A novel control method for electric power steering system based on dead-zone inverse transforming compensation in a four-in-wheel-motor drive electric vehicle

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Abstract: Since the four-in-wheel-motors drive (FWMD) electric vehicle (EV) has a four-wheel independent steering, this paper presents a novel control method for an electric power steering system with a dead-zone execution mechanism. A mathematical model was derived for execution mechanisms from the empirical data. A ‘dead-zone inverse transforming compensation’ was designed to eliminate ill effects caused by dead-zone. The experiment in a real car environment proves the proposed method.

Keywords: EV; electric power steering system; steering control; dead-zone inverse transforming compensation.

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1 Introduction
Differing from the electric power assisted steering system; the electric power steering system has a motor to offer the entire steering power and works in the line control mode (Yuan, 2007).

The advantages of the electric power steering system are as below (Dong et al., 2010):

1 The original steering device in the vehicle has been cancelled to simplify the system structure to be a ‘steering wheel-ECU-motor’.

2 The steering mechanism can be in a flexible layout. It can be fixed on the front wheel or the rear wheel or on the four wheels.

3 The pivot steering and transverse turning, which are impossible for the traditional vehicles, can be cooperated with the drive system.

However the electric power steering system also has its own disadvantages as below:

1 The electric power steering system has a heavy steering load and is unable to draw the support from the mechanical or hydraulic power therefore the motor is required to output a great torque. Though the problem can be solved by the help of retarding mechanism, its speed ratio cannot be too large otherwise the relevant speed of the steering will be severely affected. Therefore, the torque balance is required.

2 There is a non-mechanical connection between the wheels therefore an ECU is applied to control the angles of the left wheel and the right wheel separately. Any minor deviation in the angle control will leave the vehicle in risk when it runs at high speed.

3 The heavy load of the motor and the friction of the retarding mechanism may be caused to fall in the dead zone. Whilst in the ECU control, the control signal from the motor fails to be directly proportional; the theoretical and simulation effect is unable to be achieved.

This paper has offered a solution (Tian and Tao, 1997) to the third disadvantage mentioned above.

2 Object and structure of steering system
The study of the paper has been carried out on a test vehicle which is a modified Chery QQ. The test electric vehicle has a four-wheel independent drive line control. The engine, string system and suspension have been moved from the Chery QQ and an electrical driving system and an electrical steering system have been installed (Deng et al., 2010). The test vehicle weights 1,144 kg and the wheelbase is 1,318 mm. The wheel-centre-distance is 2,114 mm. The overview of the test vehicle is as shown in Figure 1.

![Figure 1](https://example.com/figure1)

The overview of the test vehicle (see online version for colours)

The designed steering mechanism is shown in Figure 2.

![Figure 2](https://example.com/figure2)

The real steering mechanism (see online version for colours)
The control system structure is shown as Figure 3. In this diagram, $\theta$ refers to the angle of the steering wheel. The control system will calculate the expected angle of the corresponding wheel according to the angle of the steering wheel and compare it with $\alpha$ (real angle of the wheel) to calculate $u$ (controlled variable) through the controlling algorithm then transfer to the controller and act on the wheel through motor, construct a closed-loop control of angle.

### 3 Mathematical model of steering control system

Based on the theoretical analysis and the requirement of simulation and parameter determination, the following obtains the mathematical models of angular speed to wheel angular and the mathematical models of steering mechanism through acquisition and analysis of the test data as well as theoretical derivation. The two kinds of models make up the control objects of the control system. As it is the computer control system, the control algorithm and object must be described with the discrete mathematics model.

1. Discrete mathematics model of angular speed to wheel angular ($\varpi \rightarrow \alpha$).

According to the definition of the angular speed, the following differential equation can be used to describe the relations between the wheel corner and the angular speed.

$$\alpha_n = \alpha_{n-1} + \varpi t_s$$  

(1)

In the above equation, $\alpha$ refers to the wheel angle, its unit is ‘angular degree’ (°); $\varpi$ refers to the rotating angular speed of the wheel, its unit is ‘angular degree per second’ (°/S); $t_s$ means the sampling cycle, its unit is ‘second’ (S).

2. Discrete mathematics model of the steering mechanism ($u \rightarrow \varpi$).

The discrete mathematics model of the steering mechanism means the relations between the controlled variable $u$ ($-5 \text{ V} \sim +5 \text{ V}$) output by the control system and rotating angular speed of the wheel $\varpi$. The characteristic curve achieved by the experiment is as shown in Figure 4.

The experiment data is shown as the following three important features of the steering mechanism:

1. the different resistance for the left steering and the right steering of the wheel results in the different angular speed $\varpi$ of the same controlled variable $u$ in different rotating directions.
2. the angular speed and the controlled variable in two directions are in linear relations.
3. there are dead zones of $\Delta^+$ and $\Delta^-$ in two directions and the two threshold values are different.

Figure 4 Characteristic of the steering mechanism

We are able to optimise the discrete mathematics model of the steering mechanism [see equation (2)] based on the above analysis (Wen et al., 2010).

$$\varpi_n = \begin{cases} \varpi^- (u_n - \Delta^-) & u_n \leq \Delta^- \\ 0 & \Delta^- < u_n < \Delta^+ \\ \varpi^+ (u_n - \Delta^+) & \Delta^+ < u_n \end{cases}$$  

(2)

In equation (2), the units of $\Delta^+$, $\Delta^-$, $u_n$ are ‘volt’(V) and the unit of $\varpi^+$, $\varpi^-$ are ‘angular degree/second · volt’ (°/SV).

In the steering system of the paper,

$$\Delta^+ = 0.47V, \quad \varpi^+ = 45^\circ / SV$$

$$\Delta^- = -0.45V, \quad \varpi^- = 41^\circ / SV$$

### 4 Dead-zone inverse transforming compensation

The essence of the dead-zone inverse transforming compensation is to change the non-linear system with dead zones to a linear system through the inverse function compensation and then achieve the best effect of PID control after transformation, which will be quicker in eliminating the offset than an invariable system (Huang et al., 2010). The dead-zone inverse transforming compensation is a kind of software algorithm which is able to eliminate the impact of the dead zone in the execution mechanism when keeping the hardware system unchanged.
to make the control object present favourable linear relations against the PID algorithm during control. Its position in the control system is as shown in Figure 5.

**Figure 5** Position of the dead-zone inverse transforming compensation in the system

The dead-zone inverse transforming compensation shall be carried out according to the following procedures:

1. determine the target linear function \( \sigma = f(u) = Ku \); and the slope \( K \) will be decided by the user
2. derive the transfer function \( \sigma = g(u') \) of the execution mechanism according to the experiment data
3. get the dead-zone inverse transforming compensation function.

\[
u' = h(u) = g^{-1}(f(u)) \tag{3}\]

Prove for the method of inverse transforming compensation is shown in equation (4).

\[
\sigma = g(u') = g(h(u)) = g(g^{-1}(f(u))) = f(u) \tag{4}\]

Based on the above analysis, when carrying out dead-zone inverse transforming compensation to the control signal \( u \), the angular speed of the wheel \( \sigma \) and the control signal are in linear relation, namely \( \sigma = f(u) = Ku \); therefore the objective of linearisation has been reached.

Taking the control objective as an example, the dead-zone inverse transforming compensation function is shown in equation (5).

\[
u' = \begin{cases} 
\Delta^* + \frac{Ku}{K^*} & u \geq 0 \\
\Delta^* + \frac{Ku}{K^-} & u < 0 
\end{cases} \tag{5}\]

The course of dead-zone inverse transforming compensation can be illustrated as the Figure 6.

Judging at Figure 6, the dead zone for the object of the PID control has been eliminated after inverse transforming. Favourable linearity has been presented. \( K = 1 \) is applied for the steering system mentioned in the paper.

5 MATLAB simulation validation and parameter setting of dead-zone inverse transforming compensation

Input the mathematical model and control algorithm deduced above into the MATLAB for software simulation (Liu, 2003) to achieve the control effect under two kinds of situations.

**Figure 6** Diagram of dead-zone inverse transforming

When not dealing with the dead zone according to the ordinary PID control method, major offset exists in the control result due to big dead zone. The proportion \( kp \) must be increased or the integration time \( ti \) must be decreased in order to eliminate the offset; therefore great overshoot may be caused. The method is to reduce the overshoot to increase the differential time; however shaking emerges which results in unsatisfactory effect (Wu and Wang, 2006). The Figure 7 shows relatively better control effect. The offset is \( 0.83^\circ ~5.97^\circ \) and the PID control parameters are \( ts = 0.5 \text{ s}, \ kp = 0.025, \ ti = 5 \text{ s}, \ td = 1 \text{ s} \).

**Figure 7** General PID control effect

After the dead zone is eliminated through inverse transforming compensation, the offset of the control effect is minor; therefore better control effect can be achieved by regulating other parameters. The offset is \( 0.04^\circ ~0.13^\circ \) as shown in Figure 8 and the PID control parameters are as follows: \( ts = 0.5 \text{ s}, \ kp = 0.7, \ ti = 600 \text{ s}, \ td = 0.1 \text{ s} \).
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Figure 8  Control effect of PID with dead-zone inverse transforming compensation

6  Vehicle validation

The parameter determined by the MATLAB simulation is used for the control of vehicle in real-time. The angle target value and the angle feedback value during the control will be uploaded to the computer through the serial communication interface in the ECU. The input data received by the computer has been converted into the EXCEL format to draw a curve to display the control effect.

The Figure 9 shows the test of real vehicle controlled by PID. It is found that during the test that the proportion coefficient $k_p$ for simulation in Figure 7 is too small due to the great impact of the dead zone; therefore $k_p$ will be modified to 0.05 in the test, i.e., the control result in the Figure 9 is obtained when the control parameters are as follows: $t_s = 0.5$ s, $k_p = 0.05$, $t_i = 5$ s, $t_d = 1$ s. The offset is then $3.95^\circ$-$4.24^\circ$.

Figure 9  General PID real vehicle control effect

Figure 10 shows the real vehicle control effect of PID with Dead-zone inverse transforming compensation. The control parameters used are the same as the simulation parameters in Figure 8; however when the method is applied in the real vehicle, sometimes the wheel may shake slightly if equation (5) is directly used. The reason lies in that there is slight shift for $\Delta^+$ and $\Delta^-$ in equation (5) due to the mechanical lubrication and electric circuit temperature shift. If the real dead zone is smaller than that in the algorithm, the control system will output the control variable which is bigger than that required actually; thus shaking results. Therefore equation (5) has been improved to the following equation (6) during the test.

$$u' = \begin{cases} 
0.95\Delta^+ + \frac{K_u}{K^*_i} & u \geq 0 \\
0.95\Delta^- + \frac{K_u}{K^-} & u < 0 
\end{cases}$$

(6)

In equation (6), the dead zone is modified to 0.95 times as large as the measured one, i.e., 0.05 $\Delta$ control allowance is reserved to prevent the shrinkage of the real dead zone. In the other hand, there is a dead zone of about 0.05 $\Delta$ after using equation (6) for dead-zone inverse transforming compensation (The dead zone may change depending on the mechanical lubrication or the electric circuit temperature shift.); however judging from the Figureure of real control effect, the method eliminates 95% of the total dead zones actually and achieves more stable control effect with smaller offset which is $0.28^\circ$-$0.42^\circ$.

Figure 10  Real vehicle control effect of PID with dead-zone inverse transforming compensation

7  Conclusions

The power of the electric vehicles is basically offered by the motor whose execution mechanism may have bigger dead zone when bearing heavier load. The dead zone affects the control effect, which may easily cause great offset. The method introduced in this paper is proved to be effective upon the simulation and real vehicle tests and can be taken as a way to eliminate the impact of the dead zone.

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