EFFECT OF BORE STROKE RATIOS ON OVERALL FRICTION MODEL OF SI ENGINE

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ABSTRACT

Bore Stroke ratios are considered one of the most significant impact on friction of SI engine. This paper outlines a method or model which provides the engine designer with an estimation of overall friction mean effective pressure due to change of bore stroke ratios in spark-ignition internal combustion engines. The general effect of overall friction on engine performance was examined during normal working conditions. A parametric study was performed covering wide range of dependent variables such as engine speed, change of bore stroke ratios with and without changing engine volume. Different bore stroke ratios are used for three different cylinder condition / model, such as, at constant volume, at variable bore and volume (constant stroke), and lastly at variable stroke and volume (constant bore). Various average friction coefficients are used considering oil viscosity property, film thickness, temperature effect and roughness of the surface. At low speed with lower bore stroke ratio and at high speed with higher bore stroke ratio, constant bore model gives less friction irrespective of compression ratio. 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.

KEY WORDS: friction, engine speed, bore stroke ratio, constant volume model, constant stroke model, constant bore model.

NOMENCLATURE

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

1.0 INTRODUCTION

Since 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 [1].

High speed engines would suffer more from extremely large bore sizes than would low speed engines. If a lesser number of larger cylinders are used, the friction and economy are improved and vice versa. So it seems to be justified that 4 and 6 cylinder engine are more efficient than that of 8, 10 or 12 cylinders. The effect of reduced contact area of the piston skirt is quite pronounced. As an example, at 4800 rpm, a 25% reduction in contact area or effective length of the piston skirt would result in a 5% change in friction and a 2.5% change in power and economy. Although the friction increases with compression ratio, the mechanical efficiency either stays constant or improves at higher ratios. Increased displacement (engine size) has an adverse effect on economy and vice versa. Large engines geared to run slowly can equal or better the economy of smaller engines with the same performance. The friction and economy effects of bearing size are relatively small [2].

At low speed, friction from the pistons, piston rings, and connecting rod bearings was predicted to account for 40 - 60% of total engine friction. At high speeds, pumping losses were predicted to be the most significant loss, accounting for up to 50% of total losses. The predicted effect of bore/stroke ratio (at constant displacement) on the distribution of friction losses between component groups is relatively minor; however, above 1000 rpm, total predicted fmep does decrease with increasing bore/stroke ratio [3].

At the same bmep, both friction and pumping mep are higher at a higher compression ratio. Friction is higher because peak cylinder pressures are higher. Pumping is higher at fixed bmep because the engine is throttles more because the efficiency is higher [4].

When the 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 the ovality or cutting back the sides might have a minor effect that might not be measurable.

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 [5].

The largest single contributor 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 [6].

The piston assemblies of most engines contribute about half of the total friction and can contribute as much as 75% at light load. The piston rings alone contribute about 20% of total friction. Most pistons have two compression rings and one or two oil rings. The second compression ring reduces the pressure differential that occurs across the first compression ring during combustion and power stroke.

Adding an additional compression ring can add about 10 kPa to fmep of an engine. Increasing the compression ratio also requires heavier bearings on the crank shaft and connecting rods and may require an additional piston compression ring. The valve train of an engine contributes about 25% of total friction, crank shaft bearings contribute about 10% of total, and engine driven accessories contribute about 15% of total [7].

For wide open throttle conditions at low engine speeds, the overall engine fmep is primarily due to piston and valve train friction. As the engine speed increases, the pmep fraction increases to about 35%, and the valve train fmep fraction decreases from 35% to 10%. Friction forces occurring during expansion are about twice as large as those occurring during any other stroke.

Friction forces tend to be high just after top and bottom dead center due to metallic contact between the rings and the cylinder wall [1].

While examining the friction results 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 [8].

It was found that the inclusion of the piston friction in efficiency evaluation can reduce the calculated values especially in high speed range and at low oil temperatures. Although the contribution of ring friction becomes more significant at low oil temperatures and high engine speeds, skirt friction remains the dominant factor in piston friction [9].

2.0 MODELING, SIMULATION AND ANALYSIS

2.1 Governing Equations of the SI Engine Friction

Ideal cycle consider only combustion at constant volume process without any heat losses and friction. When actual cycle need to simulated heat losses and friction must be considered. Although the complexities of simulating the actual cycle are many, yet most of the deviations from ideal cycle can be modeled using classical thermodynamics. Before starting to formulate the equations for the friction modeling of SI engine, the close relative terminology needs to be described first.

2.1.1 Crankshaft Friction Mean Effective Pressure

The journal bearings used in an internal combustion engine include the main crankshaft bearings, connecting rod bearings, and accessory bearings and seals.

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

Proportionality cb = 3.03 x 10^{-4} (kPa-min/rev-mm) for spark ignition engines [3].

The crankshaft bearing seals operate in a boundary lubrication regime, since the seals directly contact the crankshaft surface.

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

Proportionality constant cs = 1.22 x10^5 kPa-mm2 [3].

2.1.2 Piston Friction Mean Effective Pressure

Friction correlations for piston and ring friction have been developed which take both the boundary lubrication and the hydrodynamic friction regimes into account. The hydrodynamic friction component depends on the contact area. The skirt length scaling is based on geometrical similarity, and the clearance scaling is based on thermal expansion considerations. Therefore, the piston skirt hydrodynamic fmep scales as [1]:

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

Proportionality constant c_{ps} = 294 kPa-mm-s/m for the piston hydrodynamic friction, including the oil properties in the proportionality constant [3].

The friction force of the piston rings has two components, one resulting from the ring tension and the other component from the gas pressure loading. The component of piston friction due to ring tension in the mixed lubrication regime will have a friction coefficient inversely proportional to the engine speed. The piston ring fmep scaling is-

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

The proportionality constant recommended is cpr = 4.06 x 10 +4 kPa-mm2 [3].

A correction for the component of piston friction due to the gas pressure loading recommended is-

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

cg = 6.89, and K = 2.38 x 10-2 s/m. The correlation includes the effect of compression ratio, and a decrease in the friction coefficient in the mixed lubrication regime [2].

2.1.3 Valve Train Friction Mean Effective Pressure

The valve train frictional losses are due to the following; 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 [1].

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

Where ncs is the number of camshaft bearings, assumed equal to the product of the number of camshafts and the number of main bearings. cc = 2.44 X 102 kPa-mm3-min/rev as the proportionality constant [3].

A flat follower (ff) is assumed to operate in the mixed lubrication regime, and can be scaled with a friction coefficient inversely proportional to engine speed.

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

A roller follower (rf) operates in the rolling contact friction regime and is scaled with a friction coefficient proportional to engine speed.

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

The oscillating hydrodynamic (oh) friction in the lifters and valve guides scales with square root of the Stribeck variable, in which the relative velocity is proportional to the maximum valve lift Lv and the engine speed N.

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

The oscillating mixed (om) lubrication in the valve stem and tip can be scaled with a friction coefficient proportional to valve lift, and inversely proportional to engine speed.

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

The coefficients for the valve train friction terms are given in [1]. c_{ff} is the flat follower coefficient, c_{rf} is the roller follower coefficient, c_{oh} is the oscillating hydrodynamic coefficient and com is the oscillating mixed coefficient.

A simpler overall expression for valve train friction [2]:

$$fmep_{valvetrain} = 393\left(30-\frac{4N}{1000}\right)\frac{n_{iv} D_{iv}^{\frac{7}{4}}}{b^2 s} \quad \text{.......... (11)}$$

Decrease of the friction coefficient in the mixed lubrication regime with engine speed, and is inversely proportional to the piston stroke [2].

2.1.4 Pumping Friction Mean Effective Pressure

The pumping mean effective pressure is the sum of the pressure drops across flow restrictions during the intake and exhaust strokes. It is a measure of the work required to move the fuel-air mixture into and out of an engine. The pressure drop in the intake manifold is $\triangle P_{im}$. So

$$\triangle P_{im} = P_a - P_i \quad \text{.......... (12)}$$

Neglecting the density change across the valves, the inlet valve pressure drop therefore scales as the square of the mean piston speed

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

The exhaust valve pressure drop $\triangle P_{ev}$ scaling is

$$\Delta P_{ev} = c_v\left(\frac{P_i}{P_a}\cdot\frac{U_p b^2}{n_{ev} D_{ev}^2}\right)^2 \quad \text{.......... (14)}$$

The exhaust system pressure drop $\triangle P_{es}$ also scale with the square of the mass flow rate

$$\Delta P_{es} = K_{es}\left(\frac{P_i}{P_a}.U_p\right)^2 \quad \text{.......... (15)}$$

2.1.5 Accessory Friction Mean Effective Pressure

It includes the oil pump, water pump, and noncharging alternator friction. Suggested accessory mean effective pressure of the form [3].

$$fmep_{accessory} = a_1 + a_2\left(\frac{N}{1000}\right) + a_3\left(\frac{N}{1000}\right)^2 \quad \text{.......... (16)}$$

Where a_1 = 6.23 kPa, a_2 = 5.22 kPa-min/rev and a_3 = - 0.179 kPa-min2/rev2.

2.2 Simulation of SI Engine Friction Model

In the present study, the stuffy of friction analysis in spark ignition engine by developing the crank shaft friction model, piston friction model, valve train friction model, pumping friction model and accessories friction model. 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 such 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 1. Proportionality constants used, are included with oil properties [1] [2] [3].

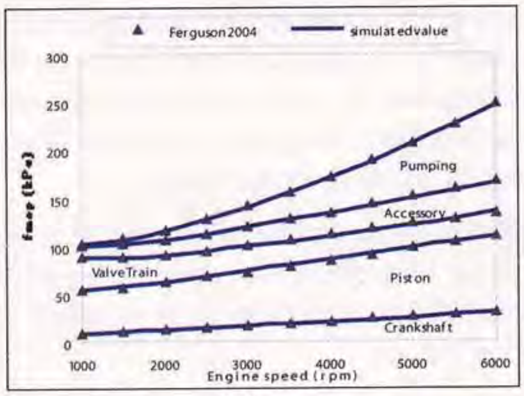
In the present study, the results obtained from the simulation of a 4 cylinder four stroke spark ignition engine are reported. Firstly, analysis of most common different components of friction occurs in SI engines have been presented. Results also verified to the established results from different sources. Secondly, the variation of engine speed, compression ratio, bore stroke ratio and different pressure has been analyzed .The preceding component analyses can be combined to form an overall engine fmep model. It should be noted that the coefficients used in the fmep model are likely to depend on oil properties, such as the viscosity, that have not been explicitly included in the analysis.

In the present study, a 4 cylinders SI engine is modeled. Specification of the engine is reported in table 1. The values of overall engine friction and its components confirm to the Figure. 1[1].

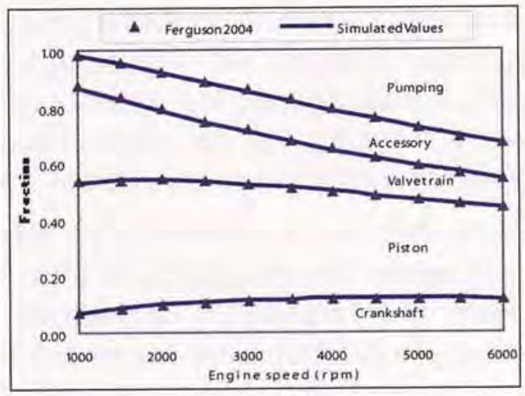
Table 1. Specifications of the Engine used for simulation [1]

Component	Dimension	Component	Dimension
Bore (mm)	86	Exhaust Valve Diameter (mm)	31
Stroke (mm)	86	Maximum Valve Lift (mm)	11
Number of Cylinders	4	Number of Crankshaft Bearings	5
Compression Ratio	9	Crankshaft Bearing Dia. (mm)	56
Atmospheric Pr essure (kPa)	101	Crankshaft Bearing Length (mm)	21
Intake Pressure (kPa)	101	Number of Connecting Rod Bearings	4
Exhaust Pressure (kPa)	103	Connecting Rod Bearing Dia. (mm)	48
Intake Valves/Cylinder	2	Connecting Rod Bearing Length (mm)	42
Exhaust Valves/Cylinder	2	Number of Camshafts	1
Intake Valve Diameter (mm)	35		

Figure. 1(a) shows, the fmep components as a function of engine speed. The friction from the piston rings and skirt is the largest component. As the engine speed increase from 1000 to 6000 rpm, the total fmep increases nonlinearly from about 100 kPa to 260 kPa, with the pumping friction contributing the largest increase. The results match the similar diagram given in [1].



(a)



(b)

Figure 1: Friction mean effective pressure (a) and Component fractions of fmep versus engine speed (b).

2.3 Estimation of Friction for Different Bore Stroke Ratio

Overall friction for SI engine and also individual engine components frictions are evaluated for different bore stroke ratios such as 0.7, 1.0 and 1.3.

Two models are considered such as constant bore and constant stroke model for variable volume with different bore stroke ratios. Constant bore model using variable stroke length as a result variable displacement volume with constant bore diameter. All variable strokes considered based on standard constant bore, 86 mm. Constant stroke model using variable bore diameter as a result variable displacement volumes with constant stroke length. Standard constant stroke length considered as 86 mm.

2.4 Estimation of Friction for Different Bore Stroke Ratio at Constant Volume

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 and under square cylinder are considered for different bore stroke ratios but displacement volume of engine always constant [10].

3.0 RESULTS AND DISCUSSIONS

Single overhead cam (SOHC), flat follower valve train configureurations are considered. Figure. 2 show, the effects of bore/stroke ratio on the distribution of component fmep and the magnitude of total fmep. Figure. 2 is a series of graphs showing the contributions of individual component groups to wide open throttle total fmep versus engine speed for seven bore/stroke ratios. The series of graphs shown is for engine with compression ratios of 8.0. Here stroke length kept constant and bore diameter changed to get varied bore/stroke ratios. Here upto 4000 rpm considered as low and medium speed and 4000 to 6000 rpm considered as higher speed.

The largest portion of the total fmep at low and medium speed was contributed by the reciprocating components group, which was 30 - 50% of the total. At higher speeds, the reciprocating group fmep was 25 - 40% of the total. The reciprocating group portion decreased a moderate amount (7-12%) 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 15 - 40% 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 bore/stroke ratio and valve train losses as a percentage of the total losses increased with increasing bore/stroke ratio for below 4000 rpm (constant above 4000 rpm). Auxiliary component losses were nearly constant at 9 - 13% 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. However, the magnitude of total friction losses was more sensitive to bore/stroke ratio. Here C = crankshaft component, R=reciprocating component, V = valve train component, A = auxiliary components and P = pumping losses.

Also shows the bore stroke ratio versus volume ratio. Here volume ratios are normalized with the volume at B/S = 1.0.

Figure. 3 show, the effects of bore/stroke ratio on the specification and configureuration described in Figure. 2 only exception is compression ratio 18.0.

The largest portion of the total fmep at low and medium speed was contributed by the reciprocating components group, which was 35 - 60% of the total. At higher speeds, the reciprocating group fmep was 25 - 50% of the total. The reciprocating group portion decreased a small amount (2-10%) with increasing bore/stroke ratio at low and medium speed but large (10-14%) at high speed. At high speeds, the largest contributor was pumping losses, which were as high as 45% of the total at high bore/stroke ratios. Valve train losses, which were as high as 12 - 30% at low and medium speeds, were less than 15% above 4000 rpm (8-10% 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 bore/stroke ratio and valve train losses as a percentage of the total losses increased with increasing bore/stroke ratio for below 4000 rpm (constant above 4000 rpm). Auxiliary component losses were nearly constant at 7 - 13% of the total throughout the speed and bore/stroke ratio ranges.

Bore/stroke ratio has a relatively small effect on the distribution of WOT losses in a typical engine.

Figure. 4(a) shows, the results based on the Figure. 2 (compression ratio 8.0), bore/stroke ratio has a relatively small effect on the distribution of wide open throttle losses in a typical engine. However, the magnitude of total friction losses was more sensitive to bore/stroke ratio. Figure. 4(a) shows the percent change in total fmep (with respect to the B/S = 1.0 engine) versus engine speed for the engines of Figure. 2. At speed lower than 1500 rpm, there was highest difference between the total fmep of the engines for different bore stroke ratios. In this speed range, total fmep greatly varied with bore diameter or engine displacement volume where stroke length kept constant. If bore diameter reduced (kept stroke length constant) then pumping and valve train losses also reduced but losses for reciprocating and crank shaft will increases.

To meet the bearing loads, crankshaft main bearings and connecting rod bearings length also varied with cylinder bore. Above 2000 rpm, the total fmep decreased when bore/stroke ratio less than 1.0 and total fmep increases for bore/stroke ratio greater than 1.0. It was shown in Figure. 2 that the change in the distribution of losses was relatively small over the range of bore/stroke relations, so it is likely that the bore/stroke ratio effect on total fmep is due to effects on all of the component groups.

Figure. 4(b) shows, the results based on the Figure. 3 (compression ratio 18.0), bore/stroke ratio has a relatively smaller effect than Figure. 4(a). Above 3000 rpm, the total fmep decreased when bore/stroke ratio less than 1.0 and total fmep increases for bore/stroke ratio greater than 1.0. However, the magnitude of total friction losses was less sensitive to bore/stroke ratio.

Figure. 5 show, the effects of bore/stroke ratio on the distribution of component fmep and the magnitude of total fmep. Others specifications are as Figure. 2. Here bore diameter kept constant and stroke length changed to get varied bore/stroke ratios.

The largest portion of the total fmep at low and medium speed was contributed by the reciprocating components group, which was 35 - 50% of the total. At higher speeds, the reciprocating group fmep was 30 - 40% of the total. The reciprocating group portion decreased a small amount (1-2%) with increasing bore/stroke ratio at higher speed but large (2-11%) at low and medium speed. At high speeds, the largest contributor was pumping losses, which were as high as 50% of the total at low bore/stroke ratios (0.7). Valve train losses, which were as high as 10 - 40% at low and medium speeds, were less than 20% above 4000 rpm. Crankshaft losses were less than 12% throughout the speed range for lower bore/stroke ratios but it were between 8 to 18% 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 - 16% of the total throughout the speed and bore/stroke ratio ranges.

Figure. 6 show, the effects of bore/stroke ratio on the specification and configureuration described in Figure. 5 only exception is compression ratio 18.0.

The largest portion of the total fmep at low and medium speed was contributed by the reciprocating components group, which was 45 - 65% of the total. At higher speeds, the reciprocating group fmep was 30 - 45% of the total. The reciprocating group portion decreased a small amount (3-5%) with increasing bore/stroke ratio at higher speed but moderate amount (3-8%) 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 8 - 30% at low and medium speeds, were less than 17% above 4000 rpm. Crankshaft losses were less than 10% throughout the speed range for lower bore/stroke ratios but it were between 11 to 16% 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 8 - 14% of the total throughout the speed and bore/stroke ratio ranges.

Figure. 7(a) shows, the results based on the Figure. 4(b) (compression ratio 8.0), bore/stroke ratio has a relatively small effect on the distribution of wide open throttle losses in a typical engine. However, the magnitude of total friction losses was more sensitive to bore/stroke ratio. Figure. 7(a) shows the percent change in total fmep (with respect to the B/S = 1.0 engine) versus engine speed for the engines of Figure. 5. At speed lower than 2000 rpm, there was little difference between the total fmep of the engines. This is probably due to the fact that most of the friction at those speeds is boundary friction, which is less sensitive to engine dimensions. Above 2500 rpm, the total fmep decreased as bore/stroke ratio increased. It was shown in Figure. 5 that the change in the distribution of losses was relatively small over the range of bore/stroke relations, so it is likely that the bore/stroke ratio effect on total fmep is due to effects on all of the component groups

Figure. 7(b) shows, the results based on the Figure. 6 (compression ratio 18.0), bore/stroke ratio has a relatively lesser effect than Figure. 7(a). However, the magnitude of total friction losses was less sensitive to bore/stroke ratio. At speed lower than 2800 rpm, there was little difference between the total fmep of the engines. Above 3200 rpm, the total fmep decreased as bore/stroke ratio increased.

Figure. 8 shows the effects of bore/stroke ratio on the distribution of component fmep and the magnitude of total fmep. Others specifications are as Figure. 2. Here Engine volume kept constant, bore and stroke length changed to get varied bore/stroke ratios.

The largest portion of the total fmep at low and medium speed was contributed by the reciprocating components group, which was 34 - 50% of the total. At higher speeds, the reciprocating group fmep was 30 - 41% of the total. The reciprocating group portion decreased a small amount (5-7%) with increasing bore/stroke ratio at higher speed but large (7-11%) at low and medium speed. At high speeds, the second largest contributor was pumping losses, which were as high as 38% of the total at low bore/stroke ratios (0.7). Valve train losses, which were as high as 10 - 40% at low and medium speeds, were less than 18% above 4000 rpm. Crankshaft losses were 7-13% 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 - 16% of the total throughout the speed and bore/stroke ratio ranges.

Also shows the bore stroke ratio versus volume ratio. Here volume ratios are constant with variable bore and stroke.

Figure. 9 show the effects of bore/stroke ratio on the specification and conFigureuration described in Figure. 8 only exception is compression ratio 18.0.

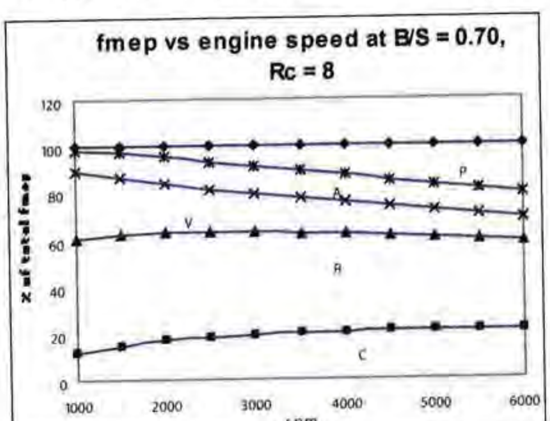
The largest portion of the total fmep at low and medium speed was contributed by the reciprocating components group, which was 43 - 63% of the total. At higher speeds, the reciprocating group fmep was 35 - 46% of the total. The reciprocating group portion decreased a small amount (2-3%) with increasing bore/stroke ratio at higher speed but large (3-7%) 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 - 30% at low and medium speeds, were less than 16% above 4000 rpm. Crankshaft losses were 5-12% 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 8 - 13% of the total throughout the speed and bore/stroke ratio ranges.

Figure. 10(a) shows the results based on the Figure. 8 (compression ratio 8.0), bore/stroke ratio has a relatively small effect on the distribution of wide open throttle losses in a typical engine. However, the magnitude of total friction losses was more sensitive to bore/stroke ratio. Figure. 10(a) shows the percent change in total fmep (with respect to the B/S = 1.0 engine) versus engine speed for the engines of Figure. 8. At speed lower than 1200 rpm, there was little difference between the total fmep of the engines. This is probably due to the fact that most of the friction at those speeds is boundary friction, which is less sensitive to engine dimensions. Above 1400 rpm, the total fmep decreased as bore/stroke ratio increased. It was shown in Figure. 8 that the change in the distribution of losses was relatively small over the range of bore/stroke relations, so it is likely that the bore/stroke ratio effect on total fmep is due to effects on all of the component groups.

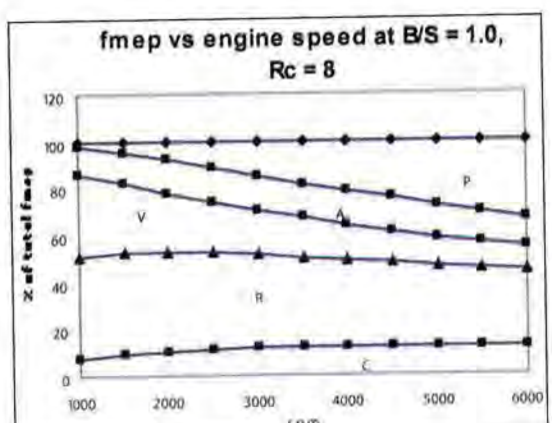
Figure. 10(b) shows the results based on the Figure. 9 (compression ratio 18.0), bore/stroke ratio has a relatively lesser effect than Figure. 10(a). However, the magnitude of total friction losses was less sensitive to bore/stroke ratio. At speed lower than 1600 rpm, there was little difference between the total fmep of the engines. Above 1800 rpm, the total fmep decreased as bore/stroke ratio increased.

From the analysis of overall fmep and its individual components considered in the present study, some of the key results obtained from all Figure. and its discussions are given below:

1. At lower B/S ratio and low speed, maximum fmep provided by at constant stroke and lowest fmep provided by engine at constant bore irrespective of compression ratio.
2. At lower B/S ratio and high speed, maximum fmep provided by at constant bore and lowest fmep provided by engine at constant stroke irrespective of compression ratio.
3. At higher B/S ratio and low speed, maximum fmep provided by at constant bore and lowest fmep provided by engine at constant stroke irrespective of compression ratio.
4. At higher B/S ratio and high speed, maximum fmep provided by at constant stroke and lowest fmep provided by engine at constant bore irrespective of compression ratio.
5. Piston fmep for constant volume, constant bore and constant stroke at lower compression ratio, much less than the fmep for higher compression ratio irrespective of B/S ratios.
6. Valve train fmep for constant volume, constant bore and constant stroke at lower compression ratio, small greater than the fmep for higher compression ratio irrespective of B/S ratios.
7. Pumping fmep for constant volume, constant bore and constant stroke at lower compression ratio, small greater than the fmep for higher compression ratio irrespective of B/S ratios.
8. Total fmep at lower B/S ratios are much higher (positive) difference from square cylinder at higher speed for constant volume and bore irrespective of compression ratio.
9. Total fmep at higher B/S ratios are small lower (negative) difference from square cylinder at higher speed for constant volume and bore irrespective of compression ratio.
10. Total fmep at lower B/S ratios are much higher (positive) difference from square cylinder at low and medium speed for constant stroke irrespective of compression ratio.
11. Total fmep at higher B/S ratios are small lower (negative) difference from square cylinder at low and medium speed for constant stroke irrespective of compression ratio.
12. 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.



(a)



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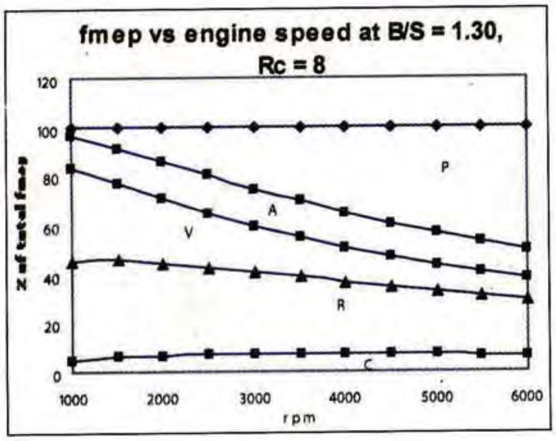
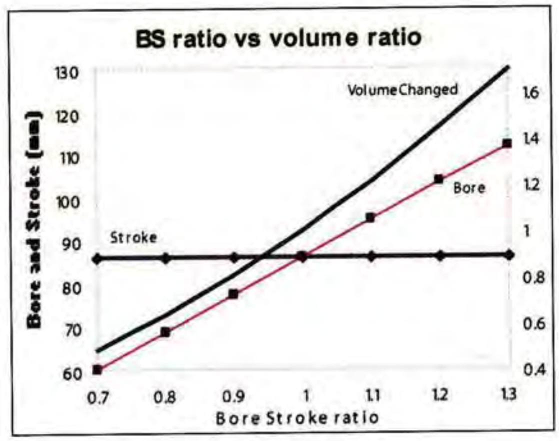
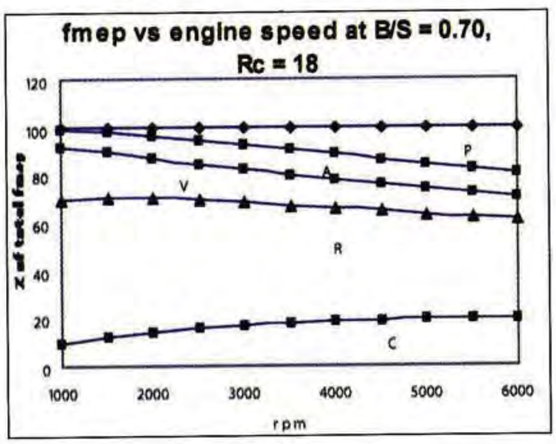


(c)



(d)

Figure 2: Effect of bore/stroke ratio on the distribution of friction for constant stroke



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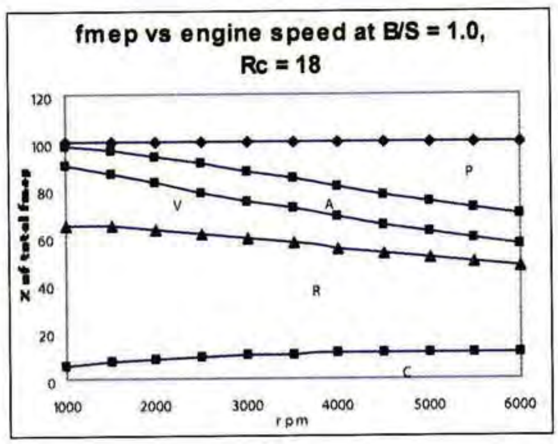

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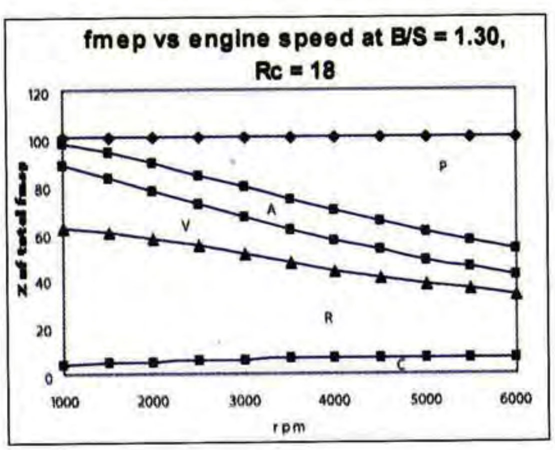
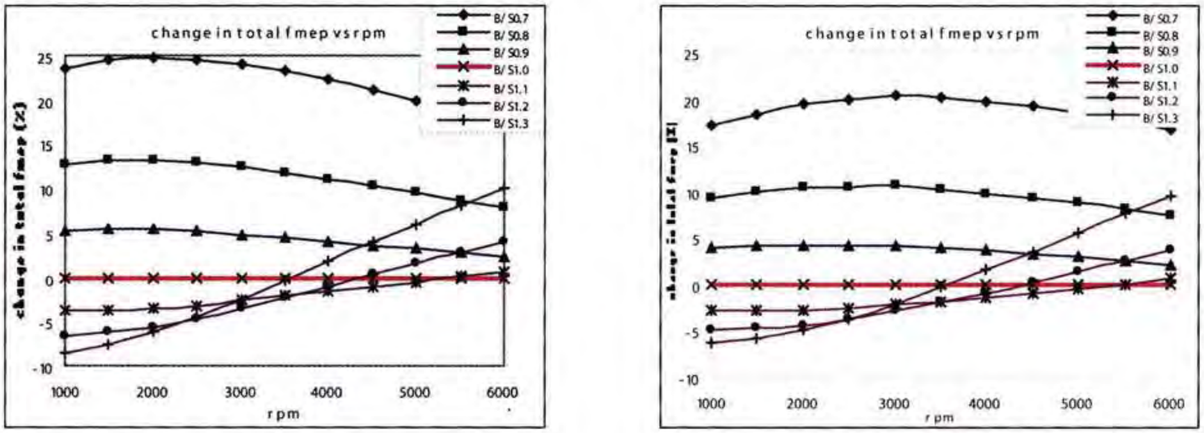


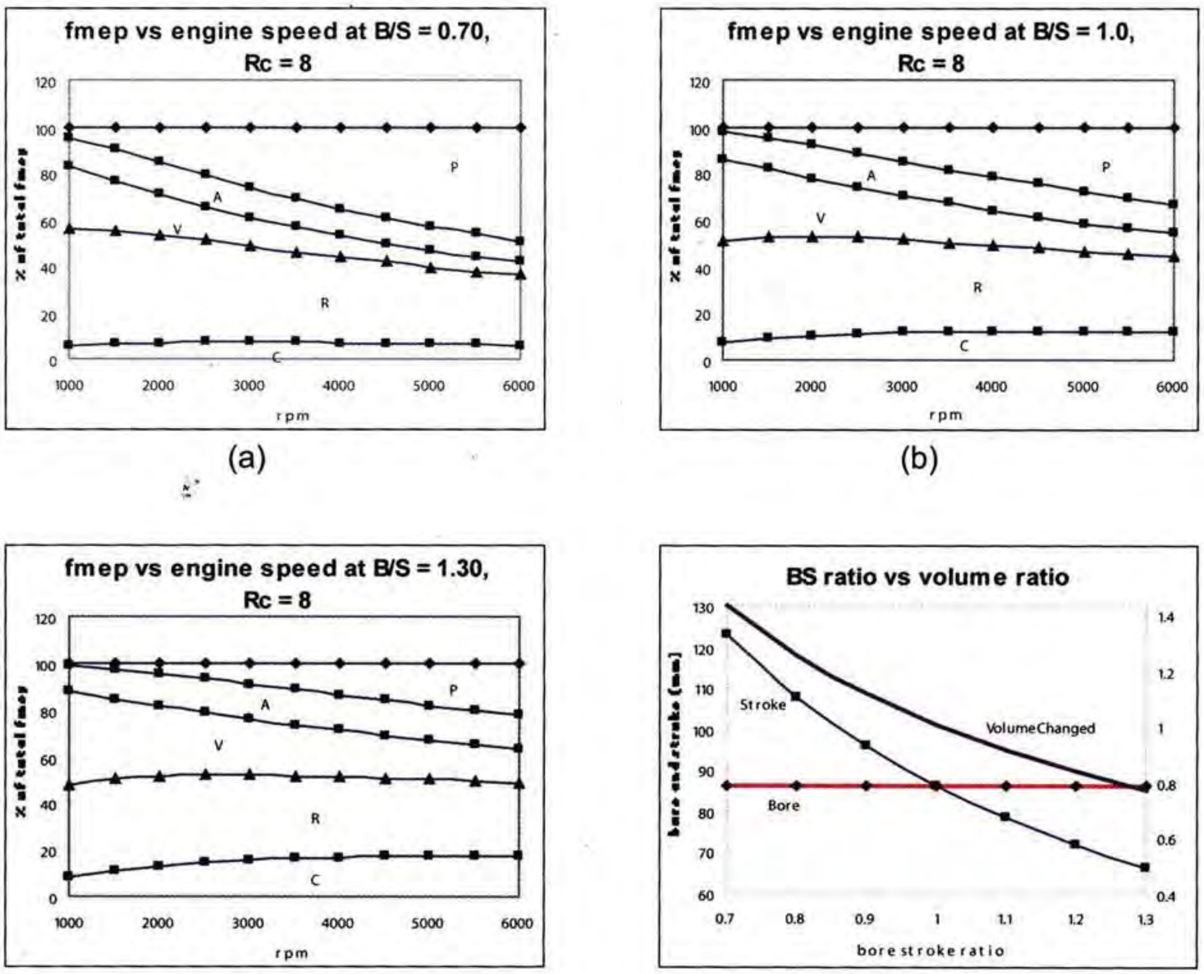
(c)

Figure 3: Effect of bore/stroke ratio on the distribution of friction for constant stroke



(a) (b)

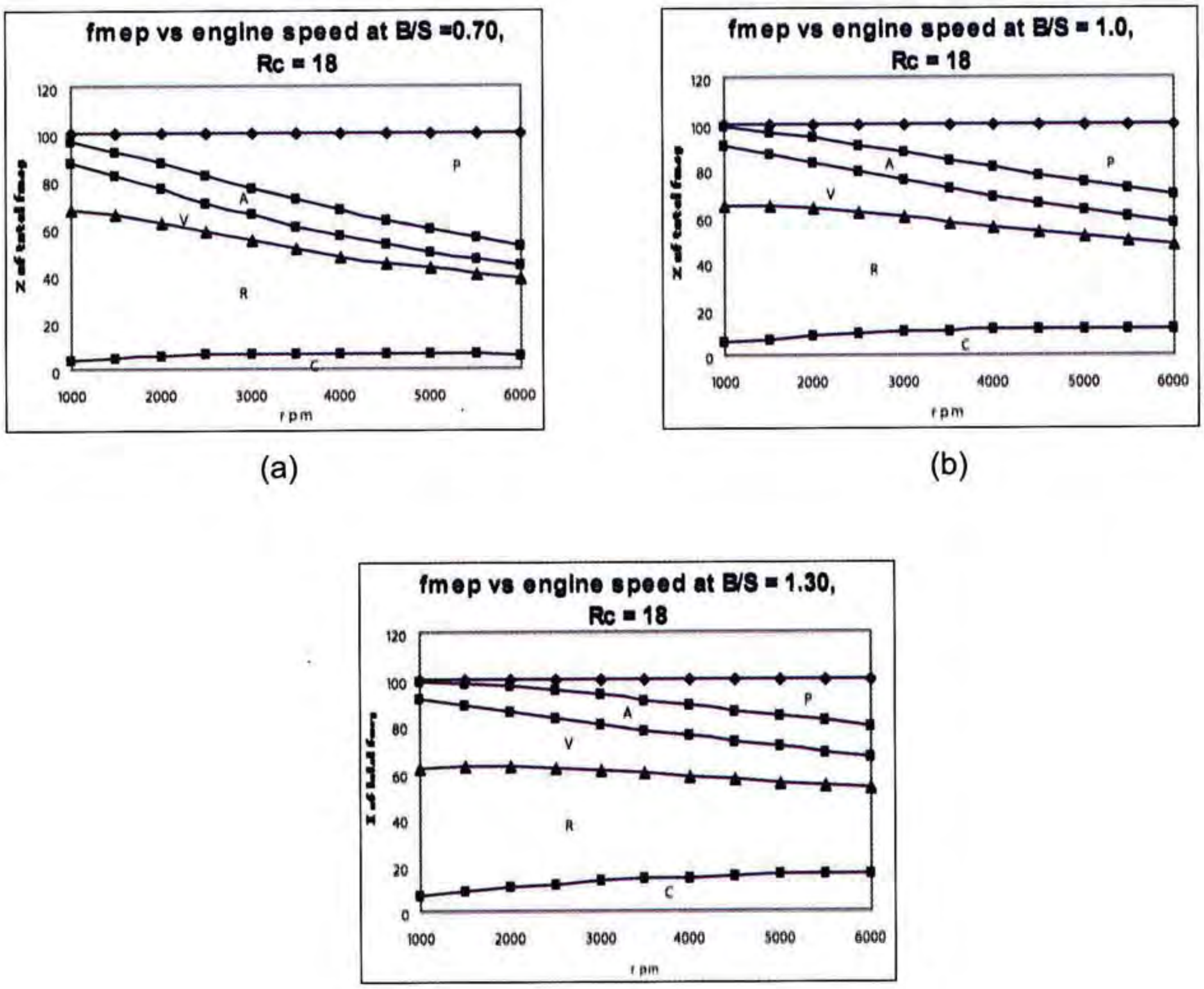
Figure 4: Predicted percent change in total fmep (constant stroke, 8 and 18 compression ratio, 101 kPa manifold pressure) as B/S varied relative to B/S = 1.00.



(a) (b)

(c) (d)

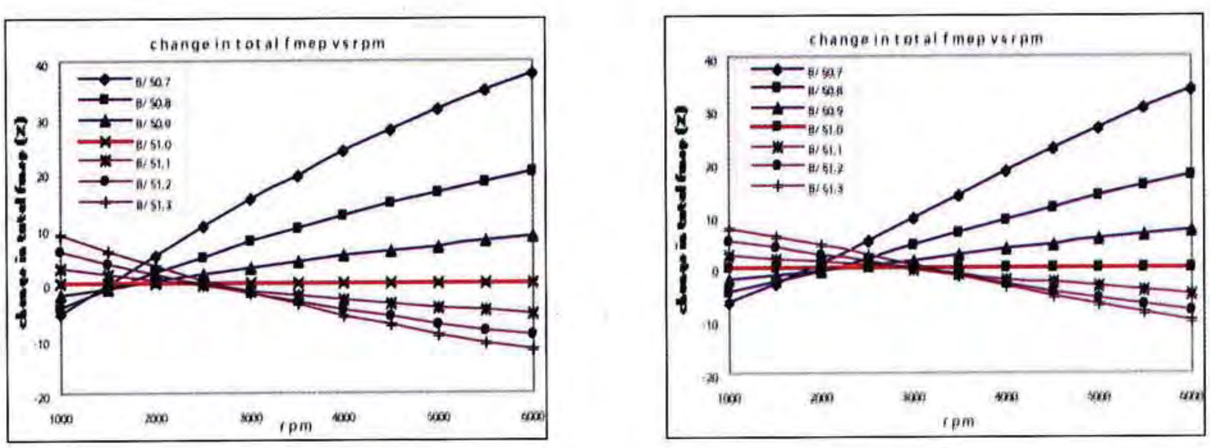
Figure 5: Effect of bore/stroke ratio on the distribution of friction for constant bore



(a) (b)

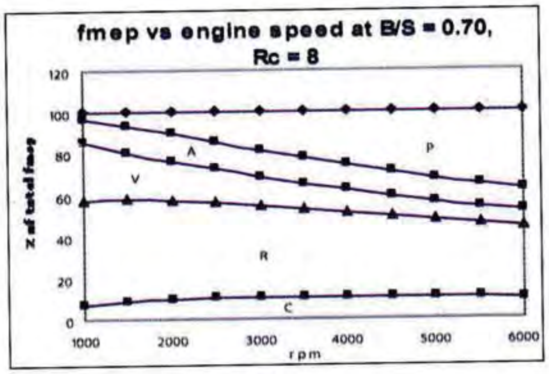
(c)

Figure 6: Effect of bore/stroke ratio on the distribution of friction for constant bore

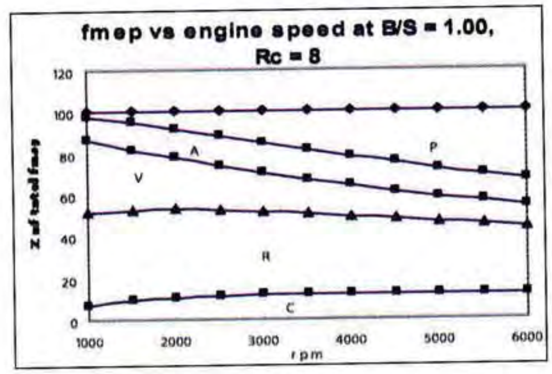


(a) (b)

Figure 7: Predicted percent change in total fmep (constant bore diameter, 8 and 18 compression ratio, 101 kPa manifold pressure) as B/S varied relative to B/S = 1.00.



(a)



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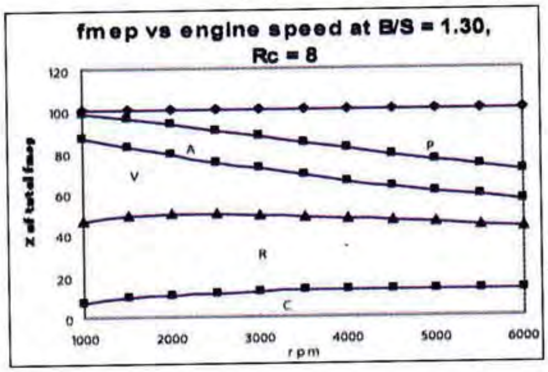
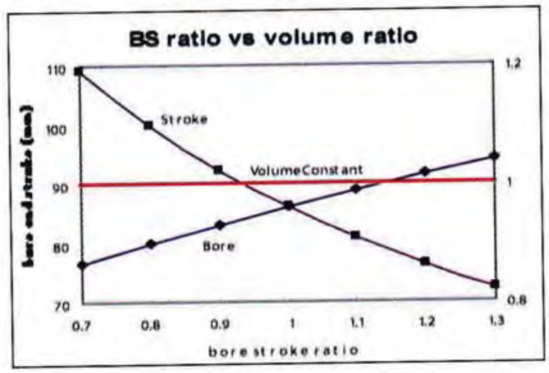
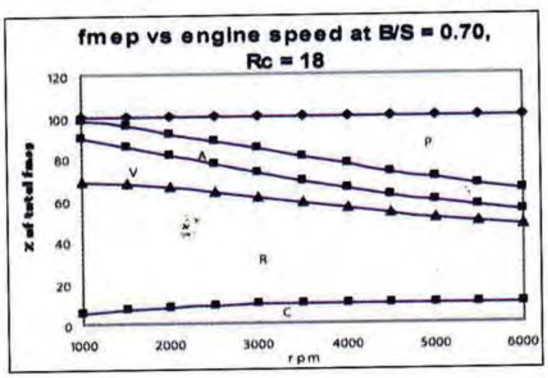


(c)

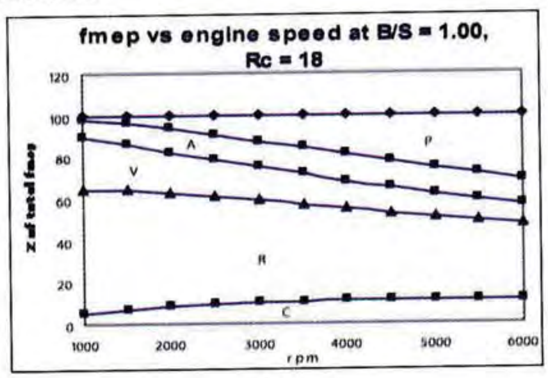


(d)

Figure 8: Effect of bore/stroke ratio on the distribution of friction for constant Engine displacement volume.



(a)



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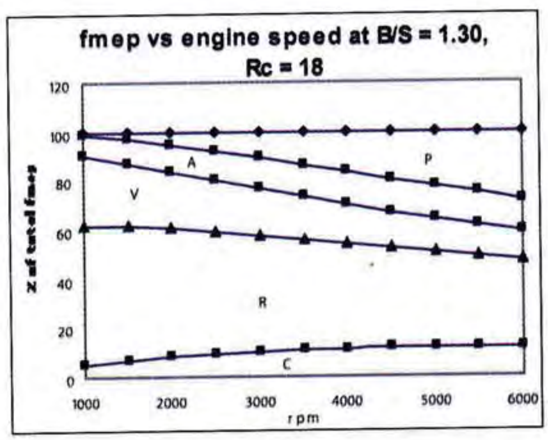
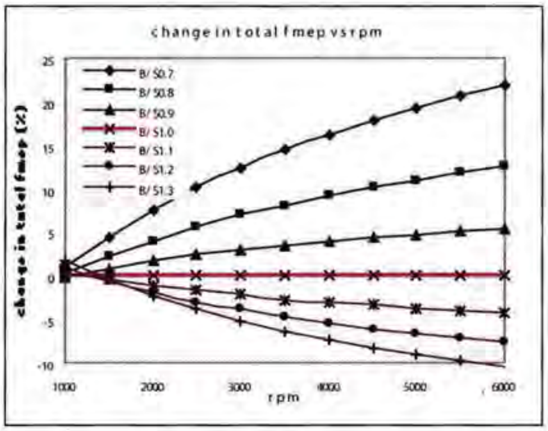
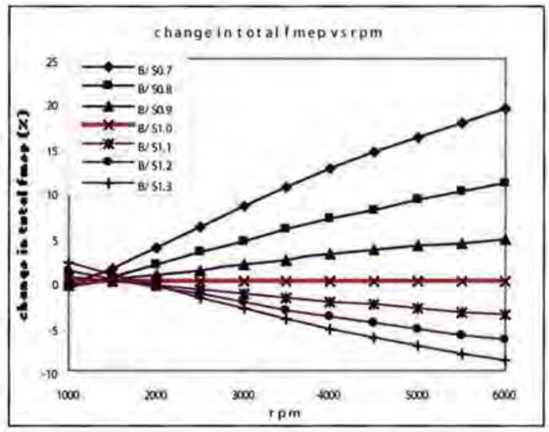


(c)

Figure 9: Effect of bore/stroke ratio on the distribution of friction for constant Engine displacement volume.



(a)



(b)

Figure 10: Predicted percent change in total fmep (constant displacement volume, 8 and 18 compression ratio ,101 kPa manifold pressure) as B/S varied relative to B/S = 1.00.

4.0 CONCLUSION

This paper represents the development and evaluation of overall and different components of engine friction model which predicts friction using basic engine design and operating parameters as inputs. Some important comments or results are given under results and discussions. The results highlight the importance of bore stroke ratio for different conditions to probe and identify the way to minimize the overall friction produced by engine.

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