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Prediction of Oil Film Thickness in Piston Ring - Cylinder Assembly in an I C Engine: A Review

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Abstract **-I C engines are the largest energy converting units for the mobile application across the globe and millions of litres of oil is converted into mechanical energy using these engines every day. It has been recognized that a large amount of energy is lost in the friction between Piston ring – Cylinder assembly. Hence this area has been the interest of research for decades. Oil film thickness formed between the piston ring and the liner has a major impact on reduction of power loss due to friction. A micro review on prediction of Oil Film Thickness in piston ring-cylinder assembly has been presented in order to have an over view of the research going on in this area. An attempt has been made to incorporate the research work of leading researchers and to focus light on the major parameters affecting OFT and their impact on the OFT. This review may be helpful for new upcoming researchers to identify the areas for future research.**

Index terms-: **Tribology, Oil film thickness, Piston ring – cylinder assembly, I C engine**

I. INTRODUCTION

Oil film thickness has been predicted by renowned researchers by identification of important variables affecting OFT and assumed certain parameters as constant having insignificant impact on OFT while developing various models. Some of the major parameters affecting the oil film thickness are, piston speed, lubricant viscosity, ring face profile, boundary conditions, surface roughness, effect of ring twist, bore distortion and ring flexibility. All the aforementioned parameters have their own effect on the oil film thickness, thus for a designer it is important to have a trade-off between all the parameters.

Reynold's equation of hydrodynamic lubrication, Stribeck curve of journal bearing and mass conserving algorithm are used for analysis and simulation of various models.

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II. LITERATURE REVIEW

Sreenath & Venkatesh[1] in their analytical work have predicted the oil film thickness for a four stroke engine at full load and no-load conditions and have compared with the results of other researchers. The film thickness is predicted by solving Reynold's equation considering the "Squeeze film effect" at TDC and BDC where it is more significant. Full flooded lubrication is assumed for the entire stroke length.

Ting & Mayer[2]&[3] developed theory including the considerations of the hydrodynamic lubrication between the rings and the cylinder liner, piston ring geometric and elastic characteristics, gas blow-by through the ring pack, minimum film thickness permitting film lubrication, piston side thrust load and Archard's wear relation. The ring face profiles for four engines with various mileages have been measured and were found to be offset parabola. A general off centered parabola is used to represent the profile of the ring running surface.

It was assumed that there is no pressure change in the divergent portion of the ring wedge in both strokes. It could be concluded that a highly curved ring has a very large film thickness at the center of the stroke, where the wedge action is predominant. The film thickness falls very rapidly at the ends of the stroke where the action changes to the squeeze film type. The flattest ring never achieves a large film thickness, even at mid-stroke, but the fall at the ends of the stroke is less severe. Hence, there must be an optimum ring shape profile existing between the highly curved and flat ring face which will give the greatest film thickness at the critical points near the top and bottom dead centers.
The work of Takiguchi[4] is mainly related to the study of friction forces in the engines however the authors have also touched the issue of oil film thickness.
In one experiment using special piston, oil holes were provided so that a sufficient amount of oil could be supplied to each ring per cycle to ensure the film thickness formation throughout the stroke.

The authors maintain that with three standard ring package plus three-piece oil ring at 200 rpm the force characteristic is mainly non-hydrodynamic lubrication at bottom dead center. At 4000 rpm peaks of friction forces at the bottom dead center disappear and mainly hydrodynamic lubrication is seen throughout all strokes of the engine operation. This is attributed to the increased oil film thickness between the ring and the cylinder due to the high piston speed resulting increase in hydrodynamic fluid pressure.

Zhou et al.[5] have proposed a model to simulate the effect of one dimensional roughness of the piston ring surface on lubrication and friction based on stochastic theory. The applications of the model to an actual diesel engine indicate that about 8.3%-9.4% increase in friction power loss can be expected when the rough surface (d = 0.6 μm) rather than the smooth surface is considered.

Miltsios et al.[6] assumed that the ring have a circular profile in the direction of motion and assumed the bore to be elliptic. Tilting of the ring is also taken into account. When the oil film is thick enough so that there is no surface to surface contact of the slider and the plane, the lubrication is hydrodynamic. When the oil film is not thick enough there is surface to surface contact (contact of the asperities of the two surfaces) and the lubrication is mixed because part of the load is carried by the oil film and part of it by asperities. The oil film thickness beyond which hydrodynamic lubrication exists cannot be determined accurately. It depends upon the topography of the surfaces and height of the asperities.

Hwu and Weng[7] have used a non-linear finite-element scheme, based on the Newton-Raphson-Murty algorithm, to analyze the problems of piston rings according to theories of both elasto-hydrodynamic and hydrodynamic lubrication. The maximum film pressure and the minimum film thickness were obtained and depicted as functions of the crank angle.

Wakuri et al.[8] assumed that the oil-film pressure at the oil inlet is equal to the ambient pressure, and the oil film breaks down on the downstream side according to the Reynolds boundary condition. After the oil film breaks down, the oil is carried into a cavitated region. The pressure in the cavitated region is constant with respect to the ambient pressure. When the piston rings are used as a ring pack, the oil supplied to each ring is dependent on the amount of oil left on the liner by the preceding ring. The subsequent rings presumably operate in a starved condition where the inlet region of a ring is incompletely filled with oil even if the leading ring is fully flooded with oil. Therefore, the analysis that the oil is copiously supplied to the rings at all times does not seem reasonable. The interaction between the rings in a ring pack is determined by the condition of oil-flow continuity if the oil is neither supplied nor extracted between the rings.

Jeng[9]&[10] derived a system of three equations to describe the starved lubrication condition. The system takes the amount of lubricant supplied as an input and treats the lubricant inlet position as an unknown. The lubricant inlet position can be obtained directly by solving the system of equations. Thus, a direct approach is provided to conduct the starved lubrication analysis with a given lubricant supply. To determine the lubricant availability for the starved lubrication analysis a postulate for the lubricant transport phenomena in a complete ring pack is proposed. This postulate considers the flow continuity, oil accumulation and the relative location of the rings in a ring pack. In some instances, especially when squeeze motion dominates, there would be no cavitation. The exit boundary condition will then be: $p(b/2,t) = p_T$ where, p is hydrodynamic pressure at a point along the width of the piston ring; b is width of the piston ring and p_T is Pressure on the trailing edge.

It is shown that boundary lubrication as indicated by film thickness ($\sigma_{comp.}$=0.37 μm) occurs in the vicinity of either top or bottom dead center due to very small sliding velocity. This qualitatively agrees with the wear pattern observed on cylinder liners which show higher wear at the top and bottom of the stroke. Effect of engine speed, ring tension, ring width, crown height and offset on power consumption has been presented.

Hu et al.[11] presented this paper with an objective to develop a more realistic non-axisymmetrical model for predicting friction and lubrication behavior of the piston ring, including the effect of various factors such as surface roughness, asperity contact, elastic asymmetry of the piston rings, static distortion of cylinder bore, variation of gas pressure in inter–ring space and axial movement of the ring in groove. The surface roughness effect is considered with the average Reynold's equation while the asperity contact pressure is calculated by Greenwood-Tripp model.

Shuzou et al.[12] have developed the Scanning Laser-Induced-Fluorescence method to measure oil film thickness on piston rings in real operating engines with high precision. Predicted oil film thickness for each ring and the friction force has been compared respectively with those measured by the scanning LIF method and the floating liner methods. The starved inlet boundary conditions have been employed for oil film on each ring based on the experimental results.

Han & Lee[13] also considered a partially lubricated ring, the following conditions are presupposed; oil starvation is applied to the inlet region and the open-end assumption to the outlet region. This algorithm confirms flow continuity and permits the pressure to go down to the saturation pressure. Using these new boundary conditions, the actual effective width participating in ring lubrication is determined and the minimum film thickness and flow rate for the ring pack can be calculated.

Gulwadi[14] considered reattachment of the oil film with the ring due to the Jakobsson-Floberg-Olsson (JFO) boundary condition after the film detachment. Over an engine cycle the scraping of rings against the liner causes oil to accumulate at the leading and trailing edges of each ring. During the flow of gases through the land/groove regions, a fraction of this lubricant is transported by these gases toward the combustion chamber during blow-back. Additionally, a fraction of the oil accumulation above the top ring is discharged toward the combustion chamber by throw-off due to inertia. The study conducted by the author involves the implementation of a mass-conserving (cavitation) scheme in the solution of the one-dimensional hydrodynamic lubrication equation for the piston ring. This scheme, in conjunction with a boundary lubrication model and oil transport model, facilitates the computation of the volume accumulation of oil for the ring at its leading and trailing edges. The trailing height of the oil film behind the ring is also computed, which is used in the estimation of the axial oil film distribution on the liner at each instant. From this information the amount of oil available for lubricating the ring at each instant of the cycle can be calculated.

Akalin & Newaz[15] investigated the effects of running speed, normal load, contact temperature and surface roughness both experimentally and numerically for conventional cast-iron cylinder bores. The author has shown the effect of two contact temperatures: 24°C and 70°C on the oil film thickness. Higher temperatures lead to lower viscosity. Therefore, lubricant film thickness and load carrying capacity of the oil film decrease significantly with increasing temperature. Mixed lubrication occurs throughout the stroke at the higher temperature 70°C and the tendency to cavitate increases over most of the stroke.

Bolander et al.[16] used twin fiber optic displacement sensors to accurately measure the lubricant film thickness experimentally. The analytical model developed by them can capture the different lubrication regimes that the piston ring and liner experience. They developed a model to investigate the lubrication condition between a piston ring and cylinder liner. It was assumed that the contact operates under fully flooded conditions except at the top and bottom dead centers and the mode of cavitation is closed form. A significant amount of surface interaction may occur near TDC and BDC where hydrodynamic action is at a minimum. In these mixed lubrication regions the applied load will be balanced by contributions from both the fluid pressure and the surface asperities in contact.

Bhatt[17] has, in his Ph. D thesis presented the effect of speed, viscosity and ring profile on co-efficient of friction. The effect of viscosity is brought about by using different grades of oil.

Jaana[18] used a commercial six-cylinder, medium-speed diesel engine as a test engine to investigate the oil film thicknesses between the compression rings and the cylinder liner at different load conditions. The engine speed for all measurements was 900 rev/min.

Harigaya et al. [19] have analyzed the effect of lubricant viscosity and temperature on the thickness of oil film between piston ring and cylinder liner in a diesel engine by using unsteady state thermo-hydrodynamic lubrication analysis, i.e., Reynolds equation and an unsteady state two-dimensional energy equation with heat generated from viscous dissipation. The effect of temperature rise and shear rate has been considered while determining OFT.

George et al.[20] predicted and compared their results with results from other semi-empirical models. In early studies, the squeeze film effect was neglected and a simplified hydrodynamic lubrication theory was applied to predict the oil film thickness. The model proposed by the authors considers that the complete ring pack can be reduced to a set of several compression rings and one twin-rail oil control ring. Each rail of the oil control ring is manipulated as a separate single ring. For the simulation of the oil film action between the piston ring and the cylinder liner, the one-dimensional Reynolds equation is used, considering sliding and squeeze ring motion.

Abu-Nada et al.[21] have approximated the shape of the oil film thickness by a trigonometric function where oil film thickness has minimum values at BDC and TDC and higher values in between. In the current study, the authors have assumed following distribution:

$h_m(\theta) = A + B|\sin(\theta)|$ where, A and B are constants.

h_m is the oil film thickness between the ring and cylinder liner. This thickness reaches a minimum value at the bottom dead center (BDC) and top dead center (TDC) and has higher values between them. Increasing the number or thickness of piston rings increases piston friction and thus reduces the brake power and the efficiency. However, the effect of changing piston ring configuration is limited in comparison to other parameters. This is due to dominance of skirt friction over ring friction. The author examined the effect of speed and temperature on efficiency for three different oil film distributions. Although decreasing oil film thickness results in a drop in efficiency, this effect becomes noticeable only at low temperatures and at high engine speeds. It is also interesting to note that the two distributions assume maximum film thicknesses of 7 μm and 12 μm resulted in nearly identical efficiency curves. However, the distribution that assumes a maximum film thickness of 2.5 μm resulted in a somewhat different curve. This suggests that there exists a threshold value in the order of 1 μm for the oil film thickness, below which ring friction can start to play significant role in piston friction.

III. CONCLUSIONS

1. The major factors affecting the oil film thickness are, piston speed, lubricant viscosity, ring face profile, boundary conditions, surface roughness, effect of ring twist, bore distortion and ring flexibility.
2. Models are developed either on the assumption of Reynold's theory of hydrodynamic lubrication or Stribeck curves, the same are established for rotary motion while in PRA system the motion is reciprocating, so results may not have the consistency.
3. A need to develop versatile model for reciprocating motion of piston assembly exists as lubrication regimes varies from Boundary to Hydrodynamic.
4. Film thickness may vary from about 2.5 to 8 micron in power stroke with SAE 10W50a oil at 1600 rpm & no load conditions as predicted by Harigaya et al. [19]

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