Some Development Aspects of Two-Stroke Cycle Motorcycle Engines*

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ABSTRACT

This paper describes aspects of YAMAHA 2 cycle, high speed, high output engines. Generally speaking, in order to obtain good results in developing engine performance, high delivery ratios and high thermal and mechanical efficiencies are essential. In addition to these, the most suitable cooling and lubricating systems must be employed. YAMAHA has developed a separate and automatic lubrication system for 2-cycle gasoline engines, which keeps YAMAHA engines well lubricated.

.NCE YAMAHA PRODUCED its first motorcycle, about twelve years have passed. The performance development of YAMAHA 125 cc engines during this period is shown in Fig. 1. As seen from this figure, the trend toward higher power outputs is remarkable. For example, the liter horsepower has increased much more than two times during this period. This trend is not a characteristic only of YAMAHA engines, but also applies to the entire motorcycle field. Moreover, the liter horsepower of the GP racing machines is over 250 and is now approaching 300.

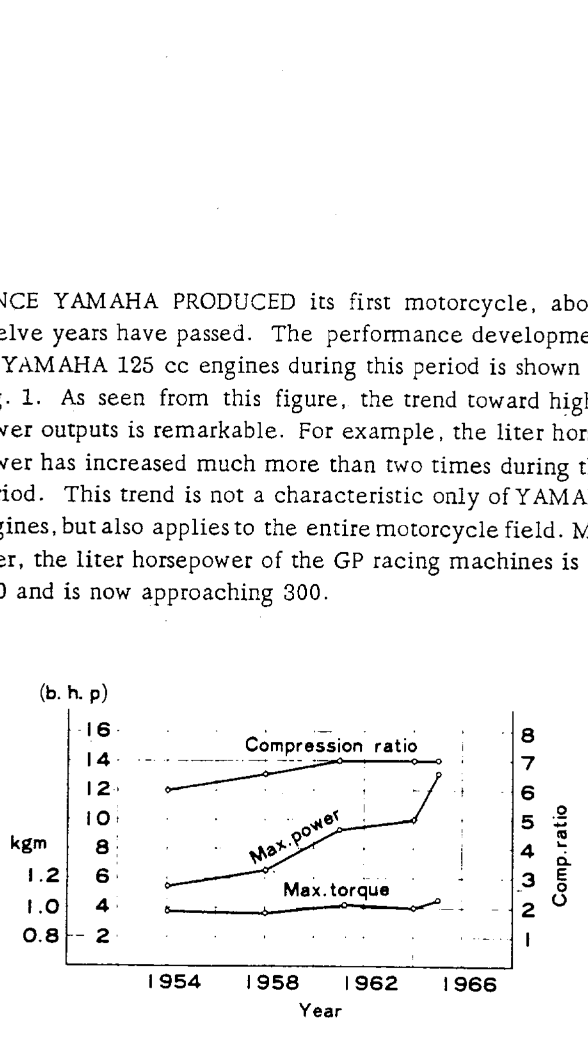

Fig. 1 - Performance trends of Yamaha 125 cc engine

Various methods have been investigated for obtaining high outputs. These include: increasing the air intake passage diameter of the carburetor, employing a rotary disc intake valve, choosing a high compression ratio, selecting the best timing for the intake valve and the scavenging and exhaust ports, and improving the exhaust system. Consequent improvements have resulted not only from continuous theoretical research, but also from numerous experiments and trials.

In spite of the intense efforts of many engineers and scientists to solve theoretically and experimentally the various difficult problems involved in small 2 cycle engine design, all the information necessary for design of an engine has not yet been obtained. Therefore, referring to the technical papers for qualitative data, the conventional "make-and-try" method is still the best approach to development.

INTAKE SYSTEM

It is well known that the intake and exhaust systems markedly influence the performance of 2 cycle engines. The usual investigations of intake systems are almost entirely limited to the pressure changes in the crankcase and the inertia and pulsation effects in the intake pipe. In the production motorcycle, owing to the restrictions of the layout for the whole vehicle, the dimensional freedom in intake system design

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is very limited, especially since the recent adoption of the rotary disc valve for controlling the intake system. The disc valve and the carburetor are connected with a very short pipe in order to avoid an increase in engine width, and the engine has no so-called intake pipe before the carburetor. The carburetor is usually put in its separate chamber and a kind of intake pipe is formed between this chamber and the air cleaner. Owing to the complicated intake system, precise theoretical calculation is hardly possible.

Moreover, it has been verified by various experiments that such an intake system does not show inertia and pulsation effects to any great extent. Hence, there are only two ways to get an effective intake system:

1. Make the area of the intake passage as large as possible.
2. Reduce the air flow resistance in the passage by all possible means.

At the present time we have not yet advanced to the point where intake effects can be analyzed positively.

In addition to these measures, another important item is the selection of the proper port timing. Except for the GP racing machines, the production 2 cyl engine usually employs the conventional piston valve system, owing to the difficult problems concerning engine width and electrical equipment.

The output change caused by intake valve timing is shown in Figs. 2 and 3 for a 125 cc rotary valve 1 cyl engine. In Fig. 4 it is shown for a 250 cc piston valve, 2 cyl engine.

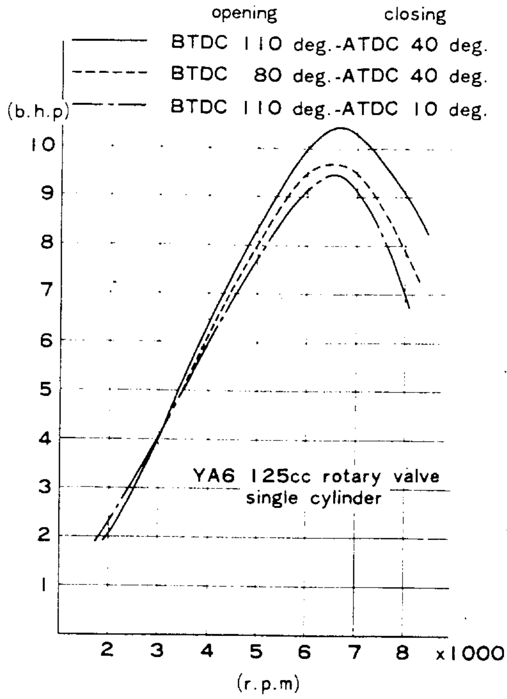

Fig. 3 - Output change caused by rotary intake valve timing

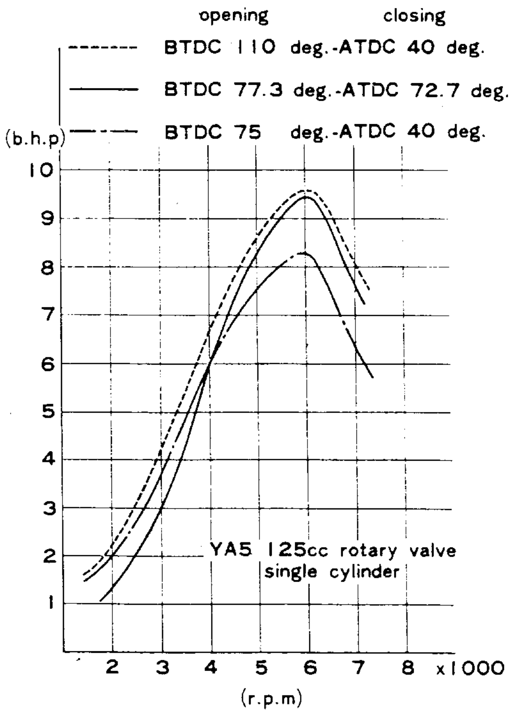

Fig. 2 - Output change caused by rotary intake valve timing

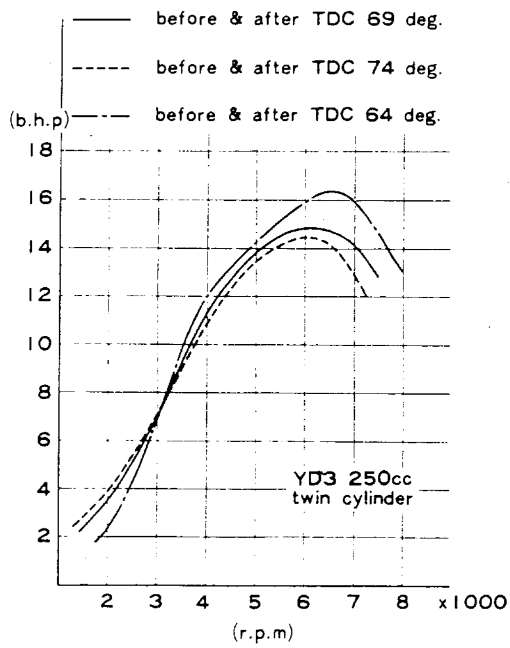

Fig. 4 - Output change caused by piston intake valve timing

ROTARY DISC INTAKE VALVE

The adoption of the rotary disc intake valve in small 2 cycle engines is considered to be one of the great recent advances in the motorcycle engineering field. The rotary disc valve has improved engine performance so much as to make it possible to compete with 4 cycle racing machines. Figs. 5 and 6 show the pressure changes in the crankcase of the conventional piston valve and rotary disc valve engines, respectively.

Because the intake port of the piston valve engine opens and closes in the symmetrical positions relative to both TDC and BDC, the opening time of the intake port is automatically determined when the closing time of the port is chosen for the maximum delivery ratio. Therefore, if the pressure in the crankcase and the pressure just before the intake port are chosen to be nearly equal at the closing of the intake port, in order to increase the delivery ratio, at the opening of the intake port the pressure in the crankcase is only slightly lower than it is in the intake pipe. Under this condition one cannot effectively utilize the negative pressure in the crankcase for the intake process. When the intake timing is selected for high engine speeds, at low engine speeds the charge blowback from the crankcase at the closing of the intake port and the delivery ratio decreases significantly. The abovementioned matters are the disadvantages of the piston valve system.

In the case of the rotary disc valve, the timing of the valve opening and closing can be decided independently of each other, the negative pressure in the crankcase can be used effectively, and the delivery ratio is improved as the result of the increased time area. In this case, even if the rotary valve timing is determined most suitable for high engine speeds, the blowback at valve closing at low engine speeds is relatively small and the delivery ratio is hardly decreased because the valve closes early in the crankcase compression stroke.

The early closing of the rotary valve means an increase in the crankcase compression ratio, and the combustion chamber is very well scavenged, owing to the increased scavenging pressure. Thus, the output can be increased by more than 20% compared with the conventional piston valve engine. A by-product of the decreased blowback at low engine speeds is that the carburetor specifications with which the optimum air-fuel mixture is obtained can be decided very simply, and at the same time the specific fuel consumption is decreased. These are great advantages.

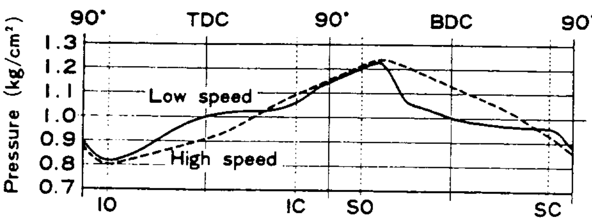

Fig. 5 - Pressure in crankcase (piston valve)

An output comparison of rotary and piston valves is shown in Fig. 7 for a 125 cc, 1 cyl engine. If one attempts to get a relatively high output at low engine speeds, one cannot help but get a low output at high speeds. The reverse is also true.

COMPRESSION IN CRANKCASE

The relation between the compression ratio of the crankcase and the delivery ratio has been made clear to some extent theoretically and experimentally by Prof. Nagaro.* According to his paper, if the compression ratio of the crankcase increases, the engine speed at which the maximum delivery ratio is obtained moves to a higher value in proportion to the square root of the crankcase volume, without changing the absolute value of delivery ratio. Hence, the crankcase volume should be as small as possible to realize a high speed and high output engine. However, because

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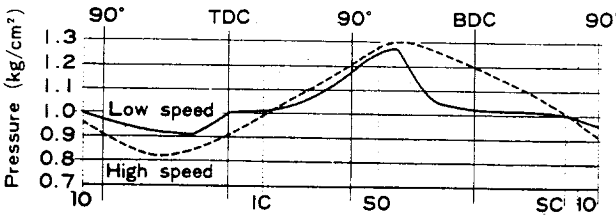

Fig. 6 - Pressure in crankcase (rotary disc valve)

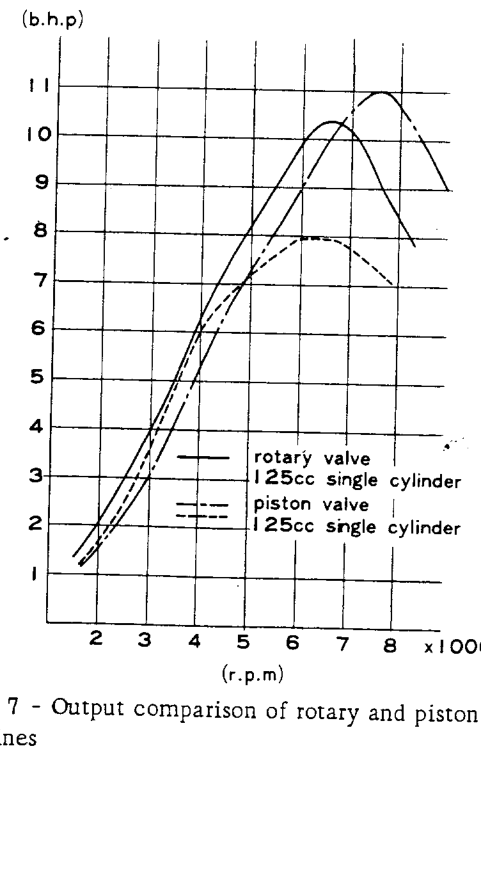

Fig. 7 - Output comparison of rotary and piston valve engines

of various restrictions, the crankcase volume cannot be reduced enough.

There are several methods to make the crankcase volume smaller:

1. Decrease the crank diameter.
2. Shorten the distance between the crank webs.
3. Fill the balancing holes in the crank webs with light metal plugs.
4. Integrate the crank webs tightly with the crankcase.

Even if all these measures are taken, the increase in engine output can be barely observed in bench tests.

TIMING OF SCAVENGING AND EXHAUST PORTS

Since the performance of the 2 cycle engine is greatly influenced by the exhaust system and also by the timing of the scavenging and exhaust ports, the engine output can be increased in most cases through improvement of the exhaust system.

The fundamental approach toward determining the port timing is that the time area should be suitable to the characteristics of the engine. It is very easy to understand that the higher the speed of the engine, the greater the time area requirements. It is therefore necessary to increase the angle area. There are two ways to increase the angle area in a given engine. One is to increase the width of the port and the other is to add to its height. Even if the same angle area is obtained for a given engine by using these two different methods, the results will be quite different. The port width and height are factors independently affecting engine characteristics. Judging from the experimental results, the height of the port has the greater influence upon engine performance.

Port Width - The main factors that should be considered in designing port width are:

1. The durability of the piston ring decreases with increased port width, owing to the protrusion of the ring into the port.
2. The fuel loss increases, owing to the "short circuiting" of the fuel mixture from the scavenging ports to the exhaust ports.

Taking the protrusion of the piston ring into consideration, the allowable opening angle of the port around the cylinder center is 70 deg maximum. In the production engine it is 65-66 deg. The distance between the scavenging and exhaust ports must be determined by experiment because this distance depends on the opening angle of the scavenging port. As explained above, the port width is chosen proportionate to a certain diameter of cylinder. Therefore, for increased time area and smoothed gas flow, multicylinder engines must be adopted.

Port Height - The port height determines essentially the port timing, and it is very important not only for acquiring the necessary time-area, but also for utilizing the inertia effect of the pulsation effect. Broadly speaking, the characteristic of the 2 cycle engine depends mostly on the exhaust timing. However, it is not advantageous to open the exhaust ports excessively early because this will decrease the effective piston stroke, and therefore a compromise must be made. From experimental results, the optimum timing of the exhaust port is about 100 deg before and after TDC, as shown in Fig. 8.

As for the scavenging port timing, even if the exhaust port timing is advanced to increase the power output at higher engine speeds, the optimum scavenging port timing will not be necessarily greatly changed, since the interval between the exhaust and scavenging should also be increased.

SCAVENGING METHOD

The scavenging method used for recent small 2 cycle gasoline engines is almost standardized in the Schnürle or loop type, with the notable exception of some outboard engines. It is preferred mainly because it minimizes the severe thermal load at the top of piston. In accordance with the trend toward high output engines, various scavenging types based on the Schnurle method have been tried.

Considerable research and experiments performed concerning gas flow, dimensions and other characteristics show that all types have nearly equal scavenging and charging efficiencies. Hence, a great increase in engine output can hardly be expected by improving the scavenging method alone.

COMBUSTION CHAMBER SHAPE AND COMPRESSION RATIO

The effective stroke volume in 2 cycle engines is small, and therefore the combustion chamber volume must be small

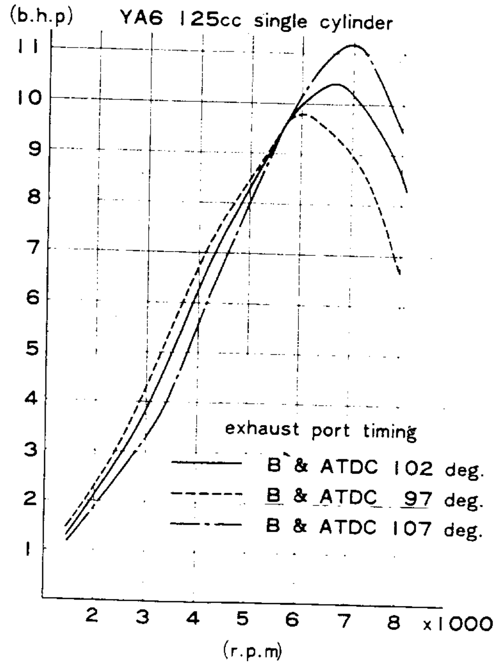

Fig. 8 - Output change caused by exhaust port timing

for a high compression ratio. This makes it difficult to give a special shape to the combustion chamber. From our experimental results, it is known that differences in the shape of the combustion chamber have no great influence upon the engine output, if the chamber volumes are equal. At present, therefore, we use the relatively simply shaped hemispherical combustion chamber, which can be easily manufactured.

The upper limit of the compression ratio is determined not only by knocking, abnormal combustion phenomena, and other indicators (as in 4 cycle engines), but also by the effects of the thermal load on the durability of the piston, spark plug, and cylinder head. The upper limit of the effective compression ratio is considered to be 7.5:1 for production engines, 8.0:1 for engines for sport use, and 9.5:1 for racing engines.

DYNAMIC EFFECTS DUE TO THE EXHAUST SYSTEM

The special purpose of the exhaust system is to increase the delivery ratio and the scavenging efficiency by utilizing certain phenomena induced in the exhaust pipe. Consequently, it results in an increase of engine output due to a higher mep that is obtained as the result of retaining as much new air-fuel mixture as possible in a given piston displacment.

With an increase in engine speed, a larger time area of the exhaust port is necessary, which in turn causes a shorter effective piston stroke. Nevertheless, it is not impossible to get a charging efficiency greater than 100%, owing to the utilization of special exhaust system effects. The importance of the influence of the exhaust system upon the performance of the 2 cycle engine has been recognized for a long time, and various types have been tested.

What must be brought to mind here is the fact that the exhaust system by itself cannot sufficiently display its effect unless the system is properly designed by carefully considering the coordination of the systems involving movement of gas intake, scavenging, and exhaust.

Each exhaust pipe system has its own natural frequency, and this is the basis of one of the theoretical explanations of the functioning of the exhaust pipe system. Generally speaking, it is desirable that the period of the natural frequency of the exhaust pipe be made equal to the period of the scavenging process. It is considered that the air-fuel mixture is pushed into the cylinder by the aid of low pressure induced in the exhaust pipe. In this case a more advantageous condition is that in the first half of the scavenging process, the local low pressure in the exhaust pipe should assist removal of the combustion gas while taking the mixture into the cylinder at the same time, and in the latter half of the exhaust process, the local high pressure in the exhaust pipe should prevent the new mixture from getting into the exhaust pipe. As the effect of this mechanism is influenced greatly by the port timing, good selection of port timing might be very effective.

The next explanation involves the use of the inertia effect of gas. In this case, although the influence of the exhaust port is great for increasing the output of the engine, the inertia effect of gas in the cylinder can be also effectively used for the same purpose by helping the gas flow through pulsation in the exhaust pipe system and by choosing the shape of pipe suitable for reducing the gas flow resistance as low as possible.

The abovementioned parameters are desirable conditions for the exhaust system of 2 cycle engines, as stated by many investigators. According to the theory, even a straight line can be used for the exhaust pipe with a rather good effect. Moreover, if the shape of the pipe is modified somewhat (in line with the characteristics of gas flow and calculations based on the theory of pulsation and also with an intention to decrease the flow resistance), a divergent exhaust pipe with a length matched to the engine will satisfy the theory.

What is the existing state of exhaust systems? For YAMAHA GP racing engines, the exhaust system is constructed by combining a tapered exhaust pipe and a large expansion chamber in series, and narrowing it again at the end. Here, the exhaust pulse supercharging mechanism is clearly shown. Owing to the throttling at the end of the exhaust system, a high back pressure is incurred by this exhaust pulse, and its reflected pressure wave brings about a high pressure in the exhaust pipe near the exhaust port at the closing of the scavenging port. Consequently, this pressure pushes back the air-fuel mixture into the cylinder, preventing it from escaping into the exhaust pipe.

In addition to exhaust pulse supercharging, exhaust energy supercharging should be discussed. This is based on the theory that the high energy of the exhaust gas can be also used effectively for increased output. When high temperature and high speed gas is ejected from the exhaust port and led into an exhaust pipe of diverged shape, the gas speed gradually decreases and the kinetic energy of the gas is changed to pressure energy. If throttling now occurs, a high pressure is produced. Under the utilization of the reflected pressure wave, the pressure at the time just previous to the closing of the exhaust port becomes very high, and it pushes the air-fuel mixture back into the cylinder.

In the practical case, since extreme throttling of the exhaust pipe disturbs the exhaust gas flow, it is recommended that a tapered part of the exhaust pipe system is utilized for the same purpose. The fact that the liter horsepower of YAMAHA GP racing engines is above 250 could not be achieved without the aid of an effective exhaust system.

Next we want to discuss the present practice with regard to exhaust systems. As described above, there are several theoretical explanations of how exhaust systems function, but many difficult phenomena still remain to be solved. Therefore, the ideal exhaust system can be established only through many successful tests. The YAMAHA GP racer is equipped with a divergent exhaust pipe; the production racer (TDI-B) and production motorcycle use a cylindrical exhaust pipe. Generally speaking, in the exhaust system with a cylindrical exhaust pipe, length has a minimal influence upon the engine speed at the maximum output; in the divergent exhaust pipe, it is essential to design the taper properly. Naturally the shapes of the expansion chamber and the throttling section are also important.

In line with the trend toward higher speeds and higher outputs, shorter exhaust systems are adopted. It should not be thought, however, that an increased engine output can be obtained by shortening the exhaust system excessively, since this measure decreases the torque and diminishes the performance at both low and high engine speed unless it matches with engine characteristics.

For example, if the exhaust system for the GP racing engine is attached to the YDS-3 "Motorcross" engine, a power output equal to that of the GP racing engine naturally cannot be expected. As shown above, the performance of a 2 cycle engine cannot be discussed without describing the exhaust system. In the future, supercharging thst uses the special effects of the exhaust system will have still greater importance for boosting performance.

The influences of the exhaust system on engine output are shown in Figs. 9-11 for a 62.4 cc, 1 cyl engine.

COMBUSTION AS AFFECTED BY TRANSISTOR IGNITION

With the conventional ignition system, the higher the engine speed, the more unstable the action of the mechanical contact breaker. Namely, the breaker cannot follow the movement of the cam. In order to avoid this defect, an ignition system without a contact breaker has been developed and is already used for some engines. We also have investigated three "breakerless" ignition systems using transistors. These three methods are as follows:

1. Transistor Cutoff Method. The signal generated by the signal generator is amplified by the signal amplifier and works as a switch in the primary transistor circuit. When the "switch-transistor" turns off the current, high voltage is generated in the ignition coil. Germanium is used for the "switch-transistor."

2. This method is almost the same as the cutoff method except that silicon is used instead of germanium for the "switch-transistor."

3. Transistor "ON" Method. The "dc-dc" transistor converter generates 300 v to charge a condenser. At the time of ignition, the output signal from the signal generator is fed to the gate of the control rectifier (SCR), turns on the SCR, and permits the electricity stored in the condenser to flow into the primary coil, which in turn induces high voltage in the secondary. The wave pattern of the voltage, measured in still air by the three needle gap method with a constant gap of 8 mm, is shown in Fig. 12.

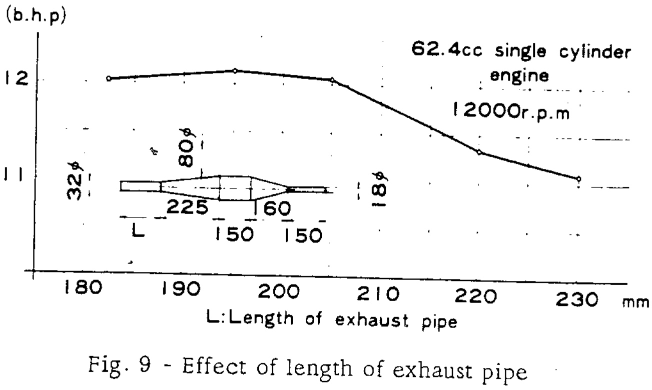

Fig. 9 - Effect of length of exhaust pipe

Of the various influences of these ignition systems upon engine performance, only the so-called wet plug problem will be discussed here.

Time intervals for voltage rise in the various systems are shown in Table 1. From the results of bench and running tests the following relationships have been made clear:

1. In the method whose time interval for voltage rise is below about 40 μsec, the wet plug phenomenon does not generally occur.

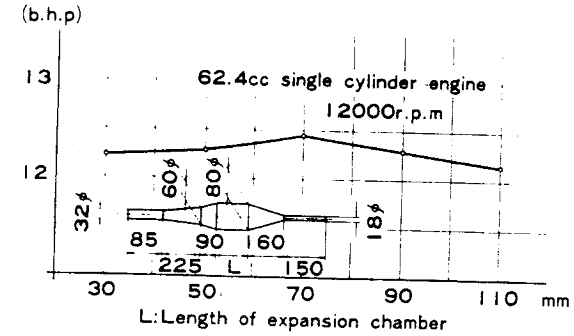

Fig. 10 - Effect of expansion chamber length

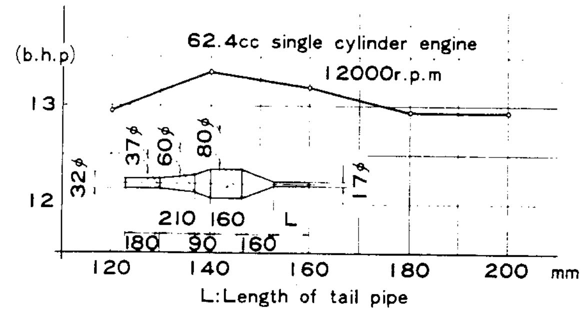

Fig. 11 - Effect of tailpipe length

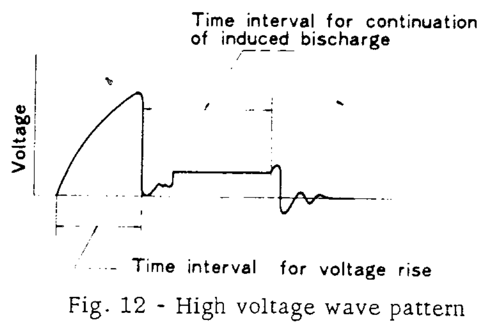

Fig. 12 - High voltage wave pattern

2. The wet plug phenomenon does not depend on the time interval for continuation of induced discharge. With a spark plug whose length of insulation is as short as that of the spark plug for racing engines, it seems that when the insulator is covered by much carbon and gasoline, the leakage current increases; because of this energy loss, the output voltage of the ignition coil does not rise and consequently causes misfire. In such a case, this phenomenon can be prevented if an electric supply with a large power output is introduced or the time interval for voltage rise is made shorter.

In the first case, the system has enough energy to produce the voltage rise, even if the current flows into the leakage resistance. In the second case, the system makes the voltage rise in a very short time interval, with relatively little energy, and decreases the energy that flows into the leakage resistance. It seems that, even if the systems have the same plug gap, the system with the smaller time interval for voltage rise has a higher discharge voltage and stronger spark than the system with the longer time interval. Thus, engines that have rich fuel mixtures, such as engines used for racing machines, have good ignition performance if ignition systems with short voltage rise time are used.

Table 1 - Time Intervals for Voltage Rise

Type of Ignition System	Interval for Continuation of Induced Discharge	Interval for Voltage Rise, μsec
Transistor Cutoff (Ge transistor)	1-1.5 msec	50-60
Transistor Cutoff (Si transistor)	1-1.5 msec	35-45
SCR Method	100 μsec	16-20
Magneto Ignition	0.6-1 msec	36-40

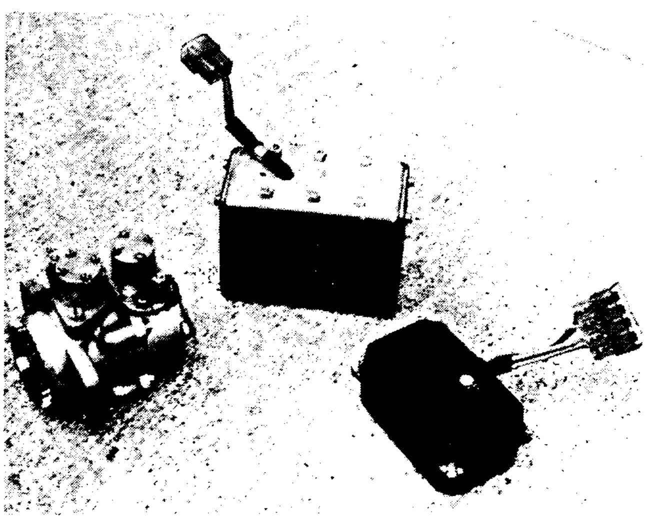

Fig. 13A - Transistor ignition assembly

COOLING

Engine performance, especially that of high speed and high output engines, depends greatly on cooling. We have investigated 2 cycle motorcycle engines ranging from air cooled engines with cast iron cylinders to water cooled engines with aluminum cylinders. A few of the problems encountered will be described below.

Years ago, all racing machines did not have cowlings, but nowadays all have cowlings to decrease the air drag. The narrowed inlet for the air intake, which was early introduced, must be made as small as possible to reduce the air drag; however, this tends to starve the engine of cooling air, so the lead-in air plate was introduced to force all cooling air to flow between the fins of cylinder. Without the lead-in air plate, part of the air flows between the fins and part is dissipated. The introduction of the lead-in air plate produced sufficient cooling with less air than that needed before, and consequently permitted us to narrow the air inlet in the cowling in order to improve styling and diminish air drag.

In our racing engines, not only is the outside of the cylinders cooled, but also the inside. In the latter, the heat of vaporization of the gasoline is used, that is, the gasoline is supplied to the cylinder with a richer mixture ratio than is necessary for maximum power in order to cool the cylinder, piston, and crankcase. When this method is used, naturally the combustion efficiency and fuel consumption become worse, but the acceleration is improved and the engine output does not decrease even with long term continuous running.

The high speed and high output engine, such as the one for the racing machine shows an output decrease of 10-20% with continuous running. It is naturally the same effect as increasing the engine output to hold this decrease as small as possible. At the time of starting the engine, as shown in Fig. 14, output is larger for the maximum power air-fuel ratio than that for the overrich air-fuel ratio, but after a few minutes this relation is reversed.

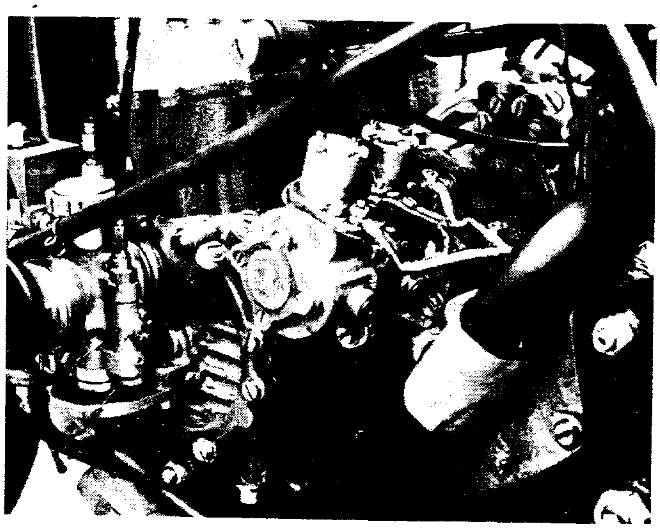

Fig. 13B - Signal generator mounted on engine

Recently, water cooled engines have been introduced to racing motorcycles. We used a water cooled 2 cyl engine in the 125 cc class of the T.T. race in 1965 and won a victory. The influences of water cooling upon engine performance as experienced by us are explained below. First, the comparison between water and air cooling is described.

For the case where engine constructions are the same except for the cooling method, almost identical outputs are initially obtained. However, as mentioned above, the output of our high speed and high output engine decreases gradually with continuous running. (As the causes of this phenomenon, various things could be pointed out, but are not referred to in this paper.) The aspects of this output decrease are shown in Fig. 15 for both water and air cooling. Considering that the decrease of output in the water cooled engine is over 50% less than that of the air cooled engine, the increase in the number of parts and weight should not be considered excessive.

Next, the higher the temperature of the cooling water, within limits, the larger the difference between the temperature of the atmosphere and the cooling apparatus becomes. Therefore, the smaller cooling apparatus utilized is advantageous in reducing the air drag. Although high temperature cooling has a tendency to lower the charging efficiency and induce abnormal combustion, its other merit should not be overlooked. This is the reduction in mechanical losses. Fig. 16 shows the relationship between the temperature of the cooling water measured at the outlet and the mechanical loss.

LUBRICATION

The trend toward high performance engines means that engine lubrication must be considered seriously. Because of the deleterious effect of the increase in piston speed and mep on the lubrication of the piston sliding on the cylinder wall and on the bearings of rotating parts such as the piston pin, crankpin, and crankshaft, the customary petroil (gas-oil) method is regarded as inadequate; consequently, a new method has been introduced.

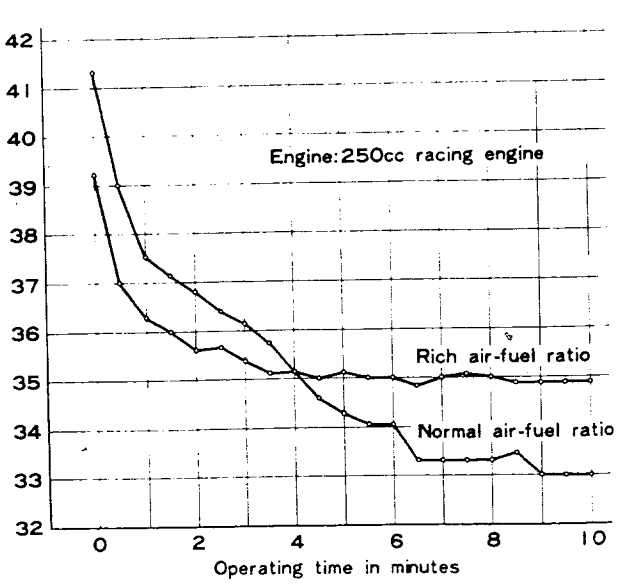

Fig. 14 - Effect of air-fuel ratio on engine performance

The racing engine is operated under such conditions as a maximum mep of $9\ kg/cm^2$, a mean piston speed of 20 m/sec, and an engine speed of 12,000-14,000 rpm. For these reasons the lubricating oil is delivered separately and forcibly by a special plunger pump, which has seven to eight delivery holes in the crankpin and crankshaft bearings, and in the rotary disc valve that opens and closes the intake port.

The oil is metered in proportion to engine speed, little by little, to lubricate and also cool these bearings. The amount of oil is adjusted to each type of engine and condition of operation. The oil is led into the crankcase after lubricating the bearings and valve, then joins with the oil that is introduced by the petroil method (mixture ratio of about 50:1) in order to lubricate the cylinder and piston, and finally is burned or escapes to the exhaust system. The minute quantities of oil involved must be delivered effectively to each part being lubricated.

The 2 cycle petroil engines have always had such problems as port blocking, piston seizure, piston ring sticking, and spark plug fouling. Furthermore, the difficulty of producing a petroil mixture and the problem of air pollution caused by rich exhaust smoke also exist. Thus, the petroil method presents many problems to be solved.

The oil injection system called "Autolube" is installed on engines for the production motorcycles. This is an application of the lubrication method introduced for the racing engines. The "Autolube" YAMAHA oil injection system operates in this manner: A special plunger pump meters the volume of oil according to engine speed and engine load, and injects oil from the nozzle into the intake pipe after the carburetor. Thus, a gasoline-oil mixture is created which is delivered to the parts in need of lubrication.

Fig. 17 shows this system installed on a 2 cyl 25 cc motorcycle engine. The pump drive is carried through the main shaft of the transmission gears to the worm gear, which rotates the worm wheel and distributor. The plunger stroke is regulated by means of a cable connected to the acceleration grip. The pump has two delivery holes, which deliver the metered oil separately. The total volume of oil is determined by the length and frequency of the plunger stroke. The frequency of the plunger stroke is always in proportion to the engine speed, and its stroke length varies from 0.25-2.2 mm by means of a cylindrical cam regulated by the same grip that controls the carburetor throttle valve.

Fig. 18 shows the limit of piston seizure in a 250 cc motorcycle engine for quarter, half and full loads. In this figure, the oil consumption curves are shown for the petroil method (20:1 mixture ratio) and for the YAMAHA "Autolube" oil injection system. In the petroil method, an insufficient quantity of oil is supplied at high speed and full load. The YAMAHA "Autolube" supplies enough oil at full load and only the required quantity of oil at partial load. It is

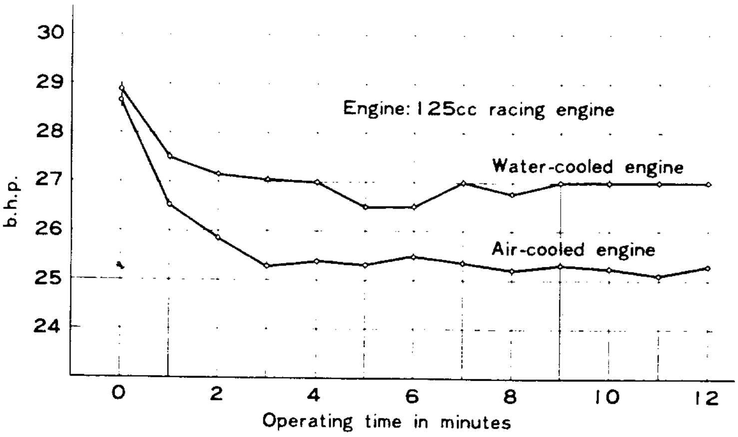

Fig. 15 - Effect of operating time on engine performance

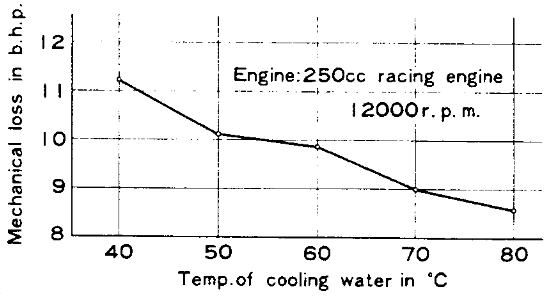

Fig. 16 - Effect of temperature of cooling water on mechanical losses

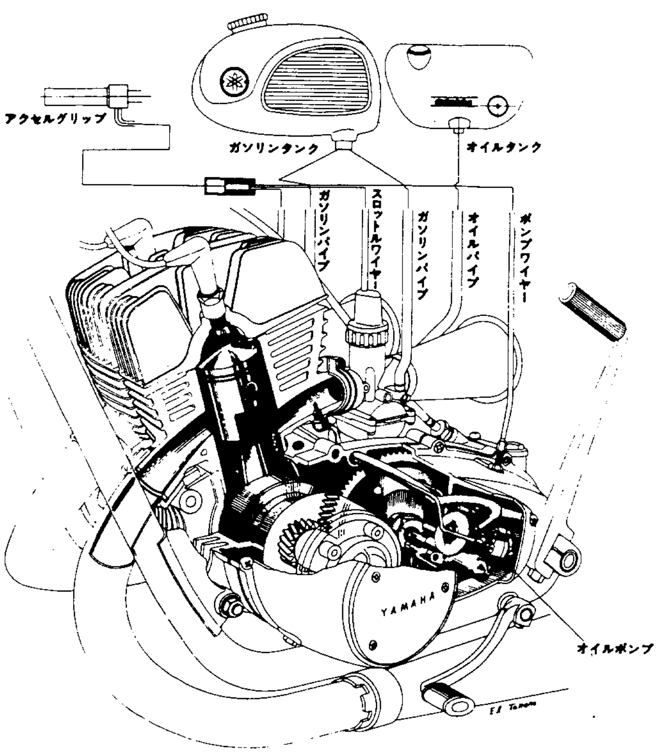

Fig. 17 - "Autolube" system installed on a twin-cylinder 250 cc engine

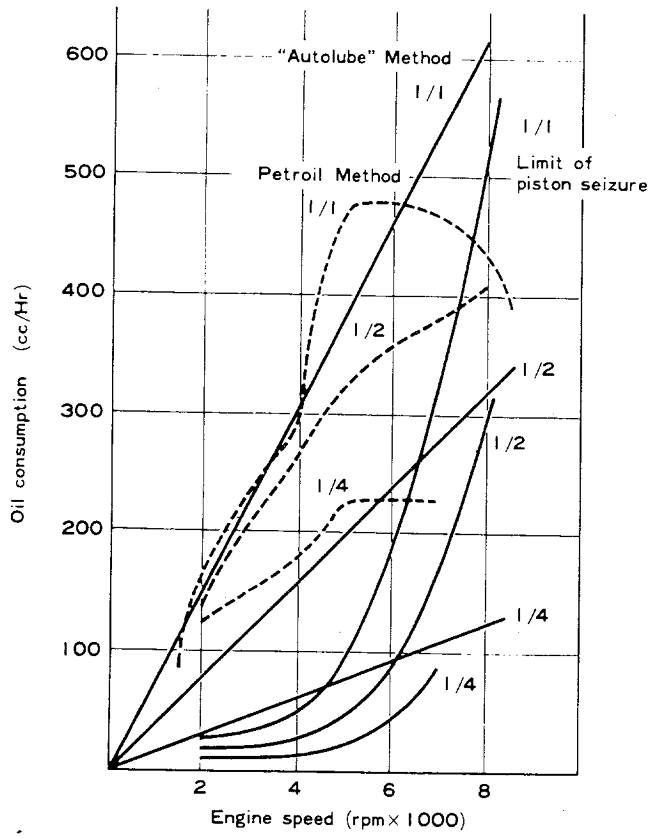

Fig. 18 - Piston seizure limits

a requirement that the pump must supply the oil at flow rates from 15 cc/hr to 640 cc/hr and deliver it equally to the two carburetors.

Fig. 19 shows the construction of the plunger pump and rotating distributor assembly. As the distributor rotates, the cylindrical cam provided at the opposite side of the distributor opens and closes the suction and delivery holes in the pump case. In order to vary the stroke length, the "adjust pulley" has a cylindrical cam fitted to it, and this cam pushes against a guide pin mounted in the pump case. The throttle, when turned, pulls the pump cable that rotates the "adjust pulley."

Fig. 20 shows the oil consumption and the fuel-oil ratio of a 250 cc motorcycle engine equipped with the YAMAHA "Autolube" system (constant speed operation on a paved level road). It is seen that the fuel-ratio changes from 70:1 to 13:1.

In the 2 cycle gasoline engine, it is still not known in detail how to lubricate the sliding and rotating parts and how to distribute the oil. Also, the movement of oil in the engine is unknown. These items are now being investigated experimentally, using radioisotopes. As a result of these tests, it is intended to consume the oil more effectively, with due consideration to improvements in materials, accuracy of parts, and structure.

OTHER PROBLEMS CAUSED BY THE TREND TOWARD - HIGHER SPEEDS AND OUTPUTS

Crankpin and Wristpin Bearings - Bearings pose difficult problems in high speed and high output engines, particularly the bearings for the small and big ends of the connecting rod. Until about two to three years ago, plain bearings made of phosphor bronze were extensively used for the connecting rod small end, but at the present time, needle bearings are used in all engines. This was brought about by the lubrication problem. On the other hand, needle bearings have always been used for the big end of the connecting rod. Generally speaking, rolling bearings function even with imperfect lubrication, in contrast to plain bearings. Naturally, there is a limitation to their use.

The technical difficulties occurring in the small and big ends of the connecting rod are the following:

1. Shock load must be endured.
2. The bearing of the small end of the connecting rod has an oscillating motion and that of the big end has a rotating motion. Large loads act on the rolling surface of the bearing.

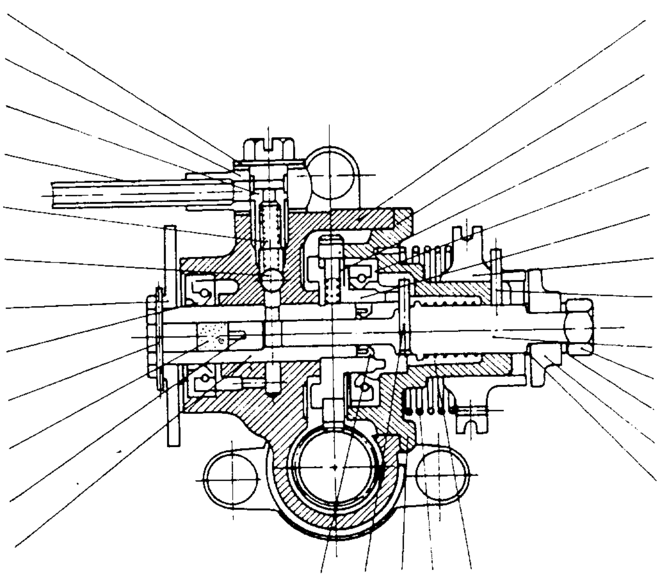

Fig. 19 - Pump and distributor construction

3. The operating temperatures are high.
4. There is not enough lubricating oil. In order to simplify construction, the production engines do not use the special lubrication device that supplies lubricating oil to these bearings, but are lubricated with fuel-oil mixture by the "Autolube" system. The special lubricating device is installed only on racing engines.

Next it will be discussed how these difficulties are being solved nowadays. Even if the bearing is damaged or burned, the cause of failure does not necessarily exist in the bearing. If the shaft supported in the bearings does not have the necessary accuracy, failure will occur. Thus, it is necessary to explain briefly the construction and accuracy of the shaft.

The present production engine for motorcycles has either one or two cylinders. The construction is shown in Fig. 21. The crankshaft is assembled by pressing the crankpin into the crank webs. In 2 cyl engines, besides the crankpin, the shaft between the first and second cylinder is pressed into place; in this case, involute splines are used for alignment purposes and to prevent rotation. The crankshaft is forged AISIC 1042 or AISIE 4135 steel. The spline and the insertion part for the ball bearing are induction hardened. The accuracy of the production crankshaft assembly is within 0.05 mm for 1 cyl and 0.05-0.08 mm for 2 cyl engines, when the crankshaft is supported between both center holes of the shaft and rotated.

The difference in radius from the crank center to the crankpin center for both ends must be within 0.005 mm by

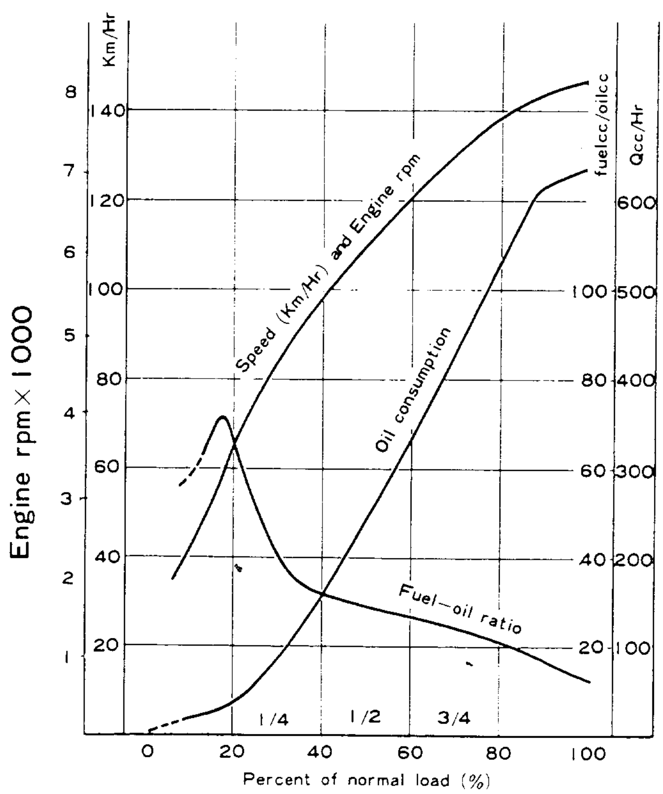

Fig. 20 - Oil consumption of a 250 cc engine with "Autolube" system

selective assembly. The material of the connecting rod is NF12CDD4 or AISI 8620, and is quenched after carburizing. The shank of the connecting rod is shielded by copper plating during the carburizing process. The tolerance for roundness and taper of the hole in both ends of the connecting rod is 0.003-0.006 mm for a hole diameter of 15-30 mm.

A very important factor in the functioning of the connecting rod is the parallelism of the two bores, which has a limit of 0.05 mm/100 mm. The material of the crankpin and the piston pin is the same as that for the connecting rod, and these parts are also quenched after carburizing. The diameter of these pins is within 0.003-0.004 mm. In the preceding description, the tolerances of the moving parts of the production engine are listed. In the racing engines, these parts must be made much more accurately.

The next description is concerned mainly with the needle bearing for the big end of the connecting rod. There are several retainer types of needle bearings and each type has good and bad points. Until two or three years ago, phosphor bronze was mainly used as the retainer material. Since phosphor bronze has good machinability, it can be accurately manufactured. It has, moreover, a superior antiburning property. But because of the copper alloy, the strength is not adequate, especially at high temperatures. The retainer made of phosphor bronze can be used without trouble only in engines with a maximum speed below about 7000-7500 rpm. Another disadvantage is that its specific weight is relatively high.

Because of the trend toward higher engine speeds, phosphor bronze has been replaced by steel. The retainer used at the present time is made of NF12CD4 and is copper plated after cyaniding. Silver plating is used for the high speed engines. The purpose of the plating is to prevent burning. The Dürkopp type is now a successful form of retainer; other new types, shown in Fig. 22, are also successful.

An important problem of the steel retainer is its dimensional accuracy. In order to prevent the skew of needles, the window edges must be parallel within 5 mm/1000 mm. Another very important problem of the steel retainer is how to limit distortion during heat treatment. As a result, considering the balance between strength, rigidity, accuracy, and the ability to maintain an oil film, one must carefully design the retainer. This precision is also applicable to the needle bearing for the small end. The types of needle bearings that are successful at the small end are INA and Dürkopp types. The needle material is AISI 52100; in special cases, high speed steel is used. The diameter of the needles is 2.0-3.5 mm. The radial clearances are:

1. Diameter of piston pin, 12-16 mm; radial clearance, 5-15 μ.
2. Diameter of crankpin, below 20 mm; radial clearance, 7-15 μ.
3. Diameter of crankpin, 20-25 mm; radial clearance, 13-21 μ.

In engines with displacement above 125 cc/cyl, plates made of phosphor bronze are inserted between the crank webs and the big end of the connecting rod to take up the thrust load. The connecting rod, crankshaft, and needle bearings must be accurate to the extent mentioned above for produc-

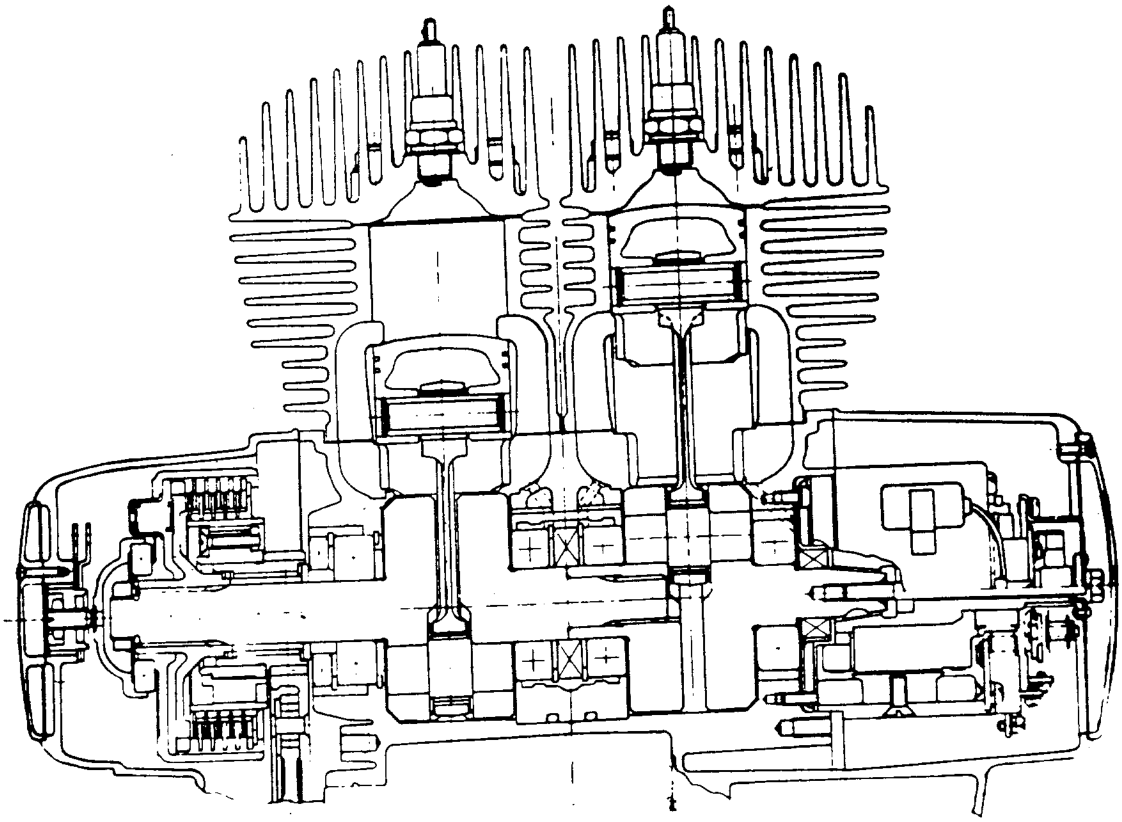

Fig. 21 - Yamaha 250 cc engine

tion engines. The needle bearing on the big end causes a thrust load whose direction continually changes and whose magnitude is said to be 30-50 kg. Thrust plates are necessary to support this load. In the production engines, which do not have the special lubricating device, these thrust plates must be carefully designed, with suitable oil grooves and holes.

Main Bearings - Ball bearings are utilized as the main bearings of the crankshaft. Since the bearing box is made of aluminum about 20-50μ clearance is given to the outer race for the purpose of preventing outer race creep. Hence, ball bearings with larger than normal internal clearances are normally used.

In such a case, the radial clearance if 2-3μ at atmospheric temperature, but when the temperature rises to about 70-90 C in operation, the most suitable radial clearance will be established. Lubrication is accomplished by the fuel-oil mixture; the oil that adheres to the wall of the crankcase is guided to the bearing by the scavenging hole. Under the poor lubricating conditions of the high speed engine, the retainer must be particularly designed so as to have a large pocket clearance for the oil.

Oil Seals - The next problem is oil seals. In 2 cycle engines that are crankcase scavenged, it is very important to maintain air tightness in order to obtain good performance.

At the present time, NBR rubber (which has some heat resisting and antiwear qualities) is used as the oil seal, but this kind of rubber cannot be utilized to the fullest extent. Polyacrylic rubber, which had superior heat resistance, was used previously, but since this type had a problem from the standpoint of swell in gasoline and friction, it is rarely used today. In the immediate future, oil seals with lips made of Teflon, which is superior with regard to heat resistance and frictional wear, may be used.

In order to decrease the mechanical losses and in order to expect perfect air tightness,we use double lip seals designed with special consideration. These are shown in Fig. 23.

Thermal Distortion - Because the 2 cycle engine is generally subjected to higher thermal loads and has many holes in the cylinder wall, thermal distortion is greater than that of the 4 cycle engine. Hence, as engine output increases, the so-called piston burning problem happens easily. When the clearance between the cylinder wall and piston is made greater to avoid this trouble, seizure of the piston ring occurs; consequently, the clearance cannot be made excessively great. Since the thermal distortion can result from a nonuniform temperature distribution, aluminum alloy for the cylinder material is more advantageous than cast iron because of its greater thermal conductivity. In fact, piston burning diminishes and smaller clearances are possible by this means.

For this reason, although production motorcycle engines use cast iron cylinders, the sport racing engines have aluminum alloy cylinders with a hard chrome plated bore. In this case, the chromium plating thickness is about 0.05-0.08 mm, and the inner surface of the cylinder is finished by honing before and after plating. Moreover, in order to improve the compatibility of the piston and piston ring with the cylinder wall and to hold the oil film securely, the plating surface is also liquid honed. But, since this type of cylinder is expensive and cannot be bored oversize in the event of failure, an aluminum cylinder with a cast iron sleeve has been recently developed.

Previous cylinders in which sleeves were used were of the following types: The first was an aluminum alloy cylinder barrel in which the cast iron sleeve was inserted with a press or shrink fit; the other had the aluminum cylinder cast around the cast iron sleeve. But in these cylinders, piston seizure was frequent. This was the result of a local gap between the sleeve and the barrel, which often occurred because of the temperature differential during engine running. Consequently, the mixture and exhaust gas were able to enter into this gap and the heat transfer to the cooling fins became lower accordingly.

On account of this problem a new type of sleeve was developed, the surface of which is specially treated with a metallic compound in order to ensure a metallurgical bond with the aluminum. Therefore, even with a high temperature differential, good thermal conductivity can be attained without gaps occurring between the sleeve and barrel.

Since the cylinder head is attached to the cylinder by

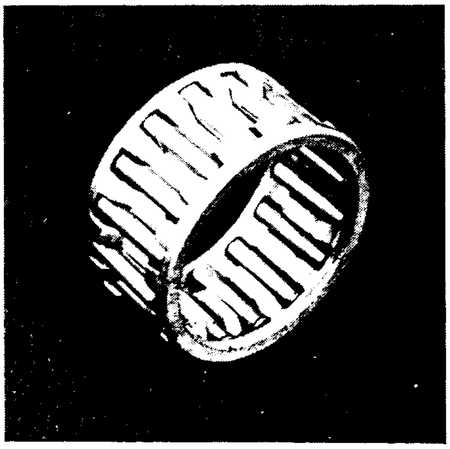

Fig. 22 - Durkopp type retainer

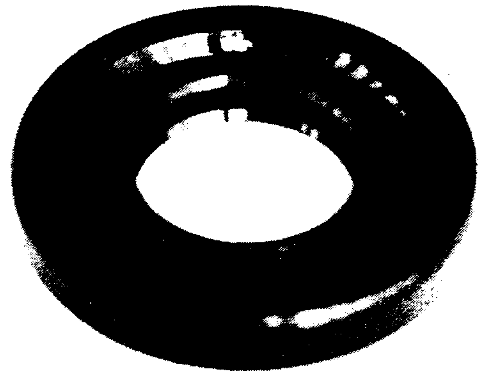

Fig. 23 - Double lips oil seal

means of long bolts, the cylinder can be deformed greatly, particularly in the case of the aluminum alloy cylinder. In order to avoid this deformation, the cylinder bore is honed with a dummy head bolted to the cylinder with the correct torque value.

As the engine output increases, the cooling fins must become larger. It then becomes necessary to prevent vibration of the fins. For this purpose a few ribs are formed in the end point of the fins to increase the natural frequency.

Pistons - As mentioned above, the so-called piston burning problem occurs in company with an increase of engine output. The countermeasures for this problem are improvement of cooling method, dimensions, and material of the piston. As regards cooling, the crown of the piston is formed so that heat can easily escape, and the piston land is made as small as possible within the admissible limit of strength. For example, the piston of the YAMAHA 305 cc (YMI) engine is shown in Fig. 24.

Since 2 cycle engines cannot adopt piston construction with slots because of strength requirements and the scavenging crankcase, the profile of the piston cannot help but be complicated. Thus, the elliptic ratio of this profile is changed continuously along its length. This suitable elliptic ratio of profile can be found only by the trial-and-error method method for each type of engine.

Furthermore, because of the trunk-type design, it is necessary to choose a material with as low a coefficient of expansion as possible. For this reason, high silicon alloy is used nowadays in most cases. This alloy has a silicon content of 18-26%, and its coefficient of expansion is rather small in comparison with SAE 321 or 332 materials; it is also superior with regard to wear and strength at high temperatures. These qualities increase with an increase in silicon percentage. In particular, in the case of contents above 20%, these properties are very remarkable. For this reason, an aluminum alloy with 20-22% silicon is used in spite of its poor castability and machinability.

The higher the piston speed, the thinner the ring width must be. This decreases the influence of the inertia force caused by the piston ring on the ring groove and prevents ring fluttering. The ring width of the racing engine is 1 mm, and in order to decrease the friction loss as much as possible, only one ring is used.

Combustion Chamber - The shape of the combustion chamber is very interesting. In order to shorten the distance of flame propagation, the chamber is hemispherical and the spark plug is positioned at the top point.

It is additionally formed so that compression turbulence is produced between the piston and cylinder head in order to increase the combustion speed. This is shown in Fig. 25.

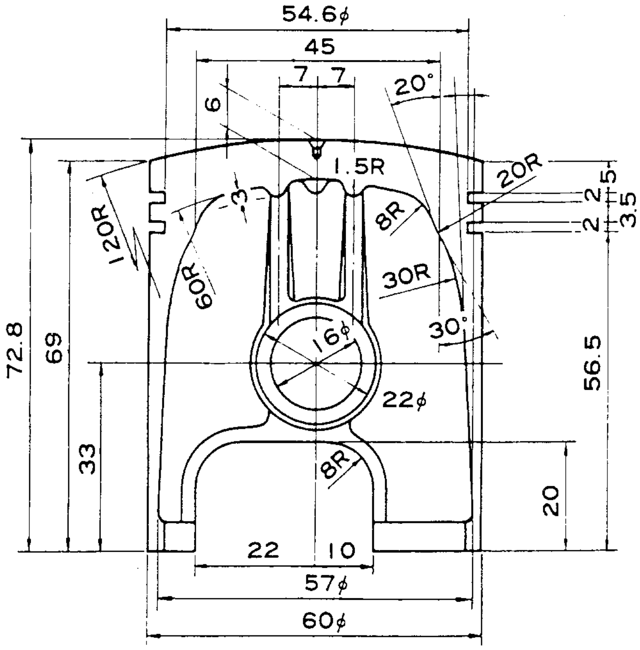

Fig. 24 - Piston details of Yamaha 305 cc (YMI) engine

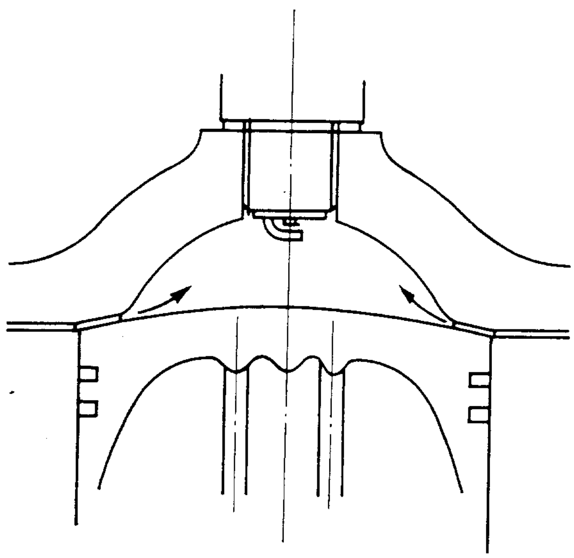

Fig. 25 - Combustion chamber configuration

(Discussion of this paper appears on next page)

WILLIAM J. HARLEY
Harley-Davidson Motor Co.

YAMAHA MOTOR CO. is generally recognized as producing excellent motorcycles for both commercial and racing purposes. This interesting and well-prepared paper shows results of an intensive research program by Yamaha.

American Outboard Engines use a "reed or diaphragm" type for intake, and in Claus Waker's paper,* he states:

"For all four series produced combinations, the particular advantage of having inlet diaphragm control was clearly evident. It adapts the crankangle at intake independently and flexibly to the load and flow conditions existing at a given time, in the charging pump. As regards torque and specific consumption, the inlet diaphragm control, measured over the entire range, may even exceed the rotary valve control."

In connection with the performance of the piston valving as compared to rotary disc valves, it is indicated, in the authors' Fig. 7, that higher horsepower is attained with piston valving than with rotary valving, although better horsepower is attained through the lower speed range with rotary valves. It seems that this does not agree with the text.

Would the authors care to comment on the relative merits of rotary valve versus reed or diaphragm control of intake? Their observation that resonant intake and exhaust systems are more important than compression ratio in the crankcase is confirmed by results published by MZ.

Would the authors care to comment on the possibility of the use of more than two cylinders, incorporating a rotary valve induction system?

In their paper, there is no mention of experiments with the use of "boost ports" in the transfer system. Ehrlich's experience with 2-cycle engines found a substantial increase in power through the use of "boost ports." Would the authors care to comment on this?

It is stated that the shape of the combustion chamber has no great effect upon the engine output, but our tests indicate that the positioning of spark plugs and the shape of the combustion chamber do affect general overall performance of the engine. Apparently the compression ratios mentioned in the paper are measured after the ports close, and this agrees with our general test pattern.

Have tests been conducted using surface discharge plugs or plugs such as Champion #UJ-17-V, which is currently used by American Outboard Engines?

It is not apparent why complete lubrication could not be attained by oil pumps rather than resorting to a combination of petroil with the oil pump. Would the authors care to comment on this? Due to the very small quantities of oil that must be metered by the oil pump, it would seem that manufacturing of the oil pump must be a very precisely controlled operation. Has this been a problem? Any contaminants in the oil supply would probably seriously affect pump performance. Are special precautions necessary for the customer to observe?

It has been our experience that bronze plating of steel connecting-rod bearing retainers is superior to silver plating both in function and cost. Also, bronze-plated steel washers on each side of connecting-rod bearings have been very successful in absorbing any side thrust generated.

We have found that a better compromise is reached in the manufacturing of steel retainers using a material such as Stressproof, having a Rockwell hardness of C-25 to C-35, rather than cyanide hardening the retainer after machining, due to distortion caused by heat treating.

Recently published test data released by NASA indicate that a significant gain in bearing capacity and durability can be attained by having the roller bearing two to three points Rockwell C harder than bearing races. Has Yamaha test experience on this?

It is hard to understand why the connecting-rod bearing should cause thrust loads of up to 50 kg, unless serious deflection is occurring in the connecting rod crankshaft assembly.

Our test experience on piston material with silicon content in excess of 20%, concurs with the authors' experience, but we are also plagued with problems in machining this alloy.

This paper is particularly interesting to Harley-Davidson, inasmuch as Yamaha's application of their research has been in the motorcycles which they manufacture. They have brought the performance of high output 2-cycle engines a long way, so that they are equal to, or exceed, the specific performance of the best 4-cycle engines.

GORDON JENNINGS
Cycle World Magazine

IN THE AREA of high specific output, 2-stroke cycle engine design, the state of the art owes much to Japan's motorcycle manufacturers generally, and to the Yamaha Motor Co. in particular. However, the paper asks as many questions as it answers.

For example, while the performance trends charted show a yearly rise in power output, and follow the course of compression ratio and torque, we see nothing about engine speed. It might be instructive to know if Yamaha has found, or can forsee any indication of, an upper limit for crankshaft speed? From all reports, it would appear that the power peak for racing 2-stroke engines is holding at around 12,000 rpm -- almost without regard for unit cylinder displacement. Is this due to a problem in maintaining volumetric efficiency, or one connected with reliability in the crankcase-scavenged engine's lightly-lubricated bearings, or both?

A convincing case is made for the disc-type rotary intake valve, but it would have been interesting to see the relative

*Claus Waker, "The Present-Day Efficiency and the Factors Governing the Performance of Small Two-Stroke Engines." Paper 660009 SAE Transactions, Vol. 75 (1967), pp. 40-59.

shapes of power curves for piston-port and rotary-valve engines having the same displacement, maximum output and peaking speed. Also, one is inclined to wonder if research has been done precisely the sort of broad-range power characteristics needed for the sports/touring motorcycle? Finally, it is rather curious that after having made a fine case for rotary intake valves, Yamaha would offer as the latest model a piston-port engine that is a more or less direct replacement for a rotary-valve model that was widely considered a success and was already in volume production.

May we assume that "various scavenging types" refers to various arrangement and number of transfer ports? It is stated that all types "have shown nearly equal scavenging and charging efficiencies." How nearly equal? Others have found benefits in controlling the transfer phase, and reducing the time requirement for transfer, with multiple transfer ports. Perhaps the authors would like to comment on bridged exhaust ports, as Yamaha produces at least one engine -- a high-output version of the YDS3 "touring" unit -- in which port width is such that ring life is adversely affected and almost instant ring failure occurs unless there is sufficient chamfer around the port window?

In regard to exhaust systems, what is the influence of shape, as distinct from length, on power output characteristics? Others have found that this is related to the angles of divergence and convergence for the opening and closing portions of the system. What have been the authors' findings in this?

The virtues of an oiling system delivering lubricant directly to the engine's bearings, as in the Yamaha Grand Prix racing engine, are easily understood. Are there advantages, beyond simple convenience and the "idiot-proof" aspect, to the automatic-metering "Autolube" system used on Yamaha's touring-type engines -- in which oil is merely injected into the airstream in the intake passage?

Would the authors care to comment on the merits of all-aluminum cylinders with chromium-plated bores relative to those having a cast-in-iron liner? This question presupposes a mechanical lock between iron and aluminum rather than the bond provided by the more expensive "Al-Fin" method.

AUTHORS' CLOSURE TO DISCUSSION

THE TWO opposite extremes for the piston valve engine are shown in Fig. 7. One line show optimum low-end torque while the other line shows maximum peak. These two lines are compared with a single line on the rotary valve engine which represents the "happy compromise." If desired, the rotary valve engine could be so designed and constructed that it would have less low-end and a slightly better peak, for instance, than the piston valve engine.

We are not able to comment on the relative merits of reed valves. Yamaha is definitely interested in more information along this line.

At this time we have no comments on the use of more than two cylinders or the use of "boost ports" in the transfer system.

There is no research data available from Yamaha, as we understand that the surface discharge spark plug is only suitable for use with an engine using a gasoline-oil ratio of 40:1 or leaner. Using the common oils, Yamaha engines require a richer mixture than that at full throttle.

With the standard Yamaha Street Machine, all oil is supplied by the Autolube Pump. There is no premixing at all. Only the special racing machines use both an oil pump and a premix.

The oil pumps have very close machining tolerances and are matched assembled. They have been quite reliable in operation, and have shown a very low incidence of failure. They have not been affected by ordinary contaminants.

The authors appreciate the suggestions and information furnished by Harley-Davidson concerning the bronze plating of steel bearing retainers. The 50 kg side thrust load is really in excess of the actual figure reached. Yamaha thought that the use of this figure would eliminate any and all failures at this point. The side thrust that is present is due to both the rocking motion of the shaft as well as accumulative misalignment of the rotating and reciprocating parts.

The machining difficulties of the high silicon material of our pistons is thought to be a necessary evil. These troubles are tolerated for the sake of the end result.

In response to Gordon Jennings, engine speed has been rising steadily over the years. For instance, in 1954, the YA1 had 5000 rpm; in 1958, the YA3 had 5100 rpm; in 1961, the YA5 had 6300 rpm; and in 1964, the YA6 had 6700 rpm, with our single cylinder 125 cc engine.

With 2-stroke engines the maximum rpm is limited by materials, tuning procedures, and possibly other factors which are directly related to reliability. The volumetric ceiling is probably more around the 16,000 rpm figure, however, because some racing engines have been turning up to this figure.

The piston port and rotary valve engine comparisons, as well as the reed intake valve situation, have been briefly touched with Mr. Harley's discussion. Concerning replacement, "more or less," of our popular rotary valve 80 cc engine with a Twin 100 cc cylinder port engine, this is partly a matter of sales appeal and the mechanical need of going to a multiple cylinder engine. The smooth running, high turning 100 cc Twin would be quite difficult and expensive to make with rotary valves because of the then necessary remote location of the electrical and primary drive system, as pointed out with our 250 cc engines.

The "various scavenging" types are roughly within 3% of each other as to charging efficiency. The Yamaha TD1B exhaust port is admittedly at its maximum permissible width. It is felt that the bridge should be avoided if possible because of heat problems with this mass.

Yamaha's findings are similar to those stated here in that the angles of divergence and convergence as well as diameters are related to power characteristics. With motorcycles there is a physical size restriction which prohibits a complete freedom size choice.

The advantages of the Autolube system over the bearing injection system are those of manufacuring economy and foolproof operation. With street machines, the bearing injection system has apparently no advantages.

The matter of all-aluminum cylinders with chromium-plated bores is still being investigated by the engineers. The bonding and locking methods used in the past by Yamaha did have problems which, if eliminated, would then produce a system with definite economical advantages over the chrome-plated bore. This crome-plated bore is ideal as to heat dissipating characteristics.

ORAL DISCUSSION

Reported by B. K. Ghandhi, Outboard Marine Corp.

Richard Booy, Outboard Marine Corp. - Regarding your exhaust system, how do you produce the optimum in combining diffuser, chamber, and tailpipe? Is this cut and try?

A.: Yes, it is trial and error, but using a proven system as the starting point and adjusting to the new engine.

Dean Thomas, Nelson Muffler Co. - What criteria are used for the selection of rpm range?

A.: We vary gearing with the rider and with the course. We wish we could vary exhaust systems too, but we compromise for the conditions.

Otto Scharpf, Outboard Marine Corp. - What is the maximum power gain due the exhaust system only? What is your experience with ring flutter?

A.: There is no figure available for the exhaust system since it was designed as a part of the engine just as is the porting. The ring flutter is a function of piston clearance, ring width, lubrication, and ring tension, but no particular rpm. As a rule of thumb, use thinner rings at higher speeds.

Q.: What type of lubricating oil do you recommend?

A.: The best possible grade SAE 30 medium detergent automotive oil for normal temperatures.