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Tire-Model-Free Control for Steering of Skid Steering Vehicle

Haolan Meng, Lu Xiong, Letian Gao, Zhuoping Yu and Renxie Zhang

***Abstract*—Skid steering vehicle has larger wheel slip ratio when steering than straight driving. Due to the lack of accurate tire model in practical application, the slip coefficient is difficult to be quantitatively studied. However, there would be a large steady state error on the skid steering vehicle, which using the wheel speed difference control if the coefficient was ignored. Aiming at this problem, a controller overcoming integral saturation for steering control based on proportional integral control method is designed, which can correct the wheel slip coefficient in real time without the tire model. In the end, the effectiveness of the algorithm is verified by real vehicle test.**

Keywords—skid steering, steer-by-wire, electric vehicles

I. INTRODUCTION

The skid steering vehicle is steered by the rotational speed difference which is relied on the difference of the longitudinal force between the two sides of the wheels. Compared to the traditional steering, it has the characteristics of simple structure, high reliability, small occupied space and small turning radius. It can also achieve the original direction to meet more operating conditions. Therefore, it has been widely used in fields such as farms, mines, and the moon [1].

The traditional skid steering vehicle is driven by the engine and transfers power to the driving wheels through the gearbox and the drive shaft. For this kind of skid steering vehicle, special steering mechanism is needed to produce different driving and braking torque on the both sides of driving wheels to realize the steering. This mode of power transfer is more complex in structure and less efficient in fuel efficiency. The use of motor drive can help to solve these defects. Moreover, the motor drive has the characteristics of fast response, which can improve the mobility of the skid steering vehicle and reduce the noise in special occasions.

At present, most of the researches on dynamic control of skid steering vehicle are aimed at trajectory tracking control of driverless vehicle, and few researches are about skid steering vehicle with human driving. The common lateral dynamics control is mainly divided into two methods: direct yaw moment control and wheel speed difference control. For example, the speed and yaw rate tracking control strategy for the unmanned six-wheels independent driven skid steering vehicle is designed in reference 2. In which the yaw rate controller based on vehicle 2-DOF model adopts sliding-mode control algorithm is designed to calculate yaw moment to achieve reference yaw rate tracking. In reference 3, the fuzzy control is adopted and the PD closed loop control is used to design the yaw rate tracking strategy. In reference 4, the reference yaw rate is tracked by calculating the torque of the two motors by a hierarchical control structure. Moreover, a yaw rate tracking controller based on state feedback method is designed in reference 5.

However, in practical application, it is found that the method of direct yaw moment control is less robust and difficult to be popularized. This is because it requires higher modeling precision, sensor quality and actuator response. On the other hand, when the skid steering vehicle turns, the side slip angle of the tire will be larger, which will cause greater cornering force. At this time, a greater resistance will impede vehicle steering. In order to reduce the lateral resistance and realize the steering of the vehicle, there will be a large slip coefficient on both sides of the wheels, while most of the related studies of the wheel speed difference control have not paid much attention to this characteristic and put forward some pertinent measures.

Skid steering vehicle often works on complex terrain, and the driving torque of differential steering is much larger than that of straight driving, which makes wheels more prone to slip. This will also hinder its driving stability. Moreover, due to the difficulty of obtaining accurate models for various tires used by the research object, and with the change of pavement and speed, the wheel slip coefficient when steering is not a certain value. Therefore, based on the original research, from the perspective of engineering practice, the proportional integral control method with anti-integral-saturation is designed to ensure vehicle driving stability and control vehicle steering. By analyzing the driver's steering intention, this algorithm corrections the wheel skidding coefficient in turn, and realizes the lateral dynamics control based on the wheel speed control by wheel speed difference control. Finally, the validity of the algorithm is verified by the real vehicle test.

II. THE WORKING PRINCIPLE OF THE CONTROLLED OBJECT

A. The Hardware Platform of The Controlled Object

The structure of a skid steering vehicle platform studied in this paper is shown in Fig. 1, which is driven by 2 motors. The platform drives both sides of the wheels by the two wheel-side motors through the reducer. By controlling the torque of the two motors independently, the wheel speed difference between the two sides of the wheel is produced to track the driver's steering intention. Due to the rigid connection between the wheels through the drive shaft, both the torque

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and speed of the wheels on the same side are consistent when driving. Moreover, the speed of inside wheels is less than that on outside.

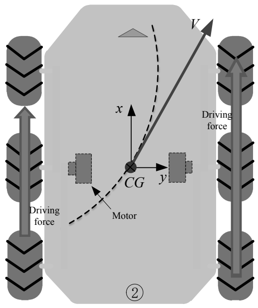

Figure 1. Structure of skid steering vehicle

B. Vehicle Dynamics Model

In order to simplify the vehicle dynamics model, it is assumed that vertical motion, the tilting movement and the pitch motion of the vehicle are ignored when the vehicle is moving on the ground. And the wheel torque and speed of each side of the wheels are the same in the movement. In addition, the vertical force of each wheels is evenly distributed.

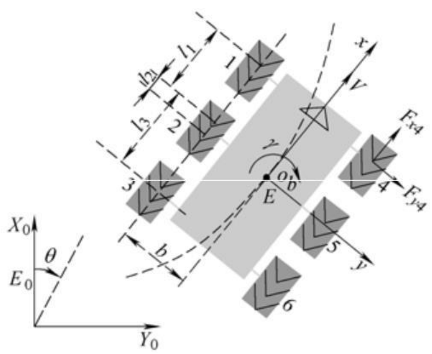

Figure 2. Coordinate system of skid steering vehicle

As shown in Fig. 2, the vehicle's attitude vector in the local coordinate system $E_0 = \{O_0, X_0, Y\}$ is represented by $(x_c, y_c, \theta)^T$, $(x_c, y_c)^T$ represents the location of the vehicle's center of mass in the local coordinate system, and θ represents the heading angle of the vehicle coordinate system relative to the local coordinate system. The absolute speed of vehicle in the local coordinate system can be transformed from $(\dot{x}_c, \dot{y}_c, \dot{\theta})^T$ to $E = \{o_b, x, y\}$ $(v, u, \gamma)^T$ under the vehicle coordinate system at the center of mass of the vehicle. It is easy to deduce the equation of vehicle kinematics, such as (1).

$$\begin{pmatrix} \dot{x}_c \\ \dot{y}_c \\ \dot{\theta} \end{pmatrix} = \begin{pmatrix} \cos\theta & -\sin\theta & 0 \\ \sin\theta & \cos\theta & 0 \\ 0 & 0 & 1 \end{pmatrix} \begin{pmatrix} v \\ u \\ \gamma \end{pmatrix} \tag{1}$$

In the equation, the v is the longitudinal vehicle speed, the u is the lateral speed of the vehicle, and the γ is the yaw rate of the vehicle.

According to Newton's second law, the dynamics of the whole vehicle can be described by the following (2)：

$$\begin{cases} \dot{v} = \dfrac{1}{m}\sum_{i=1}^{6} F_{xi} + \gamma u \\ \dot{u} = \dfrac{1}{m}\sum_{i=1}^{6} F_{yi} - \gamma v \\ \dot{\gamma} = \dfrac{1}{I_z} M_z \end{cases} \tag{2}$$

In the equation, F_{xi} and F_{yi} are tire longitudinal adhesion and lateral adhesion. M_z is the yaw moment acting on vehicle, m is the quality of vehicle, and I_z is the moment of inertia of vehicle. Among them:

$$M_z = \left(\sum_{i=1}^{3} F_{xi} - \sum_{i=4}^{6} F_{xi}\right) b + \left(\sum_{i=1}^{3} F_{yi} - \sum_{i=1}^{3} F_{y(i+3)}\right) l_i \tag{3}$$

In the equation, b is the half track, l_i is the i axis to the centroid distance.

C. Wheel dynamics model

Because the wheel speed difference control is used in this paper to realize the steering of the skid steering vehicle, it is necessary to establish the wheel model. Considering that the wheel speed of the left and right side of the vehicle can be controlled separately, and the wheel speed of each side wheel is the same. So it is only necessary to establish the dynamic equation of one wheel.

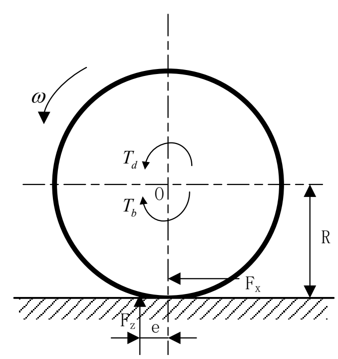

Figure 3. Force analysis of tire

As shown in Fig. 3 is a force analysis of any wheel, its simplified kinetic equation is：

$$J_\omega \dot{\omega}_j = -RF_{xi} - T_{bj} + T_j \tag{4}$$

In the equation, $j = l, r$ represents the left and right wheels, J_ω is the wheel inertia, ω_j is the wheel speed of the wheel, R is the effective radius of the wheel, F_{xi} is the force acting on the i wheel, T_{bj} is the braking torque of the wheel, T_j is the driving torque of the wheel:

$$\begin{cases} T_{Bj} = 3T_{bj},\ j = l, r \\ T_{mj} = \dfrac{3T_j}{z},\ j = l, r \end{cases} \tag{5}$$

In the equation, T_{Bj} is one side braking torque, T_{mj} is one side motor torque, and z is the drive ratio of the reducer.

I. CONTROLLER DESIGN

A. Analysis of Driver's Demand

For traditional steering vehicle, according to Ackerman steering principle, the relationship between turning radius r and front wheel angle α during vehicle steering, when the influence of wheel side deflection is ignored, is shown in (6).

$$\begin{cases} r = \dfrac{l}{\tan \alpha_{out}} \\ \tan \beta = \dfrac{l}{r - \dfrac{L_w}{2}} \end{cases} \tag{6}$$

In the equation, the l is the wheelbase and the α_{out} is the angle of the outer steering wheel. If the maximum value of the mechanical structure is obtained, the minimum turning radius $R_{\min}$ can be obtained. At the same time, because the steering wheel is connected with the steering wheel through the mechanical structure, the relationship can be converted to the relationship between the turning radius r and the steering wheel angle δ_s through the transmission ratio.

Therefore, can similar the skid steering wheel angle signal analysis for the driver's desired turning radius, which is defined as the minimum radius $R_{\min}$ of the maximum design of turning vehicles. According to the geometric relationship of the vehicle, it is not difficult to get Δv_r, which is the difference between the real speed of the ground at the two sides of the wheel:

$$\Delta v_r = \frac{2vb\delta_s}{R_{\min}\delta_{\max}} \tag{7}$$

In the equation, $\delta_{\max}$ is the maximum angle of the steering wheel angle. Because differential motors need to overcome the lateral resistance produced by larger side slip when steering, so there will be a s_s of slip coefficient of larger wheels on the two sides of the wheel. Therefore, the actual wheel speed difference Δv_w of the two sides of the wheel should be expressed as:

$$\Delta v_w = \frac{2vb\delta_s}{R_{\min}\delta_{\max}} s_s \tag{8}$$

B. Modification of Wheel Slip Coefficient of Vehicle in Steering Condition

The wheel slip coefficient curve of steering wheel with different road surface and turning radius theoretically can be obtained by analyzing the interaction between longitudinal force and lateral force of steering wheel when skid steering vehicle is steering, if tire can be accurately modeled. But the research object used in this paper is difficult to get the accurate model of the tire. Through real vehicle test, the coefficient is related to road adhesion coefficient, longitudinal driving force and vehicle speed. It can be initially identified as a change within a certain range, and it can be replaced by its value further. However, because of the changeable driving conditions and use of the research objects, the change of different types of tires will occur. Therefore, it is necessary to find a method to modify the wheel slip coefficient which is not based on the tire model and can be updated in real time.

In order to increase vehicle driving stability under complex conditions, a method of introducing yaw rate closed loop is adopted, which refers to the traditional vehicle stability control method. This method can be used to modify the wheel slip coefficient of the steering wheel, and also play a certain role in the stability control.

Calculation of Angular Velocity of Reference Yaw Rate

By analyzing the vehicle dynamics model, the steady yaw rate gain similar to that of the traditional Ackerman steering vehicle can be obtained:

$$\frac{\gamma}{\Delta v_r / v} = \frac{Au}{C + Dv^2} \tag{9}$$

In the equation, $A = 2b(k_{xf} + k_{xm} + k_{xr})(k_{yf} + k_{ym} + k_{yr})$, $C = 4b^2(k_{xf} + k_{xm} + k_{xr})(k_{yf} + k_{ym} + k_{yr}) + 4(k_{yf} + k_{ym} + k_{yr})(a_0{}^2 k_{yf} + b_0{}^2 k_{ym} + c_0{}^2 k_{yr}) - 4(a_0 k_{yf} + b_0 k_{ym} + c_0 k_{yr})^2$, and $D = -2(a_0 k_{yf} + b_0 k_{ym} + c_0 k_{yr})m$ are constant and only related to the parameters of the vehicle itself. Among them, k_{xi}, k_{yi} respectively for the i axis of the tire longitudinal slip stiffness and lateral stiffness, a_0, b_0, c_0 respectively, the rear axle from the centroid distance, it represents the current axis in the center position before, said negative centroid position, D determines the characteristics of the vehicle steering, when D turned to neutral.

Although the expression of the steady-state yaw rate gain of the skid steering vehicle is much more complex, its form is similar to that of the traditional vehicle, but the front wheel angle corresponds to the ratio $\Delta v_r / v$ of the speed difference between the wheel location and the vehicle speed, and the following relations can be obtained:

$$\Delta v_r / v = k_0 \delta_s \tag{10}$$

In the equation, k_0 is the ratio coefficient, which is similar to the ratio between the traditional steering wheel and the front wheel angle. The reference yaw rate γ_r of neutral steering can be obtained:

$$\gamma_r = \frac{Av}{C} k_o \delta_s \tag{11}$$

At the same time, in order to avoid the instability of the vehicle in the steering, the maximum lateral acceleration of the vehicle is 0.8 g. In conjunction with the steering wheel resolution in the previous section, the final defined reference yaw rate γ_{ref} is expressed as (12):

$$\gamma_{ref} = \min\left(\frac{v\delta_s}{R_{\min}\delta_{\max}}, \frac{0.8g}{v}\right) \tag{12}$$

Modification of Wheel Slip Coefficient

There are the following equations for the swinging motion of a vehicle around the z axis:

$$J_z\dot{\gamma} = M_{zy}(S) + M_z \tag{13}$$

In the equation, J_z is the moment of inertia of the vehicle running around the z axis, S is the turning coefficient of the vehicle wheel when steering, M_{zy} is the yaw moment caused by the lateral force, and M_z is the additional yaw moment generated by the longitudinal torque of the wheel. During vehicle steering, M_z is needed to overcome M_{zy} to achieve steering, while the increase of wheel slip coefficient S will weaken the lateral force of the tire, so that the steering resistance of M_{zy} will be weakened, which will help to achieve steering.

The speed error of the yaw rate is defined as $\tilde{\gamma} = \gamma - \gamma_{ref}$, and the control rate of the wheel slip coefficient S can be obtained based on the control rate of the nonlinear nominal system:

$$\begin{cases} S = -K_s sat\left(\dfrac{\tilde{\gamma} + k_s\varepsilon_s}{\theta_s}\right) + s_s \\ \dot{\varepsilon}_s = -k_s\varepsilon_s + \theta_s sat\left(\dfrac{\tilde{\gamma} + k_s\varepsilon_s}{\theta_s}\right) \end{cases} \tag{14}$$

The control rate will be described in detail in the next section. The s_s is the initial slip coefficient, which can be calibrated by the research vehicle. The design parameters of K_s, k_s and θ_s are all greater than zero, and K_s is the allowable maximum slip coefficient.

Due to the longitudinal torque exerted on the wheels of skid steering vehicle is much larger than that in straight line when steering, so it is necessary to set the slip coefficient saturation value K_s to ensure that the wheels are seriously slippery and the reference yaw rate cannot be tracked, so the integral operation is not divergent.

The final speed difference Δv_{ref} of the reference wheel can be expressed by (15):

$$\Delta v_{ref} = \frac{2vb\delta_s}{R_{\min}\delta_{\max}} S \tag{15}$$

C. Design of The Tracking Algorithm for Wheel Speed Difference

The design of the tracking control of the wheel speed difference is based on a nonlinear nominal system.

Nonlinear Nominal System

A nonlinear nominal system shown in the (16):

$$\begin{cases} \dot{x} = f(x) + u \\ y = x \end{cases} \tag{16}$$

In the equation, x is the state quantity of the system, $f(x)$ is a continuous function satisfying the local Lipschitz condition, its Lipschitz constant is L_f, u is the control input of the system, y is the controlled output of the system.

The design control rate makes the system output y tracking the reference value $y_r = x_r$, and the tracking error is $\tilde{x} = x - x_r$, then the original tracking system is transformed into a stabilization system.

$$\dot{\tilde{x}} = \frac{\hat{f}(\tilde{x})}{\tilde{x}}\tilde{x} + u + p(x_r) \tag{17}$$

In the equation, $\hat{f}(\tilde{x}) = f(x) - f(x_r)$, $p(x_r) = -\dot{x}_r + f(x_r)$ can be considered as a constant value.

By using the conditional integration method, the stabilization control rate of the stabilization system expressed by equation 18 is designed. If $s = \tilde{x} + k_0\varepsilon$,

$$\begin{cases} u = -Ksat\left(\dfrac{s}{\theta}\right) \\ \dot{\varepsilon} = -k_0\varepsilon + \theta sat\left(\dfrac{s}{\theta}\right) \end{cases} \tag{18}$$

In the equation, the controller parameters k_0, θ and K need to be greater than zero, of which K represents the saturation value of the actuator, and the value of θ should be $\dfrac{K}{\theta} > L_f$.

Stability Analysis of Control Rate

The integral operation will continue to increase, if the system cannot be stabilized for a long time. This is because the system is affected by external interference, modeling inaccuracy and actuator capacity constraints. It will affect the transient response and even the stability of the system. Therefore, it is necessary to analyze the stability of the system from two cases of unsaturation and saturation of integral operation.

When integral operation is unsaturated:

At this time $|s| < \theta$, the control rate is proportional integral control, which can guarantee the state $\tilde{x}$ of the system to be stabilized to the original point. The control rate (18) is replaced by the system equation (17), and the error system can be expressed by (19):

$$\begin{cases} \dot{\tilde{x}} = \left(\dfrac{\hat{f}(\tilde{x})}{\tilde{x}} - \dfrac{K}{\theta}\right)\tilde{x} - \dfrac{K \cdot k_0}{\theta}\varepsilon + p(t) \\ \dot{\varepsilon} = \tilde{x} \end{cases} \tag{19}$$

If $\sigma = \varepsilon - \dfrac{\theta}{K \cdot k_0} p(t)$, the system equation (19) is converted to (20).

$$\begin{cases} \dot{\tilde{x}} = \left(\dfrac{\hat{f}(\tilde{x})}{\tilde{x}} - \dfrac{K}{\theta}\right)\tilde{x} - \dfrac{K \cdot k_0}{\theta}\sigma \\ \dot{\sigma} = \tilde{x} \end{cases} \tag{20}$$

The Lyapunov function is established as (21):

$$V = \frac{1}{2}\tilde{x}^2 + \frac{1}{2}\frac{K \cdot k_0}{\theta}\sigma^2 > 0 \tag{21}$$

Equation (22) can be obtained by the simultaneous derivation of the upper side:

$$\dot{V} = \tilde{x}\dot{\tilde{x}} + \frac{K \cdot k_0}{\theta}\sigma\dot{\sigma} = \tilde{x}^2\left(\frac{\hat{f}(\tilde{x})}{\tilde{x}} - \frac{K}{\theta}\right) \tag{22}$$

Because the system function $f(x)$ is continuous and satisfies the local Lipschitz condition, so as long as (23) is set up, then $\dot{V} < 0$.

$$\frac{K}{\theta} > L_f \geq \left\|\frac{\hat{f}(\tilde{x})}{\tilde{x}}\right\| = \left\|\frac{f(x) - f(x_r)}{x - x_r}\right\| \tag{23}$$

Therefore, as long as the upper form is established, the system can be asymptotically stable.

When integral operation is saturated:

At this time $|s| \geq \theta$, the stability of the system needs to meet the following two requirements:

Integral operation does not diverge: First, the case of $s \geq \theta$ is discussed, when equation 18 can be converted to (24):

$$\begin{cases} u = -K \\ \dot{\varepsilon} = -k_0\varepsilon + \theta \end{cases} \tag{24}$$

As shown from the above, the control input u reaches the saturation value $-K$ of the actuator, and the ε tends to θ / k_0, so the integral value is stable.

In the same way, when $s \leq -\theta$, ε tends to $-\theta / k_0$, and the integral value is equally stable.

Bounded convergence: Under certain conditions, s is required to converge to $(-\theta,\ \theta)$ within a limited time, and after that, it is restored to the unsaturated state in this range, so as to ensure the asymptotic stability of the system.

Take $s \geq \theta$ as an example, so long as $\dot{s} < 0$ allows s to converge to $(0,\ \theta)$ at a limited time.

Took the derivative of this with respect to s, then

$$\dot{s} = k_0\dot{\varepsilon} + \dot{\tilde{x}} = -k_0^2\varepsilon + k_0\theta + f(x) - K - \dot{x}_r \tag{25}$$

When the integral value tends to θ / k_0, $\dot{s} = f(x) - K - \dot{x}_r$, if $\dot{s} < 0$ is required to be full of $\dot{x}_r > f(x) - K$. In the same way, when $s \leq -\theta$, if $\dot{x}_r < K + f(x)$ is satisfied, the s can be guaranteed to converge to the $(-\theta,\ 0)$ range in a limited time.

Therefore, as long as the above conditions are satisfied, the s can be reconverted to $(-\theta,\ \theta)$, and the unsaturated state can be restored.

Design of Tracking Control Rate of Wheel Speed Difference

The wheel speed difference between the two sides of the wheel can be expressed by (26), in which the left turn is positive:

$$\begin{cases} \Delta v = \omega_r - \omega_l \\ J_\omega \dot{\omega}_j = -RF_{xi} - T_{bj} + T_j \end{cases} \tag{26}$$

In the equation, $T_{mj} = \frac{3T_j}{z}, (j = l, r)$ is a single side motor torque, which can be divided into the left and right side control. The speed difference Δv_{ref} of the reference wheel can be obtained by (15), and the tracking error $\Delta\tilde{v} = \Delta v - \Delta v_{ref}$ of the wheel speed difference can be defined, then the nominal system equation of the wheel speed difference control can be obtained:

$$\Delta\dot{\tilde{v}} = \frac{\hat{f}(\Delta\tilde{v})}{\Delta\tilde{v}}\Delta\tilde{v} + \frac{T_{\Delta mj}}{J_\omega} + p(\Delta v_{ref}) \tag{27}$$

In the equation, $T_{\Delta mj}$ is a single side differential moment for the realization of the wheel speed difference. For the above system based on the nominal system control rate shown in (18), the control rate of $T_{\Delta mj}$ can be obtained:

$$\begin{cases} T_{\Delta mj} = -K_{\Delta j} sat\left(\frac{\Delta\tilde{v} + k_\Delta\varepsilon_\Delta}{\theta_\Delta}\right) \\ \dot{\varepsilon}_\Delta = -k_\Delta\varepsilon_\Delta + \theta_\Delta sat\left(\frac{\Delta\tilde{v} + k_\Delta\varepsilon_\Delta}{\theta_\Delta}\right) \end{cases} \tag{28}$$

The parameter $K_{\Delta j}$, k_Δ, θ_Δ in the equation is larger than zero, which indicates the differential torque of the maximum output of the motor.

Take the left turn as an example, the following torque signal will be sent to the motor:

$$\begin{cases} T_{bl} = T_{str} - T_{\Delta ml} \\ T_{br} = T_{str} + T_{\Delta mr} \end{cases} \tag{29}$$

In the equation, T_{str} is the longitudinal driving torque, which is not discussed in this paper.

II. Real Vehicle Verification

On the basis of meeting the requirements of controller design parameters mentioned above, the algorithm is simulated by Matlab/Simulink and other tools. Based on that, the specific controller parameters are obtained through real vehicle calibration, which is a matter of engineering at this point. For commercial reasons, the details of the real vehicle similar to Fig. 1 won't be described here. Due to the lack of relevant standards for the test and evaluation of the skid vehicle performance, considering the project requirements and actual conditions, the two conditions of steady circle and S-shaped driving are designed to verify the effect of steering control. In the test, the steering command is input by the driver through the steering wheel.

A. Steady Circle Driving

The steady circle condition is the test of the steady state tracking of the driver's steering instruction by the control algorithm. Because it is difficult to measure the steering radius of vehicle directly, it is verified by comparing the reference and actual yaw rate.

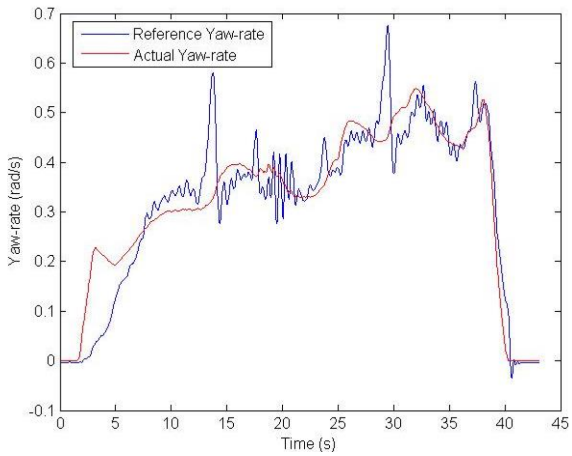

Figure 4. Yaw-rate tracking of steady circle driving

Due to the limited precision of human operation, the vehicle runs around the circle at about 20 kilometers per hour under the control of the driver. As shown in Fig. 4, the velocity tracking of the yaw rate of the whole steering process is good, the actual value is in accordance with the trend of the reference one and the response is rapid. It shows that the coefficient correction algorithm can play the role of real-time correction in the steady state and eliminate the steady state error of the steering well. There is some jitter on the yaw rate signal because of the road bump.

B. S-shaped Driving

The S-shaped condition tests the transient tracking of the driver's steering instruction by the control algorithm. In addition to contrasting yaw rate tracking, the trend and speed of steering wheel angle and vehicle yaw rate are also compared here.

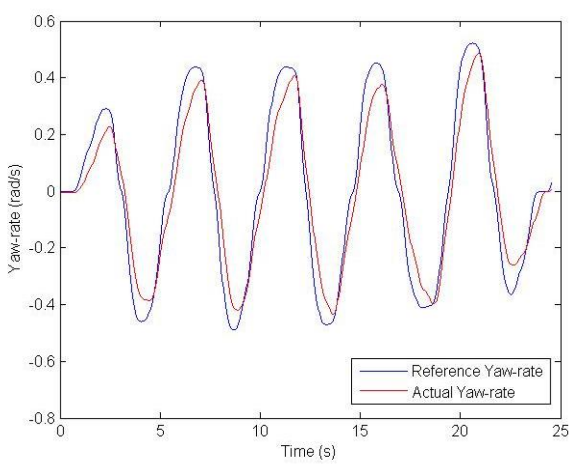

Figure 5. Yaw-rate tracking of S-shaped driving

As shown in Fig. 5, the response of the yaw rate is rapid. This is the auxiliary effect of the tire slip coefficient correction module based on the proportional integral control on the wheel speed difference tracking. The modified method can amplify the slip coefficient in the limit value and speed up the response speed in the transient tracking.

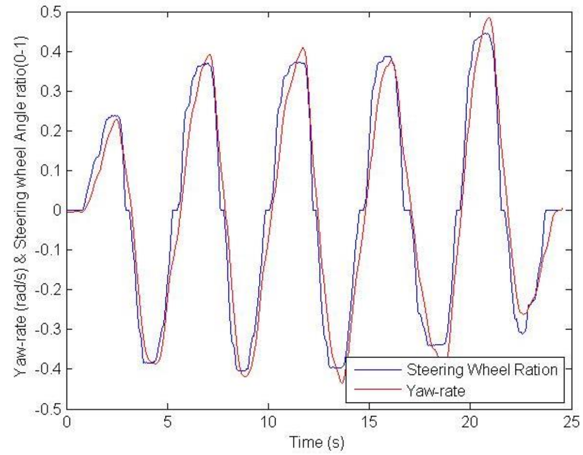

Figure 6. Relation between yaw-rate and steering input

In addition, Fig. 6 shows that due to the fast response of the algorithm, the actual yaw rate is basically synchronized with the driver's steering wheel input trend, which is more conducive to the driver's better driving experience.

III. CONCLUSION

According to the multi-purpose characteristics of the research object, combined with the driving characteristics of the skid steering vehicle, the steering control method of the skid steering vehicle, which can modify the wheel slip coefficient in real time and is not based on the tire model, is designed. In order to carry out the steering control method successfully, the corresponding control algorithm is designed based on the proportional integral control method with anti-integral-saturation. Through real vehicle verification, the algorithm has a quick response and good tracking effect. The experimental results show that the control response of the algorithm is rapid and the steady state tracking effect is satisfied. As for the further research of the wheel slip coefficient, it will be completed in the follow-up study.

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