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Lubrication analysis and abrasion prediction of a cam-roller configuration

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Abstract：In this work, a cam-roller follower system is studied with its lubrication condition. First of all, we analyze kinematic parameters of whole diesel engine valve system via numerical equations. The kinematic analysis results include velocity and acceleration. The EHL model provide the parameters for the prediction model such as the minimum film thickness at the specified cam angle, the maximum film pressure, and the minimum film thickness distribution over all the cam angle. The last work in this paper is predicting working life according to the above results. The lubrication condition of the cam-roller follower system under actual working conditions is evaluated, and the actual working life is predicted via numerical solution. The results provide the basal dates for the design process of diesel engine valve cam. What's more the models adopted of this paper can be used to different cam-roller configurations.

1. Introduction

Valve mechanism is an important part of internal combustion engines and it has a direct impact on performance of internal diesel engine. In valve mechanism, the valve cam is the core that makes the valve mechanism act as we expected. We have higher requirements on fuel efficiency, reliability requirements, emission standards in modern design and manufacture process. That would mean higher demands for cam design. Looking at it from a tribological perspective, cuts and abrasions are more likely to occur on the interface because of the transient heavy dynamic loads. Therefore continuous and good lubrication is very essential. The rapid changing of curvature radius and the entrainment velocity of lubricating oil make tribological design more challenging except the heavy loads. In the analysis process of this paper, we assumed that the temperature condition is isothermal and non-significant. On this basis, we analyzed the minimum oil film thickness, maximum oil film pressure, and abrasion loss of valve cam.

2. Analyzing and modeling

The cam-roller configuration mentioned before belongs to a valve mechanism of a marine diesel engine. The roller and the roller guide bushing can rotate freely driven by the cam, all the interfaces are well lubricated by hydraulic bearings with low friction. This work mainly focuses on the interface between cam and roller where abrasion most likely to occur. Kinematic analysis of cam and roller are carried out in the first part of this section in order to study the friction and lubrication characteristics of the interface. Kinematic analysis presented by the following aspect, such as radius of curvature, entrainment velocity, cam-roller contact loads. The second section established the lubrication model of the system based on the linear contact non-newtonian fluid elastohydrodynamic model and analyzes the lubrication condition under the corresponding working conditions. In the third section, the life prediction model of cam-roller

configuration is given intended to prevent accidents.

2.1. kinematic analysis

The kinematic analysis models used in this work are programmed and solved by MATLAB software. The lift curve of the cam roller motion is obtained by the design drawings. The lift curve and cylinder pressure curve (measured data point with increment of 1 degree cam angle) are required as the input parameter in this analysis.

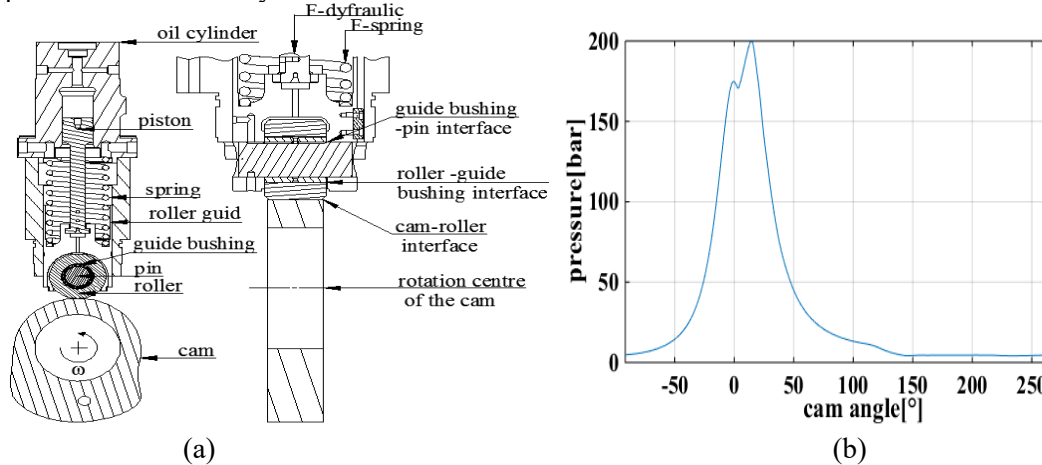
(a)   (b)

Figure 1. (a)Cam and roller follower configuration which we focus on its cam-roller interface. (b)Cylinder pressure curve with corresponding cam angle measured values with increments of half degree cam angle.

Curvature radius of cam theoretical profile can be expressed by polar coordinate expression

$$\rho_1' = \frac{(r^2 + \dot{r}^2)^{3/2}}{r^2 + 2\dot{r} + r\ddot{r}} = \frac{[r + (\frac{dr}{d\theta})^2]^{3/2}}{r^2 + 2(\frac{dr}{d\theta})^2 - r\frac{d^2r}{d\theta^2}} \tag{1}$$

Where r is radius vector of the cam, θ is cam angle. The actual radius of curvature of cams is

$$\rho_1 = \rho_1' - \rho_2 \tag{2}$$

In order to calculate the variation curve of the entrainment velocity which is required by elastohydrodynamic lubrication calculation, the entrainment velocity can be calculated by the following equation

$$U_m = \frac{U_{cam} + U_{roller}}{2} \tag{3}$$

The velocity of cam has relationship to the curvature radius of the cam and rotate velocity ω

$$U_{cam} = \rho_{cam} \cdot \omega_{cam} \tag{4}$$

Finally, the changing curve of the entrainment velocity can be obtained.

The contact force on the cam-roller interface contains kinds of loads. Such as spring force, inertia force, cylinder pressure transmitted by hydraulic circuit. At the same time, we simplified the roller and the moving parts behind the roller as single mass parts with the purpose simplify the calculation. Then the combined force can be calculated by the following equation

$$F_{totle} = F_P + F_S + F_A \tag{5}$$

Where

$$\begin{cases} F_A = M \cdot a \\ F_P = \dfrac{P_{in-cylinder}}{A_{valve}} \\ F_S = k \cdot S_{roller} + F_0 \end{cases} \tag{6}$$

With parameters defined as $P_{in-cylinder}$ is in-cylinder combustion pressure which was measured by experiment. A_{valve} is valve area, k is spring stiffness and F_0 is spring preload.

2.2. Elastohydrodynamic lubrication analysis

The elastohydrodynamic lubrication model used in this work stems from Huang Ping[1]. All variables and parameters involved in the calculation are in dimensionless form, cause dimensionless variables and parameters can simplify the process of numerical calculation. Here, the following dimensionless variables and parameters are shown as follows

$$\rho^* = \frac{\rho}{\rho_0}, \eta^* = \frac{\eta}{\eta_0}, P = \frac{p}{p_H}, X = \frac{x}{b}, H = \frac{hR}{b^2}$$

$$W_l = \frac{\eta_0 u}{ER}, Z = \frac{z}{h}, p_{H=} \frac{2w_l}{\pi b}, b = \sqrt{\frac{8 w_l R}{\pi E}} \tag{7}$$

Where ρ_0 and η_0 are initial density and viscosity of a lubricant respectively p_H is maximum Hertz contact stress h is lubricant film thickness R, E are relative radius of curvature and elasticity modus b is contact boundary w_l is linear load

Lubrication equation

$$\frac{d}{dX}(\varepsilon \frac{dP}{dX}) = \frac{d(\rho^* H)}{dX} \tag{8}$$

The dimensionally normalized film thickness equation

$$H(X) = H_0 + \frac{X^2}{2} - \frac{1}{\pi} \int_{X_0}^{X_e} \ln |X - X'| p(X') dX' \tag{9}$$

The viscosity of lubricating oil generally changes with the changing of pressure, so the viscosity pressure equation proposed by Roelands is used. Although the equation is complex, it is more accurate and more consistent with the actual condition

$$\eta^* = \exp\{[\ln(\eta_0) + 9.67][-1 + (1 + p_H P / p_0)^{0.68}]\} \tag{10}$$

The density of the lubricating oil also change with pressure

$$\rho^* = 1 + \frac{0.6p}{1 + 1.7p} \tag{11}$$

MATLB software was used to compile the program and compute the distribution of lubricating oil film and pressure through iterative calculation of the basic lubrication equation. An initial pressure distribution was given at first, then the elastic deformation of the contact surface and lubrication parameters of the oil film were computed. In this calculation process, the program will keep running until the results meet the iteration requirements. Finally, the distribution of lubricating oil film on the whole cam surface can be obtained.

2.3. working life prediction

After the previous analysis work as well as the problems encountered in the actual design, we obtained the lubrication analysis results of the cam surface. But we still need to establish the cam-roller abrasion loss calculation model and predict the actual working life of the cam-roller configuration. According to the calculation model of abrasion loss proposed by Archard[2]

$$dV = \frac{KP}{H} ds \qquad (12)$$

If we take abrasion loss **h** into account, the equation can be obtained after differentiation and modification

$$\Delta h_{pc} = \int_a^b K_C p_{PC}(x,y) \frac{|v_s|}{|v_{PE} - v_{PC}|} dx \qquad (13)$$

Where K_C is wear factor at contact point p_{PC} is maximum contact stress v_s is velocity of contact point v_{PE}, v_{PC} are tangential velocity and relative velocity respectively a, b are the boundary of Hertz contact zone.

At this moment, we assume that the contact stress of Hertz zone has constant effect of abrasion. After analyzing the lubrication system of the cam, the film thickness ratio is obtained and the relevant parameters are computed. The equation is derived to calculate the wear amount of the cam after a certain period of usage.

3. Results

In this section, the cam and roller follower configuration mentioned before is analyzed and the results are presented. First of all, we present the result of kinematics analysis include the cam and roller acceleration, velocity, and the loads on interface between cam and roller. The second part we study lubrication condition of cam-roller configuration and provide results which mainly include the film thickness, film pressure. All of the data will be used for life estimation. Finally, we give the life prediction of cam-roller under the above condition and result.

3.1. Kinematic analysis

We need the relative speed between cam and roller to carry on the elastohydrodynamic lubrication analysis. Entrainment velocity, accelerations and loads are obtained by kinematic analysis. We can see that the profile of the cam is asymmetric from the kinetic sketch. Therefore we need to analyze whole cam angle instead of half of them.
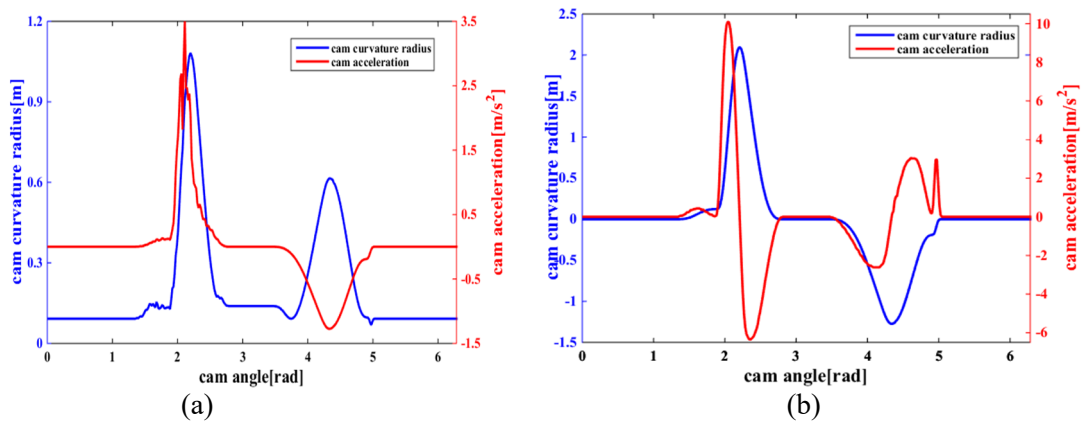

(a)   (b)

Figure 2. Radius of curvature, acceleration of roller a_{roller} and cam a_{cam}, velocity of roller v_{roller}

Cam-roller contact force can be determined by previous analysis, including cylinder pressure, spring force, and inertia force. The cylinder pressure does not generate pressure on the interface of the cam-roller when the relative angle of the roller is 0 degree during runtime. Hence $F_{cylinder}$ equal to 0. In any case, when the cam rotate speed is 100,140 and 180 rpm, the positive pressure on the cam-roller interface can be calculated.
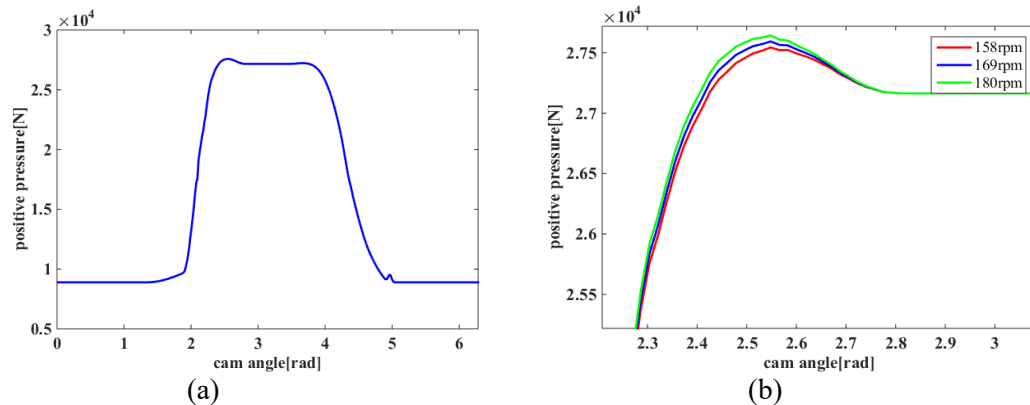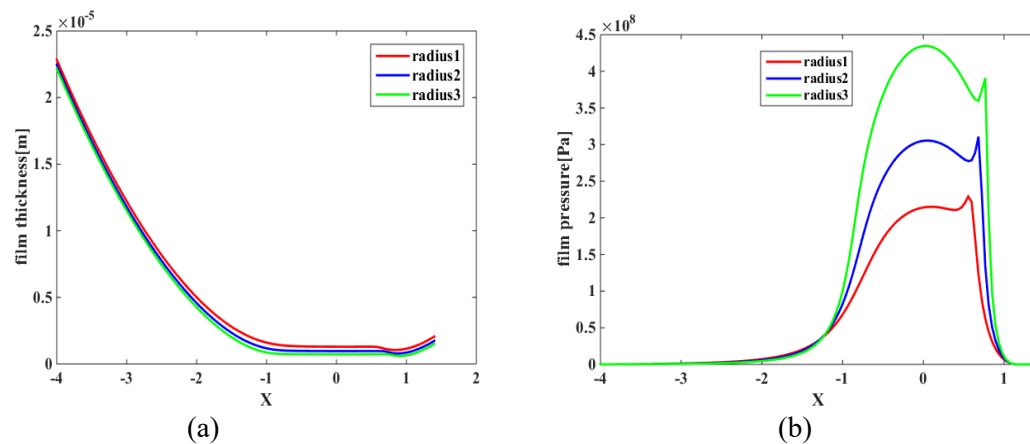
(a)   (b)

Figure 3.Positive pressure on cam-roller interface with different rotate speed.

In addition, the relations between the rotate velocity ω and loads F_{totle} are nonlinear. We can clearly see the incremental gaps between the curves. The increment between curves of 169rpm and 180rpm is different. With the rising of rotating speed, the increasing trend of load decreases gradually after the curve peak.

3.2. EHL

The analysis results of cam lubrication have great influence on the determination of the lubrication condition between cam and roller. The ratio of film thickness to roughness directly determines the condition of lubrication. The results provides the distribution of film thickness and pressure near the contact point at a certain angle of the cam.

(a)   (b)

(c)   (d)
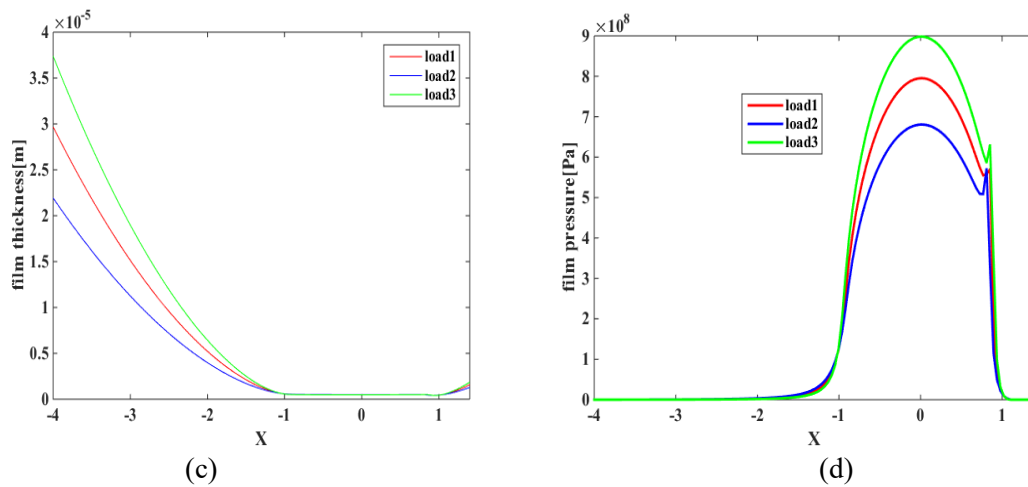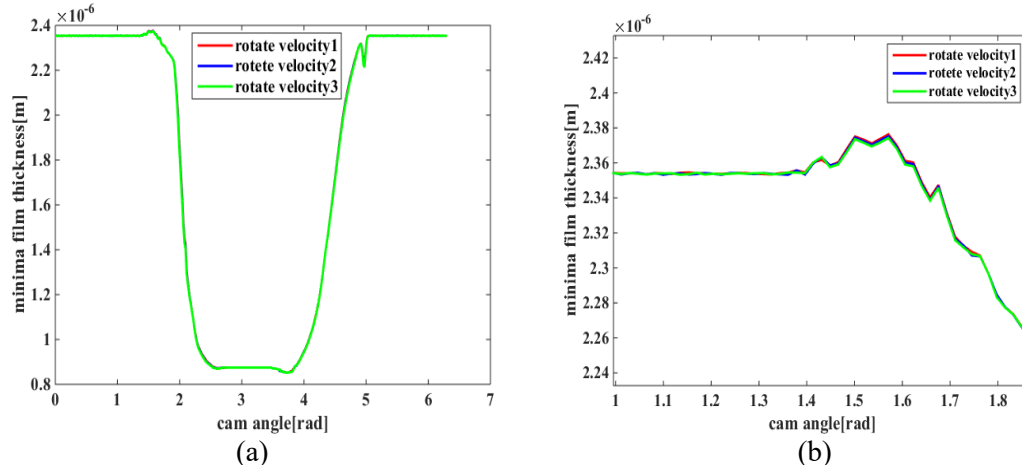
Figure 4. Distribution of film thickness and pressure near the contact point. Different color mean same contact point with different loads or different radius. Figures (a) and (b) study the distribution of film thickness and fil pressure near the contact point in different curvature radius under the same loading condition. Figures (c) and (d) are at the opposite.

We can see the effect of the changing radius or loads on film thickness and film pressure. The effect of radius on distribution of pressure is greater than that of loads. But on distribution of film thickness, this two parameters are opposite. We can analyze the distribution of film thickness and pressure to the entire cam profile. All date point will be used to estimate the abrasion of the profile. It should be pointed out that location of contact point is X=0. In this analysis, roughness of cam and roller configuration are 0.4 micron, so we can work out that the lubrication condition of cam-roller contact points at various points is based on film thickness ratio. At the same time, according to the model established by the analysis earlier, the abrasion loss of the cam can be predicted.

The theoretical analysis and numerical estimation above are all in ideal conditions, which may be somewhat different from the actual conditions, but it still can provide preliminary design advices for the designers.

(a)   (b)
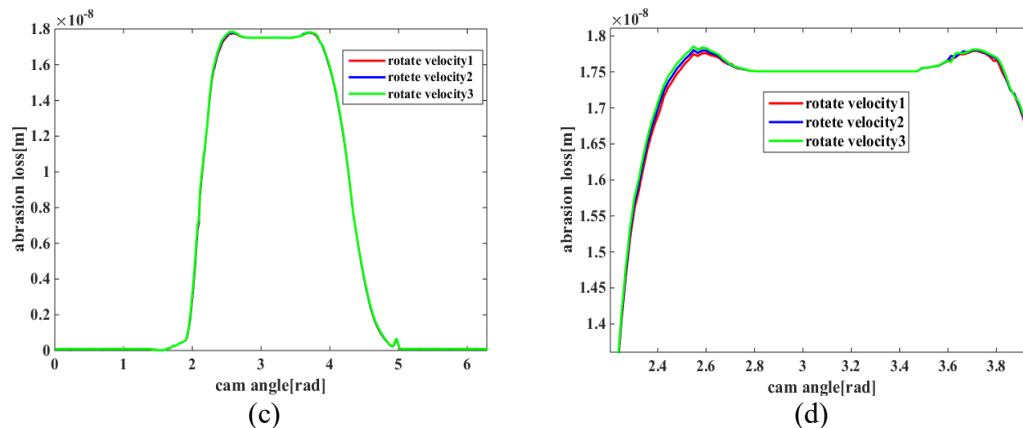
Figure 5. Distribution of minima film thickness and abrasion loss at different cam angle with different rotate velocity. Figure (a) and (c) show the holistic distribution. Figure (c) and (d) are magnified picture of corresponding distribution.

4.   Conclusion

In this work, a cam-roller configuration was analyzed which we focus on its kinematic analysis, EHL model analysis and prediction model calculation. For the kinematic analysis, we can found that curvature radius doesn't change smoothly which may lead to abrupt changes in contact stress. This part will have a great influence on life prediction. On the other hand, the relation between loads and speed is nonlinear. For the EHL model analysis, it can be concluded that the film thickness should be thinner to increase the carrying capacity with loads increment at the same curvature radius. Cam-roller contact pair runs in mixture lubrication mostly, but sometimes it runs in boundary lubrication which is detrimental to the life of cam-roller configuration. The result of lifetime prediction under lubrication have indicated that the trend of abrasion loss changing with loads and rotate speed accords with the above prediction model. And all the model can be used to other similar configurations, the mathematical analysis model and MATLAB program have some openness and universality.

Reference

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