

Integrated Control of Active Steering and Electronic Differentials in Four Wheel Drive Vehicles

Riccardo Marino and Stefano Scalzi

University of Rome Tor Vergata

Copyright © 2009 SAE International

ABSTRACT

In this paper the integrated control of front and rear active differentials with active front steering is investigated in order to improve dynamics, stability and to reduce the drawbacks of mechanical self-locking differentials. The proposed integrated centralized control feeds back both the yaw rate and the wheel speed measurements to the control inputs which are the front wheel steering angle and the torque transferred by the electronic differentials between the left and the right wheels of both vehicle's axles. The control of the electronic differentials is not only aimed at keeping the wheel speed differences at desired values but it is also integrated with the active steering control (a PI action from the yaw rate error) to produce a yaw moment (also depending on the yaw rate error) which improves handling and stability. Several simulations are carried out on a CarSim small SUV model to explore the robustness with respect to unmodelled dynamics such as pitch, roll and nonlinear combined lateral and longitudinal tire forces. In response to sudden direction changes the simulations show reduced oscillations, robustness and enlarged bandwidth for the yaw rate tracking dynamics while new stable maneuvers are allowed especially in critical conditions such as μ -split braking.

INTRODUCTION

Vehicle yaw rate and lateral velocity dynamics may show a coupled oscillatory behaviour which changes significantly and may become unstable depending on

many factors such as road and tire conditions, speed, vertical load and driver inputs. When oscillations appear, the driver has the difficult task of controlling the vehicle by means of several countersteerings. Several feedback systems have been proposed and implemented to increase vehicle stability and safety including active front and rear steering, active suspensions, selective braking actions to generate stabilizing yaw torques, electronic differentials. More recently research is focused on the integration of such feedback systems into a centralized control system. Several front steering control laws are proposed as in [1], [2], [3] and implemented on vehicles or on steer by wire prototypes. In [1] it is shown that the lateral acceleration of the front axle may be robustly triangularly decoupled from the yaw rate dynamics, using only the front wheel steering angle as a control input, by feeding back the yaw rate error through an integrator. In [2] a PI active steering control on the yaw rate tracking error with different gains for braked and unbraked driving conditions is used. In [3] the active front steering is designed in order to ensure safety also during system failures: in fact, the wheel steering angle δ_f is the sum of the designed control signal δ_c and of the driver input δ_p . The development of electro actuated differentials allows for new control strategies in vehicle dynamics control as in [4], [5], [6], [7] and [8]. In [6] the semiactive electronic control system can determine the torque transferred by the differential but not its direction; the torque is transferred from the fastest wheel to the slowest one; the control operates when the rear wheel speed difference exceeds a given threshold and it is designed as a proportional-integral control law based on the error between the measured and the desired rear wheel speed angular velocity. In [7] the controller is designed following

the Internal Model Control approach and it is active since it can generate yaw moments of every amount and direction. In [8] the rear semiactive differential is electronically controlled and integrated with front active steering according to a Lyapunov analysis. In [9] the active sport differential is introduced: a regulating system distributes torque from the engine in continuously variable proportions between the rear wheels. The controlled power flow enables the car to take corners more directly and responsively. Many papers are currently focused on the design of integrated global chassis control systems: in [10] an integrated control of active front steering and direct yaw moment generated by a distribution of braking forces is designed; in [11] the electronic stability program is integrated with the active front wheel steering, the active suspension and an active anti roll bar; in [12] four wheel steering is coordinated with wheel torque distribution using an optimization approach; a non linear optimization approach is followed by [13] to determine the optimal forces to be exerted by the tires which are controlled by active steering and brake pressures distribution; in [14] the global chassis control optimization is obtained following a gradient-based optimal control algorithm approach: the active rear steering and the active rear differential are used as actuators.

In this paper the front and rear active electronically controlled differentials and an active front steering control are designed and integrated: the goal is to improve vehicle dynamics, by suppressing resonances and enlarging the bandwidth for the yaw rate tracking dynamics so that the driver effort is reduced, and to improve stability. Moreover the control law on the electronically controlled differentials is designed in order to reduce the drawbacks of mechanical self-locking differential. The proposed integrated centralized control feeds back both the yaw rate and the wheel speed measurements to the control inputs which are the front wheel steering angle and the torque transferred by the electronic differentials between the left and the right wheels of both the vehicle's axles. The control of the electronic active differentials is not only aimed at keeping the wheel speed differences at desired values but it is also integrated with the active steering control (a PI action from the yaw rate error) to produce a yaw moment (also depending on the yaw rate error) which improves the vehicle steering dynamics: the eigenvalues are placed to be all real at every speed when parameters have nominal values. The yaw moment is provided by the front and/or the rear differentials control action on the basis of a repartition parameter which assigns the percentage of yaw moment to be generated by front and rear wheels. The robustness with respect to tire parameters such as the cornering stiffnesses, the vehicle mass and the vehicle moment of inertia is analyzed by means of the eigenvalues displacement: the perturbed controlled vehicle has real modes or complex modes with a damping coefficient greater than the uncontrolled vehicle. Realistic simulations are carried out on a CarSim small SUV model which takes into account roll, pitch, driver dynamics and combined lateral and longitudinal

tire forces which were neglected at the control design stage. In response to sudden direction changes the simulations show reduced oscillations, robustness, enlarged bandwidth for the yaw rate dynamics and new stable maneuvers especially in critical conditions. In a μ -split brake maneuver, when one side of the vehicle is on a low adherence surface, the proposed integrated control shows new stable maneuvers with improved performance in comparison with both an uncontrolled vehicle and a vehicle equipped with self locking mechanical differentials and the active steering.

VEHICLE MODELS

A detailed standard CarSim small SUV model is used in numerical simulations to analyze the responses of both the uncontrolled and the controlled vehicle. CarSim vehicle model use nonlinear tire according to combined sideslip theory [15], nonlinear spring models, and incorporates the major kinematics and compliance effects in the suspensions and steering systems including differential load transfer for each wheel. However to capture the essential vehicle steering dynamics and to design the controller a simplified non linear seventh order model and its linear approximation about uniform rectilinear motion is used; the approximated linear system can be written as in [8] in the following state space form:

$$\dot{z} = Az + Bu \quad (1)$$

with:

$$A = \begin{bmatrix} a_{44} & a_{45} & 0 & 0 \\ a_{54} & a_{55} & a_{56} & a_{57} \\ 0 & a_{65} & a_{66} & 0 \\ 0 & a_{75} & 0 & a_{77} \end{bmatrix}; \quad z = \begin{bmatrix} v_y \\ r \\ z_3 \\ z_4 \end{bmatrix};$$

$$B = \begin{bmatrix} b_{41} & 0 & 0 \\ b_{51} & 0 & 0 \\ 0 & b_{62} & 0 \\ 0 & 0 & b_{73} \end{bmatrix}; \quad u = \begin{bmatrix} \delta_f & T_{d1} & T_{d2} \end{bmatrix}^T$$

where: v_y is the vehicle lateral velocity, r is the vehicle yaw rate, z_3 is the speed difference between the front left wheel and the front right wheel, z_4 is the speed difference between the rear left wheel and the rear right wheel; the control inputs are: the front steering angle δ_f , the transferred torque between the left and the right wheel on the front axle T_{d1} and the transferred torque between the left and the right wheel on the rear axle T_{d2} . The entries in the matrices A and B , whose values are identified from a small SUV CarSim model in the simulation paragraph, depend on the vehicle physical

parameters and on the vehicle velocity. The A and B entries and their values are given in the Appendix.

INTEGRATED CONTROL DESIGN

The integrated feedback control law acting on the front steering angle and on the active front and rear differentials is presented in this section. Considering the first two equations of the linear vehicle model (1) the following equation can be written:

$$\begin{bmatrix} \dot{v}_y \\ \dot{r} \end{bmatrix} = \begin{bmatrix} a_{44} & a_{45} \\ a_{54} & a_{55} \end{bmatrix} \begin{bmatrix} v_y \\ r \end{bmatrix} + \begin{bmatrix} 0 \\ a_{56} \end{bmatrix} z_3 + \begin{bmatrix} 0 \\ a_{57} \end{bmatrix} z_4 + \begin{bmatrix} b_{41} \\ b_{51} \end{bmatrix} \delta_f$$

in which the front and the rear wheel speed difference z_3 and z_4 can be viewed as control inputs to generate a yaw moment in addition to the active front steering action. The designed active front steering control for δ_f is a proportional-integral (PI) controller as in [2]; the PI feedback control algorithm can be written as follows (K_{pf} and K_{if} are positive control parameters):

$$\begin{cases} \delta_f = -K_{pf}\tilde{r} - K_{if} \int_0^t r(\tau) d\tau = -K_{pf}\tilde{r} - K_{if}\alpha_0 \\ \dot{\alpha}_0 = \tilde{r} \end{cases} \quad (2)$$

where: $\tilde{r} = r - r_d$ and r_d is the desired yaw rate constant reference. Define the variables $\tilde{z}_3$ and $\tilde{z}_4$ as follows:

$$\begin{aligned} \tilde{z}_3 &= z_3 - z_3^* - z_{3d} \\ \tilde{z}_4 &= z_4 - z_4^* - z_{4d} \end{aligned}$$

with

$$\begin{aligned} z_3^* &= \gamma K_{pMz} (r - r_d) \\ z_4^* &= (1 - \gamma) K_{pMz} (r - r_d) \end{aligned} \quad (3)$$

in which γ , $0 \leq \gamma \leq 1$ is a repartition parameter to assign the percentage of yaw moment to be generated by front and rear wheels and z_{3d} and z_{4d} are the constant desired reference signals for the front and rear wheel speed differences. The error dynamics for the new state space variables $\tilde{z}_3$ and $\tilde{z}_4$ are:

$$\begin{aligned} \dot{\tilde{z}}_3 &= a_{65}r + a_{66}\tilde{z}_3 + a_{66}z_3^* + a_{66}z_{3d} + b_{62}T_{d1} - \dot{z}_3^* \\ \dot{\tilde{z}}_4 &= a_{75}r + a_{77}\tilde{z}_4 + a_{77}z_4^* + a_{77}z_{4d} + b_{73}T_{d2} - \dot{z}_4^* \end{aligned} \quad (4)$$

in which, from (3), $\dot{z}_3^*$ and $\dot{z}_4^*$ are equal to:

$$\begin{aligned} \dot{z}_3^* &= \frac{\partial z_3^*}{\partial r} \frac{\partial r}{\partial t} = \gamma K_{pMz} \dot{r} \\ \dot{z}_4^* &= \frac{\partial z_4^*}{\partial r} \frac{\partial r}{\partial t} = (1 - \gamma) K_{pMz} \dot{r} \end{aligned} \quad (5)$$

Substituting (5) in (4) the error dynamics for the front and rear wheel speed differences are:

$$\begin{aligned} \dot{\tilde{z}}_3 &= a_{65}r + a_{66}\tilde{z}_3 + a_{66}z_3^* + a_{66}z_{3d} + b_{62}T_{d1} \\ &\quad - \gamma K_{pMz} (a_{54}v_y + a_{55}r + a_{56}z_3 + a_{57}z_4 + b_{51}\delta_f) \\ \dot{\tilde{z}}_4 &= a_{75}r + a_{77}\tilde{z}_4 + a_{77}z_4^* + a_{77}z_{4d} + b_{73}T_{d2} \\ &\quad - (1 - \gamma) K_{pMz} (a_{54}v_y + a_{55}r + a_{56}z_3 + a_{57}z_4 + b_{51}\delta_f). \end{aligned}$$

The active front and rear differentials controls are defined as follows:

$$\begin{cases} T_{d1} = -\frac{1}{b_{62}} [a_{66}z_3^* - \gamma K_{pMz} (a_{55}r + a_{56}z_3 + a_{57}z_4 + b_{51}\delta_f)] - K_{p1}\tilde{z}_3 - K_{i1}\alpha_1 \\ T_{d2} = -\frac{1}{b_{73}} [a_{77}z_4^* - (1 - \gamma) K_{pMz} (a_{55}r + a_{56}z_3 + a_{57}z_4 + b_{51}\delta_f)] - K_{p2}\tilde{z}_4 - K_{i2}\alpha_2 \\ \dot{\alpha}_1 = \tilde{z}_3 \\ \dot{\alpha}_2 = \tilde{z}_4 \end{cases} \quad (6)$$

so that the controlled linear system (1,2,6) can be written in the state space form

$$\dot{x}_c = A_c x_c + B_c u_c \quad (7)$$

in which A_c , x_c , u_c and B_c are the following:

$$A_c = \begin{bmatrix} A_{c11} & A_{c12} \\ A_{c21} & A_{c22} \end{bmatrix} =$$