

Investigations of Piston Ring Pack and Skirt Contributions to Motored Engine Friction

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ABSTRACT

An experimental study has been carried out to examine the influence of ring tan load and piston skirt modifications on piston assembly friction under motored engine conditions for initial temperatures of -20, 0 and 30°C and motoring speeds within the range 400 to 2000 rev/min. The study has been carried out using the block, crankshaft and pistons of a 2.4l, 4 cylinder diesel engine with a bore and stroke of 89.9mm and 94.6mm respectively. The pistons examined are typical of current designs for light duty diesels. A range of ring pack and piston skirt modifications have been tested, in each case as part of a complete piston assembly. The first changes produced reductions in fmep of between 5% and 38%. The reduction was due to improved skirt and ring pack designs in equal measure, each giving improvements of up to 20%. From this baseline eliminating the tan load of the piston rings was projected to give a further reduction in fmep of between 10% and 20%. Increasing the piston diametric clearances by close to the difference between minimum and maximum grade clearance gave smaller improvements of between 2% and 10% relative to the baseline, and reducing skirt roughness to perfectly smooth gave a projected reduction of typically 4%, and more at low oil viscosities.

INTRODUCTION

Rubbing friction losses in automotive i.c. engines are a source of engine inefficiency, a cause of poor fuel economy during warm-up and an adverse influence on cold start behaviour at low ambient temperatures. The study reported here is one of a series [1-6] carried out to investigate friction in engines under motoring conditions at low temperatures and low to medium engine speeds. These are particularly relevant to cold start and warm-up operating conditions. The emphasis of the work reported in the following was placed on the influence of

piston skirt and ring pack features, in keeping with the high contribution made by the piston assembly to the total engine friction. Throughout the study, the test work was carried out on pistons with complete sets of rings to minimise changes to the lubrication regime. The pistons examined are typical of current designs for light duty diesels.

The test work was carried out on block, crankshaft and piston assemblies of Ford 4-cylinder in-line 2.4 litre direct injection diesels. The cylinder bore was 89.9mm; the piston stroke was 94.6mm. Tests were run with the cylinder head removed and the cylinders open to eliminate pumping work and the valve train contribution to friction. The cylinder head was replaced by a stiffening plate secured with head bolts to the block. The crankshaft seals were retained and the lubrication system unmodified apart from the blanking off of the risers to the cylinder head. The same, SAE 5W/30 grade, lubricating oil was used for all the tests. Previous work [5,6] showed the dependence of friction on viscosity is effectively independent of SAE grade. Oil and metal temperatures were measured using K type, sheathed thermocouples at locations which include the oil main gallery and metal temperature close to the bore surface at the top, middle and bottom of the liner.

The engine assembly was mounted in a cold cell which could be cold soaked at temperatures down to -30°C. Typically, cold soaking would reduce the spread of temperatures across the engine to less than 0.5°C prior to the start of a motoring test. An external 24kW motor provided the motoring input to the crankshaft. The motoring torque was measured using a torque tube placed in-line between the motor and the flywheel. Data acquisition started at the same time as the engine was run up to the target test speed. The engine was motored at the target speed for at least 10 minutes. Temperatures and motoring torque were recorded at a sample rate of 10 samples per second and a moving

average was taken at one second intervals. The repeatability of test results was good, generally to well within 5% variation at all test conditions. The results presented are the average of at least two tests at each test condition. Measured values of motoring torque T_o are presented as values of friction mean effective pressure (fmep) calculated from

$$\text{fmep} = 4\pi T_o / V_s$$

where V_s is the engine swept volume. The fmep contribution of the piston assembly (piston skirt and ring pack, small and big end bearings) is taken to be the difference between results for the pistons, crankshaft main bearings and seals combined, and fmep values measured with the piston assemblies removed.

GENERAL OBSERVATIONS

During motoring at constant speed from a cold start condition, fmep falls with time. Principally this is associated with the reduction in oil viscosity at rubbing surfaces as temperatures rise. Previous studies have shown that the fmep variation follows a power-law dependence on oil dynamic viscosity evaluated from the temperature variation at an appropriate reference location, and plots of fmep versus viscosity allow the effects of design changes to be separated from those of viscosity. For the piston assembly, the strongest correlation between the fmep contribution and viscosity is achieved when viscosity is evaluated using liner temperatures at the mid stroke position [5]. This is illustrated in Figure 1 by the better consistency of results from tests at different start temperatures compared to cases where the bottom or top liner positions have been used as the reference location. Dynamic viscosity values μ (in mPa.s) for the SAE 5W/30 oil were calculated using the Vogel equation [7], given by:

$$\mu = k_v \exp\left(\frac{\theta_1}{T + \theta_2}\right)$$

where the k_v is 0.078mPa.s, θ_1 is 1180°C and θ_2 is 133°C [4], and T is the temperature. The results presented in Figure 1 are from tests at 1000 rev/min and start temperatures of -20, 0 and 30°C.

Investigations have focused on original (PV2) and revised (V3) designs of piston which differ in the design of the skirt profile, the size and distribution of low friction pads, the ring profile and ring tan loads. Details are given in Figure 2. The PV2 piston design has three low friction pads on each thrust side of the skirt. The V3 piston design has a single low friction pad on each thrust side of the skirt, resulting in lower skirt contact pressure due to larger surface area. Both PV2 and V3 have the same nominal piston skirt diameter. The top ring for PV2 is a half-keystone type while for V3, this is a parallel cross-section type ring. The second rings are Napier-type scraper rings. The oil control rings are coil spring

loaded 2-piece designs, where the V3 ring has lower height compared to the PV2 ring. The PV2 piston rings have higher tan loads than the V3 piston rings. The same connecting rod was used, hence the big-end bearing friction was the same for both cases. The number of differences makes it impossible to differentiate the benefits of any one, but the combined effect of the revisions yields a significant reduction in fmep of between 5% and 38%, depending on test condition. A comparison of fmep values at four speeds and various viscosity values is given in Figure 3. The fmep reduction given by the V3 piston design varies between 0.03 and 0.45 bar across the test conditions. The ratio of V3 to PV2 fmep values is plotted in Figure 4. This is always less than unity, showing the advantage of the V3 design. The ratio is almost independent of viscosity but rises as speed increases, indicating a diminishing advantage and suggesting the benefit of the V3 design is most significant when operating in the mixed lubrication regime. It is not possible to infer from these results if the advantage of the V3 design is due to improvements in ring pack or skirt design. To examine this, the ring grooves of a set of V3 pistons were widened to accept the wider rings from PV2 pistons.

The results of motoring tests at -20°C on V3 pistons with PV2 rings are plotted in Figure 5 together with test results for the PV2 pistons and V3 pistons with their original sets of rings. These plots allow the differences in fmep associated with PV2 to V3 changes to rings and skirt to be separated as indicated on the figure. The inferred reductions in fmep attributed to the V3 ring pack and the V3 skirt design are plotted in Figure 6. At the lower speeds of 400rev/min and 700rev/min, the biggest fmep reduction is given by the changes to the ring pack. At the higher speeds of 1000rev/min and 2000rev/min, the ring pack and skirt changes contribute more or less equally to the fmep reduction, each giving an fmep reduction of the order of 10%-20%.

All the above comparisons are based on tests carried out with the pistons running in a block specially honed to produce two pairs of cylinders with different surface finishes. For this 2+2 block, cylinder numbers 2 and 4 had plateau-honed (PH) bores, and cylinders 1 and 3 had Laser honed (LH) bores. The LH bores were fine-honed and incorporated laser-structured pockets in zones near each end of the piston-bore contact length. (A second block used in later tests had bores which were all plateau-honed.) The influence of bore finish was investigated by carrying out further tests with pistons running in just one or other of the pairs of cylinders. Results from these tests are given Figures 7 and 8. These show the LH bore finish gave a systematic reduction in fmep relative to the PH bore finish of between ~2% and 20%, and ~10% on average. The ratio of LH to PH values of fmep showed no dependence on oil viscosity, but did show some sensitivity to engine speed. The advantage of the LH finish was greatest, giving the lowest ratio of fmep values, at the lowest test speeds. This is consistent with results presented in

Jeng [8] who reported that the effect of surface roughness diminishes with increasing speed.

EFFECTS OF DETAILS

TAN LOAD - Tan load is the tangential force which must be applied across the ring gap to close this. It provides a measure of the static radial load exerted by the piston ring on the cylinder bore. The effect of tan load on piston friction was investigated through trials of three designs of ring pack for the same piston. Each pack had similar types of rings. Top rings were parallel cross-section, second rings were Napier-type scraper rings and the oil control rings were coil spring loaded 2-piece designs. There were some material and small geometric detail differences across the three pack designs in addition to differences in tan loads of one or more of the rings. The tan loads are summarised in Table 1. These were highest for R1 and lowest for R3. The results of motored tests on engine builds using each ring pack are given in Figure 9. At low viscosities, fmep drops from R1 to R3 but at the higher viscosities, there is no clear or consistent trend which would indicate the change in tan load has been the dominant influence.

Table 1. Tan loads (N) for piston ring sets R1, R2 and R3.

	R1	R2	R3
Top Ring	17.7	14.3	14.3
Second Ring	19.9	19.9	13.9
Oil Control Ring	22.5	20.1	16.5
Total/cylinder	60.1	54.3	44.7

Table 2. Tan loads (N) for V3 piston ring sets with varying oil control ring values.

	R1(L)	R1	R1(H)
Top Ring	17.7	17.7	17.7
Second Ring	19.9	19.9	19.9
Oil Control Ring	14.7	22.5	42.2
Total/Cylinder	52.5	60.1	79.8

To remove uncertainty about the influence of other changes to the ring pack on the apparent effect of tan load, a further set of tests were carried out using the R1 ring pack design. In these tests only the tan load of the oil control ring was varied, from 14.7N to 42.2N. The tan loads for the ring pack are summarised in Table 2. The V3 piston assembly with the original top and second rings was used. Tests were carried out at 1000 rev/min and 2000rev/min, since these gave a larger potential change in fmep than lower speeds. Plotting measured

values of fmep against oil ring tan load for chosen values of viscosity showed that the fmep of the piston assembly fell as tan load decreased. By extrapolating the trend lines for each viscosity, values of fmep corresponding to a fictitious, zero tan load condition for the oil control ring were identified. These fmep values give the friction contribution from the piston skirt, connecting rod bearings and piston ring friction which is not due to tan load of the oil control ring. The results are plotted in Figure 10, for the two speeds. The fmep variation shows a power law dependence on mid-stroke oil viscosity with a power law index of 0.32 for 1000rev/min and 0.42 for 2000 rev/min. The index values are slightly lower than the 0.5 value determined for the baseline case, with the R1 set of tan loads.

Table 3. Piston clearance at positions down the bore

V3 MIN CLEARANCE PISTONS 80.25 microns ave @ 80mm				
mm	Cyl 1	Cyl 2	Cyl 3	Cyl 4
20	0.096	0.127	0.127	0.085
60	0.090	0.089	0.095	0.072
100	0.071	0.063	0.063	0.066
120	0.071	0.051	0.051	0.060
160	0.052	0.057	0.063	0.053
V3 MAX CLEARANCE PISTONS 96.5 microns ave @ 80mm				
mm	Cyl 1	Cyl 2	Cyl 3	Cyl 4
20	0.112	0.143	0.144	0.101
60	0.106	0.105	0.112	0.088
100	0.087	0.079	0.081	0.082
120	0.087	0.067	0.068	0.076
160	0.068	0.073	0.081	0.069

The test results described above were manipulated to estimate the per-cent contribution of the oil ring and full ring pack to the fmep of the piston assembly when the tan loads have the standard values given by R1 in Table 2. The results are plotted in Figure 11, for the two test speeds. For each speed, the line fit to the diamond data points represents the percentage of the original total fmep remaining after the tan load of the oil control ring has been reduced to zero. This is around 90% or higher. The line fit to the triangle data points represents the difference, and is the effect of the tan load of 22.5N of the oil control ring. Finally, to estimate the effect of the combined tan loads of the ring pack, the triangle values have been multiplied by the ratio of the total to oil ring tan loads, 60.1/22.5, to obtain the square data points. The line fit to these suggests that the tan loads of the ring pack give rise to a contribution of 10% to 20% of piston assembly fmep. This is similar to the measured 10% reduction for a reduction in tan load from 70N to 18N reported in [9] for a 1.8l spark ignition engine, but less than the 50% given by a 50% reduction in tan load reported in [10]. The ring pack of modern designs of piston assembly is reported in [11] to account for 28% to 45% of the fmep of the piston assembly. The 10% to

20% contribution associated with tan load reported here is lower, suggesting that eliminating the tan load did not eliminate more than perhaps 1/3 to 1/2 of ring friction. The tan load is a measure of radial load exerted under static conditions; this may not reflect the load under dynamic conditions when there is ring movement, distortion and interaction with the lubricant film on the bore wall.

PISTON-BORE CLEARANCE - Two series of tests were carried out to examine the influence of piston clearance. The first tests were run using two sets of V3 pistons. Each set had skirt diameters matched to cylinder bores to give consistent clearances across the cylinders. All cylinders of the block had plateau honed bores. The cylinder-averaged diametric clearances at 80mm from the top of the bores were 80.3 μm and 96.5 μm . Table 3 gives the diametric clearances down the bore for these pistons. The difference in clearances between the two sets is of a similar magnitude to the minimum- to-maximum clearance range of production components. However, the absolute values of the smaller clearance set were still larger than the production mean. In effect, the clearance range was shifted to an oversized mean. Tests carried out at 400rev/min and 1000rev/min showed the decrease in clearance from the larger to smaller clearance pistons produced increases in fmep of between 5% and 10% across the range of oil viscosity. At 2000 rev/min, the reduction in clearance produced the opposite effect and fmep decreased by between 5% and 10%. The higher fmep given by the larger clearance at this speed may have been due to inadequate control of oil movement at large clearances, modifying friction behaviour, as oil appeared above the piston in sufficient quantity to visibly wet the piston crown.

Table 4. Diametric clearance (mm) at the grade point, 80mm down bores

Cyl No Piston	Cyl 1	Cyl 2	Cyl 3	Cyl 4	Ave
Grade A (max)	0.044	0.051	0.031	0.065	0.048
Grade C (min)	0.018	0.028	0.007	0.040	0.023

To resolve some of the uncertainty arising from the first series of tests, further investigations were carried out. In the second series of tests, pistons giving clearances more representative of minimum clearance values were used. These were grade A and C pistons giving average clearances of 23 μm (grade C) and 48 μm (grade A) when run in the block with plateau honed, grade B bores. The values of the clearances are shown in Table 4. The clearances were measured at the 80mm below the top of the block. Results show that the minimum clearance pistons (grade C) gave rise to higher piston assembly friction than the maximum clearance pistons (grade A) by between 2 and 10%, depending on test condition. A

summary of the percentage increase in fmep for minimum clearance pistons compared to maximum clearance pistons are shown in Table 5. Generally, cold temperatures (high oil viscosity) and low speed give the greatest difference, while high temperature (low oil viscosity) and high speed give the smallest difference. Both the first and second series of tests suggest the higher clearances give fmep reductions at low speed and that the benefit reduces and may become a penalty as speed is raised.

Test results for the grades A and C pistons, at 400rev/min and three start temperatures are given in Figure 12. The figure also includes results from tests using grade B pistons with skirts which had suffered polishing in previous running. The fmep values recorded for these were the lowest of the three cases, despite having clearance values in the middle. Repeat tests confirmed the advantage and suggested that skirt surface roughness might be a significant factor. Subsequently the effect of surface roughness was investigated.

Table 5. Percentage increase in fmep when diametric clearance was reduced from 0.048mm (Grade A) to 0.023 mm (Grade C).

Start temperature (°C)	Engine speed (rev/min)		
	400	1000	2000
-20	<10%	<10%	<2%
0	<2%	<5%	-
30	<5%	<2%	<2%

PISTON SKIRT SURFACE ROUGHNESS -Motored friction tests were carried out on pistons with 3 grades of skirt roughness ranked from low (LSF), to standard (SSF) and then high surface finish (HSF) grades. The pistons were run in the block with plateau honed cylinder bores. The piston assembly fmep generally decreased with decreasing skirt roughness. The LSF skirt gave significantly lower fmep values than the SSF and HSF skirts. The difference between results for the SSF and HSF skirt cases was relatively small, particularly at higher viscosities.

Skirt roughness values were defined by measurements of the arithmetic roughness average (Ra) and maximum peak-to-valley height (Rz). The measurements were taken at several locations within the contact area of the piston skirt. The averaged values are shown in Table 6. Initially the cut-off length for Ra was taken to be 0.8mm for all cases, but should have been longer for the roughness values of the HSF pistons. The measurements were repeated using a cut-off length of 2.5mm, which gave a higher Ra value, 6.92 compared to 4.85.