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Design and Initial Development of a High Specific Output 500 CC Single-Cylinder, Two-Stroke, Racing Motorcycle Engine

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RACING MOTORCYCLE ENGINES have traditionally developed high specific power outputs from naturally aspirated gasoline-burning power units; specifically the 125 cc, V4, 2 stroke Yamaha engine, which was water-cooled, produced 42 bhp at 17,000 rpm, or 336 bhp/l. However, the bmep attained for various engines and for various cylinder sizes appeared to be oriented toward the induction characteristic. That is to say, irrespective of cylinder size, some 130 lb/in.^{2} bmep was available with asymmetrical inlet timing (or disc valving) and 100 to 110 lb/in.^{2} with normal piston-controlled induction. This pertains specifically to naturally aspirated, 2-cycle power units, using 95 to 100 octane gasoline. This discussion is exclusively concerned with these overall criteria.

In a racing motorcycle some 60 bhp is necessary to sustain 150 mph on the faster race circuits. Once the 60 bhp is attained, higher lap speeds are often best achieved by reducing machine overall weight, height, and width to improve the handling and aerodynamic characteristics. The challenge, therefore, in the academic sense, lay in designing a 500 cc single-cylinder engine that would produce 60 bhp and be a neat (small and light) package to improve the cornering characteristics of the motorcycle. The engineering problems were twofold:

1. Could the gas dynamic characteristics of small engines (up to 250 cc swept volume) be reproduced in a large cylinder of 500 cc swept volume in order to attain a bmep of 105 lb/in.^{2} at 7500 rpm?
2. Would an engine of 1.2 bore/stroke ratio perform in a mechanically reliable fashion with a mean piston speed of 3740 ft/min at 7500 rpm?

With this bore/stroke ratio (1.2) and a swept volume of 494 cc, convenient bore and stroke values of 91 mm and 76 mm, respectively, were used. With the rapid computational analysis of porting characteristics available, it has been found that bore/stroke ratios in excess of 1.2 lead to unacceptable exhaust port height/stroke ratios for the same bmep output.

Obviously, a twin-cylinder engine would provide more power (1) *. For example, a 500 cc twin-cylinder engine,

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*Numbers in parentheses refer to References at end of paper.

ABSTRACT

This paper describes the initial investigation and design of a lightweight racing motorcycle with a single-cylinder 2-stroke engine, capable of producing 60 bhp. The data discussed here pertain to the gas dynamic and mechanical parts and functions of the cycle. Designs of the various components are described and reports of tests on road and test beds verify the viability of this concept of a high specific output and large displacement cylinder for a lightweight, air-cooled motorcycle engine.

72 mm X 60 mm (bore X stroke), and producing the same bmep (105 lb/in.^{2}) at the same mean piston speed (3740 ft/min) would develop 75 bhp at 9500 rpm. But the engine would be at least 60% heavier and certainly 70% wider and require a larger fuel tank for the same bsfc. Furthermore, if such a machine were put into production, it would cost the amateur racing motorcyclist twice as much for spare parts to obtain reliability comparable to that of the single cylinder engine. Thus, unless the power/weight ratio for the complete twin-cylinder machine plus fuel plus rider was larger than that for the single-cylinder machine, no acceleration advantage would be achieved, while at the same time there would be an aerodynamic disadvantage (greater frontal area) and a handling disadvantage (higher machine c.g. and actual weight).

So the concept of a 500 cc, single-cylinder, piston-ported unit was evolved, despite the handicap that no engine of this cylinder size in 2-stroke form had been used for racing and the opinion that the mechanical design problems associated with thermal distortion in such a large cylinder could well prove to be insuperable.

Faced with the thermal problem, it might be expected that the engine would have to be liquid-cooled. But this solution would increase total machine weight to the point where the air-cooled, twin-cylinder unit would be the logical choice. Thus, the air-cooled single had to be proved acceptable or the concept would be invalid.

From discussions with engineers dealing with high-performance snowmobile engines, it was realized that the design problems of racing motorcycles are not dissimilar and that the choice of the single cylinder was obvious.

THEORY

The discussion of the design is in two parts: gas dynamic, or that part pertaining to design of all ports, crankcase/inlet system, combustion chamber, and exhaust system; and mechanical, or that section dealing with the layout of piston, connecting rod, bearings and crankshaft, cylinder and liner, and similar components.

GAS DYNAMIC DESIGN

Crankcase/Inlet System - The interesting aspect of a large single-cylinder engine like this is that the crankshaft flywheel diameter is not significantly greater than that for a 350 cc single-cylinder engine. Here the flywheel diameter, after allowing for a 28 mm crankpin diameter, a 28 X 35 X 16 mm Dürkopp type of needle roller bearing, and a suitable connecting-rod big end eye thickness for the estimated stresses was less than 5 in. With only a 10 mm space between the flywheels for the connecting rod, and 146 mm connecting rod centers, plus the fact that the flywheels were fully shrouded from the cylinder, a crankcase compression ratio [(CR) crank] of 1.65 was attained. This is defined as

$$(CR)_{crank} = (V_1 + V_2)/V_1$$

where:

V_1 = Volume under piston at bdc

V_2 = Swept volume

As Nagao (2) shows, high crankcase compression ratios are required for high delivery ratios at elevated engine speeds. Thus it was encouraging at the early design stage to find that the highest crankcase compression ratio is available, in general, from an engine with a large swept volume cylinder.

Following the analytical technique of Blair and Arbuckle (3), it was possible to deduce rapidly, using a high-speed digital computer, the optimum dimensions of inlet port width and height, carburetor flow area, downdraught angle, and total inlet tract length, to give an acceptable volumetric efficiency for an engine power spread from 6000-8000 rpm.

Based on the calculated transfer port timing and area (see next section), the following optima were obtained:

Inlet port chordal width = 1.75 in.
Inlet port timing (included) = 200 deg
Carburetor flow diameter = 1.6535 in. (42 mm)
Total inlet tract length = 7 1/4 in.

The predicted volumetric efficiency for the engine was 79.8% at 7500 rpm, with a peak crankcase pressure of 1.523 atm at 133 deg atdc (24 deg after transfer port opening) and a peak ramming pressure of 1.383 at 78 deg atdc in the plane of the inlet port. The analysis also showed that for the 1.6535 in. (42 mm) carburetor, a crankcase compression ratio higher than 1.65 would have been beneficial, whereas if a 1 1/2 in. carburetor were to be employed, a 1.5:1 crankcase compression ratio was the optimum, even though the peak volumetric efficiency was some 15% lower and the peak crankcase compression pressure down to 1.401 atm at 125 deg atdc.

Thus the analytical treatment of Blair and Arbuckle (3) allows rapid analysis of various variable combinations, and from experimental work to date it appears that a volumetric efficiency of 80-85% is essential to maintain a bmep greater than 100 lb/in.^{2}, with all other functions such as scavenging and exhaust charging being performed adequately.

The analytical treatment of Blair and Arbuckle (3) was modified to take into account the cylinder pressure behavior. In the earlier work (3) a cylinder pressure of 1.0 atm was assumed as the unit was being motored. Here a cylinder pressure versus degree crank curve was assumed for the calculations, based on experimental work taken in a geometrically similar engine. This more precise work will, hopefully, be reported to SAE at a later time.

Design of Ports, Exhaust, and Transfer - For engines designed to produce a high bmep at an elevated piston speed, almost certainly the critical factor is the transfer port time area, which must be sufficiently large to avoid excessively long transfer port timing. This implies that the area of the normal two side transfer port layout will be inadequate and that two extra transfer ports are necessary. As it is not possible to have the exhaust port side transfer port encroach on the exhaust port without incurring serious danger of short-circuiting the charge, the extra transfer ports must be accommodated near the inlet port, which, in effect, leads to a reduced inlet port width.

Another problem arises in dealing with the angles (in the planview) of these transfer ports because the jets from the four ports must cohere into a single core of fluid moving up

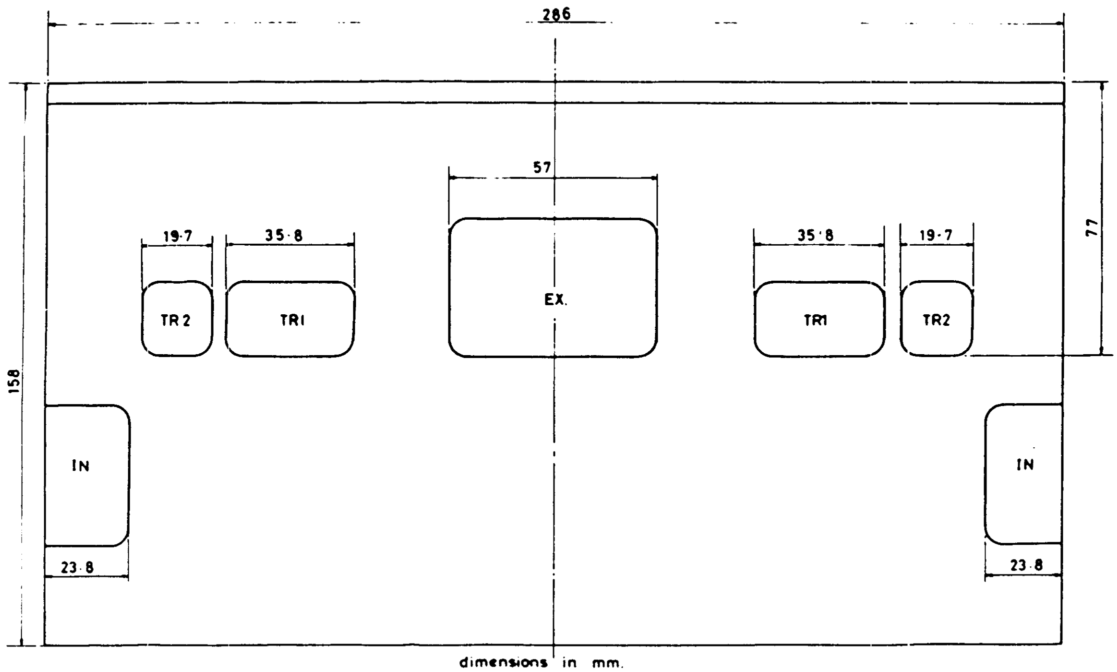

Fig. 1 - Folded-out cylinder dimensions

the rear cylinder wall to provide good loop scavenging, as discussed by Jante (4). For example, pitching the angle of transfer port TR2 in Fig. 1 toward the rear cylinder wall will seriously constrict the effective outlet area as it enters the cylinder. Furthermore, the angle of confluence of the two jets from ports TR1 and TR2 then becomes critical.

As a complement to the work on timing and areas of transfer and exhaust ports, a study of flow from transfer ports of the type outlined in Fig. 1 was initiated in two parts. The first part examined visually the path of smoke issuing from the ports and the loop scavenge mechanism in this particular configuration. A model of the cylinder was constructed in Perspex. The second part was a steady flow analysis conducted along the lines indicated by Jante (4), but with air pulsed into the transfer ports from the exhaust pulse simulator used earlier and described fully in Blair and Goulburn (5).

Some of the many photographs taken in the visual study are presented in Fig. 2. Fig. 2 (top) shows a view of the exhaust port with the piston just uncovering the transfer ducts. A central vortex appears to form with a minimum cross section halfway between the piston crown and the cylinder head level. (The downflow from the cylinder head is approaching the exhaust port in two paths.) The block construction of the transfer ducts can be easily seen. A closeup of the exhaust port is shown in Fig. 2 (bottom). Short circuiting from the transfer jets is clearly evident, as is the fluid looping in two paths toward the side transfer ports and toward the exhaust port. This trend is borne out in the static pressure studies.

Fig. 2 - Visual flow study

For the pressure contour study in the cylinder head plane, with the cylinder head removed, the piston was set at some predetermined position and the pressure profiles recorded. In Figs. 3A, 3B, and 3C the probe is positioned at 0 mm below the head surface, 30 mm below the head surface, and 50 mm below the head surface. All pictures are taken with the air pulsed into the transfer port reservoir at 6000 pulses/min. The cylinder head is removed and the piston is at bdc. Fig. 3A

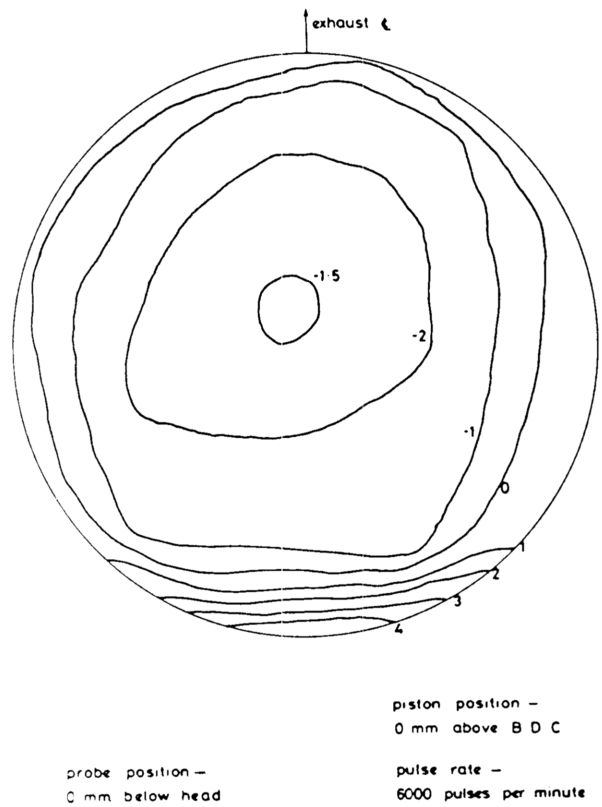

Fig. 3A - Pressure (in. water gage)–probe at head level

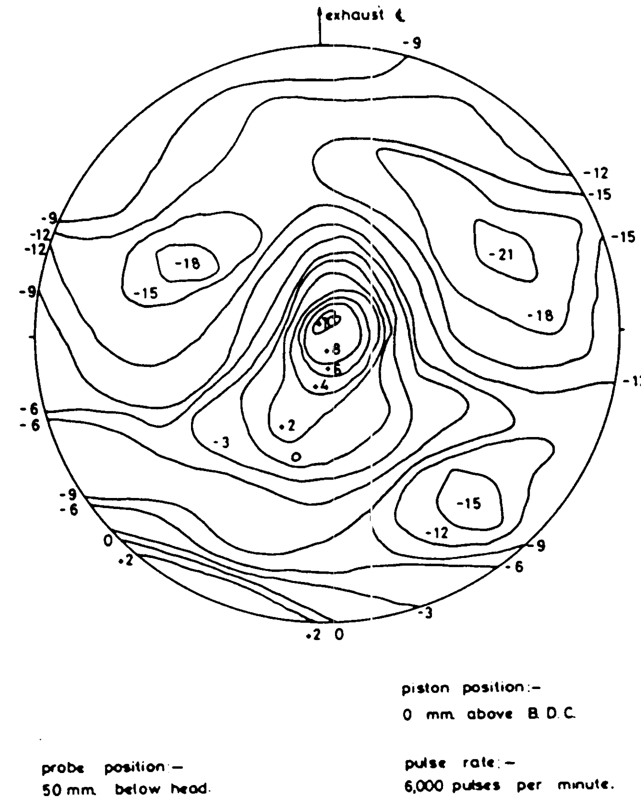

Fig. 3C - Pressure (in. water gage)–probe 50 mm below head level

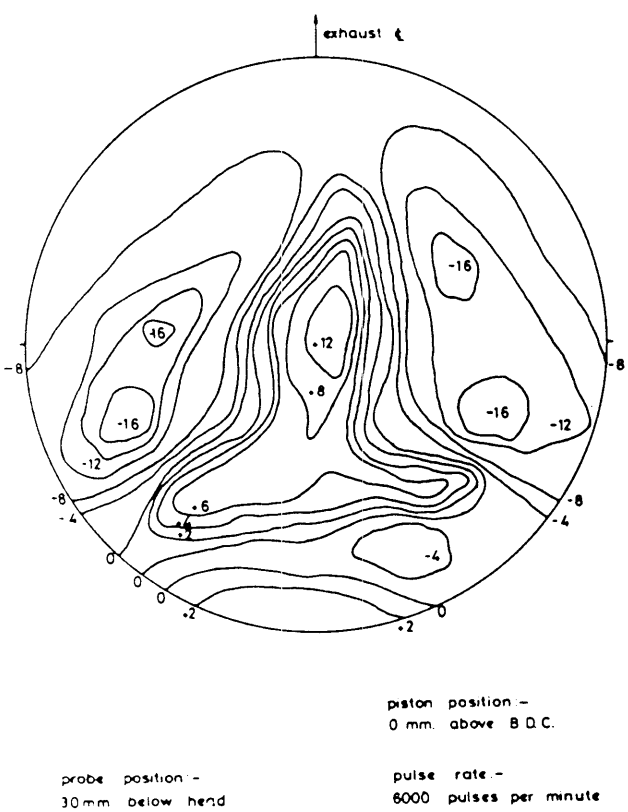

Fig. 3B - Pressure (in. water gage)–probe 30 mm below head level

shows what Jante would describe as a wall tongue scavenge system. However, Fig. 3B, taken 30 mm below the head, shows a strong central jet core of transfer fluid, as does Fig. 2 (top). Fig. 3B also shows the two downward loops above the side transfer ports. Near bdc the central core is still in evidence in Fig. 3C, with the two side downward loops nearer to the exhaust port. There is also a curious negative loop near the rear wall. The values of pressure contour shown are in inches of water gage.

The actual design of exhaust and transfer ports can be effected by one of two techniques. A complete cycle analysis can be employed, taken from maximum cylinder pressure, through expansion with heat transfer to release, then sonic followed by subsonic outflow, followed through transfer blowdown and exhaust pipe suction, then exhaust to cylinder charging, trapping, compression with heat transfer, to ignition, squish, and combustion to the initial starting point. This complex thermodynamic cycle together with the gas dynamic procedure is as yet incomplete. The complete cycle analysis, of course, is the engine designer's requirement, and many researchers are working toward this goal. However, if one examines, using digital computational procedures to simplify the arithmetic to a prepared scheme, the blowdown and transfer time areas of many engines with a high bmep output, it becomes obvious that there is a pattern that relates these variables to the engine power output. In fact, the scatter for a bmep range from 80-130 lb/in.^{2} is remarkably small. At

Queen's University this pattern relating bmep to porting time areas for any cylinder size was carried out in 1966, and besides adding to the library of information in the intervening period, the analysis pattern was incorporated in the design and was proved successful. This is also a rapid computational procedure, economical of computer time, and is recommended for industrial design application. The result of this analysis, too lengthy to present here in useful form, was, for the port plan shown in Fig. 1,

Exhaust port timing = 199 deg
Transfer port timing ≐ 142 deg

These timings were for a designed bmep of 105 lb/in.^{2} at 7500 rpm.

Design of Cylinder Head - It has been determined experimentally that the effect of squish benefits both the combustion process rate of pressure rise (this is shown in retarded ignition timings for a higher bmep) and reduction of the thermal loading on the piston. The latter effect is evidenced by the pattern on the piston surface after running at full power. This shows a considerably lower temperature condition on the squish band by comparison with the combustion zone; consequently, there is a heat extraction effect from this area toward the ring band, the piston sides, and thence to the cylinder wall.

The squish band area has been standardized at Queen's University to approximately 50% of the bore area. The resulting physical dimensions of the chamber, being segments of a sphere, are calculated by the computer for a given trapped compression ratio, based on a calculation such as that for exhaust port height and timing, with input data registering squish area proportion, piston to head clearance, piston crown radius, and combustion zone edge radius. The result gives a position of cut center and cutting radius; if necessary, computer output could be connected to a numerically controlled machine tool for automatic profiling. Both combustion zone and squish band are concentric with the cylinder axis, with a centrally located spark plug (usually, for the test bed, a Champion N54R).

Computer solution of the arithmetic is advantageous because the calculation for either the combustion zone profile radius or its center involves solving a third-order equation, with an overall boundary condition on the intersection of the zone profile with the piston crown profile to provide the correct squish area. Although the algebra is simple, the arithmetic solution is tedious, and the automatic computation of the correct profile to obtain the required clearance volume saves valuable time. Moreover, machining accuracy is well within any experimental volume-measuring technique. Other combustion chamber profiles such as flat topped cylinders, or cones, involve easier arithmetic, but suffer the disadvantage of having higher surface/volume ratios and hence run the risk of charge quenching, with consequent power loss.

The actual trapped compression ratio is, at the moment, 7.75:1, but 8:1 and 8.25:1 ratios will be used in later development work. The piston to head clearance is 1 mm. The ignition system is an Autolite contact breaker operating off a specially profiled contact cam, with an Autolite oil-filed coil delivering 15 kV from 6V battery. A sketch of the cylinder head is shown in Fig. 4.

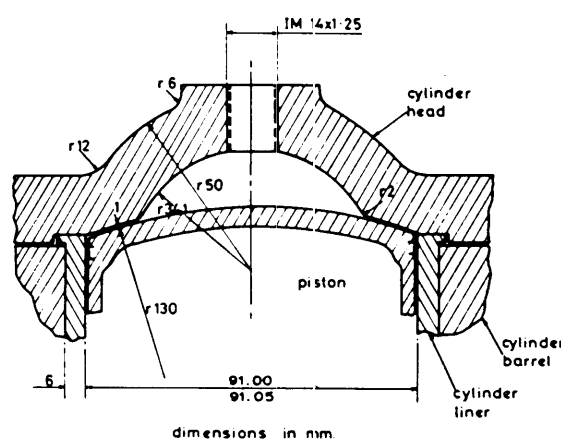

Fig. 4 - Cylinder head dimensions

Exhaust System Design - The design of the expansion chamber exhaust system for this type of engine is critical and has been discussed in detail by Blair and Johnston (6 and 7) and by Naito and Taguchi (8). The exhaust system designed here, although it could have been effected by the complete unsteady gas dynamic analysis outlined in Ref. 6, was calculated on the basis of simplified empirical design criteria explained in Ref. 7. Two exhaust systems were used to move the position of peak bmep a few hundred rpm apart. A sketch of these systems is shown in Fig. 5, and engineers interested in this area may wish to use them as sample calculations for the criteria used in Ref. 7 and for the variables listed here.

MECHANICAL DESIGN - The features of the engine are discussed and any unusual material or component is described.

Piston and Cylinder Liner Design - The basic piston layout was set out and the final working drawings, casting, and machining were done at Hepworth & Grandage of Bradford. The piston alloy is hypereutectic, containing 22-25% silicon, cast to HG416 specifications. The gudgeon pin is a case-hardened steel, En36, retained by wire circlips. Two Dykes' rings are used, the top ring forming the top or leading edge of the piston; the material is soft iron, HG303-alloy quenched and tempered as castings to give a matrix of spheroidized pearlite. After machining the ring is tin plated all over. The piston crown is slightly domed, principally to direct the gas from the squish area toward the propagating frame front from the spark plug. A flat-top piston does this ineffectually.

The cylinder liner is machined from blanks centrifugally cast in nonheat-treated gray iron. The alloy is HG 101 with approximate content 3% C, 2% Si, 0.5% P, 0.75% Mn, and 0.5% Cr. It is ideal for cylinder liner construction. The finished piston to liner clearance cold is 0.003 to 0.0035 in. on a bore of 91.05 mm. After all ports are machined, the liner is shrunk

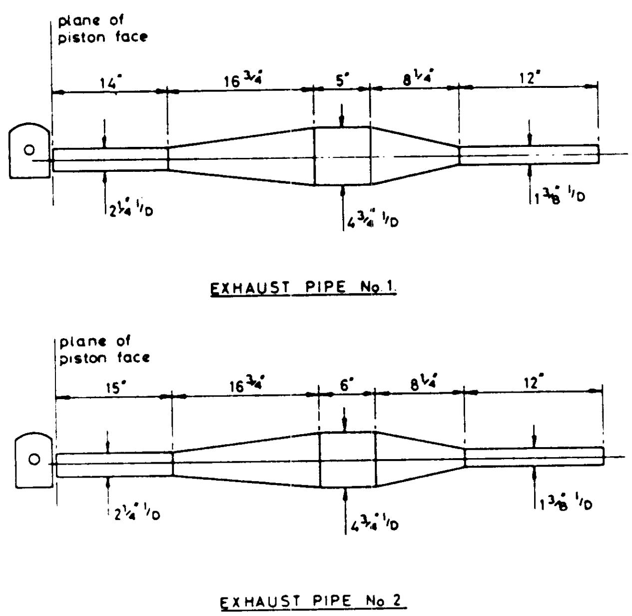

Fig. 5 - Dimensions of exhaust pipes 1 and 2

into the sand-cast aluminum alloy cylinder barrel (LM6 alloy, with a silicon content between 10% and 13%) and hone-finished.

Connecting Rod Design - The connecting rod is a thin I-section design with the maximum tensile stress on the surface of the rod, due to end load and bending between centers, calculated by a digital computer program. A rod of any I-section can be analyzed when the small end mass and the maximum engine rpm are known. Mathematically, the I-section is split into segments and whirled about the crank throw for 360 crankshaft degrees, with the small end mass attached. A printout of the following data is available for every 10 deg of crank rotation: stress at the big and small end eyes, big and small end bearing loads, piston side thrust load (friction neglected), and the maximum tensile stress on the surface of the I-section due to both reciprocating load and bending under the rod's own I-section mass, and the position of that maximum stress.

The material for the rod is KE805 (composition 0.35% C, 1.0% Cr, 1.5% Ni, 0.2% Mo) and is left in the fully annealed condition at 65 tons/in.^{2} UTS with an Izod test value of 35 ft-lb. The maximum tensile stress on the rod surface was calculated at 33,000 lb/in.^{2} at 8000 rpm for a connecting rod having a total weight of 0.740 lb. The small end bearing is an uncaged "drawn cup" Torrington bearing for an 18 mm diameter gudgeon pin. The big end eye is sleeved with a through-hardened shell for the Dürkopp type of Ransome-Hoffman-Pollard caged needle roller bearing on a 28 mm diameter crankpin. There is a slot in the big end to allow petrol access to the bearing.

Crankshaft Design - The crankshaft is of the pressed-up design. The main shafts are machined from KE805 in the same condition as the connecting rod and pressed into the flywheels with a 0.003 in./in. interference. The flywheels are machined from EN36 (composition 0.15% C, 0.1/0.35% Si, 0.3/0.6% Mn, 3.0/3.75% Ni, 0.6/1.1% Cr) and case-hardened all over. The crank pin, 28 mm diameter, is also made from EN36, which is case-hardened and ground to a hardness of Rockwell 62C and a surface finish of 12-14μ in CLA. The crank pin interference in the flywheels is similar to that for the main shafts.

The balance factor for the engine is 50%, a useful starting point for any new design; so far, vibration in the motorcycle frame with the engine running, or on the test bed during testing, is present but not objectionable.

EXPERIMENTAL WORK

TEST BED TESTS - The early development tests were confined to setting the carburetor float level to a standard position, obtaining a carburetor needle profile that would allow the engine to come onto the power band smoothly. Then, after some initial jetting tests on the Amal 42 mm "concentric" model carburetor, the optimum position of ignition advance at the calculated peak bmep position, 7500 rpm, was determined. After this, the No. 1 exhaust system was fitted and the power/torque curve shown in Fig. 6 was obtained.

It is interesting to note that the power/torque curve was recorded exactly nine months after the initial contemplation of the engine construction, and was the first real experimental test on the engine. During the test, porosity was observed on the main bearing housings on both timing and drive sides of the unit, and the cylinder head was later discovered to have been leaking slightly. The optimum ignition timing was found to be at 23 deg btdc, with some 6% fall-off in torque at 22 deg btdc and about 3% fall-off at 24 deg btdc. Power points were taken at 200 rpm intervals from 6600-8000 rpm.

The exhaust pipe was adjusted, as shown in Fig. 5, to be exhaust No. 2, and was set to lower the speed for maximum torque. The result of the power test, with the ignition timing still an optimum at 23 deg btdc, is shown in Fig. 7. This gives an increase in bmep from 101 to 104 lb/in.^{2} and with peak power relatively unchanged. The target design bmep of 105 lb/in.^{2} is fractionally less than 1% low at the design speed of 7500 rpm. Thus the engine has a specific output of 122 bhp/l and, as the complete engine weight is 52 lb, a specific output of 1.2 bhp/lb.

After test bed testing, and some considerable road-race mileage, the engine appears to be quite reliable. The piston, particularly, has shown no tendency whatsoever to seize, which, considering the bore size and the high piston speed (4000 ft/min at 8000 rpm), was earlier thought to be the major problem. Both on the test bed and the road, the engine starts easily and runs satisfactorily on a Champion N54R spark plug. The fuel used is 101-octane pump gasoline containing a half-pint to the gallon (16:1) mix of Castrol R40 (castor base oil) as lubricant for the bearing surface. Apparently a leaner oil/gasoline mix would be perfectly acceptable to the engine, but this has not yet been tried experimentally.

ROAD TESTS - The engine is installed in a frame constructed by Colin Seeley Racing Developments from Reynolds

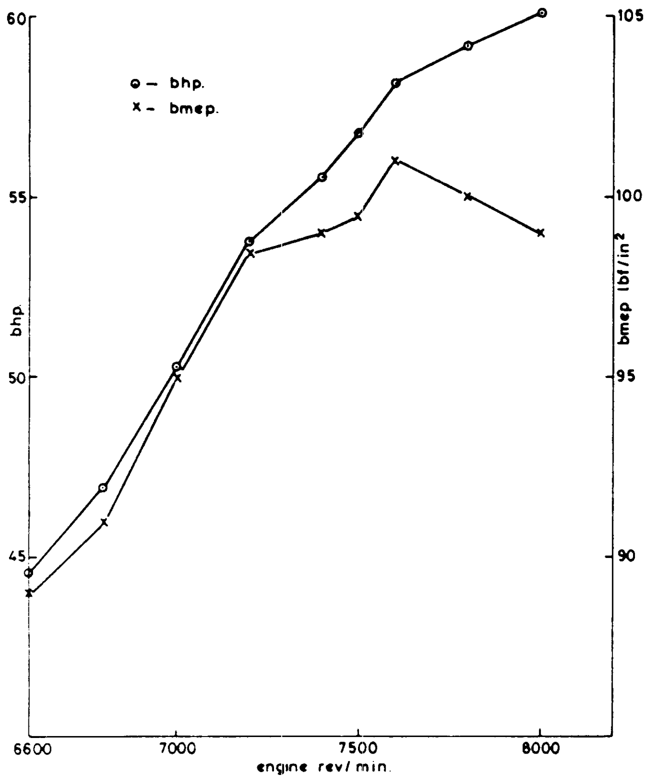

Fig. 6 - Exhaust pipe 1

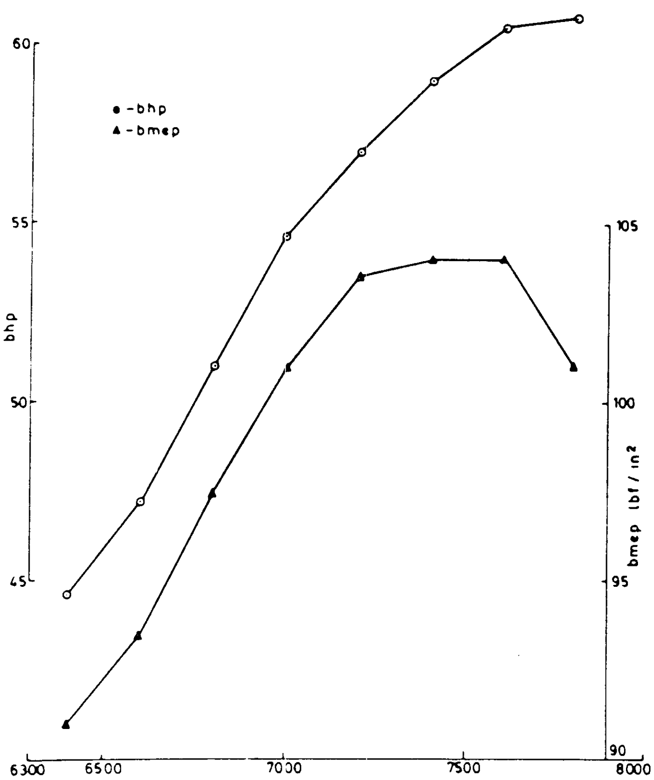

Fig. 7 - Exhaust pipe 2

531 tubing; the small total height of the engine (15 in.) allows an upswept expansion chamber location. A photograph of the complete machine shows the engine location, Fig. 8; another

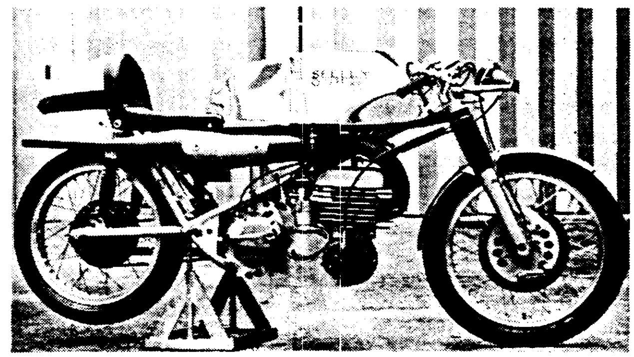

Fig. 8 - The complete machine

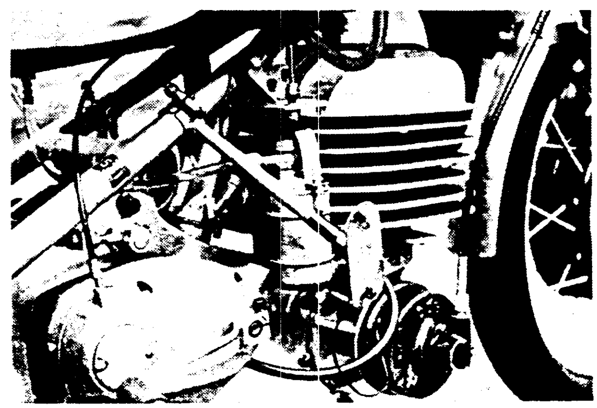

Fig. 9 - Engine closeup

view, Fig. 9, shows the 42 mm carburetor being fed from a remote float bowl, antivibration mounted. The forward placement of the engine and the resulting excellent weight distribution, combined with a low center of gravity for the complete machine (total weight, dry, 255 lb) makes for good vehicle cornering characteristics. The narrow engine allows a slim aerodynamic profile, and has been run at 145 mph in a straight line under road-race conditions.

CONCLUSION

The viability of the concept of a high specific output and large displacement cylinder appears to be vindicated. In fact, with some further attention to detail regarding scavenge and combustion characteristics, a bmep of 110 lb/in.^{2} at 8000 rpm appears to be a distinct possibility, yielding 67 bhp or 134 hp/l as a specific output. There also appears to be no reason why a single-cylinder engine of larger displacement should not perform just as successfully and provide specific outputs above 1.2 hp/lb with an engine using die castings for cylinder head, barrel, and crankcase instead of the heavier sand-cast items employed here.

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