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RESEARCH

A Review of Piston Compression Ring Tribology

P.C. Mishra[a]

[a] *Green Engine Technology Center, School of Mechanical Engineering, KIIT University, Bhubaneswar, India-751024.*

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Corresponding author:

Prakash Chandra Mishra
Green Engine Technology Center,
School of Mechanical Engineering,
KIIT University, Bhubaneswar, India
E-mail: pmishrafme@kiit.ac.in

A B S T R A C T

This paper presents studies related to piston compression ring tribology and the theoretical and experimental works developed to analyze ring-liner contact friction. The literatures revealed that the simulation and experimental work are more independently investigated. The correlation of modeling output with experimental output is presented in limited number of research works. The experimental work to capture data from a running engine is also in basic level.

1. INTRODUCTION

Mechanical Friction is one of the major forms of energy loss in an IC engine. The figure 1 shows the source contributing the overall losses of an engine. Earlier study predicts 15 % of total input fuel energy in an IC engine is lost due to friction of various relatively moving components in contact. A slight modification in component surface/geometry can save a significant part of investment in automotive sector.

Piston compression ring is placed on the top position of the piston assembly. It accounts for 80 % of piston subsystem loss due to mechanical friction developed because of simultaneous sealing and sliding action. A slight improvement in ring design and its manufacturing method could save significant amount of energy lost due to friction (Fig. 1).

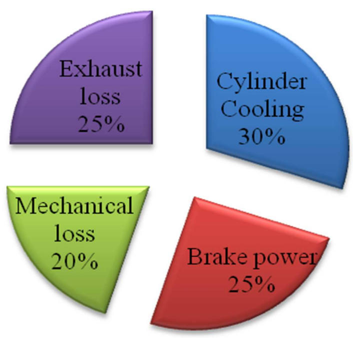

Fig. 1 Sources contributing to overall losses in an engine.

The concept of out ward springing incomplete ring and its use as piston compression ring was started since James Watt's engine invention, when horse manure, straw, asbestos cloths were used for sealing the stem to retain cylinder pressure. Because of leakage and blow by, the brake power was weak enough to develop time delayed pickup. Hence the time of journey was longer as compared to current days. Also, the wear and tear of relatively moving parts and sealing materials were of faster rate.

Based on such problem, the design, development and use of incomplete circular ring started to improve the life of piston and cylinder. Since then a chronological developments of piston ring occurred to reach its current state. This paper is a review of theoretical and experimental method of piston compression performance in reducing friction.

2. CONCEPTS OF PISTON RING LUBRICATION

Piston compression ring and cylinder lubricated contact can be analysed using transient thermoelastohydrodynamics. It is a multi-scale and multi-physics problem which needs sound numerical understanding of the engine dynamics and tribology. The literature review pertaining to this analysis can sweep from macroscopic approach of engine dynamics to the microscopic methods of micro-hydrodynamics. A reliable interface is the goal. The focus of the literature review is presented through the following block diagram (Fig. 2.):

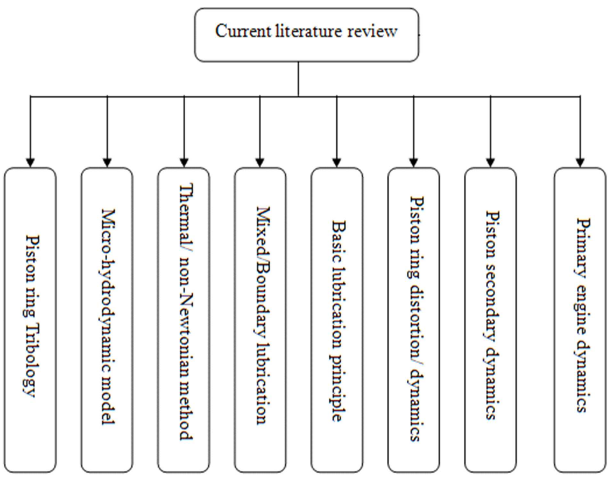

Fig. 2 Multi Physics methods for piston ring TEHL

3. BACK GROUND MOTIVATION

The earliest form of reciprocating prime mover is steam engine. The expansion of steam in this type engine cylinder is acted upon the piston [1]. These early pistons are relatively poor fitted in cylinder due to machining inaccuracy of both piston and cylinder bore. Hence gas sealing was considerably less effective, which encourage James Watt to use woollen clothing, strawboard and horse manure packing for the piston of wood and metal type. But the sealing process was so unhealthy that it necessities the need of a proper sealing device. The next step in this area is the invention of an outward springing ring [2]. This principle is still present in present piston rings, only the development is tried, without affecting the ring.

Inertia dynamics and its application to achieve useful work was an inception from 3rd century B.C. But the work done by Henri Papin towards development of steam engine became a guideline for basic engine invention. Through this invention the inertial dynamics of crank slider mechanism was converted to rotational motion by connecting rod and crank shaft assembly. From then, there was an intensive search for a device which can convert the rotational motion out of crank slider mechanism [3].

The success of Nikolaus Otto in 1876 through a patent on 4-stroke engine, using gaseous fuel, was a major landmark in engine research. In 1893, the compression ignition engine was developed by Rudolf Diesel, which opened the avenue of engine applications. But the concern for noise, vibration and harshness is still a challenging area [4], which needs a multi-body dynamics environment to involve contribution of each relatively moving engine subsystem to the overall performance of an engine.

The piston assembly undergoes secondary motion due to its eccentric positioning. Most of the engines are fitted with piston with an offset gudgeon pin where the offset is towards the major thrust side. The thrust side of the piston is at right angle to the gudgeon pin. During power stroke, the majority of side load is present in the thrust side. Hence, to prevent slamming of piston into cylinder wall, the gudgeon pin is located eccentric to bore axis. The secondary motion creates ring tilt and helps reducing scuffing.

Isothermal transient analysis of piston skirt and cylinder wall contact under combined axial, lateral movement and tilting explained the approaching and separating phenomena of piston secondary motion [5,67-68]. The analysis was carried out for both minor and major thrust sides and included transient analysis due to inertial dynamics of piston skirt to down to microscale analysis due to elastohydrodynamics of the piston skirt and cylinder liner contact. The elastohydrodynamic solution was achieved through solving the Reynolds equation by setting a suitable convergence criterion for error in pressure and load. Transient solution was carried out using Newmark linear acceleration method.

The vibration behaviour of an engine is a challenge for the present design analysis work. Multi-body dynamics and elastohydrodynamic excitation in engines especially considering piston and liner contact is a step forward for piston secondary dynamics analysis [6]. The lubrication performance of engine contact conjuction such as piston-liner, shaft-bearing etc. are largely dependent on geometric and physical properties of contact area. A multibody dynamic simulation tool is required to analyse the system. It divides this non-linear mechanical system into subsystems with elastic behaviour. In case of ring liner linear elastic body, the non-linear connection like film thickness is present. A simulation procedure is out lined by calculating the non-linear connecting forces formulated through mathematical modelling of component structure. Such method can also be used for optimization of noise in IC engine. This new simulation technique helps minimizing friction and wear. Such simulation integrates component kinetics, contact elasticity and elastic deformation of rigid body under partially lubricated contact conjuction.

FEM is useful to evaluate piston tilt effect on piston ring dynamics of IC engine. Lubrication and blow-by in engine is strongly influenced by piston as well as ring motion. Formulating an FEM model is useful for such motion that includes skirt secondary motion, ring axial, radial and titling motion.

Two dimensional average flow Reynolds equation is useful to analyse skirt liner friction, powerloss due to piston secondary motion [8]. In the mixed regime, radial clearance, engine speed, piston skirt profile and wristpin offset plays important role in determine piston secondary motion.

The materials used for piston subsystem components are highly elastic. Hence, deformation of contiguous solids during reciprocation is obvious due to high pressure generation out of hydrodynamic action [9].

The crank location 300^0 to 400^0 is known as high pressure zone of engine cycle as the combustion rises and vanishes from and after this crank location. At high pressure zone of engine cycle, Mishra [10] developed a tribodynamic technique to compute the film thickness, friction force and validated the numerical findings with the results of friction from engine test by Furuhama and Sasaki [65]. A good agreement is found. Prior to this Mishra et al. [69,70] studied the transient nature of compression ring lubrication performance at the vicinity of top and bottom dead center as well as for the entire engine cycle.

Mishra [11] further developed a model to study the friction in the four stroke four cylinder petrol engine. The compared the film, friction and power in different cylinder different due to firing order in a common time frame.

The nature of heat transfer due to thermal effect and the mixed lubrication of ring-liner conjuction because of approaching contiguous surfaces were addressed by Shahmohamadi [12]. In this context, the analytical evaluation of fitted compression ring modal behavior and the frictional assessment for three dimensional distortions [13,66].

4. SIMULATION AND MODELING IN PISTON RING TRIBOLOGY

Simulation and modelling is an important tool to understand system performance using mathematical technique. It is a method to understand the system before hand on practical test. In case of piston assembly, it is also possible to get prior information on various contact conjuctions.

4.1 Ring dynamics and deformation

Applying the method of toroidal elasticity, Lang [14] studied the pure twist of an incomplete circular ring of hollow cross section with non-isotropic orientation. An incomplete circular ring of thin cross section is tested for small amplitude vibration. End clamped rings were studied for natural frequency and mode shapes.

The angles that are subtended by the ring with reference axis were taken to be 180^0 and 360^0 respectively. As per assumption, one end of the ring is clamped and the other end is given a prescribed It was assumed that the ring was clamped at one end and given a prescribed time dependent displacement at the other end.

The vibration of piston compression ring was considered to be in-plane in extensional mode for circular incomplete ring of small cross-section. It is governed by basic equation of motion with an additional term to represent the damping effect [15]. Transient response of semi-circular ring with relatively moving support is depicted through an equation. The solutions obtained were given in terms of natural frequency and ring modal shapes through appropriate computation of differential equations for problem at hand, together with suitable boundary conditions. It is an improvement of approximated energy technique in which first four natural frequencies of the ring were determined [16].

Den Hartog [17] investigated both extensional and in-extensional vibration using Rayleigh-Ritz energy method. The first and second natural frequency of circular arc was evaluated, which was clamped and hinged at ends. It was concluded that within $60^0 < \alpha < 270^0$ the results were smaller and better compared to the other angle (i.e. for and $\alpha > 270^0$).

In an earlier attempt, Brown [19] estimated the natural frequency of a ring frame subjected to lateral vibration. He studied the nature of combined action of flexural and torsional vibration perpendicular to the plane of the ring. The solution was an approximation to the Rayleigh's energy method. Ring secondary dynamics is analogous to the flexural vibration of curved beam, which is in form of an arc of circle [20]. Using such concept a general equation with boundary condition of in-plane mode can be formulated.

The incomplete circular ring was considered with different types of applied force, such as slightly bent by an in-plane couple applied to its ends (i.e. Central line remains undeformed), the ends were subjected to tension, which were radically opposite to each other, a couple applied further at the ends, which is perpendicular to the plane of ring. Through these analyses an incomplete circular ring, bent out of its plane along with in-plane and out-of-plane flexural vibration were discussed with modal frequencies, using previous work by Mayer [21] and Timoshenko [22].

The contact pressure distribution of piston ring can be obtained by solving the governing equation of ring contour as curved beam [23]. At first the piston ring contour in a free form to be measured and then the pressure distribution on the ring circumference due to contact to be calculated. The displacements at corresponding points were evaluated from the contact force and validated with those obtained through experiments, which gave excellent agreement.

Three dimensional distortion of piston ring occurs in a running engine [24]. Such distortions in an arbitrary cross section were studied to find its importance in oil flow and blow by. This distortion is either due to installation stresses or due to operational parameters, like gas pressure, friction and thermal loads. The model was applied to several typical cross-sections. The piston ring conformability was aimed through various engineering optimisation with consideration of both Young's and shear Moduli. A model of conformability analysis of a split piston ring with arbitrary cross-section was developed with consideration of bore profile and centre-line of piston ring through Fourier harmonics.

The sign of pressure exerted on the piston ring (its centre-line) from the bore defines its conformability. If the pressure is positive, this means there exists full conformability. If the pressure is negative, the conformability is absent in some regions. The equation of three dimensional deformation of a ring was utilised to calculate a twist of the compression ring. Two sets of calculated twists were compared with conducted measurements.

Due to installation stress, there occurs the elastic distortion of piston ring in reciprocating air brake compressor [25]. An advanced finite element model of stress analysis can be used to monitor such distortion in both lubricated and unlubricated condition. The paper demonstrated analytically and with aid of FEA, the mechanical and geometrical parameters of a split ring, which was supposed to have an effect on the twist of the ring during installation in the cylinder, and calculated magnitude of this twist along the ring circumference.

Saburi et al. [26] analysed tribology of piston ring-to-cylinder liner in a marine diesel engine. This involved theoretical estimation of oil film thickness between piston ring and cylinder liner and an experiment to evaluate the effect of surface roughness on cylinder liner scuffing failure.

4.2 Ring-bore conformability

Conformability is the ability of the ring to conform on the bore with almost zero gaps. Ma et al [27] found a model to implement starved lubrication of piston ring in a distorted bores. At some crank locations piston rings experiences lubricant starvation. The non-axisymmetric lubrications models of a single ring with inlet and cavitation boundary of the oil film were located. For a distorted bore, the gas blow by could be predicted through the ring face. The mass conservation algorithm is used to estimate oil availability and lubricant accumulation. The computational method is tested extensively and is found that operating performance is significantly influenced by lubricant starvation. As the starvation increases, the blow-by increases.

Ma et al. [28] carried a three-dimensional study of piston ring lubrication to evaluate performance characteristics of piston ring by considering the bore-out-of roundness. Bore distortion lead to significant decrease of film thickness between ring and liner. In distorted bore the ring experiences hydrodynamic lubrication over most of the engine cycle. The instantaneous viscous friction force and the associated power loss is significantly less in case of a non-circular or distorted bore, as there is instantaneous oil transport rate. The non-circular bore predicted hydrodynamic pressure were not uniform and shows considerable variability along the circumference.

The cyclic variations of minimum film thickness are governed by combustion chamber gas pressure and inter ring gas pressure. In the reason of larger distortion, the hydrostatic effect of gas pressure is remarkable. The blow-by most likely occurs around TDC if the bore is elliptic in shape. Within combustion chamber and volume between first and second compression ring, there is large pressure gradient acting across low pressure zone.

Sun [29] formulated thermal elastic theory of piston ring and cylinder bore. As per Sun, the contact between piston ring and cylinder bore is generally non-uniform, due to mechanical and thermal distortion of the bore and manufacturing imperfection in shape of the free ring.

$$r_f = R_b + \frac{g}{3\pi}\left(1 + \frac{1}{2}(\pi - \theta_0)\sin\theta_0\right) \tag{1}$$

Where g is the free end gap, which can be obtained from ring handbook.

Abe and Suzuki [30] analysed the cylinder bore distortion during the operation of an engine and developed a calculation method for the same. This method can consider the sliding effect of the cylinder head on the top dock of the cylinder block. The bore distortion during engine operation calculated by this method agrees with that measured by Fujimoto. The result calculated for a 4-cylinder in-line 1.5L engine showed that thermal distortion has larger effects on the cylinder bore distortion during engine operation than cylinder head clamping distortion.

Scheider [31] studied the cylinder bore-out-of-roundness on piston ring rotation and engine oil consumption. Ginsberg [32] has discussed the advantages of using a split less piston ring. Hill and Newman [33] designed a new piston ring to reduce friction. Hu et al [34] carried out the numerical simulation of piston ring in mixed regime using a non-axi symmetric model.

Massen et al. [35] under take study on analytical and empirical methods for the optimization of cylinder liner bore distortion. Rahmani et al [36] studied the transient nature of the EHL lubrication of rough new or worn piston ring in conjunction with out of round cylinder bore. Tomanik [37] studied the conformability of ring on a distorted bore and found out.

4.3 Lubrication principle applicable to piston ring

Knoll and Peeken [38] developed the hydrodynamic lubrication model of skirt and liner contact conjuction using open end boundary condition with inclusion of piston secondary motion. Ruddy et al. [39] studied a twin land type oil control piston ring for the cyclic variation of important parameters. In the mixed lubrication regime they assumed profile of worn oil control ring and the effect of surface roughness.

Kazmierczak [40] performed a computer simulation in Piston-Piston Ring-Cylinder liner coactions. A sealing piston ring with anti-ceramic cover was developed. This simulation process uses KIVA3 for analysing load due to combustion engine process by computation of temperature and pressure distribution and motion of the changes in the combustion chamber at particular point in the work cycle. It renders the design material features of the ring seal component. The model was discretized using the EDS unigraphics software. A tetra nodal and tetrahedral component is taken for investigation. The Piston ring coating used was TiN and prepared through PVD (Physical vapour deposition) method. Piston ring coating was modelled through quadrilateral plane elements. The temperature distribution range, heat flow, reduced stresses, displacement and reaction force in a ring with coactions was estimated.

Saburi et al. [26] analysed tribology of piston ring-to-cylinder liner in a marine diesel engine. This involved theoretical estimation of oil film thickness between piston ring and cylinder liner and an experiment to evaluate the influence of surface quality of the cylinder liner on scuffing.

Ma et al. [41] developed a one dimensional EHL mixed lubrication model to study the friction and wear characteristics of piston ring and cylinder liner. Such model predicts asperity contact parameters and temperature-pressure-viscosity relationship. The compared bore wear and ring pack friction with experimental data drawn from bench test. The ring pack friction due to oil viscous shearing and asperity contact is found to reach its minimum at a certain oil temperature.

4.4 Mixed regime in ring liner contact, Roughness effect and texturing

The regime of lubrication changes because of approach or separation of relatively moving surfaces. The Figure 3 shows the location of asperity in different lubrication regime.

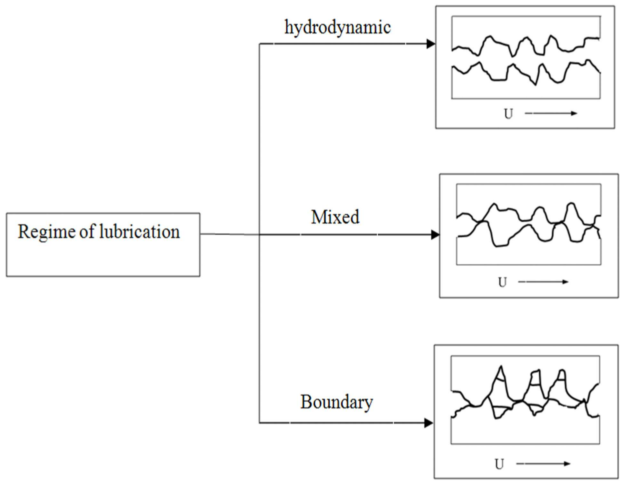

Fig. 3. Lubrication regime with asperity status.

As most important sealing action of piston ring and cylinder liner happens under partially lubricated contact. An average flow model for determining the effects of three dimensional roughnesses on partial hydrodynamic lubrication is required [42]. The average Reynolds equation is defined in terms of various flow factors, which are obtained using numerical flow simulation. But the assumption of one dimension roughness rendered this theory less useful, especially predicting failure of EHD contacts.

The extended work of such average flow model includes sliding contact by deriving the shear flow factor for various roughness configurations [43]. The shear rate is modified using the shear flow factor, which further help in friction force estimation.

Akalin and Newaz [44] developed a test rig to measure friction of piston ring and cylinder liner contact. Such test rig has the provision of measuring friction, load, crank location and contact temperature data simultaneously. It has perfect control of the speed, temperature and lubricant flow. How the measured parameters influence the friction coefficient, it is observed for a convectional cast iron cylinder liner. Friction coefficient derived from analytical method was verified with the result obtained from test bench and found to be matching.

Jacobson [46] studied a simplest rheological model of Newtonian lubricant, which is used to calculate the oil film thickness between elastrohydrodynamically lubricated smooth or rough surface without the breakdown of oil film thickness.

Sato et al. [47] studied the positive effect of surface roughness in reducing friction forces of piston and cylinder contact. Different pattern of surface roughness and treatment methods and their effect on reduction of friction is measured using floating liner. Reduction in surface roughness improves piston lubrication performance and low viscosity oil reduces lubricant degradation. Etching and DLC coating are verified and found better surface treatment to minimize friction and wear. The effect of sleeve fitting, honing pattern like, single honing, plateau honing, on friction force is analysed at 1200, 1500 and 1800 rpm. The variation of cylinder liner roughness, i.e. R_{pk} during running of the engine is found to decrease over all. Plateau honing is widely used in diesel engine cylinder surface as it provides the deep valley for the oil retention and flat plate for smoothness.

The laser texturing of engine components and it's advantage to overcome friction loss is studied by Kligerman et al. [48] through experimental investigation. Simulated plane surface configuration was modelled theoretically in starved condition which shows good correlation with experimental findings. For all range of lubricant flow low lubricant viscosity with optimum dimple depth is beneficial. At specific operating condition, high viscosity and maximum dimple depth is detrimental texture.

Blau et al. [50] has carried out the laser surface texturing of lubricated ceramics parts. They studied the micro scale dimple of rectangular pattern produced through laser to reduce friction. They outlined the extent to which the laser surface texturing (LST) process affects the structure of the ceramics materials.

Etsion and Etsion et al. [51, 52] studied the benefits of laser texturing in increasing strength while reducing weight of material. Such process improves frictional performance due to formation of micro conjuctions on the contact surfaces. Fenske et al. [53] applied the laser texturing technique to seal face to evaluate the sealing performance.

Kovalchenko [49] analysed the effect of surface texturing on transition in regime of lubrication during uni-directional sliding contact. A pulsating laser beam is used to create thousands of structured micro-dimples through ablation. Such dimples creates many micro hydrodynamic conjuctions that are parallel to sliding surfaces. The reduction in friction due to texture of micro-dimples in surface was found.

McNickle and Etsion [54] carried out experimental investigation of near-contact gas seal. A simulated seal for a high speed gas turbine was studied. With the LST, a 40 % reduction in frictional torque and at least 20 ^{0} C reduction in face temperature was achieved due to enhanced lubrication as compared to the base line simulated seal. It was also concluded that LST is fast and can be applicable to any contacting seal design. The mechanism behind it is that the micro-dimples are accommodated directly on sealing dam and do not required any additional accommodating area like that needed for lift devices in form of grooves or shrouded steps in non-contacting seals.

Some manufacturing process are conducive for lubricant retention and wear debris collection. Honing is such process to generate component friendly texture in the surface. Radil [55] studied the influence of honing on the wear of ceramic coated piston ring in contact with cylinder liner. Rahnejat et al. [4] developed an optimization technique to optimize surface finish required for incylinder friction reduction.

An experimental investigation of laser texturing in application to reciprocating automotive engine components was conducted by Kligerman et al [56]. An effective micro-structure was produced by laser texturing to improve tribological properties of these components. Wakuri et al [57] study the characterstics of piston ring friction considering oil properties and their variation.

5. EXPERIMENTAL METHODS IN PISTON RING TRIBOLOGY

Spencer et al. [58] developed a transient roughness model of piston ring and cylinder liner contact. The roughness texture is taken as semi deterministic and by help of an algorithm the texture is generated in simulation. Using white light interferometer the plateau honed liner roughness is measured. Reynolds equation is solved making local scale homogenization.

Barrel et al [59] presented the wear behaviour of the top and the second compression ring of a piston an internal combustion engine for different time slots of running.

Mufti and Priest [60] made experimental evaluation of Piston subsystem friction under motor and fired conditions. They used advanced felemetry type data acquisition to sense experimental data like cylinder pressure, crank angle, sliding velocity and connecting rod strain. Experiments were done for piston subsystem friction at a range of engine operating condition with different lubricant formulation with and without friction modifier.

Bolander et al. [61] analysed the lubrication regime transitions of piston ring-cylinder liner contact through a numerical and experimental model to determine the lubrication and frictional losses. Reynolds equation and film thickness equation subjected to suitable boundary condition were solved simultaneously. The effect of boundary and mixed lubrication were implemented using the stochastic model.

6. METHOD OF PISTON RING TRIBOLOGY

Lubricant performs to avoid the contact of relatively moving parts because it's viscous property and incompressibility. When it passes through the converging and diverging gap of mating surfaces, the hydrodynamic pressure is developed and provides cousin effect to maintain separation. Such pressure determines the load bearing ability, friction force, and flow-in to the conjuction. In order to obtain such pressure we have solved the Reynolds equation as given in equation (6).

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{\eta}\frac{\partial p_h}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\rho h^3}{\eta}\frac{\partial p_h}{\partial y}\right)=12\left(U\frac{\partial}{\partial x}(\rho h)+\frac{\partial}{\partial t}(\rho h)\right) \quad (2)$$

The lubricant in running state of the engine is subjected to piezo-viscous behavior change. It is bases on the formulation given in equation (3).

$$\eta=\eta_0 \exp^{(\ln\eta_0+9.67)\left(-1+\left(1+5.1\times10^{-9}p_h\right)\right)} \quad (3)$$

The density and pressure correlation is addressed in the model and is given in equation (4).

$$\frac{\rho}{\rho_0}=1+\frac{0.6p_h}{1+1.7p_h} \quad (4)$$

6.1 Method of Solution

The FDM type numerical computation is used along with low relaxation effective influence Newton-Raphson method with appropriate pressure convergence (given in equation (5), and with Reynolds/Swift-Steiber exit boundary condition. Load convergence is carried out together with film relaxation to find the appropriate gap according to load applied, using a suitable damping factor (given in equation (10)).

$$Error^{p_h}=\frac{\sum_i^n\sum_j^m\left|P_{hi,j}^{K+1}-P_{hi,j}^{K}\right|}{\sum_i^n\sum_j^m\left|P_{hi,j}^{K+1}\right|}\le 0.01 \quad (5)$$

Where, $error^p$ is the error of pressure convergence, K is the number of iteration step, (i,j) is position vector.

$$h_0(\theta)^k=h_0(\theta)^{k-1}-\left\{\vartheta\left|\frac{F_{ap}-W}{F_{ap}}\right|\right\} \quad (6)$$

where,

$h_0(\theta)$ is the minimum film thickness, $F_{ap}(F_g+F_e)$ is the applied external force and W is the estimated lubricant reaction force. Through this iterative method the film is made exact for corresponding applied force.

6.2 Friction Force

Due to rapid shear of lubricant layer, there occurs the fluid friction. It is the integration of shear stress developed (given in equation (7). The expression for shear stress is as per equation (8).

$$F=\iint\tau dxdy \quad (7)$$

where,

$$\tau_h=\iint\left(\frac{\eta U}{h}-\frac{h}{2}\frac{\partial p_h}{\partial x}\right)dxdy$$

$$and \quad (8)$$

$$\tau_c=\iint\frac{\eta U}{h}dxdy$$

also,

$$\tau=\tau_h+\tau_c$$

It means the total shear in the contact is equal to the sum of the shear in the fluid film region and that in the cavitation region. In this analysis we have not incorporated asperity interaction.

Power loss due to friction is given in equation (9):

$$P_f = FU \tag{9}$$

6.3 Some useful results

Friction loss is an important issue related to the fuel efficiency of an engine. The simulation states that maximum amount of friction loss (57 %) happens in power stroke of an engine cycle followed by compression stroke (26 %) as shown in Fig. 4.

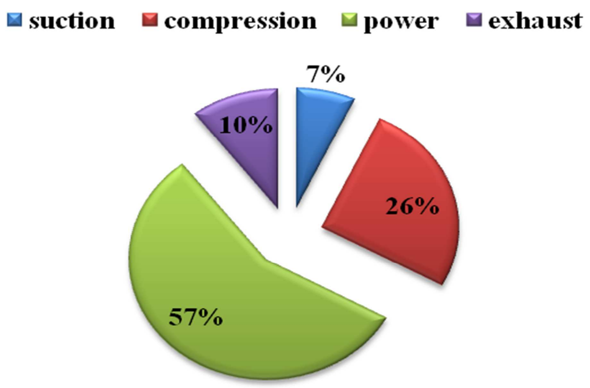

Fig. 4. Frictionloss % in various strokes.

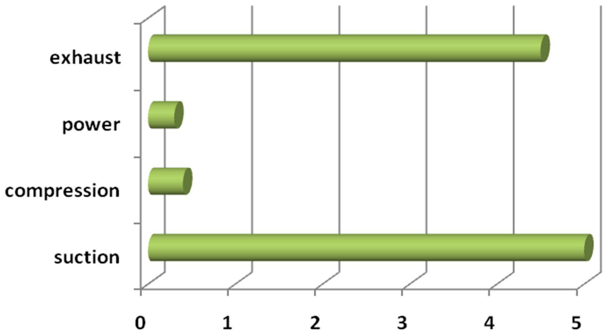

Fig. 5. Maximum value of minimum film thickness in µm.

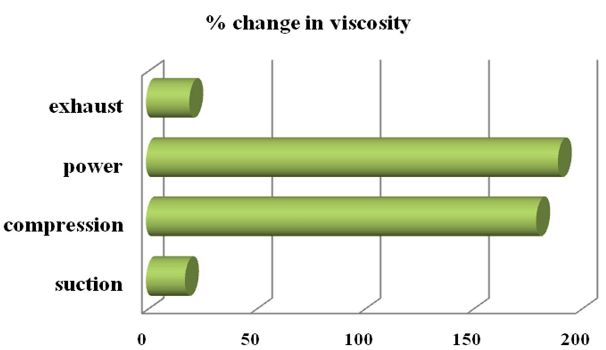

Fig. 6. Viscosity variation due to thixotropic behavior.

The nominal film thickness is near to 5 µm during compression and power stroke, where as in power stroke the film thickness is as low as 0.5 µm as per Fig. 5. Due to pizoviscous action, there is increase in viscosity the highest being 200 % in power stroke as shown in Fig. 6.

The computation is done for all 720^0 crank location. Figure 7 represents the film thickness of a rough and smooth cylinder in contact with piston ring plotted on top of each other. It is found that for a particular crank location, a rough liner and ring interface accumulates more oil during suction and exhaust stroke. But during compression and power stroke, such difference decreases. Because of increasing gas pressure, the sealing becomes stronger by reducing the gap.

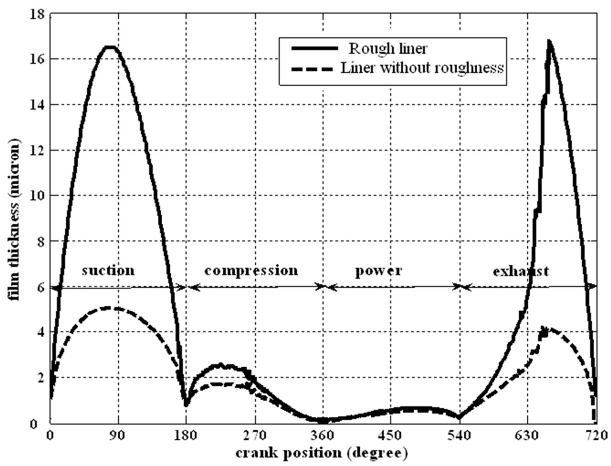

Fig. 7. Cyclic variation of oil film between.

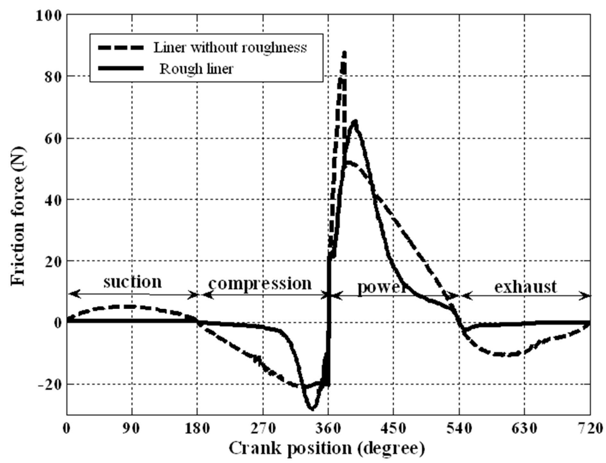

Fig. 8. Cyclic variation of friction force.

The Figure 8 gives the cyclic variation of friction for both smooth and rough liner. The friction is comparatively less in case of a rough liner because of micro conjuction effect. Again, the friction is more in compression and power stroke due to development of gas pressure in combustion chamber on the back of the ring.

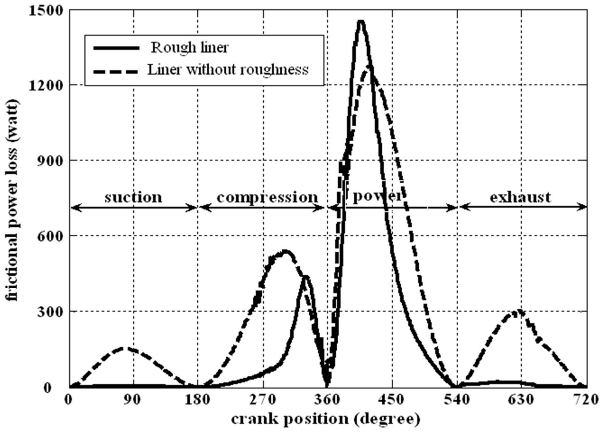

Fig. 9. Cyclic variation of friction power.

The Figure 9 represents the frictional power loss of ring liner contact in an engine cycle. The powerloss in compression and power stroke together is 80 % of total power loss in an engine cycle. The rough liner in contact with pison ring causes less friction.

7. CONCLUSION

A broad literature survey is carried out in the research area of piston compression ring to know about the simulation and experimental methods developed to study its performance. There are still limited number of experimental methods in compression ring tribology available. Application of non-contact type measuring principle, use of nano sensing device, optical measurement technique etc. are yet to be implimented.

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