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Investigation of a Design Modification for Double Helical Gears Reducing Vibration and Noise

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Abstract: To reduce vibration and noise and increase transmission efficiency, a three segment method for modifying the pinion profile was proposed. Cutter surface equations were deduced by changing the shape of the cutter-edge, substituting three segment parabolas for the line. The influence of longitudinal tooth modifications on tooth surface load distributions was discussed. Transmission error minimization and uniformity of tooth surface load distribution were chosen as optimization goals and the modified parameters were obtained by applying the complex method. Finally, an experiment comparing the loaded transmission error, vibration, and noise both before and after modifications was carried out. The results indicate that the modified design is reliable.
Keywords: tooth profile modification; tooth longitudinal modification; double helical gears; loaded transmission error
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1 Introduction

Double helical gears have been widely used in marine ships because of their advantages such as high loading capability and steady transmission (Amendola, 2006). Noise of marine ships threatens their own safety. For example, high noise can disturb sonar to detect underwater targets, also, noise of submarines can lead to that sonar can detect them, and even judge the technology and running state of gearbox. Noise of marine ship gearbox is a major component of marine ship noise, which is mainly generated by tooth meshing each other. Therefore, decreasing vibration and noise of gears is an important research direction in marine ship gears design (Wang *et al*., 2003).

Practices have been proved that through modifying addendum, dedendum and longitude, tooth meshing performance can be effectively improved including meshing impact being decreased, noise being reduced, and loading capacity being increased because of tooth surface load uniformly distributing. In addition, for raising working efficiency, modification of pinion dedendum is instead of that of gear addendum, i.e. all modifications are put in pinion (Yuan, 2006).

At present, about the study of tooth modification, mostly we use ideal force under static or quasi-static conditions to substitute for transient meshing force, so the effect of modification is not obvious in the middle and high speed gear transmission. Gear meshing vibration and noise are generated in the dynamic state of gear transmission. Therefore, combining gears modification and dynamics analysis of gears may offer reliable basis for the choice of modification parameters. However, the complexity of dynamic modeling and the gear system of nonlinear factors, especially, the dynamics study about double helical gears has not yet been developed, all of which increased the difficulty of dynamic modification.

Loaded tooth contact analysis (LTCA) is an important numerical method simulating the course of tooth meshing under loaded. Combining loaded tooth contact analysis and tooth modification, and studying loaded tooth meshing under modification can explore the best tooth surface modification design (Simon, 1989; Weck, 1990).

Based on the above analysis, in terms of the shape of pinion cutter-edge, we use three segment parabolas to substitute for line to realize three segment modifications to pinion profile. Pinion tooth longitudinal modification is realized by adopting a segment parabola along the direction of pitch circle helix angle, and the effect of tooth longitudinal modification on tooth surface distribution is discussed. The optimization goal is to minimize transmission error and uniformly distribute tooth surface load, and the modified parameters can be obtained by applying the complex method. Finally, an experiment which verifies the correctness of the modification method was done by comparison with the loaded transmission error, vibration and noise both before and after modification.

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2 Tooth modification

2.1 Three segment modifications to pinion profile

Common method of tooth profile modification is to change the shape of cutter-edge (Fang, 1997). In order to reduce vibration and noise and raise working efficiency, the method of three segment modifications to pinion profile was proposed. By changing the shape of cutter-edge, three-segment parabola was used to substitute for the linear profile. Normal profile of rack-cutter and tooth surface coordinate of rack-cutter are shown in Fig.1 and Fig.2, respectively.

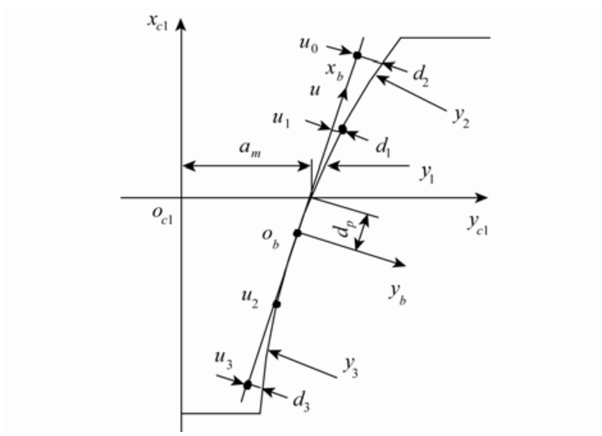


Fig.1 Normal profile of rack-cutter

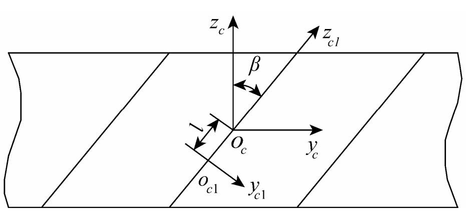


Fig.2 Tooth surface coordinate of rack-cutter

Coordinate systems $o_bx_by_b$ and $o_{c1}x_{c1}y_{c1}$ are rigidly connected to the rack-cutter profile and rack-cutter surface ($o_{c1}y_{c1}$ is machining pitch line; $o_{c1}x_{c1}$ is tooth thickness symmetric midline), respectively. y_1, y_2 and y_3 are the parabolic equation in the coordinate systems $o_bx_by_b$, respectively. They are represented as

$$\begin{cases} y_1 = a_1u^2 \\ y_2 = a_2u^2 + b_2u + c_2 \\ y_3 = a_3u^2 + b_3u + c_3 \end{cases} \tag{1}$$

By coordinate transformation, the equation of rack-cutter profile is represented in the coordinate systems $o_cx_cy_c$ as

$$r_c = \begin{cases} -y_i \sin\alpha + (u - dp)\cos\alpha \\ y_i \cos\alpha \cos\beta + l\sin\beta + [(u - dp)\sin\alpha + a_m]\cos\beta \\ -y_i \cos\alpha \sin\beta + l\cos\beta - [(u - dp)\sin\alpha + a_m]\sin\beta \end{cases} \tag{2}$$

where, i=1, 2, 3; u and l are the rack-cutter surface parameters; u is the distance between the point of rack-cutter profile and parabola pole; l is the distance between the point of rack-cutter profile and o_c; d_p is displace of parabola pole; α and β are normal pressure angle and helical angle, respectively; a_m is half normal tooth thickness of rack-cutter pitch line.

u_1 and u_2 are the position of three segment parabolas in normal profile of rack-cutter. If $u_2 \le u \le u_1$, the equation of cutter profile is y_1; If $u > u_1$, the equation of cutter profile is y_2; If $u < u_2$, the equation of cutter profile is y_3.

For the left and right pinion surface manufacture, use different modifications to rack-cutter.

2.2 Tooth longitudinal modification

The tooth longitudinal modification has important influence on tooth surface load distribution. Under the ideal condition, i.e. when there is no installation error and no deformation, tooth surface load distribution is uniform (Fig.3). However, in fact, installation error and deformation are inevitable. So in order to improve the tooth surface load distribution, tooth longitude is modified along a segment parabola which is in the direction of pitch circle helix angle, and its normal modification is represented as

$$y = ax^2 + bx + c \tag{3}$$

Similarly, as to the tooth profile modification, for the left and right pinion surface, different modifications can be used. Fig.4 and Fig.5 show load distribution while existing errors of axis alignment, respectively. Before longitudinal modification, load offsets toward one end, which can easily damage tooth. After longitudinal modification, load centralizes toward the middle, which increases the load capability of tooth.

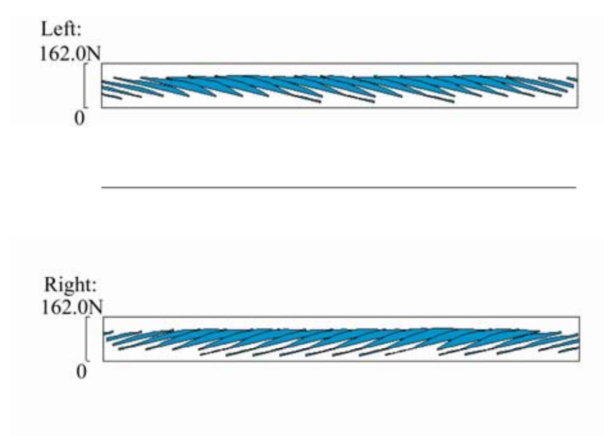
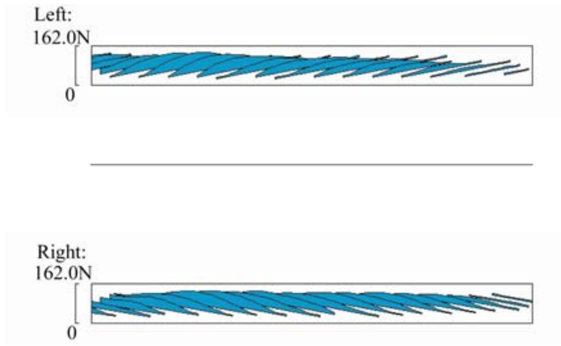
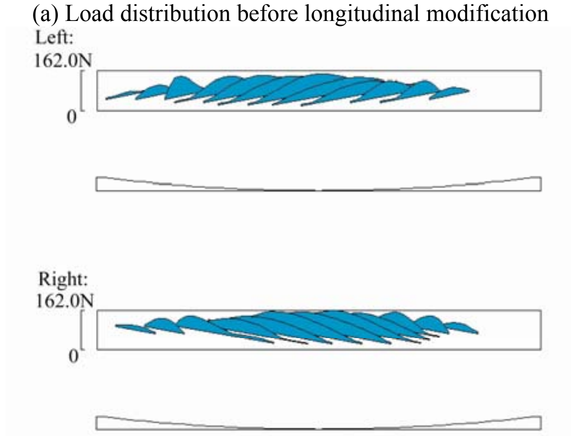


Fig.3 Load distribution of tooth surface under the ideal condition

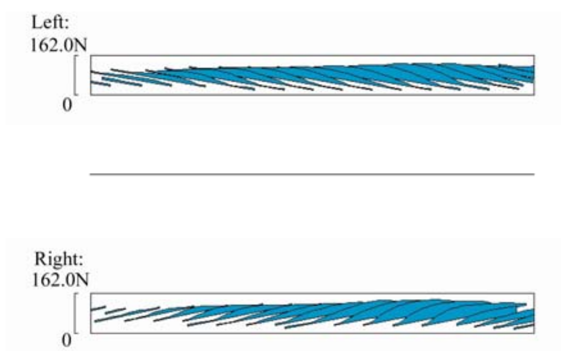


(a) Load distribution before longitudinal modification

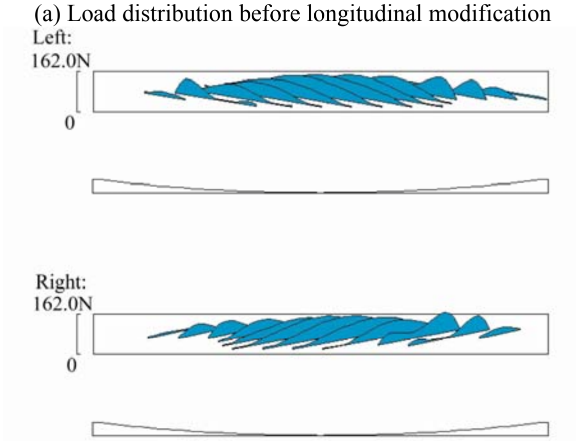


(b) Load distribution after longitudinal modification

Fig.4 Tooth surface load distribution while existing errors of axis alignment (δ=0.06')



(a) Load distribution before longitudinal modification



(b) Load distribution after longitudinal modification

Fig.5 Tooth surface load distribution while existing errors of axis alignment (δ=-0.06')

3 Modification optimization

Tooth contact analysis (TCA) and loaded tooth contact analysis (LTCA) of double helical gears had been introduced in other papers (Wang *et al*., 2009b). In addition, related references (Kin, 1994; Li, 2002; Litvin *et al*., 1999; Litvin *et al*., 2003; Litvin, 2007) also referred to them. Thus, by the calculation of TCA and LTCA of double helical gears, the loading transmission error and tooth surface load can be obtained.

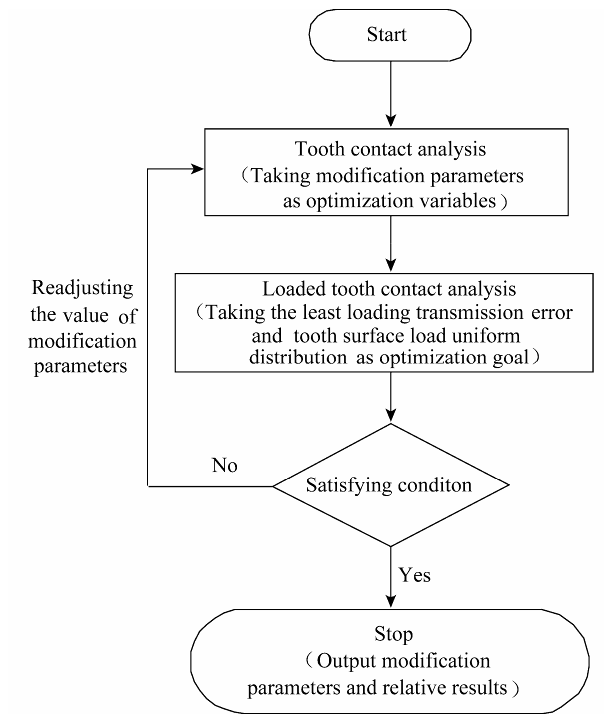


Fig.6 Flow diagram for modification optimization

Taking the least loading transmission error and tooth surface load uniform distribution as the optimization goal (Fang *et al*., 1992a; Fang *et al*., 1992b), and taking the maximum modification of rack-cutter normal profile of d_1, d_2, d_3, the position of three segment parabolas of u_1, u_2, and the tooth longitudinal parabola modification coefficient of a, b, c as optimization variables (here only one end's tooth modification parameters are given, and the others are similar), the modification parameters can be obtained by applying the complex method (Xu and Qian, 2005). Flow diagram for modification optimization is shown in Fig.6.

4 Experimental results and discussion

The comparison experiment of loaded transmission error, vibration and noise both before and after modification was carried out (with gear acted torque of 1 000 N·m and 2000N·m, respectively). Double helical gears parameters are given in Table 1. Because the assembly relation between pinion and gear are very strict and for the sake of easily machining, errors of axis alignment are not taken into account, only tooth profile modification is used (where, left and right tooth modifications are the same). Modification of pinion after optimization (Wang *et al.*, 2009a) is shown in Fig.7.

A set of equipment for detecting transmission error is designed which can detect less than 1" transmission error and experimental equipment is shown in Fig.8. By use of filter technology, the gear frequency error is separated from shaft error, and the frequency error of experiment is compared with that of calculation.

Fig.9 and Fig.10 show the loaded transmission error under 1000 N·m and 2 000 N·m, respectively. Table 2 shows the transmission error amplitude of experiment and calculation.

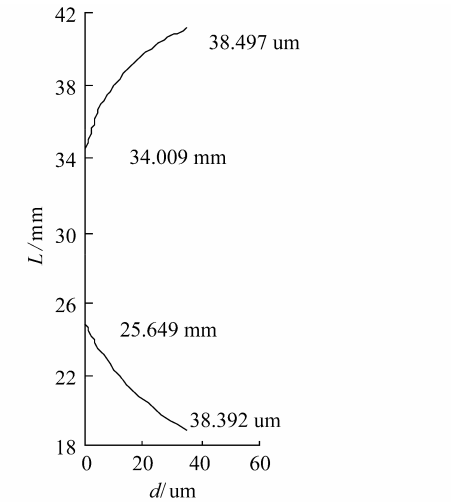
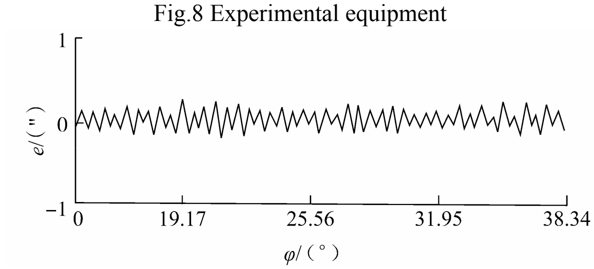


Fig.7 Modification of pinion

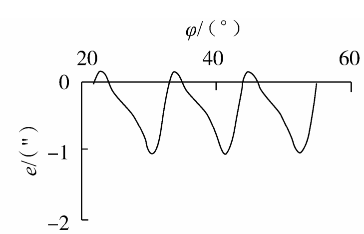
Table 1 Parameters of double helical gears

Tooth number		Normal modulus /mm	Normal pressure angle /(°)	Helical angle /(°)	Face width /mm	Helical direction		Run-out groove width /mm
Pinion	Gear					Pinion	Gear	
31	102	4.5	20	28.34	90×2	right left	left right	70

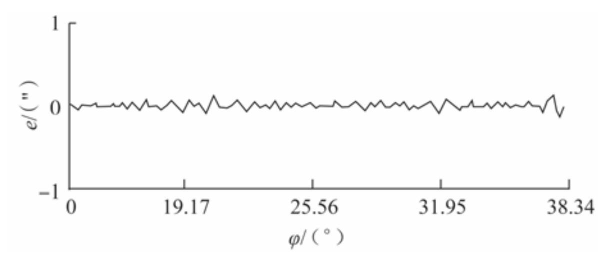

Fig.8 Experimental equipment



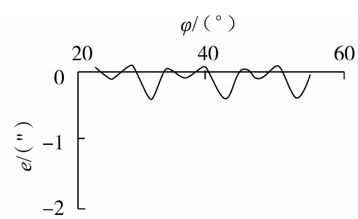
(a)Transmission error of experiment (before modification)



(b)Transmission error of calculation (before modification)

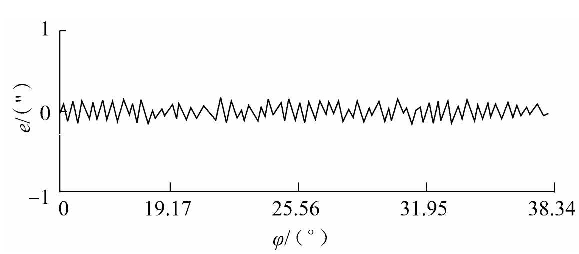


(c)Transmission error of experiment (after modification)

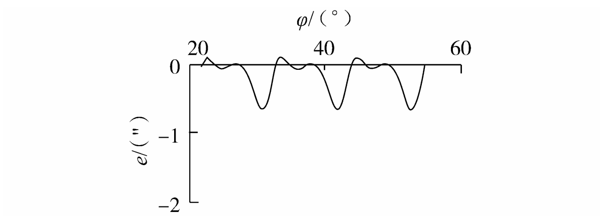


(d) Transmission error of calculation (after modification)

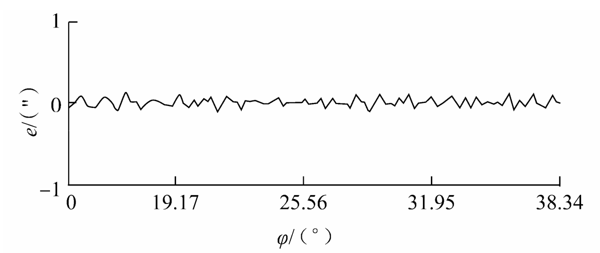
Fig.9 Loaded transmission error under 1 000 N·m



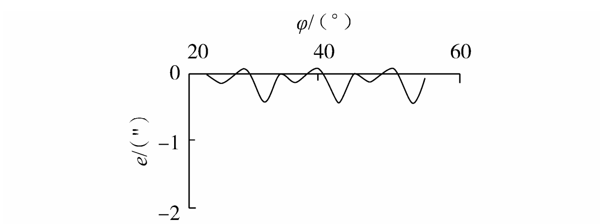
(a)Transmission error of experiment (before modification)



(b)Transmission error of calculation (before modification)



(c)Transmission error of experiment (after modification)



(d) Transmission error of calculation (after modification)

Fig.10 Loaded transmission error under 2 000 N·m

Table 2 Transmission error amplitude of experiment and calculation both before and after modification

Torque /(N·m)	Amplitude of Experiment/ (")		Amplitude of calculation/ (")	
	before	after	before	after
1 000	0.977 987	0.413 205	0.957 3	0.400 6
2 000	0.655 402	0.451 669	0.640 9	0.439

From Figs.9~10 and Table 2, it can be found that 1) Transmission error amplitude after modification is obviously reduced compared with the modification before; 2) Transmission error of experiment is bigger than that of calculation, which is because transmission error of experiment includes tooth deformation and tooth surface minuteness error, while transmission error of calculation only includes tooth deformation. In addition, meshing vibration and noise of different points disposed both before and after modification are measured (Fig.11). The average vibration acceleration and noise are given in Table 3 and Table 4, respectively. The results show that the meshing vibration and noise after modification have decreased 18% and 2.7 dB respectively in average comparing with that without modification, which indicate the modification design method proposed can lead to a significant reduction of noise and vibration.

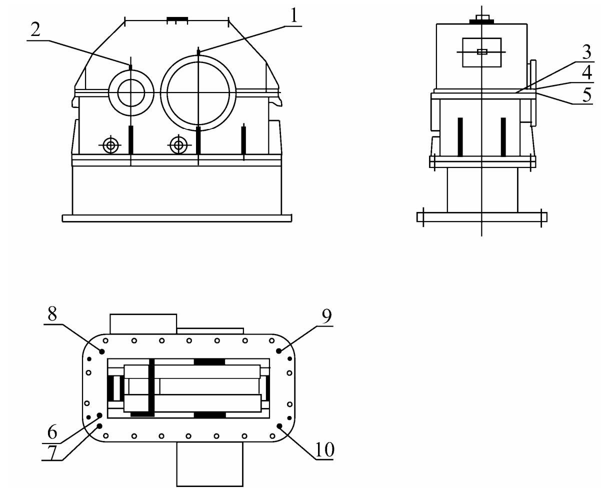


Fig.11 Measuring points placement of vibration

Table 3 The average vibration acceleration both before and after modification

Measuring points	Vibration acceleration/($m \cdot s^{-2}$)		Decrease Proportion/%
	before	after	
1	3.33	3.01	10
2	7.23	5.70	21
3	3.62	2.86	21
4	4.64	3.20	31
5	2.05	2.38	-14
6	4.98	2.96	41

Table 4 The average noise both before and after modification

Measuring points	Noise/dB		Decrease proportion
	before	after	
7	136.226 7	131.796 7	4.43
8	133.676 7	131	2.68
9	130.55	128.61	1.94
10	129.82	128.073 3	1.75

5 Conclusions

1) For reducing vibration and noise and raising working efficiency, all modifications are put in pinions. Cutter surface equation was deduced by changing the shape of cutter-edge, substituting three segment parabolas for lines.

2) The tooth longitudinal modification has important influence on tooth surface load distribution, which can improve the offset load caused by installation error and increase the load capability of tooth.

3) The optimization goal is to minimize transmission error minimizing and uniformly distribute tooth surface load, and the modified parameters can be obtained by applying the complex method.

A set of equipment for detecting transmission error is designed. The comparison experiment of loaded transmission error, vibration and noise both before and after gear modification is accomplished. The results indicate that the modified tooth obtains ideal effect on reducing transmission error, vibration and noise compared with the tooth without modification.

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