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Non-Steady Elastohydrodynamic Lubrication Analysis on Spur Gear Teeth

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A non-steady 3-dimensional elastohydrodynamic lubrication analysis was performed on the contacting teeth surfaces of involute spur gears. Kinematics of the gear and the pinion were taken into account to get accurate geometric clearance around the elastohydrodynamic lubrication region of the contacting teeth. Pressure and film thickness distribution for the whole contacting faces in lubricated condition at several time steps were obtained through the analysis. Besides the pressure spike at the outlet region, a representative phenomenon in elastohydrodynamic lubrication regime, the pressure at the inlet region was slight higher than that of the center region. The film thickness of non-steady condition was thicker than that of steady condition.

Key words: Spur gear, Elastohydrodynamic lubrication, Non-steady, Kinematics

1. INTRODUCTION

Gears are essential elements to transfer power and are widely used for most of the machinery. Many studies have been focused on design, manufacturing, and test of the gear and some standards are much useful for gear industry [1-2]. An inclination to precision and fineness of the modern mechanical industry requires that gears should have high quality and high reliability. Therefore, an actual operating condition should be considered in designing of the gear in addition to the conventional standards. In particular, as gears are normally operated in lubricated condition a study on lubrication phenomenon of gear teeth can be much helpful to improvement of the gear design. Because the contact of the gear teeth is a non-conformal contact and the contact pressures are very high due to small contact area, gear is a typical element that is working in elastohydrodynamic lubrication regime [3].

It seems that not so many studies on elastohydrodynamic lubrication of gear teeth have been executed. R. Larson calculated film thickness and pressures along line of action with consideration of transient effect and non-Newtonian fluid model [4]. Contact model is 2-dimensional line contact with assumption that gear's contacting face is two roller's contact. Dowson and Higginson suggested a formula of minimum film thickness between gear teeth with the use of their study in elastohydrodynamic lubrication [5]. In this study, the lubrication phenomenon on gear teeth contact was estimated by a non-steady 3-dimensional elastohydrodynamic lubrication analysis with consideration of gear kinematics.

2. INITIAL CLEARANES OF GEAR TEETH

The clearance shape between gear teeth should be known in advance of the elastohydrodynamic lubrication analysis. As the gear rotates the contacting points of gear teeth vary and contacting radius of curvature change and therefore, the clearance shape of gear teeth changes dependent on time. By the following procedure, the initial clearance shape for a specific time step can be calculated accurately.

1. Tooth profiles of the gear and the pinion are generated in local polar coordinates whose origins are located at the center of the gear and the pinion respectively.

2. To find the posture of the gear profile and pinion profile at initial contact, gear profile is rotated so that the end point of gear profile meet the intersection of the gear outside diameter and the line of action. Cartesian coordinates of the intersection point are:

$$x_{o1} = \frac{-2R_{p1}\tan\alpha + \sqrt{4R_{p1}^2\tan^2\alpha - 4(1+\tan^2\alpha)\left(R_{p1}^2 - R_{o1}^2\right)}}{2\left(1+\tan^2\alpha\right)}$$

$$y_{o1} = x_{o1}\tan\alpha + R_{p1} \tag{1}$$

Then, pinion profile is rotated until the initial contacting point on the pinion profile matches the above intersection point. The initial contacting point on the pinion profile is the one from which the distance to the origin of the pinion is the same as the distance from the intersection point to the origin.

3. To get the changed profile posture by the successive rotation of gear and pinion, the local Cartesian coordinates to be converted to the local polar coordinates. Several rotation angles selected along the line of action and the gear and the pinion are rotated stepwise each other. After each step rotation, the local polar coordinates are converted to the global Cartesian coordinates and the profile data are reserved.

4. To find a contacting point of two profiles at each step, the distances between two profiles are calculated. A pair of points whose distance is minimum becomes a contacting point.

5. Region for lubrication analysis is set. By letting the contacting point as reference point, the same lengths of inlet region and outlet region are taken between two profiles respectively. Because the grid size of the profiles are not even due to the conversion of polar to Cartesian coordinate system, the nodes should be rearranged so as to have an even grid size along the original profile.

6. Finally, the initial clearances are obtained by eq.(2) which is a distance between two mating nodes.

$$h_g = \sqrt{(x_2 - x_1)^2 + (y_2 - y_1)^2} \tag{2}$$

3. LUBRICATION ANALYSIS

The "Reynolds Equation" is adopted to find pressure distributions through a non-steady 3-dimensional elastohydrodynamic lubrication analysis.

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial y}\right)=12\tilde{u}\frac{\partial(\rho h)}{\partial x}+12\frac{\partial(\rho h)}{\partial t} \qquad (3)$$

The entraining velocity is calculated by eq.(4).

$$\tilde{u}=\{\omega_1(R_{p1}\sin\alpha+x)+\omega_2(R_{p2}\sin\alpha-x)\}/2 \qquad (4)$$

The pressure distribution should satisfy the following force equilibrium condition.

$$w=\iint pdxdy \qquad (5)$$

At an elastohydrodynamic lubrication regime, dependency of density and viscosity on pressure should be considered [5-6].

Newton-Rapson method was used in the analysis. First, 3-dimensional contact analysis was done and the pressure distribution that satisfy the equilibrium condition of eq.(5) is used for the initial pressure of steady elastohydrodynamic lubrication analysis. And the result of steady elastohydrodynamic lubrication analysis was used for the initial pressure of non-steady elastohydrodynamic lubrication analysis. Gear's type is involute spur gear and module is 4mm. 13 time steps were taken along the line of action for the non-steady analysis.

4. ANALYSIS RESULT AND DISCUSSION

Fig.1 is a shape of the initial clearance between two teeth profiles.

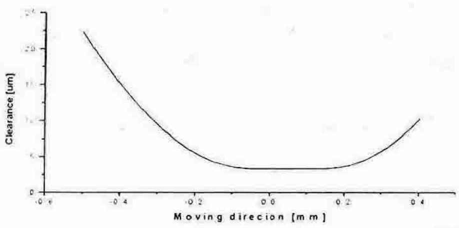

Fig.1 Initial clearance between the contacting gear teeth

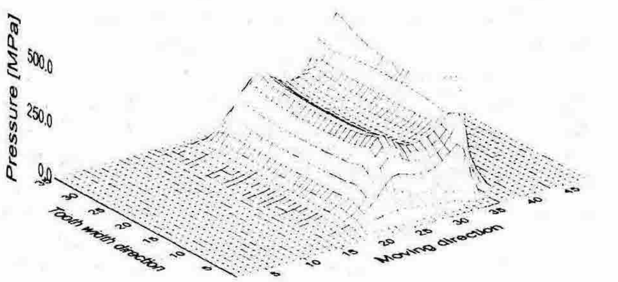

Fig. 2 Pressure distribution at line of action 3mm

Fig.2 demonstrates 3-dimensional pressure distribution of contacting region between the teeth for line of action 3mm at which load is high and entraining velocity is low relatively. We can see a remarkable pressure spike at outlet and a moderate peak pressure at inlet also. This phenomenon can be attributed to the contact pressure pattern and contacting width of the teeth. Namely, contact of two gear teeth is close to face contact rather than line contact.

Fig.3 shows pressure profiles at center of tooth face. They are similar pattern one another but peak values increase with the decreasing contact width for the same load 1540N in the range of line of action from −5mm to 3mm. Both the film profile and pressure profile are shown on Fig.4 for the line of action 1mm. There is a film constriction at outlet due to the elastic deformation of the surface and the pressure peaks just before the film constriction. The analysis results with the assumption of steady state are compared with those of non-steady analysis in Fig.5, which shows about 20% increase of non-steady film thickness.

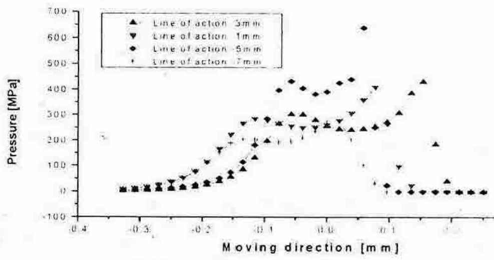

Fig. 3 Pressure profiles at center of tooth width

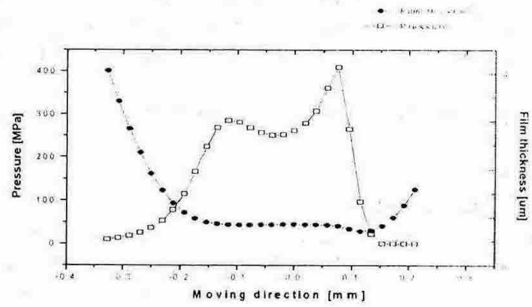

Fig. 4 Pressure and Film profiles at line of action 1mm

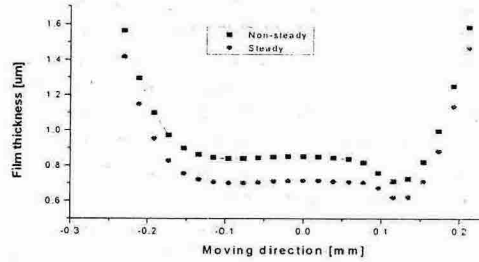

Fig. 5 Comparison of film profiles (Non-steady : Steady)

5. CONCLUSIONS

The analysis results showed high pressure at inlet region as well as pressure peak at outlet region and thicker film thickness than that of steady condition. This study may be utilized for the improvement of the gear tooth design like modification of tooth profile or profiling of the tooth edge.

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