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# Emergency Control Strategy for Distributed Skid-Steering Vehicles

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**Abstract:** Based on the lateral properties of tire and the input variable of braking force, a 2-DOF lateral dynamic analytical model of wheel skid-steered vehicle is established in this paper. Based on the model, the steady-state yaw velocity gain and steering radius in the process of skid-steering are analyzed theoretically. Then the control boundary and distribution strategy of the braking force are given. An all-terrain vehicle was used to carry out a real vehicle test, and verified that the strategy can ensure the safe steering of the vehicle when the driving force is lost.

**Key words:** distributed-drive; skid-steering; lateral dynamics; control strategy; emergency control

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Distributed driving vehicles with left and right wheels independently driven can change the moving direction of the vehicle by actively giving different tire speeds of two sides without swiping the front wheels[1]. As a unique steering method, its advantage is that the interior space of the vehicle is large, and there is no need to provide a dedicated mechanical steering system, which can simplify the vehicle structure and reduce weight[2].

Distributed-driving speed skid steered vehicles usually use two-sided motors to cooperate with each other to achieve driving and steering. If motor fails during driving, the tire on the corresponding side will not provide tractive force. In this case, the tractive force on both sides of the vehicle will be uncontrollable, and the driver cannot control the trajectory of the vehicle by adjusting the power output of the motor, the vehicle loses direction control[3].

As an important failure mode, the out-of-control direction of the vehicle caused by motor failure is fatal to vehicle safety. In this paper, by establishing an analytic model of skid steered vehicle with a input of tire forces, the vehicle motion law with motor failure is studied, and the control objectives and control boundaries under this conditions are analyzed and determined. A braking force distribution system is proposed for emergency control strategy, and the strategy is verified by experiment.

## 1 Kinematics Model

### 1.1 2-DOF analytical model for lateral dynamics

As shown in Fig. 1, a force analysis of the entire vehicle shows that the vehicle will be affected by the longitudinal and lateral forces of each tire and the road resistance $F_f$. Since the load transfer phenomenon has been ignored, the road resistance of the two tires on the same axle is equal, and at the center of mass, the combined torque generated by the two tires is zero.

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Therefore, in the situation described herein, the influence of road resistance $F_{\mathrm{f}}$ can be ignored[4-5].

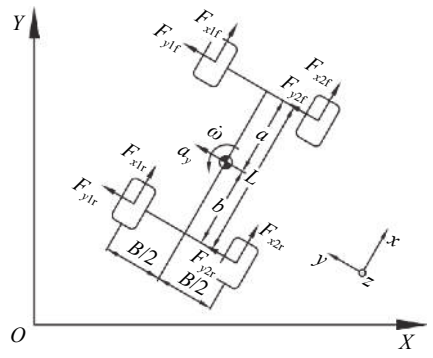


Fig. 1 Effect of tire force and road resistance on car body

Combining the above conclusions to list the force balance equations, a 3-DOF differential equation system as shown in the following formula can be established:

$$\begin{cases} ma_x = (F_{x1\mathrm{f}} + F_{x2\mathrm{f}} + F_{x1\mathrm{r}} + F_{x2\mathrm{r}}) \\ ma_y = (F_{y1\mathrm{f}} + F_{y2\mathrm{f}} + F_{y1\mathrm{r}} + F_{y2\mathrm{r}}) \\ I_z\alpha_z = \dfrac{B}{2}(-F_{x1\mathrm{f}} + F_{x2\mathrm{f}} - F_{x1\mathrm{r}} + F_{x2\mathrm{r}}) + \\ \qquad a(F_{y1\mathrm{f}} + F_{y2\mathrm{f}}) - b(F_{y1\mathrm{r}} + F_{y2\mathrm{r}}) \end{cases} \tag{1}$$

where $a_x$ and $a_y$ are the projection of the absolute acceleration of the center of mass on the $x$ and $y$ axis of the dynamic coordinate system; Subscript $x$, $y$ mean longitudinal and lateral respectively; Subscript 1, 2 mean inner and outer tires respectively; Subscript f, r mean front and rear axles respectively; $F_{\mathrm{f}}$ is the road resistance; $m$ is the quality of the whole vehicle; $I_z$ is the moment of inertia of the vehicle in the $z$ direction; $\alpha_z$ is the vertical angular acceleration; $a$ and $b$ are the distance between the center of mass of the vehicle and the front and rear axles, respectively; $B$ is the width of the vehicle.

From Eq. (1), there are

$$\begin{cases} a_x = \dot{u} - v\omega \\ a_y = \dot{v} + u\omega \\ \alpha_z = \dot{\omega} \end{cases} \tag{2}$$

where $u$ is the longitudinal velocity of vehicle; $v$ is the lateral velocity of vehicle; $\omega$ is the vertical angular velocity. So the dynamic equation of 3-DOF skid steered vehicle is

$$\begin{cases} m(\dot{u} - v\omega) = (F_{ZD1} + F_{ZD2}) \\ m(\dot{v} + u\omega) = -\dfrac{2}{u}(ak_{y\mathrm{f}} - bk_{y\mathrm{r}})\omega - \dfrac{2}{u}(k_{y\mathrm{f}} + k_{y\mathrm{r}})v \\ I_z\dot{\omega} = \dfrac{B}{2}(-F_{ZD1} + F_{ZD2}) - \dfrac{2}{u}(a^2k_{y\mathrm{f}} + b^2k_{y\mathrm{r}})\omega \\ I_z\dot{\omega} = -\dfrac{2}{u}(ak_{y\mathrm{f}} - bk_{y\mathrm{r}})v \end{cases} \tag{3}$$

where $F_{ZD}$ is the braking force of tires; $k_y$ is tire cornering stiffness.

In this process, assuming that the inside and outside braking forces are equal in magnitude and opposite in direction, and that the change in vehicle speed $u$ is small, approximately constant, then the equation degenerates into a 2-DOF form, and the results are summarized as

$$\begin{cases} \dot{v} = -\dfrac{2}{mu}(k_{y\mathrm{f}} + k_{y\mathrm{r}})v - \left[u + \dfrac{2}{mu}(ak_{y\mathrm{f}} - bk_{y\mathrm{r}})\right]\omega \\ \dot{\omega} = -\dfrac{2}{I_Zu}(ak_{y\mathrm{f}} - bk_{y\mathrm{r}})v - \\ \qquad \dfrac{2}{I_Zu}(a^2k_{y\mathrm{f}} + b^2k_{y\mathrm{r}})\omega + \dfrac{B}{I_{ZD}}F_{ZD} \end{cases} \tag{4}$$

### 1.2 Steady state steering radius

Let $\beta = \dfrac{v}{u}$, in the field of vehicle dynamics, then Eq. (4) can be rewritten as

$$\begin{cases} m(\dot{v} + u\omega) = -\dfrac{2}{u}(ak_{y\mathrm{f}} - bk_{y\mathrm{r}})\omega - 2(k_{y\mathrm{f}} + k_{y\mathrm{r}})\beta \\ I_z\dot{\omega} = -\dfrac{2}{u}(a^2k_{y\mathrm{f}} + b^2k_{y\mathrm{r}})\omega - \\ \qquad 2(ak_{y\mathrm{f}} - bk_{y\mathrm{r}})\beta + BF_{ZD} \end{cases} \tag{5}$$

From Eq. (5), let $\dot{v} = 0$, $\dot{\omega} = 0$, and eliminate $v$ to obtain the steady-state yaw rate gain expression as

$$\frac{\omega_{\mathrm{S}}}{BF_{ZD}/I_Z} = \frac{(k_{y\mathrm{f}} + k_{y\mathrm{r}})I_Zu}{2L^2k_{y\mathrm{f}}k_{y\mathrm{r}} + B(k_{y\mathrm{f}} + k_{y\mathrm{r}}) + (bk_{y\mathrm{r}} - ak_{y\mathrm{f}})mu^2} \tag{6}$$

When $ak_{\alpha\mathrm{f}} - bk_{\alpha\mathrm{r}} = 0$, the steady-state steering radius $R_{\mathrm{S}}$ at this time can be solved by

$$R_{\mathrm{S}} = \frac{u}{\omega_{\mathrm{S}}} = \frac{2L^2k_{y\mathrm{f}}k_{y\mathrm{r}} + B(k_{y\mathrm{f}} + k_{y\mathrm{r}})}{2(k_{yf} + k_{yr})}\frac{1}{F_{ZD}} = \frac{B}{2}\left[1 + \frac{2L^2k_{y\mathrm{f}}k_{y\mathrm{r}}}{B(k_{y\mathrm{f}} + k_{y\mathrm{r}})}\right]\frac{1}{F_{ZD}} \tag{7}$$

Steady-state steering radius is the reciprocal of braking force input.

## 2 Emergency Control Strategy

Due to the distributed driving of the skid steered vehicle, the available driving control means after the motor fails is only the brakes on two sides[6], so the brake at this time needs to achieve the function of reducing the vehicle speed to stop, but also need to cooperate with each other to complete vehicle trajectory control during parking.

### 2.1 Kinematics analysis of emergency condition

For the steering process of skid steered vehicles, it can be simply decomposed into straight-moving motion and yaw motion as shown in Fig. 2[7]. Next, the effect of braking force on the straight-moving motion will be considered.

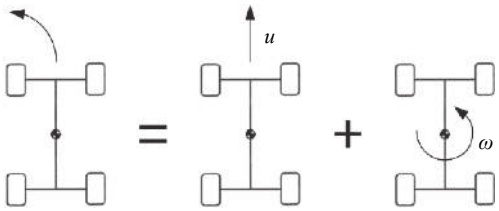


Fig. 2 Decomposition of turning process

For straight braking conditions, in addition to the braking force $F_{ZD}$, it is necessary to increase the braking force $F_{Z1}$ and $F_{Z2}$ in the negative x-axis direction of the inner and outer tires. These two forces are equal in magnitude and the same direction, and no torque in the $z$ direction is generated[8]. Let the braking force $F_{Z0} = F_{Z1} + F_{Z2}$, get

$$\dot{u} = \frac{F_{Z0}}{m} - v\omega_{\mathrm{r}} \tag{8}$$

Since it has been decomposed into linear motion in the $x$ direction, $v = 0$ and $\omega_r = 0$, so

$$\dot{u} = \frac{F_{Z0}}{m} \tag{9}$$

That is, in this process, the speed u of the vehicle in the forward direction can be obtained by dividing the braking force $F_{Z0}$ by the vehicle mass $m$ and then integrating. Combined with the conclusion of Eq. (6) steady-state yaw rate gain and Eq. (7) steady-state steering radius, the vehicle kinematic parameters at any time $T$ can be obtained as

$$\begin{cases} u(T) = \int_0^t \dfrac{F_{Z0}(t)}{m}\mathrm{d}t + u_0 \\ \omega_S(T) = \dfrac{B(k_{yf}+k_{yr})I_Z u(T)}{2L^2 k_{yf}k_{yr} + B(k_{yf}+k_{yr}) + (bk_{yr}-ak_{yf})m[u(T)]^2} F_{ZD}(T) \\ R_S(T) = \dfrac{2L^2 k_{yf}k_{yr} + B(k_{yf}+k_{yr}) + (bk_{yr}-ak_{yf})m[u(T)]^2}{(k_{yf}+k_{yr})I_Z} \dfrac{B}{I_Z} F_{ZD}(T) \end{cases} \tag{10}$$

where $F_{Z0}(t)$ is braking force, input amount; $F_{ZD}(T)$ is the balanced braking force of the inner and outer tires at time $T$, the input amount; $u_0$ is the initial value of longitudinal speed; $u(T)$ is longitudinal speed at time $T$; $\omega_s(T)$ is steady-state yaw rate at time $T$; $R_s(T)$ is steady turning radius at time $T$.

### 2.2 Boundary conditions of braking force control

The emergency control condition of the wheeled skid steered vehicle studied in this paper is characterized by the vehicle being in the combined condition of emergency braking and steering. Among them, deceleration is the main problem that needs to be solved. The steering is applied to ensure the safety and stability during parking. Therefore, the tires are mainly in the longitudinal braking condition with a large braking force, while the lateral force is small and the side deviation is not obvious. Therefore, the longitudinal slip rate of the tire in emergency control conditions should be analyzed and the value of the longitudinal slip rate of the tire in this process should be limited.

The braking force exerted on a tire is equal to the longitudinal force of this tire

$$F_{ZD} = F_x \tag{11}$$

Considering that the longitudinal force is related to the longitudinal slip rigidity and longitudinal slip rate of the longitudinal tire, there are

$$F_{ZD} = F_x = -k_x S_x \tag{12}$$

where $k_x$ is the longitudinal slip rigidity of the tire; $S_x$ is the longitudinal slip ratio of the tire, $S_x = v_{sx}/u$. And,

$$v_{sx} = v_{sx1f} = v_{sx1r} = \Delta u - \frac{B}{2}\omega \tag{13}$$

where $v_{sx}$ is the longitudinal sliding speed of the tire; $\Delta u$ is the amount of change in tire speed, and so

$$S_x = \frac{1}{u}\left(\Delta u - \frac{B}{2}\omega\right) \tag{14}$$

This article assumes that the tire slip rate is less than 10%, so $\Delta u$ should meet the following conditions[9] as

$$\Delta u = S_x u + \frac{B}{2}\omega,\ S_x < 10\% \tag{15}$$

That is, when performing emergency control operations, the value of $\Delta u$ should be monitored to satisfy at any time $T$

$$\Delta u\,(T) < 0.1u\,(T) + \frac{B}{2}\omega_s\,(T) \tag{16}$$

where $\Delta u\,(T)$ is the amount of change in tire speed at time $T$; $u\,(T)$ is longitudinal speed at time $T$; $\omega_s\,(T)$ is steady yaw rate at time $T$.

### 2.3 Emergency control process

When a motor failure occurs, the control system should immediately set the vehicle to a state where both sides of the driving force are zero, whether it is a single-sided motor failure or a double-sided motor failure. After that, the electronic hydraulic brake system (EHB) distributes different braking forces to the brakes on both sides to enable the vehicle to achieve emergency steering.

#### 2.3.1 Brake force output limitation strategy

In the emergency treatment process, it is necessary to control the braking force on both sides to realize the steering and braking of the vehicle. In order to ensure that the vehicle trajectory can be controlled during braking, the wheel braking force should not exceed the attachment limit, which is the limiting condition for the braking force in the emergency control strategy. In practice, the tire braking force during braking is not easy to measure, so this paper gives the boundary condition using the change in tire rotation speed $\Delta u$ as an independent variable as shown in Eq. (16). When the braking force is applied, the control flow of the braking force of the inner and outer wheels is shown in Fig. 3.

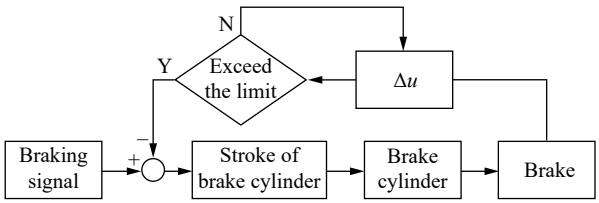


Fig. 3 Braking force output restriction strategy

#### 2.3.2 Braking force distribution strategy

Simultaneous failure of the double-sided motor is a relatively easy-to-handle situation, because at this time the vehicle is in a free-running state, there is no additional yaw moment, and the vehicle trajectory is relatively stable. When a single-sided motor fails, although the vehicle theoretically still has half of power, it is affected by the form of distributed drive system, the power can only be output to the single-sided wheels[10]. The unilateral driving force will produce a yaw moment sufficient to affect driving safety, which will have an impact on driving safety. Therefore, the degree of danger and difficulty in dealing with loss of power on one side are higher than those on two sides. At this time, in order to ensure the stable and controllable trajectory of the vehicle, the non-failed motor should be manually set to a free rotation state that neither outputs nor recovers power, and then control the vehicle. After cutting off the power output of the motor, the handling process of the two situations is the same: in the event of a motor failure, whether it is a single-sided failure or a double-sided failure, the driver should first turn on the emergency control switch to set the motors on two sides to free rotation. Motor torque target is set to zero, and depress the brake pedal to slow down the vehicle, while operating the steering wheel is in-

tended to give the vehicle a controllable steering radius. The emergency control strategy intervenes in the emergency treatment process with the flow shown in Fig. 4.

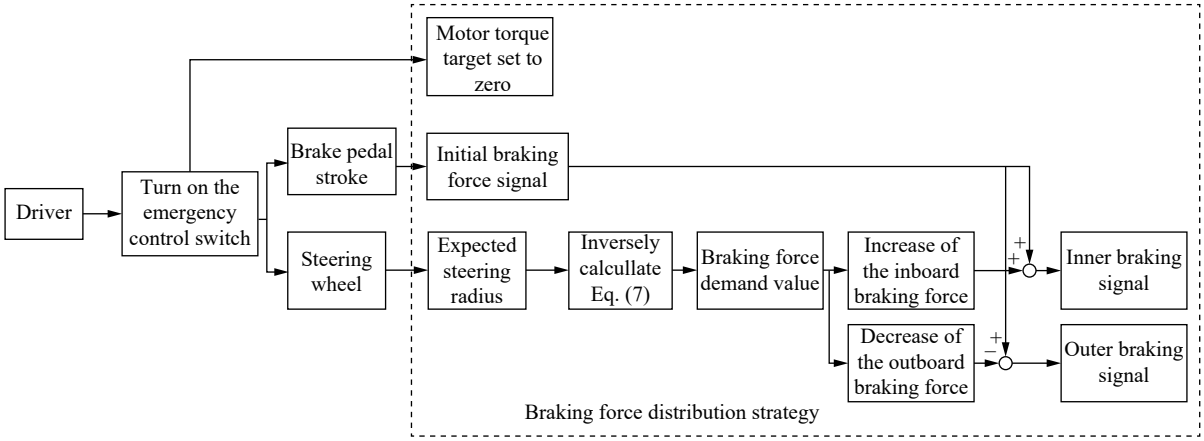


Fig. 4 Braking force distribution strategy

After the driver turns on the emergency control switch, the emergency control strategy will take over the vehicle control. The target torque of the motor is set to zero, so that the two-sided motor does not output power at the same time. The control system will calculate the driver's expected steering radius, and use the expected steering radius value to inversely calculate Eq. (7) to obtain the increase of the inboard braking force and the decrease of the outboard braking force when braking and turning. The braking force demand value is calculated based on the initial braking force signal generated based on the brake pedal stroke. The inner braking signal and the outer braking signal are calculated and output as the input of Fig. 3 to the inner and outer braking cylinders, and finally the inner and outer braking force differential control is realized.

## 3 Experimental Verification

To verify the effectiveness of the emergency control strategy, an emergency control experiment was designed and carried out for a 6×6 all-terrain vehicle. Through the subjective evaluation method, the effect of emergency control capability in the event of loss of power failure (high-voltage power failure) in straight driving is evaluated.

### 3.1 Experiment preparation

The experimental vehicle is a distributed drive 6×6 skid steered all-terrain vehicle developed and produced by Jialing Co. The experimental vehicle is shown in Fig. 5.

Fig. 5 Experimental vehicle

The experimental vehicle is loaded with a maximum total mass of 3 500 kg. In addition to the quality of the driver, experimenter, equipment, etc., the load inside the cab is arranged on the seat and the floor with a load bag; the load should be fixed firmly, and it should not be swayed or distorted during the experiment; it should not be lost due to moisture or loss When the conditions change, the quality is changed to ensure that the size and distribution of the loading quality remain unchanged. The loads outside the cab are counterweight cargo boxes and load bags. The experimental vehicle will carry out various experiments with the gross vehicle weight (GVW). During the experiment, the tire pres-

sure is the pavement air pressure, and the cold inflation pressure of the tire conforms to the vehicle technical specifications.

The experimental site is located in Chongqing city. The pavement is paved with concrete and has a total length of about 1 500 m, of which the length of the straight line is about 1 100 m, the width is 14 m, and the longitudinal slope is zero. The ambient temperature is 23±5 ℃, the air pressure is 99.8±5 kPa, and the wind speed is 2.0±1 m/s.

### 3.2 Experimental process

#### 3.2.1 Experimental conditions

The test site is a comprehensive performance test site, the road's natural environmental conditions, the component of wind speed in the direction of vehicle travel should not exceed 3m/s, and the experiment is carried out under no-load mass and full combat weight, respectively. Connect the external equipment that simulates high-voltage power-down before the experiment. The number of drivers participating in the subjective evaluation should not be less than 3.

#### 3.2.2 Experimental standard

Set 5 grades. See Tab. 1 for evaluation grade, corresponding score and evaluation standard.

#### 3.2.3 Experimental method

The test vehicle travels straight at a specified speed at a constant speed. After the speed is stable, the simulated high-voltage power-off device is activated, and the steering wheel is controlled to stop according to the driver's intended driving route. The parking brake can be stepped on at the same time during the parking process. Evaluate the data; the experimental vehicle speed starts from 20 km/h to 70 km/h and ends with the no-load experiment and then the full-load experiment.

**Tab. 1 Evaluation standard**

| Content | Standard | | | | |
|---|---|---|---|---|---|
| | 5 | 4 | 3 | 2 | 1 |
| Straight-line power-off and side parking ability | Control the direction after power off without any deviation from the driver's expected driving route | Control the direction after power off, the driving route is clear and clear | After power off, the control direction can be controlled intentionally, slightly offset from the driver's expected driving route | The direction is not controlled, and there is a significant deviation from the driver's expected driving route after power off | After power off, the direction cannot be controlled, which seriously affects driving safety |

### 3.3 Experimental results

The experiment was carried out in accordance with the method clearly defined in the experiment outline, and a subjective evaluation experiment was carried out on the ability of straight-line high-voltage power-off and side-by-side parking. The experimental results are shown in Tab. 2.

**Tab. 2 Results of the subjective evaluation experiment**

| No. | Speed/(km·h$^{-1}$) | Result | | |
|---|---|---|---|---|
| | | Driver A | Driver B | Driver C |
| 1 | 20 | 5.0 | 5.0 | 5.0 |
| 2 | 30 | 5.0 | 4.0 | 5.0 |
| 3 | 40 | 5.0 | 4.0 | 4.0 |
| 4 | 50 | 4.0 | 4.0 | 4.0 |
| 5 | 60 | 4.0 | 4.0 | 3.0 |
| 6 | 70 | 4.0 | 3.0 | 3.0 |

The photos of the experiment site are shown in Fig. 6. The experimental results show that the high-voltage power-off during straight-line driving, the vehicle speed above 30 km/h can stop sideways, and the emergency control process is basically stable.

Fig. 6 Experimental site

## 4 Conclusions

This paper selects the braking force as the input to model the skid steered vehicle, develops the emergency control strategy research of the wheeled skid steered vehicle, proposes the braking force control boundary and distribution strategy for emergency control, and performs emergency control. The actual vehicle experiment has formed a set of emergency control methods suitable for distributed-driven wheeled skid steered vehicles, which provides a theoretical basis and practical basis for the in-depth study of the design of wheeled differential vehicle control systems. The main research conclusions of this article are as follows.

① Based on the braking force input model, the 2-DOF linear differential equation of the skid steered vehicle is established, and the equation of the steady-state steering radius of the skid steered vehicle under the braking force is given, and it is pointed out that under a certain braking force, the steady turning radius changes with the speed of the vehicle.

② By analyzing the kinematic characteristics of the wheeled skid steered vehicle under emergency control conditions, combined with the actual working state of the tire during steering, the relationship between the longitudinal slip rate and the braking force is obtained, and the control boundary and distribution strategy is given. Through experiments, the emergency control strategy was verified.

③ An emergency control strategy is proposed: during emergency control, the direction stability of the vehicle should be ensured first. After the direction is controlled, the vehicle speed should be moderately reduced until it stops; the braking force should be applied on the left and right sides at the same time, and the steering direction according to the demand. Adjust the size of the braking force; in order to ensure that the tire does not slip significantly, the value of the tire rotation speed change $\Delta u$ should be controlled.

④ Some deficiencies still remain to be improve. The electronic stability programs and functional safety design will be considered to improve the safety of vehicles.

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