DEPARTMENT OF MECHANICAL & INDUSTRIAL ENGINEERING OF NORTHEASTERN UNIVERSITY

Final Reports of Engine Tribology

ME5656 Mechanics of Contact & Lubrication

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Our group is studying on internal combustion engine tribology, and through reading, we found that Cylinder-piston ringfrictionswidely used nowadays, hundreds of millions of engine uses thisfriction pair, it is one of the most important friction pair of internal combustion engine. Researches show that, the 25% ~ 50%friction work of internal combustion engines consumed by the cylinder-piston ring friction pair, this means the tribological properties quality of this friction pair directly affect the efficiency of internal combustion engine, service life and emission characteristics. Cylinder-piston ringfriction pair lubrication analysis isone of the most important tribological application areas, for many years lots of researches are done around the world, and many kinds of lubrication theory model are formed.

The key problem is how to make engines work more effective and efficiency, and so that the cylinder-piston ringfriction pair is the top priority. Part I made a more theoretical study and Part II gives us some real tests.

I. Experimental analysis of friction and wear characteristics for cylinder liner piston ring of engine

With the supercharged engine, the friction pairs of the engine's friction and wear problems become more prominent. Because the harsh conditions of the working engine, the cylinder - piston ring system is one of the most possible parts to get failure. In addition, now more and more design towards to the heavy, high-speed direction.

It is very important to make the cylinder - piston ring systems maintain high-performance in high-temperature, high-speed and overloaded conditions. Friction and lubrication to the system becomes a very important role for the engine. Therefore, the tribological behavior of the system analysis research is crucial not only for engines serve life but also for the reliability and economy of usage.

Friction and wear are not intrinsic material properties, but the comprehensive reflection of the tribological system by speed, temperature, materials and conditions and other factors. According to the similarity of friction and wear tests in the same principles to the design of the sample tests under laboratory conditions, we can simulate actual working conditions [5].

In this paper, under the conditions of temperature we focus on the system friction characteristics of piston ring simulation experiment to study the cylinder - piston ring relative motion during sliding speed, load changes, temperature changes on the system friction.

1. Structure and principle of test-bed

Test equipment contains the power and transmission system, fuel supply systems, heating systems, cylinder - piston ring simulation work system, test and data acquisition system. Structural diagram of test equipment is shown in Figure 1.

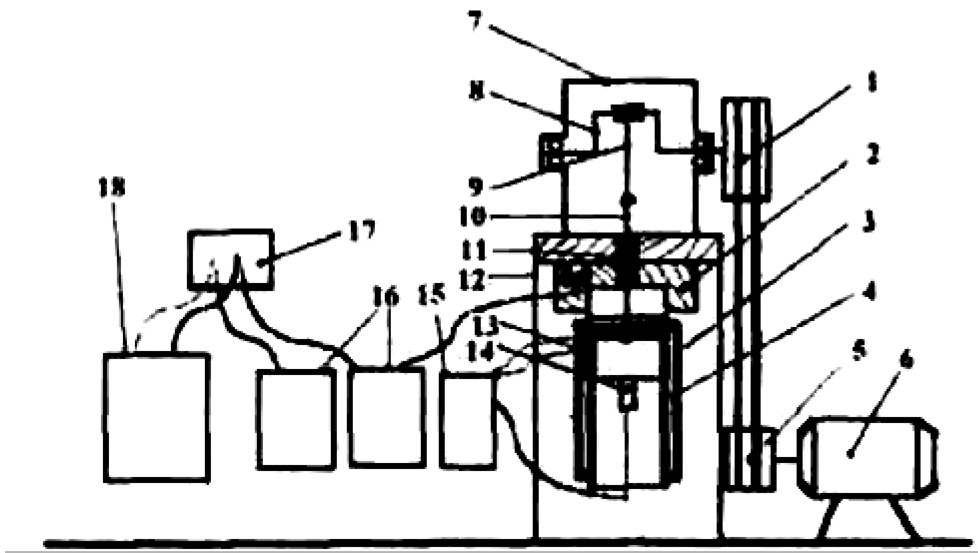


Fig.1 Structural diagram of test equipment

1. Great belt pulley; 2.flange plate; 3.heater; 4.Cylinder liner; 5.small pulleys; 6. DC motor; 7. Case; 8.crank; 9 connecting rod; 10. Slider bar; 11.Slider bar liner; 12. Pressure sensor; 13.Temperature sensor; 14.Velocity sensor; 15.Temperature controller; 16.charge amplifier; 17. Data Acquisition Card; 18.Computer.

2. Experimental results and analysis

Cylinder-piston ring friction system is a complex system. Its wear properties are influenced by multiple factors such as working temperature, loading, working speed and material lubrication. The following were investigated temperature, working speed and loading conditions affecting on the cylinder-piston ring system's lubrication condition and friction and wear characteristics.

2.1 Temperature to the cylinder - piston ring affecting friction characteristics

First consider the friction with different temperature changes. Figure 2 shows the relationship between temperature and friction by heating. After the initial running-in (Figure.2, 0 ~ 400 samples), the temperature rise directly to 200 ℃, after 1h the temperature continues to rise to 250 ℃.

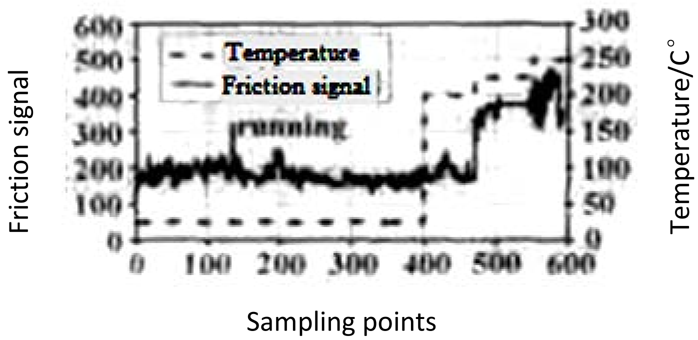


Fig. 2 The relationship between temperature and friction by heating

The figure shows, below 200 ℃, little change the temperature rises to 200 ℃, and 225 the friction increased rapidly (Figure2, 400 ~ 550 samples), the reason is that with increasing temperature, lubricant viscosity and the thickness of fluid lubrication decreased. After the temperature rise to 250℃, the friction increased further, and with large fluctuations (Figure 2, 550 ~ 600 samples). After experiment we found that the cylinder's surface sticky black residue, analysis as lubricating oil at high temperature evaporation and the residual material after carbonization. Figure 3 shows that using scanning electron microscopy (sem) to observe the surface of cylinder liner and piston ring after test [6]. The pictures show that a further increase in friction is due to the high temperature evaporation and carbonization of lubricating oil supply is not sufficient between the contact areas.

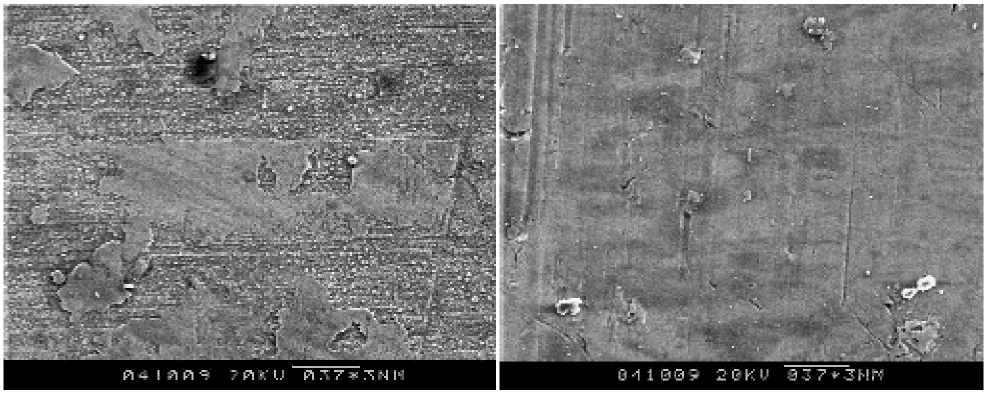


Fig.3(a) Local adhesion of the cylinder liner 3(b) The metal peeled piston ring surface

2.2 Working speed of the cylinder affecting piston ring friction characteristics

Figure 4 shows a typical friction-speed curve in experiment. When the piston move to the cylinders up and down end point, the piston has the lowest moving speed, the friction value is relatively large; when the piston move to the middle, that piston has the highest speed and it has less friction, and so forth.

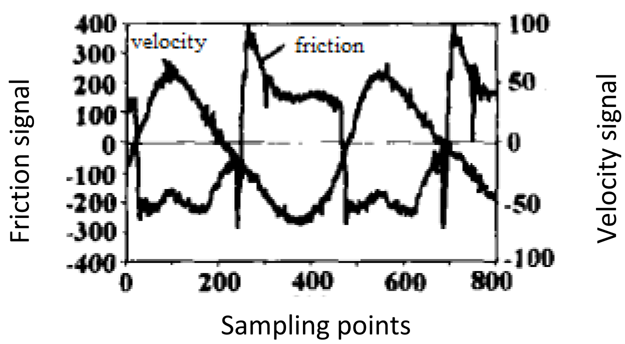


Fig. 4 Friction-speed curve

This indicates that the cylinder - piston ring friction system is a complex system, the piston at the up and down end point in the relatively low speed, lubricant film is difficult to form, while the piston is changing move direction, suffered and alternating load inertia is also large, in mixed lubrication, friction and wear caused the most serious here; in the middle of stroke, the cylinder piston speed is relatively large, in the fluid lubrication state, it has small coefficient of friction, friction and wear is also very small, it has a good working condition.

Figure 5 gives the change of friction under different rotating speed from single cylinder testing engine. Temperature fixed at 200℃and a 417N loading. From the figure we can see that the there is no apparent effect on friction for different rotating speed in the testing speed range.

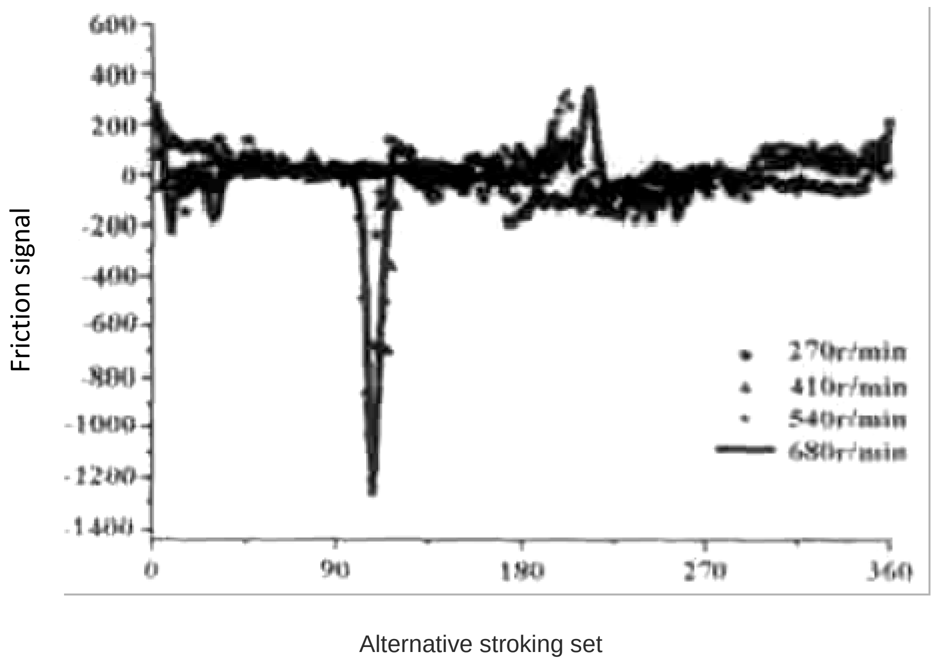


Fig.5 Relationship between friction and rotating speed

2.3 Loading on the cylinder - piston ring affecting friction characteristics

Test in spring with different stiffness installing in the piston, thus changing the section of the cylinder piston load. At a fixed temperature of 250℃ and speed 270r / min, to ensure adequate oil supply, repeat cyclic loading test. Figure 6 shows the impact of speed on friction. The figure 6 shows when the cylinder - piston ring system's loading increased, the system friction increased slightly.

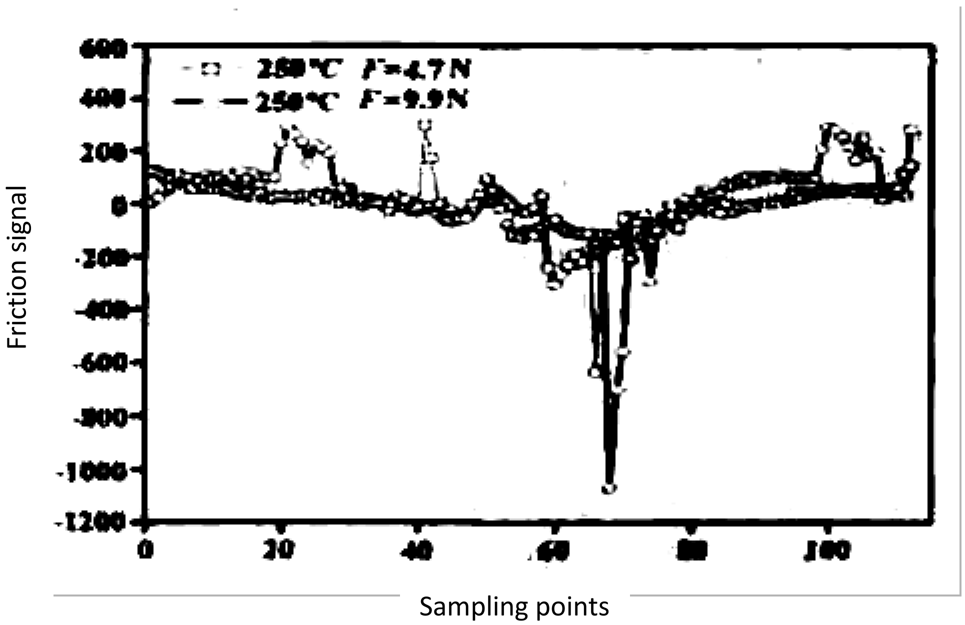


Fig.6 Relationship between friction and load

3. Conclusion

On the single-cylinder engine testing machine, with the cylinder piston ring friction and wear experimental research, testing showed that:

The state of cylinder piston-ring friction in the engine lubrication work cycle is constantly changing. When the piston is in the upper and lower stop point position, friction in mixed lubrication state, and in the middle of the stroke system is completely fluid lubrication.

Temperature of the cylinder piston ring friction pairs has a significant effect. As the temperature increases, a significant reduction in the viscosity of the lubricant and constant evaporation and carbonation, making the lack of friction work in the oil states, a significant increase in friction, wear and even violent-prone glue failure. Under the experimental conditions, load and speed on friction less affected.

II. Theoretical analysis of friction and mixed lubrication for piston ring-cylinder system of engine

Piston ring-cylinder is one of the most important parts of friction components and its lubrication property has a great effect on the performance of the engine. In the condition that well sealed of combustion chamber, with the better lubrication, the friction between piston ring and cylinder is low and friction power consumption is also in a low level. On the contrary the friction and friction power consumption should increase. The lifetime of piston ring and cylinder will decrease because of excessive wear.

The surfaces of piston ring-cylinder are assumed absolutely smooth in earlier studies to study and calculate the result of friction and lubrication property by using the theory of fluid lubrication. After Average of flow model proposed by Patir and Cheng [1], Rhode [4] apply it into problem of piston ring-cylinder. The behavior leads later generations to consider the effect of surface roughness as well as reveals the existence of mixed lubrication area in piston ring-cylinder. However, for simplification, only one dimensional Reynolds equation was used in the study talked above and led to no consider of effect of variance of contact pressure from piston ring-cylinder due to the deformation of piston ring and cylinder. Therefore there is a difference between the results obtained from the theory and the reality.

This study is based on two dimensional Reynolds equation, a model of friction and lubrication property is proposed which includes the deformation of surface roughness, asperity contact, cylinder and piston ring, along with variance of contact pressure. This model describes the mixed lubrication property of piston ring-cylinder sensitively and offers evidences for the design of low-level friction piston ring-cylinder.

1. Theory and equation of lubrication property in piston ring-cylinder

1.1 Average Reynolds equation

Due to average Reynolds equation proposed by Patir and Cheng[1], consider the effect of surface roughness, two dimensional Reynolds equation can be expressed as

$$\frac{\partial}{\partial x}(\emptyset_x h^3 \frac{\partial p}{\partial x}) + \frac{\partial}{\partial y}\left(\emptyset_y h^3 \frac{\partial p}{\partial x}\right) = 6\mu u\left[\frac{\partial h_t}{\partial y} + \sigma\frac{\partial \emptyset_s}{\partial y}\right] + 12\mu\frac{\partial \bar{h}_T}{\partial t} \quad (1)$$

Note: $\emptyset_x$,$\emptyset_y$: Pressure flow factor

$\emptyset_s$: Shear flow factor

σ: Surface roughness Comprehensive RMS,

$$\sigma^2 = \sigma_1^2 + \sigma_2^2$$

σ_1, σ_2: Surface roughness RMS of cylinder and piston ring, m

μ: viscosity of lubrication, Pa·s

p: fluid pressure, Pa

u: velocity of piston reciprocating motion, m/s

$\bar{h}_T$: mathematical expectation of actual oil film thickness,m

h: nominal oil film thickness, m

t: time, s

1.2 Model of asperity contact

Contact equation is one of the most important equations in study of mixed lubrication problems, according to Greenwood and Tripp theory [2].

$$p_c = 4.738(\eta\beta\sigma)^2 E'\sqrt{\frac{\sigma}{\beta}}A \cdot F_{\frac{5}{2}}(H_0) \quad (2)$$

Real contact area:

$$A_c = \pi^2(\eta\beta\sigma)^2 A \cdot F_2(H_0) \tag{3}$$

Note: $H_0 = \frac{h}{\sigma}$

A: nominal contact area, m^2

E': Comprehensive elasticity modulus of cylinder and piston ring, Pa

η: density of asperities, number per m^2

β: summit radius of curvature, m

$$F_n(H_0) = \frac{1}{\sqrt{2\pi}}\int_{H_0}^{\infty}(S - H_0)^n\, e^{-S^2/2}ds$$

1.3 Density of asperities$\boldsymbol{\eta}$ and summit radius of curvature$\boldsymbol{\beta}$

In earlier study of mixed lubrication problems, there is significant one-sidedness because β and η were picked based on experience. The actual value ofβ and η is measured from the surface profile in this study then used into model of contact, assume cylinder and piston ring are both isotropic. z(q) is surface profile, its comprehensive PDF is $f(\xi_1, \xi_2, \xi_3)$, in which $\xi_1 = z(q)$,$\xi_2 = z(q)$,$\xi_3 = z(q)$. ξ_1, ξ_2, ξ_3 indicates the height, slope, curvature of asperities. Spectral moment of surface profile is

$$m_n = \int_{-\infty}^{\infty} G(\omega)\, \omega^n d\omega \qquad n = 0,2,4 \tag{4}$$

in equation above, $G(\omega)$ is power spectrum density function, then[3]

$$f(\xi_1, \xi_2, \xi_3) = (2m_2 n)^{-3/2}(\alpha - 1)^{-1/2}\exp\{-\frac{1}{2}[\frac{\alpha}{\alpha-1}(\frac{\xi_1^2}{m_0} + \frac{\xi_3^2}{m_4}) + \frac{\xi_2^2}{m_2} + \frac{2\xi_1\xi_3}{m_2(\alpha-1)}]\}$$

note: $\alpha = \frac{m_0 m_4}{m_2^2}$ is bandwidth coefficient due to equation above, applying the method of conditional probability could get

$$\eta = \frac{1}{2\pi}(m_4/m_2)^{1/2}$$

$$\beta = \frac{3}{8}\sqrt{\frac{\pi}{m_4}}$$

1.4 Deformation of cylinder and piston ring effects on contact pressure

When putting the piston with ring gap into cylinder, if cylinder deforms then contact pressure between cylinder and piston ring will redistribute. Deformation of piston ring shown in fig.1. Due to deformation, the contact pressure around ring gap decrease. The decrement of contact pressure is

$$Q = p_0 g \qquad (5)$$

Note: p_0: specific pressure of piston ring

For fig.1a $g = 1 - \cos\alpha_0$

For fig.1b $g = (\frac{\sin\alpha_0}{\alpha_0 - \sin\alpha_0} - \frac{1}{1-\cos\alpha_0})^{-1}$

In above equations, α_0 indicates the region that contact pressure decreased, the value of α_0 is related to parameters of cylinder and piston ring and deformation.

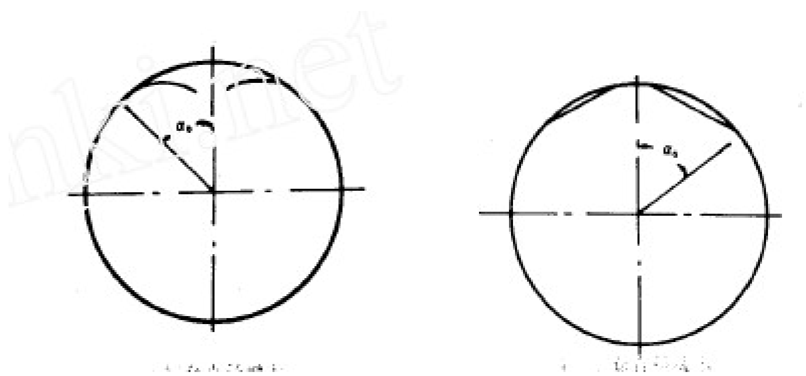


fig.1 deformation of piston ring in cylinder

1.5 Load equation of piston ring

The piston ring is divided into "m" parts along with circumference and considering the minimum oil film thickness h_i and its derivative $\dot{h}_i$ in each part are the same, then there are some forces acting on each part:

(1) load due to air pressure difference between the top and bottom ring;

(2) contact pressure between cylinder and piston ring;

(3) fluid bearing capacity in fluid lubrication region;

(4) asperities load.

for each part of ring, its equation of motion in radial direction is

$$b_i(h_i, \dot{h}_i) = 0 \tag{6}$$

in which $b_i(h_i, \dot{h}_i) = \iint (p_0 - Q)dydx + \iint p_i\, dydx - \iint pdydx - \iint p_c dydx$

then the equation of motion of whole piston ring could be expressed as

$$B(H, HD) = 0 \tag{7}$$

in which $H = [h_{1,} h_2 \cdots h_m]^T$, $HD = [\dot{h}_1 \dot{h}_2 \cdots \dot{h}_m]^T$

The equation 7 is the equation of motion for the whole piston ring and it is the leading equation for solving piston ring-cylinder lubrication problems.

1.6 Friction

Friction in piston ring-cylinder lubrication problems consists of two parts. One is fluid viscous shear stress and another is asperity shear stress.

Fluid viscous shear stress could be expressed as

$$\tau_H = \tau_1 + \tau_2 \tag{8}$$

$$\tau_1 = \frac{\mu u}{h}(\emptyset_f + \emptyset_{fs}) + \emptyset_{fp}\frac{h}{2}\frac{\partial p}{\partial y}$$

$$\tau_2 = (\frac{\sigma_2}{\sigma})^2[(\emptyset_{fp}h - \bar{h}_T)\frac{\partial p}{\partial y} - \frac{2\mu u}{h}\emptyset_{fs}]$$

Note: $\emptyset_f, \emptyset_{fs}, \emptyset_{fp}$ are shear stress factors[4].

Asperity shear stress could be expressed as

$$\tau_A = \tau_0 + fp_c \tag{9}$$

in which τ_0 is shear strength of boundary film (Pa) and f is boundary COF.

Therefore the friction of each part of ring is

$$F_i = \iint(\tau_H + \tau_A)dydx \tag{10}$$

Friction of whole ring should be the summation of the friction on each part of ring

$$F = \sum_{i=1}^{m} F_i \tag{11}$$

2. Process of numerical solution

The solution of non-linear equation (7) is shown as follow: from the "k" position of piston ring, give the values of H_k, then equation (7) is a set of equations respect with HD. Using the method of Newton to solve the value of HD_k. During this process we need to solve Reynolds equation first, finite difference method could be used. After we get the value of HD_k, there is

$$H_{k+1} = H_k + \Delta T \cdot HD_k \tag{12}$$

in which ΔT is time step. It would then follow that until accomplish a cycle. When satisfy

$$\|H_{k+T} - H_k\| \ll \varepsilon\|H_k\| \tag{13}$$

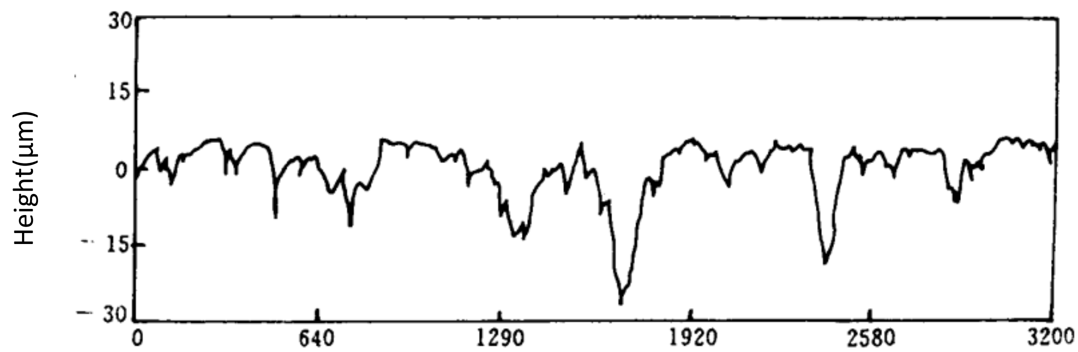
solve the problem. T is cycle period of the engine, ε is convergence factor. If equation (13) is not satisfied with inequality, use H_k and repeat the process talked above instead of H_{k+T} until satisfy the condition of convergence.

3. Result

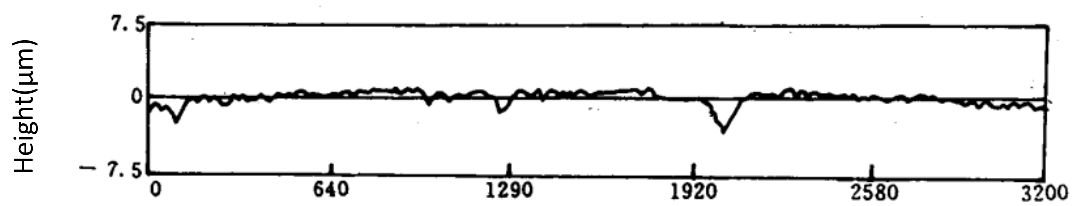
The structural and operating parameters of the engine in this paper are shown as below:

Crank radius R=0.0575m, connecting rod length L=0.217m, inner diameter of cylinder d=0.095m, crank shaft speed $n_r = 2000\text{r/min}$, width of piston ring B=0.003m, comprehensive elastic modulus of cylinder and piston ring $E^* = 200\text{GPa}$,viscosity$\mu = 0.5 \times 10^{-2}\text{Pa} \cdot \text{s}$, shear strength of boundary film $\tau = 2 \times$

10^6Pa, boundary friction coefficient f = 0.1. The surface profile curves of cylinder and piston ring are shown as fig.2a and fig.2b.



Sample points (a) cylinder



Sample points (b) piston ring

Fig.2 The surface profile curves of cylinder and piston ring

Using equation 4 we can determinem_0, m_2, m_4. When two surfaces getting contact, we need to use the comprehensive value of spectrum, the relationship with surfaces is

$$m_n = m_{nr} + m_{nc}, \quad n = 0, 2, 4$$

Where m_{nr} , $m_{nc}$are spectrum of piston ring and cylinder. Absolutely thereis $m_{0c} = \sigma_1^2, m_{0r} = \sigma_2^2$. The oil film thicknesses at the top of the ring in three rings are varied as the dimension of crank angle and central angle $\emptyset$, which is shown as fig 3. When cylinder diameter increased, the oil film thickness increased due to the deformation around the ring gap and increase amplitude is about 30%. Fig 4 shows the ratio of the oil film thickness and surface roughness at $\emptyset = 180^\circ$, which is also called film thickness ratio H_0. We can determine two surfaces are in fluid lubrication phase or mixed lubrication phase. Two surfaces are in mixed lubrication phase when H_0 is less than a certain value. Friction curve in each ring is shown as fig 5, we can determine the maximum friction happens around the stop point during power stroke. Because cylinder piston ring is in mixed lubrication phase at this moment, the asperities happen to shear and lead to increase the friction.

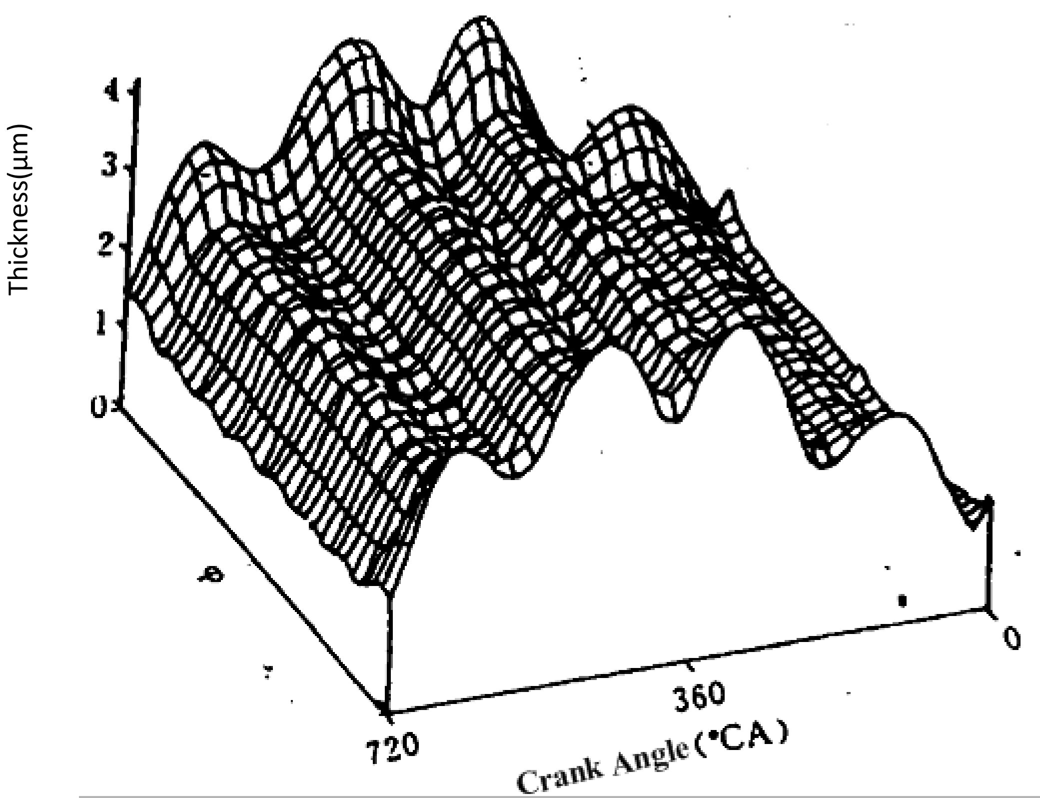


Fig. 3 Changes in 3D for the oil film thicknesses at the top of the ring in three rings

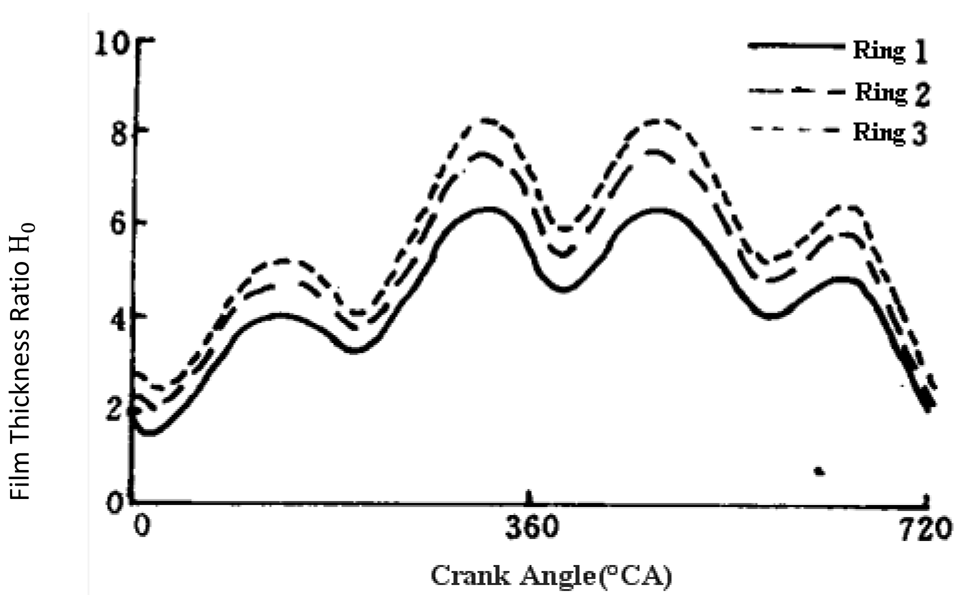


Fig.4 Changes for the oil film thicknesses at $\emptyset = 180^{\circ}$

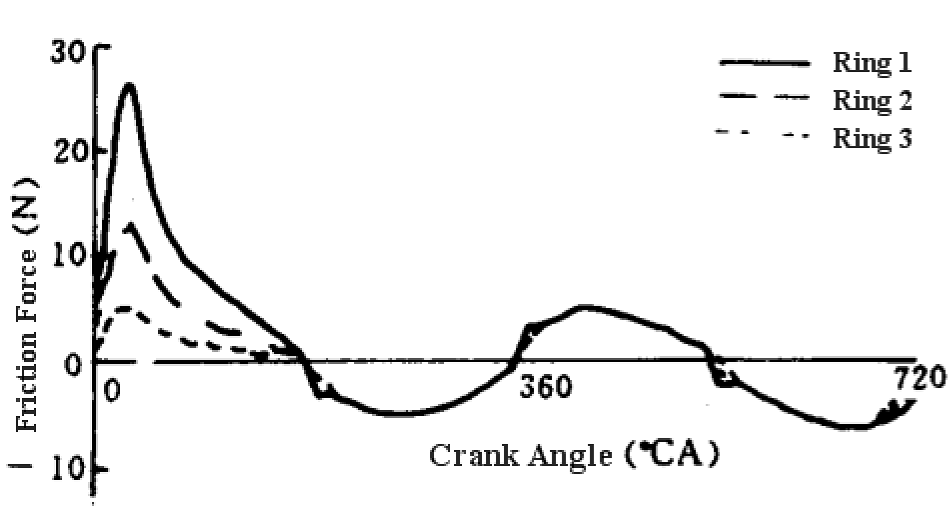


Fig.5 Friction Force Curve in each ring.

4. Conclusion

The oil film thickness between cylinder and piston ring is variational along with circumference. The oil film thickness increases around ring gap because the deformation of the cylinder and piston ring.

Around the region of power stroke, piston ring-cylinder is under mixed lubrication status and the friction is maximum at this time.

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