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Research in Torque Distribution of Dual Motor Electric Vehicle Based on PSO

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Abstract. This paper proposed real-time optimization algorithm in order to solve the problem which the dynamic nonlinear torque distribution of rear dual-hub-motor-driven vehicle. Based on the research of a company's rear-wheel drive double-wheel hub motor, build the control model by MATLAB, the vehicle model set up by CarSim, and Co-simulation test the performance of algorithm. Then hardware-in-loop test based on the VCU. The result shown that the algorithm could distributed torque according to the actual working conditions, reduced the slip ratio, improved the driving performance of vehicle.

1. Introduction

The hub motors integrated drive parts and brake parts, omitted traditional differential parts, transmission shafts and other transmission components. So, the hub motor became the hot direction of new energy vehicles because of the small size and the control flexibility [1~2].

In the driving process, the acceleration slip regulation system (ASR) could achieve greater driving force which improve the dynamic quality [3~4]. But the new energy vehicle driven by hub motor didn’t have a mechanical differential, and the small new energy vehicle could not install the structure which is necessary to the ASR due to space constraint, so this paper calculated the torque output by right and left motors in order to realize the function of differential and ASR in the driving process.

Many scholars had always ignored the increase of unsprung mass caused by the hub motors [5~8], which would change the loads of the driving wheel in steering. Driving wheels with larger loads could withstand greater torque without slip rate increased. So, based on the analysis of vehicle dynamics, the left and right driving wheel torque is analyzed and corrected in this paper.

2. Vehicle dynamics model

2.1. Rear axle drive wheel load

When the vehicle starts on road with a certain slope angle, The vehicle dynamics diagram was shown in the Figure 1. the load of the rear axle drive wheel is as shown in Formula (1) [9]:

\[
F_{2R} = G \left( \frac{L}{L} \cdot \cos \varphi \right) - \frac{h}{L} \sin \varphi + \frac{Gh \cdot a_z}{gL}
\]
In the formula (1), $F_{ZR}$—the rear axle load; $G$—actually the weight of the vehicle; $L$—the wheelbase of the vehicle; $L_1$—distance from the centroid to the front axle; $h_g$—height of the centroid; $g$—acceleration of gravity, $9.8 \text{ m/s}^2$; $\Phi_r$—the longitudinal slope angle of road, $\Phi_r > 0$ means the vehicle goes uphill.

When the transverse slope angle of the road is $0^\circ$, the loads on the left and right wheels are as follows the formula (2):

$$F_{ZL2} = F_{ZR2} = \frac{1}{2} \cdot F_{ZR}$$

(2)

In the formula (2): $F_{ZL2}$—load on the left driving wheel; $F_{ZR2}$—load on the left driving wheel.

![Diagram of Vehicle Dynamics](image)

**Figure 1.** Diagram of Vehicle Dynamics.

2.2. Load offset caused by sprung mass in steering

The sprung mass of the vehicle was distributed to the front axle and rear axle connecting center $M_F$ and $M_R$. When the vehicle in steering, the load offset on the rear axle caused by the sprung mass is shown in Formula (3):

$$F_{ms} = \frac{m_s \cdot u^2 \cdot L_1 \cdot h_R}{R \cdot L \cdot B_R}$$

(3)

In the formula (3), $m_s$—sprung mass; $h_R$—length of centroid to the ground; $B_R$—the track of rear axle; $R$—Steering radius, calculating $R$ by Ackermann steering model, shown as the formula (4).

$$R = \sqrt{\left(\frac{L}{\tan \delta}\right)^2 + L_1^2}$$

(4)

2.3. Load offset caused by suspension roll stiffness recovery torque

In steering, the rear axle suspension generates a restoring moment by the suspension mass, which also affects the vertical load of the left and right drive wheels. The moment shown in formula (5):

$$F_{TR} = \frac{T_{\phi_s}}{B_R} - \frac{K_{\phi_s} \cdot \Phi_s}{B_R} = \frac{K_{\phi_s} \cdot (\Phi_s - \Phi_0)}{B_R}$$

(5)

In the formula (5): $\Phi_s$—the roll angle of the vehicle body; $\Phi_0$—landscape slope angle of the road surface; $K_{\phi_s}$—roll stiffness of the rear axle of the body.

2.4. Load offset caused by unsprung mass in steering

The hub motors change the traditional drive motor as suspension mass to non-suspension mass. With the increase of unsprung mass, it’s necessary to consider the load offset on the left and right wheel loads caused by the centrifugal force during steering. The load offset shown in formula (6):
In the formula (6): \( m_{\text{Rus}} \)—unsprung mass on the rear axle; \( h_{\text{Rus}} \)—height of the rear axle unsprung mass centroid; Here, \( h_{\text{Rus}} \) is equal to the rolling radius \( r \) of the driving wheel.

3. Rear drive wheel torque distribution

3.1. Torque distributed by the vertical load ratio of the drive wheel

The electronic acceleration pedal was often used in new energy vehicles to replace the mechanical throttle pedal, which facilitates the torque distribution. The output voltage of the electronic accelerator pedal is 0~5V [10], the pedal opening angle could be calculated by the output voltage signal of the electronic pedal through AD converter. Sign \( K_P \) is the voltage of pedal output, Multiply the maximum torque of the hub motor, inferred the torque required by the driver, and mark it as \( T_E \).

Mark \( C \) as the ratio of right rear wheel to left rear wheel vertical load in steering, shown in the formula (7). The preliminary torque distribution shown in formula (8).

\[
C = \frac{F_{ZL2}}{F_{ZR2}}
\]

\[
\begin{align*}
T_{L21} &= T_E \frac{C}{1+C} \\
T_{R21} &= T_E \frac{1}{1+C}
\end{align*}
\]  

In the formula (8): \( T_{L21} \)—the preliminary torque of left drive wheel; \( T_{R21} \)—the preliminary torque of right drive wheel.

3.2. Torque correction

According to the vehicle dynamics model, the torque distribution in the formula (8) is only proportionally distributed according to the load of drive wheels. However, the condition of the road surface is complicated and varied. In order to make full use of the road adhesive force, it’s necessary to identify the type of road.

3.2.1. Pavement observation system. Burckhardt et al. [11] proposed the \( \mu-S \) curve that accurately describes the relationship between the wheel slip ratio \( S \) and the road surface adhesion coefficient \( \mu \) under different road conditions:

\[
\mu(S) = C_1 \left( 1 - e^{-C_2 \cdot S} \right) - C_3 \cdot S
\]

\[
S_{\text{max}} = \frac{1}{C_1} \cdot \log \left( \frac{C_1 \cdot C_2}{C_3} \right)
\]

The formula (9) (10) shown that the optimum slip rate and the corresponding optimum attachment coefficient. In the formula (10): \( C_1, C_2, C_3 \) are fitting curve coefficient. The road surface recognition system is not the main research content, so in this paper, the on-line monitoring method of the road surface observation system quoted the reference [12]. The observation system output the optimal slip rate for the left and right driving wheels, and respectively marked as \( S_{R1} \) and \( S_{R2} \) respectively.

3.3. Torque output optimized by PSO

The driving condition variation was complex and non-linear. It’s necessary to adjust the output torque adaptively according to the road type. The robustness of the driving system is too low when using the conventional fixed parameter controllers, just like PID controller, couldn’t adjust the control model parameters according to the driving condition in time, so this paper brought the real-time optimization algorithm to corrects the torque output. The Particle Swarm Optimization (PSO) had many advantages,
such as easily implementation, less parameters and didn’t need decode and code. Experts and scholars in many fields had a lot of research [13–14], so PSO was used to correct the torque on-line in running process.

The objects need to optimized in this paper were correction coefficient $a_{L2}$ and $a_{R2}$, shown in the formula (11):

$$
\begin{align*}
\begin{cases}
a_{L2} = \frac{T_{L2}}{T_{L21}} \\
a_{R2} = \frac{T_{R2}}{T_{R21}}
\end{cases}
\end{align*}
$$

(11)

In the formula (11): $T_{L2}$—the final torque output of rear left driving wheel; $T_{R2}$—the final torque output of rear right driving wheel.

In the actual drive ASR control, the dominant relationship between the ASR system and the driver’s demand for torque must be considered, and the ASR system couldn’t affect the driver’s dominance of the vehicle driving torque control [15]. According to the above considerations, the range of PSO search is set to $[0 \leq a_{L2} \leq 0.8] \cup [0 \leq a_{R2} \leq 0.8]$.

After randomly distributing all the particles in the search area, calculate the value of fitness function, obtain the global optimal solution by comparison, and named $g_{best}$; with the iterations gradual increase, every particle had its best solution, and named $p_{best}$; the global optimal solution and the particle optimal solution are updated to the next iteration. At the same time, the searching direction and searching speed of a particle are updated by formula (12), while the absolute value of $g_{best}$ less than 0.01, output the optimized $a_{L2}$ and $a_{R2}$. That was the general process of optimization.

$$
\begin{align*}
\begin{cases}
v_{j}(t+1) = \omega \cdot v_{j}(t) + c_{1} r_{1} (p_{bestj}(t) - x_{j}(t)) + c_{2} r_{2} (g_{bestj}(t) - x_{j}(t)) \\
x_{j}(t+1) = x_{j}(t) + v_{j}(t)
\end{cases}
\end{align*}
$$

(12)

In the formula (13): $v_{j}$—the velocity of single particle; $x_{j}$—the position of single particle; $c_{1}$, $c_{2}$—the learning factor, based on the literature [14], the value were 2; $\omega$—inertia weight, In order to solve the defect that PSO has some probability falling into partial optimum value, nonlinear curves used to change $\omega$ during the calculation, shown in formula (13), the $\omega_{max}$=0.9, $\omega_{min}$=0.4, $d_{1}$ and $d_{2}$ were control factor, in this paper, the values were $d_{1}$=0.2, $d_{2}$=0.7. The fitness function was as follows the (14).

$$
\omega = (\omega_{max} - \omega_{min} - d_{1}) \cdot e^{\frac{(t+1) \cdot d_{2}}{\omega_{max}}}
$$

(13)

$$
S = \frac{1}{4} \left[ \left( \frac{r \cdot \omega_{L2} - u_{L2} - S_{R1}}{r \cdot \omega_{L2} - S_{R1}} \right)^{2} + \left( \frac{r \cdot \omega_{R2} - u_{R2} - S_{R2}}{r \cdot \omega_{R2} - S_{R2}} \right)^{2} \right]^{1/2}
$$

(14)

In the formula (14), there were some parameters need to confirmed: the next CAN message cycle rotation speed $(\omega_{L2}, \omega_{R2})$ and driving speed $(u_{L2}, u_{R2})$. Then calculated the left rotation speed $(\omega_{L2})$, driving speed $(u_{L2})$ as example.

The following is based on the left rear wheel 1/4 vehicle dynamics model to infer the driving speed and rotating speed of the next sampling time, the 1/4 vehicle model shown in formula (15):

$$
\begin{align*}
\begin{cases}
m_{L2} \cdot u_{L2} = F_{L2} \cdot \mu(S) \\
J \cdot \omega_{L2} = T_{L2} - r \cdot F_{L2} - T_{f}
\end{cases}
\end{align*}
$$

(15)

$$
m_{L2} = \frac{1}{2} \cdot \frac{L_{1}}{L_{1} + L_{2}} \cdot m_{s} = \frac{L_{1} \cdot m_{s}}{2L}
$$

(16)

In the formula (15): $u'_{L2}$ is the acceleration of driving wheel; $\omega'_{L2}$ is angular acceleration; $T_{f}$ is rolling resistance torque of tire; $\mu(S)$ shown in formula (9); $J$ is the moment of inertia of wheel; $m_{L2}$ is the load of 1/4 vehicle dynamics model and shown in formula (16).

Through the above calculation, the next sampling time of rotation speed($\omega_{L2}$) and the drive speed ($u_{L2}$) calculated under the output torque are as follows:
\[
\begin{align*}
\dot{u}_{e2} &= u_{e2} + \frac{2F_{e2} \cdot \mu(S_e)}{L_2 \cdot m_2} \cdot L_1 \cdot m_1 \\
\dot{\omega}_{e2} &= \dot{\omega}_{e2} + \frac{1}{2} \left( \frac{F_{e2} \cdot \mu(L_1) \cdot r - T_j}{m_2} \right) dt
\end{align*}
\]

In the formula (17): \(u_{e2}\) — the driving speed of left rear driving wheel in real time, \(\omega_{e2}\) — the rotation speed in real time. If the speed of the driving wheel was collected directly from the sensor, because of the slip ratio of the driving wheel, there would be deviation with the \(u_{e2}\). Therefore, a hall sensor was set up on the left front wheel, and the drive speed of the left rear wheel was calculated by Ackermann steering model (Figure 2) from the real-time collected left front wheel rotation speed \(\omega_{eF1}\) and the result shown in formula (18):

\[
u_{eR2} = \omega_{eF1} \cdot r \cdot \frac{2L - B_L \cdot \tan \delta}{2 \cdot \tan \delta \left[ \left( \frac{L}{\tan \delta} - \frac{B_L}{2} \right)^2 + L^2 \right]^{1/2}}
\]

The real-time rotation speed could be measured directly by a speed sensor mounted near the wheel brake disc.

![Figure 2. Ackermann steering model.](image)

In the same way, the rotation speed \(\omega_{eR2}\) and drive speed \(u_{eR2}\) could be calculated.

### 3.4. The second calculation of torque output by hub motor

After the PSO optimized the torque distribution according to the real-time driving condition, the output torque of the driving wheel is shown in the formula (19):

![Figure 3. Schematic diagram of vehicle driving control.](image)
\[
\begin{align*}
T_{L2} &= \alpha_{R1} \cdot T_{L21} \\
T_{R2} &= \alpha_{R2} \cdot T_{R21}
\end{align*}
\]  

(19)

As shown in the Figure 3, The output torque of the left and right driving wheels in formula (19) is calculated in VCU and transmitted to the MCUs through P-CAN. (MCU: motor control unit), the speed of driving wheel collected by hall sensor, and the speed signals transferred to VCU through P-CAN. The E-motor control was not the main research contents in this paper, so the calibrated PID controller was used to control the hub motor.

4. Simulation and result analysis

4.1. Vehicle model parameters

CarSim is an advanced driving and dynamic test tool. It could test the driving and dynamic performance of the vehicle before the prototype developed. So, chose CarSim to build vehicle model in this paper. The vehicle parameters were shown in Table 1.

Table 1. The vehicle parameters.

| Item                      | Symbol | Value | Unit |
|---------------------------|--------|-------|------|
| Complete vehicle mass     | \(m\)  | 1330  | kg   |
| Sprung mass               | \(m_{us}\) | 1130  | kg   |
| Wheelbase                 | \(L\)  | 2610  | mm   |
| Tread(front)              | \(B_F\) | 1535  | mm   |
| Tread(rear)               | \(B_R\) | 1540  | mm   |
| Height of center of mass  | \(h_g\) | 515   | mm   |
| Unsprung mass of rear axle| \(m_{Rus}\) | 110   | kg   |
| Power of single hub motor | \(P_h\) | 35    | kW   |

4.2. Virtual driving simulation analysis

This paper mainly focused on the state of driving during starting, so, the test scenario chose the modified double-line-shifting condition. Accelerate from zero at the start of the double-line-shifting path. The gradient of the whole pavement was set to uphill 5 degrees; The throttle pedal was set to fixed value of 0.6; The pavement was set as wet asphalt pavement, the optimal pavement utilization coefficient \(\mu_{max}\) was 0.8, and the optimal slip ratio \(\mu_{max}\) was 0.13.

And the double-shift line test results without optimized as comparison.

![Figure 4. Slip ratio optimized by PSO.](image)
Figure 5. Slip ratio without optimized PSO.

Figure 6. Torque optimized by PSO.

Figure 7. Torque not optimized

Figure 4 showed the slip rate curve of driving wheel optimized by PSO. Figure 5 shows the slip rate curve of distribute torque only according to the load on driving wheel; Figure 6 and Figure 7 were the torque of driving wheel in different situations; Figure 8 was driving speed in the above two situations. The conclusions could draw from the simulation results: the slip ratio optimized by PSO generally stable at 0.13, the maximum deviation was 0.01, the slip ratio not optimized by PSO had a larger fluctuate, the maximum deviation was nearly 0.11, the robustness of torque optimized system
was higher under comparison. And the result of driving speed showed that the dynamic performance of optimized by PSO was better, it had better acceleration performance under the same test environmental.

![Figure 8. Driving speed comparison.](image)

5. Hardware-in-loop test

After above analysis, the torque distribution system required much data to run, so in this paper the torque distribution unit integrated in the VCU (vehicle control unit). Multi-core processor Infineon TC 265D used to build VCU in order to meet the computing power. Its innovative multicore architecture, based on up to three independent 32-bit TriCore CPUs. The development process of the hardware was not the main research content of this paper, it would not discuss here, and the controller development test process was shown in the Figure 9 and Figure 10.

The approximate process of software generation as follow: The torque distribution unit built by Simulink; optimized algorithm compiled by S-function in Matlab; Finally, Matlab RTW tool used to convert the control model into code. At the same time, superimposed the external chip code generated by Infineon DAve 4 tool, after debugging by the HighTEC tool, the control code was rewrite to the VCU through UDE.

![Figure 9. The VCU in debugging.](image) ![Figure 10. The VCU in HIL test.](image)

The ASM Vehicle Dynamics Model was used to build the vehicle model in the HIL (hardware in loop) test, the road model selected the road spectrum which collected in Yancheng test field.

The test results are as shown in Figure 11, Figure 12, Figure 13.

According to the reference (14) the optimized slip ratio on this road was about 0.17, although there was spectrum and noise in the HIL test, the test results could be clearly analyzed that: the slip ratio optimized by PSO was fluctuated in a small range, and approximately close to the optimized value. The slip ratio curve of Figure 11: there was a larger tremor in steering, the maximum slip ratio was almost 0.35, and the slip ratio was larger than the optimized ratio in the hole test process.

So, the distribution strategy designed in this paper could effectively reduce the slip rate in the HIL test, increasing adhesion coefficient of pavement utilization, improve the driving stability.
6. Conclusions
This paper disturbed the torque depend on vehicle dynamics analysis and optimized by PSO. Frist, calculated the load of driving wheel, and according that disturbed torque preliminary. Second, due to the complexity of driving condition, the PSO was used to optimize the torque output. Built the control model in Simulink, built the vehicle model of small SUV in CarSim. Combine the CarSim and Simulink into virtual driving test. The result shown that the torque distribution algorithm designed in this paper could reasonably distribute the torque output in steering, the result show that the slip ratio optimized by PSO generally stable at the slip with the maximum attachment coefficient. HIL test was carried out after the simulation test, VCU build with Infineon TC265D processor, the control code was rewrite to the VCU through UDE. The result of HIL test shown that the torque distribution algorithm could reasonably realize the optimize torque distribute function on the hardware, improving the driving performance of the vehicle.
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