Coordinated control of multi-actuated electric vehicle

Barys Shyrokau

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**Coordinated Control
of
Multi-actuated Electric Vehicle**

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Summary

Increasing requirements to vehicle safety, performance and environmental friendliness lead to the development of complex vehicle subsystems to achieve better operation characteristics. As a result, modern vehicles have the tendency to embed multiple actuators. This feature allows the implementation of stand-alone and joint operation of actuators to control stability of motion, vehicle handling, energy consumption, energy losses and others; therefore, the integrated control of vehicle as multi-actuated system is gaining importance year after year.

Literature review describes the relevant state-of-the-art of integrated vehicle control systems including their historical development process, publication statistics and various application techniques. Since the modern passenger car is over-actuated, a detailed review of control allocation methods and their automotive applications is presented.

The research goal of this study is the development and evaluation of a control system coordinating priorities between vehicle subsystems for a multi-actuated passenger car and demonstration of the possibility to improve several vehicle operation characteristics during vehicle motion. The subsystem coordination under discussion covers (i) friction brake system, (ii) individual near-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. It should be noted, that this is the first study in literature exploring dynamic tire pressure system for integrated vehicle control.

For the preliminary investigations, the original vehicle model was developed. The vehicle model has 14 DoF including 6 DoF of vehicle body in the space and 2 DoF of rotational and vertical movements of each wheel. The actuator models are selected using the experimental test results and technical literature information. The verification of the developed model is carried out under transient manoeuvre. The developed model demonstrates a similar behaviour compared to a multibody vehicle model.

Since friction brakes are the main subsystem to ensure vehicle safety and stability, the special chapter of the thesis introduces detailed analysis of the effect of hysteresis in the brake mechanism. The effect of hysteresis in the brake mechanism was first described by the Bouc-Wen model and verified by experimental investigations of disc brake mechanism for passenger car. Using the computational model of the brake mechanism with hysteretic elements, the analysis of hysteresis influence on vehicle dynamics during

ABS straight-line braking and stability control of curvilinear motion was carried out. The analysis shows that the effect of hysteresis on stopping distance for new hydraulic disc brake mechanism does not exceed 5% in the most critical cases. Hysteresis influence on stability control of curvilinear motion is insignificant due to the duration and frequency of brake torque actuations. The developed hysteresis model was integrated into the model of the friction brake system and used in subsequent investigations.

Next part of the thesis describes a multi-layer control system including high-level controller for vehicle dynamics, middle-level control allocation, low-level individual controllers for each vehicle subsystem and vehicle state observer. The high-level controller consists of a reference vehicle model and vehicle dynamics controller to calculate generalized demands of longitudinal and lateral forces, and yaw torque using discrete second-order sliding mode control. The middle layer is optimization-based control allocation with a special block related to the coordination of priorities between vehicle subsystems. The fixed-point method with termination conditions is used as an optimization solver. The low-level actuator control guarantees a precise tracking to reference control signals obtained from the middle layer. Moreover, it is used to estimate boundary conditions for control allocation taking into account actuator physical limits and wheel slip control.

Analysis of stand-alone / joint operation, the prioritization of vehicle subsystems and their influence on vehicle motion performance, energy consumption and losses was carried out. It demonstrates properly distributed operation of all considered subsystems can guarantee a reasonable compromise in simultaneous maintenance of criteria of motion performance, energy consumption and energy losses. Novel algorithms of the coordination of priorities between vehicle subsystems using the restriction weights into the control allocation were proposed. The first algorithm is based on fuzzy logic and the second one is an iterative approach with the various variants of performance criteria prioritization.

The developed control system with the proposed algorithms was investigated using a hardware-in-the-loop test rig with hardware components of conventional brake system and dynamic tire pressure system in real-time domain. Vehicle test manoeuvres covering longitudinal and lateral motion were considered. Results indicate that the proposed algorithms of the coordination of priorities between vehicle subsystems provide better energy consumption or higher potential recuperated energy for all manoeuvres.

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List of Symbols

Δ_{ax}	dead zone threshold for the error of longitudinal acceleration, m/s ²
$\Delta_{d\psi/dt}$	dead zone threshold for the error of yaw rate, rad/s
$(Nv)_{ref}$	reference system power, W
ΔV_{xa}	loss of velocity during the manoeuvre, km/h
Δ_{β}	dead zone threshold for the error of sideslip angle, rad
a	distance between front axle and total CoG, m
A_{front}	frontal area of vehicle, m ²
A_i^{inlet}	maximum orifice area of inlet valve, m ²
A_i^{outlet}	maximum orifice area of outlet valve, m ²
A_j^{block}	orifice area of block valve, m ²
A_j^{compen}	orifice area of compensative valve, m ²
A_{mc}	area of the tandem master cylinder, m ²
a_s	distance between front axle and CoG of sprung mass, m
a_x	longitudinal acceleration of vehicle, m/s ²
a_x^{max}	maximum acceleration constrained by friction limits, m/s ²
a_x^{ref}	reference deceleration, m/s ²
a_y	lateral acceleration of vehicle, m/s ²
a_z	vertical acceleration of vehicle, m/s ²
b	distance between rear axle and total CoG, m
B_{br}	control effectiveness matrix for friction brake torques, [-]
B_{em}^{neg}	control effectiveness matrix of brake torques for electric motors, [-]
B_{em}^{pos}	control effectiveness matrix of traction torques 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_s	distance between rear axle and CoG of sprung mass, m
BSF	bump stop force to strut force, [-]
B_u	control effectiveness matrix, [-]
B_u^{slip}	control effectiveness matrix defining wheel slips, [-]
B_{γ}	control effectiveness matrix for camber angle actuators, [-]
B_{δ}	control effectiveness matrix for wheel steer actuators, [-]
c', α', γ'	empirical parameters which define temperature behaviour, [-]
c_l	translational damping coefficient, Nm ⁻¹ s ⁻¹
C_B	battery capacity, F
$C_{complex}$	criterion combining vehicle handling, stability, energy consumption and losses, [-]
C_d	aerodynamic drag coefficient, [-]
C_{energy}	criterion of energy consumption and energy losses, [-]
C_f	cornering stiffness of front tires of reference model, N/rad
C_{flow}	flow coefficient, [-]
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 force, [-]
C_{Rzwy}	coefficient related to changing of lateral force depends on normal force, [-]
$C_{s\&h}$	criterion of vehicle stability and handling, [-]
C_{sf}, C_{sr}	front and rear suspension vertical damping coefficient, Nm-s/rad
c_{tire}	rotational damping coefficient, kgm ² s ⁻¹ rad ⁻¹
C_{ywi}	tire cornering stiffness, N/rad

C_{ai}	cornering tire stiffness, N/rad
$C_{\gamma i}$	camber tire stiffness, N/rad
$C_{\gamma i}$	tire camber stiffness, N/rad
$C_{\phi f}, C_{\phi r}$	front and rear suspension roll damping coefficient, Nm-s/rad
d_{brake}	brake pedal displacement, m
d_{cyl}	diameter of brake piston/cylinder, m
$dV_{bwc i}/dt$	volume changes in the i^{th} calliper, m ³ /s
$d\beta_{bound}/dt$	boundary limit of derivative of vehicle sideslip angle, rad
$d\beta_{est}/dt$	derivative of vehicle sideslip angle, rad
$d\theta/dt$	vehicle pitch rate, rad/s
$d\phi/dt$	vehicle roll rate, rad/s
$d\psi/dt$	vehicle yaw rate, rad/s
$d\psi/dt$	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	control error, [-]
e_{ax}	control error of longitudinal acceleration, m/s ²
$e_{d\psi/dt}$	control error of yaw rate, rad/s
E_{em}^{total}	total energy consumption by electric motors, kJ
e_M	piecewise function, [-]
e_{mem}	memory variable, [-]
E_{tire}^{total}	total tire energy dissipation in longitudinal and lateral directions, kJ
e_{β}	control error of sideslip angle, rad
F^{aero}	aerodynamic drag force, N
$F_{bump,i}$	bump stop force, N
$F_{damp,i}^{ext}$	external damping force, N
$F_{damp,i}^{ref}$	reference damping force from the control system, N
F_{fi}	rolling resistance force, N
F_p	pedal force, N
F_{si}	suspension force at strut, N
$F_{spring,i}^{ext}$	external spring force, N
$F_{spring,i}^{ref}$	reference spring force from the control system, N
$F_{wi}^{actuators}$	force components generated by actuators, N
F_{wi}^{sys}	force components related to vehicle motion, N
F_{xi}	resultant longitudinal force acting on wheel in x-direction, N
F_{xwi}	longitudinal tire force, N
F_{xwi}^{ss}	longitudinal tire force from steady-state model, N
F_{yi}	resultant lateral force acting on wheel in y-direction, N
F_{ywi}	lateral tire force, N
F_{ywi}^{ss}	lateral tire force from steady-state model, N
F_{zi}	normal tire force, N
F_{zi}^{static}	static normal tire force, N
F_z^{lat}	front and rear load transfer owed to the lateral acceleration, N
F_z^{long}	load transfer owed to the longitudinal acceleration, N
F_z^{pitch}	front and rear load transfer owed to the pitch behaviour, N
F_z^{roll}	front and rear load transfer owed to the roll behaviour, N
h_f	height of front roll center, m
h_o	height of origin above ground, m
h_r	height of rear roll center, m
h_s	distance from the roll center to the sprung mass center, m
h_{uf}	height of front unsprung mass CoG above origin, m
h_{ur}	height of rear unsprung mass CoG above origin, m

h_θ	height of pitch center, m
I_B	instantaneous battery current, A
I_{xxs}	moment of inertia of sprung mass about x-axis, kgm^2
I_{xzs}	moment of inertia of sprung mass about x- and z-axis, kgm^2
I_{yys}	moment of inertia of sprung mass about y-axis, kgm^2
I_{zz}	total moment of inertia of vehicle about z-axis, kgm^2
I_{zzs}	moment of inertia of sprung mass about z-axis, kgm^2
I_{zzuf}, I_{zzur}	moment of inertia of front and rear unsprung masses about own CoG, kgm^2
J_{bd}	moment of inertia of brake disc, kgm^2
J_{em}	moment of inertia of the electric motor, kgm^2
J_{hs}	moment of inertia of the half-shaft, kgm^2
J_{tr1}	moment of inertia of the input transmission shaft, kgm^2
J_{tr2}	moment of inertia of the output transmission shaft, kgm^2
J_v	moment of inertia, which is equivalent to vehicle mass, kgm^2
J_{wi}	moment of inertia of one wheel, kgm^2
k_l	slope coefficient, kg^{-1}
k_l	translational contact stiffness, Nm^{-1}
k_2	initial deceleration caused by aerodynamic drag and rolling resistance, m/s^2
K_d^{br}	derivative gain of the controller for friction brake system
k_{fi}	rolling resistance coefficients, [-]
K_{hs}	half-shaft torsion stiffness, Nm/rad
k_i^{CSFx}	compliance steer depending on longitudinal force, [-]
k_i^{CSFy}	compliance steer depending on lateral force, [-]
k_i^{CSMz}	compliance steer depending on self-aligning torque, [-]
K_i^{em}	integral gain of e-motor controller, [-]
K_I^{react}	integral gain of wheel slip controller
K_p^{br}	proportional gain of the controller for friction brake system
K_p^{em}	proportional gain of e-motor controller, [-]
K_p^{react}	proportional gain of wheel slip controller
K_{sf}, K_{sr}	front and rear suspension vertical stiffness, Nm/rad
K_{sm}	control gain to calculate generalized demand, [-]
$K_{sm,ax}$	control gain to calculate generalized longitudinal force, [-]
$K_{sm,d\psi/dt}$	control gain to calculate generalized yaw torque, [-]
$K_{sm,\beta}$	control gain to calculate generalized lateral force, [-]
K_{ti}	tire vertical stiffness, Nm/rad
k_{tire}	rotational contact stiffness, Nmrad^{-1}
$K_{\phi f}, K_{\phi r}$	front and rear suspension roll stiffness, Nm/rad
L	wheelbase of vehicle, m
l_{cgs}	longitudinal distance from sprung CoG to total mass CoG, m
L_{xi}	longitudinal tire relaxation length, m
L_{yi}	lateral tire relaxation length, m
m_a	total vehicle mass, kg
m_{eq}	lumped mass of elements with translational motion, kg
M_{res}	resistance torque from the road, Nm
m_s	sprung mass, kg
m_{uf}	front unsprung mass, kg
m_{ur}	rear unsprung mass, kg
M_{zwi}	self-aligning torque, Nm
M_{zwi}^{ss}	self-aligning torque from steady-state model, Nm
M_θ	total pitch torque, Nm
M_ϕ	total roll torque, Nm
$N(t)$	actual normal load, N

N_{subs}	number of vehicle subsystems, [-]
nv	dimensionless parameter, describing the actual normal load and actual tangential velocity in the friction contact, [-]
$p(t)$	input pressure, MPa
p_{acc}	accumulator gas pressure, Pa
$p_{acc,0}$	initial accumulator gas pressure, Pa
$p_{br,i}$	brake pressure in the calliper, Pa
p_{br0}	push out brake pressure, Pa
$P_{em,i}$	instantaneous power consumption of electric motor, W
P_{emax}	maximal power of electric motor, kW
$P_{loss,swi}$	longitudinal slip power loss, W
p_{low}	constant reservoir pressure, Pa
p_{mc}	master cylinder pressure, Pa
p_{mcp}	pressure in primary section of the tandem master cylinder, Pa
p_{mcs}	pressure in secondary section of the tandem master cylinder, Pa
$p_{br,i}^{ref}$	reference brake pressure, Pa
$p_{w,i}$	tire pressure, bar
$p_{w,i}^{CA}$	additional dynamic tire pressures from control allocation, bar
$p_{w,i}^{ext}$	additional tire pressure, bar
$p_{w,i}^{nom}$	nominal tire pressure, bar
Q_i	volumetric flow from accumulator to the calliper, m ³ /s
Q_i^{dump}	volumetric flow from the calliper to the reservoir across the outlet valve, m ³ /s
Q_j^{block}	volumetric flow from the tandem master cylinder to the calliper across the block valve, m ³ /s
Q_j^{compen}	volumetric flow from the right wheel cylinder to the left wheel cylinder across the compensative valve, m ³ /s
Q_{pump}	pump volumetric flow, m ³ /s
R_{0B}	internal resistance of the battery, Ohm
R_{IB}	overvoltage resistance, Ohm
r_{eff}	effective radius of friction, m
r_{wi}	tire rolling radius, m
R_{zi}^{CA}	additional normal forces from control allocation, N
SWF	suspension force to wheel force, [-]
s_{wi}	longitudinal slip, [-]
$T_{br,i}^{CA}$	friction brake torques from control allocation, Nm
$T_{br,i}^{lim}$	position limits for friction brake torque, Nm
$T_{em,i}$	transient torque of electric motor, Nm
$T_{em,i}^{CA}$	torques of electric motors from control allocation, Nm
$T_{em,i}^{CA,neg}$	brake torques of electric motors from control allocation, Nm
$T_{em,i}^{CA,pos}$	traction torques of electric motors from control allocation, Nm
$T_{em,i}^{lim}$	position torque limits of an electric motor, Nm
$T_{em,i}^{ss}$	torque from steady-state model of electric motor, Nm
T_{em}^{max}	maximum overload torque, Nm
t_f	front track width, m
$T_{hs,i}$	half-shaft torque, Nm
T_i	winding temperature, °C
T_i^{constr}	constructive limits related to friction brake torque, Nm
T_{max}	maximum temperature, when only nominal torque is generated, °C
T_{min}	minimum temperature defining the starting point to reduce the maximum overload torque, °C
t_r	rear track width, m
T_{react}	reactive torque founded according to wheel slip control, Nm

T_{wi}	transmitted torque to the wheel, Nm
U_{0B}	ideal no-load battery voltage, V
u_{valve}^{act}	control signal for valve, [-]
U_B	battery voltage, V
u^{CA}	control input
U_{Cb}	voltage across capacitor C_B , V
$u_{em,i}$	control signal for electric motor, [-]
u_i^{inlet}	control signal for inlet valve, [-]
u_i^{outlet}	control signal for outlet valve, [-]
u_j^{block}	control signal for block valve, [-]
u_j^{compen}	control signal for compensative valve, [-]
$u_{low,up}^{lim}$	final constraints for optimization
$u_{min,max}^{pos}$	constraints related to position limitations of actuators
u_{rate}	rate limits of actuators
u_{sm}	gear ratio of steering mechanism, [-]
u_{valve}^{ss}	control signal for valve from steady-state model, [-]
$u_{min,max}^{tire}$	constraints related to longitudinal tire forces for electric motor and friction brake system, N
u_{tr}	transmission ratio, [-]
V_{acc}	accumulator volume, m ³
v_w	input vector combining performance criteria, [-]
V_{xa}	longitudinal velocity of vehicle, m/s
V_{ya}	lateral velocity of vehicle, m/s
V_{za}	vertical velocity of vehicle, m/s
w_{bdi}	angular velocity of brake disc, rad/s
W_{slip}	weighting matrix defining the restriction of wheel slips, [-]
W_u	weighting matrix defining actuators' restriction, [-]
W_v	weighting matrix defining a priority among generalized forces/torque, [-]
x_i	displacement of equivalent mass, m
z_a	vertical body displacement, m
z_{bi}	vertical displacement of body section, m
z_{ui}	vertical displacement of wheel, m
α, β, γ, n	coefficients describing the hysteresis in rotational direction, [-]
$\alpha_1, \beta_1, \gamma_1, n_1$	coefficients of hysteretic model for loading curve, [-]
$\alpha_2, \beta_2, \gamma_2, n_2$	coefficients of hysteretic model for unloading curve, [-]
α_{wi}	wheel slip angle, rad
α_w^{net}	lateral slip angle corresponding to maximum pure cornering force of linear regime of vehicle motion, rad
β_{bound}	boundary limit of vehicle sideslip angle, rad
β_{est}	estimated vehicle sideslip angle, rad
β_{ref}	reference vehicle sideslip angle, rad
γ_i	wheel camber angle, rad
γ_i^{CA}	additional camber angles from control allocation, rad
γ_i^{ext}	external wheel camber angle, rad
$\delta_{av,f}$	average steer angle of front axle, rad
δ_i	wheel steer angle, rad
δ_i^{CA}	wheel steer angles from control allocation, rad
δ_i^{ext}	external steering angle, rad
δ_{steer}^{driver}	steering wheel angle generated by driver, rad
ε	constant dimensionless power, [-]
ε_i^{roll}	roll axle steer, [-]
η	parameter for the temperature in the friction contact, [-]

η_0	initial normalized temperature in the friction contact, [-]
η_{con}	dimensionless criterion related to power consumption of electric motor, [-]
η_j^{inner}	efficiency of the inner constant velocity joint, [-]
η_j^{outer}	efficiency of the outer constant velocity joint, [-]
η_{lat}	dimensionless criterion represented the net lateral power, [-]
η_{lon}	dimensionless criterion represented the net longitudinal power, [-]
η_{rec}	dimensionless criterion related to power recuperation of electric motor, [-]
η_{tr}	efficiency of the transmission, [-]
θ	vehicle pitch angle, rad
θ_{bdi}	angular displacement of brake disc, rad
θ_{wi}	angular displacement of tire, rad
Λ	scaling vector, [-]
μ	dynamic friction coefficient in the brake pad-disc contact, [-]
μ_0	static friction coefficient in the brake pad-disc contact, [-]
μ_{road}	tire-road friction coefficient, [-]
v_*	generalized demand excluding a nonlinear component of vehicle motion, [-]
v_{bdi}	actual tangential velocity between the brake pad and disc, m/s
v_d	final generalized demand from high-level controller, [-]
v_{lat}	generalized lateral force, N
v_{lon}	generalized longitudinal force, N
v_{yaw}	generalized yaw torque, Nm
ζ	parameter defining a significance of minimization of control actuations, [-]
ζ_{slip}	parameter defining a significance of minimization of wheel slips, [-]
ρ_{air}	air density, kg/m ³
ρ_{br}	brake fluid density, kg/m ³
σ_{ax}	threshold for the saturation of longitudinal acceleration, m/s ²
$\sigma_{d\psi/dt}$	threshold for the saturation of yaw rate, rad/s
σ_{β}	threshold for the saturation of sideslip angle, rad
ϕ	vehicle roll angle, rad
χ_{st}	stability index during the manoeuvre, [-]
χ_{xi}	kinematic ratios for longitudinal direction, [-]
χ_{yi}	kinematic ratios for lateral direction, [-]
ψ	vehicle yaw angle, rad
ω_{wi}	wheel angular velocity, rad/s

Chapter 1

1 Introduction

The chapter includes motivation, the main contributions and the description of the contents of the thesis.

1.1 Motivation

The tightening requirements for the reduction of environmental pollution is among many factors defining new ground vehicle technologies. The limitations of greenhouse gas emissions are becoming stricter every year. The CO₂ emissions should be reduced 90% by the year 2050 as compared to 2000 (Figure 1.1a), when could only be feasible by the production of zero-emission vehicles such as electric vehicles (EV) or fuel cell vehicles (FCEV) [1]. Hybrid electric vehicles reduce but do not solve this problem (Figure 1.1b) [2]. As to fuel cell vehicles, hydrogen production on a large scale has economic implications and still underdeveloped infrastructure. Therefore, an extensive development of battery electric vehicles will be expected.

Figure 1.1 Vehicle CO₂ emissions [1, 2]

An important challenge is to extend the application area of battery electric vehicles; however, the current technologies of automotive batteries cannot provide sufficient energy density [3]. Due to the limits of electric battery, instantaneous power requirements higher than it can be provided by battery under emergency manoeuvres. In this case, the installation of supercapacitor on the vehicle will be necessary. As a result, the total vehicle weight and final energy consumption will be increased. It stimulates to seek internal reserves to minimize energy consumption not only in the drive cycle, but also

during curvilinear and emergency manoeuvres. Therefore, a modern system of integrated vehicle control should achieve a set of control objectives.

The industrial example of such a fusion of control objectives was first announced in October 2013, when Jaguar Land Rover initiated the development of a control system to recover energy from braking on corners and stability controls. The first prototype is expected in 2015 [4].

An academic example is the minimization of tire energy dissipation in the framework of G-vectoring control [5]. It allows to reduce total dissipation energy during the vehicle motion. For example, the reduction of wheel slip (longitudinal energy losses) improves energy efficiency. The minimization of tire energy dissipation has a positive influence on tire wear during vehicle operation. In the long-term operation, its lifetime is increased reducing the vehicle operation costs.

The achievement of optimal energy consumption and minimal energy losses in modern electric vehicles can be realized through embedded mechatronic systems. These systems have the tendency to implement multiple actuators. Especially, this feature is related to premium-class and sport-modified vehicles. As a result, the vehicle can have several active subsystems operating both in stand-alone and joint modes. Referring to Figure 1.2, the multi-actuation implies that the change of tire contact forces can be simultaneously achieved through the parallel operation of different vehicle subsystems.

Figure 1.2 Various active vehicle subsystems

From constructive point of view, the parallel operation can be realized by integrated control of conventional active vehicle subsystems [6] or development of special wheel corners like Autonomous Corner Module [7, 8], which allows to reach a new level of vehicle autonomy and control of spin, steering, suspension and camber.

As a result, the multi-actuator vehicle configuration provides more freedom and flexibility to achieve desired characteristics related to vehicle operation characteristics.

Taking into account above-mentioned aspects regarding energy consumption and energy losses, vehicle dynamics control should reach a set of control targets related to stability, handling, energy consumption and energy losses. In this thesis the selected approach to reach this target is a multi-actuator vehicle configuration (Figure 1.3).

Figure 1.3 Fusion of control objectives for vehicle dynamics control

1.2 Major contributions of the Thesis

The primary goal of this thesis is the development and evaluation of a control system coordinating priorities between vehicle subsystems for a multi-actuated vehicle to reach concurrent improvement of several vehicle operation characteristics. In accordance with this goal, the following contributions are obtained:

- Development of a computational model of the brake mechanism with hysteretic elements, based on the Bouc-Wen method, and its verification with the experiments. The analysis of hysteresis influence on vehicle dynamics was carried out both for ABS straight-line braking and curvilinear motion with stability control.
- Design of the control system for a multi-actuated vehicle with the coordination between friction brake system, near-wheel electric motors, wheel steer actuators, camber angle actuators, dynamic tire pressure system and actuators generating additional normal forces through external spring, damping and stabilizer forces. It should be noted, that this is the first study exploring dynamic tire pressure system for integrated vehicle control.
- Analysis of stand-alone and joint operation, the prioritization of vehicle subsystems and their influence on vehicle motion performance, energy consumption and losses. It was demonstrated that properly distributed operation of all considered subsystems can guarantee a reasonable compromise in simultaneous maintenance of criteria of vehicle operation characteristics.
- Novel algorithms of the coordination of priorities between vehicle subsystems are proposed using the restriction weights into the real-time optimization-based control allocation. The restriction weights are changed according to performance criteria, such as longitudinal and lateral efficiency, energy consumption and recuperation.
- Experimental validation of the proposed algorithms in the framework of hardware-in-the-loop testing with the hardware components of conventional brake system and tire pressure system in real-time domain. The experimental results demonstrate better energy consumption compared to conventional control allocation without impairment of other vehicle operation characteristics.

1.3 Organization of the Thesis

The rest of the thesis is organized as follows:

- Chapter 2 describes a literature study of integrated vehicle control and, especially, the application of control allocation methods for multi-actuated vehicle. The thesis objectives are formulated in the context of this review.
- Chapter 3 focuses on the models of vehicle and its subsystems. The actuator models are selected using the experimental test results and technical literature information. The verification and the comparison of the developed model with a multibody vehicle model are carried out.
- Chapter 4 focuses on the investigation of an effect of hysteresis in the brake mechanism. The computational model of the brake mechanism with hysteretic elements, based on the Bouc-Wen method, is developed and verified with the experiments. The analysis of hysteresis influence on vehicle dynamics is provided for ABS straight-line braking and curvilinear motion with stability control.
- Chapter 5 presents an overall control system including high-level vehicle dynamics control, middle-level optimization-based control allocation with the coordination of priorities between vehicle subsystems, low-level actuator controllers for each vehicle subsystem and vehicle state observer.
- Chapter 6 includes the analysis of stand-alone and joint operation of vehicle subsystems and their influence on vehicle operation characteristics; introduction of the subsystem prioritization and the analysis of its influence on vehicle operation characteristics; novel algorithms of the coordination of priorities between vehicle subsystems.
- Chapter 7 demonstrates experimental investigations of the proposed control system based on real-time hardware-in-the-loop testing with the hardware components of conventional brake system and dynamic tire pressure system. The considered manoeuvres cover longitudinal and lateral vehicle motion.
- Chapter 8 presents a summary of the thesis and identifies the key conclusions to be drawn in the context of the thesis objectives. Possible areas for future research are highlighted.

Chapter 2

2 Literature review

The chapter introduces an overview of integrated vehicle control including publication statistics and various applied control techniques. Since the control allocation technique is selected to distribute control demand, a detailed review is conducted on control allocation methods and their automotive applications. Then the thesis objectives are formulated.

2.1 Overview of integrated vehicle control

The idea of integrated control of vehicle subsystems appeared more than 20 years ago [9]. The main requirements are formulated as follows: the integration between various vehicle subsystems, the adaptation to a wide range of road conditions, driver feedback, compliance with national standards and others [9]. For integrated control systems, the system architecture is one of the key issues and it can be represented by the following types [10]:

- centralized control, where the main controller is responsible for all control actions and decisions;
- supervisory control, where the control structure is divided by upper and lower levels and there is a master controller, which manages other subsystems' control structures;
- decentralized control, where the control structure is parallel and no interactions between subsystems occur, but the controllers can communicate with each other.

Supervisory control dominates in recent publications due to its advantages over other control types. The challenges of the control system architecture not only include a choice of the control structure, but also the requirements of modularity, autonomy and integration of subsystem controllers, the issues of online estimation, fault diagnostics and others [10]. Statistics on publications in this field from 1999 to 2012 is obtained by SciVerse Scopus and it is shown in Figure 2.1.

Figure 2.1 Publications related to integrated control

The pie chart of the combination of various vehicle subsystems is shown in Figure 2.2. The following variants are depicted: 1 – brakes and suspension; 2 – brakes, powertrain and steering; 3 – steering and suspension; 4 – brakes and steering; 5 – brakes, steering and suspension; 6 – steering and powertrain; 7 – brakes and powertrain; 8 – suspension and powertrain; 9 – brakes, powertrain and suspension; 10 – brakes, powertrain, steering and suspension.

Figure 2.2 Integration of various vehicle subsystems

An analysis of the state-of-the-art literature review shows that the integration between two to four subsystems is considered in most of known case studies. The most global integration of active steering angles, brake/drive torques, wheel loads and camber angles is investigated in [11].

Increase in the number of interacting subsystems, especially with different actuator dynamics, calls for search of complex control approaches, which are acceptable to receive on-line solution and have sufficient operation speed and robustness for real-time applications.

The main groups of integrated control methods are:

- classical control methods to minimize a control error;
- modern control methods to find an extremum of complex cost function.

The following methods for integrated control can be applied:

- state feedback control [12, 13];
- frequency-domain multivariable control [14];
- Model Predictive Control (MPC): linear, hybrid [15] and nonlinear [16];
- fuzzy and rule-based control (incl. combinations with other methods) [17-20];
- robust control (μ -synthesis and H_∞ control) and backstepping method [21];
- sliding mode control (incl. combinations with other methods) [22-24];
- model based control [25, 26] and model-matching control [27, 28];
- optimization methods and other [29-33].

The main stages of integrated/coordinated control are depicted in Figure 2.3. Control demand including generalized vehicle forces and torques is obtained from high-level vehicle dynamics controller.

The next stage is a distribution of control demand between four wheels, taking into account tire grip margin of each wheel, normal force distribution, power minimization and other objectives. Several solutions for force allocation are proposed in [11, 34, 35].

The subsystem coordination succeeds the procedure of force distribution. Existing vehicle subsystems are shown in Figure 2.3 in the outer ring, which also includes some hypothetic subsystems, such as regulation of tire contact temperature, changing of down force using aerodynamic spoilers, and others.

Figure 2.3 Control demand distribution

Thereby, control demand from high-level controller can be realized by various ways according to control tasks in current conditions of vehicle motion. The following approaches can be applied to distribute control demand between vehicle subsystems:

- rule-based integration [23];
- fuzzy logic based approach [10, 17];
- multi-agent approach [36];
- control allocation, which is one of the most common control techniques to distribute control demand for over-actuated system [37].

2.2 Review of control allocation methods for vehicle motion control

2.2.1 Control allocation for automotive tasks

The control allocation methods, such as direct, pseudo-inverse, daisy-chain, optimization-based and other, are well described theoretically in [37]. For the area of vehicle motion control, some of them have a wide application. Figure 1 shows statistics of relevant peer-reviewed publications from the SciVerse Scopus database and SAE Technical Paper Series.

Figure 2.4 Analysis of publications related to the control allocation methods in integrated vehicle control: SciVerse Scopus (blue); SAE Technical Paper Series (red)

Pseudo-inverse control allocation based on Moore-Penrose inversion is an effective technique from the point of view of computational cost, because it requires only algebraic computation [38, 39]. For instance, in [39] the coordination of three systems, such as friction brake system, front steering and slip differential at the rear axle, is considered. The P-controller is a high-level controller calculating generalized yaw torque. The simulation of medium-sized luxury sports car is carried out for ‘Sine with Dwell’ manoeuvre.

A flatness-based tracking controller operating with yaw rate, vehicle side slip angle and vehicle velocity is applied in [40]. Control demand is allocated between single-wheel steering, brake, drive and suspension subsystems using Moore-Penrose inversion. The simulation manoeuvre is 0.5 Hz single-sine steer input.

However, the major concern of classical pseudo-inverse control allocation is the neglect of control constraints omitting the actuator dynamics. Taking into account the actuator constraints or tire friction limits, the modified pseudo-inverse control allocation can be advanced with the introduction of variable weights [41].

Optimization-based control allocation with position, rate and acceleration constraints can effectively solve the allocation problem. For instance, the authors [29] present the yaw control, obtained from the Lyapunov function, with limitation of the roll angle for light commercial van. The control allocation is based on weighted-norm constraints and a quadratic programme. The simulation manoeuvre is ‘Fishhook’ test with an initial speed of 80 km/h.

The allocation of generalized longitudinal and lateral forces and yaw torque between engine and brake torques is presented in [42]. The control law for the high-level controller is based on feedback linearization of a nonlinear vehicle model and PI-controller with anti-windup based on back calculation. To solve the optimization problem the weighted least square method is used with constraints including position and rate limits of actuators, and force limits for each tire. The simulation manoeuvre is ‘Sine with Dwell’.

In paper [43] a control allocator based on a quadratic programming (QP) problem with inequality constraints is proposed. The linear quadratic regulator and full state feedback are used for the high-level controller of yaw rate and sideslip angle. To prevent unfeasible solutions of the QP problem, the sign preserving QP method is used with the introduction of slack variables for each control attempt. The proposed controller is investigated by the simulation of a double lane change manoeuvre. Three different scenarios of subsystem failures are used to show reconfiguration ability.

The control structure in [33] includes a proportional controller of vehicle dynamics and control allocation based on static mapping and multi-parameter nonlinear programming. The piecewise linear approximate solution functions are obtained offline. The cost function represents a tracking error of generalized yaw torque and longitudinal wheel slips. This control objective can be considered as the minimization of energy loss. Control demand is allocated between front wheel steer angles and brake torques.

In [11] a control algorithm based on dynamic inversion of a non-linear vehicle model and force allocation using non-linear optimization with optimization constraints including adhesion potential utilization and limits of actuator dynamics is developed. The considered control system is investigated in the framework of single-lane change manoeuvre on wet road with vehicle velocity of 100 km/h and deceleration of 2 m/s^2 after 0.5s.

The main disadvantage of optimization-based control allocation is the high requirement to computational performance and load in the case of real-time application. To increase computational speed of optimization-based control allocation, accelerated fixed-point method with termination of iterations while solution distance exceeds allocation tolerance to improve convergence rate and reduce computational complexity is

proposed in [31]. Another approach is directed to combine various optimization methods. The hybrid optimization solver which combines pseudo-inverse and fixed-point iteration methods is discussed in [44]. The hybrid steepest descent method for dynamic allocation of tire forces is discussed in [30]. The tire forces in this method trend in the optimal direction, but their values are not necessary optimal at each instant. As is alleged by authors, the advantage of hybrid steepest descent method is the reduction of computational load.

Instead of the application of optimization-based control allocation, which requires a solution at each instant, *dynamic control allocation method* using dynamic update laws for control inputs is proposed in [32]. The comparison between static and dynamic control allocation demonstrates higher computational efficiency of dynamic control allocation; however, its performance during manoeuvres with high nonlinear behaviour of vehicle and tires is lower compared to static control allocation [45]. The authors compare static allocation with nonlinear saturation constraints of tire forces solved by interior-point method and dynamic allocation by Newton-like update law and Lagrange multipliers.

Depending on parameters included in the input control vector, the control allocation can be ‘direct’ or ‘indirect’ [46]. The first type is defined by actuator parameters. The second one uses intermediate parameters into the control input vector. The type of control allocation depends on actuator control inputs, desired cost functions, inverse tire model, number of active subsystems and other factors.

Some examples of allocation configurations of input control are given below:

- longitudinal and lateral tire forces [29, 30]. Only the problem of force allocation has been solved;
- kinematic parameters of the tire (wheel slip and slip angles). Based on tire inversion model, tire kinematic parameters can be obtained for force allocation [31, 47];
- reference control inputs of actuators (brake pressure / torque, drive torque, steering angle) [48, 49]. Both, tasks, such as force allocation and subsystem coordination, can be solved jointly;
- mixed force/actuator output, for instance, longitudinal forces and steering angle [43, 50].

2.2.2 Research issues in control allocation

The following research issues in the field of control allocation can be pointed:

- representation of actuator dynamics;
- approximation of nonlinear allocation;
- adaptation of control allocation to uncertainties and disturbances;
- achievement of several objectives known as multi-objective problem.

Although optimization-based methods consider actuator position and rate limits, actuator dynamics is still statically represented. Thereby, to handle actuator dynamics several solutions can be applied:

- penalization of incremental steps of control inputs taking into account previous behaviour can be applied [51];
- compensation of phase shift between desired and actual control effort [52];
- application of Model Predictive Control Allocation [53, 54].

In addition, when the desired demands have different frequencies, the frequency-apportioned control allocation [55] can separate control demand between high and low frequency components by a low-pass filter matrix.

One of the significant aspects of control allocation is the approximation of tire forces and characteristics of electric motors. The tire constraints can be obtained by piecewise linear approximation (rhombus) [30], friction ellipse concept [56] or more complex approximation techniques [57]. For the approximation of motor characteristics the 3rd and 4th-order polynomial can be used [58]. The losses of electric motor can be represented by the 2nd order polynomials [59].

Due to measurement noise, external disturbances and probability of actuator failure, a dynamic adaptive law of control effectiveness matrix to minimize uncertainties and to increase adaptability is proposed in [60]. One of the special issues of control allocation adaptability is its effectiveness in the case of presence of actuator failures. The algorithms [61] of active fault diagnosis and control are proposed to maintain vehicle stability in the case of the fault of one electric motor using control gain estimation. The non-iterative fault tolerant control of electric motors based on off-line control allocation laws to maintain vehicle stability is proposed in [62]. In [63] the different scenarios of failures in the brake-by-wire system are considered to guarantee braking performance by the changing of the elements of weighting matrix.

2.2.3 Review of multi-objective problem in control allocation

The cost function of control allocation problem typically covers two terms related to the minimization of allocation error and control actuations. The additional objectives, which can be introduced into cost function, can be:

- total instantaneous power consumption for all in-wheel motors in different modes. The method called “energy-efficient control allocation”. The additional term for one actuation mode is represented as [58]:

$$J_{add} = \sum_{i=1}^n P_{em,i} \quad (2.1)$$

- longitudinal wheel slip ratios [64]:

$$J_{add} = \xi_{slip} \|W_{slip} (B_u^{slip} u^{CA} - T_{wi})\| \quad (2.2)$$

- electric and mechanical losses of electric motors [65]:

$$J_{add} = \int_t^{t+T_s} (P_{inv,const} + (\tau_{fric} + k_{hyst})|\omega| + k_{eddycurrent}\omega^2 + a|\tau| + b\tau^2) d\tau \quad (2.3)$$

- simplified cost function with the minimization of steering rate and the maximization of recuperation [65]. Both the cost terms demonstrate similar results in overall energy consumption.

In [66] the following auxiliary cost terms are investigated:

- minimization of overall input motor power as in Eq. 2.1;
- minimization of the standard deviation of wheel slips:

$$J_{add} = \sqrt{\sum_{i=1}^n 0.25s_{wi}^2 - \bar{s}_w^2} \quad (2.4)$$

- minimization of the total longitudinal slip power loss:

$$J_{add} = \sum_{i=1}^n P_{loss,swi} \quad (2.5)$$

- minimization of the sum of the tire force coefficients:

$$J_{add} = \sum_{i=1}^n \frac{\sqrt{F_{xwi}^2 + F_{ywi}^2}}{F_{zi}} \quad (2.6)$$

The simulation-based analysis on vehicle steady-state performance [66] shows that cost functions with wheel slips have insignificant influence on energy consumption compared to energy-based criteria.

Depending on additional control objectives, a number of auxiliary terms into the cost function can be increased. Multi-term cost functions have the following undesired features:

- characteristic of auxiliary term should be known and described by the approximation polynomial or presented as a look-up table. As an example, it can be the efficiency map of electric motor, characteristic of electric losses or tire power dissipation. Some of them like efficiency map can be found from the specification of electric motor, when others like tire power dissipation require additional complex investigation;
- mixture of auxiliary terms into the cost function can cause its smoothing even non-convexity in the case when the weights of auxiliary terms are close to the weights of main terms. As a result, a number of required iterations and total computational load will increase;
- extreme of cost function depends on the pre-defined weights of all terms, when their values are independent from vehicle manoeuvres.

Instead of adding cost terms, another way is to select a subset of these cost terms based on the intended manoeuvres and to use a penalization of control inputs. The following examples demonstrate possible applications:

- coordination of brake pressures according to normal force distribution during heavy vehicle braking to obtain better vehicle stability [67];
- weighting matrix is a function of friction circles for the investigation of manoeuvrability of six-wheeled and skid-steered vehicle for on-road and off-road conditions [68];
- elements of weighting matrix are used as reliability indicators of actuators and the distribution of control demand is carried out taken into account actuator health and failures [69];
- saturation of longitudinal forces is realized by the definition of the weighting matrix taking into account normalized tire slip [70];
- saturation of longitudinal and lateral forces taking into account wheel slips and slip angles [41];
- dynamic weight scheduling to achieve lower energy consumption without significant impairment of stability of motion and vehicle handling compared to control allocation with fixed weight distribution [71].

2.2.4 Implementation of control allocation for automotive tasks

The results of control allocation investigation in the majority of research publications is limited by the simulation investigations in Simulink [38, 72] or in the software fusion of Simulink with commercial software, such as CarSim [31, 41, 64], IPG CarMaker [73], veDyna [74] and others [11, 29]. The investigation of control allocation for electro-hydraulic brake system and axle electric motor using the real-time simulation via dSPACE 1006 and DS MicroAutoBox is discussed in [47]. The investigations of longitudinal dynamics of experimental electric ground vehicle with independently actuated in-wheel motors based on ‘energy-efficient control allocation’ method from [58] is proposed in [75]. The allocation of friction brake torques for lane-keeping assistance control of real test vehicle during double lane change is considered in [76].

From industrial point of view there are several patents related to control allocation. The control system including control allocation and phase compensation of actuator commanded signal is patented by SAAB AB [77]. The solution has a general view and it covers control allocation of actuations of any vehicle subsystem. The real-time allocation of actuator torque by friction brakes and wheel motors is patented by GM [78]. The control allocation method is based on a Least Squares formulation. It should be noted, that in the patent the prioritization of vehicle subsystems using weighting matrices is mentioned without the detailed recommendations regarding the organization of prioritization.

2.3 Thesis Objectives

Taking into account the study motivation and literature review, the overall research goal of the thesis is the development and evaluation of a control system coordinating priorities between vehicle subsystems for a multi-actuated vehicle and the demonstration of concurrent improvement of several vehicle operation characteristics. In order to achieve the abovementioned goal, the following objectives are formulated:

- to derive detailed models of a nonlinear vehicle and its subsystems for the study of the effect of subsystem integration;
- to investigate an effect of hysteresis in the brake mechanism of friction brake system, which can cause significant delays and, as a result, a wrong actuation;
- to design a multi-layer control system for maintaining vehicle stability and handling and the minimization of energy consumption and energy losses using subsystem prioritization;
- to analyse necessity to use multiple actuators to reach several vehicle operation characteristics;
- to investigate the influence of subsystem coordination with prioritization on various assessment criteria of vehicle dynamics, energy consumption and energy losses;
- to propose an algorithm of the coordination of priorities between vehicle subsystems during vehicle motion;
- to evaluate the proposed algorithms using a hardware-in-the-loop test rig with hardware components of conventional brake system and dynamic tire pressure system in real-time domain.

2.4 Chapter summary

To summarize the literature review, the following conclusions can be inferred:

- two to four vehicle subsystems involved in the regulation process are typically considered;
- the prevalent technique to distribute control demand for over-actuated systems is static optimization-based control allocation;
- together with the minimization of allocation error and control efforts, the additional objectives related to energy consumption and energy losses can be introduced;
- instead of the introduction of additional terms into the cost function, a penalization/prioritization of control inputs according to certain objectives can be applied;
- evaluation of a control system with control allocation is typically carried out only by the simulation investigations.

Thesis objectives have been formulated taking into account the study motivation and literature review.

Chapter 3

3 Modelling of vehicle and its subsystems

The models of a passenger car and its actuators are discussed in this chapter. Two vehicle models are considered: original 14 DoF vehicle for preliminary investigations [71, 72] and a multibody vehicle model in commercial software IPG CarMaker for final investigations presented in the thesis.

It should be noted, that the following assumptions related to vehicle and subsystems models were taken into account:

- it is assumed that the measured signals from vehicle sensors are available and the measured noise can be ignored;
- the motion is investigated on a flat road without external disturbances like wind, fluctuation of friction coefficient and others. The following assumption is reasonable, because the manoeuvres will be investigated as considered in the ISO standards;
- dynamics of the actuators can be simplified and, in some cases, described by transfer functions using the recommendations provided by other researchers.

The original vehicle model describes a vehicle body motion in space as well as vertical and rotational motions of each wheel. The description of vehicle subsystems includes:

- wheel steer actuators generate additional steer angles by first order plus time delay (FOPTD) transfer function;
- camber angle actuators are described by FOPTD transfer function;
- electric powertrain includes battery, near-wheel electric motors with single-speed gearboxes and half-shafts. The model of an electric motor is represented by technical characteristics of a real electric motor. The transient behaviour of the electric motors is described by FOPTD transfer function. The transmission and half-shaft dynamics is represented by a system of differential equations;
- electro-hydraulic friction brake system is described by a system of differential equations;
- additional normal forces are generated by the actuators through external spring, damping and stabilizer forces. They are generated by actuator models of variable

suspension, active damping and active anti-roll bars described by FOPTD transfer functions;

- dynamic tire pressure system is represented by the FOPTD transfer function based on step response characteristics obtained from experimental tests;
- to characterize the steady-state tire behaviour, the Pacejka tire model including inflation/deflation pressures is used. For transient behaviour, it is extended by a relaxation length model.

To verify the developed model, the transient test manoeuvre with different amplitudes of steering angle is investigated. The developed model is compared to a multibody vehicle model from commercial software. The investigated vehicle is a passenger car with a mass of 1534 kg, weight front/rear distribution 55/45 and tire size 205/60R15. The numerical parameters of vehicle, tire and subsystems are listed in Appendix A.1 (original model, tire and subsystems) and Appendix A.2 (IPG CarMaker model).

3.1 A nonlinear model for vehicle and tire

3.1.1 A nonlinear vehicle model

An adequate model of vehicle dynamics is the first requirement for the investigations of vehicle motion. The vehicle behaviour is defined by weight distribution, tire properties, suspension, external applied forces and other factors. The number of degrees of freedom (DoF) of the vehicle model and, accordingly, the number of differential equations varies from two for the classical bicycle model up to 38 DoF models for complex investigations [79]. In case of multibody commercial software, for instance, MSC.ADAMS/Car, it can be up to 160 DoF. The typical approach is the application of the 8 DoF [23, 80] vehicle model, and rarely the 14 DoF model [81-83]. The advantage of 14 DoF is that vehicle subsystems influencing on pitch and vertical dynamics can be involved in the process of regulation. The second advantage is a more realistic description of vehicle behaviour in comparison with low-order DoF models.

Original model

The vehicle body motion in space has six DoF (longitudinal, lateral, vertical, roll, pitch and yaw motions). Assuming the vehicle is travelling on a smooth flat road and the

road disturbances are equal to zero, the following equations of the motion describe the dynamics of the vehicle body [83, 84]. The schematic model of the vehicle is shown in Figure 3.1.

Figure 3.1 14-DoF vehicle model

Longitudinal motion:

$$m_a \left(\frac{dV_{xa}}{dt} - V_{ya} \frac{d\psi}{dt} + V_{za} \frac{d\theta}{dt} \right) - m_s \left(\frac{d\phi}{dt} \frac{d\psi}{dt} + \frac{d^2\theta}{dt^2} \right) h_s \cos \phi = \sum F_x \quad (3.1)$$

Lateral motion:

$$m_a \left(\frac{dV_{ya}}{dt} + V_{xa} \frac{d\psi}{dt} - V_{za} \frac{d\phi}{dt} \right) - m_s \left(\frac{d\theta}{dt} \frac{d\psi}{dt} - \frac{d^2\phi}{dt^2} \right) h_s \cos \phi = \sum F_y \quad (3.2)$$

Vertical motion:

$$m_s \left(\frac{dV_{za}}{dt} - V_{xa} \frac{d\theta}{dt} + V_{ya} \frac{d\phi}{dt} \right) + m_s \left(\frac{d\phi^2}{dt} + \frac{d\theta^2}{dt} \right) h_s \cos \phi = \sum F_z \quad (3.3)$$

Roll motion:

$$I_{xss} \frac{d^2\phi}{dt^2} - I_{xzs} \left(\frac{d\phi}{dt} \frac{d\theta}{dt} + \frac{d^2\psi}{dt^2} \right) - m_s a_y h_s \cos \phi = \sum M_\phi \quad (3.4)$$

Pitch motion:

$$I_{yys} \frac{d^2\theta}{dt^2} - (I_{zss} - I_{xss}) \frac{d\phi}{dt} \frac{d\psi}{dt} + I_{xzs} \left(\frac{d\phi^2}{dt} - \frac{d\psi^2}{dt} \right) + m_s a_x h_p \cos \theta = \sum M_\theta \quad (3.5)$$

Yaw motion:

$$I_{zss} \frac{d^2\psi}{dt^2} + I_{xzs} \left(\frac{d\theta}{dt} \frac{d\psi}{dt} - \frac{d^2\phi}{dt^2} \right) + m_s a_x h_s \sin \phi = \sum M_z \quad (3.6)$$

The right part of the equation system is given below:

$$\begin{aligned}
\sum F_x &= \sum_{i=1}^4 F_{xi} - F_x^{aero}; & \sum F_y &= \sum_{i=1}^4 F_{yi} - F_y^{aero}; & \sum F_z &= \sum_{i=1}^4 F_{zi} \\
\sum M_\phi &= m_s g h_s \sin \phi - \frac{t_f}{2} (F_{s1} - F_{s2}) - \frac{t_r}{2} (F_{s3} - F_{s4}) \\
\sum M_\theta &= m_s g h_p \sin \theta - a (F_{s1} + F_{s2}) + b (F_{s3} + F_{s4}) \\
\sum M_z &= a (F_{y1} + F_{y2}) - b (F_{y3} + F_{y4}) + \frac{t_f}{2} (F_{x1} - F_{x2}) + \frac{t_r}{2} (F_{x3} - F_{x4}) + \sum_{i=1}^4 M_{zwi}
\end{aligned} \tag{3.7}$$

The full vehicle mass is calculated as:

$$m_a = m_s + m_{uf} + m_{ur} \tag{3.8}$$

The longitudinal distance from sprung center of gravity (CoG) to total mass CoG is:

$$l_{cgs} = \frac{m_{ur} b_s - m_{uf} a_s}{m_a} \tag{3.9}$$

The distances from total CG to front axle and to rear axle are:

$$a = a_s + l_{cgs}; \quad b = b_s - l_{cgs} \tag{3.10}$$

Inertia of front and rear unsprung masses about their own CG:

$$I_{zzuf} = 0.25 m_{uf} t_f^2; \quad I_{zzur} = 0.25 m_{ur} t_r^2 \tag{3.11}$$

Total yaw inertia of the vehicle is:

$$I_{zz} = I_{zss} + m_s l_{cgs}^2 + I_{zzuf} + m_{uf} a^2 + I_{zzur} + m_{ur} b^2 \tag{3.12}$$

Height of origin above ground is:

$$h_o = h_f + a \frac{h_r - h_f}{a + b} \tag{3.13}$$

Distance from the roll center to the sprung mass center is:

$$h_s = h_{cgs} - h_o \tag{3.14}$$

Height of front and rear unsprung masses above the origin are:

$$h_{uf} = h_{cuf} - h_o; \quad h_{ur} = h_{cur} - h_o \tag{3.15}$$

The tire projections along the vehicle axes F_{xi} and F_{yi} are related to tire forces along wheel axes:

$$\begin{bmatrix} F_{xi} \\ F_{yi} \end{bmatrix} = \begin{bmatrix} \cos \delta_i & -\sin \delta_i \\ \sin \delta_i & \cos \delta_i \end{bmatrix} \begin{bmatrix} F_{xwi} \\ F_{ywi} \end{bmatrix}, i = 1, \dots, 4 \tag{3.16}$$

The tire normal forces are defined by static weight distribution, suspension forces at struts and the load transfers owed to the longitudinal and lateral accelerations. The tire normal forces F_{zi} are calculated as [85]:

$$\begin{aligned}
F_{z1} &= \frac{m_a g b}{2L} - F_{s1} SWF_1 - F_z^{long} - F_{z,front}^{lat} \\
F_{z2} &= \frac{m_a g b}{2L} - F_{s2} SWF_2 - F_z^{long} + F_{z,front}^{lat} \\
F_{z3} &= \frac{m_a g a}{2L} - F_{s3} SWF_3 + F_z^{long} - F_{z,rear}^{lat} \\
F_{z4} &= \frac{m_a g a}{2L} - F_{s4} SWF_4 + F_z^{long} + F_{z,rear}^{lat}
\end{aligned} \tag{3.17}$$

The suspension force F_{si} at strut is calculated as [85]:

$$F_{si} = K_{si} (z_{ui} - z_{bi}) + C_{si} (\dot{z}_{ui} - \dot{z}_{bi}) + F_{bump,i} BSF \tag{3.18}$$

Vertical wheel motion is defined as [85]:

$$\begin{aligned}
0.5m_{uf}\ddot{z}_{u1} &= K_{t1} (\xi_1 - z_{u1}) - F_{s1} SWF_1 - F_z^{long} - F_{z,front}^{lat} \\
0.5m_{uf}\ddot{z}_{u2} &= K_{t2} (\xi_2 - z_{u2}) - F_{s2} SWF_2 - F_z^{long} + F_{z,front}^{lat} \\
0.5m_{ur}\ddot{z}_{u3} &= K_{t3} (\xi_3 - z_{u3}) - F_{s3} SWF_3 + F_z^{long} - F_{z,rear}^{lat} \\
0.5m_{ur}\ddot{z}_{u4} &= K_{t4} (\xi_4 - z_{u4}) - F_{s4} SWF_4 + F_z^{long} + F_{z,rear}^{lat}
\end{aligned} \tag{3.19}$$

Vertical body motion at small angles can be simplified to:

$$\begin{aligned}
z_{b1} &= z_a + 0.5t_f \phi - a\theta \\
z_{b2} &= z_a - 0.5t_f \phi - a\theta \\
z_{b3} &= z_a + 0.5t_r \phi + b\theta \\
z_{b4} &= z_a - 0.5t_r \phi + b\theta
\end{aligned} \tag{3.20}$$

The load transfer F_z^{long} owed to the longitudinal acceleration is calculated as:

$$F_z^{long} = \frac{m_s h_s + m_{uf} h_{uf} + m_{ur} h_{ur}}{2L} a_x \tag{3.21}$$

The load transfers $F_{z,front}^{lat}$ and $F_{z,rear}^{lat}$ owed to the lateral accelerations are defined as:

$$\begin{aligned}
F_{z,front}^{lat} &= \frac{a_y}{t_f} \left(\frac{m_s h_f b}{L} + m_{uf} h_{uf} \right) \\
F_{z,rear}^{lat} &= \frac{a_y}{t_r} \left(\frac{m_s h_r a}{L} + m_{ur} h_{ur} \right)
\end{aligned} \tag{3.22}$$

The load transfers $F_{z,front}^{roll}$ and $F_{z,rear}^{roll}$, owed to the roll stiffness and roll damping, are defined by suspension springs and dampers [86]:

$$\begin{aligned} F_{z,front}^{roll} &= -\frac{1}{t_f} \left(K_{\phi f} \phi + C_{\phi f} \frac{d\phi}{dt} \right) \\ F_{z,rear}^{roll} &= -\frac{1}{t_r} \left(K_{\phi r} \phi + C_{\phi r} \frac{d\phi}{dt} \right) \end{aligned} \quad (3.23)$$

The load transfers $F_{z,front}^{pitch}$ and $F_{z,rear}^{pitch}$ owed to the pitch behaviour are:

$$\begin{aligned} F_{z,front}^{pitch} &= 0.5a \left(K_{\theta f} \theta + C_{\theta f} \frac{d\theta}{dt} \right) \\ F_{z,rear}^{pitch} &= 0.5b \left(K_{\theta r} \theta + C_{\theta r} \frac{d\theta}{dt} \right) \end{aligned} \quad (3.24)$$

The rolling resistance force F_{fi} under curvilinear motion is defined by the combination of mathematical models in [87] and [88]:

$$F_{fi} = p_{wi}^{n_1} F_{zi}^{n_2} \left(k_{f1} + k_{f2} V_{xwi} + k_{f3} V_{xwi}^2 + \mu_y \sin \alpha_i \right) \quad (3.25)$$

The air drag force F^{aero} is calculated as [89]:

$$F^{aero} = \frac{1}{2} \rho_{air} C_d A_{front} V_{xa}^2 \quad (3.26)$$

IPG CarMaker model

For the final investigations of the coordination of priorities between vehicle subsystems and HIL testing, a vehicle simulator realized in the IPG CarMaker software has been developed. The simulator allows the integration of diverse mathematical and physical models of the vehicle and the vehicle subsystems on a unified software platform:

- vehicle body – rigid model taking into account the mass distribution between wheel carriers, wheels, and body;
- 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 the models for spring, damper and stabilizer;
- steering – dynamic steer ratio model and wheel steer actuators generating external steer angles, which are added to steer angles generated by the IPG CarMaker;

- camber angle actuators – external camber angles are added to camber angles generated by the IPG CarMaker;
- tires – Pacejka tire including tire pressure with relaxation model (described below);
- powertrain – electric powertrain (described below);
- brake system – electro-hydraulic brake system (described below);
- aerodynamics – 1D look-up table for x, y and z axis;
- road – numerical model of road geometry.

3.1.2 Transient tire model

The tire behaviour plays a key role in vehicle dynamics, stability and handling. The tire forces have a higher impact on vehicle behaviour in comparison with air drag, rolling resistance and other forces. Therefore, the precise model of tire behaviour is the primary requirement to investigate vehicle motion control successfully.

Kinematic parameters

The wheel dynamics is represented as:

$$J_{wi} \frac{d\omega_{wi}}{dt} = T_{wi} - T_{bri} - F_{xwi} r_{wi} - F_{ywi} r_{wi} \quad (3.27)$$

The velocities of wheel centres along the vehicle fixed axes V_{xi} and V_{yi} are:

$$\begin{aligned} V_{x,i} &= V_{xa} \mp 0.5t_f \dot{\psi}, & i = 1, 2 & \quad V_{y,i} = V_{ya} + a\dot{\psi}, & i = 1, 2 \\ V_{x,i} &= V_{xa} \mp 0.5t_r \dot{\psi}, & i = 3, 4 & \quad V_{y,i} = V_{ya} - b\dot{\psi}, & i = 3, 4 \end{aligned} \quad (3.28)$$

The wheel slip angles α_{wi} are calculated as:

$$\alpha_{wi} = \tan^{-1} \left(\frac{V_{yi}}{V_{xi}} \right) - \delta_i \quad (3.29)$$

The velocities in the tire-road contact V_{xwi} and V_{ywi} are calculated as:

$$\begin{bmatrix} V_{xwi} \\ V_{ywi} \end{bmatrix} = \begin{bmatrix} \cos \delta_i & \sin \delta_i \\ -\sin \delta_i & \cos \delta_i \end{bmatrix} \begin{bmatrix} V_{xi} \\ V_{yi} \end{bmatrix}, i = 1, \dots, 4 \quad (3.30)$$

The longitudinal wheel slips s_{wi} can be found as:

$$s_{wi} = \begin{cases} \frac{\omega_{wi} r_{wi} - V_{xwi}}{\omega_{wi} r_{wi}} & \omega_{wi} r_{wi} \geq V_{xwi} \quad (acceleration) \\ \frac{\omega_{wi} r_{wi} - V_{xwi}}{V_{xwi}} & \omega_{wi} r_{wi} < V_{xwi} \quad (braking) \end{cases} \quad (3.31)$$

Steady-state and transient tire model

There are hundreds of different mathematical models of tire behaviour. The first and most popular class is steady-state models. The models are mainly derived from different variants of the brush model and they are based on empirical approximations of the experimental results. Predominated one is the ‘Magic Formula’, later known as the ‘Pacejka Tire model’. The model uses up to 160 parameters to describe force and torque contact components [90].

To simulate the changing of inflation pressure, the extension of this model for longitudinal slip and cornering stiffness, longitudinal and lateral maximum friction, and magnitude of the pneumatic trail are used from [91]. The longitudinal / lateral force and self-aligning torque obtained by the extended tire model are shown in Figure 3.2. The normal force is 5000 Nm, friction – 1.0 and nominal inflation pressure – 3 bar.

When the loads are applied, owing to elasticity the tire does not produce forces instantaneously. Therefore, there is a delay between applied torque and the final reaction of the wheel. The Pacejka tire model is appropriate only for the steady-state vehicle motion; however, the vehicle control always starts to operate during the transient motion in nonlinear or near limit of friction conditions. To improve the accuracy of the steady-state model, the behaviour of transient tire dynamics should be considered. There are various ways to implement transient behaviour. The first-order force lag tire dynamics demonstrates suitable accuracy and it can be used for the description of transient process [92, 93]. Some researchers noted that the first-order force lag can be insufficient to describe precisely the wide range of vehicle motion. Therefore, the higher-order force lag [94] or the dynamic tire model like the LuGre model [95] can be used.

The lag of forces and torque are calculated for small slip values:

$$\begin{aligned} \frac{L_{xi}}{V_{xwi}} \dot{F}_{xwi} &= F_{xwi}^{ss} - F_{xwi} \\ \frac{L_{yi}}{V_{xwi}} \dot{F}_{ywi} &= F_{ywi}^{ss} - F_{ywi} \\ \frac{L_{zi}}{V_{xwi}} \dot{M}_{zwi} &= M_{zwi}^{ss} - M_{zwi} \end{aligned} \quad (3.32)$$

Figure 3.2 Longitudinal, lateral forces and self-aligning torque depending on wheel slip, slip angle, tire pressure, camber angle and normal force

In this thesis, the influence of tire pressure changing on tire transient behaviour is neglected, because the dynamics of tire pressure system is slow up to 1 bar/s. However, this issue can be critical, when the system provides fast pressure variation.

3.2 Vehicle subsystems

3.2.1 Friction brake system

The electro-hydraulic brake system [96] consists of a tandem master cylinder with spring pedal travel simulator, brake fluid reservoir, block valve, compensative valve, hydraulic pump with pump motor, high-pressure hydraulic accumulator, control inlet/outlet valves for pressure build-up and decrease at each wheel, and brake mechanisms. The effects related to fluid inertia and brake line are neglected. The half of hydraulic scheme for one brake circuit combined to a control system of brake pressure is depicted in Figure 3.3.

The mathematical model of the electro-hydraulic brake system is based on [97]. The pump volumetric flow Q_{pump} depends on accumulator pressure and it is represented by a look-up table [98]. The typical operation range of the pressure accumulator is from 160 bar to 180 bar. The following control logic is used to control the pump:

- if accumulator pressure is less than 160 bar, then the pump is switched on;
- if accumulator pressure is more than 180 bar, then the pump is switched off.

The volume change in the accumulator $\dot{V}_{acc}$ can be found as:

$$\dot{V}_{acc} = Q_{pump} - \sum_{i=1}^4 Q_i \quad (3.33)$$

Figure 3.3 Electro-hydraulic brake system (half view)

Volume changes in the i^{th} calliper $\dot{V}_{bwc i}$ is calculated as:

$$\begin{aligned}\dot{V}_{bwci} &= Q_i + Q_j^{compen} - Q_i^{dump} & i &= \{1, 3\}; j = 1, 2; \\ \dot{V}_{bwci} &= Q_i + Q_j^{block} - Q_j^{compen} - Q_i^{dump} & i &= \{2, 4\}; j = 1, 2;\end{aligned}\quad (3.34)$$

As the braking process occurs during a short-time period, the accumulator does not have a sufficient time to transfer heat to the environment. Therefore, its behaviour can be described as an adiabatic process [99, 100]. The second assumption is that gas pressure equals to fluid pressure at the accumulator inlet [99].

Pressure in the accumulator p_{acc} is calculated as:

$$p_{acc} = p_{acc,0} \left(\frac{V_{acc,0}}{V_{acc}} \right)^\gamma \quad (3.35)$$

$$V_{acc} = V_{acc,0} - \int \dot{V}_{acc} dt \quad (3.36)$$

Volumetric flow Q_i from accumulator to the i^{th} calliper across the i^{th} inlet valve is:

$$Q_i = u_i^{inlet} C_{flow} A_i^{inlet} \sqrt{\frac{2}{\rho_{bf}} |p_{acc} - p_{br,i}| \text{sign}(p_{acc} - p_{br,i})} \quad i = 1, 4; \quad (3.37)$$

Volumetric flow Q_i^{dump} from the i^{th} calliper to the reservoir across the i^{th} outlet valve is:

$$Q_i^{dump} = u_i^{outlet} C_{flow} A_i^{outlet} \sqrt{\frac{2}{\rho_{bf}} |p_{br,i} - p_{low}|} \quad i = 1, 4; \quad (3.38)$$

Volumetric flow Q_j^{block} from the tandem master cylinder to the i^{th} calliper across the j^{th} block valve is:

$$Q_j^{block} = u_j^{block} C_{flow} A_j^{block} \sqrt{\frac{2}{\rho_{bf}} |p_{mc} - p_{br,j}^{right}| \text{sign}(p_{mc} - p_{br,j}^{right})} \quad j = 1, 2; \quad (3.39)$$

Volumetric flow Q_j^{block} from the right wheel cylinder of j^{th} axle to the left wheel cylinder of j^{th} axle across the j^{th} compensative valve is:

$$Q_j^{compen} = u_j^{compen} C_{flow} A_j^{compen} \sqrt{\frac{2}{\rho_{bf}} |p_{br,j}^{right} - p_{br,j}^{left}| \text{sign}(p_{br,j}^{right} - p_{br,j}^{left})} \quad j = 1, 2; \quad (3.40)$$

Neglecting small inertias of the pistons in the master cylinder, return spring forces and seal friction forces, the pressures p_{mcp} and p_{mcs} for the primary and secondary sections of the tandem master cylinder can be written as:

$$p_{mcp} = p_{mcs} = \frac{F_p}{A_{mc}} \quad (3.41)$$

Pressures p_{bri} in the callipers versus volume change is taken from the experimental data as look-up tables [98].

The dynamics of electromagnetic valves are considered as the second order transfer function with bandwidth is 60 Hz and damping ratio of 1.3 [101]:

$$\frac{u_{valve}^{act}}{u_{valve}^{ss}} = \frac{\omega_n^2}{s^2 + 2\xi\omega_n s + \omega_n^2} \quad (3.42)$$

The normal contact force generated by the calliper pressure between the brake pad and disc on each wheel is:

$$F_{ni}^{contact} = \begin{cases} (p_{br,i} - p_{br0}) \frac{\pi d_{cyl}^2}{4}, & p_{br,i} > p_{br0} \\ 0, & p_{br,i} \leq p_{br0} \end{cases} \quad (3.43)$$

The model of the brake mechanism is extended by the Bouc-Wen model. The model description and analysis of the influence of hysteresis effect on vehicle dynamics are presented in the Chapter 4.

3.2.2 Electric powertrain

An electric powertrain includes a battery and near-wheel electric motors with single-speed gearboxes. Thevenin battery model is used [102]. It consists of an ideal no-load battery voltage U_{0B} , internal resistance R_{0B} , capacity C_B and overvoltage resistance R_{1B} :

$$\begin{cases} U_B = U_{0B} - I_B R_{0B} - U_{C_B} \\ \dot{U}_{C_B} = \frac{I_B}{C_B} - \frac{U_{Th}}{R_{1B} C_B} \end{cases}, \quad (3.44)$$

A schematic view of powertrain layout for one wheel is shown in Figure 3.4.

Figure 3.4 A schematic view of powertrain layout for one wheel [103]

The model of an electric motor describes steady-state behaviour by a look-up table of motor characteristics and its transient behaviour by FOPTD transfer function [104]. The torque-speed characteristics for the electric motor are shown in Figure 3.5. The part-load characteristics are defined by the level of the control signal $u_{em,i}$ obtained from the controller of the electric motor.

Figure 3.5 Torque-speed and efficiency characteristics of the electric motor

The transient behaviour of the electric motor is represented as [105]:

$$\frac{T_{em,i}}{T_{em,i}^{ss}} = \frac{1}{0.0022s + 1} e^{-0.002s} \quad (3.45)$$

The powertrain layout consists of near-wheel electric motors with single-speed gearboxes and half-shafts to transmit output torque from electric motor to the wheel. The transmission dynamics can be described as [105]:

$$\left(J_{em} + J_{tr1} + \frac{u_{tr}^2}{\eta_{tr}} \left(J_{tr2} + \frac{0.5J_{hs}}{\eta_j^{inner}} \right) \right) \ddot{\theta}_{em,i} = T_{em,i} - \frac{u_{tr}}{\eta_{tr}} T_{hs,i} \quad (3.46)$$

The half-shaft torques $T_{hs,i}$ are calculated as [105]:

$$T_{hs,i} = K_{hs} (u_{tr} \omega_{em,i} - \omega_{wi}) + C_{hs} (u_{tr} \theta_{em,i} - \theta_{wi}) \quad (3.47)$$

The transmitted torque T_{wi} to the wheel by the half-shaft can be founded [105]:

$$0.5J_{hs} \ddot{\theta}_{w,i} = T_{hs,i} - \frac{T_{wi}}{\eta_j^{outer}} \quad (3.48)$$

3.2.3 Four-wheel steering system and camber angle actuators

The steering system consists of conventional steering system and wheel steer actuators. They can generate individual steer angles for each wheel. The average steer angle of front axle is calculated as a driver input divided by the gear ratio of steering mechanism. In original model, the dynamics of conventional steering system is described as a second-order transfer function with a bandwidth of 10 Hz and a damping ratio of 0.707 [106]:

$$\frac{\delta_{av,f}}{\delta_{steer}^{driver}} = \frac{\omega_a^2}{s^2 + 2\xi_a \omega_a s + \omega_a^2} \frac{1}{u_{sm}} \quad (3.49)$$

The steer angles of the wheels δ_i are equal in total to the sum of the steer angles $\delta_i^{kinematic}$ generated by driver input, the steer angles $\delta_i^{dyn.effects}$ caused by dynamic effects and the steer angles δ_i^{ext} generated by individual wheel steer actuators:

$$\delta_i = \delta_i^{kinematic} + \delta_i^{dyn.effects} + \delta_i^{ext} \quad (3.50)$$

Based on Ackerman steering the kinematic steer angles are found as [107]:

$$\begin{cases} \delta_1^{kinematic} + \delta_2^{kinematic} = 0.5\delta_{av,f} \\ \cot \delta_1^{kinematic} - \cot \delta_2^{kinematic} = \frac{t_f}{L} \end{cases} \quad \delta_3^{kinematic} = \delta_4^{kinematic} = 0 \quad (3.51)$$

The steer angles $\delta_i^{dyn.effects}$ caused by dynamic effects are related to roll steer and compliance steer [108]. Roll steer is caused by wheel steering owed to the rolling motion of the sprung mass relative to the unsprung mass [109]. Additional parameter for the detailed description of steering dynamics is the compliance steer. It is caused by a number of suspension design factors like bushings at junctions of the suspension linkage or the flexibility:

$$\delta_i^{dyn.effects} = \epsilon_i^{roll} \phi_i + k_i^{CSFx} F_{xi} + k_i^{CSFy} F_{yi} + k_i^{CSMz} M_{zi} \quad (3.52)$$

The external front and rear steering angles generated by wheel steer actuators are calculated as [86]:

$$\frac{\delta_{f,i}^{ext}}{\delta_{f,i}^{ref}} = \frac{1}{0.01s + 1} e^{-0.002s}, i = 1, 2 \quad (3.53)$$

$$\frac{\delta_{r,i}^{ext}}{\delta_{r,i}^{ref}} = \frac{1}{2.53 \cdot 10^{-4} s^2 + 0.011s + 1} e^{-0.003s}, i = 3, 4 \quad (3.54)$$

Since only the prototypes of camber angle actuators are known, the information regarding actuation frequency and delays can be varied in the wide range. The frequency of variable camber suspension should be below 5-7 Hz [110]. The prototype of active camber system [111] allows to realize a camber actuation time of 0.3 s. The transfer function of camber angle actuator is selected on the basis of concept design as:

$$\frac{\gamma_i^{ext}}{\gamma_i^{ref}} = \frac{1}{0.15s + 1} e^{-0.020s} \quad (3.55)$$

3.2.4 Additional normal forces generated by actuators

To change distribution of normal forces, variable stiffness suspension, electrorheological dampers and electromechanical anti-roll bars can be applied. The dynamics of variable stiffness suspension is described as [112]:

$$\frac{F_{spring,i}^{ext}}{F_{spring,i}^{ref}} = \frac{1}{0.03s + 1} e^{-0.008s} \quad (3.56)$$

Since electrorheological dampers are considered instead of semi-active hydraulic dampers, the dynamics of active damping system will be faster [113]:

$$\frac{F_{damp,i}^{ext}}{F_{damp,i}^{ref}} = \frac{1}{0.005s + 1} e^{-0.005s} \quad (3.57)$$

The dynamics of active anti-roll bar depends on type of actuator. For hydraulic actuator time constant is more than 100 ms. For electromechanical actuator of active anti-roll bar time constant is experimentally around 50 ms [114]. The several publications propose to use time constant from 20-60 ms [86, 115]. The system dynamics is represented by:

$$\frac{F_{arb,i}^{ext}}{F_{arb,i}^{ref}} = \frac{1}{0.05s + 1} e^{-0.010s} \quad (3.58)$$

3.2.5 Dynamic tire pressure system

Dynamic tire pressure system can change pressure in each tire independently. The system configuration includes a pressure compressor (or compressed air bottle), pneumatic reservoirs and relay valves. In details, the description of dynamic tire pressure system is discussed in the Chapter 7.

Step response tests with different amplitudes of pressure changing were carried out. The example of the experimental results is shown in Figure 3.6. To obtain a transfer function of dynamic tire pressure system, the experimental results were processed by Matlab System Identification Tool. The system behaviour is represented by:

$$\frac{p_{w,i}^{ext}}{p_{w,i}^{CA}} = \begin{cases} \frac{1}{0.556s + 1} e^{-0.054s}, & \text{phase of pressure increasing} \\ \frac{1}{1.306s + 1} e^{-0.063s}, & \text{phase of pressure decreasing} \end{cases} \quad (3.59)$$

The increasing rate of pressure is approximately 2.3 times higher than decreasing rate.

The final tire pressure is calculated as:

$$p_{w,i} = p_{w,i}^{nom} + p_{w,i}^{ext} \quad (3.60)$$

Figure 3.6 Experimental results of tire pressure changing

3.3 Model verification

The original vehicle and subsystem models are realized in Matlab/Simulink. The verification of the developed model is carried out by the comparison of the developed vehicle behaviour with the multibody vehicle model in commercial software IPG CarMaker.

For the verification a transient test ‘Sine with Dwell’ was selected. This test is the basis of the National Highway Traffic Safety Administration (NHTSA) regulation, which describes test procedures for Electronic Stability Control (ESC) systems. It is performed by the applying a 0.5 or 0.7 Hz sine wave with a 3/4 cycle 500 millisecond pause to the

steering wheel. The steering amplitude is calculated as a product of the steering angle that develops 0.3 g of lateral acceleration and a multiplying coefficient [116].

The comparison between developed and multibody models was carried out at different steering wheel amplitudes. The simulation results at initial velocity of 80 km/h and maximal amplitude of steering wheel angle of 70, 80, 90 and 100 deg on high friction surface of 0.8 are shown in Figure 3.7. The distinction between the simulation results of two models can be explained by the usage of the different models of suspension and tire transient behaviour.

Figure 3.7 Verification under different velocities

3.4 Chapter summary

This chapter has presented the nonlinear models of vehicle, vehicle subsystems and tire. The model of friction brake system was developed by means of analytical approach using fluid volume balance. The model of dynamic tire pressure system was obtained from the experimental results. Other models are taken from technical literature sources. The proposed models were realized in Matlab/Simulink. Under transient manoeuvre, the original vehicle model demonstrates behaviour close to the multibody vehicle model. The original vehicle model with its subsystems was used for preliminary investigations of different variants of integrated vehicle subsystems published in [71, 72] and Chapter 4 to analyse influence of hysteresis in the brake mechanism on stability control during curvilinear motion. The multibody vehicle model in IPG CarMaker with above-mentioned subsystems is used for the investigation of the coordination of priorities between vehicle subsystems in the Chapter 6. The models of vehicle subsystems, excluding friction brake system and dynamic tire pressure system, is used as virtual models for a test rig introduced in the Chapter 7.

Chapter 4

4 Control of friction brake system with hysteresis

This chapter investigates the effect of hysteresis in the friction brake mechanism. Since the operation of a friction brake system is a key action to ensure safety and stability of vehicle motion, a reaction delay due to hysteresis is significant. The effect of hysteresis in the brake mechanism was first described by the Bouc-Wen model and verified by experimental investigations of disc brake mechanism for passenger car. Using the simplified model of the brake mechanism with the hysteretic element, an analysis of hysteresis influence on vehicle dynamics during ABS straight-line braking and stability control of curvilinear motion is carried out.

4.1 Problem statement

In a hydraulic brake system, there are several hysteretic elements such as a vacuum booster, pipelines, valves, brake cylinders and others. The description of static hysteresis in the vacuum booster using a reduced-order model of brake system dynamics is proposed in [117]. The experimental investigation of hydraulic brake components of the light commercial vehicle (vacuum booster, brake master cylinder, pipeline) demonstrates a noticeable influence of hysteresis in the term of hysteresis losses [118]. It should be noted that the hysteresis characteristics of vacuum booster and master cylinder are well known and, due to their location in the brake drive, have no influence, when brake pressure is generated using the electro-hydraulic unit of ABS or ESC. The investigation of medium and heavy trucks with pneumatic S-cam drum and disc mechanisms shows that air disc mechanisms with lower hysteresis provide better braking performance [119]. The attempt to analyse an influence of hysteresis on vehicle dynamics using a HIL test rig is proposed in [120]; however, the research hypothesis is that the increasing of time of pressure-holding phase in the ABS algorithm is equivalent to the hysteresis value. This hypothesis is controversial and the approach does not achieve an analysis of a pure effect of hysteresis. The results only demonstrate that the duration of pressure-holding phase influences on braking dynamics. Moreover, the characteristics of brake pressure in [120] have the different levels of maximal brake pressures (up to 30 bar). Thus, the increasing

of brake distance is caused by an overall impact of the abovementioned factors, and not only by hysteresis.

In the study of vehicle dynamics, the hysteresis is generally represented by a constant time delay [121, 122], look-up tables [123] or a linear characteristic (backlash). While in other cases, it is neglected. Meanwhile, there is a wide range of hysteresis models proposed by Preisach, Duhem, Bouc, Wen and others for a precise description of the mechanical system behaviour. For instance, the Maxwell-slip model for hysteresis caused by friction and implemented real-time calibration technique for the clamp-force model in electromechanical brakes is proposed in [124]. An effective way of describing hysteresis is a hysteretic semi-physical model developed by Bouc and generalized by Wen. The Bouc-Wen model is applied for a wide range of the various objects such as magnetorheological dampers, piezoelectric actuators and others [125].

The investigation of hysteresis influence in the brake mechanism is experimentally complicated due to an uncontrolled setting of its parameters. Therefore, the simplified mathematical model of the brake mechanism with hysteretic components is developed in the chapter and then experimentally validated. The Bouc-Wen method is used as the basis for the development of the proposed model. Using the developed model the hysteresis influence on vehicle dynamics is investigated.

The focus of the opening part of this dissertation is to analyse how an effect of hysteresis in the brake mechanism influences on vehicle dynamics and how significant it is for the controlled generation of brake torque. The second objective is to show a difference between various representational forms of hysteresis.

4.2 Description of test rig and experimental program

Hysteresis characteristics in dynamics are provided by experimental testing of a new hydraulic brake mechanism of a passenger car. For this purpose, the series of experimental tests are carried out using a controlled law of input pressure. The schematic diagram of the brake test rig and a general view of the dynamometer part are shown in Figure 4.1.

Figure 4.1 Schematic diagram and general view of the test rig

- 1 – motor, 2 – inertia disc assembly, 3 – air-hydraulic actuating mechanism,
 4 – inertia hub assembly, 5 – rotating torque sensor, 6 – brake disc,
 7 – brake mechanism, 8 – flow meter, 9 – pressure sensor.

The brake mechanism under study is mounted and controlled by a drive system of the test rig with the maximum velocity up to 2500 rpm. The maximum of input controlled pressure can be generated up to 21 MPa by an air-hydraulic actuating mechanism of the dynamometer. The measurement system included a flow meter, pressure and torque sensors. The flow meter is located before the testing brake mechanism and it measures the amount of volumetric displacement. The pressure transducer is used to measure actual pressure entering to the calliper. The torque sensor is connected with an inertia hub assembly. It is isolated from influence of temperature environment by a phenolic plate. The measured torque signal is acquired by telemetry. It represents output torque in the contact between brake pad and brake disk. The sensors accuracy is displayed in Table 4.1.

Table 4.1 Accuracy of measurement system

Pressure Sensor		
Max. Brake Pressure	21 MPa	
Accuracy	+/- 0.5% FS	+/- 0.105 MPa
Flow meter (gear type)		
Maximum Displacement	50 cm ³	
Accuracy	0.3% FS	0.15 cm ³
Torque Sensor (calibration test)		
Rated load torque	5000 N	
Accuracy		
Linearity deviation	0.004	
Linearity deviation incl. hysteresis	0.006	
Relative hysteresis	0.011	55 Nm

Since the width of hysteresis loops in the static mode is the largest within the middle range of working pressure, the pressure law is chosen as a pulse signal with

minimum and maximum amplitudes of 3.2 and 5.5 MPa respectively, and frequency of 6 Hz. Such kind of pressure signal corresponds to the braking of the vehicle equipped with ABS and two-phase rule-based algorithm (without pressure-holding phase) under wet road conditions. To reduce measurement noise, three replications are done for each test. The measurement sampling frequency is 10 kHz. The initial velocity is 90 and 120 km/h; contact temperature of friction pair (brake pad and brake disk) – 100 and 300 °C; testing modes – “in-stop” mode (free rotation of brake disc related to emulation of emergency braking) and “drag” mode (constant rotational velocity related to emulation of service braking).

The experimental results are shown in Figure 4.2. The control input pressure is shown in Figure 4.2a. The relation between the piston linear displacement and the input pressure is displayed in Figure 4.2b. The piston linear displacement is calculated from the volumetric displacement, and the accuracy of piston displacement is 0.06 mm. It can be seen that hysteresis exists between the phases of pressure increasing and decreasing. It should be noted that in the range of lower pressures the hysteresis loop is wider in comparison with the range of higher pressures. The results confirm that hysteresis has an asymmetric form. Figure 4.2c presents the brake torque versus the linear displacement of the piston. The fluctuation of brake torque along the linear displacement is non-ordered. Furthermore, the points of one pressure phase overlap another phase. It indicates that the relationship between torque and displacement has weak nonlinearity due to an apparent fluctuation of friction coefficient during braking [126]. Meanwhile, the results might include measurement noise due to a low accuracy of the torque sensor (55 Nm). The summarized characteristic between input (pressure) and output (torque) signals is shown in Figure 4.2d.

Figure 4.2 Experimental results:

(a) input pressure versus time, (b) piston linear displacement versus input pressure, (c) brake torque versus piston linear displacement and (d) brake torque versus input pressure
x(blue) – phase of pressure increasing; o(red) – phase of pressure decreasing.

4.3 Computational model of brake mechanism and verification

In accordance with the experimental results, the hysteresis characteristic has a nonlinear and asymmetric form. Therefore, there is the hypothesis that the representation of hysteresis by a constant shift of characteristic or in a linear form is inadequate to describe a hysteresis effect. The aim of the section is to develop the simplified mathematical model of the brake mechanism including hysteretic elements and to verify the proposed model with experiments.

A dynamic model of the brake mechanism can include brake rotor, piston and other elements [127, 128]. The model refinement depends on the requirements of research tasks. In this chapter, the first objective is the representation of hysteresis characteristic for the

task of vehicle braking dynamics. To reduce complexity of the mathematical model, a simple representation of the brake mechanism is considered as shown in Figure 4.3.

Figure 4.3 A simplified model of the brake mechanism

The subsequent discussion assumes that the friction processes are the same for the primary and secondary brake pads. The elements having translational motion are lumped to one equivalent mass m_{eq} . It is represented by the second-order differential equation:

$$m_{eq}\ddot{x}_i + c_c\dot{x}_i + \Phi(\dot{x}_i, x_i, z_1) = u_i(t) \quad (4.1)$$

The control input force is calculated as:

$$u_i(t) = p_i(t) \frac{\pi d_{cyl}^2}{4} \text{ (for verification) or } u_i(t) = F_{ni}^{contact} \text{ (for simulation)} \quad (4.2)$$

The contact restoring force $\Phi(x)$ has nonlinear and asymmetric behaviour in a “load-unload” cycle and it is described by the modified version of the Bouc-Wen model [129]:

$$\begin{aligned} \Phi(x_i) &= k_c (x_i + z_1) \\ \dot{z}_1 &= \begin{cases} \alpha_1 \dot{x}_i - \beta_1 |\dot{x}_i| |z_1| |z_1|^{n_1-1} - \gamma_1 \dot{x}_i |z_1|^{n_1}, & \dot{x}_i \geq 0 \\ \alpha_2 \dot{x}_i - \beta_2 |\dot{x}_i| |z_1| |z_1|^{n_2-1} - \gamma_2 \dot{x}_i |z_1|^{n_2}, & \dot{x}_i < 0 \end{cases} \end{aligned} \quad (4.3)$$

Contact friction torque for the brake mechanism with a fixed calliper is given by:

$$T_{br,i} = M_{\mu i} = 2\mu(nv, \eta)\Phi(x_i)r_{eff} \quad (4.4)$$

Friction coefficient depends on pressure (normal contact force), temperature and velocity in contact. During braking process, friction coefficient changes dynamically. The dynamics can be modelled by a system of differential equations combining thermal and mechanical behaviour. For instance, Ostermeyer [130] or Chichinadze [131] models for friction coefficient can be applied. The results obtained from abovementioned models are sensitive to a small variation of input parameters. Dynamic friction coefficient is calculated [130] as:

$$\dot{\mu} = -a((nv + \varepsilon)\mu - \eta), \quad (4.5)$$

The parameter nv is found as:

$$nv = \frac{\Phi(x_i)v_{bdi}}{(Nv)_{ref}}, \quad (4.6)$$

Actual tangential velocity between the brake pad and the disc is calculated as:

$$v_{bdi} = w_{bdi}r_{eff} = w_{wi}r_{eff}, \quad (4.7)$$

Normalized temperature in the contact is found as:

$$\dot{\eta} = -c(\eta - \eta_0 - \alpha nv) - \gamma(\eta^4 - \eta_0^4), \quad (4.8)$$

To involve tire hysteresis (dashed part of Figure 4.3), the system of equations can be expanded. In this case, the rotational dynamics is represented as:

$$\begin{aligned} J_d \frac{d^2\theta_{bd}}{dt} &= M_\mu - c_{tire}\Delta\dot{\theta} - M(\Delta\dot{\theta}, \Delta\theta, z_{tire}) \\ J_w \frac{d^2\theta_w}{dt} &= c_{tyre}\Delta\dot{\theta} + M(\Delta\dot{\theta}, \Delta\theta, z_{tire}) + M_{res} \\ \Delta\theta &= \theta_{bd} - \theta_w \end{aligned} \quad (4.9)$$

For the verification with the experimental data: 1) the resistance torque is equal to zero; 2) the additional moment of inertia J_v , which is equivalent to vehicle mass, is attached to the required inertia.

Output torque (equivalent tire torque) is equal to:

$$\begin{aligned} M_{tire} &= M(\Delta\dot{\theta}, \Delta\theta, z_{tire}) = k_{tire}(\Delta\theta + z_{tire}) \\ \dot{z}_{tire} &= \alpha\Delta\dot{\theta} - \beta|\Delta\dot{\theta}|z_{tire}|z_{tire}|^{n-1} - \gamma\Delta\dot{\theta}|z_{tire}|^n \end{aligned} \quad (4.10)$$

The influence of the proposed model parameters on the hysteresis characteristic, its area and its width is shown in Figure 4.4.

Figure 4.4 Characteristic and parameters of the hysteresis loop

The parameters of the proposed simplified model of the brake mechanism are shown in Table 4.2. The results of model verification are displayed in Figure 4.4. The input pressure law is the same as shown in Figure 4.2a. The proposed model demonstrates a good comparability with the experimental data in Figure 4.5.

Figure 4.5 . The results of model verification:

x(blue) – phase of pressure increasing; o(red) – phase of pressure decreasing.

Table 4.2 Parameters of simplified model of brake mechanism

mass of piston, kg	0.18	n_1	0.75
mass of pad, kg	0.24	$\alpha_1 = \alpha_2$	2.0
piston/cylinder diameter d_{cyl} , m	0.059	β_1	0.2
effective radius r_{eff} , m	0.13	γ_1	0.5
inertia moment of brake disc J_d , kgm ²	0.11	n_2	0.65
actual inertia J_v , kgm ²	47.46	β_2	4.2
required inertia, kgm ²	61.79	γ_2	0.7
static coefficient of friction, μ_0	0.33	α'	0.1
translational contact stiffness k_c , 10 ⁷	1.35	c'	0.3
translational damping c_c , 10 ⁵	2.0	α'	0.8
initial velocity, rads ⁻¹	90.42	γ'	0.35
initial disc temperature, °C	96	η_0	0.1

4.4 Analysis of the influence of hysteresis parameters on vehicle dynamics

4.4.1 Straight-line braking

To minimize the influence of normal load transfer, changing of wheel radius and other factors on vehicle dynamics, a quarter-model of vehicle with the fixed vertical force is considered. The mathematical description of longitudinal dynamics is given:

$$\begin{aligned} m_a \frac{dV_x}{dt} &= -F_{xw} \\ J_w \frac{d\omega_w}{dt} &= -T_{br} + F_{xw} r_w \end{aligned} \quad (4.11)$$

For the purpose of longitudinal dynamics simulation discussed further, the pressure is calculated without brake actuator dynamics (fluid behaviour in master brake cylinder, pipelines, valve dynamics and others). Thereby, a pure effect of hysteresis is investigated.

The PID control and the rule-based algorithms are considered. The PID control is wheel slip-based one and it provides two phases of regulation related to increase/decrease of brake pressure.

The rule-based algorithm is based on thresholds by slip and wheel acceleration similar to [132]. The additional phases, related to pressure holding during build-up and release phases, are neglected.

The simulation was carried out for three types of road surfaces (dry (0.8) and wet (0.6) asphalt, snow (0.2)) with different initial velocities. The parameters of hysteresis model were chosen as for the initial experimental test data from Table 4.2.

The simulation results are represented in Table 4.3. The type of hysteresis shows how hysteresis effect is taken into account:

- “zero” means that hysteresis is neglected;
- “delay” means that a constant transfer delay for brake torque is applied;
- “backlash” means that the hysteresis characteristic is linear and it is represented as backlash;
- “model” means the application of abovementioned model.

The parameters of “delay” and “backlash” are chosen so that the maximal width of hysteresis is the same for all models under the braking with the initial velocity of 75 km/h and on wet road.

Table 4.3 Simulation results of straight-line braking with ABS

Road	V_x	Hysteresis Type	$\bar{A}$	w_{max}	s_{br}	t_{br}	j	f_{br}
snow	45	zero	-	-	37.2	4.9	-2.04	6.28
		delay	54.1	0.373	39.1	5.2	-1.94	5.29
		backlash	54.4	0.548	40.7	5.4	-1.87	3.89
		model	42.8	0.360	38.8	5.2	-1.96	5.23
	60	zero	-	-	66.8	6.9	-2.05	5.83
		delay	54.2	0.438	70.2	7.3	-1.95	5.05
		backlash	60.3	0.593	73.3	7.6	-1.88	3.82
		model	44.3	0.420	69.9	7.3	-1.96	4.95
wet asphalt	75	zero	-	-	41.0	3.4	-5.35	4.94
		delay	265.3	0.695	42.5	3.6	-5.15	4.33
		backlash	229.3	0.695	42.7	3.6	-5.13	3.76
		model	212.1	0.695	42.4	3.6	-5.18	4.22
	100	zero	-	-	72.4	4.7	-5.39	4.57
		delay	259.6	0.813	75.0	4.9	-5.19	3.99
		backlash	249.9	0.824	75.5	4.9	-5.16	3.56
		model	215.0	0.828	74.8	4.9	-5.21	4.01
dry asphalt	90	zero	-	-	40.4	2.9	-7.91	4.38
		delay	499.3	0.821	41.8	3.0	-7.65	3.90
		backlash	416.1	0.768	41.6	2.9	-7.67	3.57
		model	406.7	0.857	41.6	2.9	-7.68	3.91
	120	zero	-	-	71.0	3.9	-8.00	4.01
		delay	488.8	0.993	73.2	4.0	-7.74	3.63
		backlash	437.9	0.956	73.1	4.0	-7.75	3.39
		model	412.2	1.038	73.1	4.0	-7.77	3.64

Hysteresis has more influence on brake distance and braking time in the cases of low velocities and slippery roads. This is mainly caused due to the fact of the actuation frequency increasing and enlarging of the total number of regulation cycles. Nevertheless, the average area and maximal width are decreased in comparison with the braking on the road with a higher friction coefficient. For dry asphalt, the difference between the “zero” and “model” cases is less than 3% and it is up to 4.5% for the slippery road. The description of hysteresis by a delay causes less than 1% error for all surfaces accordingly. The backlash description causes 0.1% and 5% errors for asphalt and snow accordingly. In the simulation results, the effect of hysteresis has less than 5% error. However, the parameters of hysteresis were taken for a new brake mechanism. As known, the area and width of hysteresis can increase during the continuous brake operation. Therefore, the additional investigation related to the hysteresis variation is required.

Figure 4.6 Brake distance and frequency of wheel deceleration versus hysteresis parameter α :

(a), (b) for PID control; (c), (d) for rule-based control

The hysteresis loop was expanded using parameter α . The simulation results are shown in Figure 4.6 for both control strategies under different velocities and roads. The higher value of parameter α means that the hysteresis is decreased (Figure 4.5, a-b). It causes a reduction of brake distance and increase in frequency of wheel deceleration simultaneously (Figure 4.6). The effect has a strong impact and it is approximately similar for both control algorithms: the PID and the rule-based strategies.

The hypothesis is that the hysteresis can be controlled by a proportional gain for the PID control. For the rule-based control, the issue is how a period of pressure-holding phase is related to brake distance. The simulation with the fixed parameters of hysteresis and the variation of the proportional gain for the PID control and the time of pressure-holding phase for the rule-based control is carried out. The simulation results are shown in Figure 4.7.

Figure 4.7 Brake distance versus proportional gain for PID control (a), (b); brake distance versus time of pressure-holding phase for rule-based control (c), (d)

Due to a decreased settling time of control process, brake distance is reduced as long as the proportional gain is equal to 5. From other side, the high proportional gain induces a growth of the average area of hysteresis (Table 4.3). In addition, the high proportional gain calls for control overshoot. It has a negative influence on braking performance and increases brake distance. One of the possible ways to minimize this effect is to introduce a derivative gain.

For the rule-based control (Figure 4.7, c-d) the growth of the time of pressure-holding phase invokes a significant rise of brake distance. The time of pressure-holding phase has a higher impact on brake distance than the parameters of hysteresis loop such as area and width. Pressure-holding phase is usually used to stabilize oscillations in the system. The time of pressure-holding phase is significant to minimize consequences of hysteresis effect; however, it is useless for the regulation of hysteresis. Apparently, the possible way to improve brake efficiency and to minimize hysteresis is a variable pressure rate during ABS cycles.

Figure 4.8 Brake distance and frequency of wheel deceleration versus proportional gain for different representational forms of hysteresis:

(a), (b) for snow (45 km/h); (c), (d) for dry asphalt (90 km/h)

To investigate the influence of the representation of hysteresis characteristics on braking performance, abovementioned tests were provided for two types of road conditions (Figure 4.8, a-b for snow road with the initial velocity 45 km/h and Figure 4.8, c-d for dry asphalt with the initial velocity 90 km/h). The neglect of hysteresis causes a monotonic decrease of brake distance under the increase of the proportional gain. Thereby, the distortion of real system behaviour is observed. The linear representation provides the same behaviour as “zero” mode. The delay representation qualitatively shows the similar results as the developed model; however, the increase of brake distance is higher under greater proportional gain.

4.4.2 Stability control of curvilinear motion with friction brake actuation

To analyse the influence of hysteresis in the brake mechanism on stability control during curvilinear motion, the original vehicle model with friction brake system from Chapter 3

and the control system from Chapter 5 is used. The torque demand is generated only by friction brake system. The ‘Sine with Dwell’ test is carried out at different values of hysteresis in the brake mechanism.

The characteristics of brake pressure versus volumetric flow for front brake mechanism under different levels of hysteresis are shown in Figure 4.9a. The relationships between yaw rate and sideslip angle are shown in Figure 4.9b. Despite more than two times of hysteresis width increasing, its influence on sideslip angle and yaw rate is insignificant.

The explanation of this fact is that the duration and frequency of brake torque actuations for stability control is lower than for the straight-line braking case with ABS.

Figure 4.9 Brake pressure versus volumetric flow for front brake mechanism (a); vehicle sideslip angle versus yaw rate (b) under different levels of hysteresis

4.5 Chapter summary

Experimental investigations of disc brake mechanism clearly show the nonlinear and asymmetric form of hysteresis. The computational model of the brake mechanism with hysteretic elements, based on the Bouc-Wen method, is developed and verified with experiments.

Using the original vehicle model, the analysis of hysteresis influence on vehicle dynamics (ABS straight-line braking and curvilinear motion with stability control) was provided. The influence of the variation of the hysteresis parameters on the main vehicle brake characteristics such as brake distance, time of braking, average deceleration and others are shown. In addition, the results with various ways of hysteresis representation (non-consideration, signal delay and linear form of hysteresis) are deduced. The main outcomes of the presented chapter are:

- experimental analysis using the dynamometer test rig confirmed the nonlinear and asymmetric form of hysteresis in the brake mechanism;
- hysteresis in the brake mechanism is significant during specific braking manoeuvres, such as braking on slippery roads with high initial velocities or with the sharp changing of friction coefficient; however, it can be neglected during the braking on dry road due to the less impact;
- effect of hysteresis on stopping distance for new hydraulic disc brake mechanism does not exceed 5% in the most critical cases; however, ignoring this fact can cause incorrect control. For instance, a false demand can increase frequency of brake torque actuation;
- to simplify the initial procedures of the ABS and brake design, the influence of the brake hysteresis on vehicle dynamics can be represented by the emulating hysteresis as a signal delay, because it provides sufficient accuracy;
- effect of hysteresis on vehicle dynamics can be controlled by frequency or rate of brake pressure. It allows to improve the adaptive properties of ABS control;
- influence of hysteresis on stability control is insignificant due to the duration and frequency of brake torque actuation.

The proposed model of hysteresis in the brake mechanism is further used for the computational simulation of brake system for the proper generation of brake torque.

Chapter 5

5 Control system

This chapter presents a multi-layer control system to regulate vehicle dynamics. The inputs of the control system are obtained from the driver. The outputs are control signals for vehicle subsystems.

5.1 Control objectives

The main objectives of vehicle dynamics control are [133]:

- guarantee of vehicle stability;
- obtain a vehicle response according to the reference model.

The additional control objectives are related to energy characteristics and represented the energy consumption of electric motors and tire energy dissipation in tire-road contact. In addition, they should be achieved without the complication of the cost function.

Since the considering vehicle is multi-actuated and the integrated control will be applied, the additional task will be related to avoiding of cross-couplings and conflicts during integrated control. Figure 5.1 displays the conflict indicators in the form of rating expert appraisals. The appraisals lay in the range 0 to 5 points and they are composed from the analysis of ground vehicle dynamics. These indicators estimate a grade of possible conflict situations within the ground vehicle. A greater value of the indicator indicates a more anticipated conflict between vehicle subsystems ensuring a specified property [134]. This task is solved by the application of weighting matrix defining a priority among generalized forces/torque and specific logical conditions for the control demand correction depending on driver inputs.

Figure 5.1 Conflict indicators

5.2 Multi-layer control architecture

The multi-layer architecture has several advantages compared to others [135]:

- coordinate vehicle subsystems to realize a command from high-level controller;
- implementation of vehicle subsystems from different producers.

The overall structure of control system is shown in Figure 5.2. It consists of a state observer; high-level vehicle dynamics controller; middle-level control allocation and low-level individual actuator control.

Figure 5.2 Overall control structure

Steering angle and displacement of brake pedal from driver are the input parameters to the reference vehicle model. The high-level controller compensates a difference between reference and actual vehicle behaviour to guarantee braking performance, handling and stability of motion. The middle layer allocates control demand between vehicle subsystems. The low-level actuator control guarantees a precise tracking to

reference control signals obtained from the middle layer. Moreover, it is used to estimate boundary conditions for control allocation taking into account actuator physical limits and wheel slip control.

5.3 High-level vehicle dynamics controller

The high-level vehicle dynamics controller consists of two parts: a generator of reference signals and a controller to produce demands of generalized longitudinal and lateral forces and yaw torque to minimize the deviation between actual and reference vehicle behaviour. The reference signals are reference longitudinal deceleration related to braking performance, reference yaw rate for vehicle handling, and reference sideslip angle, when the vehicle is outside of the boundaries of vehicle stability. The standard on-board sensor can measure yaw rate, but the measurement of the sideslip angle is a more complex task. Therefore, vehicle sideslip angle is calculated using the estimated longitudinal and lateral velocities from the vehicle state observer.

5.3.1 Reference vehicle model

Reference vehicle deceleration a_x^{ref} depends on pedal force F_p . The desired pedal force F_p is calculated by a 1-D lookup table where brake pedal displacement is the input parameter. The relationship between pedal force and pedal displacement is shown in Figure 5.3 [96].

Figure 5.3 Pedal force versus displacement and reference deceleration versus pedal force

Reference deceleration for full laden vehicle is described as:

$$a_x^{ref} = \begin{cases} -k_1 F_p - k_2, & a_x < a_x^{\max} \\ a_x^{\max}, & a_x \geq a_x^{\max} \end{cases}, \quad (5.1)$$

To describe the curvilinear motion, a 2 DoF classical bicycle vehicle model is used as the reference model. Tire stiffness in the reference generator is constant and correspond to the motion in the high friction condition.

The reference yaw rate can be found as follows [136]:

$$\begin{bmatrix} \dot{\beta} \\ \ddot{\psi} \end{bmatrix} = \begin{bmatrix} -\frac{C_f + C_r}{m_a V_{xa}} & -1 + \frac{-aC_f + bC_r}{m_a V_{xa}^2} \\ \frac{-aC_f + bC_r}{I_{zz}} & -\frac{a^2 C_f + b^2 C_r}{I_{zz} V_{xa}} \end{bmatrix} \begin{bmatrix} \beta \\ \dot{\psi} \end{bmatrix} + \begin{bmatrix} \frac{C_f}{m_a V_{xa}} \\ \frac{aC_f}{I_{zz}} \end{bmatrix} \delta_{av,f} \quad (5.2)$$

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

$$\dot{\psi}_{ref} = \begin{cases} \dot{\psi}, & |\dot{\psi}| \leq |\dot{\psi}_{max}| \\ \pm \dot{\psi}_{max}, & \text{otherwise} \end{cases} \quad (5.3)$$

The maximum of the reference yaw rate, obtained beyond the adhesion tire limit, is calculated as [137]:

$$\dot{\psi}_{max} = 0.85 \frac{\mu_{road} g}{V_{xa}} \quad (5.4)$$

The controller maintains vehicle stability only during emergency manoeuvres, therefore, sideslip angle control should be bounded using phase-plane approach [23]. When sideslip angle is inside of the stability boundaries, no control action is required. The reference sideslip angle β_{ref} is calculated from estimated β_{est} , boundary β_{bound} and derivative $d\beta_{bound}/dt$ values as:

$$\beta_{ref} = \begin{cases} \beta_{est} - \beta_{bound} \left(1 - \frac{\dot{\beta}_{est}}{\dot{\beta}_{bound}} \right), & \beta_{est} \geq 0 \wedge \dot{\beta}_{est} \geq 0 \\ \beta_{est} + \beta_{bound} \left(1 + \frac{\dot{\beta}_{est}}{\dot{\beta}_{bound}} \right), & \beta_{est} < 0 \wedge \dot{\beta}_{est} < 0 \\ \beta_{est} - \beta_{bound} \left(1 + \frac{\dot{\beta}_{est}}{\dot{\beta}_{bound}} \right), & \beta_{est} \geq 0 \wedge \dot{\beta}_{est} < 0 \\ \beta_{est} + \beta_{bound} \left(1 - \frac{\dot{\beta}_{est}}{\dot{\beta}_{bound}} \right), & \text{otherwise} \end{cases} \quad (5.5)$$

5.3.2 Second-order sliding mode control

The control error of longitudinal acceleration e_{ax} is calculated as:

$$e_{a_x} = \begin{cases} a_x^{ref} - a_x^{mes}, & |a_x^{ref} - a_x^{mes}| > \Delta_{a_x} \\ 0, & |a_x^{ref} - a_x^{mes}| < \Delta_{a_x} \end{cases} \quad (5.6)$$

The control error of sideslip angle e_β can be found as:

$$e_\beta = \begin{cases} |\beta_{est} - \beta_{ref}| \text{sign}(\beta_{est}), & \beta_{est} > \beta_{ref} \wedge \beta_{ref} > 0 \\ |\beta_{est} - \beta_{ref}| \text{sign}(\beta_{est}), & \beta_{est} < \beta_{ref} \wedge \beta_{ref} < 0 \\ 0, & \text{otherwise} \end{cases} \quad (5.7)$$

The control error of yaw rate $e_{d\psi/dt}$ is calculated as:

$$e_{\dot{\psi}} = \begin{cases} \dot{\psi}_{ref} - \dot{\psi}_{mes}, & |\dot{\psi}_{ref} - \dot{\psi}_{mes}| > \Delta_{\dot{\psi}} \\ 0, & |\dot{\psi}_{ref} - \dot{\psi}_{mes}| < \Delta_{\dot{\psi}} \end{cases} \quad (5.8)$$

The thresholds Δ_{a_x} and $\Delta_{d\psi/dt}$ define dead zones to eliminate the demand generation when control errors are insignificant.

Various control techniques can be applied to minimize control error. The second-order sliding mode control (SOSMC) is selected as the control method to stabilize a closed-loop system. In contrast with the first-order sliding mode control, the SOSMC defines a sliding surface using the second time derivative of the system deviation. As mentioned in [138], the main advantages of SOSMC are chattering reduction and high accuracy under the same robustness in comparison with the first-order sliding mode control.

The proposed controller is formulated in discrete-time domain, taking into account its subsequent investigation using hardware-in-the-loop test rig. Based on the algorithm of discrete SOSMC [139], the algorithmic sequence for each control error consists of the following steps:

1. calculation of the parameter of operation condition $\Delta(k)$:

$$\Delta(k) = -(e(k) - e(k-1))e(k) \quad (5.9)$$

2. find memory variable e_{mem} :

if $\Delta(k) \leq 0$ then

$$e_{mem}(k) = e(k)$$

else

$$e_{mem}(k) = e_{mem}(k-1)$$

(5.10)

3. calculate a piecewise function describing the value of the last singular

point of $e(k)$:

if $\Delta(k) \leq 0$ then

if $e_{mem}(k)e_M(k-1) > 0$ AND $|e_{mem}(k)| < |e_M(k-1)|$

$$e_M(k) = e_{mem}(k)$$

else

$$e_M(k) = e_M(k-1)$$

else

$$e_M(k) = e_{mem}(k)$$

(5.11)

4. the control law $v(k)$ will be calculated as:

$$v(k) = -K_{sm} \operatorname{sgn}(e(k) - 0.5e_M(k)) \quad (5.12)$$

5. to reduce the chattering effect, instead of the application of the sign function, the saturation function is applied:

$$v(k) = -K_{sm} \operatorname{sat}(e(k) - 0.5e_M(k)) \quad (5.13)$$

The generalized longitudinal force v_{lon} is calculated using sampling time T_s :

$$\begin{aligned} v_{lon}(k+1) &= f(d_{brake}) [v_{lon}(k) - K_{sm,ax} m_a \operatorname{sat}(f_{ax}) T_s] \\ \operatorname{sat}(f_{ax}) &= \begin{cases} \frac{e_{ax}(k) - 0.5e_{Max}(k)}{\sigma_{ax}}, & |e_{ax}(k) - 0.5e_{Max}(k)| < \sigma_{ax} \\ \operatorname{sgn}(e_{ax}(k) - 0.5e_{Max}(k)), & \text{otherwise} \end{cases} \end{aligned} \quad (5.14)$$

The function $f(d_{brake})$ is a switching function to prevent a false generation of longitudinal force demand. It is equal to one, when brake pedal is pushed, and zero in other cases.

The generalized lateral force v_{lat} is calculated as:

$$\begin{aligned} v_{lat}(k+1) &= v_{lat}(k) - K_{sm,\beta} m_a \operatorname{sat}(f_{\beta}) T_s \\ \operatorname{sat}(f_{\beta}) &= \begin{cases} \frac{e_{\beta}(k) - 0.5e_{M\beta}(k)}{\sigma_{\beta}}, & |e_{\beta}(k) - 0.5e_{M\beta}(k)| < \sigma_{\beta} \\ \operatorname{sgn}(e_{\beta}(k) - 0.5e_{M\beta}(k)), & \text{otherwise} \end{cases} \end{aligned} \quad (5.15)$$

The generalized yaw torque v_{yaw} is calculated as:

$$\begin{aligned} v_{yaw}(k+1) &= v_{yaw}(k) - K_{sm,\psi}(\dot{\psi}) I_{zz} \text{sat}(f_{\dot{\psi}}) T_s \\ \text{sat}(f_{\dot{\psi}}) &= \begin{cases} \frac{e_{\dot{\psi}}(k) - 0.5e_{M\dot{\psi}}(k)}{\sigma_{\dot{\psi}}}, & |e_{\dot{\psi}}(k) - 0.5e_{M\dot{\psi}}(k)| < \sigma_{\dot{\psi}} \\ \text{sgn}(e_{\dot{\psi}}(k) - 0.5e_{M\dot{\psi}}(k)), & \text{otherwise} \end{cases} \end{aligned} \quad (5.16)$$

The controller gain for generalized torque is proportionally scheduled using the values of yaw rate. Since gain-scheduling might introduce instability, the variation limits of the controller gain were selected taking into account a stable operation of the control system. This procedure reduces the generation of control demand under lower control errors. The parameters of controller are obtained from simulation-based offline optimization with interior-point method to minimize tracking errors. They are listed in Table 5.1.

Table 5.1 Controller gains

$K_{sm,ax}$		15	σ_{ax}	1e-2	Δ_{ax}	0.1 m/s ²
$K_{sm,\beta}$		10	σ_{β}	2.5e-2	Δ_{β}	0.2 deg
$K_{sm,d\psi/dt}$	$d\psi/dt < 3 \text{ deg/s}$	5	$\sigma_{d\psi/dt}$	5e-2	$\Delta_{d\psi/dt}$	0.25 deg/s
	$d\psi/dt > 10 \text{ deg/s}$	25				

The final demand v_d from the high-level controller is defined as:

$$v_d = \begin{bmatrix} v_{lon} & v_{lat} & v_{yaw} \end{bmatrix}^T \quad (5.17)$$

5.4 Middle-level control allocation

The aim of middle layer is a distribution of generalized control demand between vehicle subsystems. This problem is solved by an optimization procedure. The coordination of priorities between vehicle subsystems using restriction weights is introduced to achieve alterable priorities of vehicle subsystems during vehicle motion. The selection procedure of restriction weights is discussed in Chapter 6.

The relationship between generalized forces and yaw torque (control demands from high-level controller) and control input vector (outputs from control allocation) is nonlinear. According to [31, 50] a nonlinear relationship can be linearized about an operating point by first-order Taylor series expansion:

$$\mathbf{v}^* = \mathbf{v}_d - \mathbf{g}(\boldsymbol{\xi}, u_{k-1}) = \frac{\partial \mathbf{g}(\boldsymbol{\xi}, u_{k-1})}{\partial u} u_{k-1} \approx \mathbf{B}_u(\boldsymbol{\xi}, u_{k-1}) u_k \quad (5.18)$$

Tire forces are attributed to vehicle motion and they are under influence of subsystem actuation. Assuming that tire forces along wheel axes can be separated between force components related to vehicle motion and subsystem actuation:

$$\mathbf{F}_{wi} = \mathbf{F}_{wi}^{sys} + \mathbf{F}_{wi}^{actuator} \quad (5.19)$$

The force components $\mathbf{F}_{wi}^{sys}$ related to vehicle motion can be written in the following form:

$$\begin{aligned} \mathbf{F}_{xwi}^{sys} &= -\frac{\mathbf{J}_{wi}}{r_w} \frac{d\omega_{wi}}{dt} \\ \mathbf{F}_{ywi}^{sys} &= \mathbf{C}_{ywi} \left(\tan \frac{V_{yi}}{V_{xi}} - \boldsymbol{\delta}_i^{kinematic} \right) \end{aligned} \quad (5.20)$$

Neglecting transient processes in actuators and considering their simplified dynamics, the force components $\mathbf{F}_{wi}^{actuators}$ generated by actuators are equal to:

$$\begin{aligned} \mathbf{F}_{xwi}^{actuators} &= \frac{1}{r_w} T_{em,i}^{CA} + \frac{1}{r_w} T_{br,i}^{CA} + \mathbf{C}_{p_{wx}} p_{wi}^{CA} + \mathbf{C}_{R_{zx}} R_{zi}^{CA} \\ \mathbf{F}_{ywi}^{actuators} &= \mathbf{C}_{ywi} \boldsymbol{\delta}_i^{CA} + \mathbf{C}_{\gamma_i} \gamma_i^{CA} + \mathbf{C}_{p_{wy}} p_{wi}^{CA} + \mathbf{C}_{R_{zy}} R_{zi}^{CA} \end{aligned} \quad (5.21)$$

The nonlinear component of vehicle motion $\mathbf{g}(\boldsymbol{\xi}, u_{k-1})$ can be calculated as:

$$g(\xi, u_{k-1}) = \begin{bmatrix} m_a V_{ya} \dot{\psi} + \sum_{i=1}^4 F_{xi}^{sys} \\ -m_a V_{xa} \dot{\psi} + \sum_{i=1}^4 F_{yi}^{sys} \\ a \sum_{i=1}^2 F_{yi}^{sys} - b \sum_{i=3}^4 F_{yi}^{sys} + \sum_{i=f,r} t_i \frac{-F_{x,il}^{sys} + F_{x,ir}^{sys}}{2} \end{bmatrix} \quad (5.22)$$

Since the six vehicle subsystems are considered, and electric motors can develop positive (traction) and negative (brake) torques, the control input vector is given as:

$$u^{CA} = \begin{bmatrix} \delta_i^{CA} & T_{em,i}^{CA,pos} & T_{em,i}^{CA,neg} & T_{br,i}^{CA} & \gamma_i^{CA} & p_{wi}^{CA} & R_{zi}^{CA} \end{bmatrix}^T \quad (5.23)$$

The control effectiveness matrix B_u is equal to:

$$B_u = \begin{bmatrix} B_\delta & B_{em}^{pos} & B_{em}^{neg} & B_{br} & B_\gamma & B_{p_w} & B_{R_z} \end{bmatrix} \quad (5.24)$$

The parameters of control effectiveness matrix B_δ for wheel steer angles δ_i^{CA} can be formulated as:

$$B_\delta = \begin{bmatrix} C_{ywfl} \sin \delta_{fl} & C_{ywfr} \sin \delta_{fr} & C_{ywl} \sin \delta_{rl} & C_{ywr} \sin \delta_{rr} \\ C_{ywfl} \cos \delta_{fl} & C_{ywfr} \cos \delta_{fr} & C_{ywl} \cos \delta_{rl} & C_{ywr} \cos \delta_{rr} \\ C_{ywfl} \chi_{yfl} & C_{ywfr} \chi_{yfr} & C_{ywl} \chi_{yrl} & C_{ywr} \chi_{yrr} \end{bmatrix} \quad (5.25)$$

Using front t_f and rear t_r track widths and wheel steering angles δ_i , the kinematic ratios χ_{yi} are defined as:

$$\begin{aligned} \chi_{yfl} &= -0.5t_f \sin \delta_{fl} + a \cos \delta_{fl} \\ \chi_{yfr} &= 0.5t_f \sin \delta_{fr} + a \cos \delta_{fr} \\ \chi_{yrl} &= -0.5t_r \sin \delta_{rl} - b \cos \delta_{rl} \\ \chi_{yrr} &= 0.5t_r \sin \delta_{rr} - b \cos \delta_{rr} \end{aligned} \quad (5.26)$$

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

$$B_{br} = \begin{bmatrix} r_w^{-1} \cos \delta_{fl} & r_w^{-1} \cos \delta_{fr} & r_w^{-1} \cos \delta_{rl} & r_w^{-1} \cos \delta_{rr} \\ r_w^{-1} \sin \delta_{fl} & r_w^{-1} \sin \delta_{fr} & r_w^{-1} \sin \delta_{rl} & r_w^{-1} \sin \delta_{rr} \\ r_w^{-1} \chi_{xfl} & r_w^{-1} \chi_{xfr} & r_w^{-1} \chi_{xrl} & r_w^{-1} \chi_{xrr} \end{bmatrix} \quad (5.27)$$

The kinematic ratios χ_{xi} are:

$$\begin{aligned}
\chi_{xfl} &= -0.5t_f \cos \delta_{fl} + a \sin \delta_{fl} \\
\chi_{xfr} &= 0.5t_f \cos \delta_{fr} + a \sin \delta_{fr} \\
\chi_{xrl} &= -0.5t_r \cos \delta_{rl} - b \sin \delta_{rl} \\
\chi_{xrr} &= 0.5t_r \cos \delta_{rr} - b \sin \delta_{rr}
\end{aligned} \tag{5.28}$$

The control effectiveness matrices for electric motors and brake system are the same:

$$B_{em}^{pos} = B_{em}^{neg} = B_{br} \tag{5.29}$$

Control effectiveness matrix B_γ for camber angle actuators γ_i^{CA} can be calculated as:

$$B_\gamma = \begin{bmatrix} C_{\gamma fl} \sin \delta_{fl} & C_{\gamma fr} \sin \delta_{fr} & C_{\gamma rl} \sin \delta_{rl} & C_{\gamma rr} \sin \delta_{rr} \\ C_{\gamma fl} \cos \delta_{fl} & C_{\gamma fr} \cos \delta_{fr} & C_{\gamma rl} \cos \delta_{rl} & C_{\gamma rr} \cos \delta_{rr} \\ C_{\gamma fl} \chi_{yfl} & C_{\gamma fr} \chi_{yfr} & C_{\gamma rl} \chi_{yrl} & C_{\gamma rr} \chi_{yrr} \end{bmatrix} \tag{5.30}$$

Control effectiveness matrix B_{pw} for dynamic tire pressures p_{wi}^{CA} is defined as:

$$\begin{aligned}
B_{pw} &= \begin{bmatrix} B_{pw}^{front} & B_{pw}^{rear} \end{bmatrix} \\
B_{pw}^{front} &= \begin{bmatrix} C_{p_{wx}}^{fl} \cos \delta_{fl} - C_{p_{wy}}^{fl} \sin \delta_{fl} & C_{p_{wx}}^{fr} \cos \delta_{fr} - C_{p_{wy}}^{fr} \sin \delta_{fr} \\ C_{p_{wx}}^{fl} \sin \delta_{fl} + C_{p_{wy}}^{fl} \cos \delta_{fl} & C_{p_{wx}}^{fr} \sin \delta_{fr} + C_{p_{wy}}^{fr} \cos \delta_{fr} \\ C_{p_{wx}}^{fl} \chi_{xfl} + C_{p_{wx}}^{fl} \chi_{yfl} & C_{p_{wx}}^{fr} \chi_{xfr} + C_{p_{wx}}^{fr} \chi_{yfr} \end{bmatrix} \\
B_{pw}^{rear} &= \begin{bmatrix} C_{p_{wx}}^{rl} \cos \delta_{rl} - C_{p_{wy}}^{rl} \sin \delta_{rl} & C_{p_{wx}}^{rr} \cos \delta_{rr} - C_{p_{wy}}^{rr} \sin \delta_{rr} \\ C_{p_{wx}}^{rl} \sin \delta_{rl} + C_{p_{wy}}^{rl} \cos \delta_{rl} & C_{p_{wx}}^{rr} \sin \delta_{rr} + C_{p_{wy}}^{rr} \cos \delta_{rr} \\ C_{p_{wx}}^{rl} \chi_{xrl} + C_{p_{wx}}^{rl} \chi_{yrl} & C_{p_{wx}}^{rr} \chi_{xrr} + C_{p_{wx}}^{rr} \chi_{yrr} \end{bmatrix}
\end{aligned} \tag{5.31}$$

Control effectiveness matrix B_{R_z} 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 the 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 $C_{\gamma i}$, the coefficients $C_{p_{wx}}$ and $C_{p_{wy}}$ depending on tire pressure, and the 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. Tire model is represented in Chapter 3.

The control allocation problem is formulated as the minimization of allocation error and control actuations, taking into account actuator constraints [51]:

$$u^{CA} = \arg \min_{u_{lim}^{low} \leq u^{CA} \leq u_{lim}^{up}} \left(\|W_v(B_u u^{CA} - v^*)\|_2^2 + \xi \|W_u u^{CA}\|_2^2 \right) \tag{5.32}$$

The parameter ξ defines a significance of the penalization of control actuations and it is equal to 0.1 according to the recommendations from [51]. Matrix W_v is used to set up a priority among generalized longitudinal and lateral forces and yaw torque according to vehicle manoeuvre. A description of weighting matrix W_u to restrict actuators is described below in the Chapter 6. The fixed-point method is selected as an optimization solver, because it is suitable for the application in the real-time systems [47]. The fixed-point method is terminated when the allocation error is lower than the predefined tolerance.

Limitation constraints related to longitudinal tire forces are considered only for electric motor and frictional brake system. The reason is that both these subsystems can generate significantly higher demand compared to dynamic tire pressure system and additional normal force actuators. In this case, limitation constraints can be formulated as:

$$\begin{aligned} u_{\min}^{tire} &= \begin{bmatrix} 0 & -F_{xw}^{\lim} r_w - T_{br,i} & -F_{xw}^{\lim} r_w \end{bmatrix} \\ u_{\max}^{tire} &= \begin{bmatrix} F_{xw}^{\lim} r_w & 0 & 0 \end{bmatrix} \end{aligned} \quad (5.33)$$

The limit of longitudinal force $F_{xw}^{\lim}$ is calculated from friction circle [137] using normal and lateral forces, and friction coefficient.

The actuator position and rate limits are used as optimization constraints to represent static actuator dynamics. Their values are shown in Table 5.2. Since the powertrain architecture has near-wheel motors, the rate limit of torque generated by the electric motor is less compared to the friction brake system due to a low natural frequency of the half-shafts. In the case of the application of in-wheel motor, the torque rate of electric motor can be higher [140].

Table 5.2 Position and rate limits of actuators

Subsystem	Position limit u^{pos}	Rate limit u_{rate}
front wheel steer actuators	-30...30 deg	50 deg/sec
rear wheel steer actuators	-15...15 deg	30 deg/sec
camber angle actuators	-10...10 deg	10 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 0.6 / 1 bar /sec
additional normal force actuators	1000 N	3000 N/sec

The final constraints for optimization can be written as:

$$\begin{aligned} u_{\lim}^{low} &= \max \left(u_{\min}^{pos}, u(T - T_s) - u_{rate} T_s, u_{\min}^{tire} \right) \\ u_{\lim}^{up} &= \min \left(u_{\max}^{pos}, u(T - T_s) + u_{rate} T_s, u_{\max}^{tire} \right) \end{aligned} \quad (5.34)$$

Control effectiveness matrix B_u is ill-conditioned due to a high-order difference between B_δ and others matrices. Therefore, the following problem-solving procedure is applied:

- scale final constraints and control effectiveness matrix:

$$B_u = \Lambda \otimes B_u; \quad u_{\text{lim}} = \Lambda^{-1} \otimes u_{\text{lim}}; \quad u_0 = \Lambda^{-1} \otimes u_0 \quad (5.35)$$

- solve optimization problem (5.32);
- rescaling of output control vector and control effectiveness matrix:

$$u^{CA} = \Lambda \otimes u^{CA}; \quad B_u = \Lambda^{-1} \otimes B_u \quad (5.36)$$

The scaling factor is calculated for each subsystem independently and it is equal to:

$$\Lambda_j = \|B_j\|_F^{-1}, \quad j = \{\delta, \gamma, p_w, R_z\} \quad (5.37)$$

To take into account normal weight distribution for the torques generated by friction brake system and electric motors, the scaling factor is calculated as:

$$\Lambda_j = \frac{F_{zi}}{F_{zi}^{static}} \|B_j\|_F^{-1} \quad (5.38)$$

5.5 Low-level actuator control

Low-level control is used to generate control actuator signals and to calculate constraints for middle-level control allocation. Since several vehicle subsystems are represented as transfer functions, which were described in the Chapter 3, the reference signals from wheel steer actuators, camber angle actuators and actuators generating additional normal forces are directly assigned to vehicle model.

5.5.1 Electric motor controller and its constraints

The controller of the electric motor is the PI controller [141]. The reference signals are torques obtained from control allocation. The actual signal is estimated real torque. It is assumed that estimated real torque is obtained from the estimation procedure with sufficient accuracy.

$$u_{em,i} = K_p^{em} (T_{em,i}^{CA} - T_{em,i}) + K_i^{em} \int (T_{em,i}^{CA} - T_{em,i}) dt \quad (5.39)$$

The position torque limits of an electric motor can be found as [142]:

$$T_{em,i}^{\lim} = T_{em,i}^{\max} (\omega_{emi}) w_1(T_i) w_2(SOC) w_3(V_{xa}) w_4(U_{vi}) w_5(fault) \quad (5.40)$$

The maximum overload torque $T_{em}^{\max}$ of an electric motor is obtained from the look-up table of the torque-speed map, which is stored in the controller, according to motor speed.

Maximum overload torque is limited by its thermal time constant [143]. According to the thermal analysis in [144], the winding is the most heated element of the electric motor. The weight function $w_1(T_i)$ describes the influence of winding temperature on maximum overload torque, and it is calculated as:

$$w_1(T_i) = \begin{cases} 1, & T_i < T_{\min} \\ 1 - 2 \left(\frac{T_i - T_{\min}}{T_{\max} - T_{\min}} \right)^2, & T_{\min} \leq T_i \leq \frac{T_{\min} + T_{\max}}{2} \\ 2 \left(\frac{T_i - T_{\max}}{T_{\max} - T_{\min}} \right)^2, & \frac{T_{\min} + T_{\max}}{2} \leq T_i \leq T_{\max} \\ 0, & T_i > T_{\max} \end{cases} \quad (5.41)$$

The second correction of maximum overload torque is related to the state-of-the-charge (SOC). As the battery SOC decreases, battery voltage and discharge capability are being reduced:

$$w_2(SOC) = \begin{cases} 1, & SOC < SOC_{min} \\ SOC_{max} - SOC, & SOC_{min} \leq SOC \leq SOC_{max} \\ 0, & SOC > SOC_{max} \end{cases} \quad (5.42)$$

SOC_{min} is defined as the level of SOC, when reduction of battery voltage influences on the value of maximum overload torque. The parameter SOC_{max} corresponds to the situation, when only nominal torque is generated.

The electric motor provides insignificant recuperation at low vehicle velocities. High power is generated on high velocity, because beyond the base velocity, voltage remains constant and the flux is weakened [145].

$$w_3(V_{xa}) = \begin{cases} 0, & V_{xa} < V_{xa}^{non-recup} \\ \frac{V_{xa} - V_{xa}^{non-recup}}{10}, & V_{xa}^{non-recup} \leq V_{xa} \leq V_{xa}^{recup} \\ 1, & V_{xa} > V_{xa}^{recup} \end{cases} \quad (5.43)$$

The reduction of battery voltage causes the increasing of current and, as a result, the rise of temperature. Therefore, the weight function $w_4(U_{vi})$ provides the undervoltage and overvoltage protection of an electric motor [146]:

$$w_4(U_{vi}) = \begin{cases} 0, & U_{vi} < U_{vmin} \\ 1, & U_{vmin} \leq U_{vi} \leq U_{vmax} \\ 0, & U_{vi} > U_{vmax} \end{cases} \quad (5.44)$$

The weight function of fault control is proposed as:

$$w_5(fault) = \begin{cases} 0, & \text{failure detected} \\ 1, & \text{no failure} \end{cases} \quad (5.45)$$

The parameters of electric motor controller are listed in Table 5.3.

Table 5.3 Controller parameters of electric motor

K_p^{em}	K_I^{em}	T_{min}, C	T_{max}, C	SOC_{min}
1	150	40	60	0.5
SOC_{max}	$V_{xa}^{non-recup}, \text{ km/h}$	$V_{xa}^{recup}, \text{ km/h}$	U_{vmin}, V	U_{vmax}, V
0.9	30	40	300	320

5.5.2 Friction brake system controller and its constraints

On failure of the electro-hydraulic unit, the control signal to the j^{th} block valve is calculated as:

$$u_{j,block} = \begin{cases} 1, & \text{failure detected} \\ 0, & \text{no failure} \end{cases} \quad (5.46)$$

Under normal working conditions, the PD control of the inlet and outlet valves is applied to realize the pressure tracking control. In the case when the reference pressure equals to zero, the outlet valves are opened. The error between measured and reference brake pressure for each brake mechanism is defined as:

$$e_{br,i} = p_{br,i}^{ref} - p_{br,i}^{mes} \quad (5.47)$$

The reference brake pressure $p_{br,i}^{ref}$ is found as:

$$p_{br,i}^{ref} = \frac{T_{br,i}^{CA}}{0.5\mu_0\pi d_{cyl}^2 r_{eff}} \quad (5.48)$$

The control law for the pressure controller is described as:

$$u_{br,i} = K_p^{br} e_{br,i} + K_d^{br} \frac{de_{br,i}}{dt} \quad (5.49)$$

The ideal control signal to the i^{th} inlet valve is calculated as:

$$u_{i,inlet}^{ref} = \begin{cases} 1, & e_{br,i} \geq 0 \ \& \ |u_{br,i}| \geq 1 \\ |u_{br,i}|, & e_{br,i} \geq 0 \ \& \ |u_{br,i}| < 1 \\ 0, & e_{br,i} < 0 \end{cases} \quad (5.50)$$

The ideal control signal to the i^{th} outlet valve is calculated as:

$$u_{i,outlet}^{ref} = \begin{cases} 0, & e_{br,i} \geq 0 \\ |u_{br,i}|, & e_{br,i} < 0 \ \& \ |u_{br,i}| < 1 \\ 1, & e_{br,i} < 0 \ \& \ |u_{br,i}| \geq 1 \end{cases} \quad (5.51)$$

The control signal to the j^{th} compensating valve is calculated as:

$$u_{j,compen} = \begin{cases} 1, & p_j^{right} \approx p_j^{left} \\ 0, & p_j^{right} \neq p_j^{left} \end{cases} \quad (5.52)$$

For HIL test rig, the configuration of the brake system incorporates the hydraulic control unit that operates the relay valves. Its control law combines PI gain-scheduling

control to demand calculation (5.53) and state machine algorithm (Figure 5.4) to define pressure build-up, hold and release stages for the demand realization.

$$v_{br,i} = K_p^{br} (p_{br}^{ref}) e_{br,i} + K_i^{br} (p_{br}^{ref}) \int e_{br,i} dt \quad (5.53)$$

Figure 5.4 State machine algorithm
for the controller of friction brake system (rear left wheel)

The constructive parameters of brake mechanisms and the thresholds of wheel slip control define the limitation of friction brake torque. Therefore, the position limits for friction brake torque $T_{br,i}^{lim}$ are calculated taking into account the constructive limits T_i^{constr} and wheel slip:

$$T_{br,i}^{lim} = \begin{cases} T_i^{constr}, & s_{wi} < s_w^{threshold} \\ T_{br,i}^{CA} - T_{react}, & s_{wi} > s_w^{threshold} \end{cases} \quad (5.54)$$

The reactive torque T_{react} is calculated using PI algorithm and it is similar to control of anti-lock brake system [147]:

$$T_{react} = \begin{cases} 0, & s_{wi} < s_w^{threshold} \\ K_p^{react} (s_w^{threshold} - s_{wi}) + K_i^{react} \int (s_w^{threshold} - s_{wi}) dt, & s_{wi} > s_w^{threshold} \end{cases} \quad (5.55)$$

The parameters of brake pressure controller are listed in Table 5.4.

Table 5.4 Parameters of brake pressure controller

K_p^{br}	0.3	T^{constr} , Nm	front	2000
K_d^{br}	1		rear	1400
K_p^{react}	500	μ_0		0.42
K_i^{react}	1000	$s_w^{threshold}$		0.1

5.5.3 Controller of dynamic tire pressure system

For simulation-based analysis the additional tire pressure is calculated from the Chapter 3; however, in the test rig the tire pressure controller should be implemented. As compared to other vehicle subsystems, the performance of investigated dynamic tire pressure system is slow up to 1 bar/s. The present configuration of the system includes the rule-based controller. The error between measured and reference tire pressure is calculated as:

$$e_{p_w,i} = p_{w,i}^{ref} - p_{w,i}^{mes} \quad (5.56)$$

Valve control signals to the inlet valve $u_{i,inlet}^{pw}$ are defined as:

$$u_{i,inlet}^{pw} = \begin{cases} 1, & e_{p_w,i} > 0.5 \\ 0, & e_{p_w,i} < 0.5 \& e_{p_w,i} > 0.1 \& t_{hold} > 0.02 \\ 1, & e_{p_w,i} < 0.5 \& e_{p_w,i} > 0.1 \& t_{hold} < 0.02 \\ 0, & e_{p_w,i} < 0.1 \end{cases} \quad (5.57)$$

Valve control signals to the outlet valve $u_{i,outlet}^{pw}$ are formulated as:

$$u_{i,outlet}^{pw} = \begin{cases} 1, & e_{p_w,i} > -0.5 \\ 0, & e_{p_w,i} < -0.5 \& e_{p_w,i} > -0.1 \& t_{hold} > 0.02 \\ 1, & e_{p_w,i} < -0.5 \& e_{p_w,i} > -0.1 \& t_{hold} < 0.02 \\ 0, & e_{p_w,i} < -0.1 \end{cases} \quad (5.58)$$

The holding time t_{hold} is used to generate step changing of tire pressure, when the error is smaller than 0.5 bar.

5.6 Vehicle state observer

5.6.1 Brief description of state-of-the-art of state observers for vehicle dynamics

Due to lack of direct measurement, several vehicle states like sideslip angle, roll angle should be calculated during vehicle motion. Uncertainties and measurement noise can distort calculation accuracy. Therefore, various estimation techniques are applied to estimate vehicle states to actualize them in control algorithms more precisely.

To design a state observer, the main research stages can be formulated as:

- definition of estimated states;
- type of observer structure;
- algorithm of estimation;
- complexity of process models, which will be involved into the estimation procedure.

The main estimated vehicle states are longitudinal and lateral velocities [148], sideslip angle [149], tire forces [150] and others. The measurement of these states is expensive and it requires an installation of optical sensor, wheel force hubs and other non-standard equipment. The observer structure for one or two parameters includes typically one observer. For the parameter set, two and more observers can be applied, which are working in parallel [151] or in series [152].

The different estimation techniques can be used starting from deterministic (rule-based, recursive least square method (RLSM)) to probabilistic approach (Kalman filter (KF), particle filter). The application of recursive estimation like KF or RLSM is effective and it is the most common technique to solve such problems in the automotive field. There are three different types of KF: linear KF, extended KF (EKF) for nonlinear systems and unscented KF (UKF). The main benefit of UKF is that the model linearization is not necessary and model equations can be non-differentiable. However, UKF requires higher computational load [153]. Nevertheless, the application of UKF for real-time estimation of vehicle motion is demonstrated [154]. In this section, the EKF is selected as an estimation technique for vehicle longitudinal and lateral velocities and tire forces.

To achieve a precise estimation without significant computational budget and unjustified complexity, the selection of a process model is important. The first step is the selection of vehicle model. The common approach is an application of classical bicycle vehicle model [149, 155]. The two-track vehicle models from 7 DoF [156] up to 13 DoF

[157] are seldom used. The comparison between single-track and four-wheel models in EKF is shown in [152, 158]. In slalom test both models ensure a good estimation compared to experimental measurements, but under high load transfer and steering angle, the four-wheel model demonstrates much better performance. It should be noted that in [152] rear longitudinal forces are neglected. For the all-wheel drive vehicle, this assumption is unsuitable. For proposed state observer the 7 DoF vehicle model is selected to take into account effects of longitudinal and lateral load transfers.

The second step is the selection of tire model. Various tire models can be implemented in EKF, such as the trigonometric model [149], Magic Formula and TMeasy [151], UniTire model [155] and others. Instead of tire model, the stochastic Gauss-Markov model of friction forces can be applied [150]. In [158] the significant impact of relaxation length on estimation accuracy is demonstrated. Therefore, the relaxation lengths for longitudinal and lateral forces are introduced into the state observer. The tire model is an extended Pacejka model, as described in the Chapter 3.

5.6.2 Vehicle states and tire forces estimation based on EKF

The estimated vehicle states of the observer are:

- longitudinal V_{xa} and lateral V_{ya} velocities;
- longitudinal F_{xw} and lateral F_{yw} tire forces.

The input parameters for the observer are steering angles of front wheels, yaw rate, longitudinal and lateral accelerations, wheel speeds and reference wheel torques from the control system. Normal tire forces F_z are calculated based on estimated kinematic states of the vehicle motion.

The following parameters of the vehicle are set up into the observer: sprung vehicle mass, front and rear unsprung masses, distance from front axle and from rear axle to CoG of sprung mass, front and rear track width, moment of inertia of sprung mass about z-axis, height of sprung mass CoG above ground, height of front and rear unsprung masses CoG above ground, distance between ground and front / rear static roll center, wheel inertia, tire rolling radius, longitudinal and lateral relaxation lengths, roll stiffness and damping of front and rear suspension.

The simplified vehicle model with 3 DoF represents longitudinal, lateral and yaw dynamics. The vehicle behaviour is described by the following equations:

$$\begin{aligned}
\frac{dV_{xa}}{dt} &= V_{ya} \frac{d\psi}{dt} + \frac{1}{m_a} \sum_{i=1}^4 F_{xi} \\
\frac{dV_{ya}}{dt} &= -V_{xa} \frac{d\psi}{dt} + \frac{1}{m_a} \sum_{i=1}^4 F_{yi} \\
\frac{d^2\psi}{dt^2} &= \frac{1}{I_{zz}} \left(a \sum_{i=1}^2 F_{yi} - b \sum_{i=3}^4 F_{yi} + 0.5t_f (F_{x1} - F_{x2}) + 0.5t_r (F_{x3} - F_{x4}) \right)
\end{aligned} \tag{5.59}$$

The equations describing wheel dynamics, tire steady-state and transient behaviour are described in the Chapter 3.

The normal force distribution is calculated as:

$$\begin{aligned}
F_{z1} &= \frac{m_s g b_s}{2L} + \frac{m_{uf} g}{2} - F_z^{a_x} - F_z^{KF a_{yf}} \\
F_{z2} &= \frac{m_s g b_s}{2L} + \frac{m_{uf} g}{2} - F_z^{a_x} + F_z^{KF a_{yf}} \\
F_{z3} &= \frac{m_s g a_s}{2L} + \frac{m_{ur} g}{2} + F_z^{a_x} - F_z^{KF a_{yr}} \\
F_{z4} &= \frac{m_s g a_s}{2L} + \frac{m_{ur} g}{2} + F_z^{a_x} + F_z^{KF a_{yr}}
\end{aligned} \tag{5.60}$$

Load transfer owed to the longitudinal acceleration is shown in the Chapter 3. Load transfers owed to the lateral accelerations are defined as:

$$\begin{aligned}
F_z^{KF a_{yf}} &= \frac{a_y}{t_f} \left(\frac{m_s b h_f}{L} + m_{uf} h_{uf} \right) + \frac{m_s a_y}{t_f} (h_{cgs} - h_f) \frac{K_{\phi f} \phi}{K_{\phi f} \phi + K_{\phi r} \phi} \\
F_z^{KF a_{yr}} &= \frac{a_y}{t_r} \left(\frac{m_s a h_r}{L} + m_{ur} h_{ur} \right) + \frac{m_s a_y}{t_r} (h_{cgs} - h_r) \frac{K_{\phi r} \phi}{K_{\phi f} \phi + K_{\phi r} \phi}
\end{aligned} \tag{5.61}$$

The vehicle state observer consists of an EKF to estimate the state vector $x(t)$, comprising:

$$x(t) = [V_x \ V_y \ \psi \ \omega_{w1} \ \omega_{w2} \ \omega_{w3} \ \omega_{w4} \ F_{xw1} \ F_{xw2} \ F_{xw3} \ F_{xw4} \ F_{yw1} \ F_{yw2} \ F_{yw3} \ F_{yw4}]^T$$

The measured vector $z(t)$ is:

$$z(t) = [a_x \ a_y \ \dot{\psi} \ \omega_{w1} \ \omega_{w2} \ \omega_{w3} \ \omega_{w4}]^T$$

Based on vehicle model description, Jacobians of process matrix A and measurement matrix C are found and they are depicted in Appendix A.3.

For a discretization of continuous time system, its discrete equivalent with sampling period of T_s is found in the form as:

$$\begin{aligned}
\Phi &= I + A\Lambda \\
H &= C \\
\Lambda &= \int_0^{T_s} e^{As} ds = IT_s + \frac{AT_s^2}{2!} + \frac{A^2 T_s^3}{3!} + \dots + \frac{A^i T_s^{i+1}}{(i+1)!} + \dots
\end{aligned} \tag{5.62}$$

The initial value of state vector $x(t)$ is defined as:

$$x_0 = \begin{bmatrix} 0.25 \sum_{i=1}^4 \omega_{wi} r_w & 0 & 0 & \omega_{w1} & \omega_{w2} & \omega_{w3} & \omega_{w4} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}^T$$

The initial value of error covariance matrix P_0 is set up as:

$$P_0 = \text{diag}([1 \ 1 \ 1 \ 1 \ 1 \ 1 \ 1 \ 1 \ 1 \ 1 \ 1 \ 1 \ 1 \ 1 \ 1])$$

The calculation algorithm of EKF has the following consequence [159]:

- prediction step of the EKF uses the discrete versions of equations (5.59)-(5.61):

$$\hat{x}_k^- = f(\hat{x}_{k-1}) \quad (5.63)$$

- *a priori* estimate error covariance is computed:

$$P_k^- = \Phi P_{k-1} \Phi^T + Q \quad (5.64)$$

- Kalman gain is computed based on Jacobians from Appendix A.3:

$$K_k = P_k^- H^T (H P_k^- H^T + R)^{-1} \quad (5.65)$$

- *a posteriori* estimate error covariance is computed:

$$P_k = (I - K_k H) P_k^- \quad (5.66)$$

- predicted measurement using the discretized versions of the measurement equations:

$$\hat{z}_k = f(\hat{x}_k^-) \quad (5.67)$$

- new predicted state updates using the Kalman gain:

$$\hat{x}_k = \hat{x}_k^- + K_k (z_k - \hat{z}_k) \quad (5.68)$$

The measurement noise matrix R and process noise matrix Q are selected from the recommendations in [159]:

$$R = \text{diag}([5e-2 \ 5e-2 \ 5e-2 \ 5e-2 \ 5e-2 \ 5e-2 \ 5e-2])$$

$$Q = \text{diag}([1e^{-4} \ 1e^{-4} \ 4e^{-4} \ 1 \ 1 \ 1 \ 1 \ 5e^{+3} \ 5e^{+3} \ 5e^{+3} \ 5e^{+3} \ 5e^{+3} \ 5e^{+3} \ 5e^{+3} \ 5e^{+3}])$$

5.7 Chapter summary

This chapter describes the multi-layer control system, which consists of high-level controller for vehicle dynamics, middle-level control allocation, low-level actuator controller and vehicle state observer.

The high-level controller includes reference generator and vehicle dynamics controller. The reference longitudinal acceleration, reference sideslip angle, when vehicle moves outside stability boundaries, and reference yaw rate are generated by the reference model using steering angle and brake pedal displacement from driver. Generalized control demands of longitudinal and lateral forces, and yaw torque are calculated using discrete second-order sliding mode control.

The middle layer is optimization-based control allocation, which distributes generalized control demands between vehicle subsystems: wheel steer actuators, electric powertrain, friction brake system, camber angle actuators, dynamic tire pressure system and actuators generating additional normal forces. Optimization constraints are formulated taking into account static actuator dynamics (position and rate limits) and tire friction limits. The fixed-point method with termination conditions is used as an optimization solver.

The low-level control allows to achieve precise tracking between reference control signals obtained from the middle layer and generated actuations. Moreover, it is used to estimate boundary conditions for control allocation taking into account actuator physical limits and wheel slip control.

Longitudinal and lateral velocities are estimated using a vehicle state observer based on extended Kalman filter. The 7 DoF vehicle model and Pacejka model with relaxation lengths in longitudinal and lateral directions are used as a process model.

Chapter 6

6 Subsystem coordination and its simulation-based analysis

This chapter includes the analysis of stand-alone and joint operation of vehicle subsystems and their influence on vehicle motion performance, energy consumption and losses. After the demonstration of the necessity to apply a multi-actuator configuration, the subsystem prioritization is introduced using weight coefficients into the control allocation. The influence of the introduced weights on vehicle operation characteristics is analysed. After the analysis, the algorithms of the coordination of priorities between vehicle subsystems based on the restriction weights are proposed.

6.1 Assessment criteria for testing manoeuvres

The following assessment criteria are selected to evaluate vehicle operation characteristics for simulated manoeuvres:

- mean absolute error of longitudinal acceleration, m/s^2 ;
- brake distance for straight-line braking manoeuvre, m ;
- mean absolute error of yaw rate, deg/s ;
- maximum of absolute sideslip angle during the manoeuvre, deg ;
- longitudinal velocity at the end of the manoeuvre, km/h ;
- energy consumption by electric motors, kJ , is calculated as potential energy, assuming there are no rate constraints of energy accumulator in the case of recuperation:

$$E_{em}^{total} = 10^{-3} \sum_{i=1}^4 \int_0^T P_{em,i} dt ; \quad (6.1)$$

- total tire energy dissipation in longitudinal and lateral directions, kJ :

$$E_{tire}^{total} = 10^{-3} \sum_{i=1}^4 \int_0^T (F_{xwi} V_{swxi} + F_{ywi} V_{swyi}) dt ; \quad (6.2)$$

- maximum value of stability index during the manoeuvre [23], which is related to the vehicle sideslip motion and can be found from the phase-plane $\beta_{est} - \dot{\beta}_{est}$:

$$\chi_{st} = \left| \frac{1}{24} \dot{\beta}_{est} + \frac{4}{24} \beta_{est} \right| < 1; \quad (6.3)$$

6.2 A need for multi actuators in vehicle dynamics control

Since a multi-actuated vehicle is considered, it allows to implement various subsystems for the generation of control demand to compensate control error. Many authors noted [39, 41, 115, 134] that the integration of several vehicle subsystems helps to achieve better vehicle motion characteristics and energy consumption compared to stand-alone operation.

Investigations of stand-alone and joint operation of vehicle subsystems are considered for steady-state and transient vehicle behaviour. The following subsystems are investigated:

- IFS – individual front wheel steering;
- IRS – individual rear wheel steering;
- EP – traction torques generated by electric motors;
- EBS – 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.

Individual wheel steer actuators are separated between front and rear actuators due to different actuator dynamics. The reference additional normal forces, defined by the control system, are separated in equal portions between external spring, damping and stabilizer forces. The dynamics of vehicle subsystems are described in Chapter 3.

6.2.1 Stand-alone operation of vehicle subsystems

‘Slowly Increasing Steer Test’ manoeuvre

To analyse the influence of each vehicle subsystem on steady-state vehicle behaviour, the ‘Slowly Increasing Steer Test’ manoeuvre is considered. It evaluates vehicle behaviour in both linear and nonlinear regimes of vehicle motion. In accordance with the procedure proposed in [160], steering angle increases from zero at a rate of 13.5 deg/s. The goal of the test is to reach driving conditions near to the limits defined by tire-road friction. In addition, the results of the test are input data for transient tests that have

been further performed in this study. A vehicle moves with the constant speed of 80 km/h. The tracking of yaw rate is only controlled.

The simulation results are shown in Figure 6.1. The dash black line corresponds to behaviour of linear reference model from Chapter 5.3.1. The solid black line shows nonlinear vehicle motion without any actuations. Other lines demonstrate vehicle motion in the case of the application of certain subsystem. As can be seen from Figure 6.1, the application of each subsystem enlarges the linear range of steering characteristic.

Figure 6.1 Simulation results of steady-state manoeuvre

Deeper analysis of obtained characteristics allows to formulate a number of conclusions:

- all considered subsystems provide a similar vehicle behaviour in the linear area, they have visible influence on the steering characteristic in the nonlinear area, when lateral acceleration is more than 6 m/s². In particular, a steering wheel angle gradient δ/a_y for $a_y=8$ m/s² is varied from 8,2 (IFS) to 9,17 (w/o control) deg/m/s²;
- all subsystems can keep peak sideslip angle during the manoeuvre near to the threshold of 3 deg. Small increase of angle is observed only for ACS, IRS and EP;
- FBS causes high energy consumption up to 64.4 kJ to support constant speed,

while it improves tracking performance of yaw rate up to 0.3 deg/s compared to the motion without control;

- minimal energy consumption of 186.4 kJ is obtained in the case of EP; however, maximum of sideslip angle is one of the highest in this case up to 3.6 deg;
- under high level of lateral acceleration of 8.75 m/s^2 , APS causes sharp changing of steering characteristic that influences unfavourably not only the vehicle handling but also the steering feel. Therefore, its application in this range is ineffective.

‘Steep Steer’ manoeuvre

The manoeuvre is carried out at constant velocity of 100 km/h and maximal steering amplitude of 80 deg [160]. Only yaw rate tracking is carried out. The lateral acceleration reached the friction limits.

Figure 6.2 Simulation results of ‘Steep Steer’ manoeuvre at constant speed

An analysis of simulation results (Figure 6.2) points to the following inferences:

- comparing to the motion without control, each variant provides better tracking performance of yaw rate;
- the highest energy consumption of 339.2 kJ due to the compensation of velocity

reduction is obtained for FBS, while it allows to achieve good tracking performance of 1.22 deg/s and sideslip angle of 4.79 deg;

- minimal sideslip angle of 3.72 deg is obtained by IRS;
- APS causes the high sideslip angle of 7.57 deg, which is close to the motion w/o control;
- joint operation of actuators can provide a compromise solution between motion tracking performance, energy consumption and losses.

Regarding normal driving conditions, the investigation of stand-alone operation of vehicle subsystems on vehicle handling performance, energy consumption and tire energy dissipation was analysed in frequency domain in author's publication [161].

6.2.2 Joint operation of vehicle subsystems

The second step is to investigate how joint operation of several subsystems influences on motion performance and energy characteristics during transient manoeuvres. Since the friction brake system is a typical component of any electronic stability system, the investigations are started from it. Each vehicle subsystem is added to the process of regulation on the each simulation step by step. The schematic sequence of subsystem adding is shown in Figure 6.3. Firstly, all subsystems generating wheel torque '...+EBS' and '...+EP' are considered. Then the subsystems generating additional wheel steer angles '...+IFS' and '...+IRS' and camber angles '...+ACS' are introduced. At the end, the additional normal forces '...+ARzS' and tire pressures '...+APS' are taken into account. Such kind of sequence is caused by the analysis of stand-alone operation from previous section. Final variant '...+APS' means the integration of all vehicle subsystems. All considering subsystems have the same priority during the operation.

Figure 6.3 Schematic sequence of subsystem integration

For more objective analysis of subsystem integration, complex manoeuvre like 'Sine with Dwell' is chosen for further investigations.

‘Sine with Dwell’ manoeuvre

‘Sine with Dwell’ manoeuvre is considered at initial velocity of 80 km/h and maximal steering amplitude of 120 deg on the road with high friction coefficient. Both yaw rate and sideslip angle are controlled. The quantitative values of mean absolute error of yaw rate, maximum of sideslip angle, maximum of stability index, longitudinal velocity at the end of the manoeuvre, energy consumption of electric motors and tire energy dissipation are shown in Figure 6.4. Yaw rate and sideslip angle versus time are depicted as well. The detailed description of assessment criteria are shown in the Section 6.1.

Figure 6.4 Simulation results of ‘Sine with Dwell’ manoeuvre

An assessment of their behaviour leads to a number of observations:

- all variants guarantee vehicle handling and stability of motion;
- application of FBS provides good tracking of yaw rate of 0.54 deg/s; however, it causes the reduction of longitudinal velocity up to 62 km/h;
- integration with EBS allows to recuperate energy up to -22.5 kJ, while the performance of yaw rate tracking is declined to 0.59 deg/s;
- inclusion of EP increases final vehicle velocity to 66 km/h; however, more

energy is consumed up to 1.4 kJ;

- application of wheel steer actuators reduces sideslip angle deg largely due to IRS;
- the combination of abovementioned subsystems allows to reach the best tracking performance of 0.58 deg/s, stability of motion (maximal sideslip angle of 3.3 deg) and low energy consumption of -4.7 kJ;
- implementation of ACS increases final vehicle velocity up to 69.5 km/h and reduces sideslip angle to 3.15 deg, when more energy is consumed up to 0.3 kJ;
- additional usage of normal forces and tire pressures allows to improve tracking performance to 0.48 and to a lesser extent other characteristics;
- operation of all considered subsystems provides the positive compromise between motion performance and energy characteristics. This compromise means that the joint operation keeps vehicle motion in required stability limits with simultaneous providing of higher longitudinal velocity at the end of manoeuvre and lower energy consumption.

Modified 'Double Lane Change' manoeuvre

The principal difference of modified 'Double Lane Change' manoeuvre [160] from ISO 3888 procedure is that no traction demand is applied to support vehicle speed. The initial longitudinal velocity is 120 km/h on high friction road. The wheel steering law is generated by a driver model from commercial software IPG/Carmaker. Both yaw rate and sideslip angle are controlled. The relationships between lateral versus longitudinal distance, yaw rate and sideslip angle versus time are depicted in Figure 6.5. The similar assessment criteria are used as in 'Sine with Dwell' manoeuvre.

The following conclusions are formulated:

- the manoeuvre is passed successfully for all considered variants;
- usage of FBS reduces significantly vehicle velocity up to 94 km/h;
- due to the application of electric motors (EBS and EP), vehicle velocity at the end of manoeuvre can be increased to 98 km/h;
- additional wheel steer angles helps to significantly increase vehicle velocity at the end of manoeuvre to 101.5 km/h, the best tracking performance of 0.99 deg/s, energy consumption of 11.2 kJ;

Figure 6.5 Simulation results of modified 'Double Lane Change' manoeuvre

- opposite to 'Sine with Dwell' manoeuvre, ACS application provides worse characteristics excluding vehicle velocity at the end of manoeuvre;
- implementation of ARzS and APS improves slightly all criteria of vehicle motion;
- final variant with all subsystems provides the tracking performance of 1.14 deg/s, the highest velocity of 104.5 km/h and minimal tire dissipation energy of 45.9 kJ, while other criteria are suitable.

6.2.3 Energy consumption of vehicle subsystems

The energy consumption of each vehicle subsystem during ‘Sine with Dwell’ manoeuvre is roughly calculated to evaluate how significant is energy consumption of other subsystems compared to electric powertrain. 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:

$$\begin{aligned} P_{IWS} &= \sum_{i=1}^4 M_{zwi} \delta_i^{ext}; & E_{IWS} &= 1.3 \int_0^T P_{IWS} dt; \\ P_{ACS} &= \sum_{i=1}^4 M_{\phi wi} \gamma_i^{ext}; & E_{ACS} &= 1.3 \int_0^T P_{ACS} dt \end{aligned} \quad (6.4)$$

The voltage and current levels of pump and valves of hydraulic unit into friction brake system are listed on Table 6.1.

Table 6.1 Voltage and current of pump and valves of hydraulic unit

Voltage, V	Current, A			
	Pump	Inlet valve	Outlet valve	High pressure and switch valves
12	3.2	1.2	2.4	2.4

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 pressure compressor is 50 W. The electric power consumption of active dampers is below 100 W for the entire vehicle [162]. For active suspension average power consumption is under 1.0 kW [163]. The maximal current of actuator of electromechanical anti-roll bar is 40 A under voltage of 17 V [164].

The simulation results are shown in Figure 6.6. The sum of energy consumptions of all actuators is less than 9.2% of energy consumption of electric motor. Therefore, energy consumption of other actuators can be neglected.

Figure 6.6 Energy consumption of all actuators during ‘Sine with Dwell’ manoeuvre

6.2.4 Mass of vehicle subsystems and customers' view

The total vehicle mass is increased due to the addition of active vehicle subsystems. The masses of each vehicle subsystem are listed on Table 6.2.

Table 6.2 Mass of each vehicle subsystem

Vehicle Subsystem	Mass, kg
Front steering	5.0
Rear steering	5.5 [165]
Camber system	6-7 per wheel [166]
Active damper	2.39 per wheel [162]
Active anti-roll bar	10-18 per axle [114]
Dynamic tire pressure system: all valves, pipelines, balloons and compressor	50 [167]

The increasing of vehicle mass, price and operation costs due to additional vehicle subsystems stimulates to categorize a need for multi actuators from customers' points of view. Since the all-wheel drive vehicle with near-wheel electric powertrain is considered as specified research object, the actuator configurations can be proposed for three vehicle categories:

- *city and family cars*: the specified configuration of all-wheel drive powertrain is unnecessary. The mandatory component is controlled friction brake system according to the EC regulation on General Safety [168];
- *executive and mid-luxury cars*: abovementioned configuration can be extended by active front and rear steering system to improve handling and vehicle stability;
- *luxury and sport cars*: introduction of vehicle subsystems, such as active camber control, variable suspension, active dampers, active anti-roll bars and dynamic tire pressure system, allows to improve ride comfort, to increase the final velocity at the end of manoeuvre, and, as a result, to receive more scores from typical drive tests for high-performance cars. Thereby, it increases also customers' acceptance for this kind of vehicles;
- extension of multi-actuated vehicle technology to small- and middle-class cars makes sense in the case of considerable reduction of the weight and cost of mechatronic vehicle subsystems that can be expected in near future.

6.3 Concept of vehicle subsystem prioritization

The research idea of the thesis is to prioritize each vehicle subsystem, when control demand is developed, in accordance with specific performance criteria, which is related to vehicle operation characteristics. On the left-hand graph of Figure 6.7 the illustrative example of the demand distribution is shown, when all vehicle subsystems have the same priority. As a result, the ratios between demands v_1 , v_2 and v_3 are constant.

Figure 6.7 Illustrative example of actuator coordination/prioritization

The right-hand graph of Figure 6.7 demonstrates a situation when priority of each subsystem changes according to the performance criterion. In this case, the ratios between demands v_1 , v_2 and v_3 are different. Their values depend on which subsystem has higher priority for the current range of performance criterion. For explanation, three subsystems are considered, which form a certain control demand like generalized yaw torque. If the performance criterion, for instance – control error of yaw rate, is small, control demand will be mainly generated only by steering and drive system. It allows to reduce the usage of the friction brake system to decrease non-recuperative energy losses. When the level of control error is increased, the participation of the friction brake system rises, because steering and drive subsystems cannot develop control demand precisely due to the position and rate limits.

The following arguments for the subsystem prioritization can be formulated:

- dynamics of each actuator is unique;
- each type of actuator has own internal energy consumption and energy efficiency;
- load levels acting on the vehicle during actuator operation are different, for instance, longitudinal and lateral rolling resistance;
- some actuators have a possibility to recuperate energy.

6.4 Influence of subsystem prioritization on vehicle operation characteristics

As noted in the Section 5.4, the weight coefficients in matrix W_u are used for the prioritization of vehicle subsystems. The higher value of the weight coefficient means higher restriction or lower priority of vehicle subsystem. To analyse the subsystem prioritization, the weight coefficients are varied within the normalized range.

6.4.1 Subsystem prioritization during ‘Sine with Dwell’ manoeuvre

The study is performed for the case of the vehicle manoeuvre ‘Sine with Dwell’ with the same initial conditions as in the Section 6.2.2.

It should be noted that simulation results will be dependent on a sequence of subsystem consideration. The prioritization of vehicle subsystems is analysed in three stages:

1) Prioritization between friction brake system and electric powertrain (traction and brake torques) is considered. Other subsystems are deactivated. The weight coefficients r_i are changed from 0 (no restriction) to 3 (max. restriction). Assuming that the restriction weights can be defined by the following rule:

$$r_1 + r_2 + r_3 = 3; \quad 0 \leq r_i \leq 3 \quad (6.5)$$

the matrix W_u is:

$$W_u = \text{diag} \left(\left[1^{1 \times 2}, 1^{1 \times 2}, r_3^{1 \times 4}, r_2^{1 \times 4}, r_1^{1 \times 4}, 1^{1 \times 4}, 1^{1 \times 4}, 1^{1 \times 4} \right] \right) \quad (6.6)$$

This assumption allows to present graphically three variants together using ternary plots. The simulation results of the prioritization between FBS, EBS and EP with other deactivated subsystems are shown in Figure 6.8. It can be seen that during the whole range of simulations, mean absolute error of yaw rate has been varied from 0.5 to 0.7 deg/s (Figure 6.8a), maximum of sideslip angle – from 3.8 to 4.4 deg (Figure 6.8b), longitudinal velocity at the end of the manoeuvre – from 62 to 75 km/h (Figure 6.8c), total consumed energy by electric motors – from -20 to 50 kJ (Figure 6.8d) and total tire energy dissipation– from 37 to 41 kJ (Figure 6.8e). This example shows that a wide spread of cost function values can be observed even for only three actuated variants.

The best tracking performance of yaw rate and minimal sideslip angle are obtained at the range of the restriction of FBS (0.3-0.9) and EP (0.3-1.5). Meantime, low restriction of FBS (0.3-0.6) induces reduction of longitudinal velocity. The increasing of

longitudinal velocity can be reached by more usage of EP, but it causes higher energy consumption.

To define the restriction weights for the next stage, the criterion combining vehicle handling, stability, energy consumption and losses is introduced. It consists of two parts related to abovementioned characteristics:

$$C_{complex} = 0.7C_{s\&h} + 0.3C_{energy} \quad (6.7)$$

It is assumed within the framework of this study that the criterion of vehicle stability and handling has higher priority compared to criterion of energy consumption and energy losses from Eq. (6.7).

Criterion of vehicle stability and handling is defined as:

$$C_{s\&h} = 0.7 \frac{\bar{e}_{\dot{\psi}}}{\max(\bar{e}_{\dot{\psi}})} + 0.3 \frac{\beta_{est}^{\max}}{\max(\beta_{est}^{\max})} \quad (6.8)$$

The weight factors in Eq. (6.8) are given for a high friction road, where mean error of yaw rate should have high priority. For a low friction road, the weight factor of sideslip angle should be higher.

Criterion of energy consumption and energy losses is calculated as:

$$C_{energy} = 0.5 \frac{E_{em}^{total}}{\max(E_{em}^{total})} + 0.3 \frac{\Delta V_{xa}}{\max(\Delta V_{xa})} + 0.2 \frac{E_{tire}^{total}}{\max(E_{tire}^{total})} \quad (6.9)$$

Energy consumption and energy losses during the motion have the same priority and the same weights. The criterion of energy losses includes loss of velocity during the manoeuvre and tire energy dissipation.

The criterion $C_{complex}$ is shown in Figure 6.8f. According to its global extremum within the normalized range, the restriction weights for the next stage are selected as: 1.25 for EP, 1.25 for EBS and 0.5 for FBS (marked as red point).

2) The next stage considers the prioritization between subsystems from previous stage *gr#1* and individual wheel steer actuators IFS and IRS. The explanation of chosen set is that wheel steer actuators can potentially generate higher lateral force and yaw torque. Consequently, they have more influence on lateral dynamics in general compared to ACS, ARzS and APS, which are deactivated on this stage. The matrix W_u is defined as:

$$W_u = diag \left(\left[r_2^{1 \times 2}, r_3^{1 \times 2}, 1.25r_1^{1 \times 4}, 1.25r_1^{1 \times 4}, 0.5r_1^{1 \times 4}, 1^{1 \times 4}, 1^{1 \times 4}, 1^{1 \times 4} \right] \right) \quad (6.10)$$

Figure 6.8 Prioritization of FBS, EBS and EP for 'Sine with Dwell' manoeuvre:

- (a) mean absolute error of yaw rate; (b) maximum of absolute sideslip angle during manoeuvre; (c) longitudinal velocity at the end of the manoeuvre; (d) energy consumption by electric motors; (e) total tire energy dissipation; (f) complex criterion

Figure 6.9 Prioritization of gr#1, IFS and IRS for 'Sine with Dwell' manoeuvre:

- (a) mean absolute error of yaw rate; (b) maximum of absolute sideslip angle during manoeuvre; (c) longitudinal velocity at the end of the manoeuvre; (d) energy consumption by electric motors; (e) total tire energy dissipation; (f) complex criterion

Figure 6.10 Prioritization of ACS, ARzS and APS for 'Sine with Dwell' manoeuvre:

- (a) mean absolute error of yaw rate; (b) maximum of absolute sideslip angle during manoeuvre; (c) longitudinal velocity at the end of the manoeuvre; (d) energy consumption by electric motors; (e) total tire energy dissipation; (f) complex criterion

Simulation results of the prioritization between gr#1, IFS and IRS are shown in Figure 6.9. Mean absolute error of yaw rate has been changed from 0.52 to 0.7 deg/s (Figure 6.9a), maximum of sideslip angle – from 3.2 to 3.5 deg (Figure 6.9b), longitudinal velocity at the end of the manoeuvre – from 62 to 69 km/h (Figure 6.9c), total consumed energy by electric motors – from 0 to 8 kJ (Figure 6.9d) and total tire energy dissipation – from 37 to 39 kJ (Figure 6.9e).

In the case of the restriction of gr#1 (0.6-0.9) and IRS (0.9-2.1) the best tracking performance of yaw rate can be achieved. Meantime, the minimal sideslip angle and the highest velocity at the end of manoeuvre is obtained under low restriction of IFS (0-0.6) and IRS (0-0.6). The lowest energy consumption is in the range of IFS (0-0.6) and IRS (1.5-2.4).

Using the same criterion combined vehicle handling, stability, energy consumption and losses and its extreme plateau, as before (Figure 6.9f), the restriction weights for the next stage are selected as: 0.7 for gr#1, 0.7 for IFS and 1.6 for IRS (marked as red point).

3) On the next stage, the prioritization between ACS, ARzS and APS will be investigated at the fixed restriction weights of other subsystems, because, as obtained in the Section 6.3, the influence of considered subsystems ACS, ARzS and APS on vehicle operation characteristics is not so high for the selected manoeuvre. All vehicle subsystems are activated. The matrix W_u is defined as:

$$W_u = \text{diag} \left(\left[0.7^{1 \times 2}, 1.6^{1 \times 2}, 0.875^{1 \times 4}, 0.875^{1 \times 4}, 0.35^{1 \times 4}, r_1^{1 \times 4}, r_3^{1 \times 4}, r_2^{1 \times 4} \right] \right) \quad (6.11)$$

The results of subsystem prioritization are shown in Figure 6.10. As can be noted from the simulation results, the influence of considered subsystems on vehicle operation characteristics is lower compared to previous stages. Mean absolute error of yaw rate is in the range from 0.55 to 0.8 deg/s (Figure 6.10a), maximum of sideslip angle – from 3.6 to 4.2 deg (Figure 6.10b), longitudinal velocity at the end of the manoeuvre – from 66.1 to 66.9 km/h (Figure 6.10c), total consumed energy by electric motors – from 0 to 4 kJ (Figure 6.10d) and total tire energy dissipation– from 37.5 to 39 kJ (Figure 6.10e).

The best tracking performance of yaw rate and minimal sideslip angle can be obtained for the range of APS (0-0.6) and ACS (0.3-1.2). Close to this range, the lowest energy consumption can be achieved as well. The higher restriction of APS (2.1-2.7) allows to obtain maximal longitudinal velocity.

Based on the obtained results, the range of restriction weight variation can be selected in the interval between 0.5 and 2.

6.4.2 Subsystem prioritization during ‘Double Lane Change’ manoeuvre

The similar sequence of subsystem activation is used to analyse the subsystem prioritization during modified ‘Double Lane Change’ manoeuvre. The assessment criteria are the same as used in ‘Sine with Dwell’ manoeuvre.

Simulation results of prioritization between FBS, EBS and EP with other deactivated subsystems are shown in Figure 6.11. Mean absolute error of yaw rate belongs to the range from 1.1 to 1.2 deg/s (Figure 6.11a), maximum of sideslip angle – from 3.0 to 3.5 deg (Figure 6.11b), longitudinal velocity at the end of the manoeuvre – from 92 to 102 km/h (Figure 6.11c), total consumed energy by electric motors – from -40 to 80 kJ (Figure 6.11d) and total tire energy dissipation– from 42 to 50 kJ (Figure 6.11e). The best tracking performance of yaw rate and minimal sideslip angle can be reached, when FBS is in the interval of 0.3-2.1 and EBS is 0-0.3. Using the complex criterion combined vehicle handling, stability, energy consumption and energy losses (Figure 6.11f), the restriction weights for the next stage are selected as: 1.9 for EP, 0.4 for EBS and 0.7 for FBS (marked as red point). Compared to ‘Sine with Dwell’ manoeuvre, EBS will have higher priority.

Prioritization between previous subsystems gr#1, IFS and IRS are shown in Figure 6.11. Mean absolute error of yaw rate has been varied from 1.1 to 2 deg/s (Figure 6.12a), maximum of sideslip angle – from 3 to 5 deg (Figure 6.12b), longitudinal velocity at the end of the manoeuvre – from 92 to 97 km/h (Figure 6.12c), total consumed energy by electric motors – from -40 to 40 kJ (Figure 6.12d) and total tire energy dissipation– from 45 to 55 kJ (Figure 6.12e). The restriction weights for the last stage are selected as: 0.6 for gr#1, 1.8 for IFS and 0.6 for IRS (marked as red point). Compared to ‘Sine with Dwell’ manoeuvre, IFS and IRS have opposite the restriction weights.

ACS, ARzS and APS prioritization is shown in Figure 6.13. The complex criterion demonstrates that the extremum is close to the area, when all subsystems have the same restriction weights. Mean absolute error of yaw rate is in the range from 1.1 to 2 deg/s (Figure 6.13a), maximum of sideslip angle – from 3.3 to 4.0 deg (Figure 6.13b), longitudinal velocity at the end of the manoeuvre – from 95 to 98 km/h (Figure 6.13c), total consumed energy by electric motors – from -35 to -40 kJ (Figure 6.13d) and total tire energy dissipation– from 39.4 to 50 kJ (Figure 6.13e).

Figure 6.11 Prioritization of FBS, EBS and EP for 'Double Lane Change' manoeuvre:

- (a) mean absolute error of yaw rate; (b) maximum of absolute sideslip angle during manoeuvre; (c) longitudinal velocity at the end of the manoeuvre; (d) energy consumption by electric motors; (e) total tire energy dissipation; (f) complex criterion

Figure 6.12 Prioritization of gr#1, IFS and IRS for ‘Double Lane Change’ manoeuvre:

- (a) mean absolute error of yaw rate; (b) maximum of absolute sideslip angle during manoeuvre; (c) longitudinal velocity at the end of the manoeuvre; (d) energy consumption by electric motors; (e) total tire energy dissipation; (f) complex criterion

Figure 6.13 Prioritization of ACS, ARzS and APS for 'Double Lane Change' manoeuvre:

- (a) mean absolute error of yaw rate; (b) maximum of absolute sideslip angle during manoeuvre; (c) longitudinal velocity at the end of the manoeuvre; (d) energy consumption by electric motors; (e) total tire energy dissipation; (f) complex criterion

6.4.3 Subsystem prioritization during straight-line braking

Prioritization between FBS, EBS and APS is considered. Other subsystems are deactivated and does not considered for straight-line braking. The vehicle brakes from initial velocity of 95 km/h on the road with high friction coefficient of 0.8. The weights change in the same interval as for above-considered manoeuvres. The matrix W_u is defined as:

$$W_u = \text{diag} \left(\left[1^{1 \times 2}, 1^{1 \times 2}, 1^{1 \times 4}, r_2^{1 \times 4}, r_1^{1 \times 4}, 1^{1 \times 4}, r_3^{1 \times 4}, 1^{1 \times 4} \right] \right) \quad (6.12)$$

The simulation results of the subsystem prioritization are shown in Figure 6.14. Braking distance is changed from 66 m to 71 m (Figure 6.14a), mean error of longitudinal acceleration – from 0.1 to 0.5 m/s² (Figure 6.14b), total regenerated energy – from -20 to -120 kJ (Figure 6.14c) and total tire energy dissipation – from 14 to 18 kJ (Figure 6.14d). The minimal brake distance is obtained, when APS is restricted (2.4-3).

Figure 6.14 Prioritization of FBS, EBS and APS for straight-line braking

6.5 Algorithm of coordination of priorities between vehicle subsystems

The aim of proposed algorithms is to define the restriction weights w_{ui} into the matrix W_u for the prioritization of vehicle subsystems. The restriction weights should be changed according to performance criteria characterizing energy consumption and energy losses.

The advantages of the proposed approach are: 1) characteristic of auxiliary term like efficiency map of electric motor or characteristic of tire power dissipation are unnecessary, because they are already included in the analysis of system behaviour; 2) since the dimension of cost function remains invariant as in the case of ordinary control allocation, the computational budget is not increased.

However, the drawbacks of the proposed solution are: 1) pre-designed definition of restriction weights based on offline-simulations; 2) the solution will be not indeed optimal compared to multi-term cost function and it will strongly depend on selected performance criteria and their relationships with restriction weights.

6.5.1 Performance criteria

The criteria allowing real-time assessment of energy consumption and energy losses should be primarily formulated. To evaluate tire energy dissipation from longitudinal forces, the dimensionless criterion η_{lon} , called longitudinal efficiency, is introduced. It demonstrates the net longitudinal power, which can be potentially realized into the tire-road contact without longitudinal slip losses. In Figure 6.15, the relation between longitudinal energy dissipation rate and the selected criterion is demonstrated by the example of ‘Sine with Dwell’ manoeuvre.

$$\eta_{lon} = 1 - \frac{\sum_{i=1}^4 |\hat{F}_{xwi} \hat{V}_{swi}|}{\sum_{i=1}^4 |\hat{F}_{xwi} \hat{V}_{xwi}|} \quad (6.13)$$

Figure 6.15 Total energy dissipation rate and selected criterion in longitudinal direction

To evaluate tire energy dissipation from lateral forces, the dimensionless criterion η_{lat} , called lateral efficiency, is proposed. It represents the net lateral power, which is limited by maximal lateral forces of linear vehicle model. The lateral slip angle α_w^{net} , corresponding to maximum pure cornering force of linear regime of vehicle motion, is pre-defined and it is equal to 8° in accordance with analysis of reference data [137]. The lateral energy dissipation rate and lateral efficiency are shown in Figure 6.16 by the example of ‘Sine with Dwell’ manoeuvre.

$$\eta_{lat} = 1 - \frac{\sum_{i=1}^4 |\hat{F}_{ywi} \hat{V}_{ywi}|}{0.5 \left(C_f \sum_{i=1}^2 |\hat{V}_{xwi}| + C_r \sum_{i=3}^4 |\hat{V}_{xwi}| \right) \alpha_w^{net} \tan \alpha_w^{net}} \quad (6.14)$$

Figure 6.16 Total energy dissipation rate and selected criterion in lateral direction

As it was demonstrated in the Section 6.2.3, energy consumed by electric powertrain is much higher as total energy demand of actuators of the remaining subsystems. Since the manoeuvres of the short duration are further considered, energy consumption of other

actuators is neglected. The dimensionless criterion η_{con} related to power consumption of electric powertrain can be calculated as:

$$\eta_{con} = 1 - \frac{\sum_{i=1}^4 \max(0, P_{em,i})}{4P_{e\max}} \quad (6.15)$$

The instantaneous power consumption and selected criterion are shown in Figure 6.17 by the example of ‘Sine with Dwell’ manoeuvre.

Figure 6.17 Instantaneous power consumption and consumption criterion

The dimensionless criterion η_{rec} related to power recuperation of electric powertrain can be calculated as:

$$\eta_{rec} = 1 - \frac{\sum_{i=1}^4 |\min(0, P_{em,i})|}{4P_{e\max}} \quad (6.16)$$

Figure 6.18 Instantaneous power consumption and recuperation criterion

The instantaneous power consumption and criterion related to power recuperation are shown in Figure 6.18 by the example of ‘Sine with Dwell’ manoeuvre.

6.5.2 Test manoeuvre and procedure of weight definition

To define the relationships between selected performance criteria and restriction weight for each vehicle subsystem, the following procedure is applied (Figure 6.19):

Figure 6.19 Procedure to find the relationships between performance criteria and restriction weight

1) The manoeuvre characterizing the performance of vehicle dynamics, especially, its transient behaviour, should be selected. Based of ISO standard recommendations, an extreme ‘Step Steer’ manoeuvre with initial velocity of 100 km/h is carried out. Such kind of manoeuvre characterizes transient lateral response of vehicle, and, therefore, it is specified to define the relationships between selected performance criteria and restriction weight. The value of steer amplitude of 70 deg is selected to keep vehicle motion in the nonlinear region near friction limits.

2) The simulation procedure is carried out for stand-alone operation of each vehicle subsystem. The Figure 6.20 shows the corresponding results of stand-alone operation of each vehicle subsystem when its restriction weight w_{ui} is equal to 1.

Figure 6.20 Mean values of longitudinal and lateral efficiency, consumption and recuperation efficiency versus restriction weight

3) Varying a value of restriction weight for each vehicle subsystem, the simulation procedure is repeated. The Figure 6.20 represents the relationships between mean values of performance criteria and restriction weight for each subsystem. As it can be seen, some subsystems have insignificant influence on the performance criteria. As an example, mean value of longitudinal efficiency depends on longitudinal forces and longitudinal slip velocity, therefore, only electric powertrain and friction brake system have influence on it. In this case, their priority is dependent from the longitudinal efficiency, while the priority of other subsystems will be fixed.

Using the relationships between mean values of performance criteria and restriction weight from the simulation results of the series of extreme 'Step Steer' manoeuvres, the algorithms of dynamic change of restriction weights can be proposed. These algorithms are introduced in next sections.

6.5.3 Fuzzy-based subsystem prioritization

The first approach is based on soft-computing technique, which uses control laws without exact mathematical formulation. As mentioned in literature review from Chapter 2, fuzzy logic is one of the typical methods, which are used for control demand distribution.

Therefore, one of the variants of subsystem prioritization is proposed based on fuzzy logic. The membership functions for input parameters – longitudinal and lateral efficiency, consumption and recuperation efficiency – are formalized (Figure 6.21, a-d). The membership functions of restriction weight for each vehicle subsystem are the identical and shown in Figure 6.21e.

Figure 6.21 Membership functions for: a) longitudinal efficiency, b) lateral efficiency, c) consumption efficiency, d) recuperation efficiency, e) restriction weight

Table 6.3 Fuzzy rules

η_{lon}	η_{lat}	η_{con}	η_{rec}	w_{ifs}	w_{irs}	w_{ep}	w_{ebs}	w_{fbs}	w_{acs}	w_{aps}	w_{aRzs}
LW	LW	-	-	HI	HI	HI	LW	LW	HI	LW	LW
	ME			ME	ME	HI	LW	LW	ME	ME	ME
	HI			LW	LW	HI	LW	LW	LW	HI	HI
ME	LW			HI	HI	ME	ME	ME	HI	LW	LW
	ME			ME	ME	ME	ME	ME	ME	ME	ME
	HI			LW	LW	ME	ME	ME	LW	HI	HI
HI	LW			HI	HI	LW	HI	HI	HI	LW	LW
	ME			ME	ME	LW	HI	HI	ME	ME	ME
	HI			LW	LW	LW	HI	HI	LW	HI	HI

The fuzzy rules from Table 6.3 cover the most significant relationships between input and output parameters. They are defined using heuristic analysis of behaviour of performance criteria under different restriction weights. The examples of rule surfaces based on Figure 6.21 and Table 6.3 for individual front wheel steering, traction torques generated by electric motors and friction brake system are shown on Figure 6.22. Criteria η_{con} and η_{rec} are relevant only to the electric powertrain. Therefore, corresponding fuzzy rules for η_{con} and η_{rec} are proportionally related to w_{ep} and w_{ebs} .

Figure 6.22 Examples of rule surfaces

6.5.4 Iterative subsystem prioritization

The second approach is based on an iterative procedure of subsystem prioritization. This method can be explained as follows. The number of considered vehicle subsystems is higher than the number of performance criteria. Therefore, the system is underdetermined. Moreover, an exact formulation of the relationships between performance criteria and restriction weights during vehicle motion is the complex task to be mathematically described. Therefore, mean values of performance criteria can be used as qualitative indicators of these relationships. It has been proved with the simulation results of extreme ‘Step Steer’ manoeuvre (Figure 6.20). Next paragraphs introduce how to detect the links between performance criteria and the restriction weights by the example of lateral efficiency. It is further assumed that:

- current value of performance criterion is linked to mean value and restriction weight;
- current value of performance criterion can be represented as a linear product between ratio $\chi(\bar{\eta}_i)$ and restriction weight.

In this case, the relationship between performance criterion and restriction weight is:

$$\eta_{lat} = f(\bar{\eta}_{lat}, w_i) = \chi(\bar{\eta}_{lat}) w_i \quad (6.17)$$

As an example, the dependence of ratio $\chi(\bar{\eta}_{lat})$ from restriction weight at different mean values of lateral efficiency is considered for front wheel steer actuators (IFS). The simulation results are shown in Figure 6.23. This data is obtained from modified ‘Step Steer’ manoeuvre with different steering amplitudes of 70, 80 and 90 deg. The highest mean value of lateral efficiency refers to the lowest steering amplitude.

Figure 6.23 Ratio $\chi(\bar{\eta}_{lat})$ versus restriction weight at different mean values of lateral performance for IFS

The control effectiveness matrix B_{wui} combining the ratios $\chi(\bar{\eta}_{lon})$, $\chi(\bar{\eta}_{lat})$, $\chi(\bar{\eta}_{con})$ and $\chi(\bar{\eta}_{rec})$ is represented as rescaled polynomials obtained from Figure 6.24. The shift of each characteristic depends on current value of performance criteria. The variation of longitudinal efficiency ratio $\chi(\bar{\eta}_{lon})$ is scaled taking into account the range of criterion changing from 0.9 to 1.

Figure 6.24 Ratio $\chi(\bar{\eta}_{lon})$, $\chi(\bar{\eta}_{lat})$, $\chi(\bar{\eta}_{con})$ and $\chi(\bar{\eta}_{rec})$ versus restriction weight for each vehicle subsystem (modified ‘Step Steer’ manoeuvre)

To find restriction weight for each vehicle subsystem, the allocation problem is formulated as:

$$w_u = \arg \min_{0.5 \leq w_u \leq 2} \left(\|W_{wv}(B_{wui} w_u - v_w)\|_2^2 \right) \quad (6.18)$$

The input vector v_w includes performance criteria taking into account a number of vehicle subsystems N_{subs} :

$$v_w = [N_{subs} \eta_{lon} \quad N_{subs} \eta_{lat} \quad \eta_{con} \quad \eta_{rec}]^T \quad (6.19)$$

To solve allocation problem, the combined solver is applied:

1) on the first step, the restriction weight are calculated by pseudo-inverse matrix method:

$$w_u = B_{wu}^T (B_{wu} B_{wu}^T)^{-1} v_w \quad (6.20)$$

2) then the obtained value w_u is used as the initial input to the fixed-point iteration method with termination conditions:

$$w_u(k+1) = sat \left(\left(I_{4 \times 4} - \frac{B_{wui}^T W_{wv}^T W_{wv} B_{wui}}{\|B_{wui}^T W_{wv}^T W_{wv} B_{wui}\|_2} \right) w_u(k) + \frac{B_{wui}^T W_{wv}^T W_{wv}}{\|B_{wui}^T W_{wv}^T W_{wv} B_{wui}\|_2} v_w \right) \quad (6.21)$$

From the observations, the combined solver finds the suitable solution of abovementioned allocation problem in the framework of one iteration. Therefore, the total computational load is not increased.

The range of weight variation is saturated in the interval from 0.5 up to 2 according to Section 6.4. To eliminate an effect of noisy behaviour or sharp change of the subsystem priority, the rate of weight changing can be introduced into the algorithm.

In addition, the performance criteria can have different priorities during vehicle motion. To investigate this issue, the matrix W_{wv} is used to define a priority of each performance criterion. The five variants of matrix W_{wv} are considered and listed in Table 6.4. The elements of matrix W_{wv} related to performance criteria η_{lon} and η_{lat} are changed proportionally in accordance to the yaw rate $d\psi/dt$ in the range between 2.5 deg/s and 15 deg/s.

Table 6.4 Strategies of prioritization of performance criteria

	$W_{wv}(\eta_{lon})$			$W_{wv}(\eta_{lat})$		
$d\psi/dt, \text{deg/s}$	<2.5	[2.5; 15]	>15	<2.5	[2.5; 15]	>15
str#1	1	1	1	1	1	1
str#2	5	$-0.36\dot{\psi} + 5.9$	0.5	0.5	$0.36\dot{\psi} - 0.4$	5
str#3	0.5	$0.36\dot{\psi} - 0.4$	5	5	$-0.36\dot{\psi} + 5.9$	0.5
str#4	0.5	$0.36\dot{\psi} - 0.4$	5	0.5	$0.36\dot{\psi} - 0.4$	5
str#5	5	$-0.36\dot{\psi} + 5.9$	0.5	5	$-0.36\dot{\psi} + 5.9$	0.5

6.6 Chapter summary

To analyse the advantages of multiple actuators, the stand-alone operation of several vehicle subsystems was investigated under ‘Slowly Increasing Steer Test’ manoeuvre. It was demonstrated that all considered subsystems have visible influence on the steering characteristic of the vehicle in the nonlinear area. However, the application of dynamic tire pressure system causes sharp changing of steering wheel angle gradient in the area of friction limits. This phenomenon has an unfavourable influence not only on the vehicle handling but also on steering feel. The simulation results from ‘Sine with Dwell’ and ‘Double Lane Change’ manoeuvres allow to argue that joint operation of actuators can provide a compromise solution between motion tracking performance, energy consumption and energy losses. To investigate how joint operation of several subsystems influences the motion performance and energy characteristics during transient motion, ‘Sine with Dwell’ and ‘Double Lane Change’ manoeuvres were repeated to demonstrate that properly distributed operation of all considered subsystems can guarantee a reasonable compromise in simultaneous maintenance of criteria of motion performance, energy consumption and energy losses. The energy consumption of all actuators were analysed and it demonstrates that compared to electric powertrain, the consumption of other actuators can be neglected. The advantages from customers’ view for multi actuators allows to recommend various subsystem configurations for three classes of passenger cars (city, executive and premium-class cars).

The research idea of the thesis is to prioritize of each vehicle subsystem in accordance with specific performance criteria related to energy characteristics. For this purpose, the weight coefficients, called restriction weights, in the control allocation procedure are used to prioritize vehicle subsystems. The higher value of restriction weight means higher restriction or lower priority of vehicle subsystem. The joint operation of vehicle subsystems with restriction weights was investigated in transient lateral manoeuvres and straight-line braking. The analysis covers feasible ranges of restriction weights and demonstrates their influence on vehicle operation characteristics.

The restriction weights are changed according to the performance criteria, such as longitudinal and lateral efficiency, energy consumption and recuperation. Instead of auxiliary cost terms (e.g. abovementioned performance criteria) in the cost function of control allocation, the multi-objective problem is solved by the prioritization of vehicle subsystems during the formation of control demand. Such kind of approach does not increase computational load; however, it requires pre-designed definition of restriction

weights and their influence on performance criteria using offline-simulations. The first algorithm is based on fuzzy logic and the second algorithm is an iterative approach with different variants of performance criteria prioritization; which are investigated using a HIL test rig and reported in the Chapter 7.

Chapter 7

7 Experimental evaluation of proposed control system

A brief overview and description of hardware-in-the-loop test rig is considered in this Chapter. Using HIL test rig, the proposed algorithms of the subsystem prioritization are investigated under vehicle manoeuvres covering longitudinal and lateral motion.

7.1 Description of experimental test rig

7.1.1 Brief description of state-of-the-art of hardware-in-the-loop applications for the investigations of integrated vehicle control

Most of known automotive hardware-in-the-loop applications relate to the cases of development and testing of single, stand-alone subsystems. In particular, solutions are known for advanced driver assistance systems [169], active suspension [170], active anti-roll bars [114], brake control applications [171], steering control [172] and others.

The usage of stand-alone HIL solutions constrains crucially the development of control technique for multi-actuated vehicles. The integrated or combined HIL test rigs can provide more adequately account real operation conditions as compared to traditional software-based modelling. However, the integration of several hardware-in-the-loop test rigs or their combination with other experimental tools is rarely introduced in the literature. The most relevant applications are single test platforms for testing of two and more subsystems, for example, steering and brakes [173], or for testing of vehicle dynamics controllers [174]. Some solutions are also known where two or more physically spaced HIL tests are linked by means of wireless communications. For example, the fusion of the powertrain HIL setup and the vehicle HIL setup by means of Internet-based data exchange is described in [175]. Another communication example is the coupling of HIL devices with dynamometer test rigs like combined performance technique introduced in [176]. Analysis of relevant approaches allows to propose a topological classification of possible HIL solutions.

Three classification levels can be distinguished in accordance with Figure 7.1. The object level defines typical test targets as ECUs, hardware subsystems (brakes, steering

etc.), or a complex system like a vehicle. The second level identifies the integration of tested objects. The corresponding classes are non-integrated HIL test rigs for a stand-alone object and integrated HIL tools for simultaneous testing of several objects. These objects can have either dependent (for example, subsystems of the same vehicle) or independent (for instance, subsystems of different vehicles involved in a cooperative driving) character. The connection level classifies the kind of coupling of several objects tested on the integrated HIL platforms. In the case of the single platform, the common HIL processor board is used for all tested objects. In more complicated cases, the tested objects have individual HIL processor boards, which can be linked directly through a hardware interface or through remote, wireless connection.

Figure 7.1 Topological classification of HIL test platforms

Based on the classification analysis, requirements to the configuration are formulated:

- single HIL platform with distributed processor resources enabling parallel inclusion of more automotive control systems with full-scale hardware components;
- availability of wireless plug-in of remote sensors and actuators;
- implementation of several independent communication protocols like CAN;
- inclusion of software extensions for virtual vehicle simulators;
- flexible interface for interaction of the HIL platform with other physical test setups like dynamometer, performance, or NVH test rigs.

Additional analysis has been performed to define the functional requirements to the HIL test rig under development. A number of related qualitative and quantitative

indicators on Table 7.1 are chosen on the basis of works [177-182] to characterize the HIL platform.

Table 7.1 Functional indicators of HIL platform

Main quantitative indicators	Main qualitative indicators
software-to-hardware ratio	fidelity
HIL transparency and robustness	level of multiformalism and multisolver
multirate index of system integration	level of modelling
integrated HIL accuracy	hardware shareability

7.1.2 Concept and architecture of the test rig

The test rig consists of real and virtual components, which are intercommunicated using hardware-in-the-loop technique. Hardware subsystems considered in this chapter are conventional hydraulic brake system and dynamic tire pressure system. Other vehicle subsystems, multibody vehicle model, and the proposed multi-layer control system have software realization in MATLAB/Simulink and IPG CarMaker. The architecture of test rig is shown in Figure 7.2.

Figure 7.2 Architecture of HIL test rig

The dSPACE hardware components for digital and analogue data input/output as well as for CAN bus protocol communication compose a framework of test rig architecture. The main board (ds1006) is built up on a quad core processor for

multiprocessing running at 2.8 GHz. It is capable of distributing computing procedures between four core processors that guarantee high-performance real-time simulation. Analogue input signals information from the sensors of brake pressure sensors are digitalized by multi-channel A/D board DS2002. The interaction between dynamic tire pressure system and HIL test rig is realized through controller area network (CAN) environment. For this communication, the CAN interface board DS4302 is used. The control on the valves of hydraulic control unit in friction brake system and dynamic tire pressure system is realized with the help of the HIL digital input/output board DS4004.

The specification of hardware subsystems, integrated in the test rig are outlined below.

Friction brake system

Friction brake system of the HIL test rig is conventional brake system, which consists of the master cylinder, four brake assemblies and hydraulic control unit (Figure 7.3). On the vehicle force applied by the driver on pedal is gained by the vacuum booster and transferred to the master cylinder. The primary and secondary sections of the master cylinder are hydraulically coupled with HCU. The HCU includes eight 2/2-way switching valves, two high-pressure and two switchover valves, a hydraulic pump with an electric motor and a low-pressure accumulator. A set of pressure sensors allows to measure brake pressure in each circuit as well as in each brake calliper.

Figure 7.3 Conventional friction brake system

The measurement range of pressure sensors is 0...200 bar with cut-off frequency of 1 kHz. The original electronics of HCU is removed. On the represented hardware-in-the-loop platform the HCU is controlled by analogue I/O interface via dSPACE and an amplifier board. Sending the binary signals a virtual controller actuates the solenoid valves in the HCU in terms to establish the required pressure level by feedback from the

pressure sensors. The investigated configuration of friction brake system has a feature compared to electro-hydraulic brake system investigated in the Chapter 6. The valves in hydraulic control unit are relay type instead of the proportional type found in electro-hydraulic brake system.

Dynamic tire pressure system

The inclusion of dynamic tire pressure system into the proposed control system gives an opportunity to change tire forces and, as a consequence, leads to additional potential benefits in vehicle stability and performance. The system allows taking into account the real loads on four wheels and constructive features of the system like pipeline lengths. A pneumatic part of the dynamic tire pressure system consists of compressor, air storage device, inlet and outlet valves, and rotary feedthrough. The required tire pressure level is achieved by actuation of the inlet/outlet valves provided for each wheel individually. Each valve is controlled by analogue I/O interface via dSPACE. The feedback signals are received via wireless communication from the sensors, which are mounted inside each tire. They allow to measure tire pressure in the range of 1...5 bar and temperature in the range of -40...+120 C with a sampling rate of 100Hz. The sensor plate is placed on tire rim and includes a radio transmitter for wireless communication via CAN.

The technical data of the dynamic tire pressure system are as follows [167]:

- control each individual tire pressure of the test vehicle, taking into account static load on each wheel;
- dynamic rate of tire inflation and deflation is around 1 and 0.6 bar/s respectively and achieved by relay valve control.

The principal scheme of dynamic tire pressure system and its elements are shown in Figure 7.4.

Emulated vehicle subsystems and vehicle model

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7.2 Results for “Sine with Dwell” manoeuvre

‘Sine with Dwell’ manoeuvre as in Chapter 6 is carried out at initial velocity around 80 km/h and max. steering amplitude of 120 deg on dry asphalt.

Figure 7.5 Motion characteristics during ‘Sine with Dwell’ manoeuvre

The experimental results of yaw rate (Figure 7.5a) and sideslip angle (Figure 7.5b) demonstrate that all variants guarantee stability of vehicle motion. Compared to the motion without the subsystem prioritization, the mean error of yaw rate is 6.5% higher for ‘fuzzy’ variant and 0.5% lower for ‘str#4’, when weights of longitudinal and lateral performance criteria increase proportionally to yaw rate error. The maximum of sideslip angle is 1.3% higher for ‘fuzzy’, and 0.9% lower for ‘str#5’ compared to ‘stnd’. The longitudinal velocity at the end of manoeuvre for all considered variants is relatively same. In Figure 7.5c the maximal longitudinal velocity of 68.3 km/h is reached for ‘str#5’, which is 1.1% higher than ‘stnd’. It should be noted, all variants except ‘fuzzy’ provide higher velocity at the end of manoeuvre compared to the motion without the subsystem prioritization. The energy characteristics of vehicle motion, such as energy

consumption of electric motor and tire energy dissipation, are shown in Figure 7.6. Negative value means potential energy, which can be recuperated. For all variants with the subsystem prioritization the value of potential recuperated energy is roughly 2.5 times higher compared 'stnd'. Meantime, the lower tire energy dissipation can be reached. For instance, 'str#4' provides 1.8% lower tire energy dissipation than 'stnd'.

Figure 7.6 Energy characteristics during 'Sine with Dwell' manoeuvre

Percentage of average level of subsystem actuation in comparison with the motion without the subsystem prioritization is shown in Figure 7.7. The positive value means increasing of actuation level, whereas negative one represents decreasing. The application of friction brake system (FBS) is reduced and simultaneously actuation of electric motors (EBS) and wheel steer actuators (IWS) is increased. The subsystem actuation for 'str#4' variant is shown in Figure 7.8.

Figure 7.7 Percentage level of subsystem actuation during 'Sine with Dwell' manoeuvre

Figure 7.8 Subsystem actuations during 'Sine with Dwell' manoeuvre

7.3 Results for “Double Lane Change” manoeuvre

‘Double Lane Change’ manoeuvre is started at initial velocity 125 km/h on dry asphalt. The driver (IPG model) steers the vehicle according to course layout and no traction demand is permitted during manoeuvre. The motion characteristics, such as yaw rate, vehicle sideslip angle and longitudinal velocity, are shown in Figure 7.9.

Figure 7.9 Motion characteristics during ‘Double Lane Change’ manoeuvre

The best reference tracking performance for yaw rate is reached by ‘str#4’. Meantime, the lowest maximum of sideslip angle is provided by the variant without prioritization. The longitudinal velocity at the end of the manoeuvre is highest for ‘str#1’. The simulation results for some variants (‘str#2’ and ‘str#5’) are qualitatively different from the results of ‘Sine with Dwell’ manoeuvre. One of the reasons is a sensitiveness to the criterion of longitudinal performance. Its weight is proportionally reduced according to the yaw rate error for these two variants.

The energy characteristics of vehicle motion are shown in Figure 7.10. All variants with the subsystem prioritization demonstrate lower energy consumption up to 3 times compared to 'stdn'; however, the increasing of tire energy dissipation for several of them is observed.

Figure 7.10 Energy characteristics during 'Double Lane Change' manoeuvre

Percentage of average level of subsystem actuation with respect the variant without subsystem prioritization is shown in Figure 7.11. As in 'Sine with Dwell' manoeuvre, the participation of friction brake system is reduced, whereas applied brake torque generated by electric motor, steer and camber angles are increased. In addition, the average values of applied traction torque generated by electric motor and additional normal forces are reduced. The subsystem actuation for 'str#4' variant is shown in Figure 7.12.

Figure 7.11 Percentage level of subsystem actuation during 'Double Lane Change' manoeuvre

Figure 7.12 Subsystem actuations during 'Double Lane Change' manoeuvre

7.4 Results for straight-line braking

The vehicle brakes on the surface with high friction coefficient of 0.8 at initial velocity of 95 km/h. The driver releases brake pedal, when vehicle velocity is below 20 km/h. For the repeatability of braking manoeuvre, the pump and hydraulic control unit generate brake pressures. Since the 'str#2' – 'str#5' variants depend on yaw rate, they are omitted in the considering manoeuvre.

Figure 7.13 Motion and energy characteristics during straight-line braking manoeuvre

The simulation results are shown in Figure 7.13. The maximum of reference vehicle deceleration is around 7.5 m/s^2 . The considered variants show suitable tracking

performance. The minimal brake distance is provided by ‘fuzzy’ variant. The highest potential recuperated energy and the lowest tire energy dissipation is achieved by ‘str#1’ due to higher levels of brake torque generated by electric motors and tire pressure (Figure 7.14). Due to the higher level of tire pressure, brake distance is increased up to 1.2 m compared to the variant without subsystem prioritization. Thereby, the application of dynamic tire pressure system during emergency braking should be reduced. Otherwise, it can cause a rise in brake distance. The subsystem actuation of ‘str#1’ variant is shown in Figure 7.15.

Figure 7.14 Percentage level of subsystem actuation during straight-line braking manoeuvre

Figure 7.15 Subsystem actuations during straight-line braking manoeuvre

7.5 Results for brake-in-the-turn

The simulation scenario is comprised of vehicle motion at initial velocity at 95 km/h. The lateral acceleration is around 5.0 m/s^2 when the driver pushes brake pedal. The target longitudinal deceleration is equal to 5.5 m/s^2 . The simulation results are shown in Figure 7.16.

Figure 7.16 Motion characteristics during brake-in-the-turn manoeuvre

The mean errors of longitudinal acceleration and yaw rate are lower than 0.23 m/s^2 and 0.6 deg/s accordingly. The maximal sideslip angle is close to 1.82 deg . The variants with the subsystem prioritization demonstrate better tracking performance of yaw rate compared to one without the subsystem prioritization.

The energy characteristics are shown in Figure 7.17. The potential recuperated energy for each variant of the proposed prioritization is higher by up to 26% compared to the variant without the subsystem prioritization. Meanwhile, the lowest tire energy

dissipation is observed for the variant without the subsystem prioritization; however, this difference is smaller than 3% for the worst case ('str#1').

Figure 7.17 Energy characteristics during brake-in-the-turn manoeuvre

Percentage of average level of subsystem actuation in comparison with the motion without the subsystem prioritization is shown in Figure 7.18. The qualitative behaviour of subsystem actuation is similar to 'Sine with Dwell' and 'Double Lane Change' manoeuvres. The main distinctions are high usage of dynamic tire pressure system and low application of camber angle actuators. The subsystem actuation for 'str#4' variant is shown in Figure 7.19. During braking, the application of additional normal forces is minimized.

Figure 7.18 Percentage level of subsystem actuation during brake-in-the-turn manoeuvre

Figure 7.19 Subsystem actuations during brake-in-the-turn manoeuvre

7.6 Chapter summary

The presented Chapter has introduced a brief overview and a classification of test rigs based on hardware-in-the-loop technology. The test rig includes hardware components of conventional brake system with hydraulic control unit and dynamic tire pressure system. Other vehicle subsystems are realized as computational models in MATLAB/Simulink. The hardware and software components are connected to a multibody vehicle model and the virtual environment in the IPG CarMaker software using dSPACE. The above-mentioned experimental setup is used for the investigation of the proposed control system with variants of the subsystem prioritization methods from Section 6.5.

Standard vehicle manoeuvres covering longitudinal and lateral motion are considered. The following conclusions can be inferred:

- Proposed algorithms of the subsystem prioritization provide better energy consumption or higher potential recuperated energy for all manoeuvres. For braking manoeuvres, the potential recuperated energy is 15-25% higher in comparison with the variant without the subsystem prioritization, for lateral motion it is 2-3 times more accordingly.
- For ‘Sine with Dwell’ manoeuvre all variants of iterative subsystem prioritization reach higher longitudinal velocity at the end, with lower tire energy dissipation.
- For ‘Double Lane Change’ manoeuvre only the variant of iterative subsystem coordination with same priority weights is effective from the point of longitudinal velocity at the end of the manoeuvre and tire energy dissipation. Meanwhile, tracking performance is lower compared to the variant without the subsystem prioritization.
- During emergency braking the application of dynamic tire pressure system should be reduced, otherwise it can cause a rise in brake distance.
- During brake-in-the-turn manoeuvre all variants of the subsystem prioritization demonstrate better tracking performance of yaw rate compared to the variant without the subsystem prioritization.
- The achieved results are due to the reduction of the average level of friction brake torque and traction torques from electric motors, while average levels of other subsystem actuations, such as brake torque from electric motor, steer and camber angles and others, are increased.
- effectiveness of the developed control system with the proposed algorithms was demonstrated within simulation-based analysis for the various road conditions

and vehicle manoeuvres (Appendix A.4).

The future work will be oriented to investigate the following issues:

- The application of certain vehicle subsystems provides a low impact on vehicle stability for a specific emergency manoeuvre due to their position/rate limits. Therefore, an issue of their deactivation for lateral motion and activation for longitudinal motion will be considered.
- The tuning of weighing factors for varying friction conditions will be analysed to improve adaptability of the proposed method.
- The application of adaptive control methods allows to omit a pre-definition of weighing factors and finally it can significantly improve an effectiveness of the subsystem prioritization.

Chapter 8

8 Conclusion and recommendations

The conclusions and achievements of the thesis are summarized in this chapter. Possible areas for future research are discussed.

8.1 Conclusion

As the outset of this thesis, it was mentioned that the modern system of vehicle dynamics control should achieve a set of control objectives related to stability, handling, energy consumption and energy losses, where the instrument to reach this target is a multi-actuator vehicle configuration and its integrated control. This statement is the main leitmotif of the thesis.

Literature review demonstrated various control techniques and configurations of vehicle subsystems involved in the regulation process. It helps to detected a prevalent technique to distribute control demand for over-actuated system, which is static optimization-based control allocation; demonstrate additional objectives related to energy consumption and energy loss into the tire-road contact; formulate main research goal and objectives of the thesis. The main research goal of the thesis is the development and evaluation of a control system coordinating priorities between vehicle subsystems for a multi-actuated vehicle and the demonstration of the possibility to improve concurrently several vehicle operation characteristics during vehicle motion. The subsystem coordination under discussion covers (i) friction brake system, (ii) individual-wheel drive electric powertrain, (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. Dynamic tire pressure system was first considered for integrated vehicle control.

For the preliminary investigations, the original vehicle model was developed. The vehicle model has 14 DoF including 6 DoF of vehicle body in the space and 2 DoF of rotational and vertical movements of each wheel. The conventional steering system is represented by Ackerman kinematic model with dynamic effects related to roll and compliance steer. Camber angle actuators are defined by the FOPTD. The electric

powertrain includes battery, four electric motors with single-speed gearboxes and half-shafts. The model of an electric motor is represented by technical characteristics of a real electric motor. The transient behaviour of the electric motors is described by FOPTD transfer function. The dynamics of powertrain and electro-hydraulic brake system are described by a system of differential equations. Dynamic tire pressure system is presented by FOPTD transfer function based on step response characteristics obtained on the test rig. Actuators generating additional normal forces through external spring, damping and stabilizer forces are characterized by FOPTD transfer function. The verification of the developed model is carried out by transient test manoeuvre. The developed model demonstrates a similar behaviour compared to a multibody vehicle model.

Since friction brake system is the main vehicle subsystem in any vehicle dynamics control system to ensure vehicle safety and stability, the consideration of critical components influencing the braking performance is of importance. Hence, the special chapter of the thesis introduces a detailed analysis of the effect of hysteresis in the brake mechanism. The effect of hysteresis in the brake mechanism was first described by the Bouc-Wen model and verified by experimental investigations of disc brake mechanism for passenger car. Experimental investigations clearly show the nonlinear and asymmetric form of hysteresis. Using the computational model of the brake mechanism with hysteretic elements, the analysis of the influence of hysteresis on vehicle dynamics during the straight-line braking with ABS and stability control of curvilinear motion is provided. The analysis shows that the effect of hysteresis on stopping distance for new hydraulic disc brake mechanism does not exceed 5% in the most critical cases; however, ignoring this fact can cause incorrect control, for instance, a false demand to increase the control frequency of brake torque actuation. Hysteresis influence on stability control of curvilinear motion is insignificant due to the duration and frequency of brake torque actuations. The developed hysteresis model was integrated into the model of the friction brake system.

Next part of the dissertation describes the multi-layer control system that includes high-level controller for vehicle dynamics, middle-level control allocation, low-level individual controllers for each subsystem and vehicle state observer. The high-level controller includes the reference generator and vehicle dynamics controller. The reference longitudinal acceleration, reference sideslip angle, when vehicle moves outside stability boundaries, and reference yaw rate are generated into the reference model using steering angle and displacement of brake pedal from driver. Generalized control demands of longitudinal and lateral forces, and yaw torque are calculated using discrete second-order

sliding mode control. The middle layer is optimization-based control allocation, which distributes generalized control demands between vehicle subsystems. Optimization constraints are formulated taking into account static actuator dynamics (position and rate limits) and tire friction limits. The fixed-point method with termination conditions is used as an optimization solver. The low-level actuator control guarantees a precise tracking to reference control signals obtained from the middle layer. Moreover, it is used to estimate boundary conditions for control allocation taking into account actuator physical limits and wheel slip control. Longitudinal and lateral velocities are estimated in the vehicle state observer based on extended Kalman filter. The 7 DoF vehicle model and Pacejka model with relaxation lengths in longitudinal and lateral directions are used as a process model.

The stand-alone operation of several vehicle subsystems was investigated under ‘Slowly Increasing Steer Test’ and ‘Steep Steer’ manoeuvres. It was demonstrated that all considered subsystems have visible influence on the steering characteristic of the vehicle in the nonlinear area, allowing us to argue that joint operation of actuators can provide a compromise solution between motion tracking performance, energy consumption and energy losses. To investigate how joint operation of several subsystems influences on motion performance and energy consumption during transient behaviour, ‘Sine with Dwell’ manoeuvre was simulated. It was demonstrated that properly distributed operation of all considered subsystems can guarantee a reasonable compromise in simultaneous maintenance of criteria of motion performance, energy consumption and energy losses. In addition, a need for multi actuators from customers’ point of view was categorized for three classes of passenger cars. To prioritize each vehicle subsystem, weight coefficients, called restriction weights, are used in the control allocation procedure. The joint operation of vehicle subsystems with the restriction weights was investigated for transient lateral manoeuvres and straight-line braking. The analysis covers the feasible ranges of restriction weights and demonstrates their influence on vehicle operation characteristics. The algorithms of the coordination of priorities between vehicle subsystems using restriction weights into the control allocation were proposed. The restriction weights are changed according to performance criteria, such as longitudinal and lateral efficiency, power consumption and recuperation. Instead of the supplement of auxiliary cost terms related to energy consumption and energy losses into the cost function of control allocation, the multi-objective problem is solved by the prioritization of vehicle subsystems forming control demand. Such kind of approach does not increase computational load; however, it requires pre-designed definition of restriction weights and analysis of their influence on performance criteria using offline-simulations. The first

algorithm is based on fuzzy logic and the second one is iterative approach with various variants of performance criteria prioritization.

The developed control system was investigated using hardware-in-the-loop test rig with hardware components of conventional brake system and tire pressure system in real-time domain. Vehicle test manoeuvres covering longitudinal and lateral motion were considered. The proposed algorithms of the subsystem prioritization provided better energy consumption or higher potential recuperated energy for all manoeuvres. For braking manoeuvres the potential recuperated energy with proposed algorithms is increased up to 15-25%. For lateral motion it is 2-3 times higher in comparison with the variant without the subsystem prioritization. For ‘Sine with Dwell’ manoeuvre all variants of iterative subsystem prioritization reach higher longitudinal velocity at the end and lower tire energy dissipation. For modified ‘Double Lane Change’ manoeuvre only variant of iterative subsystem prioritization with same priority weights is effective from the point of longitudinal velocity at the end of the manoeuvre and tire energy dissipation. Meanwhile, tracking performance is lower compared to the variant without the subsystem prioritization. The explanation of achieved results is the reduction of the average level of friction brake torque and traction torques from electric powertrain, while average levels of other subsystem actuations, such as brake torque from electric powertrain, steer and camber angles and others, are increased.

8.2 Recommendations for further research

The following problematic issues can be considered in the future directions:

- global goal of future investigations regarding integrated control system is to control not only considered vehicle motions, but also roll, pitch and vertical dynamics together and allocate generalized demands between maximum of vehicle subsystems with the different varied priorities depending on current driving situation and driver behaviour;
- experimental investigations of control allocation methods based on a real multi-actuated vehicle are still open issues;
- detailed representation of actuator dynamics including time delays, hysteresis, backlashes and other factors in allocation procedure can be an interesting point for further investigation. The precise models of actuators can help to achieve more accurate distribution between vehicle subsystems;
- consideration of energy consumption of other actuators, besides electric motors, and an energy loss due to inertial and resistance forces during curvilinear motion

can improve total energy efficiency of vehicle motion and provides more accurate assessment of the current level of energy consumption for entire vehicle;

- influence of control allocation methods on driver feeling and comfort is a key point on the way to the commercial application of such kind of control systems;
- application of control allocation technique for control demand distribution in the case of multi-vehicle interaction. As an example, activation of ESP for one vehicle and active steering for another one in the case of emergency situation;
- based on the correlation between driver physiological condition (heart rate, blood glucose level, electrocardiogram, etc) and vehicle behaviour, a novel vehicle subsystem coordination can be proposed to reduce physiological workload. Such kind of control system can be the first step towards the concept of “e-health safe vehicle”.

Authors' Publications

Journal Publications

1. **B. Shyrokau**, D. Wang, D. Savitski, K. Hoepping and Ivanov V., Vehicle Motion Control with Subsystem Prioritization, *International Journal of Mechatronics*, 2014 (accepted for publication)
2. **B. Shyrokau** and D. Wang, Coordination of Steer Angles, Tyre Inflation Pressure, Brake and Drive Torques for Vehicle Dynamics Control, *SAE International Journal of Passenger Cars – Mechanical Systems*, 6 (1), 2013, doi:10.4271/2013-01-0712
3. **B. Shyrokau**, D. Wang, D. Savitski and Ivanov V., Vehicle Dynamics Control with Energy Recuperation based on Control Allocation for Independent Wheel Motors and Brake System, *International Journal of Powertrains*, Vol. 2, No. 2/3, pp. 153-181, 2013
4. **B. Shyrokau**, D. Wang, K. Augsburg and V. Ivanov, Vehicle Dynamics with Brake Hysteresis, *Proc. IMechE, Part D: J. Automobile Engineering*, Vol. 227, No. 2, pp. 139-150, 2013

Conference papers

5. M. Polesel, **B. Shyrokau**, M. Tanelli, D. Savitski, V. Ivanov, A. Ferrara, Hierarchical control of overactuated vehicles via sliding mode techniques, 2014 IEEE 53rd Annual Conference on Decision and Control, 15-17 December 2014, Los Angeles, USA
6. **B. Shyrokau**, D. Savitski, D. Wang, Influence of Active Subsystems on Vehicle Behaviour and Energy Characteristics, SAE 2014 World Congress & Exhibition, Detroit, MI, USA, 2014-01-0876
7. L. Heidrich, **B. Shyrokau**, D. Savitski, K. Augsburg, V. Ivanov, D. Wang, Hardware-in-the-loop test rig for integrated vehicle control systems, 7th IFAC Symposium on Advances in Automotive Control, Tokyo, Japan, 4-7 September, 2013
8. **B. Shyrokau**, D. Wang, M. Lienkamp, Integrated Vehicle Dynamics Control Based On Control Allocation With Subsystem Coordination, 23rd International Symposium on Dynamics of Vehicles on Roads and Tracks, Qingdao, China, 19-23 August, 2013
9. **B. Shyrokau**, D. Savitski, D. Wang, V. Ivanov, K. Augsburg, Analysis of Coordination and Blending Strategy between Frictional Brake System and Electric Motors, EuroBrake2013, Dresden, Germany, 17-19 June, 2013
10. **B. Shyrokau**, D. Wang, L. Heidrich and K. Hoepping, Analysis of subsystems coordination for electric vehicle during straight-line braking and brake-in-turn, *Proc. of*

the IEEE Symposium on Computational Intelligence for Engineering Solutions, Singapore, 16-19 April, 2013

11. **B. Shyrokau** and D. Wang, Control allocation with dynamic weight scheduling for two-task integrated vehicle control, Proc. of the 11th International Symposium on Advanced Vehicle Control, Seoul, South Korea, 9-12 September, 2012

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Posters

14. **B. Shyrokau**, D. Wang, Subsystem coordination based on control allocation for integrated vehicle dynamics control, Scientific Workshop, Munich, Germany, 7-8 March, 2013

15. **B. Shyrokau**, D. Wang, Coordinated control of multi-actuated electric vehicle, Scientific Workshop, Singapore, 9-10 September, 2013

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Appendix A.1

```

%% Common Constants
density=850;
Cd_flow=0.7;
%% Vehicle Parameters
m_s = 1360;
m_uf=87;
m_ur=87;
ma = m_s+m_uf+m_ur;
a_s = 1.43;
b_s = 1.21;
L=a_s+b_s;
h_f=0.13;
h_r=0.11;
h_cgs = 0.58;
h_cguf=0.30;
h_cgur=0.30;
t_f=1.571;
t_r=1.569;
tw=0.5*(t_f+t_r);
l_cgs= (m_ur*b_s - m_uf*a_s)/ma;
a = a_s + l_cgs;
b = b_s - l_cgs;
h_o= h_f + a*(h_r - h_f)/L;
h_s= h_cgs - h_o;
h_uf = h_cguf - h_o;
h_ur = h_cgur - h_o;
h_cg = (m_s*h_cgs + m_uf*h_cguf + m_ur*h_cgur)/ma;
I_xxs = 480;
I_yys = 1910;
I_zzs = 2050;
I_xys =0.0;
I_xzs =0.0;
I_yzs =0.0;
I_zzuf = m_uf*(t_f/2)^2;
I_zzur = m_ur*(t_r/2)^2;
I_xxso = I_xxs + m_s*h_s^2;
I_zz= I_zzs + m_s*l_cgs^2 + I_zzuf + m_uf*a^2 + I_zzur + m_ur*b^2;
I_xzso = I_xzs + m_s*h_s*l_cgs;
I_xz = I_xzs + m_s*h_s*l_cgs + m_uf*h_uf*a - m_ur*h_ur*b;
%% Suspension characteristics
Krf=900*180/pi;
Krr=900*180/pi;
Crf=90*180/pi;
Crr=90*180/pi;
Ksf=25e+3;
Csf_push=2.5e+3;
Csf_pull=5e+3;
Ksr=25*10^3;
Csr_push=3.0e+3;
Csr_pull=6e+3;
SWF=[0.959 0.959 0.797 0.797];
BSF=[1.3 1.3 1.2 1.2];
%% Road parameters
f0=0.01;
f1=0.00001;
f2=0.0000001;
%% Tire parameters
Ktire=320000;
Ctire=50;

% fluid density, kg/m^3
% flow coefficient, m^3/(s-Pa)

% sprung vehicle mass, kg
% front unsprung mass, kg
% rear unsprung mass, kg
% total vehicle mass, kg
% distance front axle to CoG, m
% distance rear axle to CoG, m
% wheelbase, m
% distance between the ground and the front static roll center, m
% distance between the ground and the rear static roll center, m
% height of sprung mass CG above ground, m
% height of front unsprung mass above ground, m
% height of rear unsprung mass above ground, m
% front track width, m
% rear track width, m
% average track width, m
% longitudinal distance from sprung CG to total mass CG, m
% distance from total CG to front axle
% distance from total CG to rear axle
% height of origin above ground, m
% distance from the roll center to the sprung mass center, m
% height of front unsprung mass above the origin, m
% height of rear unsprung mass above the origin, m
% height of total vehicle CG, m
% moment of inertia of sprung mass about x-axis, kg-m^2
% moment of inertia of sprung mass about y-axis, kg-m^2
% moment of inertia of sprung mass about z-axis, kg-m^2
% product of inertia of the sprung mass about x&y axis, kg-m^2
% product of inertia of the sprung mass about x&z axis, kg-m^2
% product of inertia of the sprung mass about y&z axis, kg-m^2
% inertia of front unsprung mass about its own CG, kg-m^2
% inertia of rear unsprung mass about its own CG, kg-m^2
% roll inertia of sprung mass about origin (x-axis), kg-m^2
% total yaw inertia, kg-m^2
% inertia of sprung mass about the origin, kg-m^2
% inertia of total vehicle, kg-m^2

% front suspension roll stiffness, Nm/rad
% rear suspension roll stiffness, Nm/rad
% front suspension roll damping coefficient, Nm-s/rad
% rear suspension roll damping coefficient, Nm-s/rad
% vertical stiffness of front springs, N/m
% push vertical damping of front springs, N-s/m
% pull vertical damping of front springs, N-s/m
% vertical stiffness of rear springs, N/m
% push vertical damping of rear springs, N-s/m
% pull vertical damping of rear springs, N-s/m
% strut force to wheel force
% bump force to suspension force

% rolling resistance factor
% rolling resistance factor
% rolling resistance factor

% vertical stiffness of tire, N/m
% vertical damping of tire, N-s/m

```

rk = 0.3025;	% tire radius, m
L_w=1.6;	% wheel inertia, kg-m^2
% Pacejka tire model with inflation/deflation	pressure [values from IPG CarMaker]
LMUX = 1;	% scale factor of Fx peak friction coefficient
LMUY = 1;	% scale factor of Fy peak friction
LFZO = 1;	% scale factor of nominal (rated) load
LCX = 1;	% scale factor of Fx shape factor
LEX = 1;	% scale factor of Fx curvature factor
LKX = 1;	% scale factor of Fx slip stiffness
LHX = 1;	% scale factor of Fx horizontal shift
LVX = 1;	% scale factor of Fx vertical shift
LGAX = 1;	% scale factor of camber for Fx
LCY = 1;	% scale factor of Fy shape factor
LEY = 1;	% scale factor of Fy curvature factor
LKY = 1;	% scale factor of Fy cornering stiffness
LHY = 1;	% scale factor of Fy horizontal shift
LVY = 1;	% scale factor of Fy vertical shift
LGAY = 1;	% scale factor of camber for Fy
LTR = 1;	% scale factor of Peak of pneumatic trail
LGAZ = 1;	% scale factor of camber for Mz
LXAL = 1;	% scale factor of alpha influence on Fx
LYKA = 1;	% scale factor of alpha influence on Fy
LVYKA = 1;	% scale factor of kappa induced Fy
LS = 1;	% scale factor of Moment arm of Fx
LMX = 1;	% scale factor of overturning couple
% Longitudinal coefficients	
PCX1 = 1.6055;	% shape factor Cfx for longitudinal force
PDX1 = 1.1703;	% longitudinal friction Mux at Fznom
PDX2 = -0.081328;	% variation of friction Mux with load
PDX3 = 0.0;	% variation of friction Mux with camber
PEX1 = 0.53409;	% longitudinal curvature Efx at Fznom
PEX2 = -0.019956;	% variation of curvature Efx with load
PEX3 = 0.18089;	% variation of curvature Efx with load squared
PEX4 = 2.9951e-07;	% factor in curvature Efx while driving
PHX1 = 0.0058715;	% horizontal shift Shx at Fznom
PHX2 = eps;	% variation of shift Shx with load
PKX1 = 36.411;	% longitudinal slip stiffness Kfx/Fz at Fznom
PKX2 = 0.12615;	% variation of slip stiffness Kfx/Fz with load
PKX3 = 0.51289;	% exponent in slip stiffness Kfx/Fz with load
PVX1 = -0.18;	% vertical shift Svz/Fz at Fznom
PVX2 = 0.0;	% variation of shift Svz/Fz with load
RBX1 = 18.456;	% slope factor for combined slip Fx
RBX2 = 16.314;	% variation of slope Fx reduction with kappa
RBX3 = 0.0;	% variation of slope Fx reduction with kappa
RCX1 = 1.0;	% shape factor for combined slip Fx
REX1 = 0;	% curvature factor of combined Fx
REX2 = 0;	% curvature factor of combined Fx with load
RHX1 = 0.0066313;	% shift factor for combined slip Fx reduction
PPX1 = -0.12;	% pressure factor
PPX2 = -0.05;	% pressure factor
PPX3 = -0.18;	% pressure factor
PPX4 = -0.08;	% Pressure factor
% Overturning coefficients	
Qsx1 = 2.3155e-04;	% lateral force induced overturning
Qsx2 = 0.51574;	% camber induced overturning couple
Qsx3 = 0.046399;	% Fy induced overturning couple
% Lateral coefficients	
PCY1 = 2.1322;	% shape factor Cfy for lateral forces
PDY1 = 1.0283;	% lateral friction Muy
PDY2 = -0.16758;	% variation of friction Muy with load
PDY3 = -1.5821;	% variation of friction Muy with squared camber
PEY1 = 0.33443;	% lateral curvature Efy at Fznom
PEY2 = -1.8733;	% variation of curvature Efy with load

PEY3 = -0.13136;	% zero order camber dependency of curvature Efy
PEY4 = -11.677;	% variation of curvature Efy with camber
PHY1 = 0.0031377;	% horizontal shift Shy at Fznom
PHY2 = 0.00051596;	% variation of shift Shy with load
PHY3 = 0.039251;	% variation of shift Shy with camber
PKY1 = -20.505;	% maximum value of stiffness Kfy/Fznom
PKY2 = 2.0284;	% load at which Kfy reaches maximum value
PKY3 = 0.89994;	% variation of Kfy/Fznom with camber
PVY1 = 0.0526365;	% vertical shift in Svy/Fz at Fznom
PVY2 = -0.0062119;	% variation of shift Svy/Fz with load
PVY3 = -0.41389;	% variation of shift Svy/Fz with camber
PVY4 = -0.048038;	% variation of shift Svy/Fz with camber and load
PPY1 = 0.33;	% pressure factor
PPY2 = -0.15;	% pressure factor
PPY3 = 0.039;	% pressure factor
PPY5 = -0.032;	% pressure factor
RBV1 = 22.003;	% slope factor for combined Fy reduction
RBV2 = -13.623;	% variation of slope Fy reduction with alpha
RBV3 = -0.0093616;	% shift term for alpha in slope Fy
RCY1 = 0.98294;	% shape factor for combined Fy reduction
REY1 = 0.0;	% curvature factor of combined Fy
REY2 = 0.0;	% curvature factor of combined Fy with load
RHY1 = -9.1492e-11;	% shift factor for combined Fy reduction
RHY2 = 0;	% shift factor for combined Fy reduction with load
RVY1 = 22.965;	% kappa induced slide force Svyk/Muy*Fz at Fznom
RVY2 = 0.37981;	% variation of Svyk/Muy*Fz with load
RVY3 = 1.8552;	% variation of Svyk/Muy*Fz with camber
RVY4 = 0.08767;	% variation of Svyk/Muy*Fz with alpha
RVY5 = -8.8234e-11;	% variation of Svyk/Muy*Fz with kappa
RVY6 = 0.90374;	% variation of Svyk/Muy*Fz with atan(kappa)
% Rolling coefficients (developed by another model)	
% Aligning coefficients	
QBZ1 = 19.588;	% trail slope factor for trail Bpt at Fznom
QBZ2 = -4.1671;	% variation of slope Bpt with load
QBZ3 = -0.34132;	% variation of slope Bpt with load squared
QBZ4 = -0.023607;	% variation of slope Bpt with camber
QBZ5 = -0.0793;	% variation of slope Bpt with absolute camber
QBZ9 = -50.0;	% slope factor Br of residual torque Mzr
QBZ10 = 0.0;	% slope factor Br of residual torque Mzr
QCZ1 = 1.2012;	% shape factor Cpt for pneumatic trail
QDZ1 = 0.12819;	% peak trail Dpt" = Dpt*(Fz/Fznom*R0)
QDZ2 = -0.0052433;	% variation of peak Dpt" with load
QDZ3 = -0.099544;	% variation of peak Dpt" with camber
QDZ4 = -15.0;	% variation of peak Dpt" with camber squared
QDZ6 = 6.2499e-05;	% peak residual torque Dmr" = Dmr/(Fz*R0)
QDZ7 = 5.0135e-04;	% variation of peak factor Dmr" with load
QDZ8 = -0.28564;	% variation of peak factor Dmr" with camber
QDZ9 = -0.025056;	% variation of peak factor Dmr" with camber and load
QEZ1 = -0.0044308;	% trail curvature Ept at Fznom
QEZ2 = 0.022579;	% variation of curvature Ept with load
QEZ3 = -0.16199;	% variation of curvature Ept with load squared
QEZ4 = -0.5576;	% variation of curvature Ept with sign of Alpha-t
QEZ5 = -87.296;	% variation of Ept with camber and sign of Alpha-t
QHZ1 = 8.2436e-04;	% trail horizontal shift Sht at Fznom
QHZ2 = 0.00273 ;	% variation of shift Sht with load
QHZ3 = 0.15;	% variation of shift Sht with camber
QHZ4 = 0.11608;	% variation of shift Sht with camber and load
QPZ1 = 0.47;	% pressure factor
SSZ1 = -5.3899e-10;	% nominal value of s/R0: effect of Fx on Mz
SSZ2 = -2.2371e-09;	% variation of distance s/R0 with Fz/Fznom
SSZ3 = 3.6029e-08;	% variation of distance s/R0 with camber
SSZ4 = 3.9826e-08;	% variation of distance s/R0 with load and camber
% Tire transient dynamics	

Lx= 0.1;	% tire longitudinal relaxation length, m
Ly= 0.5;	% tire lateral relaxation length, m
%% Electric Powertrain Parameters	
% MAP of torque vs. speed	
max_torque = 50;	% max. torque of the electric motor, Nm
max_power = 20000;	% max. power of the electric motor, W
speed_max = 14000;	% maximum motor allowed angular velocity, rpm
% Transient motor behaviour	
tm = 5/1000;	% electric motor rise time, s
taum = -tm/log(0.1);	% electric motor drive torque time constant, s
slew_rate = 350000;	% motor slew rate, Nm/s
tm_delay = 3/1000;	% electric motor time delay, s
% Torsion dynamics	
utr = 1 / 8.5;	% single gear ratio
Chs = 7000;	% half-shaft torsion stiffness, Nm/rad
Khs = 2;	% half-shaft torsion damping coefficient, Nm-s/rad
etajin = 0.99;	% efficiency of the inner constant velocity joint
etajout = 0.99;	% efficiency of the outer constant velocity joint
etag = 0.98;	% efficiency of the transmission
Jem = 0.05;	% motor inertia, kg-m ²
Jtr1 = 0.002;	% gear drive inertia (motor side), kg-m ²
Jtr2 = 0.003;	% gear drive inertia, (output shaft), kg-m ²
Jhs = 0.1;	% half-shaft inertia, kg-m ²
% Battery	
n=180;	% number of cells
SOC0 = 50;	% initial SOC, %
U0B = n*2.2;	% no-load voltage, V
R0B = n*1.5*10 ⁻³ ;	% internal resistance of the battery, Ohm
C1B = n*10;	% battery capacity, F
R1B = n*0.1;	% overvoltage resistance, Ohm
%% Steering system	
u_steer=20;	% kinematic ratio of steering system
K_cs_Fy=[-0.27e-3 -0.27e-3 -0.10e-4 -0.10e-4];	% compliance steer depending on lateral force
K_cs_Mz=[0.38e-2 0.38e-2 0.190e-2 0.190e-2];	% compliance steer depending on self-aligning torque
K_rs=[0.12 0.12 0.12 0.12];	% roll steer coefficients
freq_sw=5*2*pi;	% natural frequency of conventional steering, rad/s
damp_sw=0.707;	% damping ratio of conventional steering
freq_rear_steer=10*2*pi;	% natural frequency of IRS, rad/s
damp_rear_steer=0.707;	% damping ratio of IRS
%% Hydraulic Brake System Parameters	
pedal_ratio=5;	% Pedal ratio
failure_state=0;	% Failure activation (0 - no fail, 1 - fail)
% Pedal Travel Simulator	
% Pedal force vs. pedal travel - lookup table without local extrema	
pedal_travel=[0 2.5 5 10 20 30 40 50 60 64 66 67 68 69 69.5 69.9];	% Pedal travel, mm
pedal_force=[0 6.5 11 22 32 55 90 130 175 200 215 225 235 250 320 600];	% Pedal force, N
ped_trl=0:0.1:69.9;	
ped_frc=interp1(pedal_travel,pedal_force,ped_trl,'spline');	
ped_trl=[ped_trl 70];	
ped_frc=[ped_frc 800];	
% Pump characteristic	
delta_pressure_pump=linspace(0,18e+6,10);	% difference between pressures of accumulator and reservoir, Pa
Q_pump=linspace(8,6,10)*10 ⁻⁶ ;	% pump flow, m ³ /s
% Valve dimension	
din_front=0.71/1000;	% diameter of front inlet valve, m
dout_front= 1.5*din_front;	% diameter of front outlet valve, m
din_rear=0.40/1000;	% diameter of rear inlet valve, m
dout_rear=1.5*din_rear;	% diameter of rear outlet valve, m
dblock=1.2/1000;	% diameter of block valve, m
dcompen=1.2/1000;	% diameter of compensative valve, m
% Accumulator	
Vacc0= 200e-6;	% initial gas volume, m ³
pacc0=17.1e+6;	% initial accumulator gas pressure, Pa


```

plow=0.1e+6; % reservoir pressure, Pa
% Master Cylinder
d_mc=16/1000; % diameter of tandem master cylinder, m
% Valve dynamics (the second order transfer function)
eps_valve=1.3; % damping ratio of valve
wn_valve=2*pi*60; % natural frequency, rad/s
tdelay_valve=1/1000; % time delay, s
% Wheel brake pressure as a function of the fluid volume in the cylinder (experimental testing)
pref=[eps 0.5 1 2.5 5 10 15 20 30 40 60 80 100 120 160]*1e5; % Pressure, Pa
VFm=[eps 0.15 0.3 0.46 0.66 0.88 1.09 1.29 1.56 1.79 2.13 2.46 2.78 2.95 3.15]/100^3; %sm^3->m^3
VRm=[eps 0.14 0.26 0.39 0.5 0.61 0.71 0.79 0.93 1.03 1.2 1.35 1.53 1.63 1.73]/100^3; %sm^3->m^3
VcylF=max(VFm); % front cylinder volume, m^3
VcylR=max(VRm); % rear cylinder volume, m^3
% Brake mechanism
p_pushout=5e+5; % push out pressure, Pa
re=0.13; % effective radius of friction, m
dcyl=57/1000; % diameter of brake piston, m
effcyl=0.96; % wheel cylinder efficiency
% Friction model
mu0=0.35; % initial static friction coefficient
in_temp=100; % initial temperature, C
refPower=2*10^5; % reference system power, W
eta0=0.1; % initial normalized temperature in the friction contact
epsil=0.225; % constant normalized power
a_fric=0.1; % empirical parameter defines the influence of power
alpha_fric=0.8; % empirical parameter defines the temperature behaviour
c_fric=0.3; % empirical parameter defines the temperature behaviour
gamma_fric=0.35; % empirical parameter defines the temperature behaviour
% Hysteresis model
mcyl=2*(180+240)/1000; % mass of piston and pad, kg
K1=1.35*10^7; % translational contact stiffness, N/m
C1=1.5*K1*10^-2; % translational contact damping, N-s/m
z0=eps; % initial hysteresis value
n1=0.75; % coefficient of hysteretic model for loading curve
a1=2; % coefficient of hysteretic model for loading curve
b1=0.2; % coefficient of hysteretic model for loading curve
g1=0.5; % coefficient of hysteretic model for loading curve
n2=2; % coefficient of hysteretic model for unloading curve
a2=2; % coefficient of hysteretic model for unloading curve
b2=4.2; % coefficient of hysteretic model for unloading curve
g2=0.7; % coefficient of hysteretic model for unloading curve
%% EHCUC Pressure Controller
Kp_PID_brake=[0.3 0.3 0.2 0.2]; % proportional gain of EHCUC controller
Kd_PID_brake=ones(1,4); % derivative gain of EHCUC controller
Ki_PID_brake=0; % integral gain of EHCUC controller
%% Motor PI Controller
Kp_motor=1; % proportional gain of e-motor controller
Ki_motor=150; % integral gain of e-motor controller
%% Vehicle Reference model
aref=-0.0526*(ped_frc-20); % reference acceleration
aref (aref>-0.0) = 0; aref (aref<-1.35*g) = -1.35*g;
Cf=2*5.0e+4; % cornering stiffness of front tires of reference model, N/rad
Cr=2*7.8e+4; % cornering stiffness of rear tires of reference model, N/rad
%% Vehicle Dynamics SOSMC controller
Klon = 15; % gain of SOSMC of longitudinal dynamics
thr_lon = 1e-2; % threshold of SOSMC of longitudinal dynamics
Klat = 10; % gain of SOSMC of lateral dynamics
thr_lat = 2.5e-2; % threshold of SOSMC of lateral dynamics
Kyaw_lg = 5; % lowest gain of SOSMC of yaw dynamics
Kyaw_hg = 25; % highest gain of SOSMC of yaw dynamics
thr_yaw = 5e-2; % threshold of SOSMC of yaw dynamics

```

Appendix A.2

Vehicle Body	Bodies	Engine	Suspensions	Steering	Tires	Brake	Powertrain	Aerodynamics	Sensors
Body	x [m]	y [m]	z [m]	Mass [kg]	lxx [kgm ²]	lyy [kgm ²]	lzz [kgm ²]		
Wheel Carrier FL	3.58	0.7854	0.3	18.	0.2	0.2	0.2		
Wheel Carrier FR	3.58	-0.7854	0.3	18	0.2	0.2	0.2		
Wheel Carrier RL	0.94	0.7847	0.3	13	0.2	0.2	0.2		
Wheel Carrier RR	0.94	-0.7847	0.3	13	0.2	0.2	0.2		
Wheel FL	3.58	0.7854	0.3	28	0.8	1.6	0.8		
Wheel FR	3.58	-0.7854	0.3	28	0.8	1.6	0.8		
Wheel RL	0.94	0.7847	0.3	28	0.8	1.6	0.8		
Wheel RR	0.94	-0.7847	0.3	28	0.8	1.6	0.8		
Number of Trim Loads:			0					Mounting	

Position	x [m]	y [m]	z [m]
Origin Fr1	0.0	0.0	0.0
Aero Marker	3.65	0.0	0.6
Hitch	0.0	0.0	0.4
Jack FL	0.0	0.0	0.0
Jack FR	0.0	0.0	0.0
Jack RL	0.0	0.0	0.0
Jack RR	0.0	0.0	0.0

● Origin Fr1
 ● Positions
 ● Geometry Bodies
 ● Geometry Trim Loads

Vehicle Body: **Rigid**

Rigid Vehicle Body

☐ Override internally computed vehicle body proportioning

	x [m]	y [m]	z [m]	Mass [kg]	lxx [kgm ²]	lyy [kgm ²]	lzz [kgm ²]
Vehicle Body	2.15	0	0.58	1360	480	1910	2050
Vehicle Body B	2.15	0.0	0.58	650.5	180.0	900.0	900.0
Joint A - B	2.15	0	0.58				

Calculated vehicle overall mass [kg] **1534.00**

Model: **Dynamic Steer Ratio (GenTorque)**

Steering Gear Ratio

Mode: **Characteristic Value**

Rack to Steering Wheel [rad/m] **100.0**

Mechanical Module

Power Assist

Model: **Dynamic Steer Ratio (GenTorque)**

Steering by Torque

Inertia of Steering Wheel [kgm²] **0.001**

Rotational Damping [Nms/rad] **0.1**

Mass of Steering Rack [kg] **10.0**

Translational Damping [Ns/m] **50.0**

Max. Travel of Steering Rack [m] **0.08**

Amplification [-] **3.0**

Front Rear

Stiffness Mode: 1D Look-Up Table

Stiffness [N/m] 25000.0

compress [m]	force [N]
-0.01	-240.0
0.0	0.0
1.0	24000.0

Amplification [-] 1.0

☒ Length l0 [m] 0.32

Front Rear

Stiffness 15000.0 N/m Nm/rad

Amplification 1.0

Front Rear

Damping Mode: 1D Look-Up Table

Damping [Ns/m] 2500.0 Push

velocity [m/s]	force [N]
0.00	0
0.05	100
0.13	150
0.26	250
0.39	330

Amplification [-] 2.0

Damping [Ns/m] 5000.0 Pull

velocity [m/s]	force [N]
0.00	0
0.05	110
0.13	350
0.26	650
0.39	725

Amplification [-] 2.0

Front Rear

Stiffness Mode: 1D Look-Up Table

Stiffness [N/m] 50000.0 Push

compress [m]	force [N]
0.0	0.0
0.002	32.0
0.004	88.0
0.006	167.0

Amplification [-] 1

Length l0 [m] 0.055 parallel

Stiffness [N/m] 50000.0 Pull

compress [m]	force [N]
0.0	0.0
0.002	32.0
0.004	88.0
0.006	167.0

Amplification [-] 1

Length l0 [m] -0.09 parallel

Model: 1D Look-Up Table

General

Reference Point x [m] 2.25 y [m] 0.0 z [m] 0.6

Reference Area [m²] 2.0 Crosswind Build-up

Reference Length [m] 2.508 Kind of Build-up: Step

Aerodynamics Characteristics

tau [deg]	cD [-]	cS [-]	cL [-]	cRM [-]	cPM [-]	cYM [-]
-180	-0.4	0.0	0.1	0.0	-0.01	0.0
-120	-0.2	-1.4	0.7	-0.2	-0.021	0.06
-90	0.0	-1.7	0.9	-0.2	0.0	0.0
-45	0.3	-1.2	0.6	-0.16	0.025	-0.1
0	0.2	0.0	0.1	0.0	-0.03	0.0
45	0.3	1.2	0.6	0.16	0.025	0.1
90	0.0	1.7	0.9	0.2	0.000	0.0
120	-0.2	1.4	0.7	0.2	-0.021	-0.06
180	-0.4	0.0	0.1	0.0	-0.010	0.0

Driver

Mode: User parameterized Driver Racing Driver

Standard Parameters Traffic Race Driver Misc. / Additional Parameters

General

Cruising Speed 80 km/h dt Change of Pedals 0.5 s

Corner Cutting Coefficient 0.5 Min. dt Accel./Decel. 4 s

Accelerations, g-g Diagram

Max. Long. Acceleration 3.0 m/s²

Max. Long. Deceleration -4.0 m/s²

Max. Lat. Acceleration 4.0 m/s²

Exponent of g-g Diagram (ax/ay dependency)

Speed [km/h]	Accel.	Decel.
50	1.0	1.0

☐ Traction Control: reduce throttle if wheelspin occurs

Appendix A.3

Jacobian of process matrix A can be founded as:

$$A = \begin{bmatrix} 0 & 0 & x_2 & 0 & 0 & 0 & 0 & \frac{\cos \delta_{fl}}{m_a} & \frac{\cos \delta_{fr}}{m_a} & \frac{1}{m_a} & \frac{1}{m_a} & -\frac{\sin \delta_{fl}}{m_a} & -\frac{\sin \delta_{fr}}{m_a} & 0 & 0 \\ 0 & 0 & -x_1 & 0 & 0 & 0 & 0 & \frac{\sin \delta_{fl}}{m_a} & \frac{\sin \delta_{fr}}{m_a} & 0 & 0 & \frac{\cos \delta_{fl}}{m_a} & \frac{\cos \delta_{fr}}{m_a} & \frac{1}{m_a} & \frac{1}{m_a} \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{-0.5t_f \cos \delta_{fl} + a \sin \delta_{fl}}{I_{zz}} & \frac{0.5t_f \cos \delta_{fr} + a \sin \delta_{fr}}{I_{zz}} & \frac{-0.5t_r}{I_{zz}} & \frac{0.5t_r}{I_{zz}} & \frac{a \cos \delta_{fl} + 0.5t_f \sin \delta_{fl}}{I_{zz}} & \frac{a \cos \delta_{fr} - 0.5t_f \sin \delta_{fr}}{I_{zz}} & \frac{-b}{I_{zz}} & \frac{-b}{I_{zz}} \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & -\frac{r_w}{J_w} & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -\frac{r_w}{J_w} & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -\frac{r_w}{J_w} & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -\frac{r_w}{J_w} & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & -\frac{x_4}{L_x} & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -\frac{x_5}{L_x} & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -\frac{x_6}{L_x} & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -\frac{x_7}{L_x} & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -\frac{x_4}{L_y} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -\frac{x_5}{L_y} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -\frac{x_6}{L_y} & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -\frac{x_7}{L_y} \end{bmatrix}$$

Jacobian of measurement model C is:

$$C = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{\cos \delta_{fl}}{m_a} & \frac{\cos \delta_{fr}}{m_a} & \frac{1}{m_a} & \frac{1}{m_a} & -\frac{\sin \delta_{fl}}{m_a} & -\frac{\sin \delta_{fr}}{m_a} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{\sin \delta_{fl}}{m_a} & \frac{\sin \delta_{fr}}{m_a} & 0 & 0 & \frac{\cos \delta_{fl}}{m_a} & \frac{\cos \delta_{fr}}{m_a} & \frac{1}{m_a} & \frac{1}{m_a} \\ 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}$$

Appendix A.4

The simulation scenario is selected as the ‘Sine with Dwell’ manoeuvre due to the fact that the requirements of ISO standard related to longitudinal and lateral accelerations for braking-in-the-turn manoeuvre cannot be realized under low friction conditions. The steering law is kept the same for all friction conditions. Initial vehicle velocity is selected as: 80 km/h for concrete (0,8); 70 km/h for gravel (0,6); 60 km/h for earth road (0,4); 50 km/h for snow (0,2).

The simulation results are presented in Figure A.4.1. Mean error of yaw rate and root mean square slip angle for the proposed algorithms of subsystem prioritization is kept close to the variant without the subsystem prioritization. Based on this fact, it can be concluded that vehicle stability is achieved under various friction conditions. Consumed / recuperated energy of electric powertrain for the proposed algorithms is better compared to the motion without the subsystem prioritization. Tire dissipation energy is not significantly changed within one road type.

Figure A.4.1 – Motion and energy characteristics during ‘Sine with Dwell’ manoeuvre under various friction conditions

Based on the recommendations from NHTSA test procedure for FMVSS 126, Electronic Stability Control Systems, a parameter, which defines aggressiveness of the manoeuvre, is maximal steering amplitude. The vehicle moves with an initial velocity of 80 km/h under high friction conditions with various max. steering amplitudes. The simulation results are presented in Figure A.4.2. The proposed algorithms demonstrate a small increasing of mean error of yaw rate due to more aggressive steering input. Velocity at the end of the manoeuvre for the proposed algorithms, except fuzzy approach, is higher and potential recuperated energy for all manoeuvres with prioritization is higher compared to the motion without the subsystem prioritization. Therefore, the effectiveness of the proposed solution is independent from varying driver input.

Figure A.4.2 – Motion and energy characteristics during ‘Sine with Dwell’ manoeuvre under various maximal steering amplitude