Transmission Design and Kinematics Simulation of Fixed Gear and Planetary Gear

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Abstract: Based on the theory of 2K-H planetary gear transmission, a fixed-axis gear planetary gear transmission scheme is designed, the gear meshing parameters are determined, and the theoretical transmission ratio and output speed are calculated. The 3D model software UG NX8.0 is used to build the solid model, and the virtual prototype model is built based on ADAMS 2013 and the kinematics simulation is carried out. The simulation results are compared with the theoretical values. The results show that the errors of speed and gear ratio are 0.43% and 0.50%. The correctness of the design parameters and the credibility of the virtual prototype model are validated, which lays the foundation for further research on the dynamic characteristics of the planetary gear mechanism.

1. Introduction
Planetary gear transmission is a common transmission mechanism, which is characterized by small mass, small size, large transmission ratio, large carrying capacity, smooth transmission and high transmission efficiency. It can be applied to a variety of mechanical transmission devices to complete the variable speed drive, motion synthesis, motion decomposition and other special features [1]. At present, the research focus of the power coupling device of hybrid electric vehicle gradually shifts from ordinary gear to planetary gear, but the literature of the transmission mechanism combining the two is few. In view of the above problems, based on the traditional transmission theory of 2K-H planetary gear mechanism, the transmission scheme of fixed-shaft gear and planetary gear is designed and simulated.

2. Design of planetary gear mechanism

2.1 Transmission scheme of planetary gear mechanism
The fixed-axis gear and planetary gear transmission is shown in Figure 1. When fixed-axis gear 1 is a power input, the planet carrier H is output unit, it has a good dynamic characteristics and high transmission efficiency [2].
2.2 Selection of meshing parameters

2.2.1 Transmission ratio conditions
According to Figure 1 which shows the fixed-axis gear and planetary gear drive, the gear and gear ratio as follows:

\[ i_{AH} = 1 - i_H^H = 1 + \frac{Z_B}{Z_A} \]  
\[ i_1^2 = \frac{Z_2}{Z_1} \]  

In the type: \( i_B AH \) is the transmission ratio of the wheel system when the inner gear B is fixed, the sun wheel A is input and the planetary frame H is output, and the \( i_1 \ 2 \) is the transmission ratio of the fixed axis gear \( Z_1 \) and \( Z_2 \).

2.2.2 Adjacent conditions
In order to reduce the gear train of the planetary gear train so that the planetary gears do not collide with each other, it is necessary to ensure that there is a certain gap between the tops of the gears on the connecting line, that is, the sum of the top circle radii of two adjacent planetary gears should be smaller than the center distance \( L_C \).

\[ \begin{align*}
2r_{AC} &< L_C \\
d_{AC} &< 2a_{AC} \sin \frac{\pi}{n_p} 
\end{align*} \]  

In the type: \( r_{AC} \), \( d_{AC} \) are the addendum circle radius and diameter of the planet wheel C; \( n_p \) is the number of planet wheels; \( a_{AC} \) is the center distance of gear pair A and C; L is the distance between two adjacent planet wheel. In practical use, the gap value general takes as \( \Delta_c = L_C - d_{AC} \geq 0.5m \), \( m \) is the modulus of the gear.

2.2.3 Concentric conditions
The concentric condition is that the actual center distances of all the meshing gear pairs of the center wheel A, B and the planet wheel C must be equal. For standard planetary gears, concentric conditions are: \( a_{AC} = a_{CB} \).

\[ \frac{m(Z_A + Z_C)}{2} = \frac{m(Z_A - Z_C)}{2} \]  

To ensure concentricity, the number of teeth \( Z_A \) and \( Z_B \) of both sun gears must be even or odd at the same time, otherwise the number of planet gears \( Z_C \) can not be an integer.

2.2.4 Assembly condition
The mounting condition is the condition that the number of teeth of each gear should be satisfied when \( np \) planetary gears mounted on the boom H are evenly distributed around the center wheel. The assembly condition is:

Figure 1 Shaft gear and planetary gear transmission diagram
\[ \frac{Z_1 + Z_\beta}{n_p} = I \quad \text{(Integer)} \quad (5) \]

Refer to the national standards (GB 1357-1987), (GB 1356-1988) and (GB 1357-1988) through literature [1]; select \( i_B AH = 2.8 \), \( n_p = 3 \) and \( i l = 1.3 \). Table 1 shows the meshing parameters of fixed-axis gear and planetary gear drive. And this data is put into equation (3) to have a check. This condition is satisfied by calculation.

| Table 1 Gear meshing parameters |
|----------------------------------|
| **Planet gear** | **Sun gear** | **Ring gear** | **Z1** | **Z2** |
|------------------|--------------|---------------|-------|-------|
| Modulus (m) | 4 | 4 | 4 |
| Number of teeth | 19 | 47 | 85 |
| Pressure angle (\( \alpha \)) | 20° | 20° | 20° |
| Tooth width (mm) | 40 | 40 | 40 |

2.3 Transmission ratio and output speed calculation

2K-H planetary gear mechanism center wheel A, ring gear B, planet carrier H around the main axis rotation speed of rotation \( n_a, n_b, n_h \). By the conversion mechanism method, the ratio of the angular velocities of the members A, B with respect to the movement of the member H is:

\[ i_{AH} = \frac{\omega_A - \omega_H}{\omega_B - \omega_H} = \frac{-Z_B}{Z_A} \quad (6) \]

Due to \( i_B AH=-Z_B/Z_A=-K \), the formula (1) and (6) can be combined 2K-H-type planetary gear movement equation is:

\[
\begin{align*}
\omega_A + K \omega_B - (1 + k) \omega_H &= 0 \\
\omega_B + \frac{1}{K} \omega_A - \frac{1}{1 + K} \omega_H &= 0 \\
\omega_H - \frac{1}{1 + K} \omega_A - \frac{K}{1 + K} \omega_B &= 0
\end{align*}
\quad (7)
\]

In the type: K is the characteristic parameter of the 2K-H type planetary transmission.

In this paper, 2K-H planetary gear components A, B, H, internal gear B input, the center wheel A fixed \( (n_a=0) \), the planet carrier H output, the planetary gear mechanism gear ratio:

\[ i = i_C^A \cdot i_{BH} = \frac{Z_2}{Z_1} \cdot (1 + \frac{1}{K}) = -2 \quad (8) \]

3. Modeling of planetary gear mechanism

Planetary gear mechanism models can be built using UG NX8.0. According to the meshing parameters designed in Table 1, select the corresponding model in UG interface, input the above model parameters to the GC Toolkits and set the position of each model [3]. The three-dimensional model is shown in Figure 2.
To guide the model built by UG into ADAMS, you need to convert the model file into a Parasolid-formatted text file. In the UG user interface, select File, Export, Parasolid, then select the model and enter the file name and save * x-t format, click OK to complete the physical output. Kinematics simulation for ADAMS environment.

4. Kinematics Simulation of Planetary Gear Mechanism

4.1 Establishment and Constraint of Virtual Prototype

ADAMS (Automatic Dynamic Analysis of Mechanical Systems) is a virtual prototype simulation software developed by American MDI Company. Open the ADAMS software, select the file (File), import (Import), UG output Parasolid format file, the entity model into [4]. Set the rigid body material, impose constraints on the virtual prototype according to the design requirements. It is shown in Figure 3:

(1) The rotation pairs of the fixed axle gears 1 and 2 relative to the ground, the revolving pairs of the internal gears relative to the ground, the revolving pairs of the planet wheels relative to the planet carrier, and the revolving pairs of the planet carrier relative to the ground.

(2) The fixed pair of the shaft 1 relative to the fixed axis gear 1, the fixed pair of the shaft 2 relative to the sun wheel and the ground, and the fixed pair of the fixed axis gear 2 relative to the internal gear.
4.2 Contact force parameter settings

ADAMS contact force calculation method is based on the impulse function method (IMPACT function) or based on the compensation method (Restitution function), the IMPACT function method is used to define the collision between gear meshing contacts, the contact type is entities and entities, K is contact parameters Stiffness coefficient, $K = \frac{4}{3} R^{1/2}E$. The basic physical quantity of the elastic deformation of material under external force depends on the structure shape and material of the collision object [5], in which:

$$
\frac{1}{E} = \frac{1}{R} + \frac{1}{R_2} \left( \frac{1 - \mu_1^2}{E_1} + \frac{1 - \mu_2^2}{E_2} \right)
$$

(9)

In the type: $R_1$ and $R_2$ are the equivalent radius of the contact object at the contact point; $\mu_1$ and $\mu_2$ are the Poisson's ratio of the two contacted materials; $E_1$ and $E_2$ are the elastic modulus of the two contact object materials.

Gear material between gears has a Poisson's ratio of $\mu_1=\mu_2=0.29$ and elastic modulus of $E_1=E_2=2.07\times10^5\,\text{N/mm}^2$. The contact stiffness coefficient $K_1=1.26\times10^6\,\text{N/mm}^{3/2}$ of the fixed gear 1 and 2 can be obtained by calculation. The sun Wheel and planetary wheel's contact coefficient of rigidity is $K_2=7.83\times10^5\,\text{N/mm}^{3/2}$, planetary gear and ring gear’s contact stiffness coefficient is $K_3=1.05\times10^6\,\text{N/mm}^{3/2}$, contact force index takes 1.5; damping coefficient takes 50N-s/mm. The depth of penetration is 0.1, the static friction coefficient and the dynamic friction coefficient are 0.15 and 0.1 [6]. According to the parameters of the contact force, a total of 7 pairs of physical contact forces are added between the fixed-shaft gears 1 and 2, the center wheel A and the planetary wheel C, the inner gear B and the planetary wheel C.

4.3 Simulation results

Applying a constant rotational speed of 3500 r/min to the fixed-axis gear 1, a load torque of 402 Nꞏm is applied to the planetary carrier opposite to the angular velocity. In order to ensure that the application process does not change suddenly, Step function is used and the load torque is Step $(\text{time}, 0, 0, 0, 1, 4, 0.02e5)$. The simulation time is set 1s, which uses I3 integrator, and its step is 0.001. The input speed curve of fixed-axis gear 1 is shown in Fig.4.

Figure 4 Fixed-axis gear 1 input speed curve

Planetary carrier output speed curve shown in Figure 5.
It can be seen from the simulation results that the input rotational speed of the fixed-axis gear 1 is positive, and the speed is constant during the simulation time, which is in consistent with the design requirements. The planetary carrier as the output shaft has a negative value and meets the planetary gear meshing relationship. Within 0-0.2s, the fluctuation is large, and the rotation speed fluctuates around -10500 °/s and it tends to be stable.

4.4 Error Analysis
The actual input and output components in the planetary gear mechanism speed and transmission ratio shown in Table 2.

| Transmission ratio | Input speed (°/s) | Output speed (°/s) |
|--------------------|-------------------|--------------------|
| Theoretical value  | 21000             | -10500.0           |
| 2.00               |                   |                    |
| Actual value       | 21000             | -10454.7           |
| 2.01               |                   |                    |
| Error (%)          | 0                 | 0.43               |

By comparing the theoretical values, it can be found that the gear speed direction meets the meshing relationship. The simulation values and theoretical values of the speed and the transmission ratio are compared. The error is within a reasonable range. It mainly comes from the gear assembly precision and collision and the planetary frame applied load torque caused by the gear meshing into the lead in the transmission process and the meshing impact caused by the factors, which is consistent with the actual gear, the correctness of the proof of the reliability and the design parameters of the prototype model.

5. Summary
(1) The design and simulation of fixed-shaft gear drive and 2K-H planetary gear drive provide a reference for further research on the combination of planetary gear drive.

(2) The three-dimensional solid model was established by UG NX8.0 and the kinematics simulation was carried out by using ADAMS 2013 simulation software. Comparison between simulation results and theoretical values, this conforms to the dynamic characteristics of gear transmission. The feasibility of combining the UG with the ADAMS software and the accuracy of the design parameters are verified. It provides a reference for the design of multi star gear mechanism.

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