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# *Construction and Analysis of the Stress Model of Key Components of Involute Planetary Reducer with Less Gear Difference*

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***Abstract.*** To clarify the force characteristics of involute planetary reducer core components and provide a theoretical basis for structural design and performance optimization, a 2KV involute planetary reducer (outer diameter 570 mm, width 245 mm, rated output torque 14700 N·m) was studied. Multibody dynamics software verified model accuracy. Results show the force model accurately reflects component force transmission; the small toothdifference planetary gear meshing force is dynamically balanced with crankshaft support force, and each crankshaft fulcrum force changes steadily with phase angle, laying a mechanical foundation for subsequent reducer structural optimization and reliability design.

***Keywords:*** involute less tooth difference, planetary reducer, less tooth difference planetary gear, stress model, mechanical analysis

## 1. Introduction

Precision reducers, as core transmission components of high-end equipment, are widely used in industrial robots and heavy machinery, whose performance determines equipment operation accuracy and reliability [1,2]. Driven by growing domestic demand, China's robot reducer industry has long been restricted by foreign core technologies. Traditional RV reducers suffer from high cycloidal wheel machining costs and post-load transmission accuracy degradation [3-6]. Involute planetary reducers adopt involute gear transmission, featuring mature theory, reliable processing and high load capacity [7,8], making them ideal for heavy-duty precision transmission. Planetary gears and crankshafts endure complex composite loads [9], whose force characteristics directly affect transmission efficiency and structural reliability. Taking a 2KV involute planetary reducer as the research object, this paper establishes an accurate force model for small toothdifference planetary gears and crankshafts, derives force formulas, verifies model accuracy via dynamic simulation, and clarifies force transmission laws, providing theoretical support for structural design, performance optimization and domestic independent research of such reducers.

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## 2. The transmission structure and core parameters of the reducer

The small tooth-difference involute planetary reducer is a closed differential train composed of input shaft, sun gear, high-speed planetary gear, crankshaft, small tooth-difference planetary gear, ring gear, output disc and support bearing.

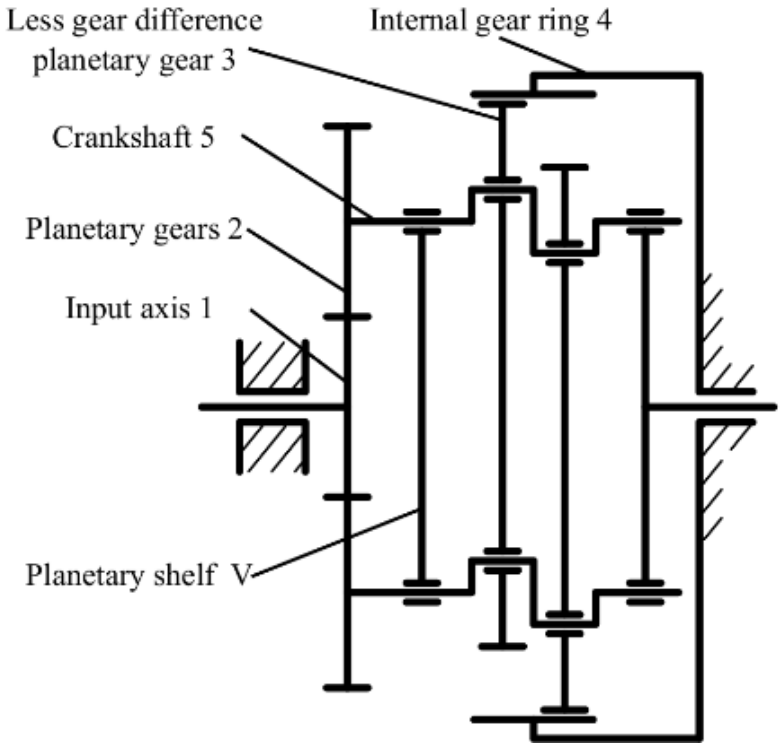


Figure 1. Structural diagram of the reducer

Power is input through the center gear shaft. After reduction by three evenly distributed planetary gears, the crankshaft converts rotation into eccentric revolution of the small toothdifference planetary gear. Meshing with the fixed ring gear causes rotation; torque is transmitted to the crankshaft via bearings, and the planet carrier integrates motion for output. The multifunctional crankshaft distributes lowstage torque and enables power output, serving as the transmission core.

Key gear parameters are determined by tooth matching, geometric verification and optimization (Table 1), laying a foundation for force modeling and calculation.

Table 1. Key design parameters of reducer gears

| Components | Modulus | Number of teeth | pressure angle | Top gap coefficient | Tooth top height coefficient |
|---|---|---|---|---|---|
| center gear | 2 | 18 | 20 | 0.25 | 1 |
| High-speed stage planetary gears | 2 | 105 | 20 | 0.25 | 1 |
| Less gear difference planetary gear | 2.5 | 166 | 20 | 0.25 | 0.65 |
| Internal gear ring | 2.5 | 170 | 20 | 0.25 | 0.65 |

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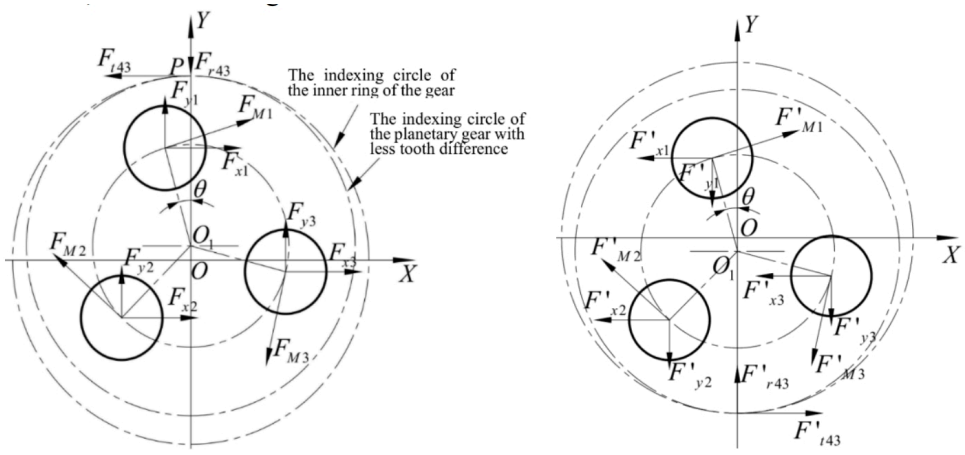
## 3. Construction and analysis of the force model of planetary gears with less tooth difference

### 3.1. Construction of stress model

Under ideal conditions, the supporting forces from the three crankshaft arm bearings on the small tooth-difference planetary gears act at the bearing hole centers, evenly distributed circumferentially. The ring gear meshing force acts at node P, decomposing into tangential and radial components. The arm bearing supporting force can be resolved into horizontal, vertical and tangential components to balance the torque from meshing. The two small tooth-difference planetary gears exhibit similar force laws, with only a 180° phase difference in ring gear meshing force. Thus, a single small tooth-difference planetary gear force model is established, as shown in Fig. 2.




(a) The force of the first piece of the planetary gear with less tooth difference

(b) The force of another planetary gear with less tooth difference

Figure 2. Schematic diagram of the force of a planetary gear with less tooth difference

### 3.2. Derivation of force calculation formulas

The two small tooth-difference planetary gears evenly share the reducer's rated output torque $T_0$, so the torque acting on a single planetary gear is $0.5T_0$. On this basis, the tangential meshing force and radial meshing force exerted by the inner gear ring are derived as follows:

$$F_{t43} = \frac{2000T_0}{2d_3} \tag{1}$$

$$\mathrm{F_{r43}} = \mathrm{F_{t43}} \tan \alpha_{34} \tag{2}$$

Where $d_3$ is the pitch circle diameter (mm) of the small tooth-difference planetary gear; α is the pressure angle (°) of the small tooth-difference involute planetary transmission.

To balance tangential and radial meshing forces, three crankshaft rotary arm bearings exert reaction forces at the bearing holes, with equal tangential and radial reaction forces at each hole. Let the tangential reaction force of a single bearing hole be $F_X$ and the radial reaction force be $F_Y$. The total reaction forces of the three bearing holes satisfy:

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$$\begin{cases} F_{t43} + \sum_{i=1}^{3} F_{xi} = 0 \\ F_{r43} + \sum_{i=1}^{3} F_{yi} = 0 \end{cases} \tag{3}$$

Due to the action point positions, the ring gear meshing force and crankshaft support reaction generate a derived torque M about the planetary gear center, balanced by the extra tangential force from three crankshafts. Let the crankshaft distribution circle radius be R and the torque-balancing tangential force per bearing hole is $F_M$; the torque balance equation is established via the torque balance principle:

$$\begin{cases} \sum_{i=1}^{3} F_{Mi} = 0 \\ \frac{d_3}{2} F_{t43} + 3R \bullet F_M = 0 \end{cases} \tag{4}$$

Using the above formula, the meshing force and rotor arm bearing support force can be calculated, providing input load data for the subsequent crankshaft force analysis.

## 4. Construction and analysis of crankshaft stress model

Treating the crankshaft as a spatial rigid shaft and assuming the angle between its axis and the vertical direction is θ, a 3D Cartesian coordinate system is established along the crankshaft axis to define the spatial positions of each load point: point 0 is the high-speed gear meshing point, points 1 and 4 are the planet carrier support bearing fulcrums, points 2 and 3 are the rotating arm bearing fulcrums connected to the small tooth-difference planetary gear, thus constructing the 3D force model of the crankshaft as shown in Fig. 3.

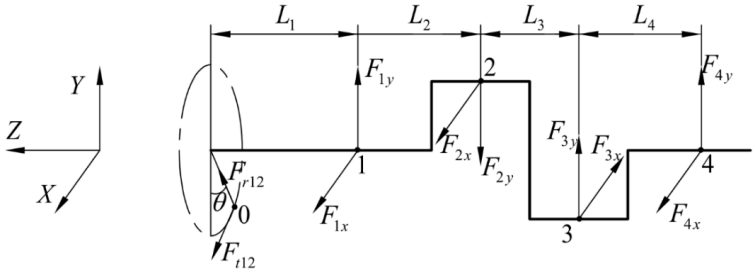


Figure 3. Force diagram of a single crank shaft

By the principle of action and reaction, the force exerted by the rotating arm bearing on the crankshaft is equal in magnitude and opposite in direction to the support force of the small tooth-difference planetary gear. The meshing force between the high-speed planetary gear and the central gear is calculable via gear meshing characteristics, while the support bearing constraint reaction is unknown; the overall crankshaft force is quantitatively analyzed by establishing spatial mechanical equilibrium equations.

The tangential force $F_{t12}$ and radial force $F_{r12}$ generated by the meshing of the high-speed planetary gear and central gear serve as the crankshaft input loads, with the calculation formulas as follows:

$$F_{t12} = \frac{T_1}{3R_1} \tag{5}$$

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$$F_{r12} = F_{t12} \bullet \tan\alpha_{12} = \frac{T_1 \bullet \tan\alpha_{12}}{3R_1} \tag{6}$$

$T_1$ is input torque (N·m), $R$ is indexing circle radius of input shaft center wheel (mm), $\alpha_{12}$ is pressure angle of high-speed planetary gear pitch circle (°). To simplify establishing the space mechanics equilibrium equation, tangential and radial forces are resolved into X- and Y-axis components.

$$\begin{cases} F_{12x} = F_{\mathrm{t}12}cos\theta + F_{r12}sin\theta \\ F_{12y} = -F_{t12}sin\theta + F_{\mathrm{r}12}cos\theta \end{cases} \tag{7}$$

The reaction force of the rotor arm bearing on the crankshaft, transmitted by the force of the planetary gear with less tooth difference, depends on the planetary gear's phase angle θ and is resolved into X- and Y-axis force components; the calculation formula is:

$$\begin{cases} F_{2x} = F_{xa} + F_M cos\theta \\ F_{2y} = F_{ya} + F_M sin\theta \end{cases} \tag{8}$$

$$\begin{cases} F_{3x} = F_{xb} - F_M cos\theta \\ F_{3\mathrm{y}} = F_{yb} - F_M sin\theta \end{cases} \tag{9}$$

Based on the spatial equilibrium conditions of a rigid body, the resultant forces in the X and Y axes are 0, and the resultant moment about the axis is 0. The following equilibrium equation is established to solve for the unknown support bearing reaction forces $F_1$ and $F_4$ .

$$\begin{cases} \sum F_x = 0 \\ \sum F_y = 0 \\ \sum M_x = 0 \\ \sum M_y = 0 \\ \sum M_z = 0 \end{cases} \tag{10}$$

$$F_{12x} + F_{1x} + F_{2x} - F_{3x} + F_{4x} = 0$$

$$F_{12y} + F_{1y} - F_{2y} + F_{3y} + F_{4y} = 0$$

$$F_{12y}\left(L_1 + L_2 + L_3 + L_4\right) + F_{1y}\left(L_2 + L_3 + L_4\right) - F_{2y}(L_3 + L_4) + F_{3y}L_4 = 0$$

$$F_{12x}\left(L_1 + L_2 + L_3 + L_4\right) + F_{1x}\left(L_2 + L_3 + L_4\right) + F_{2x}\left(L_3 + L_4\right) - F_{3x}L_4 = 0$$

$$R_2F_{t12} - e(F_{2x} + F_{3\mathrm{x}}) = 0$$

By solving the equations, the reaction forces of each support bearing are obtained, and the resultant forces at each fulcrum are synthesized for quantitative analysis of crankshaft forces. Using the 1500e reducer under rated load, computer programming determines the forces at each fulcrum and their variation with phase angle; the results are shown in Fig. 4.

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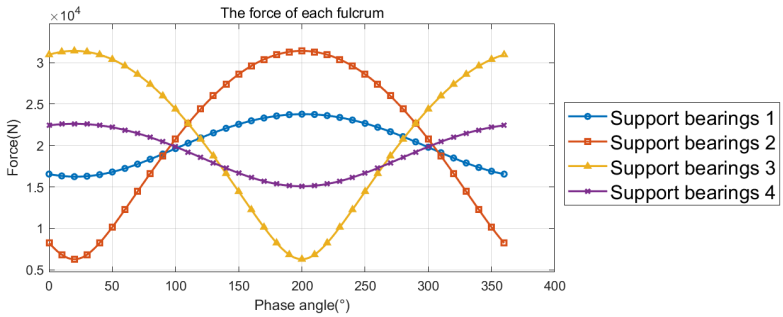


Figure 4. Stress results of each fulcrum of the crank shaft under rated load

As shown in Fig. 4, the force at each crankshaft fulcrum varies sinusoidally with phase angle, with no abrupt changes, indicating stable power transmission under rated conditions, consistent with design expectations. The two rotary arm bearings exhibit identical force fluctuations with opposite phases, ranging from 6000 N to 31,000 N. The forces on the two support bearings differ slightly; bearing 1 (near the input end) experiences a slightly higher force due to the additional load from the high-speed gear meshing force, which aligns with the force model derivation.

## 5. Conclusion

This paper studies the 2KV involute planetary reducer with less tooth difference, focusing on the planetary gear and crankshaft. Key conclusions: (1) Aiming at the planar rigid motion characteristics of the planetary gear with less tooth difference, a force model considering meshing force, rotor arm bearing support force, and derived torque balance is established, calculation formulas for each force component are derived, and the force law of the two planetary gears is clarified. (2) For the crankshaft's threedimensional spatial force characteristics, a spatial force model including highspeed gear meshing force, rotor arm bearing reaction force, and support bearing constraint reaction force is established.

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