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Shockwave swing oscillating gear transmission with isometric polygonal profile

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ABSTRACT

Transmission principle is the base of design theory research. The work proposed a new type of swing oscillating gear transmission—swing oscillating gear transmission of shockwave isometric polygonal profile. This transmission was convenient for optimization with simple design method, which could implement transmission of arbitrary tooth difference. The basic structure and transmission principle of this oscillating gear transmission were introduced with analysis on the main character. The transmission was proved as the one with constant transmission ratio. Based on ways of installation and the number of oscillating gear, twelve transmission ratios can be obtained to derive theory and actual tooth profile equations of shockwave gear and inner gear. Meanwhile, design conditions were proposed to avoid tooth interference of central gear. Therefore, MATLAB program was used to imitate the tooth profile based on the design of this type of transmission.

Key words: Arbitrary tooth difference, isometric polygonal profile, tooth profile equations, curvature radius, swing oscillating gear transmission

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INTRODUCTION

Small-tooth difference swing oscillating gear transmission is an active branch of mechanical transmission arena [1,2]. Different from one-tooth difference oscillating gear transmission, wave exciter of two-tooth difference oscillating gear transmission is axisymmetric and self-balanced to adopt single-row teeth. Thus, Two-tooth difference oscillating gear transmission has more promising future than one-tooth difference oscillating gear transmission because of its axisymmetric shockwave generator, small axial dimension and compact structure [3]. Oscillating gear transmission with small tooth number difference has various structures, such as cycloidal cam oscillating gear transmission [4], space cam oscillating gear planetary transmission [5], cam shockwave swing oscillating gear transmission [6,7,8], two-tooth-difference putt transmission [9], two-tooth-difference cam shockwave swing oscillating gear transmission [10-14].

Swing oscillating gear transmission is an important type of oscillating gear transmission. A distinct feature of the transmission is that swing-oscillating teeth revolve around evenly distributed dowel pins in oscillating-tooth carrier. Slip of tooth inside tooth gullet and sliding wear can be effectively reduced, thus improving the transmission efficiency of devices [3,11,12]. Oscillating gear transmission with two-tooth difference is an advanced version of that with one-tooth difference. However, transmission ratio selection range of multi-tooth difference transmission is much broader with more research value. The cosine curve of tooth profile was studied in Reference [15]. Equidistant line of cosine curve is the profile of both shockwave gear and solid gear, for pure swing oscillating gear transmission of arbitrary tooth difference in Reference [16]. Isokinetic conjugate transmission of arbitrary tooth difference could be operated in this type of transmission. This design is simple for optimization. Rolling bearing components were used in oscillating gear to achieve pure rolling contact transmission. Reference [17] is about the matching curve of dual-cosine tooth transmission. Reference [18] focused on the three-shockwave roller tooth transmission, which was actually a special case of cosine shockwave. The work studied cosine shockwave swing oscillating gear transmission, briefly introducing the principle and structure of shockwave swing oscillating gear transmission with isometric polygonal profile. Tooth equation of engagement member was derived based on speed conversion and envelope principle, providing a foundation for such transmission devices. Moving parts of the new transmission, such as input shafts, are self-balanced, achieving transmission of arbitrary tooth difference. This design is specifically suitable for mechanic transmission device for heavy load due to high removability and great transmission torque.

PRINCIPLE OF SHOCKWAVE SWING OSCILLATING GEAR TRANSMISSION WITH ISOMETRIC POLYGONAL PROFILE

Shockwave gear, inner gear and oscillating tooth group share a rotation center. Each one of them can be fixed while other two are acting as input link and output link separately to obtain different speed variation. The principle of swing oscillating gear cosine with cam shockwave can be explained as follows: shockwave cam rotates at a uniform speed and generates radial forces under the effect of input drive torque; it promotes oscillating gear to swing around stud centers; meanwhile, the forces, generated by engagement of swing gear and working tooth profile of inside gear, drives tooth rack to rotate with a given ratio; the transmission from motion to driving force is realized after the torque is transmitted to output shaft of gear rack. Restricted by shockwave gear, inner gear and live gullet, the oscillating gear will have rotary motion at constant speed ratio. Rotary motion of oscillating gear is periodically driven by shockwave wave gear. Shock gear oscillating gear and inner gear maintain a continuous relationship with each other at a fixed transmission ratio.

Fig.1 shows the structure of shockwave swing oscillating gear transmission with isometric polygonal profile. It mainly consists of shockwave gear H, oscillating gear G (composed of oscillating gear and tooth rack) and inner gear K. The transmission efficiency of this device can be significantly enhanced by effectively reducing the slippage of tooth inside oscillating-tooth carrier and sliding gear.

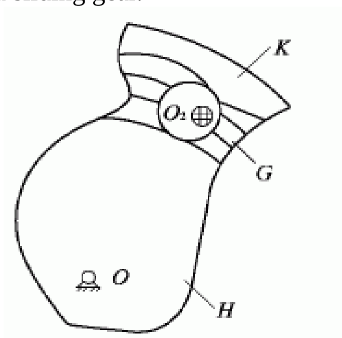

Fig. 1: Principle of shockwave swing oscillating gear transmission with cosine curve

Shock cam is connected with input shaft through flat keys, with offset curves of its profile cosine curve. Oscillating gear rack consists of output tray and floating tray, which are fixed together by circumferentially-arranged oscillating-tooth pins. Profiles of central inner gear are swinging movement locus of tooth around shockwave rack pin driven by shockwave cam. Meanwhile, shockwave cam and oscillating gear rack draw envelope line of uniform circular motion at a given ratio. Tooth number of central inner tooth and oscillating gear is defined as arbitrary tooth difference. As shown in Fig.2, three-section composite structure is used in swing oscillating gear, consisting of tooth matrix 3 and two needle bearings 1 and pinhole 2. The middle section of oscillating tooth is a part of tooth matrix. Its excircle surface is meshed with shockwave cam. The two needle bearings, as truncated needle bearings without inner ring, are installed on both sides of the middle segment of tooth matrix; its outer surface is meshed with central inner gear. Tooth matrix is eccentrically mounted on tooth rack pin roll. Moreover, a shallow annular groove is placed in the center of inner gear to avoid contact friction of excircle surface of the middle tooth with inner gear. Meanwhile, the width of shockwave cam should be smaller than the middle section of tooth, thus avoiding intervention of shockwave cam and outer surface of the two needle bearings.

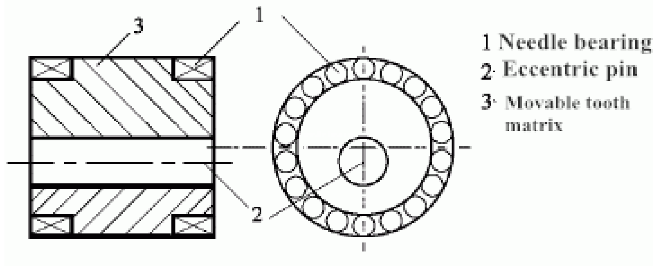

Fig. 2: Structure sketch of swing oscillating gear

There are several major features of this transmission device: (1) Profile of shockwave cam is axisymmetric cosine curve, which can achieve shockwave motion through rotation around its geometric center of the tooth. Meanwhile, it ensure self-balancing force of shockwave cam and central inner gear. Whether using single or multiple-piece shockwave cam, excitation force and overturning moment will not exist in transmission. Vibration excitation can be avoided based on the principle. (2) Double needle bearing was placed on matrix, appropriately reducing the friction among oscillating gear, central inner gear and shockwave cam gear; swing oscillating gear is connected with shockwave cam by line, enhancing entrainment rate between tooth and cam. The transmission efficiency is effectively improved with simple structure of tooth rack, making the structure more compact. (3) The cam doesn't need to be eccentrically mounted on shaft due to the application of symmetrical structure. Therefore, the angular position of each chain can keep consistent when using multiple-row transmission chain composed of shockwave cam, oscillating gear, tooth rack and inner gear. It is convenient for assembling transmission devices. Moreover, the wave number of the cam is different from traditional oscillating gear of one tooth difference or two-phase cam shockwave of two tooth difference. It can be any teeth difference with broader selection range of transmission ratio. Therefore, it has more research value.

CALCULATION OF TRANSMISSION RATIO

If the shockwave gear is fixed, transmission mechanism of shockwave swing oscillating gear with isometric polygonal profile can be regarded as conversion mechanism without planetary gear. Then the whole system of shockwave gear was given with an additional angular velocity ω_H which is equal to that of shockwave gear but with opposite direction. According to transmission ratio formula of fixed axis gear, it can be calculated as:

$$i_{GK}^{H} = \frac{\omega_G^H}{\omega_K^H} = \frac{\omega_G - \omega_H}{\omega_K - \omega_H} = \frac{Z_K}{Z_G} \tag{1}$$

From Formula (1), we know

$$\omega_G = \omega_K i_{GK}^{H} + \omega_H (1 - i_{GK}^{H}) \tag{2}$$

where H is shockwave wave gear; G oscillating gear; K inner gear; i transmission ratio. The superscript letters indicate corresponding fixing member, while subscript letters indicate the state of corresponding members of left letters with respect to that of the right letters. For example, i_{GK}^{H} indicates the ratio of relative angular velocity of G and H to that of H and G. Superscript letters of ω indicate corresponding fixing member, while subscript letters indicate corresponding elements; Z_H is the wave number of shockwave gear; Z_G tooth number of oscillating gear, and Z_K the wave number of inner gear. Tab.1 showed transmissions ratios of different installation forms.

TAB.1 Transmission ratios by changing ways of installation

Transmission scheme	Transmission ratio	Steeling direction	Tooth number of oscillating gear	Deceleration or acceleration
Fixed inner gear $\omega_K = 0$	$i_{HG}^{K} = \frac{Z_G}{Z_G - Z_K}$	$Z_G > Z_K$ same direction	$Z_G = Z_K + Z_H$	Deceleration
		$Z_G < Z_K$ opposite direction	$Z_G = Z_K - Z_H$	Deceleration
	$i_{GH}^{K} = \frac{Z_G - Z_K}{Z_G}$	$Z_G > Z_K$ same direction	$Z_G = Z_K + Z_H$	Acceleration
		$Z_G < Z_K$ opposite direction	$Z_G = Z_K - Z_H$	Acceleration
Fixed oscillating gear $\omega_G = 0$	$i_{HK}^{G} = \frac{Z_K}{Z_K - Z_G}$	$Z_G > Z_K$ opposite direction	$Z_G = Z_K + Z_H$	Deceleration
		$Z_G < Z_K$ same direction	$Z_G = Z_K - Z_H$	Deceleration
	$i_{KH}^{G} = \frac{Z_K - Z_G}{Z_K}$	$Z_G > Z_K$ opposite direction	$Z_G = Z_K + Z_H$	Acceleration
		$Z_G < Z_K$ same direction	$Z_G = Z_K - Z_H$	Acceleration
Fixed Shockwave gear $\omega_H = 0$	$i_{GK}^{H} = \frac{Z_K}{Z_G}$	$Z_G > Z_K$ same direction	$Z_G = Z_K + Z_H$	Acceleration
		$Z_G < Z_K$	$Z_G = Z_K - Z_H$	Deceleration

$i_{KG}^{H}=\frac{Z_G}{Z_K}$	same direction $Z_G > Z_K$ same direction	$Z_G = Z_K + Z_H$	Deceleration
	$Z_G < Z_K$ same direction	$Z_G = Z_K - Z_H$	Acceleration

PROFILE EQUATION OF SHOCKWAVE GEAR AND INNER GEAR

The profile of shockwave cam in swing oscillating gear transmission with isometric polygonal profile is an equidistant cosine shockwave curve. Profile of swing oscillating gear adopts standard cylindrical curve. Profile of central inner gear is swinging movement around shockwave rack pin driven by shockwave cam. Meanwhile, shockwave cam and oscillating gear rack draw envelope line of uniform circular motion at a given ratio.

TOOTH PROFILE EQUATION OF SHOCKWAVE GEAR

Isometric polygonal profile equation of shockwave gear is

$$\begin{cases} x_1 = (R + e\cos(N\theta))\cos\theta + Ne\sin(N\theta)\sin\theta \\ y_1 = (R + e\cos(N\theta))\sin\theta - Ne\sin(N\theta)\cos\theta \end{cases} \quad (3)$$

where R is nominal radius of isometric polygonal profile, e difference of nominal diameter and inscribed circle diameter; $\theta \in [0, 2\pi]$, N is shockwave number of shockwave gear, $N = Z_H$.

Actual tooth profile of shockwave gear is the inner equidistant curve of theoretical profile with offset r_1. Therefore, actual tooth profile equation can be deduced from mechanical principles [21]

$$\begin{cases} x_2 = x_1 - \dfrac{r_1 \dfrac{dy_1}{d\theta}}{\sqrt{(\dfrac{dx_1}{d\theta})^2 + (\dfrac{dy_1}{d\theta})^2}} \\ y_2 = y_1 + \dfrac{r_1 \dfrac{dx_1}{d\theta}}{\sqrt{(\dfrac{dx_1}{d\theta})^2 + (\dfrac{dy_1}{d\theta})^2}} \end{cases} \quad (4)$$

where

$$\frac{dx_1}{d\theta} = N^2 e\cos(N\theta)\sin\theta - \sin\theta(R + e\cos(N\theta)),$$

$$\frac{dy_1}{d\theta} = -N^2 e\cos(N\theta)\cos\theta + \cos\theta(R + e\cos(N\theta))$$

TOOTH PROFILE EQUATION OF INNER GEAR

In Fig.3, XOY is set as the fixed coordinate system connected with annular gear while the origin coordinate O is the geometric center. $X'OY'$ and X_1OY_1 are defined as conjoined coordinate systems for shockwave cam and oscillating-tooth carrier separately. They share the same origin coordinate with the fixed coordinate system XOY. In the initial configuration of transmission, coordinate systems XOY, $X'OY'$, X_1OY_1 and X_2OY_2 coincide with each other. Pinhole center O_1 is on OX_1, while the gear center O_2 is on OX_2. According to transmission principle of cam shockwave swing oscillating gear, the motion curve of O_2 around $X'OY'$ in transmission is the theoretical profile of shockwave cam. The inner equidistant curve, with gear radius r_1 as the offset, is called as working profile of shockwave cam. The motion curve of O_2 around XOY is the theoretical tooth profile of central inner gear (motion curve of tooth center). The working tooth profile of central inner gear is the equidistant curve of theoretical tooth profile with offset radius r_2.

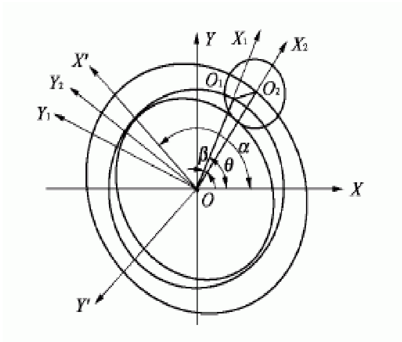

Fig. 3: Generation principle of tooth profile of inner gear

Fig.3 shows the pinhole center of oscillating gear rack O_1 is on the standard circle with the center of coordinate origin O and radius R_0; the offset of gear is d (distance between O_1 and O_2); the angle of lines connecting two centers with O (OO_1 and OO_2) is β. Preconditions are set as following: making coordinate system $X^{'}OY^{'}$ turn around fixed coordinate system XOY at any time with angle α ($\angle XOX^{'}$); turning X_1OY_1 around angle θ ($\angle XOX_1$); driving gear center O_2 to turn around angle $\alpha-\theta$ ($\angle XOX_2$) relative to coordinate system XOY. Moreover, α and θ meet the given condition of input-output ratios: $i_{HG}^{K}=\dfrac{\alpha}{\theta}$ (i_{HG}^{K} is the transmission ratio). In Figure 4, the angle of OO_2 with $OX^{'}$ is $(\alpha+\beta-\theta)$, while it is $(\theta-\beta)$ of OO_2 and OX. Thus, the length of OO_2 can be calculated according to Formula (3) as follows:

$$\left|OO_2\right| = \rho = \sqrt{x_{30}^2 + y_{30}^2} \tag{5}$$

where

$$\begin{cases} x_{30}=(R+e\cos(N(i-1)\theta+N\beta))\cos((i-1)\theta+\beta) \\ \qquad +Ne\sin(N(i-1)\theta+N\beta)\sin((i-1)\theta+\beta) \\ y_{30}=(R+e\cos(N(i-1)\theta+N\beta))\sin((i-1)\theta+\beta) \\ \qquad -Ne\sin(N(i-1)\theta+N\beta)\cos((i-1)\theta+\beta) \end{cases}$$

$$\begin{cases} x_3=\rho\cos(\theta-\beta) \\ y_3=\rho\sin(\theta-\beta) \end{cases} \tag{6}$$

In the triangle OO_1O_2, $\left|OO_2\right|=\rho,$ $\left|O_1O_2\right|=d,$ and $\left|OO_1\right|=R$. Then, according to cosine law, we can know that,

$$d^2=R^2+\rho^2-2R\rho\cos\beta \tag{7}$$

If θ is known, β and ρ can be calculated from Formulas (5) and (7) using Function fsolve.m of Matlab platform. Meanwhile, $\dfrac{dx_3}{d\theta}$ and $\dfrac{dy_3}{d\theta}$ in Formula (6) can be deducted through numerical differentiation.

Actual tooth profile of inner gear is the equidistant curve of theoretical tooth profile with offset r_2. Therefore, based on mechanical principle of inner gear, the actual tooth profile equation is as follows [21]

$$\begin{cases} x_4 = x_3 + \dfrac{r_2 \dfrac{dy_3}{d\theta}}{\sqrt{(\dfrac{dx_3}{d\theta})^2 + (\dfrac{dy_3}{d\theta})^2}} \\ y_4 = y_3 - \dfrac{r_2 \dfrac{dx_3}{d\theta}}{\sqrt{(\dfrac{dx_3}{d\theta})^2 + (\dfrac{dy_3}{d\theta})^2}} \end{cases} \tag{8}$$

CALCULATION INSTANCE

Parameters of isometric polygonal profile are given as follows: cosine shockwave number $Z_H = 3$; nominal radius of isometric polygonal profile R=120mm; the difference of nominal diameter and inscribed circle diameter e=5mm; inner offset radius of shockwave wave gear r_1=20mm; outer offset radius of inner gear r_2=20mm, tooth number of inner gear Z_K = 8; tooth number of oscillating gear Z_G=11; tooth eccentricity d=20mm; the inner gear is fixed; input $i = i_{HG}^K = 3$. MATLAB programming is used to calculate theoretical and practical tooth profile of shockwave gear. In Fig.4, the inner curve is actual tooth profile, while the outer curve is theoretical tooth profile. Fig.5 shows the theoretical and practical tooth profile of inner gear; the outer curve is actual tooth profile, while the inner curve is theoretical tooth profile. The curvature radius of convex segment near the top of theoretical tooth profile of gear was 28.0333mm, while that of actual tooth profile is 8.0333mm.

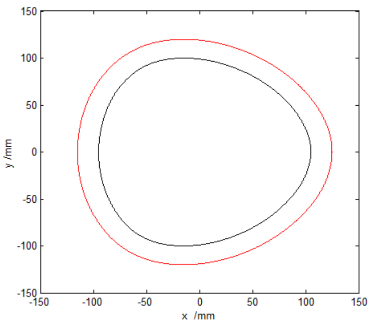

Fig. 4: Tooth profile curve of shockwave cam

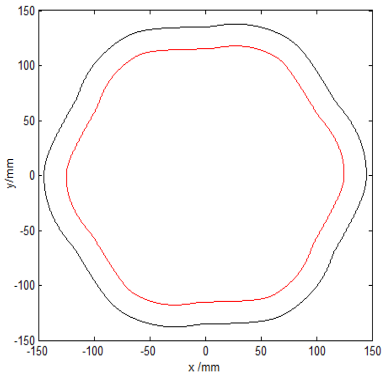

Fig. 5: Tooth profile curve of inner gear

Matlab programme is presented as follows:

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
global R N e seta1 R0 x30 y30;
global i d;
d=5;R0=120;
R=120;N=3;e=5;i=3;r01=20;r02=20;
step=3600;
zh=N;
seta_ok=linspace(0,2*pi,step);
```

```
x1=(R+e*cos(N*seta_ok)).*cos(seta_ok)+N*e.*sin(N*seta_ok).*sin(seta_ok);%
y1=(R+e*cos(N*seta_ok)).*sin(seta_ok)-  N*e*sin(N*seta_ok).*cos(seta_ok);
dx1=N^2*e*cos(N*seta_ok).*sin(seta_ok) - sin(seta_ok).*(R + e*cos(N*seta_ok));
dy1=cos(seta_ok).*(R + e*cos(N*seta_ok)) - N^2*e*cos(N*seta_ok).*cos(seta_ok);
x11=x1-r01*dy1./(dx1.^2+dy1.^2).^(1/2);
y11=y1+r01*dx1./(dx1.^2+dy1.^2).^(1/2);
r10=max(x11.^2+y11.^2).^(0.5)+0.5; plot(x1,y1,'r');
hold on;plot(x11,y11,'k');
if i>0,zg=i*zh;zk=zg-zh;else zg=-i*zh;zk=zg+zh;end
pho=[];beta=[];seta=[];
seta11=5*pi/180;RR=R0+d;
dx2=[];dy2=[];dx21=[];dy21=[];
%%%%%%%%%%%%%%%%%%%%
for j=1:step+1
   seta1=(j-1)*(2*pi)/step;
   sseta1=seta1;
   alfa1=i*seta11;
    xx=fsolve('my_eq_ShapedContour',[RR seta11]);
   pho=[pho xx(1)];
   beta=[beta xx(2)];
   seta=[seta seta1];
   seta11=xx(2);
   RR=xx(1);
   seta1=(j-1)*(2*pi)/step+dseta1;
   yy=fsolve('my_eq_ShapedContour',[RR seta11]);
dx20=((yy(1).*cos(seta1-yy(2)))-(xx(1).*cos(sseta1-xx(2))))/dseta1;
dy20=((yy(1).*sin(seta1-yy(2)))-(xx(1).*sin(sseta1-xx(2))))/dseta1;
    dx2=[dx2 dx20]; dy2=[dy2 dy20];
    seta1=(j-1)*(2*pi)/step-dseta1;
yy2=fsolve('my_eq_ShapedContour',[RR seta11]);
dx220=-((yy2(1).*cos(seta1-yy2(2)))-(xx(1).*cos(sseta1-xx(2))))/dseta1;
dy220=-((yy2(1).*sin(seta1-yy2(2)))-(xx(1).*sin(sseta1-xx(2))))/dseta1;
    dx21=[dx21 dx220];dy21=[dy21 dy220];
end
x2=pho.*cos(seta-beta);
y2=pho.*sin(seta-beta);
x22=x2+r02*dy2./(dx2.^2+dy2.^2).^0.5;
y22=y2-r02*dx2./(dx2.^2+dy2.^2).^0.5;
r20=min(x22.^2+y22.^2).^(0.5)-0.5;%
 plot(x2,y2,'r');
 hold on;plot(x22,y22,'k');
 hold on;plot(r20.*cos(seta),r20.*sin(seta),'r',r10.*cos(seta),r10.*sin(seta),'r');
 hold off;
dxseta21=(dx2-dx21)/(dseta1);
dyseta21=(dy2-dy21)/(dseta1);
rho=(dx2.^2+ dy2.^2).^(3/2)./(dx2.*dyseta21-dxseta21.*dy2);
plot(seta,rho,'r');
rhomin=min(abs(rho));
disp(['%%%%%%%%%',date,'%%%%%%%%%'])

function f=my_eq_ShapedContour(x)
pho=x(1);beta=x(2);
global R N e seta1 R0 x30 y30;
global i d;
seta=seta1;
temp=i*seta-seta+beta;
f=[pho-((R+e*cos(N*temp))^2+N*e.*sin(N*temp)^2).^0.5;-d.^2+R0^2+pho.^2-…
2*R0.*pho*cos(beta)]
%%%%%%%%%%%%%%%%%%%%%%%%%
```

CONCLUSION

With isometric polygonal profile of arbitrary tooth difference, the swing oscillating gear transmission adopts standard cylindrical curve as the profile of equidistant line and swing oscillating gear for cosine curve of shockwave gear. Profile of central inner gear is swinging movement locus of tooth around shockwave rack pin driven by shockwave cam. Meanwhile, shockwave cam and oscillating gear rack draw envelope line of uniform circular motion at a given ratio. This type of transmission enables isokinetic conjugation transmission with arbitrary tooth differences. Its design methods are simple for optimization. Programming of Matlab software is used to calculate tooth profile equation. Solid model of cosine shockwave wave swing oscillating gear can be established using Solidworks parametric modeling. The application of motion simulation of Solidworks enables transmission to perform well without interference, indicating that the above deduction is theoretically correct. Moreover, the deducing formulas are applicable to any shockwave swing oscillating gear transmissions with isometric polygonal profile. The transmission has broad prospects of promotion and application.

Acknowledgments
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