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Friction Characteristics of Piston Skirt: Parametric Investigation

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Abstract : The purpose of this paper is to investigate the effects of design parameters on the friction loss in piston skirt. An analytical model to describe the friction characteristics of piston skirt has been presented, which is based on the secondary motion of piston and mixed lubrication theory. It could be shown that the skirt friction closely depends on the side force acted on the piston pin. The side force is influenced by cylinder pressure at low engine speed, but by inertia force at high engine speed. The usage of extensive skirt area and low weight piston is effective to reduce the friction loss at high speed. The low viscosity oil considerably decreases viscous friction as engine speed increases, but it increases boundary friction at low engine speed. From the parametric study, it is found that the skirt axial profile is the most important design parameter related to the reduction of skirt friction.

Keywords : Piston skirt, friction loss, secondary motion, mixed lubrication, side force, profile

Introduction

There is much possibility of improvement in fuel economy by reduction of mechanical friction losses in engine accounted for a 15~20% of total energy input. It is reported that a 10% reduction in overall friction that would give a 1~1.5% improvement in fuel consumption [1]. The friction losses in engine results from the relative motion of each moving component and direct metal-to-metal contact. A 40~50% in total engine friction occurs at piston assembly which composed of piston skirt and ring pack, and the piston skirt occupies a 15~25% in total engine friction [2].

Although piston skirt occupies large part of engine friction, the most of researches are focused on the slap motion in piston. However, the active progress being made in the study of skirt friction with consideration of secondary motion. Li *et al.* [3] considered hydrodynamic lubrication theory to investigate secondary motion of piston skirt. Oh *et al.* [4] and Dursunkaya *et al.* [5] developed elastohydrodynamic lubrication model in conjunction with piston motion. Zhu *et al.* [6, 7] performed a significant analysis for the secondary motion and friction characteristics of piston skirt. In recent years, Liu *et al.* [8], and Kim, *et al.* [9] analyzed the friction and motion of piston with consideration of surface roughness effects.

The skirt friction closely relates to secondary motion of piston that influenced by various design parameters. Therefore, the parametric study is performed to investigate the effects of design parameters on the skirt friction. The mathematical model takes into account dynamic behavior of piston skirt and mixed lubrication theory. The presented results will be very useful to understand friction characteristics of skirt, and will be a great contribution to the reduction of friction loss in piston skirt.

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Theoretical Model

Average reynolds equation

To take into account surface roughness effects, the average Reynolds equation derived by Patir and Cheng is used [10,11].

$$\frac{1}{R^2}\frac{d}{d\theta}\left(\frac{\phi_x h^3}{\eta}\frac{dp}{d\theta}\right)+\frac{d}{dy}\left(\frac{\phi_y h^3}{\eta}\frac{dp}{dy}\right) \tag{1}$$

$$= -6U\frac{d\bar{h}_t}{dy}-6U\sigma\frac{d\phi_s}{dy}+12\frac{d\bar{h}_t}{dt}$$

The nominal oil film thickness is defined as following form:

$$h_s(\theta, y, t) = C+e_t\cos(\theta-\alpha) + (e_b-e_t)\frac{y}{L}\cos(\theta-\alpha)+R_s(\theta, y) \tag{2}$$

In Eq. (2), represents the function of axial profile of skirt.

The classical Reynolds boundary conditions are used to solve the Ea. (1). The piston skirt has two lubrication areas which are thrust and anti-thrust side. Therefore, the axial and circumferential boundary conditions are given by:

$$p(y=0) = p(y=L) = p_{cr}$$
$$\frac{\partial p}{\partial \theta}(\theta=0) = \frac{\partial p}{\partial \theta}(\theta=\pi) = 0 \tag{3}$$
$$p(\theta=\pm\alpha) = p(\theta=\pi\pm\alpha) = p_{cr}$$

In Eq. (3), α represents the angular extend of lubricated area and p_{cr} means the pressure in crankcase.

The normal force and its moment due to oil film pressure can be expressed by following integrations:

$$F_{h,s} = R_s \iint_A p\cos(\theta - \alpha)d\theta dy \quad (4)$$

$$M_{h,s} = R_s \iint_A p(a-y)\cos(\theta - \alpha)d\theta dy \quad (5)$$

Asperity Contact Model

From the Greenwood and Tripp[12]s asperity contact theory, the average contact pressure and contact area are defined as follows:

$$p_c(h/\sigma) = \frac{16\sqrt{2}}{15}\pi(\mu\beta\sigma)^2 E\sqrt{\frac{\sigma}{\beta}}F_{2.5}(h/\sigma) \quad (6)$$

$$A_c(h/\sigma) = \pi^2(\mu\beta\sigma)^2 F_2(h/\sigma) \quad (7)$$

The reaction force and its moment due to contact can be expressed by following integrations:

$$F_{c,s} = R_s \iint_A p_c\cos(\theta - \alpha)d\theta dy \quad (8)$$

$$M_{c,s} = R_s \iint_A p_c(a-y)\cos(\theta - \alpha)d\theta dy \quad (9)$$

Friction Force

On the mixed lubrication regions, the friction is consist of viscous and boundary friction. The shear stress by hydrodynamic action in oil film is defined as following form:

$$\tau_h = -\frac{\eta U}{h}[\phi_f + (1-2V_{r2})\phi_{fs}] + \frac{dp}{dy}\left[h\phi_{fp}\left(\frac{1}{2}+V_{r2}\right) - V_{r2}\bar{h}_t\right] \quad (10)$$

The viscous friction and its moment are defined by following integrations:

$$F_{fh,s} = R_s \iint_A \tau_h d\theta dy \quad (11)$$

$$M_{fh,s} = R_s \iint_A \tau_h(R_s\cos(\theta-\alpha) - C_p)d\theta dy \quad (12)$$

And, boundary friction and its moment by asperity contact are defined as following form:

$$F_{fc,s} = \mu_f F_{c,s} \quad (13)$$

$$M_{fc,s} = \mu_f M_{c,s} \quad (14)$$

Therefore, the total friction force in piston skirt is given by:.

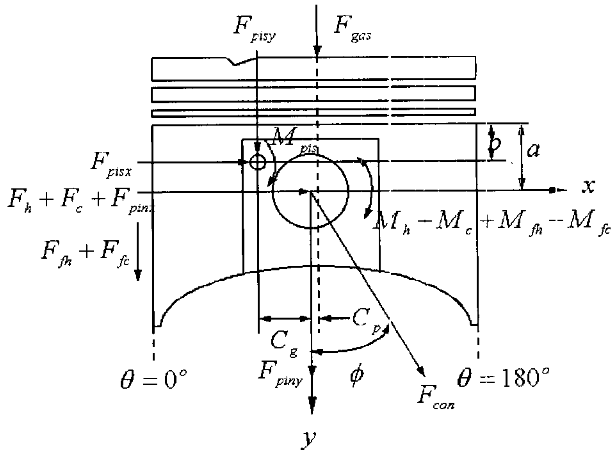

Fig. 1. Schematic diagram of force balance in piston ring and piston skirt.

$$F_{f,total} = F_{fc,s} + F_{fh,s} \quad (15)$$

In the above equations, the flow factor (ϕ_x, ϕ_s), contact factor (ϕ_c), and shear stress factor (ϕ_f, ϕ_{fs}, ϕ_{fp}) can be determined according to the previous study of Patir and Cheng [11].

Force Equilibrium

As can be seen in Fig. 1, the equations of secondary motion can be derived by equilibrium of all forces and moments acted on the piston, and that can be expressed in the following matrix form about top and bottom of skirt.

$$\begin{bmatrix} m_{pin}\left(1-\frac{a}{L}\right) + m_{pis}\left(1-\frac{b}{L}\right) & m_{pin}\frac{a}{L} + m_{pis}\frac{b}{L} \\ \frac{I_{pis}}{L} + m_{pis}(a-b)\left(1-\frac{b}{L}\right) & m_{pis}(a-b)\frac{b}{L} - \frac{I_{pis}}{L} \end{bmatrix}\begin{Bmatrix} \ddot{e}_t \\ \ddot{e}_b \end{Bmatrix}$$

$$= \begin{bmatrix} F_{h,s} + F_{c,s} - (F_{fh,s} + F_{fc,s} + F_{gas} + F_{pisy} + F_{piny})\tan\phi \\ M_{h,s} + M_{c,s} + M_{fh,s} + M_{fc,s} + F_{gas}C_p - F_{pisy}C_g \end{bmatrix}$$

Numerical procedure

The numerical procedure for obtaining the trajectory of piston motion is as follow:

1. The initial value of e_t, e_b, $\dot{e}_t$ and $\dot{e}_b$ is assumed.
2. The fluid film thickness and flow factors are calculated.
3. The pressure distribution is calculated by using the SOR method for solving the Eq. (1), and asperity contact pressure is obtained by using the Eq. (6).
4. All applied forces and moments of the oil film and the asperity contact are obtained by using the numerical integration.
5. The new values of and are determined by using the fourth-order Runge-Kutta method for solving the Eq. (16). The procedure is repeated, until the convergence for the trajectory of piston motion is achieved.

Table 1. shows the specification of test skirt and operating conditions. The measured cylinder pressure is used in the calculation of applying load.

Table 1. Specification of test engine.

Bore × Stroke	75.5 × 83.5 mm
Mass (piston + piston pin)	250 g
Mass (con-rod)	450 g
Reference clearance	30 μm
Oil temp.	90 °C
Lubricant	SAE5 W / 10 W
Engine speed	1500~6000 rpm
Load	Motoring / BMEP 2 bar / 4 bar/ WOT

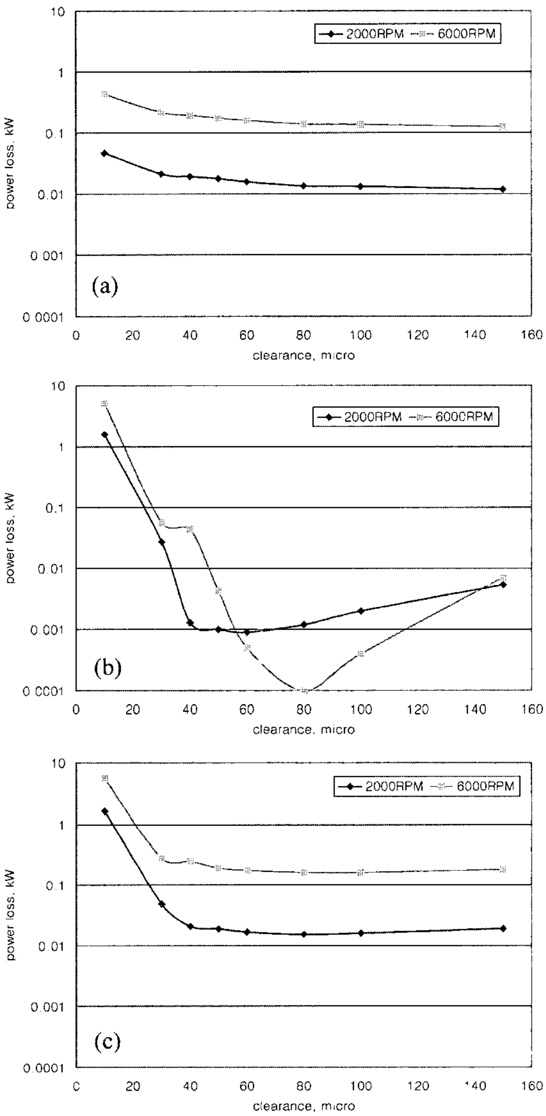

Fig. 2. The effect of radial clearance (a) hydrodynamic (b) boundary (c) total.

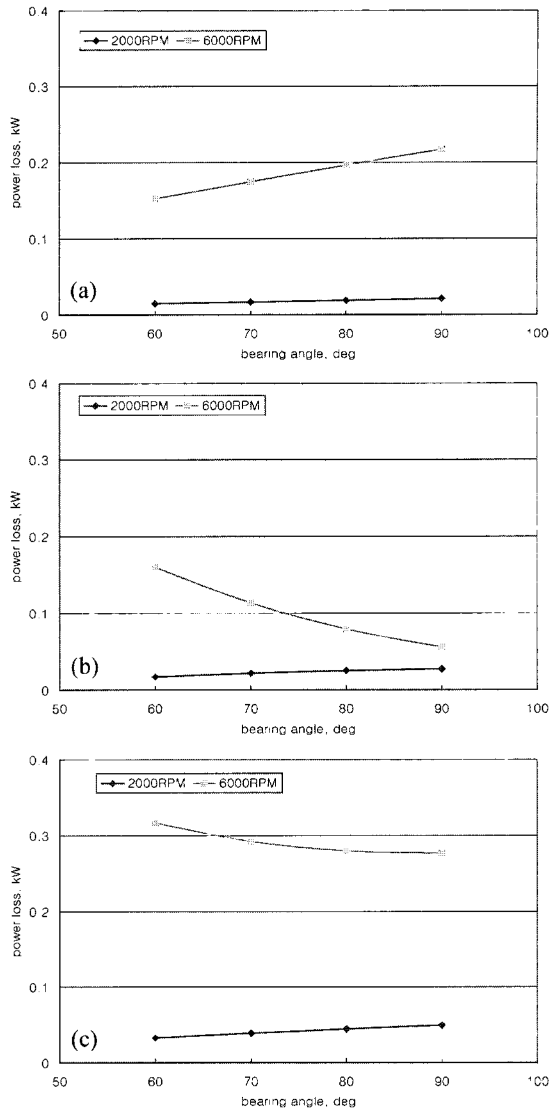

Fig. 3. The effect of bearing area (a) hydrodynamic (b) boundary (c) total.

Results and Discussion

The effect of radial clearance

The skirt friction loss decreases with increase in radial clearance, but there is optimal point to minimize friction loss. If the radial clearance increases too much, the load carrying capacity of oil film is seriously decreased, and results in the increase of solid to solid contact as can be seen in Fig. 2.

The increase of lubricated area usually results in the increase of oil film thickness. This means reduction of shear in oil film. However, in spite of increasing in oil film thickness, the viscous friction increases as lubrication area increases. In boundary friction, it increases at low engine speed with skirt area, but reverse tendency is shown at high speed. The small lubrication area is effective for the reduction of skirt friction at low engine speed.

The effect of mass

The mass of piston and connecting-rod has little effect on the skirt friction at low engine speed. However, the friction loss in skirt increases with increase of mass. The skirt friction depends

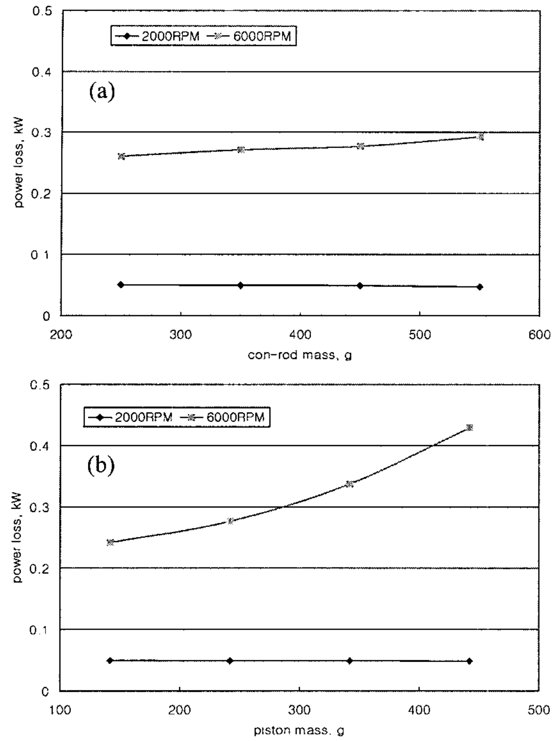

Fig. 4. The effect of mass (a) con-rod mass (b) piston mass.

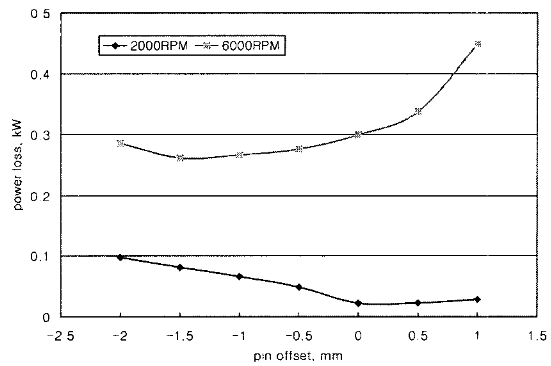

Fig. 5. The effect of pin offset.

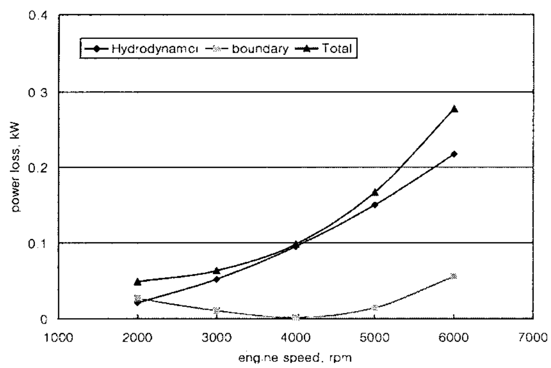

Fig. 6. The effect of engine speed.

on the side force acted on the piston pin. The side force is generally affected by inertia force except expansion stroke, and

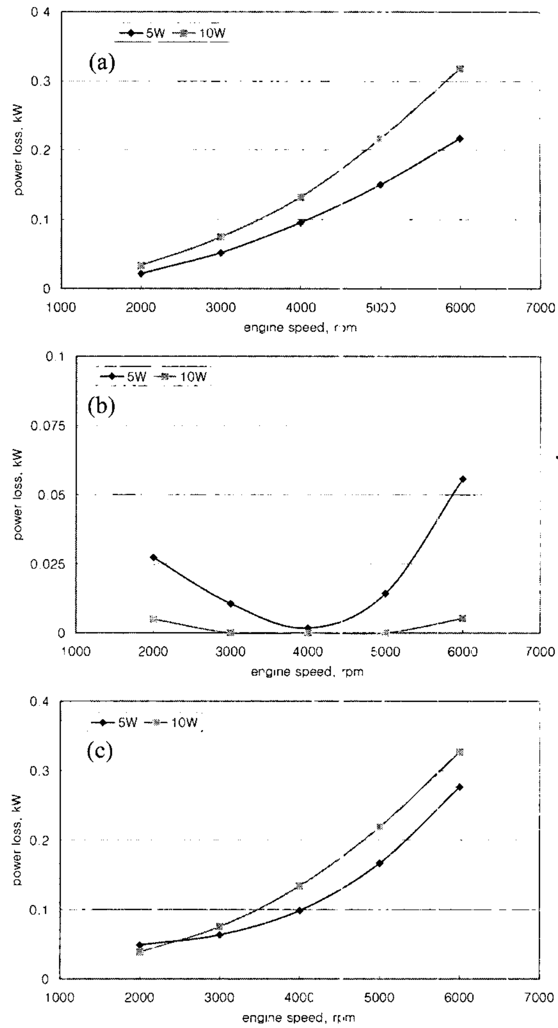

Fig. 7. The effect of oil viscosity (a) hydrodynamic (b) boundary (c) total.

the inertia effect increases with increasing in speed. Therefore, reducing mass is very effective for reduction of skirt friction at high engine speed, and piston mass is superior to connecting-rod mass in the reduction of skirt friction.

The effect of pin offset

The zero offset has advantage of friction loss in skirt at low engine speed. However, the negative offset located at thrust side is effective for the reduction of skirt friction at high speed. And it is of great benefit to slap noise reduction. The selection of optimal pin offset to minimize the skirt friction simultaneously requires consideration in slap motion.

The effect of engine speed

The friction loss by hydrodynamic friction increases with increasing speed because hydrodynamic friction is proportional to the sliding speed. As engine speed increases the oil film thickness increases and friction loss by solid contact decreases. However, a point is reached as speed is increased at which the side force by inertia becomes so great that the boundary friction increases again.

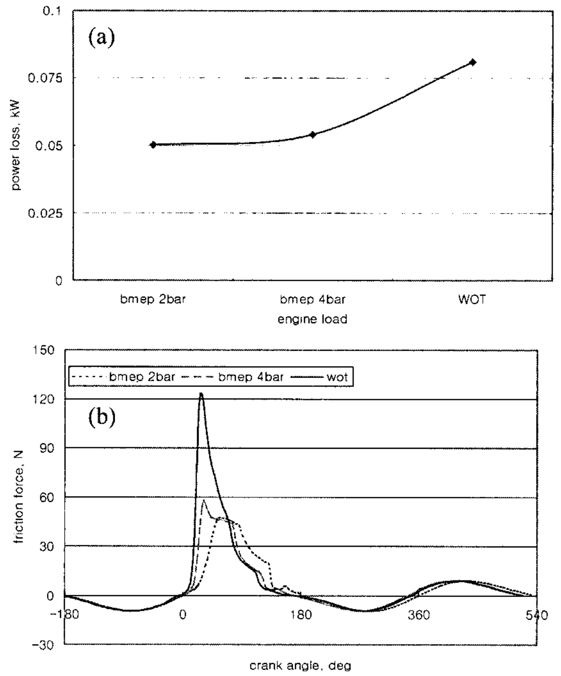

Fig. 8. The effect of engine load (a) power loss (b) friction force.

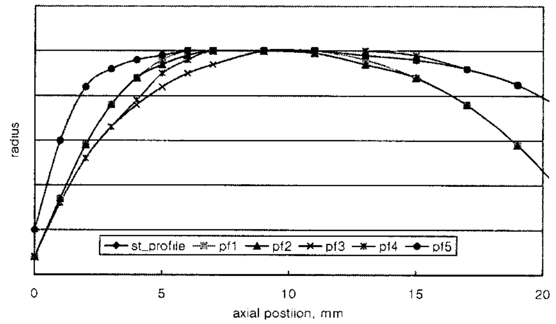

Fig. 9. The various skirt profile.

The effect of oil viscosity

The usage of low viscosity oil is effective to reduce the hydrodynamic friction loss. However, it increases boundary friction because the oil film thickness decreases with decreasing viscosity. As engine speed becomes lower and load higher, the boundary friction becomes greater by low viscosity oil.

The effect of engine load

As engine load is increased the cylinder pressure increases and so the side force increases. This increases in side force causes the skirt friction to increase. Since the cylinder pressure increase is mostly occurred at expansion stroke, the friction at this area is considerably increased as can be seen in Fig. 8 (b).

The effect of skirt profile

The example of various skirt profile is illustrated in Fig. 9, and the effect of skirt profile on the skirt friction is shown in Fig. 10. The skirt friction is determined by oil film thickness that is

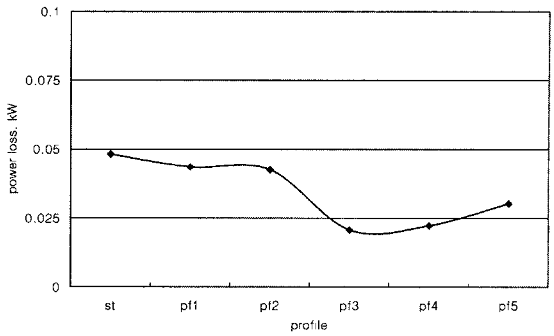

Fig. 10. The effect of skirt profile.

greatly influenced by skirt profile. The profile 3 can improve skirt friction by a 50%.

Conclusion

The effects of design parameters and operating conditions are investigated to improve the skirt friction. From the parametric study, the following conclusions are derived.

1. As radial clearance increases, the skirt friction decreases. A point is reached as clearance is increased at which the oil film thickness becomes so small that the friction increases.
2. Smaller lubrication area is effective to reduce the skirt friction at low engine speed.
3. Smaller mass is effective to reduce the skirt friction at high engine speed.
4. The skirt friction increases with increasing load
5. The usage of low viscosity oil is effective to reduce the skirt friction except low engine speed and high load.
6. The best way to improve the skirt friction is to optimize the skirt axial profile.

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Nomenclature

A	:bearing surface area of skirt
A_c	:contact area per unit area
a	:distance from the top of skirt to the pin
b	:distance from the top of skirt to the C.G.
C	:clearance between skirt and cylinder
C_g	:distance between C.G. and wrist-pin
C_p	:distance between. wrist-pin and piston center
E	:Young's Modulus
e_b	:eccentricity of piston bottom
e_t	:eccentricity of piston top
$\dot{e}_b$	:radial velocity of piston bottom
$\dot{e}_t$	:radial velocity of piston top
F	:normal and friction force
F_{gas}	:combustion gas force
F_{pinx}	:inertia force due to pin mass in x-direction
F_{pisy}	:inertia force due to pin mass in y-direction
h	:inertia force due to piston mass in x-direction
h_m	:inertia force due to piston mass in y-direction
h_t	:nominal film thickness
h_m	:nominal minimum oil film thickness
h_t	:local oil film thickness
$\bar{h}_t$	:average gap
I_{pis}	:piston moment of inertia
L	:piston-skirt length
l	:length
l_{con}	:length of connecting-rod
M	:moment by normal and friction force
M_{pis}	:inertia moment of piston-skirt
m_{pin}	:wrist-pin mass
m_{pis}	:piston-skirt mass
p	:hydrodynamic pressure
p_c	:asperity contact pressure
R	:nominal radius of skirt
r_{cr}	:crank radius
t	:time
U	:sliding speed
V_{r1}, V_{r2}	:variance ratio
$\dot{Y}, \ddot{Y}$	:piston-skirt velocity, acceleration
y	:axial coordinate
β	:asperity radius of curvature
ϕ_x, ϕ_y	:pressure flow factor
ϕ_s	:shear flow factor
$\phi_f, \phi_{fp}, \phi_{fs}$	:shear stress factor
η	:oil viscosity
μ	:asperity density
μ_f	:friction coefficient
θ	:angular coordinate around piston-skirt
σ	:composite rms roughness
τ_h	:hydrodynamic component of shear stress
ω	:rotational speed
ψ	:crank angle

subscript

c	:asperity contact
fc	:friction by asperity contact
fh	:friction by hydrodynamic
h	:hydrodynamic
s	:skirt