# Mixed Lubrication Analysis of Cam/Tappet Interface on the Direct Acting Type Valvetrain System

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This paper reports on the mixed lubrication characteristics between the cam and the tappet contact surface of direct acting type valve train systems. First, the dynamic characteristics are solved by using the lumped mass method to determine the load conditions at the contact point. Then, the minimum oil film thickness is calculated with consideration of elastohydrodynamic line contact theory and the friction force is obtained by using the mixed lubrication model which separates the hydrodynamic and the boundary friction. Finally, the average surface temperatures are calculated by using the flash temperature theory. The results show that, there are some peaks in the friction force due to the asperity contact friction, and flash temperature at the position of minimum oil film thickness. It is thought that there is a relationship between the surface temperature and cam surface wear, and therefore, the analysis on the worn cam profile has been performed.

**Key Words:** Cam, Tappet, Valvetrain, Mixed Lubrication, Temperature, Wear

## Nomenclature

$A$ : Contact area, $m^2$
$A_c$ : Asperity contact area, $m^2$
$b$ : Half width of contact rectangle, m
$C$ : Damping coefficient, Ns/m
$E'$ : Reduced Youngs modulus, Pa
$F$ : Total friction force, N
$F_b$ : Asperity contact friction force, N
$F_h$ : Hydrodynamic friction force, N
$h$ : Nominal oil film thickness, m
$h_{cen}$ : Central oil film thickness, m
$h_{min}$ : Minimum oil film thickness, m
$K$ : Spring coefficient, N/m
$K_c$ : Thermal contact coef. of cam, $Ws^{1/2}/m^2$ ℃
$K_t$ : Thermal contact coef. of tappet, $Ws^{1/2}/m^2$ ℃
$L$ : Length of contact rectangle, m
$M$ : Mass of each element, Kg
$m$ : Coefficient of lubricant-limiting shear stress-pressure relation
$p_b$ : Asperity load per unit area, $N/m^2$
$p_h$ : Hydrodynamic load per unit area, $N/m^2$
$p_{max}$ : Maximum Hertzian stress, Pa
$R'$ : Reduced radius of curvature, m
$R_0$ : Radius of base circle, m
$R_c$ : Radius of curvature of cam, m
$R_t$ : Radius of curvature of tappet, m
$S$ : Lift of cam, m
$T$ : Average surface temperature, ℃
$T_{in}$ : Inlet temperature of working fluid, ℃
$V_c$ : Velocity of cam surface, m/s
$V_e$ : Entrain velocity, m/s
$V_s$ : Sliding velocity, m/s
$V_t$ : Velocity of tappet surface, m/s
$W$ : Total contact load, N
$W_b$ : Asperity contact load, N

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$W_h$ : Hydrodynamic contact load, N
$\alpha$ : Pressure coefficient of viscosity, $m^2/N$
$\beta$ : Radius of asperity tip, m
$\gamma$ : Pressure coefficient of boundary shear strength
$\delta$ : Temperature coefficient of viscosity, 1/℃
$\eta$ : Number of asperity per unit area, $1/m^2$
$\theta$ : Rotational angle of cam, rad
$\mu$ : Lubricant viscosity, Pa.s
$\mu_0$ : Lubricant viscosity at ambient viscosity, Pa.s
$\sigma$ : Standard deviation of asperity height, m
$\tau_0$ : Boundary shear strength of ambient pressure, $N/m^2$
$\tau_b$ : Boundary shear stress, $N/m^2$
$\tau_h$ : Hydrodynamic shear stress, $N/m^2$
$\omega$ : Angular velocity, rad/s

## 1. Introduction

In recent years there has been a tendency to design engines to be small and light weight in order to increase fuel efficiency. These trends have resulted in adopting higher valve lift and variable valve system, and usage of low viscosity engine oil. According to these situations, the operating conditions of valve train systems have become more and more severe. Such changes bring about a failure in valve component and deteriorates the engine durability. Therefore, the investigation of the tribological characteristics of these systems is important in improving their performance.

As a result, theoretical prediction and experimental measurement of the lubrication characteristics of cam and tappet interface has been performed extensively. Staron and Willermet (1983) presented the analytic model for the friction loss on the rocker arm type valve train system, and Helden et al. (1985) investigated method to measure the friction at the cam and tappet interface. Also, Crane and Meyer (1990) proposed the various approximate equations to calculate the friction force on the center pivot type OHC (overhead cam) valve train system. Ji et al. (1998) has performed the theoretical study for the effects of design parameters on the friction loss. Besides, much research has been performed to examine the effect of tappet rotation on the friction loss (Gecim, 1992; Kim, et al., 1998; Monteil, et al., 1996; Pieprzak, et al., 1992).

This paper reports the theoretical model to understand the mixed lubrication characteristics at the cam and tappet interface of a direct acting type OHC valvetrain system. This model includes kinematics, dynamics, asperity contact, Hertzian stress, oil film thickness, friction, and surface temperature analysis. Using the above model, we have investigated the main effects of cam surface failure, and performed comparative analysis between the designed and worn cam profile. We believe that this study is very useful for the initial design of valvetrain system.

## 2. Theory

### 2.1 Kinematics and dynamics analysis

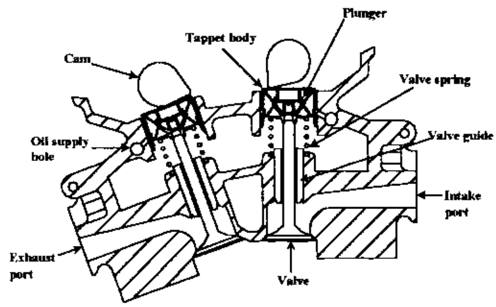



**Fig. 1** Schematic diagram of OHC valvetrain system equipped with HLA

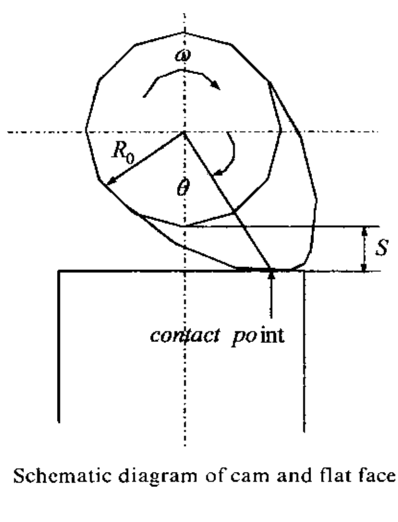



**Fig. 2** Schematic diagram of cam and flat face tappet

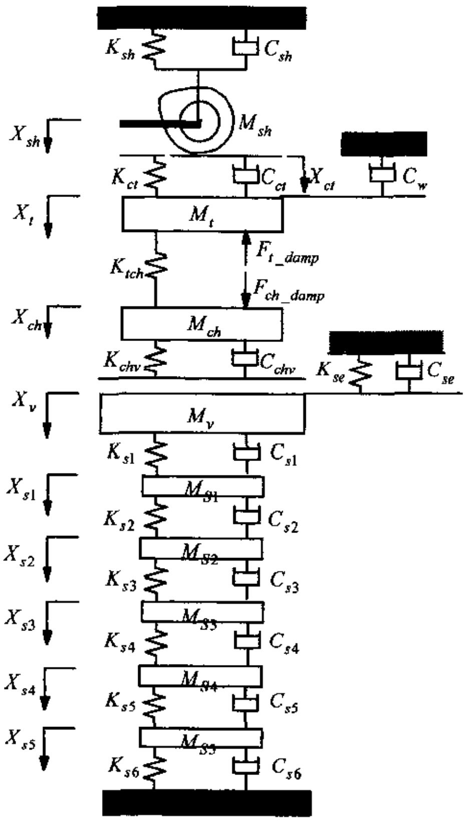



**Fig. 3** Dynamic model of valvetrain system

Figure 1 shows the schematic diagram of OHC valvetrain system used in this study. The tappet used in this study is equipped with a hydraulic lash adjuster. Figure 2 shows the contact configuration of cam and tappet for the kinematic analysis. In Fig. 2, the radius of curvature of the cam at the contact point is defined as follows:

$$R_c = R_o + S + \frac{d^2S}{d\theta^2} \tag{1}$$

Therefore, the relative velocities of cam and tappet at the contact point are written as:

$$V_c = \omega(R_0 + S)$$
$$V_t = 0 \tag{2}$$

At the contact point, the equivalent radius of curvature, entrain velocity, and sliding velocity are defined as follows:

$$\frac{1}{R'} = \frac{1}{R_c} + \frac{1}{R_t} = \frac{1}{R_c} \tag{3}$$

$$V_e = \frac{\omega}{2}\left(R_o + S + 2\frac{d^2S}{d\theta^2}\right)$$
$$V_s = V_c - V_t \tag{4}$$

Figure 3 shows the dynamic model of the valvetrain system for the analysis of acting force at the cam and tappet interface. The valvetrain system is divided into 9 lumped mass elements, and the HLA chamber and plunger are considered as nonlinear dampers. The valve spring is divided into five mass elements to reflect the spring surge phenomenon. The dynamic equation of the above model is represented as a matrix form, which is presented as follows:

$$[M][\ddot{X}] + [C][\dot{X}] + [K][X] = [f] \tag{5}$$

The solutions of Eq. (5) are obtained by using the numerical integration method.

### 2.2 Mixed lubrication analysis

If the acting forces at the cam and tappet interface are calculated using Eq. (5), the contact area and maximum contact pressure are calculated through the Hertzian line contact theory. The minimum and central oil film thicknesses between the cam and the tappet under the isothermal condition are calculated by using the elastohydordynamic theory, which was previously presented by Dowson et al. (1986).

$$h_{min} = 2.6R'\left(\frac{\mu V_e}{E'R'}\right)(\alpha E')^{0.54}\left(\frac{W}{E'R'l}\right)^{-0.1} \tag{6}$$

$$h_{cen} = 3.06R'\left(\frac{\mu V_e}{E'R'}\right)^{0.69}(\alpha E')^{0.56}\left(\frac{W}{E'R'l}\right)^{-0.1} \tag{7}$$

For the mixed lubrication analysis, we make the following assumption. The friction between the cam and the tappet is composed of viscous friction due to hydrodynamic and boundary friction due to asperity contact. According to the asperity contact theory presented by Greenwood and Tripp (1971), the asperity contact load acting through the elastically deformed asperity is defined as:

$$V_b = \frac{8\sqrt{2}}{15}\pi(\eta\beta\sigma)^2 E'\sqrt{\frac{\sigma}{\beta}}AF_{\frac{5}{2}}\left(\frac{h}{\sigma}\right) \tag{8}$$

In Eq. (8), the upper limit of the asperity contact load, $W_b$, is the acting force between the cam and the tappet, $W$. If we assume Gaussian distribution for the surface roughness, the function $F_{5/2}\left(\frac{h}{\sigma}\right)$ in Eq. (8) is defined as follows:

$$F_{\frac{5}{2}}\left(\frac{h}{\sigma}\right)=\frac{1}{2\sqrt{2\pi}}\int_{\frac{h}{\sigma}}\left(s-\frac{h}{\sigma}\right)^{5/2}\exp\left(-\frac{s^2}{2}\right)ds \quad (9)$$

The asperity contact area, $A_c$, is written by:

$$A_c=\pi^2(\eta\beta\sigma)^2AF_2\left(\frac{h}{\sigma}\right) \quad (10)$$

In Eqs. (8), (9), and (10), the central oil film thickness between the cam and the tappet is adopted as the nominal oil film thickness $h$. Therefore, the friction force due to boundary lubrication is induced by shear of thin film and asperity contact. In this case, the shear stress and friction force are defined as follows:

$$\tau_b=\tau_0+\gamma p_b \quad (11)$$

$$F_b=\tau_0A_c+\gamma W_b \quad (12)$$

However, the shear stress due to hydrodynamic lubrication is induced by sliding movement of the cam and the tappet, which is defined as:

$$\tau_{hh}=\frac{\mu}{V_s}h_{cen} \quad (13)$$

In Eq. (13), $\mu$ is the viscosity of the working oil, which is the function of pressure and surface flash temperature.

$$\mu=\mu_0\exp[\alpha(p_h)-\delta(T-T_i)] \quad (14)$$

If the oil film is very thin, the shear stress becomes extremely high. Therefore, the shear stress cannot be calculated by using Eq. (13). In this case, the shear stress can be calculated by using the limit shear stress theory (Rohde, 1981), which can be written as:

$$\tau_{hb}=\tau_0+\gamma p_h \quad (15)$$

$$\tau_L=\tau_0+mp_h \quad (16)$$

where

$$p_h=(W-W_b)/(A-A_c) \quad (17)$$

Therefore the shear stress under the hydrodynamic lubrication condition is defined according to the oil film thickness, and the viscous friction force is calculated by numerical integration of Eq. (18). The definition of Staron and Willermet (1983) is used in the classification

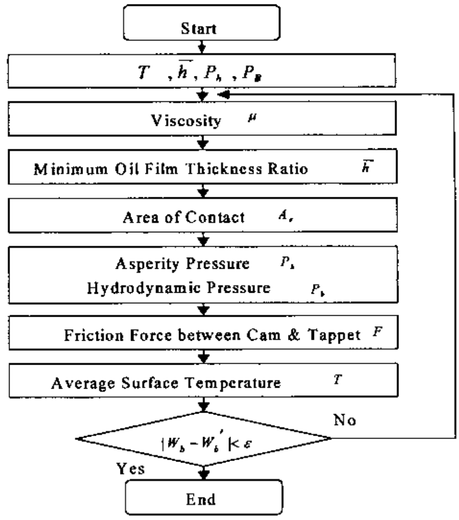



**Fig. 4** Flow chart for the mixed lubrication analysis

of lubricating condition.

$$\tau_h=\begin{bmatrix} \tau_{hb}, & h\le h_1 \\ \frac{h-h_1}{h_2-h_1}\tau_{hb}+\frac{h_2-h}{h_2-h_1}\tau_{hh}, & h_1\le h\le h_2 \\ \tau_{hh}, & h\ge h_2 \end{bmatrix} \quad (18)$$

The total friction force is the sum of viscous and boundary friction forces, and that is written as.

$$F=F_b+\tau_h(A-A_c) \quad (19)$$

The temperature rise in the contact surface due to the shearing of oil film and asperity contact is estimated based on the flash temperature concept, and the average surface temperature of the cam and the tappet can be defined as follows (Gecim, 1992):

$$T=T_i+1.064\frac{FV_s}{A}\frac{\sqrt{b}}{K_c\sqrt{V_c}+K_t\sqrt{V_t}} \quad (20)$$

The flow chart of calculation procedure is shown in Fig. 4. Table 1 shows the specification of the test valvetrain system and the properties of the working fluid.

## 3. Results and Discussion

Figure 5 displays the cam lift curve and equivalent radius of curvature used in the analysis.

Figure 6 shows the result of acting force and minimum oil film thickness between the cam and

**Table 1** Input data for the valvetrain

| | |
|---|---|
| Base circle radius (mm) | 18 |
| Cam width (mm) | 14 |
| Reduced Youngs modulus (GPa) | 165 |
| Composite surface roughness ($\mu$m) | 0.4 |
| Product of ($\eta\beta\sigma$) | 0.056 |
| Ratio of ($\sigma/\beta$) | 0.001 |
| Thermal contact coef. of cam ($Ws^{1/2}/m^2$ ℃) | $1.26\times10^4$ |
| Thermal contact coef. of tappet ($Ws^{1/2}/m^2$ ℃) | $126\times10^4$ |
| Oil supply temperature (℃) | 120 |
| Dynamic viscosity at 120℃ (Pa.s) | 0.0057 |
| Boundary shear stress of lubricant (MPa) | 2 |
| Rate of change of shear stress with pressure | 0.08 |
| Limiting shear stress-pressure relation | 0.17 |
| Temperature coef. of viscosity (1/℃) | 0.01 |
| Pressure coef. of viscosity (1/GPa) | 14.3 |

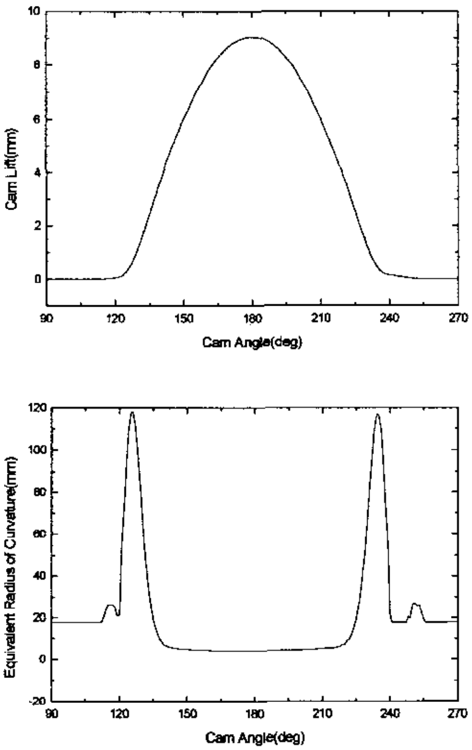



**Fig. 5** Cam lift curve and equivalent radius of curvature

the tappet at 1500 rpm. The variation of acting force is due to the dynamic characteristics of the spring surge phenomenon. The minimum oil film thickness is not formed at the maximum force area. It is thought that the oil film thickness is

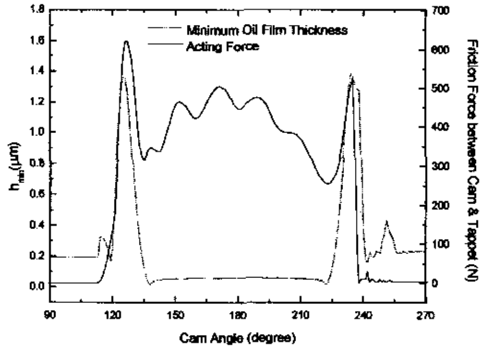



**Fig. 6** Calculation results of acting force and minimum oil film thickness at 1500crpm

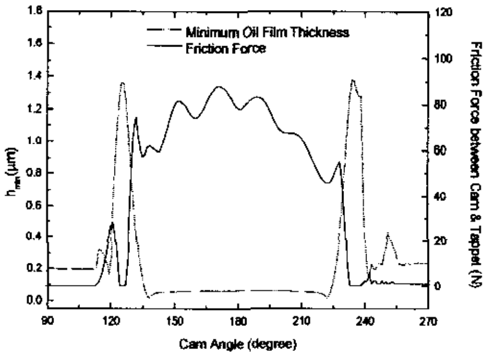



**Fig. 7** Calculated results of minimum oil film thickness and friction force at 1500crpm

not affected by the acting force but by the entrain velocity according to the equivalent radius of curvature.

Figure 7 shows the calculated result of the minimum oil film thickness and the total friction force. The maximum friction force occurs at the cam nose area, where the film thickness is relatively large. The abnormal peak in the acting force occurs at the relatively thick oil film position, but the peak is shifted to the direction of the minimum oil film area.

Figure 8 shows the friction components of the total friction force at 1500rpm. The degree of boundary friction force in the total friction is about 10%. The abnormal peak in the asperity contact friction force occurs at the minimum oil film thickness position. This peak increases the peak of the total friction at the valve opening area.

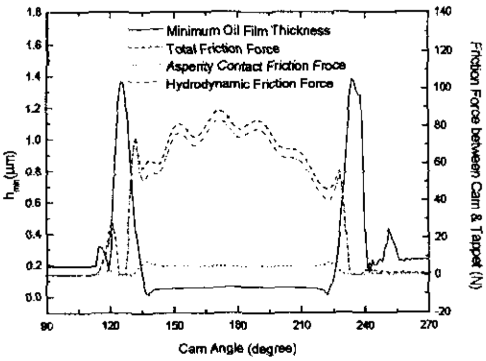
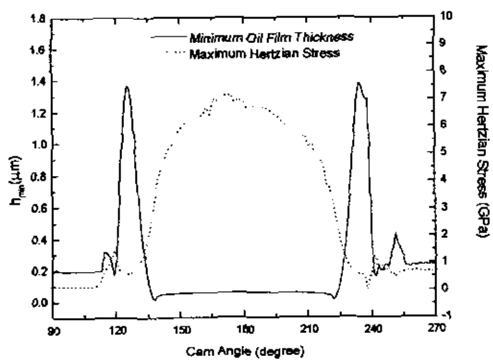
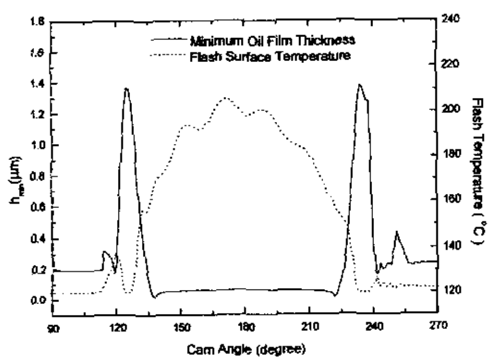



**Fig. 8** Calculated results of minimum oil film thickness and friction force at 1500crpm




(a) Maximum Hertzian stres



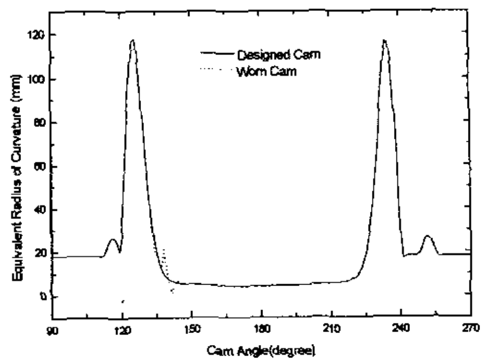

(b) Surface temperature

**Fig. 9** Calculation results of max. Hertzian stress and average surface temperature

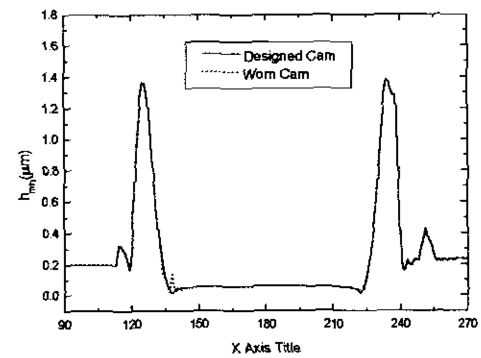
The maximum Hertzian contact stress and the average surface temperature variation at the cam and tappet interface are shown in Fig. 9. The maximum stress and surface temperature occur at the cam nose area.

In the analytical results, the maximum contact stress occurs at the cam nose. However, in the operating engine the failure of cam surface occurs at the flank area, and then it propagates towards the nose area. From this fact, we deduce the following point. It is thought that the main factor of initial wear in the cam surface is not the maximum stress due to the acting force but it is the abnormal friction force due to minimum oil film thickness. This phenomenon was examined by experiment by Soejima et al. (1995). To understand the propagation of cam surface failure, an analysis on a cam with a worn cam profile is performed. It is assumed that certain amount of polishing wear occurs from -10° to 10° range with respect to the minimum oil film thickness position on the designed cam profile.

Figure 10 shows the comparative result of the equivalent radius of curvature and the minimum oil film thickness for the worn and the designed




(a) Equivalent radius of curvature




(b) Minimum oil film thickness

**Fig. 10** Comparison of equivalent radius of curvature and minimum oil film thickness between the designed and worn cam profile

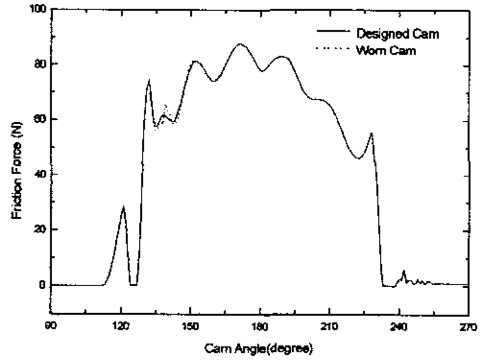



**Fig. 11** Comparative results of total friction force between designed and worn cam

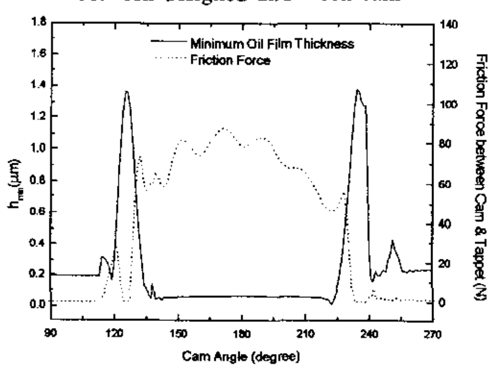
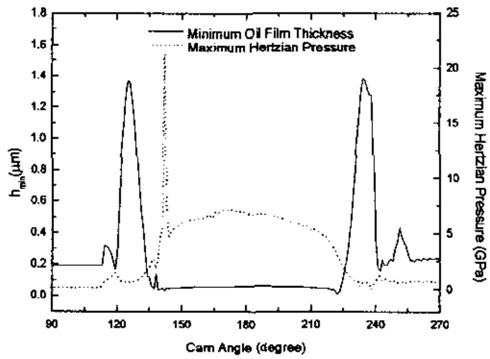
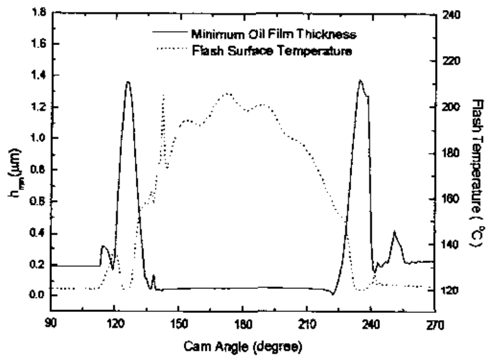



**Fig. 12** Calculated results of minimum oil film thickness and friction force in the worn cam profile




(a) Maximum Hertzian stress




(b) Average surface temperature

**Fig. 13** Comparative results of max. Hertzian stress and surface temperature in the worn cam

cam profiles. The polishing of cam surface changes the radius of curvature at the worn area. In Fig. 10, the position of minimum oil film thickness is changed due to the variation in the radius of curvature.

The comparative result of the total friction force between the designed and the worn cam profiles is shown in Fig. 11. The extraordinary peak point in friction is increased in the worn cam at the altered minimum oil film thickness position (see Fig. 12).

Figure 13 shows the maximum Hertzian contact stress and surface temperature of the worn cam. The stress is abruptly increased at the minimum oil film position, and the average surface temperature is also increased at the same position. From the above results, if the slightly polishing wear occurs at the cam surface, the stress and temperature are concentrated at that point. Therefore it is thought that the initial wear of the cam surface occurs at that point, and then it propagates along the cam surface to the nose area.

## 4. Conclusion

From the mixed lubrication analysis of the cam and tappet interface, the following results are derived.

(1) The ratio of the asperity contact friction is about 10% of the total friction force.

(2) It is thought that the initial wear occurs at the abnormal peak point in friction force in the minimum oil film thickness region, and it propagates to the nose direction.

(3) Modification of the cam profile by initial wear at the abnormal point changes the position of the minimum oil film thickness, and the friction force, surface temperature, and contact stress become concentrated at that point.

(4) It is thought that the above analytical model is a useful tool to verify and enhance the tribological performance of designed valvetrain system.

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