International Research Journal of Engineering and Technology (IRJET) e-ISSN: 2395-0056
Volume: 08 Issue: 09 | Sep 2021 www.irjet.net p-ISSN: 2395-0072

A Comparative Analysis of Effect of Friction on Four Stroke 4 Cylinder Petrol Engine using Different Engine Oil

Chetan tare[1], Dr. Ankur Geete[2], Santosh Rathore[3]

[1]M.TECH STUDENT, Dept. of Mechanical Engineering, S D Bansal College of Technology,Indore
[2]Associate Prof., Dept. of Mechanical Engineering, S D Bansal College of Technology,Indore
[3]Assistant Prof. Dept. of Mechanical Engineering, SAGE University, Indore

---

Abstract: In the internal combustion engine not all the work transferred to the piston from the gases contained inside the cylinder the Indicated work is available at the drive shaft for actual work.. 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 force. This work attempts to evaluate the frictional force at the piston ring liner assembly by using different engine oil. To evaluate the frictional force a model prepared by the YUKIO is considered. The experimental work was conducted on a four stroke 4-cylinder SI engine 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 of 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.

Key words:- PRA , internal combustion engine , friction, YUKIO model, SAE.

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. Mechanical losses due to friction account for between 4 to 15 % of the total energy consumed in modern internal combustion engine 40-50% of those total mechanical losses occur in the power cylinder and half of the power cylinder friction losses come from friction generated by the piston ring as a result, a reduction in piston ring friction has the potential to improve engine efficiency lower fuel consumption and reduce emissions. 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. 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.

DATA COLLECTION AND ANALYSIS OF DATA

1 Engine specification:

Name	Description
Manufacturer	The premier automobiles ltd. Pune
Engine type	4-cylinder,4-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
Cooing	Cooling water circulated by centrifugal pump
Lubrication	Force lubrication with gear pump by pass oil filter
Fuel supply	Fuel supplied by mechanical pump, down draught carburetor with economy setting.
Ignition system	Battery ignition system

ENGINE SETUP OF 4-STROKE PETROL ENGINE

Estimation of Indicated Power, Brake Power and Friction Loses using Morse Test. [13]

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 $$B.P. = \frac{WN}{2000} \text{ kW} \quad (5.1)$$

Where,

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

N= Engine Speed (rpm)

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

There is 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

$$I.P. = [(B - B_1) + (B - B_2) + (B - B_3) + (B - B_4)] \text{, kW} \quad (5.2)$$

$$I.P. = [(I.P.)_1 + (I.P.)_2 + (I.P.)_3 + (I.P.)_4] \quad (5.3)$$

Where, $I.P._1$=23.9 kW , $I.P._2$=17.71 kW , $I.P._3$=12.47 kW , $I.P._4$=9.08 kW

The total friction power may be calculated as follows;

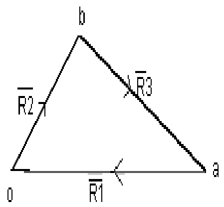
$$F.P. = [(B_1 - (I.P.)_1) + (B_2 - (I.P.)_2) + (B_3 - (I.P.)_3) + (B_4 - (I.P.)_4)] \quad (5.4)$$

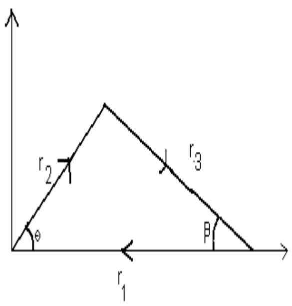
The value of Indicated Power, Brake Power and Frictional Power loss are shown in Table no.5.2

S. NO.	Engine RPM	B.P. (kW)	I.P. (kW)	FP (kW)	P_I (MPa)	% of total F.P. in the PRA
1	3800	11.4	23.9	12.4	0.34630	64.85
2.	3000	9	17.71	8.71	0.32101	64.76
3.	2250	6.75	12.47	5.72	0.31182	64.055
4.	1550	4.65	9.08	4.43	0.32260	47.29

Estimation of Instantaneous Piston Velocity.

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





Vector Diagram Phase Diagram

From the above phase diagram (5.4a)

$$R_1 + R_2 + R_3 = 0 \quad (5.6)$$

Expressing the above vector diag.(5.4b) into the complex rectangular notation

$$(-r_1 + j0) + (r_2 + jr_2 \sin\theta) + (r_3 \cos\theta - jr_3 \sin\theta) = 0 \quad (5.7)$$

Where,

r_1= linear displacement of slider (piston) =0.075m

r_2= radius of crank=0.0375m

r_3=length of connecting rod= 0.156m

θ=angular displacement of crank, degree

β= angular displacement of connecting rod, degree

ω_2= angular speed of crank, rad/s

ω_3 = angular speed of connecting rod, rad/s

Now, from the above equation

$$-r_1 + r_2\cos\theta + r_3\cos\beta = 0 \quad (5.8)$$

$$r_2\sin\theta - r_3\sin\beta = 0. \quad (5.9)$$

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

$$\sin\beta = \frac{r_2}{r_3}\sin\theta$$

So angular displacement of connecting rod at any angle θ is

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

(5.10)

Angular speed of connecting rod is $\omega_3 = \frac{r_2\omega_2}{r_3\cos\beta}$, rad/s

(5.11)

Piston velocity at any crank angle position is given as $V_p = \dot{r_1} = r_2\omega_2\sin\theta + r_3\,\omega_3\sin\beta$, m/s

(5.12)

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 ω_3, ω_2, β and V_p are calculated as below.

r_2 = 0.075/2 m

r_3 =0.156 m

$$\omega_2 = \frac{2\pi N}{60} \quad \text{rad/sec}$$

= 2*π*3800/60

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

Using Eq.(5.10) and (5.11) connecting rod angle and angular speed of connecting rod at θ=30° may be calculated as follows

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

$$\beta = \sin^{-1}\left(\frac{.0375}{0.0156}\right) * \sin 30$$

$\boldsymbol{\beta}$ **= 6.90°**

$$\omega_3 = \frac{0.0375 * 397.93}{0.156\cos 6.90}$$

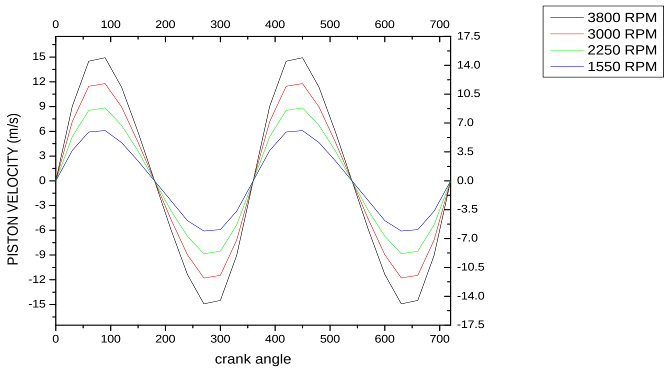
$\boldsymbol{\omega_3}$ **= 83.44 rad/s**

And using

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

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

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



Similarly at the different engine speed the piston speed has been calculated

Graph showing piston velocity for various crank angle in one complete cycle at speed3000-1550 rpm

Estimation of Static Ring Tension of Existing Piston Ring

The static ring force (Tension) is easily obtained using Castigliane's theorem. Castigliane's theorem [4] 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.

Using Castiglione's theorem the gap closure of the piston ring is derived as

$$g = \frac{\partial SE_r}{\partial T} = \int_0^{\pi} \frac{M_r R dy}{2EI} \left(\frac{\partial M_r}{\partial T}\right) dy$$

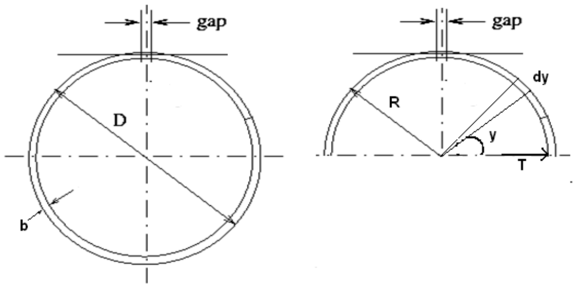


Fig.5.5Diagram Showing Forces Acting on the Section of the Ring

Now, from Fig No.5.5,

$$M_r = TR \sin y \qquad (5.13)$$

Where,

$M_r =$ is the bending moment of the ring (N-m)

$T =$ is the static ring tension (N)

So substituting M_r into equation 5.12 yields

$$T = \frac{2CEIg}{\pi R^3} \quad N \qquad (5.14)$$

Where

$R = \frac{D}{2} - \frac{b}{2}$ is the mean radius of the ring.

I= moment of inertia of the ring

C=1.778 is the correction factor [16].

By substituting these values in equation (5.12) the static tension can be expressed as

$$T = \frac{Eg}{7.07D\left[\frac{D}{b}-1\right]^3} \pi D b \quad N \qquad (5.15)$$

$$= \frac{107*10^9*.001*.068*.00225*3.14}{7.07*.068\left[\frac{0.068}{0.00225}-1\right]^3}$$

The static ring tension is calculated as $T = 42.05\ N$

Where,

g= 0.001 m (piston ring gap)

E=107*10^9 N/m^2

Friction force analysis

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

$$F_p = C_1 * [\,\mu_k * V_p * 200 * \left(\frac{T}{D}\right)]^{0.5}] \qquad (5.16)$$

Where,

C_1= constant value for the lubricating oil [2]

For SAE15W C1=8.136

SAE30W C1= 8.135

T = ring tension

μ_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.

RESULT:

The variation of the friction force seems to be simulated in nature and the maximum value of friction force are seen at crank angle position 60°, 270° , 450°, and 630°resptively for the existing piston ring profile. (Referring fig no.--- at various engine speed).

The maximum value of the friction force is seen to be high for the lubricant SAE15W when compared with that for the lubricant SAE30W for the existing ring profile.

Friction Force Calculation

TABLE NO. 5.8: Friction force value for SAE15W

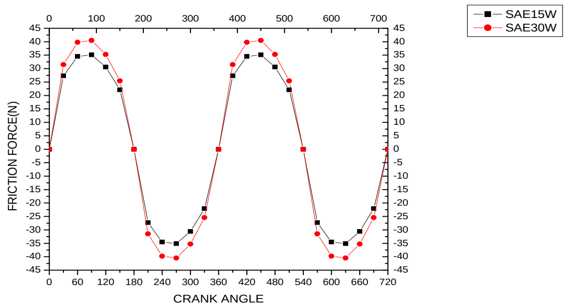
Sr.	Crank angle θ (deg.)	Piston velocity (m/s)	Fp (N)
1	0	0	0
2	30	9.02399	27.30
3	60	14.490	34.522
4	90	14.922	35.115
5	120	11.336	30.606
6	150	5.895	22.0712
7	180	0	0
8	210	-5.895	-27.30
9	240	-11.336	-34.522
10	270	-14.922	-35.115
11	300	-14.490	-30.606
12	330	-9.02390	-22.0712
13	360	0	0
14	390	9.02399	27.30
15	420	14.490	34.522
16	450	14.922	35.115
17	480	11.336	30.606
18	510	5.895	22.0712
19	540	0	0
20	570	-5.895	-27.30
21	600	-11.336	-34.522
22	630	-14.922	-35.115
23	660	-14.490	-30.606
24	690	-9.02390	-22.0712
25	720	0	0

Table 5.9: Friction force value for SAE30W

Sr.	Crank angle θ°	Piston velocity m/s	Fp (N)
1	0	0	0
2	30	9.02399	31.462
3	60	14.490	39.774
4	90	14.922	40.458
5	120	11.336	35.263
6	150	5.895	25.429
7	180	0	0
8	210	-5.895	-31.462
9	240	-11.336	-39.774
10	270	-14.922	-40.458
11	300	-14.490	-35.263
12	330	-9.02390	-25.429
13	360	0	0
14	390	9.02399	0
15	420	14.490	31.462
16	450	14.922	39.774
17	480	11.336	40.458
18	510	5.895	35.263
19	540	0	25.429
20	570	-5.895	0
21	600	-11.336	-31.462
22	630	-14.922	-39.774
23	660	-14.490	-40.458
24	690	-9.02390	-35.263
25	720	0	-25.429

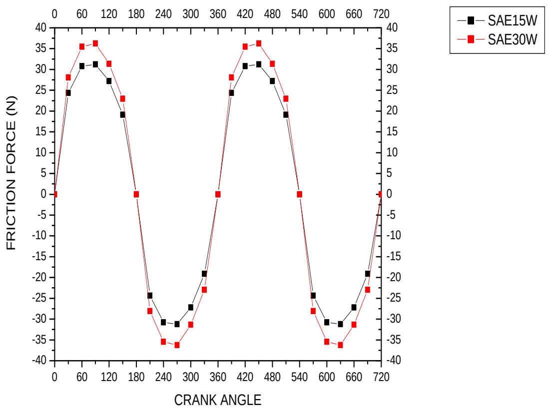
Friction force graph using different lubricating oil at different engine speed.

At 3800 rpm

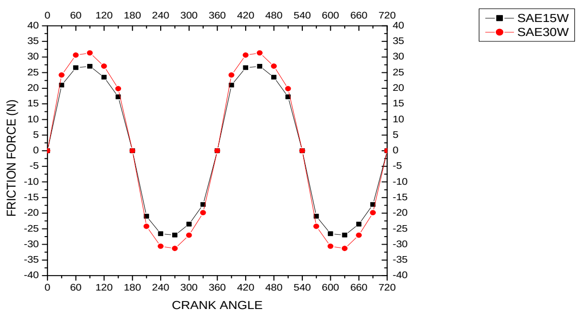


Similarly at different engine speed.

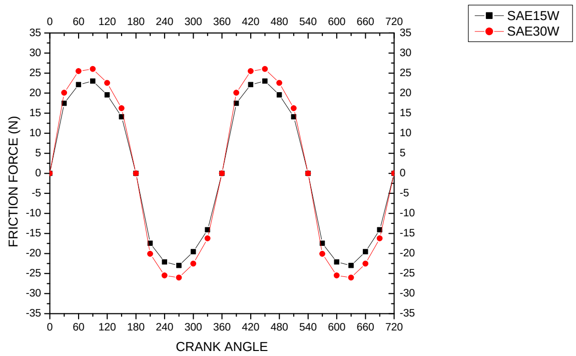
At 3000 rpm



At 2250 rpm



At 1500 rpm



Conclusions

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 speed range from 1550 rpm to 3800 rpm.. It was inferred that piston ring geometry plays an important role to reduce the PRA friction.

As engine speed increases the friction force also increases. It is seen that the maximum value of the frictional force is higher for the lubricant SAE30W which may cause of 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.

References:-

1. H.D. Desai "Computer Aided Kinematic and Dynamic Analysis of a Horizontal Slider Crank Mechanism Used For Single Cylinder Four Stroke Internal Combustion Engine". Proceeding of world congress on engineering 2009 vol.II WCE 2009, July 01-03-2009.
2. B.M. Sutaria, D.V. Bhatt and K.N. Mistry "Simulation of Piston Ring Friction Model of Single Cylinder Internal Combustion Engine" world applied science journal 7(8) 998-1003, 2009
3. D.V. Bhatt, M.A.Bulsara and K.N. Mistry "Prediction of Oil Film Thickness in Piston Ring Cylinder Assembly in an I.C. Engine: A Review" Proceeding of world congress on engineering 2009 London U.K.
4. Muntaser Momani, Sayel M. Fayyad, Suleiman Abu-Ein, Waleed Momani, and Hisham Mujafet "Computerized Mathematical Model for Studying Geometric Shape and Stresses Applied on the Pistons' Rings of ICE." Journal of Applied Sciences Research, 6(7): 905-908, 2010.
5. Li Ming Chu Yuh Pang Chang and Jung-Hua yang "Profile Design of Piston Ring Using Inverse

Method" Journal of Marine Science and Technology; vol. 16 no. 1 pp. 64-70 2008.

6. Richard Mittler, Albin Mierbach and Dan Richardson "Understanding the fundamentals of Piston Ring Axial Motion and Twist and the Effects on Blow-by" Proceeding of the ASME Internal Combustion Engine Division 2009 spring technical conferenceICES2009.
7. Yeau-Ren Jeng "Theoretical Analysis of the Piston Ring Lubrication Part 1- Fully Flooded Lubrication". Tribology Transaction, vol.35 (1992), 4,696-706.
8. Rebecca M. Hoffman "Robust Piston Design and Optimization Using Piston Secondary Motion Analysis". SAE technical paper series 2003-01-0148.
9. F.S.Silva "Fatigue Engine Pistons- A Compendium of Case Studies" Engineering Failure Analysis, 13(2006) 480-492.
10. Grant Smedley "Piston Ring Design for Reduce Friction in Modern Internal Combustion engine" Master Thesis, Massachusetts Institutes of Technology published in May 2004 Page no.16
11. Eric J. Deutsch "Piston Ring Friction Analysis from Oil Film Thickness Measurement" Master Thesis, Massachusetts Institutes of Technology published in February 1994.
12. John B. Heywood Chapter no.13 Engine Friction and lubrication in "Internal Combustion Engine Fundamental" Tata McGraw-hill series. Page no.711-722
13. Kirpal Singh, Chapter no.2.5 Constructional Details of Piston Ring Assembly in "Automobile Engineering vol.2" A.K. JAIN publication, 9th edition 2004. Page no. 36-45