AUTOMOBILE DIVISION

A PASSENGER CAR DIESEL ENGINE FOR AMERICA

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This paper describes the conversion of the 3.6 l (225 in) Chrysler gasoline engine to diesel operation for use in passenger cars. The requirements for an acceptable passenger car diesel engine are discussed and the design modifications necessary to ensure these are described. The modified engine was subjected to a detailed development programme resulting in a performance approaching that of the single carburettor gasoline engine together with the expected improvement in vehicle fuel economy and adequate mechanical durability. Reduction of gaseous emissions was investigated and the resulting demonstration vehicles proved very acceptable, and deserving of further development towards production.

1 INTRODUCTION

In the mid 1970s the Chrysler Corporation, along with some other sectors of the United States automobile industry, recognized the need to offer diesel engines as power plants for their passenger cars and light trucks to enable them to comply with the fuel economy standards. Together with the fuel saving requirement it was necessary to be able to meet the projected gaseous emission standards whilst maintaining an acceptable performance level. It was also important to achieve acceptability in other areas such as noise, cold starting, driveability, etc.

With these aims in view Chrysler enlisted the assistance of Ricardo who have wide experience from involvement, in some measure, in most of the European light duty diesel engine designs and more particularly for their expertise in the areas of light duty diesel combustion, namely the Comet Vb system.

As the project gathered momentum Ricardo became responsible for more aspects of the engine including mechanical development and some proving tests in vehicles enabling definitive fuel economy and emission data to be obtained.

This paper describes the development of a diesel version of the Chrysler RG225 gasoline engine as fitted to the Chrysler Volare, Dodge Aspen etc., cars.

2 BASIC AIMS OF THE PROGRAMME

2.1 Performance

Based on European experience some US engineers have become resigned to an apparently inevitable sacrifice of some 30 per cent in specific output from gasoline engines converted to diesel as part of the cost of achieving diesel fuel economy levels.

However, in reviewing the general performance levels of US gasoline engines as fitted to the majority of popular automobiles, Ricardo had concluded that, in the American scene, the sacrifices of power would be much less than in Europe. Typically the American six and eight cylinder gasoline engines produce in the region of 18–22.5 kW/l (0.4 – 0.5 h.p./cu inch displacement), a level which has come to be expected from European light duty diesel engines (**1**). The generally larger swept volume per cylinder involving either longer strokes and lower rotational speeds or over-square configurations will however result in some sacrifice of performance when compared with the smaller higher speed diesels in Europe.

The performance level predicted for the RG225 engine compared with the gasoline versions is represented in the following table:

	Gasoline		Diesel
	Single carb.	Twin carb.	predicted
Power, kW (SAE NET)	7.46 @ 3600 (100 h.p.)	82.1 @ 3600 (110 h.p.)	74.6 @ 3600 (100 h.p.)
Torque, Nm (SAE NET)	230 @ 1600 (170 lbf ft)	244 @ 2000 (180 lbf ft)	232 @ 2000 (171 lbf ft)

2.2 Exhaust emissions

The above diesel performance should be accompanied by a maximum smoke level of 2–2.5 Bosch and NO_x of 1.5–2.0 g/mile in a 4000 lb inertia weight vehicle. It was also predicted that to meet a 1.0 g/mile NO_x level a sacrifice of some 10 per cent would have to be made in both performance and fuel economy, depending on the strategy used.

Hydrocarbon emissions will very much depend on the capability of the fuel injection equipment to give the required injection timing plan whilst also retaining clean injection without secondary nozzle openings, etc.

2.3 Noise

With the predicted performance level achieved, noise would probably be the major deterrent to most car owners choosing the diesel option. Under drive-by conditions European diesel engined cars generally radiate 1–2 dBA more than the equivalent gasoline vehicle. There was no reason to expect that the

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Chrysler engine would do better than this, especially as the engine was basically of gasoline origin and therefore rather less robust than European diesels. However, the effect of injection retard is also to reduce noise and it was expected that when set for low NO_x and with typical American interior trim the presence of the diesel, once warm, would not be obvious to the driver and passengers.

2.4 Cold starting

The combination of a compression ratio of 21.5:1 and modern glow plugs would ensure reliable starting down to −20°C (−4°F). Below this temperature, in common with all diesels some supplementary starting aid would be required. It was therefore important not to have to reduce the compression ratio in order to control maximum cylinder pressures. A combustion system with a low peak firing/compression pressure ratio was therefore essential.

2.5 Driveability

No problems were foreseen in this area especially under cold starting conditions where modern emission controlled gasoline engines have to be set at high idle speeds to avoid stalling. Once running the diesel can be relied upon to produce cold engine power on demand. Nevertheless it was important to ensure that in all other respects the driveability would be at least as good as the equivalent gasoline engine.

3 DESIGN CHANGES FOR THE DIESEL

The list of engine data is provided in the Appendix.

A primary design requirement with all conversions of existing gasoline engines, in the interest of minimizing capital expenditure, is to ensure that major components such as the cylinder block can be machined on the same transfer line as the gasoline engine and that as many parts as possible remain common to both engines.

A major advantage of indirect or divided combustion systems and the Ricardo Comet in particular is that the best performance can be achieved with peak pressures over the piston only 10–15 per cent above compression pressure. Retard of injection timing as a means of NO_x and noise control often results in the peak firing pressure not exceeding compression pressure. In these circumstances it is usually possible to use the gasoline lower end and running gear with only small changes in design or material.

Design changes were mainly in the cylinder head and piston, to accommodate the modified combustion system and the provision of a drive for the fuel injection pump.

3.1 Cylinder Block

The only change to the design of the cylinder block at this stage was the provision of additional metal in the tappet chamber, between the push rods to support the gasket immediately below the combustion chamber insert or hotplug. The 'four-bolt' head fastener pattern, common to most gasoline engines was retained and it was considered that the $^{7}/_{16}$ in bolts in a grade 8 material should enable a pre-load/gas load ratio of 3.5–5.0 to be attained. This would be adequate in Ricardo's experience.

3.2 Cylinder Head

A completely new design was prepared to accommodate the combustion system with vertical valves and with the uni-sided port arrangement necessary to retain the gasoline manifold layout.

Figure 1 is a section through the engine showing the details of the combustion chamber. The main dimensions of the chamber were computed using the Comet chamber design program. The under-square cylinder configuration resulted in chamber proportions of a more or less ideal form i.e. 50/50 division of volume at top dead centre (TDC) between the swirl chamber and that above the piston including piston cavities 3.56 mm (0.14 in) deep. The injector angle of 30° to the vertical was also within the preferred limits and the injector was located outside the rocker cover. The glow plug was located on the upstream side of the injector. Provision was made for an alternative downstream glow plug position to be tried during development.

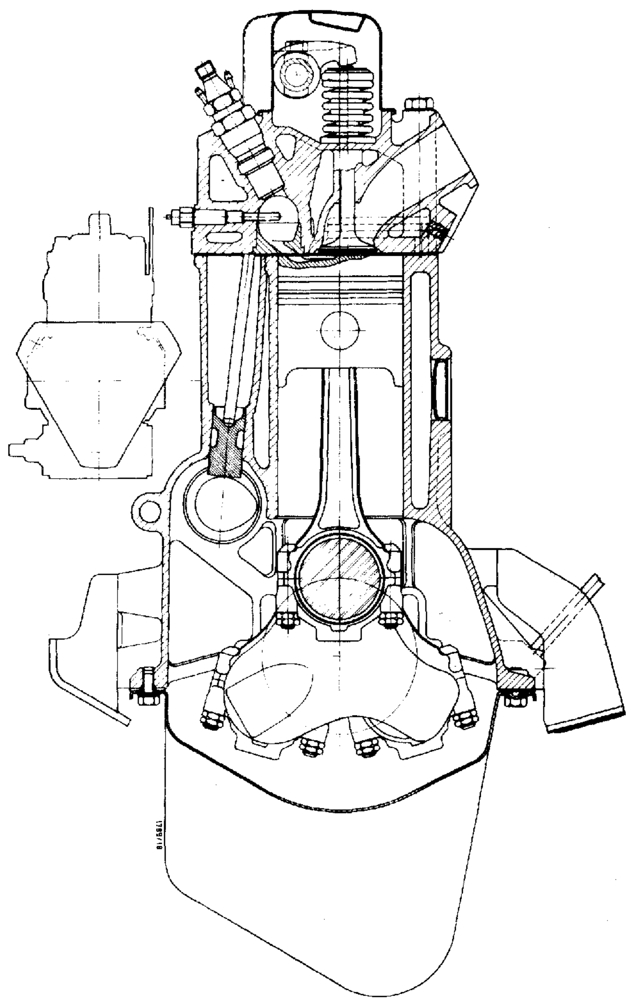

Fig. 1 Cross section 3.6 litre diesel engine showing combustion system

The cylinder head cooling system included a drilled passage between the valve seats. This feature has been found to be necessary to ensure adequate cooling in the area bounded by the valves and the combustion chamber hotplug where the highest thermal loading occurs with the Comet system (2).

The hotplug is also of classical form and is made from heat resistant material.

For light duty applications it is generally possible to use integral valve seats provided the head material is adequately hard. Separate inserts were not therefore included in this design although space was provided should it prove necessary to resort to inserts for some applications.

3.3 Piston

At an early stage in the development programme it was decided to fit expansion controlled pistons of European manufacture. This minimized the need for profile development as well as ensuring minimum piston clearance, and therefore noise, due to piston slap, when cool.

Fully floating gudgeon pins of 25.4 mm (1.0 in) diameter replaced the smaller pressed-in gasoline pins. This required some enlargement of the small end of the connecting rod to accommodate a bronze bush together with an oil collecting hole in the top of the rod.

No piston cooling was provided for the light duty application, although for truck duty and if turbocharged, this would be necessary.

3.4 Camshaft and valve train

The camshaft design was modified with new intake and exhaust cam profiles and timings giving the small overlap necessary to avoid the valves hitting the piston with the close diesel piston/head clearance. The valve timings are given in the Appendix.

Other modifications included new rocker arms, push rods and rocker shaft pedestals.

3.5 Fuel injection equipment

The fuel pump was a Roosamaster Type DB2 supplied by Stanadyne with cam form and plunger sizes to cover the range needed for development purposes. Alternative delivery and reverse flow damping valves were also available.

The injector design was that in use in a number of European engines, incorporating a zero angle pintle nozzle type DNOSD220 set to open at 125 bar and was as shown in Fig. 1.

3.6 Fuel pump drive

To facilitate early commencement of development work the fuel injection pump was provided with a toothed belt drive direct from the crankshaft nose, the pump being mounted on a casting bolted to the front end of the cylinder head. The pump was located on the camshaft and therefore combustion chamber side of the engine thus avoiding the need to carry the high pressure pipes over the rocker cover, as well as keeping the pipes to an acceptable length.

In parallel with the early development phase a more definitive fuel pump drive was designed for use on later development engines including those to be fitted in cars.

Several arrangements were considered, the two main constraints being a limit on the possible increase in length of the engine, from the vehicle installation point of view, and the need for good durability in respect of maintaining correct injection timing over a long period and therefore ensuring minimum change of gaseous emissions due to injection retard.

Spur gear drives tend to be unacceptably noisy, particularly at idle (3) whilst a toothed belt solution would have meant a major departure from the gasoline camshaft drive and would also have exceeded the space limitation.

A multiple chain arrangement as shown in Fig. 2 was therefore selected to ensure quiet operation whilst staying within the space constraint.

The relative positions of the crankshaft, camshaft and fuel injection pump precluded the use of a single chain drive. The only way of ensuring an adequately durable chain for the fuel pump was to drive it directly from the crankshaft using a 3/8 pitch duplex roller chain. The fuel pump drive was placed behind the existing camshaft chain in order to minimize the overhand on the fuel pump drive bearing. Retaining the existing camshaft chain gave the shortest engine as well as meeting the requirement for as much commonality as possible with the gasoline engine.

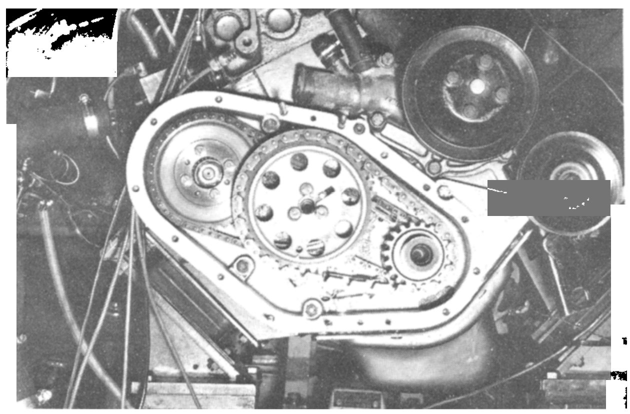

Fig. 2 Definitive timing drive using a Duplex chain for the fuel pump drive

Experience has shown that in a light duty diesel engine this type of chain drive will extend, due to wear, between 0.15 and 0.3 per cent per 1000 hours running. Simple calculations show that in this design a timing change of between 2.0 and 4.0 crankshaft degrees can be expected in 50000 miles or 0.5–1.0 crankshaft degrees per 12500 miles. This is considered to be small enough to hold the emission levels for 50000 miles with the permitted frequency of adjustment.

The pump drive sprocket is supported by its own bearing thus enabling the fuel injection pump to be removed without having to dismantle the timing drive.

3.7 Manifolds

Two types of inlet manifold were prepared for testing. One was of a fairly conventional rake or log form having a cylindrical plenum chamber and individual branches to each port. The other had a circular plenum chamber containing the air cleaner and long octopus type tubes connecting with the inlet ports.

The exhaust manifold was of single piece cast iron construction as for the gasoline engine. The hot spot was, of course, eliminated.

4 PERFORMANCE DEVELOPMENT

An initial calibration of the first prototype engine showed it to be lacking in both smoke limited power and fuel economy when compared with other well developed engines of a similar size. This performance is shown in curves A of Fig. 3.

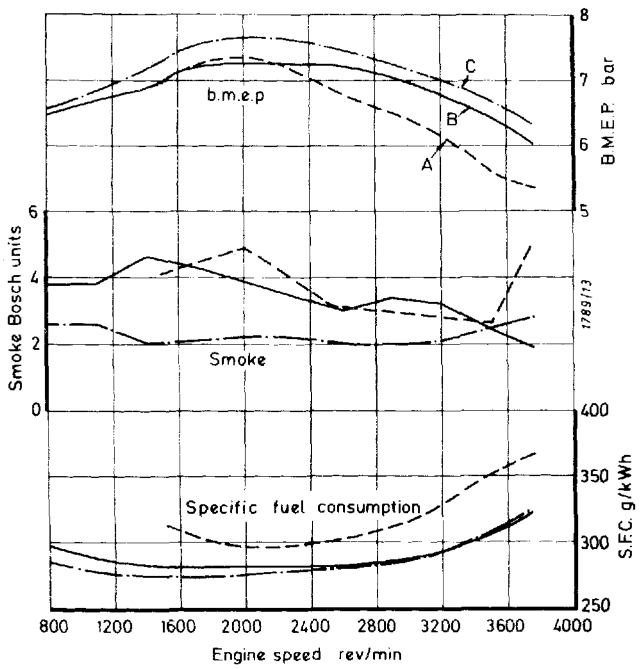

Fig. 3 Performance at various stages of development

4.1 Volumetric efficiency and friction

Checks of both breathing and friction showed these to be satisfactory as is seen in Fig. 4. The volumetric efficiency peaks at 87.5 per cent, the octopus type manifold giving the better result. The motoring friction was *circa* 0.6 bar lower than the prediction based on European light duty diesels (**4**). This latter feature is common to most diesel conversions of large American gasoline engines. With little scope for improvement in either breathing or friction, development was concentrated on the combustion system and fuel injection specification.

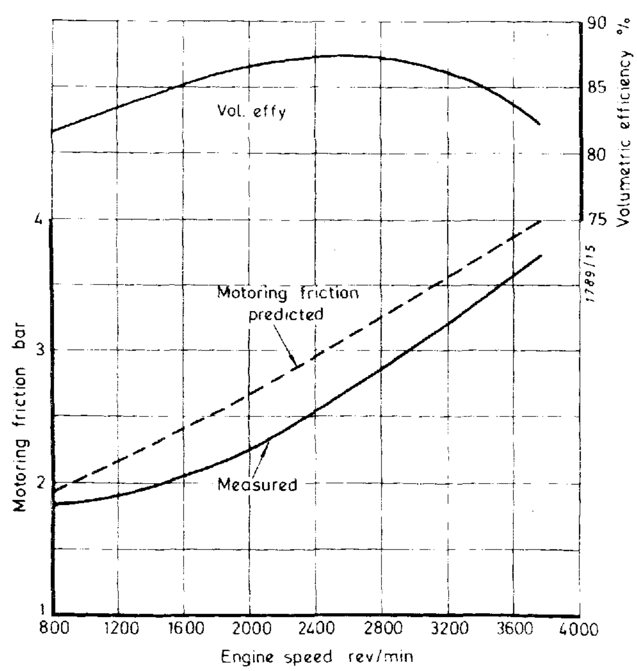

Fig. 4 Volumetric efficiency and motoring friction over the speed range

4.2 Chamber geometry

Examination of the chamber geometry revealed two features known to be detrimental to good combustion. The first was excessive piston/head or bump clearance. This results in too much of the air charge being inaccessible for mixing with the fuel during the early stages of combustion. This was corrected by machining the top deck of the cylinder block to bring the clearance as near to 1.0 per cent of the engine stroke as possible. The valve seats were then recessed further into the head to avoid the valves touching the pistons.

The chamber form in the vicinity of the injector heat shield included a lenticular space in front of the injector with a relatively small exit passage through which the fuel spray was directed. This is shown in scheme A of Fig. 5. Such cavities are thought to disturb the precise mixing of the fuel and the air in this region. As is shown in scheme B the chambers were modified by opening the exit hole to the maximum the heat shield would allow.

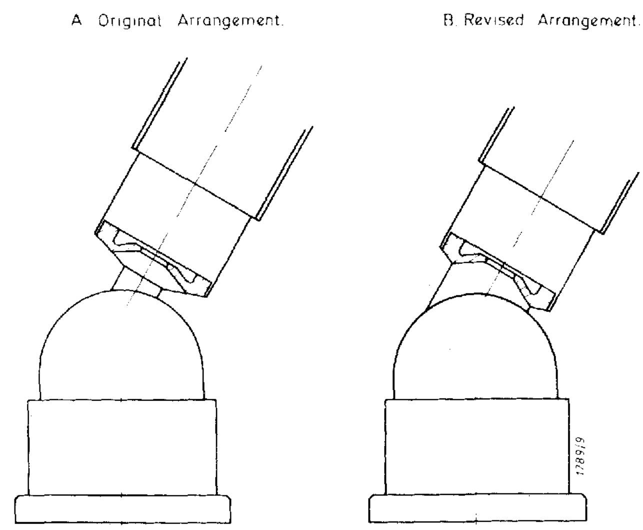

Fig. 5 Swirl chamber design and modification

These two modifications resulted in the performance shown in curves B of Fig. 3 where the high speed performance in particular has been very much improved.

Provision had been made for an alternative glow plug position to the horizontal one normally used. It was assumed that in the downstream position the plug would have less detrimental effect on combustion than in the upstream location. Back-to-back tests were run with the glow plug in the two positions indicated in Fig. 6. The results are shown in Fig. 7 where it is seen that the vertical or downstream position gave a superior smoke condition over the whole speed range with a better fuel economy at low speed when compared with the horizontal position.

It was decided, however, not to persist with this vertical plug position since it was thought that there

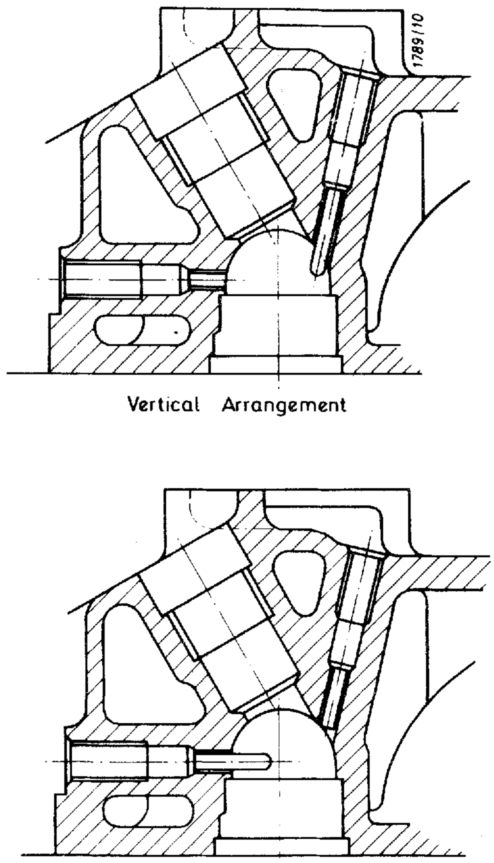

Fig. 6 Alternative glow plug locations

would be serious practical problems associated with locating the plugs within the rocker cover with regard to oil leakage. It was also suspected that a glow plug in this vertical location would be subjected to direct fuel impingement at high speeds resulting in overheating and consequent damage. In the normal location the

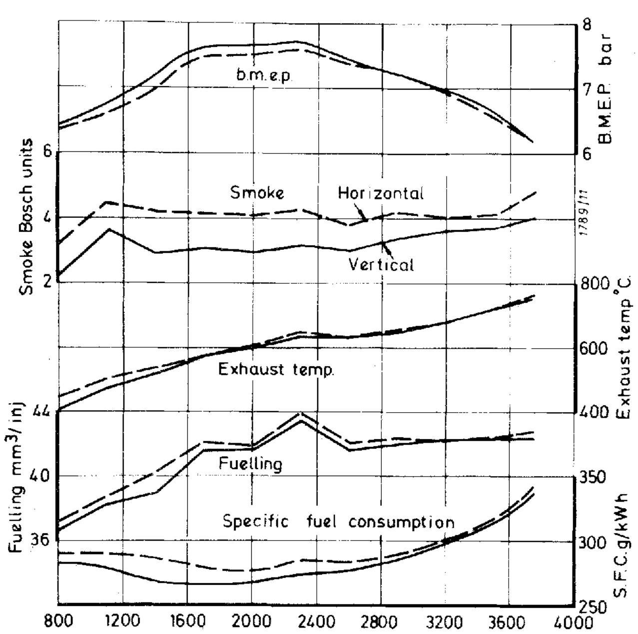

Fig. 7 Effect on performance of changing the glow plug location

spray is deflected away from the plug tip at all but cranking speeds.

4.3 Fuel injection specification

The remainder of the improvement of performance indicated by curves C of Fig. 3 comprising a reduction in smoke of up to two Bosch numbers and some further reduction in fuel consumption at low speed was accomplished by careful matching of the fuel injection equipment. Much of this improvement resulted from the elimination of after injections and achieving the optimum injection period and start of injection timing. The best performance shown in Fig. 3 was achieved from an engine fitted with the definitive chain timing drive which was considered also to have contributed to the good all round performance. Figure 8 shows that in this optimum performance build the engine achieves 74.6 kW (100 h.p.) and a peak torque of 230 Nm (170 lb ft) at 2000 rev/min and this is close to the rating of the single carburettor version of the gasoline engine.

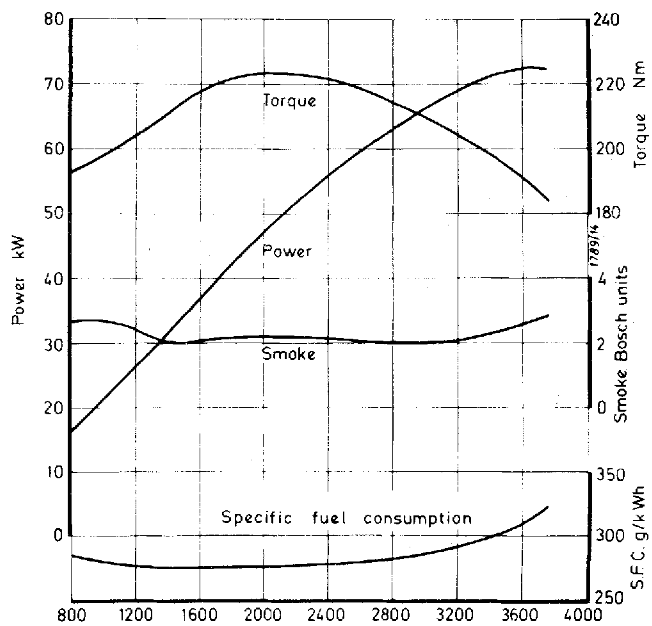

Fig. 8 Final performance level achieved

5 MECHANICAL DEVELOPMENT

In parallel with the performance development a second and third engine were engaged in proving and improving the mechanical viability of the engine in diesel form. This part of the programme comprised a series of 100 hour full load/full speed durability tests aimed at identifying weak points in the design or construction when subjected to mechanical and thermal loads inflicted by the diesel system. These tests culminated in a 1000 hour durability test using a load and speed cycle considered to represent that experienced in light duty applications.

Several problems were encountered, as might be expected, when attempting to use as many gasoline parts as possible. The more significant occurrences are included in this paper.

5.1 Crankcase

The engine has a four bearing crankshaft which would not normally be considered ideal for diesel cylinder

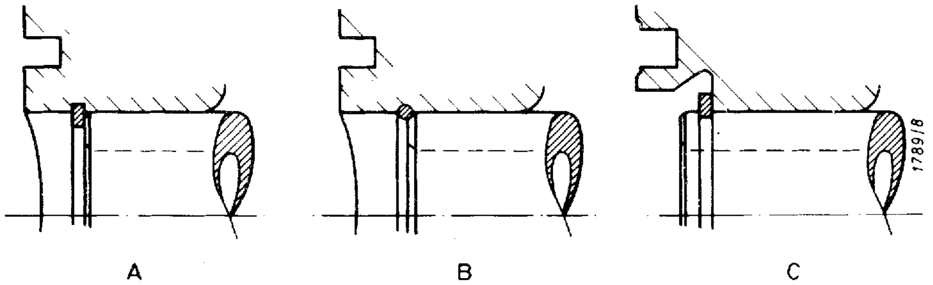

Fig. 9 Alternative wrist pin retainers

pressures in the region of 60–80 bar over the piston. It was therefore not surprising that the front main bearing bulkhead developed cracks near the main bearing housing. This was remedied by the addition of metal in areas which did not affect its commonality with the gasoline block.

5.2 Pistons

A persistent problem with the pistons during the early stages involved the wrist pin circlips or snap rings being worn badly and driven out of their grooves in a period of a few hours running at full load. The worst instance resulted in the fully floating gudgeon pin wearing its way into the cylinder bore and wrecking the engine.

Careful examination revealed a pattern of damage in which one snap ring in each piston appeared to be subjected to most of the hammering and indicated that the four bearing crank was probably again the culprit. Bending of the crank between the main bearings would cause the small ends of adjacent pairs of connecting rods to nod towards each other thus urging the gudgeon pins towards one snap ring. Static deflection measurements were rather inconclusive showing less crank deflection than had been anticipated. Nevertheless a solution had to be found. Among those tried were heavier section and deeper seated circlips of the type shown in Fig. 9a and light alloy buttons to locate the wrist pins against the cylinder wall in two-stroke fashion. Neither was successful. The two successful solutions were provided by the piston suppliers Mahle and Karl Schmidt. The former replaced the conventional circlips with round wire snap rings in semi-circular grooves which were used in conjunction with a small chamfer on the gudgeon pin as shown in arrangement B of Fig. 9. The other alternative supplied by Karl Schmidt was to fit the circlips in grooves in the wrist pin on the outside of the piston as in Fig 9c. This allowed more generous contact areas both on the piston and in the pin. Both arrangements completed the 100 hour test satisfactorily.

Tests were carried out to determine the full load temperatures of the pistons. These tests involved the use of eutectic fusible plugs in drilled holes in the crown and down the skirt on all four sides of the piston. The missing and remaining plugs after a carefully controlled run at full load indicated the temperature within fairly close limits and allowed the chart shown in Fig. 10 to be drawn. This indicates a centre crown temperature of 350–360°C and top ring groove temperatures of 230–260°C, the highest of the latter being on the combustion chamber side. The temperatures are within acceptable limits, avoiding the need for oil cooling. However, any increase in rating, as by turbocharging would exceed the limits and require some form of oil cooling.

5.3 Cylinder heads

As mentioned earlier, the heads were not provided with seat inserts but it was found necessary to induction harden the material of the seats to prevent rapid wear.

The cooling in the centre of the heads proved entirely satisfactory, there being no incidence of cracking of the valve bridge even after the 1000 hour cyclic test.

5.4 1000 hour durability test

As a final demonstration of the viability of the design, one engine was set to run for 1000 hours operating on the following thirty minute cycle:

Mode	Time, min
Start	0
Low idle	2
Full power	1
Part load/half speed	2
Full power	1

After each four six-minute cycles as above, the engine was idled for one minute and stopped for five.

The lubricating oil (Rimula 20/20W) was checked for viscosity at regular intervals and changed when it had increased by 20–25 per cent. This resulted in changes at approximately 100 hour intervals or the equivalent of

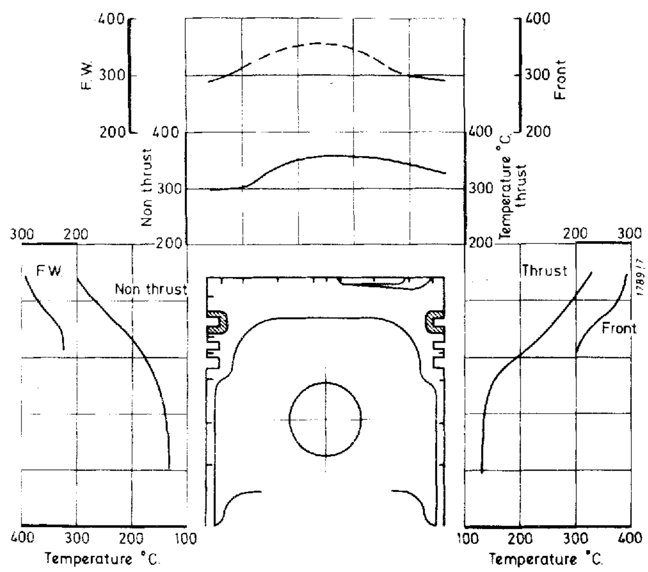

Fig. 10 Piston temperatures at full load/full speed

every 3000 miles albeit with the system volume on the test bed greater than in a vehicle.

The performance was checked before and after the 1000 hours. A performance check was also made after correcting the injection timing. To compensate for 1.5–2.0° retard, due partly to chain stretch. These checks showed that the performance deterioration was not significant. The main effect stemmed from a reduction in fuel delivery of some 2.5 mm per injection.

Referring to the log sheet in Fig. 11 the main features are a serious loss of performance at point 1 at 74 hours due to a blocked air filter and again at point 2 at 441 hours. At 990 hours (point 3) the injection timing was corrected for the final performance check.

On inspection at the end of the 1000 hours all the major components were in satisfactory condition. Apart from some minor trouble spots such as rocker pad wear and exhaust manifold warping the engine could clearly have continued for at least a further 1000 hours.

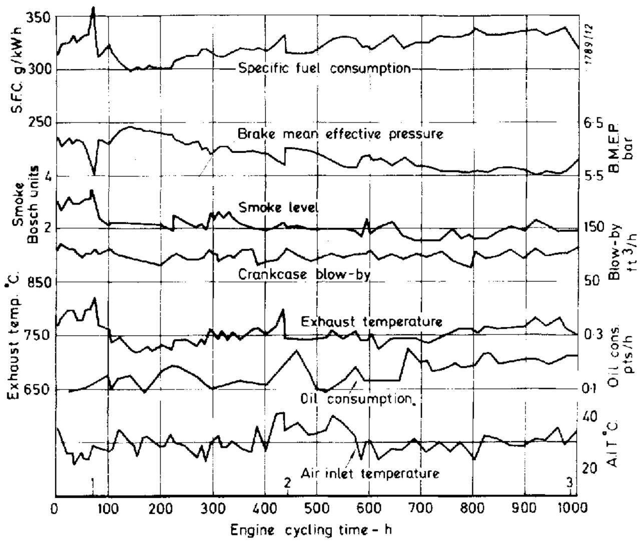

Fig. 11 1000 hour durability test log sheet

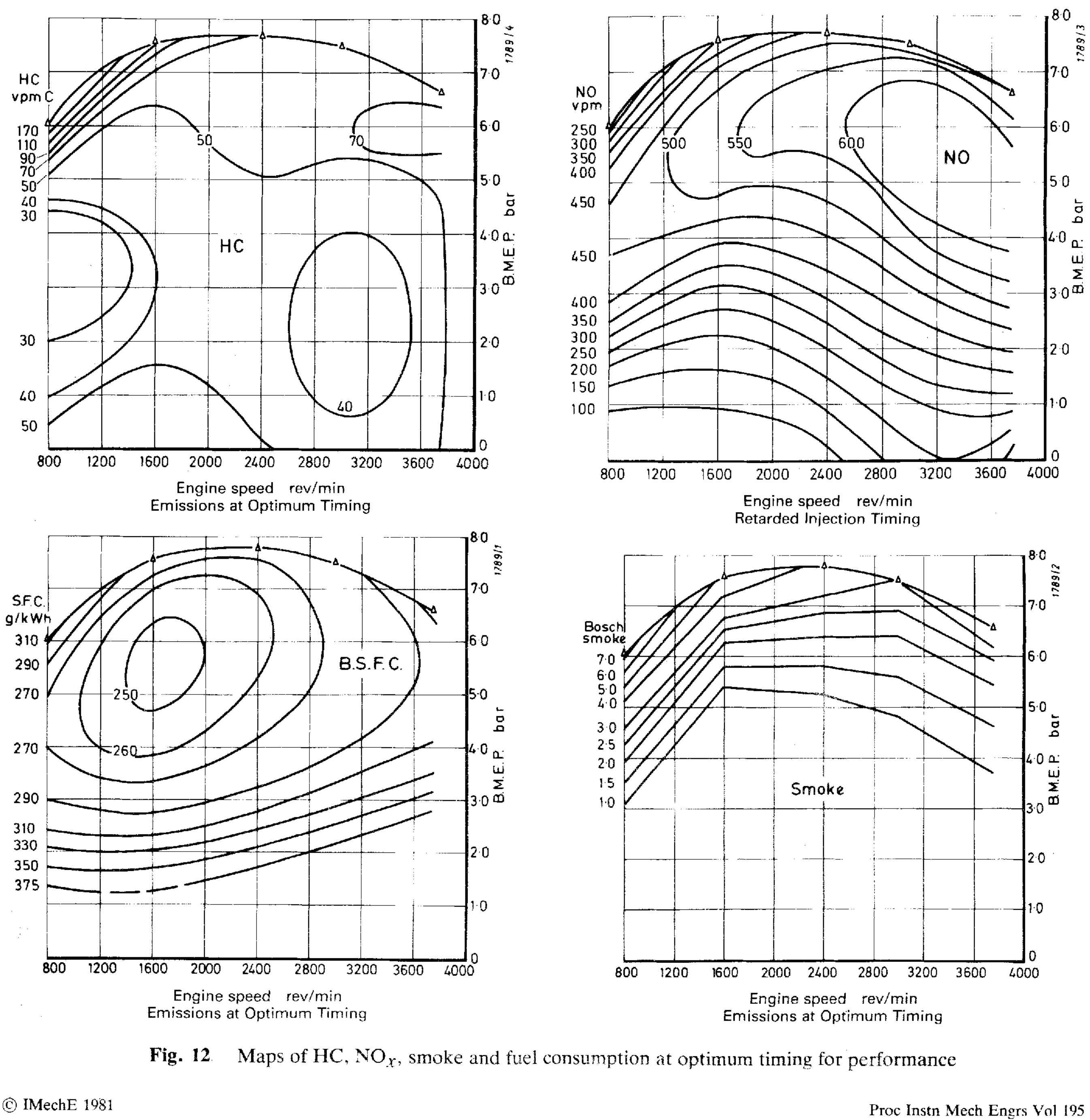

Fig. 12 Maps of HC, NO_x, smoke and fuel consumption at optimum timing for performance

6 GASEOUS EMISSION REDUCTION

Having eliminated secondary injections and established the optimum injection specification in terms of rate, timing, etc, the response of the engine to retard of injection timing as a means of NO_x control was investigated.

In optimum performance build, the engine was mapped for steady state unburned hydrocarbon (HC) oxides of nitrogen (NO_x), brake specific fuel consumption (BSFC) and smoke. Figure 12 shows these maps plotted against load and speed.

Hydrocarbon levels remained below 50 v.p.m. C over that part of the map that is pertinent to the US Federal Test Procedure. The NO peaked, typically around 75 per cent load, the highest concentrations being 600 v.p.m. towards full speed.

Smoke was generally below Bosch 3.0, the highest value of 7.0 being recorded at the lowest speed which was of little significance with an automatic transmission.

The minimum BSFC was 250 g/kW h rising to about 280 at full load in the mid speed range.

The effect of retard of timing on the trade-off between HC and NO is shown in Fig. 13. The initial

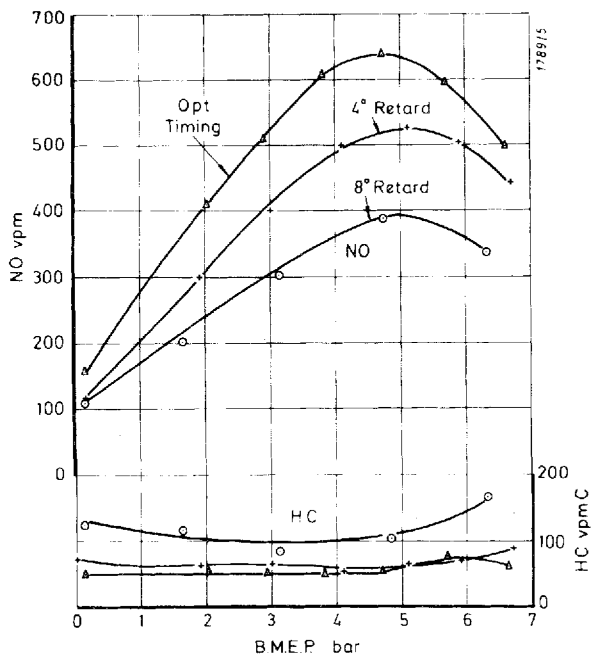

Fig. 13 Effect of timing retard on gaseous emissions at full speed

retard of 4° crankshaft reduced NO by some 20 per cent generally for very little increase in HC. A further retard of 4° however resulted in doubling of HC for a further reduction of 150 v.p.m. NO. There was clearly a 'best' timing for minimum combined HC and NO_x.

Full emission maps were prepared for all three injection timings and the HC and NO data at the 8° retard condition is presented in Fig. 14 for comparison with the data in Fig. 12.

The retard of timing was accompanied by some deterioration of BSFC but an improvement in smoke of one Bosch number which is a typical Comet system characteristic.

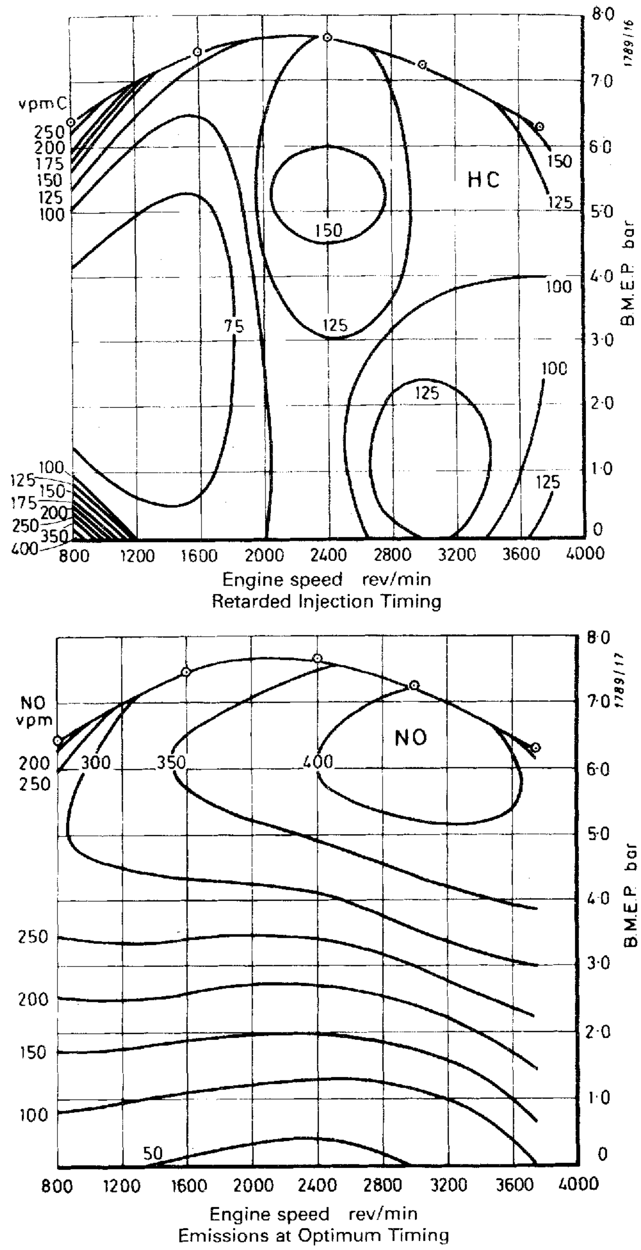

Fig. 14 HC and NO_x maps at the most retarded injection timing

The final selection of the best injection timing for low emissions was left, to be established in the vehicle.

7 VEHICLE TESTS

Two vehicles were provided for conversion to diesel in order to demonstrate the acceptability of the engine and establish the performance, fuel economy and gaseous emissions. The engines were accommodated with only minor resiting of some components to clear the fuel injection pump, etc. The specification data for the Plymouth Volare which was used for most of the testing are given in the Appendix.

After some settling-in adjustments and checking of the various systems the car was committed to a range of road and chassis dynamometer tests. Injection timing was varied, as was the rear axle ratio. The resulting data were recorded and the best compared with the gasoline vehicle before conversion:

	Gasoline	Diesel
Performance		
Acceleration 0–60	18.5	20.5 s
Gaseous emissions		
HC	2.59	0.21 g/mile
CO	33.80	0.90 g/mile
NO_x	2.48	1.60 g/mile
Fuel economy		
Urban cycle	15.0	25.5 miles/gal (US)
Highway cycle	20.6	33.8 miles/gal (US)
Composite	17.6	28.7 miles/gal (US)

7.1 Noise

Drive-by noise averaged 2 dBA above the gasoline vehicle whilst internally the diesel engine was 1–3 dBA noisier than with the gasoline engine.

7.2 Cold starting

Starting with normal glow plugs was found to be satisfactory down to 0°F.

7.3 Passenger comfort

It was recognized that, in common with all diesel passenger cars, the low heat rejection at part load would result in slower warm-up and therefore a less effective heater on short journeys. A possible solution to this problem is to apply exhaust back-pressure which would load the engine and also ensure that most of the increased energy would find its way into the coolant rather than down the exhaust pipe.

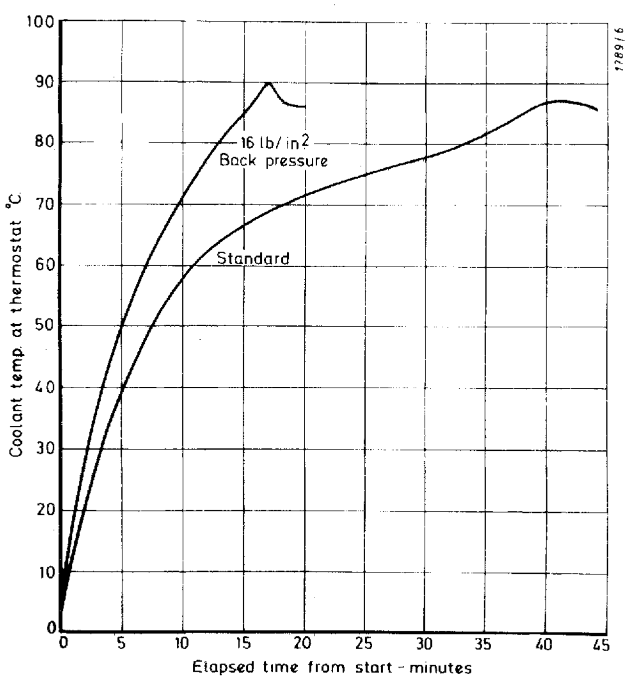

Fig. 15 Effect of exhaust back-pressure on warm-up rate

Figure 15 shows the result of this experiment. The application of approximately one bar back-pressure resulted in the coolant reaching 'thermostat open' temperature in seventeen minutes of idling instead of forty minutes. The car heater was set to warm.

The application of this level of back-pressure also improves cold starting by 5–6°C provided the starter can carry the extra load.

8 CONCLUSIONS

This exercise has demonstrated that a very acceptable passenger car diesel engine can be developed from a typical US gasoline engine with minimum sacrifice of acceptability criteria for the sake of fuel economy.

In particular it has been shown that specific performance similar to that expected from European light duty diesels and equal to US gasoline engines can be achieved without durability limitations whilst retaining a high degree of commonality with the original gasoline engine.

Gaseous emission levels close to the 0.41, 3.4, 1.5 emission control requirement can be achieved from a 4000 lb vehicle without recourse to exhaust gas recirculation, making the eventual achievement of 1.0 g/mile NO_x that much easier.

In areas where the diesel is still falling short in acceptability, such as in car heating, means exist for further improvement.

ACKNOWLEDGEMENTS

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APPENDIX

Engine data

Bore and stroke	86.4 × 104.8 mm (3.401 in × 4.125 in)
Number of cylinders	6
Swept volume	3.69L 225 CID
Combustion system	Comet Mk Vb
Compression ratio	21.5
Valve timings	I.O. 2° BTDC
	I.C. 37° ABDC
	E.O. 52° BBDC
	E.C. 4° ATDC
Fuel injection pump	Roosamaster type DB2
Injector	Bosch KCA type fitted with DNOSD220 nozzle

Vehicle data

Model	Plymouth Volare four-door sedan
Year	1977
Weight	1720 kg (3788 lb)
Transmission	Three-speed automatic with lock-up torque converter
Rear axle ratio	2.76
N/V	37.6 (26.6 m/h per 1000 rev/min)
Inertia class	4000 lb

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