Paper 4

THE EFFECT OF COMBUSTION CHAMBER SHAPE AND OTHER ENGINE DESIGN FACTORS ON EXHAUST EMISSIONS

By P. L. Dartnell*, C. L. Goodacre* and P. V. Lamarque*

A Heron combustion chamber engine of 2 litre capacity has been utilized to investigate the effect of combustion chamber shape, increased mixture movement, valve timing, mixture formation, and reaction in the exhaust system on engine performance and level of exhaust emissions using the seven-mode U.S. Federal cycle.

Such factors as carburettor weakening and limitation of intake manifold vacuum during overrun have been included in this investigation, and it has been shown that it is possible to reduce exhaust emissions and also satisfy the current U.S. requirements with an engine giving acceptable performance, improved economy, and unaffected reliability.

Much of the information reported may be negative in terms of improvement to exhaust emissions by detailed engine design. Nevertheless, some positive conclusions have been reached as a result of this work, and it is hoped that this will draw forth more informed discussion than the authors have been able to assemble from the work attempted with one basic engine.

INTRODUCTION

A STANDARD PRODUCTION ENGINE having the Heron form of combustion chamber, i.e. essentially a flat cylinder head with the combustion chamber being formed in the piston crown, was used for this investigation.

A report of the early stages of this work (1)† dealt with the comparison of this form of combustion chamber to other forms, the effect on engine performance of increasing compression ratio and altering combustion chamber bowl shape and, in a limited manner, the effect of certain of these aspects on exhaust emissions of unburnt hydrocarbons and carbon monoxide.

The present work relates to a more detailed study of the effect of combustion chamber shape, increased mixture movement, valve timing, mixture formation, and reaction in the exhaust system on engine performance and level of exhaust emissions.

For the earlier work, a single-carburettor engine was used; for much of the later work the engine was equipped with two carburettors, and exhaust emissions were examined using the seven-mode U.S. Federal cycle. Such factors as carburettor weakening and limitation of intake manifold vacuum during overrun have been included to produce an engine which will satisfy the current U.S. exhaust emission requirements while giving acceptable performance, improved economy, and unaffected reliability.

TEST PROCEDURE AND EQUIPMENT

A four-cylinder, liquid-cooled engine of 3·375 in (85·6 mm) bore and 3·375 in (85·6 mm) stroke was used throughout, and is shown in section in Fig. 4.1. This was basically a standard production engine of 2 litre capacity, having the Heron form of combustion chamber. For all tests the engine was coupled to an eddy current motoring-absorbing dynamometer. Some of the work relates to the engine in single carburettor form (nominal size 1·75 in, 44·5 mm), and a large proportion of the results relate to the engine fitted with twin carburettors (nominal size 2·0 in, 50·8 mm).

Full and part throttle (where applicable) power and economy data were obtained at progressive points in the speed range in the conventional manner by the adjustment of the air/fuel ratio from over-rich to lean, with the spark timing adjusted to minimum advance for best torque; alternatively, the spark timing was set for maximum 'knock-free' power. The gasoline used throughout the tests comprised a normally marketed Super Premium of 103 octane number Research, 93 octane number Motor.

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† *References are given in Appendix 4.1.*

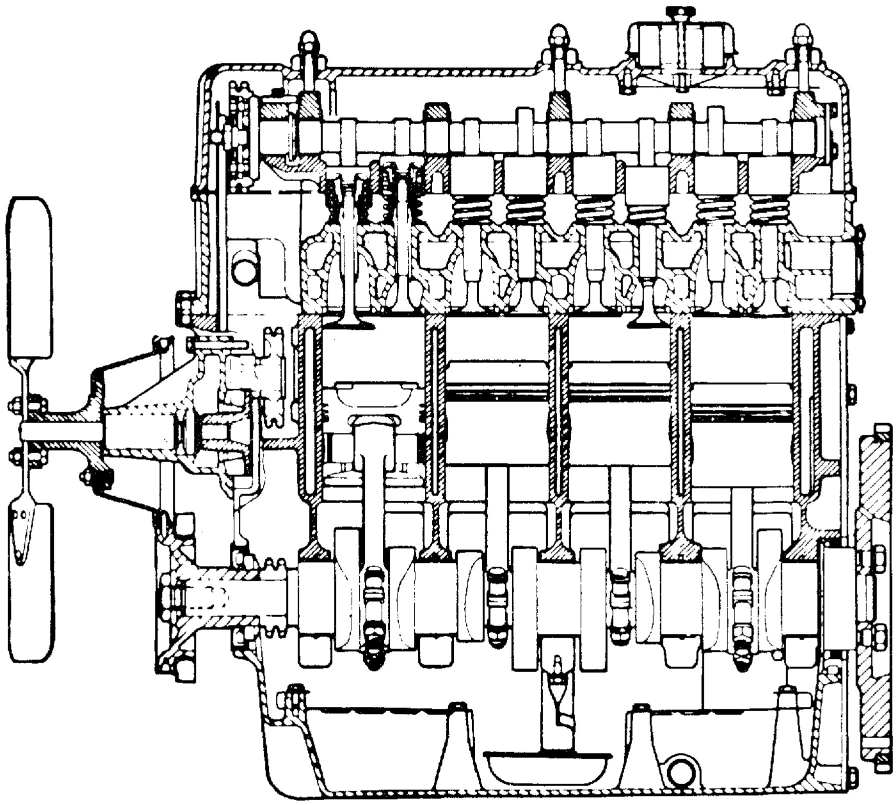


Fig. 4.1. 2-litre engine with Heron combustion chamber

The steady-state conditions used for exhaust emission tests with the single-carburetted engine are shown in Table 4.1.

Continuous analysis of hydrocarbon exhaust emissions was made using a Grubb–Parsons non-dispersive infrared gas analyser sensitized with *n*-hexane. Carbon monoxide emissions were measured on grab samples collected in p.v.c. plastic bags and analysed on equipment comprising a thermistor (thermal conductivity) detector and a 5A molecular sieve chromatographic column which enabled the carbon monoxide to be separated from the hydrocarbons, oxygen, nitrogen, and carbon dioxide.

Hydrocarbon repeatability, ± 15 p.p.m. *n*-hexane
Carbon monoxide repeatability, $\pm 0{\cdot}5$ per cent volume

An additional technique was used for measuring the exhaust gas emissions during the work with the twin-carburetted engine. This involved the use of a dynamometer system which provided inertia characteristics closely resembling those of the vehicle. Such characteristics were employed to simulate the seven-mode U.S. Federal or Californian cycle. Hot cycles were run with the engine direct-coupled to the dynamometer (without gearbox), and p.v.c. plastic bag samples were collected from individual modes—the unburnt hydrocarbon and carbon monoxide emissions being subsequently determined by non-dispersive infrared gas analysers. Oxygen content was determined with a Servomex Type OA/150 paramagnetic analyser. Composite hot cycle values for emissions were then determined by introducing the appropriate weighting factors to the seven modes.

Table 4.1. Road-load conditions

Road-load setting, b.h.p.	Road speed, mile/h	Engine rev/min
3·4	20	1000
6·25	30	1500
10·18	40	2000
15·3	50	2500

DISCUSSION OF RESULTS

Effect of direction swirl produced during the intake stroke

Examination of increased mixture movement in the combustion chamber was made using the method of promoting directional swirl during the intake stroke by means of shrouded valves. A 120° shroud was selected and positioned to symmetrically face the axis of the intake port. This increased the rate of swirl from almost negligible values of 150 rev/min to well in excess of 4000 rev/min at valve lifts near the maximum, which is coincident with the period when the piston is at mid-stroke and air flow into the cylinder is at maximum. No attempt has been made to determine mean swirl speed at the end of the induction stroke, but it is accepted that, during compression, swirl

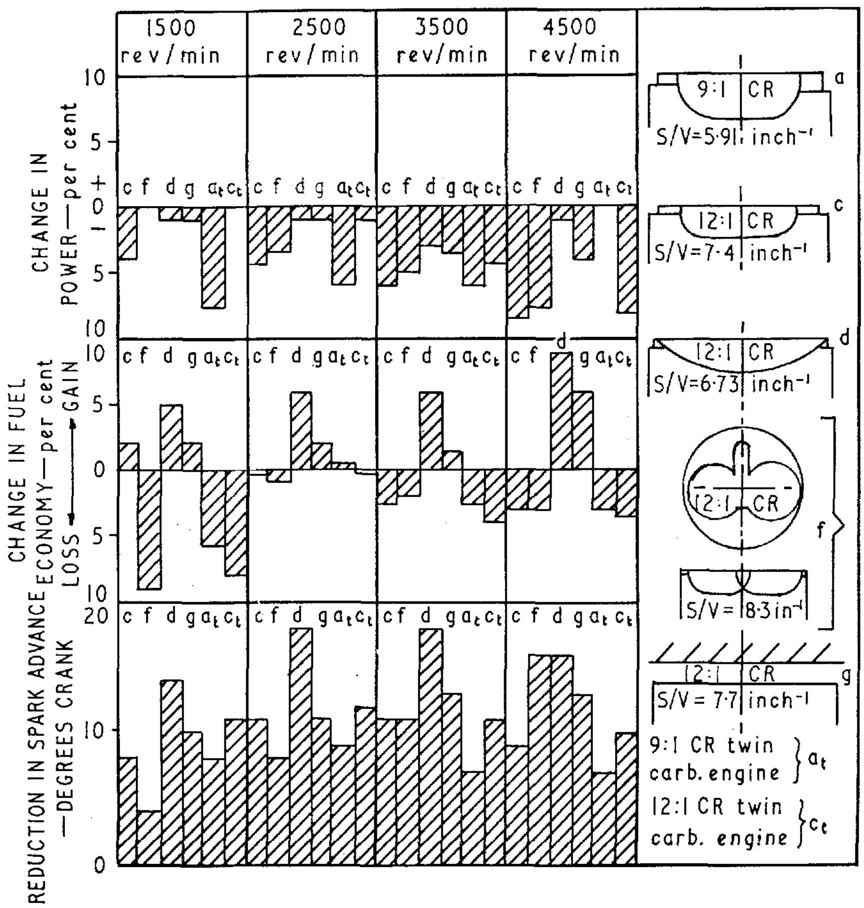


Fig. 4.2. Full throttle performance comparison for various combustion chamber forms with shrouded inlet valves compared to similar combustion chamber base-line forms without shrouds

is accelerated as the charge is forced into the smaller diameter combustion chamber (**2**), which would apply with forms 'a' and 'c' shown in Fig. 4.2.

The effects of directional swirl for a number of combustion chamber configurations (most of them employed with a single-carburetted engine) are compared with identical forms without the use of shrouds to the inlet valves; these are shown in Fig. 4.2. The results relate to full throttle operation.

As a result of the lowered volumetric efficiency, power loss with intake valve shrouds occurred at most speeds with all the forms of chamber shape and intake configuration. By using intake valve shrouds, the specific fuel consumption was worse at most speeds with chambers which had maximum 'squish'. Directional intake swirl with chambers 'd' and 'g', which were both 'quiescent' configurations, showed a significant improvement in fuel economy together with a minimum of power loss, form 'd' being the better at all engine speeds. Since there was no significant change in average exhaust gas temperature at full throttle it is suggested that the improvement was probably caused by a reduction in cyclic dispersion resulting from increased turbulence and mixture movement (**3**).

Spark advance for best torque was retarded significantly with intake valve shrouds for all chamber forms, but more particularly with the quiescent form 'd', suggesting an increase in the rate of burning.

The effect of intake valve shrouds at part throttle is shown in Fig. 4.3. The settings chosen represent road-load conditions from 1000 to 4000 rev/min; it was found that fuel economy was significantly improved for all combustion chamber forms considered and at most engine speeds with directional intake swirl. Maximum improvement occurred below 2000 rev/min, and, as in the case of the full load results, chamber form 'd' showed the maximum gain. A reduction in spark advance for best torque was evident for all the chamber configurations, amounting to an average value of 15° crankshaft up to and including engine speeds of 2500 rev/min. There was indication of a 5–10 per cent reduction in exhaust gas temperature at part load conditions with the shrouded intake valves.

The results shown in Fig. 4.4, which were obtained under steady-speed road-load conditions, suggested that the use of shrouded intake valves, when fitted to a single-carburetted engine at 12/1 compression ratio, provided a reduction in unburnt exhaust hydrocarbon emissions,

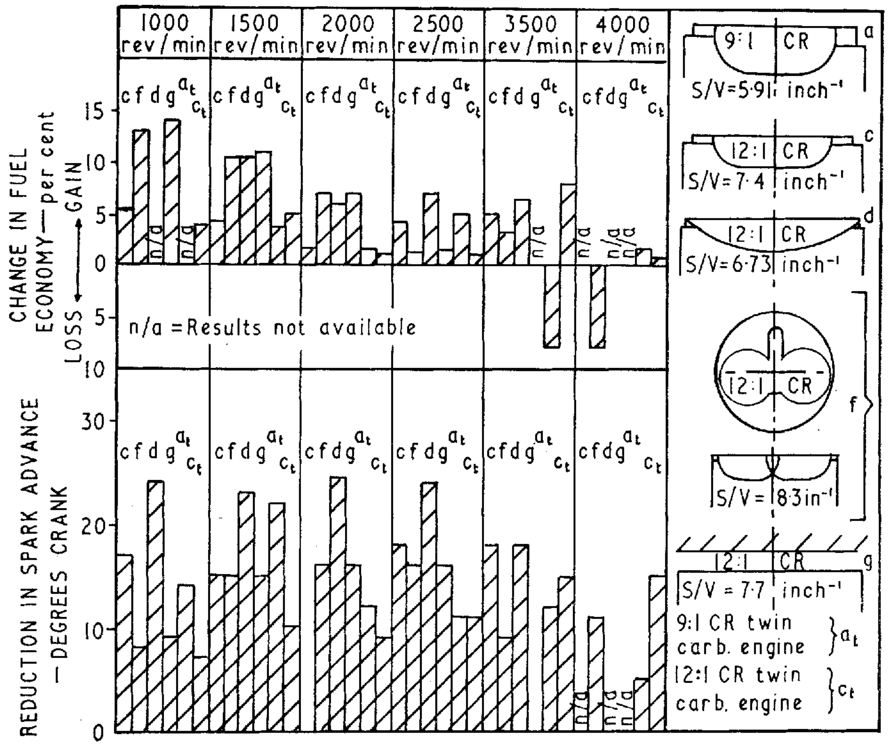


Fig. 4.3. Road-load performance comparison for various combustion chamber forms with shrouded inlet valves compared to similar combustion chamber base-line forms without shrouds

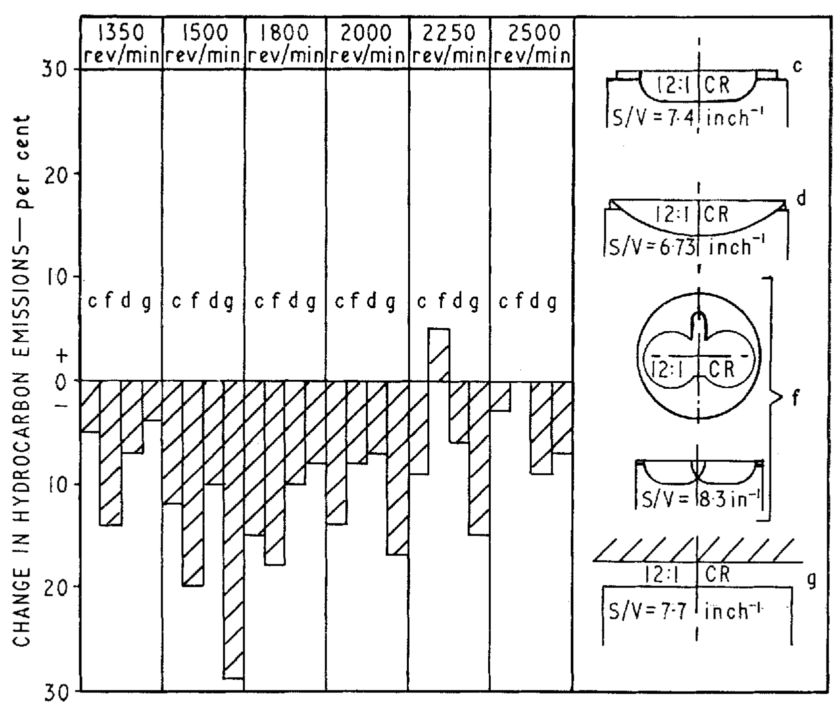


Fig. 4.4. Change in hydrocarbon emissions—steady-speed road load for various combustion chamber forms with shrouded inlet valves—compared to similar combustion chamber base-line forms without shrouds

Table 4.2. Weighted average hydrogens with and without shrouded intake valves

Combustion chamber form	Composite values of hydrocarbon emissions—seven-mode cycle	
	With shrouds	Without shrouds
a_t (9/1 compression ratio) . .	647	660
c_t (12/1 compression ratio—engine 'de-toxed')	411	390

amounting to an average reduction of more than 10 per cent at 1500, 1800, and 2000 rev/min. When exhaust emissions were obtained under the transient conditions of the U.S. Federal seven-mode cycle, with the twin-carburetted engine at 9/1 and 12/1 compression ratio, the composite hydrocarbon emission values shown in Table 4.2 indicated that, with the chamber configurations tested (forms 'a_t' and 'c_t'), no reduction in hydrocarbon emissions resulted from the use of increased directional intake swirl.

This result was disappointing since it did not support the trend obtained under steady-speed conditions for the engine in single-carburetted form, and failed to provide positive support for the theory that the greater turbulence in the combustion chamber should lead to a reduction of the non-combustible strata close to the wall (4). Nevertheless, such an effect could have been obscured in terms of a discernible reduction in unburnt hydrocarbons as measured downstream of the engine exhaust manifold, owing to the lowered exhaust gas temperature which accompanied increased turbulence at part load.

It should be emphasized that, for comparison purposes, similar air/fuel ratios were used for each test with and without shrouded intake valves. It is evident that examination of the amount by which the engine working range could be extended in the weak direction by the use of greater turbulence arising from intake swirl, would be worthy of further investigation.

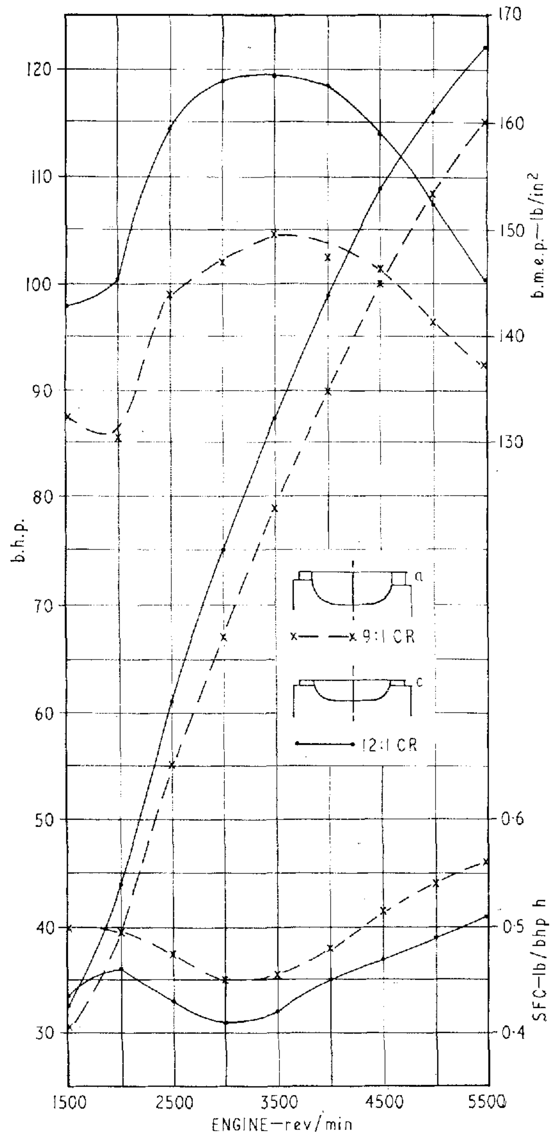


Fig. 4.5. Full throttle performance of twin-carburetted engine

Full throttle performance comparison for twin-carburetted engine at 9/1 and 12/1 compression ratios

Performance of the engine in twin-carburetted form was established at 9/1 and 12/1 compression ratios, the chamber form in the piston being concentric as for the standard engine, but with shallower valve recesses and reduced depth of chamber at 12/1 compression ratio. Fig. 4.5 shows the full throttle performance comparison, and by increasing compression ratio to 12/1 a gain of approximately 10 per cent in engine performance and economy was possible over a wide band of engine speed range. In addition, compared to the engine equipped with a single carburettor, power had increased at both 9/1 and 12/1 compression ratios by at least 7 per cent over the mid-speed range and by an average of 15 per cent at higher engine speeds.

Base-line exhaust emission values over the U.S. Federal seven-mode cycle at 9/1 and 12/1 compression ratios

Unburnt hydrocarbon and carbon monoxide exhaust emissions were obtained for the seven-mode cycle, as described under 'Test procedure and equipment'. These tests were made with the engine at 9/1 compression ratio, with symmetrical chamber of reduced depth, and with the shallower valve recesses in the piston crown. Fig. 4.6 shows the contribution of the individual modes for unburnt hydrocarbons and carbon monoxide plotted against a base of percent volume for the complete cycle. The weighted average for these hot cycles is included and compared to the Californian and U.S. Federal limiting composite values (cold and hot cycles) for this size of

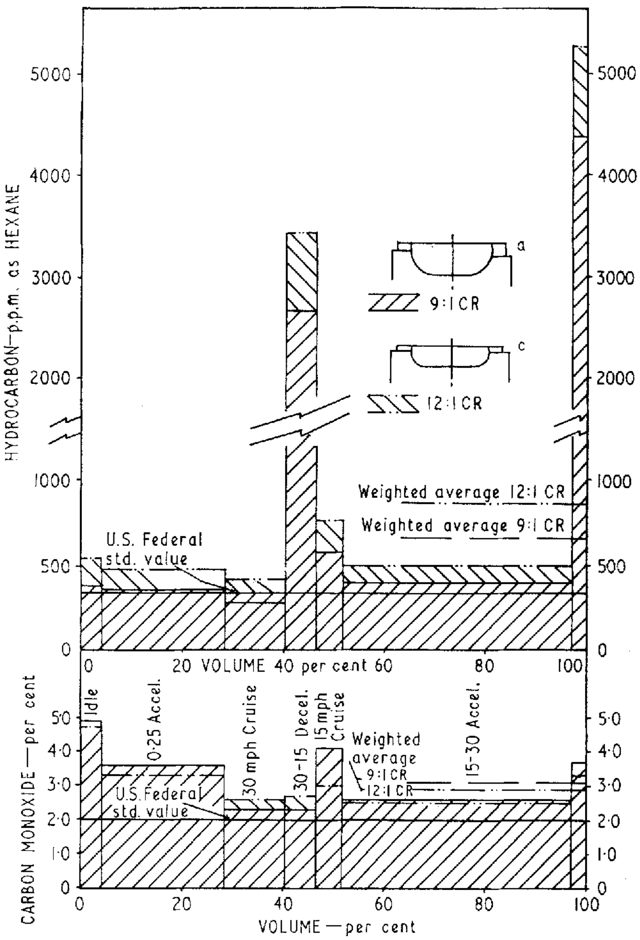


Fig. 4.6. Contribution of modes with engine in unmodified form

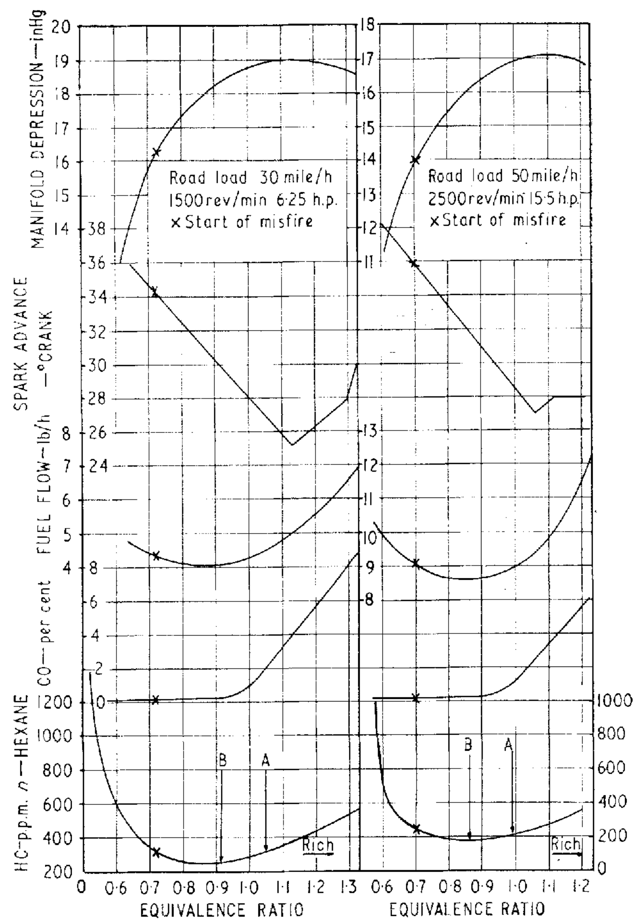


Fig. 4.7. Effect of mixture strength on performance and exhaust emissions at part load

engine of 350 p.p.m. hydrocarbons as hexane and 2·0 per cent carbon monoxide (CO). In the standard form (9/1 compression ratio) the engine exhaust emissions were considerably in excess of these limiting values, the weighted average being 660 p.p.m. hydrocarbons and 3·1% CO. With the compression ratio increased to 12/1, exhaust emissions increased during all modes, and the weighted average values were 830 p.p.m. hydrocarbons and 2·9% CO. Other investigators have shown that increased compression ratio results in increased unburnt hydrocarbon emissions (**5**).

Effect of weaker mixture setting on exhaust emissions

The effect of varying mixture strength was examined at engine speeds of 1500 and 2500 rev/min, maintaining a constant load (road load) over the mixture range at each speed. The results are shown (Fig. 4.7) plotted against a base of mixture strength or 'equivalence ratio', expressed as the percent fuel weak or rich of the stoichiometric condition—stoichiometric being taken as 100 per cent. The terms 'weak' and 'rich' express, respectively, the deficiency and the excess of the fuel in a given quantity of air entering the engine, by comparison with the stoichiometric quantity of fuel. The air/fuel ratio was determined from the measured quantity of burnt mixture, including hydrocarbon content, using an equation evolved by Spindt (**6**). Account of hydrocarbon measurement by infrared analysers expressed as *n*-hexane equivalent was taken as described by Robison *et al.* (**7**). It will be noted that spark advance for best torque over the range 90 per cent weak to 130 per cent rich remained effectively constant. The weak limit of each loop represented the conditions where engine misfire and surge increased to the point where any further weakening resulted in completely unstable running. However, it was possible to exceed the 80 per cent weak position and still maintain stable operation without misfire, and on this basis the carburettors were adjusted at part load to the fuel delivery at 'B' compared to the original setting shown at 'A' in Fig. 4.7. At the revised setting, unburnt hydrocarbons and carbon monoxide were both at a minimum. It is interesting to note that the carbon monoxide concentration at the chemically correct mixture ratio was approximately 1 per cent or 20 times greater than the theoretical quantity (**8**).

Effect of various devices to reduce exhaust emissions during overrun

Two methods of limiting intake manifold vacuum during the overrun periods were tried, and both have been referred to by Dietrich (**9**).

One was to permit air to enter the intake manifold downstream of the carburettor by use of an air valve which was energized by manifold vacuum and operated at the vacuum prevailing during overrun conditions. The use of this valve effectively reduced the intake manifold vacuum, and the hydrocarbon emissions during the overrun modes were reduced by 20–30 per cent. However, it was evident that, although the beginning of the overrun cycle was probably receiving a more combustible mixture by permitting extra air into the manifold, part of the cycle was receiving a weaker and even more incombustible mixture than without the valve fitted.

The simplest and most effective method of vacuum limiting device was found to be with a spring-loaded poppet valve, situated in the throttle plate and energized by manifold depression. By this means, manifold vacuum was reduced, but in addition a combustible air–fuel mixture was supplied during overrun which led to a reduction in unburnt hydrocarbons of up to 90 per cent.

By the use of the weaker mixture setting described previously, the limitation of manifold vacuum during overrun employing a poppet valve in the throttle plate, and a retarded spark at idle (6° a.t.d.c.) to provide an increased throttle opening during idle and overrun, it was possible to substantially reduce carbon monoxide and unburnt hydrocarbon exhaust emissions. Fig. 4.8 shows the average values for a number of separate hot cycle tests including air/fuel ratio. Weighted average values for these hot cycles were 332 p.p.m. unburnt hydrocarbons and 1·9% CO.

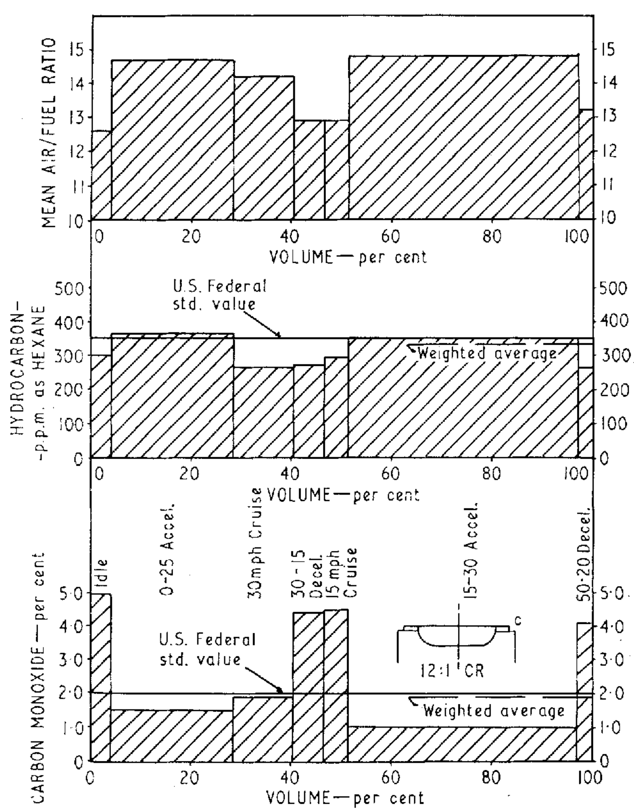


Fig. 4.8. Contribution of modes, engine in 'de-toxed' form

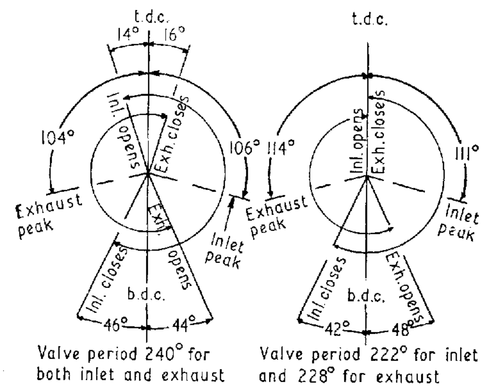


Fig. 4.9. Valve timing diagram

Camshaft

The effect of a camshaft having zero valve overlap (as shown in Fig. 4.9 and compared with the standard valve timing) was examined with the twin-carburetted engine. When used with combustion chamber form 'g' and without a vacuum-limiting device, the zero overlap camshaft did provide a reduction in the order of 5 per cent in the weighted average unburnt hydrocarbon emissions, but it significantly contributed the least improvement during the overrun modes. Tested with a vacuum-limiting device fitted to the engine, the zero overlap camshaft showed no improvement during the overrun modes. Nevertheless, because of the possibility of such a valve timing arrangement reducing emissions during the acceleration modes—and even a marginal improvement here is worth pursuing because of the high weighting factors—it was considered worthy of further investigation.

Further tests were made with the engine fitted with pistons to form 'c', and at steady-speed road-load conditions, unburnt hydrocarbon emissions were reduced by up to 10 per cent with the zero overlap camshaft. However, when examined using the simulated U.S. Federal cycle, unburnt hydrocarbon emissions increased during all modes, the weighted average value for hot cycles being 18 per cent greater than the base-line emissions for the standard camshaft of 350 p.p.m.

Significantly, engine breathing at low speeds was improved with the zero overlap camshaft to such an extent that engine 'idle' was obtained with a manifold vacuum of more than 19 inHg compared with a vacuum of little more than 15 inHg at 'idle' with the standard camshaft. This reduced throttle opening would certainly be expected to

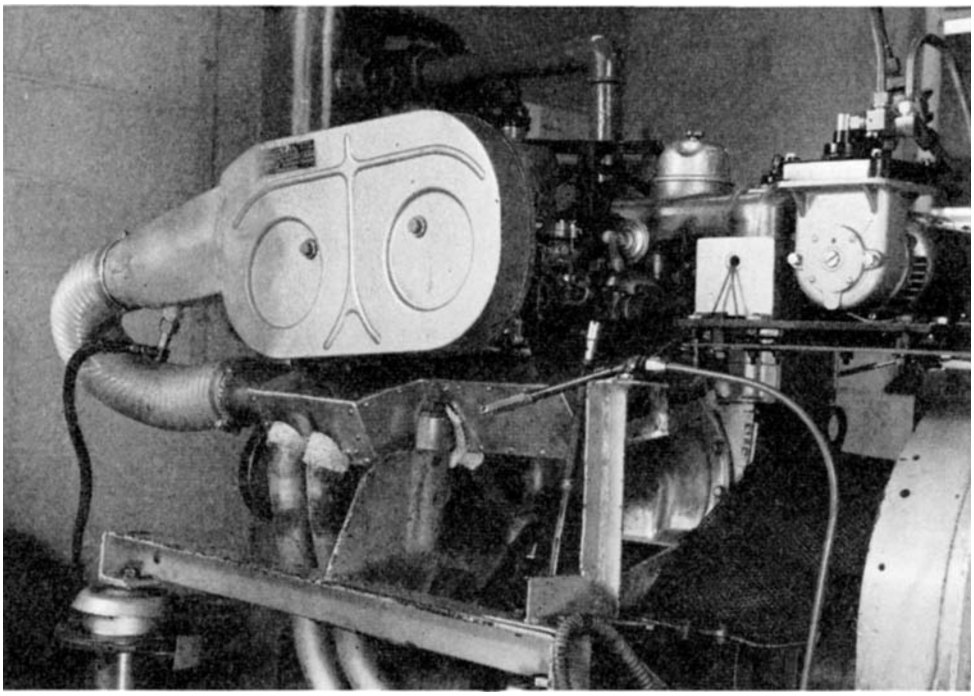


Fig. 4.10. Exhaust heated induction air layout

adversely influence overrun or deceleration modes, and may be reflected in the cruise and acceleration modes.

In terms of full throttle performance, zero valve overlap increased power in the low- and medium-speed range in the order of 5 per cent, but gave a reduction approaching 4 per cent at engine speeds above and including 4500 rev/min.

The effect of increased air intake temperature on exhaust emissions

A box was constructed to surround the exhaust pipes at outlet from the engine and the whole of the induction air was passed through this box, as shown in Fig. 4.10. No attempt was made during this initial investigation to regulate the amount of heat taken up by the air, either by diverting some of the air from passing over the exhaust pipes or by introducing cold air with the heated air. Engine air intake temperature, as measured at the air cleaners, was in fact increased by more than 80 degC at road-load conditions.

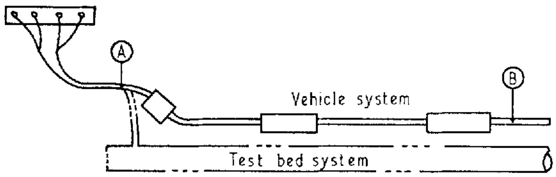
Significant changes in air/fuel ratio occurred with the heated air, and it was concluded that the fuel delivery characteristic of the carburettors was being influenced by the increased air volume required to support a given load, giving rise to changes in piston lift and needle displacement. A further factor may have been the influence of the increased temperature on fuel viscosity. A compromise setting of the needle–jet relationship was chosen when using heated air which closely matched the mixture strength obtained with the standard air intake system, but which was in fact slightly weaker at 'idle'.

With heated induction air, there was a reduction of between 60 and 100 p.p.m. unburnt hydrocarbons in the individual modes of the cycle. The weighted average value of unburnt hydrocarbons for the hot cycles was reduced by 100 p.p.m. or approximately 30 per cent from the baseline of 350 p.p.m. for the standard induction system. Carbon monoxide was reduced by 0·4 per cent. It is assumed that these reductions in exhaust emissions were attributable to improvement in mixture distribution with the heated induction air.

The effect of exhaust system and sampling position on exhaust emissions

The engine on the test bed was coupled in turn to the test bed exhaust system and to a vehicle exhaust system, as shown diagrammatically in Fig. 4.11. When run at steady speeds and loads representing road-load conditions, the table in the figure shows that unburnt hydrocarbon emissions were substantially lower with the vehicle exhaust system. In addition, this reduction was evident whether the exhaust gas was sampled 3 ft from the engine (position 'A') or at the end of the tail-pipe (position 'B').

This observation supports the findings of Lamont Eltinge *et al.* (**10**), in which they claim that raising exhaust gas back pressure increases the reaction potential. In our tests, the exhaust gas back pressure at full power was significantly increased from less than 1 lb/in^2 with the test bed system to more than 3 lb/in^2 with the vehicle system. At the engine speed and loads shown there was also an increase of up to 5 per cent (20 degC) in exhaust gas temperature with the vehicle exhaust system.



	Unburnt Hydrocarbon—p.p.m		
	Test bed system position 'A'	Vehicle system position 'A'	Vehicle system position 'B'
1500 rev/min 30 mile/h 6·25 h.p.	300	200	200
2000 rev/min 40 mile/h 10·2 h.p.	—	160	160
2500 rev/min 50 mile/h 15·5 h.p.	190	139	135

Fig. 4.11. The effect of exhaust system and sampling position on unburnt hydrocarbon emissions—steady speed, road load

The use of twisted metal tape inserts in the inlet ports

Strips of metal approximately 0·03 in thick (11) were twisted 180° to produce a spiral equal in width to the port diameter, and were inserted into the inlet ports (as shown in Fig. 4.12) with the direction of twist and resulting swirling motion such that maximum directional swirl would be promoted in the cylinder.

Port efficiency, as measured on a steady-flow air rig, was lower with the twisted tapes, but when tested on the engine, volumetric efficiency and power output were only significantly affected at 5000 rev/min and above, a loss of 2–3 per cent in power being observed.

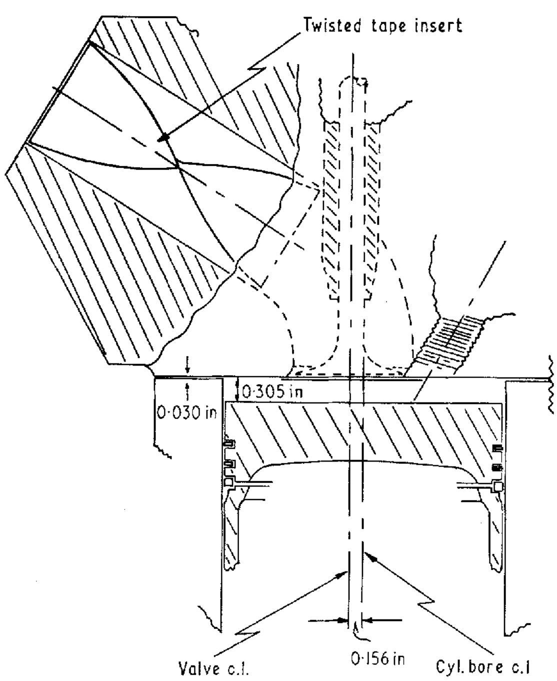


Fig. 4.12. Arrangement of combustion chamber and induction port, showing twisted tape insert (piston at t.d.c.)

With the configuration shown in Fig. 4.12 (form 'g'), unburnt hydrocarbon emissions measured on simulated hot cycles on the test bed were found to be lower by 20 per cent with twisted tapes. Subsequent results for the complete Federal test (cold and hot cycles) with the vehicle equipped with combustion chamber configuration to form 'd' confirmed an advantage with the twisted tapes, with composite value being reduced by 30 p.p.m. for unburnt hydrocarbons to 330 p.p.m. The use of twisted tape did not influence carbon monoxide emissions during any of the tests.

The reason for the reduction in hydrocarbon emissions with the twisted tapes is not completely understood, particularly in view of the disappointing results obtained for the 'cycle' when using shrouded inlet valves to promote directional swirl within the cylinder. It is suggested that probably part of the advantage with twisted tape arises from improvement in mixture preparation before entering the cylinder.

COMPARISON OF FULL AND PART THROTTLE PERFORMANCE DATA FOR VARIOUS FORMS OF COMBUSTION CHAMBER

A number of combustion chamber bowl forms were tested with the engine at 12/1 compression ratio and in twin-carburetted form, and related at full throttle (in Fig. 4.13) to the results obtained for the standard or symmetrical bowl shape.

It was evident from this comparison that some gain in performance was obtainable by offsetting the bowl so that the spark plug occupied a more central position (form 'h'). In addition, combustion chamber configurations which involved minimum or zero squish area (forms 'd' and 'g') were not unduly penalized on the basis of power loss or lower economy, although it must be acknowledged that, on the given test fuel, such forms were operating closer to the trace knock point and would have a slightly higher octane requirement. There would not seem to be any significant advantage in the configuration tested (form 'd_1') where the valves were recessed into the cylinder head and valve cutouts in the piston crown eliminated.

A configuration was tested with the squish region situated directly beneath the spark plug (form 'i'). This layout resulted in a faster burning rate, as indicated by the spark retard for optimum and some power loss (up to 4 per cent), but there was an extension of the spark advance for trace knock, and octane requirement was two to three numbers less than for form 'c'.

Comparison of results at part load (road load) have been related to form 'h' and are shown in Fig. 4.14. At speeds up to 3000 rev/min a significant loss in fuel economy was apparent with forms 'g' and 'i'.

The configurations outlined in Figs 4.13 and 4.14 were examined in relation to unburnt hydrocarbon emissions

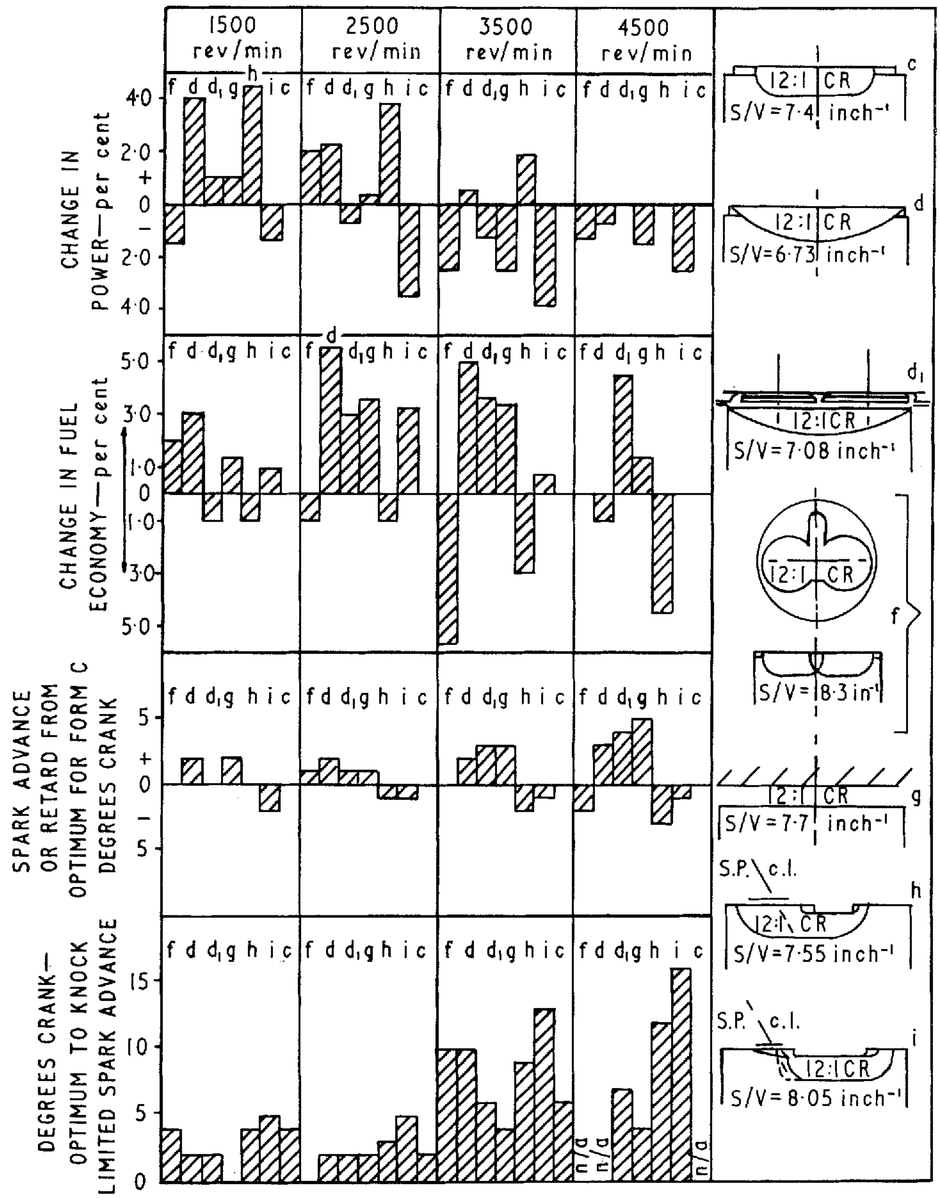


Fig. 4.13. Full throttle performance comparison for the various forms of combustion chamber related to base-line form 'c' (engine fitted with twin carburettors)

with the vacuum-limiting device and weak mixture setting, utilizing the seven-mode cycle in the manner already discussed. Forms 'h' and 'i' gave similar results to the standard form 'c', but some reduction was noted with forms 'f' and 'g', the weighted average value for hydrocarbons being approximately 320 p.p.m. The form which involved the lowest surface area to volume ratio, form 'd', provided the lowest measured emissions at 290 p.p.m., and some further slight reduction of 10 p.p.m. was possible by altering the ring belt so that the top ring land was reduced in depth from 0·8 in (20·3 mm) to 0·3 in (7·6 mm).

The overall changes to exhaust emissions have been summarized in Table 4.3.

CONCLUSION

Of the combustion chamber forms examined in this work, form 'd' (at 12/1 compression ratio) appears the most promising for power economy and low emissions. When installed in the vehicle weighted average values for hot cycle emissions of approximately 320 p.p.m. for hydrocarbons, 1·7% CO have been obtained by the U.S. Federal seven-mode cycle. The composite value for both cold and hot cycle emissions was approximately 360 p.p.m. hydrocarbons and 1·4% CO. By the use of twisted tape in the inlet ports, the composite value was reduced to 330 p.p.m. hydrocarbons.

Exhaust emissions were not reduced for the cycle with high rates of directional swirl imparted to the mixture

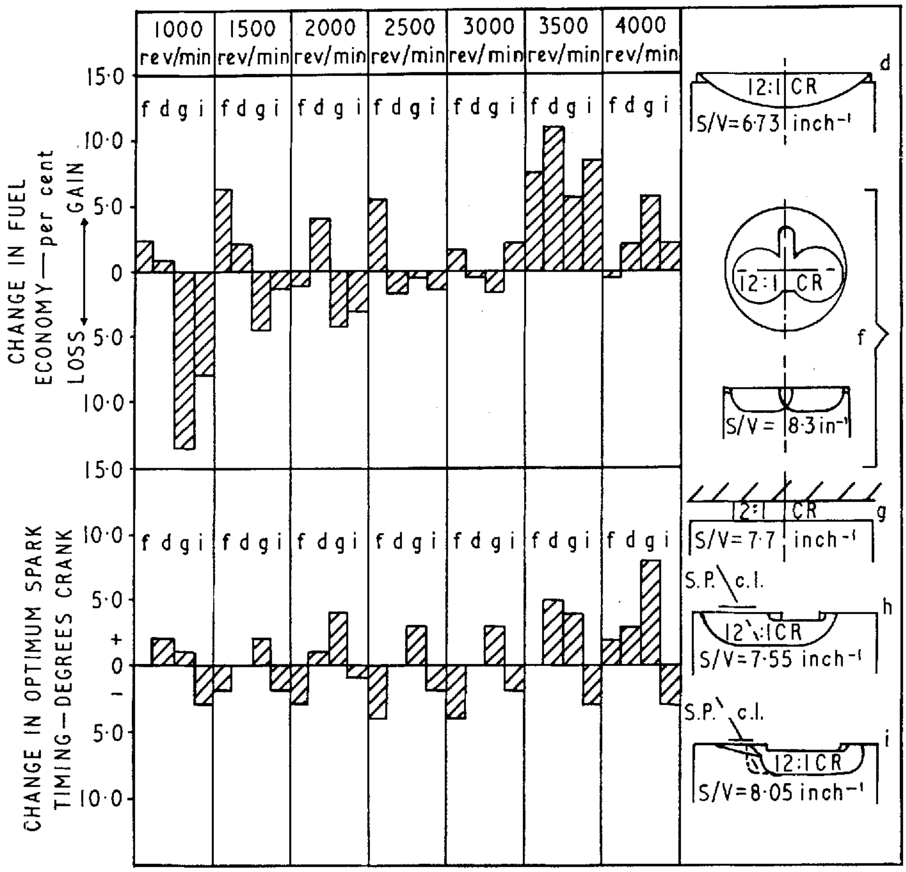


Fig. 4.14. Road-load performance comparison for various forms of combustion chamber compared to base-line form 'h'

Table 4.3. Summary of exhaust emission results obtained on the test bed using the Federal seven-mode test

(Hot cycles only—engine in twin-carburetted form)

Test number	Unburnt hydro-carbons, p.p.m.	Carbon monoxide, %
(1) 9/1 compression ratio (form 'a')	660	3·1
(2) 12/1 compression ratio (form 'c')	830	2·9
(3) 12/1 compression ratio (form 'c') with weaker mixture setting and vacuum limiting during overrun	332	1·9
(4) 12/1 compression ratio (forms 'f' and 'g')—otherwise as for (3)	320	1·8
(5) 12/1 compression ratio (form 'd')—otherwise as for (3)	290	1·8
(6) With twisted tapes in induction ports —otherwise as for (4) (form 'g')	280	1·8
(7) With exhaust heated induction air—otherwise as for (3) (form 'c')	250	1·4

entering the combustion chamber, or by eliminating the valve overlap period.

Significantly lower unburnt hydrocarbon emissions were obtained by supplying heated air to the intake system. Initial results appear to indicate that reduction in hydrocarbons and carbon monoxide may be possible through increased exhaust gas back pressure and temperature.

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APPENDIX 4.1

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