Torque Distribution Control Strategy for Electric Vehicles with Axles Separately Driven

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Abstract. This paper presents a control strategy of torque distribution for electric vehicles with axles separately driven. With two motors driving the front and rear axle separately, the torque distribution between axles can be easily achieved. To obtain good traction performance and stability of the electric vehicle, three control modes appropriate for different driving conditions are adopted: the routine control of equal distribution; the distribution based on Sliding Mode Control (SMC) to minimize wheel slip difference between axles; axles separately antiskid control based on SMC. With MATLAB/SimDriveline software, a forward vehicle simulation model was set up. The simulation results show that the torque distribution control strategy of three control modes can maintain the wheel slip in a reasonable range regardless of driving conditions, improving both vehicle traction ability and stability.

Introduction

For traditional vehicles, the Four Wheel Drive (4WD) system distributes the engine torque to the front or rear wheels according to the driving condition and vehicle statement to improve traction ability and stability of the vehicle [1]. In this paper, a research on the electric vehicles with axles separately driven was conducted. Unlike the torque generated by classic internal combustion engines, the torque of electric motors is available almost instantaneously [2]. In addition, it can be measured on-line, which means that advanced control techniques can be applied. With two motors driving the front and rear axle separately, the torque distribution control between axles is of large importance for improving longitudinal driving performance.

Antiskid control is necessary for safety driving on low adherent road surfaces. Vehicle traction control system (TCS) and anti lock brake System (ABS) are two of the most important components of vehicle longitudinal control in providing safety and achieving desired vehicle motion [2]. For the proposed 4WD electric drive system, drive torque distribution between axles is another effective way to avoid wheel slipping.

The remaining paper is organized as follows, section 2 is devoted to introduce the dynamic vehicle model and tire model. In section 3 are exposed the control strategy we are proposing: routine control of equal distribution, a torque distribution control between axles under light wheel slipping and axles separately antiskid control under deep wheel slipping. Some illustrative examples and scenarios are detailed in section 4 and some concluding remarks are presented in section 5.

System Modeling

In this paper, a nonlinear model of the vehicle is adopted [3]. As only longitudinal traction control is discussed, the dynamic longitudinal model of the vehicle is described in Fig.1. The difference between the left and right tires is ignored, making reference to a so-called bicycle model. Ignoring the influence of suspension, the dynamic equations of the wheel and the vehicle take the forms as follows:
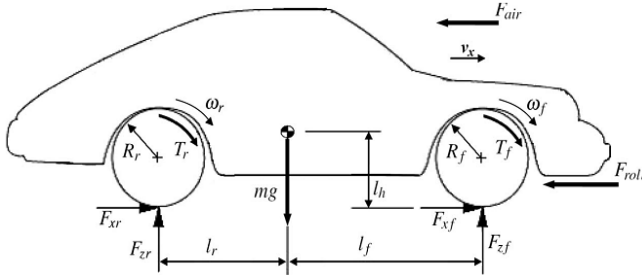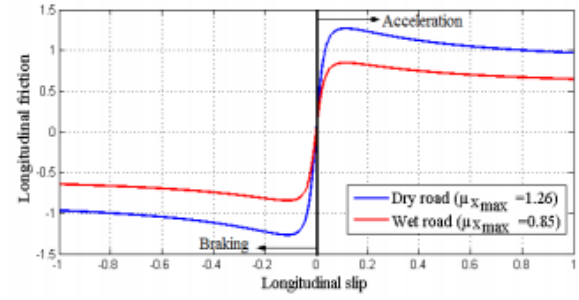
Fig.1 Vehicle model.

Fig.2 Friction coefficient vs. longitudinal slip.

$$m\dot{v}_x = F_{xf}(\lambda_f, F_{zf}) + F_{xr}(\lambda_r, F_{zr}) - F_{loss}(v_x). \tag{1}$$

$$J_f\dot{w}_f = T_f - R_f F_{xf}(\lambda_f, F_{zf}). \tag{2}$$

$$J_r\dot{w}_r = T_r - R_r F_{xr}(\lambda_r, F_{zr}). \tag{3}$$

$$F_{loss}(v_x) = F_{air}(v_x) + F_{roll} = c_x v_x^2 * sign(v_x) + f_{roll}mg. \tag{4}$$

$$F_{zf} = (l_r mg - l_h m\dot{v}_x)/(l_f + l_r). \tag{5}$$

$$F_{zr} = (l_f mg + l_h m\dot{v}_x)/(l_f + l_r). \tag{6}$$

$$\lambda_i = \begin{cases} (w_i R_i - v_x)/(w_i R_i), & w_i R_i > v_x, \quad w_i \neq 0, \quad \text{acceleration} \\ (w_i R_i - v_x)/v_x, & w_i R_i < v_x, \quad v_x \neq 0, \quad \text{braking} \end{cases} \quad i \in \{f, \ r\}. \tag{7}$$

$$F_{xi} = \mu_{xi}(\lambda_i) * F_{zi}. \tag{8}$$

where v_x is the longitudinal vehicle velocity of the vehicle center of gravity, w_f and w_r are the front and rear wheel rotation, T_f and T_r are the front and rear input torque, λ_f and λ_r are the slip ratio of front and rear wheel, F_{xf} and F_{xr} are the front and rear longitudinal tire-road contact forces, F_{zf} and F_{zr} are the normal force on the front and rear wheels, F_{air} is the air drag resistance, and F_{roll} is the rolling resistance (see Fig.1).

The vehicle parameters are the following: m is the vehicle mass, c_x is the longitudinal wind drag coefficient, f_{roll} is the rolling resistance coefficient, J_f and J_r are the front and rear wheel moments of inertia, R_f and R_r are the front and rear wheel radius, l_f and l_r are the distance from the front/rear axle to the center of gravity, and l_h is the height of the center of gravity. The normal force calculation method in this paper [see (5) and (6)] is based on a static force model, giving a fairly accurate estimate of the normal force, particularly when the road surface is fairly paved and not bump.

The longitudinal slip λ_i, $i \in (f, r)$ for a wheel is defined as the relative difference between a driven wheel angular velocity and the vehicle absolute velocity, and the longitudinal force F_{xi} of the tire-road contact is a nonlinear function of the longitudinal slip λ_i and of the normal force applied at the tire F_{zi}, as shown in equation (7) and (8), where μ_x is the longitudinal friction coefficient between the road and the tire.

The relationship of longitudinal friction coefficient (μ_x) and longitudinal slip λ_i can be described as Fig.2, which is significant in describing tire-road friction models. Nowadays different longitudinal tire-road friction models for vehicle motion control have been proposed in the literature [4, 5]. One of the most well-known models of this type is Pacejka's "magic formula" model (see, Pacejka and Sharp [6]). This model is a semi-empirical static slip/force model, obtained under particular conditions of constant liner and angular velocity. The Pacejka model has the form

$$F(\lambda) = c_1 \sin(c_2 \arctan(c_3\lambda - c_4(c_3\lambda - \arctan(c_3\lambda)))). \tag{9}$$

Where c_i, i=1,…,4, are parameters characterizing this model, which can be identified by matching experimental data. The parameters c_i depend on the tire characteristics(such as compound, tread type, tread depth, inflation pressure, temperature), on the road conditions(such as type of surface, texture, drainage, capacity, temperature, lubricant, i.e., water or snow), and on the vehicle

operational conditions (velocity, load).

However, the traditional simulations with static tire model face numerical problems at low speeds because the longitudinal speed of the wheel hub appears in the denominator of the expressions for longitudinal slip. Thus zero speed becomes an impossible calculation. This phenomenon has been reported in the work of Clover and Bernard [7], where they develop a differential equation for the slip coefficient to be a state variable rather than a kinematic function, starting from a simple relationship of the relative reflections of the tire elements in the tire contact patch. So transients of the static magic tire can be more reasonable.
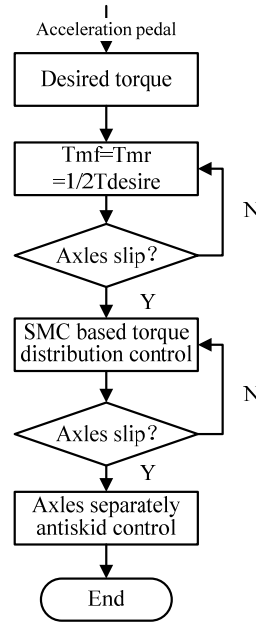
Control Strategy

Fig.3 Control strategy of the axles separately driven vehicle

In this paper, the desired torque from driver is always the first target to be followed during motors' torque control. To obtain the desired traction, $T_{desire} = T_f + T_r$ should be approximately met, where T_{desire} is the desired torque analyzed by pedal opening α_{pedal}, i.e. $T_{desire} = \alpha_{pedal} \times 2 \times T_{m-max}$, where T_{m-max} is the motor peak torque. In this paper, the front and rear motors have the same performance parameters, so it can be rewrote that $T_{desire} = (\alpha_f + \alpha_r)T_{m-max}$, where α_f, α_r is the corresponding motor load signal. Therefore, $T_{desire} = T_f + T_r$ can be replaced by $\alpha_{pedal} = 1/2(\alpha_f + \alpha_r)$. Then once α_{pedal} is given, the torque distribution control is merely the assign between α_f and α_r.

For time-varying road surfaces, $T_f + T_r \leq T_{road}$ should be always met to avoid slippage and keep vehicle stability, where T_{road} is the overall maximum road adhesion of certain road surface. And the crucial factor of estimating $T_f + T_r \leq T_{road}$ is the phenomenon of wheel slip, as shown in Fig.2. Once wheel slippage accruing ($\lambda > \lambda_{ux-max}$), the corresponding drive torque is out of road adhesion. Only within the stable region, i.e. $\lambda < \lambda_{ux-max}$, vehicle could drive safely.

The driving conditions are divided into three categories: $T_{desire} \leq 2 \times T_{m-max} \leq T_{road}$, $T_{desire} \leq T_{road} \leq 2 \times T_{m-max}$ and $2 \times T_{m-max} \geq T_{desire} \geq T_{road}$. The detailed analysis of each category is as follows:

A) Actually, when the vehicle is driving on dry road, i.e. $T_{desire} \leq 2 \times T_{m-max} \leq T_{road}$, there is no need to control torque distribution between axles, just giving two motors the same load signal, i.e. $\alpha_f = \alpha_r = \alpha_{pedal}$. No vehicle traction ability and stability would be affected. The only result is that, for different axle vertical loads, the front and rear tires would always be in different wheel slip,

resulting in different tire attrition. In this paper, the above average torque distribution is called 'routine control' and it is in the top priority in torque distribution control strategy.

B) However, when the vehicle is driving on wet road in hard acceleration or on ice road in light acceleration, i.e. $T_{desire} \leq T_{road} \leq 2 \times T_{m-max}$, the axle with lower vertical load may suffer slippage. In these situations, a flexible and reasonable torque distribution control should be applied to avoid slippage and keep $T_f + T_r = T_{desire}$. In this paper, a control method based on Sliding Mode Control is proposed and called 'SMC based torque distribution control'.

C) The worst situation is vehicle driving on ice road in hard acceleration, i.e. $2 \times T_{m-max} \geq T_{desire} \geq T_{road}$, resulting in slippage of both axles, which is very dangerous in longitudinal control. At this time, axles separately antiskid control is the best way to obtain maximum traction, making full use of the road adhesion, i.e. $T_f + T_r = T_{road}$. This control stage is called 'axles separately antiskid control'.

The whole torque distribution control strategy is shown in Fig.3.

SMC Torque Distribution control. The optimal result of torque distribution control is to keep the front axle slip and rear axle slip in the same. From equation (1), it can be seen that the slip difference is resulted from speed difference between front and rear wheel. To minimize speed difference, the following SMC strategy is researched.

A. Sliding Mode Control. The summary of Sliding Mode Control is as follows. Define the function in equations (9) and (10) that satisfies Condition 1 and 2 below. The function that fulfills Condition 2 meets requirements for a Lyapunov function, thereby guaranteeing that converges to 0. The control aim is achieved if Condition 1 is also satisfied, i.e., e converges to 0[1].

$$s = e + c \int_0^t e d\tau . \tag{9}$$
$$\dot{s} = \dot{e} + ce . \tag{10}$$

Where e is a deviation and c is a constant.
-Condition 1
If s=0, e converges to 0 with time.
-Condition 2
Differentiation of $V \equiv (1/2) * s^2$ satisfies the condition below.

$$\dot{V} = s\dot{s} \leq 0 . \tag{11}$$

B. The Speed Difference Minimize Control Between Front and Rear Wheel. In order to minimize the speed difference between front and rear wheels, the error term is defined as below.

$$e_1 = w_r - w_f . \tag{12}$$

Consider the system model represented in above section, the first-order sliding surface is designed as follow:

$$s_1 = e_1 + c_1 \int_0^t e_1(\tau) d\tau . \tag{13}$$

Eq.(14) is formulated by Eq.(13) and wheel dynamics equation.

$$J_f \dot{w}_f = T_f - R_f F_{xf}$$
$$J_r \dot{w}_r = T_r - R_r F_{xr}$$
$$J_w = J_r = J_f, R = R_f = R_r . \tag{14}$$

Differentiate the s_1 and use Eq.(14), it can be simplified as follow:

$$\dot{s}_1 = \dot{e}_1 + c_1 e_1$$
$$= \frac{1}{J_w}[T_r - T_f - R(F_{xr} - F_{xf})] + c_1 e_1 \cdot \tag{15}$$

In Eq.(14), using the relationship, $T_{desire} = T_f + T_r$, the equation can be simplified again.

$$\dot{s}_1 = \frac{1}{J_w} T_{desire} - \frac{2}{J_w} \cdot T_f - \frac{R}{J_w}(F_{xr} - F_{xf}) + c_1 e_1 \tag{16}$$

From these equations, the control law is designed as follow:

$$T_{f_{desire}} = \frac{1}{2} T_{desire} - \frac{R}{2}(F_{xr} - F_{xf}) + \frac{J_w}{2}(c_1 e_1 + k_1 \operatorname{sgn}(s_1)). \tag{17}$$

The system trajectories reach to the sliding surfaces within a finite time when the control law is designed to satisfy the following sliding condition, and then the error converges to zero.

$$\dot{V} = s_1 \dot{s}_1 = s_1[-k_1 \operatorname{sgn}(s_1)] = -k_1 |s_1| \le -\eta |s_1|. \tag{18}$$

Where $k_1 \ge \eta$ and $\eta > 0$.

Axles Separately antiskid control. When the SMC torque distribution control cannot prevent slippage, it means the desired torque is over maximum road adhesion. Then axles separately antiskid control should be applied to maintain the front and rear wheel slip in the optimal slipping zone, obtaining the maximum traction to be close to the desired traction. For the sake of simplicity, take the front axle antiskid control for example.

For this controller design, the new error is defined as follow:

$$e_f = \lambda_f - \lambda_r. \tag{19}$$

Where λ_r is the target constant slip ratio. Similarly in the speed minimize control, the first-order sliding surface is designed as follow:

$$s_f = e_f + c_2 \int_0^t e_f(\tau) d\tau. \tag{20}$$

Differentiate the s_f and use the wheels dynamic formulation in Eq.(13), it can be simplified.

$$\dot{s}_f = \dot{e}_f + c_2 e_f = \dot{\lambda}_f + c_2 e_f = \frac{1}{v_x}[\dot{w}_f R - (1 + \lambda_f)\dot{v}_x] + c_2 e_f$$
$$= \frac{1}{v_x}[\frac{T_f R - F_{xf} R^2}{J_w} - (1 + \lambda_f)\dot{v}_x] + c_2 e_f \tag{21}$$

From these equations, the control law is designed as follow:

$$T_{f_{desire}} = \frac{J_w \dot{v}_x}{R}(1 + \lambda_f) - \frac{J_w v_x c_2 e_f}{R} + F_{xf} R + k_2 \operatorname{sgn}(s_f). \tag{22}$$

The system trajectories reach to the sliding surfaces within a finite time when the control law is

designed to satisfy the following sliding condition, and then the error converges to zero.

$$\dot{V} = s_f \dot{s}_f = s_f [\frac{Rk_2}{v_x J_w} \text{sgn}(s_f)] = \frac{Rk_2}{v_x J_w} |s_f| \le -\eta |s_f|, \quad k_2 \le \frac{-\eta J_w v_x}{R} \text{ and } \eta > 0. \tag{23}$$

Simulation and Analysis

To verify the accuracy and effectiveness of the proposed control algorithm, we have simulated several scenarios with the vehicle in different acceleration on roads of different adherence conditions, as shown in Fig.4.-Fig.6.
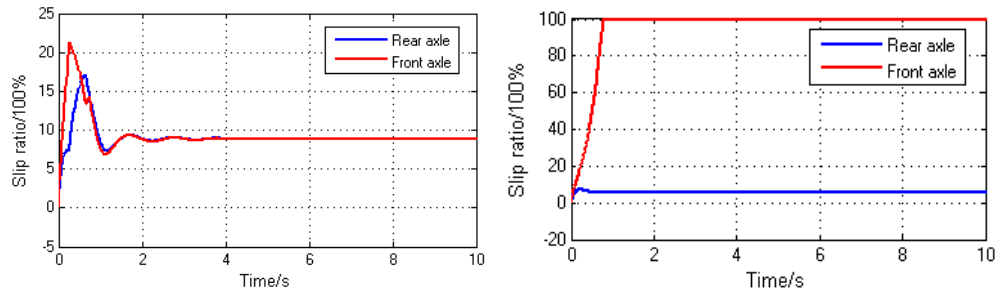
(a) Axles slip with torque distribution control   (b) Axles slip with non-control

Fig.4 Simulation results on ice road in light acceleration
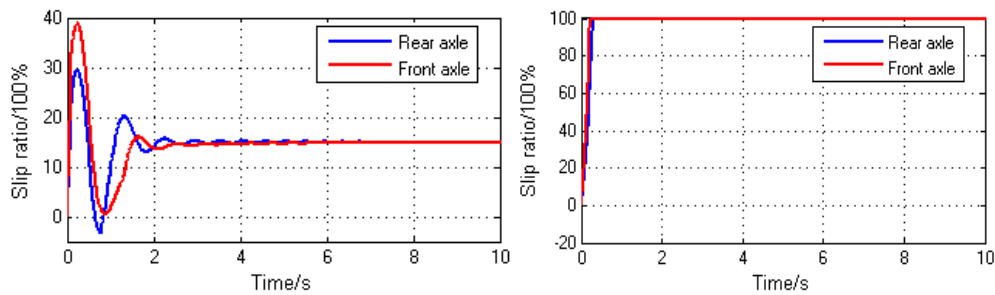
(a) Axles slip with torque axles separately control   (b) Axles slip with non-control

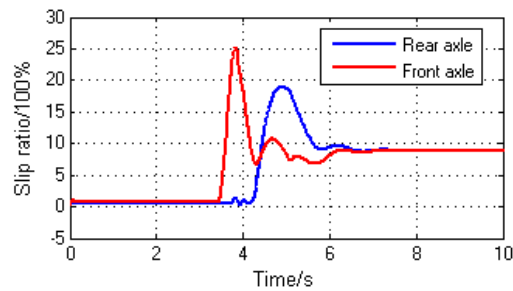Fig.5 Simulation results on ice road in hard acceleration

Fig.4 and Fig.5 show the performance of torque distribution controlled vehicle and non-controlled vehicle on ice road in light and hard acceleration respectively. It can be seen that, in the former case, the desired torque is within road adhesion, but if with no control the axle with lower vertical load would be in serious slippage (λ_f=100%). Apparently, road adhesion can be made full use with SMC torque distribution control. In the latter case, the desired torque is out of road adhesion and with axles separately antiskid control, the slip ratio of each axle is well controlled in the optimal slipping zone (referring to around 15%).

When the vehicle is driving on changeable road, the torque distribution control strategy can effectively switch between the three control modes, as shown in Fig.6. Fig.6(a') and Fig.6(b') are the traction performance of corresponding Fig.6(a) and Fig.6(b). It is clear in Fig.6(a') that when the vehicle drives from dry road (μ_{max}=0.85) to ice road(μ_{max}=0.10) in light acceleration, the SMC torque distribution control can maintain the acceleration same as it on dry load. Fig.6(b') shows that when the vehicle is in hard acceleration, the SMC torque distribution control can not prevent slippage, which means $T_{desire} \ge T_{road}$, then the axles separately antiskid control takes effect, keeping the vehicle with maximum traction.
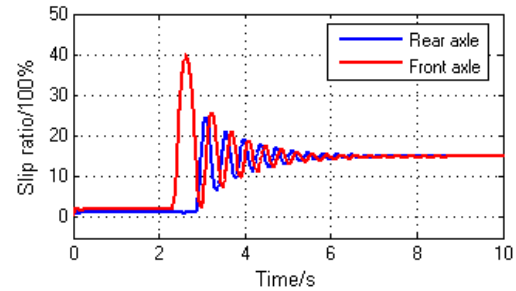
Conclusion

This work is focused on traction distribution control strategy of electric vehicle with axles separately driven. To simulate vehicle dynamics accurately, a modified magic tire model is adopted. A sliding mode control based torque distribution strategy of three modes appropriate for different driving conditions is proposed. Some simulation results are presented to demonstrate that with a
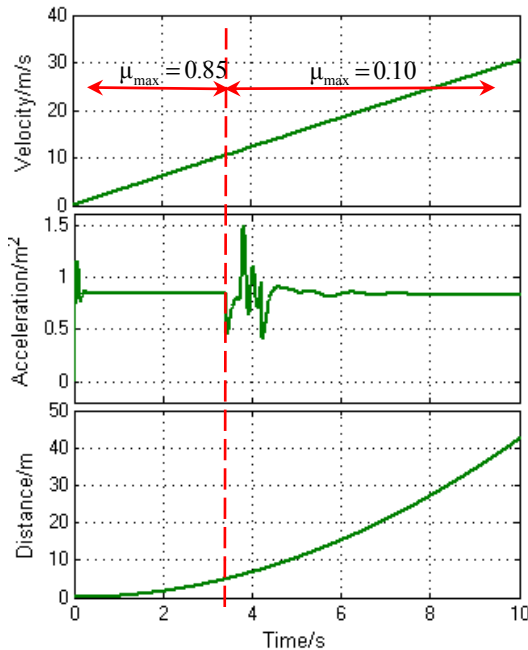
reasonable switch between the three modes, electric vehicles can run without slippage regardless of driving conditions, improving both vehicle traction ability and stability.
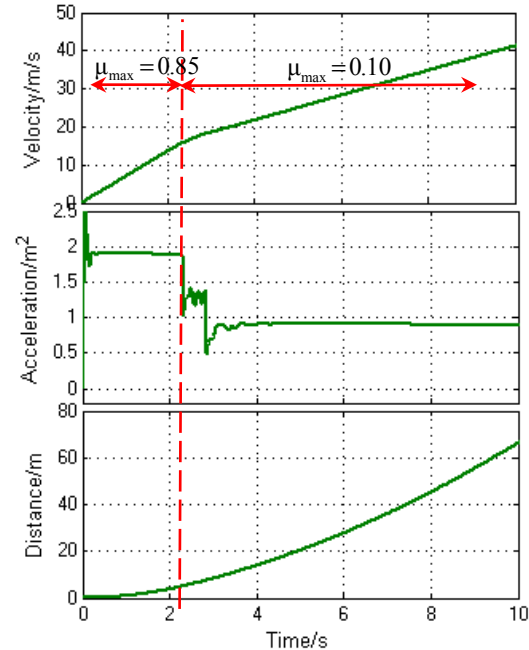

(a) Axles slip in light acceleration   (b) Axles slip in hard acceleration

(a') Traction performance in light acceleration (b') Traction performance in hard acceleration

Fig.6 Simulation results on road with changeable adhesion

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