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# Effect of Surface Texture on Compressor Piston Lubrication

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## ABSTRACT

Due to small piston size and small piston – cylinder clearances, a partial lubrication regime is present in hermetically sealed reciprocating compressors. Hence the surface roughness effects on lubrication attract attention. In the current study, a model that solves the average Reynolds' equation for piston-cylinder bearing for a compressor piston with circumferential oil feed groove is developed. A parametric study is carried out to investigate the effect of surface roughness by implementation of average Reynolds' equation in the solution of the lubrication problem do define different surface textures of piston-cylinder.

## 1. INTRODUCTION

The solution of piston lateral motions requires the solution of slider-crank dynamics together with the Reynolds' equation, which solves the lubrication problem between the piston and cylinder. For the lateral force on piston formed by slider-crank dynamics, the piston makes lateral and rotational motions which create the film pressure in oil which fills the piston-cylinder clearance. But when the clearance is small the effect of surface texture which influences the lubrication should be considered. When partial lubrication is present the texture of the surfaces affect the oil flow through the piston-cylinder clearance. According to orientation of the asperities the flow is restricted in the direction of piston motion or in transverse direction. The effect of surface texture in lubrication can be defined with the average Reynolds' equation.

In this study, both Reynolds' and average Reynolds' equations are solved using a rigid piston-cylinder assumption. Using the calculated oil film hydrodynamic pressure, the equations of motion in lateral direction are solved then including the boundary contact pressure, cylinder gas pressure and inertia forces determined from slider crank dynamics and lubrication analysis. Finally a parametric study is performed to investigate the effect of surface texture for selected compressor piston designs.

In the field of lubrication phenomenon of a reciprocating piston many studies were carried out in recent years especially for automotive applications. Goenka and Meernik [1] described the analysis methods developed for piston lubrication analysis and compared the friction prediction of each of these methods with data obtained from an experimental rig designed to measure piston-assembly friction. Keribar and Dursunkaya [2] solved the hydrodynamic lubrication problem and presented a general model for the solution of secondary motion analysis of conventional and articulated piston assemblies. In the study of Keribar and Dursunkaya [3] that is a continuation of [2], a comprehensive model of piston is developed for use in conjunction with piston secondary dynamic analysis to characterize the effects of the skirt-cylinder oil film on piston motions. For the Reynolds' equation a finite difference solution is used and an asperity contact model is implemented to the solution to calculate the oil and contact pressure distribution in the skirt-bore oil film as a function of all input design parameters and positions and motions of the skirt relative to the cylinder. An integrated simulation methodology for the analysis of piston tribology was presented by Keribar, Dursunkaya and Ganapathy [4] that is comprised of coupled models of piston secondary dynamics, skirt oil film elasto-hydrodynamic lubrication and wristpin bearing hydrodynamics, developed earlier by the authors. The model predicts piston assembly secondary

motions, piston skirt friction, skirt and wristpin oil film pressures, transient deformations, skirt-cylinder contact/impact pressures and skirt and cylinder wear loads. Duyar et. Al. [5] improved the model of Dursunkaya et al.[4] including a mass conserving algorithm to the piston lubrication model.

In the field of lubrication of compressor bearings, Duyar and Dursunkaya [6, 7] solved elasto-hydrodynamic lubrication problem for compressor small end bearing using finite difference solution of the Reynolds' equation for elastic pin problem. With a parametric study they searched the effects of design parameters of the connecting rod small end bearing. The solution of the compressor piston lubrication for rigid surfaces including different oil feed groove applications is given by Hacıoğlu and Dursunkaya [8].

For partial lubrication case Patir and Chang [9, 10] introduced flow coefficients for average Reynolds' equation in order to define the effect of different surface roughness contacts on lubrication. By numerical solutions they obtained pressure and shear flow factors that define the average Reynolds' equation.

In this study that is continuation of [8] the model of Patir and Chang [9, 10] is used to determine the roughness affect in partially lubricated compressor piston.

## 2. MODELLING

The equation of the piston hydrodynamic lubrication problem is the Reynolds' equation for the oil film thickness and oil film pressure distributions. The following form of the Reynolds' equation is used, where  $\theta$  the piston circumferential coordinate,  $z$  is the pistons axial coordinate.  $R$  is the piston radius and  $U$  is the piston sliding velocity in axial direction.

$$\frac{\partial}{\partial z} \left( h_T^3 \frac{\partial P}{\partial z} \right) + \frac{1}{R^2} \frac{\partial}{\partial \theta} \left( h_T^3 \frac{\partial P}{\partial \theta} \right) = -6\mu U \frac{\partial h_T}{\partial z} + 12\mu \frac{\partial h_T}{\partial t} \quad (1)$$

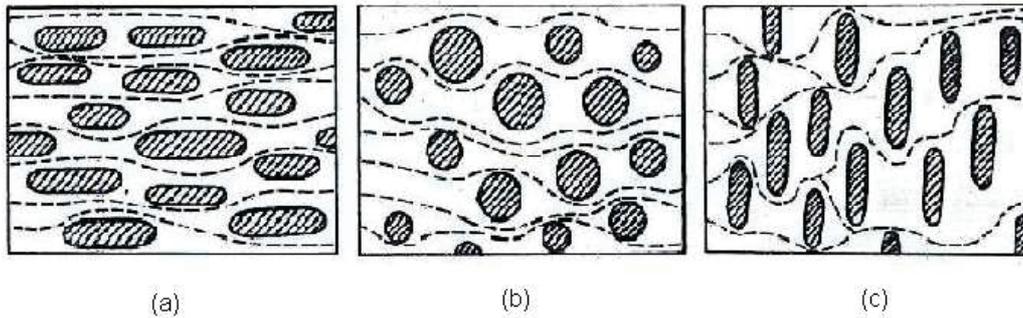
The average Reynolds' equation is derived through flow simulation which is based on numeric solution of the Reynolds' equation on a model bearing with a randomly generated surface roughness and then deriving the average Reynolds' equation from mean flow quantities as given in [9,10].

$$\frac{\partial}{\partial z} \left( \phi_z h^3 \frac{\partial P}{\partial z} \right) + \frac{1}{R^2} \frac{\partial}{\partial \theta} \left( \phi_\theta h^3 \frac{\partial P}{\partial \theta} \right) = -6\mu U_z \frac{\partial \bar{h}_T}{\partial z} - 6\mu U_z \sigma \frac{\partial \phi_s}{\partial z} + 12\mu \frac{\partial \bar{h}_T}{\partial t} \quad (2)$$

where,  $\phi_z$ ,  $\phi_\theta$ ,  $\phi_s$  are flow factors obtained by flow simulation.

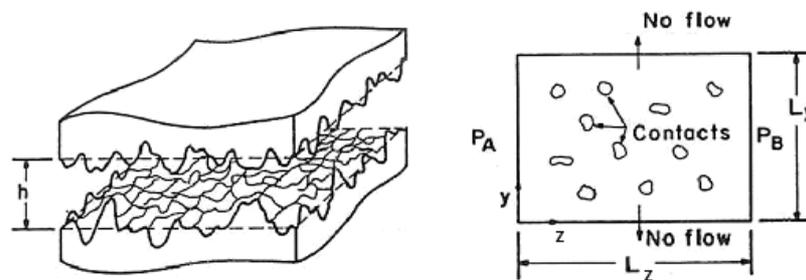
Engineering surfaces have directional properties resulting from different manufacturing processes or because of running-in. These directional properties are mostly in the longitudinal or transverse directions. The contacts of a general three dimensional surface that is partially lubricated can be modeled as ellipses with the mean ellipticity ratio  $\gamma$ . Flow factors are dependent on ellipticity ratio and nominal film thickness.

Longitudinally oriented contact areas ( $\gamma > 1$ ), offer little resistance to the pressure flow, permitting only a small side flow. Since the average gap in the valleys is greater than the compliance, the resulting flow is greater than a smooth surface bearing. The resistance to main flow together with main side flow increases as the surface becomes transversely oriented ( $\gamma < 1$ ). Increasing the resistance means thereby reducing the  $\phi_x$ . For isotropic surfaces ( $\gamma = 1$ ) the local side flow is of the same order of magnitude as the main flow. Hence the resistance to main flow also increases as the flow has to pass around the asperities.



**Figure 2.4** Typical contact areas for (a) longitudinally oriented surface  $\gamma > 1$ , (b) isotropic surface  $\gamma = 1$ , and (c) transversely oriented surface  $\gamma < 1$  [9]

For a given surface roughness characteristics, flow coefficients are calculated in Patir, N. and Cheng, H. S. [18,19] for the model problem shown in Figure 2.5.



**Figure 2.5** Model problem for simulation (Patir, N. and Cheng, H. S. [9])

$$\phi_z = \frac{\frac{1}{L_y} \int_0^{L_y} \left( \frac{h_r^3}{12\mu} \frac{\partial p}{\partial z} \right) dy}{\frac{h^3}{12\mu} \frac{\partial p}{\partial z}} \quad (3)$$

where

$$\frac{\partial p}{\partial z} = \frac{P_B - P_A}{L_z} \quad (4)$$

$$\phi_\theta = \phi_\theta(h, 1/\gamma) \quad (5)$$

$$\phi_s = \frac{2}{U_s \sigma} E \left( - \frac{h_r^3}{12\mu} \frac{\partial p}{\partial z} \right) \quad (6)$$

The solution of Reynolds' equation using finite difference formulation for the second order derivatives is straightforward. The average Reynolds' equation requires the discretization of the flow coefficient for the finite difference formulation. For the numerical solution of Reynolds' equation, the same approach described in [8] is used. The boundary contact friction, power loss and wear rate calculation carried are similar.

Piston dynamics parameters such as; piston axial velocity, piston axial acceleration, piston inertial behavior versus crank rotational speed, are solved by analyzing the slider crank mechanism.

The approach presented in [2] is used for the overall solution of the piston lubrication;

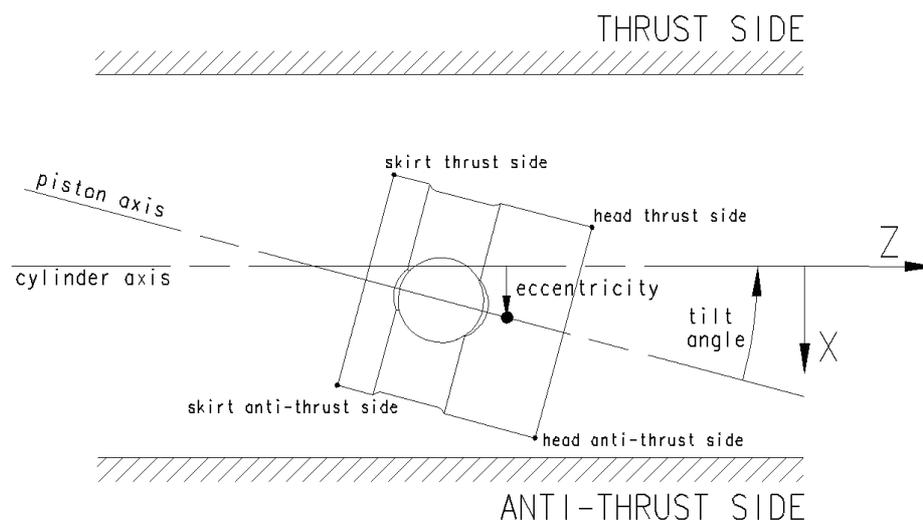
- Given cylinder pressure, slider crank dynamics are solved and the side load on the piston is calculated.
- For a given side load Newton Raphson Method is used for the solution of equations of motion.
- At every perturbation of the Newton Raphson piston eccentricity and piston tilt with respect to cylinder axis are estimated and the clearance between piston and cylinder is calculated.
- Knowing the film thickness around the piston, oil hydrodynamic pressure distribution is calculated by the solution of discretized form of Reynolds' equation.
- After integration of hydrodynamic and boundary contact pressures, the force balance in lateral direction and moment balance are solved.
- The iteration is continued until convergence, and the solution is advanced to next time step.
- The entire solution is repeated until cyclic convergence is achieved.

### 3. RESULTS AND DISCUSSION

It is well known that the oil film pressure formation is based basically on wedge effect and squeeze effect of two surfaces in relative motion. For a piston moving in cylinder hole, the wedge effect is achieved by the tilt of the piston axis with respect to cylinder axis and the change in eccentricity of the piston axis relative to cylinder axis. However, the squeeze effect is created due to the change in eccentricity.

In Figure 1 the nomenclature of the piston is given. For an exaggerated cylinder-hole clearance, the distance of center of mass of the piston from the cylinder axis is eccentricity. The angle between the cylinder axis and piston axis is piston tilt. Knowing these two parameters for different load cases in each crank angle, the distribution of film thickness around the piston can be calculated. Similarly the minimum film thickness, which defines the lubrication condition of the piston, for each crank angle can be found easily.

For the parametric study real data of a compressor manufactured by Arçelik Compressor Plant for different piston designs is used. The results describing the piston dynamics are given as the film thicknesses of the piston thrust plane head and skirt sides and piston anti thrust plane head and skirt sides.

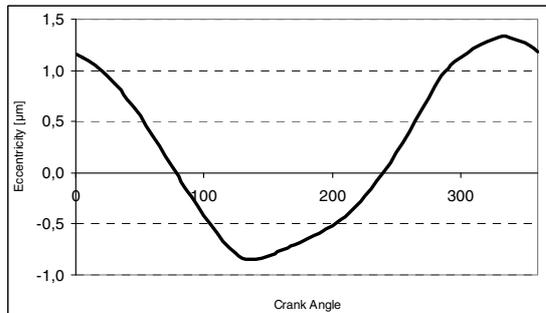


**Figure 1** Piston locations of which film thickness values are presented

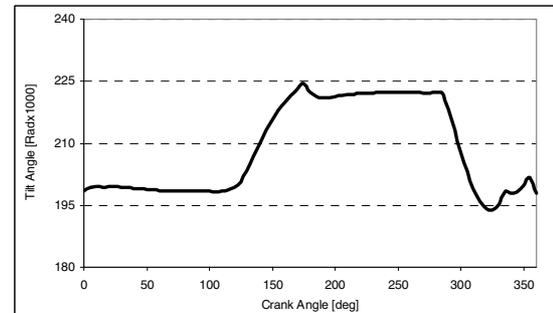
### 3.1 Results for Smooth Surfaces

The lubrication dynamics of the base piston model with smooth surfaces is given in Figures 2, 3, 4 and 5. A piston design that works in partial lubrication regime is selected since it is known that as the asperities of two adjacent surfaces get closer, the effect of surface roughness is more prominent.

In Figures 2 and 3 the tilt and eccentricity of the piston are given respectively. The variation of piston tilt is minimal and the primary motion is achieved by the change in eccentricity. As a result, it can be concluded that oil film formation is due to the squeeze effect formed by the change in piston eccentricity.

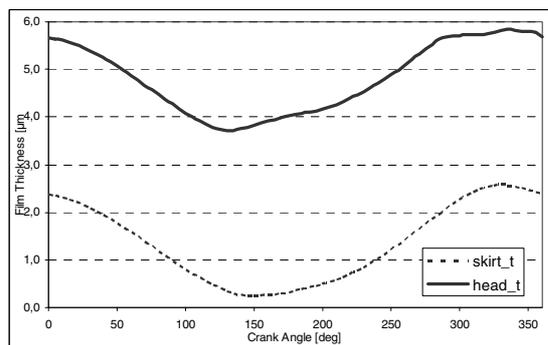


**Figure 2** Cyclic variation of piston eccentricity for a smooth surface

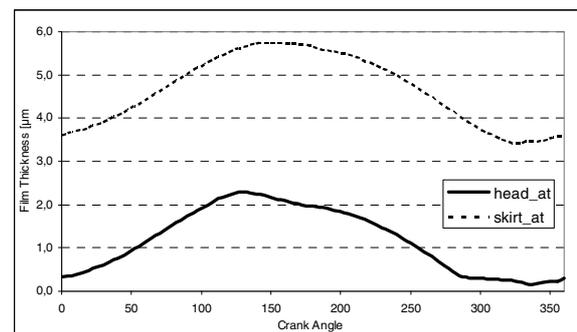


**Figure 3** Cyclic variation of piston tilt angle for a smooth surface

In Figure 4 and 5, piston skirt and head thrust and anti-thrust side the film thickness values for the piston-cylinder with smooth surfaces is given. When the figures are investigated it can be seen that the profiles of skirt and head sides are very similar with small differences. This is the result of small piston tilt variation illustrated in Figure 3. As a result the piston skirt and head sides' film thickness variations have almost the same profile.

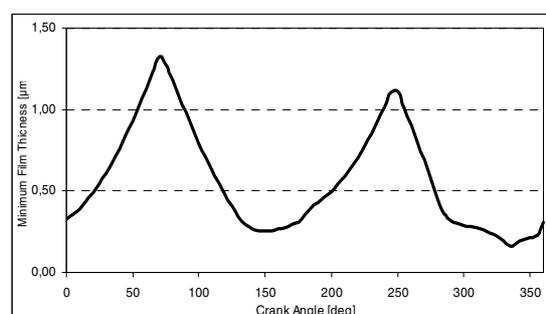


**Figure 4** Cyclic variation of piston thrust side skirt and head film thicknesses a smooth surface



**Figure 5** Cyclic variation of piston anti thrust side skirt and head film thicknesses a smooth surface

The minimum film thickness variation of the piston for smooth surfaces case is given in Figure 6. It is just the minimum values of Figure 4 and Figure 5. It is clear that the film thickness drops below  $0.2 \mu\text{m}$  where asperity contact will be significant. Hence the surface roughness must affect the lubrication for this piston design.

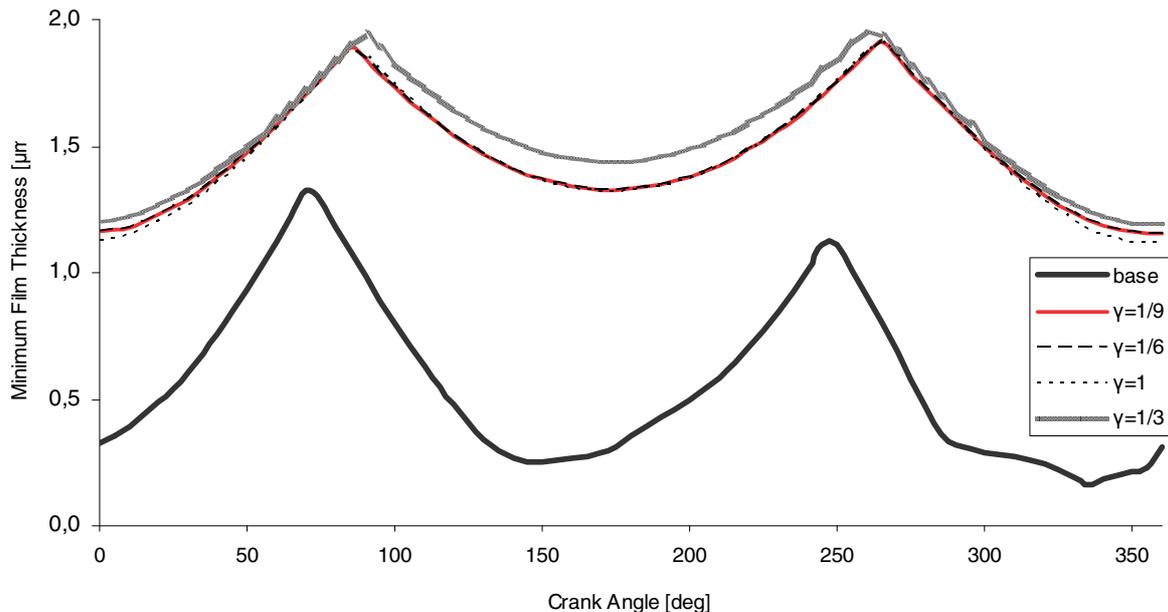


**Figure 6** Cyclic variation of minimum film thickness for a smooth surface

### 3.2 Results for Rough Surfaces

The effect of surface roughness is included numerically as explained in Section 2. The asperity orientation is given in Figure 2.4 for a longitudinally oriented surface  $\gamma > 1$ , for an isotropic surface  $\gamma = 1$ , and for a transversely oriented surface  $\gamma < 1$ . For different ellipticity ratios  $\gamma$ , the effect is investigated. The minimum film thickness variations for different surfaces are taken into consideration to understand the effect of surface roughness.

In Figure, 7 the minimum film thickness values are given for different transversely oriented roughness ( $\gamma < 1$ ) surfaces. As  $\gamma$  goes from 1/3 to 1/9 the surface has larger transverse direction.

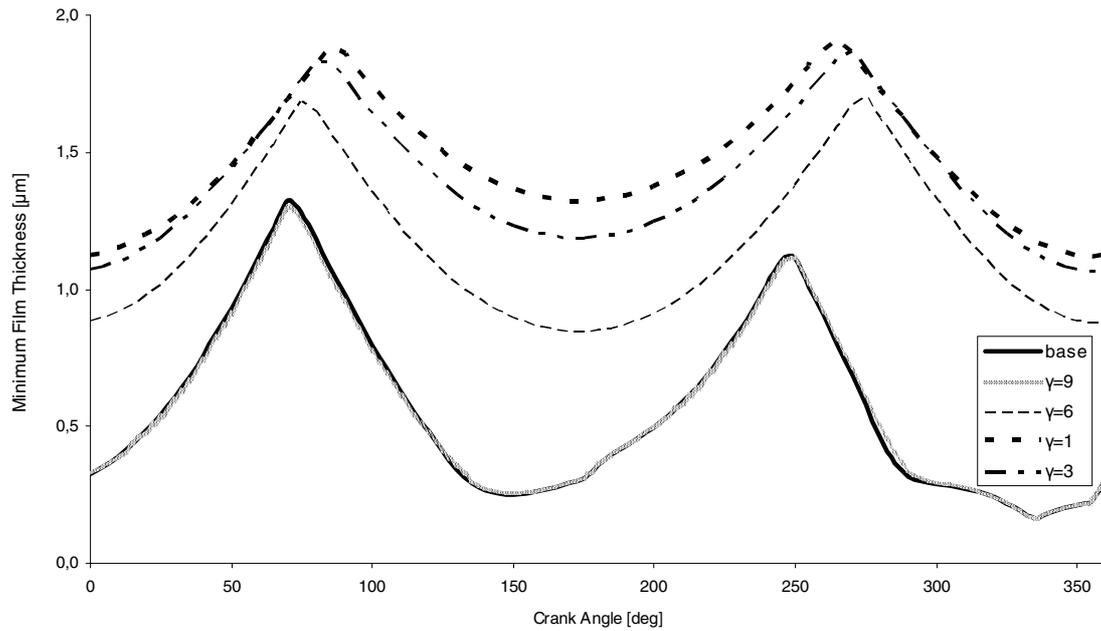


**Figure 7** Cyclic variations of piston minimum film thicknesses for different transversely oriented rough surfaces

From investigation of Figure 7, it is clear that the minimum film thickness is higher for rough surfaces with respect to baseline, which is the value of smooth surface. It is expected that the transversely oriented surfaces might decrease the oil film pressure as the asperities act as barriers which restrict the main oil flow. However as the oil film pressure is primarily due to the squeeze effect, the oil film restriction is beneficial. The oil film flow is still restricted, but for the squeeze motions this causes higher oil film pressures. As a result the minimum film thickness values are higher for transversely oriented surfaces. Moreover, the differences between different transverse orientations are minimal.

For longitudinally oriented surfaces ( $\gamma > 1$ ), the effect is shown in Figure 8. In this case different longitudinal surfaces have different effects. For an isotropic surface, the minimum film thickness values are higher and close to transversely oriented surface. As the longitudinal direction of the roughness increases, the minimum film thickness gets thinner and as it reaches its maximum value at  $\gamma = 9$ , the effect is minimum and film thickness is the same as the base piston model without roughness. As a result longitudinal roughness is still advantageous until it is highly longitudinal.

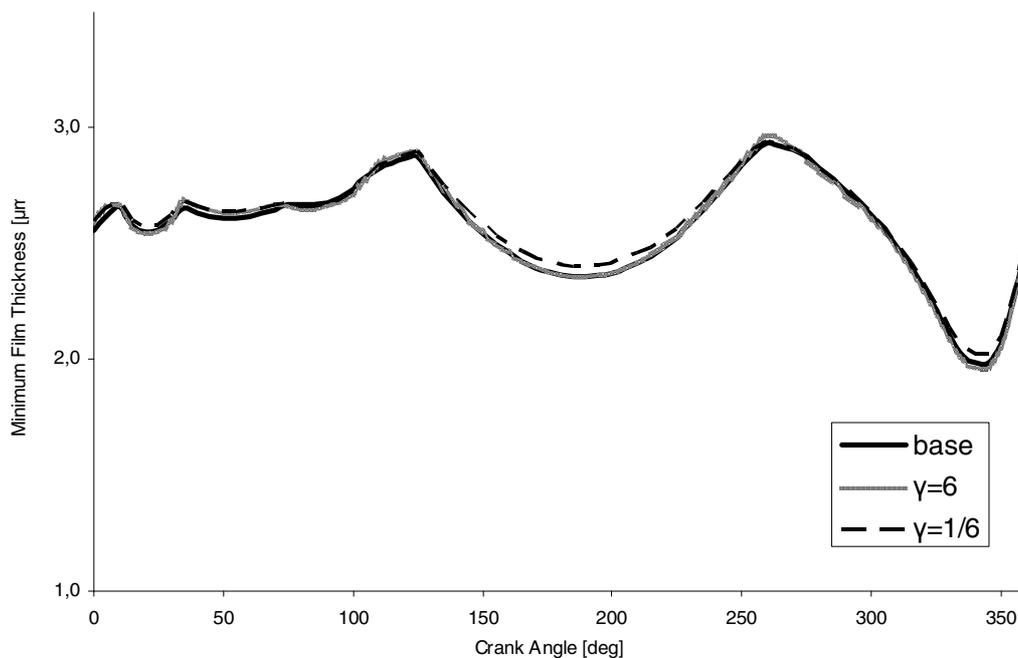
Because of manufacturing techniques, the roughness is transversely oriented on piston and cylinder whole. During operation, because of running in, a longitudinal character of the surface will form and the lubrication will worsen. This can be concluded by the comparison of roughness affects in Figures 7 and 8.



**Figure 8** Cyclic variations of piston minimum film thicknesses for different longitudinally oriented rough surfaces

### 3.3 Results for Rough Surfaces in Hydrodynamic Lubrication Regime

A different design in which there is no surface contact is investigated for rough piston-cylinder surfaces in a comparison with smooth surfaced piston-cylinder model. Since working in hydrodynamic lubrication regime, the minimum film thickness is higher with respect to RMS surface roughness, it is expected that the surface texture effect to be minimal.



**Figure 9** Cyclic variations of piston minimum film thicknesses for hydrodynamic lubrication operating conditions

It is clear in Figure 9 that there is no significant difference between the baseline piston model with smooth surfaces and roughness effect included models. For longitudinally oriented roughness value  $\gamma=6$  and transversely oriented roughness surface  $\gamma=1/6$  the minimum film thickness values given in Figure 9 are very similar to the baseline piston case.

#### 4. CONCLUSION

For the piston-cylinder bearing model developed, a parametric study for different surface roughness texture models is carried out. In order to understand the effect of different surface textures, two different piston models working in partial lubrication and hydrodynamic lubrication regimes are selected. For rigid surface assumption, longitudinally and transverse oriented textures are studied.

The effect of surface texture on hydrodynamic lubrication is minimal; however, it is more effective for the partial lubrication regime. The expectation of transversely oriented surfaces is to decrease the film pressure hence to decrease the minimum film thickness since the asperities act as barriers and restrict main oil flow. But for the case in the current study, oil film pressure is created due to the squeeze effect; hence the oil flow restriction in such a case is advantageous. As the surface roughness orientation becomes longitudinal this effect still continues but in a decreasing manner with increasing ellipticity ratio. Finally for highly longitudinally oriented surface, this advantage disappears and the piston works like the smooth surface.

For a small size piston with small clearances between piston-cylinder, roughness in transverse orientation is beneficial. In addition, the manufacturing techniques of piston and cylinder cause transverse oriented texture surfaces. But because of running in and wear during extended operation time, the asperities will shape into a new longitudinal form which will worsen the lubrication, resulting in a decreased the minimum film thickness.

#### NOMENCLATURE

$c$	radial clearance	m
$e$	piston eccentricity	m
$h$	nominal film thickness	m
$hT$	local film thickness	m
$h$	average gap	m
$P$	hydrodynamic oil pressure	Pa
$R$	piston radius	m
$t$	time	s
$T$	period of one cycle	s
$U, U_z$	linear velocity of the piston	m/s
$z$	piston axial position	rad
$\lambda$	piston tilt angle	rad
$\mu$	lubricant viscosity	Pa.s
$\theta$	angular coordinate on the piston	rad
$\varphi_x, \varphi_y$	pressure flow factors	
$\varphi_s$	shear flow factor	
$\varphi$	ellipticity ratio	
$\sigma$	standard deviation of combined roughness	

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