# EXPERIMENTAL ANALYSIS OF PISTON RING TO REDUCE FRICTION BY USING DIFFERENT LUBRICANTS (SAE15W and SAE30W) FOR A FOUR STROKE FOUR CYLINDER PETROL ENGINE

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**ABSTRACT**

*The friction at the piston ring cylinder liner assembly is a major contributor in the total friction losses in the I.C. engine. The friction produced due to PRA has a major contribution in total frictional losses of the engine. The piston ring assembly is dominant sources of the engine rubbing forces. The components in Piston Ring Assembly are compression ring, oil control ring and piston skirts are the responsible for the frictional force. The present work deals with evaluation of the friction in the piston ring cylinder liner assembly for a 4-stroke four cylinder petrol engine. In this study Friction is evaluated for existing rectangular shaped piston ring for two different lubricating oils at various engine speeds. An attempt was also made to evaluate the friction for a piston ring of different geometry. Friction is evaluated for an existing piston ring design (rectangular shaped) for two different lubricating oils at various engine speeds. It is also attempted to evaluate friction for a parabolic ring face profile. To evaluate the frictional force a model proposed by the YUKIO is considered for the evaluation of friction in the existing (rectangular shaped) piston rings. It is seen from the results that the friction for the rectangular shaped piston ring profile is more as compare to the parabolic piston ring profile. The experimental work was conducted on a four stroke 4-cylinder SI engine with an application of different lubricants (SAE15W and SAE30W) on given piston ring geometry from speed range from 1550 rpm to 3800 rpm. It is seen that the maximum value of the frictional force is higher for the lubricant SAE30W which may cause - higher wear of the rings. The use of lubricant SAE15W in the given setup would cause less wear of piston rings due to reduce frictional forces.*

**Keywords-** ***Piston Ring, Friction, Lubricating Oil (SAE15W and SAE30W), 4-Stroke Four Cylinder Petrol Engine***

## I INTRODUCTION

The purpose of the internal combustion engine is the production of the mechanical power from the chemical energy contained in the fuel. In the internal combustion engine, as distinct from external combustion engine, this energy is released by burning or oxidizing the fuel inside the engine. The internal combustion engine is an engine in which the combustion of a fuel (normally a fossil fuel) occurs with an oxidizer (usually air) in a combustion chamber. In an internal combustion engine, the expansion of the high-temperature and -pressure gases produced by combustion applies direct force to some component of the engine, such as pistons, turbine blades, or a nozzle [14]. This force moves the piston over a distance, generating useful mechanical energy. The friction is an important consideration taken into account in the internal combustion engine operation.

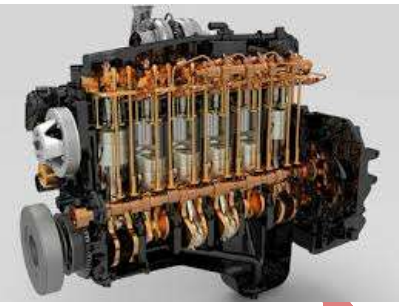


**Fig 1.1 A Diagram of the 4-Stroke 4- Cylinder Petrol Engine**

In an internal combustion engine major proportion of energy of fuel is dissipated as heat either from the engine surface or from exhaust pipe. Mechanical action accounts for further loss as friction leaving reduced brake power. The breakdown of the mechanical losses in the engine suggests that the piston ring assembly (PRA) friction is the major contributors. There are also losses associated with pumping and accessories.

## II. PROBLEM STATEMENT

The piston assembly is the dominant source of the engine rubbing friction. The component that contribute to the friction are compression rings, oil control rings, piston skirt and piston pin. The present work attempt to evaluate the friction at the piston ring liner assembly as it is the major contributor of the piston assembly friction. The forces acting on the piston assembly include; static ring tension (which depend on ring design and material), the gas pressure forces (which depend on the engine load) the inertia forces (which are related to component mass and engine speed). The major design factors which influence piston assembly friction are the following; ring width, ring face profile, ring tension, ring gap (which governs the inter ring gas pressure), cylinder liner temperature, ring land width and clearance.

## III.OBJECTIVES

The objective of the present work is to evaluate the ring friction at various engine speeds and at various crank angle position for two different lubricating oils SAE15W and SAE30W. The effect of variation in ring geometry on the ring friction is also studied. Two important parameters in ring design viz, ring pressure and the maximum bending stresses that the piston ring is exposed during operation, are also evaluated for the existing ring.

The scope of the present work is limited to the theoretical evaluation of piston ring friction at various engine speeds and at various crank angle positions for two different lubricating oil SAEI5W and SAE30W. The piston ring pack of the Premier-Padmini 4-stroke SI engine has been considered for the present study. The existing piston ring in the above mentioned setup has rectangular cross section and is made of nodular cast iron.An attempt was made to study the effect of variation in piston ring geometry or the friction by considering parabolic ring made of same material.

## IV. METHODOLOGY

The following methodology is adopted to achieve the objectives stated .

- Collection of data related to the engine cylinder block and piston ring.
- Estimation of indicated power, brake power and friction power in the 4-stroke 4- cylinder S.I. engine setup using MORSE test.
- Estimation of instantaneous piston velocities at the various crank angle position of the different speed (rpm) of the engine.
- Evaluation of static ring tension of the existing piston ring.
- Theoretical evaluation of friction force on the basis of model proposed by YUKIO [2] to understand the effect of various variables. The variable model parameters considered are piston velocity, engine rpm, kind of lubricant and crank angle.
- Theoretical evaluation of friction for two different ring geometry. Here we have considered cast iron ring of parabolic ring face.
- Evaluation of pressure distribution of ring on cylinder wall and calculation of maximum bending stress for the existing (rectangular shaped CI) piston ring.
- Comparison of the evaluated friction force for different types of lubricants SAE15W and SAE30W for the existing piston ring.
- Comparison of the friction force two ring geometry considered in present work.

## V. ENGINE SPECIFICATION

The geometrical data related to the test setup is presented below.

**Table 1.1 Engine Specifications**

| Name | Description |
|---|---|
| Manufacturer | The premier automobiles ltd, Pune |
| Engine type | 4-cylinders-stroke, petrol engine |
| Cylinder bore | 68mm |
| Stroke | 75mm |
| Connecting rod length | 156mm |
| Capacity | 1089cc |
| Compression ratio | 7.3:1 |
| Cylinder head | Aluminum with valve seat insert, overhead valves |
| Cylinder block | Cast iron |
| Cooling | Cooling water circulated by centrifugal pump |

## VI. PISTON RING NOMENCLATURE

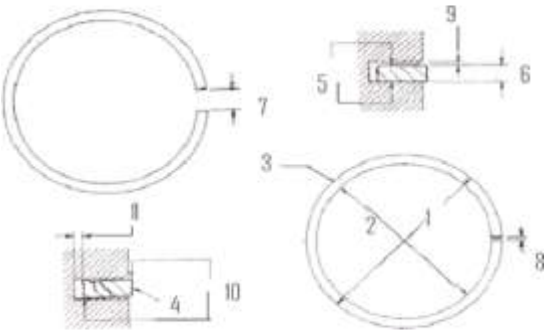


**Fig: 1.2 Engine Setup of 4-Stroke Four Cylinder Petrol Engine.** **Fig.1.3 Piston Ring**

1. Diameter-The outside diameter of a piston ring when compressed to actual bore size. (D=0.068m)
2. Inside Diameter-The inside diameter of a piston ring when compressed to actual bore size.
3. Radial Wall Thickness-The material thickness between the inside and outside diameters of the ring. (t=0.0025)
4. Ring Face-That portion of the ring which contacts the cylinder wall.
5. Ring Sides-The top and bottom surfaces of the piston ring.
6. Ring Width-The width of the area across the face of the ring. (b=0.00225m)
7. Free Gap-The end clearance when the ring is in its free state. (g=0.02m)
8. Compressed Gap-The end clearance when the ring is compressed to bore diameter.
9. Side Clearance-The clearance between the piston ring and groove with the ring in its free state.
10. Twist-Built in unbalance in the ring which causes the ring to twist in the groove when compressed. This is done to seal the groove as well as the ring face and cylinder.
11. Back Clearance-Area between the inside diameter of the ring and the bottom of the Ring groove in the piston with ring compressed to bore size.

**Table 1.2: Piston Ring Material and Properties**

| Material | Tensile strength(MPa) | Young's modulus(MPa) |
|---|---|---|
| Gray cast iron | 350 | 85-115 |
| KV1/GOE52 Ductile cast iron | 1300 | 150 |
| Steel | 1150 | 230 |

### 6.1 Estimation Of Indicated Power, Brake Power And Friction Loses Using Morse Test.

This is simple, quick and quite accurate test is used for determining the mechanical efficiency of the engine the equipment is required is only a water brake dynamometer and a tachometer, as used for brake power determination.

The Morse test consist of determining brake power of the engine at any particular speed, then cutting one cylinder at a time and measuring B.P. of the rest.

Now $\quad B.P.= \frac{WN}{2000}$ KW

Where, W= Load on the engine (kg) =6.5kg

N= Engine Speed (rpm)

I.P. = B.P. + Engine losses

There as one cylinder is cut out, the losses in the cylinder must be supplied by the by the other cylinder. Thus, the difference between the B.P. measured for the whole engine and for the engine with one cylinder cut gives the I. P. power of the engine. With the help of this I.P. calculated friction power of the engine as described below,

BRAKE POWER

B.P. with cylinder working = B, KW

B.P. with cylinder no 1 is cut out = $B_1$, KW

B.P. with cylinder no 2 is cut out = $B_2$, KW

B.P. with cylinder no 3 is cut out = $B_3$, KW

B.P. with cylinder no 4 is cut out = $B_4$, KW

Indicated Power of the Engine

$I.P. = [(B - B_1) + (B - B_2) + (B - B_3) + (B - B_4)]$, kW

$I.P. = [(I.P.)_1 + (I.P.)_2 + (I.P.)_3 + (I.P.)_4]$

Where, I.P. 23.9 kW, $I.P._2$=17.71 kW, $T.P._3$=12.47 kW, $I.P._4$=9.08 kW

The total friction power may be calculated as follows;

$F.P. = [(B_1 - (I.P.)_1) + (B_2 - (I.P.)_2) + (B_3 - (I.P.)_3) + (B_4 - (I.P.)_4)]$

$F.P. = F.P_1 + F.P_2. + F.P_3 + F.P_4$

F.P. = 12.5+8.41+5.72+4.42

F.P. = 31.06 kW

**Table 1.3 Brake Power And Indicated Power Values**

| S. NO. | Engine RPM | BP (kW) | I.P. (kW) | FP (kW) | P. (MPa) | Total P.P. in the PRA system (kW) | % of total P.P. in the PRA of the P.P. of the |
|---|---|---|---|---|---|---|---|
| 1 | 3800 | 11.4 | 23.9 | 12.4 | 0.34630 | 8.042 | 64.85 |
| 2. | 3000 | 9 | 17.71 | 8.71 | 0.32101 | 5.641 | 64.76 |
| 3. | 2250 | 6.75 | 12.47 | 5.72 | 0.31182 | 3.664 | 64.055 |
| 4. | 1550 | 4.65 | 9.08 | 4.43 | 0.32260 | 2.095 | 47.29 |

## 6.2 Estimation of Instantaneous Piston Velocity.

The expression for the piston velocity at a giver crank angle position is presented as below. [1]

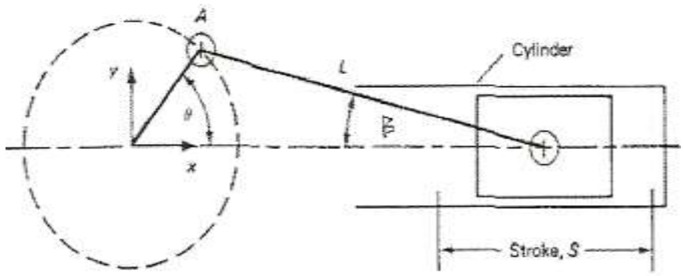
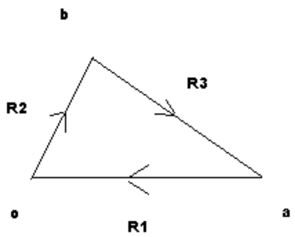



**Fig. 1.4Kinematic Diagram of the Crank Slider Mechanism.[14]**



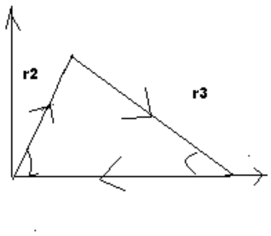




**Fig. No. 1.4a Phase Diagram of piston velocity Fig. No. 1.4 b Vector Diagram of piston velocity**

From the above phase diagram (1.4a)

$R_1$-$R_2$-$R_3$=0

Expressing the above vector diagram (1.4b) into the complex rectangular notation

(-r1-jΘ)+ (r2+jr2 sinΘ) + (r3cosΘ-jr3sinΘ) =0

Where

r1= linear displacement of slider (Piston)=0.075 m

r2= radius of crank=0.0375 m

r3= length of connecting rod =0.156m

Θ= angular displacement of crank, degree

ß= angular displacement of connecting rod, degree

ω2= angular speed of crank= 2ΠN/60 rad/s

ω1= angular speed of connecting rod, rad/s

Now from the abovre equation

r1+r2cosΘ+r3cosΘ=0

$r_2 \sin\theta - r_3 \sin\beta = 0.$

on differentiating equation (5.5) and (5.6) we obtain,

$\text{Sin}\ \beta = \frac{r2}{r3}\sin\theta$

So angular displacement of connecting rod at any angle θ is

$\beta = \sin^{-1}(r_{2/}\ r_3)^* \sin\theta$

Angular speed of connecting rod is

$\omega_3 = r3\omega2 / r3 \cos\beta$ rad/s

Piston velocity at any crank angle position is given as

$y_p = r_1 = r_2\omega_2 \sin\theta + r_3\, \omega_3 \sin\beta$, m/s

Using the above equations from 5.6-5.12 we can easily find out the piston velocity and angular velocity of the connecting rod at the different speed of the engine or crank speed.

For 3800 rpm of the engine and the crank angle $\theta°=30°$ the value of $\omega_3$, $\omega_2$, $\beta$ and $V_p$ are calculated as below.

$r_2 = 0.075/2$ m

$r_3 = 0.156$ m

$\omega_2 = 2\,\pi\, N/60$ rad/sec

$= 2*\pi*3800/60$

$\boldsymbol{\omega_2 = 397.93}$ **rad/sec**

Using Eq. (5.10) and (5. 1) connecting rod angle and angular speed of connecting rod at $\theta=30$ may be calculated as follows

$$\beta = \sin^{-1}\left(\frac{r_2}{r_3}\right) * \sin\theta$$

$\beta = \sin^{-1}(0.0375/0.0156) * \sin 30$

$\beta = 6.90^0$

$$\omega_3 = \frac{0.0375 * .93}{0.156 \cos 6.90}$$

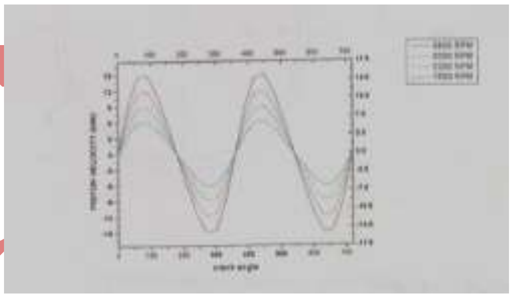
$\boldsymbol{\omega_3 = 83.44}$ **rad/s**

And using Eq.(5.12) the piston speed may be evaluate as follows;

$V_p = r_1 = 0.0375 * 397.93 * \sin 30 + 0.156 * 83.44 * \sin 6.90$

$\boldsymbol{V_p = 9.02399}$ **m/s**

Similarly at the different crank angle position the respective values are evaluated and tabulated as follows;



**Graph 1.1 Figure showing piston velocity for various crank angle in one complete cycle at speed 3800-1550 rpm**

### 6.3 Estimation of Static Ring Tension of Existing Piston Ring.

The static ring force (Tension) is easily obtained using Castiglione's theorem. Castiglione's theorem [16] states that, when force act on an elastic system subject to small displacement, the displacement corresponding to any force, collinear with the force, is equal to partial derivatives of total strain energy with respect to that force.

Three terms contribute to the total strain energy due to total strain energy $SE_r$, a term due to bending moment, anther due to normal force and third due to shear energy. Both the strain energy due to shear and normal force has a minor effect, so the effect of the bending moment is Considered. [4]

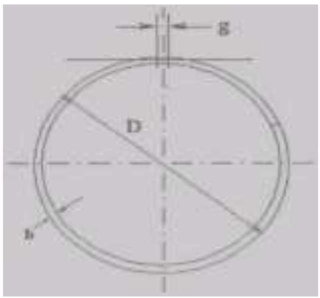



**Fig No. 1.5 Diagram Showing Forces Acting on the Section of the Ring**

### 6.4 Evaluation of Friction Force on the basis of Model Proposed by YUKIO. [2]

In internal combustion engine a major mechanical loss occurs at piston ring assembly (PRA).To evaluate this friction loss different researcher have explained friction phenomenon in PRA with different theories and mathematical relationship based either on experiment result or by simulation of model. Here an attempt is made to evaluate the friction force on the basis of the model prepared by YUKIO [2] to understand the effect of various parameters. YUKIO has presented the model to friction force by considering piston velocity, engine rpm. Crank angle, lubricating oil as variable parameter and keeping the compression ratio, reciprocating mass and ring tension as non variable parameter. [2]

Friction force at different speed of the engine is given by

$$F_P = C_1 *[\mu_k * V_P * 200 * \left(\frac{T}{D}\right)]^{0.5}] \quad N$$

Where,

$C_1$= constant value for the lubricating oil [2]

For SAE15WC1=8.136

SAE30WC1=8.135

T = Ring tension

$\mu_k$ = kinematic viscosity [14] =$64*10^{-6}$ $m^2/s$ for SAE15W =$69*10^{-6}$ $m^2/s$ for SAE30W

D=cylinder bore in mm.

For the calculation of friction force in the piston ring assembly there are two lubricating oil used in the engine SAE15W, SAE30W

## VII . FRICTION FORCE CALCULATION AT DIFFERENT SPEED OF THE ENGINE.

Existing piston ring profile is shown in the figure no.5.€ and the hydrodynamic lubrication between the piston ring face and cylinder wall is assumed.

The value of the friction force for the two different lubricating oils at speed range from 3800rpm-1550rpmhave been presented.

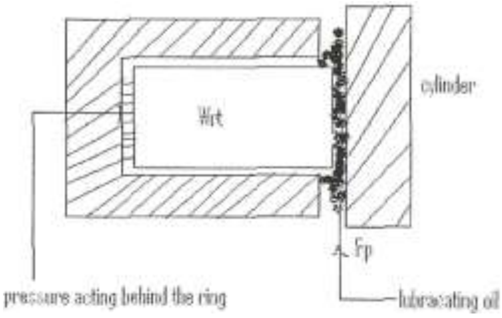
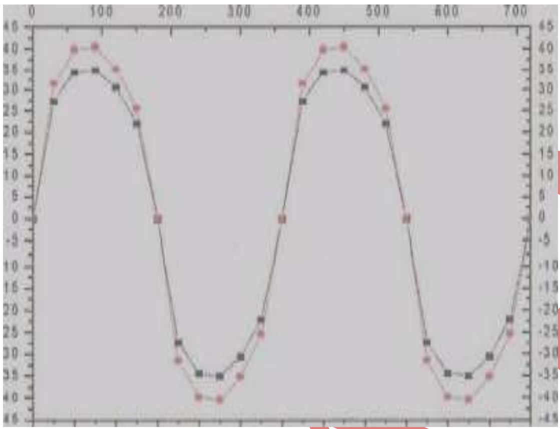
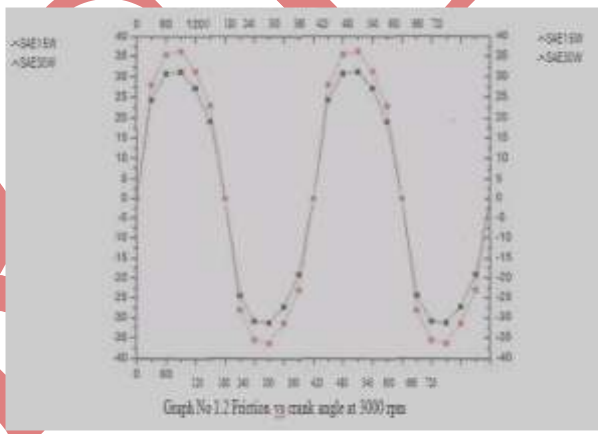



**Fig No. 1.6 Piston Ring Profile**

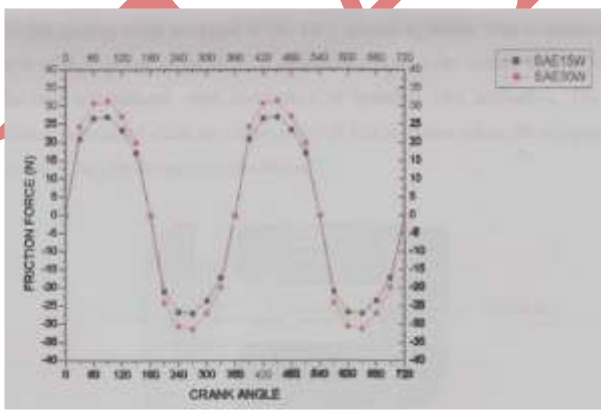
**Speed, at 3800rpm**



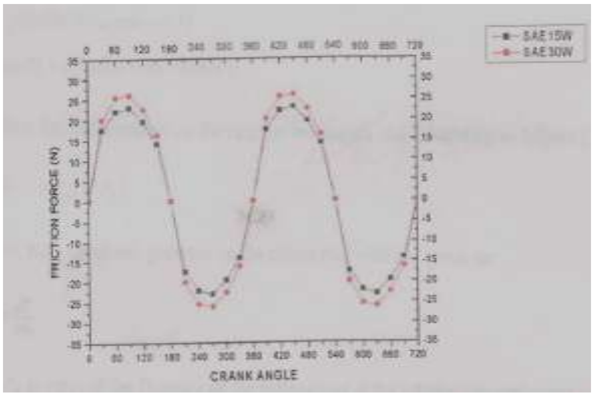
**at 3000 rpm**




**at 2250 rpm**



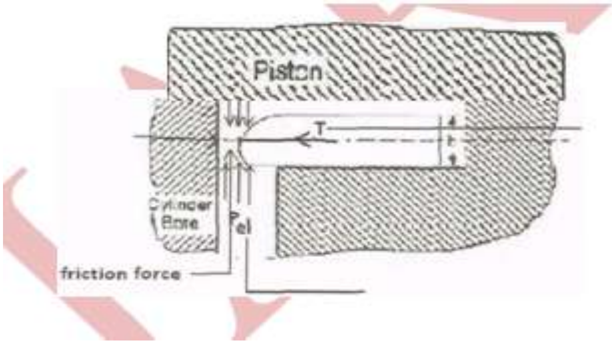

**at l550rpm**




**Graph No. 1.2 Friction force vs. crank angle at 2250 rpm**

**Graph No1.3 Friction force vs. crank angle at 1550 rpm**

### 7.1 Evaluation of Friction Force for the Parabola Ring Face Profile.




The friction force produced in the PRA system in which ring is axis symmetric in shape is used as shown in fig.1.7. In this PRA system the viscous friction force acting on the ring is calculated with assumption of boundary film lubrication. The boundary lubrication is occurred when the value of the oil film thickness below the composite surface roughness of the piston ring and cylinder wall [7].

The friction force on the ring is obtained as [7]

$$Fp=\mu\ [\Pi\ DW\ (\Theta)\ \text{sign}\ (-Vp)]$$

Where, D = cylinder bore =0.068 m

W (Θ) is the load imposed on the ring per unit length and calculating as follows [7]

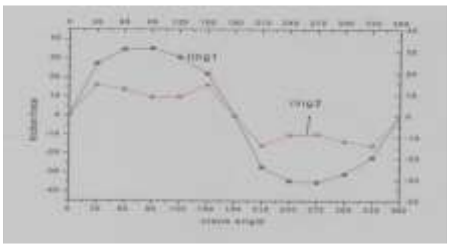
$$W\ (\Theta) = b[Pel+P\ \Theta\ ]$$

Where, Pel is the elastic pressure on the piston ring which is given as:

$$Pel=2T/Db$$

And P Θ is value of the Pressure on the pistons ring at the internal circumference with any crank angle Θ. π- is the coefficient of friction whose value is taken as 0.08. [7] The calculated value of the friction force FP for different crank angle position has been presented in Table no.5.16

Friction force graph for the different ring face profile




**Graph No. 1.4 Friction force vs. crank angle for the ring geometry**

**TABLE NO. 1.4 The friction force Fp value at different crank angle**

| S.No. | e | W(B} | $F_P$ |
|---|---|---|---|
| 1 | 0 | 946.500 | 0 |
| 2 | 30 | 929.400 | 15.8836 |
| -i | 60 | 792.756 | 13.55818 |
| 4 | 90 | 570.618 | 9.7520122 |
| 5 | 120 | 587.780 | 10.045144 |
| 6 | 150 | 929.350 | 15.887138 |
| 7 | 180 | 1151 | 0 |
| 8 | 210 | 929.5 | -15.8836 |
| 9 | 240 | 587.693 | -10.045144 |
| 10 | 270 | 570.615 | -9.7520122 |
| 11 | 300 | 792.713 | -13.55818 |
| 12 | 330 | 929.400 | -15.8836 |
| 13 | 360 | 946.500 | 0 |

### 7.2 Evaluation of Pressure Distribution of Ring on Cylinder Wall for the Existing (Rectangular face Profile) Piston Ring

The geometry of the piston ring depends on the pressure distribution applied on the internal surface of the cylinder. The mean (average) pressure of the pistons ring on the internal surface of the cylinder can be given as. The load on the piston ring is changes with the variation of the pressure distribution during the engine cycle; the load per unit length is calculated on this pressure at the different crank angle.

$$P_{el} = \frac{0.142 * E * G}{\left\{\left(\frac{D}{b}-1\right)^{3} * D\right\}}$$

Where,

E: is the elastic coefficient of the piston material =107*10 $^9$N/m$^2$

G: is the clearance of the pistons ring in Free State = 0.02 m

\D: is the cylinder diameter = 0.068 m

t: is the diametrical thickness of the pistons ring=0.00225 m

Pressure of the pistons ring at any point on the circumference of radius R with an angle Ѳ can be given as:

$$P_\theta = P_{el}\,(1 + 0.42\cos 2\theta - 0.18\cos 3\theta)$$

Where:

$\theta$ : is the polar coordinate of the point at which is calculated [4]

Also the radius of the pistons ring (R) at the free situation can be calculated by ;

$$R = r_m + \frac{t}{2} + G\left\{\left(\frac{g}{2r_m}\right)Y + X\right\}$$

Where;

$r_m$ : Is the mean piston rind radius, which can be calculated as

$$r_m = \left(\frac{D-t}{2}\right)$$

Using MATLB program pressure distribution around the ring is determined, X and Y values can be determined from table no (5.17) ,where the relation between X and Y values and $\theta$ can be given as

$$X = 9*10^{-8} - 2*10^{-5}\,\theta + 0.1109$$

$$Y = 6*10^{-10}\,\theta^4 - 3*10^{-7}\,\theta^2 - 0.0006\theta + .0024$$

Maximum bending stress for the existing (rectangular face Profile) Piston ring

$$\sigma_{\max} = 2.16 * P_{el} * \left(\frac{D}{t} - 1\right)^2 \; MPa$$

Equations will be used to calculate:

- The pressure caused by the piston on the surface of the cylinder
- The outer pistons ring radius, and
- The maximum stress the ring exposed

**Table No.1.5 X and Y values**

| $\theta$ | X | Y |
|---|---|---|
| 0 | 0.1107 | 0.0000 |
| 10 | 0.1117 | 0.006 |
| 20 | 0.1149 | 0.0023 |
| 30 | 0.1201 | 0.0050 |
| 40 | 0.1270 | 0.0083 |
| 50 | 0.1352 | 0.0121 |
| 60 | 0.1443 | 0.0157 |
| 70 | 0.1536 | 0.0190 |
| 80 | 0.1628 | 0.0199 |
| 90 | 0.1711 | 0.0189 |
| 100 | 0.1778 | 0.0148 |
| 110 | 0.1823 | 0.0070 |
| 120 | 0.1840 | 0.0050 |
| 130 | 0.1823 | 0.0211 |
| 140 | 0.1766 | 0.0409 |
| 150 | 0.1667 | 0.0635 |
| 160 | 0.1522 | 0.0876 |
| 170 | 0.1338 | 0.1114 |
| 180 | 0.1112 | 0.1331 |

Pressure distribution for the existing rectangular shaped ringh is determined as

$$P_{el} = \frac{0.142 * E * G}{\left\{\left(\frac{D}{t} - 1\right)^3 * D\right\}}$$

Pressure of the pistons ring at any point on the interal circumferece of radius R with an angle $\theta$ can be given as

$$P_\theta = P_{el}\,(1 + 0.42\cos 2\theta - 0.18\cos 3\theta)$$

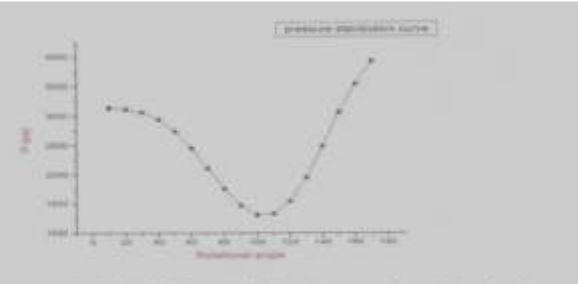
Stress calculation

$$\sigma_{\max} = 2.16 * P_{el} * \left(\frac{D}{t} - 1\right)^2$$

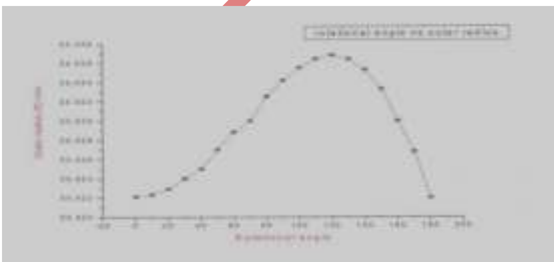
$$= 2.16 * .253123 * [(0.068/0.0025) - 1]^2$$

$$= 466 MPa$$

The value of the stress for the piston ring is within the limit of bending stress of the material used



Graph No. 5.7 Pressure distribution versus the rotational angle



Graph No. 5.8 Piston ring outer radius versus the rotational angle

## VIII RESULT & DISCUSSION

The variation of the friction force seems to be sinusoidal in nature and the maximum values of friction force are seen at crank angle position 60°, 270°, 450°, and 630°respctively for the existing piston ring profile. The maximum value of the friction force at various engine speeds are seen to be high for the lubricant SAE30W when compared with that for the lubricant SAE15W for the existing ring profile. The graph no.1.6 showing the variation for friction force for the parabolic ring face profile at various crank angle position suggest that the frictional force variation is not sinusoidal, the maximum values of friction force are seen at 30°, 150° , 210°, and 330°.The graph No 5.6 suggests higher value of frictional force for the existing (rectangular cross section) piston ring when compared with the parabolic ring profile.

## IX CONCLUSION

The experimental work was conducted on a four stroke 4- cylinder SI engine with an application of different lubricants (SAE15W and SAE30W) on given piston ring geometry for speeds ranging from 1550 rpm to 3800 rpm. It was inferred that the piston ring geometry plays an important role to reduce the PRA friction.As engine speed increases the friction force also increases.The power loss due to the friction at the interface of the piston ring and the cylinder liner is significant and is the major contributor to the total frictional.Power losses in I.C. engine. The friction power loss at piston ring assembly seems to increase with engine speed. It is seen that the maximum values of the frictional force are higher for the lubricant SAE30W which may cause higher wear of the rings. The use of lubricant SAE15W in the given setup would cause less wear of piston rings due to reduce frictional forces. The existing (rectangular shaped) piston ring profile may undergo more wear as compared to the barrel shape or parabolic profile due to increases friction at the piston ring liner interface CAD models can be developed for the slider crank mechanism of the SI engine and can be simulated to perform the kinematic analysis. Piston side thrust transmitted to the liner via rings and piston skirt could be computed to study the wear pattern. The frictional power losses associated with different ring geometries and ring face coating can be studied experimentally. The effect of variation in the number of rings and their location on the piston surface could be studied. Finite element analysis of the piston ring assembly could be carried out to study. The stress pattern of rings at various operating conditions.

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