Friction Characteristics of Piston Ring Pack with Consideration of Mixed Lubrication: Parametric Investigation

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This paper reports on the friction characteristics of a piston ring pack with consideration of mixed lubrication. The analytical model is presented by using the average flow and asperity contact model. The effect of operating condition, and design parameters on the MOFT, maximum friction force, and mean frictional power loss are investigated. Piston ring pack shows mixed and hydrodynamic lubrication characteristics. From the predicted results, it was fand that the ring tension and height of surface roughness have great influence on the frictional power losses in a ring pack. Especially, ring tension is a dominant factor for the reduction of friction loss and maintenance of oil film thickness.

Key Words: Mixed Lubrication, Piston Ring Pack, Asperity Contact, Minimum Oil Film Thickness, Surface Roughness, Power Loss, Tension

Nomenclature

- A_{ac} : Real contact area per unit area
- A_c : Real contact area
- b : Ring width
- E : Young's Modulus
- F_b : Boundary friction force
- F_c : Asperity contact force
- F_{oil} : Radial force of oil film
- F_{total} : Total friction force
- F_v : Viscous friction force
- h : Nominal film thickness
- h_m : Nominal minimum oil film thickness
- h_t : Local oil film thickness
- $\bar{h}_t$: Average gap
- p : Film Pressure
- p_b : Back pressure
- p_c : Asperity contact pressure
- p_{TE} : Elastic pressure of ring
- R : Ring radius
- r_c : Crank radius
- t : Time
- U : Sliding speed
- V_{r1}, V_{r2} : Variance ratio of parameter
- α : Rate of the film shear strength with pressure
- β : Asperity radius of curvature
- δ : Composite roughness amplitude
- γ : Roughness pattern parameter
- η : Oil viscosity
- μ : Asperity density
- σ : Composite rms roughness
- τ_0 : Shear strength constant
- τ_v : Hydrodynamic component of shear stress
- ω : Rotational speed

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1. Introduction

The fuel efficiency is the most important aspect of modern engine design and development. To achieve such aim, it is necessary to reduce the frictional power losses in the internal combustion engine.

Piston ring generally operates in the hydrodynamic lubrication regime. However contact between the ring and liner occurs at the TDC and BDC because of relatively low sliding speed. The direct contact at these regimes results in liner and ring wear, and increae in the frictional losses. Frictional power loss from the piston assembly accounts for 40-45% of the total engine frictional losses. The friction losses in ring pack amoun to 60% in the total friction losses of the piston assembly (Hu, et al., 1994). Therefore, accurate understanding of friction characteristics of piston ring is essential for the reduction of frictional losses, and the insurance of engine durability.

Lloyd (1968), Ting and Mayer (1974), and Dowson, et al. (1979) performed a significant hydrodynamic lubrication analysis for piston ring. They calculated the minimum oil film thickness and viscous friction force between the ring and the liner. In the recent years, the hydrodynamic lubrication theory has been extended to the mixed lubrication regime by combining the average Reynolds equation presented by Patir and Cheng (1978, 1979) with Greenwood and Tripp's (1971) asperity contact model. Rohde (1980) investigated the ring friction and film thickness in the mixed lubrication regime, and found friction force spikes at the TDC or BDC. They are induced by asperity contact, and depend on the height and directional pattern of surface roughness. Hu, et al. (1994) found that contact pattern and film thickness between the ring and the liner are not exactly axisymmetrical. Wakuri, et al. (1995) investigated the effect of design parameter on the mixed lubrication characteristics in the piston ring pack. Hamatake, et al. (1995) and Yun, et al. (1995) performed other studies for mixed lubrication in ring pack.

In this paper, a mixed lubrication model for piston ring pack is presented. The analytical model uses the average flow model for calculating the mean hydrodynamic film pressure, and asperity contact model for determining the asperity contact forces. The effects of various conditions on the nominal MOFT (Minimum Oil Film Thickness), total friction force of ring pack, and mean power loss are investigated. From the predicted results, it is found that ring tension and surface roughness are dominant factors for reducing the frictional losses in the ring pack. Especially, it is very important for the ring tension to reduce friction and maintain the sufficient film thickness. The presented results in the paper may be very useful guidelines to design low frictional ring pack.

2. Basic Theory

2.1 Average reynolds equation

The average Reynolds equation derived by Patir and Cheng (1978, 1979) is used in this study to consider the surface roughness effects. One-dimensional form of equation is written by

$$\frac{d}{dx}\left(\frac{\phi_x h^3}{\eta}\frac{dp}{dx}\right) = 6U\frac{d\bar{h}_t}{dx} + 6U\sigma\frac{d\phi_s}{dx} + 12\frac{d\bar{h}_t}{dt} \tag{1}$$

The nominal oil film equation is expressed as follows

$$h(x,\ t) = h_m(t) + R(x) \tag{2}$$

In Eq. (2), the second term of right hand is the function of the ring profile. In this study, barrel type top ring, napier type second ring, and 3-piece type oil ring is used. Equations (3) and (4) define the local oil film thickness and the average gap, respecfively.

$$h_t = h + \delta_1 + \delta_2 \tag{3}$$

$$\bar{h}_t = \int_{-h}^{\infty} (h+\delta) f(\delta)\, d\delta \tag{4}$$

where δ and $f(\delta)$ are composite roughness and probability density function of δ (Patir and Cheng, 1978).

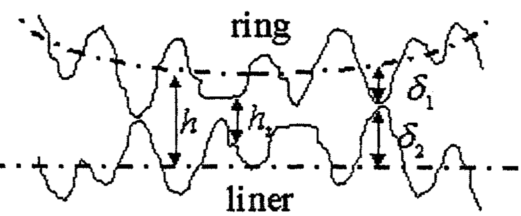


Fig. 1 Schematic diagram of surface roughness

By adapting the following nondimensional variables,

$$\bar{x}=\frac{x}{b},\ H=\frac{h}{\sigma},\ \bar{p}=\frac{p(\sigma/b)^2}{\eta_0\omega},$$
$$\bar{U}=\frac{U}{r_c\omega},\ \tau=\omega t,\ \phi_c=\frac{d\bar{h}_t}{dh}$$

Equation (1) is normalized as the following form

$$\frac{d}{d\bar{x}}\left(\frac{\phi_x H^3}{\eta^*}\frac{d\bar{p}}{d\bar{x}}\right) =6\frac{r_c}{b}\bar{U}\left[\phi_c\frac{dH}{d\bar{x}}+\frac{d\phi_s}{d\bar{x}}\right]+12\phi_c\frac{dH}{d\tau} \tag{5}$$

Equation (5) is solved for pressure distribution using classical Reynolds boundary condition, which are given by:

$$\bar{p}(-1/2)=\bar{p}_1$$
$$\bar{p}(\bar{x}_2)=\bar{p}_2,\ \left.\frac{\partial\bar{p}}{\partial\bar{x}}\right|_{\bar{x}_2}=0 \tag{6}$$

In Eq. (6), the boundary pressures $\bar{p}_1$ and $\bar{p}_2$ means cylinder pressure and inter-ring pressure, respecfively. Inter-ring pressures can be obtained by blow-by analysis.

2.2 Asperity contact model

From the Greenwood and Tripp's (1971) asperity contact model, the average contact pressure and contact area are expressed as follows:

$$p_c(H)=\frac{8\sqrt{2}}{15}\pi(\mu\beta\sigma)^2E\sqrt{\frac{\sigma}{\beta}}F_{2.5}(H) \tag{7}$$
$$A_{ac}(H)=\pi^2(\mu\beta\sigma)^2F_2(H) \tag{8}$$

In Eqs. (7) and (8), for the surface roughness with Gaussian distributed asperity height, the function $F_n(H)$ is defined as:

$$F_n(H)=\frac{1}{\sqrt{2\pi}}\int_H^\infty(s-H)^n e^{\frac{s^2}{2}}ds \tag{9}$$

The asperity contact force and real contact area can be expressed as follows

$$F_c=2\pi R\int_0^b p_c(H)\,dx \tag{10}$$
$$A_c=2\pi R\int_0^b A_{ac}(H)\,dx \tag{11}$$

2.3 Force equilibrium

The radial force by oil film pressure can be obtained by integration of pressure, which can be expressed as follows

$$F_{oil}=2\pi R\int_0^b p\,dx \tag{12}$$

Therefore, the radial force equilibrium has the form of

$$F_{total}\left(H_m,\frac{dH}{d\tau}\right)=F_{oil}+F_C-2\pi Rb(p_{TE}+p_b) =0 \tag{13}$$

In Eq. (13), p_b represents behind ring pressure, which can be obtained by blow-by analysis.

2.4 Friction force

The friction force between the piston ring and liner in mixed lubrication regime consists of the viscous shearing force of hydrodynamic oil film and boundary shearing force due to asperity contact.

The viscous shear stress caused by hydrodynamic lubrication is defined as follows

$$\tau_v=-\frac{\eta U}{h}[\phi_f+(1-2V_{r2})\phi_{fs}] +\frac{dp}{dx}\left[h\phi_{fp}\left(\frac{1}{2}+V_{r2}\right)-V_{r2}\bar{h}_t\right] \tag{14}$$

and viscous friction force can be obtained by integration of viscous shear stress

$$F_V=2\pi R\int_0^b \tau_v dx \tag{15}$$

The boundary friction force due to the asperity contact is written by (Rohde, 1980)

$$F_b=\tau_0 A_c+\alpha F_c \tag{16}$$

Therefore the total friction force is expressed as follows

$$F_{fric}=F_v+F_b \tag{17}$$

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 (1979).

Figure 2 shows the numerical procedure for calculating the nominal minimum oil film thickness and friction force during the engine cycle.

Table 1 shows the specification of test ring pack.

Table 1 Specification of test ring pack

Bore × Stroke	75.5 × 83.5mm
Total Ring Tension	43N
RMS Roughness	1.0μm
Ring Width	1.5/1.5/2.5mm
Oil Viscosity	10.0cP (90℃)

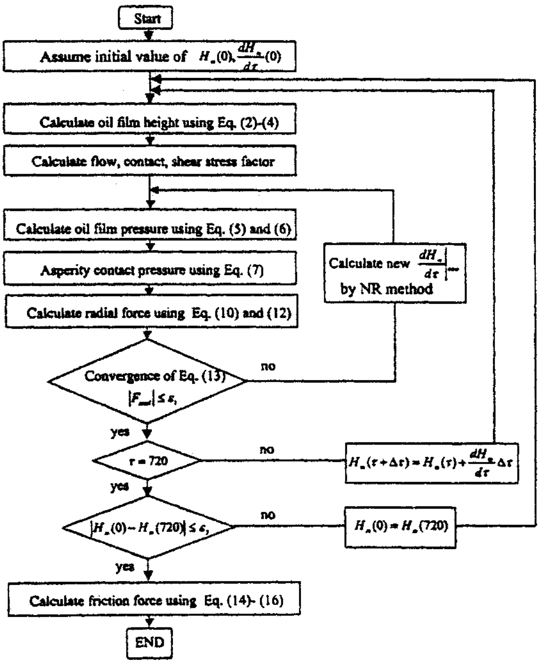


Fig. 2 Flow chart for numerical procedure

3. Result and Discussion

A cyclic variation of minimum oil film thickness for the ring pack is shown in Fig. 3. The minimum oil film thickness occurs near the TDC of expansion stroke, where the gas pressure is high and the sliding speed is relatively low. In the TDC and BDC areas, the sliding velocity is instantly zero, and squeeze velocity is dominant; therefore the oil film thickness becomes thinner. In a second ring, there is some difference between up-stroke and down-stroke due to the nonaxisymmertic ring profile.

Figure 4 shows the variation of the total friction force for each ring. The results present hydrodynamic lubrication characteristics during mid-stroke, but mixed lubrication characteristics appear at the TDC and BDC. At these areas,

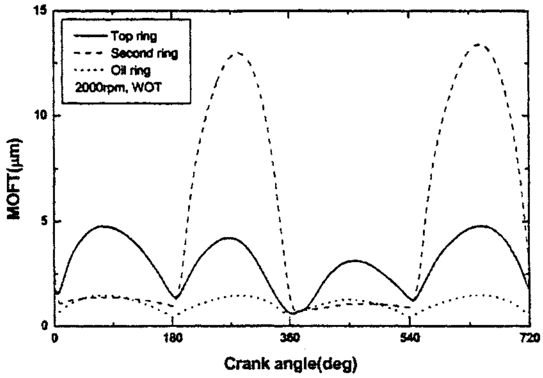


Fig. 3 Variation of MOFT for ring pack

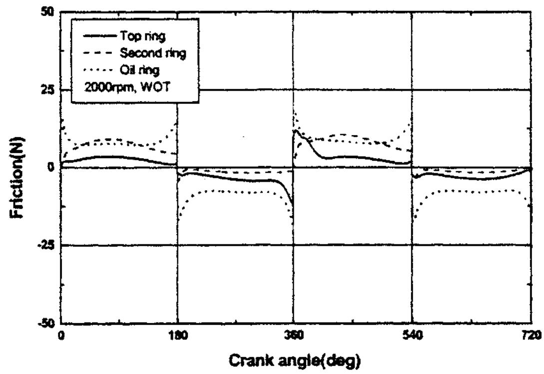


Fig. 4 Variation of total friction force for each ring

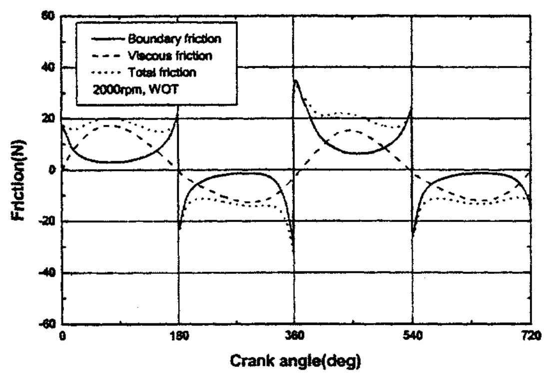


Fig. 5 Variation of viscous and boundary components in total ring pack friction

partial asperity contact occurs because of the low sliding velocity, and then the spike of friction force appears. The maximum value of friction in the mid-stroke is related to the friction loss, but the value of friction spike at the TDC and BDC will be related to the wear of the ring and liner. Figure 5 displays the viscous and boundary friction terms in total the ring pack friction, and the variation of total power loss in the ring pack is shown in Fig. 6.

Figure 7 shows the effect of oil viscosity on the MOFT at the top ring and the total friction in the ring pack. The minimum oil film thickness of top ring tends to increase with an increase of oil viscosity. With an increase of oil viscosity, the boundary component of the total friction at the TDC and BDC is reduced, but the viscous component increases. Therefore the total power loss increases with usage of high-viscosity oil because of stronger hydrodynamic action of oil film and higher shear rate of high-viscosity oil.

In spite of reduced friction loss, wear of ring and liner could be increased by higher boundary friction with usage of low-viscosity oil. Above results correspond to the mixed and hydrodynamic lubrication characteristics in *Stribeck* diagram. From the results of Fig. 7, it is more effective to use low viscosity oil for the reduction of friction loss in the ring pack.

The effect of ring tension on the MOFT and total friction in ring pack is shown in Fig. 8. The reduction of ring tension results in the decrease in the applied force to the ring. Therefore, minimum oil film thickness and total friction decreases with

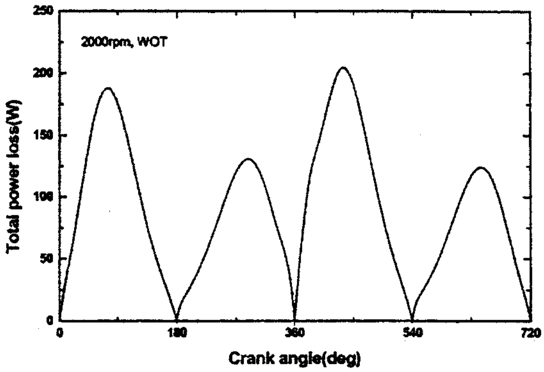
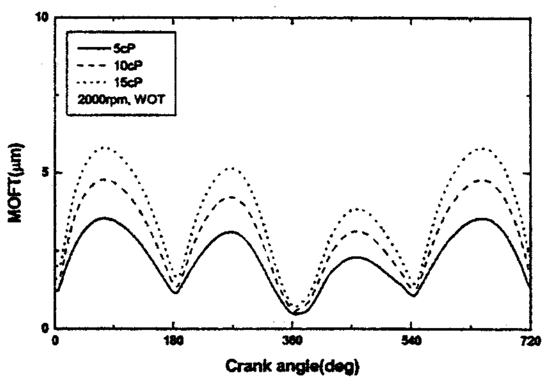
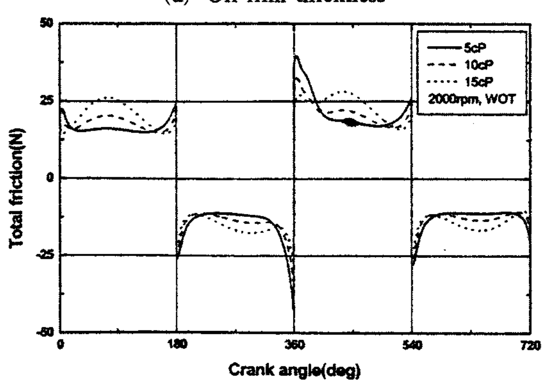


Fig. 6 Variation of total power losses in a ring pack

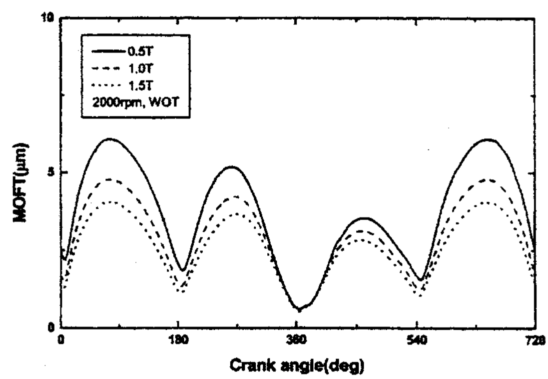


(a) Oil film thickness

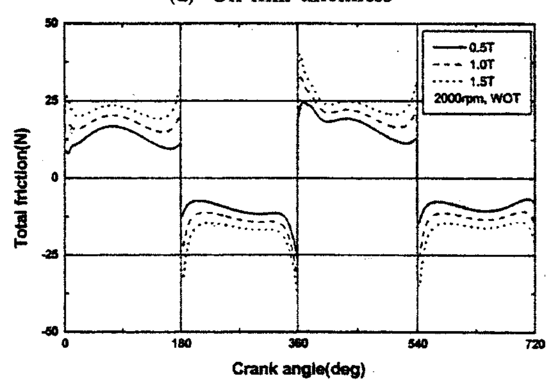


(b) Friction force

Fig. 7 Variation of MOFT in a top ring and total ring pack friction with oil viscosity

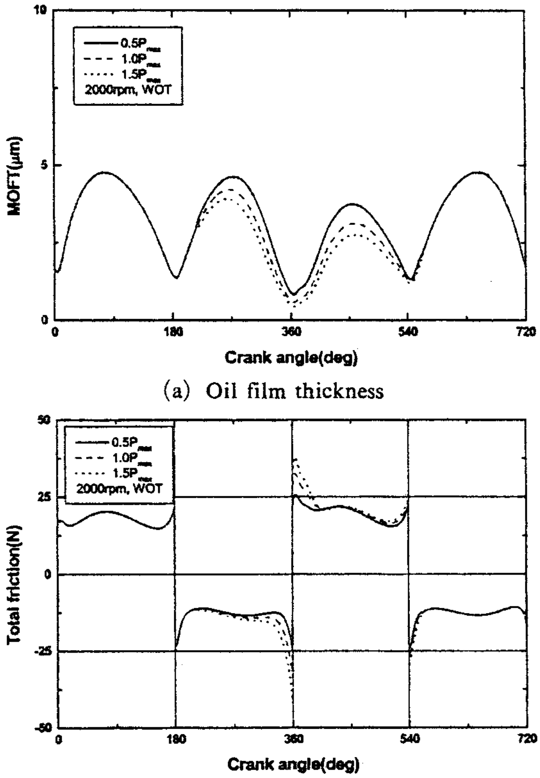


(a) Oil film thickness



(b) Friction force

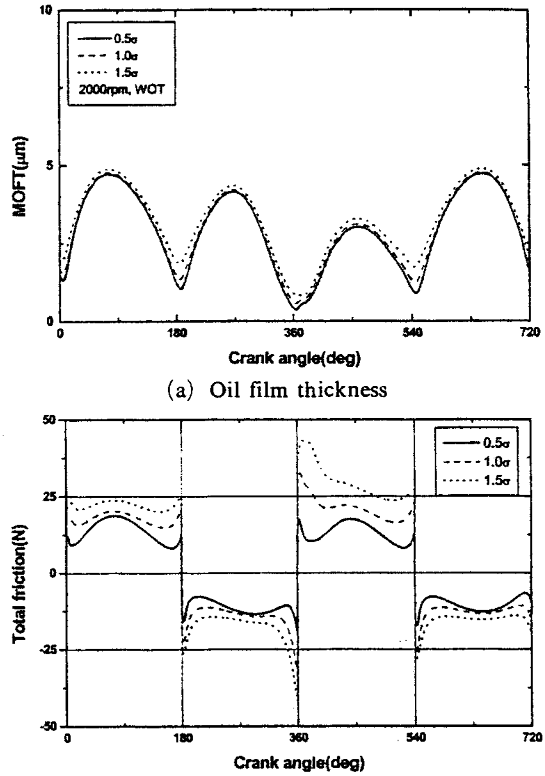
Fig. 8 Variation of MOFT in a top ring and total ring pack friction with ring tension



(a) Oil film thickness

(b) Friction force

Fig. 9 Variation of MOFT in a top ring and total ring pack friction with maximum cylinder pressure



(a) Oil film thickness

(b) Friction force

Fig. 10 Variation of MOFT in a top ring and total ring pack friction with roughness height

decrease in the ring tension. The reduction of ring tension is shown to be very effective to reduce the friction loss, and to maintain sufficient oil film thickness.

Figure 9 shows the effect of maximum cylinder pressure. The applied force to the ring is increased in the case of higher cylinder pressure. Therefore, minimum oil film thickness and friction force increase with an increase of maximum pressure, but the effect is not significant, and primarily appears during the compression and expansion strokes.

The effect of composite rms roughness height is displayed in Fig. 10. A high roughness height decreases the local film thickness, and increases the asperity interaction. Therefore, minimum oil film thickness and total friction force increase with an increase in roughness height. This tendency is in good agreement with previous studies (Rohde, 1980).

Figure 11 displays the sensitivity charts for the effects of variaus parameters on the MOFT and mean power loss. The cylinder pressure marginally affects the friction loss, and low-viscosity oil and low roughness height are bad for sufficient oil film. But it is the most important for ring tension to reduce friction loss, and maintain the minimum oil film thickness.

4. Conclusion

The mixed lubrication analysis for the piston ring pack is performed with consideration of average Reynolds equation and asperity contact model. The effects of various parameters on the MOFT, maximum friction force, and mean power loss are investigated. The following conclusions

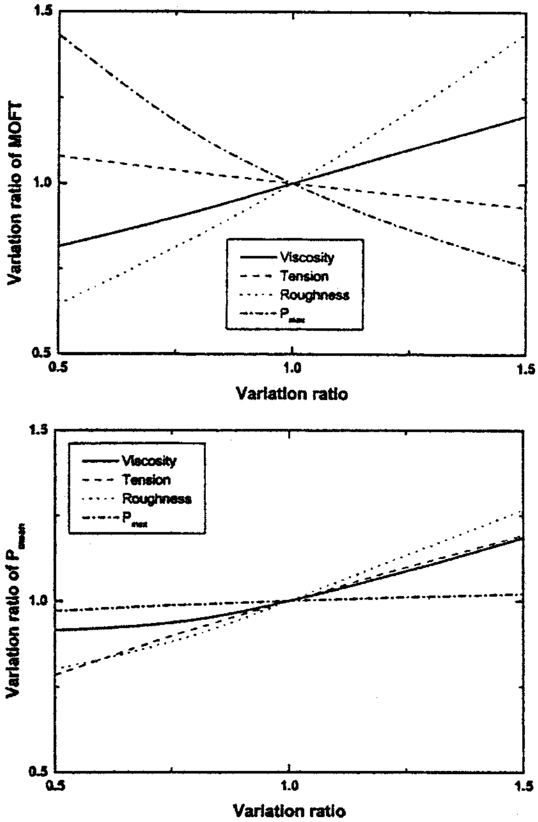


Fig. 11 Effects of parametric variations on the MOFT in a top ring and mean power loss in a ring pack

are derived.

(1) The ring pack shows that it is under hydrodynamic lubrication characteristics during the mid-stroke, but mixed lubrication characteristics appear at TDC and BDC.

(2) The low viscosity oil is effective to reduce the total friction losses, but it increases the maximum friction force at TDC and BDC, and reduces MOFT.

(3) The low height of surface roughness decreases the total friction losses, but nominal MOFT decreases.

(4) The total friction losses are reduced with decrease of cylinder pressure, but the effect is very small.

(5) From the predicted results, it is very effective for ring tension and surface roughness to reduce the frictional losses in a ring pack.

(6) It is thought that the above results are very useful for design of low frictional ring pack.

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