, under and overvoltage protection and fault control [22]:

$$M_i^{em,lim} = f(u_{em,i}^c, w_m) \prod w_i(T_i, SOC, V_{xa}, U_i, fault)$$
(5)

The constructive parameters of brake mechanism and the level of wheel slip restrict the torque limit of frictional

brakes. The correction taking into account wheel slip is carried out based on the threshold of wheel slip. The algorithm is rule-based and similar to control of anti-lock brake system [26].

Optimization-Based Control Allocation

The relationship between generalized forces and yaw torque from control demand and control inputs is nonlinear. To approximate this relationship into linear case, the feedback linearization by cancelling of nonlinear term is used [10]:

$$v - f(x) = v^* = \frac{\partial g(x, u)}{\partial u} u \approx B_u u$$
(6)

Tyre forces can be written in the following form:

$$F_{xwi} = \underbrace{\varepsilon_{Fxi}}_{\text{system}} + \underbrace{\frac{1}{\eta_{Fxi}} T_i^{em} + \frac{1}{\eta_{Fxi}} T_i^{br}}_{\text{actuators}} + \tilde{C}_{p_{ws}} \Delta p_{wi}$$

$$F_{ywi} = \underbrace{\tilde{C}_{ywi} \left(\tan \frac{V_{yi}}{V_{xi}} - \delta_{wi}^{w/o corr} \right)}_{\text{system}} + \underbrace{\tilde{C}_{ywi} \delta_{afs} + \tilde{C}_{ywi} \delta_{ars} + \tilde{C}_{\gamma_i} \gamma_i + \tilde{C}_{p_{wy}} \Delta p_{wi}}_{\text{actuators}}$$
(7)

Vehicle velocities projections V_{xi} and V_{yi} are calculated based on information from state observer. The wheel steer angles without corrections from active front and rear steering systems are:

$$\delta_{wi}^{w/o corr} = \delta_{wi} - [\delta_{afs} \quad \delta_{ars} \quad \delta_{ars} \quad \delta_{ars}]^T$$
(8)

To describe wheel dynamics in braking mode, the coefficients ε_{Fxi} and η_{Fxi} , are founded as [27]:

$$\eta_{Fxi} = \frac{J_w V_{xw}}{\tilde{C}_{xwi} r_w t_s}; \quad \varepsilon_{Fxi} = \left(1 - \frac{r_w}{\eta_{Fxi}} \right) \hat{F}_{xwi}$$
(9)

Since the maximal drive torque is lower than brake one, in driving mode wheel dynamics is neglected.

Tyre longitudinal stiffness $\tilde{C}_{xwi}$, cornering stiffness $\tilde{C}_{ywi}$, camber stiffness $\tilde{C}_{\gamma_i}$ and coefficient $\tilde{C}_{p_{ws}}$, related to changing of longitudinal force depends on inflation pressure, are numerically computed as partial derivatives of longitudinal and lateral tyre forces from Pacejka model by Taylor series explanation with sampling time of 1ms.

To find nonlinear term $f(x)$, only tyre forces related to system behavior is used. Tyre projections along the vehicle axes F_{xi}^{sys} and F_{yi}^{sys} are related to tyre forces along wheel axes: