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An Innovative Planetary Gear Reducer with Overcoming the "Dead Point"

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Abstract: This paper proposes an innovative planetary gear reducer with overcoming the "dead point", which is to be able to maintain expected reduction ratio by overcoming the "dead point" during the transmission motion. The motion characteristic is combined with the features of parallelogram mechanism transmission and planetary transmission mechanism with small teeth number difference. Structure analysis and force analysis of the mechanism are provided to explain working principle and kinematic feasibility of the mechanism. Three basic structure forms are proposed with a simple ring plate, two ring plates and three ring plates. In addition, the force analysis and the stress state analysis of ring plate based on a series of engineering software are carried on in detail. This study has the significance on design calculation and structure optimization of this kind of innovative planetary gear reducer.

Keywords: Planetary gear reducer, small teeth number difference, inner gear ring plate, force analysis, stress state analysis.

1. INTRODUCTION

Speed reducers are widely used in various applications for speed and torque conversion purposes [1]. A common speed reducer can realize speed reduction between input shaft and output shaft which occupy fixed relative positions. Due to its lower transmission ratio, the application of this kind of reducer is limited. In the late 20th century, with the development of industry automation, gear devices with large reduction ratio were used widely. Planetary drives with small teeth number difference (PDSTD) were also adopted widely in automation industry. For example, planetary drive were considered by Nabtesco *et al.* [2], and Gomà Ayats *et al*. [3]. Research on the effect of mesh phase on wave vibration of spur planetary ring gear was carried out by Wang *et al.* [4]. As a kind of PDSTD, three-ring gear reducer has a big single stage gear ratio (up to 99), high mechanical efficiency (up to 0.92~0.96), the strong bearing capacity and compact structure, etc. However, it is difficult to process inner gear ring plate by slotting machining, and it is also difficult to guarantee the precision since the lacks of its structure. In addition, there are only two cranks. But in order to achieve static balance and across the "dead point" position, three ring plates are needed necessarily. Since the three ring plates motion with 120° phase angle between each other can just realize the static balance and is unable to realize the dynamic balance, the large vibration will emerge in high speed working conditions for this kind of reducer.

A new type of planetary gear reducer, whose features can overcome the "dead point" of three-ring gear reducer, is introduced in this paper. This mechanism has been granted a patent by the People's Republic of China [5]. Three basic structure forms were proposed: A simple ring plate, two ring plates and three ring plates. For the two ring plates, it can realize the static balance, so the small and medium load working condition is suitable for this kind; for three ring plates, high speed and overloaded working condition is suitable for this type due to their easy realization of the dynamic balance. Researches on this three crank ring plate gear reducer were carried out about 3D parameterized modeling of the major parts and the whole device [6], and design calculation software were develop by Guan *et al.* [7].

In this paper, we intend to demonstrate the performance of the speed reduction mechanism which mentioned above has been proven by theoretical analysis and the finite element analysis.

2. STRUCTURE AND TRANSMISSION PRINCIPLE INTRODUCTIONS

Fig. (**1**) shows the structure of one kind planetary gear reducer consisting of one inner gear ring plate, one external spur gear, four ball bearings, three eccentric sleeves, one input shaft, one output shaft and two support shafts. In order to let teeth of the external gear mesh the teeth of the inner gear ring plate, a radial movement of the external gear relative to the internal gear is needed. This radial movement is realized through rotational movement of a crankshaft. Here the input shaft and two support shafts are crankshafts, so it has three cranks, and the eccentric distance of the three eccentric sleeves assembled on three crankshafts is the length of crank.

Fig. (**2**) shows a three crank ring plate transmission mechanism. In Fig. (**2**), O_1ABO_2 and O_1ACO_3 are two parallelogram double crank connecting rod mechanisms; O_1A, O_2B and O_3C are three cranks; BAC is the connecting rod, where the inner gear ring plate is just installed; The length of three cranks is equal; what's more, the value is just the center distance between the external gear and the inner gear ring plate. When the driving crank O_1A rotates at a high speed, the motion of inner gear ring plate with the

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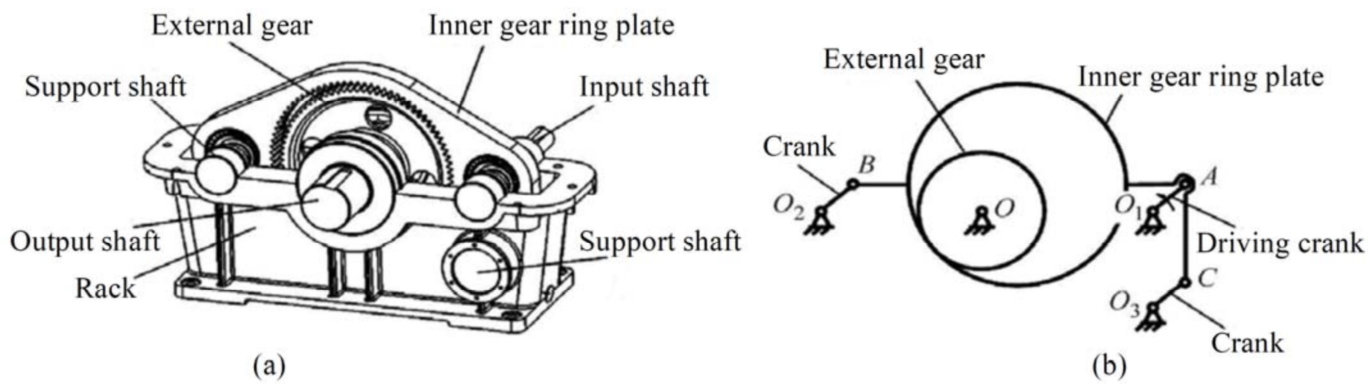


Fig. (1). Structure of one kind planetary gear reducer with a single ring plate and three cranks.

connecting rod is translation with revolution and no rotation, thus promotes external gear to do reverse fixed axis rotation at low speed. From the reference [8], we can see the transmission ratio of the mechanism is $i = \frac{Z_1}{Z_2 - Z_1}$,

where Z_1 is tooth of output external spur gear and Z_2 is tooth of inner gear ring plate. Obviously the tooth number difference of the two gears is not great, And the smaller the tooth number difference, the greater the absolute value of velocity ratio. In order to obtain larger transmission ratio, usually the tooth number difference $Z_2 - Z_1$ is from 1 to 4.

Actually, the mechanism can work smoothly in general position even in the absence of the crank O_3C, but when the driven crank O_2B and the rod AB are collinear, it will be in a state of "dead point". Here the driven crank O_3C is set up for overcoming the "dead point", which is just shown as Fig. (**2a**).

The relative position of rotation center of the three crank is right triangle arrangement, when a group of parallelogram double crank connecting rod mechanism is in the "dead point" position, the transmission angle of another group of parallelogram double crank connecting rod mechanism is equal to 90° (Fig. **2**), which is just in the best state of movement transmission and can overcome the "dead point". Then the minimum transmission angle of this mechanism is 45°, which can meet the requirement of the minimum transmission angle no less than 40°.

3. FORCE ANALYSIS OF INNER GEAR RING PLATE

Because the inner gear ring plate is the key component which affects this kind of reducer transmission performance, the force analysis is premise of the strength analysis for this inner gear ring plate, which is the necessary work to do for researching this kind of reducer. The main structure parameters of this inner gear ring plate must be found out first before the force analysis.

3.1. Structure Size

The object of this study is the inner gear ring plate of three crank single ring plate gear reducer. For the inner gear ring plate, the modulus is 4mm, addendum coefficient is 0.6, pressure angle is 20°, angle of engagement is 58.34°, the modification coefficient of external gear x_1 is 0.18, the tooth number of external gear Z_1 is 75, the tooth number of inner gear ring plate Z_2 is 76, the thickness of inner gear ring plate is 180mm, and the other main parameters are as follows in Table **1**.

3.2. Force Analysis

Fig. (**3**) shows the force analysis diagram. The bearings are installed in A, B and C holes, which respectively connects to the input shaft, the support shaft and the eccentric sleeve. Middle hole is the internal gear, which forms meshing pair with the external gear installed on the output shaft. F_i is the inertial force of inner gear ring plate, whose direction is parallel to the position lines of crank;

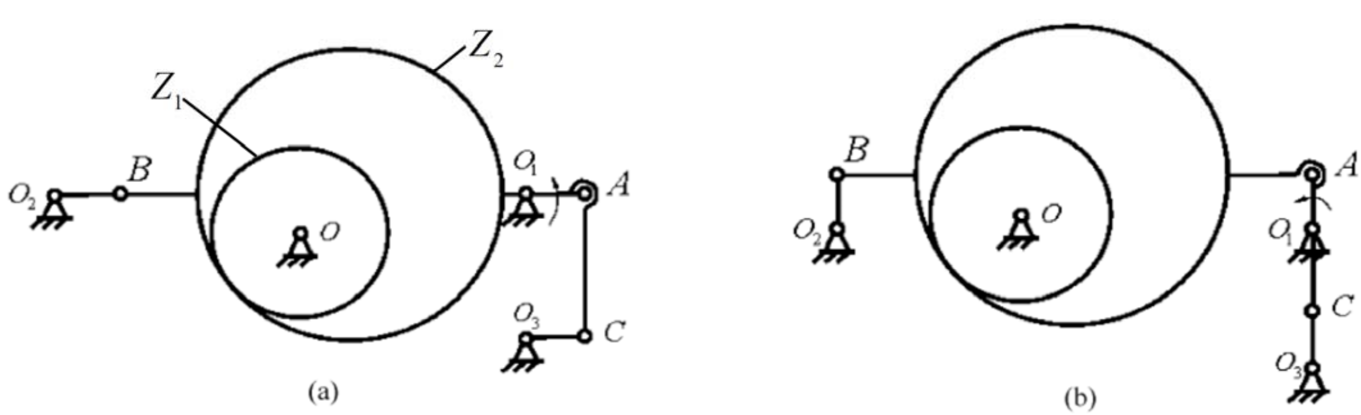


Fig. (2). Position of overcoming the "dead point".

F_n is the meshing force between external gear and internal gear, whose direction is along the two gear meshing line; F_{Ar}, F_{Br} and F_{Cr} are the load components along the direction of the cranks of each turning arm bearing; F_{At} is load component perpendicular to the direction of the crank for turning arm bearing on the input shaft. Because cranks of two support shafts are two dowel bars, the load component perpendicular to the direction of the cranks are all zero. According to Fig. (**3**), we have

$$F_i = m\omega^2 a \tag{1}$$

$$F_n = \frac{T_w}{r_{b1}} \tag{2}$$

$$F_{nr} = F_n \sin\alpha', F_{nt} = F_n \cos\alpha' \tag{3}$$

Table 1. Main Parameters of the Inner Gear Ring Plate for the Object of this Study

Serial Number	Parameters	Values
1	Modification coefficient x_2	1
2	Reference diameter d_2(mm)	304
3	Addendum circle diameter d_{a2}(mm)	300.92
4	Dedendum circle diameter d_{f2}(mm)	318.8
5	Outside diameter of planetary bearing D(mm)	200

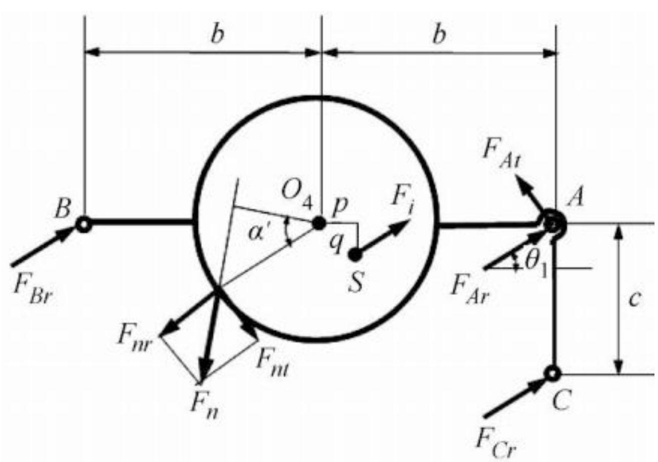


Fig. (3). Force analysis diagram.

According to the balance condition of inner gear ring plate in the direction of perpendicular to the crank, we can derive

$$F_{At} = F_{nt} = F_n \cos\alpha' \tag{4}$$

For simplicity, from Fig. (**3**), F_n, F_i and F_{At} are combined as an equivalent resultant force F along the direction of the crank and an equivalent torque M which acts on the center O_4 of the internal gear, shown as Fig. (**4a**), we can derive

$$\begin{cases} F = F_{nr} - F_i \\ M = F_n \cdot r_{b1} + F_{At} \cdot b \cdot \cos\theta_1 + (p\sin\theta_1 + q\cos\theta_1) \cdot F_i \end{cases} \tag{5}$$

$$F_{Ar} + F_{Br} + F_{Cr} - F = 0 \tag{6}$$

$$F_{Ar} b\sin\theta_1 - F_{Br} b\sin\theta_1 + F_{Cr}(b\sin\theta_1 + c\cos\theta_1) - M = 0 \tag{7}$$

To facilitate the solution of the above equations, the ring plate is assumed to be a rigid body, the turning arm bearings and the input shaft are assumed to be elastomers, and they have the same stiffness on the direction of the crank. Therefore, the forces of F_{Ar}, F_{Br} and F_{Cr} are proportional respectively to the elastic displacement x_A, x_B and x_C of each bearing on the direction of the force, shown as Fig. (**4b**). According to the elastic deformation of the ring plate, we can obtain compatibility equation

$$\frac{x_A - x_C}{x_B - x_C} = \frac{c\cos\theta_1}{2b\sin\theta_1 + c\cos\theta_1} = d \tag{8}$$

That is

$$x_A - d \cdot x_B + (d-1)x_c = 0$$

So

$$F_{Ar} - d \cdot F_{Br} + (d-1)F_{Cr} = 0 \tag{9}$$

According to Eqs. (6), (7), and (9), we have

$$F_{Ar} = \frac{F \cdot b \cdot \sin\theta_1 + F \cdot c \cdot d\cos\theta_1 + M(2d-1)}{2b(2-d)\sin\theta_1 + c(d+1)\cos\theta_1} \tag{10}$$

$$F_B = F_{Br} = \frac{F \cdot b \cdot (2\text{-d})\sin\theta_1 + F \cdot c \cdot \cos\theta_1 + M(2-d)}{2b(2-d)\sin\theta_1 + c(d+1)\cos\theta_1} \tag{11}$$

$$F_C = F_{Cr} = \frac{F \cdot b \cdot (1-d)\sin\theta_1 - M(d+1)}{2b(2-d)\sin\theta_1 + c(d+1)\cos\theta_1} \tag{12}$$

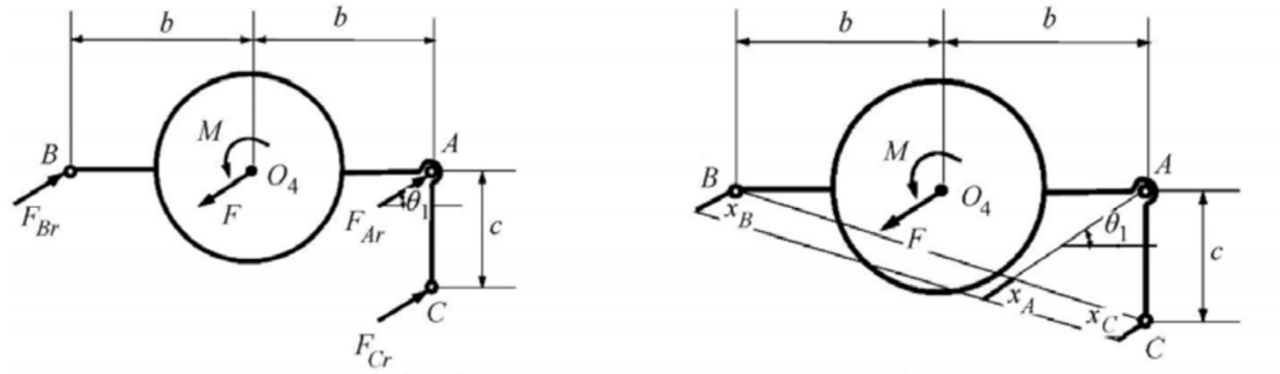
The force of bearing A is

$$F_A = \sqrt{F_{Ar}^2 + F_{At}^2} \tag{13}$$

In this study, the output torque is known as 15065 $N \cdot m$, transmission ratio is 75, and revolving speed of input shaft n1 is 980rpm. And it is assumed that horizontal distance from point of F_i to center point p is 56.58mm, vertical distance from point of F_i to center point q is 56.64mm, total quality of inner gear ring plate is 222.53kg.

Due to the complexity of calculation process and the difficulty of solving modified coordination equation, we use a special programming method is adopted to solve Eqs. (6), (7), and (9). According to the actual programming calculation process, it is found that the value of force F_A, F_B, F_C caused by parameters p, q and m's changes very small relative to the three force themselves, about one thousandth of them, thus it can be neglected. So the values of p, q and m here are hypothetical, and it can be thought that it has no effect on the calculation results.

Fig. (**5**) shows the force state calculated by programming during the crank rotating from 0° to 360°.



(a). Force analysis simplified diagram. **(b).** Elastic displacement assumption.

Fig. (4). Equivalent diagram of force analysis.

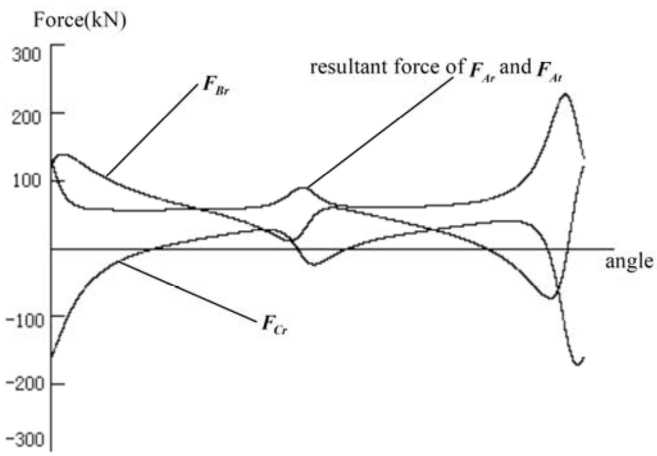


Fig. (5). Stress calculation results graph of the inner gear ring plate.

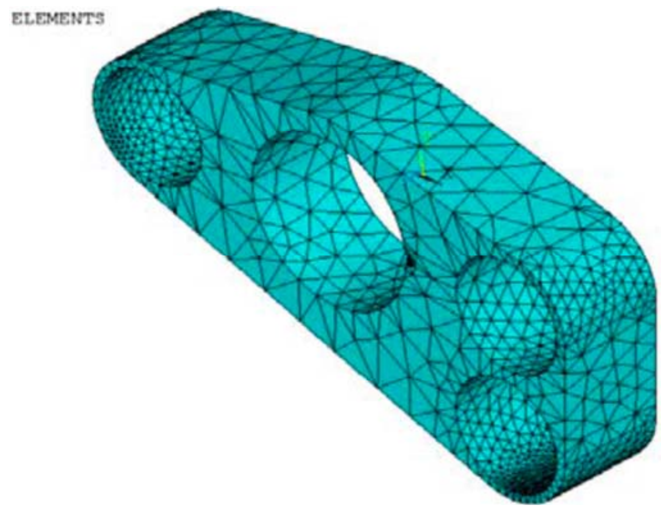


Fig. (6). Meshing situation of the inner gear ring plate.

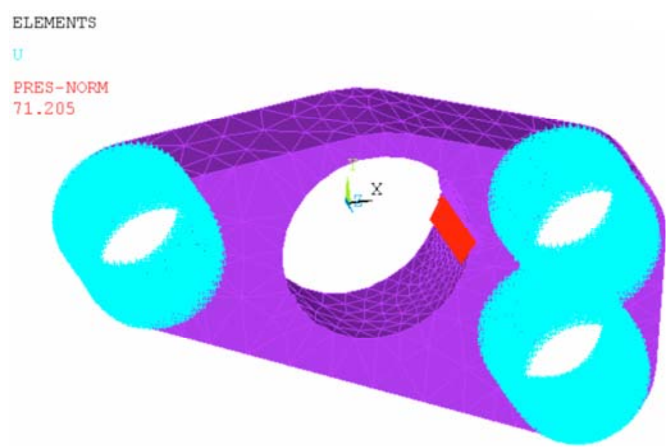


Fig. (7). Finite element model of the inner gear ring plate.

4. FINITE ELEMENT ANALYSIS OF INNER GEAR RING PLATE

4.1. Establishment of Model and Mesh

The CAD software SolidWorks is employed to establish the 3D solid model of the inter gear ring plate, and the plate model is saved as the iges file, which is then imported into the FEA software ANSYS, where the finite element analysis happens. Before the beginning of the analysis, we take the plate material as ASTM1045 and modified treatment, the center distance between the input shaft and output shaft is 350mm, distance between the supporting shaft and input shaft is 220mm, tooth difference between internal and external gear tooth is 1. Some parts are simplified for modeling, and the author also converts meshing F_n to pressure on the surface of tooth mesh of the inner gear ring plate, the value of which is 106879N, the direction of which is tangent to base circle of the external gear. Since constraints can be considered circumferential constraints along three inner holes, in ANSYS, it is in the form of bearing constraints. There are 22210 nodes and 12744 elements for the FEM model, which is shown in Fig. (**6**). The tetrahedral element named solid92 with 10 nodes is used in the FEM model. The load and constraint conditions are shown in Fig. (**7**).

4.2. Results Analysis

Based on the above modeling situations, the finite element analysis results of inner gear ring plate are as follows: Equivalent stress diagram is shown in Fig. (**8**), and composite deformation diagram is shown in Fig. (**9**).

By observing the composite deformation diagram, the largest composite deformation is 0.013412mm, which just

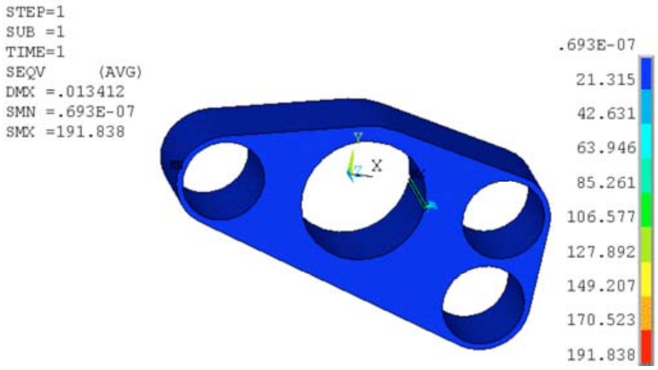


Fig. (8). Equivalent stress diagram of the inner gear ring plate.

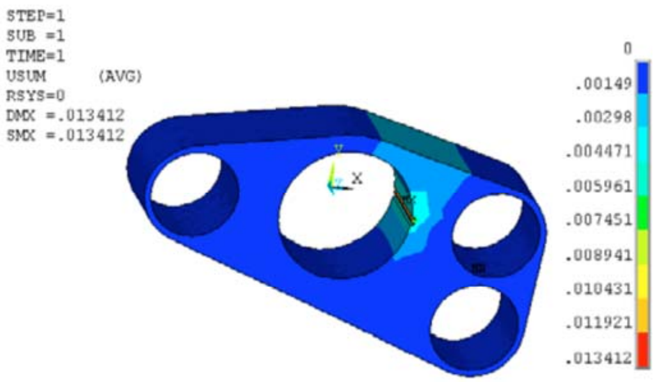


Fig. (9). Composite deformation diagram of the inner gear ring plate.

occurs on the addendum of the inner gear ring plate, according to the result, the deformation is small. According to the equivalent stress diagram, the maximum value is 191.838MPa, which just takes place on the diagram of inner gear ring plate, and stress of the other parts is far less than the limit value. By Handbook of Mechanical Design [9], the yield strength of ASTM1045 is 355MPa. Then the maximum stress value of inner gear ring plate is less than the yield strength, so it is safe.

5. VIRTUAL ASSEMBLY OF THE MACHINE

Although the structure of the three crank single ring plate reducer is simple, the design and calculation is complex, in order to test rapidly whether the design results are appropriate or not, and to improve the efficiency of product design, it is necessary to quickly implement virtual assembly of the real product. By using SolidWorks, the low-level sub assembly is first carried out through a bottom-up approach, then some small assembly units can be got by this method, then the series of small assembly units are assembled into a higher level assembly, finally the whole machine assembly is completed. Fig. (**10**) shows the virtual assembly of the whole machine, and the exploded view of the overall units is generated in the process of virtual assembly, shown in Fig. (**11**), which helps to observe directly the assembly relationships between parts of this reducer.

6. CONCLUSIONS

On the basis of analyzing the structure and working principle of the three crank ring plate reducer, we firstly discuss the forces in detail, and then show the concrete calculation process of the internal gear plate by the VB software. Then we take the single ring plate reducer as the study object, employ SolidWorks to establish the 3D solid model for the internal gear plate of the reducer, adopt ANSYS to analyze the internal gear plate with finite element analysis, and finally derive the composite deformation diagram and equivalent stress diagram. From the analysis result, we find that the strength of the ring plate meets the requirement and its structural design is reasonable.

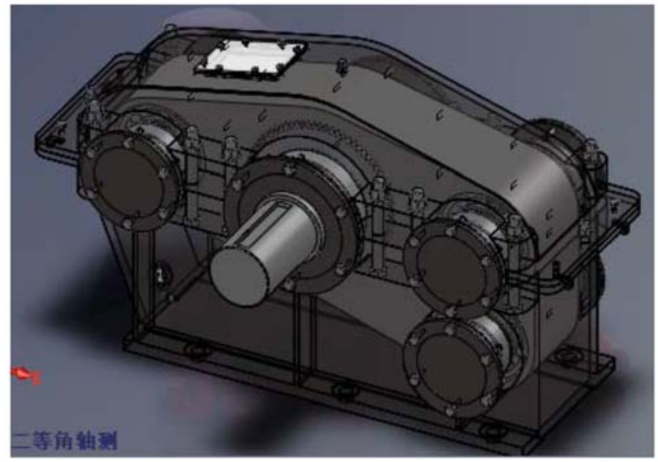


Fig. (10). Virtual assembly of the whole machine.

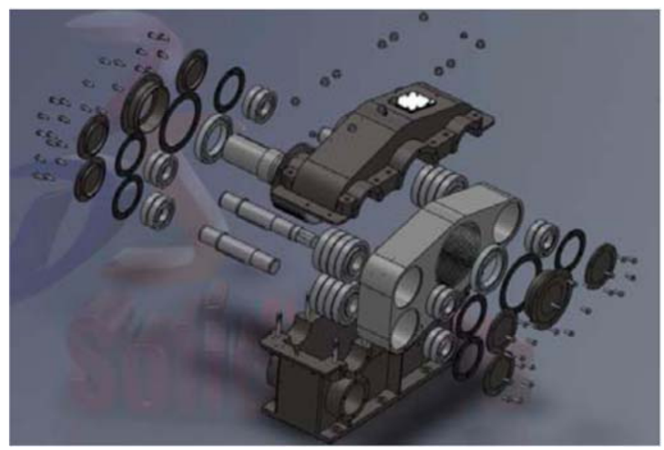


Fig. (11). Exploded view of the overall units.

NOMENCLATURE

ω	Angular velocity of crank
T_w	Output torque
r_{b1}	Radius of base circle for external gear
F_i	Inertial force of inner gear ring plate
F_{nr}	Component along the direction of the crank
F_{nt}	Component perpendicular to the direction of the crank
α'	Meshing angle
F_n	Meshing force between external gear and internal gear

F_{Ar} Load component along the direction of crank for bearing A

F_{At} Load component perpendicular to the direction of crank for bearing A

F_{Br} Load component along the direction of crank for bearing B

F_{Cr} Load component along the direction of crank for bearing C

F Equivalent resultant force

M Equivalent torque

x_A Elastic displacement on the x direction for bearing in hole A

x_B Elastic displacement on the x direction for bearing in hole B

x_C Elastic displacement on the x direction for bearing in hole C

θ_1 Angle between x and horizontal direction

b Half the center distance between the hole A and B

c Center distance between the hole A and C

m Total mass of ring plate

a Center distance between external gear and internal gear

p Horizontal distance between F_i and center point

q Erect distance between F_i and center point

CONFLICT OF INTEREST

The authors confirm that this article content has no conflict of interest.

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