A New Approach to Traction Control of EV Without Velocity Sensors

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This paper presents a new control algorithm that prevents electrical vehicle from wheel skidding in presence of uncertainties of tire-road condition. An estimator of the maximum effective torque is realized by wheel velocity and input torque, which is based on a pure kinematic relationship between the wheel and the chassis. In order to enhance the stability of the control system and handing performance of the driver, a half-closed loop controller makes use of the estimator to limit the maximum torque output to wheel, whose results to date indicate that it is an effective approach to prevent slip.

Keywords: Electrical vehicle, Traction control, Anti-skid, Maximum effective torque estimation, Half-closed loop control

1. Introduction

Due to the drastically increasing price of oil and the growing concern about global environment problems, more and more attention has been being paid to the research on the electrical vehicle, which includes hybrid electrical vehicle, Fuel-cell vehicle, pure electrical vehicle, etc. Among these different types of electrical vehicle, driven by motors is one of their common and instinct features, that is, pure electrical vehicle can be treated as the core prototype.

Comparing with internal combustion engine vehicle, the advantages of the electrical vehicle can be summarized as the following three points[1].

(1) Quick torque generation

This feature should be the most exciting advantage to the control of electrical vehicle. Electric motor's torque response is several milliseconds, which is 10-100 times faster than that of the internal combustion engine or hydraulic braking system. Not surprisingly, shorter delay permits more accurate control on the output torque.

(2) Independently equipped motors

Small but powerful electric motors equipped into each wheel can generate even anti-directional torques on left and right wheels at the same time. Independently controlled motors afford higher freedom to enhance the performance of vehicle stability control. Even from an economic point of view, it is not permitted ICV (internal combustion engine vehicle) to use four engines, but do electrical vehicle to use four electric motors.

(3) Easy torque measurement

While vehicle driving or braking, the uncertainty of torques output of motor is much smaller than that of IC engine or hydraulic brake. Motor torque can be easily calculated with the motor current. Therefore, simply driving and braking force observer can be designed to estimate the driving and braking forces between tire and road surface in real time. This advantage will contribute a great deal to application of new control strategies based on road condition estimation.

Fig. 1. COMS3-Electrical Vehicle for Experiment.

Thanks to improvements in motors and power electronics, the research and commercialization of pure electric vehicle have achieved much development. But

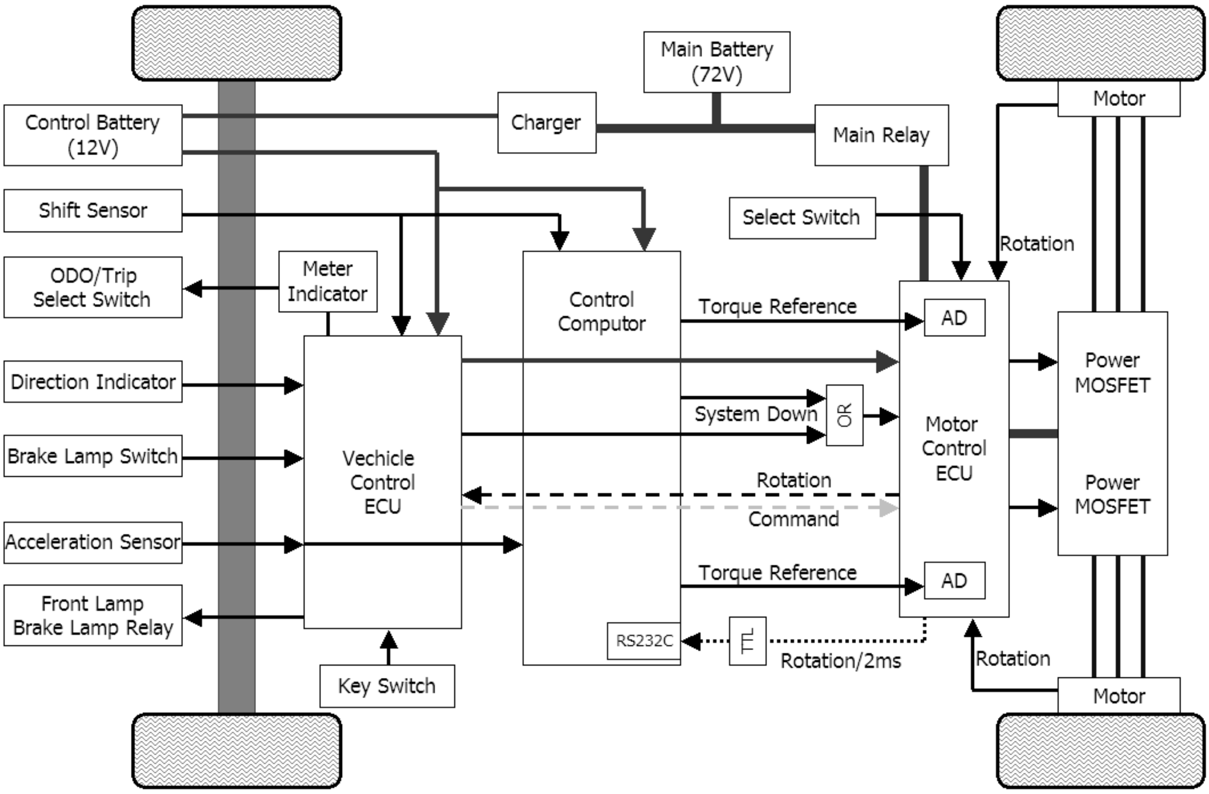

Fig. 2. Electrical System of COMS3.

because of the limited capacity of battery at the present, considering the cruise range, it is more reasonable of the automotive manufactures to product the light and small pure electrical vehicle. As well known, the light pure electrical vehicle is more inclined to skid on the slippery road and roll in the wide curve.

Table 1. Specification of COMS3.

Total Weight	340kg
Max. Power	2000W × 2
Max. Torque	100Nm × 2
Wheel Inertia	0.5Nms^2 × 2
Wheel Radius	0.22m
Controller	PentiumM1.8G, 1GB RAM
Real-OS	ART-Linux
Sampling Time	0.01s

By introducing computer control technology, many vehicle chassis control systems have been made significant technological progress over the last decade to enhance vehicle stability and handling performance in critical dynamic situations.(2) Among these controllers are systems such as the anti-lock braking system, direct yaw control system, integrated vehicle dynamic control system, etc. Effective operation of each of these control systems is based on some most basic conditions, for example, the output torque being able to accurately work on the vehicle. With this aim, traction control is developed to control the effectiveness of the torque output. The key of the traction control is anti-skid control or slip ratio control(3) when the vehicle travels on a slippery road, which must not only guarantee the effectiveness of the torque output, but also afford the parameters about the road situation to other chassis control systems.

In this paper, a new traction control system is proposed based on the estimation of the maximum effective torque. In the general anti-skid control systems, due to the physical and economic reasons, only the non-driven wheels can actually afford an approximate vehicle velocity. However, this method is not applicable when the vehicle decelerates by brakes equipped in these wheels or accelerates by 4WD systems. On the other hand, the velocity integrated by accelerometer value cannot avoid the offset problem and the curve error problem. Therefore, in this paper, use is made of the wheel accelerate to detect the maximum effective torque to the road, and the estimated torque is applied to anti-skid control implementation. Considering integrated with other control system easily, the application is realized in an independent controller instead of inside of the motor controller, the inverter(4)(5).

2. Experimental Electrical Vehicle - COMS3

In order to implement and verify the proposed control system, a commercial electrical vehicle, COMS shown as Fig. 1 is modified to fulfill the experiment requirement, which is made by TOYOTA AUTO BODY Co., Ltd. Shown as in Fig. 2, a control computer is added to take the place of the previous ECU to operate the motion control. The computer receives the acceleration reference signal from the acceleration pedal sensor, the forward/back signal from the shift sensor and the feedback values of the wheel rotation from inverter, and then calculates the most suitable torque for the motors. At last, the torque reference values of the left and the right rear wheel are independently given to inverter by two analog signal lines. Therefore, this experimental vehicle is named COMS3. Its main specification is list in Table 1. The select switch in Fig. 2 permits the driver select the driving mode between previous COMS mode and computer control mode.

On the other hand, in order to make the inverter not only receive the torque reference from the control computer, but also send the wheel rotation values to the control computer, the inverter manufacture was invited to modify the inverter. The inverter receives the torque reference values of the two rear wheels with AD converters, and sends the wheel rotation values to the control computer using serial communication (RS232C) based on a special protocol every 0.002s. The effective word length of the wheel rotation value is set as 10 bits of absolute rotation value + 1 bit of rotation direction flag. The main customized specification of the inverter is list as Table. 2.

Table 2. Modification of Inverter.

	BEFORE	AFTER
CPU for Current Calculation	SH7044 @ 28.7Mhz	SH7047 @ 50MHz
Torque Reference Interface	Special Serial Port	AD Converter
Min. Refresh Time of Torque Reference	10ms	2ms
Max. Delay Time of Torque Output	0.1ms	0.1ms
Max. Vary Rate of Torque Output	±1Nm/10ms, |T|< 5Nm ±5Nm/10ms, |T|≥ 5Nm	±5Nm/2ms
Rotation Output Interface	Special Serial Port	RS232C, 38400bps
Min. Refresh Time of Rotation Output	15ms	2ms

The most outstanding adventure of the modified inverter is that the minimum refresh time of torque reference is decreased from 10ms to 2ms, which makes it possible that the torque reference can be actualized more fast and accurately. And the developed maximum vary rate of torque output permits more intense torque variation for motion control.

The essence of the motion control of electrical vehicle is motor control. Therefore, to a common sense, the shorter sampling time or calculation period is expected. In some researches on the motion control of electrical vehicle, the calculation period of 0.001s has been realized. But in this paper, it must be point that the calculation period of control computer is constrained to 0.01s at the present time, due to the refresh rate of the RS232C buffer limited to 0.01s in ART-Linux OS. This problem is expected to be resolved in the future research.

3. Longitudinal Motion Control

Because of only longitudinal motion discussed in this paper, the dynamic model of the vehicle can be simplified as Fig. 3, and the parameter definition is list in Table 3.

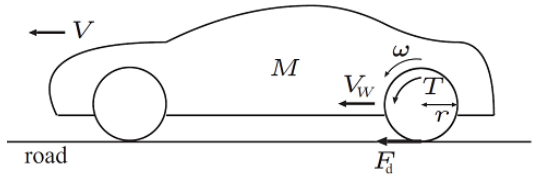

Fig. 3. Dynamic Model of Vehicle.

Table 3. Parameter List.

J_w	Wheel Inertia
V_w	Wheel Velocity
ω	Wheel Rotation
T	Driving Torque
r	Tire Radius
F_d	Driving Force
M	Vehicle Mass
V	Vehicle Velocity
F_{dr}	Cruise Resistance
λ	Slip Ratio
μ	Friction Coefficient

The dynamic differential equations for the calculation of longitudinal motion of the vehicle are described as follows.

$$J_w \dot{\omega} = T - rF_d \quad (1)$$

$$M\dot{V} = F_d - F_{dr} \quad (2)$$

$$V_w = r\omega \quad (3)$$

$$F_d(\lambda) = \mu N \quad (4)$$

The interrelationships between the slip ratio and friction coefficient can be described as various formulas.

Here, the most generously adopted Magic Formula, shown as Fig. 4, is applied to build up a vehicle model in the following simulations.

4. Maximum Effective Torque Estimation (METE)

Some anti-skid control systems make use of the interrelationships between the slip ratio and friction coefficient to control the wheel velocity. No matter these various formulas describing the interrelationship are accurate or not in any state, at least, the friction coefficient between the tire and the road surface is much different in different condition and difficult to measured in real time. In this paper, in order to avoid the complicated problem, only the relationship of driving force distribution is considered based on the following recognition.

(1) Whatever kind of tire-road condition the vehicle is driven in, the kinematic relationship between the wheel and the chassis is always fixed and available.

(2) If the wheel and the chassis accelerations are well controlled, the difference between the wheel and the chassis velocities, i.e. the slip is also well done.

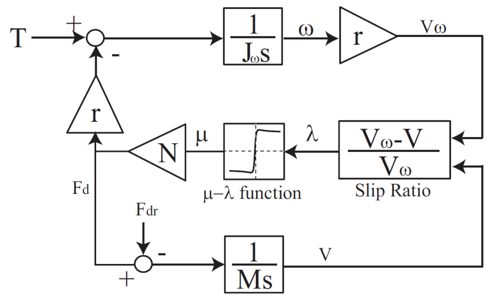

Fig. 4. Magic Formula in One Wheel Vehicle Model.

Using the equations (1)-(3), the kinematic relationship between the wheel and the chassis can be always available, shown as equation (5).

$$\hat{\dot{V}} = \frac{rT - J_w \dot{V}_w}{r^2 M} \quad \text{......(5)}$$

So, the estimation of the chassis acceleration can be obtained by equation (5), which indicates that when the knowledge of T is available, the bigger the acceleration of the wheel increases, the smaller the acceleration of the chassis decreases.

While slip starting to occur, the difference between the wheel and the chassis velocities will become bigger and bigger, i.e. the acceleration of the wheel is larger than the one of the chassis. While no slip, the difference between the two accelerations should be nearly zero. So, the condition in which the slip does not start or become bigger is that acceleration of wheel nearly equals the one of the chassis. In most case, the acceleration of the chassis can be treated as the maximum effective acceleration of the whole vehicle system that comprises with wheel and chassis, no matter whether slip occurs or not. Considering the Magic Formula, it needs a subtle difference between them, so a relaxation factor, α is introduced here.

In equation (1)-(3), apply

$$\dot{V} = \alpha \dot{V}_w \quad \text{......(6)}$$

then, the maximum effective torque for the whole vehicle system can be estimated as follows. Here, cruise resistance, F_{dr} is assumed to 0. But F_{dr} is a variable related with the chassis velocity and the vehicle shape, and can be calculated or estimated in real time.

$$T_{\max} = \left(\frac{J_w}{\alpha r} + rM \right) \hat{\dot{V}} \quad \text{......(7)}$$

Since the maximum effective torque, T_{max} is smaller than T in equation (5) when slip occurs, it can be utilized as the limited value when the system detects the slip starts. And due to $\alpha < 1$, T_{max} is a little bigger than T in common state. A torque controller is designed as Fig. 5.

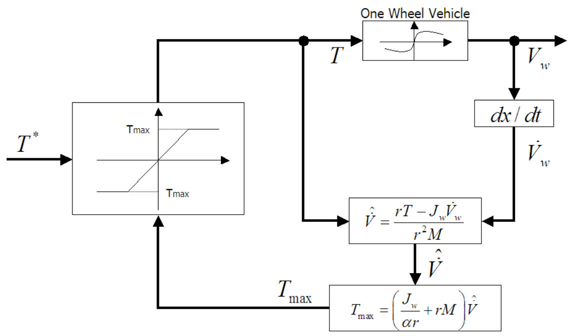

Fig. 5. Control System using METE.

Fig. 5 shows a limiter constrains the torque reference to the motor. As driven in a common road, the T_{max} is little bigger than T^*, so the system seems to be an open loop control system without the feedback T_{max}. While the vehicle accelerates in a slippery road, the T_{max} will become smaller than T^*. As a result, the estimated T_{max} instead of T^* will be treated as the input value to the motor. In this case, the system automatically transforms into a closed feedback system, shows as Fig. 6. In Fig. 5, D is used to indicate the extent of the slip. F_d, under this condition, equaling to the maximum friction force that the road and the tire can afford, is the only input reference, while T_{max} transforms into a self-active parameter.

In the actual control system, shown as Fig. 6, a low pass filter (LPF) is added to erase the noise of V_w, and T

is delayed to ensure the coincidence with it and system delay.

Here, it must be pointed that, considering both the acceleration and the deceleration case, strictly to say, when slip occurs, $|T^*| >> |T_{max}|$. In this paper, only acceleration is discussed and the T^* is always a positive value, so the controller is designed as $T^* >> T_{max}$ here.

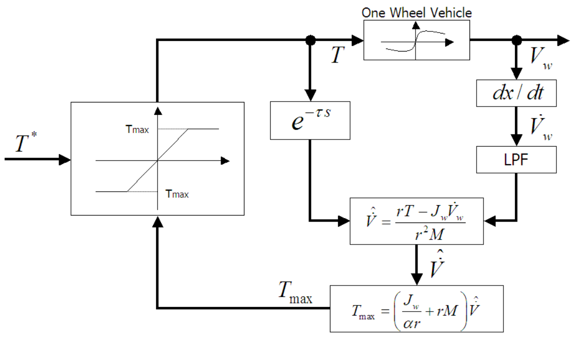

Fig. 6. Actual Control System using METE.

It can been found that in the transfer function (8), D is depended by M and α. In an ideal state, if α near to 1, D will be rendered to 0.

$$\frac{D}{F_d} = \frac{1}{Ms}\left(\frac{1}{\alpha} - 1\right) \qquad (8)$$

In the transfer function (9), T_{max} is decided by M and α, and have a linear relation with the F_d.

$$\frac{T_{max}}{F_d} = -r\left(\frac{J_w}{\alpha M r^2} + 1\right) \qquad (9)$$

Some merits of the half-closed loop control system can be concluded as follows.

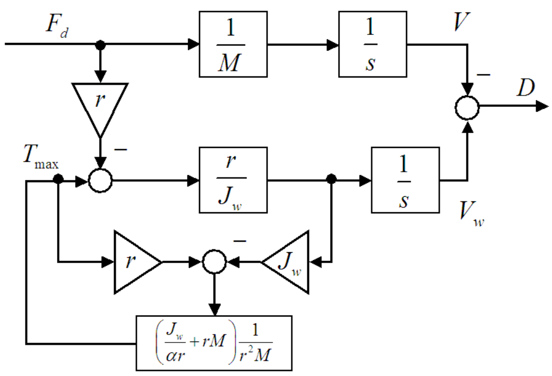

Fig. 7. Equal Control System while Slipping.

(1) The driver can directly control the acceleration at any time because only if T^* becomes smaller than T_{max}, the driver will be given the priority to take back the control right to the motor from controller immediately.

(2) Stability can be ensured in any condition. There is no error reference amplified or integrated to act as the input signal to the actuator, which exists commonly in the closed-loop control systems.

(3) Good performance due to no consideration of the worst case to ensure the stability.

(4) Suit for any road because only the kinematic relationship is utilized as the theory basis, which is always correct.

(5) Easy to apply in a real vehicle control system because of few parameters need adjusting.

(6) Afford the maximum friction force to other control systems in real time. METE indicates it can not only control the torque to the wheel in a critical condition, but also inform the controllers of other wheels and other control systems the state of the tire-road.

5. Results

A simulation system is built up using the scheme of Fig. 6. Slip starting and stopping time is set to 3rd second and 8th second, and the friction coefficient of the slippery road is 0.3.

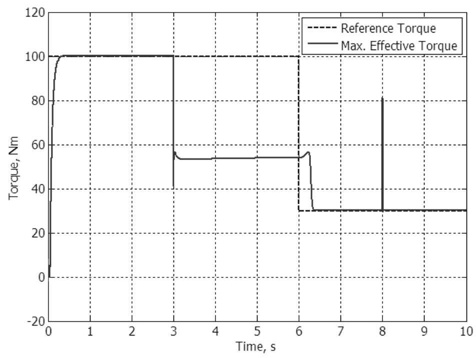

Fig. 8. T^* vs. T_{max} in Simulation.

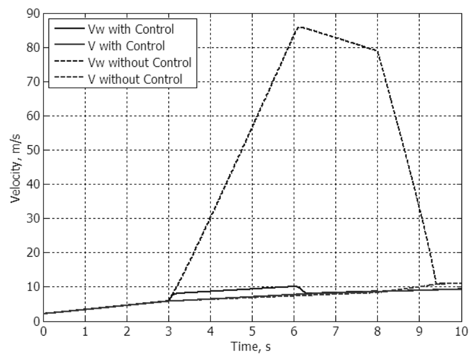

Fig. 9. V_w vs. V in Simulation.

The simulation result, shown in Fig. 8-9, proves the effectiveness of both the estimator and the controller. Contrast to the no control case, the difference between the wheel velocity and the chassis velocity caused mainly by the delay of control systems does not become bigger. The estimated maximum effective torque is near to the input reference toque in the common road, and fits to the maximum friction force afforded by slippery road surface to the tire.

A controller designed based on the algorithm the same as the simulation is applied to COMS3. The experiment result is shown as Fig. 10-12. In this experiment, the slippery road is simulated by acryl sheet with the length of 1.2m and lubricated with water. The initial velocity of the vehicle is set higher than 1m/s to avoid the immeasurable zone of the velocity meters equipped in the wheels. Here, it must be point that, in order to detect the vehicle velocity, only the left rear wheel is driven by the motor, while the right rear wheel is at a free state to give a reference value of the vehicle velocity for comparison.

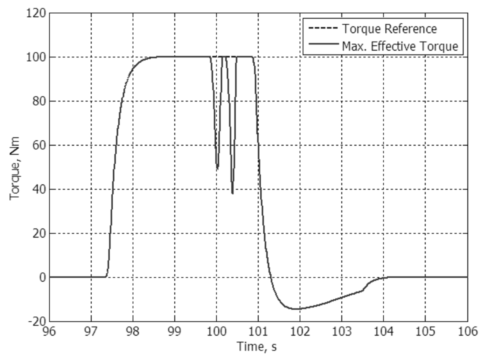

Fig. 10. T^* vs. T_{max} in Experiment with Control.

As shown in Fig. 10, when the vehicle driven into the slippery road, the torque controlled by METE is rendered suddenly to decrease the slip and increase the acceleration of whole vehicle system. But there is a little vibration the same as the simulation, whether the vibration will disappear or not must be test in a longer slippery road in future experiment.

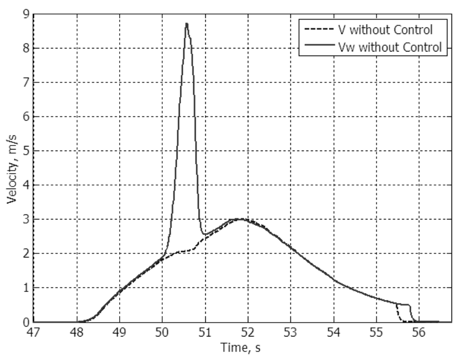

Fig. 11. V_w vs. V in Experiment without Control.

Since COMS3 is developed into an electrical vehicle with fast and independent torque response, in next experiments, some parameters such as τ, α, LPF, etc. will be varied to examine their affections to the performance of whole control system further.

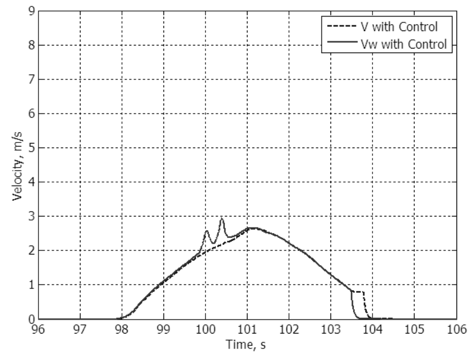

Fig. 12. V_w vs. V in Experiment with Control.

6. Conclusion

In this paper, an estimator of the maximum effective torque is proposed and applied to control the motor driving the vehicle. The accuracy of the estimator and the effectiveness of the control system are proved by both the computer simulation and the real experiment, which also affords a good basis for other more advanced vehicle motion control systems to enhance their performance and effectiveness. However, this implementation is based on the assumption that the vehicle is driven with a straight line and all the driving wheels on the road with the same friction condition. So, the next research will focus on such problems.

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