Disturbance Compensation and Torque Coordinated Control of Four In-Wheel Motor Independent-Drive Electric Vehicles

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ABSTRACT Functional overlaps and conflicts between Electronic Power Steering (EPS) and Torque Vectoring Control (TVC) for Distributed Drive Electric Vehicles (DDEVs) are issues that need to be addressed urgently. This paper deals with the interaction between EPS and TVC from the view of the influence mechanism and control objectives. The disturbance steering torque caused by the differential torque is estimated and a unified steering characteristic is proposed. To validate the analysis results, a torque compensation strategy is modified based on the exiting control system. And a robust vehicle state estimator is built thanks to the aligning torque estimation to provide the necessary information. Typical ground test results show that the torque oscillation on the steering wheel is suppressed. A faster and more linear steering response can be seen, which means that the proposed disturbance compensation strategy can comfort the contradiction between EPS and TVC, also improve the handling performance of the vehicle.

INDEX TERMS Coordinated control, disturbance compensation, TVC, EPS.

I. INTRODUCTION

Active chassis control systems, such as Electronic Power Steering (EPS), Anti-lock Brake Systems (ABS), Electronic Stability Control (ESC), Active Suspension Systems (ASS), are usually designed separately. Under most conditions, different active control systems can work independently of each other. For example, AFS tracks the desired yaw rate, while ESC makes slip angles within the stability margin [1], [2]. But things become different when it comes to critical conditions. In recent years, there is a trend of integrating different active control systems to further enhance vehicle handling and stability. Once a vehicle is equipped with several controllable chassis subsystems, it is important to synchronize the control actions of subsystems. An integrated chassis control system by coordinating the differential braking force, front and rear axle driving force, and active roll moment was proposed in [3], which improved the vehicle’s steering speed and lateral stability to a certain extent. But the fixed order of intervention restricted the overall performance. A new control architecture [4] was designed to control the multi-dimensional motions separately, but the coupling relationship was omitted. The integrated architecture of nonlinear model predictive control was adopted to coordinate the control of vehicle front wheel angle and four-wheel torque distribution [5], but the dependency on the model’s accuracy impaired the robustness of the control system in critical conditions. So far, EPS focuses on the realistic steering feel and active return function [6], [7], while ESC mainly concerns about vehicle stability [8], [9]. The coupling relationship is still poorly considered, leading to functional overlaps and interference.

Still, ongoing research and development are oriented to coordinated control. Bosch GmbH proposed a dynamic steering torque control system, which determined whether the vehicle is under critical situations and changed the assist torque to adjust the steering wheel angle. United chassis control system proposed by MANDO Company regulated the additional yaw moment and assist torque in the case of μ-split roads. In [10], the assist torque was adjusted through differential torque and the reference value was calculated according to the modified steering angle. A coordinated control system based on function allocation was designed [11] to decide the control weight of EPS and ESC. However, the two subsystems still worked independently without
unified dynamic control objectives. Nevertheless, AFS and ESC can join together through integrated control and minimize mutual interference.

The birth of Distributed Drive Electric Vehicle (DDEV) has promoted the demand for the integration of EPS and Torque Vectoring Control (TVC). Wang et al. [12] studied the integrated control of four-wheel drive and steering. The generalized longitudinal force and yaw moment demand were calculated firstly, and the target slip rate and slip angle for each wheel were achieved by AFS and ESC. But the control effect was determined by the tire-road friction estimation precision and tire model accuracy. The coordinated control strategy of the EPS and Traction Control System (TCS) was proposed in [13], which limited the assist torque and the target slip ratio to ensure the vehicle steering stability and longitudinal acceleration performance. But the influence on one subsystem from another was considered poorly. Besides, due to scrub radius and mechanical connection, differential torque would affect steering wheel torque, thereby, influence the driving performance [14]. So the driver’s torque feedback needs more consideration, especially with a DDEV. And the steering response lag of vehicles driven by wheel hub motors is much more severe because of the extra unsprung mass, which makes the disturbance worse. For this type of vehicle, it is of great significance to study the coordinated control of EPS and TVC.

To reveal the coupling relationship between EPS and TVC, the structural change of the steering system of DDEVs was analyzed, including the mechanism and control objectives. Based on these analyses, the consistent driving characteristic was identified. After that, an integrated coordinated control architecture was proposed. In the rest of the paper, a dynamic model of the vehicle system was presented first, and then the coupling relationship was analyzed to give the design objective in section 3. Key states and parameters estimation method was presented in section 4, while section 5 gave the coordinated control structure. Finally, test results were presented to prove the effectiveness of the proposed system.

II. VEHICLE DYNAMIC MODEL
A. EPS MODEL
A column-type EPS system is adopted here, shown in Fig. 1. The dynamic model can be observed in the following equation (1).

\[ J_s \ddot{\delta}_H + b_s \dot{\delta}_H = i_s \cdot M_H + i_p \cdot M_p - F_{\text{rack}} \cdot R_p \]  

where \( J_s \) is the equivalent inertia of steering system, \( b_s \) is the equivalent damper coefficient, \( \delta_H \) is the steering wheel angle, \( i_p \) is the power motor transmission ratio, \( i_s \) is the steering wheel transmission ratio, \( M_H \) is the steering wheel torque, \( M_p \) is the torque output of power motor, \( F_{\text{rack}} \) is the tensile force at the rack, \( R_p \) is the radius of the pinion.

From the side view of the steering system (Fig. 1), ignoring the effect of gravity and caster angle, the aligning moment at kingpin can be calculated as:

\[ M_k = M_{fl} + M_{fr} = M_{ZT} + F_{yfl} \cdot n_{fl} + M_{ZTr} + F_{yfr} \cdot n_{fr} \]  

where \( M_{ZT} \) is the self-aligning torque of tire, \( F_y \) is the lateral force of tires, \( n_k \) is the kinematic trail. Subscripts \( fl \) and \( fr \) stand for front left and front right respectively.

The force delivered to the rack from the kingpin torque of steering the system is:

\[ F_{\text{rack}} = \frac{M_{fl} \cdot \cos \phi_{fl}}{l_{fl}} + \frac{M_{fr} \cdot \cos \phi_{fr}}{l_{fr}} \]  

where \( l_k \) is the normal distance between the tie rod and the kingpin, \( \phi \) is the angle between the tie rod and the rack.

In order to identify the steering system parameter values, a force sensor is mounted on the tie rod to measure the tensile force and the aligning moment. Then, the results of fitting for each parameter are obtained. The dynamic geometric changes are considered and approximated through a parallel wheel travel simulation in ADAMS.

B. 2.2. VEHICLE DYNAMIC EQUATION
To simplify the analysis, without loss of generality, a yaw-plane two degree of freedom model to capture the vehicle lateral dynamics is depicted in Fig. 2. The dynamics equations are written as:

\[
\begin{align*}
\dot{v}_y &= v_x \sin \beta + v_x \cos \beta \cdot \omega_z \\
\dot{\omega}_z &= \frac{m}{I_z} \left( F_{yfl} \cos \delta_{fl} + F_{yfr} \cos \delta_{fr} \right) \cdot l_f - (F_{yfr} + F_{yrr}) \cdot l_r + \Delta M_z
\end{align*}
\]

\[ \frac{d}{dt} \begin{bmatrix} v_x \cr v_y \cr \omega_x \cr \omega_y \cr \omega_z \end{bmatrix} = \begin{bmatrix} 
\end{bmatrix} \]

FIGURE 1. Electric power steering system.

FIGURE 2. Vehicle dynamics model.
where, $y_i$ is the distance between the preview point of the forward-looking camera and the lane line, $v_i$ is lateral velocity, $v_x$ is longitudinal velocity, $\omega_z$ is yaw rate, $v$ is the course angle, $l_p$ is the preview distance, $m$ is the mass of the vehicle, $\Delta M_z$ is the additional direct yaw moment, $l_f$ is the front wheelbase, $l_r$ is the rear wheelbase, $l_z$ is the yaw inertia.

### C. TIRE MODEL

The test vehicle is equipped with Chaoyang tires 185/55R15. With the test data fitting results, magic formula tire model [15] is adopted here:

$$y(x) = D \sin[C \tan^{-1}[B(x + s_h) - E(B(x + s_h)] - \tan^{-1}(B(x + s_h)))] + s_v$$

$$M_{ZT} = -D_1 \cos[C_1 \tan^{-1}[B_1(\alpha + s_{th})] - E_1(B_1(\alpha + s_{th})] - \tan^{-1}(B_1(\alpha + s_{th})))] \cos(\alpha) \cdot F_y$$

(5)

where $y(x)$ stands for the longitudinal force or lateral force separately, $x$ stands for the slip rate $s$ or slip angle $\alpha$ separately. The parameters (B/C/D/E/...) are shown in Table 1.

### TABLE 1. Tire parameters of magic formula tire model.

| Parameters for $F_r$ | Parameters for $F_l$ | Parameters for $M_{ZT}$ |
|----------------------|----------------------|--------------------------|
| $B$, $C$, $D$        | $E$, $s_h$, $s_v$    |                          |
| 20.38                | 1.45                 | 5306.70                  |
| -9.89                | 1.42                 | 4515.35                  |
| 11.11                | 1.08                 | 0.04                     |

The slip angle for each wheel is calculated as:

$$\alpha_{fl} = \frac{v_x + l_f \cdot \omega_z}{v_x - 0.5b \cdot \omega_z} \delta_{fl}$$

$$\alpha_{fr} = \frac{v_x + l_r \cdot \omega_z}{v_x - 0.5b \cdot \omega_z} \delta_{fr}$$

$$\alpha_{rl} = \frac{v_x - l_r \cdot \omega_z}{v_x + 0.5b \cdot \omega_z} \delta_{rl}$$

$$\alpha_{rr} = \frac{v_x - l_f \cdot \omega_z}{v_x + 0.5b \cdot \omega_z} \delta_{rr}$$

(6)

where, subscripts $fl$, $fr$, $rl$, $rr$ stand for front left, front right, rear left, and rear right respectively, $b$ is the wheel track.

Both the tire lateral force and self-aligning torque are related to the tire vertical load. When there is load transfer, the vertical load of each wheel can be estimated by the following formula:

$$F_{ij} = \frac{l_i}{2l} mg - \frac{h_e}{2l} ma_x - \gamma_i ma_y + \frac{bh_i}{l} + K_i(h_g - h_v)$$

$$i = f, r \quad j = l, r \quad \gamma_f = -\gamma_r = 1 \quad \gamma_l = -\gamma_r = 1$$

(7)

Considering the limit of friction, it is necessary to use the longitudinal/lateral coupling characteristic of tire forces to modify the calculation results of the magic formula:

$$F_y = -\frac{s_v}{\sigma} y(x)$$

$$\sigma = \sqrt{\sigma_x^2 + \sigma_y^2}, \quad \sigma_x = \frac{s}{1 + s}, \quad \sigma_y = \frac{\tan \alpha}{1 + s}$$

(8)

Combining (4)-(7), the lateral force and self-aligning torque of each wheel can be obtained.

### III. COUPLING RELATIONSHIP ANALYSIS AND CONTROL TARGET DESIGN

The main function of EPS is to generate realistic steering feeling with good feedback behavior and steering return ability during daily driving. While the TVC system focuses on the improvement of yaw rate gain under normal situations, but the magnitude of sideslip angle under critical situations. It seems that there is little interaction between these two systems. However, the force acting (Fig. 3) reveals that a traction force difference would change the steering moment about the kingpins because of the mechanical connection between the steering wheel and front wheels. On the other hand, the power assistance characteristic of EPS influences the steering wheel angle, thus affecting the lateral dynamics of the vehicle, which may conflict with the control target of the TVC system. Therefore, it is of great significance to take the interaction into consideration.

### A. DISTURBANCE ON THE STEERING WHEEL TORQUE

The computation of the assist torque, which is based on the steering torque and the vehicle speed, complies with the boost assist curves [16]. There is little adjustment for DDEVs in the existing EPS control strategy. As (1), the desired steering torque is realized by regulating the magnitude of power assist torque. But the steering moment about kingpin is no longer as shown in (2) with the TVC system.

$$M_s = M_{ZTfl} + F_{xfl} \cdot n_{fl} + M_{ZTfr} + F_{xfr} \cdot n_{fr} + F_{zfr} \cdot n_{zfr} \cos \sigma_{fr} - F_{zfr} \cdot n_{zfr} \cos \sigma_{fr}$$

(9)

where $F_x$ is the longitudinal force of each wheel, $r_k$ is the scrub radius, $\sigma$ is the inclination angle.

As is well known, the in-wheel motor causes a larger kingpin offset, thus enhances the disturbance. A similar situation could be found on a wheel side motor drive vehicle with the equivalent acting point moved to the wheel center [14]. A natural approach for this problem is to equalize the driving torques between the inside and outside wheels of the front axle. But the differential torque caused by the rear axle is insufficient, and it may lead to the saturation of the rear tire force. Another solution is to design a differential power assist steering strategy accordingly. That is to use differential torque to supply a part of assist torque. But the conflict between additional yaw moment and assist torque still exists.
Besides, since the steering resistance at the spin turn is usually beyond the capability of drive torque, the EPS system is still necessary.

Take the sinusoidal steering angle input simulation for example, comparing with the original assist torque ($T_{c-EPS}$), additional torque caused by TVC ($T_{c-TVC}$) could be assistive or resistant, which depends on the scrub radius, additional yaw moment and steering wheel velocity. In most cases, the disturbance is nearly proportional to the steering angle, as (10) shows.

$$T_{c-TVC} \propto \Delta M_Z \propto (\frac{v_x}{l} \cdot \frac{\delta H}{i_s} \cdot \frac{1}{1 + K \cdot \sqrt{v_x^2}} - \omega_c)$$  \hspace{1cm} (10)

where $l$ is the wheelbase, $K$ is the desired stability factor.

Based on (9)-(10), the variation trend of $T_{c-TVC}$ and $T_{c-EPS}$ is shown in Fig. 4 (a). The green line stands for the total moment on the column, and the red zone stands for $T_{c-TVC}$. Due to the phase lag between the steering wheel torque and angle, there is a similar phase lag between $T_{c-TVC}$ and $T_{c-EPS}$ according to (9)-(10). As the brown zone in Fig. 4 (a) shows, the disturbance is not obvious at the beginning of the corner, whereas it becomes considerable at the beginning of return. This could lead to a larger overshoot of steering return. Besides, the disturbance reduces the steering torque level and gradient (Fig. 4 (b)), thus, affecting the feedback behavior of the driver.

To sum up, the disturbance $T_{c-TVC}$ caused by the differential torque at the front axle is non-negligible for the EPS system, which would conflict with the original steering power assist characteristic. So the influence of TVC on steering wheel torque needs to be impaired.

### B. DIFFERENT DESIRED STEERING CHARACTERISTIC

Traditional power assist steering strategy has taken the steering angular velocity into consideration. Specifically, as shown in Fig. 6, assist torque increases with the steering wheel angular velocity. In other words, compared with the steady-state condition, EPS will adjust the steering characteristic by changing the assist torque during the transient condition. However, most existing TVC algorithms are developed according to the steady-state cornering characteristic. And the steady-state gain of yaw rate is commonly used to describe the desired steering characteristic [3], [4], [5].

$$\omega_c,_{ref} = \begin{cases} \frac{v_x}{l} \cdot \frac{\delta H}{i_s} \cdot |\omega_c| \leq \frac{0.85 \mu g}{v_x} \\ \frac{0.85 \mu g}{v_x} \cdot |\omega_c| > \frac{0.85 \mu g}{v_x} \end{cases}$$  \hspace{1cm} (11)

**FIGURE 4.** Variation trend of $T_{c-TVC}$ and $T_{c-EPS}$ (a) and steering torque gradient (b).

What’s worse, from the simulation results of a double lane change test in Fig. 5, obvious fluctuations and even unexpected growth of steering wheel torque can be seen after the intervention of TVC. It may make the driver feel nervous and cause a steering failure.

**FIGURE 5.** Simulation results for double lane change close loop test ($v_x = 50km/h$), steering wheel angle (a), steering wheel torque (b).

**FIGURE 6.** Steering expected response with different steering speed.

As shown in (11), the desired yaw rate above is determined by the steering angle. As a result, no matter how fast the driver turns the steering wheel, the additional yaw moment could not adjust accordingly. Also, there is a certain lag and overshoot at the beginning of steering and the returning phase. The mismatch between the desired yaw rate and the real yaw rate grows with the speed of turning the steering wheel. That may lead to inferior tracking performance on sharper turns. As long as TVC and EPS are trying to track the different steering characteristics, it leads to the interference between TVC and EPS. Moreover, the combined effect of both subsystems may cause the overshoot and vibration of the steering response.

In a word, it is necessary to modify the reference in the existing TVC strategy and solve the mismatch between the expected driving characteristics of the two subsystems.

### IV. ESTIMATOR DESIGN OF VEHICLE STATES

Knowledge of the tire-road friction coefficient and sideslip angle is important for the design and analysis of the vehicle control systems. Many approaches to estimate these states have been proposed based on different dynamics [17], [18]. In this study, with the benefit of EPS and a camera installed...
on the windshield, the measurement of aligning moment and lateral displacement is available, improving the estimation precision of the tire-road friction coefficient and side slip angle, especially during daily drive situation. It’s worth noting that, the introduction of the aligning moment and lateral displacement is no longer useful once the lateral excitation exceeds a certain threshold. After that, vehicle lateral dynamics is more robust to noises and disturbances, that can be used to estimate the sideslip angle.

Considering the modeling error, a robust algorithm is needed to guarantee the convergence of the estimator. Thus, a nonlinear estimator based on the lyapunov function is proposed to estimate the tire-road friction and sideslip angle simultaneously (12). Where, lateral acceleration, yaw rate, kingpin torque, and lateral displacement are fed back to the estimator. The details and the proof of convergence can be found in the author’s previous paper [18].

A. DAILY DRIVING

\[
\begin{align*}
\hat{\mu} &= k_1 \text{sgn}(\hat{T}_p) \cdot (T_p - \hat{T}_p) + k_2 \text{sgn}(\hat{a}_y) \cdot (a_y - \hat{a}_y) \\
\dot{\hat{y}} &= \cos \nu \cdot \hat{\nu} + \omega_z l_p \cos \nu + \nu_v \sin \nu + k_3 (y_v - \hat{y}_v) \\
\hat{\nu} &= \dot{\hat{y}} - \omega_z \nu_v + k_4 (y_r - \hat{y}_r) + k_5 (a_y - \hat{a}_y) + k_6 \int_0^t (a_y - \hat{a}_y) dt
\end{align*}
\]

where \(\mu\) is the tire-road friction coefficient, \(T_p\) is the torque applied to the pinion, \(a_y\) is the lateral acceleration.

The kingpin torque is calculated according to steering system dynamics (1), the self-aligning torque and lateral force are obtained through the fitting data of the tire test (4)-(7). And thanks to motors’ torque feedback, the longitudinal forces are calculated easily (16). The superscript ^ corresponds to the estimated value.

\[
\begin{align*}
T_p &= i_s \cdot M_H + i_p \cdot M_p - J_H \ddot{\theta}_H - b_H \dot{\theta}_H \\
\hat{T}_p &= \left( \frac{\hat{M}_{fr} \cos \phi_{fr}}{l_{kg}} + \frac{\hat{M}_{fr} \cos \phi_{fr}}{l_{kg}} \right) \cdot R_p \\
\hat{a}_y &= \frac{1}{m} \left[ \hat{F}_{gy} \cos \phi_{fr} + \hat{F}_{gy} \cos \phi_{fr} + \hat{F}_{yrl} + \hat{F}_{yrr} \right] \\
F_{si} &= \frac{T_i - J_{oa} \dot{\omega}_i}{r} = i = fl, fr, rl, rr
\end{align*}
\]

B. LARGE LATERAL ACCELERATION

\[
\begin{align*}
\dot{\hat{y}} &= \hat{\nu} - \omega_z \nu_v + k_7 [a_y - \hat{a}_y] + k_9 (\omega_z - \hat{\omega}_z) \\
\dot{\hat{\nu}} &= f_r(\dot{\hat{y}}, \hat{\omega}_z) + k_9 (\omega_z - \hat{\omega}_z) \\
f_r(\dot{\hat{y}}, \hat{\omega}_z) &= \frac{1}{l_z} \left[ (F_{yrl} \cos \phi_{fr} + F_{yrr} \cos \phi_{fr}) \cdot l_f \\
&- (F_{yrl} + F_{yrr}) l_r - (F_{yrl} - F_{yrr} + F_{xrl} - F_{xrr}) \frac{b_1}{2} \right]
\end{align*}
\]

Here, the tire-road friction coefficient is supposed to be constant (\(\dot{\mu} \approx 0\)), so the estimation of the tire-road friction coefficient is no longer updated under the large lateral acceleration, but the estimated friction coefficient under daily driving remains. The architecture of the estimator is shown in Fig 7. \(k_1 - k_9\) are feedback gains.

V. COORDINATED CONTROL ARCHITECTURE DESIGN

According to the analysis results above, two main design objectives of system coordination control are obtained: 1) suppress disturbance from TVC without affecting basic steering characteristic, 2) modify the original reference model of TVC to match the expected driving characteristic of EPS. The coordinated control architecture is shown in Fig. 8.

Firstly, the torque command of the booster motor is adjusted to compensate for the steering wheel torque fluctuation caused by the differential driving torque based on the traditional EPS power strategy. On the other hand, considering the steering characteristic of EPS, the desired yaw rate is adjusted according to the steering wheel velocity, and a sliding mode controller is designed to guarantee the tracking performance.

A. DISTURBANCE COMPENSATION STRATEGY DESIGN

1) COMPENSATION FOR DAMPER AND INERTIA

The characteristic boost curve is the most basic guarantee for the EPS system to achieve the expected functions. In practical application, the basic control method is shown in Fig. 9. It contains a phase corrector, a signal filter, and the basic boost curve, which are the same as the traditional ones [16].

In-wheel motors have changed the steering structure, so the damping torque has to be adjusted. Since the vehicle sensitivity to steering inputs varies with the vehicle speed, it is
reasonable to adjust the damping coefficient with speed. This could contribute to improvement of the vehicle maneuverability at low speed by reducing system drag and friction, and also the vehicle stability at high speed by avoiding uncontrollable large steering wheel motion.

\[
T_{m-damp} = K_{damp}(v) \cdot \delta_H
\]  

(18)

where \(T_{m-damp}\) is the compensation torque of damper, \(K_{damp}\) is the damper gain coefficient varying with velocity, as the first calibration table in Fig. 9 shows.

The in-wheel motor causes a greater moment of inertia of the steering system than that of the ordinary EPS system, which will cause a sluggish response of the steering system. In order to improve the response rate of the steering system, the inertia compensation strategy based on the angular acceleration of the steering wheel has been applied, as the second calibration table in Fig. 9 shows.

\[
T_{m-inertia} = K_{inertia}(v) \cdot \delta_H
\]  

(19)

where \(T_{m-inertia}\) is the compensation torque of inertia, \(K_{inertia}\) is the inertia gain coefficient varying with velocity, which is determined by the additional moment of inertia.

2) COMPENSATION FOR INTERFERENCE OF TVC

Referring to the friction compensation strategy of EPS system, the basic idea of the compensation strategy is that a disturbance torque is extracted out of the steering model to generate a compensating counter-torque via the boost motor of EPS system, in consideration of the traction force difference and the variation of steering parameters. Combining (3) and (9), the equivalent moment about the steering column caused by the traction force difference can be calculated as:

\[
T_c-TVC = \frac{F_{sfr} r_{sfr} \cos \phi_{fr} \cos \sigma_{fr}}{l_{kfr}} - \frac{F_{sfl} r_{sfl} \cos \phi_{fl} \cos \sigma_{fl}}{l_{kfl}} \cdot R_p
\]  

(20)

In order to verify the analysis and suppress the disturbance, a simple approach is to apply the opposite acting moment to the column, which is added to the assist torque of the EPS system. The control strategy block diagram is shown in Fig. 9.

B. TVC SYSTEM WITH MODIFIED TRACKING REFERENCE

Firstly, according to the analysis of the desired driving characteristic of EPS, the steering wheel angular velocity is introduced to modify the original desired yaw rate so as to match the desired driving characteristic. Then the feedforward control based on steady-state gain and feedback control based on sliding mode algorithm are used together as the TVC algorithm to ensure the motion tracking performance and improve the response speed of the system. Finally, by weighting the average adhesion utilization rate of all wheels and the tracking errors, the optimal allocation algorithm is realized.

1) MOTION TRACKING REFERENCE MODEL WITH CONSIDERATION OF STEERING WHEEL SPEED

It can be seen from the simulation results in Fig. 10 that the vehicle’s yaw rate response slightly lags behind the steering wheel angle input. Positive additional yaw moment could help to reduce the response delay. Contrarily, when the driver starts to return, the sideslip angle keeps increasing (1.475s∼1.92s), leading to the delay of steering response and the increase of tracking error. And this conflict may cause the excessive steering input of the driver which then may make the vehicle loses its stability. A negative additional yaw moment is required in this case. Therefore, when the driver turns the steering wheel quickly, it could be useful to amplify the yaw rate tracking error and bring about more additional yaw moment. Obviously, introducing the steering angular velocity is simple yet effective.

On the basis of (11), the existing tire-road friction constraint remains, and the reference yaw rate is modified as shown in (21).

\[
\omega_{z,ref} = \begin{cases} 
\frac{v_x/l}{1 + (K - \Delta K(\delta_H))} \frac{\delta_H}{i_s} & |\omega_{z}| \leq \frac{0.85 \mu g}{v_x} \\
\frac{0.85 \mu g}{v_x} \cdot \text{sgn}(\delta_H) & |\omega_{z}| > \frac{0.85 \mu g}{v_x}
\end{cases}
\]  

(21)

where, \(\Delta K(\delta_H)\) is a dynamic correction based on the steering wheel angular velocity, the modified desired vehicle stability...
factor is between 0 and $K$, as shown in (22).

$$\Delta K (\delta_H) = \begin{cases} K \cdot G_{\delta_H} & \delta_H \cdot \delta_H \geq 0 \\ (K_0 - K) \cdot G_{\delta_H} & \delta_H \cdot \delta_H < 0 \end{cases}$$ \hspace{1cm}(22)$$

where $K_0$ is the original vehicle stability factor, $G_{\delta_H}$ is the gain coefficient of steering wheel angular velocity. It is not difficult to find that this gain coefficient should be positive and less than one, also, comparing with the high steering wheel angular velocity, the gain coefficient should change slowly at low steering wheel angular velocity. So the sinusoidal function is used to depict the variation trend of the gain coefficient with the steering wheel angular velocity, as shown in (23).

$$G_{\delta_H} = \begin{cases} 0.5 + 0.5 \sin\left(\frac{2}{3} \delta_H - \frac{\pi}{2}\right) & |\delta_H| \leq \frac{3\pi}{2} \\ 1 & |\delta_H| > \frac{3\pi}{2} \end{cases}$$ \hspace{1cm}(23)$$

The desired stability factor decreases with the increasing of the steering wheel angular velocity at the beginning of a corner and recovers to the initial value at the end. So, the deviation between the modified desired yaw rate and the actual yaw rate is enlarged, thereby increasing the additional yaw moment, assisting the driver to turn/return, and further enhancing the vehicle’s response to the driver’s steering intention.

2) FEEDFORWARD CONTROL

Feedforward control adopts the desired steady-state response to calculate the additional yaw moment needed as the feedforward input. By assuming that the derivative of states is zero, the steady state sideslip angle is obtained according to (4):

$$\beta_{ss} = \arg \min ||F_{yyf}(\alpha_r) \cos \delta_f + F_{yfr}(\alpha_r) \cos \delta_f + F_{yrl}(\alpha_r) + F_{yrr}(\alpha_r) - m \cdot v_\gamma \cdot \omega_z||_2$$ \hspace{1cm}(24)$$

where $\beta$ is the sideslip angle, and subscript $ss$ means steady state.

Ignoring the longitudinal motion, it is supposed that the steady-state lateral acceleration equals to the product of velocity and yaw rate. The steady lateral force can be obtained with the steady-state response. Assuming the yaw rate keeps unchanged, the desired additional yaw moment is obtained as:

$$\Delta M_{FF} = (F_{yrl} + F_{yrr}) \cdot I_r - (F_{yyf} \cos \delta_f + F_{yfr} \cos \delta_f) \cdot I_f$$ \hspace{1cm}(25)$$

It is difficult to obtain the analytical solution of a steady lateral force due to the nonlinear tire model. Therefore, a numerical solution is adopted here, and the value of feedforward yaw moment is stored in a lookup table, whose inputs are steering wheel angle, tire-road friction coefficient, and vehicle speed.

3) FEEDBACK CONTROL

Considering the modeling error and the nonlinearity of the vehicle, it is necessary to design a feedback control algorithm to improve the system’s robustness. The anti-windup sliding mode algorithm is adopted as the direct yaw torque controller. Ignoring the change of velocity, the dynamic equation of vehicle is rewritten as:

$$\dot{\beta} = F_{\alpha}(\beta, \omega_z, \delta_H) - \omega_z + D_{\beta}$$

$$\dot{\omega}_z = F_{\gamma}(\beta, \omega_z, \delta_H) + \mu + D_{\gamma}$$

$$F_{\alpha}(\sim) = F_{yyf} \cos \delta_f + F_{yfr} \cos \delta_f + F_{yrl} + F_{yrr}$$

$$F_{\gamma}(\sim) = \frac{(F_{yyf} \cos \delta_f + F_{yfr} \cos \delta_f) l_f - (F_{yrl} + F_{yrr}) l_r}{I_z}$$ \hspace{1cm}(26)$$

where $D = [D_{\beta} D_{\gamma}]^T$ is the disturbance of the system.

The desired side slip angle is obtained by (24) and restrained by the tire-road friction:

$$\beta_{ref} = \min(\beta_{s\beta}, \beta_{th}) \cdot \text{sgn}(\beta_{ss})$$

$$|\beta_{th}| \leq 10^5 \text{ if } \mu = 0.9, \quad |\beta_{th}| \leq 4^5 \text{ if } \mu = 0.35 \hspace{1cm}(27)$$

where subscript $ref$ and $th$ mean reference and threshold respectively.

Referring to the sliding mode control method, the tracking error of the system is defined as:

$$e = \omega_z - \omega_{z,ref} + \xi |\beta - \beta_{ref}| \cdot \text{sgn}(\delta_H)$$

$$\text{if } |\beta| \leq |\beta_{ref}|, \quad \text{then } \xi = 0 \hspace{1cm}(28)$$

where $\xi$ is an artificial weight coefficient.

According to (26) and (27):

$$\dot{e} = u + F_{\gamma}(\beta, \omega_z, \delta) - F_{\gamma,ss}(\beta_{ss}, \omega_{z,ref}, \delta) - \frac{\Delta M_{FF}}{I_z}$$

$$+ D_{\gamma} + \xi \left| F_{\alpha}(\beta, \omega_z, \delta_H) - \omega_z + D_{\beta} \right| \cdot \text{sgn}(\delta_H)$$ \hspace{1cm}(29)$$

Referring to the input and output linearization method, the main nonlinear part of the error equation is eliminated by designing the input of the system:

$$u = \frac{\Delta M_{FB}}{I_z} + \frac{\Delta M_{FB}}{I_z} - F_{\gamma}(\sim) + F_{\gamma,ss}(\sim)$$

$$- \xi \left| F_{\alpha}(\sim) - \omega_z \right| \cdot \text{sgn}(\delta_H)$$ \hspace{1cm}(30)$$

Combining (29) and (30), the error equation can be obtained as follows:

$$\dot{e} = \frac{\Delta M_{FB}}{I_z} + D_{\gamma} + D_{\beta}$$ \hspace{1cm}(31)$$

Due to modeling errors and disturbances, the feedback linearization method is often unable to effectively track the reference model of the vehicle system, and the sliding mode control often brings violent system oscillation. To cope with this difficulty, a sliding-mode formulation is adopted, endowed with conditional integrators [19].

$$\Delta M_{FB} = -k_p \text{sat} \left( \frac{e + k_i \rho}{\theta} \right) \cdot I_z$$

$$\dot{\rho} = -k_i \rho + \theta \text{sat} \left( \frac{e + k_i \rho}{\theta} \right)$$ \hspace{1cm}(32)$$
where \( \rho \) is an intermediate integral variable, \( k_p, k_i, \theta \) are the design parameters of the controller, the convergence of the error can be guaranteed as long as certain conditions (33) are satisfied. The proof can be found in [20].

\[
k_p > \bar{D} + 2k_i \cdot \theta, \quad k_i > 0, \quad 0 < \theta < \frac{4k_p k_i}{4k_i L_f + (k_i L_f + 1)^2}
\]  

(33)

Finally, the desired additional yaw moment is solved as:

\[
\Delta M_y = u \cdot I_z
\]  

(34)

C. SIMULATION TESTS

Simulation tests of typical conditions were carried out in the CarSim-Simulink joint simulation platform to verify the above control strategy. Referring to ISO 3888-2:2011, emergency avoidance simulation test was conducted (\( v_x = 40 \text{km/h}, \mu = 0.85 \)). In comparison, the original control strategy (TVC) was proposed in the authors’ previous paper [22]. Simulation results are shown in Fig. 11.

At the beginning of this simulation, the tracking error of yaw rate brought about auxiliary yaw moment to satisfy the driver’s steering intention, and the average of the peak steering wheel angle input during the whole operation was reduced by 10.5%. But the differential torque of the front axle was transmitted to the steering wheel, causing a great torque fluctuation, especially when the driver tried to return the steering wheel quickly (Fig. 11 (b)). After a compensation torque was applied, the torque fluctuation was suppressed.

On the other hand, the response curves of the yaw rate and sideslip angle and the hysteresis curve of steering response have revealed that the steering response has been improved with the optimizing strategy. When the driver tried to return the steering wheel, the steering angular velocity reduced the desired yaw rate and brought about an opposite yaw moment, thus reducing the overshoot of the yaw rate and sideslip angle (Fig. 11 (c-d)). From the hysteresis curve in Fig. 11 (e-f), the linearity of steering response has been improved, verifying the effectiveness of the control system.

Since the driver model in the simulation environment is quite simplified, vehicle tests have to be conducted to verify the coordinated strategy.

VI. VEHICLE TESTS

A. TEST PLATFORM VEHICLE

The test platform vehicle was adapted from a Roewe E50, equipped with 4 in-wheel motors and EPS, as shown in Fig. 8. The relevant parameters of the vehicle are shown in Table 2.

B. OPEN-LOOP TEST

To prove the ability of the proposed controller to stabilize the steering wheel torque, an open-loop test referring to ISO 7401 was carried out. Considering the size of the testing ground, the test speed was limited to 35km/h, the steering wheel angle amplitude was 90° with a frequency of 0.5Hz. The test results including step and sinusoidal input are shown in Fig. 12.
Although the fluctuation cannot be completely eliminated because of the un-modeled dynamics and external disturbance, the effectiveness of the controller is verified to some extent. Moreover, from the sinusoidal steering input test results, the overall steering torque gradient is increased by 27% (0.78Nm/rad-0.99Nm/rad), which guarantees good road feedback for the driver.

C. CLOSED-LOOP TEST

In order to further verify the effectiveness of TVC and EPS coordinated control system, the double lane change obstacle avoidance test (ISO 3888-2:2002) was carried out. The test speed was 40-45km/h, and the test results are shown in Fig. 13.

From the hysteresis curve of steering response, it can be seen that the control strategy can improve the response speed of vehicle yaw rate and lateral acceleration to the steering wheel angle, so that the vehicle can better respond to the driver’s steering intention. A good estimation result of aligning torque contributes to the suppression of the steering torque fluctuation. Besides, the modified reference improves the response speed and strength of additional yaw moment, helping to accelerate the yaw rate response and also reduce the overshoot during the period of lane changing, thus the mean peak steering wheel angle input during the whole operation has been reduced by 13% (108°-98°). Judging by the estimation results (Fig. 13 (d-f)), with the help of aligning torque estimation and lateral displacement error brought by the front camera, the estimation result of sideslip angle converges to the real one, and the estimation results of tire-road friction coefficient fit the real road conditions, which are necessary for the torque compensation and coordinated control system.

VII. CONCLUSION

Separate design of EPS and TVC of DDEVs has restricted the application of the integrated chassis control system. This work analyses the coupling relationship between EPS and TVC of DDEVs firstly. Then, a coordinated control strategy of EPS and TVC from the perspective of vehicle handling stability has been proposed. The main differences and contributions of this work are twofold:

1. The difference in design target and control effect between existing EPS and TVC strategy is pointed out, and the main contradiction between them is given. First, TVC intervention will produce certain interference to steering wheel torque, which violates the expected driver characteristic in EPS and affects the normal steering control of the driver. The second is the conflict and mismatch between the expected steering characteristic of two subsystems. Finally, the design goal of coordinated control based on steering rate correction is proposed.

2. The proposed compensation strategy suppresses the disturbance caused by TVC. Even though the proposed method is dependent on the precision of the model, lacking robustness, the effectiveness of the controlling idea has been verified. Then, the steering wheel speed information is introduced into the existing TVC control strategy to dynamically modify the reference model, so as to meet the driver’s steering/aligning intention and make it agree with the expected driving characteristics of EPS, thereby improving the overall control stability of the vehicle.

Several simulations and tests were conducted to verify the effectiveness of the control system. However, the robustness still needs further validation for the overall estimation precision and feasibility based on the aligning torque estimation. Meanwhile, the mechanical coupling effect of steering system restricts the coordination and control ability of EPS and TVC to a certain extent, further research can be conducted on the intelligent vehicle with drive by wire chassis in the future.
