ADVANTAGES OF ROLLER FOLLOWER FOR DIFFERENT BORE STROKE RATIOS AND EFFECTS ON OVERALL FRICTION OF SI ENGINE

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ABSTRACT:
This paper presents the advantages of roller follower specially over flat follower considering overall friction analysis in spark-ignition internal combustion engines. Overall friction on engine performance was examined during normal working conditions and wide range of dependent variables such as engine speed, change of bore stroke ratios with and without changing engine volume. Three different cylinder conditions / models, such as, at constant volume, at variable bore and volume (constant stroke), and lastly at variable stroke and volume (constant bore) are used. The valve train frictional losses are due to the hydrodynamic friction in the camshaft bearing, mixed lubrication in the flat followers, rolling contact friction in the roller followers, and both mixed and hydrodynamic friction due to the oscillating motion of the lifters and valves. Flat follower and roller follower friction also examined and found that flat follower gives greater friction mean effective pressure then roller follower. At low engine rpm, when intake manifold pressure increases, it leads to decreases the pumping losses, but very less decrease at high engine rpm.

keywords: *Engine friction, Roller follower, Flat follower, Bore stroke ratio, Constant volume model, Constant stroke model, Constant bore model.*

1.0 INTRODUCTION

The frictional processes in an internal combustion engine can be categorized into three main components: mechanical friction, pumping work and accessory work. The mechanical friction includes the friction of internal moving parts such as the crank shaft, piston, rings, and valve trains. The pumping work is the net work done during the intake and exhaust strokes. The accessory work is the work required for operation of accessories such as the oil pump, fuel pump, alternator and fan.

At low speed with lower bore stroke ratio and at high speed with higher bore stroke ratio, constant bore model gives less friction. At high speed with lower bore stroke ratio, constant bore model gives less friction irrespective of compression ratio [1].

On air standard power cycles, it is shown that friction is generally neglected for simplicity of the analysis. However, due to the high speed of the engine, this assumption becomes less realistic where a large percentage of engine power is dissipated into friction [2].

Frictional losses are a significant fraction of the power produced in an internal combustion engine, minimization of friction has been a major consideration in engine design and operation. Engines are lubricated to reduce friction and prevent engine failure. The friction energy is eventually removed as waste heat by the engine cooling system [3].

Piston skirt area was reduced axially generally a friction benefit was observed. When the skirt area was reduced circumferentially there was little benefit. This may be because a piston is typically made with a large ovality. Increasing skirt clearance was another way to reduce friction, but it was noted that this might increase cavitations problems. Reduced mass of the piston was cited by many as a way to reduce friction [4].

In SI engine the largest single contributor for friction is the pistons, pins, rings and rods, and after that at low speeds the valve train (but at high speeds, the crankshaft and seals, followed by the valve train). The approximate percentage at mid range are 12% for crankshaft and seals, 46% for pistons, rings, pins and rods, 23% for the valve train, 6% for the oil pump and 13% for water pump and alternator [5].

For a given displacement shorter stroke requires a larger bore, which results in greater heat losses due to the larger cylinder surface area. Greater flame travel distance also increases knock problems. This is why most medium sized engines (automobile engines) are close to square; with b ≈ s. Piston friction force will be proportional to oil viscosity, engine speed and imep (indicated mean effective pressure) [6].

It is important to bear in mind that the measured piston-assembly friction force is the summation of four main components: two compressions rings, an oil control ring, and the piston skirt. Therefore a change of a variable may produce different and even conflicting effects for each component. For example at moderate lubricant temperatures, the piston skirt operates in the hydrodynamic regime, whereas the piston rings operate in the boundary to hydrodynamic lubrication regimes. Any increase in lubricant temperature would bring the piston-ring lubrication conditions more toward boundary, increasing the friction loss, whereas the decrease in viscosity would reduce the friction contribution from the piston skirt because of a reduction of shear loss [7].

2.0 MODELING AND SIMULATION OF SI ENGINE:

2.1 Governing Equations of the SI Engine Friction:

Any cycle can be explained by some governing equations with or without boundary conditions. Ideal cycle consider only combustion at constant volume process without any heat losses and friction. But actual cycle need to considered heat losses and friction. Although the complexities of simulating the actual cycle are many, yet most of the deviations from ideal cycle can be modeled using classical thermodynamics. There are main five types of friction calculations are considered here as per [1], [3] and [8]:

Crankshaft Friction Mean Effective Pressure

$$fmep_{bearings} = c_b \frac{n_b N D_b^{\,3} L_b}{n_c b^2 s} \quad \text{(1)}$$

$$fmep_{seals} = c_s \frac{D_b}{n_c b^2 s} \quad \text{(2)}$$

Piston Friction Mean Effective Pressure

$$fmep_{skirt} = c_{ps} \frac{U_p}{b} \quad \text{(3)}$$

$$fmep_{rings} = c_{pr} \left(1 + \frac{1000}{N}\right) \frac{1}{b^2} \quad \text{(4)}$$

$$fmep_{gasload} = c_g \frac{P_i}{P_a} \left[0.088r + 0.182r^{(1.33 - KU_p)}\right] \quad \text{(5)}$$

Valve Train Friction Mean Effective Pressure

$$fmep_{cam} = c_c \frac{N n_{cs}}{n_c b^2 s} \quad \text{(6)}$$

$$fmep_{ff} = c_{ff} \left(1 + \frac{1000}{N}\right) \frac{n_v}{n_c s} \quad \text{(7)}$$

$$fmep_{rf} = c_{rf} \frac{n_v N}{n_c s} \quad \text{(8)}$$

$$fmep_{oh} = c_{oh} \frac{n_v L_v^{\,3/2} N^{1/2}}{n_c b s} \quad \text{(9)}$$

$$fmep_{om} = c_{om} \left(1 + \frac{1000}{N}\right) \frac{n_v L_v}{n_c s} \quad \text{(10)}$$

Pumping Friction Mean Effective Pressure

$$\Delta P_{im} = P_a - P_i \quad \text{(11)}$$

$$\Delta P_{iv} = c_v \left(\frac{P_i}{P_a} \cdot \frac{U_p b^2}{n_{iv} D_{iv}^2}\right)^2 \quad \text{(12)}$$

$$\Delta P_{ev} = c_v \left(\frac{P_i}{P_a} \cdot \frac{U_p b^2}{n_{ev} D_{ev}^2}\right)^2 \quad \text{(13)}$$

$$\Delta P_{es} = K_{es} \left(\frac{P_i}{P_a} U_p\right)^2 \quad \text{(14)}$$

Accessory Friction Mean Effective Pressure

$$fmep_{accessory} = a_1 + a_2 \left(\frac{N}{1000}\right) + a_3 \left(\frac{N}{1000}\right)^2 \quad \text{(15)}$$

All proportionality constant and additional abbreviations are as per [1] and [3]. When calculating valve train friction including the flat and roller follower's effects table 1 and table 2 were considered:

Table 1 Valve train types [3].

Type of Valve	Position of Cam	Rocker arm / Follower
Type I	OHC	Direct acting/flat or roller follower
Type II	OHC	End pivot rocker/flat or roller follower
Type III	OHC	Center pivot rocker/flat or roller follower
Type V	CIB	Rocker arm/flat or roller follower

Table 2 Coefficients for valve train friction terms [8].

Configuration	Type	Flat follower, c_{ff} (kPa-mm)	Roller follower, c_{rf} (kPa-mm-min/rev)	Oscillating hydrodynamic, c_{oh} (kPa-mm-min/rev)$^{1/2}$	Oscillating mixed, c_{om} (kPa)
Single overhead cam (SOHC)	Type I	200	0.0076	0.5	10.7
Double overhead cam (DOHC)	Type I	133	0.0050	0.5	10.7
Single overhead cam (S OHC)	Type II	600	0.0227	0.2	42.8
Single overhead cam (SOHC)	Type III	400	0.0151	0.5	21.4
Cam in block (CIB)	Type IV	400	0.0151	0.5	32.1

2.2 SI Engine Friction Model:

In this present article, the overall friction analysis in spark ignition engine was modeled by developing the crank shaft friction model, piston friction model, valve train friction model, pumping friction model and accessories friction model. Flat follower and roller follower friction also considered separately. The simulation is done by using FORTRAN 95. Developed fundamental equations and impirical relations have been used for the dependence of the various modes of friction work on overall engine parameters as bore, stroke and engine speed, and then construct an overall engine friction model.

Before the start of the computation, the design characteristics of the engine are given in Table 3. Proportionality constants are used which was included with oil properties [3], [8] and [9].

The results obtained from the simulation of a 4 cylinder four stroke spark ignition engine are reported. Firstly, friction for flat and roller follower were presented and then its overall effect also produced. Effects of intake manifold pressure also presented. Results also verified to the established results from different sources [3].

Secondly, the variation of engine speed, compression ratio and bore stroke ratio are considered. The preceding component analysis can be combined to form an overall engine fmep. It should be noted that the coefficients used in the fmep model are likely to depend on oil properties that have not been explicitly included in the analysis.

Table 3 Specifications of the Engine used for simulation [3].

Bore (mm) & Stroke (mm)	86 & 86
Intake & Exhaust Valves /Cylinder	2 & 2
Intake & Exhaust Valve Diameter (mm)	35 & 31
Maximum Valve Lift (mm)	11
No of Crankshaft Bearings, Dia. (mm) & Length (mm)	5, 56 & 21
No of Connecting Rod Bearings, Dia. (mm) & Length (mm)	4, 48 & 42
Number of Camshafts	1

Figure 1 shows that the relative magnitudes of the three piston friction terms. The skirt and rod bearing fmep increase linearly with engine speed, while the piston ring fmep decreases with engine speed. At low speed, most of the friction is due to the piston rings, and at higher speeds, the majority of the friction is from the piston skirt. The results are in excellent agreement with [3].

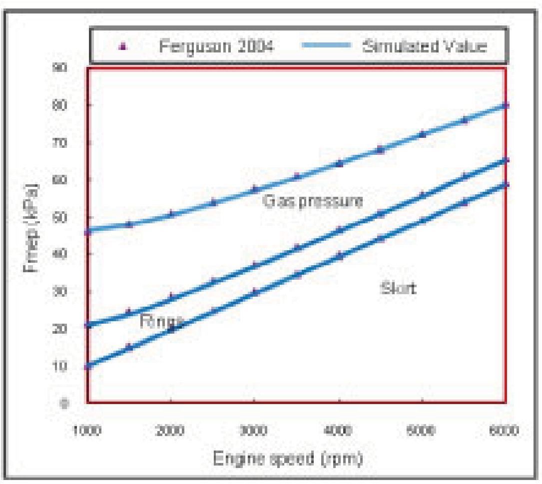


Fig. 1. fmep and Components of piston friction versus engine speed.

2.3 Estimation of Friction for Flat and Roller Follower:

The effect of flat and roller follower on valve train friction are calculated separately. Table 1 and Table 2 are used for calculating the frictions.

2.4 Estimation of Friction for Different Intake Manifold Pressure:

Overall frictions were evaluated with the effect of varying intake manifold pressure. Here also presented the percent change in total fmep (with respect to the 100 kPa intake pressure case) versus engine speed for the previous varying intake manifold pressure.

2.5 Estimation of Friction for Different Bore Stroke Ratio at Constant Volume, Stroke and Bore Models:

Overall friction for SI engine and also individual engine components frictions are evaluated for different bore stroke ratios such as 0.8 and 1.2. Three models are considered such as constant bore, constant stroke and constant volume model with different bore stroke ratios [1].

Constant bore model is using variable stroke length as a result variable displacement volume with constant bore diameter. All variable strokes were considered based on standard constant bore, 86 mm. Constant stroke model is using variable bore diameter as a result variable displacement volumes with constant stroke length, 86 mm. Constant volume model also used to evaluate the overall and individual components friction in SI engine. Here bore diameter and stroke length are variable with constant volume based on standard bore and stroke length (square cylinder). Over square (b>s) and under square (b<s) cylinder are considered for different bore stroke ratios but displacement volume of engine are always constant as [10].

3.0 RESULTS AND DISCUSSIONS:

Figure 2(a) shows that the friction for flat follower provides the major portion of the valve train friction. Camshaft bearings and oscillating hydrodynamic friction linearly increasing with engine speed but rate is very less, while the flat follower and oscillating mixed friction decreases with engine speed. A flat follower is assumed to operate in the mixed lubrication regime, and can be scaled with a friction coefficient inversely proportional to engine speed. At low speed, most of the friction is due to the flat follower, and at higher speeds, the majority of the friction is from the flat follower, camshaft bearings and oscillating mixed.

Figure 2(b) shows that the friction for roller follower and oscillating hydrodynamic provides very less portion of the valve train friction at low speed. Camshaft bearings, roller follower and oscillating hydrodynamic friction linearly increases with engine speed but rate is very less, while the oscillating mixed friction decreases with engine speed. A roller follower is assumed to operate in the rolling contact friction regime and is scaled with a friction coefficient proportional to engine speed. At low speed, most of the friction is due to the oscillating mixed, and at higher speeds, the majority of the friction is from the camshaft bearings, oscillating mixed and also roller follower.

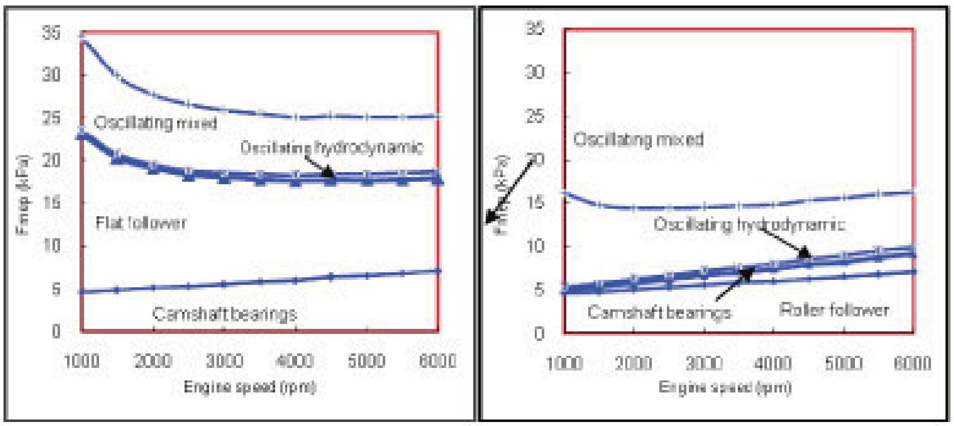


Fig. 2. Components of Valve Train frictions for (a) Flat follower and (b) Roller follower.

Figure 3 shows that the valve train fmep versus rpm at compression ratio 9 and bore stroke ratio 1. The curve accurately reflected the difference of total fmep for valve train between the flat and roller follower configurations.

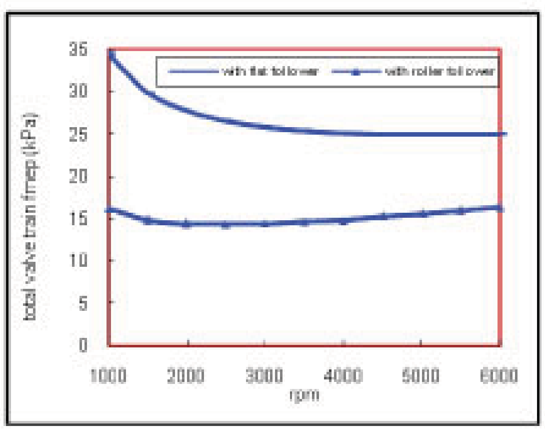
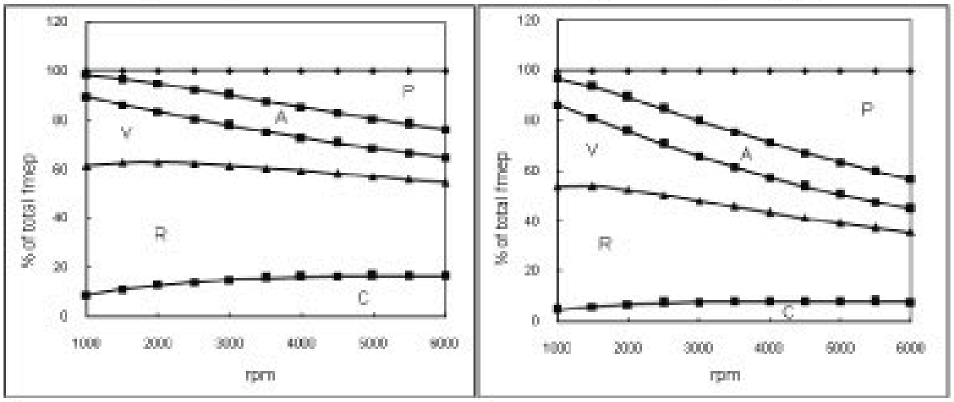


Fig. 3. Comparison of flat follower and roller follower valve train fmep at $r_c = 9.0$ and b/s = 1.0.

Figure 4 shows the largest portion of the total fmep at low and medium speed was contributed by the reciprocating components group, which was 30 – 55% of the total. At higher speeds, the reciprocating group fmep was 25 – 45% of the total. The reciprocating group portion decreased a moderate amount (5-12%) with increasing bore/stroke ratio at low and medium speed but large (12-15%) at high speed. At high speeds, the largest contributor was pumping losses, which were as high as 50% of the total at high bore/stroke ratios. Valve train losses, which were as high as 12 - 35% at low and medium speeds, were less than 15% above 4000 rpm (10% at 6000 rpm). Crankshaft losses were less than 20% throughout the speed range for lower bore/stroke ratios but it were less than 10% for higher bore/ stroke ratios. Crankshaft losses as a percentage of the total losses decreased with increasing of bore/stroke ratio and valve train losses as a percentage of the total losses increased with increasing of bore/stroke ratio for below 4000 rpm (constant above 4000 rpm). Auxiliary component losses were nearly constant at 8 – 15% of the total throughout the speed and

bore/stroke ratio ranges (note that since auxiliary fmep was a function of engine speed only, the auxiliary fmep contribution was the same for all bore/ stroke ratios). Bore/stroke ratio has a relatively small effect on the distribution of WOT losses in a typical engine. In figure, C for crank-shaft, R for reciprocating, V for valve train, A for auxiliary and P for pumping losses are defined.

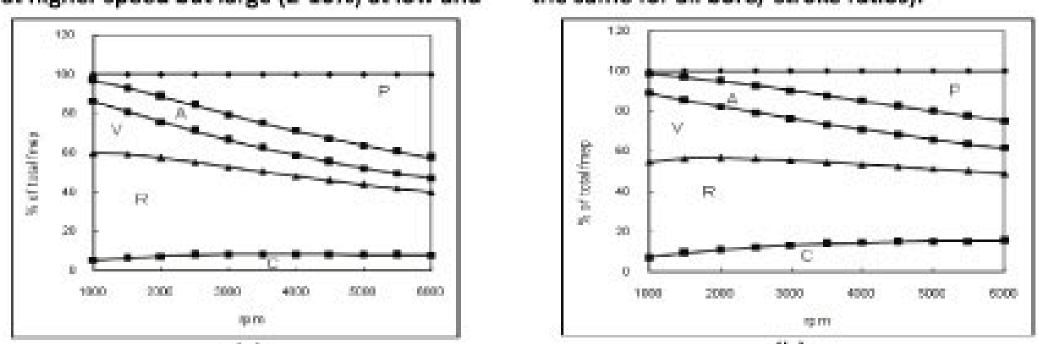


(a) (b)

Fig. 4. Effect of bore stroke ratio on the distribution of friction for constant stroke, r_c = 12 for (a) b/s = 0.8 and (b) b/s = 1.2.

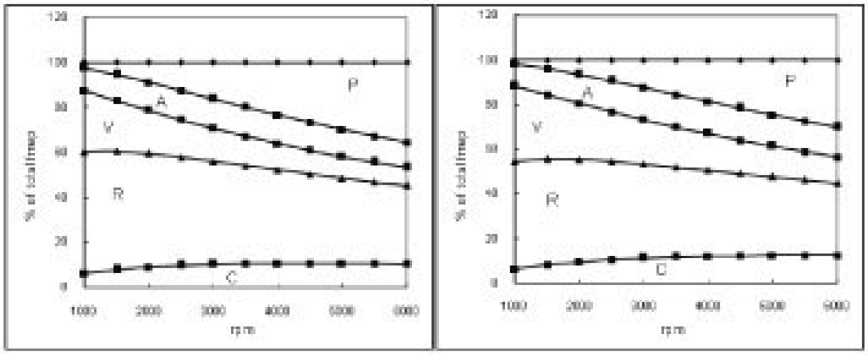
Auxiliary component losses were nearly constant at 8 – 15% of the total throughout the speed and bore/stroke ratio ranges (note that since auxiliary fmep was a function of engine speed only, the auxiliary fmep contribution was the same for all bore/ stroke ratios). Bore/stroke ratio has a relatively small effect on the distribution of WOT losses in a typical engine. In figure, C for crank-shaft, R for reciprocating, V for valve train, A for auxiliary and P for pumping losses are defined.

Figure 5 shows the largest portion of the total fmep at low and medium speed was contributed by the reciprocating components group, which was 40 – 55% of the total. At higher speeds, the reciprocating group fmep was 30 – 40% of the total. The reciprocating group portion decreased a small amount (2%) with increasing bore/stroke ratio at higher speed but large (2-10%) at low and medium speed. At high speeds, the largest contributor was pumping losses, which were as high as 50% of the total at high bore/stroke ratios (0.7). Valve train losses, which were as high as 9 - 36% at low and medium speeds, were less than 20% above 4000 rpm. Crankshaft losses were less than 11% throughout the speed range for lower bore/stroke ratios but it were between 11 to 17% for higher bore/ stroke ratios. Crankshaft losses as a percentage of the total losses increased with increasing bore/stroke ratio and valve train losses as a percentage of the total losses increased with increasing bore/stroke ratio throughout the speed range. Auxiliary component losses were nearly constant at 9 – 15% of the total throughout the speed and bore/stroke ratio ranges (note that since auxiliary fmep was a function of engine speed only, the auxiliary fmep contribution was the same for all bore/ stroke ratios).



(a) (b)

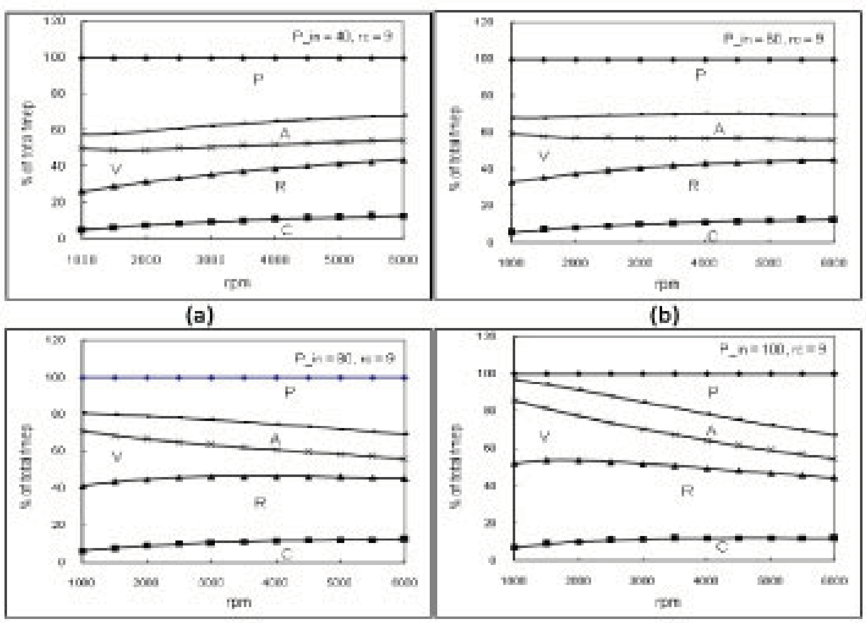
Fig. 5. Effect of bore stroke ratio on the distribution of friction for constant bore, r_c = 12 for (a) b/s = 0.8 and (b) b/s = 1.2.



(a) (b)

Fig. 6. Effect of bore stroke ratio on the distribution of friction for constant Engine displacement volume, r_c = 12 for (a) b/s = 0.8 and (b) b/s = 1.2

Figure 6 shows the largest portion of the total fmep at low and medium speed was contributed by the reciprocating components group, which was 38 – 56% of the total. At higher speeds, the reciprocating group fmep was 32 – 43% of the total. The reciprocating group portion decreased a small amount (4-5%) with increasing bore/stroke ratio at higher speed but large (5-9%) at low and medium speed. At high speeds, the second largest contributor was pumping losses, which were as high as 37% of the total at low bore/stroke ratios (0.7). Valve train losses, which were as high as 10 - 35% at low and medium speeds, were less than 17% above 4000 rpm. Crankshaft losses were 6-13% throughout the speed range and bore/stroke ratios. Crankshaft and valve train losses as a percentage of the total losses increased with increasing bore/stroke ratio throughout the speed range. Auxiliary component losses were nearly constant at 10 – 15% of the total throughout the speed and bore/stroke ratio ranges (note that since auxiliary fmep was a function of engine speed only, the auxiliary fmep contribution was the same for all bore/stroke ratios).



(a) (b) (c) (d)

Fig. 7. Effect of intake manifold pressure on the distribution of friction for an engine; r_c = 9.0, b/s = 1.0, SOHC, flat follower for (a) 40, (b) 60, (c) 80 and (d) 100 kPa.

Figure 7 and 8 show the effects of intake pressure on fmep were evaluated for an engine with a bore/stroke ratio of 1.0, compression ratio of 9.0 and 18, and a SOHC with flat followers. Both Figures show fmep versus engine speed results for four intake pressures ranging from 40 kPa to 100 kPa. Intake pressure had a major effect on the relative importance of the component group friction contributions, primarily because changes in intake pressure resulted in large changes in predicted pumping losses and reciprocating boundary friction.

Increases in intake pressure affected pumping losses by decreasing the intake manifold vacuum (which directly decreased the intake pumping loss) and by increasing the mass flow rate of intake and exhaust flow. Reciprocating fmep increased with higher intake pressures because reciprocating boundary friction increases proportionately to cylinder gas pressure loads.

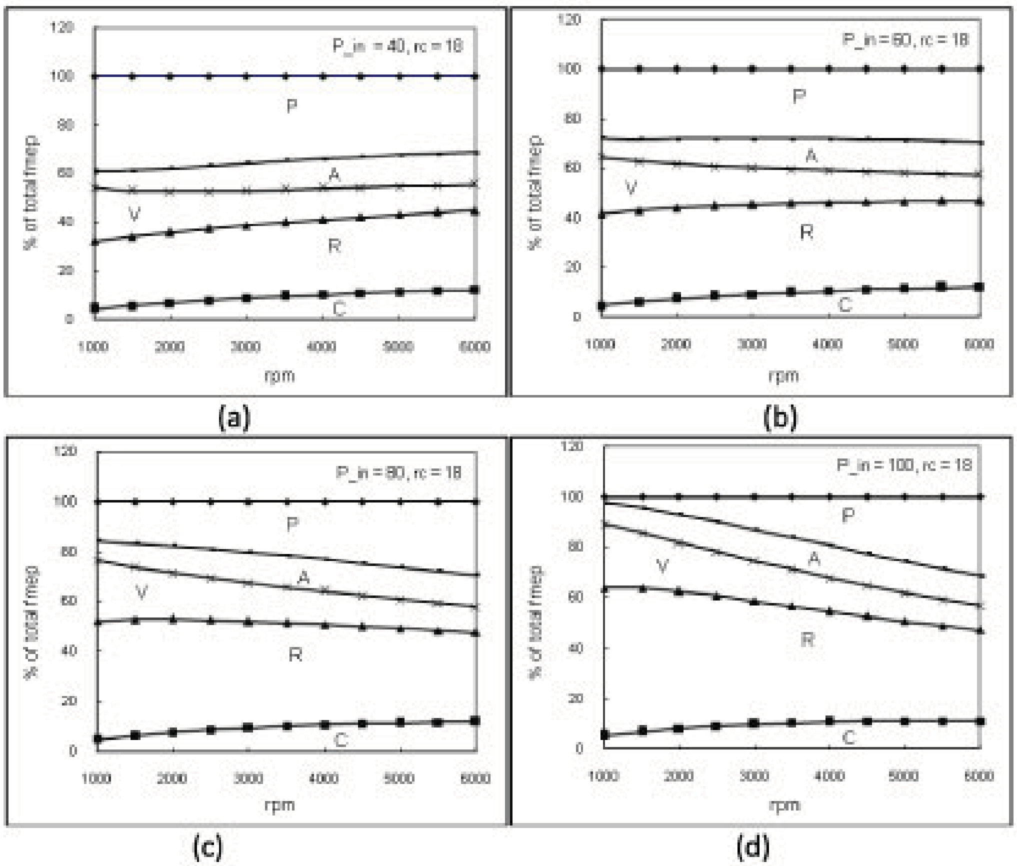
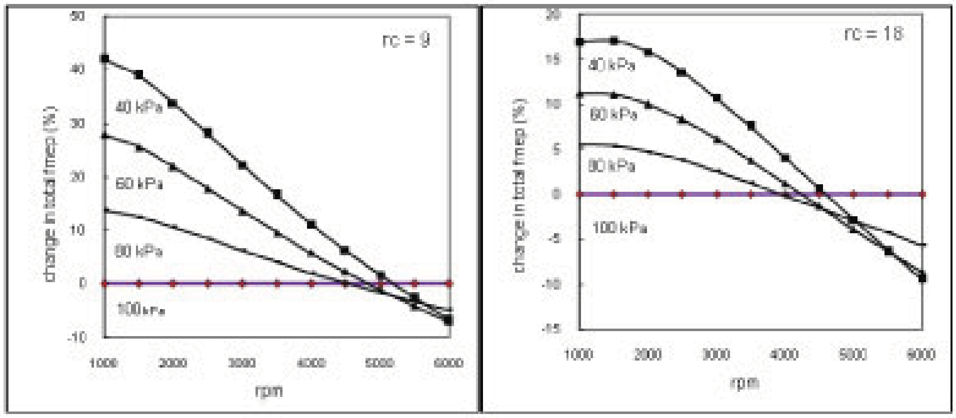


Fig. 8. Effect of intake manifold pressure on the distribution of friction for a engine; r_c = 18.0, b/s = 1.0, SOHC, flat follower for (a) 40, (b) 60, (c) 80 and (d) 100 kPa.

As shown in fig 7 and 8 the low speed contributions of the pumping and reciprocating groups respectively decreased and increased as intake pressure was raised. At higher speeds, mass flow dominated the pumping loss contribution, so the pumping loss percentage increased as intake pressure increased. Because of the reduced importance of reciprocating boundary friction at high speed, there was little change in the reciprocating group contribution above 5000 rpm.

Figure 9 shows the percent change in total fmep (with respect to the 100 kPa intake pressure case) versus engine speed for the intake pressures of figure 7 and 8. At low speed where manifold vacuum dominates the pumping losses, fmep was much higher for low intake pressures. At high speed, where mass flow rates are more important, fmep was higher for high intake pressures.



(a) (b)

Fig. 9. Predicted percent change in total fmep for an engine; b/s = 1.0, SOHC, flat follower, intake manifold pressure varied relative to 100 kPa for (a) r_c = 9.0 and (b) rc = 18.0

4.0 CONCLUSION:

This paper evaluates engine overall friction and specially emphasis given to find out the effect of flat and roller follower. Friction for flat and roller follower also estimated considering different compression ratios. The results highlight the importance of bore stroke ratios and flat/roller followers for different conditions to probe and identify the way to minimize the overall friction produced by engine.

DEFINITIONS, ACRONYMS, ABBREVIATIONS

b	Bore (mm)	*bmep*	Break mean effective pressure
D_b	Crankshaft bearing diameter (mm)	D_{cb}	Connecting rod bearing diameter (mm)
D_{ev}	Exhaust valves diameter (mm)	D_{iv}	Intake valves diameter (mm)
fmep	Friction mean effective pressure	L_b	Crankshaft bearing length (mm)
L_{cb}	Connecting rod bearing length (mm)	L_v	Maximum valve lift (mm)
mep	Mean effective pressure	N	Speed (rpm)
n_b	Number of crankshaft bearings	n_c	Number of cylinders
n_{cs}	Number of camshafts	n_{ev}	Exhaust valves per cylinder
n_{iv}	Intake valves per cylinder	*pmep*	Pumping mean effective pressure
P_a	Atmospheric pressure (kPa)	P_e	Exhaust pressure (kPa)
P_i	Intake pressure (kPa)	r_c	Compression ratio
s	Stroke (mm)	U_p	mean piston speed

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