Design and simulation of an integrated active yaw control system for road vehicles

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Abstract: In this paper, design methodology and simulation results of an active yaw control system for road vehicles are presented. The main objectives of the yaw control system are to estimate the desired yaw behaviour of the vehicle according to the demand of the driver by means of a two degree-of-freedom vehicle model and track this desired yaw rate while considering vehicle steerability parameters. Based on vehicle yaw rate error and vehicle sideslip angle, the controller system applies brake torques to individual wheels in order to create a yaw moment, thus maintaining the desired vehicle behaviour. The control system is based on fuzzy logic control and consists of a two-level controlling scheme: the high-level controller deals with the yaw rate control and vehicle sideslip angle limitation, while the low-level controller calculates and applies the appropriate brake torques to the appropriate wheels requested by the high-level controller for yaw moment generation. An eight degree-of-freedom vehicle model is used to represent the real vehicle in the simulations. Results are compared with those of similar studies in the literature and the observed differences are discussed.

Keywords: active yaw control; AYC; integrated vehicle safety systems; fuzzy logic control; FLC; wheel slip control; vehicle sideslip angle limitation.

Reference to this paper should be made as follows: Tekin, G. and Samim Ünlüsoy, Y. (2010) 'Design and simulation of an integrated active yaw control system for road vehicles', *Int. J. Vehicle Design*, Vol. 52, Nos. 1/2/3/4, pp.5–19.

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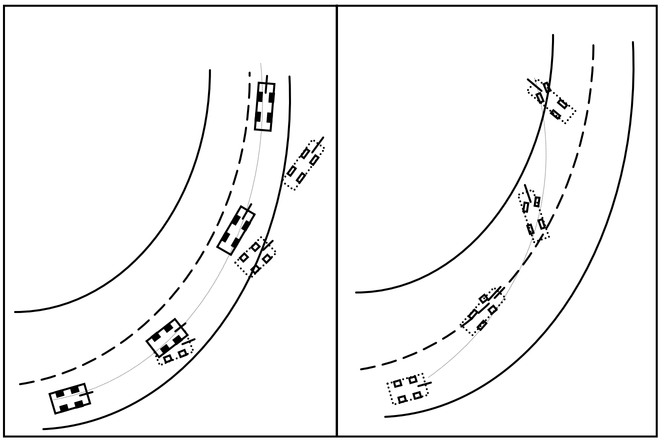
1 Introduction

Safety systems applied to road vehicles can be classified as active and passive safety systems. Passive safety systems are designed to prevent or lessen the damage to the passengers, pedestrians, and the vehicle(s) involved after the accident has occurred. Typical applications for passive systems are seat belts, air bags, door shock absorber bars, etc.

Figure 1 (a) Understeering (b) oversteering



(a) (b)

Active safety systems, on the other hand, are designed to prevent the accident by means of detecting any possible threats of instability and help the driver to regain control of the vehicle. The foremost major development was the development of anti-lock brake system (ABS). After the successful integration of ABS into road vehicles, several other active safety systems designed for different instability or 'loss of control' cases are introduced. Most commonly encountered problems in the control of road vehicles are disproportionate understeering or oversteering behaviours (Figure 1). Active yaw control (AYC) systems, designed to regulate the yaw rate of the vehicle in case of any excessive over- or understeering behaviour can be shown as an example of these active safety systems. Although many approaches have been proposed for the application of the control, independent wheel braking and engine torque control methods are the most preferred approaches (Boada et al., 2005; Park and Kim, 1999; Drakunov et al., 2000; Güvenç et al., 2003; Esmailzadeh et al., 2003; Shim and Margolis, 2001; Mokhiamar and

Abe, 2002, 2006). Attempts at developing a basic framework for active steering systems are also in progress and these systems are waiting for a certain level of maturity to be reached (Buckholtz, 2002).

While there have been successful applications of the individual active safety systems, recent work has focused on the design procedure, methodologies, and performance analyses of integration of these independent safety systems (Osborn and Shim, 2006; Gordon et al., 2003; Zheng et al., 2006). In view of the fact that these systems share similar information from common sensors and have similar control actions, it is logical to attempt their integration.

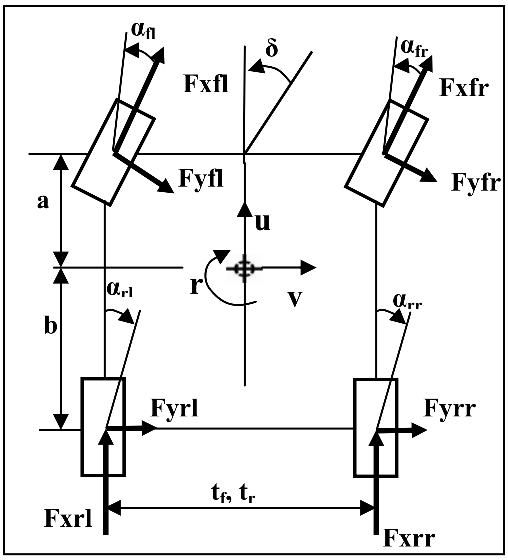
2 Vehicle model

To simulate the performance of the designed controller and make a throughout analysis, an eight degree-of-freedom (DOF) vehicle model is prepared in MATLAB/Simulink environment. Further, a simpler two DOF vehicle model (commonly known as bicycle model) is prepared and utilised in order to predict driver's intentions.

2.1 Eight DOF vehicle model

This eight DOF model is generated to represent the real vehicle behaviour under simulation conditions. These DOFs are the longitudinal and lateral motions, yaw, roll, and four wheel rotations. The pitch motion, suspension motions, suspension geometry, and body bounce motions are neglected since this study is basically on controlling the handling behaviour of vehicles. The model and the DOFs involved are shown in Figure 2.

Figure 2 Eight DOF vehicle model



Note: Roll DOF is not shown.

The differential equations for the detailed vehicle model are given below:

Longitudinal motion

$$\dot{u} = \frac{1}{M}\left(F_{xfl}\cos\delta_{fl} + F_{xfr}\cos\delta_{fr} + F_{xrl} + F_{xrr} - F_{yfl}\sin\delta_{fl} - F_{yfr}\sin\delta_{fr}\right) - h_s r\dot{\phi} + vr \tag{1}$$

Lateral motion

$$\dot{v} = \frac{1}{M}\left(F_{yfl}\cos\delta_{fl} + F_{yfr}\cos\delta_{fr} + F_{yrl} + F_{yrr} + F_{xfl}\sin\delta_{fl} + F_{xfr}\sin\delta_{fr}\right) - h_s\ddot{\phi}\cos(\phi) - ur \tag{2}$$

Yaw motion

$$\dot{r} = \frac{1}{I_{zz}}\Big[a\left(F_{xfl}\sin\delta_{fl} + F_{xfr}\sin\delta_{fr} + F_{yfl}\cos\delta_{fl} + F_{yfr}\cos\delta_{fr}\right) + b\left(-F_{yrl} - F_{yrr}\right) + \frac{t_r}{2}\left(F_{xrl} - F_{xrr}\right) + \frac{t_f}{2}\left(F_{yfl}\sin\delta_{fl} - F_{yfr}\sin\delta_{fr} - F_{xfl}\cos\delta_{fl} + F_{xfr}\cos\delta_{fr}\right)\Big] \tag{3}$$

Roll motion

$$\dot{p} = \ddot{\phi} = \frac{1}{I_{xx}} m_S h_S\left(\dot{v} + ur\right) + m_S h_S g\phi - K_{\phi f}\phi - C_{\phi f}\dot{\phi} - K_{\phi r}\phi - C_{\phi r}\dot{\phi} \tag{4}$$

Wheel motion (valid for all four wheels)

$$\dot{\omega} = \frac{1}{I_\omega}\left(-FR - T_{wh}\right) \tag{5}$$

Here, F_{ijk} represents the forces generated by the tyres and applied to vehicle, δ represents the front tyre steering angle and T_{wh} represents the torque applied to wheels by the driver and/or the controller. The vehicle parameters used in the simulation are given in Table 1.

Table 1 Eight DOF vehicle model parameter values

Parameter name		*Value*
M	Vehicle mass	1,300 kg
t_f	Front track length	1.45 m
t_r	Rear track length	1.45 m
a	Distance between COG and front axle)	1.1 m
b	Distance between COG and rear axle	1.35 m
R	Wheel radius	0.33 m
I_{zz}	Rotational inertia – yaw	1,620 kg m^2
I_{xx}	Rotational inertia – roll	750 kg m^2
I_w	Wheel rotational inertia	2.03 kg m2
h_s	Sprung mass COG height	0.2575 m

2.2 Load transfers

As previously mentioned, suspension forces and pitch motion are neglected in the detailed vehicle model study. However, the load transfers due to the longitudinal and lateral should be taken into account for accurate force generations. The equations provided below are utilised in order to calculate static and dynamic tyre forces.

Front left tyre

$$F_{ZFL} = \frac{mgb}{2(a+b)} - \frac{ma_x h}{2(a+b)} - \frac{ma_y bh}{2(a+b)t_f} + \frac{M_{\phi f}}{t_f} \tag{6}$$

Front right tyre

$$F_{ZFR} = \frac{mgb}{2(a+b)} - \frac{ma_x h}{2(a+b)} + \frac{ma_y bh}{2(a+b)t_f} - \frac{M_{\phi f}}{t_f} \tag{7}$$

Rear left tyre

$$F_{ZRL} = \frac{mga}{2(a+b)} + \frac{ma_x h}{2(a+b)} - \frac{ma_y ah}{2(a+b)t_r} + \frac{M_{\phi r}}{t_r} \tag{8}$$

Rear right tyre

$$F_{ZRR} = \frac{mga}{2(a+b)} + \frac{ma_x h}{2(a+b)} + \frac{ma_y ah}{2(a+b)t_r} - \frac{M_{\phi r}}{t_r} \tag{9}$$

In these equations, a_x represents longitudinal acceleration, a_y represents lateral acceleration, h represents height of the centre of mass of the vehicle, while M_ϕ represents the roll moment.

2.3 Tyre model

In this study, tyre model developed by Allen et al. (1987) is selected to simulate the tyre behaviour. Allen tyre model can provide realistic non-linear tyre cornering force characteristics even in combined slip cases, when longitudinal and lateral tyre slips occur simultaneously. Allen tyre model is preferred as it provides for effortless simulation in combined slip, when compared to other tyre models found in the literature.

2.4 Simple reference vehicle model (bicycle model)

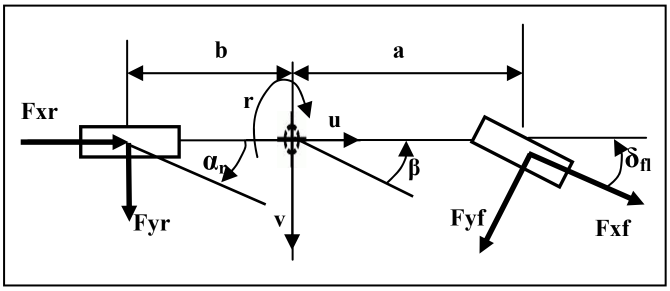
A major difficulty in yaw control problem is the prediction of the driver's intention. There exist no single solution for this particular problem. Indeed, the solution can be varied so that the vehicle may show desired over- or understeering characteristics. In this study, a two DOF single track model is modelled to represent the desired characteristics and used to predict the driver's demands. This model accepts throttle/brake and steering wheel angle signals coming from the driver as inputs and generates a desired yaw rate as the output. The two DOF vehicle model is shown in Figure 3.

The equations used to estimate the desired yaw rate is shown below:

$$m\dot{v} = \left(C_f + C_r\right)\frac{v}{U} + \left(aC_f - bC_r - mU^2\right)\frac{r}{U} - C_f\delta \tag{10}$$

$$J\dot{r} = \left(aC_f - bC_r\right)\frac{v}{U} + \left(a^2C_f + b^2C_r\right)\frac{r}{U} - aC_f\delta \tag{11}$$

Figure 3 Two DOF vehicle model



3 Controller design

The controller used to regulate the yaw rate of the vehicle consists of two levels. The first level controller compares the desired yaw rate with actual yaw rate and (if necessary) generates a yaw moment signal, resulting in calculation of independent brake torque signals. The second level controller (also called low-level controller) is a tyre slip controller, which assesses brake torque signals and regulates the brake so that undesired locking of wheels is avoided. The aim of this cascaded structure is to enable the designer to easily integrate different tyre slip controllers to the designed yaw rate control systems. The design of the tyre slip controller utilised in this study is based on the study of Şahin (2007). The general structure of the integrated controller system is shown in Figure 4.

3.1 Yaw rate controller design

The main objective of the yaw rate controller design is to compare the vehicle yaw rate and other stability related states with the desired states obtained from driver commands and manipulate the vehicle by means of brake, throttle regulation, etc., in case of any possible undesired behaviours. The designed controller consists of a simple vehicle model used to predict the driver's intention and a fuzzy-based controller comparing the desired and actual behaviours and generates a signal, which reduces this comparison error. The most important variable compared in this system is the yaw rate. Besides, float angle (also called vehicle sideslip angle), which affects vehicle stability and steerability drastically is also observed and taken into account. Since no direct measurement is possible, vehicle sideslip angle is estimated using measured variables such as lateral acceleration and longitudinal velocity parameters. Fuzzy-based controller evaluates these input variables and decides whether the error signals are enough to initiate the manipulation stage. If needed, the yaw moment and corresponding individual brake torques are calculated and applied by the controller. The general scheme of the yaw rate controller with the integrated tyre slip controller is shown in Figure 5.

Figure 4 General structure of the integrated controller

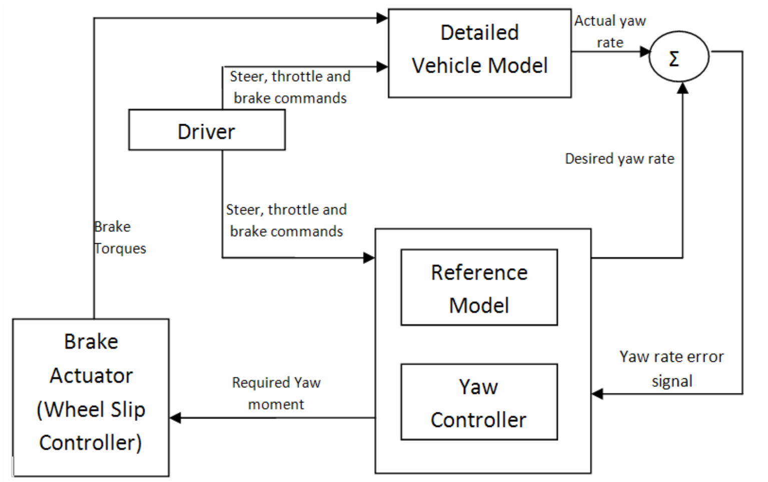


Figure 5 Yaw rate controller integration scheme (see online version for colours)

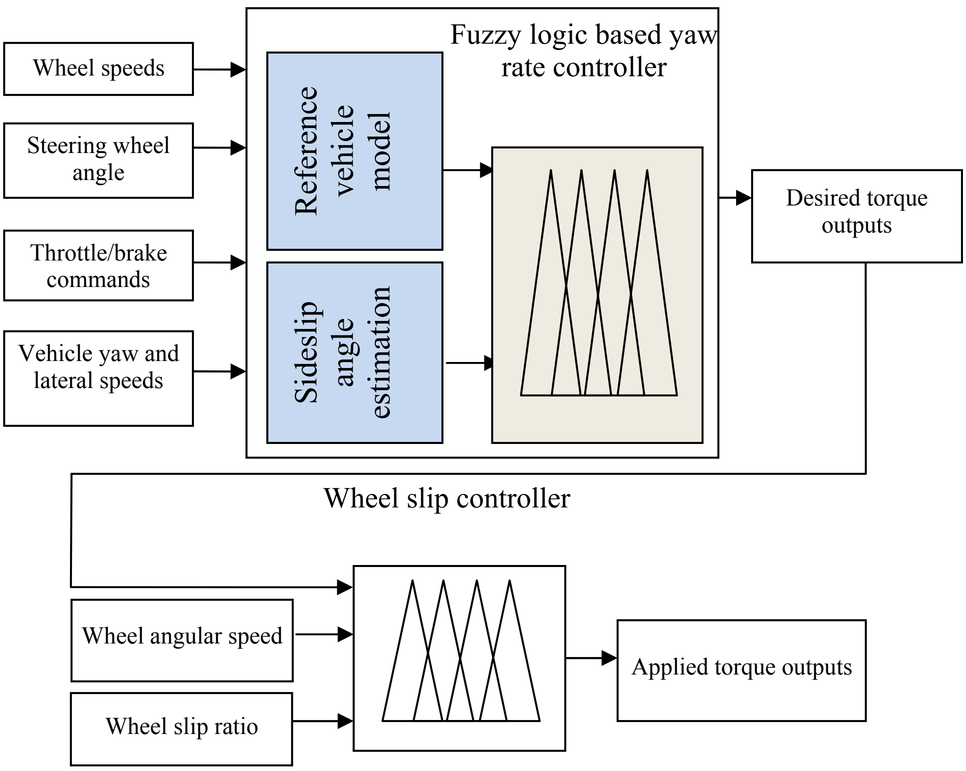
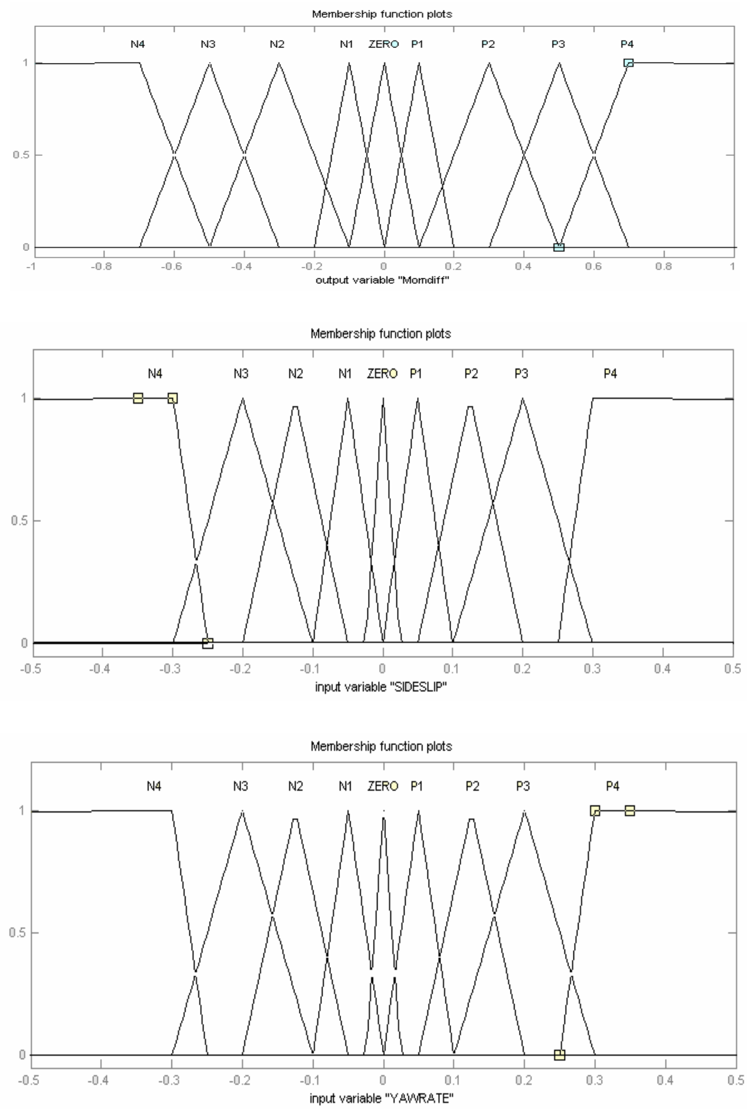


Figure 6 Yaw control fuzzy logic membership sets (see online version for colours)



For the design and application of fuzzy control, MATLAB fuzzy toolbox is used. Mamdani inference method is used for implementing the inference mechanism and centroid algorithm is used for the defuzzification process. Nine different levels are defined for each input and output membership functions. Membership functions have narrow ranges near to the zero point while they cover a broader range far from the central zero point. Membership functions are shown in Figure 6. The first two sets present yaw rate error and vehicle sideslip angle inputs correspondingly. The last set represents the output signal, which is the required normalised yaw moment to stabilise the vehicle. These set are conserved qualitatively, however, for different road conditions, the numerical values differ. For example, the yaw rate error signal is defined between

±0.9 rad/s for dry road conditions while it is defined between ±0.1 rad/s for icy road conditions. With this manipulation, effective control is obtained for all road conditions.

In order to use in fuzzy logic control (FLC) algorithm, an 'IF-THEN' based rule table is prepared. The rule base covers the whole input domain while the output signal is unique for every input pair, which means that 81 rules corresponding to each nine membership functions of the inputs are defined for the control algorithm. After time-consuming decision and trial-error processes, the tweaking of the rule base is completed. Designed rule base is shown in Table 2.

Table 2 Yaw controller rule base

Sideslip angle/yaw rate error	*N4*	*N3*	*N2*	*N1*	*Z*	*P1*	*P2*	*P3*	*P4*
N4	N4	N4	N4	N3	N2	N3	N3	N3	N3
N3	N4	N4	N3	N2	N2	N2	N3	N3	N3
N2	N4	N3	N2	N2	N2	N2	N3	N2	N2
N1	N3	N2	N2	N1	N1	N1	N2	N1	N1
Z	N2	N2	N1	N1	Z	P1	P1	P2	P2
P1	P3	P2	P2	P1	P1	P1	P2	P1	P1
P2	P4	P3	P2	P2	P2	P2	P3	P2	P2
P3	P4	P4	P3	P2	P2	P2	P3	P3	P3
P4	P4	P4	P4	P3	P2	P3	P3	P3	P3

3.2 Tyre slip controller design

Tyre slip controller is designed to regulate the brake torque according to the brake torque demand, wheel acceleration and wheel slip information. Following the initial application of the torque demand received from the yaw controller, wheel acceleration is compared with the vehicle longitudinal acceleration. Evaluating this information with the longitudinal slip, final brake torque is regulated and applied to the corresponding wheel. Rule base obtained after a long trial and error procedure for this tyre slip controller is shown in Table 3.

Table 3 Tyre slip controller rule base

Wheel acceleration/wheel slip error	*NL*	*NS*	*ZERO*	*PS*	*PL*
NL	NM	PL	PL	PL	PL
NS	NL	PL	PL	PM	PL
ZERO	NL	NM	Z	PM	PL
PS	NL	NL	NM	Z	Z
PL	NL	NL	NL	NM	Z

4 Simulation results

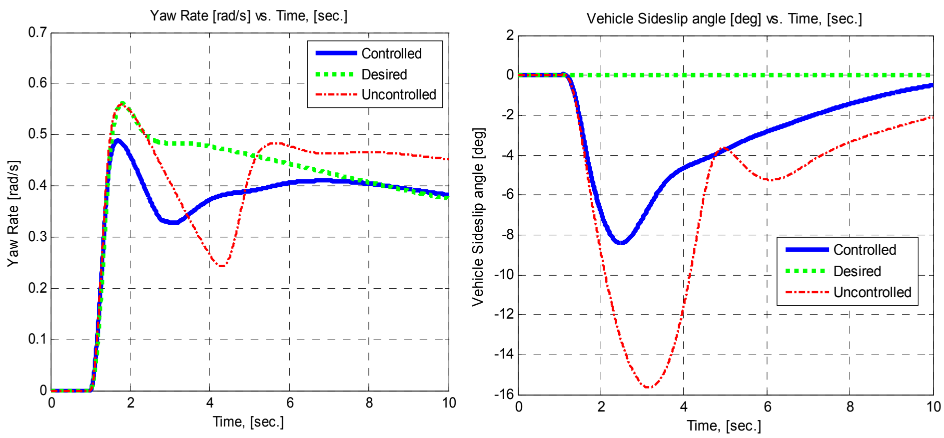
In this study, extensive simulation studies have been presented in order to evaluate the performance of the designed integrated AYC system and compare the results with similar studies in the literature. These simulation studies cover the combinations of two different manoeuvre types and three different road conditions. J-turn and lane change manoeuvre tests are commonly used in evaluating AYC systems. Road conditions are defined as dry and wet. After the application of these conditions, the same manoeuvres are applied for 0.2 g negative acceleration and dry road conditions.

Simulation results and evaluation of the results obtained are as follows.

J-turn – dry road conditions (μ = 0.9)

J-turn manoeuvre is a test type manoeuvre consisting of a constant steer angle while the vehicle velocity is kept constant. The aim of the manoeuvre is to make a turn with a nearly constant radius curve. Simulation results for 90 km/h vehicle velocity and 90° steering wheel angle (steering ratio = 1/18) are shown in Figure 7. Obtained yaw velocity is around 0.4 rad/s, while vehicle sideslip angle is kept about 8° at maximum. In similar studies found in literature, lower vehicle sideslip angles are obtained at the cost of lower yaw rates (Boada et al., 2005; Mokhiamar and Abe, 2002).

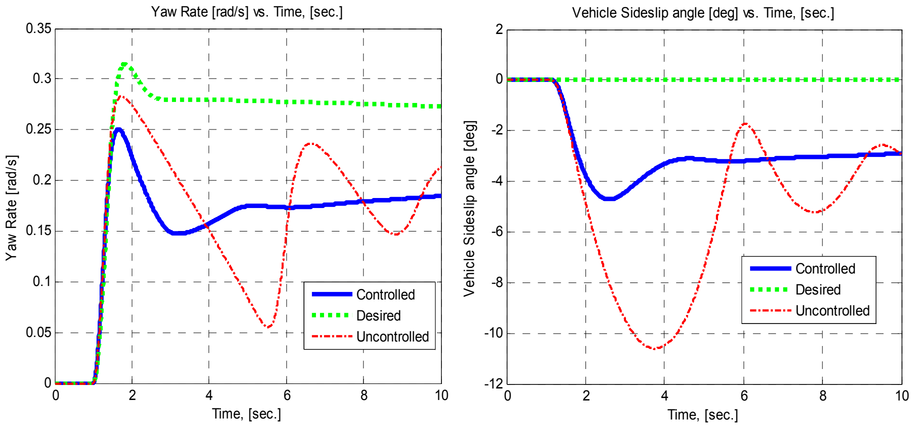
Figure 7 Raw rates and vehicle sideslip angles (J-turn dry road) (see online version for colours)



J-turn – wet road conditions (μ = 0.4)

In a similar simulation study, the vehicle model is tested under wet road conditions with an initial velocity of 90 km/h and 55° steering wheel angle. The simulation results are shown in Figure 8. The controlled and uncontrolled vehicles display similar behaviours, however, the oscillations in uncontrolled vehicle sideslip angle and yaw rate are eliminated.

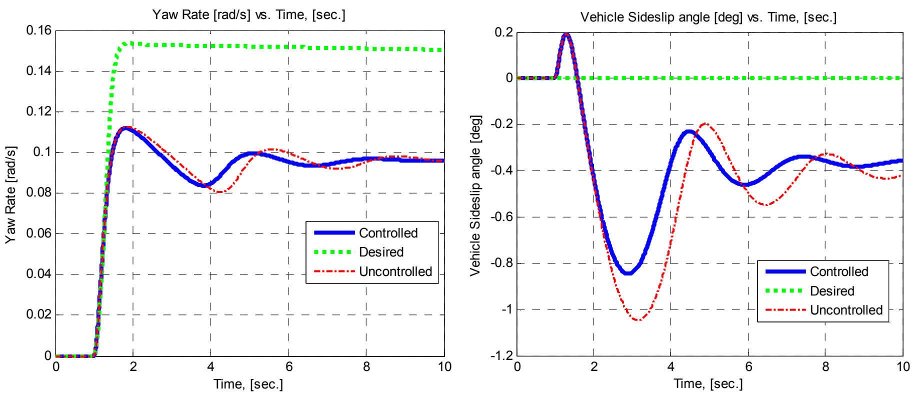
Figure 8 Yaw rates and vehicle sideslip angles (J-turn wet road) (see online version for colours)



J-turn – icy road conditions ($\mu = 0.1$)

J-turn manoeuvre is applied to icy road conditions. The vehicle velocity is initially 40 km/h while the steering wheel angle is kept as 45°. The simulation results are shown in Figure 9. In this simulation, the AYC system performs poorly with respect to previous simulation results due to the severe road conditions, which make the desired behaviour impossible to follow. On the other hand, active control system manipulates the vehicle to lower the sideslip angle, which is crucial for steerability on especially low friction surfaces. For comparison, studies of Esmailzadeh et al. (2003) can be examined.

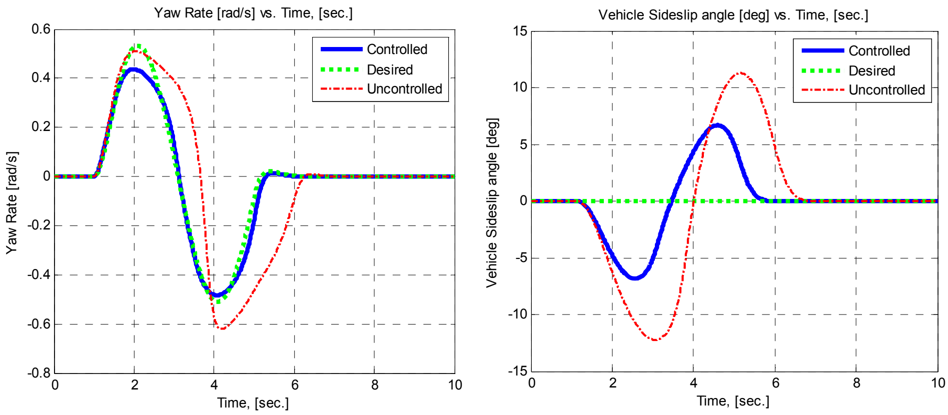
Figure 9 Yaw rates and vehicle sideslip angles (J-turn icy road) (see online version for colours)



Double lane change – dry road conditions ($\mu = 0.9$)

Double lane change (DLC) manoeuvre is obtained by varying the steering angle to one direction first and to opposite direction in series, with the steering angle variation in the shape of a sine wave. This manoeuvre simulates emergency lane changes when faced with an obstacle, which needed to be avoided. For the first simulation, with 90 km/h vehicle velocity and a maximum of 90° wheel angle, the simulation results are shown in Figure. 10. As it can be observed in Figure 11, obtained yaw rate with the controller is kept close to the desired yaw rate while the vehicle sideslip angle is limited to lower angles, which is beneficial for steerability of the vehicle. In similar simulations, De Li et al. (2005) reported yaw rates are near to 0.4 rad/s while the vehicle sideslip angles are limited to 10°. The yaw controller designed for this study reduces sideslip angles, hence providing for better controllability.

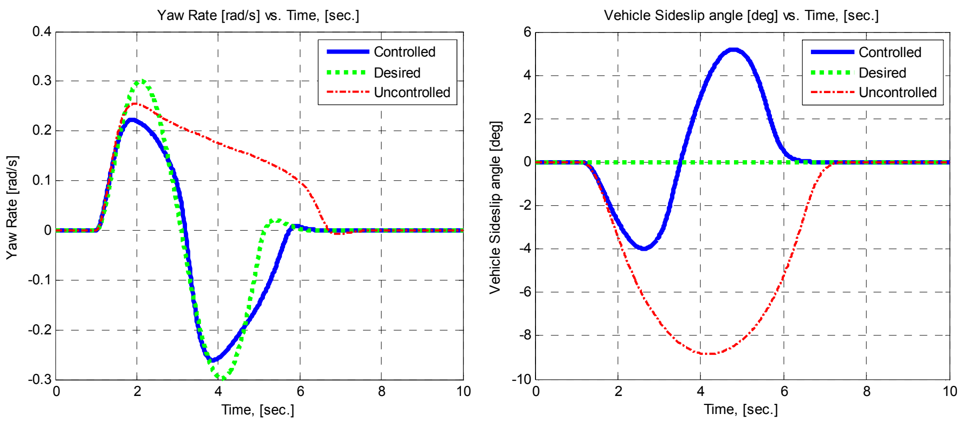
Figure 10 Yaw rates and vehicle sideslip angles (DLC – dry road) (see online version for colours)



DLC – wet road conditions ($\mu = 0.4$)

For this simulation, vehicle velocity is kept as in dry road conditions (90 km/h) while the steering wheel angle is taken as 55°. Simulation results are shown in Figure 11. Results achieved at the end of this simulation are in agreement with similar simulations in the literature (Boada et al., 2005; Shim and Margolis, 2001).

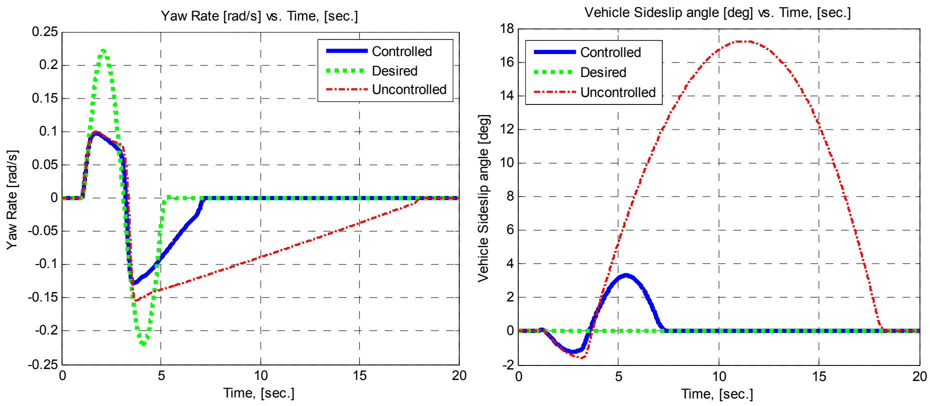
Figure 11 Yaw rates and vehicle sideslip angles (DLC – wet road) (see online version for colours)



DLC – icy road conditions (μ = 0.1)

In this simulation, DLC manoeuvre is repeated for icy road conditions. The vehicle velocity is initially 50 km/h while the maximum steering wheel angle is 55°. Results obtained with these parameters are shown in Figure 12. Vehicle yaw rate can only be raised to moderate values indicating the limits of controller performance, while the limits of the vehicle sideslip angle is lowered to acceptable values, hence maintaining steerability.

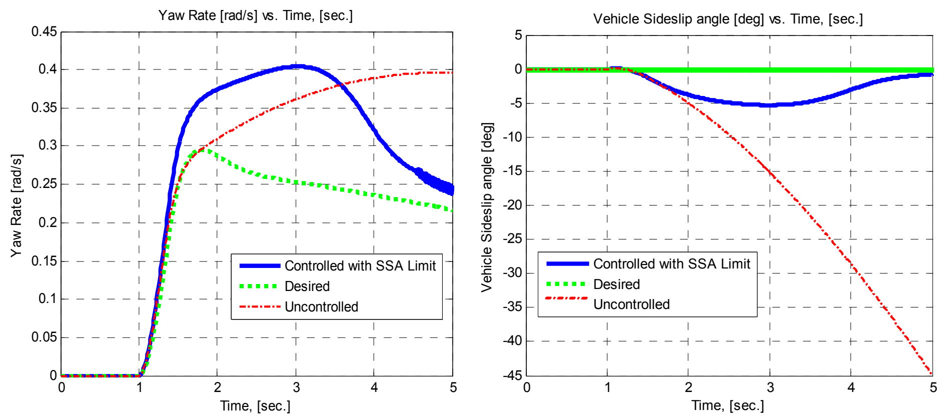
Figure 12 Yaw rates and vehicle sideslip angles (DLC – icy road) (see online version for colours)



J-turn – dry road conditions (μ = 0.9): (0.2 g deceleration)

In this simulation, J-turn manoeuvre is tested on the vehicle model on dry road conditions while brakes are applied resulting in a 0.2 g deceleration. Initial vehicle velocity is 90 km/h. As it can be observed from Figure 13, yaw rate controller has performed a successful manipulation in order to preserve the vehicle behaviour similar to the driver's intentions while maintaining the vehicle sideslip angle to acceptable limits.

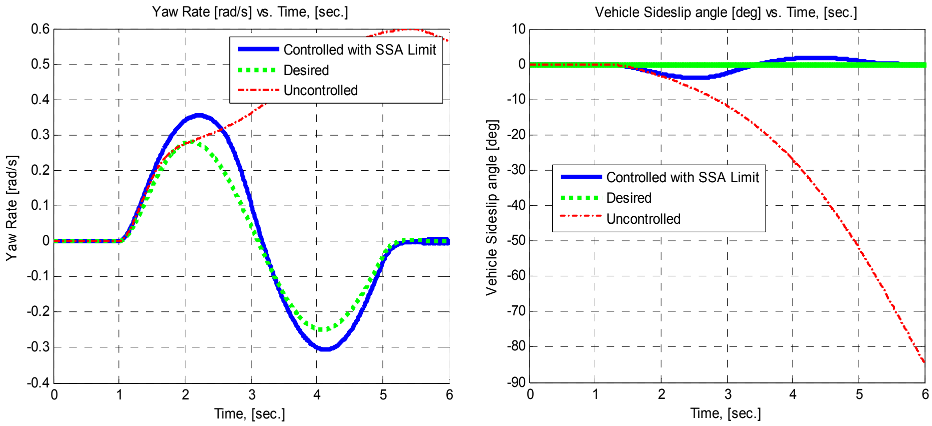
Figure 13 Yaw rates and vehicle sideslip angles (J-turn – dry road, 0.2 g) (see online version for colours)



DLC – dry road conditions (μ = 0.9): (0.2 g deceleration)

Last simulations are held with DLC manoeuvre with a 0.2 g deceleration on dry road conditions. Initial parameters are kept the same as in the previous simulations; the only changing parameter is the manoeuvre type. Simulation results are shown in Figure 14. According to the simulations, the vehicle may become unstable and the driver may lose steerability of the vehicle if yaw rate controller is not initiated. With the intervention of the controller, the vehicle behaviour is kept close to the driver's demand and steerability is preserved.

Figure 14 Yaw rates and vehicle sideslip angles (DLC – wet road, 0.2 g) (see online version for colours)



5 Conclusions and future work

The main objective of this study is to design an integrated AYC system and evaluate its performance with extensive simulations. Integration of the active yaw controller is realised with an appropriately designed tyre slip controller. Resulting system is tested via different simulations with various road conditions and manoeuvre types. Simulation results point out satisfactory outcomes with respect to yaw rates and vehicle sideslip angle limitations. These results are also compared with similar studies and simulations found in the literature, finding out that the designed system generally performs at least at the same level with the systems reported in the literature, if not better. The designed active yaw controller has a positive effect on vehicle stability and steerability. On the other hand, the integration of the active yaw controller is limited to only tyre slip controller. Extensions such as traction control systems, hill climb control, etc., may also be integrated to overall active safety systems in order to decrease output intervention and attain a more secure level of control and safety.

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