Direct Torque Control for Electronic Differential in an Electric Racing Car

C. Fu, R. Hoseinnezhad*, Simon Watkins, and Reza Jazar

School of Aerospace, Mechanical and Manufacturing Engineering, RMIT University, Victoria 3083, Australia
rezah@rmit.edu.au

Abstract. This paper presents a new method for development of electronic differentials for electric racing vehicles. Most electronic differential solutions focus on maintaining the vehicle stability as the first and dominant priority, and are designed to keep some stability-related quantity (e.g. wheel slip) in a "safe region". With racing cars however, the main focus is on the responsiveness of the vehicle and its capability to cope with extreme steering and accelerating demands from the driver. Our focus is on designing a controller to achieve neutral-steer (avoiding over- or under-steer) in race car driving conditions. We show a direct relationship between the steering condition and the difference of the longitudinal tire-road friction forces for the driven wheels. We mathematically derive the desired difference in the tire-road frictions that would achieve neutral-steer and show that it is directly related to the difference in the driving torques provided by motors. A closed-loop-control system is proposed for direct control of the motor torques. The simulation results show a close-to-neutral steering performance of the car (while maintaining its stability) in challenging steering scenarios.

1 Introduction

The two major problems caused by internal combustion engine vehicles, energy scarcity and environment contamination, have become public concerns already. Measures are being taken to address these problems, and one of the main focuses of current research and development trends in sustainable automotive technologies is on developing fully electric vehicles.

In electric cars, each driven wheel is individually actuated by an electric motor, which makes it possible to employ an electronic differential instead of the heavy mechanical differential and to benefit from the swift response time of the electric motors. An electronic differential is a torque and wheel speed controller for managing multiple drives. It supervises the distribution of torques and speeds between driven wheels in accordance with the state of the vehicle and the driver's commands. Therefore, the stability and dynamic performance of the car can be

* Corresponding author, Tel: +61-3-9925-6135.

enhanced by fine tuning of the difference between the torques applied on driven wheels. With a conventional mechanical differential, the stability and performance merits of such a differential torque are implausible.

Most electronic differential solutions, reported so far in the vehicular technology literature, focus on maintaining the vehicle stability as the first and dominant priority, and design to keep some stability-related quantity (e.g. wheel slip) in a "safe region". With racing cars however, the priorities are different. Here, the main focus is on the responsiveness of the vehicle and its capability to cope with extreme steering and accelerating demands from the driver. Vehicle stability can be pushed to its margins (yet maintained) and slips can go beyond the "safe region" levels usually applied with passenger car design.

The easiest method to design an electronic differential is the "equal torque strategy" proposed by Guillermo A. Magallan [1]. The principle of this method is to emulate the behavior of a mechanical differential. Thus, this method always applies even torque to both driven wheels for all vehicle maneuvers. To achieve a better design, many reported electronic differential strategies utilize Ackerman condition to calculate the desired angular speed for each driven wheel. Ackerman condition is a kinematic relation between the inner and outer wheels, allowing them to turn slip free [2]. The "zero slip" condition that is presumed necessary by Ackerman condition is not always the optimal choice in terms of achieving the best vehicle maneuverability. In high performance vehicle dynamic arrangements, some level of wheel slip is allowed to achieve maximum traction while maintaining vehicle stability. For example, in the strategy proposed by Jeongmin Kim and Hyunsoo Kim [3], control commands are generated to produce a desired yaw rate and a desired lateral acceleration which are computed from a vehicle planar dynamic model. In their method, wheel slip explicitly contributes to the control commands generated. Other examples of control methods include an independent motor control strategy using fuzzy logic [4], and an independent motor control method using sliding mode control and a vehicle state observer [5].

In this paper, a closed-loop control system is proposed for direct control of the motor torques. The loop is closed using observers that estimate the tire-road friction and the error signal is the distance between the desired and actual (estimated) values of the difference between the tire-road frictions for the driven wheels. The simulation results show a close-to-neutral steering performance of the car (while maintaining its stability) in challenging steering scenarios.

2 Electronic Differential via Direct Torque Control

Consider a rear-driven electric car with a local coordinate frame attached to the vehicle at its center of mass, as shown in Fig. 1. We will show that the reaction forces exerted on the two rear driven wheels, F_{x3} and F_{x4}, are directly related to the steering performance of the vehicle, and can be tuned to achieve neutral-steering. Our method is based on tuning F_{x3} and F_{x4} to attain their desired values via controlling the driving torques produced by the two motors.

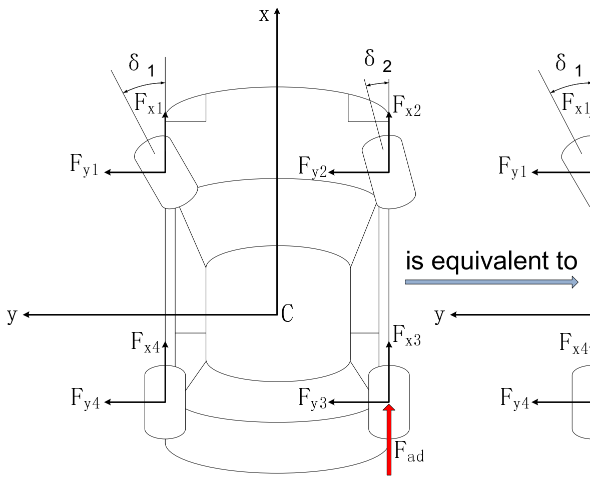

Fig. 1. Top view of the local coordinate frame and the force system acting on the vehicle

Let us assume that, in the first place, both reaction forces F_{x3} and F_{x4} are equal. Then we have the following set of equations of motion:

$$\sum F_x = m\dot{v}_x - mrv_y\,;\sum F_y = m\dot{v}_y + mrv_x;\sum M_z = I_z\dot{r}\,;\sum M_x = I_x\dot{p} \qquad (1)$$

where m, v_x, v_y, I_z and I_x denote the vehicle mass, the longitudinal velocity, the lateral velocity, the yaw moment of inertia and the roll moment of inertia respectively. Expanding the left-hand side of the last three equations, we get the complete version of the equations of motion that govern the lateral, the yaw and the roll motion of the car.

$$\begin{aligned} C_r r + C_p p + C_\beta \beta + C_\varphi \varphi + C_\delta \delta &= m\dot{v}_y + mrv_x \\ E_r r + E_p p + E_\beta \beta + E_\varphi \varphi + E_\delta \delta &= I_z \dot{r} \\ D_r r + D_p p + D_\beta \beta + D_\varphi \varphi + D_\delta \delta &= I_x \dot{p} \end{aligned} \qquad (2)$$

where r, p, β, φ and δ represent the yaw rate, the roll rate, the vehicle sideslip angle, the roll angle and the cot-average steering angle of the front wheels ($\cot\delta = (\cot\delta_1 + \cot\delta_2)/2$) respectively. The coefficients are explicitly expressed in [2].

When we have electronic differential on-board, we are able to send different torque commands to the two driving motors, so the reaction forces F_{x3} and F_{x4} can be different. Let's define $F_{ad} = F_{x3} - F_{x4}$, then an additional moment $M_{ad} = F_{ad} \times w/2$ (w is the rear track) is applied on the rear axle, as indicated in Fig. 1. Therefore, the equations of motion governing the lateral, the yaw and the roll motion – equation (2) – can be modified as follows:

$$\begin{aligned} &C_r r + C_p p + C_\beta \beta + C_\varphi \varphi + C_\delta \delta = m\dot{v}_y + mrv_x \\ &E_r r + E_p p + E_\beta \beta + E_\varphi \varphi + E_\delta \delta = I_z \dot{r} \\ &D_r r + D_p p + D_\beta \beta + D_\varphi \varphi + D_\delta \delta + \frac{w}{2} \cdot F_{ad} = I_x \dot{p} \end{aligned} \tag{3}$$

In order to evaluate the cornering performance of the vehicle, we analyze its steady-state response to steering commands δ and the extra tire forces F_{ad} induced by an electronic differential. This will simplify equations (3) via replacing all time-derivatives with zero which leads to:

$$\begin{bmatrix} C_\beta & C_r - mv_x & C_\varphi \\ E_\beta & E_r & E_\varphi \\ D_\beta & D_r & D_\varphi \end{bmatrix} \begin{bmatrix} \beta \\ r \\ \varphi \end{bmatrix} = \begin{bmatrix} -C_\delta & 0 \\ -E_\delta & 0 \\ -D_\delta & -\frac{w}{2} \end{bmatrix} \begin{bmatrix} \delta \\ F_{ad} \end{bmatrix} \tag{4}$$

Solving the above system of equations, the following yaw rate response is derived in terms of the control inputs δ and F_{ad}:

$$r = \frac{Z_{\delta r}}{Z_0} \delta + \frac{Z_{Fr}}{Z_0} F_{ad} \tag{5}$$

$$\begin{aligned} Z_{Fr} &= \frac{w}{2}(E_\beta C_\phi - E_\phi C_\beta) \\ Z_0 &= E_\beta (D_\phi C_r - D_r C_\phi - mv_x D_\phi) + E_\phi (D_r C_\beta - D_\beta C_r + mv_x D_\beta) + E_r (D_\beta C_\phi - D_\phi C_\beta) \\ Z_{\delta r} &= E_\beta (D_\delta C_\phi - D_\phi C_\delta) + E_\phi (D_\beta C_\delta - D_\delta C_\beta) + E_\delta (D_\phi C_\beta - D_\beta C_\phi) \end{aligned} \tag{6}$$

Equation (5) shows that the yaw rate is directly related to the difference between the two reaction forces on the rear wheels, F_{ad}. By controlling the driving torques generated by the two motors, we can tune F_{x3} and F_{x4} to attain the desired difference between their values.

In racing conditions, responsiveness and accuracy become the dominant requirements on the steering performance of the electric car. Hence, it is sensible to maintain the car always in the neutral-steer condition so that it follows the driver's steering command accurately and swiftly. With a neutral-steer vehicle, the yaw rate is expressed as follows:

$$r^* = \frac{v_x}{l} \delta \tag{7}$$

From equations (5) and (7), the desired reaction force difference F_{ad}, needed to achieve neutral-steer, is derived as follows:

$$F_{ad} = F_{x3} - F_{x4} = \frac{\delta}{Z_{Fr}} \left(\frac{v_x}{l} Z_0 - Z_{\delta r} \right) \tag{8}$$

Equation (8) is the control goal of our control system design. Figure 3 shows a block diagram of the structure of the proposed control system. The actual reaction force difference F_{ad} is obtained by the wheel dynamics block, with torque commands and vehicle roll angle being its inputs. Then, this actual F_{ad} is compared to the desired reaction force difference F_{ad}^{*} which is calculated based on vehicle longitudinal velocity, steering angle command and vehicle parameters. The error between the desired value and actual value goes to a PID controller which generates the difference between the two motor torques and feeds it back to the torque command generation block.

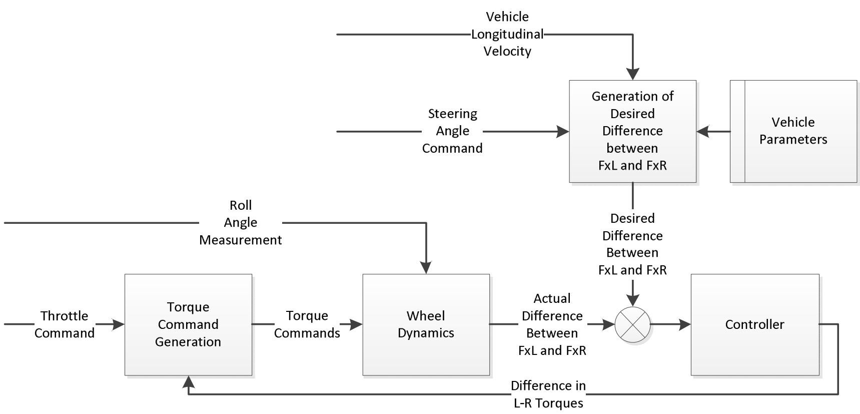

Fig. 2. A block diagram of our proposed direct torque control method for electronic differential.

3 Simulation Results

The electronic differential control system is constructed, based on the structure shown in Fig. 2, in MATLAB Simulink environment. In our Simulink model shown in Fig. 3, SimMechanics blocks are used to simulate the behavior of wheels, and a vehicle dynamics block (a State-Space block) is employed to calculate the vehicle states in the simulation environment. The vehicle state (roll angle) from the vehicle dynamics block is fed back to the tire dynamics blocks in which the wheel slips and the reaction forces F_{x3} and F_{x4} are calculated. A PID controller is used to track the desired value of the difference between F_{x3} and F_{x4}.

Using the Simulink model shown in Fig. 3, we have conducted two simulation studies to verify the effectiveness of the proposed control system. In the first simulation, a fixed steering angle value (0.1 rad) is sent to the model, and the actual yaw rate is observed to be maintained by the closed loop control system at the desired value.

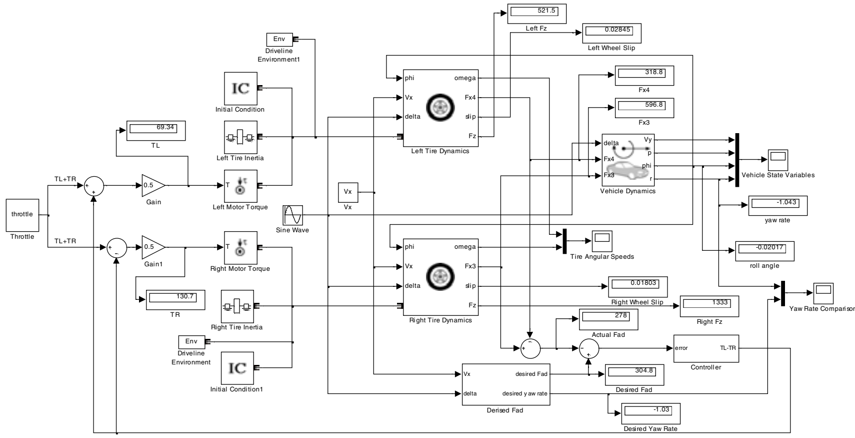

Fig. 3. The Simulink model of our proposed direct torque control method for electronic differential.

In the second simulation, a sinusoidal-type steering angle is input to the model. The result shown in Fig. 4 manifests that with a slight lag, the actual yaw rate (shown in blue) is very close to the desired yaw rate (shown in red). Also, as indicated by the wheel slip scopes, the wheel slips of both tires are contained in a reasonable range. Therefore, the proposed control system provides the electric racing car with a close-to-neutral steering performance, while maintaining its stability and keeping slip ratios in a reasonable range in challenging steering scenarios.

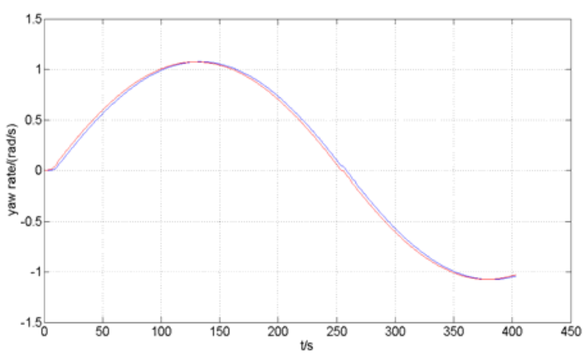

Fig. 4. The actual and desired yaw rate response with sinusoidal-steering input

4 Conclusions

In this paper, a closed-loop-control system is proposed for direct control of the motor torques, in order to achieve enhanced handling (namely neutral-steer) of an electric racing car. By individually controlling the driving torques produced by the two motors, the reaction forces exerted on the two rear driven wheels, F_{x3} and F_{x4}, can be tuned to attain their desired values. As can be seen from the simulations results, the proposed control system provides the electric racing car with a close-to-neutral steering performance, while maintaining its stability and keeping slip ratios in a reasonable range in challenging steering scenarios.

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