UNIVERSITY
OF WEST BOHEMIA

Applied and Computational Mechanics 1 (2007) 411 - 416

Applied and Computational Mechanics

Vehicle lateral dynamics control

V. Drobný [a,*], M. Valášek [b]

[a] TÜV SÜD Auto CZ s.r.o., Novodvorská 994/138, 142 21 Praha 4, Czech Republic
[b] Department of Mechanics, Biomechanics and Mechatronics, CTU in Prague, Karlovo nam. 13, 121 35 Praha 2, Czech Republic

Received 10 September 2007; received in revised form 10 October 2007

Abstract

This document deals with the controlled system of lateral vehicle dynamics regulated in terms of a reference model outputs. The reference model means a simplified mathematical vehicle model, which describes the desired lateral movements of a vehicle. The output of the reference model produces the most optimal trajectory in various driving conditions. The target of the control strategy is to hold the vehicle on this trajectory. Mathematical model of the vehicle and the reference model were created in the Matlab – Simulink software. The final product of the regulation was the corrective torque that had the stabilization effect on the controlled vehicle. By the simulations there were used user-defined maneuvres such as curve driving, avoidance maneuvre together with maneuvres defined by directives. The result was the comparison of the produced corrective torques of the control strategy for the defined driving states.

Keywords: vehicle lateral dynamics, controlled systems, active safety

1. Introduction

In the field of active safety there are a lot of known solutions how to keep the vehicle stable. By the growth of electronics and mechatronics applied in the automotive industry there is always distinguished potential source for new solutions. There are three basic ways how to directly or indirectly affect behaviour of the vehicle in yaw direction. All the methods have the same principle – to find optimal tire forces distribution among the four corners in the vehicle. Active rear steering changes the tire lateral forces through the wheel slip angles differences. ESP system [4] uses brake forces to change the longitudinal and lateral tire forces. Dynamic body control modifies the vertical forces distribution [3] in the way to change the adhesion threshold up and down. The mentioned systems have a similar control strategy based on tracing the reference model outputs and minimizing the control deviation. The system starts to operate when the desired path of the vehicle is deviated from the actual path. All these solutions may have a different efficiency and energy consumption. That's because there is the question how different energy demands are for extreme driving situations. The value of the added corrective torque may give a sufficient answer.

2. Vehicle model and reference model

The mathematical formulation of a vehicle and a reference model was given by the following equations described in [1], where β means vehicle slip angle, $\dot{\psi}$ means yaw rate, variables α_f (α_r) represents tire side slip angles front and rear, *Sf* (*Sr*) are lateral tire forces front and

*Corresponding author. Tel.: +420 239 046 977, e-mail: vladislav.drobny@tuv-sud.cz.

rear. Parameter *lp* (*lf*) specifies front resp. rear axle distance from vehicle centre of gravity. The vehicle loaded to a certain mass *m* corners with steering angle δ_f as well as with longitudinal velocity *v*. M_{Corr} correspond with corrective torque used as a control strategy product.

$$\dot{\beta} = \frac{-m\dot{v}\sin\beta + Sf\,\cos\delta_f + Sr}{mv\cos\beta} - \dot{\psi}$$
$$\ddot{\psi} = \frac{Sf\cdot lp\cos\delta_f - Sr\cdot lz + M_{Corr}}{J} \tag{1}$$

$$\alpha_f = \delta_f - \beta - \frac{lp}{v}\dot{\psi}$$
$$\alpha_r = -\beta + \frac{lz}{v}\dot{\psi} \tag{2}$$

3. Control strategy, Corrective torque definition

The main task for the control strategy is to provide the tracking of the reference model outputs as well as minimizing the control deviation. Regulated model output parameters are the yaw velocity and the slip angle of the vehicle. According to the reasons mentioned in [4] there were used both parameters for controlling, with the help of weighting factors, which modify the parameter significance. There could appear undesirable situations if the control algorithm controls the yaw rate and at the same time the slip angle grows. This leads to the fact that vehicle moves outwards from the ideal trajectory. The control strategy receives the impulse to start controlling both parameters and make correction of the slip angle in the desired range. The weighting factors are now switched to another priority.

Fig. 1 describes the scheme of the vehicle control system algorithm where the *State and Friction Estimator* evaluates the yaw velocity together with the slip angle of the vehicle by the transformation of the measured inputs. The *Reference model* evaluates the desired yaw rate and the desired slip angle of the vehicle in terms of driver inputs. The *Control Law and Logic* provides corrections through the appropriate actuators.

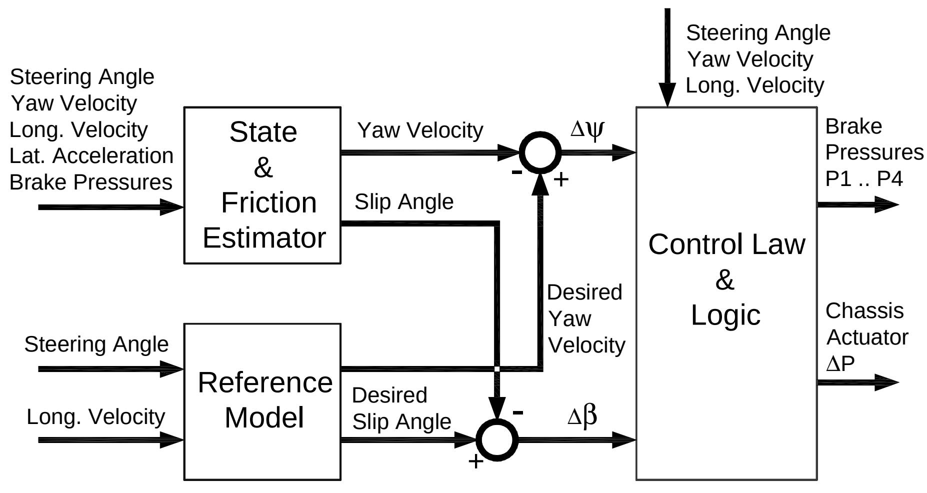

Fig. 1. Control system scheme.

4. Driving maneuvres

The study of driving maneuvres was the first step to choose reduced amount of observed characteristics focused on extremely dangerous driving situations. Avoidance maneuvres as the lane change or the double lane change belong to the worst conditions. Previous studies, which were done at various vehicle models, resulted into following maneuvres sets.

The first one was the circle driving with the constant steering wheel angle together with growing velocity. This quasi-static situation was chosen to compare the other tests coming from the dynamical range.

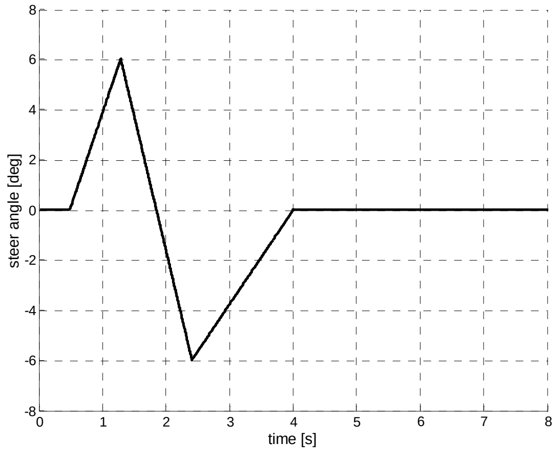

Fig. 2. Lane change.

The second test case (fig. 2) uses steering wheel actuation created by the simplification of the MBS *Lane change* simulation results with the closed loop driver model. In this study, the driver model outputs were considered to be a linear approximated input function.

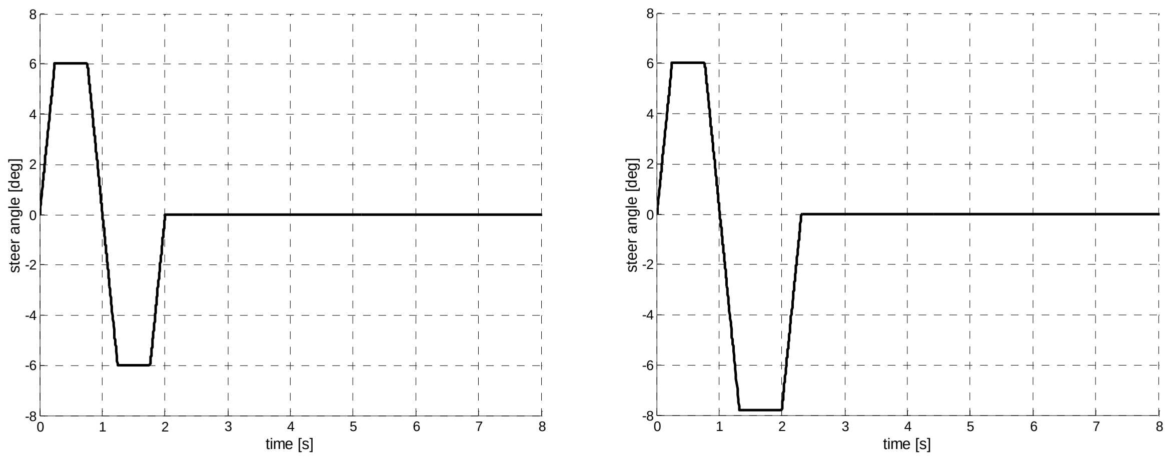

Fig. 3. Yaw acceleration steering reversal (left), Increasing yaw acceleration steering reversal (right).

Other driver inputs borrow from NHTSA experiences described in [2]. They are the *Yaw acceleration steering reversal* and the *Increasing yaw acceleration steering reversal*. Transformations of the desired steering wheel angle time dependency to the front wheel angles provide inputs described on fig. 3.

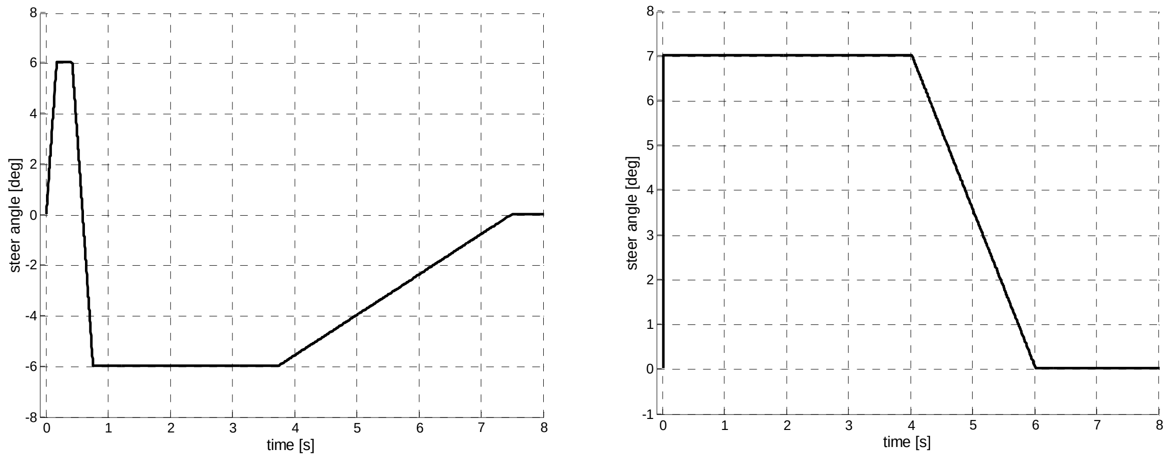

Fig. 4. Fishhook (left), J – Turn (right).

The final observed fig. 4 describes driver inputs. They are driving maneuvres often used for the vehicle stability evaluation as well as for the vehicle rollover prevention testing. The vehicle velocity used for all the dynamical tests was as high as overriding of adhesion limits occurred.

5. Simulation results

Time simulations of a common passenger car with 6 various inputs defined in the previous chapter were realized. The time dependencies of corrective torque were observed (fig. 5, fig. 6, fig. 7). The purpose was to evaluate the peak values needed for the vehicle stabilization and to determine their duration at these limits.

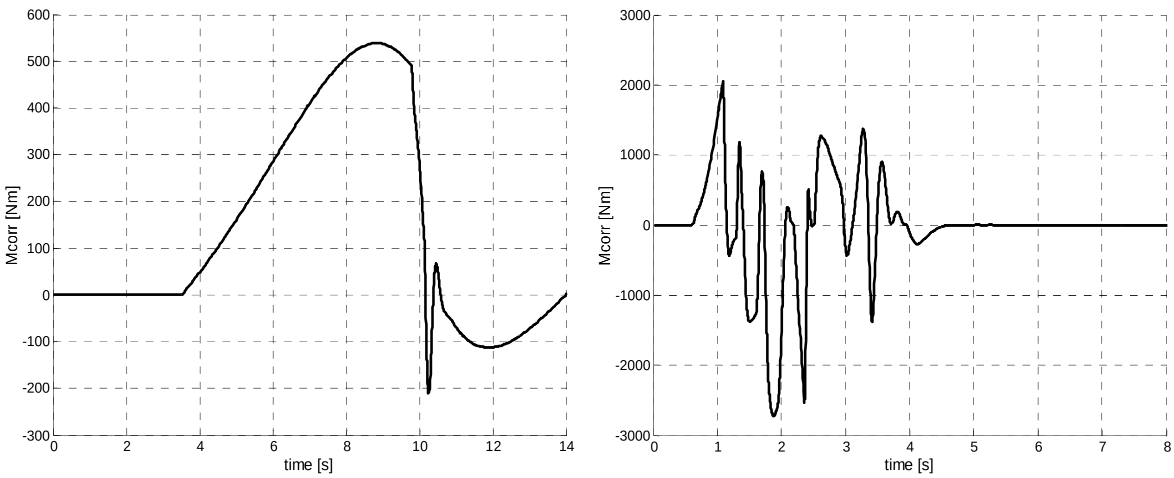

Fig. 5. Circle driving (left), Lane change (right).

The visible changes in the corrective torque time dependencies are given by switching of the weighting factors, where only small deviations from the desired trajectory were allowed.

As for the real vehicle setup there are the characteristics smoother because the time delay in actuators is generated. Partially the smoothing is also caused by the tire contact forces hysteresis.

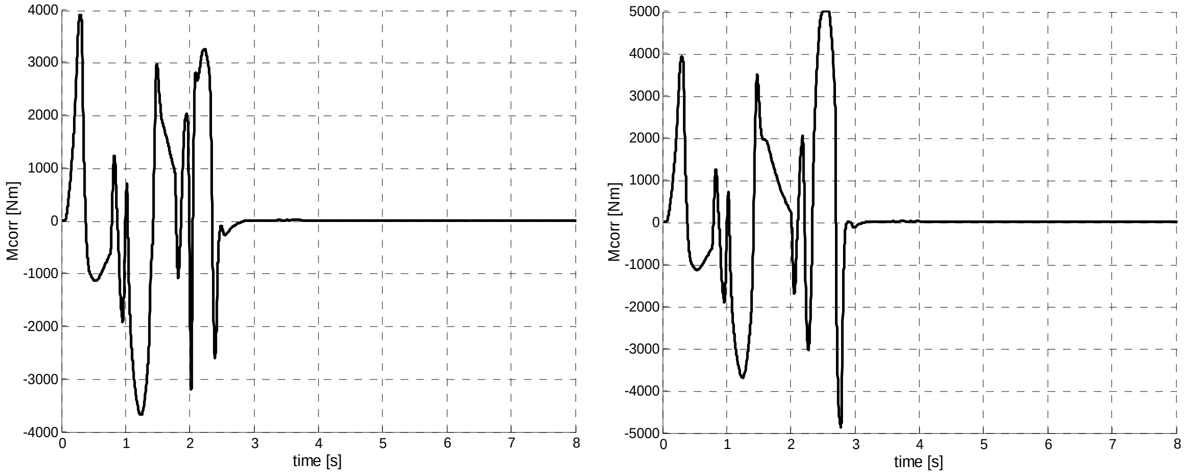

Fig. 6. Yaw acceleration steering reversal (left), Increasing yaw acceleration steering reversal (right).

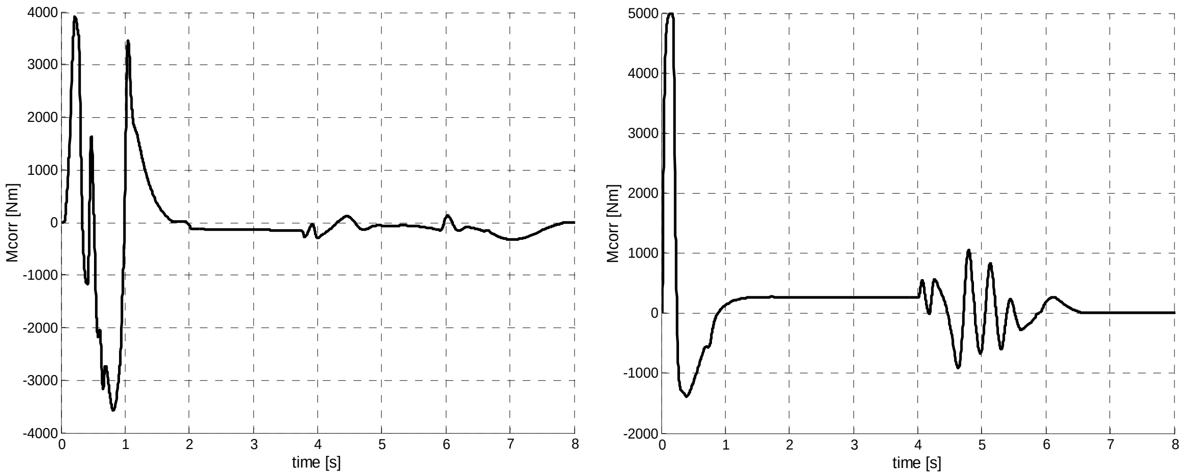

Fig. 7. Fishhook (left), J – Turn (right).

As the result figures show, the peak value of 5000 Nm occurs by the *Increasing yaw acceleration steering reversal* maneuvre and by the *J – Turn* maneuvre. The peak value durations are in size of milliseconds. Compared with quasi-static circle driving maneuvre, the dynamical correction torque needs are approximately 10-times higher. The worst of the chosen maneuvers seems to be the *Increasing yaw acceleration steering reversal* maneuvre where the peak threshold with approximately 0.5 s duration occurs.

The future work based on these results will be focused on finding such a control actuator, which has the ability to supply stabilization forces into suspension in the way to produce the similar corrective torque as mentioned in this paper.

6. Conclusion

This paper presents various driving tests to obtain extreme driving characteristics. Applied control strategy stabilizes the vehicle in the desired path using additional corrective torque. The time dependency of corrective torque was observed to get its peaks and average values, which were subsequently compared with each other. The generated results give us significant information about the additional corrective torque demand for the vehicle stabilization.

Acknowledgement

The work has been supported by the Josef Božek Research Center of Engine and Automotive Engineering II.

References

[1] V. Drobný, Reference model for Vehicle Dynamics Control, Engineering mechanics 2007 Conference, Svratka, 2007.
[2] G. J. Forkenbrock, D. Elsasser, B. O'Harra, NHTSA's Light Vehicle Handling and ESC Effectiveness Research program, International Technical Conference on the Enhanced Safety of Vehicles, 2005
[3] A. Hac, Influence of Active Chassis Systems on Vehicle Propensity to Maneuver-Induced Rollovers, SAE Paper 2002-01-0967, 2002.
[4] A. van Zanten, R. Erhardt, G. Pfaff, VDC, The Vehicle Dynamics Control System of Bosch, SAE Paper 950759, 1995.