Vehicle Lateral Dynamics Stabilization Using Active Suspension

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Abstract

The paper deals with the investigation of active nonlinear suspension control in order to stabilize the lateral vehicle motion in similar way as systems like ESP do. The lateral stabilization of vehicle based on braking forces can be alternatively provided by the different setting of suspension forces. The basis of this control is the nonlinear property of the tyres. The vehicle has at least four wheels and it gives one or more redundant vertical forces that can be used for the different distribution of vertical suspension forces in such a way that resulting lateral and/or longitudinal forces create the required correction moment for lateral dynamic vehicle stabilization.

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1. Introduction

Current vehicles are equipped with many electronic control devices assisting driver in daily and emergency situations. The devices are partially connected to each other by networks. The communication between earlier stand-alone devices will bring added value not only in operational comfort of the vehicles, but particularly in active and passive safety. Among the others, synergetic effects between vehicle controllable suspensions and lateral dynamic vehicle system stability are expected [5, 6]. In other words the lateral stabilization of vehicle, that uses the braking forces in ESP system, can be alternatively provided by the different setting of suspension forces.

The vehicle has at least four wheels and it gives one or more redundant vertical forces in order to comply with balance of gravity and vertical force. This freedom can be used for the different distribution of vertical suspension forces in such a way that resulting lateral and/or longitudinal forces create the required correction moment for lateral dynamic vehicle stabilization.

This has been already studied by semi-active [1] suspension control and proposed for fully-active [6] suspension control. The novelty of this study is the investigation of such lateral dynamic vehicle stabilization using fully active and limited-active suspension control and the investigation of the energy requirements of such control for different manoeuvres and/or emergency situations.

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2. Lateral Dynamics Stabilization

The basis of this control of vehicle lateral stability is the nonlinear property of the tyres. The longitudinal F_x and/or the lateral forces F_y depend on the vertical load F_z, sideslip α and the slip κ: $F_{x,y} = F_{x,y}(F_z, \alpha, \kappa)$ in softening manner. As a result of this nonlinearity, the different distribution of the vertical force results in a production of yaw moment.

The suspension control consists of several subsystems. The first one is the subsystem of vehicle and reference models. The purpose of this subsystem is to detect the difference between the driver manoeuvre intention and real vehicle behaviour. The second one is the subsystem of corrective yaw torque. The purpose of this subsystem is to determine the corrective yaw torque that minimizes the difference between behaviour of the reference vehicle model and the estimated vehicle behaviour. The third one is the subsystem of distribution of vertical suspension forces. The purpose of this subsystem is to determine such distribution of vertical suspension forces that produces the yaw moment nearest to the required one.

3. Reference model

The first subsystem of vehicle and reference models fulfils the difficult task of the estimation of the driver manoeuvre intention. The driver intention is not just an instantaneous vehicle state but a complete vehicle behaviour. The reference model is based on the single track vehicle model (fig. 1). This describes the ideal lateral vehicle behaviour.
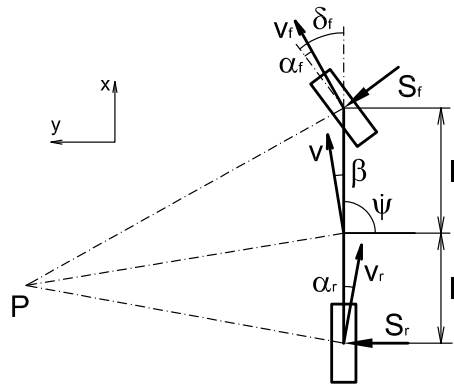

Fig. 1. Reference vehicle model

The mathematical formulation of a vehicle and a reference model was given by the following equations described in [2], where β means vehicle slip angle, $\dot{\psi}$ means yaw rate, variables α_f (α_r) represents tire side slip angles front and rear, S_f (S_r) are lateral tire forces front and rear. Parameter l_f (l_r) 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. Later corresponds with corrective torque used as a control strategy product. The lateral tire forces are described by linear dependencies on slips of tires by the lateral tire stifnesses C_{S1} and C_{S2}.

$$S_f = C_{S1} \cdot \alpha_f, \quad S_r = C_{S2} \cdot \alpha_r \tag{1}$$

where linearized tire side slip angles are

$$\alpha_f = \delta_f - \beta - \frac{l_f}{v}\dot\psi, \quad \alpha_r = -\beta + \frac{l_r}{v}\dot\psi \tag{2}$$

The resulting equations of motion of the vehicle reference model are

$$\dot\beta = \frac{-\beta v\left(C_{S1} + C_{S2}\right) + \dot\psi\left(C_{S2}l_r - C_{S1}l_f - mv^2\right) + vC_{S1}\delta_f}{mv^2} \tag{3}$$

$$\ddot\psi = \frac{\beta v\left(-l_f C_{S1} + l_r C_{S2}\right) - \dot\psi\left(l_f^2 C_{S1} + l_r^2 C_{S2}\right) + \delta_f l_f C_{S1} v}{vJ_z} \tag{4}$$

This model based on the input of velocity and steering angle predicts the ideal behavior of the vehicle.

4. Vehicle estimation model

4.1. Lateral dynamics model

Important part of the control system is the estimator-observer. For this study it is replaced by the following simplified vehicle model. The vehicle model used for the estimation of the vehicle state is based on the two track vehicle model from the fig. 2. Its equations of motion are following:

$$\dot v = \frac{1}{m\cos\beta}\left(mv(\dot\psi + \dot\beta)\sin\beta - S_{FL}\sin\delta_f - S_{FR}\sin\delta_f\right) \tag{5}$$

$$\dot\beta = \frac{-m\dot v\sin\beta + \cos\delta_f\left(S_{FL} + S_{FR}\right) + S_{RL} + S_{RR}}{mv\cos\beta} - \dot\psi \tag{6}$$

$$\ddot\psi = \frac{1}{J_z}[S_{FL}\left(l_f\cos\delta_f + \frac{T}{2}\sin\delta_f\right) + S_{FR}\left(l_f\cos\delta_f - \frac{T}{2}\sin\delta_f\right) - \tag{7}$$
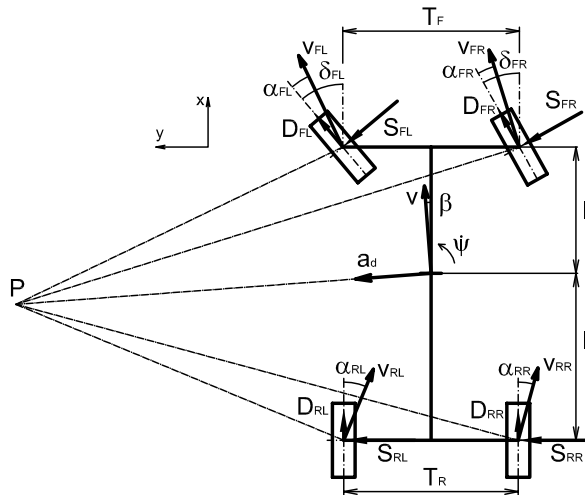$$- S_{RL}l_r - S_{RR}l_r + M_{corr}]$$

Fig. 2. Scheme of the lateral mathematical model of the vehicle

Parameter δ_f means the average value of the front wheels steering angles δ_{FL}, δ_{FR}. Variables S_{FL}, S_{FR}, S_{RL}, S_{RR} are the tire lateral forces, l_f and l_r are the CG (centre of gravity) distances from front and rear axle. J_z means the moment of inertia around the z-axis of the vehicle. As for the parameter T, the simplification which is based on the average value of the front and rear track T_F, T_R was used. The tire lateral forces are computed from the Pacejka magic formula. As a result of the equations (6) and (7), there are evaluated two parameters: the sideslip angle of the vehicle β and the yaw rate $\dot\psi$.

If M_{corr} is used as an input variable, then it has the ability to affect turning of the vehicle in positive or negative way directly. Positive M_{corr} makes the vehicle more oversteered, negative M_{corr} makes the vehicle more understeered. This subject of corrective torque magnitude, that represents the corrective torque demand for vehicle stabilization as an idealized problem, was described in [3]. The direct control of the vehicle using the M_{corr} input variable can be solved from the mathematical point of view only. As for the real vehicle behavior modeling, there rises the certain demand on finding the energy source for vehicle dynamics control in relation to M_{corr}. This paper extends the last obtained results [4, 7] and is focused on the mentioned source of M_{corr} generation using the vertical forces variation with the mathemathical model of the vehicle, which behaves more realistic than the previous ones.

4.2. Vertical forces model

Into the 3DOF lateral dynamic model was added the vertical suspension model of the vehicle chassis. This model contains next 3DOF, that in the result enlarges the whole vehicle model to 6DOF. The added unlocked coordinates are vertical displacement of the chassis z, roll angle γ and inclination angle ε. The final system of three equations (8) was derived using Lagrange's equations and uses among the parametres explained before, the parameters which are used to describe the suspension dynamics. The spring rates front/rear k_f/k_r together with the damping ratios front/rear c_f/c_r affect the dynamical behavior and also the final values of the tire forces compared to the previous studies provided with 3DOF models. There is together with the rolling of the chassis also induced the inclination of the chassis on a turning vehicle. It is caused by x axis component of the side forces acting as a braking forces on front steered tires.

As shows fig. 3, the used parameter h_2 means the distance between the centre of gravity CG and the roll centre P. Parameters J_x and J_y are the moments of inertia around x and y axis.
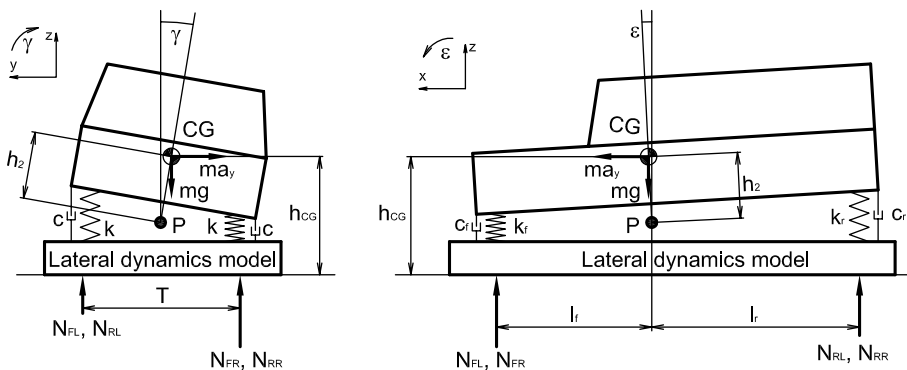

Fig. 3. Vertical forces model

$$m\ddot{z}_p - mh_2 \left(\dot{\gamma^2} + \gamma\ddot{\gamma} + \dot{\varepsilon^2} + \varepsilon\ddot{\varepsilon} \right) + 2k_f \left(z_p - l_f\varepsilon \right) + 2k_r \left(z_p + l_r\varepsilon \right)$$
$$+2c_f \left(\dot{z}_p - l_f\dot{\varepsilon} \right) + 2c_r \left(\dot{z}_p + l_r\dot{\varepsilon} \right) = 0$$
$$\left(J_x + mh_2{}^2 \right) \ddot{\gamma} + mh_2{}^2\gamma \left(\dot{\varepsilon^2} + \varepsilon\ddot{\varepsilon} \right) - mh_2\gamma\ddot{z}_p + \frac{T^2}{2}\gamma \left(k_f + k_r \right)$$
$$-mgh_2\gamma + \frac{T^2}{2}\dot{\gamma} \left(c_f + c_r \right) = 0 \tag{8}$$
$$\left(J_y + mh_2{}^2 \right) \ddot{\varepsilon} + mh_2{}^2\varepsilon \left(\dot{\gamma^2} + \gamma\ddot{\gamma} \right) - mh_2\varepsilon\ddot{z}_p - 2z_p \left(k_f l_f - k_r l_r \right)$$
$$+2\varepsilon \left(k_f l_f{}^2 + k_r l_r{}^2 \right) - mgh_2\varepsilon - 2\dot{z}_p \left(c_f l_f - c_r l_r \right) + 2\dot{\varepsilon} \left(c_f l_f{}^2 + c_r l_r{}^2 \right) = 0 \,.$$

5. Control strategy and corrective torque computation

The second subsystem computes the corrective yaw torque. The vehicle model has two output parameters — the yaw velocity and the slip angle. These parameters possess different importance during the vehicle motion. Therefore the weighting factor that modifies the parameter significance is used.

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. 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. 4 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.
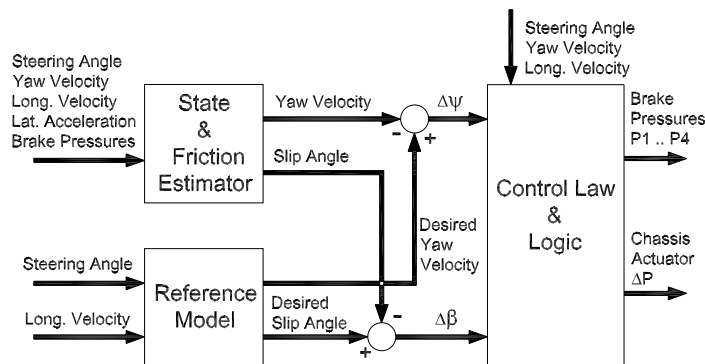

Fig. 4. Control system scheme

6. Vertical forces variation

The Control Law and Logic of the presented stability control system is based on the controlling of the lateral tire forces transfer among the four corners of the vehicle. The tire vertical load affects longitudinal and lateral forces, which are acting on the tire. By increasing the vertical tire force, the lateral force increases nonlinearly with apparent degradation at high vertical forces. The lateral force dependency generated by using Pacejka Magic Formula is represented on fig. 5.
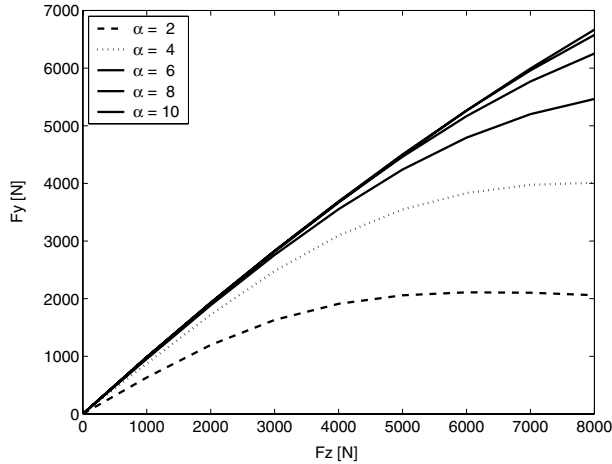

Fig. 5. Lateral tire force F_y over wheel load F_z at different slip angles

With the usage of the lateral force degradation on the appropriate tire, it is possible to generate the yaw stabilization movement of the vehicle. Described situation occurs at the vertical force lowering. On the other side, by increasing the tire vertical force, the response leads to enlarging the tire lateral force. The resulted dynamic effect of the 4-wheel vertical force variations is sufficient to be described by parameter M_{corr}. It characterizes the stabilization demand and among others it allows to make the comparison to the passive vehicle.

By the formulation of this problem there was expected, that the weight distribution among the 4 wheels did not cause any distinctive movement in vehicle roll and pitch. In the ideal case the driver wouldn't be able to recognize any change of the vehicle's behavior. Development of the expressed problematic led to the system of four equations. The first equation is the moment equilibrium written with respect to right side, the second equation is the moment equilibrium written with respect to rear axle, the third is the sum of vertical forces, the fourth equation expresses the equality of the change of vertical forces on front and rear axles.

$$\begin{bmatrix} 1 & 0 & 1 & 0 \\ 1 & 1 & 0 & 0 \\ 1 & 1 & 1 & 1 \\ 0 & 0 & 0 & 1 \end{bmatrix} \cdot \begin{bmatrix} N_{FL} \\ N_{FR} \\ N_{RL} \\ N_{RL} \end{bmatrix} = \begin{bmatrix} -\frac{h_{CG}ma_y \cos\beta}{T} + \frac{mg}{2} \\ \frac{mgl_r}{l_f+l_r} \\ mg \\ \left(\frac{h_{CG}ma_y \cos\beta}{T} + \frac{mg}{2}\right) \cdot \left(1 - \frac{l_r}{l_f+l_r}\right) + dN \end{bmatrix}$$

The variable dN represents one input parameter. Solving the equation for specific dN results to the actual values of the vertical forces N_{FL}, N_{FR}, N_{RL}, N_{RR}. Defined formula was

integrated to the vehicle control system as the actuator and was connected as a part of the feed-back control loop, which compares the movement of the vehicle with the movement of the reference model.

7. Stabilization process

Simulation outputs of the controlled vehicle model were analyzed. The correlation between dN and M_{corr} was determined. The corrective torque is generated in positive direction, if the vertical load at inner front and outer rear wheel increases and at the same time the outer front and inner rear tire vertical force decreases. The vertical force shift dN has the same absolute value for all wheels. The difference consists in the force direction. The final conclusions for positive and for negative corrective torque generation are shown on fig. 6. Plus sign on the appropriate wheel means the increase of the vertical tire force and the minus sign means the decrease.

The limitation of the control system is given by the value of static load of the appropriate tire. The controlled lateral movement of the vehicle keeps as stable as the potential to transfer the lateral forces is sufficient. By wheel lift is the effectivity of such controlled system reduced.
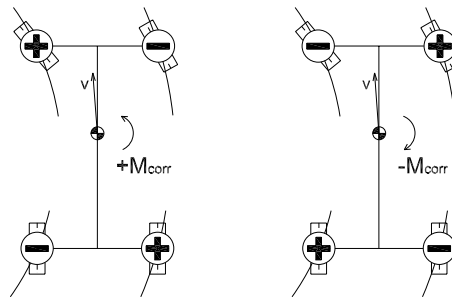

Fig. 6. Vertical tire force shift (dN) and corrective torque (M_{corr}) correlation

8. Equivalent torque

The corrective torque M_{corr}, which is the product of the vertical forces' change, could not be simply measured. There was used a approximative method to obtain the value of corrective yaw torque. It was assumed, that the state parameters (β, $\dot{\beta}$, $\dot{\psi}$, $\ddot{\psi}$, v_x) observed from movement of the controlled vehicle will be the same for passive vehicle. That means that passive vehicle should move along the same trajectory as the controlled vehicle. For these input state parameters were in the case of passive vehicle model computed suspension and tire forces. The equations of the moment equilibrium $\ddot{\psi} J_z = \Sigma M_z$ for both passive and controlled models were compared and the appeared difference produces the instantaneous corrective torque M_{corr}.

9. Driving maneouvers

The study of driving maneuvres was the first step to choose reduced amount of observed char-acteristics 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.
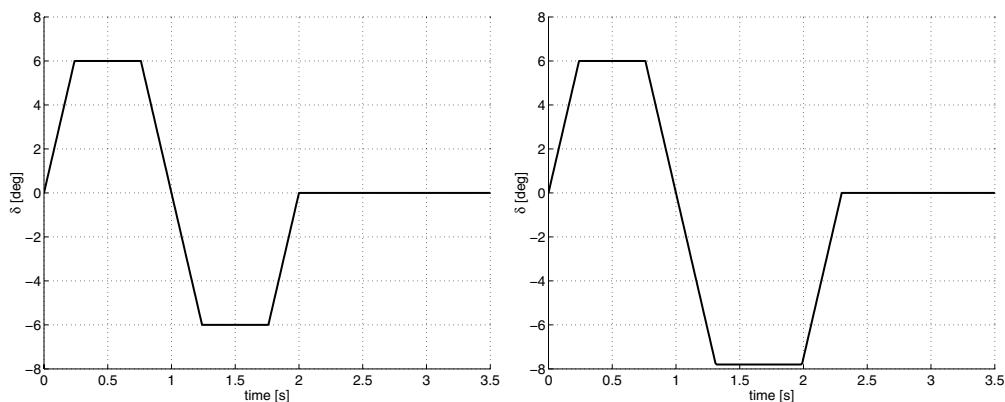
Fig. 7. Yaw acceleration steering reversal (left), Increasing yaw acceleration steering reversal (right)

The following results are based on the driver inputs borrowed from NHTSA experiences described in [5]. 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. 7. These inputs were combined with the increase of the velocity for defined adhession coefficient μ and were simulated for the passive vehicle. If the increase of the velocity caused, that the vehicle slip angle crossed the treshold of $\beta = 20 \ deg$ and induced loosing of the vehicle stability, then this velocity was declared as the critical velocity. This critical velocity together with the inputs on fig. 7 were used for the excitation of the active vehicle with the lateral dynamic stability control.

10. Simulation results

The results of two selected maneuvres for the wet surface ($\mu = 0.7$) and for the dry surface ($\mu = 1.0$) are shown on fig. 8 and on fig. 9. Comparison to the passive vehicle demonstrates the significant improvement for the stability behavior of the vehicle using controlled system. In the situation, where passive vehicle looses stability and slips out from the desired track, the controlled vehicle remains stable. As well as the observed parameters β and $\dot{\psi}$, the demanded corrective torque M_{corr} was evaluated. The M_{corr} time dependencies of followed maneuvers are collected on fig. 10. The gained maximal peak values about $3\,000$ Nm are sufficient for vehicle stabilization.

11. Conclusion

The clear dependency was found between the corrective torque of the vehicle and the tire vertical forces redistribution among four corners of the vehicle. The results give to the dynamic body control system the opportunity to control the lateral dynamics of the vehicle. The limitation of the system is caused by the contact of the tire. The maximal actuated force dN should be less then the tire static load force.

The efficiency of the mentioned control system will be changed, if the longitudinal driving or braking tire forces will be acting to the mathematical model of the vehicle. Driving and braking forces cause destabilization of the vehicle lateral movement. Compared to the presented model it will occur at lower vehicle speeds where the potential for stabilizing is higher. It is an object
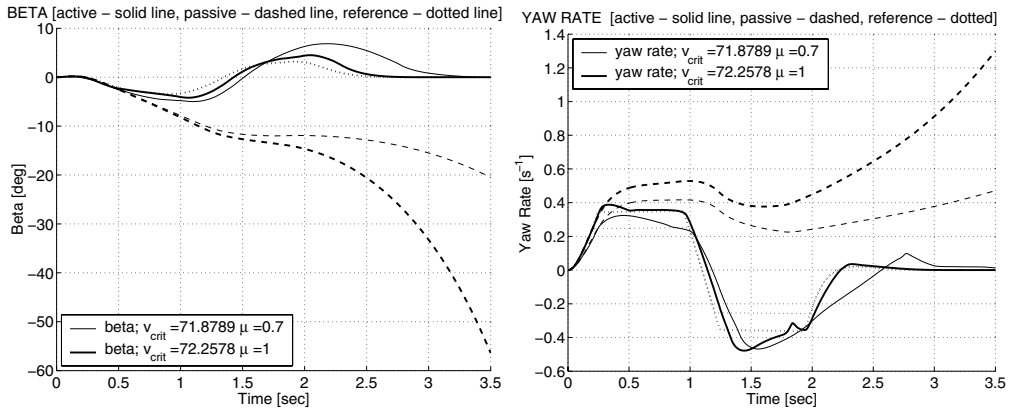
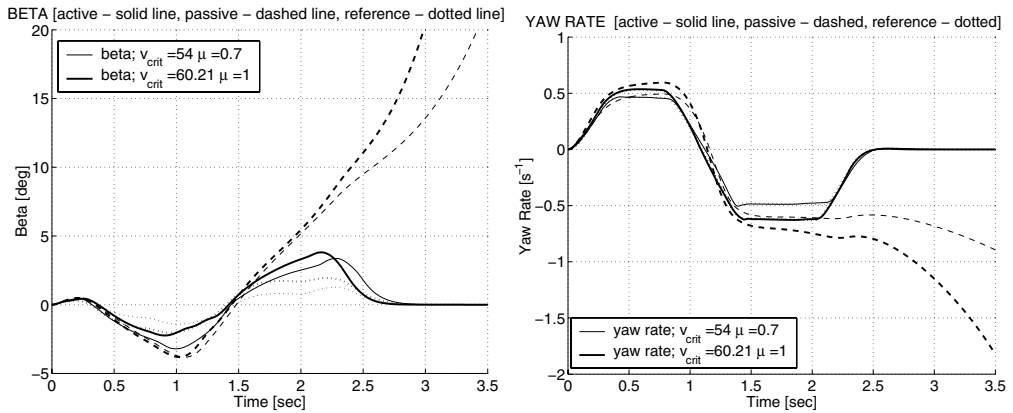Fig. 8. Yaw acceleration steering reversal simulation results

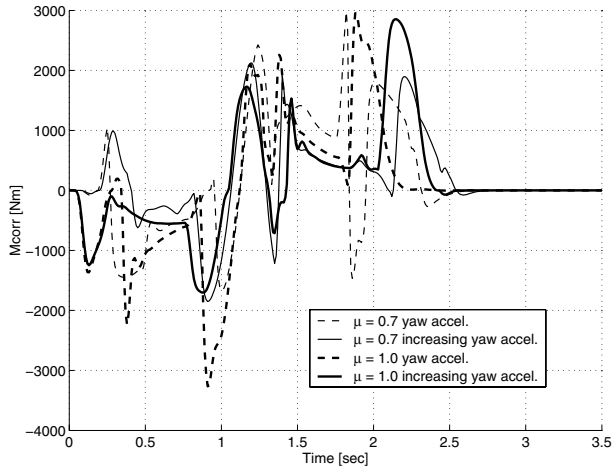Fig. 9. Increasing yaw acceleration steering reversal simulation results

Fig. 10. Required corrective torque for vehicle stabilisation

for the future work. The simulation results indicate that the usual maneouvers can be corrected using limited active suspension.

Another topic for future work is the investigation of the energy demand for the limited-active suspension control and the influence of energy source limitation on the stabilization influences.

Acknowledgements

References

[1] M. O. Bodie, A. Hac, Closed Loop Yaw Control of Vehicles Using Magneto-Rheological Dampers, SAE Paper 2000–01–0107, 2000.

[2] V. Drobný, Reference Model for Dynamics Control, In: Proc. of Engineering Mechanics 2007, Svratka 2007, CD ROM, pp. 1–7.

[3] V. Drobný, M. Valášek, Vehicle Lateral Dynamics Control, Applied and Computational Mechanics. vol. 1, no. 2, 2007, p. 411–416.

[4] V. Drobný, M. Valášek, Wheel Vertical Forces Distribution as the Source of Vehicle Stabilization, In: Proc. of Engineering Mechanics 2008, Svratka 2008, CD ROM, pp. 1–5.

[5] G. J. Forkenbrock, D. Elsasser, B. O'Harra, NHTSA's Light Vehicle Handling and ESC Effectiveness Research program, In: Proc. of International Technical Conference on the Enhanced Safety of Vehicles, 2005.

[6] M. Rau, D. Ammon, Vernetzung von aktiven Fahrwerk-Regelsystemen am Beispiel Aktive Federung (ABC) und Fahrdynamik-Regelung (ESP), In: H. Wallentowitz, (ed.): Proc. of 5. Tag des Fahrwerks 09. Oktober 2006, IKA RWTH, Aachen, 2006, pp. 5/1–16.

[7] M. Valášek, V. Drobný, O. Vaculín, Vehicle Lateral Dynamic Stability by Limited-Active Nonlinear Suspension Control, 9th International Symposium on Advanced Vehicle Control AVEC'08, Japan 2008 (accepted).

[8] A. van Zanten, R. Erhardt, G. Pfaff, VDC, The Vehicle Dynamics Control System of Bosch, SAE Paper 950759, 1995.