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Design and Kinematics Analysis of the Reduction Mechanism for Hub Drive System

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ABSTRACT

Distributed hub drives electric vehicles highly integrate the drive motor, deceleration mechanism, etc. in the wheel. As a key component of the hub drive system, the structure and performance of the reduction mechanism will directly affect the driving and safety of the vehicle. How to realize the integrated design of the reduction mechanism and other parts of the hub drive system in limited wheel space is one of the basic problems to be solved in hub drive system design. In this paper, the NGW-type planetary mechanism is used as the hub reduction mechanism, combined with other parts of the hub drive system to design the specific structure of the planetary reduction mechanism, establish the kinematic model of the reduction mechanism, and carry out the kinematic simulation analysis.

Keywords: hub drive system; reduction mechanism; structure design; kinematic analysis

INTRODUCTION

As an important part of the hub drive system, the structure and performance of the reduction mechanism have an important impact on the power and safety of the hub drive system and even the vehicle. The reduction mechanism is installed inside the wheel, making the reduction mechanism in the actual process of the vehicle carrying more complex, so the design of the reduction mechanism of the hub drive system needs to choose the bearing capacity and high transmission efficiency layout. Mitsubishi Corporation of Japan developed a hub motor using a lithium-ion battery to decelerate and drive the electric vehicle Colt EV. The two rear wheels of this electric vehicle are equipped with permanent magnet three-phase AC synchronous servo motors and planetary gear reduction mechanism with a transmission ratio of 6 to obtain sufficient power [1].

Toyota exhibited the fuel cell hybrid vehicle FINE-T on the New York Auto Show. The four wheels of this vehicle are equipped with a high-speed internal rotor motor and hub reduction mechanism, which is composed of two reduction mechanisms, namely reverse gear and planetary gear, with a total reduction ratio of 8.5. The motor offset is realized by using the reverse gear [2]. Shilin et al. proposed a highly integrated deceleration transmission scheme, which can avoid the reduction mechanism being directly affected by the impact load from the road surface, and adopted a multi-objective optimization algorithm to optimize the reduction mechanism and reduce the unsprung mass [3]. The structure will cause the unsprung mass to be too large, affecting the ride comfort of the vehicle, and the unsprung mass should be controlled as small as possible when designing the reduction mechanism of the hub drive system. Ma Yaqing et al. completed the integrated matching design of the hub motor and reduction mechanism based on the hub drive system of a 6×6 light unmanned vehicle. The reduction mechanism adopts the NGW primary planetary deceleration transmission mode, and the lightweight design and strength check simulation verification was carried out [4]. Jiang Zhang designed a reduction mechanism of a wheel drive system based on an off-road vehicle, compared and analyzed the equal-load transmission performance of different floating reduction mechanisms, and determined that the double floating reduction mechanism was the best deceleration transmission scheme [5].

In summary, domestic and foreign researchers have carried out various aspects of research on the reduction mechanism of the hub drive system and obtained many research methods with reference significance, which provide a certain theoretical basis for the design of the hub drive reduction mechanism, but the layout of the reduction mechanism is mostly the reduction mechanism and hub motor are connected in parallel, the structure is not compact enough, the axial size is large. In this paper, the dynamic index of the vehicle as the design conditions, for the hub drive system reduction mechanism transmission ratio determination. At the same time, At the same time, the layout scheme of the reduction mechanism is designed based on the spatial structure of the hub drive system, the kinematics are analyzed in the multibody dynamics software, and the effectiveness of the structural design is verified.

VEHICLE STRUCTURE AND DESIGN OBJECTIVES

Hub-driven electric vehicles install the hub motor and reduction mechanism inside the wheel, cancel the transmission parts between the traditional electric vehicle traction motor and wheel, simplify the transmission system, shorten the power transmission path, and improve transmission efficiency. The power output of the hub motor is transmitted to the wheel through the reduction mechanism, and the speed reduction and torque increase are realized according to different driving conditions, to meet the power requirements of the hub-driven vehicle under different driving conditions.

Hub-driven electric vehicles are mainly composed of a battery pack, chassis, body, control unit, and other parts [6]. The chassis is mainly composed of three parts: a hub drive system, a braking system, and a steering system. The basic structure of hub-driven electric vehicles is shown in Figure 1. M is the hub motor, R is the reduction mechanism, and B is the brake.

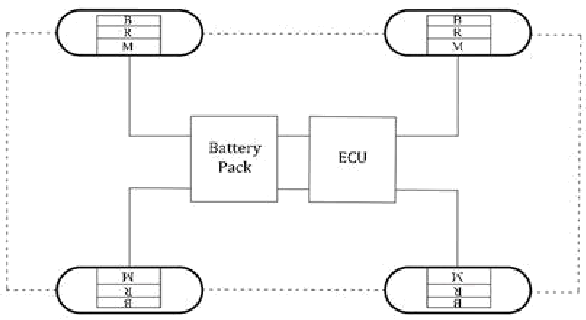


FIGURE 1: Basic structure of a hub-driven electric vehicle.

This paper takes a four-wheel hub-driven electric vehicle as the research object, and its structural parameters are shown in Table 1. In order to ensure that the hub drives the electric vehicle with sufficient power, the performance objectives of the electric vehicle are determined by referring to the performance specifications of the electric vehicle in GB/T 28382-2012 Technical Conditions for Electric Passenger Vehicles [7], as shown in Table 2.

TABLE 1: Basic Parameter of Vehicles.

Name	Value
Quality of maintenance	1319 kg
Windward area	2.65 m²
Driving form	4WD
Rolling resistance coefficient	0.015
Air drag coefficient	0.28
Wheel rolling radius	0.33 m

TABLE 2: Performance Objectives of Vehicles.

Name	Value
Maximum speed	≥135km/h
0-100km/h acceleration time	≤10s
Maximum climbing grade	0.3

The hub motor used in this paper is an internal rotor-type permanent magnet brushless DC motor, whose performance parameters and basic structural parameters are shown in Table 3 and Table 4. A reduction mechanism is installed in the hub motor, so the reduction mechanism design is not only to meet the vehicle power needs but also to ensure that the reduction mechanism of the outside diameter is less than the hub motor rotor diameter.

TABLE 3: Performance Parameters of Hub Motor.

Name	Value	Name	Value
Rated power	7.5kW	Peak power	21kW
Rated speed	1881r/min	Peak speed	4500r/min
Rated torque	38.1N·m	Peak torque	106.6N·m

TABLE 4: Basic Structure Parameters of Hub Motor.

Name	Value	Name	Value
Stator inner diameter	240mm	rator inner diameter	200mm
Stator outside diameter	310mm	Rotor core length	71mm

DETERMINATION OF TRANSMISSION RATIO OF THE REDUCTION MECHANISM

The setting of the gear ratio of the reduction mechanism mainly considers three factors: The first one is to meet the needs of the maximum climbing slope of the vehicle; The second is to meet the needs of vehicle acceleration time; Third, ensure that the vehicle can reach the maximum speed [8].

1) The transmission ratio that meets the maximum climbing slope of the vehicle

$$i \geq \frac{mg(f\cos\alpha_{\max}+\sin\alpha_{\max})r+\dfrac{C_D A u^2}{21.15}r}{4T_{\max}\eta_T} \quad (1)$$

In the formula, m is full load mass, assuming that there is one driver and four passengers in the car, whose average weight is set to 60kg, that is, m=1619kg; α is the angle between the road surface and the horizontal plane, the climbing speed u is 20km/h, and the maximum climbing slope is 0.3, that is, α is 16.7°. η_T is transmission efficiency; f is rolling resistance coefficient; C_D is air resistance coefficient; A is the windward area; T_{max} is the peak torque of the hub motor and r is the wheel radius. By substituting the values in Tab. 1-3 into Equation (1), we can get $i \geq 3.87$.

2) The transmission ratio that meets the acceleration performance of the vehicle

When the vehicle accelerates, it can be obtained from the driving equation of the vehicle:

$$F_t = mgf + \frac{C_D A u^2}{21.15} + \delta m\frac{du}{dt} \quad (2)$$

Where, F_t is the driving force; m is the test mass, is the sum of the preparation mass and the additional test mass of 180kg; δ the conversion coefficient of rotating mass.

According to the kinematics, we can know:

$$\begin{cases} t = \dfrac{1}{3.6}\displaystyle\int_0^{u_e}\frac{\delta m}{F_t - mgf - \dfrac{C_D A u^2}{21.15}}du \\ \quad + \dfrac{1}{3.6}\displaystyle\int_{u_e}^{100}\frac{\delta m}{F_t - mgf - \dfrac{C_D A u^2}{21.15}}du \\ F_t = \begin{cases} 3600\dfrac{P_{\max}\eta_t}{u_e} & (u > u_e) \\ 3600\dfrac{P_{\max}\eta_t}{u} & (u \leq u_e) \end{cases} \end{cases} \quad (3)$$

Therefore, the maximum driving torque required by a single wheel of the vehicle during acceleration is:

$$T_j = 3600\frac{P_{\max}\eta_t}{4u_e}r \quad (4)$$

Then the transmission ratio satisfying the acceleration performance of the vehicle is:

$$i \geq \frac{T_j}{T_{max}} = 3600 \frac{P_{max}\eta_t}{4u_e T_{max}} r \qquad (5)$$

The u_e corresponding to the rated speed of the hub motor is 60km/h. The output power Pmax of the vehicle during acceleration and the peak torque T_{max} of a single hub motor are substituted into equation (5) to obtain: $i \geq 3.59$.

3) The transmission ratio that meets the maximum speed of the vehicle

$$i \leq \frac{0.377 r n_{max}}{u_{max}} \qquad (6)$$

By substituting the maximum speed u_{max}, wheel rolling radius r and peak speed n_{max} of the hub motor into Equation (6), we can get: $i \leq 4.15$.

Considering the requirements of maximum climbing slope, acceleration time and maximum speed of vehicles, the transmission ratio of reduction mechanism should meet the following requirements: $3.87 \leq i \leq 4.15$. In this paper, the transmission ratio of reduction mechanism is 4.

STRUCTURAL DESIGN OF THE REDUCTION MECHANISM

Hub drive system basic structure scheme

The development trend of "two-in-one" and "multi-in-one" integrated design of key components such as hub motor, brake, and deceleration mechanism is remarkable. At present, the integrated configuration of the hub deceleration drive system is mostly parallel with the reduction mechanism and the hub motor, which has a large axial size and low utilization rate of the internal space of the motor. In this paper, a new hub drive system is designed. The reduction mechanism is installed inside the hub motor to make full use of the internal space of the motor, shorten the power transmission path from the hub motor to the wheel, and improve the transmission efficiency to a certain extent. Figure 2 shows the structure scheme of the hub drive system.

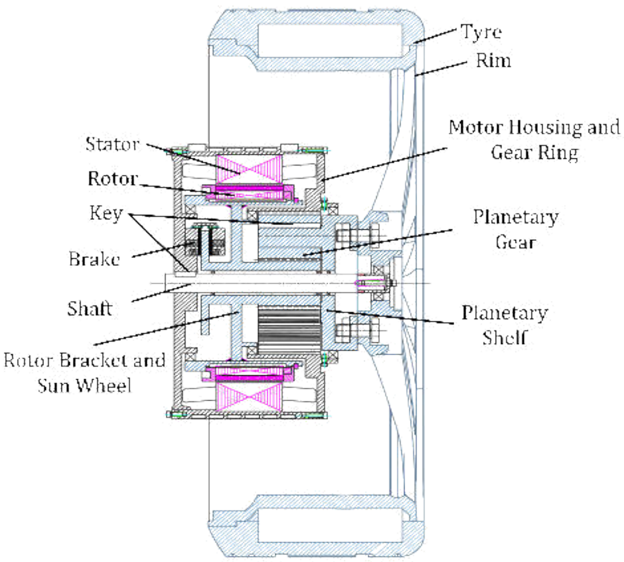


FIGURE 2: Structure Scheme of Hub Drive System.

As shown in Figure 2, the hub drive system uses an internal rotor motor, and the planetary gear reduction mechanism is completely embedded in the hub motor.

The rotor bracket is integrated with the brake disc and sun wheel, and the motor shell is integrated with the brake caliper and gear ring. The power transmission route of the hub deceleration drive system is: the gear ring is fixed, the power output of the hub motor is taken as the input of the sun wheel, and then through the planetary gear to transfer the power to the planetary frame, the planetary frame will output the power to the wheel, driving the vehicle.

Determination of reduction mechanism parameters

(1) Structural design of each component of the reduction mechanism

Firstly, the gear type is selected. The gears in this paper all adopt standard spur gear. According to the precision standard of involute cylindrical gear and the working environment of the reduction mechanism of the hub drive system, the precision of the gear is selected as 6 levels. The material of each reduction mechanism is 40MnB and tempered, and the hardness is 241~286HB. The number of primary planetary wheels is 3, and the number of solar wheels is 27. The parameters are designed according to the design requirements of tooth surface contact strength and tooth root bending strength.

In the planetary gear drive, the gear ring is fixed, the sun wheel is input, and the planet rack is output. The transmission ratio of the planetary gear reduction can be written:

$$i_{ac}^{b} = 1 + p \qquad (7)$$

$$p = \frac{z_b}{z_a} \qquad (8)$$

Where, i_{ac}^{b} is the transmission ratio of the planetary gear reduction mechanism, $i_{ac}^{b} = 4$; p is the characteristic parameter of planetary row and is related to the given transmission ratio. It is unreasonable for the p-value to be too large or too small, generally p=3~8. In this paper, the transmission ratio of the planetary gear reduction mechanism is 4, p=3, then the number of ring teeth $z_b = pz_a = 81$.

According to the concentric condition of the planetary gear train, the tooth number z_c of planetary gear can be calculated as:

$$z_c = \frac{z_b - z_a}{2} = \frac{81-27}{2} = 27 \qquad (9)$$

1) Preliminary calculation of gear parameters

The diameter of the solar wheel:

$$d_a \geq A_d \sqrt[3]{\frac{T_1}{\varphi_d [\sigma_H]^2} \frac{u+1}{u}} \qquad (10)$$

Where, A_d value is 90; T_1 is the nominal torque of meshing between the solar wheel and a single planetary wheel $T_1 = T_{max} / n = 35533.33\text{N}\cdot\text{mm}$; φ_d is the tooth width coefficient, and is 1; $[\sigma_H]$ is the allowable contact stress of solar wheel material, the allowable contact stress is preliminarily calculated, $[\sigma_H] = 0.9\sigma_{Hlim}$, and the contact fatigue limit of each gear can be obtained by looking up the table $\sigma_{Hlim} = 750\text{MPa}$. u is the tooth ratio, $u = z_c / z_a = 27/27 = 1$.

By substituting the data into Equation (10), the diameter d_a of the solar wheel can be obtained ≥48.45mm, then the module m of the gear is:

$$m = d_a / z_a \geq \frac{48.45}{27} = 1.79\text{mm} \qquad (11)$$

Allowable bending stress of solar wheel material ：

$$[\sigma_F] = \frac{\sigma_{Flim} Y_N Y_X}{S_{Fmin}} \qquad (12)$$

Where σ_{Flim} is the bending fatigue limit of gear, which can be obtained by looking up the hardness table of solar wheel materials $\sigma_{Flim} = 320\text{MPa}$; Y_N is bending fatigue coefficient, is $Y_N = 1.16$; Y_X is the size coefficient, $Y_X = 1.0$, S_{Flim} and is the minimum bending safety coefficient, obtained by referring to the table $S_{Flim} = 1.25$. Substitute the data into equation (12) to obtain.

The modulus m of the solar wheel is preliminarily calculated:

$$m \geq \sqrt[3]{\frac{2KT_1}{\psi_d z_a^2 [\sigma_F]} Y_{Fa} Y_{Sa} Y_{\xi}} \qquad (13)$$

Where, K is the load coefficient, and K=1.5; Y_{Fa} is the tooth profile coefficient, and is 2.65; Y_{Sa} is the stress correction coefficient, which is 1.56; For straight gear drive, Y_{ε} is the contact coefficient of 1. By substituting the data into Equation (13), we can obtain:

$$m \geq 1.27\text{mm} \qquad (14)$$

So m ≥ 1.79mm, according to the national standard modulus, take the modulus of 2mm, then the sun wheel indexing circle diameter $d_a = mz_a = 54\,\text{mm}$.

In two meshing gear pairs (solar gear a -- planetary gear c and planetary gear c -- ring b), the standard center distance a is:

$$a_{a\text{-}c} = \frac{1}{2} m(z_a + z_c) = 54\text{mm} \qquad (15)$$

$$a_{c\text{-}b} = \frac{1}{2} m(z_b - z_c) = 54\text{mm} \qquad (16)$$

It can be seen from equations (15) and (16) that the standard center distance of the two meshing gear pairs is the same. Therefore, the planetary gear transmission meets the concentric condition of non-displacement and does not need to undergo angular displacement.

The geometric parameters of the gear train were calculated according to the concentric conditions, assembly conditions, adjacency conditions, and hub spatial structure size of the planetary gear train, as shown in Table 5.

TABLE 5: Geometrical Parameters of The Planetary Gear Train.

Name	Value
Number of solar gear	27mm
Number of ring teeth	81mm
Number of planetary gear teeth	27mm
The diameter of the solar circle	54mm
Diameters of planetary rings	54mm
Diameter of tooth ring indexing circle	162mm
Sun gear apex diameter	58mm
Diameter of the top circle of planetary gear teeth	58mm
Ring tooth apex diameter	158mm
Diameter of the root circle of the solar wheel	49mm
Diameter of root circle of planetary gear	49mm
Ring diameter of root circle	167mm
Tooth height	4.5mm
Tooth thickness	3.14mm

As can be seen from Table 5, the diameter of the root circle of the gear ring is smaller than the inner diameter of the hub motor rotor, which can ensure that the reduction mechanism is installed inside the hub motor. Therefore, the reduction mechanism with a transmission ratio of 4 and solar gear number of 27 meets the design requirements.

2) Size design of solar wheel input shaft

The solar wheel input shaft mainly bears the torque output from the hub motor, and its shear stress is:

$$\tau_T = \frac{T}{W_T} = \frac{9550 \times 10^3 P / n}{(1-\beta^4) 0.2 d_o^{\,3}} \leq [\tau_T] \qquad (17)$$

Where, W_T is the torsion resistance section coefficient of the shaft; $[\tau_T]$ is allowable shear stress; β is the ratio of the inner and outer diameters of the hollow shaft, and β=0.6 is taken. Rewrite equation (17) into the design formula, and the outer diameter of the input shaft is:

$$d_o \geq \sqrt[3]{\frac{9550 \times 10^3 P / n}{(1-\beta^4) 0.2 [\tau_T]}} = C \sqrt[3]{\frac{P}{(1-\beta^4) n}} \qquad (18)$$

Where, C is the material correlation coefficient of the axis. In this paper, the material of the input axis of the solar wheel is 40MnB, and C=102 is obtained by referring to the table. By substituting the peak working condition parameters of the driving motor into equation (18), it can be obtained that: $d_o \geq 23.88$mm; considering its assembly conditions, d_o=48.33mm, and d_i=29mm.

3) Size design of the output shaft of the planetary gear

When the torque of the input shaft of the solar wheel is increased by the transmission ratio of 4, the torque becomes 426.4 N·m, then the output torque of a single planetary gear is 142.13 N·m, and the material of the output shaft of the planetary gear is 45# steel, C=112 can be obtained from the table. And as a solid axis, β is 0, plug the data into equation (18), and it can be obtained: d ≥ 27.54mm, rounded, d=28mm.

(2) Check the strength of each component of the reduction mechanism

1) Check the contact fatigue strength of the tooth surface

① Sun gear - planet gear meshing gear pair

Tooth surface contact stress:

$$\sigma_H = \sigma_{H0} \sqrt{K_A K_V K_{H\beta} K_{H\alpha} K_{Hp}} \qquad (19)$$

Where, σ_{H0} is the basic value of contact stress; K_A is the use coefficient; K_V is the dynamic load coefficient; $K_{H\beta}$ is the distribution coefficient of tooth load; $K_{H\alpha}$ is the load distribution coefficient between teeth; K_{Hp} is the uneven load distribution coefficient between planetary gears.

Basic value of contact stress:

$$\sigma_{H0} = Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_{ac}}{d_a b} \times \frac{u+1}{u}} \quad (20)$$

Where, F_{ac} is the tangential force acting on the gear; Z_H is node region coefficient; Z_E is the elastic coefficient; Z_ε is the coefficient of coincidence; Z_β is helical angle coefficient; u is the tooth ratio.

Tangential force:

$$F_{ac} = \frac{2T_1}{d_a} \quad (21)$$

End contact degree of gear:

$$\varepsilon_\alpha = \left[1.88 - 3.2\left(\frac{1}{z_a} + \frac{1}{z_c}\right)\right]\cos\beta \quad (22)$$

Where, β is the helical angle of the gear. In this paper, spur gear is used, β=0°.

Gear contact coefficient:

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3}} \quad (23)$$

According to the working characteristics and material characteristics of the meshing gear pair of solar wheel and planet gear, K_A is 1.5, K_V is 1.08, K_β is 1.1 by looking up the table and calculating. $K_{H\alpha}$ is 1, K_{Hp} is 1.2, Z_H is 2.5, Z_E is $189.8\sqrt{\text{MPa}}$, Z_ε is 0.89, Z_β is 1.

Substituting the above parameters into Equation (20), we can obtain:

$$\sigma_{H0} = 2.5 \times 189.8 \times 0.89 \times \sqrt{\frac{1316.05}{54 \times 54} \times 2} \quad (24)$$
$$= 401.22\text{MPa}$$

Therefore, the contact stress between the solar wheel and the planetary gear is:

$$\sigma_H = 401.22 \times \sqrt{1.5 \times 1.08 \times 1.1 \times 1 \times 1.35} \quad (25)$$
$$= 622.31\text{MPa}$$

Assuming that the stress cycle number of the gear is greater than or equal to 10^7, the contact life coefficient ZN of the gear is less than or equal to 1.18. Minimum safety factor of gear contact $S_{Hmin} = 1.05$. Therefore, the allowable contact stress of gear is:

$$[\sigma_H] = \frac{\sigma_{Hlim} Z_N}{S_{Hmin}} = \frac{750 \times 1.18}{1.05} = 842.86\text{MPa} \quad (26)$$

$\sigma_H \le [\sigma_H]$, which meets the requirements of tooth surface contact fatigue strength.

② Planetary gear - ring gear pair

According to the force of reduction mechanism, tangential force at the meshing of planetary gear and ring meets the following requirements:

$$F_{bc} = F_{ac} \quad (27)$$

Planetary gears and rings are internally engaged, and the end surfaces of the gears are overlapped:

$$\varepsilon_\alpha = \left[1.88 - 3.2\left(\frac{1}{z_c} - \frac{1}{z_b}\right)\right]\cos\beta \quad (28)$$

Basic value of contact stress：

$$\sigma_{H0} = Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_{ac}}{d_a b} \times \frac{u-1}{u}} \quad (29)$$

Similarly, according to the working characteristics and material characteristics of planetary gear - ring meshing gear pair, K_A is 1.5, K_V is 1.02, $K_{H\beta}$ is 1.1 by looking up the table and calculating. $K_{H\alpha}$ is 1, K_{Hp} is 1.35, Z_H is 2.5, Z_E is $189.8\sqrt{\text{MPa}}$, Z_ε is 0.86, Z_β is 1.

Substituting the above parameters into Equation (29), we can obtain:

$$\sigma_{H0} = 2.5 \times 189.8 \times 0.86 \times \sqrt{\frac{1316.05}{54 \times 54} \times \frac{2}{3}} \quad (30)$$
$$= 223.84\text{MPa}$$

Therefore, the contact stress between planetary gear and gear ring is:

$$\sigma_H = 223.84 \times \sqrt{1.5 \times 1.02 \times 1.1 \times 1.35} \quad (31)$$
$$= 337.40\text{MPa}$$

$\sigma_H \le [\sigma_H]$, which meets the requirements of tooth surface contact fatigue strength.

2) Check the tooth root bending fatigue strength

① Sun gear - planet gear meshing gear pair

The structural parameters of the solar wheel are the same as those of the planetary gear, so only the bending fatigue strength of the solar wheel or planetary gear can be calculated when the bending fatigue strength of the meshing gear pair of the solar wheel-planetary gear is checked.

Root bending stress:

$$\sigma_F = \sigma_{F0} K_A K_V K_{F\beta} K_{F\alpha} K_{Fp} \quad (32)$$

Where, σ_{F0} is the basic value of root bending stress; $K_{F\beta}$ is the distribution coefficient of tooth load; $K_{F\alpha}$ is the load distribution coefficient between teeth; K_{Fp} is the uneven load distribution coefficient between planetary gears.

Basic value of root bending stress:

$$\sigma_{F0} = \frac{F_{bc}}{bm} Y_{Fa} Y_{Sa} Y_\varepsilon Y_\beta \quad (33)$$

Where, Y_{Fa} is the tooth profile coefficient; Y_{Sa} is stress correction coefficient; Y_ε is coincidence coefficient; Y_β is the helix Angle coefficient.

Coincidence coefficient:

$$Y_\varepsilon = 0.25 + \frac{0.75}{\varepsilon_\alpha} \quad (34)$$

Load distribution coefficient between teeth:

$$K_{F\alpha} = \frac{1}{Y_\varepsilon} \quad (35)$$

Uneven load distribution coefficient among planetary gears:

$$K_{Fp} = 1 + 1.5(K_{Hp} - 1) \quad (36)$$

K_A is 1.5, K_V is 1.08, $K_{F\beta}$ is 1.24, according to the working characteristics and material characteristics of the meshing gear pair of solar wheel and planet gear, by looking up the table and calculating. $K_{F\alpha}$ and K_{Fp} are 1.35, 1.53, Y_{Fa} is 2.58, Y_{Sa} is 1.62; Y_ε is 0.71; Y_β is 1.

Substituting the above parameters into Equation (33), we can obtain:

$$\sigma_{F0} = \frac{1316.05}{54\times 2}\times 2.58\times 1.62\times 0.71 = 36.16\text{MPa} \quad (37)$$

Therefore, the bending stress of planetary gear and ring is:

$$\sigma_F = 36.16\times 1.5\times 1.08\times 1.24\times 1.35\times 1.53 = 150.03\text{MPa} \quad (38)$$

Permissible bending stress of gear:

$$[\sigma_F] = \frac{\sigma_{Flim} Y_N Y_X}{S_{Flim}} \quad (39)$$

Where, S_{Flim} is the minimum bending safety factor; Y_N is the bending life coefficient; Y_X is the size coefficient. Look up the table and get $S_{Flim} = 1.25$; $Y_X = 1$; $Y_N = 1.16$, and the data is substituted into equation (39) to obtain:

$$[\sigma_F] = \frac{320\times 1.16\times 1}{1.25} = 296.96\text{MPa} \quad (40)$$

$\sigma_F \le [\sigma_F]$, which meets the requirements of root bending fatigue strength.

② Planetary gear - ring gear pair

Gear contact coefficient:

$$Y_\xi = 0.25 + \frac{0.75}{\varepsilon_\alpha} \quad (41)$$

Load distribution coefficient between teeth:

$$K_{F\alpha} = \frac{1}{Y_\varepsilon} \quad (42)$$

Similarly, according to the working characteristics and material characteristics of the planetary gear-ring meshing gear pair, K_A is 1.5, K_V is 1.02, $K_{F\beta}$ is 1.24, $K_{F\alpha}$ is 1.49, K_{Fp} is 1.1, Y_{Fa1} is 2.58, Y_{Fa2} is 2.053, Y_{Sa1} is 1.62, Y_{Sa2} is 2.65, Y_ε is 0.67, Y_β is 1 by looking up the table and calculating.

Substituting the above parameters into Equation (33), we can obtain:

$$\sigma_{F01} = \frac{1316.05}{54\times 2}\times 2.58\times 1.62\times 0.67\times 1 = 34.12\text{MPa} \quad (43)$$

$$\sigma_{F02} = \frac{1316.05}{54\times 2}\times 2.053\times 2.65\times 0.67\times 1 = 44.42\text{MPa} \quad (44)$$

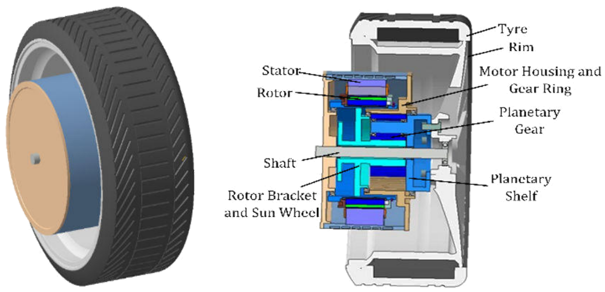
Therefore, the bending stress of planetary gear and ring is:

$$\sigma_{F1} = 34.12\times 1.5\times 1.02\times 1.24\times 1.49\times 1.1 = 106.01\text{MPa} \quad (45)$$

$$\sigma_{F2} = 44.42\times 1.5\times 1.02\times 1.24\times 1.49\times 1.1 = 138.12\text{MPa} \quad (46)$$

Where, σ_{F1} is the bending stress of the tooth root of the planetary gear when the planetary gear - gear ring is engaged, and σ_{F2} is the bending stress of the tooth root of the planetary gear - gear ring when the planetary gear - gear ring is engaged. $\sigma_F \le [\sigma_F]$, which meets the requirements of root bending fatigue strength.

According to the structure size parameters, the three-dimensional assembly drawing of the hub drive system is drawn, as shown in Figure 3.



a) Overall Structure b) Internal Structure

FIGURE 3: Hub Drive System 3D Assembly Drawing.

Kinematic analysis of reduction mechanism

The 3D model of the reduction mechanism is imported into Adams, the constraint of fixed earth is applied in the center of the gear ring, the rotation constraint relative to the earth is applied in the center of the solar wheel, planetary gear, and planetary frame, and the gear pair between each gear is set at the same time. The driving conditions are 0-2s. The angular velocity of the solar wheel accelerates from 0rad/s to 400rad/s, and the load torque applied to the planetary shelf is 59.9N·m, as shown in Figure 4. Set the simulation time to 10s and the simulation steps to 200. The simulation results are shown in Figure 5.

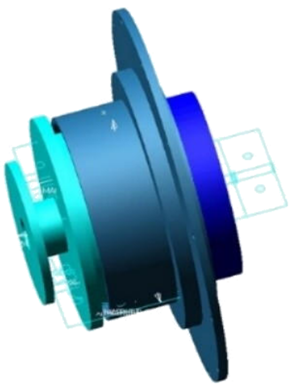


FIGURE 4: Kinematics Model of Reducer.

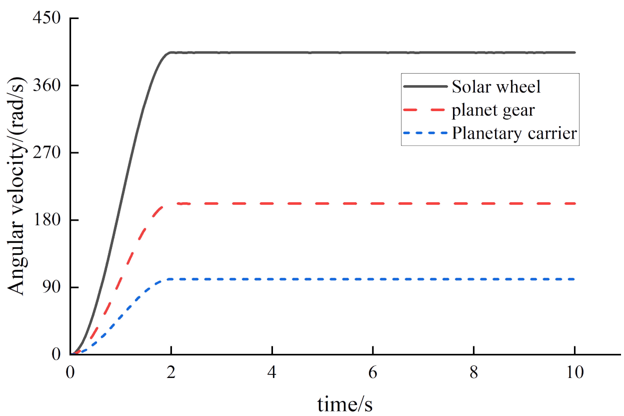


FIGURE 5: Kinematics Simulation Results.

Assume that the solar wheel is a, the planetary wheel is b, the gear ring is c, and the planetary shelf is X. When the gear ring is fixed, the solar wheel is the input and the planetary shelf is the output,

The ratio of angular velocity between the solar wheel and the planetary shelf is:

$$\frac{\omega_a}{\omega_x} = 1 + p \tag{47}$$

The angular velocity relationship between the planetary shelf and the planetary wheel is:

$$\omega_c - \omega_x = \frac{2p}{1-p}\omega_x \tag{48}$$

By putting the number of gears and the angular velocity of the solar wheel into equations (47) and (48), the angular velocity of the planetary wheel and the planetary frame are 200rad/s and 100rad/s, respectively. After the simulation is completed, the reduction mechanism has no motion interference reduction mechanism model and the accuracy of the simulation can be verified.

SUMMARY

In this paper, the hub-driven electric vehicle is taken as the research object, first based on the vehicle parameters, dynamic indicators, and hub motor performance parameters to determine the transmission ratio of the reduction mechanism, and then based on the vehicle parameters and hub motor structure characteristics, the layout design of reduction mechanism and structural parameters are determined, and the strength is checked. Finally, the reduction mechanism of the hub drive system with a deceleration ratio of 4 and high integration degree is obtained, and the three-dimensional model of the reduction mechanism is established. In addition, the 3D model of the reduction mechanism is analyzed in the multi-body dynamics software, and the results verify the rationality and effectiveness of the structure design, which provides the basis for the subsequent analysis and research of the reduction mechanism.

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