Research of parameter optimization of double trailing arm suspension mechanism for integrated wheel side deceleration electric drive system

Chen Xinbo\textsuperscript{1,\textordmasculine}, Lu Yanjia\textsuperscript{2,\textordmasculine}, Xu Naiwen\textsuperscript{3,\textordfeminine}, Xu Xiang\textsuperscript{4,\textordmasculine}

\textsuperscript{1}School of Automotive Studies, Tongji University, Shanghai 201804, China
Clean Energy Automotive Engineering Center, Tongji University, Shanghai 201804, China
\textsuperscript{2,4}School of Automotive Studies, Tongji University, Shanghai 201804, China
\textsuperscript{3}ETAS Automotive Technology (Shanghai) Co., Ltd.
\textsuperscript{\textordmasculine}email: austin_1@163.com, \textsuperscript{\textordfeminine}email: yanjia_lu@aliyun.com
\textsuperscript{\textordfeminine}Xu Naiwen: \textit{email:naiwen.xu@etas.com}

ABSTRACT—This paper put forward the solution of the wheel side deceleration electric drive system of double trailing arm suspension, through the combination of the trailing arm suspension and the electric drive system of wheel side deceleration. And the geometric mathematical model of the double trailing arm suspension based on the four-bar mechanism was established.
MATLAB executable file and the program interface was compiled, which quickly located the motor center layout position under the given conditions. When the wheel jumps, the center distance between the motor and the wheel remains unchanged. According to the reasonable parameters of the trailing arm structure provided by MATLAB, CATIA three-dimensional modeling was carried out, and ADAMS was imported for kinematics simulation, which verified the feasibility of the design scheme of the double trailing arm suspension and the MATLAB program.

1. Introduction
People pay more and more attention to environmental protection. It is imperative to save energy and reduce emissions. Looking for a new energy-saving and environmental protection "zero emission" vehicle has become a trend. In particular, electric vehicle has attracted great attention because of its low emission and low energy consumption.

Based on the motor drive mode, electric vehicles are generally divided into two types: centralized drive and distributed wheel drive. The centralized drive adopts the traditional automobile structure layout, but its structure is complex and its transmission efficiency is low, so the left and right driving wheels cannot be controlled independently. The wheel side drive system directly arranges the drive motor in or near the wheel. Compared with the centralized drive system, it has the potential of high integration, high efficiency, strong controllability, simple structure and so on. However, the unsprung mass of the electric wheel increases significantly, and the ratio of the sprung mass to the unsprung mass is unbalanced, which causes the vehicle performance deterioration [1].

In reference [2], the wheel side electric drive system with single swing arm suspension was proposed,
and it was proved by theoretical derivation that through the reasonable arrangement of the mass center position of the wheel side deceleration electric drive assembly on the suspension swing arm, the equivalent unsprung mass of the wheel side deceleration electric drive system could be greatly reduced.

This paper introduces a structure of wheel side electric drive system of double trailing arm suspension. Under this structure, the motor is arranged in the frame, as shown in Figure 1. So the unsprung mass of the motor is transferred to the sprung mass to further reduce the unsprung mass, which can effectively improve the driving performance of the vehicle[3]. Due to the high requirements of the gear transmission accuracy, the center distance is limited and the noise is loud, and its lubrication, sealing and other limitations have to be considered. Therefore, the chain drive or synchronous belt drive is considered when the transmission mechanism is arranged between the motor and the wheel, which has the advantages of adjustable center distance, strong impact resistance, stable operation, light weight and low cost. However, as the motor is arrange on the frame, the distance from the motor output shaft center to the wheel center will change with the suspension structure parameters along with the wheel jumping.

Therefore, it is necessary to design and analyze the wheel side suspension structure so as to find the appropriate layout of the suspension structure. Meanwhile, selecting the transmission mode, with either variable center distance or the center distance changing in a small range, can effectively reduce the impact of the road excitation on the wheel side drive transmission system.

Figure 1. Structure diagram of wheel side electric drive system of double trailing arm suspension with motor arranged on frame

2. Theoretical design

The theoretical structure design scheme of double trailing arm suspension wheel side deceleration electric drive system is displayed in Figure 2. The motor is fixed on the frame, and the motor and driving wheel have the same axis, $AD$ is the frame; $AB$ and $CD$ are the upper and lower trailing arms; $BC$ is the kingpin, $A$ and $D$ are the hinge points of upper and lower trailing arms and frame; $B$ and $C$ are the hinge points of upper and lower trailing arms and kingpin; $E$ is the wheel center. The axle is fixed on the kingpin. In Figure 2, the frame, upper and lower trailing arms and kingpin constitute a hinged four-bar mechanism. The rod $BE$ and kingpin 6 are consolidated as a whole, and the $E$ point is on the connecting rod; the $M$ point is on the frame, which is the center of the motor output shaft. When the wheel jumps
up and down, $S$ is the arc trajectory to be realized for the center point $E$ of the wheel. It is the arc trajectory with the center $M$ of the circle and the radius $r$. Therefore, the problem is transformed into solving the four-bar mechanism design parameters in accordance with the predetermined trajectory. If we can find the reasonable parameters of the double trailing arm length, the kingpin length and the equivalent frame $AD$ length to make the wheels jump along the ideal arc track $S$ under the road excitation, and to arrange the center of the motor at the point $M$ (the center of the driving wheel), then the center distance between the driven and the driving wheels will not change, and therefore the system scheme is feasible.

![Figure 2. System diagram of theoretical structure design scheme of double trailing arm suspension](image)

Establish the plane rectangular coordinate system, the point $A$ is the origin, the frame $AD$ and axis $x$ coincide. Let the length of the rod $AB$, $BC$, $CD$, $AD$, $BE$ be $L_1$, $L_2$, $L_3$, $L_4$, $L_5$ respectively, $\theta$ is the angle between the horizontal line $AD$ and the rod $BA$, $\mu$ is that between the horizontal line and the rod $CB$, and $\alpha$ is that between the rod $BE$ and the rod $BC$, $\angle BDA = \beta$.

In $\triangle BAD$, using cosine theorem:

$$L_{BD}^2 = L_1^2 + L_4^2 - 2L_4L_1\cos(\theta)$$  \hspace{1cm} (1)

also

$$\tan \beta = \frac{L_4\sin(\theta)}{L_4 - L_1\cos(\theta)}$$  \hspace{1cm} (2)

From equation (2), we get

$$\beta = \arctan\left(\frac{L_4\sin(\theta)}{L_4 - L_1\cos(\theta)}\right)$$  \hspace{1cm} (3)

In $\triangle BCD$, based on the cosine theorem, it can be calculated as:

$$L_3^2 = L_2^2 + L_3^2 - 2\sqrt{L_2L_3}\cos(\mu + \beta)$$

$$= L_2^2 + L_4^2 + L_4^2 - 2L_4L_1\cos(\theta) - 2L_2\sqrt{L_4^2 + L_4^2 - 2L_4L_1\cos(\theta)}\cos(\mu + \beta)$$  \hspace{1cm} (4)

Therefore

$$\mu + \beta = \arccos\left(\frac{L_2^2 + L_4^2 + L_4^2 - L_2L_4\cos(\theta)}{2L_2\sqrt{L_4^2 + L_4^2 - 2L_4L_1\cos(\theta)}}\right)$$  \hspace{1cm} (5)

$$\mu = \arccos\left(\frac{L_2^2 + L_4^2 - L_2L_4\cos(\theta)}{2L_2\sqrt{L_4^2 + L_4^2 - 2L_4L_1\cos(\theta)}} - \beta\right)$$  \hspace{1cm} (6)
Based on the above formula, the coordinates of the point \( E \) can be expressed as:
\[
Ex = L_1 \cos(\theta) + L_2 \cos(\alpha + \mu) \\
Ey = L_1 \sin(\theta) + L_2 \sin(\alpha + \mu)
\] (7)

3. Parameter optimization

The variable parameters of the optimization design of the double trailing arm suspension mechanism are: the length of the upper trailing arm \( L_1 \), the length of the kingpin \( L_2 \), the length of the lower trailing arm \( L_3 \), the length of the equivalent frame \( L_4 \), the coordinates \((x_m, y_m)\) of the point \( M \) of the motor center, and the radius \( R \) of the wheel runout fitting circle. The general mathematical expression of the optimization is as equation (8), where the optimization objective is to minimize the difference between the root mean square value of the distance from each point \( E(i)(x(i), y(i)) \) on the track \( S \) of the wheel motion center to the \( M \) point of the motor center and the radius \( R \) of the fitting circle when the wheel jumps. Constraint is the four-bar mechanism length condition, that is, the length sum of any three bars is greater than the fourth.

\[
\min \left| \sqrt{\frac{1}{m} \sum_{i=1}^{m} (x(i) - A)^2 + (y(i) - B)^2} - R \right|
\]

\[
\text{s.t.}
\]
\[
L_1 \leq L_2 + L_3 + L_4 \\
L_2 \leq L_1 + L_3 + L_4 \\
L_3 \leq L_1 + L_2 + L_4 \\
L_4 \leq L_1 + L_2 + L_3 \\
\text{where}
\]
\[
i = 1, 2, \ldots, m
\]

On the basis of the above theoretical analysis of double trailing arm suspension wheel side deceleration electric drive system, in order to quickly locate the motor center layout under given conditions, MATLAB executable file is compiled. The optimization flow chart is displayed in Figure 3, and the initial executable program interface is displayed in Figure 4.

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**Figure 3.** The optimization flow chart
Figure 4. initial interface of MATLAB executable program

The program uses five cycle optimization. The cycle variables are the length of upper trailing arm $L_1$, kingpin length $L_2$, lower trailing arm length $L_3$, equivalent rack length $L_4$ and connecting rod length $L_5$. The cycle step is 1mm. Firstly, according to the theoretical analysis formula in the previous section, the corresponding MATLAB program is compiled. Also, the corresponding expressions of mechanism length and angle parameters are listed. Then verify whether the corresponding mechanism parameters can meet the four-bar mechanism existence condition, that is, the length sum of any three bars is greater than the fourth. If it does not meet the condition, a warning will be output, and if it is satisfied, the execution will continue. Then the least square method is used to fit the circle center coordinate and arc, and the error is limited in a given range. Then the optimization parameters of the four-bar mechanism and the position, size and corresponding error of the motor layout are obtained. Finally, the limit position of the electric drive system, the track of the wheel center and the motor contour curve are drawn.

The program window got by programming is displayed in Figure 5. The image on the left side is the structure diagram of the double trailing arm suspension electric drive system; the lower left area is used to display the program image; the red line and the blue line are the upper and lower limit positions when the wheel is jumping, the green line is the corresponding track of the wheel center, the pink circle represents the outer contour of the motor, and the center represents the motor center. After executing the program, click the yellow ‘Clear’ button to clear the interface. First, you need to input the size range of the initial double trailing arm structure parameters in the upper blue text box. Here, the upper trailing arm length $L_1$ variation range is [240,350]mm. For kingpin length $L_2$, it is [140,250]mm, for lower trailing arm length $L_3$, it is [240,350]mm, for equivalent frame length $L_4$, it is [170,260]mm and for connecting rod length $L_5$, it is [20,100]mm. The angle $\alpha$ between connecting rod and kingpin is 30 °, the initial rotation angle of upper trailing arm $\theta$ is 60 °, the sum of given wheel runout $t_d$ is 80mm, and the motor radius $R$ is 90mm. Click the yellow ‘Optimize’ button, and the optimization results are shown in Figure 4. The optimization results are output in the green text box at the bottom left of the program window. The optimization results are the length of upper trailing arm $L_1=240mm$, kingpin $L_2=140mm$, lower trailing arm $L_3=240mm$, equivalent frame $L_4=180mm$ and connecting rod $L_5=100mm$. The motor center coordinates are (112.013,90.0594) mm, the fitting error is $2.84 \times 10^{-14}$, and the fitting circle radius is 199.077mm.

MATLAB visual interface (GUI) is programmed. It is very convenient, and the interface is clear and intuitive. Just input the initial parameter range, and then click the ‘Optimize’ button to quickly find the motor center position and the appropriate parameters of related components.
4. Entity model establishment and verification

4.1. CATIA parameter modeling

Based on the optimized parameters obtained in the previous section, the three-dimensional model is built through CATIA software, as displayed in Figure 6. The upper trailing arm length is \( L_1 = 240 \text{mm} \), the kingpin length is \( L_2 = 140 \text{mm} \), the lower trailing arm length is \( L_3 = 240 \text{mm} \), the equivalent frame length is \( L_4 = 180 \text{mm} \), the connecting rod length is \( L_5 = 100 \text{mm} \), the angle between the kingpin and the connecting rod \( \alpha \) is 30 °, the initial rotation angle of the upper trailing arm \( \theta \) is 60 ° and the radius \( R \) of the motor is 90mm. The motor center is arranged according to the optimized center obtained in the previous section. The model of large and small synchronous pulleys is omitted, and the equivalent mass of the large and small pulleys is transferred to the equivalent upper and lower trailing arms and kingpin, so as to achieve the simplicity.

4.2. ADAMS parameter verification

Import the CATIA 3D model into ADAMS, then it is necessary to define the attributes and the constraints of the related parts since the relationship between the parts is unconstrained.
After importing the model of double trailing arm suspension wheel side deceleration electric drive system into ADAMS, according to the motion relationship of double trailing arm suspension wheel side structure, the constraint relationship as displayed in Figure 7 is considered. Among them, a fixed pair constraint is added between the motor d and the upper trailing arm c, a fixed pair constraint is added between the wheel b and the kingpin e, and a rotation pair constraint is added between the upper trailing arm c and the kingpin e and the body a. The rotation points are the two hinged ends of the upper trailing arm, and the rotation axis direction is perpendicular to the paper. A rotation pair constraint is correspondingly added between the lower trailing arm f, the kingpin e and the car body a. A moving pair along the up and down direction is added between the vehicle body a and the ground, and it is also added between the test bench g and the ground. The body a and kingpin e, the wheel b and the test bench g are connected by springs, and the spring stiffness and damping are configured in accordance with the parameters in Table 1. The mass of each part is defined according to table 1 to establish the constraint relationship.

| Table 1 simulation model parameters of double trailing arm suspension system |
|------------------|------------------|
| Motor quality/kg | 12               |
| Body mass/kg     | 137              |
| Upper trailing arm mass/kg | 6      |
| Lower trailing arm mass/kg | 6      |
| Mass of kingpin and connecting rod/kg | 6        |
| Stiffness of upper spring damper/ N.mm⁻¹ | 20        |
| Damping of upper spring damper/(N.s).mm⁻¹ | 1.6        |
| Stiffness of lower spring damper/ N.mm⁻¹ | 200        |
| Damping of lower spring damper/(N.s).mm⁻¹ | 0          |
| Length of upper trailing arm/mm | 240        |
| Length of lower trailing arm/mm | 240        |
| Kingpin length/mm | 140          |
| Center distance between motor and wheel/mm | 199.077  |

Since the sum of the given wheel runout in the previous section is 80mm, the road input excitation is simulated by adding drive \(40mm * \sin(t)\) to the moving pair between the wheel and the test bench. Set
the simulation time as 10s, the simulation step size as 500, and run the simulation. Using the measurement and post-processing module in ADAMS, the curve of the center distance between the motor and the wheel changing with the up and down runout of the wheel is obtained, as displayed in Figure 8.

![Figure 8. curve of center distance between motor and wheel of double trailing arm suspension model changing with up and down runout of the wheel](image)

It can be observed that the average value of the center distance between the motor and the wheel is 199.2003mm, the minimum value is 199.1813mm, and the maximum value is 199.2098mm. The maximum variation of center distance is 0.0285mm, and the error is minor, which is within the acceptable variation range (<0.1mm). Therefore, within the acceptable error range, the size parameters of the trailing arm and the position of the motor center can be designed reasonably, to ensure that the center distance between the wheel and the motor center changes within the acceptable range. The simulation in this section also verifies the MATLAB program in the previous section.

5. Conclusion

(1) In this paper, a wheel side electric drive scheme of double trailing arm suspension is proposed, and the geometric mathematical model of double trailing arm suspension based on four-bar mechanism is established. By transferring the motor mass to the frame, the unsprung mass is effectively reduced, and the transmission efficiency and space utilization are improved.

(2) The four-bar structure parameters of double trailing arm suspension electric drive system are optimized by using MATLAB theory programming method, which solves the problem that the distance between the wheel center and the motor output shaft center will change with the suspension structure parameters when the wheel is bouncing and the motor is arranged on the frame. The optimized four-bar structure parameters are selected to make the center distance unchanged or within the acceptable range so that it is practical. GUI program is developed to make the program windowed and the interface clear and intuitive.

(3) According to the optimized four-bar structure parameters, the model is built and simulated in ADAMS, and the variation range of center distance is within the acceptable range of transmission mechanism. Through comprehensive verification, the feasibility of this group of parameters is verified. Meanwhile, the feasibility and accuracy of theoretical derivation and programming implementation are also verified.

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