Simulation and Modeling of Friction Force and Oil Film Thickness in Piston Ring – Cylinder Liner Assembly of an I. C. Engine

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Abstract-**The piston ring assembly is one of the key parts of internal combustion engine. Its performance decides engine power loss to over come friction force and determines the performance of the whole engine. As per the literature review it is reported 40 to 50 % of the total mechanical friction losses in piston ring - cylinder wall interface.**

The present work mainly focused on study of basic tribological parameters that influences performance of an internal combustion engine. Mathematical model is developed using average Reynolds equation. Parametric study is performed on 150 CC, 2 Stroke Internal Combustion Engine. The oil film thickness (OFT), piston friction forces (PFF), and Ring friction variations are simulated under different variable i.e engine speed, lubricants and different ring geometry. The simulated results of piston friction force, ring friction force and oil film thickness are compared with published literature.

Key wards: **Oil film thickness, Piston friction force, piston, ring friction force**

I INTRODUCTION

A significant contribution of the total power loss in a reciprocating engine is due to piston ring and cylinder liner friction. On the other hand, durability of engine materials for improved friction and wear characteristics remain an area of strong research. A set of piston rings is used to form a dynamic gas seal between the piston and cylinder wall. The sliding motion of the piston forms a thin oil film between the piston ring and cylinder wall, which lubricates the sliding components. The hydrodynamic force generated by this thin oil film is opposed by a combination of the gas pressure acting on the back side of each ring and the ring stiffness. Due to the dynamic nature of these forces, each individual ring is periodically compressed and extended during piston movement.

The study of the friction between a lubricated piston ring and cylinder wall has naturally attracted many investigators, who have sought to establish the nature of the oil film under the different conditions, and to determine variation of friction force varies with the piston speed, oil viscosity as well as the design of the ring itself. The main aim of this work is to determine the friction force occurring at the piston ring and predicting the oil film thickness during the movement of the piston.

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II PRA FRICTION REVIEW

The First calculations on piston ring and cylinder liner lubrication were made by Castleman [1]. Eilon and Saunders [2] calculated the lubricant film thickness based on the balance of radial forces. The squeeze film effect was incorporated into the analysis by Furuhama [3]. He considered the ring surface model as two circular arcs connected by a flat section. Ting and Mayer [4 & 5] developed an analytical model for determining the ring-bore wear mechanism for a reciprocating piston engine over a complete running cycle. They used hydrodynamic lubrication theory to analyze the flow between ring and cylinder bore.

Patir and Cheng [6 & 7], Greenwood and Tripp [8] modified the average Reynolds equation for rough surfaces. They defined pressure and shear flow factors, which were obtained independently by numerical flow simulation using randomly generated or measured surface roughness profiles. This approach has been used by several authors in order to predict the lubricant film thickness of rough cylinder liner surfaces in piston ring and cylinder liner contact.

Rohde *et al.* [9] used Patir and Cheng’s [6] averaged Reynolds equation and developed a mathematical model to study friction performance of dynamically loaded contacts operating in the Hydrodynamic / mixed lubrication regime. They applied the model to piston ring lubrication including flow factors and the contribution of the asperities. They concluded that piston ring friction is dependent on surface topography when contact is in the mixed lubrication regime.

Priest and Taylor [10] have reviewed the tribological design and friction associated with the tribological components of the engine with a specific focus upon surface topography and surface interaction considerations. They have found that the film thickness ratio and the surface topography have a significant role in the performance and durability of engine components. Mufti *et al.* [11] have developed a theoretical model for estimating the oil film thickness, based on the assumptions that the surfaces of the liner and the rings are smooth and have good circumferential conformity. The ring lift during the sliding is neglected, thereby assuming the axial velocity of the piston ring and the axial velocity of the piston are same. Priest *et al.* [12] have predicted film thickness throughout the engine cycle. The fully flooded model of lubricant flow in a ring pack assumes that there is an unlimited supply of lubricant available to each ring at all stages in the engine cycle such that the inlet region of the ring profile is always full of lubricant. However, in reality, they have observed that the quantity of lubricant available to each ring is the thin film smeared on the cylinder wall by preceding ring and consequently the inlet region of the ring profile may starve.

The majority of the literature on prediction of the oil film thickness in Piston Ring assembly is based on assumption of the smooth contact surface. This is mainly because, in hydrodynamic lubrication regime, if the oil film thickness is nearly 3 to 4 times greater than the standard deviation 'σ' of a Gaussian distribution of heights of surface roughness, then the effect of surface roughness can be conveniently neglected. However, in a piston ring assembly, the variations in lubrication regimes varies from boundary to mixed and hydrodynamic during the travel from TDC to BDC and thus develops the complex lubrication regimes.

Rohde [9] used the "averaged" Reynolds equation as developed by Patir and Cheng [5 & 6], which takes accounts of the surface topography. The estimation of the average film thickness given as,

$$\bar{h}_T = \int_{-h}^{\infty} (h+\delta).f(\delta).d\delta \qquad (2.1)$$

The integration of the probability Function varies from $-\delta_{min}$ to $+\delta_{max}$, and it is a complementary error function, whose solution as per Maclaurin series results into an approximation.

$$f(\delta) = \frac{1}{\sigma\sqrt{2\pi}} e^{-(\delta^2/2\sigma^2)} \qquad (2.2)$$

Figure 1 indicates the oil film thickness function with respect to cylinder liner and piston ring contact surface, the average film thickness,

$$h_T = h + \delta_1 + \delta_2 \qquad (2.3)$$

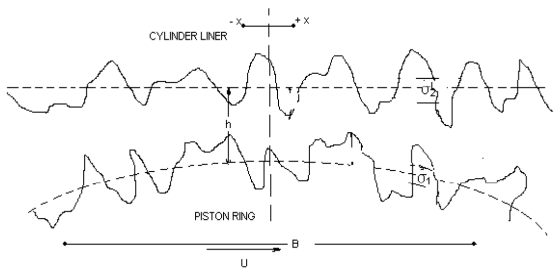

Figure 1 Oil film thicknesses with respect to liner [15]

The combined roughness variance

$$\sigma = [(\sigma_{ring})^{2} + (\sigma_{liner})^{2}]^{0.5} \qquad (2.4)$$

Hydrodynamic lubrication regime is exist till $h/\sigma > 3$, when $3 > h/\sigma > 1$, effect of surface roughness becomes important due to interacting of asperities of two surfaces and lubrication regimes may either mixed or boundary. The model of average oil film thickness given by Mistry and Priest as under

$$h_T = h - \frac{\sigma}{\sqrt{2\pi}} \qquad (2.5)$$

III AVERAGE REYNOLDS EQUATION

It is assumed that a thin oil film separates the compression rings from the liner and thus Reynolds equation can be used to determine the film thickness throughout the engine cycle. For predicting the oil film thickness by solving the Reynolds equation, the shape of the piston ring face in the direction of sliding, piston ring sliding speed, piston ring loading and lubricant viscosity must be known. The modified Reynolds equation as given by Patir and Cheng [6] is considered taking an account of the influence of surface Roughness through a series of flow factors. i.e

$$\frac{d}{dx}\left(\varphi_x (\bar{h}_T)^3 \frac{dp}{dx}\right) = 6\eta U \frac{d\bar{h}_T}{dx} + 6\eta U\sigma \frac{d\varphi_S}{dx} + 12\eta \frac{d\bar{h}_T}{dt} \qquad (2.6)$$

where,

$$U = r\,\omega\left(\sin\varphi + \frac{\lambda}{2}\sin 2\varphi\right), \text{m/sec}$$

Φ_x = Pressure Flow Factor = 1- 0.9 exp $^{(-0.56H)}$ as $H \to \infty$, $\Phi_x \to 0$

Φ_s = Shear Flow Factor = 1.12*e$^{-(0.256H)}$, H > 5, or = 1.899 H 0.98 *exp$^{(-0.92*H+0.5\ H\char`^2)}$ as $H \leq 5$

IV MODEL DEVELOPMENT FOR PARABOLIC RING

Lubrication of the ring was analyzed using the traditional approach based upon numerical solution to the modified Reynolds' equation. It is briefly summarized here for completeness. The top ring was represented by a parabola, while the scraper ring was represented by a plane incline slider bearing. The contacting faces of the oil control ring rails are assumed to be sections of cylinders, so that each rail is considered as a parabolic ring.

The following assumptions were considered in the model:

- Body forces are neglected i.e there are no extra fields of forces acting on the lubricant.
- The curvature of surfaces is large compared with film thickness surface velocities need not to be considered as varying in the direction.
- The lubricant is Newtonian i.e stress is proportional to rate of shear.
- The viscosity is constant through out film thickness.
- The Reynolds hydrodynamic lubrication concept is applicable to piston ring assembly system.
- Piston ring dimensions are assumed to be constant for width, axial height, length, outer and inner diameter, clearance between the ring and piston etc.
- Piston – cylinder assumed to be perfect concentric assembly.

The parabolic profile causes high pressure to be built on the converging portion of the ring face, but hardly any pressure generated on the divergent portion as shown figure 2. In these circumstances only about half of the rings area is responsible for ring friction and the fact is considered in derivation of the friction force expression.

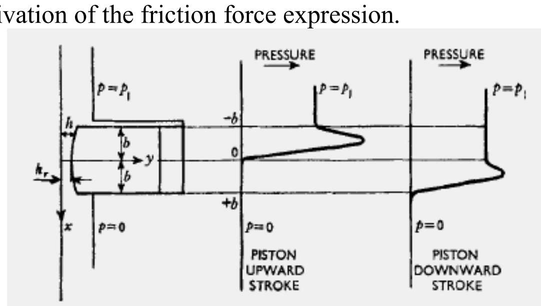

Figure 2. Parabolic ring face and pressure distribution.

Using one dimensional Reynolds equation for hydrodynamic flow for parabolic ring profile, the integrated expression for the pressure distribution on the ring's face, as

$$\frac{dp}{dx} = 6 * \eta * U * \left(\frac{1}{h^2} + \frac{c}{h^3}\right) \tag{2.7}$$

$$p = 6 * \eta * U \int \left(\frac{1}{h^2} + \frac{c}{h^3}\right) dx \tag{2.8}$$

For curvature ring h is expressed as, $h = h_r * \left(1 + \frac{x^2}{2Rh_r}\right)$

When the integration of above equation is performed after substituting for h, the following expression for the pressure is obtained as under,

$$p = 6*\eta*U*\left[\begin{array}{l}\frac{1}{2h_r^2}\frac{x}{1+\frac{x^2}{2Rh_r}}\left(1+\frac{3}{4}\frac{c}{h_r}\right)+\frac{c}{4h_r}\frac{x}{\left(1+\frac{x^2}{2Rh_r}\right)^2}+ \\ \left(1+\frac{3}{4}\frac{c}{h_r}\right)\frac{\sqrt{2Rh_r}}{2h_r^2}\tan^{-}\left(\frac{x}{\sqrt{2Rh_r}}+c_1\right)\end{array}\right] \tag{2.9}$$

The magnitude of constant c and c_1 is determined by the boundary conditions, maximum pressure will occur when,

(dp/dx)=0, $x = \sqrt{-2Rh_r\left(1 + \frac{c}{h_r}\right)}$ (2.10)

When the piston is upward stroke, approximate boundary conditions are $p = p_1$ at x = + b and p = 0 at x = 0,

The load capacity of ring is

$$W = \pi * D \int_0^b p\,dx \tag{2.11}$$

and the mean pressure exerted on the ring face, by oil film can be given as,

$$p_m = \frac{W}{2b\pi D} = \frac{1}{2b}\int_0^b p\,dx \tag{2.12}$$

This expression may be transformed in to a dimensionless form after multiplying both sides by R.

$$P_m \equiv \frac{p_m * R}{6*\eta*U}, \quad r \equiv \frac{h_r}{R}, \quad H \equiv \frac{b}{\sqrt{2Rh_r}} \tag{2.13}$$

During Upward Stroke,

$$P_m r^{3/2} * \left[1 - \frac{p_1}{2p_m} f(H)\right] = 0.3536 * \left[\left(\frac{H}{1+H^2} + \tan^{-}H\right) f(H) - \tan^{-}H\right] \quad \text{(A)}$$

$$\text{where, } f(H) = \frac{H + 3*(1+H^2)*\tan^{-}H}{5H + 3H^3 + 3(1+H^2)^2\tan^{-}H}(1+H^2)$$

Boundary condition under downward stroke, $p = p_1$ at x = 0, and P = 0 at x = - b, Hence, the mean pressure on the ring's face is now

$$p_m = \frac{1}{2b}\left[\int_{-b}^{0} p\,dx + p_1 b\right] \tag{2.14}$$

The limits between which the integration is performed as

$$\frac{p_m}{6\eta U} = -\frac{1}{4h_r^2}\left(1 + \frac{3}{4}\frac{c}{h_r}\right)\sqrt{2Rh_r}\tan^{-}H - \frac{bc}{16h_r^3}\frac{1}{H^2} + \frac{p_1}{6\eta U} \tag{2.15}$$

The final result in the nondimentional form during downward stroke

$$(p_m - p_1) r^{3/2} * \left[1 + \frac{p_1}{2(p_m - p_1)} f(H)\right] = 0.3536\left[\left(\frac{H}{1+H^2} + \tan^{-}H\right) f(H) - \tan^{-}H\right] \quad \text{(B)}$$

Friction force Calculation:

The friction forces between ring face and cylinder wall in the mixed lubrication regimes result from the viscous shearing force of hydrodynamic film, the horizontal components of contact pressure between asperities and friction between asperities. The friction force for a unit segment can be written as the shear stress on the piston assembly is presented by $\tau = \eta * U\left(\frac{4}{h} + \frac{3c}{h^2}\right)$

The piston friction is obtained by substituting $h = h_p$ giving,

$$F_p = A_p * \eta * U\left(\frac{4}{h} + \frac{3c}{h^2}\right) \tag{2.17}$$

During the stroke, pressure is building up to convergent part of the ring with hardly a pressure on the divergent part as shown in the figure 2.

$$F = \pi * D \int_0^b \tau \, dx \quad \text{and giving the force}$$

$$F_r = \pi * D * \left[\frac{3cb}{2 * h^2 \left(1 + \frac{b^2}{2Rh_r} \right)} + \left(4 + \frac{3c}{2h_r} \right) \frac{\sqrt{2Rh_r}}{h_r} \tan^{-} \left(\frac{b}{\sqrt{2Rh_r}} \right) \right] \quad (2.18)$$

The computer program has been developed to compute the oil film thickness, piston friction and piston ring friction for the single cylinder, two stroke petrol engine for the complete working cycle at the crank angle interval of 10^0. The 'C' program starts with the initial assumption of minimum lubricant film thickness. The input parameters are shown in the table 1. Lubrication inlet-outlet pressures are given as inputs to the program for each crank angle degree changing the direction of the pressure depending on the direction of the piston motion.

Table 1 Input parameters:

Cylinder bore diameter (m)	0.056
Crank Radius (m)	0.050
Connecting rod length (m)	0.110
Ring Thickness axial direction (m)	0.002
Effective ring thickness in axial direction(m)	0.001
Piston ring tension (N)	10.5
Gas pressure (N/m^2)	p_1=7.0*10^5, p_2=1.103*10^5
Young modules of elasticity, E (N/m^2)	2.01*10^{11}
Nominal piston clearance, C_b (m)	2.5*10^{-4}
Ring surface roughness, σ_{ring} (μ m)	0.15
Cylinder liner surface roughness, σ_{liner} (μ m)	0.785
System variables:	
Operating speeds (rpm)	500,750, 1000, 1250,1500
Lubricants	2 T , SAE 20, SAE 30

V RESULT AND DISCUSSION

The model presented is used to simulate the lubrication and friction force behaviuor of the first compression ring of a single cylinder petrol engine.The piston Velocity, piston friction force (PFF), ring friction force (RFF), and average minimum oil film thickness(OFT) are predicted using the proposed model for the single cylinder reciprocating system of engine.

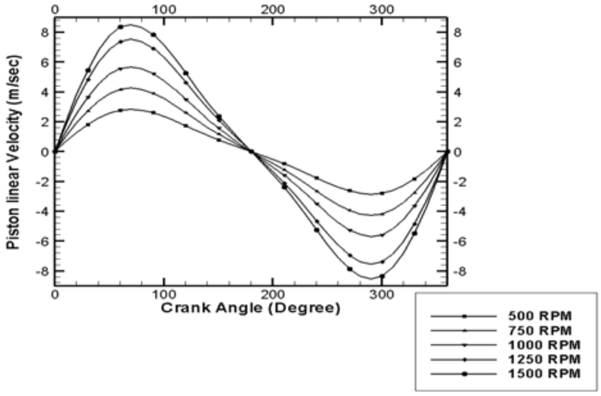

Figure 3 Piston velocity v/s crank angle

It is observed from figure 3 that as engine speed increases from 500 rpm to 1500 rpm, in the step of 250 rpm increment of speed, the piston velocity increases in the sinusoidal nature. It means that apparently constant rpm rotation, the piston undergoes acceleration / retardation motion. This variation plays important role in film thickness.

Oil film thickness with respect to crank angle:

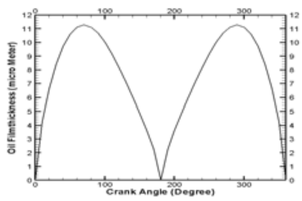

Figure 4 Oil film thickness V/s Crank angle

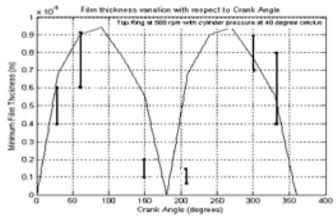

Figure 5 Oil film thickness V/s Crank angle, [14].

The figure 4 shows the calculated cyclic variation of the oil film thickness between the ring and cylinder liner with the crank angle and is noted maximum 12 micron at the middle of the stroke at 1500 rpm of the engine speed, where the sufficient oil could be existing. This trend of the curve is similar nature with that of published literature [14]. The figure 6 indicates the ranking of the oil, and 2 T oil offers better film thickness compared to others.

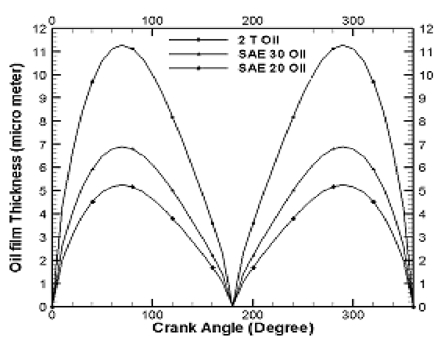

Figure 6 Comparison of oil film thickness V/s Crank angle, at 1500 rpm.

PFF v/s crank angle Comparison:

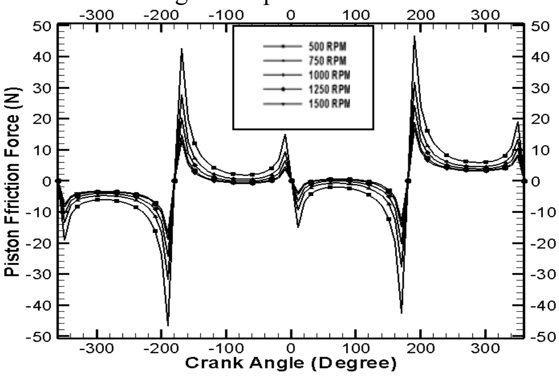

Figure 7 Piston friction force V/s Crank angle, at variation of engine Speed.

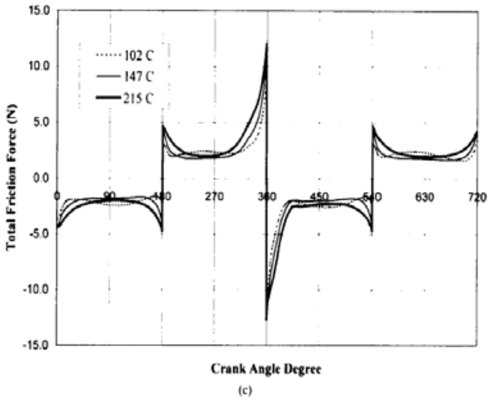

Figure 8 Friction force V/s Crank angle, [16].

The figure 7 shows the comparison of the piston friction force under variation of engine speed from 500 rpm to 1500rpm v/s crank angle. The common observation may be drawn that all nature of curves are similar. The nature of the curve is observed in line with the published work [16] as shown in figure 8.

Ring Friction Force:

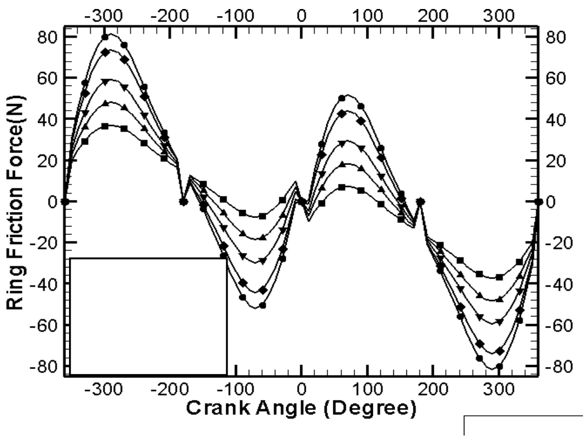

Figure 9 Piston ring friction force V/s Crank angle, with variation of the engine speed.

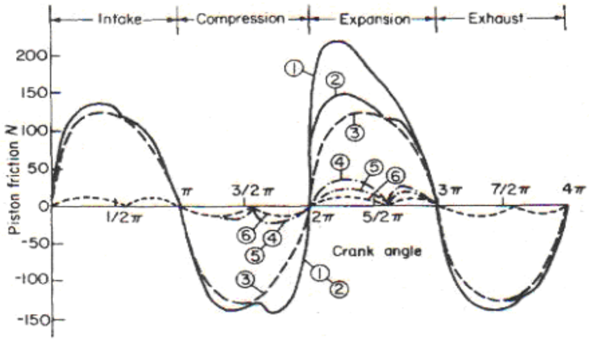

Figure 10 Ring friction force v/s crank angle (graph no. 3) [17].

The figure 9 indicates the comparisons of the piston ring friction force with the variation of the crank angle with variation of the engine speed from 500 rpm to 1500 rpm. The RFF in piston liner assembly comes higher as the engine speed increases. This result is good agreement with published result [17] as shown in the figure 10.

VII CONCLUSIONS

Under the Mathematical model proposed and simulation along with result comparison is made with the published literatures.

Oil film thickness:
The oil film thickness maximum at the middle of the stroke where the hydrodynamic lubrication and at the dead centers is minimum because of the boundary or mixed lubrications. The simulated results are in good agreement with reported by researcher Mistry [14].

Piston friction force:
The variation of Piston friction force is decreases linearly with increases engine speed. The initial part of the curve is negative sign this could be of the piston moves in the upward direction and friction force in downward direction. The nature of simulated graph at different speed range is in good agreement with work reported by Zeng *et al.* [18]. It confirms the validation of simulated models.

Ring friction force:

The nature of the simulated result of ring friction force at different crank angle is in the form of the sinusoidal nature. As the piston in the downward motion, the ring friction force acting in the upwards direction during the half working cycle. The next half cycle, it shows negative direction. It is also in good agreement with work reported by Hoshi M [19]. Hence it confirms the proposed mathematical model.

ISBN:978-988-18210-1-0

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