Advanced Materials Research *Online: 2012-04-12*
ISSN: 1662-8985, Vol. 502, pp 351-354
doi:10.4028/www.scientific.net/AMR.502.351

Numerical Simulation of Water-Lubricated Rubber/Chromium Coating Pair Test

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Keywords: Water lubrication; Rubber/Chromium Pair; Elastohydrodynamic lubrication (EHL); Lubrication Mechanism.

Abstract. The elastohydrodynamic lubrication (EHL) geometric model of the water lubricated rubber bearings under standard experimental conditions with water lubrication was established. The complete numerical solution of the EHL model was obtained by using multi-grid technique. The results show that the EHL state can be only established under high relative velocity and mild load conditions between the rubber block and the chromium coating on steel with sufficient water. The numerical results are in good agreement with the experimental data.

Introduction

Water-lubricated rubber bearing is one of the most appropriate bearings for under water use. The bearings can be found in many applications, including stern tube bearings on ships, submarines and small craft, hydroelectric power plants and pumps. At present, much of the work has been done on the friction, wear and lubrication between the rubber liner and the steel shaft. Zhang et al [1] studied the friction and wear mechanism between rubber and steel. Wang [2] developed a new structure rubber bearing with water lubrication and analyzed the lubrication mechanism of water-lubricated rubber bearings. Duan [3] focused on the EHL analysis of water-lubricated rubber bearings. Wang [4] also studied the water lubrication mechanism of rubber and nickel coating pair because the shaft was often made of middle carbon steel with plated nickel or chromium for corrosion resistance. But so far relatively little work has been carried out on water lubrication mechanism of rubber and chromium coating pair. So, systematically studying hydrodynamic lubrication mechanism between rubber block and steel ring plated with chromium under water lubrication is a supplement of the research on lubrication theory of water lubricated rubber bearings, which provide a theoretical basis to the application of water lubricated rubber bearing and has important practical usage.

Geometric Model

The 45 # steel ring specimen plated with chromium coating was fixed in the rotating spindle in the ring piece of friction and wear tester. The rubber slide block specimen was settled and pressed by the load on the ring which is illustrated in Fig.1. Both the ring and the block were immerses in the water for lubrication. So the geometric model of the tribological test can be simplified as a line contact problem of the elastohydrodynamic lubrication.

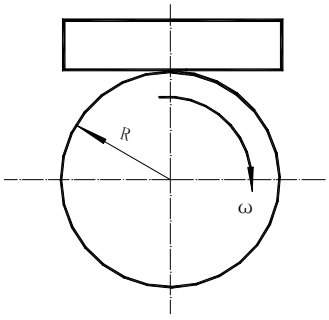


Fig.1 The EHL model of the ring/block tribological test

Governing Equations and Boundary Conditions

Reynolds equation. The Reynolds equation for steady isothermal line contact problem [5] can be written as follows,

$$\frac{\mathrm{d}}{\mathrm{d}x}\left(\frac{\rho h^3}{\eta}\cdot\frac{\mathrm{d}p}{\mathrm{d}x}\right)=12U\frac{\mathrm{d}(\rho h)}{\mathrm{d}x} \tag{1}$$

Where, p is the water film pressure (pa), h is water film thickness (m), η is the water dynamic viscosity (pa·s), ρ is the water density (kg/m^3), U is the entrainment velocity of the contact zone and $U=(u_1+u_2)/2$, where u_1 and u_2 are the line velocity which is tangential to the contact surface, i.e. u_1 is the line velocity of the rubber block and u_2 is the circumference velocity of the rotating ring. So $u_1=0$ and $u_2=\omega R$. The boundary conditions for the Reynolds equation is,

$$\begin{cases} p(x_{\mathrm{in}})=p(x_{\mathrm{out}})=0 \\ p(x)\geq 0 \left(x_{\mathrm{in}}<x<x_{\mathrm{out}}\right) \end{cases} \tag{2}$$

Where x_{in} and x_{out} is the inlet and outlet coordinate of the contact zone respectively.

Film thickness equation. Film thickness equation and the force balance equation read respectively as,

$$h(x)=h_0+\frac{x^2}{2R}-\frac{2}{\pi E'}\int_{-\infty}^{x^*}p(s)\ln(x-s)^2\,\mathrm{d}s \tag{3}$$

$$w=\int_{x_{\mathrm{in}}}^{x_{\mathrm{out}}}p\,\mathrm{d}x \tag{4}$$

The water is a special lubricant. The pressure and temperature have very little effect on water viscosity and density. Roelands viscosity-pressure relationship and Dowson-Higginson density-pressure relationship [6] were adopted in the paper.

In order to simplify the calculation and improve the stability of the numerical method, all the equations above should be disposed by dimensionless normalization. The dimensionless parameters are defined as follows, $X=x/b$, $\overline{h}=hR_0/b^2$, $P=p/p_{\mathrm{H}}$, $W_0=w_0/(ER_0)$, $U_0=\eta_0 u_0/(ER_0)$, $\overline{h}_0=h_0R_0/b^2$, $\overline{\eta}=\eta/\eta_0$, $\overline{\rho}=\rho/\rho_0$. where, w_0 is reference load; b semi-width of Hertzian contact under load w_0, $b=R_0\sqrt{8W_0/\pi}$; p_{H} the ultimate Hertzian pressure under load w_0, $p_{\mathrm{H}}=E\sqrt{W_0/(2\pi)}$; ρ_0, η_0 lubricant ambient density and viscosity respectively; R_0 reference curvature; u_0 reference velocity; h_0 rigid body central film thickness; h lubricant film thickness; E comprehensive elastic module.

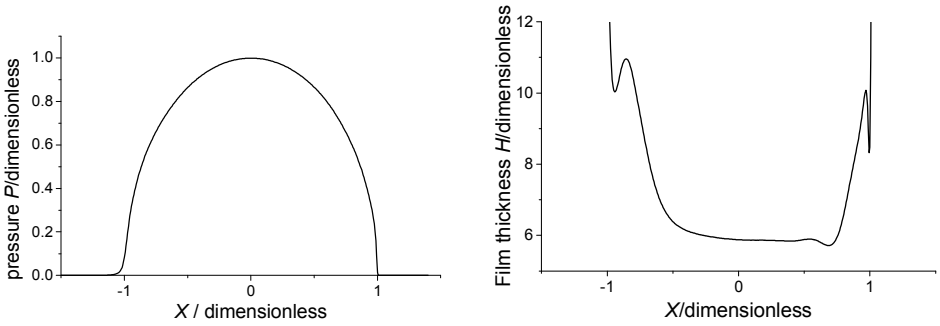
Numerical Methods

The governing equations were discretized in the calculated domain using finite difference method via having dimensionless disposal. The numerical solution of the dimensionless governing equations described above is carried out by the Gauss-Seidel iteration with their boundary conditions. The Reynolds equation is solved on a uniform grid by the finite difference method. To begin a solution a set of starting values of pressure is required, initially these values were given by the Hertzian contact theory. The multigrid technique has been utilized to improve the convergence rate. The elastic deformation is calculated by the multilevel multi-integration method [7]. Six levels of grids are used for the calculation of pressure and film thickness. The number of nodes on the finest level is 961 along the X direction. The real calculating domain of X direction for the infinite line contact is $X_{in}=-4.6$ and $X_{out}=1.6$. The convergent criterion for pressure and load are adopted as follows,

$$ERR_p=\sum_{i=0}^{n}\left|\widetilde{P}_i-\overline{P}_i\right| \Big/ \sum_{i=0}^{n}\overline{P}_i<0.001 \text{ and } ERR_W\leq 0.001$$

Results and Discussions

The parameters used in numerical calculation are as follows, water ambient viscosity $\eta_0 = 0.001$ pa·s, the Yang's elastic modulus of steel plated with chromium is 2.1×10^{11} pa, Possion's ratio is 0.3, and the Yang's elastic modulus of rubber used in bearing is 8.4×10^{6} pa, Possion's ratio is 0.47, so the comprehensive modulus is 2.17×10^{7} pa, the viscosity-pressure factor is $\alpha = 6.0\times10^{-9}$ pa^{-1}. The radius of curvature of the rubber block is infinite, and the radius of curvature of the steel ring plated with chromium is 0.02 m, so the comprehensive radius of curvature is $R = 0.02$ m. For convenient to comparison in analysis, all of the calculated results are given in dimensionless. The dimensionless pressure is expressed as $P=p/p_{\mathrm{H}}$ and p_{H} is the Hertz contact pressure. The dimensionless film thickness is signified as $H = 10^5 \times h/R$. Fig.2 shows the distribution of the dimensional pressure and film thickness when the experimental steel ring rotating speed is 360 r/min and the load on the rubber block is 196 N.

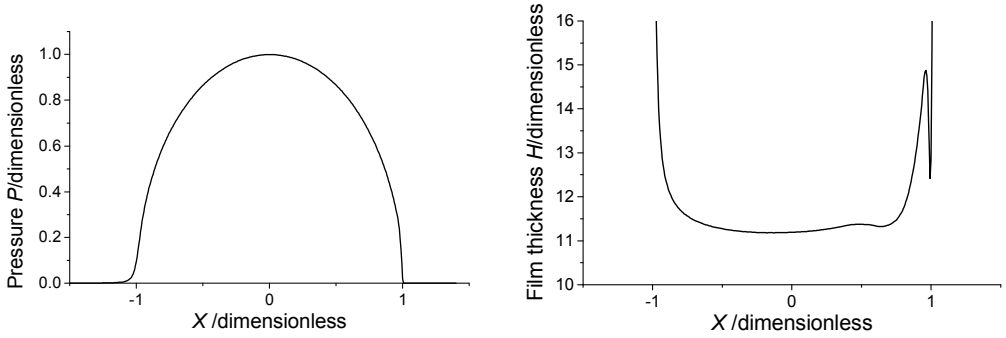


(a) pressure (b) film thickness

Fig.2 Pressure and film thickness profile distribution under water lubrication(velocity $\omega = 360\,\mathrm{r/min}$,load $w = 196\mathrm{N}$)

We can see that there is no obvious second pressure spike in the pressure distribution illustration in the contact zone from Fig.2(a). But there appears necking phenomenon in the outlet of contact zone in film thickness profile distributions from Fig.2(b). That is the typical features of elastohydrodynamic lubrication. So we deemed that the elastohydrodynamic lubrication was occurred in the contact zone between the rubber and the chromium coating. It is also shown from Fig.2(b) that there is undulation or sag in film thickness profile in the inlet and outlet of the contact zone. This is because the elasticity of rubber is so good that little pressure can lead to large compression deformation of film thickness.

In order to obtain the variation of the velocity with the pressure and film thickness, the pressure and film thickness profile distribution under water lubrication at different rotating speed and the same load was calculated numerically and plotted in Fig.3. The rotating speed of the steel ring plated with chromium was 720 r/min.



(a) pressure (b) film thickness

Fig.3 Pressure and film thickness profile distribution under water lubrication(velocity $\omega = 720\,\mathrm{r/min}$,load $w = 196\mathrm{N}$)

It can be seen from Fig.3 that the rotating speed have little effect on the pressure distribution large effect on the film thickness. When the rotating speed of the ring increased into double, the pressure increased very tiny, but the minimum film thickness became thicker and stable. The film undulation only occurred in the outlet of the contact zone. This is because with the increase of the rotating speed, the elastohydrodynamic lubrication is more apt to emerge in the contact zone.

In order to examine the effect of the load on the pressure and film thickness, to remain the velocity unchanged and decrease the load from 196N to 98N, the numerical results shows that there still exist elastohydrodynamic lubrication features. But if the load was increased up to 392N, or the rotating velocity was decreased to 100 r/min, the numerical program began to be unstable or non-convergence. The numerical results were good agreement with the experimental data. These suggested that the elastohydrodynamic lubrication could not be formed when the load was very heavy or the shaft rotating speed was very low even if the water was abundantly provided between the steel ring plated with chromium and the rubber block.

Summary

The elastohydrodynamic lubrication between the rubber block and steel plated chromium ring can be established under light load, high ring rotating speed and fully provided water. The numerical results show that there is no obvious second pressure spike in the pressure distribution in the contact zone, film thickness profile appears obvious necking in the outlet and undulation in the inlet and outlet of the contact zone. In the heavy load and low rotating speed region, it is very hard to establish stable elastohydrodynamic lubrication water film even if the water is sufficiently provided between the steel ring plated with chromium and the rubber block.

Acknowledgment

This work was financially supported by the Natural Science Foundation of P.R.China through contract no.51175275 and Taishan Scholar Special Fund of Shandong Province Government.

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