A Torque Vectoring Control System for Maneuverability Improvement of 4WD EV

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***Abstract*—This paper studies the improvement of the handling performance of 4WD EV driven by in-wheel motors under regular driving conditions. Fundamentally the structure of torque vectoring control (TVC) system for handling control consists of two control layers. The upper layer is a model following controller which makes the vehicle follow the desired yaw rate limited by the side slip angle and lateral acceleration. The torque distribution constitutes the lower layer. Several simulations based on veDYNA/Simulink are conducted to verify the effectiveness of the control system. It is clarified that the control system exhibits satisfactory performance in both open and closed loop maneuvers and the agility of the electric vehicle is improved.**

Keywords-component; torque vectoring control; 4WD; maneuverability

I. INTRODUCTION

The steering characteristic is one of the most important factors for distinguishing a vehicle from its rivals on the market. The main customer values to define an overall good tractability are precise steering without delay and vibration, an appropriate steering wheel torque, a good sense of the road conditions through steering feedback.

In order to assure the driving safety at high speeds the chassis of the vehicle is designed to decrease the yaw rate before the actual physical limit of the tires is reached. With TVC, the behavior of the chassis can be tuned without compromising on stability or comfort, by distributing the wheel torque individually to the wheels and thus creating an additional yaw moment on the vehicle. [1]The target of the TVC is to optimally utilize the different road-tire adhesion at each wheel and thus making the cornering more stable and increasing the agility of the vehicle [2].

Existing TVC systems, like the Bosch Dynamic Wheel Torque Control are either realized by brake, by differential or by wheel individual electric motors intervention. TVC systems like the Continental XDS and Cross Differential System are using the existing components of the ESC system to achieve a handling improvement. Systems based on differential are for example available at Honda, BMW and Audi. The TVC based on individual electric motors has the largest operation range as shown in the Fig.1, meaning that it can intervene well under regular driving conditions.

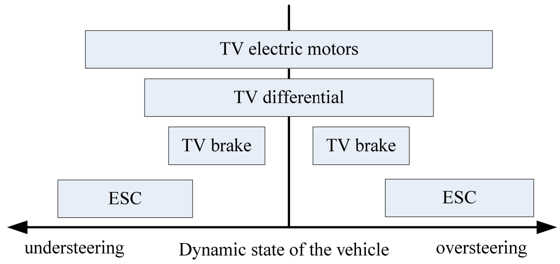

Figure 1. Torque vectoring range of action compared to ESC.[3]

In this paper, we focus on the improvement of agility with TVC under regular driving conditions and discuss the benefits with the TVC system through the simulation results. We propose a reference yaw rate model following control strategy for 4WD EV. The strategy consists of two control layers. The reference yaw rate calculation module and optimal feedback control compose the upper control layer. The lower control layer is torque allocation and this layer is used to calculate control inputs for four driving motors.

II. CONTROL SYSTEM DESIGN

The control structure is shown in Fig.2. Handling control is at the top level of the vehicle control architecture. The reference inputs of the control strategy are $\delta_f, a_y, \beta, v_x$, and its outputs are the reference torque of wheels. The 4 individual electric motors serve as the actuators.

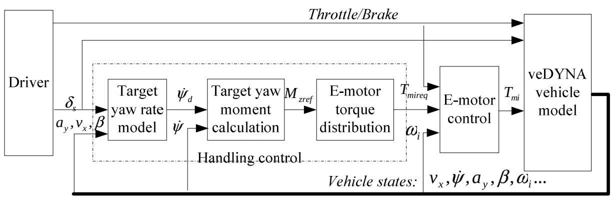

Figure 2. Torque vectoring control system structure

A. Target yaw rate calculation

The reference yaw rate $\dot{\varphi}_{ref}$ is used in the following control layers as the control target.

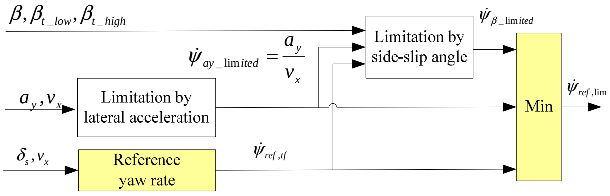

Figure 3. Target yaw rate calculation module

The basic reference yaw rate can be calculated through a linear vehicle model [4].

$$\dot{\varphi}_{ref,tf} = G(s)_{ref} \cdot \delta_F = \frac{G\varphi(0)\cdot(1+T_{\varphi}s)}{1+\frac{2\zeta}{\omega_n'}+\frac{1}{\omega_n'^2}s^2}\cdot\delta_F \tag{1}$$

Where, $G\varphi(0)$ is the steady gain. $G\varphi(0) = \frac{v}{l}\cdot\frac{1}{1+Kv^2}$, T_{φ} is the time constant, $T_{\varphi} = \frac{ml_f v}{slC_r}$, K is the stability factor of the vehicle, $K = -\frac{m\cdot(l_f C_f - l_r C_r)}{2\cdot l^2\cdot C_f\cdot C_r}$, ω_n is the natural frequency of the vehicle, $\omega_n = \frac{2l}{v}\sqrt{\frac{C_f\cdot C_r}{m\cdot I_Z}}\cdot\sqrt{1+Kv^2}$, ω'_n is the natural frequency of the reference model, $\omega'_n > \omega_n$. In this study, we put $\omega'_n = 1.5\omega_n$. ζ is the damping coefficient. $\zeta = \frac{m\cdot(l_f^2 C_f - l_r^2 C_r) + I_Z(C_f + C_r)}{2l\sqrt{mI_Z C_f C_r (1+Kv^2)}}$

Considering the relationship between lateral acceleration a_y, vehicle longitudinal speed v_x, changing rate of side-slip angle $\dot{\beta}$ and yaw rate $\dot{\psi}$:

$$a_y = v_x(\dot{\beta}+\dot{\psi}) \tag{2}$$

The target yaw rate in steady state should be limited by lateral acceleration as $\dot{\psi}_{ay\_limited} = \frac{a_y}{v_x}$

Moreover, because of side-slip angle restrict on yaw rate, the target yaw rate should also be limited as:

$$\dot{\psi}_{\beta\_limited} = \left(|\dot{\psi}| - \frac{|\beta| - \beta_{t\_low}}{\beta_{t\_high} - \beta_{t\_high}}\cdot\left(|\dot{\psi}| - |\dot{\psi}_{ay\_limited}|\right)\right)\cdot \mathrm{sgn}(\dot{\psi}) \tag{3}$$

B. Target yaw moment calculation

The feedback control compares the actual state of the vehicle with a desired state and minimizes the deviation. Based on the yaw rate control target from the reference model, the required yaw moment is calculated according to the difference between actual and target yaw rate. As controlled variable the yaw moment M_{Zref} of TVC is used.

$$M_{Zref} = \frac{(\dot{\varphi}_{ref,\lim} - \dot{\varphi}_{act})1+\frac{2\zeta}{\omega_n}+\frac{1}{\omega_n^2}s^2}{G_M(0)\cdot(1+T_M s)\cdot(1+T_{small}s)} \tag{4}$$

Where, $G_M(0) = \frac{V(C_f+C_r)}{2l^2 C_f C_r(1+KV^2)}$, $T_M = \frac{MV}{2(C_f+C_r)}$ [5], T_{small} is used to reduce the influence of this additional transfer function.

C. Control allocation layer

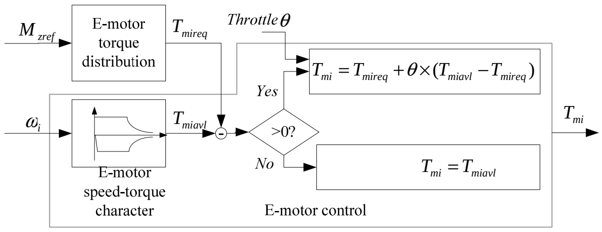

Figure 4. E-motor torque distribution module

The e-motor torque distribution module calculates the required e-torques to realize the desired yaw moment. Required yaw moment for target yaw rate is firstly distributed into e-motor torque in E-torque distribution module; the driver acceleration demand also needs to be considered as a second priority if there's still e-motor torque available. The yaw moment on vehicle is generated by longitudinal forces of four wheels as follow,

$$M_{zx,direct} = F_{xfl}\cdot\frac{b}{2} - F_{xfr}\cdot\frac{b}{2} + F_{xrl}\cdot\frac{b}{2} - F_{xrr}\cdot\frac{b}{2} \tag{5}$$

Therefore, the reference yaw moment from handling control target $M_{zref} = M_{zx,direct}$, the required e-torque of each motor is set as $T_{mijreq} = F_{ij}\cdot r$.

In this paper the TVC moment is distributed in a ratio of 0.5 to the front and 0.5 to the rear axle.

A simple strategy is designed to realize the required yaw moment, i.e. the changes of two wheels torque are at the same level (ΔT), that is: $T_{mireq} = T_{di} \pm 0.5\Delta T$. Where the normal wheel driven torque $T_{dfl} = T_{dfr}$, $T_{drl} = T_{drr}$. Consequently, $\Delta T = M_{zref}\cdot\frac{r}{b}$, and $T_{mireq} = T_{di} \pm 0.5 M_{zref}\cdot\frac{r}{b}$.

If there is still torque available according to the torque-speed characteristics of e-motors, additional driving toque $\theta\times(T_{miavl} - T_{mireq})$ should be given to single wheel. Therefore, the total torque for each wheel can be calculated as:

$$T_{mi} = \theta\cdot T_{mi\max} + (1-\theta)\cdot(T_{di} \pm 0.5M_{zref}\cdot\frac{r}{b}). \tag{6}$$

III. Simulation and Comparison

The simulation of the TVC controller is realized with the Software of TESIS veDYNA 3.10.2 based on the Matlab platform R2006b. As a vehicle model of the default veDYNA model full-size sedan is extended by individual drive and brake torque at each wheel. And the controller structure is shown in Fig.2.

TABLE I. Thr Specification of Vehicle and Motor

Vehicle mass		1296kg
Inertia moment (I_x, I_y, I_z)		305, 1520, 1750kgm2
Cornering stiffness front/rear		3500N/rad, 3500N/rad
Distance CG to front/rear axle		1.25m, 1.32m
Track width of the front/rear axle		1.405m, 1.399m
Dynamic tire radius		0.27m
Friction coefficient		0.9
Motor(/unit)	Max. power	10kW
	Max. torque	388Nm
	Max. speed	800rpm

The simulation is performed to compare the proposed control method with the one which does not use the torque vectoring control method. We use open and closed loop maneuvers for the comparison to verify the effectiveness and efficiency of the control strategy.

A. *Steering step response*

Steering step of 60° with a steering rate of 300°/s at the vehicle velocity of80 km/h is chosen in this driving maneuver.

As Fig.5 shows, the vehicle with TVC control has a faster peak response time and yaw rate oscillations are almost completely avoided. At the beginning the yaw rate is increasing through a torque distribution to the outer wheels, and at the moment where the actual yaw rate exceeds the reference yaw rate the torque is distributed to the inner wheels to prevent the overshoot of the yaw rate.

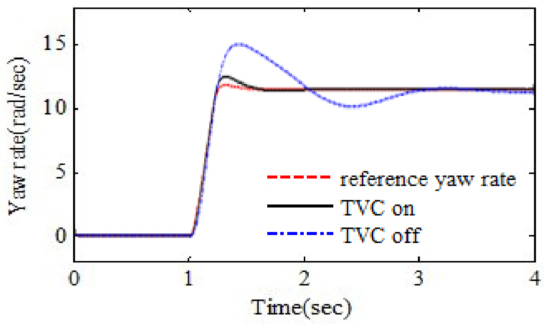

Figure 5. Yaw response in steering step

B. *Accelerating out of steady-state circular run*

At the beginning of the maneuvers the vehicle is driving on a circle with a constant speed of 20km/h with a steering angle of 60° and a lateral acceleration of 0.6m/s².Then the vehicle is accelerated with different longitudinal accelerations. For the evaluation the yaw rate difference 1s after the acceleration is measured and plotted in Fig.6.

Performing the same maneuver at a velocity of 40km/h clearly shows the understeering setup of the vehicle in Fig.7.

With the TVC, the yaw rate response is linearised, the higher the linearization degree of longitudinal acceleration to yaw rate difference, the more neutral the setup of the vehicle can be evaluated. Both maneuvers, at 20km/h, shown and at 40km/h show a change in the steering behavior and improve the agility while drifting out of corners. At higher speeds, the TVC has a higher influence on the steering behavior because the front tires are more likely to reach the traction limit.

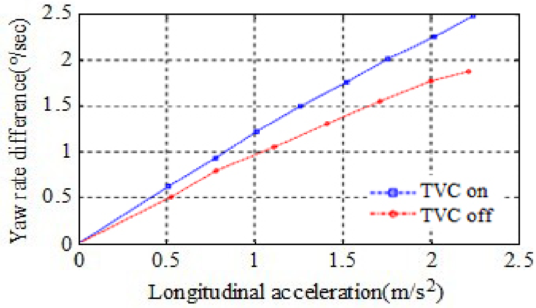

Figure 6. Yaw reponse to longitudinal acceleration, at the intitial speed of 20km/h, steering wheel angle 60°.

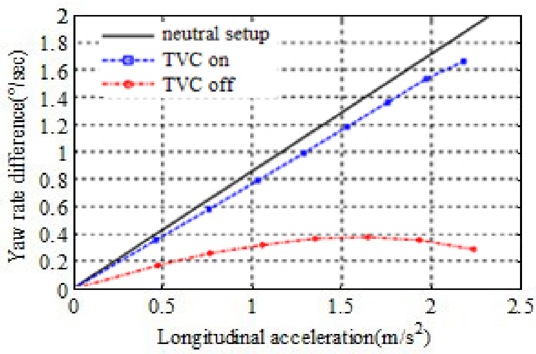

Figure 7. Yaw reponse to longitudinal acceleration, at the intitial speed of 40km/h, steering wheel angle 60°.

C. *The double lane change*

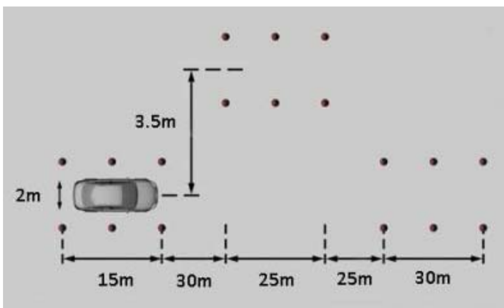

Figure 8. Setup of the double lane change.

The double lane change is a closed-loop maneuver, meaning the driver is adjusting his steering to the response of the vehicle. The vehicle is driving through three gates, according to Fig.8, with a constant speed of 70km/h.

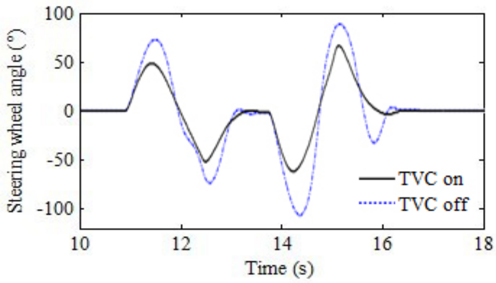

Figure 9. Steering wheel angle in double lane change.

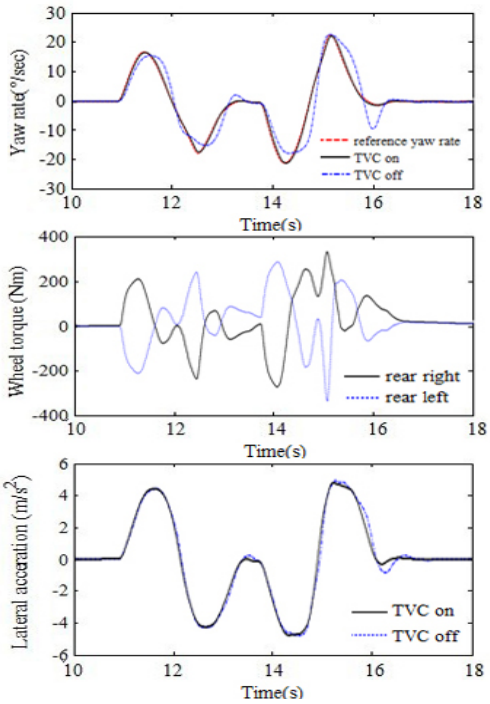

Figure 10. Yaw respone, rear wheel torque and lateral acceleration

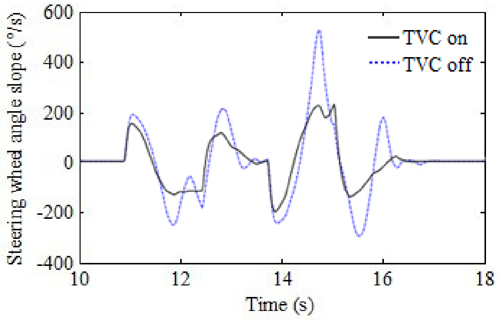

Figure 11. Lateral acceleration in double lane change.

As Fig.9 shows, the absolute maximum steering wheel angle is decreased with the TVC.

Steering correction is not necessary (counter steering) when entering the third gate with the TVC, because the yaw rate does not overshoot. And different left to right torque distributions are compensating over- and under steering of the vehicle while lateral accelerations remain the same as shown in Fig.10.

As Fig.11 shows, the maximum slope of the steering wheel angle is decreased with TVC, indicating a lower steering rate required for the driver to control the vehicle

IV. Conclusions and Future Works

Simulation results show that the proposed control strategy can improve the maneuverability of 4WD EV under the regular driving conditions. Torque vectoring control can be an effective method taking full advantage of the potentiality of lateral force.

Using the TVC system with 4 individual drive e-motors, the precision of the steering was enhanced with a faster response time, a lower overshoot and almost no oscillations of the yaw rate at a steering step. Moreover the acceleration understeering was compensated while the vehicle is drifting out of a circular run.

The steering effort for the driver was decreased because a part of the yaw rate was produced by the torque distribution and less yaw rate had to be produced by the steering of the front wheels. Additional smaller steering wheel angles, slower steering wheel travel and less counter steering was needed for dynamic maneuvers. Therefore the driver did not have to adjust his steering in the actual driving situation.

For future work, this control strategy needs to be refined and examined by various simulations. And it has to be tested at the presence of the sensors, motors and generators failures and it needs a failsafe layer which deactivates the TVC and switches the conventional hydraulic braking system. And the allocation layer can be extended with an intelligent torque distribution to further increase agility and stability by balancing the use of tire forces.

Acknowledgment

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