Evaluating Coordinated Cooperative Control of Three Active Car Systems Using Fuzzy-Logic

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Abstract. Vehicle systems automation is an inevitable process as a result of the growing safety requirements and the trend of increasing car operation autonomy by reducing driver’s influence. The paper covers the coordinated control operation of three active systems to assess the improvement of vehicle response. Direct yaw control through selective wheel braking (DYC), active front steering control (AFS) and active suspension normal force control (NFC) are included. The control method is fuzzy logic. The modelling is done in Matlab/Simulink, where 14-DOF nonlinear full vehicle model and 3-DOF reference model were introduced. Simulations were done for three manoeuvres: cornering event, single lane change and fishhook manoeuvre, at a speed of 130 km/h and dry road surface with adhesion coefficient of 0.9 was assumed. Coordination was done with adjusting scaling factors for each control. Results show improvements in the transient response behaviour of the vehicle compared with four vehicles: vehicle without any active system, vehicle with coordinated DYC and AFS, vehicle with coordinated DYC and NFC and vehicle with coordinated AFS and NFC. The biggest impact can be seen in results for sideslip angle and roll angle, coordinated cooperative control here gives lowest values in the transient period during the manoeuvre. Although it gives better results for the other variables as well. The goal of the coordinated cooperative control is fulfilled by improving the stability and handling of the vehicle, which helps the driver to maintain the feeling of manoeuvrability and the sense for the direction of movement of the vehicle. The system for direct yaw control through selective wheel braking has major impact but the coordination with the other two systems lowers the braking force per wheel. The system for active front steering control has lower impact but at lower speeds helps for maintaining better direction of vehicle movement. Considering these three active systems act simultaneously during high speeds and critical manoeuvres next step would be to define sequence of activation depending on the currently measured vehicle behaviour variables.

1. Introduction

Safety is still the main purpose in the research area of motor vehicles, but also automotive technology is going into direction of making vehicles autonomous. Steps between include automated vehicle’s systems. Automation of vehicle systems lowers the human (driver) influence in order to increase safety and stability of the vehicle, considering the human effect a main purpose for majority of car accidents [1]. Vehicle’s system automation starts with main vehicle systems which enable and support its drivability: steering, braking, suspension, but also automation of powertrain subsystems. Integrated control system of the front steering angle compensation and the braking force distribution with the application of model-matching control technique based on optimal control theory is proposed in [2].
Authors also worked on improved behavior under severe driving conditions with active control of either front or rear angle combined with yaw moment in [3]. Integrated vehicle chassis control is proposed in [4] to coordinate steering, braking/traction and active stabilizer via loop structure. The main-loop calculates the stabilizing force using sliding mode control, while the servo-loop optimally distributes force to each tire, using SQP approach. Three different cases of combining lateral force and direct yaw moment control have been investigated in [5]. A simulation of a closed-loop driver–vehicle system during quick lane change with braking is used to show significant influence of the combined control on vehicle stability and responsiveness. Coordinated control of AFS and DYC is presented in [6] based on optimal guaranteed cost theory. A rollover prevention controller for vehicles with a high center of gravity is proposed in [7]. Active suspension with lateral acceleration as a disturbance was designed and controller gains were obtained through the LQ SOF method. And ESP was designed to maintain the maneuverability of the controlled vehicle. A two-layer hierarchical control architecture is proposed in [8] for integrated control of ASS and ESP where the upper-layer is designed to coordinate the interactions between the ASS and the ESP. Coordination of active front steering and rear braking in a driver-assist system for vehicle yaw control is investigated in [9]. The coordination is achieved through a suitable gain scheduled Linear Parameter Varying controller. A combined AFS and DYC controller is presented in [10] with good robustness against in-vehicle network-induced time-varying delays for 4 wheel-independent-drive electric vehicles. An H∞-based LQR tracking controller is introduced and adopted in the control synthesis. A review study on integrated active steering and braking control for vehicle yaw stability is conducted in [11]. Gain-scheduled vehicle handling stability control via integration of active front steering and suspension systems is presented in [12]. Model and controller for an active front-wheel steering (AFS) system is proposed in [13] which are designed to be investigated in integrated dynamics control systems. Article [14] introduces the β phase plane method to determine the stability state of the vehicle and electronic stability program fuzzy controller to improve the stability of vehicle driving on a low adhesion surface at high speed. An advanced control system is designed in [15] which integrate three fuzzy logic controllers including Direct Yaw-moment Control (DYC), Active Roll-moment Control (ARC) and Active Front Steering (AFS) to enhance vehicle cornering and overturning stability. In this paper step forward is made from [15] adding one more active system in order to improve vehicle lateral dynamics and rollover stability. This paper uses fuzzy-logic control for coordinated cooperative control of three active systems. Developed control algorithm includes electronic stability control system, active front steering and front active suspension system.

2. Vehicle Dynamic Model
The mathematical model used for representation of the vehicle is based on a 14-DOF full vehicle model, developed by [16] and used by [17] (Figure 1). The 14-DOF nonlinear model is implemented in Matlab/Simulink to describe the dynamic behavior of the vehicle. Vehicle mass is taken as a lumped mass at the center of gravity and has six degrees of freedom, including linear and angular movements along X, Y and Z axis. It represents the sprung mass of the vehicle. There are another four lumped masses of wheels i.e. unsprung mass, each one having two degrees of freedom: vertical travel and wheel rotation. The 14-DOF vehicle model is sufficient to study the dynamic behavior of the vehicle in longitudinal, lateral, and vertical direction. Assumptions include that vehicle body is being modelled as rigid, steer angles of the outer and inner wheel are same, tires have contact with the ground all the time and the aerodynamic force is neglected.

A spring and a damper at each corner connect sprung mass to four unsprung masses. Thus, the vertical motion of sprung mass is given by:

\[ m_{pm} \ddot{z}_{pm} = F_{pr, pt} + F_{am, pt} + F_{pr, pd} + F_{am, pd} + F_{pr, zt} + F_{am, zt} + F_{pr, zd} + F_{am, zd} \] (1)

Equation of sprung mass pitch motion:


\[ I_y \cdot \dot{\omega}_y = (F_{pr.zl} + F_{am.zl} + F_{pr.zd} + F_{am.zd}) \cdot l_z - (F_{pr.pl} + F_{am.pl} + F_{pr.pd} + F_{am.pd}) \cdot l_p \]  

Equation of sprung mass roll motion:

\[ I_x \cdot \dot{\omega}_x = (F_{pr.pl} + F_{am.pl} + F_{pr.pd} + F_{am.pd}) \cdot \frac{b}{2} - (F_{pr.pd} + F_{am.pd} + F_{pr.zd} + F_{am.zd}) \cdot \frac{b}{2} \]  

Equation of vertical motion of each wheel:

\[ m_{nm.ij} \cdot \ddot{z}_{nm.ij} = F_{t.ij} - F_{pr.ij} - F_{am.ij} \]  

where index \( ij \) stands for the wheel position (i - front, rear; j - left, right).

The handling model has longitudinal, lateral and yaw motion of the sprung mass and wheel spin motion at each of the four wheels. The equation of longitudinal motion according to Newton’s second law is given by:

\[ m \cdot a_x = F_{xp.t} \cos \theta_p - F_{xp.i} \sin \theta_p + F_{xp.d} \cos \theta_p - F_{yp.d} \sin \theta_p + F_{xzl} + F_{xzd} \]  

Analogously to the (5), lateral motion is given by:

\[ m \cdot a_y = F_{yp.t} \cos \theta_p + F_{xp.t} \sin \theta_p + F_{yp.d} \cos \theta_p + F_{xp.d} \sin \theta_p + F_{xyl} + F_{xyz} \]  

The sum of yaw moment is given with:

\[ l_z \cdot \dot{\omega}_z = -\frac{b}{2} F_{xp.t} \cos \theta_p + \frac{b}{2} F_{xp.d} \cos \theta_p - \frac{b}{2} F_{xzl} + \frac{b}{2} F_{xzd} + \frac{b}{2} F_{yp.t} \sin \theta_p - \frac{b}{2} F_{yp.d} \sin \theta_p + l_f F_{xp.t} \sin \theta_p + l_f F_{xp.d} \sin \theta_p + l_f F_{yp.t} \cos \theta_p + l_f F_{yp.d} \cos \theta_p - l_z F_{xyl} - l_z F_{xyz} + M_{zpl} + M_{zpd} + M_{zzl} + M_{zzd} \]  

The equation of motion for each wheel rotation is:

\[ I_t \cdot \dot{\omega}_{ij} = T_{vi.j} - T_{ki.j} - F_{xij} \cdot r_t \]  

Geometry and mass parameters used in the vehicle model correspond to vehicle with higher centre of gravity, such as sport utility vehicles or crossover vehicles.

\[ \text{Figure 1.} \ 14 \text{ DOF vehicle model} \]

2.1. Tire Model

The longitudinal and lateral tire forces are modelled using Paceka’s magic formula tire model. Pacejka’s model is implemented in Matlab/Simulink to obtain the nonlinear behavior of tires [18, 19, 20].

2.2. Tire Model

A reference linear model with three degrees of freedom is adopted in order to obtain the desired values in relation to which the control algorithm makes calculations for corrective actions. The three degrees of freedom are lateral acceleration, yaw velocity and roll angle. The reference model calculates the
desired: yaw velocity \((\omega_{z,\text{pos}})\), side-slip angle \((\beta_{c,\text{pos}})\) and roll angle \((\phi_{\text{pos}})\) according to the change in steering angle and vehicle speed, given by 9 to 11 [21].

\[
\omega_{z,\text{pos}} = \frac{v_x}{t + \left(\frac{m(l_xk_{\delta_x} - lpk_{\delta_p})}{2k_{\delta_p}k_{\delta_x}l_x}\right)}v_x^2 \cdot \theta_p
\]

\[
\beta_{c,\text{pos}} = \frac{l_x - \left(\frac{lp}{k_{\delta_p}k_{\delta_x}}\right)v_x^2}{t + \left(\frac{m(l_xk_{\delta_x} - lpk_{\delta_p})}{2k_{\delta_p}k_{\delta_x}l_x}\right)}v_x^2 \cdot \theta_p
\]

\[
\phi_{\text{pos}} = 0
\]

3. Control Design

Fuzzy-logic is a method of nonlinear control and is effective in manipulating the uncertainty and nonlinearity associated with complex system control. According to [22] fuzzy-logic is a proven effective tool for control of vehicle dynamics. By adjusting the membership functions and linguistic rules through testing a real vehicle, best combination of control sets can be obtained. It has been concluded that Gaussian and triangular membership functions give approximately equally well results and generally better than other types of membership functions.

3.1. Modeling of Electronic Stability Control

Control actions are performed by calculating yaw moment \(T_{yaw}\) and roll moment \(T_{\phi}\) according to (12) and (13), where \(F_{kpi}\) and \(F_{kzi}\) denote the braking force of the front and rear wheel, respectively.

\[
T_{yaw} = F_{kpi} \cdot \frac{b_p}{z} + F_{kzi} \cdot \frac{b_z}{z}
\]

\[
T_{\phi} = T_{\phi ff} + T_{\phi fb}
\]

The principle of brake distribution is to brake the front and rear wheel on the same side of the vehicle. According to the steering wheel angle and the difference between the actual and the reference yaw velocity, the direction of the compensation moment can be determined. Thus can be determined which wheels should be braked. The principle for defining which wheels need to be braked is given in Table 1 [14].

| Reference yaw velocity \(\omega_{z,\text{ref}}\) | Yaw velocity deviation \((\omega_z - \omega_{z,\text{ref}})\) | Characteristic of steering | Braked wheels |
|------------------------------------------|---------------------------------|---------------------|----------------|
| \(\omega_{z,\text{ref}}>0\) (left turn) | \((\omega_z - \omega_{z,\text{ref}})\geq0\) | oversteer | front and rear right side |
| \(\omega_{z,\text{ref}}<0\) (right turn) | \((\omega_z - \omega_{z,\text{ref}})<0\) | understeer | front and rear left side |
| | \((\omega_z - \omega_{z,\text{ref}})>0\) | oversteer | front and rear right side |
| | \((\omega_z - \omega_{z,\text{ref}})\leq0\) | oversteer | front and rear left side |

To avoid frequent activation of the electronic stability control system, \(\beta\cdot\dot{\beta}\) phase plane method is used [14]. Fuzzy logic control calculates the compensation yaw moment \(T_{yaw}\) based on the yaw velocity error \(\omega_z\) and the sideslip angle \(\beta_c\). Five membership functions are assigned to each control input, two trapezoidal and three triangular. Direct yaw moment is obtained with a scaling coefficient, with defined eleven membership functions, of which two trapezoidal and nine triangular.
Rollover stability control regime has a feedforward and feedback control. Feedforward control inputs are steering wheel angle $\theta_{ut}$ and vehicle longitudinal speed $v_x$. Feedback control inputs are roll angle error $e(\phi)$ and roll rate error $\dot{e}(\phi)$. The corrective active roll moment $T_\phi$ is the sum of both outputs: feedforward and feedback moments. Rollover control uses only Gaussian membership functions.

### 3.2. Modeling of Active Front Steering Control

In active front steering system the actual turning angle $\theta_p$ is a difference between the input angle by the steering wheel $\theta_{ut}$ and the corrective angle generated by the fuzzy-controller $\theta_{corr}$. The steering actions are performed by accurate calculations of the correct turning angle of the front wheels as follows:

$$\theta_{corr} = \theta_{ut} - \theta_p$$  \hspace{1cm} (14)

Fuzzy-logic control estimates the corrective angle based on yaw velocity error $e(\omega_z)$ and the steering wheel angle generated by the driver $\theta_{ut}$. Scaling factor is also used. The number of membership functions and linguistic rules for both inputs and for the output are defined the same manner like in direct yaw moment control [15].

### 3.3. C. Modeling of Suspension Normal Force Control

Studies by [22], as well as [23] have shown the potential benefits of adapting the normal force through active suspension system control during turning maneuvers. Figure 2 shows a schematic representation of a single wheel suspension system where the force generated by the active and passive components can be determined by:

$$F_{sus} = c(z_{np} - z_p) + F_a + k(\dot{z}_{np} - \dot{z}_p)$$  \hspace{1cm} (15)

Figure 2. Quarter car model

In 15 $F_a$ is the controlling force in the system. Positive force causes acceleration of the body up and negative vertical acceleration of the wheel. By including the input acceleration and deceleration values, it has been observed that the application of activation force on the front angle, opposite to the turn direction, gives the best results, in terms of the lateral force impact during turning. It also gives the lowest roll movement, according to the combinations examined [22]. Therefore, an active system only on the front axle of the vehicle is chosen to model.

First controller considers yaw velocity error $e(\omega_z)$ and its derivative $\dot{e}(\omega_z)$. The input values have the opposite sign for left and right wheel. Both inputs have assigned per five Gaussian membership functions. Inputs for the second controller are steering wheel angle and longitudinal speed of the vehicle. In addition, Gaussian membership functions are assigned, five for the speed and seven for the angle. The system output is the normal control force, which is a sum from both controllers.
4. Coordinated Cooperative Control of Car Systems

Primary coordinated control was made for pairs of systems and their impact was analysed. Then coordinated cooperative control of all three active systems was comprised. It is based on coordinated control of electronic stability control (ESC) and active suspension normal force control (ASNFC) and then the active front steering control (AFSC) was added. Coordination was made by adaptation of the used scaling coefficients. In that manner every system contributes with a certain share in dynamic behaviour corrective actions. The structure of the coordinated control of those three systems: ESC, ASNFC and AFSC, is given on Figure 3 [24].

![Figure 3. Structure of active control system](image)

For the needs of the research three maneuvers were used in simulations to analyze vehicle's response in critical conditions, which belong in the category of open-loop test methods. First two are turning in a curvature and single lane-change according to ISO 7401 and the third one is fishhook maneuver developed by NHTSA.

5. Simulation Results and Discussions

All manoeuvres are performed at a speed of 130 km/h as it is limit speed on national highways. It is assumed a dry road surface with adhesion coefficient $\phi=0.9$. This paper only includes results for step input and sinusoidal input of one period.

![Figure 4. Wheels input angle (left - step steer, right - sine input)](image)

Figure 4 shows the input angle on the wheels transferred through the steering wheel by the driver for the vehicle without active control (black line) and for the vehicle with coordinated cooperative control (brown line). Vehicles’ dynamic response is given by yaw velocity (Figure 5), sideslip angle (Figure 6) and roll angle (Figure 7). Also, trajectory of vehicles is given on Figure 8.
Figure 5. Yaw velocity (left - step steer, right - sine input)

Figure 6. Sideslip angle (left - step steer, right - sine input)

Figure 7. Roll angle (left - step steer, right - sine input)
Figures give comparison of vehicle without active control systems, vehicles with coordinated control of pair of active systems and vehicle with coordinated control of all three systems.

Improvement of vehicle dynamic response is visible for all variables considered: yaw velocity, sideslip angle and roll angle. It leads to the conclusion that stability and thus safety is increased. Improvements are seen not just in lowering of variable transient magnitude but also in the time response.

Coordinated control of ESC, AFSC and ASNFC gives slightly better results than coordinated control of ESC and AFSC or of ESC and ASNFC. It points out the effectiveness of the inclusion of ESC into vehicle dynamics control.

The main benefit is seen in the values of the side slip angle and the roll angle, where the coordinated control of three active systems provides superior results, i.e. gives lowest values in the transient response. Coordinated control requires lower braking forces on wheels compared to single ESC system and also lower normal control forces in the suspension system compared to single ASNFC system. Since ESC works through individual wheel braking, coordinated action of the three active systems affects the transverse forces of wheels, which means it does not lead to a large reduction in the longitudinal velocity of the vehicle. AFSC has less impact at high speeds but its corrective actions provide better understanding of the direction of motion for the driver.

Further improvement of the defined control system can be done through defining sequence of activation depending on the currently measured vehicle behaviour variables.

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