C142/78

# FACTORS ENCOUNTERED IN THE DEVELOPMENT OF A HIGH OUTPUT SMALL ENGINE

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SYNOPSIS This paper gives an account of the performance characteristics of a single cylinder, air cooled, 350 cc. engine which initially developed a specific power output of 78 kW/litre (105 bhp/litre) and which, after a period of development, was increased to 90 kW/litre (120 bhp/litre). There were other desirable development features in prospect which, unfortunately at that time, fell outside the limits of company policy. However, one of these viz. 4 valves vs. 2 valves per cylinder is discussed briefly, since the subject still seems to be somewhat controversial. There is also a short account of an experimental "one-off" engine which involved a partial redesign of the original engine.

## 1.0 INTRODUCTION

A discussion about the many factors encountered during the development of an internal combustion engine tends, quite naturally, to prompt the question "but has this not all been said before?" and the answer is almost invariably "Yes". But because, in this instance, the development work was directed almost entirely towards achievement of reliability, it is hoped that an account of the work and the results obtained, may be of some interest.

In this case, the engine had been designed as the power unit for a road racing motorcycle and was manufactured in small quantity; prospective buyers had to be of proven ability to be entered on the sales list.

As originally designed, the swept volume was 348 cc., with air cooled cylinder head and barrel in aluminium alloy and a magnesium alloy crankcase.

The head had one inlet and one exhaust valve, each operated through a rocker by a single chain driven overhead camshaft. A combined suction/delivery gear type oil pump operated the dry sump lubrication system. The connecting rod big-end and the mainshafts were carried in roller bearings.

Further details of the specification are shown in Table 1 and Fig. 1 is a sectioned drawing of the complete engine.

## 2.0 LOCATING THE LIMITING FEATURES

Even in its original form the engine had developed about 28 kW at a crank speed of approximately 124 rev/s (37 bhp at 7400 rev/min) which was good enough to enable the motorcycle to be reasonably competitive - but there was no doubt that its performance would have to be improved considerably if it was ever to qualify as a potential Grand Prix race winner.

An intensive development project was set up to study and, if possible, identify the areas most likely to be responsible for preventing the achievement of at least 90 kW/litre (120 bhp/litre) which seemed to be a not unreasonable target, despite a rather severe condition which was that any changes must be within the capacity of existing major casting patterns and forgings. The study of a number of full and part throttle power and fuel consumption curves provided indications of less than satisfactory operation in two distinct areas.

(a) in the valve operating mechanism.

(b) in respect of gas flow resistance in the ports and possibly as regards the direction and entry angle of the ports into the combustion chamber.

A third item was arbitrarly added to this list. Precise measurement was not necessary to see that the combustion head area was too small to prevent the cylinder walls from interfering with the inlet gas flow into the combustion head. In other words, as a starter, the cylinder bore diameter would have to be increased.

It turned out that it could not be increased beyond 75.5 mm (compared with the existing bore of 74 mm). However, even so small a difference was advantageous and this change was made immediately.

## 3.0 TEST APPARATUS

To investigate within the areas listed as (a) and (b) above, two simple pieces of apparatus were designed and made up in the Experimental Dept. workshop.

(A) A test rig, constructed so that a


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complete cylinder head could be mounted on a massive block and the camshaft driven through rubber belts by a 7.5 kW (approx. 10 hp) variable speed electric motor. A pump delivered oil maintained at about 80°C to the cam housing and valve spring cases. Perspex windows were arranged to enable the valves, springs and rockers, illuminated by an intense light beam (powered from a direct current source) to be observed with a stroboscope.

(B) An air-flow test rig, shown in (Fig. 2) and diagrammatically in (Fig. 3). The principle has been fully described by J.C. Livengood et al. (Ref. 1) and so requires only a brief mention here. In short, the results of flow tests expressed in terms of a coefficient enable the same author's concept of "Mach Index" to be used to estimate the probable effect of a change in flow conditions on volumetric efficiency. The method consists of comparing the known coefficient of a calibrated orifice with the results observed when, on the rig, the valve is fixed, successively, at a number of different valve-lift position between fully closed and fully open; the results when integrated give the average flow coefficient Cav (see Appendix).

Livengood's "Mach Index" (Z) was written

$$\frac{S}{C} \times \frac{A_p}{a_v} \times \frac{1}{C_{av}}$$

where c = sonic velocity in mixture at inlet conditions

$A_p$ = Piston area. $a_v = d^2 \times 0.786$

s = mean piston speed

d = mean seat diameter

$C_{av}$ = average flow coefficient.

It has been shown that volumetric efficiency $\eta_v$ falls off steeply when "Z" exceeding a value of 0.5 approached 0.6. (see Appendix).

## 4.0 EXPLORATORY TESTING OF VALVE MECHANISM

Having installed and tested the apparatus described at (3) above, the first step was to undertake a critical examination of the valve operating mechanism since, in order to achieve the target power output of about 90 kW/litre, it was of course essential that the valves should move in a prescribed manner and that the mechanism, including the springs, should be completely reliable at sustained crank speeds up to at least 140 rev/s (approx 8500 rev/min).

In the initial design stage there had been some conflicting views regarding the respective merits of hair-pin type valve springs relative to the more conventional helical coil type. The final consensus was in favour of the former and the cylinder head was accordingly designed to accommodate them.

There was already a fairly extensive experience with this engine to indicate that, with the original cam lift and at crank speeds up to about 130 rev/s (approx. 7600 rev/min), the hair-pin springs were not over-stressed. However, with crank speeds up to 140 rev/s (approx. 8500 rev/min) in mind, together with possibly increased valve lift, it was necessary to prove their reliability when subjected to this more severe duty.

In the first of a series of rig tests, the springs were operated at a sustained CAMSHAFT speed of 62.5 rev/s (equivalent to 7500 rev/min crank speed). Some, but slight, surging of the spring coils was observed. The inlet spring failed at approx. $0.9 \times 10^6$ reversals and the exhaust spring at approx. $1.1 \times 10^6$ reversals. In the second test the speed was raised from 62.5 rev/s to 67.5 rev/s. Surging of a high amplitude was observed with the stroboscope, and both springs failed after about $0.7 \times 10^6$ reversals.

These springs, supplied by a reputable spring manufacturer, were in plain carbon steel, prehardened and tempered and shot-peened. It was decided to change the wire material and springs were obtained in Swedish STA 1, a chrome vanadium low alloy steel, prehardened and tempered; they were shot-peened and also heat stabilized. Tested at 67.5 rev/s the high amplitude surge in the spring coil was still in evidence and both springs failed shortly after $1 \times 10^6$ reversals; however, although this result confirmed the expected superiority of the Swedish steel wire, the spring life did not measure up to requirements and it was now certain that the calculated stresses were, in practice, being vastly exceeded, doubtless due to the high amplitude surging which was probably initiated by unfavourable harmonics of the particular cam contour: it was borne in mind that a contributory reason could be because of the relatively low natural frequency of the hair-pin type spring.

## 5.0 CAM MODIFICATIONS

The next step, therefore, was to design and produce two camshafts, one giving the same opening and closing periods as the original camshaft (Camshaft Mark 1) and the other with a longer inlet, but shorter exhaust period (Camshaft Mark 11). The cam profiles however, did not follow the original but were chosen to match, as far as possible, the characteristics of the hairpin springs to which we were now committed.

In passing, it seems worth mentioning that, although admirable in many other respects, the largest convenient diameter roller cam-follower introduces a negative curve in the flank of the actual cam, so that when being finish-ground the grinding wheel tends to "dwell" over this short interval and unless special precautions are taken, may produce "glazing". Such cams are unpopular with cam grinders.

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## 6. CAM/VALVE SPRING PROVING TESTS

The series of valve-spring rig-tests were continued, now with modified camshaft (Mark 1) and the springs in Swedish STA1 wire. Though relatively small in amplitude there was still some surging in the coils to be observed with the rig set to maintain 67.0 rev/s (equivalent to a crank speed of 8000 rev/min). Surging increased in amplitude if the speed was increased, but even at a camshaft speed of 71 rev/s (crank equivalent 8500 rev/min) the amplitude of surge was noticably less than with the original cam.

Finally, this combination (Cam Mk. 1 and the new valve spring) was set up on the test rig to run at a cam speed of 71 rev/s (crank equivalent 8500 rev/min) until failure occurred. After about $1.8 \times 10^6$ reversals, (approx. 7 hours) the test was discontinued, the springs being still undamaged.

At the conclusion of this test, spring load relaxation was measured to be in the order 10 to 12 per cent which was acceptable since this had been allowed for when recalculating the new valve springs.

## 7.0 EXPLORATORY TESTING OF GAS FLOW IN PORTS

To facilitate development of improved port and valve shapes (particularly on the inlet side) a number of jigwood port blocks (Fig. 4) were made by a pattern maker to drawings showing the required dimensions and shapes of the ports: shapes could be altered, if necessary, with the aid of files and plasticine. One of these port blocks had an inlet port which was a replica of the port in the head as originally designed. Tested on the air-flow rig, it gave the same result as the port in the actual head, thus establishing a reliable datum. Fig. 5 (a) shows the outline of this port shape dimensioned to suit an AMAL 35 m/m carburettor.

A port shape, as shown in Fig. 5 at 'b' was indicated to have higher valve/port flow resistance than (a), while (c), although better than (b) still did not equal the performance of port (a). In fact, no other port shape tried, showed as good a value, in terms of mass flow, as the original port shape (a). The comparative results of the air-flow rig tests are shown in Fig. 6. However, before deciding that the apparent superiority of port shape (a) necessarily meant that this would result in improved engine performance, it was decided to attempt to determine not only the port's mass flow capability, but also the manner in which unvapourized fuel droplets might be distributed in the combustion head. A very simple and somewhat crude method was employed in the hope that it might provide, if only, a rough guide.

With the particular port block installed on the test rig set to give a constant pressure difference across the valve/port combination, a precisely measured and timed quantity of coloured fluid was injected at a fixed pressure and fixed distance from the inlet valve. White card lining the interior of the dummy cylinder indicated whether the fluid spray was reasonably well dispersed or, whether it tended to concentrate in a small area high up on the cylinder wall.

Fig. 7 (a), (b) and (c) show photographs of three flow-pattern cards which correspond to the three port shapes (a), (b) and (c) at Fig. 7. It seemed that the flow pattern (c) was indicative of being more favourable than the flow patterns at (a) and (b) and was attributable to the influence of the compound change in entry angle into the head as shown at (b) and (c) in Fig. 5.

To sum up on the information obtained up to this stage, it appeared that the original port shape (a) Fig. 5, gave the least resistance to flow, while (c) which apparently gave the best droplet dispersal pattern, displayed a rather greater resistance to flow. Only actual engine performance tests on the dynamometer would show whether the rig tests were providing valid guidance. Accordingly, three cylinder head castings were machined to standard dimensions except for the ports, which were contrived by hand work until they conformed faithfully with the jigwood models in respect of both dimensions and flow characteristics.

## 8. DYNAMOMETER TESTING

Performance curves (a), (b) and (c) Fig. 8 each show results corresponding to ports (a), (b) and (c) Fig. 5; these are stages of improving performance due to improvements in air flow and combustion characteristics. Except for changes to carburation and ignition advance, all other conditions were held as constant as was practicable throughout these comparative tests. It will be seen that the changes in performance are roughly in line with those suggested by the rig-test results. Improvement in specific fuel consumption (SFC) indicated at (b) Fig 8, was attributed to the port's "steeper" angle, immediately before the valve, having the effect of producing a more favourable dispersal of unvapourised fuel droplets in the cylinder and combustion space, while a further improvement in both SFC and power output was considered to be the result of "straightening" the inlet tract, (c) Fig. 5, to give a tangential entry of gas as it flowed into the cylinder, so creating a tendency to promote swirl.

Confirming evidence of improving combustion efficiency was given by the ability to reduce ignition advance from 39° to 37° BTDC.

## 9.0 CONCERNING THE PISTON

It had been found just possible to machine the 75.5 mm diameter piston from the original 74mm piston casting (not a forging at that time); but there was insufficient metal above the top land to do more than form



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a domed crown, with rather deep valve-head clearance recesses, which resulted in a long attenuated combustion space, as shown in Fig. 9 at "A". This piston carried two 0.062 inch (approx. 1.6 mm) plain compression, and one slotted oil scraper ring.

Since further progress required it, a redesigned piston was requisitioned, since a piston was not considered to be a "major casting".

The new piston was designed with a crown formed as shown in Fig. 9 at 'B' which gave a more compact combustion space, with a compression ratio of 12.2 to 1. It had also a squish band to promote further agitation of the combustion head contents (albeit, squish action occurs rather late in the combustion event).

With the object of reducing piston friction losses it was decided to take the opportunity provided by a new piston to use only one Dykes-type compression ring and one "twin-rail" oil scraper ring (Ref. 2).

The redesigned piston, combined with modifications described earlier, raised the performance curve levels from those shown at (c) to (d) Fig. 8. Ignition advance came back from $37^{\circ}$ to $34^{\circ}$ BTDC and SFC reduced in the general order of about 60 g/kWh (approx. 0.1 lb/bhp/h).

The effect on combustion-gas "blow-by" and oil control by the somewhat adventurous decision to use only one compression ring was watched with some anxiety. However, there was no visible exhaust smoke when running on the test bench and when, after some hours at a variety of speeds, the cylinder head was removed for examination, the piston and head surfaces were dry and of a light colour. Depression in the crankcase varied over the speed range between 25 and 5 cm water.

## 10. FINAL PROVING TESTS ON THE DYNAMOMETER

Overall performance improvement, due to the combined effect of the various modifications to port and combustion head shapes, piston redesign and the smoother valve operation due to camshaft Mk. 1, is shown in Fig. 8, curve (d). Camshaft Mk 11, with slightly longer inlet and shorter exhaust periods, provided a further small gain, but only in the lower part of the speed range between 108 rev/s (6500 rev/min) and 123 rev/s (7400 rev/min) as shown in (f) Fig. 8. This Mk 11 camshaft had no effect on maximum power output or on SFC. However, the longer inlet period did have a further advantage since it enabled the final spring design to be adjusted to a marginally lower stress level.

Differences in respect of period and lift between camshafts Mk. 1 and Mk. 11 were:-

| | Mk.1 | Mk.11 |
|---|---|---|
| Inlet period | $305^{\circ}$ | $312^{\circ}$ |
| Exhaust period | $307^{\circ}$ | $304^{\circ}$ |

The maximum valve lift of camshaft Mk.11 had been increased for the inlet valve from 11.7 mm (0.460 in.) to 12.2 mm (0.480 in.) and for the exhaust, from 10.15 mm (0.400 in.) to 10.66 mm (0.420 in.).

Valve acceleration curves relative to the original inlet cam and the redesigned cam are shown in Fig. 10.

## 11. SUMMARY OF RESULTS OF STAGE 1 DEVELOPMENT

The two test-rigs had provided means for more easily identifying the areas where modifications were required - and later, to show the measure of a change.

The combined effect of the modifications described raised the maximum power output from 27.6 kW (37 hp) at 123 rev/s (7400 rev/min) to 31.3 kW (42 hp) at 130 rev/s (7800 rev/min) and reduced the specific fuel consumption.

Changes effected in the valve operating mechanism resulted in much reduced chances of valve spring failure, especially if, by accident, the max. power crank speed was exceeded.

The improved results had been obtained without changes to major casting patterns or forgings.

NOTE Table 1 lists the relevant dimensional differences prior to and after development.

## 12. INLET AND EXHAUST SYSTEMS

Throughout the foregoing description of the various means employed to improve the performance of this particular engine, no mention has been made of a most important factor in breathing efficiency, namely, the inlet and exhaust systems.

The reason for this was because, at the commencement it had been considered that these systems should not be changed while experimenting with other features.

A considerable amount of time had been spent at an earlier date to obtain "near-to-optimum" dimensions for the inlet and exhaust systems which are shown, respectively in Figs. 11 and 12 and which were used when carrying out the final performance tests shown in Fig. 8.

## 13. DISCUSSION REGARDING OTHER POSSIBLE MEANS FOR FURTHER IMPROVEMENT

Following the development which has been described it was possible that modifications of a more extensive character might be permitted, including conversionfrom two to four valves in the cylinder head. Lengthy discussion concluded with a written "Argument For and Against 4 valves" of which the following is a brief, paraphrased version.

It is included because the subject still appears to be somewhat controversial.

Some of the obvious advantages of four small valves vs two large ones, are:-

(1) Due to their smaller mass, less highly stressed springs are required to maintain valve control at higher operating speeds.

(2) Because of their lower lift, valve head clearance recesses in the piston crown can be relatively shallow.

(3) It has been shown (Ref. 3) that the volume of air flowed through two closely adjacent ports can be superior to one port of equivalent valve area.

(4) Because the sparking plug can be positioned in the centre of the combustion head, the flame travel distance is shortened.

However, despite these advantages, means for producing an organized movement of the charge within the combustion head to aid combustion efficiency is less easy than it is in the case of the two-valve head. To achieve a similar effect with two (approx. parallel) inlet ports, it seems that the only feasible notion is to attempt to induce the charge issuing from the two valves (which is influenced by the cylinder walls and the descending piston) to perform in a manner which, in two-stroke parlance, would be described as "loop-scavange". If this can be achieved (and it can be, as implied by the excellent performance of a liquid cooled engine reported in Ref. 4) then there is no corresponding loss of breathing efficiency as is inevitable, with an angled entry, in the case of a single port arrangement.

In general, therefore, there are advantages to be obtained from having two inlet and two exhaust valves (or possibly, one exhaust); even so, a decision in favour of more than two valves would be taken after considering that the included angle between the inlet and exhaust valves could not exceed about 40$^{o}$ if the required direction of flow in the cylinder was to be realised. Consequently, for air cooling, space to erect fins of adequate cooling surface area, would be extremely limited - in fact, as a motorcycle engine, the only recourse might be to position the engine in its frame so that cooling air would be directed on to the top of the head; alternatively, the engine might have to be liquid cooled.

## 14 ONE FURTHER PRACTICAL DEVELOPMENT

Although not envisaged as a replacement for the engine which has been described, sanction was given, on a strictly experimental "one-off" basis to partially redesign it; although four valves in the cylinder head were not included.

Little if any increase in maximum power was to be expected; the main object was to widen the speed band over which maximum power would be available and so, possibly, provide the ability to maintain maximum power at higher crank speeds in the intermediate gear ratios. In short, the nature of the changes envisaged would be directed towards the achievement of higher crank speeds while not exceeding known safe stress levels in the connecting rod and big-end bearing. Differences in the leading particulars, between the 75.5 mm bore engine and the experimental 81 mm bore engine, are shown in Table 2.

It will be seen that the maximum power output of 32 kW (43 bhp) occurred at 136.5 rev/s (8200 rev/min) and it remained at approximately that value up to a crank speed of 150 rev/s (9000 rev/min) when mean piston speed is 21.3 m/s (4200ft/min). Although maximum power was only about 0.75 kW (approx one hp) above that of the "standard" engine, the wider speed range enabled four different riders to reduce lap times, at a race circuit in Norfolk, by between two and three seconds. Timed on the "straight" or fastest part of the track, maximum speed of the machine with the experimental engine was approx. 200 km/h (126 mph) at a crank speed of 145 rev/s (8700 rev/min) compared with 185 km/h (115 mph) at a crank speed of 130 rev/s (7800 rev/min for a similar model, but which had the shorter "constant power" range. Contributing largely to these results was the use of Titanium (Ti) for the connecting rod of this experimental engine. The small end (reciprocating) part of the Ti rod weighed only 120 g compared with 172g, for the steel rod.

## 15. CONCLUDING REMARKS

In the course of developments in the cylinder head, it appeared that:-

(1) If requirements to aid volumetric efficiency conflicted with those for improved combustion, a net gain would be obtained by concentrating on means favourable to the latter - even at the expense of sacrificing some small measure of volumetric efficiency. The results emphasised the importance of discerning interpretation of indications given by an air flow test-rig.

(2) It was shown also that the results given by single experiments in isolation are often disappointing - and that it is only when a number of potentially improving features combine, that the individual effects can be fully appreciated.

(3) The beneficial effect of a cam contour which avoided infinite rates of change of acceleration (infinite jerk) was very noticeable. It was to this factor that improved valve spring life was mainly attributed.

(4) Simplicity of design and exceptionally close inspection of dimensions of parts, contributed in a large measure to the



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consistent performance and reliability of this engine.

(5) Finally, the performance results of the experimental engine seemed only to confirm that, while marginal improvement in performance of a single cylinder engine is possible, it is a futile exercise if the aim is success in competition with multi-cylinder four or two-stroke cycle engines.

## APPENDIX

Dimensions taken for calculation of Mach Index "Z"

Referring to Fig. 5, port(c) was selected on account of its apparently better gas distribution qualities. The average flow coefficient, $C_{av}$, for this port/valve combination turned out to be 0.310 (as against 0.318 for port (a) and 0.298 for port (b).

Taking $C_{av}$ as 0.310; piston area $A_p$=44.8cm$^2$ (6.945 in$^2$); inlet valve area, $a_v$, (based on a mean seat diameter of 4.37 cm (1.72 in.) ) = 15 cm$^2$ (2.325 in$^2$); sonic velocity (c) at inlet conditions as 366 m/s (1200 ft/sec.) and mean piston speed (s) at max. power crank speed of 130 rev/s (7800 rev/min.), then

$$\text{Mach Index "Z"} = \frac{s}{c} \times \frac{A_p}{a_v} = \frac{20.3}{366.0} \times \frac{44.8}{15} \times \frac{1}{0.31} = 0.534$$

thus, insofar as the three predominating variables, viz, mean piston speed (s), valve area ($a_v$), and inlet port flow coefficient ($C_{av}$) were concerned, the conditions predicted by this value of Z appeared to be satisfactory.


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**Table 1 Single-cylinder air-cooled engine**

| | Specification Data | | Prior to development | After development |
|---|---|---|---|---|
| 1. | Piston displacement | cm$^3$ | 348 | 349.5 |
| 2. | Bore x Stroke | mm | 74 x 81 | 75.5 x 78 |
| 3. | Stroke/bore ratio | | 1.095 | 1.033 |
| 4. | Con. rod length between centres (L) | mm | 162 | 162 |
| 5. | L/crank radius | | 4.0 | 4.15 |
| 6. | Inlet valve timing | | 50-70 | 55-75 |
| 7. | Exhaust valve timing | | 70-50 | 78-44 |
| 8. | Inlet valve lift | mm | 11.7 | 12.2 |
| 9. | Exhaust valve lift | mm | 10.15 | 10.66 |
| 10. | Inlet valve head diameter | mm | 44.5 | 44.5 |
| 11. | Exhaust valve head diameter | mm | 39 | 39 |
| 12. | Piston area | cm$^2$ | 43 | 44.8 |
| 13. | Compression ratio | | 10.8 | 12.2 |
| 14. | Carburettor bore diameter | mm | 32 | 35 |
| 15. | Dry weight with carburettor and ignition system | kg | 38 | 38 |

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**Table 2 Differences in the leading particulars between the 75.5 mm bore engine and the experimental "one off" 81 mm bore version**

| | | | 75.5 mm bore engine | "One off" engine |
|---|---|---|---|---|
| 1. | Bore x Stroke | mm | 75.5 x 78 | 81 x 68 |
| 2. | Stroke/bore ratio | | 1.033 | 0.838 |
| 3. | Con. rod centres | mm | 162 | 149 |
| 4. | Piston (complete) | g | 455 | 509 |
| 5. | Weight of reciprocating parts with steel con. rod | g | 627 | - |
| | with titanium (Ti) con.rod | g | - | 629 |
| 6. | Piston acceleration TDC 130 rev/s (7800 rev/m) | $m/s^2$ | 32300 | - |
| | 138 " (8300 " ) | " | - | 31400 |
| 7. | Inertia loading at TDC | kN | 20.3 (4560lbf | 19.8 (4440lbf) |
| 8. | Valve timing | deg. | I 56-77<br>E 78-44 | I 56-77<br>E 78-44 |
| 9. | Ignition advance | | $34^{o}$ BTC | $35^{o}$ BTC |
| 10. | Carburettor bore diameter | mm | 34 | 36 |
| 11. | Valve head dia. (inlet) | mm | 44.5 | 47.5 |
| 12. | Compression ratio | | 12.2/1 | 11.8/1 |

**Relative performance**

| | | | A | B |
|---|---|---|---|---|
| A | Crankshaft power output at approx. 130 rev/s (7800 rev/m) | kW | 31.3 (42 hp) | - |
| B | Crankshaft power output at approx. 138 rev/s (8300 rev/m) | kW | - | 32.1 (43 hp) |

Note: The principle difference in performance between A and B was an increase in the crank speed range of approx. 8 rev/s (500 rev/m) over which approx. 'constant power' was maintained at substantially equal stress levels.



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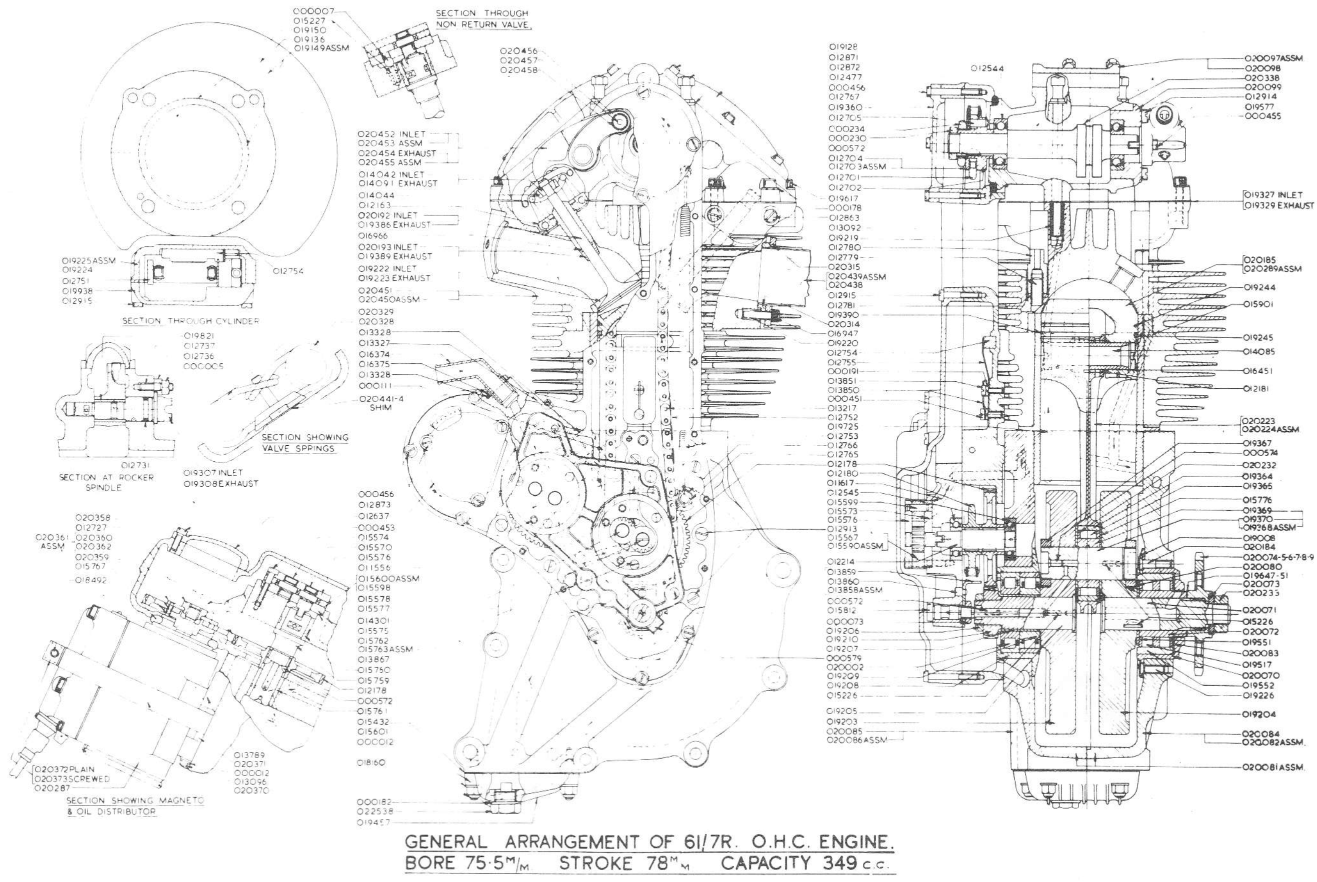



Fig. 1 Sectioned drawing of complete engine

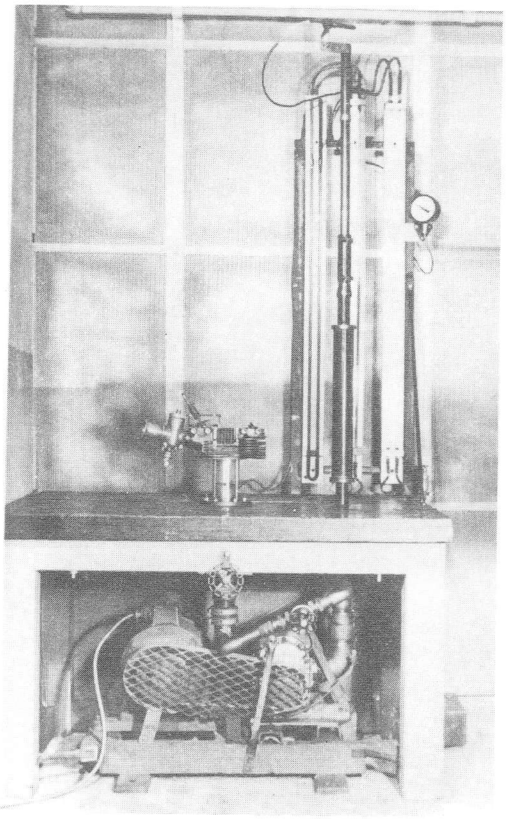
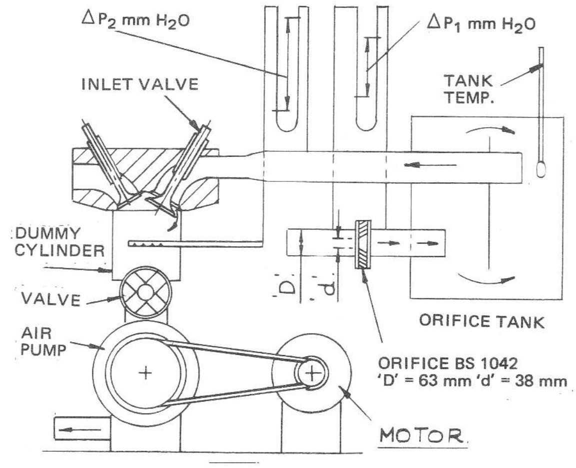


Fig. 2 Air flow test rig (general view)




$$Cv = \frac{Ao\ Co}{Av}\sqrt{\frac{\Delta P_1}{\Delta P_2}} \quad \text{WHERE}$$

Cv = VALVE/PORT AIR FLOW COEFFICIENT
Ao = AREA OF ORIFICE $mm^2$
Av = BASIC AREA OF VALVE $mm^2$
Co = COEFFICIENT OF ORIFICE
$P_1$ = PRESSURE DROP ACROSS ORIFICE, mm $H_2O$
$P_2$ = PRESSURE DROP ACROSS VALVE, mm $H_2O$

Fig. 3 Air flow test rig (diagrammatic)

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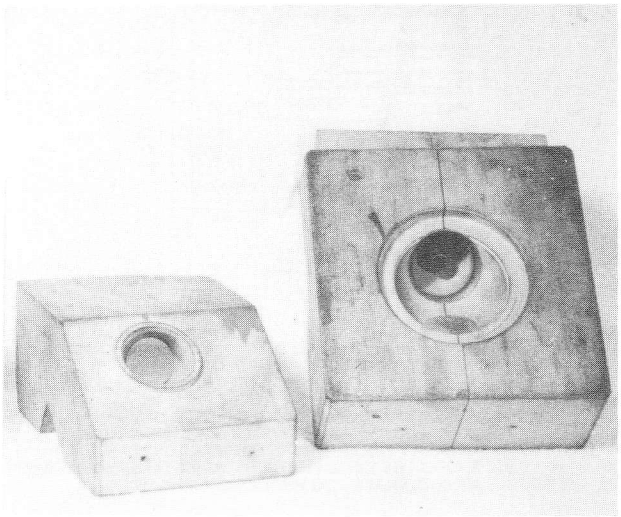

Fig. 4 Jig-wood port blocks

INLET PORT
a GOOD AIR FLOW EFFICIENCY : POOR FUEL DISTRIBUTION
b REDUCED AIR FLOW EFFICIENCY : IMPROVED FUEL DISTRIBUTION
c IN CONJUNCTION WITH (b), FURTHER IMPROVEMENT IN COMBUSTION EFFICIENCY

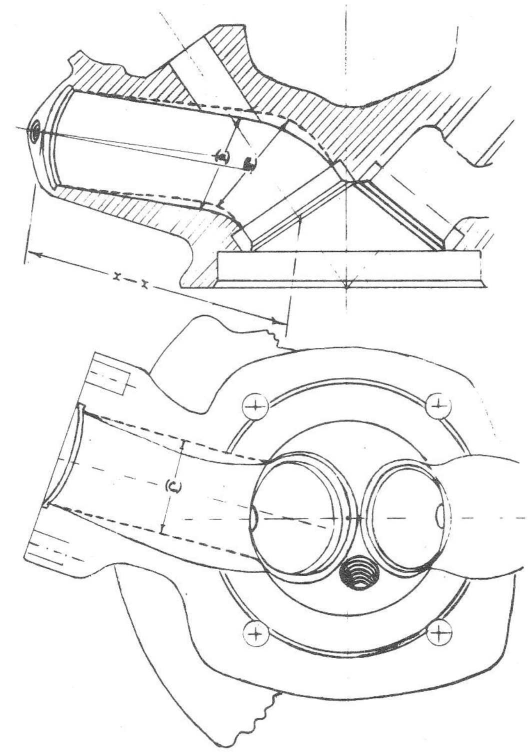


Fig. 5 Comparative inlet port shapes

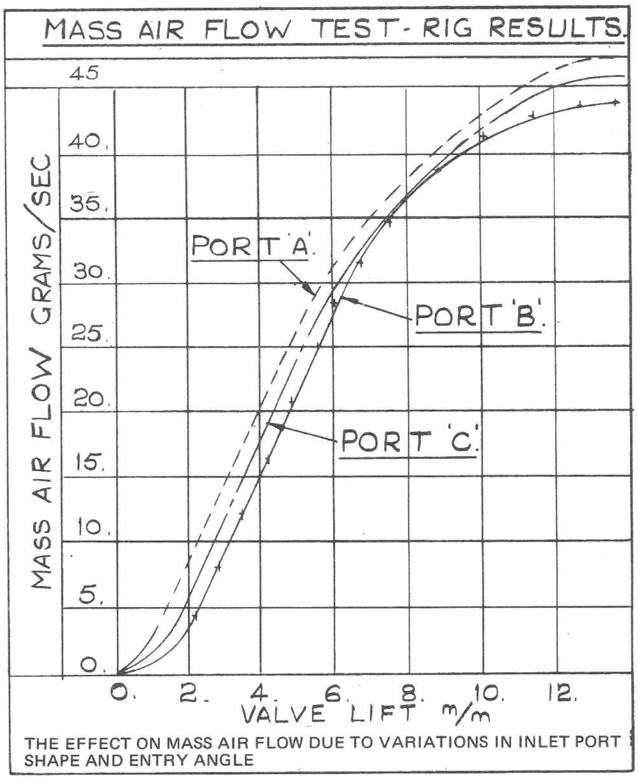


Fig. 6 Results due to alterations of inlet port shape

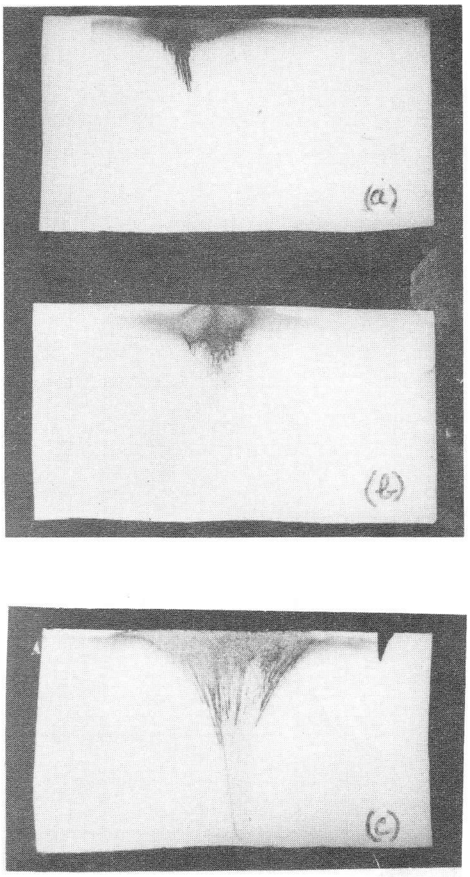


Fig. 7 Flow pattern indicator cards



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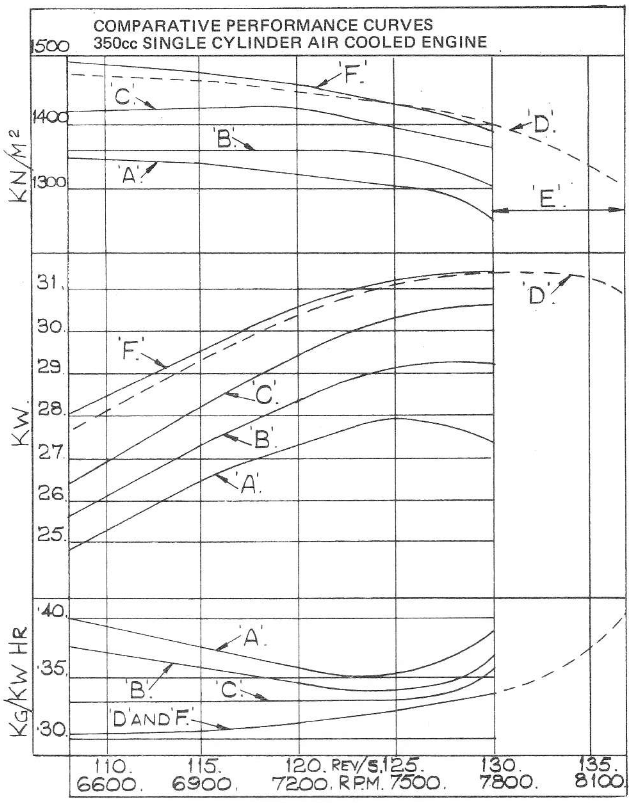
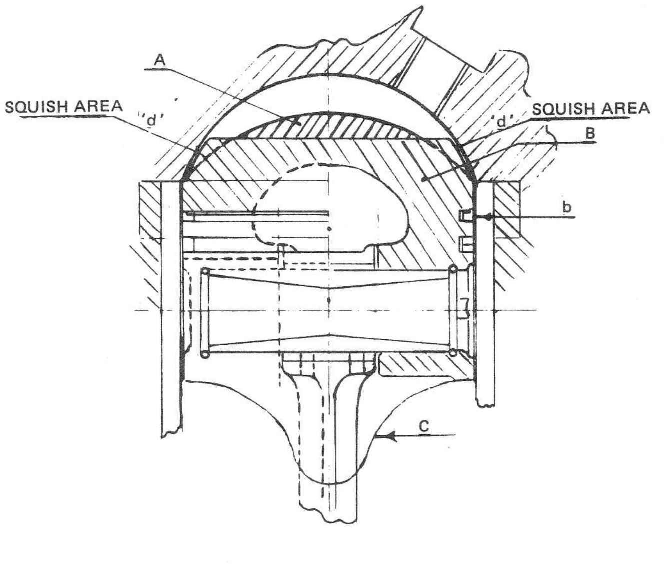



'A' = PERFORMANCE CURVES PRIOR TO DEVELOPMENT. 'B' = EFFECT OF INLET PORT MODIFICATIONS. 'C' = EFFECT OF SECOND PORT MODIFICATION. 'D' = HIGHER COMPRESSION RATIO – COMPACT ENGINE SPACE. 'E' = EXTENDED CRANK SPEED RANGE MK 1. CAMSHAFT AND MODIFIED VALVE SPRINGS. 'F' = EFFECT OF MK. 2 CAMSHAFT AND MODIFIED VALVE SPRINGS

**Fig. 8 Comparative performance curves**




VIEW OF MODIFIED PISTON CROWN B, SUPERIMPOSED ON EARLIER SHAPE A – SHOWING COMPACT COMBUSTION SPACE AND SQUISH LANDS d – d. THE SINGLE DYKES TYPE PRESSURE BACKED COMPRESSION RING IS SHOWN AT b. C SHOWS THE LONG PISTON SKIRT EXTENDING BETWEEN THE FLYWHEELS AT BDC, SHAPED TO REDUCE OIL SHEAR DRAG

**Fig. 9 Piston crown and combustion chamber variants**

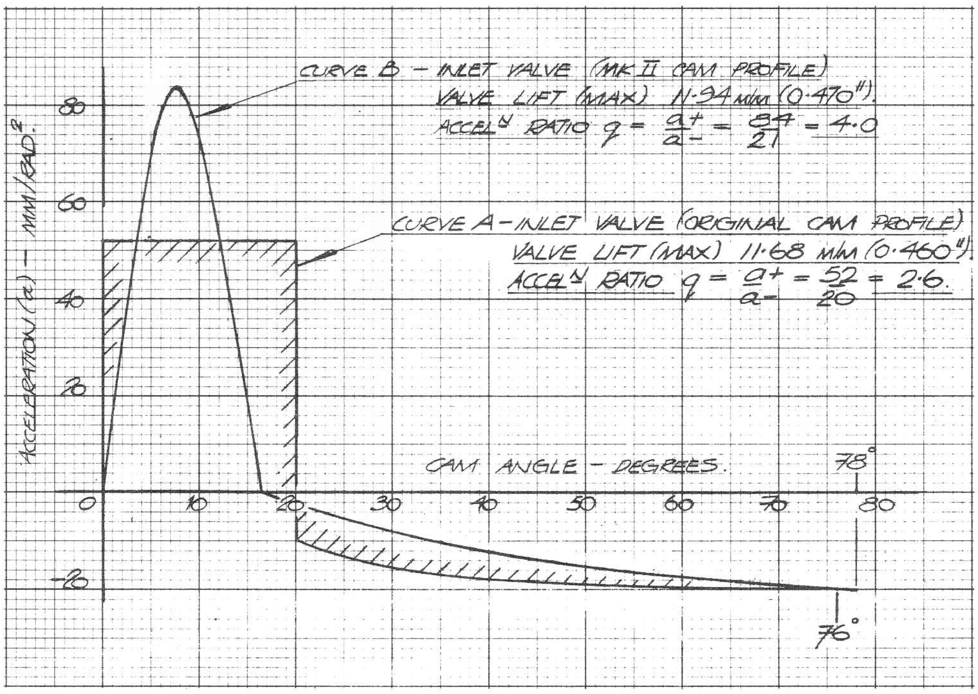



**Fig. 10 Inlet valve acceleration curves**

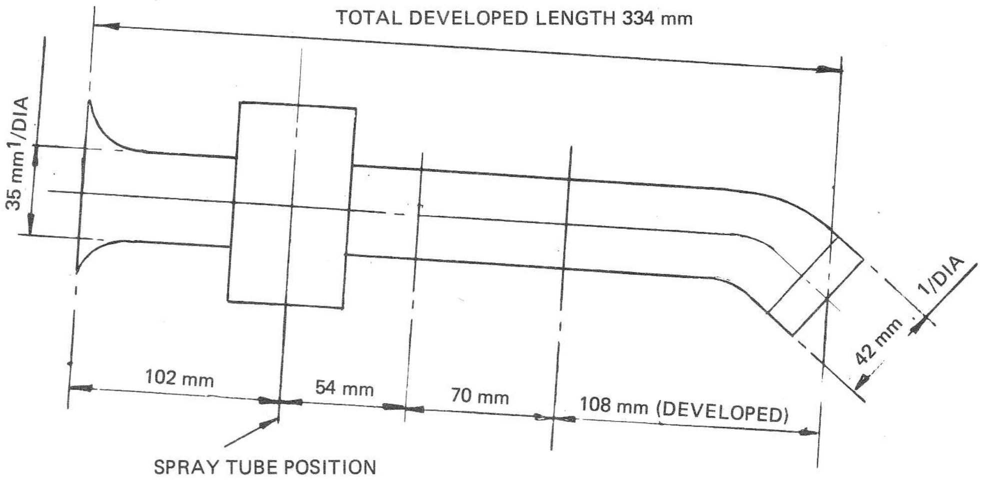



**Fig. 11** **Inlet system dimensions. Applicable to Design 'A'. Effective range 5200–7800 rev/min**

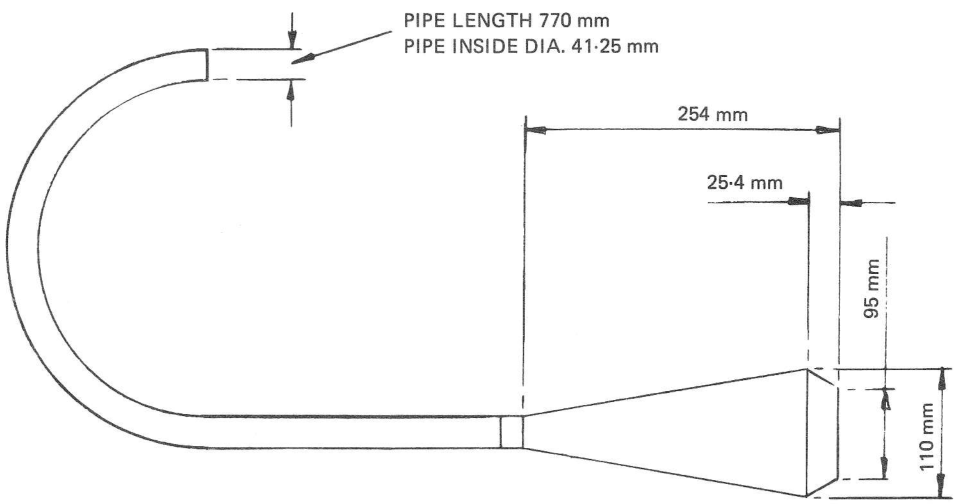



**Fig. 12** **Exhaust system dimensions**



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