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Simulation of Piston Ring Friction Models of Single Cylinder Internal Combustion Engine

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Abstract: The friction losses in the piston ring assembly (PRA) vary from 13% to 18% as reported in the literature. To reduce these frictional losses, various parametric approaches are made particularly at design stage and at experimental level. PRA friction is very complex phenomena under dynamic condition. Proper lubrication and selection of material pair are contributes to optimize the PRA friction. The theoretical model for PRA friction as reported in literature are being developed by considering different variables. These models are developed either theoretically or experimentally. The models are for specific PRA system with different capacity. The variable parameters are piston velocity, engine speed, oil viscosity, gas pressure, crank angle, film thickness and coefficient of friction. Non-variables are system constant, bore diameter, ring tension, ring width, compression ratio, reciprocating mass, piston ring area and piston ring profile. The major assumptions for developing models are either hydrodynamic lubrication theory or mixed lubrication theory of Reynolds equation. Determine variable and non-variable parameters for 100 cc reciprocating systems. Efforts are to identify the system constants and co-related for common system under consideration.

Key words: PRA · Lubrication · Friction force

INTRODUCTION

The tribological behavior of piston ring has long recognized as an important influence on the performance of internal combustion engines in terms of friction power loss, fuel consumption and harmful exhaust emissions. The primary role of the PRA is to maintain an effective gas seal between the combustion chamber and crankcase. The reciprocating motion of piston ring in cylinder liner creates drastic change in the pressure, the combustion event generates large amount of heat. The secondary role of PRA is to transfer this heat from the piston in to the cylinder wall. These can be effective by considering alloy materials, which may lead to prolong life of the cylinder liner and piston. The tertiary role of PRA is to limit the amount of oil that transported from the crankcase to the combustion chamber by oil control ring. This flow path is probably largest contributor to oil consumption of an engine leads to increase in harmful exhaust emission as the oil mixes and react with the other contents of the combustion chamber.

The drawback of I.C. Engine includes relatively low thermal and mechanical efficiencies with a very significant proportion of energy of the fuel is being dissipated as heat. About 60% of energy is dissipated in the form of heat, either from the engine surface or from exhaust pipe. Mechanical action may account for a further friction loss of the order of the 18%, leaving only a quarter of the original energy in terms of brake power. The break down of the mechanical losses is in the engine with PRA being responsible. It is noted almost 50% of mechanical losses because of PRA. Further, other losses associated with accessories and pumping indicated about 20% or more of the mechanical losses. It is clear that the energy and mechanical loss distribution depends on engine capacity, lubricants and operating conditions.

The PRA must fulfill cited three roles with a minimum of frictional power loss, most probably at sliding interface with the cylinder wall and minimum wear in order to maximized component life. Piston ring pack frictional losses are grater than those of piston skirt at low to moderate engine speeds but the situation may be reversed at high engine speeds due to larger wetted area of the piston skirt contributing to viscous friction. In term of wear, there is insufficient lubricating oil and to understand the interaction with lubrication mechanism. Theoretical and experimental study of piston ring lubrication has received much attention in the literatures. The

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mathematical analysis of piston ring lubrication is complex and required simplifying assumptions. However, rapid development in numerical methods over the last two decays has result in sophisticated PRA friction model.

In this paper, efforts are made to study the importance of engine performance and to analysis PRA friction. Theoretical study of piston-ring lubrication has received much attention in the literature. The mathematical analysis of PRA friction is complex and necessities require simplify assumptions. Three different models are applied to a predefined PRA system i.e 100 CC. Simulated results have explained by considering different lubricating oil for different engine speed.

Review of Pra Friction: Castleman [1] made earliest calculations on piston ring and cylinder liner lubrication. Eilon and Saunders [2] have calculated the lubricant film thickness based on the balance of radial forces. The squeeze film effect incorporated into the analysis by Furuhama [3], the ring surface was model as two circular arcs connected by a flat section. Ting and Mayer [4 and 5] developed an analytical model for determining the ring-bore wear mechanism for a reciprocating piston engine over a complete running cycle. They used hydrodynamic lubrication theory to analyze the flow between ring and cylinder bore. They included geometry and elasticity of the ring, blow-by through piston ring pack, minimum film thickness permitting film lubrication and piston side thrust load. In this model, assumed that there is no pressure change in the divergent portion of the ring wedge.

Hill and Newman [6] have developed reduced friction piston rings based on simple analysis. They determined five design features that could be altered to reduce the friction, which include axial width, surface pressure, number of rings, effective face ring profile and geometry of ring and bore. They conclude that surface pressure should be reduced for low friction, but that would result in higher levels of the oil consumption and blow by.

Nakada *et al.* [7] released a study that focused on surface treatments to reduce friction and wear. He also suggested removing one piston ring out of available on piston ring assembly system to reduce friction. Cullen *et al.* [8] have investigated in the cross-section compressing rings made for steel rather than cast iron. They found that steel rings with a significantly larger free gape could be used because of higher strength of steel compared to cast iron and hence the wear rate is reduces. Tomanik *et al.* [9] have performed a study, which reduced oil ring, tensions such that unit pressure decreased from 1.1 to 0.8 N/mm^2. Result indicates in 30 % reduction in the friction power losses.

Bhatt and Mistry [10] have experimented on 150 cc motorized piston-cylinder system with an application of different lubricants and piston ring geometry for speed ranges from 500 to 2000 rpm and found that ring geometry played important role to reduce PRA friction with same lubricant. Sharma [11] has experimentally studied the various parameters of engine tribology, experimented with various application of the piston ring geometry at low profile ring edge and offered a co-relation in the form of equations. He concluded that oil film thickness does not depend on ring thickness but is highly dependent on the ring curvature. Bolander *et al.* [12] have developed numerical model to investigate effects of surface modifications on lubrication condition and frictional loss at the interface. The modified cylinder liner, reduced cycle-average friction coefficient by 55-65%, while total energy loss per cycle reduced by 20 to 40%. In internal combustion engine the piston ring is the perhaps most complicated tribological component. It is subjected to large, rapid variations of load, speed, temperature and lubrication availability. In one single stroke of the piston, the piston ring may experience boundary, mixed and full film lubrication as per the Stribeck curve [13]. Elasto hydrodynamic lubrication of piston ring is possible in both gasoline and diesel engine on the highly loaded expansion stroke after firing.

Pra Friction Mechanism: In internal combustion engine, a major mechanical friction loss occurs at piston ring assembly (PRA). To evaluate this friction loss different researcher have explained friction phenomenon in PRA with different theories and developed mathematical relationship based either on experiment result or by simulation of a model. Sharma [11] has explained PRA mechanism and regime of lubrication with the help of Srtibeck curve. This curve explained variation of friction co-efficient and duty parameters (non-dimensional) for various regimes of lubrication in different part of the engine. The value of ë (ratio of “h” the oil film thickness and “a” the asperity height) decide type of lubrication. The oil film thickness can vary significantly between different locations and during engine cycle. The friction force, power loss and wear on piston ring depend upon the kind of lubrication present during the operation of engine. When oil film thickness is thick, no surface to surface contact between piston ring and cylinder liner, hence hydrodynamic lubrication consideration where, oil film thickness carries the load. The friction force and power loss in hydrodynamic lubrication is determined by oil film shear and oil pressure gradient in the oil film. When oil film thickness is not enough, the asperities of the two sliding surfaces each other and mixed lubrication takes place. In mixed lubrication, load is partly carried by oil film and partly carried by asperities. The friction between ring and cylinder liner increases due to this asperity contact at dead centers. This led to increases power loss and wear.

Pra Friction Models: The simulation of PRA friction models based on empirical relationship develop by various researchers. For the comparison of PRA, friction three different friction models are considered such as Hoshi[13] model, Yukio[14] model and Sharma[11] model. The efforts are made to simulate these models to understand the effect of various variables. The variable model parameters are piston velocity, engine rpm, oil viscosity, gas pressure, crank angle, film thickness, coefficient of friction and other system constants. The non-variables are bore diameter, ring tension, ring width, compression ratio, reciprocating mass, piston ring area and piston ring profile. Theoretical PRA friction models based on empirical relationship has been develop are summarized below.

Hoshi model [1] have performed on 1300 cc petrol engine and suggested that temperature of the cylinder wall changing the oil film viscosity has the greatest effect on piston friction and noted that increases of engine speed resulted in higher friction. These characteristic can be adequately represented by a formula as below.

$$F_p = \pm 0.6 D r^{0.5} N^{0.5} \mu_k^{0.5} \left[\sin\theta + \frac{\rho}{2}\sin 2\theta\right]^{0.5} X \left\{ \begin{array}{l} \left[\{\beta(Px+\mathrm{Pr})\}^{0.5} + 0.20 D^{0.5} \rho_0^{0.5} \rho^{0.5}\right] \\ X(\sin\theta) X \left[P + 0.0014(\frac{r}{D^2}) W N^2 \left\{\cos\theta + \frac{\rho}{2}\cos 2\theta\right\}\right] \end{array} \right\}^{0.5} \quad (1)$$

Yukio model [2] determined friction as function of the crank angle in actual engine operation provides essential information on the comparative effectiveness of various to reduce friction in piston rings by following relation ship.

$$F_p = C_1 \text{ x } [\mu_k \text{ x } U \text{ x } (200\ W_{RT} / D)]^{0.5} \quad (2)$$

Sharma model [3], developed a friction model of a multi cylinder diesel engine. The accuracy of the friction model has a significant in predicting transient behavior of the engine and the fuel consumption which represented as,

Table 1: Technical Specifications

1	Type	Air cooled 4 Strokeengine
2	Cylinder pressure	13.5 ±1.5 kg/ cm2 at1000 rpm
3	Engine Power	5.44 KW@ 8000RPM
4	Bore × stroke	50 × 49.5 mm
5	Compression ratio	8.8:1
6	Engine displacement	97.2 cm 3
7	Idling Speed	1400 ± 100 RPM
8	Dimensions	1885× 770× 1060 mm
9	Valve Clearance	Inlet: 0.05 mm, and exhaust: 0.05 mm

Table 2: PRA Friction Models Parameter Comparison

Sr, No.	Parameters	Models		
(A) VARIABLES		I	II	III
1.	Piston velocity	-	Y	Y
2.	Engine RPM	Y	-	-
3.	Oil viscosity	Y	Y	Y
4.	Gas Pressure	Y	-	-
5.	Crank Angle	Y	-	-
6.	Co-Efficient of friction	-	-	Y
(B) NON-VARIABLE	I	II	III	
1.	System Constant	Y	Y	Y
2.	Bore Diameter	Y	Y	-
3.	Ring Tension	Y	Y	Y
4.	Ring Width	Y	-	Y
5.	C/R ratio	Y	-	-
6.	Reciprocating Mass	Y	-	-
7.	Ring Profile (c/a)	-	-	Y

$$F_p = C_2 \times C \times S^m \quad (3)$$

Where,
$S = \mu_d \times U / (P_R / b)$
=Sommer feld no.
C and m function of "c/a" of 0.03 to 0.2 are,
C =1.9 to 2.25 and m = 0.425 to 0.525

Analysis of Pra Friction Models: The models are evaluated using variable and non-variables systems parameters. The Table 2 lists six variable parameters and seven non-variables parameters or constants for a reciprocating system under consideration.

Simulation of Models: Above-refereed models are simulated to available system under study. The simulation is done with engine speed base on crank angle v/s friction force and coefficient of friction. The computer program with simplified approach was applied to four-stroke single cylinder internal combustion (petrol) engine for study to parametric behavior of 100 cc engine. The detailed technical specifications of engine are as under.

Technical Specification: The technical specifications of 100 cc internal combustion engine used for experiment are presented in Table 1.

RESULT AND DISCUSTION

The simulated results shows 2T oil offered minimum friction force with compared to other lubricants at all engine speeds. This may be due the effect of lower viscosity of oil. The observations from Figure 1, 2 and 3 are summarized as under,

- From Figure no. 1, indicates friction force v/s engine speeds in case of 2 T oil. It is noted that as engine speed increases the friction force is subsequently increases in all friction models, how ever the model no 3 shows maximum friction occurs and minimum friction takes in friction model 1.
- From figure, no 2 and 3 are shows similar nature of friction force in all friction models for all engine speed in case of SAE 15W oil and SAE 30W lubricants. The maximum friction force is noted in case of SAE 30W oil under consider all friction models.
- From figure no 4 indicates comparison is made between theoretical models and experimental results. The experimental results are highest among all theoretical friction models. This may be over all friction force in the engine speeds.

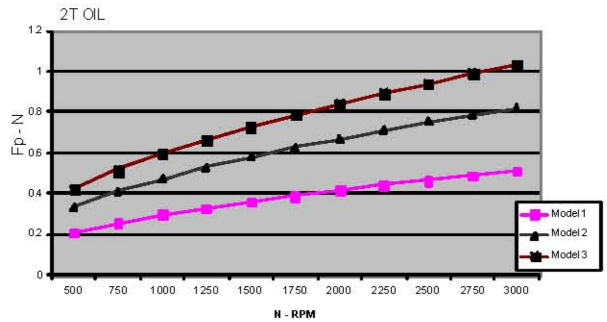

Fig. 1: Graph of Fp v/s N

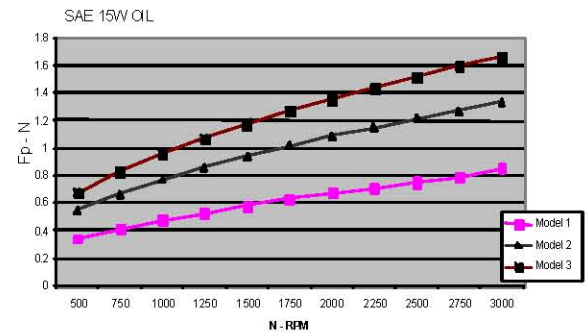

Fig. 2: Graph of Fp v/s N

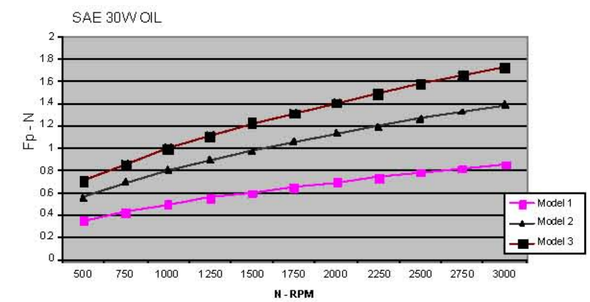

Fig. 3: Graph of Fp v/s N

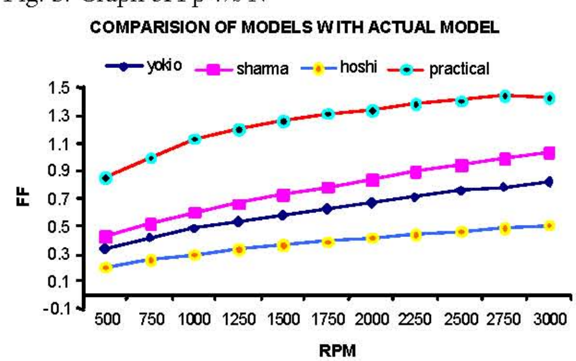

Fig. 4: Comparison of theoretical models with actual working model

Considering the above results, it is possible to establish relationship among these three models by taking Hoshi model is a basic equation. Table No 2 and 3 shows

Table 2: System constant C_1 of model 2 with model 1 under different lubricating oils

		C_1		
Engine RPM	Piston Velocity (m/s)	2T oil	SAE 15W	SAE30W
500	1.3089	8.132	8.136	8.133
750	1.9634	8.127	8.136	8.133
1000	2.6179	8.128	8.136	8.133
1250	3.2720	8.133	8.136	8.136
1500	3.9260	8.127	8.137	8.134
1750	4.5810	8.132	8.136	8.138
2000	5.2358	8.137	8.137	8.137
2250	5.8902	8.133	8.137	8.134
2500	6.5447	8.128	8.139	8.139
Average C_1 constant value		8.131	8.136	8.135

Table 3: System constant C_2 of model 3 with model 1 under different lubricating oils

		C_2		
Engine RPM	Piston Velocity (m/s)	2T oil	SAE 15W	SAE30W
500	1.3089	7.051	4.359	4.208
750	1.9634	7.046	4.356	4.205
1000	2.6179	7.048	4.357	4.205
1250	3.2720	7.051	4.362	4.209
1500	3.9260	7.047	4.357	4.206
1750	4.5810	7.047	4.317	4.206
2000	5.2358	7.046	4.354	4.206
2250	5.8902	7.046	4.357	4.205
2500	6.5447	7.048	4.360	4.207
Average C_2 constant value		7.047	4.358	4.206

Fig 4. Comparison of theoretical models with actual working model

the calculated value of system constant C_1 and C_2 for the corresponding lubricating oil. The calculated constant used to predict the PRA friction force of Yukio Model and Sharma Model. The Figure 4 shows the comparison of all theoretical models with actual working system of 100 cc S.I. Engine at laboratory scale. The practical measured value of friction force shows higher value with compare to consider three friction models at all the engine speeds. This may be because of accounting other losses in actual working conditions.

CONCLUSIONS

Following are the finding derived from the analytical and experimental study under different lubrications oil (2T, SAE15W and SAE30 W).

- Friction force increases with the speed in all the three simulated models.
- Hoshi Model has given minimum friction force value while Sharma model has given maximum friction force value at the all-respective speed.
- Yukio Model shows friction force between the value of the Hoshi and Sharma model respectively.
- The rate of increase of the friction with respect to speeds in all friction models is observed more or less constant in comparison of each model.
- The experimental results shows highest friction force with compared to all theoretical models.

Nomenclature:

a	= Height of the profile (m)
b	= Length of piston ring (m)
C	= Profile recess at the ring edge (m)
C_1,C_2	= System constant
D	= Cylinder bore diameter (m)
Fp	= Piston ring friction force (N)
f	= Coefficient of friction
L	= Length of connecting rod (m)
N	= Engine speed (RPM)
P	= Gas pressure in cylinder (Pa)
$P_{R,}W_{RT}$	= Piston ring tension (N)
P_x	= Gas pressure on top ring (Pa)
r	= Crank shaft radius (m)
S	= Sommerfeld No.
U	= Piston speed (m/s)
W	= Equivalent mass of piston in (kg)

β = Piston ring width (m)
μ_k = Kinematics viscosity (m^2/sec)
μ_d = Dynamic viscosity (Pa)
θ = Crank angle (radian)
ρ = r / L

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