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Design of helical gear transmission systems with high power density

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ABSTRACT

High power density has become an important development direction of gear transmission system. The current research is solely focused on the volume or the efficiency, but the combination of the two is extremely lacking. In the paper, the design method of helical gear transmission system with high power density is put forward. Firstly, the transmission efficiency model of helical gear is proposed by the calculation of sliding friction power losses of meshing point and the further integral along the meshing line. Secondly, volume formulas of different structure of helical gears are derived, and the volume model of helical gear transmission system is proposed. Finally, the optimization to minimize the volume and power loss (i.e., the highest transmission efficiency) of helical gear transmission system is set as the target. The linear weighted combination method is used to construct the target function; the design parameters of helical gear are set as the optimization variables. To meet the helical gear design and requirements of transmission are set as the constraint conditions; the design method of high power density of helical gears transmission system is proposed. A single stage helical gears transmission system is used to demonstrate the proposed method; the optimal design is completed. It can be found that the power loss of the optimized gear system is reduced by 18.07%, and the volume is reduced by 11.39%.

Keywords: Helical gear; high power density; optimization; transmission efficiency; volume.

INTRODUCTION

The definition of power density is the maximum output power of system divided by the weight or volume (or area) of the entire system. The power density is an important performance index of the transmission system. For the gear transmission system, high power density has become an important development direction. High power density means that the device has the small volume and high transmission efficiency. However, the current research is solely focused on volume or efficiency, but the combination of the two is rather lacking. The following is a more detailed introduction.

Many researches have been done about volume optimization. An optimal method for gear weight by using a highly efficient genetic algorithm was proposed (Yokota *et al.,* 1998). The optimal design of a heavy duty helical gear reducer for minimum volume by using the method of Particle Swarm Optimization was completed (Tamboli *et al.,* 2014). Aiming at the problems existing in the method proposed by Yokota, the particle swarm optimization/simulated annealing algorithms was used to optimize the weight of the gear (Savsani *et al.,* 2010). A nonlinear optimization model to minimize the volume of gear with a fixed load coefficient was proposed (Zhang *et al.,* 2013). A generalized optimization formulation for the minimum volume design of multi-stage spur gear reduction units was proposed (Thompson *et al.,* 2000). A method of helical gear volume precise modeling was proposed (Wang *et al.,* 2015).

Research on transmission efficiency of gear system is of great significance to reduce the energy consumption and improve the performance of the gear system (Wang *et al.,* 2015). A large number of related researches have been done. For example, a model for prediction of friction related mechanical efficiency losses of parallel-axis gear pairs was presented (Xu *et al.,* 2007). Aiming at the problems existing in the method proposed by Xu, a new hypoid gear mechanical efficiency model was presented (Kolivand *et al.,* 2010). A calculation method of sliding friction losses by tooth contact analysis (TCA) and loaded tooth contact analysis (LTCA) was proposed (Wang *et al.,* 2015). On this basis, a more accurate method considering the dynamic characteristics of gear was further proposed (Wang *et al.,* 2016).

Although there are some questions such as low calculation accuracy existing in the method of calculating gear transmission efficiency along the meshing line integration, it is still widely used because of its simplicity and feasibility. More importantly, we can directly get the relationship between the gear design parameters and the transmission efficiency by the method, which lays the foundation for the next optimization design. For example, the sliding friction power loss at the meshing point, integrated along the meshing line, was calculated (Cheng *et al.,* 2012). The expression of meshing efficiency of the helical gear was obtained, and the relationship between the design parameters and the meshing efficiency was revealed (Fig. 1). In Fig. 1, the horizontal ordinate represents the design parameter; the longitudinal coordinate represents the meshing efficiency.

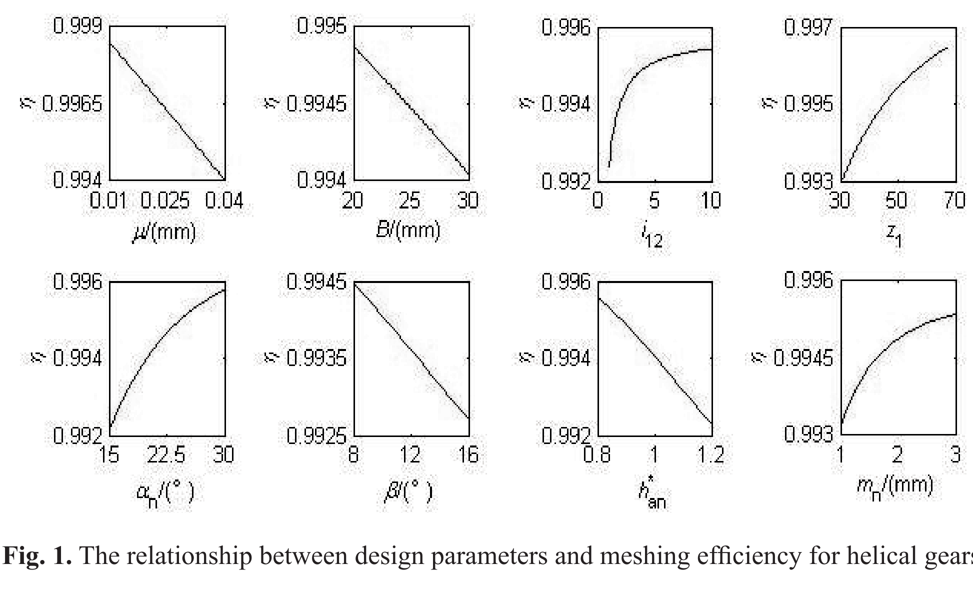


Fig. 1. The relationship between design parameters and meshing efficiency for helical gears.

At present, the research of high power density gear transmission system is rather lacking. More research is focused on the volume (miniaturization) or the transmission efficiency. Therefore, in this paper, the single stage helical gear transmission system is treated as the target, and the design method of high power density is proposed; the analysis process of this paper is shown in Fig. 2.

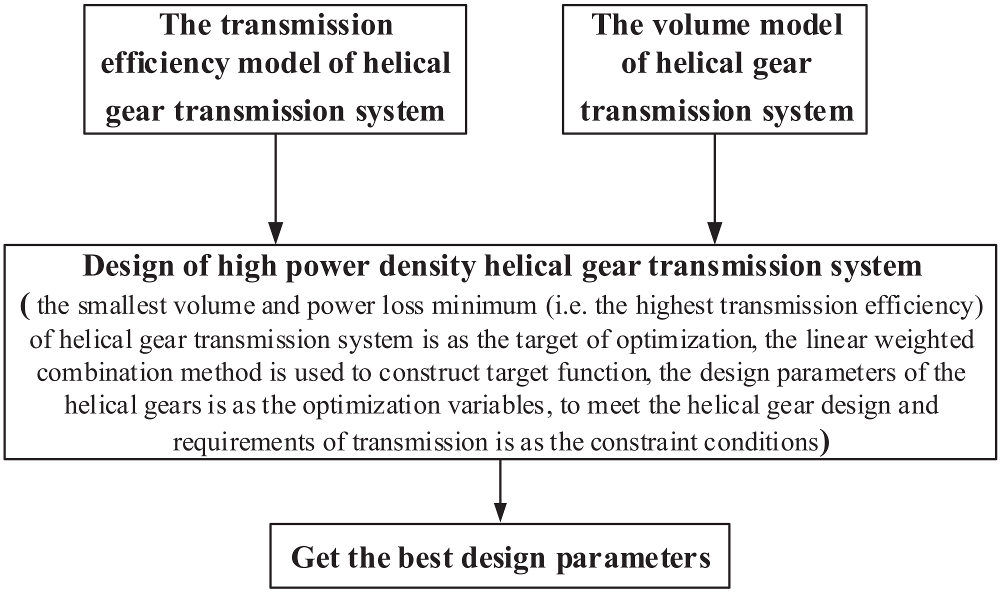


Fig. 2. Flow diagram for high power density design of helical gears transmission system.

TRANSMISSION EFFICIENCY MODEL OF A HELICAL GEAR TRANSMISSION SYSTEM

For the helical gear drive, it is impossible for the whole tooth width to engage in simultaneously; the front end of tooth first engages in, and the other subsequent end engages in. it is the same for the engage-out. Fig. 3 is the meshing surface of a standard helical gear. In the front face, B_1 point firstly engages in. *E'* point in back end face finally engages out. It is clear that the length of helical gear drive is more than B_2E segment compared with that of spur gear drive.

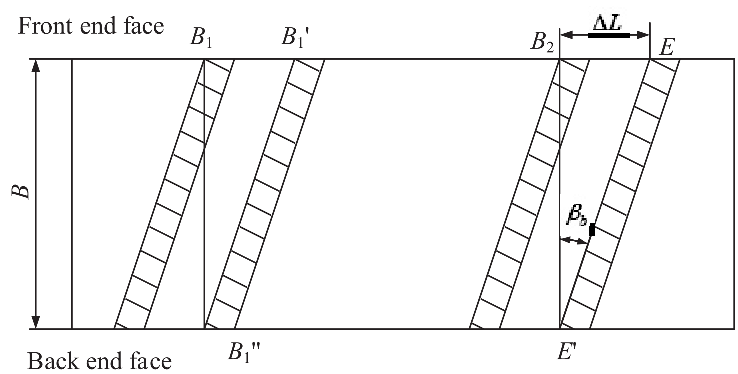


Fig. 3. Meshing face of standard helical gears.

A model of meshing efficiency of helical gear was proposed (Cheng *et al.,* 2012). However, the model was simplified. The problems include that ΔL was entirely placed in the engage-out end, and the normal load on the helical gear tooth surface is Fn/a in arbitrary meshing segment, where the model of meshing efficiency is modified.

The calculation of relative sliding speed at the meshing point of helical gears

In Fig. 4, the node *P* is the instantaneous center of the two helical gears; the relative sliding speed of the meshing point Q can be expressed as,

$$v_Q = PQ(w_1 + w_2) = e(w_1 + w_2) \tag{1}$$

where v_Q is the instantaneous sliding speed of meshing point Q; e is the distance between instantaneous meshing point Q and *P*; w_1 and w_2 are the angle speed of driving and driven gears, respectively.

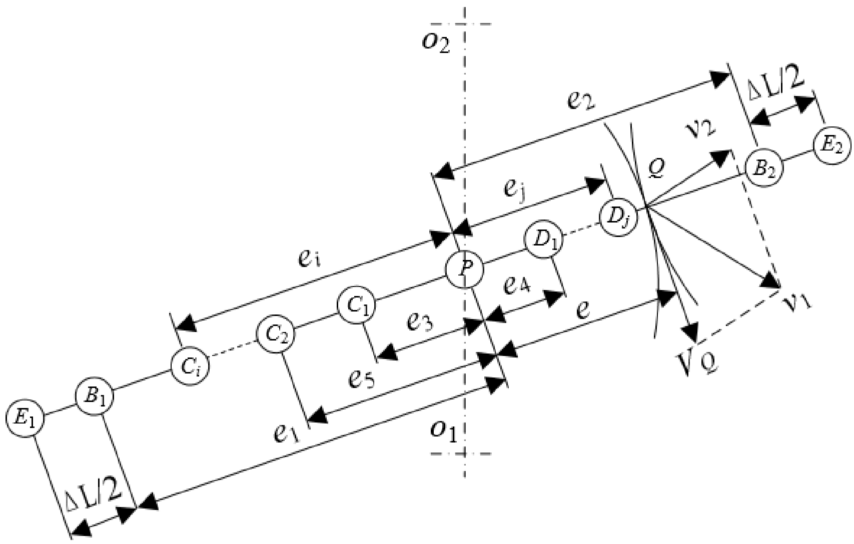


Fig. 4. The contact path and relative sliding speed at the meshing points.

The calculation of normal load on the helical gear tooth surface

For the simultaneous engagement teeth, the normal load on the teeth surface is assumed to be equal. In this paper, it is required to simplify when the coincidence degree $a \le \varepsilon \le a+1$, and the load is F_n/a in the C_1D_1 segment and is $F_n/(a+1)$ in the others.

The determination of average friction factor f on the helical gear tooth surface

The actual friction factor is not constant due to a variety of lubrication states in the helical gear meshing cycle. In the elastohydrodynamic lubrication, the friction coefficient *f* is usually between 0.03 and 0.07. In the paper, a simplified method is used, in which the friction factor *f* is taken as 0.05.

The calculation of sliding friction power loss of helical gears in a unit time

The sliding friction power loss of helical gears in a unit time can be expressed as,

$$dP_{SQ} = F_n f v_Q dt = F_n f e(w_1 + w_2)dt \tag{2}$$

where F_n is the normal load on the tooth surface; *f* is the friction factor on the tooth surface.

The moving distance of meshing point in a unit time can be expressed as,

$$de = r_{b1} w_1 dt = r_{b2} w_2 dt \tag{3}$$

Substituting (3) with (2), we obtain

$$dP_{SQ} = F_n f (1/r_{b1} + 1/r_{b2}) e de \tag{4}$$

The sliding friction power loss for a pair of helical gear teeth from engaging-in to engaging-out

Sliding friction power loss is the main component of power loss of gear transmission, where we only consider the influence of the sliding friction power loss in the transmission efficiency model of the gear transmission system. According to Fig. 3, the calculation of sliding friction power loss of a pair of helical gear teeth from engaging-in to engaging-out can be expressed as,

$$\begin{aligned} P &= \int_0^{E_1E_2} dP_{SQ} = \left(\int_{e_1}^{e_1+\frac{\Delta L}{2}} + \int_{e_i}^{e_1} + \cdots \int_0^{e_3} + \int_0^{e_4} + \cdots \int_{e_j}^{e_2} + \int_{e_2}^{e_2+\frac{\Delta L}{2}}\right) dP_{SQ} \\ &= \frac{F_n}{2} f\left(\frac{1}{r_{b1}} + \frac{1}{r_{b2}}\right)\left[\frac{e_3^2 + e_4^2}{a} + \frac{(e_1 + \frac{\Delta L}{2} - e_3)^2 + (e_2 + \frac{\Delta L}{2} - e_4)^2}{a+1}\right] \end{aligned} \tag{5}$$

where $e_1 = r_{b2}(\tan\alpha_{at2} - \tan\alpha_t)$; $e_2 = r_{b1}(\tan\alpha_{at1} - \tan\alpha_t)$; $e_3 = ap_{bt} - e_2$; $e_4 = ap_{bt} - e_1$; $p_{bt} = \pi m_t \cos\alpha_t$; $a \le \varepsilon \le a+1$, ε is the contact ratio, a is positive integer; $z_2 = i_{12} z_1$; $r_{b1} = \frac{m_t z_1}{2}\cos\alpha_t$; $r_{b2} = \frac{m_t z_2}{2}\cos\alpha_t$; $\tan\alpha_{at1} = \tan\arccos[z_2\cos\arctan\frac{\tan\alpha_n}{\cos\beta}/(z_2 + 2h_{an}^*\cos\beta)]$; $\Delta L = b\tan\beta\cos\alpha_t$; $\tan\alpha_{at2} = \tan\arccos[z_1\cos\arctan\frac{\tan\alpha_n}{\cos\beta}/(z_1 + 2h_{an}^*\cos\beta)]$.

The driving power of a pair of helical gear teeth from engaging-in to engaging-out can be expressed as,

$$P = \int_0^{E_1E_2} Fde = \left(\int_{e_1}^{e_1+\frac{\Delta L}{2}} + \int_{e_i}^{e_1} + \cdots \int_0^{e_3} + \int_0^{e_4} + \cdots \int_{e_j}^{e_2} + \int_{e_2}^{e_2+\frac{\Delta L}{2}}\right) Fde = p_{bt} F_n = \pi m_t \cos\alpha_t F_n \tag{6}$$

Therefore, the transmission efficiency of a pair of helical gear teeth from engaging-in to engaging-out can be expressed as

$$\eta = 1 - P/P_0 \tag{7}$$

THE VOLUME MODEL OF A HELICAL GEAR TRANSMISSION SYSTEM

The structure of a gear is related to its geometrical size. According to the diameter of addendum circle, the gear structure can be divided into solid structure, web-type structure, spoke-type structure with cross shape section, and spoke-type structure with H-shape section. Details are shown in Table 1.

Table 1. The structure of a gear.

Diameter of addendum circle	Structure category
$d_a \leq 160$	Solid structure
$160 < d_a \leq 500$	Web-type structure
$400 < d_a < 1000$	Spoke-type structure with cross shape section
$d_a \geq 1000$	Spoke-type structure with H-shape section

The volume model of solid structure

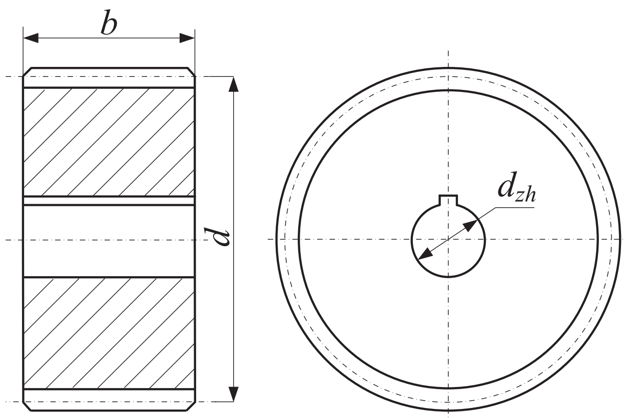


Fig. 5. The size structure of a helical gear with solid structure.

The size structure of a helical gear with solid structure is built, as shown in Fig. 5. The gear volume is defined as follows:

$$V_s = \frac{\pi b}{4}\left(\frac{m_n}{\cos\beta}z\right)^2 - \frac{\pi b}{4}d_{zh}^2 - V'' \tag{8}$$

Here, symbol b represents tooth width, symbol z represents tooth number, symbol m_n represents normal modulus, symbol d_{zh} represents axial diameter matched with gear, symbol β represents helix angle, and symbol V" represents clearance volume, with its detailed calculating method being found in the reference (Cheng *et al.,* 2015).

The volume model of web-type structure

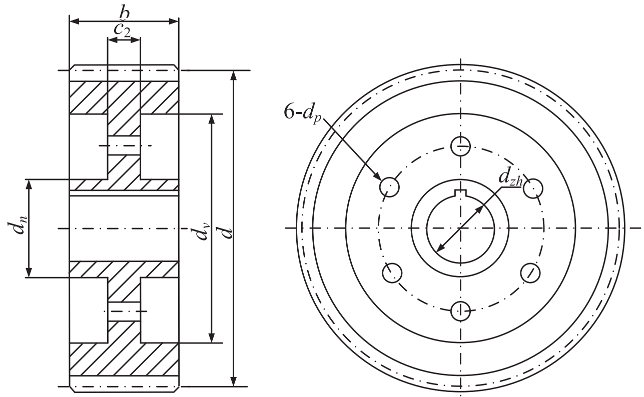


Fig. 6. The size structure of helical gear with the web-type structure.

The size structure of a helical gear with the web-type structure is built, as shown in Fig. 6. The gear volume is defined as follows:

$$V_w = \frac{\pi b}{4}\left[\left(\frac{m_n z}{\cos\beta}\right)^2 - d_v^2\right] + \frac{\pi c}{4}\left(d_v^2 - d_n^2\right) + \frac{\pi b}{4}\left(d_n^2 - d_{zh}^2\right) - \frac{n c_2 \pi}{4} d_p^2 - V'' \tag{9}$$

Here, symbol d_v represents inside diameter of rim, $d_v \approx d_a$-(1014-)m, symbol d_n represents outside diameter of wheel hub, $d_n \approx 1.7 d_{zh}$, symbol c_2 represents web thickness, c_2=(0.2~0.3)b, symbol dp represents drill hole diameter, d_p=(0.25~0.35)(d_v - d_n), and symbol n represents the number of drill holes.

The volume formula of spoke-type structure with cross shape section

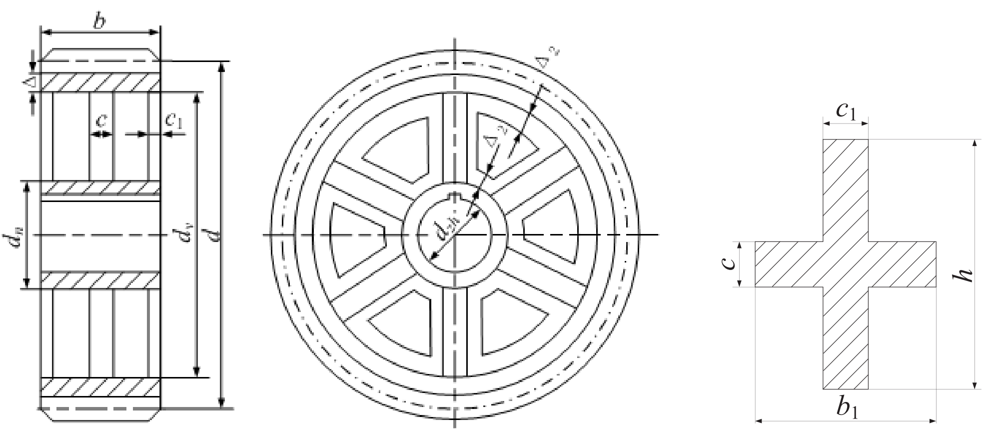


Fig. 7. The size structure of a helical gear with cross shape structure.

The helical gear size structure of spoke-type structure with cross shape section is built, as shown in Fig. 7. The gear volume is defined as follows:

$$V_{+} = \frac{\pi b}{4}\left[\left(\frac{m_n z}{\cos\beta}\right)^2 - d_v^2\right] + \frac{\pi b}{4}\left(d_n^2 - d_{zh}^2\right) + V' - V'' \tag{10}$$

Here, symbol Δ represents rim thickness, $\Delta \approx (3\sim4)m$, symbol d_v represents inside diameter of rim, $d_v = d - 2(h_a^* + c^* - y)m - 2\Delta$, $\Delta_2 \approx (1\sim1.2)\,\Delta$, symbol d_n represents outside diameter of wheel hub, $d_n \approx 1.7 d_{zh}$, symbol c, c_1, b_1, h represents the size of cross shape spoke, $c \approx 0.8 d_{zh}/5$, $c_1 \approx 0.8 d_{zh}/6$, $b_1 \approx 0.9 d_{zh}$, $h = b - 0.8 d_{zh}/3$, and symbol V' represents the volume formula of cross shape spoke; it is defined as follows:

$$\begin{aligned} V' = {} & c_1 h * (d_v - d_n)/2 * 6 + (cb_1 - cc_1)(d_v/2 - d_n/2 - 2\Delta_2) * 6 + \\ & \{\pi[(d_v/2)^2 - (d_v/2 - \Delta_2)^2]c - 6\Delta_2 c_1 c\} + \{\pi[(d_n/2 + \Delta_2)^2 - (d_n/2)^2]c - 6\Delta_2 c_1 c\} \end{aligned} \tag{11}$$

Eq. (11) can be simplified as

$$V' = 3c_1 h * (d_v - d_n) + 3(cb_1 - cc_1)(d_v - d_n - 4\Delta_2) + \pi(d_v + d_n)\Delta_2 c - 12\Delta_2 c_1 c \tag{12}$$

The volume model of spoke-type structure with H-shape section

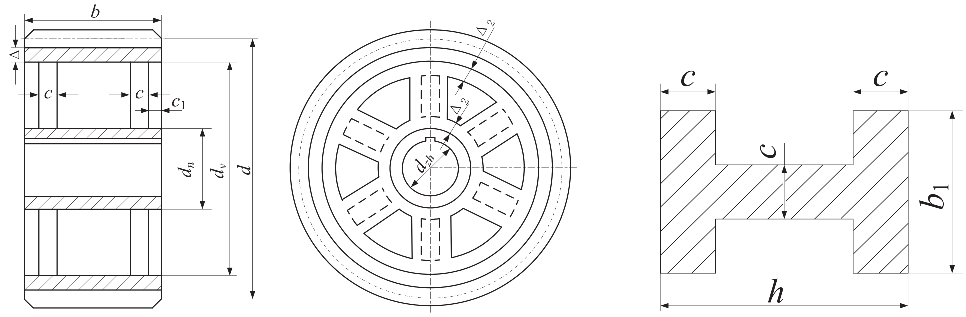


Fig. 8. The size structure of a helical gear with H-shape structure.

The helical gear size structure of spoke-type structure with H-shape structure is built, as shown in Fig. 8. The gear volume is defined as follows:

$$V_{H} = \frac{\pi b}{4}\left[\left(\frac{m_n z}{\cos\beta}\right)^2 - d_v^2\right] + \frac{\pi b}{4}\left(d_n^2 - d_{zh}^2\right) + V' - V'' \tag{13}$$

Here, symbol V' represents the volume formula of H-shape spoke; it is defined as follows:

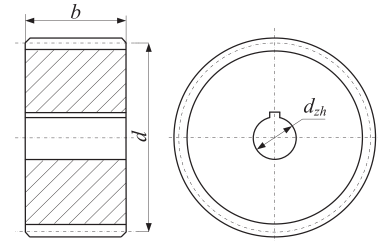
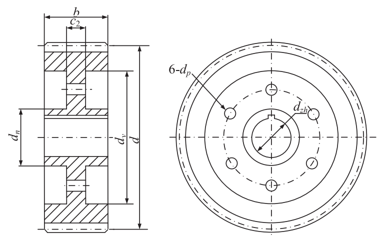
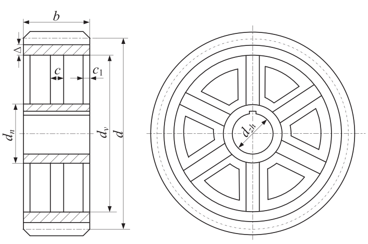
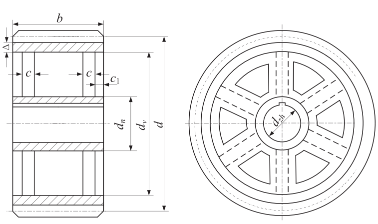
$$\begin{aligned} V' = {} & 2cb_1(d_v/2 - d_n/2 - 2\Delta_2) * 6 + c(h - 2c)(d_v/2 - d_n/2 - 2\Delta_2) * 6 \\ & + \pi[(d_v/2)^2 - (d_v/2 - \Delta_2)^2]h + \pi[(d_n/2 + \Delta_2)^2 - (d_n/2)^2]h \end{aligned} \tag{14}$$

Eq. (14) can be simplified as,

$$V' = 6cb_1(d_v - d_n - 4\Delta_2) + 3c(h - 2c)(d_v - d_n - 4\Delta_2) + \pi\Delta_2(d_v + d_n]h \tag{15}$$

The volume calculation formulas of helical gears transmission system are summarized and shown in Table 2.

Table 2. The volume calculation formula of helical gears transmission system.

Structure category	The schematic diagram of structure size	The formula of gear volume
Solid structure (d_a≤160)		$V_s = \frac{\pi b}{4}(\frac{m_n}{\cos\beta}z)^2 - \frac{\pi b}{4}d_{zh}^2 - V''$
Web-type structure (160<d_a ≤500)		$V_w = \frac{\pi b}{4}[(\frac{m_n z}{\cos\beta})^2 - d_v^2] + \frac{\pi c}{4}(d_v^2 - d_n^2) + \frac{\pi b}{4}(d_n^2 - d_{zh}^2) - \frac{n c_2 \pi}{4}d_p^2 - V''$
Spoke-type structure with cross shape section (400< d_a <1000)		$V_+ = \frac{\pi b}{4}[(\frac{m_n z}{\cos\beta})^2 - d_v^2] + \frac{\pi b}{4}(d_n^2 - d_{zh}^2) + V' - V''$
Spoke-type structure with H-shape section (d_a ≥1000)		$V_H = \frac{\pi b}{4}[(\frac{m_n z}{\cos\beta})^2 - d_v^2] + \frac{\pi b}{4}(d_n^2 - d_{zh}^2) + V' - V''$

DESIGN OF A HIGH POWER DENSITY HELICAL GEAR TRANSMISSION SYSTEM

Construct of objective function

The smallest volume and power loss minimum (i.e., the highest transmission efficiency) are set as the target of optimization; the linear weighted combination method is used to construct target function. It is shown in formula (16):,

$$f(x) = w_v f_v(x) + w_{1-\eta} f_{1-\eta}(x) \tag{16}$$

where w_v and $w_{1-\eta}$ are the weighting factors; $f_v(x)$ is the volume function; $f_{1-\eta}(x)$ is the power loss function.

Determination of the optimal variables

The design parameters of a helical gear (the normal parameters) include tooth number z, modulus $m_{n,}$ pressure angle $\alpha_{n,}$ spiral angle β, top gap coefficient c_n*, and addendum coefficient h_{an}*. Among them, α_n、c_n* and h_{an}* usually take the standard value. $z_2=i_{12}*z_1$; therefore, z_2 can be determined by z_1 and i_{12}. In the volume model and transmission efficiency model of helical gear transmission system, the tooth width b (assuming the tooth widths of driving gear and driven gear are equal) and the shaft diameter of d_{zh1} and d_{zh2} are also included. Therefore, the design variables of the helical gear transmission system are taken as,

$$x=\{x_1, x_2, x_3, x_4, x_5, x_6\}=\{m_n, z_1, b, d_{zh1}, d_{zh2}, \beta\} \quad (17)$$

Selection of constraint conditions

The constraint conditions include (1) meeting the contact fatigue strength requirements of tooth surface; (2) meeting the bending fatigue strength requirements of tooth root; (3) meeting the requirements of tooth width coefficient; (4) meeting the requirements of helix angle; (5) meeting the non-undercutting requirements of tooth; (6) meeting the requirements of torsional strength of gear shaft.

The design flow diagram of a helical gear transmission system with high power density is shown in Fig. 9.

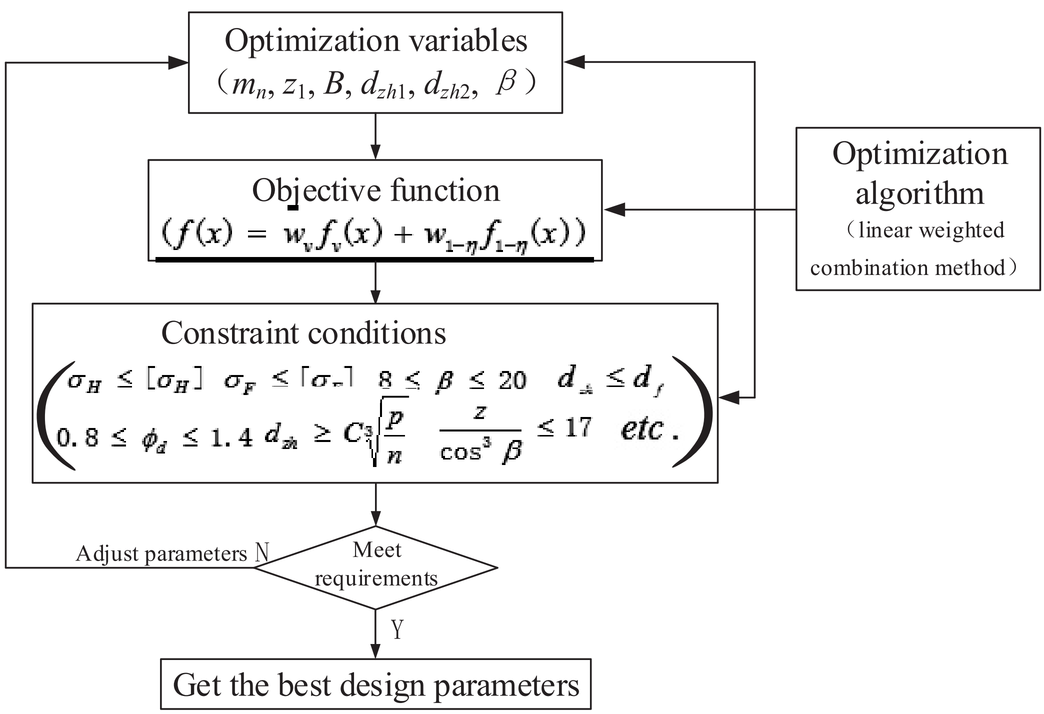
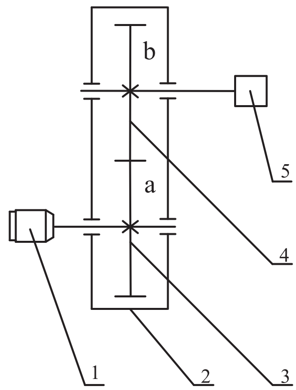


Fig. 9. Flow diagram of optimization.

AN EXAMPLE

A single stage helical gears transmission system is used to demonstrate the proposed method (Fig. 10). The input power P=10kw, the rotating speed of pinion n_1=960r/min, and the transmission ratio i_{12}=3.2. Other parameters are shown in Table 3.



1—Motor; 2—Gearbox;

3—Pinion; 4—Gear; 5—Loader

Fig. 10. A helical gear transmission system.

Table 3. The range of the optimal variables.

	m_n	z_1	b	d_{zh1}	d_{zh2}	β
Initial value	2	31	65	25	50	14
Lower limit value	2	29	60	25	50	8
Upper limit value	3	75	250	50	120	14

In this paper, the minimum power loss (i.e., the maximum transmission efficiency) is a very important objective. Therefore, w_v is taken as 0.4; $w_{1-\eta}$ is taken as 0.6. The objective function is constructed as,

$$f(x) = 0.4 f_v(x) + 0.6 f_{1-\eta}(x) \tag{18}$$

To eliminate the difference in the order of magnitude, the genetic algorithm toolbox (GA) in Matlab software is used to optimize magnitude. The range of optimization variables is shown in Table 3. The optimization results are standardized and rounded. The results before and after optimization are shown in Table 4. From Table 4, it is found that the power loss of the optimized gear system is reduced by 18.07%, and the volume is reduced by 11.39%.

Table 4. Comparison of results before and after optimization.

	m_n	z_1	b	d_{zh1}	d_{zh2}	β	$1\text{-}\eta$	Reduction ratio	Volume/ mm^3	Reduction ratio
Initial values	2	31	65	25	50	14	0.0106	18.07%	1026800	11.39%
Optimal value	2	30	61	34	51	9.9	0.008685		909850	

CONCLUSION

(1) By the method of integration along the meshing line, the transmission efficiency model of a helical gear transmission system is proposed; the relationship between the gear design parameters and the transmission efficiency is obtained.

(2) The volume model of a helical gear transmission system is proposed; the relationship between the gear design parameters and the volume model is obtained.

(3) The optimization for the smallest volume and power loss minimum (i.e., the highest transmission efficiency) is set as the target, and the linear weighted combination method is used to construct target function, in which the design parameters of the helical gears are set as the optimization variables, the helical gear design and requirements of transmission are set as the constraint conditions, and the design method of high power density of helical gears transmission system is proposed.

(4) A single stage helical gears transmission system is used to demonstrate the proposed method. The optimal design is carried out. From the results, it can be found that the power loss of the optimized gear system is reduced by 18.07%, and the volume is reduced by 11.39%.

The research work of this paper can provide a theoretical guidance for the design of a helical gear transmission system with high power density.

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تصميم أنظمة نقل الحركة الحلزونية ذات الكثافة عالية القدرة

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الخـلاصة

أصبحت الكثافة عالية القدرة توجه هام لتطوير أنظمة تروس نقل الحركة. وتركز الأبحاث الحالية على الحجم أو الكفاءة كل على حده، ولكن هناك نقص شديد في الأبحاث التي تجمع بين الاثنين. في هذا البحث، تم طرح طريقة لتصميم نظام تروس نقل الحركة الحلزونية ذات الكثافة عالية القدرة. أولاً، تم اقتراح نموذج لتروس حلزونية ذات كفاءة في نقل الحركة من خلال حساب مفقودات قدرة الاحتكاك الانزلاقي لنقطة التعشيق والتكامل الإضافي على طول خط التعشيق. ثانياً، تم اشتقاق معادلات الحجم لأشكال مختلفة من التروس الحلزونية، وتم اقترح نموذج الحجم لنظام نقل الحركة الحلزونية. وأخيراً، فإن الهدف الأمثل هو تقليل خسائر الحجم والقدرة (بمعنى، أعلى كفاءة لنقل الحركة) لنظام نقل الحركة الحلزونية. تم استخدام طريقة الارتباط الخطي المرجح لبناء دالة الهدف، وتم استخدام تصميم المعلمات للتروس الحلزونية بمثابة متغيرات للأمثلية. ولتصميم معدات التروس الحلزونية وتوفير متطلبات نقل الحركة كشروط تقييد، تم اقتراح طريقة تصميم كثافة عالية القدرة لنظام تروس نقل الحركة الحلزونية. وتم استخدام نظام نقل حركة حلزوني أحادي المرحلة لإثبات الطريقة المقترحة، وتم الانتهاء من التصميم الأمثل. ويمكن ملاحظة أن خسارة القدرة في نظام التروس المُثلى قد انخفضت بنسبة 18.07٪، وتم انخفاض الحجم بنسبة 11.39٪.