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Experimental and numerical Study of Plastic Worm Meshed with Steel Helical Gear

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Abstract Transmission mechanism of the plastic worm meshed with steel helical gear is applied to achieve power transmission and motion transfer in this paper. Tooth profiles equations of gear pair are derived and general meshing conditions are proposed in terms of gear geometry. Mathematical model of tooth profiles of gear pair is established. Contact stress analysis and general evolution law of tooth profiles are discussed by finite element method. Material characteristic, mesh generation, contact definition and constraint conditions are given. The comparisons with analysis results of steel worm meshed with steel helical gear are also provided. Through the hobbing process, injection molding, and machining center, samples of steel helical gears, plastic worm and steel worm are completed respectively. The characteristic test of transmission mechanism of plastic worm meshed with steel helical gear is carried out based on the micro transmission experimental platform. The contrast results show that it has better transmission performance.

Keywords Plastic worm; Steel helical gear; Mathematical model; Contact stress analysis; Performance experiment
1. Introduction

The worm drive has the advantages of large single-stage transmission ratio, small operation noise, low vibration, self locking and high bending strength. And it is of great interests to carry out the geometric design, theoretical characteristic analysis and experimental study of the worm gear drive currently by scholars. Litvin et al. provided a worm gear drive with improved bearing contact, reduced level of transmission errors and lessened sensitivity to errors of alignment. The generated hob and worm tooth profiles were mismatched which is different with the existing method. Design, simulation of meshing, and contact stresses for the developed gear pair were carried out. In addition, the new geometry of face worm gear drives with conical and cylindrical worms was also proposed. The generation of the face worm gear was based on a tilted head-cutter (grinding tool) (Litvin et al. 2002, 2007). Simon researched the influence of tooth errors and shaft misalignments on loaded tooth contact in different types of cylindrical worm gears. The geometric generation and theoretical descriptions were presented. And tooth contact conditions were discussed based on the numerical example (Simon 2015). Sohn et al. investigated the influence of gear hobbing and shaft misalignments on the geometric interference of cylindrical worm gear set. Utilizing the surface separation topology, the interference can be avoided not only when the hob oversize is larger than the upper limit, but also when the hob oversize is smaller than the lower limit (Sohn and Park 2017). Pawlus et al. studied the geometry and machining of concave profiles of the ZK-type worm thread. The concave thread was made by tool of straight line, introducing twist tool setting in
relation to worm. Surface roughness of worms formed by rotary file was comparable with that of commercial worms (Skoczylas and Pawlus 2016). Chen et al. proposed a novel worm drive with planar internal gear and crown worm. Superior meshing performance of this worm drive was demonstrated and its performance test was also carried out through the machined gear prototype (Chen et al. 2013). Deng et al. presented a new type of end face engagement worm gear which can minimize or overcome the gear backlash. Mathematical models and theoretical analysis were given through considering the different factors (Deng et al. 2015).

However, the practical applications for worm gear drive in industry are limited due to low transmission efficiency, poor lubrication, and high manufacturing cost. Over the past ten years, significant progress has been made in the development and application of plastic gear, a new type of non-metallic gear form, which contains the advantages of light weight, small inertia, low noise, good self-lubricating, high production efficiency, and low cost, etc. And transmission performance of the modified plastic gear can approximately equivalent to the metal gears under special conditions (Bravo et al. 2015; Bravo et al. 2018; Hasl et al. 2017; Koffi et al. 2016). To maintain the advantages of worm gear drive and to overcome its shortcomings, scholars at home and abroad used helical gears instead of traditional worm gears to form worm helical gear transmission. Koide et al. investigated the transmission characteristics of steel worm meshed with plastic helical gear, including the generation principle of engagement temperature, transmission efficiency, load capacity and service life. And the calculation methods of contact strength and life
were also put forward (Koide et al. 2010, 2011). Nomura et al. obtained the prediction method of fatigue life through discussing the meshing conditions of different types of worm and plastic helical gears (Nomura et al. 2015). Horiuchi et al. proposed a new type of worm meshed with involute helical gear, and verified that the transmission efficiency is the same as that of traditional worm gear drive (Horiuchi et al. 2009). Hao et al. studied the meshing theory and thermodynamic between the plastic helical gear and steel worm. The relationship among the various factors in the heat balance process was obtained (Hao et al. 2010).

At present, more researches on the worm helical gear focus on the form that plastic helical gear meshed with steel worm. There are few researches on the engagement form between plastic worm and steel helical gear. In this paper, the plastic worm meshed with steel helical gear is applied to achieve power transmission and motion transfer. Mathematical model of tooth profiles of gear pair is established. Contact stress analysis and general evolution law among different tooth flank are discussed using finite element method. Samples of steel helical gears, plastic worm and steel worm are completed respectively, and characteristic test of tooth profiles is carried out based on the micro transmission experimental platform.

2. Mathematical Model of Plastic Worm Meshed with Steel Helical Gear

2.1 Tooth profiles equation

For spur gear drive, the engagement in and out of whole tooth width can result in poor gear transmission stability and high vibration and noise. The spiral gear which
meshes in axial direction from one end of the tooth width to the other end is designed, and it can improve the transmission stability. It is known that the meshing conditions of the worm and helical gear are as follows:

1. Normal modulus $m_{nw}$ of the worm is equal to the normal modulus $m_{nh}$ of the helical gear.
2. Lead angle $\gamma$ of the worm is equal to the helical angle $\beta$ of helical gear.
3. Spiral direction of the worm is the same as that of the helical gear.

Considering the worm as a helical gear with large helix angle, the worm meshed with helical gear is essentially crossed-axis helical gear transmission with large helix angle. So the geometric calculation formula of the crossed-axis helical gear drive is also suitable for that of the proposed transmission form.

Four spatial coordinate systems are established in Fig.1 for tooth profiles derivations. The movable coordinate system $S_1(o_1, x_1, y_1, z_1)$ is connected to the plastic worm and the movable coordinate system $S_2(o_2, x_2, y_2, z_2)$ is connected to steel helical gear. The fixed coordinate systems are $S_f(o_f, x_f, y_f, z_f)$ and $S_p(o_p, x_p, y_p, z_p)$.

Axis $z_1$ is coinciding with worm axis and axis $z_2$ is coinciding with helical gear axis. The space distance between axis $z_f$ and axis $z_p$ is $a$, which is also the central distance.

The shaft angle between axis $z_f$ and axis $z_p$ is $90^\circ$. $\omega^{(1)}$ and $\omega^{(2)}$ are the rotational angle velocity of plastic worm and steel helical gear, respectively. $\phi_1$ and $\phi_2$ are the respective rotational angles. Meanwhile, the worm moves with velocity $v_{01}$ along the axis $z_f$, and the distance is $L$. The initial position of center point $O_1$ is coincidence with point $O_f$.  

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The transformation matrix from coordinate system \( S_1 \) to coordinate system \( S_2 \) can be expressed as follows.

\[
M_{21} = \begin{bmatrix}
\cos \phi_1 \cos \phi_2 & -\sin \phi_1 \cos \phi_2 & -\sin \phi_2 & a \cos \phi_2 - L \sin \phi_2 \\
-\cos \phi_1 \sin \phi_2 & \sin \phi_1 \sin \phi_2 & \cos \phi_2 & -L \cos \phi_2 - a \sin \phi_2 \\
\sin \phi_1 & \cos \phi_1 & 0 & 0 \\
0 & 0 & 0 & 1
\end{bmatrix}
\tag{2}
\]

And similarly the transformation matrix from coordinate system \( S_2 \) to coordinate system \( S_1 \) is written as follows.

\[
M_{12} = \begin{bmatrix}
\cos \phi_1 \cos \phi_2 & -\cos \phi_1 \sin \phi_2 & \sin \phi_1 & -a \cos \phi_1 \\
-\sin \phi_1 \cos \phi_2 & \sin \phi_1 \sin \phi_2 & \cos \phi_1 & a \sin \phi_1 \\
-\sin \phi_2 & -\cos \phi_2 & 0 & -L \\
0 & 0 & 0 & 1
\end{bmatrix}
\tag{3}
\]

To deriving the equation of tooth profiles of the worm, the coordinate system \( S_u \) is established by spiral motion along the direction of axis \( z_1 \) relative to the coordinate system \( S_1 \) in Fig.2.

According to above relationship, the transformation matrix from coordinate system \( S_u \) to coordinate system \( S_1 \) can be expressed as follows.

\[
M_{1u} = \begin{bmatrix}
\cos \theta & -\sin \theta & 0 & 0 \\
\sin \theta & \cos \theta & 0 & 0 \\
0 & 0 & 1 & \rho \theta \\
0 & 0 & 0 & 1
\end{bmatrix}
\tag{4}
\]

and the transformation matrix from coordinate system \( S_1 \) to coordinate system \( S_u \) is obtained as follows.

\[
M_{u1} = \begin{bmatrix}
\cos \theta & \sin \theta & 0 & 0 \\
-\sin \theta & \cos \theta & 0 & 0 \\
0 & 0 & 1 & -\rho \theta \\
0 & 0 & 0 & 1
\end{bmatrix}
\tag{5}
\]

Where \( \theta \) is the shaft angle and \( \rho \) is helix parameter.
We mention that the worm meshed with helical gear can be considered as the crossed-axis helical gear drive with large helix angle. So the equation of tooth profiles of the proposed gears can be expressed with the same form.

Taking the worm as the object, it is derived utilizing the manufacturing approach shown in Fig.3. Supposed that the tooth surface of the worm can be machined with a straight blade tool, the spiral motion is conducted based on the generation straight line which is the tangent curve of the spatial helix curve on the base cylinder. If the front knife face of the worm cutter is tangent to the worm base cylinder, the tooth profile angle of the tool is also the spiral angle of the worm base cylinder.

The arbitrary point $M$ on the generation straight line is written as

$$
\begin{align}
    x_u &= r_b \\
    y_u &= -u \cos \lambda_b \\
    z_u &= -u \sin \lambda_b
\end{align}
$$

where $r_b$ is the radius of basic circle of the worm. $\lambda_b$ is the lead angle of helix curve. $u$ is the straight line parameter.

Based on the transformation relationship in Eq. (3), tooth profile equation of the worm is obtained in coordinate system $S_1$ as follows.

$$
\begin{align}
    x_i &= r_c \cos \alpha + u \cos \lambda_b \sin \alpha \\
    y_i &= r_c \sin \alpha - u \cos \lambda_b \cos \alpha \\
    z_i &= -u \sin \lambda_b + pa
\end{align}
$$

where $p=r_b \tan \lambda_b$ and it is also the helix parameter. $\alpha$ is the tooth profile parameter.

The meshing equation is calculated through considering the normal vector and relative velocity. Normal vector $n$ is derived using Eq. (6) and it is

$$
\begin{align}
    n &= -\sin \lambda_b \sin \alpha i + \sin \lambda_b \cos \alpha j - \cos \lambda_b k
\end{align}
$$
Relative velocity $v_{12}$ of tooth profiles of the worm with respect to the helical gear is represented as follows.

$$v_{12} = \left[-\omega_{1} x_{1} - \omega_{2} (L + z_{1}) \cos \phi_{1}\right] i + \left[\omega_{1} y_{1} + \omega_{2} (L + z_{1}) \sin \phi_{1}\right] j$$

$$+ \left[\omega_{2} (a + x_{1} \cos \phi_{1} - y_{2} \sin \phi_{1}) + v_{01}\right] k$$

(9)

So the meshing equation is

$$\Phi = n g_{12} = \sin \lambda_{3} \left[ (p o_{1} + v_{01}) (x_{1} \cos \alpha + y_{1} \sin \alpha) + p o_{2} (L + z_{1}) \sin (\phi_{1} + \theta) \right]$$

$$- p o_{2} \cos \lambda_{3} (a + x_{1} \cos \phi_{1} - y_{1} \sin \phi_{1}) - p v_{01} \cos \lambda_{3} = 0$$

(10)

2.2 Numerical example

Given the basic parameters in Table 1, we use the Mathematica software to get the data points of the left and right side tooth surfaces according to the derived equation of the tooth profiles. Based on the surface function in software UG, the initial tooth surfaces are generated in Fig. 4(a). The solid model of the worm is established based on the further Boolean operation and solid generation function and it is depicted in Fig. 4(b).

The model of helical gear is provided in terms of the similar generation method. The result is shown in Fig.5.

Under UG assembly environment, the worm meshed with helical gear is assembled, and the assembly model is shown in Fig. 6(a). For the motion simulation, the worm and helical gears are respectively set as rotary pairs, and a constant speed is also set for the worm. Motion pair is provided and the solving parameters are set up. The animation of meshing process of the developed gear pair is obtained. The correctness of geometric modeling can be proved by simulation motion, and the point contact engagement is displayed in Fig. 6(b).
3. Stress Analysis of Gear Pair

3.1 Pre-processing of finite element analysis

The HyperMesh software can convert the geometric model into a high quality finite element model through a series of processing processes and it can lay the foundation for accurate and effective analysis.

3.1.1 Material and unit selection

The plastic worm material is set to PEEK 450G and the material of steel helical gear is 45°. The characteristics of two different materials are shown in Table 2.

For gear solid part, due to the limitation of the current contact algorithm, the application of high-order solid elements will cause the equivalent node contact force to oscillate between corner nodes and side nodes, which is unfavorable for checking and judging the contact state. In practical, the application of high-order hexahedral element for contact analysis may result in difficult convergence. Therefore, considering the calculation efficiency and accuracy, the low-order hexahedral element SOLID185 are applied to divide the solid mesh.

3.1.2 Mesh generation

The geometric model constructed in UG software is imported into HyperMesh through a special interface between UG and HyperMesh. In order to reduce the calculation amount, three teeth of helical gear are selected for analysis. For contact area of tooth profiles, the change of stress has an important influence on the accuracy of calculation result of contact stress. Under keeping the computation precision and
efficiency, the mesh size of tooth contact area is set to 0.08mm after trial, and the other regional unit is set to 0.2~0.5mm. The finite element models of the helical gear and worm are shown in Fig.7 and Fig.8, respectively.

3.1.3 Definition of tooth profile contact

For the finite element model, the TARGE170 element is used to simulate the tooth surface of the helical gear, and the CONTA173 element is used to simulate the tooth surface of the worm. After the trial calculation, the contact stiffness FKN is set to 1, which can satisfy the requirements of calculation efficiency and precision. The results are displayed in Fig.9.

3.1.4 Load and constraint

To achieve the rotational motion and applied torque, multipoint constraint element MPC184 unit is used in finite element analysis of the worm meshed with helical gear pair. Based on the multi-point constraint algorithm, the contact pair is established in the inner wall of the helical gear and the rotation center, and it is also established in the worm's inner wall and the rotation center. For the boundary load, the 0.1Nm is applied to the steel helical gear and the 6.3Rad/s speed is applied to the plastic worm.

3.2 FEA results

After solving the finite element model, the force distribution and contact situation of the worm meshed with helical gear pair can be obtained. Figure 10 shows the single tooth contact stress of a plastic worm meshed with steel helical gear. And the double teeth contact stress of a plastic worm meshed with steel helical gear is
displayed in Fig. 11. In addition, Contact stress of the single tooth of a steel worm meshed with steel helical gear is shown in Fig. 12. Contact stress of double teeth of a steel worm meshed with steel helical gear is depicted in Fig. 13. According to the Hertz contact theory, the shape of the contact area is ellipse in theory due to the effect of elastic deformation between the action surfaces. However, the tooth root bending deformation has the influence on the contact condition of tooth profiles. So the contact area is only an approximate ellipse in the finite element analysis.

For plastic worm meshed with steel helical gear, the maximum contact stress with the change of rational angle of the worm from two teeth to single tooth and from single tooth to two teeth is expressed in Fig. 14, where A and B are the double teeth and single tooth meshing area, respectively. For steel worm meshed with steel helical gear, the maximum contact stress with the change of rational angle of the worm from two teeth to single tooth and from single tooth to two teeth is also expressed in Fig. 15.

The following results can be obtained:

(1) During the meshing process, contact situation changes from theoretical contact point to actual contact region. And the plastic worm has greater contact area than the steel worm, which can effectively reduce contact stress in meshing process.

(2) The meshing area of double teeth of the plastic worm is increased by about 60% compared with the steel worm, which can improve the contact ratio of gear pair.

(3) The maximum stresses of plastic worm and steel worm appear in the meshing transition between single tooth and double teeth. The maximum stress of plastic worm meshed with helical gear drive is 300.5 MPa and for steel worm meshed with helical
gear drive, it is 770.2MPa. The contact stress on the top part of the helical gear is greatly increased because of the meshing impact. However, from double teeth to single tooth, the contact stress changes gently. The engagement of the helical gear with one tooth has little effect on the maximum contact stress change.

(4) Compared with the steel worm, the contact stress of plastic worm changes slowly, which reduce the vibration and noise during the transmission process.

(5) For the plastic worm meshed with helical gear, the maximum contact stress of tooth profiles of the plastic worm is always lower than that of the helical gear. Due to the plastic material characteristic, the elastic deformation can reduce the maximum contact stress of tooth profiles which can improve the load capacity.

4. Experimental Study

Based on the provided parameters in Tab.1, the steel helical gear is manufactured using the hobbing process. The steel worm is generated by the machining center and the plastic worm is produced with injection molding. The sample results are shown in Fig.16.

The experimental research on the transmission performance of plastic worm meshed with steel helical gear is carried out according to the developed micro transmission experimental platform. Meanwhile, the comparison experiment with the steel worm meshed with steel helical gear is also conducted. The input and output speeds, input and output torques, and running time of the transmission prototype are test. The status of the sample's tooth profiles after the test is analyzed.
Micro transmission platform test site is displayed in Fig. 17. The drive motor speed and magnetic powder brake load can be adjusted by the PC machine. The connection between the motor and the tested parts is high-speed torsion, and the requirement for the coaxial degree is relatively high. The mobile platforms 6 and 7, the lifting platform 8 and the rotary platform 9 are also adjusted by the PC machine.

Supposed that the output torque is 0.1Nm, the input speed is set to 500r/min, 1000r/min, 1500r/min, 2000r/min, 2500r/min and 3000r/min. The efficiency values under every speed should be recorded and the results are shown in Fig. 18(a).

In addition, supposed that the input speed is 1000r/min, the load torque is set to 0.05 Nm, 0.1 Nm, 0.15 Nm, and 0.2Nm. The efficiency values under every load should be recorded and the results are shown in Fig. 18(b).

It can be concluded from Fig. 18(a) that transmission efficiency results gradually decrease with the increase of the input speed, and the transmission efficiency of plastic worm meshed with steel helical gear is always higher than that of steel worm meshed with steel helical gear. At the speed of 500r/min, the transmission efficiency for plastic worm meshed with steel helical gear can reach 52%, while for the steel worm meshed with steel helical gear is only 42%. The reason is that the self lubrication characteristics of plastic material can reduce the tooth profile friction loss, thereby improving the transmission efficiency. And meshing impact occurring for steel worm meshed and steel helical gear is also other influence factor.

Under a certain speed, transmission efficiency results gradually increase with the increase of the load torque in Fig. 18(b). But for the plastic worm meshed with steel
helical gear, transmission efficiency declines at the torque of 0.15Nm because of the excessive load and temperature rise. It produces elastic deformation making the incorrect meshing state.

We find that the plastic worm meshed with steel helical gear shows excellent transmission performance under low power transmission. In addition, to improving transmission efficiency, it is a feasible way to increase the applied torque and reduce the speed of transmission.

Meanwhile, the tooth profiles conditions after test are displayed in Fig.19. Compared with the steel worm, the plastic worm meshed with steel helical gear has slighter wear and it exhibits better wear resistance and service life.

Usually, the plastic gear usage is limited by not only poor mechanical properties but also equally poor temperature limits and poor heat conduction properties. For a long and continuous work process, the plastic worm meshed with steel helical gear may show the engagement instability because of the aforementioned reasons. The working conditions and requirements of the device should be considered for practical applications. And the further study on dynamic characteristics will be carried out in the future to get the transmission performance evolution results.

5. Conclusions

Generation of the plastic worm meshed with steel helical gear is provided based on the gear geometry and applied theory. Spatial relationships are written in terms of the established coordinate systems. Tooth profiles equations of gear pair are derived and
meshing condition is also calculated. Numerical example is illustrated according to the given design parameters and three-dimensional solid models are established.

By finite element method, the contact stress of tooth profiles of plastic worm meshed with steel helical gear is analyzed. Material characteristic, mesh generation, contact unit definition and constraint conditions are proposed. Meanwhile, the comparisons with steel worm meshed with steel helical gear are also carried out. The results are summarized as follows: (1) During the meshing process, the plastic worm has greater contact area than the steel worm. (2) The meshing area of double teeth of the plastic worm is increased by about 60% compared with the steel worm. (3) The maximum stress of plastic worm meshed with helical gear drive is 300.5MPa and for steel worm meshed with helical gear drive, it is 770.2MPa. (4) Compared with the steel worm, the contact stress of plastic worm meshed with steel helical gear changes slowly. And the maximum contact stress of tooth profiles of the plastic worm is always lower than that of the helical gear.

Through the hobbing process, injection molding, and machining center, samples of steel helical gears, plastic worm and steel worm are completed respectively. Transmission mechanism test of plastic worm meshed with steel helical gear is carried out based on the micro transmission experimental platform. The results are summarized as follows: (1) Transmission efficiency of plastic worm meshed with steel helical gear is higher than that of steel worm meshed with steel helical gear. (2) The plastic worm meshed with steel helical gear has slighter wear and it exhibits better wear resistance and service life.
Acknowledgments

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Table 1 Basic parameters of the worm meshed with helical gear

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<th>Module (mm)</th>
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<th>Helix angle (°)</th>
<th>Pressure angle (°)</th>
<th>Diameter of pitch circle (mm)</th>
<th>Thread direction</th>
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<td>0.95</td>
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<td>20</td>
<td>12.675</td>
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<tr>
<td><strong>Helical gear</strong></td>
<td>0.95</td>
<td>13</td>
<td>13</td>
<td>20</td>
<td>8.446</td>
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Table 2 Material characteristics

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<tr>
<th></th>
<th>Material</th>
<th>Elastic modulus (MPa)</th>
<th>Poisson ratio</th>
<th>Density (g/cm³)</th>
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<td><strong>Plastic worm</strong></td>
<td>PEEK 450G</td>
<td>3.8e3</td>
<td>0.4</td>
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<td><strong>Steel worm</strong></td>
<td>45°</td>
<td>2.09e5</td>
<td>0.269</td>
<td>7.85</td>
</tr>
<tr>
<td><strong>Steel helical gear</strong></td>
<td>45°</td>
<td>2.09e5</td>
<td>0.269</td>
<td>7.85</td>
</tr>
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Figure captions:

Fig.1 Spatial coordinate systems
Fig.2 Coordinate system \( S_u \)
Fig.3 Manufacturing method of ZI worm
Fig.4 Solid model of the worm
Fig.5 Solid model of the helical gear
Fig.6 Engagement condition of the worm meshed with helical gear
Fig.7 Finite element model of the steel helical gear
Fig.8 Finite element model of the plastic worm
Fig.9 Contact definition of gear pair
Fig.10 Single tooth contact stress of a plastic worm meshed with steel helical gear
Fig.11 Double teeth contact stress of a plastic worm meshed with steel helical gear
Fig.12 Single tooth contact stress of a steel worm meshed with steel helical gear
Fig.13 Double teeth contact stress of a steel worm meshed with steel helical gear
Fig.14 The maximum contact stress for plastic worm meshed with steel helical gear
Fig.15 The maximum contact stress for steel worm meshed with steel helical gear
Fig.16 Samples of the steel worm, plastic worm and steel helical gear
Fig.17 Micro transmission platform test site

1-Drive motor; 2-Rotating speed and torque sensor I; 3-Test gearbox; 4-Rotating speed and torque sensor II; 5-magnetic powder brake; 6-mobile platform with X direction; 7-mobile platform with Y direction; 8-the lifting platform with Z direction; 9-Rotational platform

Fig.18 Test results of transmission efficiency
Fig.19 Conditions of tooth profiles after test
Fig. 1 Spatial coordinate systems

59x87mm (300 x 300 DPI)
Fig. 2 Coordinate system $S_u$

51x56mm (300 x 300 DPI)
Fig. 3 Manufacturing method of ZI worm

51x65mm (300 x 300 DPI)
Fig. 5 Solid model of the helical gear

121x118mm (96 x 96 DPI)
Fig. 7 Finite element model of the steel helical gear

211x150mm (96 x 96 DPI)
Fig. 8 Finite element model of the plastic worm

165x158mm (96 x 96 DPI)
Fig. 9 Contact definition of gear pair

101x87mm (96 x 96 DPI)
Fig. 10 Single tooth contact stress of a plastic worm meshed with steel helical gear

174x65mm (96 x 96 DPI)
Fig. 11 Double teeth contact stress of a plastic worm meshed with steel helical gear

171x65mm (96 x 96 DPI)
Fig. 12 Single tooth contact stress of a steel worm meshed with steel helical gear

175x67mm (96 x 96 DPI)
Fig. 13 Double teeth contact stress of a steel worm meshed with steel helical gear

174x67mm (96 x 96 DPI)
Fig. 14 The maximum contact stress for plastic worm meshed with steel helical gear

97x56mm (300 x 300 DPI)
Fig. 15 The maximum contact stress for steel worm meshed with steel helical gear

102x57mm (300 x 300 DPI)
Fig. 16 Samples of the steel worm, plastic worm and steel helical gear

165x76mm (96 x 96 DPI)
Fig. 17 Micro transmission platform test site

180x131mm (96 x 96 DPI)
Fig. 18a

Transmission efficiency (°C)

Rotational speed (r/min)

- Plastic worm
- Steel worm