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Research on Thermal Elastohydrodynamic Lubrication of Cylindrical Gears with Curvilinear Shaped Teeth

Yong-Qiao Wei*, Deng-Qiu Ma**, Yong-Ping Liu* and Li Hou***

Keywords： CGCST, TEHL, Involution Velocity, Numerical Calculation, Oil Film Pressure,Oil Film Thickness

ABSTRACT

Based on meshing theory and the machining theory of cylindrical gears with curvilinear shaped teeth (CGCST), the tooth surface equation of CGCST is obtained. The modeling of arc tooth cylindrical gear pair was built by UG. Considering the contact geometry, relative velocity, entrainment velocity and load etc, the thermal elastohydrodynamic lubrication model of arc tooth cylindrical gear pair is established. The paper researched the arc tooth cylindrical gear pair in the meshing process by numerical analysis, including the law of oil film pressure, oil film thickness and temperature variation. It investigated the influence of the entrainment velocity and load on the oil film pressure, thickness and temperature of the tooth surface. The result shows that though the two pressure peaks, thickness and temperature rise of the arc tooth cylindrical gear pair's oil film are greatly influenced by the entrainment velocity and load, the corresponding laws are existing. The results of this study have a guiding role in the tribological design of arc tooth cylindrical gear and provide theoretical basis and engineering application value for the industrial applications of arc tooth cylindrical gears under the high speed and heavy load.

INTRODUCTION

With the rapid development of modern industry and the proposing of the "Industry 4.0",Mechanical transmission systems of higher requirement are required to have a greater bearing capacity and higher transmission efficiency in engineering fields of aerospace, construction machinery , mining machinery and so on, which gives many development opportunities for new gear technology[1]. CGCST researched by our group has the advantages of both traditional involute spur and helical cylindrical gears, CGCST has arc contact line along the direction of the tooth line and arc tooth surface. Compared with involute spur gears, they have a longer line of action and greater degree of coincidence in the same situation, which lead to better performance of meshing, higher contact strength and larger carrying capacity. Because of the "arch effect" of cylindrical gears with curvilinear shape teeth, their flexural strength has been largely improved. In addition, CGCST also has other characteristics, such as smooth transmission, no axial force, low noise, long life and low installation requirements. Based on those good features, the application prospect of those gears is very bright[1，2].

Since the characteristic and manner of engagement and more complex manufacturing than involute cylindrical gears, CGCST is susceptible to failures of pitting, gluing, and wear caused by poor lubrication in the transmission process. Through researching literature [4,5],it is known that current research on the lubrication performance of CGCST is almost blank. Therefore, in-depth study of lubrication parameters and lubrication conditions of CGCST is very important for improving their transmission lubrication properties and their application in aviation and aerospace field.

In 1916, Martin applied Reynolds equation to the research in gear lubrication for the first time. Gears were simplified as two cylindrical rollers, and the change regularity of pressure distribution and film thickness with velocity and load was obtained, yet which was quite different from the practice. Using inverse method, Dowson and Higginson have gotten the full numerical solution line contact lubrication model in 1950s. EHL theory considering Contact deformation at the interface in document [8] provides basis for study on gear lubrication. Bevel gear thermal EHL was analyzed by means of finite difference method in document [9], where load

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* *School of Mechanical & Electronical Engineering,Lanzhou University of Technology,Lanzhou,China*

** *School of Engineering and Technology, Zunyi Normal College, Guizhou Zunyi, China*

****School of Manufacturing Science and Engineering, Sichuan University, Chengdu, China*

capacity of gear, maximum oil film temperature and formula for coefficient formula were fitted regressively .With Multi-grid method, it was in document[10] that spur gear thermal EHL was discussed. In document[11] analysis model of finite length thermal EHL was established, and change of oil film pressure, thickness and temperature was considered. Thermal EHL of bevel gear, hypoid gear and worm gear was analyzed in Document[12], where the influence of rotational speed on load capacity, oil film pressure and temperature was obtained. It was V. Simon who established the thermal EHL analysis model of hypoid gear, according to its characteristic [13]. Wang Jiaxu and others have conducted research on interfacial lubrication of hypoid gear under heavy load [14]. Hou Li and others have established EHL model of CGCST. And the effect of load, velocity, and viscosity on oil film pressure and thickness was discussed in their research. The above is research of EHL and Thermal EHL on spur, bevel and hypoid cylindrical gear. However, there isn't any study on thermal EHL analysis of CGCST that was published.

In this paper, based on meshing principle of CGCST and the EHL theory, the equation of CGCST tooth surface was established, while the tooth flank curvature and relative velocity (Entrainment velocity) of meshing surface was calculated. By means of adding energy equation into isothermal EHL model in line contact, the thermal EHL model was set up, and the effect of load and entrainment velocity on the oil film pressure, thickness and temperature was discussed in this paper. The result of study plays a guiding role in the tribology design of CGCST.

THE EQUATION OF TOOTH SURFACE

The machining principle with rotary cutter head for cylindrical gears with arcuate tooth trace is generation method. When the cylindrical gears with arcuate tooth trace is machined by the rotary cutter head, the rotary cutter head is in the rotation and the gear blank is in the rotation and moving along the rotary cutter head. The position relationship, coordinate system and coordinate transformation of rotary cutter head and gears are shown in figure 1. The coordinate system $\Sigma_{d0}=\left[O_{d0};x_{d0},y_{d0},z_{d0}\right]$ is the static coordinate system, which is rigidly connected to the cutter head. The $\Sigma_d=\left[O_d;x_d,y_d,z_d\right]$ is the moving coordinate system which is fixed with the axis of the cutter central position. The coordinate system $\Sigma_1=\left[O_1;x_1,y_1,z_1\right]$ is the moving coordinate system which is rigidly connected to the gear blank. The coordinate system $\Sigma=\left[O;x,y,z\right]$ is the fixed coordinate system. The coordinate system $\Sigma_f=\left[O_1;x_f,y_f,z_f\right]$ is the reference coordinate system.

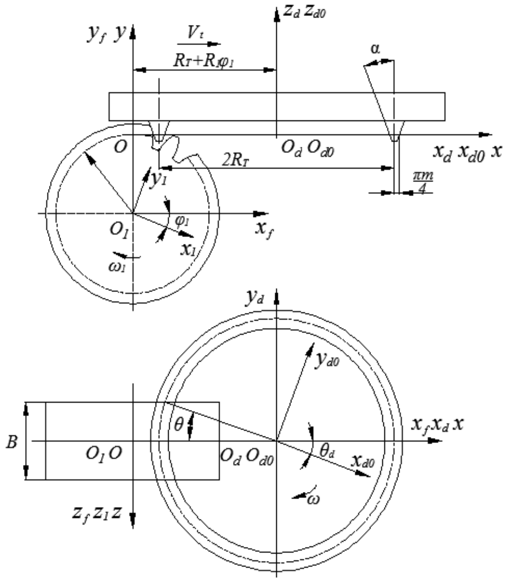


Fig.1 Forming principle of CGCST

In the coordinate system of rotary cutter, the parameter equation of the cutter surface is given as follows based on the fig.1:

$$\Sigma_d:\begin{cases} x_d=-(R_T\pm\pi m/4\pm u\sin\alpha)\cos\theta \\ y_d=(R_T\pm\pi m/4\pm u\sin\alpha)\sin\theta \\ z_d=u\cos\alpha \end{cases} \quad (1)$$

The parameter μ is the distance in cutter coordinate system from point on the cutting surface to the envelop reference point along cutting cone generatrix. Parameter α is the pressure angle of tool (equal to the cone semi-angle).Parameter θ is the angle that gear blank rotates by from the middle section to current enveloping point. When the parameter θ equals to the zero, the reference point is on the middle section. Parameter R_T is the nominal cutter radius.

The transformation matrix A_{od} from $\sum_d=\left[O_d;x_d,y_d,z_d\right]$ to $\sum=\left[O;x,y,z\right]$ is given as following:

$$A_{od}=\begin{bmatrix} 1 & 0 & 0 & R_1\varphi_1+R_T \\ 0 & 0 & 1 & 0 \\ 0 & -1 & 0 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (2)$$

The parameter R_1 is the pitch radius of gear blank. The motion parameter φ_1 is the gear rotational angle.

The transformation matrix A_{fo} from

$\sum = [O; x, y, z]$ to $\sum_f = [O_f; x_f, y_f, z_f]$ is given as following:

$$A_{fo} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & R_1 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (3)$$

The transformation matrix A_{1f} from $\sum_f = [O_f; x_f, y_f, z_f]$ to $\sum_1 = [O_1; x_1, y_1, z_1]$ is given as following:

$$A_{1f} = \begin{bmatrix} \cos\varphi_1 & -\sin\varphi_1 & 0 & 0 \\ \sin\varphi_1 & \cos\varphi_1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (4)$$

So the transformation matrix A_{1d} from $\sum_d = [O_d; x_d, y_d, z_d]$ to $\sum_1 = [O_1; x_1, y_1, z_1]$ is given as following:

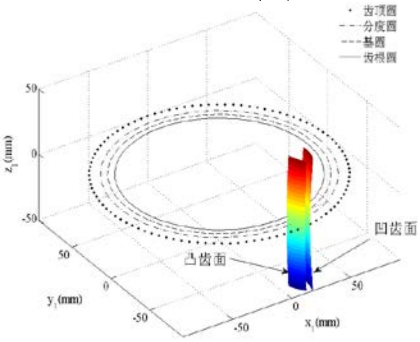
$$A_{1d} = A_{1f} A_{fo} A_{od} \quad (5)$$

Based on the spatial engagement principle and differential geometry, using the movement relationship between cutter head and gear blank, the tooth surface equation of CGCST is obtained according to the tool surface parameter equation and corresponding coordinate transformation matrix .

$$\begin{cases} x_1 = \left[-(R_T \pm \pi m/4 \pm u\sin\alpha)\cos\theta + R\varphi_1 + R_T\right]\cos\varphi_1 \\ \quad -(u\cos\alpha + R)\sin\varphi_1 \\ y_1 = \left[-(R_T \pm \pi m/4 \pm u\sin\alpha)\cos\theta + R\varphi_1 + R_T\right]\sin\varphi_1 \\ \quad +(u\cos\alpha + R)\cos\varphi_1 \\ z_1 = -(R_T \pm \pi m/4 \pm u\sin\alpha)\sin\theta \\ u = \mp\sin\alpha(R_T \mp \pi m/4) + (R\varphi_1 + R_T)\sin\alpha/\cos\theta \end{cases} \quad (6)$$

MESHING MODEL

The point cloud of tooth surface is generated in MATLAB on the basis of the principle of meshing and tooth surface equation, as shown in Fig.2. The tooth surface is formed from the point cloud using the curved surface fitting function in the NX 10.0. And the gear model is form the tooth surface. The rack straight line profile in meshing movement formed in the space of linear combination is the essence of the CGCST by the point cloud. The ideally instantaneous contact lines of gear is the space line, the gear is the arc curve in the tooth line direction and the involute in the tooth profile direction. As shown in Fig.3 is the meshing contact model of arc tooth cylindrical gears, the contact form is the line contact, and the contact line length is equal to the corresponding the arc length of the circular radius R_a (R_b).



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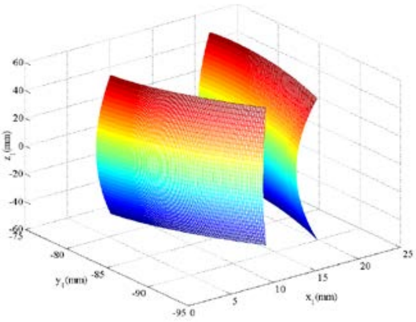

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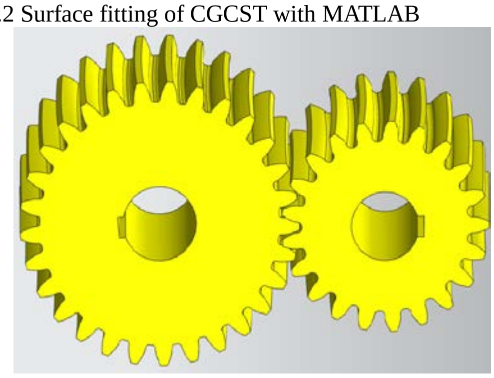
Fig.2 Surface fitting of CGCST with MATLAB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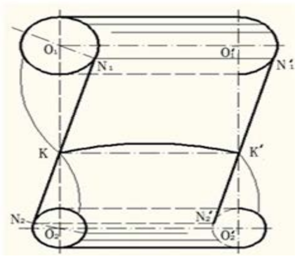
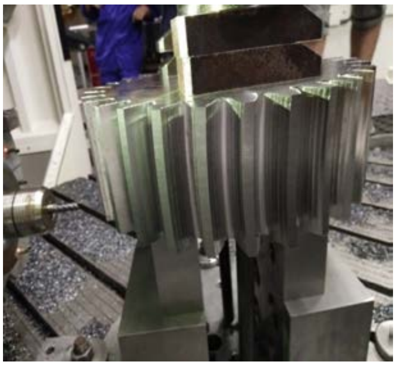


(b)

Fig.3 Meshing model of CGCST

According to the tooth surface equation, the CGCST gears are manufactured with NCN machine tool, it's the gear's processing sample in Fig.4(a) and the installed CGCST gear pairs in Fig.4(b). The gear pair meets the meshing properties and geometric features by applying loads, which verifies the correctness of the tooth surface equation.



(a)

(b)

Fig.4 Sample processing of the CGCST gears

The meshing transmission diagram gear is shown in fig.5, two gears were rotating in the opposite direction, M is the space meshing points, a is the center distance. w_1 and w_2 are gear's angular velocity respectively.

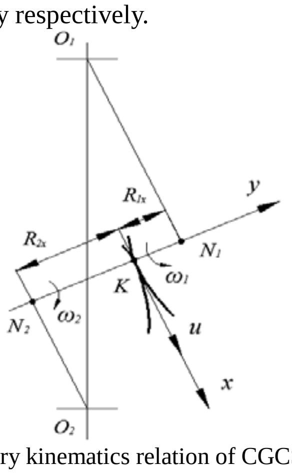


Fig.5The geometry kinematics relation of CGCST

Based on the elastohydrodynamic lubrication theory, the tooth surfaces speed of the CGCST gear at the meshing-point are given as follows:

$$\mu_1 = R_{1x}\omega_1 = \frac{\pi n_1}{30K_2^I} \qquad (7)$$

$$\mu_2 = R_{2x}\omega_2 = \frac{\pi n_2}{30K_2^{II}} \qquad (8)$$

According to the reference [15], the entrainment velocity of the CGCST gears at the meshing point can be obtained:

$$\mu = \frac{\mu_1 + \mu_2}{2} = \frac{\pi}{60}\left(\frac{n_1}{K_2^I} + \frac{n_2}{K_2^{II}}\right) \qquad (9)$$

The rolling sliding ratio is an important parameter used to estimate the motion state of the gear transmission. According to the reference [15], the rolling sliding ratio of the CGCST gear is obtained:

$$\kappa = 2\left|\frac{K_2^{II}n_1 - K_2^I n_2}{K_2^{II}n_1 + K_2^I n_2}\right| \qquad (10)$$

LUBRICATION MODEL

The analysis of thermal EHL for CGCST is obtained based on the EHL of CGCST in line contact which is added energy equation and the temperature of upper and lower interfaces.

The Reynolds equation of isothermal EHL in line contact is expressed as follows:

$$\frac{d}{dx}\left(\frac{\rho h^3}{\eta}\cdot\frac{dp}{dx}\right) = 12u\frac{d(\rho \mathrm{h})}{dx} \qquad （11）$$

Where p is the film pressure, h is the film thickness, ρ is the density of lubrication oil, η is the viscosity of lubrication oil, u is the entrainment velocity.

The energy equation is expressed as follows:

$$c_p\rho u\frac{\partial T}{\partial x} = k\frac{\partial^2 T}{\partial z^2} - \frac{T}{\rho}\frac{\partial \rho}{\partial T}u\frac{\partial p}{\partial x} + \eta\left(\frac{\partial u}{\partial z}\right)^2 \qquad (12)$$

Where the temperature of the upper and lower interfaces for gear is given as follows[17]:

$$\begin{cases} T(x,0) = \dfrac{k}{\sqrt{\pi\rho_1 c_1 k_1 u_1}}\displaystyle\int_{-\infty}^{x}\frac{\partial T}{\partial z}_{(x,0)}\frac{ds}{\sqrt{x-s}} + T_0 \\ T(x,h) = \dfrac{k}{\sqrt{\pi\rho_2 c_2 k_2 u_2}}\displaystyle\int_{-\infty}^{x}\frac{\partial T}{\partial z}_{(x,h)}\frac{ds}{\sqrt{x-s}} + T_0 \end{cases} \qquad (13)$$

The formula (11) is normalized by the introduced dimensionless parameters.

$$X = x/b, W = \omega/(E^{'}.R), H = h\cdot R/b^2$$

$$P = p/p_h, \bar{\eta} = \eta/\eta_0, \bar{\rho} = \rho/\rho_0$$

$$\frac{1}{E^{'}} = \frac{1}{2}\left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right)$$

Where b is the contact half-width. η_0 is the initial viscosity. p_h is the maximum hertz contact pressure. $E^{'}$ is synthetic elastic modulus of gear pairs， E_1、E_2 and ν_1、ν_2 are respectively the elastic modulus and Poisson's ratio of the two gears.

The normalized equation（11）is given as follows：

$$\frac{d}{dX}\left(\varepsilon\frac{dP}{dX}\right)=\frac{d(\bar{\rho}\cdot H)}{dX} \quad (14)$$

Where

$$\varepsilon=\frac{\rho H^3}{\eta\lambda},\lambda=\frac{12\eta_0 UR^2}{\mathrm{p_h}\cdot \mathrm{b}^2}$$

$$P(X_0)=0,P(X_e)=0,\frac{dP(X_e)}{dX}=0$$

The normalized energy equation is given as follows：

$$A\bar{\rho}u\frac{\partial T}{\partial X}=B\frac{\partial^2 T}{\partial Z^2}+C\frac{u\bar{T}}{\bar{\rho}}\frac{\partial\bar{\rho}}{\partial\bar{T}}\frac{\partial P}{\partial X}+D\eta\left(\frac{\partial u}{\partial Z}\right)^2 \quad (15)$$

where，

$$A=\rho_0 c_p u_s T_0/b, B=kT_0 R/b^2$$

$$C=u_s p_H/b, D=\eta_0\left(\frac{u_s R}{b^2}\right)$$

The following formulas are rewritten according to the above normalization method. As the limited of the length, the original equations are omitted and the normalized equations are given.

The normalized film equation is given as follows：

$$H(X)=H_0+\frac{X^2}{2}-\frac{1}{\pi}\int_{x_0}^{x_e}\ln\left|X-X'\right|P(X')\mathrm{d}X' \quad (16)$$

The normalized viscous pressure density equation is given as follows:

$$\bar{\rho}=1+\frac{0.6\times10^{-9}p}{1+1.7\times10^{-9}p}+D(T-T_0) \quad (17)$$

The normalized viscous pressure temperature equation is given as follows:

$$\bar{\eta}=\exp\{(\ln\eta_0+9.67)\}\left[(1+p_H P/p_0)^{0.68}\left(\frac{T-138}{T_0-138}\right)^{-1.1}-1\right] \quad (18)$$

The normalized loading equation is given as follows:

$$W=\int_{X_0}^{X_e}P(X)\mathrm{d}X=\frac{\pi}{2} \quad (19)$$

NUMERICAL CALULATION

For the study of line contact EHL numerical solutions, differential method is used for the lubrication model in this paper. The above equations are discretized by using central difference, forward and backward difference. Then discrete equations are obtained:

$$\frac{\varepsilon_{i-1/2}\cdot P_{i-1}-(\varepsilon_{i-1/2}+\varepsilon_{i+1/2})P_i+\varepsilon_{i+1/2}\cdot P_{i+1}}{\Delta X^2}=\frac{\nabla\bar{\rho}_i\mathrm{H_i}}{\Delta X} \quad (20)$$

Where

$$\nabla X=X_i-X_{i-1},\nabla\bar{\rho}_i\mathrm{H_i}=\frac{\bar{\rho}_i H_i-\bar{\rho}_{i-1}H_{i-1}}{\Delta X},$$

$$\varepsilon_{i\pm1/2}=\frac{(\varepsilon_i\pm\varepsilon_{i\pm1})}{2}$$

$$A\bar{\rho}u\left(\frac{T_{i,k}-T_{i-1,k}}{\Delta X}\right)=B\frac{T_{i,k+1}-2T_{i,k}+T_{i,k-1}}{\Delta Z^2}$$

$$+C\frac{u\bar{T}}{\bar{\rho}}\frac{\partial\bar{\rho}}{\partial\bar{T}}\frac{\Delta P_{i,j}-\Delta P_{i-1,j}}{\Delta X}+D\eta\left(\frac{u_{i,k+1}-u_{i,k}}{\Delta Z}\right)^2$$

$$H_\mathrm{i}=H_0+\frac{\mathrm{x}^2}{2}-\frac{1}{\pi}\cdot\sum_{\mathrm{j=1}}^{\mathrm{n}}K_{\mathrm{ij}}\cdot P_\mathrm{j}$$

$$\Delta X\sum_{\mathrm{i=1}}^{n}\frac{P_i+P_{i+1}}{2}=\frac{\pi}{2}$$

In the form, $K_{i,j}$ is elastic deformation stiffness coefficient, which indicates elastic deformation at the node i caused by the unit pressure at the node j .After the discretization of the equation, the calculation flow chart shown in Fig. 6 is used for iterative solution of the model.

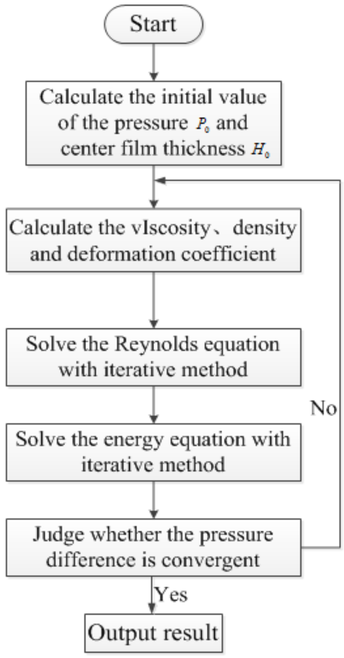


Fig.6 The calculating flow chart

Based on the mathematical model established above, arc tooth cylindrical gear pair is calculated, and the gear pair parameters are shown in Table 1:

Table1 the parameter of gear pair

Parameter	Driving wheel	Driven wheel
Tooth Number	21	29
Modulus	4	4

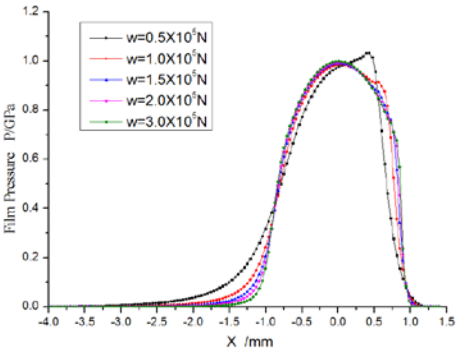
tooth width/mm	30	30
Pressure angle	20	20
Tooth line radius /mm	200	200

The elastic modulus of gear material is $2.21\times10^{11}Pa$; The poisson's ratio is 0.3; The initial viscosity of lubricating oil is $0.03Pa.s$ .The computational boundary is $-2.0\le x\le1.5$ .Node number N is set as 129;Driving wheel torque M=520N.m. The entrainment velocity can be calculated according to the formula (5) (6).According to the gear thermal EHL model by iterative calculation with the above-mentioned parameters, we get the oil film pressure distribution and the shape of arc tooth cylindrical gear line side thermal EHL.

The characteristic parameters of lubrication characteristics of CGCST are mainly thermal EHL oil film pressure, film thickness and temperature change. It can be seen that the characteristic parameters of effect were mainly entrainment speed, load, temperature, and the material elastic modulus and viscosity of lube base oil parameters. The impact of the entrainment velocity, load and temperature rise on the oil film pressure and oil film thickness of the CGCST gear will be mainly analysed in this paper.

Parameter analysis of film pressure

Fig. 7 shows the influence of entrainment velocity and load on the oil film pressure. The pressure distribution of oil film is in the middle part of the contact area, which is close to the Hertz pressure distribution from the figure. Oil film pressure appears obvious second pressure peak near the exit zone, but the second pressure decreases rapidly to the ambient pressure. From Fig.7 (a), it shows that the second pressure peak of the oil film increases rapidly with the increasing of the entrainment velocity, and its position moves to the entrance zone. In Fig.7 (b), it shows that the second pressure peak of oil film decreases slowly with the increase of load, and its position moves to the entrance zone.



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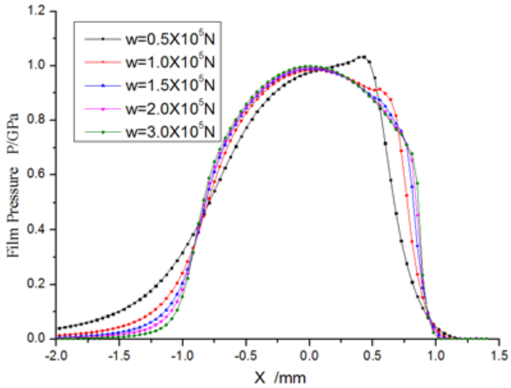

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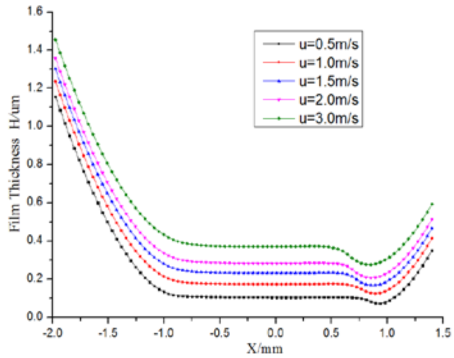
Fig.7 Influence of entrainment velocity and load on the oil film pressure

Parameter analysis of film thickness

Fig.8 reflects the influence of entrainment velocity and load on the film thickness, and it could be seen from the figure that the necking of film thickness occurs in the vicinity of the second pressure peak, and the film thickness reaches minimum at the necking position. It could be seen from Fig.8 (a) that the film thickness increases with the increasing of entrainment velocity. The necking of film thickness corresponding to the second pressure peak also occurs in advance and moves towards the direction of entrance zone. It could be seen from Fig.8 (b) that the film thickness decreases with the increasing of load, the necking of film thickness corresponding to the second pressure peak occurs lately and moves towards the direction of exit zone. With film thickness decreasing, gear surfaces are more susceptible to wear failures, so reasonable choosing of operating speed of gears and load on gears is very important to avoid the occurrence of wear failure on gear surfaces maximally.



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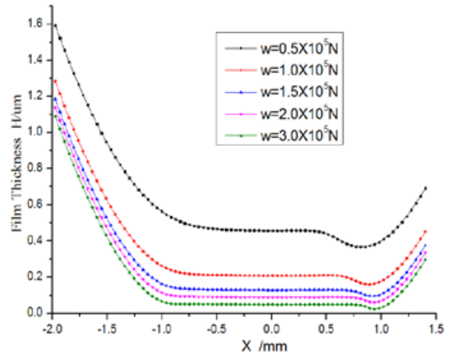

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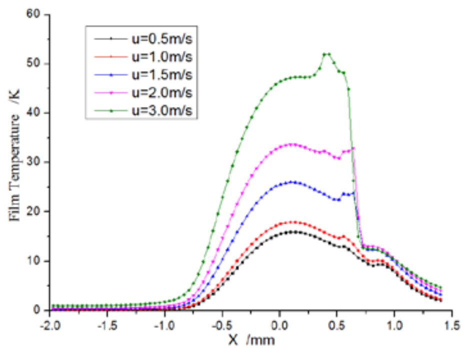
Fig.8 Influence of entrainment velocity and load on the oil film thickness

Parameter analysis of film temperature

Fig.9 reflects the influence of entrainment velocity and load on temperature of oil film. It could be seen from Fig.9 that the temperature of oil film increases with the increasing of entrainment velocity and load. The change amplitude of temperature become larger and the position of temperature jump moves towards the entrance zone. With the temperature rising, gear surfaces are more susceptible to scuff, meanwhile the effect of load change on temperature rising is stronger than that of entrainment velocity. So the reasonable choosing of operating speed and load of gears is very important to avoid the occurrence of scuffing on gear surfaces maximally.



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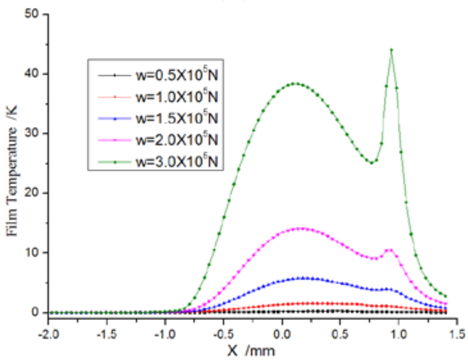

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Fig.9 Influence of entrainment velocity and load on the oil film temperature

CONCLUSIONS

1) Based on the meshing principle and machining principle of CGCST, the tooth surface equation of CGCST has been derived. Considering the contact characteristics, relative velocity, entrainment velocity, load and other factors synthetically, the mathematical model of thermal EHL for CGCST has been established.

2) Taking numerical method as the calculation foundation, the influence regularity of entrainment velocity and load on oil film pressure, thickness and temperature of CGCST have been discussed.It could be known that the temperature of the oil film have obviously positive correlation with velocity and load. And the greater the change of velocity and load is, the more likely tooth surface scuffs.So in order to avoid possible scuffing of gears maximally, the operating velocity and load of gears should be chosen reasonably to make the lubrication of CGCST close to full EHL and ensure the best lubrication properties.

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變雙曲圓弧齒線圓柱齒輪熱彈流潤滑特性研究

魏永峭　劉永平
蘭州理工大學機電工程學院

馬登秋
遵義師範學院工學院

侯力
四川大學製造科學與工程學院

摘 要

本文以啮合原理和弧齿圆柱齿轮加工原理为基础，得到弧齿线圆柱齿轮的齿面方程，利用 UG 实现了弧齿圆柱齿轮副建模；综合考虑了弧齿线圆柱齿轮传动的接触几何、相对运动速度、卷吸速度和载荷等因素，建立了弧齿线圆柱齿轮副的热弹流润滑模型，数值分析了弧齿线圆柱齿轮副在啮合过程中的齿面油膜压力、油膜厚度和温度变化规律，探讨了卷吸速度和载荷对齿面油膜压力、厚度和温度的影响. 结果表明：卷吸速度和载荷对弧齿线圆柱副油膜的二次压力峰、厚度和温升影响较大，但都存在相应的规律。该结果对弧齿线圆柱齿轮的摩擦学设计分析有指导作用，并为弧齿圆柱齿轮在高速重载等工况下的工业应用提供理论依据和工程应用价值。