Influence of active subsystems on electric vehicle behavior and energy characteristics

Shyrokau, Barys; Savitski, Dzmitry; Wang, Danwei

2014

Shyrokau, B., Savitski, D., & Wang, D. (2014). Influence of active subsystems on electric vehicle behavior and energy characteristics. SAE Technical Papers, 2014-01-0876-. doi: 10.4271/2014-01-0876

<https://hdl.handle.net/10356/106006>

<https://doi.org/10.4271/2014-01-0876>

© 2014 SAE International. This paper was published in SAE Technical Papers and is made available as an electronic reprint (preprint) with permission of SAE International. The published version is available at: [<http://dx.doi.org/10.4271/2014-01-0876>]. One print or electronic copy may be made for personal use only. Systematic or multiple reproduction, distribution to multiple locations via electronic or other means, duplication of any material in this paper for a fee or for commercial purposes, or modification of the content of the paper is prohibited and is subject to penalties under law.

Downloaded on 26 Apr 2023 21:51:41 SGT

Influence of Active Subsystems on Electric Vehicle Behavior and Energy Characteristics

2014-01-0876
Published 04/01/2014

Barys Shyrokau

Nanyang Technological University

Dzmitry Savitski

Ilmenau University of Technology

Danwei Wang

Nanyang Technological University

CITATION: Shyrokau, B., Savitski, D., and Wang, D., "Influence of Active Subsystems on Electric Vehicle Behavior and Energy Characteristics," SAE Technical Paper 2014-01-0876, 2014, doi:10.4271/2014-01-0876.

Copyright © 2014 SAE International

Abstract

Nowadays there is a tendency to implement various active vehicle subsystems in a modern vehicle to improve its stability of motion, handling, comfort and other operation characteristics. Since each vehicle subsystem has own limits to generate supporting demand, their potential impact on vehicle dynamics should be analyzed for steady-state and transient vehicle behavior. Moreover, the additional research issue is the assessment of total energy consumption and energy losses, because a stand-alone operation of each vehicle subsystem will provide different impact on vehicle dynamics and they have own energy demands.

The vehicle configuration includes (i) friction brake system, (ii) individual-wheel drive electric motors, (iii) wheel steer actuators, (iv) camber angle actuators, (v) dynamic tire pressure system and (vi) actuators generating additional normal forces through external spring, damping and stabilizer forces. A passenger car is investigated using commercial software. The actuator models are defined using experimental test results and technical literature information.

The selected open-loop maneuvers cover steady-state and transient vehicle behavior. Slowly Steer Increasing maneuver demonstrated that all considered subsystems have notable influence on the steering characteristic of the vehicle in the transient area. For normal operation conditions, Step Steer maneuver in time-based domain and Sine Sweep maneuver in frequency domain are also investigated. For steady-state area, the cases of equivalent impact of each vehicle subsystem on the vehicle dynamics are defined. In such conditions the influence of stand-alone operation of vehicle subsystem on total energy consumption and energy losses was demonstrated for different steering frequencies.

Introduction

Active vehicle subsystems help to improve vehicle handling and stability performance. Several authors noted [1,2] that integration of various vehicle subsystems provides better vehicle motion characteristics and energy consumption compared to their stand-alone operation. Nevertheless, to develop integrated control between several vehicle subsystems, their individual effect on vehicle dynamics during stand-alone operation should be investigated. In the last two decades many investigations of influence of vehicle subsystems have been done using different control techniques taking into account own subsystem capacity to realize supporting demand. We refer the reader to the publications [2, 3, 4, 5] for some examples of such kind of investigations.

In the same time, the each actuator has own energy consumption and influence on the tire-road interaction. The analysis of energy characteristics such as total energy consumption and energy losses under stand-alone operation of vehicle subsystem is still weak-investigated.

Based on abovementioned fact, the research aim of this paper is to investigate performance of stand-alone operation of several vehicle subsystems and their influence on energy consumption and tire energy dissipation. The second aim is to analyze which vehicle subsystem has more potential to achieve a target behavior of a reference model with *a priori* better handling performance compared to the investigated vehicle without control.

The paper is organized as follows: models of vehicle and its subsystems are presented in next section. The control system structure and control laws are discussed in Section 'Description of Control System'. Thereupon investigated maneuvers and

simulation results are shown in Section 'Discussion of Simulation Results'. Conclusions and future work is discussed in Section 'Conclusions'.

Models of Vehicle and Subsystems

The investigated vehicle is a passenger car with a mass of 1534 kg, weight front/rear distribution 55/45 and tire size 205/60R15.

It should be noted, that the following assumptions, applicable conditions and limitations of the vehicle and subsystems models were taken into account for the proposed paper:

- it is assumed that the measured signals from vehicle sensors are available and the measured noise can be ignored;
- the applicable conditions are related to the motion on the flat road without external disturbances like wind, fluctuation of road friction coefficient and others. Since the maneuvers based on ISO standards will be investigated, the following assumption is reasonable;
- dynamics of the actuators can be simplified and, in some cases, it can be described by transfer functions using experimental investigations or the recommendations provided by other researchers;
- the energy consumption models of actuators are simplified for the rough assessment of energy characteristics and they can be detailed for more precise analysis.

The main features of simulation setup based on commercial software IPG CarMaker and Matlab/Simulink are concluded below:

- vehicle body - rigid model taking into account mass distribution between wheel carriers, wheels, and body;
- brake system - electro-hydraulic friction brake system;
- powertrain - electric powertrain includes a battery and near-wheel electric motors with transmission;
- steering - wheel steer actuators generating driver effort and external steer angles, which are added to steer angles generated by the IPG CarMaker;
- suspension - 2 DOF model of kinematics and actuators generating additional normal forces through external suspension forces (spring, damper and stabilizer forces). These forces are added to the forces generated by the IPG CarMaker in models for spring, damper and stabilizer;
- camber angle actuators - external camber angles are added to camber angles generated by the IPG CarMaker;
- tires - Pacejka tire model including tire inflation/deflation pressures with relaxation model [6].

The electro-hydraulic brake system consists of a tandem master cylinder with spring pedal travel simulator, brake fluid reservoir, block valve, compensative valve, hydraulic pump with electric motor, high-pressure hydraulic accumulator, control inlet/outlet proportional valves for pressure build-up and decrease at each wheel, and brake mechanisms. The effects related to fluid inertia and brake line are neglected. Pressures in the calipers versus volume change are taken from

experimental data as look-up tables. The detailed mathematical model of the electro-hydraulic brake system is represented in [7]. The current level of pump is 3 A, inlet valves of electro-hydraulic unit consume 1.2 A and outlet ones - 2.4 A under maximal load.

An electric powertrain includes a battery and near-wheel electric motors with transmission. The dynamics of electric motor is represented by a look-up table of real motor characteristics for steady-state behavior [7] and the first-order transfer function with time delay for transient behavior. The part-load characteristics are defined by the level of the control signal obtained from the controller of the electric motor. The transient behavior of the electric motor is represented as [8]:

$$\frac{T_{em,i}}{T_{em,i}^{ss}} = \frac{1}{0.0022s + 1} e^{-0.002s} \quad (1)$$

To transmit output torque $T_{em,i}$ from electric motor to the wheel, near-wheel electric motors with transmission and half-shafts are used. The transmission dynamics is described as [8]:

$$\left(J_{em} + J_{tr1} + \frac{u_{tr}^2}{\eta_{tr}} \left(J_{tr2} + \frac{0.5J_{hs}}{\eta_{joint}^{inner}} \right) \right) \ddot{\theta}_{em,i} = T_{em,i} - \frac{u_{tr}}{\eta_{tr}} T_{hs,i} \quad (2)$$

The half-shaft torques $T_{hs,i}$ and wheel torque T_{wi} are calculated as follows [8]:

$$\begin{aligned} T_{hs,i} &= K_{hs} (u_{tr} \omega_{em,i} - \omega_{wi}) + C_{hs} (u_{tr} \theta_{em,i} - \theta_{wi}) \\ 0.5J_{hs} \ddot{\theta}_{wi} &= T_{hs,i} - \frac{T_{wi}}{\eta_{joint}^{outer}} \end{aligned} \quad (3)$$

Energy consumption by electric motors is calculated as potential energy, assuming there are no rate constraints of battery in the case of recuperation:

$$P_{em} = \sum_{i=1}^4 T_{em,i} \omega_{em,i}; \quad E_{em}^{total} = \int_0^T P_{em} dt \quad (4)$$

The external steer angles (IWS) generated by wheel steer actuators are described by the second-order transfer function with time delay:

$$\frac{\delta_i^{ext}}{\delta_i^{ref}} = \frac{\omega_{n,steer}^2}{s^2 + 2\xi_{steer} \omega_{n,steer} s + \omega_{n,steer}^2} e^{-\tau_{d,steer} s} \quad (5)$$

where $\omega_{n,steer}$ - natural frequency (15 Hz), ξ_{steer} - damping coefficient (0.707) and $\tau_{d,steer}$ - time delay (3 ms).

The transfer function of camber angle actuator (ACS) is selected on the basis of concept design as:

$$\frac{\gamma_i^{ext}}{\gamma_i^{ref}} = \frac{1}{0.15s + 1} e^{-0.005s} \quad (6)$$

The energy consumption of wheel steer and camber angle actuators can be calculated as requested energy to overcome inertia and road resistance plus additional energy for the compensation of internal power loss:

$$P_{IWS} = \sum_{i=1}^4 M_{zwi} \delta_i^{ext}; E_{IWS} = 1.3 \int_0^T P_{IWS} dt;$$

$$P_{ACS} = \sum_{i=1}^4 M_{\phi wi} \gamma_i^{ext}; E_{ACS} = 1.3 \int_0^T P_{ACS} dt$$
(7)

Dynamic tire pressure system controls pressure in each tire individually. To obtain a transfer function of dynamic tire pressure system, the experimental results were used [9]. The updated hardware system setup is used to reach pressure changing rate up to 1.5 bar/s. The system behavior is represented as:

$$\frac{p_w^{ext}}{p_w^{ref}} = \frac{1}{0.371s + 1} e^{-0.036s}$$
(8)

The current level of inlet/outlet valves of dynamic tire pressure system is 1.2 A and voltage is 12 V. The power of air compressor is 50 W.

To realize external normal forces, the variable stiffness suspension, electrorheological dampers and active anti-roll bars are used. The dynamics of variable stiffness suspension is described as [10]:

$$\frac{F_{spring,i}^{ext}}{F_{spring,i}^{ref}} = \frac{1}{0.03s + 1} e^{-0.004s}$$
(9)

The dynamics of active damping system is formulated as [11]:

$$\frac{F_{damp,i}^{ext}}{F_{damp,i}^{ref}} = \frac{1}{0.005s + 1} e^{-0.002s}$$
(10)

The dynamics of active anti-roll actuators is represented by [12]:

$$\frac{F_{arb,i}^{ext}}{F_{arb,i}^{ref}} = \frac{1}{0.1s + 1} e^{-0.006s}$$
(11)

The electric power consumption of active dampers is below 100 W for the entire vehicle [13]. For active suspension average power consumption is less than 1.0 kW [14]. The maximal current of actuator of electromechanical anti-roll bar is 40 A under voltage of 17 V [15].

Tire energy dissipation in longitudinal and lateral directions is calculated as:

$$E_{tire}^{total} = \sum_{i=1}^4 \int_0^T (F_{xwi} V_{swxi} + F_{ywi} V_{swyi}) dt$$
(12)

Description of Control System

The architecture of control system is presented in Figure 1 and includes reference generator, vehicle dynamics controller, control allocation, individual actuator control and state observer.

Figure 1. Architecture of control system

Reference Generator

Steering angle from the driver is the input parameter to the reference vehicle model. To describe the reference curvilinear behavior, the reference model can be described by the first-order or higher-order transfer functions. One of the common methods is the application of bicycle model. In the proposed work according to [16], the bicycle model is considered taking into account tire relaxation lengths in lateral direction. In this case, the reference yaw rate is formulated as:

$$\begin{bmatrix} \ddot{\psi} \\ \dot{\beta} \\ \dot{F}_y^f \\ \dot{F}_y^r \end{bmatrix} = \begin{bmatrix} 0 & 0 & \frac{a}{I_{zz}} & -\frac{b}{I_{zz}} \\ -1 & 0 & \frac{1}{m_a V_{xa}} & \frac{1}{m_a V_{xa}} \\ -\frac{C_f a}{L_{yf}} - \frac{C_f V_{xa}}{L_{yf}} & -\frac{V_{xa}}{L_{yf}} & 0 & 0 \\ \frac{C_r b}{L_{yr}} - \frac{C_r V_{xa}}{L_{yr}} & 0 & -\frac{V_{xa}}{L_{yr}} & 0 \end{bmatrix} \begin{bmatrix} \psi \\ \beta \\ F_y^f \\ F_y^r \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ \frac{C_f V_{xa}}{L_{yf}} \\ 0 \end{bmatrix} \delta_f$$
(13)

To obtain more agile vehicle behavior, the constructive parameters of target model were changed comparing to the original model. Thereby, the application of active subsystems should ensure behavior of target vehicle, which has to achieve more sportive characteristics. The yaw rate response of several reference models is shown in Figure 2.

The reference yaw rate is constrained according to the friction limits [17]:

$$\dot{\psi}_{ref} = \begin{cases} \dot{\psi}, & |\dot{\psi}| \leq |\dot{\psi}_{max}| \\ \pm \dot{\psi}_{max}, & \text{otherwise} \end{cases} \quad \dot{\psi}_{max} = \frac{\mu_{road} g}{V_{xa}}$$
(14)

Figure 2. Behavior of reference model

Vehicle Dynamics Controller

The vehicle dynamics controller compensates a difference between reference and actual vehicle behavior to guarantee yaw rate tracking. The selected controller is proportional one with resetting back-calculation to take into account rate limitations of actuators:

$$\dot{v}_{yaw} = K_p \Delta \dot{\psi} + \frac{1}{T_i} \int (v_{yaw}^{act} - v_{yaw}) dt \quad (15)$$

The realized demand v_{yaw}^{act} by certain subsystem is calculated based on actuator output signals. When yaw rate error is crossing a zero, the integrator is reset.

Control Allocation

The control demand V_{yaw} is disturbed between vehicle subsystems and wheels converting to reference actuator signals in control allocation. Since stand-alone operation is considered, the task of demand distribution between vehicle subsystems is omitted. More details regarding this issue is presented in [9]. The control allocation problem is formulated as a minimization of allocation error and control actuations, taking into account actuator constraints:

$$u^{CA} = \arg \min_{u_{lim}^{low} \leq u^{CA} \leq u_{lim}^{up}} \left(\|W_v (B_u u^{CA} - v^*)\|_2^2 + \zeta \|W_u u^{CA}\|_2^2 \right) \quad (16)$$

The parameter ζ defines a significance of minimization of control actuations and is equal to 0.1. Matrix W_v is defined a priority between generalized longitudinal force and yaw torque, because in further considered maneuvers the velocity support is required. A weighting matrix W_u can be used to prioritize actuators. The fixed-point method is selected as optimization solver. It is terminated when the allocation error is lower than predefined tolerance.

Due to consideration of stand-alone operation, the control effectiveness matrix B_u is equal to control effectiveness matrix of considered subsystem:

$$B_u = \{B_\delta \parallel B_{em} \parallel B_{br} \parallel B_\gamma \parallel B_{p_w} \parallel B_{R_z}\} \quad (17)$$

The similar approach is used for the control input vector:

$$u^{CA} = \{\delta_i^{CA} \parallel T_{em,i}^{CA} \parallel T_{br,i}^{CA} \parallel \gamma_i^{CA} \parallel p_{wi}^{CA} \parallel R_{zi}^{CA}\} \quad (18)$$

The presented control effectiveness matrices are formulated only for yaw rate. The full description of these matrices is presented in [9]. The control effectiveness matrix B_δ for external wheel steering angles δ_i^{CA} can be formulated as:

$$B_\delta = \begin{bmatrix} C_{ywf} (-0.5t_f \sin \delta_{fl} + a \cos \delta_{fl}) \\ C_{ywf} (0.5t_f \sin \delta_{fr} + a \cos \delta_{fr}) \\ C_{ywr} (-0.5t_r \sin \delta_{rl} - b \cos \delta_{rl}) \\ C_{ywr} (0.5t_r \sin \delta_{rr} - b \cos \delta_{rr}) \end{bmatrix}^T \quad (19)$$

Control effectiveness matrix B_γ for camber angle actuators γ_i^{CA} is the same as control effectiveness matrix B_δ , only instead of lateral stiffness C_{ywi} , the camber stiffness $C_{\gamma i}$ should be used.

Control effectiveness matrix B_{br} for friction brake torques $T_{br,i}^{CA}$ and matrix for electric motors B_{em} can be written using wheel radius r_w :

$$B_{br} = B_{em} = \begin{bmatrix} r_w^{-1} (-0.5t_f \cos \delta_{fl} + a \sin \delta_{fl}) \\ r_w^{-1} (0.5t_f \cos \delta_{fr} + a \sin \delta_{fr}) \\ r_w^{-1} (-0.5t_r \cos \delta_{rl} - b \sin \delta_{rl}) \\ r_w^{-1} (0.5t_r \cos \delta_{rr} - b \sin \delta_{rr}) \end{bmatrix}^T \quad (20)$$

Control effectiveness matrix B_{pw} for dynamic tire pressures

p_{wi}^{CA} is defined as:

$$B_{pw} = \begin{bmatrix} C_{p_{wx}}^{fl} (-0.5t_f \cos \delta_{fl} + a \sin \delta_{fl}) + C_{p_{wx}}^{fr} (-0.5t_f \sin \delta_{fl} + a \cos \delta_{fl}) \\ C_{p_{wx}}^{fr} (0.5t_f \cos \delta_{fr} + a \sin \delta_{fr}) + C_{p_{wx}}^{fl} (0.5t_f \sin \delta_{fr} + a \cos \delta_{fr}) \\ C_{p_{wx}}^{rl} (-0.5t_r \cos \delta_{rl} - b \sin \delta_{rl}) + C_{p_{wx}}^{rr} (-0.5t_r \sin \delta_{rl} - b \cos \delta_{rl}) \\ C_{p_{wx}}^{rr} (0.5t_r \cos \delta_{rr} - b \sin \delta_{rr}) + C_{p_{wx}}^{rl} (0.5t_r \sin \delta_{rr} - b \cos \delta_{rr}) \end{bmatrix}^T \quad (21)$$

Control effectiveness matrix B_{RZ} for the actuators generating additional normal forces R_{zi}^{CA} is found in the same way as control effectiveness matrix B_{pw} , when, instead of coefficients $C_{p_{wx}}$ and $C_{p_{wy}}$, the coefficients $C_{R_{zwx}}$ and $C_{R_{zwy}}$ are used.

Tire cornering stiffness C_{ywi} , camber stiffness C_{yi} , coefficients $C_{p_{wx}}$ and $C_{p_{wy}}$ depending on tire pressure, and coefficients $C_{R_{zwx}}$ and $C_{R_{zwy}}$ depending on normal forces, are numerically computed as partial derivatives of longitudinal and lateral tire forces from tire model by Taylor series explanation with sampling time of 1 ms.

The actuator position and rate limits are used as optimization constraints to represent static actuator dynamics. Their values are listed in [Table 1](#).

Table 1. Position and rate limits of actuators

Subsystem	Position limit u^{pos}	Rate limit u_{rate}
wheel steer actuators	-15...15 deg	50 deg/sec
camber angle actuators	-10...10 deg	30 deg/sec
electric motor	from torque-speed map	5000 Nm/sec
friction brake system	depends on slip controller	release / build-up 8000 / 12000 Nm/sec
dynamic tire pressure system	1.5 bar	release / build-up 1.5 bar / sec
additional normal force actuators	2000 N	5000 N/sec

Actuator Control

For electro-hydraulic brake system PD control of the systems' valves is applied to realize the brake pressure regulation. The control law for the pressure controller uses the error between measured and reference brake pressure for each brake caliper [7].

The controller of the electric motor is a PI controller and the detailed description of controller of the electric motor and the position torque limits can be found in [7].

The dynamic tire pressure system is controlled using the rule-based controller [9].

Since several vehicle subsystems are represented as transfer functions, the reference signals from wheel steer actuators, camber angle actuators and actuators generating additional normal forces are directly assigned to actuator models.

Discussion of Simulation Results

Please, note that in the printable version of the paper the following [Figures 3, 4, 5, 6, 7, 8, 9, 10, 11, 12](#) could have reduced readability. The color version of Figures is in electronic edition of the paper.

The following subsystems are considered to investigate their stand-alone operation:

- IFS - individual front wheel steering;
- IRS - individual rear wheel steering;
- IEM - traction / brake torques generated by electric motors;
- FBS - friction brake system;
- ACS - active camber system;
- APS - dynamic tire pressure system;
- ARzS - additional normal forces generated by external spring, damping and stabilizer forces.

The stand-alone operation is investigated for three different maneuvers represented below and covering vehicle responses (steady-state and transient) in the time domain and in the frequency domain: i) Slowly Increasing Steer Test, ii) Step Steer Test; iii) Sine Sweep Test.

Slowly Increasing Steer Test

The Slowly Increasing Steer test evaluates vehicle behavior in both linear and nonlinear regimes of vehicle curvilinear motion. In accordance with the procedure proposed in [18], steering wheel angle increases from zero to 360 deg at a rate of 13.5 deg/s. In addition, from the results of this test the steering wheel angle value is extracted for Step Steer test that has been further considered in this study. Vehicle moves with the constant speed of 100 km/h. The simulation results are shown in [Figure 3](#). The dash black line corresponds to behavior of reference model. The solid black line shows a motion of investigated vehicle without any actuations. Other lines represent vehicle motion in the case of the application of certain subsystem. The energy characteristics including total energy consumption (electric motor plus actuator consumption) and tire energy dissipation are shown in [Figure 4](#).

Figure 3. SWA versus lateral acceleration during Slowly Increasing Steer Test

Figure 4. Energy characteristics during Slowly Increasing Steer Test

As can be seen from Figure 1, the application of each subsystem increases a linear range of steering characteristic. More deep analysis of obtained characteristics allows us to formulate a number of conclusions:

- all considered subsystems provide a similar vehicle behavior in the linear area;
- stand-alone operation of each vehicle subsystem has notable influence on the steering characteristic in the nonlinear area, when lateral acceleration is more than 5 m/s^2 ;
- APS and ARzS have higher steering-wheel angle gradient $\dot{\psi}^{act}_{yaw}$ in the nonlinear area;
- IFS, IRS and APS cannot provide precise tracking close to friction boundary conditions;
- IEM and ACS allow to realize accurate tracking of reference characteristic in the area of friction limits;
- since the constant vehicle velocity should be maintained during the maneuver, all subsystems have higher total energy consumption in comparison to the motion without control;
- FBS leads to the maximal total energy consumption of 221 kJ due to nature of friction brake impact on vehicle motion;
- from the point of view of tire energy dissipation, ACS has the highest value of energy loss of 195 kJ.

Step Steer Test

The maneuver is carried out at initial velocity of 100 km/h and steering wheel rate of 200 deg/s for all considered configurations. Vehicle velocity can be reduced during the maneuver. Corresponding steering wheel angles are selected from Figure 1 to reach lateral acceleration of 4 m/s^2 (Figure 5). The yaw rate response is shown in Figure 6. Loss of longitudinal velocity, actuator energy consumption and tire energy dissipation are shown in Figure 7. In accordance with [19], the characteristic values are summarized in Table 2: steady-state yaw rate response gain $(d\psi/\delta_H)_{ss}$, response time of yaw rate $T_{d\psi}$ and lateral acceleration T_{ay} , peak response times of yaw rate $T_{d\psi, max}$ and lateral acceleration $T_{ay, max}$, overshoot values of yaw rate $U_{d\psi}$, lateral acceleration U_{ay} and sideslip angle U_{β} , steady-state β_{ss} and peak β_{max} sideslip angle.

The analysis of simulation results points to the following inferences:

- comparing to the motion without control, each variant provides yaw rate response close to reference characteristic, except APS and ARzS;
- APS provides similar characteristic of yaw rate as the motion without control due to actuator rate limits. ARzS reaches faster yaw rate response than APS;
- IRS and ACS have the similar shortest response time of yaw rate, which demonstrates better handling performance;

Figure 5. Lateral acceleration during Step Steer Test

Figure 6. Yaw rate during Step Steer Test

Figure 7. Energy characteristics during Step Steer Test

Table 2. Characteristic values of Step Steer Test

Value	Subsystems							
	w/o	IFS	IRS	IEM	FBS	ACS	APS	ARzS
$(d\psi/\delta_H)_{ss}$	0.349	0.351	0.351	0.351	0.353	0.351	0.348	0.350
$T_{d\psi, s}$	0.145	0.143	0.141	0.144	0.148	0.142	0.144	0.145
$T_{d\psi, max, s}$	0.305	0.252	0.290	0.255	0.249	0.226	0.305	0.292
$T_{\delta r, s}$	0.351	0.386	0.397	0.392	0.390	0.392	0.348	0.387
$T_{\delta r, max, s}$	0.587	0.683	1.132	0.756	0.928	0.926	0.584	0.596
$U_{d\psi}$	0.172	0.074	0.061	0.072	0.111	0.058	0.175	0.158
$U_{\delta r}$	0.058	0.020	0.008	0.011	0.016	0.013	0.062	0.081
β_{ss}, deg	1.083	1.090	1.102	1.099	1.085	1.095	1.071	1.086
β_{max}, deg	1.214	1.131	1.120	1.119	1.129	1.118	1.210	1.225
U_{β}	0.121	0.037	0.017	0.019	0.041	0.022	0.129	0.128

- the lowest overshoot values of yaw rate are obtained with ACS and IRS, while the lowest overshoot of lateral acceleration and of sideslip angle is reached by IRS. Such kind of overshoot values will provide better driver comfort;
- ACS, IEM and IRS ensure the lowest peak value of sideslip angle;
- IEM provides the lowest loss of longitudinal velocity of 0.85 km/h, when the highest one of 2.4 km/h is obtained with FBS;
- the highest actuator energy consumption of 4.7 kJ is obtained with IEM and the lowest one of 0.14 kJ with IRS (except motion without control);
- the lowest tire energy dissipation of 12.45 kJ is provided by FBS and the highest one of 12.63 kJ by IEM.

Sine Sweep Test

The Sine Sweep test is carried out according to the procedure described in [19]. This test allows to obtain frequency response of the main characteristics of lateral dynamics. The input steering wheel angle is a sine with variable frequency increasing from 0.2 to 4 Hz and steering amplitude of 25 deg. During the maneuver constant vehicle velocity of 100 km/h is maintained. The frequency range for normal driving conditions is typically limited by 1.5-2 Hz [16].

Based on the results some conclusions in terms of yaw rate response (Fig.8) for each of subsystems can be done:

- ARzS and APS have the worst tracking over the whole frequency range up to 4 Hz. In the meantime, their phases are shifted relatively to uncontrolled vehicle from 1 Hz to 2 Hz;
- due to the actuator dynamics limitations the application of ACS can be reasonable only in lower frequencies up to 1.6 Hz;
- other subsystems provide a similar response with some phase shifts, which tends to the reference characteristic;
- more agile vehicle behavior compared to IFS, IEM and FBS, is produced by IRS with the lowest phase lag and similar magnitude;
- FBS has the highest phase lag in higher frequencies after 2 Hz. Hence, vehicle will have slower reaction in this range compared to IFS, IRS and IEM.

In terms of sideslip angle response (Fig.9) the following conclusions can be received:

- IRS has the lowest magnitude of sideslip angle in lower frequencies up to 2 Hz and the lowest phase lag in the entire frequency range, which allows to achieve better transient response compared with other configurations;
- ACS, IEM, FBS and IFS have higher magnitudes and produce worse transient response in the mentioned order compared to IRS; in the same time, magnitude and phase lag of ACS is sharply changed after 2 Hz;

Figure 8. Yaw rate / SWA during Sine Sweep Test

Figure 9. Sideslip angle / SWA during Sine Sweep Test

Figure 9. (cont.) Sideslip angle / SWA during Sine Sweep Test

Figure 10. Lateral acceleration / SWA during Sine Sweep Test

- APS has higher magnitude in lower frequencies up to 1.2 Hz, while ARzS will have worse response in the range from 1 to 1.8 Hz compared to motion without control.

Lateral acceleration response, which is represented in Fig.10, leads to the next conclusions:

- APS has the similar lateral acceleration response as the uncontrolled vehicle;
- ARzS causes a significant variation of lateral acceleration response passing an excitation frequency range from 1.5 to 1.8 Hz;
- IRS provides the best tracking of lateral acceleration response in frequencies up to 3 Hz. In higher frequencies it has a wide variation of phase and drop of magnitude;

- IFS, IEM, FBS, ACS produce an appropriate behavior close to the target over the whole frequency range;
- having a higher magnitude, IFS produces sharper response in frequencies over 3 Hz than the reference characteristic;

Authors of the presented research propose to analyze the power consumption and tire power dissipation for each subsystem as the response to steering wheel sinusoidal excitations. Such approach makes feasible to carry out power analysis in terms of different disturbance frequencies and define the most energy efficient subsystems in different frequency ranges.

Figure 11. Total power consumption / SWA during Sine Sweep Test

Figure 12. Tire power dissipation / SWA during Sine Sweep Test

Figure 13. Energy characteristics during Sine Sweep Test

The total power consumption is shown in [Figure 11](#) and tire power dissipation in [Figure 12](#). The energy characteristics obtained at the end of maneuver are presented in [Figure 13](#). The obtained relationships allow us to formulate the following inferences:

- FBS produces the highest magnitude of total power consumption up to 3 Hz and, as a result, it consumes maximal energy. After 3 Hz IEM has higher total power consumption;
- other subsystems have similar total power consumption up to 2 Hz;
- APS has the lowest magnitude of total power consumption and minimal energy consumption; in the meantime it has insignificant impact on vehicle dynamics as it has been admitted previously;
- all subsystems generate similar tire power dissipation in lower frequencies up to 0.7 Hz;
- ARzS and APS demonstrate tire power dissipation close to uncontrolled motion;
- IEM and ACS have lower tire power dissipation after 2 Hz compared to other subsystems, when power dissipation for IFS and IRS increases. Due to this fact, final tire energy dissipation at the end of maneuver is the highest one for IFS and IRS.

Subjective Assessment of Influence of Vehicle Subsystems

Summarizing vehicle behavior during all maneuvers, the subjective assessment of vehicle subsystem effectiveness from various points of view can be proposed. The following criteria are considered in [Table 3](#):

- “Range of applicability” represents how the application of vehicle subsystem reaches target behavior in different operational modes. For example, the application of IFS, IRS and APS can cause a loss of tracking performance in the area of friction boundary conditions;
- “Agility” demonstrates how fast vehicle response can be in any particular case;
- “Energy Efficiency” and “Tire Energy Losses” evaluate total energy consumption and energy dissipation in tire-road contact.

Table 3. Subjective assessment of vehicle subsystems during all maneuvers for normal driving conditions

Subsystem	Range of Applicability	Agility	Energy Efficiency	Tire Energy Losses
IFS	+	+	++	-
IRS	++	++	++	--
IEM	+	+	-	-
FBS	+	0	--	++
ACS	-	++	0	0
APS	--	--	+	+
ARzS	--	-	+	-

Conclusions

The proposed paper covers the investigation of stand-alone operation of several vehicle subsystems and their influence on handling performance, energy consumption and tire energy dissipation under different operational modes. The considered subsystems are: (i) friction brake system, (ii) individual-wheel drive electric motors, (iii) wheel steer actuators, (iv) camber angle actuators, (v) dynamic tire pressure system and (vi) actuators generating additional normal forces through external spring, damping and stabilizer forces. The investigated maneuvers are related to vehicle responses during steady-state and transient motions in the time and frequency domains. The obtained results can be used for the development of integrated control strategy to operate vehicle subsystems in joint mode and, moreover, for selection of subsystems composition.

References

1. Yim, S., Choi, J. and Yi, K., “Coordinated Control of Hybrid 4WD Vehicles for Enhanced Maneuverability and Lateral Stability,” *IEEE Transactions on Vehicular Technology*, 61(4), pp. 1946-1950, 2012.
2. Schiebahn, M., Zegelaar, P., Lakehal-Ayat, M. and Hofmann, O., “The yaw torque influence of active systems and smart actuators for coordinated vehicle dynamics controls,” *Vehicle System Dynamics*, 48(11), pp. 1269-1284, 2010.
3. Hac, A., and Bodie, M.O. “Improvements in vehicle handling through integrated control of chassis systems,” *Int. J. of Vehicle Design*, 29(1/2), pp. 23-50, 2002.
4. Traechtler, A., “Integrated vehicle dynamics control using active brake, steering and suspension systems,” *Int. J. of Vehicle Design*, 36(1), pp. 1-12, 2004.
5. Schroeder, W., Knoop, M., Liebemann, E., Deiss, H. and Krimmel, H., “Collaboration of active chassis and drivetrain systems for improved vehicle dynamics control,” *Proc. of 15th Aachen Colloquium Automobile and Engine Technology*, pp. 1671-1682, 2006.
6. Besselink, I., Schmeitz, A. and Pacejka, H., “An improved Magic Formula/Swift tyre model that can handle inflation pressure changes,” *Vehicle System Dynamics*, 48(Sup.1), pp. 337-352, 2010.
7. Shyrokau, B., Wang, D., Savitski D., and Ivanov V., “Vehicle Dynamics Control with Energy Recuperation based on Control Allocation for Independent Wheel Motors and Brake System,” *International Journal of Powertrains*, 2 (2/3), pp. 153-181, 2013.
8. Bottiglione, F., Sorniotti, A. and Shad, L., “The effect of half-shaft torsion dynamics on the performance of a traction control system for electric vehicles,” *Proc IMechE Part D: J Auto. Eng.*, 226 (9), pp. 1145-1159, 2012.
9. Shyrokau, B., Wang, D. and Lienkamp, M., “Integrated Vehicle Dynamics Control Based On Control Allocation With Subsystem Coordination,” *Proc. of 23rd Int. Symposium on Dynamics of Vehicles on Roads and Tracks, Qingdao, China, 19-23 August, 2013.*

10. Alleyne, A. "Improved vehicle performance using combined suspension and braking forces," *Vehicle System Dynamics*, 27(4), pp. 235-265, 1997.
11. Dixon, J., The shock absorber handbook: Wiley, 2007
12. Sornioti, A., "Electro-Mechanical Active Roll Control: A New Solution for Active Suspensions," SAE Technical Paper 2006-01-1966, 2006, doi:10.4271/2006-01-1966.
13. Reybrouck, K., Vandersmissen, I. and Six, I., "ACOCAR: Ultimate Comfort and Safety Through the Energy-Efficient Ective Damping System of Tenneco," Proc. of 21st Aachen Colloquium Automobile and Engine Technology, 2012, pp. 1-15.
14. Westhuizen, S. and Els, P., "Slow active suspension control for rollover prevention," *Journal of Terramechanics*, 50(1), pp. 29-36, 2013.
15. Buma, S., Ookuma, Y., Taneda, A., Suzuki, K., Cho, J. and Kobayashi, M., "Design and development of electric active stabilizer suspension system," *Journal of System Design and Dynamics*, 4(1), pp. 61-76, 2010.
16. Morgando, A., Velardocchia, M., Danesin, D., and Rossi, E., "An Innovative Control Logic for a Four Wheel Steer Vehicle - Part 1: Analysis and Design," SAE Technical Paper 2005-01-1267, 2005, doi:10.4271/2005-01-1267.
17. Rajamani, R. Vehicle dynamics and control: Springer, 2006
18. ISO 4138:2004. Passenger cars - Steady-state circular driving behaviour - Open-loop test methods", 2004.
19. ISO 7401: 2003. Road vehicles - lateral transient response test methods, 2003.

Contact Information

Barys Shyrokau
Nanyang Technological University
Division of Control and Instrumentation
50 Nanyang Avenue, S2-B6a-01, Singapore 639798
barys1@e.ntu.edu.sg

Dzmitry Savitski
Ilmenau University of Technology
Department of Automotive Engineering
Gustav-Kirchhoff-Platz 2, Ilmenau, Germany 98693
dzmitry.savitski@tu-ilmenau.de

Professor Danwei Wang
Nanyang Technological University
Division of Control and Instrumentation
50 Nanyang Avenue, S2-B6a-01, Singapore 639798
edwwang@ntu.edu.sg

Acknowledgments

This publication is made possible by the Singapore National Research Foundation under its Campus for Research Excellence And Technological Enterprise (CREATE) programme. The views expressed herein are solely the responsibility of the authors and do not necessarily represent the official views of the Foundation. This work is also supported in part by National Natural Science Foundation of China (91120308). This work also relates to the scientific activities of

the research group PORT at Ilmenau University of Technology funded by the European Social Fund ESF (project No. 2011 FGR 0120).

Definitions/Abbreviations

- a - distance between front axle and total CoG, m
- b - distance between rear axle and total CoG, m
- B_{br} - control effectiveness matrix for friction brake torques, [-]
- B_{em} - control effectiveness matrix for electric motors, [-]
- B_{pw} - control effectiveness matrix for dynamic tire pressures, [-]
- B_{Rz} - control effectiveness matrix for the actuators generating additional normal forces, [-]
- B_u - control effectiveness matrix, [-]
- B_v - control effectiveness matrix for camber angle actuators, [-]
- B_δ - control effectiveness matrix for wheel steer actuators, [-]
- C_f - cornering stiffness of front tires of reference model, N/rad
- C_{hs} - half-shaft torsion damping coefficient, Nms/rad
- C_{pwx} - coefficient related to changing of longitudinal force depends on tire pressure, N/bar
- C_{pwy} - coefficient related to changing of lateral force depends on tire pressure, N/bar
- C_r - cornering stiffness of rear tires of reference model, N/rad
- C_{Rzwx} - coefficient related to changing of longitudinal force depends on normal forces, [-]
- C_{Rzwy} - coefficient related to changing of lateral force depends on normal forces, [-]
- C_{ywi} - tire cornering stiffness, N/rad
- C_{yi} - tire camber stiffness, N/rad
- $d\beta/dt$ - vehicle sideslip rate, rad/s
- $d\psi/dt$ - vehicle yaw rate, rad/s
- $d\psi_{max}/dt$ - maximum of the reference yaw rate according to adhesion tire limit, rad/s
- $d\psi_{ref}/dt$ - reference yaw rate, rad/s
- E_{em}^{total} - total energy consumption by electric motors, kJ
- E_{tire}^{total} - total tire energy dissipation in longitudinal and lateral directions, kJ
- $F_{damp,i}^{ext}$ - external damping force, N
- $F_{damp,i}^{ref}$ - reference damping force from the control system, N
- $F_{spring,i}^{ext}$ - external spring force, N
- $F_{spring,i}^{ref}$ - reference spring force from the control system, N
- F_{xwi} - longitudinal tire force, N
- F_{ywi} - lateral tire force, N
- I_{zz} - total moment of inertia of vehicle about z-axis, kgm²
- J_{em} - moment of inertia of the electric motor, kgm²
- J_{hs} - moment of inertia of the half-shaft, kgm²
- J_{tr1} - moment of inertia of the input transmission shaft, kgm²
- J_{tr2} - moment of inertia of the output transmission shaft, kgm²
- K_{hs} - half-shaft torsion stiffness, Nm/rad
- K_p - proportional control gain to calculate generalized demand, [-]

L_{yi} - lateral relaxation length of tire, m	γ_i^{ext} - external wheel camber angle, rad
m_a - total vehicle mass, kg	δ_i - wheel steer angle, rad
M_{zwi} - tire self-aligning torque, Nm	δ_i^{CA} - wheel steer angles from control allocation, rad
$M_{\phi wi}$ - tire overturning torque, Nm	δ_i^{ext} - external steering angle, rad
P_{em} - instantaneous power consumption of electric motors, W	δ_{steer}^{driver} - steering wheel angle generated by driver, rad
$p_{w,i}^{CA}$ - additional dynamic tire pressures from control allocation, bar	η_j^{inner} - efficiency of inner constant velocity joint, [-]
$p_{w,i}^{ext}$ - additional tire pressure, bar	η_j^{outer} - efficiency of outer constant velocity joint, [-]
r_{wi} - tire rolling radius, m	η^{tr} - efficiency of the transmission, [-]
R_{zi}^{CA} - additional normal forces from control allocation, N	θ_{emi} - angular displacement of electric motor, rad
T_{ay} - response time of lateral acceleration, s	θ_{wi} - angular displacement of tire, rad
$T_{ay,max}$ - peak response time of lateral acceleration, s	μ_{road} - tire-road friction coefficient, [-]
$T_{br,i}^{CA}$ - friction brake torque from control allocation, Nm	v_{yaw}^{act} - generalized yaw torque realized by actuators, Nm
$T_{d\psi}$ - response time of yaw rate, s	v_{yaw} - generalized yaw torque, Nm
$T_{d\psi,max}$ - peak response time of yaw rate, s	ξ - parameter defining a significance of minimization of control actuations, [-]
$T_{em,i}$ - transient torque of electric motor, Nm	ω_{emi} - angular velocity of electric motor, rad/s
$T_{em,i}^{CA}$ - torque of electric motor from control allocation, Nm	ω_{wi} - wheel angular velocity, rad/s
$T_{em,i}^{ss}$ - torque from steady-state model of electric motor, Nm	
t_f - front track width, m	
$T_{hs,i}$ - half-shaft torque, Nm	
T_i - anti-windup gain to calculate generalized demand, [-]	
t_r - rear track width, m	
T_{wi} - transmitted torque to the wheel, Nm	
U_{ay} - overshoot value of lateral acceleration	
u^{CA} - control input	
$U_{d\psi}$ - overshoot value of yaw rate	
u^{pos} - constraints related to position limitations of actuators	
u_{rate} - rate limits of actuators	
u_{tr} - transmission ratio, [-]	
U_{β} - overshoot value of sideslip angle	
V_{swxi} - longitudinal slip velocity in the tire-road contact, m/s	
V_{swyi} - lateral slip velocity in the tire-road contact, m/s	
V_{xa} - longitudinal velocity of vehicle, m/s	
W_u - weighting matrix defining actuators' restriction, [-]	
W_v - weighting matrix defining a priority among generalized forces/torque, [-]	
β - sideslip angle, rad	
β_{max} - peak sideslip angle, rad	
β_{ss} - steady-state sideslip angle, rad	
γ_i^{CA} - additional camber angles from control allocation, rad	

The Engineering Meetings Board has approved this paper for publication. It has successfully completed SAE's peer review process under the supervision of the session organizer. The process requires a minimum of three (3) reviews by industry experts.

All rights reserved. No part of this publication may be reproduced, stored in a retrieval system, or transmitted, in any form or by any means, electronic, mechanical, photocopying, recording, or otherwise, without the prior written permission of SAE International.

Positions and opinions advanced in this paper are those of the author(s) and not necessarily those of SAE International. The author is solely responsible for the content of the paper.

ISSN 0148-7191

<http://papers.sae.org/2014-01-0876>