Paper No. 31 – Centrifugal Filtration of Lubricating Oil – Laboratory Test Results and Fleet Experience

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1. INTRODUCTION

The targets being set for the life between overhaul of commercial vehicle diesel engines are continually being extended. Manufacturers and users of diesel engines are seeking half a million kilometres for light duty engines, with targets for heavy duty engines in excess of one million kilometres. Lubricating oil contamination is the primary reason for engine wear and its effect on engine life; it also leads to reduced fuel efficiency, reduced oil life and loss of engine performance through increased frictional losses.

The demand for reductions in diesel exhaust emissions also means reductions in lubricating oil consumption levels, achieved by taking the top ring higher up the piston, reducing piston top land clearances and operating with hotter cylinder bores. Since oil consumption serves as a means for removing contamination from an engine, these developments have also put an increased burden on lubricating oils to contain contamination as well as resist oxidation. Furthermore recent developments of oils to increase their high-temperature oxidation resistance have caused some oils to give worse anti-wear and soot prevention properties.

The combination of all these effects has made engine wear and oil cleanliness, today, one of the most important aspects of diesel engine design.

This paper describes laboratory accelerated wear tests conducted to study the effect in diesel engines of a centrifugal by-pass lubricating oil separator which cleans the oil and hence reduces the wear rate. The rate at which abrasive particles, less than 15 microns in size, cause wear of engine components is investigated, particularly in the valve train area, which is a part of the engine not covered in previous accelerated engine wear tests (1,2). Bench tests and field results are presented which support the laboratory analysis of engine wear and centrifuge efficiency.

2. ENGINE TEST CONDITIONS AND OBJECTIVES

Previous accelerated engine wear tests have been on heavy duty engines running at rated speed (1,2,4), with A.C. fine test dust (ACFTD) added to the lubricating oil in discrete amounts at regular intervals. Addition of contaminant was done either manually or using a quadrant feed system, with typically 4 grams added every four hours over a test duration of 150 hours. With an oil system capacity of 35 litres this equates to approximately 0.03 grams per litre per hour.

The accelerated wear tests described were run on smaller engines over a standard duty cycle (Fig. 31.1). The test period was 150 hours, or until failure, and the contaminant was added to the oil in a continuous manner throughout the test thereby eliminating the effect of discrete additions. A device was constructed to meter into the engine oil exact amounts of ACFTD made up into a paste, by means of a motor and gear box which drives a piston in a cylinder to squeeze the paste through an orifice into the oil circuit.

Two engines were used in the tests, one with full-flow filtration only, the other with full-flow filtration and a centrifugal separator added. The engines chosen for the tests were of two litre capacity, four cylinders, direct injection, turbocharged and rated at 59.5 kW at 4,500 rev/min. As the oil system capacity in the engines was only 5 litres, a contaminant addition rate of 0.15 grams per hour would have been required to achieve the level described previously. It was considered that this rate was too small to meter accurately and to see discernable wear at the end of a 150 hour test run; a contaminant addition rate of 0.75 grams per hour was therefore chosen. This high addition rate would allow for a study of a centrifuge's ability to stabilise sump contamination levels over a long test period, given the effect of contaminant accumulation in the centrifuge rotor bowl and the effect of time on cumulative efficiency of a by-pass system.

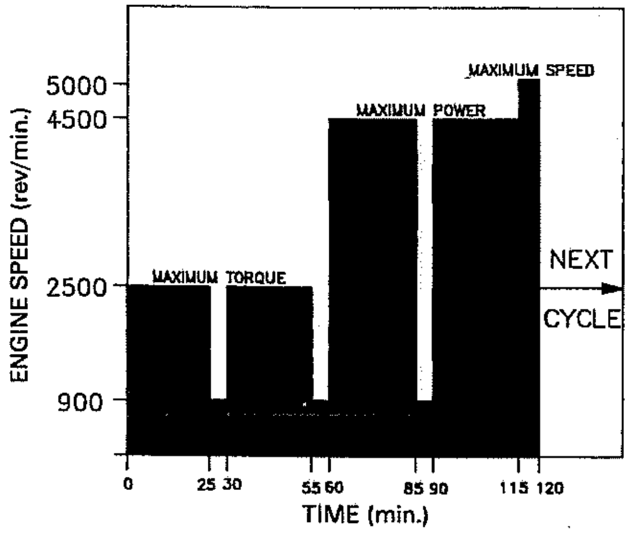

Fig. 31.1 Engine Test Cycle

It should be noted that the levels of contaminant used in the tests were not representative of that seen in practice, and the tests were not therefore designed to reproduce expected field service.

The effect of the two filtration systems on engine wear rate were assessed by measuring the dimensions and weight loss on all critical components, and by assessing oil contamination levels at regular intervals throughout the tests.

A major objective was to study wearing effects with oil containing abrasive particles below the nominal screening capacity of the standard full-flow filter, that is, less than 15 microns in size. The filter change intervals were established during the running of the engine with standard full-flow filtration and without the centrifuge, by monitoring differential pressure drop across the filter to determine its plugging characteristics. Since the relief valve in the filter fitted to the test engines is designed to open when the pressure drop is between 0.5 and 1.2 bar, it was decided that the filter should be changed as soon as the pressure drop reached 0.5 bar. This filter change interval was then applied in the running of the engine with the centrifuge fitted, and at each filter change an oil sample was taken.

3. ENGINE MODIFICATIONS, TEST METHOD AND METROLOGY

For the purposes of this report, Engine No. 1 refers to the standard engine run with full-flow filtration only, Engine No. 2 refers to the engine modified by the addition of a centrifugal separator.

The lubrication circuits of both engines were modified to be identical, allowing for contaminant addition, oil sampling and oil cooling. The oil circuit, as designed, is shown in Fig. 31.2 and a photograph of the engine installation shown in Fig. 31.3. Because the contaminant addition in Engine No. 2 was into the drain pipe from the centrifuge back to the sump (to facilitate the mixing of contaminant and oil), the same set-up was used on Engine No. 1 but with the rotor within the centrifuge replaced with a single nozzle, sized to give the same by-pass flow-rate as that through the centrifuge.

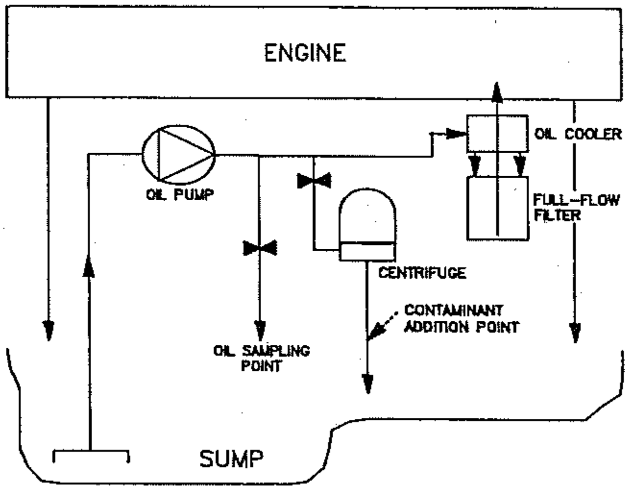

Fig. 31.2 Schematic of Lubricating Oil System

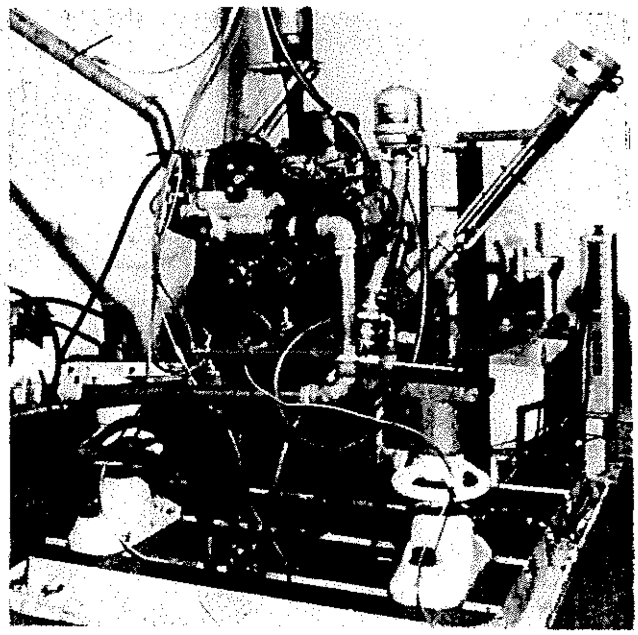

Fig. 31.3 Wear Test Engine

To ensure test conditions were as similar as possible in the two engines, all pipework external to the engines was the same.

The two engines were installed in computer controlled test cells allowing for on-line measurement of all essential parameters and, in particular, blowby and oil pressure at pump exit and in the main gallery, indicating ring wear and bearing wear respectively.

The lubricating oil used in the tests was Shell Rimula 15W/40.

Each engine was subject to a 50 hour run-in cycle with no contaminant addition, the purpose of which was to ensure that the intitial and possibly unrepresentative wear in the engines did not effect the wear tests, and that all built-in dirt was removed and the engines were operating correctly.

After running-in, the engines were removed from the test cell and stripped down for measurement. All the main components were measured and weighed; cam followers, tappet pads, camshaft, pistons, piston rings, piston pins, bores, bearings, thrust washers and crankshaft. Surface finish and flatness measurements were also taken of critical components. Methods used in making the measurements were the same for each engine.

The tests were run continuously, and the time during which the engines were shut down for filter changes and oil samples was kept to a minimum, to prevent contaminant settling out.

4. ENGINE TEST OBSERVATIONS

Immediately before the running of each test, an oil pressure check was undertaken to ensure that the oil pressure variation with engine speed was the same as it was before the engines were stripped down and rebuilt.

During the running of Engine No. 1, blowby gradually increased and oil pressure dropped with time. During the tenth cycle, at 20 hours, the oil pressure drop across the full-flow filter exceeded 0.5 bar and the filter was changed. This happened again at 44 hours. At 58 hours the engine stopped automatically due to low oil pressure. Upon inspection it was found that the oil level was well below the minimum. To ascertain the reason for the low oil pressure the sump was refilled and the engine restarted. The gallery oil pressure, power output and torque readings were well below normal and blowby was over 100% higher than at the beginning of the test. The test was stopped and the engine stripped down for inspection. It was found that failure was due to turbo-charger spindle breakage; subsequent disintegration of the turbocharger and loss of oil pressure had caused damage in the combustion chambers and big-end bearings.

Engine No. 2 was then run. The filter was changed at 20 and 44 hours as with Engine No. 1 and then at every 24 hours until test completion at 150 hours. The engine then was still running satisfactorily.

Fig. 31.4 shows a comparison of blowby measurements for Engine No. 1 and the first 60 hours of Engine No. 2. Fig. 31.5 shows gallery oil pressure measurements for both engines over the same period.

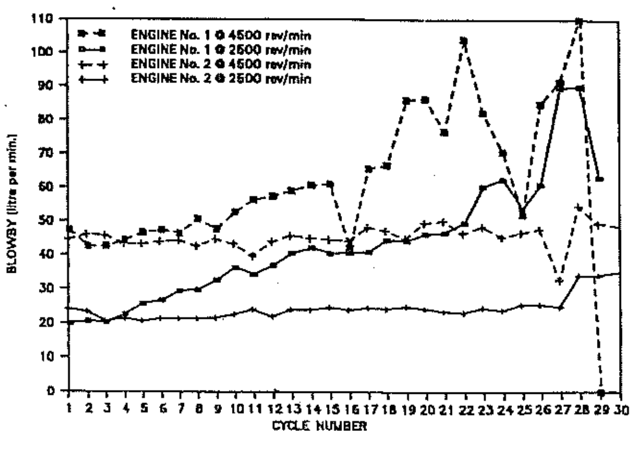

Fig. 31.4 Engine Blowby

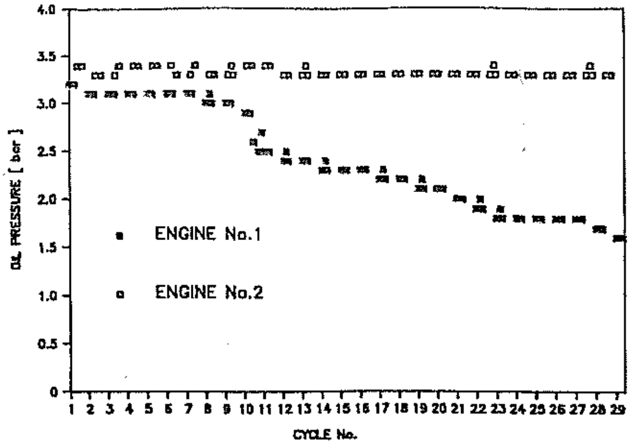

Fig. 31.5 Gallery Oil Pressure at 2500 rev/min

5. APPLICATION OF THE CENTRIFUGAL SEPARATOR TO ENGINE No. 2

The centrifugal separator selected for application to Engine No. 2 was of 0.32 litre oil capacity and 0.25 litre contaminant capacity (Fig. 31.6). Flow-rate through the unit varied between 2.3 and 3.4 litres per minute at inlet pressures of 2 and 4 bar respectively; flow-rates being taken on a bench test of the unit at an oil temperature of 75°C. The oil pressure seen by the centrifuge was between 2.3 and 3.3 bar for the majority of the cycle that is corresponding to engine speeds of 2500 and 4500 rev/min; the centrifuge rotor speed at these pressures being 6400 and 7700 rev/min.

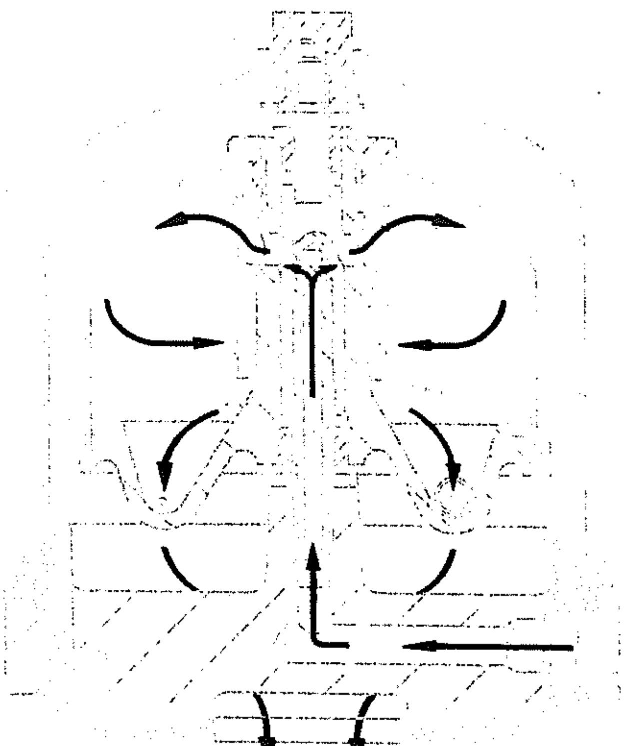

Fig. 31.6 Centrifugal Separator

With an engine system capacity of 5 litres, the total volume of oil would be processed in the centrifuge between 27 and 41 times per hour depending on engine speed and oil temperature during the test cycle. It should be noted that one of the advantages of applying a centrifuge to an engine of this size is that the rate at which the sump contents are circulated through the centrifuge is three to four times higher than that achieved by a centrifuge fitted to a larger heavy-duty diesel engine. This significantly increases the probability of a contaminant particle passing through the centrifuge and therefore increases the likelihood of its extraction. In addition, smaller engines tend to run at higher oil temperatures which increases the efficiency of the centrifuging process.

Excluding the contamination removed from the system by the full-flow filter the contamination in the sump would be given by the equation:

$$x = \frac{C}{nN}\left(1-e^{-nNt}\right)$$

where: x = contamination in the sump, g
C = rate at which contaminant enters the sump, g/h
n = centrifuge instantaneous efficiency,
N = flow rate through centrifuge per litre of oil system capacity, h^{-1}
t = time, h

The above equation would be different for each particle size and mass but, taking maximum and minimum values of n of 0.05 and 0.02, the system stabilises at levels of x of 0.37 and 1.39 grams after 3 to 15 hours respectively, assuming no deterioration in centrifuge efficiency with time (Fig. 31.7). On running the tests for 150 hours the system would pass through the transient phase described in the above equation and the contamination level would therefore have stabilised in Engine No. 2.

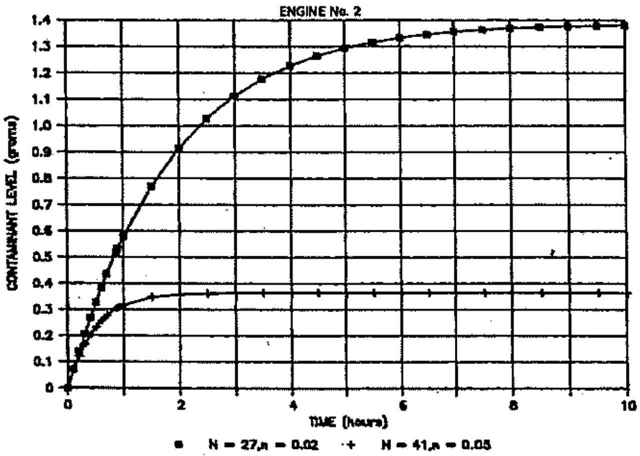

Fig. 31.7 Theoretical Level of ACFTD in Sump

Practical verification of the stabilising of contaminant level was made by analysing silicon levels in the oil samples taken at each filter change (silicon dioxide is the major constituent of ACFTD).

Figure 31.8 shows a particle count by volume of the ACFTD used in the test. As can be seen, the majority of the particles are in the size range of 3 to 20 microns. As the

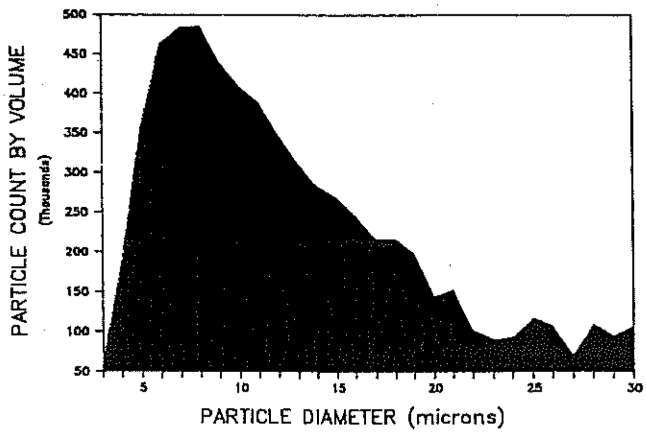

Fig. 31.8 Size Distribution of AC Fine Test Dust by Volume

nominal screening capacity of the full-flow filters used on the test engines is 15 microns, it would be expected that the majority of particles above this size would be taken out by the full-flow filter leaving 60 to 80% of the ACFTD remaining in circulation in the engine to be separated out by the centrifuge.

6. ENGINE TEST RESULTS

Of all the measurements taken to analyse wear rates in the two engines, representative samples of the results covering critical components are shown in Figs. 31.9 to 31.18. The only components that could not be measured after testing were the big-end bearings and turbocharger components from Engine No. 1, due to the damage they suffered.

The results as presented use wear rates and weight losses per hour to allow direct comparison between the two engines. Wear rates on circular components are expressed as changes to the radius, rather than the diameter, to allow for comparison with linear wear rates.

Figs. 31.9 and 31.10 show the wear rates on the piston rings. Apart from cylinder No. 4 the greatest weight loss per hour was seen on the top two rings in Engine No. 1, both of which suffered a wear rate twice as high as that seen on each of the oil control rings. Most noticeably however the wear rates seen in Engine No. 1 were between 15 and 50 times higher than those in Engine No. 2. Similar differences between the two engines can be seen from the measurements of ring gap. In this case the wear on the periphery of the top rings was less than that on the second rings due to the top ring periphery being chrome plated; by inference side face and back face wear would have been greater than on the second rings.

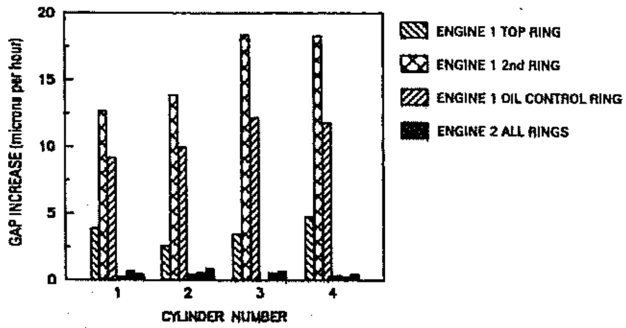

Fig. 31.9 Piston Rings – Fitted Gap Increase/Hour

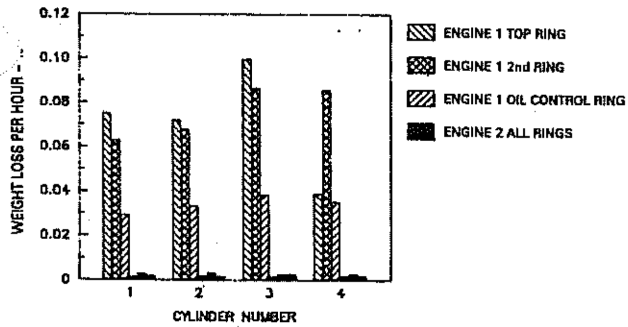

Fig. 31.10 Piston Rings – % Weight Loss/Hour

Figs. 31.11 and 31.12 show the wear variation in the cylinder bores of both engines. The wear was measured on both in-line and transverse axes at five positions down the bore at 8, 14, 57, 111 and 156 mm from the cylinder head face. The measurement planes at 14 mm and 111 mm represent, respectively, the position of the top ring at TDC and the oil control ring at BDC.

Fig. 31.11 shows averaged cylinder bore wear taken at three of these planes; the wear at 8 mm and 156 mm was negligible and is not included. The wear rate for Engine No. 1 is between 7 and 10 times that of Engine No. 2 throughout the

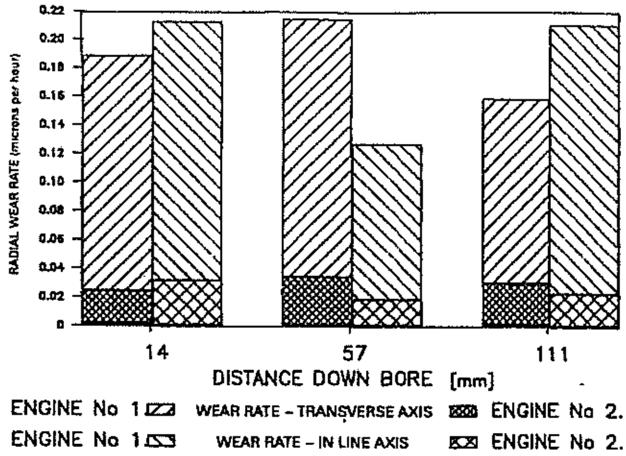

Fig. 31.11 Average Cylinder Bore Wear

worn portion of each cylinder bore. Wear at TDC and BDC reversal points was, as expected, uniformly distributed around the bore being caused by the collapse at those points of the hydrodynamic oil film under the ring peripheral face. By comparison wear at the mid-stroke point, 57 mm down the bore, is reduced on the in-line axis, but displays similar wear on the transverse axis arising from the piston skirt side loading.

Fig. 31.12 represents typical surface profile traces along the bores of the two engines before and after the tests; notice the wear scar of about 12 microns depth at the position of the top ring reversal point in Engine No. 1 after the test.

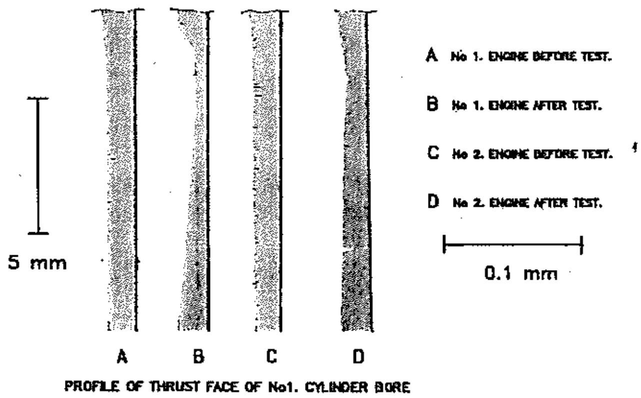

Fig. 31.12 Cylinder Bore Surface Profile

Figs. 31.13 and 31.14 show the wear on the piston pins in the region of the small end bush. The average material removed on Engine No. 1 was 14 to 16 microns after the 58 hours of operation; sufficient to be felt by hand. The results from Engine No. 2 showed an average 90% less wear after 150 hours of engine running; this can be seen by the worn surface profiles shown in Fig. 31.13.

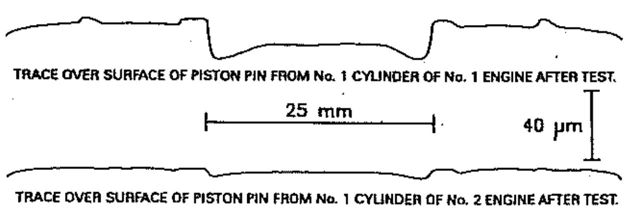

Fig. 31.13 Piston Pin Surface Profile

Figs. 31.15 to 31.17 depict the wear of the cam followers and tappet pads. As with the other components measured, the wear rate and weight loss of the cam followers and tappet pads in Engine No. 2 was significantly less than in Engine No. 1. For example, after 50 hours, the average weight loss per cam follower for Engine No. 1 was 0.07% compared with 0.005% for Engine No. 2. The surface traces, taken

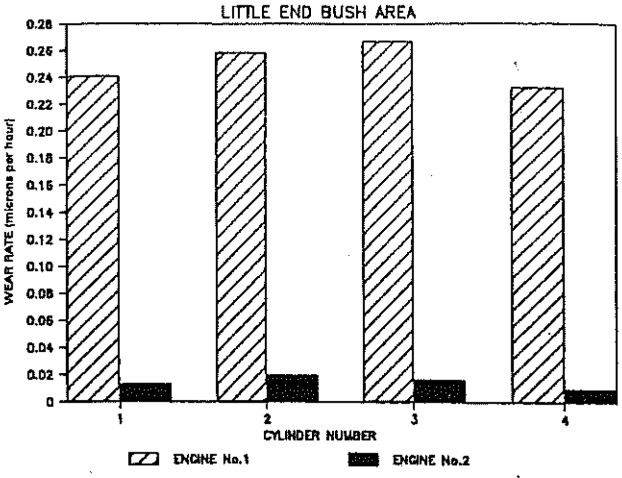

Fig. 31.14 Wear Rate on Radii of Piston Pin

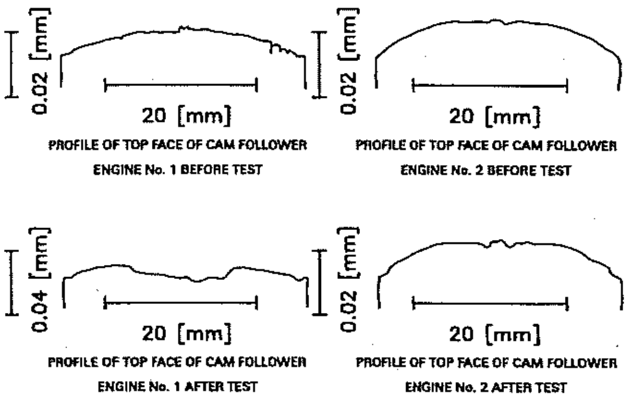

Fig. 31.15 Cam Follower Surface Profile

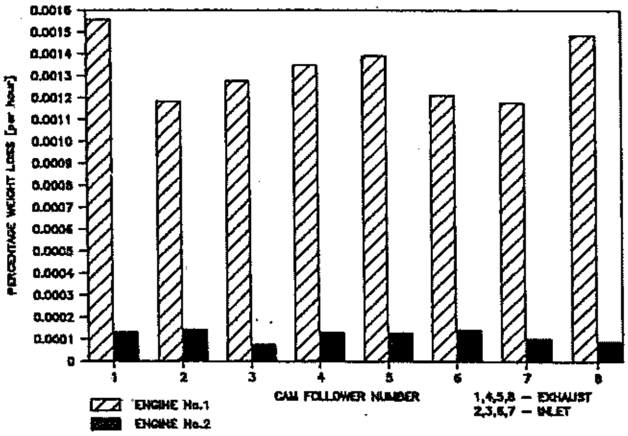

Fig. 31.16 Cam Followers – % Weight Loss/Hour

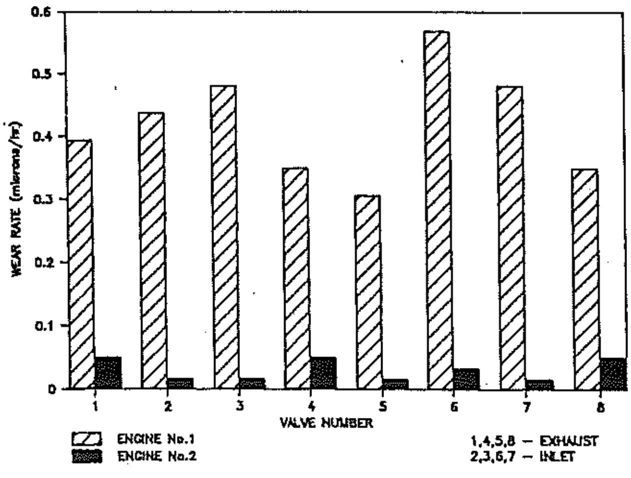

Fig. 31.17 Linear Wear Rate of Contact Area of Tappet Pad

across the face of the cam followers, show a noticeable area of wear at the cam contact point on Engine No. 1 after the 58 hours of operation (Fig. 31.15).

The loss of lift was measured on the camshafts in three positions; at the nose and at 50 degrees on either side. For Engine No. 1 the loss of lift at plus and minus 50 degrees, averaged across the eight cams, was 0.1775 mm and 0.1643 mm respectively; for Engine No. 2 it was 0.0653 mm and 0.0495 mm. The wear rates, therefore, in these two positions on the cams in Engine No. 2 were 0.44 and 0.33 microns per hour compared with 3.1 and 2.8 microns per hour on the cams in Engine No. 1, that is between 7 and 8.5 times greater. The differences in loss of lift on the camshaft nose were less pronounced, being 0.0538 mm for Engine No. 1 and 0.0406 mm for Engine No. 2, giving wear rates of 0.93 and 0.27 microns per hour respectively. This is due to the very small oil film (less than 0.1 microns) that occurs at the nose allowing contamination only up to this size to be introduced into the wear mechanism.

The wear profiles on the crankshaft journals and main bearing shells were complex. Because the wear profiles vary so much with angular position, bearing surface area, crankshaft deflection and journal position in each engine they are not directly comparable; a complete analysis therefore is outside the scope of this report. However, in all the measurements taken there was significantly less wear in Engine No. 2, and Fig. 31.18 is an example of this, showing for both engines, radial wear rate of the five crankshaft main bearing journals measured in the vertical axis. The wear rate in Engine No. 2 was 15 to 20% of that seen in Engine No. 1. With the wear on the main bearing shells it was evident that the greatest wear was taking place at the point of minimum oil film thickness and that, in these areas, the wear rate seen in Engine No. 2 was between 14 and 52% of that measured in Engine No. 1.

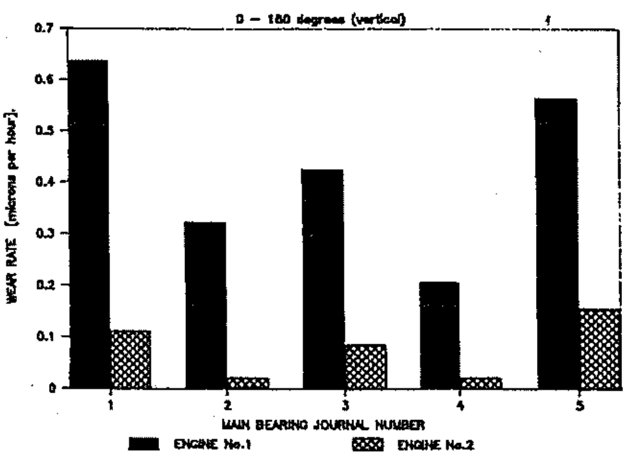

Fig. 31.18 Wear Rate on Radii of Crankshaft Main Bearing Journal

7. OIL CONTAMINANT ANALYSIS

Oil samples were taken from Engine No. 1 at 20 and 44 hours and at 20, 44, 70, 94, 118, 142 and 150 hours from Engine No. 2. These samples were then analysed both by spectrographic analysis and particle counting to assess amount and size of contaminant particles in the oil. A sample of the contaminant collected in the centrifuge was also taken and analysed.

All the oil samples analysed by particle counting showed inconsistent results, even though care was taken in obtaining representative samples and in the calibration of the equipment. This could have been due to the well reported problems (3) associated with a large concentration of fine soot particles and the possible existence of water droplets in the sensing zone of the apparatus, which lead to coincident error with concentrations higher or lower than actually exist and with measured size. A more likely reason for the inconsistency in particle count is the extent to which ACFTD particles break down into smaller particles in the engine; since the equipment was set to sense only particles above 3 microns, the validity of the results could not be relied upon. The following results therefore are from spectrographic analyses only.

Fig. 31.19 shows the concentration of particles in parts per million (ppm) for the six main metals in the wear debris in the two oil samples taken from Engine No. 1. This is a good example of the 'chain reaction of wear' which takes place in engines. As the ACFTD causes wear on the various components the wear particles themselves cause further damage ultimately leading to catastrophic failure.

OIL SAMPLES – ENGINE No. 1

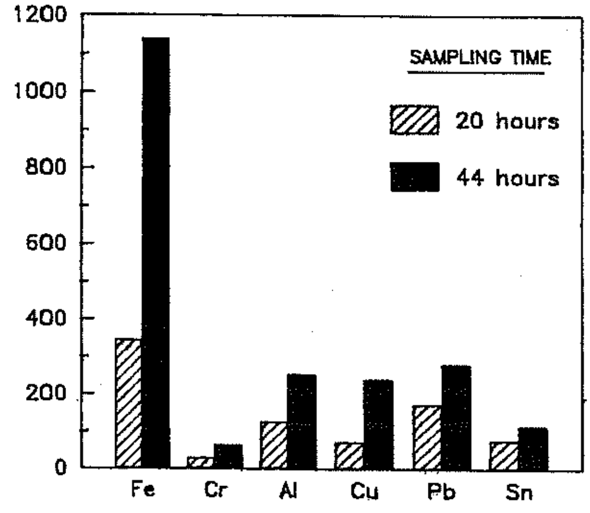

Fig. 31.19 A.A. Spectrographic Analysis – Oil Samples Engine No. 1

Fig. 31.20 compares the readings for silicon content from both engines and shows that the addition of the centrifuge to Engine No. 2 resulted in the amount of silicon being stabilised at a low level throughout the test. The silicon is present in the form of silicon dioxide being the major constituent of ACFTD.

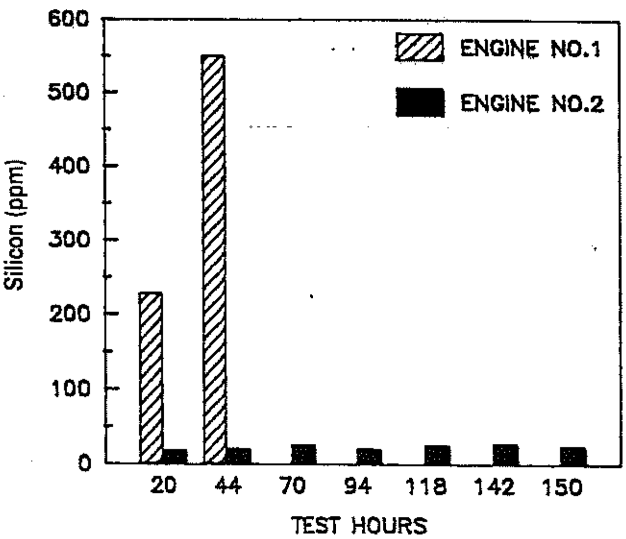

Fig. 31.20 A.A. Spectrographic Analysis – Silicon

It should be noted that atomic absorption techniques for analysing contamination in oil are most sensitive to smaller particles, below 5 microns in size, and will not detect particles above 10 microns. The results quoted should therefore be viewed with this in mind. As the centrifugal force on a particle increases with its size it would be expected that particles above 10 microns would have been extracted by the centrifuge, even more efficiently than the sub-10 micron particles represented by the results in Fig. 31.20.

Fig. 31.21 shows the full analysis of the constituents of the contaminant collected in the centrifuge. These results were obtained after diluting the contaminant 4:1 with clean oil. As expected the most prevalent elements are silicon from the ACFTD, the count for which exceeded the range of the measuring instrument, followed by iron and aluminium wear particles.

(ENGINE No. 2 after 150 hours)

		ppm
WEAR ELEMENTS	IRON	1681
	CHROMIUM	97
	ALUMINIUM	1419
	COPPER	107
	LEAD	390
	TIN	480
	NICKEL	44
	MANGANESE	36
	TITANIUM	28
	SILVER	0
	MOLYBDENUM	20
CONTAMINANT ELEMENTS	SILICON	>5000
	SODIUM	450
	BORON	152
	VANADIUM	2

Fig. 31.21 A.A. Spectrographic Analysis of Centrifuge Contents

The mass of contaminant collected in the centrifuge was 91.6 grams, equivalent to an average collection rate of 0.61 grams per hour (in addition to the contaminant collected in the seven full-flow filters used throughout the test). With an engine oil system capacity of 5 litres this mass of contaminant represents 1.8% solids by weight. In addition to the contaminant particles listed in Fig. 31.21, the soot that was extracted by the centrifuge influenced oil thickening, as can be seen by the viscosity of the oil through the test.

In Engine No. 2, the viscosity of the oil (measured in Centistokes at 40°C) increased from 105 at 20 hours to 125 at 150 hours (an average 0.15% increase per hour). The viscosity of the oil in Engine No. 1 increased from 110 at 20 hours to 116 at 44 hours (an average 0.23% increase per hour).

8. ANALYSIS OF ENGINE TESTS

The tests in a small direct injection diesel engine have shown that substantial wear can occur between all close fitting lubricated components leading to catastrophic failure, caused by abrasive contaminant with an average particle size less than 15 microns in diameter. In these conditions full-flow filters have a limited capacity to protect an engine against abrasive wear, because they cannot trap particles of a size range equal to or just greater than the oil film thicknesses in a running engine.

Engine No. 1 equipped with standard full-flow filtration only, which failed after 58 hours of operation, had received a total of 43.5 grams of ACFTD, 60 to 80% of which was less than the screening capacity of the full-flow filter (15 microns), and this caused catastrophic wear.

Engine No. 2, equipped with a by-pass centrifugal separator in addition to the full-flow filter, under the same operating conditions as Engine No. 1, maintained a very low level of contamination in the system throughout the 150 hours of testing and suffered negligible wear. Wear rates were between 3 and 52% of those seen in Engine No. 1; the greatest wear reduction being in the top half of the engine, particularly in the ring pack and cam and cam follower areas, where the smallest oil films occur.

As the rotor speed of a self-driven centrifugal separator is dependent on engine oil pressure, and the tests on Engine No. 2 were conducted over a variable duty cycle and hence variable oil pressure, the results gained are more representative than would have been the case with tests run at constant rated speed.

By the continual addition of contamination to the engine during the test, it has been possible to show that a centrifugal separator, applied to an engine of this size and allowed to operate over a representative test period, can pass through the transient phase of a by-pass system and stabilise contaminant level. Its ability to maintain the same flow-rate and efficiency throughout the operating period is a feature which is not possible with media type by-pass filters.

9. MULTI-PASS AND SINGLE-PASS BENCH TESTS

Multi-pass and single-pass bench tests (test standard SAE J806) have been conducted on centrifugal separators, sized for heavy-duty engine applications, which, in terms of the centrifuge efficiency, show similar results to those obtained in these engine tests.

In the first test, 18.9 litres of RFO-3-79 (SAE J1260) test oil, maintained at a constant temperature of 82°C, was continually circulated through a centrifuge for 24 hours. The presssure maintained in front of the centrifuge was 4.22 bar and the flow-rate was 8.3 litres per minute, equivalent to a system recirculation rate of 26.3 times per hour. ACFTD was added to the oil at the rate of 20 grams per hour.

At the end of the test, of the 480 grams of ACFTD added to the oil, 464.86 grams were removed by the centrifuge giving a cumulative weighed efficiency of 96.8%.

The second test, for single-pass particle retention, was conducted to assess the instantaneous efficiency of a centrifuge with large particles. For this purpose A.C. coarse test dust (ACCTD) was used which has characteristics similar to ACFTD but with a larger particle size. The test set-up was the same as in the first test but a single amount of ACCTD of 0.3 grams was added to the oil and the rig run for 15 minutes. At the end of the test 79.7% of the ACCTD had been removed by the centrifuge showing the efficiency of a centrifuge with large, dense particles.

10. FIELD EXPERIENCE

10.1 Analysis of Contamination

To study the rotor contents of a centrifuge applied to a diesel engine in service (for comparison with the above engine tests), a full qualititive and quantitive analysis was undertaken of the contamination collected in a centrifuge applied to a diesel engine in service in a coach operating between Chicago and Winnipeg in North America.

This engine had completed 15,200 kilometres since its last service and the contaminant retained in the centrifuge was removed and the constituent fractions analysed. It was found that 25% was metal oxides, 54.7% was carbon (soot) and 5% was retained water, the rest being 'spent' additive and oil.

A spectrographic analysis of a sample of the contaminant revealed the following relative levels of wear metals:

Iron	437 ppm
Lead	60 ppm
Copper	44 ppm
Chromium	80 ppm
Aluminium	15 ppm
Tin	165 ppm

A ferrograph was also taken of a sample of the contaminant, after having been solvent washed in toluene. This method has the advantage of being able to reveal wear particle concentration, size distribution and shape and is most sensitive to particles sized around 10 microns. A photograph taken at 400x magnification is shown in Fig. 31.22. This shows large volumes of iron and copper particles, many below 10 microns in size, plus a large 55 micron silica particle in the centre. The full analysis showed red oxides in the 20 to 30 micron size range indicating water corrosion, ferrous particles less than 1 micron in size, indicating corrosive wear, lead alloys in the 5 to 8 micron size range, copper particles in the 5 to 10 micron size range and clusters of spherical particles indicating that cracks in rolling contact components had begun to spall.

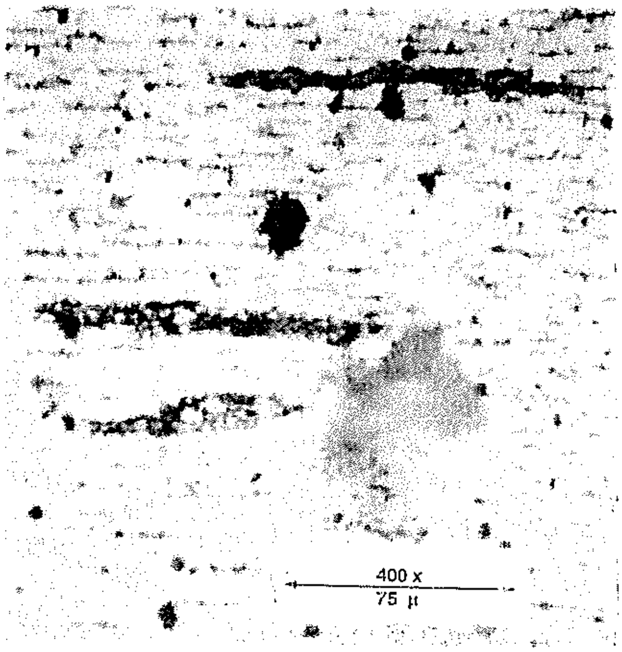

Fig. 31.22 Ferrograph of Centrifuge Contents

A sample of contaminant collected in a centrifuge applied to another engine of the same make was analysed by a resistive method of particle counting. This showed that 78% of the particles were below 15 microns in size.

10.2 Wear Rate Reduction

Several large truck fleets in the USA specify by-pass centrifuges on their heavy-duty truck diesel engines, and have had the opportunity, over many years, to evaluate the effect a centrifuge has in increasing component and engine life.

One fleet in particular comprises many Class 8 trucks powered by Caterpillar, Cummins and Detroit Diesel engines all of which are fitted with by-pass centrifuges. The engines are all replaced after six years service during which they will have covered between 1,520,000 and 1,920,000 kilometres with one overhaul typically at 1,000,000 kilometres. This compares with 10 years ago when this fleet was not achieving engine total life of 750,000 kilometres. Also such average engine life to rebuild is now being achieved with oil change intervals of 40,000 kilometres. This experience should be compared with the current industry averages of 700,000 kilometres life to rebuild and oil change intervals of 25,000 kilometres (10). As an example of relative component life, this truck fleet has turbocharger life of 800,000 to 1,000,000 kilometres compared with the industry average of 385,000 kilometres.