Lightweight design of the planetary gear reducer of reduction drive hub motors

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Abstract—Hub motor is the development direction of new energy vehicles in the future. Usually, the reduction drive hub motor is directly arranged in the narrow rim space, so there are strict restrictions on the volume and weight of its reducer. This paper presents a highly integrated transmission scheme which can improve the working life of the reducer. For the planetary gear reducer integrated by the reduction drive hub motor, a multi-objective optimization function with minimum volume and large coincidence degree is constructed. In terms of constraints, the necessary conditions of planetary gear transmission are fully considered, the gear module meets the requirements of national standards, and the strength conditions of gear transmission. Using genetic optimization algorithm, the optimal solution satisfying the constraints is obtained. The optimization results show that the coincidence degree of the engagement between the sun gear and the planetary gear increases by 67% and the equivalent volume decreases by 31%, which can reduce the volume and reduce the transmission noise at the same time.

1. Introduction

At present, the automobile industry is experiencing unprecedented changes, and the electrification, intelligence, networking, and sharing of automobiles are developing rapidly [1]. Compared with traditional fuel vehicles, electric vehicles driven directly by motors can effectively reduce air pollution and help achieve the goal of carbon neutrality. Electric vehicles have no complex and lengthy transmission system, so it has fast control response speed and high control accuracy. It is a good automatic driving platform [2]. In the drive system of an electric vehicle, the hub motor can be directly arranged at the wheel, which can shorten the transmission chain to the greatest extent, improve the transmission efficiency, and realize the independent and controllable performance of four wheels of an electric vehicle, so it has broad development prospects. Hub motor is generally divided into direct drive and deceleration drive. Direct drive of the hub motor refers to the direct output of power from the motor to the wheel. Therefore, this kind of motor generally adopts the form of low speed and external rotor transmission, with a compact structure and the highest transmission efficiency, but its output torque is small and it is easy to damage the permanent magnet of the motor under the working condition of high torque [3]; Hub motor deceleration drive refers to the integration of the motor and the reducer with fixed speed ratio into the rim, and the power is transmitted to the wheel after deceleration and torque increase [4]. The reduction drive hub motor has the characteristics of high torque and high power density, but it increases the unsprung mass of the vehicle, resulting in the deterioration of vehicle ride comfort [5]. Therefore, to sum up, the reduction drive hub motor needs to reasonably select the reducer. On the one hand, it needs to ensure the reliability and rationality of the reducer in the transmission process. On the other hand, it also needs to pay attention to reducing the quality of the reducer, reducing the unsprung quality of the vehicle as much as possible, and improving the integration of the reduction drive hub...
motor and the smoothness of the vehicle.

There are a large number of examples at home and abroad to optimize the volume and reliability of the reducer. Xu et al. [6] proposed an improved adaptive weighted particle swarm optimization algorithm, which takes the minimum volume and maximum transmission efficiency as the multi-objective optimization functions, so that the reducer has a smaller volume and better transmission efficiency. Wen et al. [7] used MATLAB optimization toolbox to carry out multi-objective optimization design for the reducer of dual-purpose vehicle, which increased its coincidence degree under the condition of reducing the volume of the reducer and effectively improved the working performance of the reducer. Zhang et al. [8] can optimize the volume and transmission efficiency of planetary gear reducer by introducing variance into grey correlation degree and mutation particle swarm optimization algorithm, so as to avoid premature entry of the algorithm into local optimization. Zhou et al. [9] used the analytic hierarchy process (AHP) to get the weight of multi-objective function, and optimized the parameters of the reducer through genetic algorithm. The process is simple and reliable, which effectively improves the bearing capacity of the reducer.

In view of the above difficulties, this paper proposes a design scheme for the reduction drive hub motor. The power transmission scheme can well avoid the direct impact of the tire, and has the advantage of high integration. On this basis, the gear parameters of hub reducer are optimized to realize the lightweight of the hub motor and reduce its impact on vehicle ride comfort.

2. Optimization model of the planetary gear reducer

2.1. Reduction drive hub motor model

Compared with general gear reducers, planetary gear reducers have significant advantages such as strong bearing capacity, stable transmission, large power mass ratio, and high integration. Therefore, they are more and more used in environments with high space and quality constraints [10]. The hub motor reducer adopts planetary gear transmission in this paper. Wan et al. [11] proposed an integrated electric wheel structure of inner rotor deceleration drive for fuel cell vehicles. In this structure, the motor rotor is connected with the sun gear to input power, the ring gear is fixed on the motor shell, and the power is connected and output by the planet carrier and the wheel hub. The transmission scheme is simple and convenient, but because the planetary gear reducer is directly connected with the hub, it will inevitably lead to the impact of the road on the tire, which will be directly transmitted to the reducer through the hub. The reducer is a precision transmission device. Under a large number of irregular impacts, it will seriously affect the service life of the reducer, and may also directly affect the noise and smoothness of the electric wheel. In view of this, we redesigned the hub motor transmission scheme. As shown in Fig. 1, the power is input to the sun gear of planetary gear reducer through the inner rotor of the motor, the ring gear is fixed with the motor housing, and the power is output through the planetary carrier. In this scheme, the hub bearing is mainly placed outside the planetary gear reducer, so that the impact of the road on the wheel is mainly transmitted through the hub bearing, so as to protect the planetary gear reducer. In addition, the original parameters of the planetary gear reducer integrated by the hub motor are shown in the Table 1.
Fig. 1 Transmission scheme of reduction drive hub motor

Table 1 Original parameters of planetary gear reducer

| Parameters                              | Value           |
|----------------------------------------|-----------------|
| Number of sun gear teeth               | 23              |
| Number of planetary teeth              | 22              |
| Number of ring gear teeth              | 67              |
| Gear width                              | 20 mm           |
| Helix angle                            | 20.4°           |
| Modulus                                | 2               |
| Reduction ratio                        | 3.913           |
| Number of planetary gears              | 3               |
| Gear pressure angle                    | 20°             |
| Reconnection degree of sun gear and planetary gear | 2.6450         |
| Equivalent volume of reducer           | 141620 mm³      |

2.2. Objective functions

When the manufacturing material of planetary gear reducer is determined, its mass is directly proportional to its own volume. Therefore, it is hoped that the gear parameters at the minimum mass of the reducer are equivalent to the gear parameters at the minimum volume. Generally, the size of gear indexing circle can be used to calculate its volume, which can well simplify the volume calculation process of gear. For the planetary gear reducer, the volume of the sun gear and the planetary gear determines the size of the ring gear. Therefore, the volume sum of the first two can generally be used to represent the volume of the planetary gear reducer, and the minimum volume sum is taken as an objective function [7].

\[ v = v_a + n_p v_g = \frac{h \pi m_n^2}{4 \cos \beta_2} (z_a + n_p z_g) \]  

(1)

Where, \( b \) is the gear tooth width; \( m_n \) is the module of the gear; \( \beta \) is the helix angle of the gear; \( z_a \) is the number of sun gear teeth; \( n_p \) is the number of planetary gears; \( z_g \) is the number of planetary teeth.

The coincidence degree can represent the average value of the number of teeth participating in meshing at the same time when a pair of gears are engaged. The greater the coincidence degree, the more pairs of teeth participating in meshing at the same time, which is more conducive to the smooth transmission of gears, the improvement of bearing capacity and the reduction of transmission noise. Therefore, in this optimization process, the coincidence degree is taken as the second objective function. For helical gear transmission, the coincidence degree is equal to the sum of normal surface coincidence...
degree and end surface coincidence degree, that is:

\[
\varepsilon = \varepsilon_\alpha + \varepsilon_\beta \\
\varepsilon_\alpha = \frac{1}{2\pi} \left[ z_a (\tan \alpha_{ata} - \tan \alpha_t) + z_g (\tan \alpha_{atg} - \tan \alpha_t) \right] \\
\varepsilon_\beta = \frac{b \sin \beta}{n m_n} 
\]  
(2)

Where, \( \varepsilon_\alpha \) is the coincidence degree of end face; \( \varepsilon_\beta \) is the axial plane coincidence degree.

For the multi-objective function optimization problem, several sub objective functions need to be unified to transform the multi-objective optimization problem into single objective optimization. In this paper, the total objective function is given by using the method of linear weighting.

\[
f = u_1 v + u_2 \varepsilon
\]  
(3)

Where: \( u_1 \) and \( u_2 \) are the weighting factors of the sub-objective functions, given based on practical experience. The gear coincidence degree needs to take a larger value, so \( u_2 \) takes a negative value.

### 2.3. Constraint functions

In order to obtain the reducer parameters that can meet the actual needs of the project, the objective function must be limited by reasonable constraints. The specific constraints are introduced below:

1. **Transmission ratio condition**

   The transmission ratio of the reducer can generally determine the approximate range. As long as the speed of the motor and the speed required by the car are determined, the range of the transmission ratio can be obtained. In this design process, the planetary gear reducer is input by the sun gear, the planetary carrier outputs power, and the motor speed should not be too high. Therefore, the value range of transmission ratio is:

   \[
i_{adh}^b = 3.1 \leq i_{adh} \leq 4.7
\]  
(4)

Where \( i_{adh}^b = 1 + \frac{z_b}{z_a} \); \( z_b \) is the number of teeth of the ring gear.

2. **Concentric condition**

   For involute cylindrical planetary gear transmission, the rotation axis of sun gear, planetary gear and ring gear must be consistent with the main shaft line. Therefore, the center distance between the sun gear and the planetary gear must be equal to the center distance between the planetary gear and the ring gear, that is: \( a_{wg} = a_{gb} \).

   \[
   \frac{m_n}{2 \cos \beta} (z_a + z_g) = \frac{m_n}{2 \cos \beta} (z_b - z_g)
   \]  
(5)

Simplify the above formula to obtain:

\[
z_b - z_a = 2z_g
\]  
(6)

3. **Assembly conditions**

   In this planetary gear reducer, three planetary gears are evenly distributed and installed around the sun gear. At this time, the requirements for the number of teeth of each gear are as follows:

   \[
   \frac{z_a + z_b}{3} = N \text{(Integer)}
   \]  
(7)

4. **Adjacency condition**

   In order to make each planetary gear rotate normally without interference, it must be ensured that the center distance of any two adjacent rows of planetary gears is greater than the sum of the radius of its tooth top circle. In this paper, in order to leave enough clearance, it is necessary to meet the following formula:

   \[
l - d_b \geq m_n
\]  
(8)

That is: \( 2a_{wg} \sin(\pi / 3) - d_b \geq m_n \). Bring the calculation formula of tooth tip circle and center distance into the above formula for simplification:
5

(5) Non undercutting condition
The designed gear needs to meet the following conditions without undercutting during processing:
\[
\frac{m_n}{\cos \beta} (z_a + z_g) \sin \left( \frac{\pi}{3} \right) - \frac{m_n z_g}{\cos \beta} = 3m_n \geq 0
\]  

(9)

(6) Tooth width and coincidence constraints
The gear width and coincidence degree need to meet certain range constraints. For the reduction drive hub motor, because it needs to be arranged in a narrow hub space, it has strict requirements on the axial space.
\[
10 \leq b \leq 22
\]  

(11)

\[
\varepsilon_a \geq 1.2; \quad \varepsilon \geq 2.8
\]  

(12)

(7) Modulus constraint
Gear modulus is an important parameter of gear, and its value directly affects the performance of gear, such as gear coincidence degree, gear strength and so on. In automobile design, the gear module generally has some standard values, so the module can only choose some fixed values. Gear module \( m \in \{1.00, 1.25, 2.00, 2.50, 3.00, 4.00, 5.00\} \).

(8) Helix angle constraint
The helix angle of the gear has a great impact on the gear strength, working noise and axial force. Generally, the above performance will become better when the helix angle increases. However, when the helix angle increases to 30 degrees, its bending strength will drop suddenly, and the axial force will be too large. Therefore, it is necessary to limit the size of the helix angle.
\[
15^\circ \leq \beta \leq 25^\circ
\]  

(13)

(9) Contact strength constraint
When the gear is engaged, the surface of the gear produces the interaction force, and the force is transmitted through the interaction force. Therefore, in the transmission process, the tooth surface contact stress of the gear needs to be less than the allowable contact stress of the tooth surface. In the process of planetary gear transmission, the gear ring and the planetary gear are in internal meshing, and their coincidence degree is greater than that between the sun gear and the planetary gear. Therefore, as long as the sun gear and the planetary gear meet the meshing strength requirements, it means that the planetary gear reducer meets the strength requirements. The allowable tooth surface contact stress of gear is 1300 MPa in this paper. The contact stress of gear tooth surface is calculated as follows:
\[
\sigma_f = 0.418 \sqrt{\frac{F_a E}{b \left( \frac{1}{\rho_a} + \frac{1}{\rho_g} \right)}} \leq 1300 \text{ MPa}
\]  

(14)

Where, \( F_a \) is the normal force of tooth surface; \( E \) is the elastic modulus of gear material, \( E = 207000 \text{ MPa} \); \( T \) is the torque transmitted by the sun gear, \( T = 167000 \text{ N} \cdot \text{mm} \).

(10) Bending strength constraint
When the gear transmits power in meshing, it will have a torque effect on the tooth root. When the torque is too large, it will cause damage to the tooth root. Therefore, it is necessary to restrict the bending stress of the tooth root. The root bending stress is calculated as follows:
\[
\sigma_w = \frac{2T \cos \beta K_e}{3 \pi z_d m_n^2 y w x_e K_e} \leq 340 \text{ MPa}
\]  

(15)

3. Optimization design using genetic algorithm
At present, intelligent algorithm is widely used in engineering practice. Genetic algorithm can well deal with the nonlinear, large space and global optimization problems described in this paper. Therefore, this paper uses genetic algorithm to solve the problem.

3.1. Principle of genetic algorithm
Genetic algorithm is a process of using computer coding to simulate the process of biological natural evolution and natural heredity, and finally get the global optimal solution. When editing the algorithm,
it needs to include the generation of initial population, calculation of fitness, sorting, crossover, mutation and other processes. It should be noted that since the modulus can only take a few specific values, the gene needs to be transformed in these values. In the process of initialization, the parameter values in the genetic algorithm need to be set. The specific parameters and their values are shown in Table 2.

### Table 2 Parameter values

| Parameter          | Value   |
|--------------------|---------|
| Number of genes    | 5       |
| Population number  | 100     |
| Maximum genetic algebra | 200 |
| Crossover probability | 0.1  |
| Variation probability | 0.1   |

#### 3.2. Optimization results and discussion

The fitness change curve in the optimization process of genetic algorithm is shown in Fig. 2. We can see that the fitness function only uses 37 iterations from the large value at the beginning, and quickly converges to -3.265, with good convergence speed and results. In this optimization process, the parameters of the planetary gear reducer are shown in Table 3.

![Fitness evolution curve](image)

**Fig. 2** The historical iteration curve of the fitness function

### Table 3 Optimized design parameters

| Parameters                                      | value |
|------------------------------------------------|-------|
| Number of sun gear teeth                       | 31    |
| Number of planetary teeth                      | 35    |
| Number of ring gear teeth                      | 101   |
| Gear width                                      | 22 mm |
| Helix angle                                     | 24.9799° |
| Modulus                                        | 1     |
| Reduction ratio                                 | 4.26  |
| Number of planetary gears                       | 3     |
| Gear pressure angle                             | 20°   |
| Reconnection degree of sun gear and planetary gear | 4.4180 |
| Equivalent volume of reducer                    | 97477 mm³ |

By comparing the data before and after the optimization of the planetary gear reducer of the reduction drive hub motor, it can be seen that the coincidence degree of the engagement between the sun gear and the planetary gear has increased from 2.645 to 4.418, an increase of 1.773, with a growth rate of 67%, which can well improve the transmission stability and reduce the noise of the reducer. The equivalent volume of the reducer is obviously reduced, from 141620 mm³ to 97477 mm³, with a reduction rate of 31%, which is conducive to the integrated design of the reduction drive hub motor and the optimization of the ride comfort of the vehicle. The optimized reduction ratio of the reducer is 4.26, which is within the reasonable allowable range.
4. Conclusion
Through the analysis of the transmission path of the reduction drive hub motor, a transmission scheme which can improve the service life of the reducer is put forward. On this basis, the optimization objective of the planetary gear reducer is determined, and the constraints affecting gear design are discussed in detail, especially considering that the selection of gear modulus should meet the national standard. Finally, the genetic algorithm is used to obtain the optimal solution that meets the constraints. The coincidence degree of the meshing between the sun gear and the planetary gear is increased by 67%, and the equivalent volume is reduced by 31%. While improving the meshing degree of the gear, the volume of the reducer is reduced, which is conducive to the lightweight of the reduction hub motor and reducing the transmission noise.

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