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The Present-Day Efficiency and the Factors Governing the Performance of Small Two-Stroke Engines

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THE EXTRAORDINARY INCREASE of mobile power requirements in industry and road traffic has greatly contributed to the development of small internal combustion engines. An entirely new complex of problems was posed in particular by the demand for small internal combustion engines of low weight, small space requirements, and low manufacturing costs, that would also provide a specific high output during continuous operation. Since a mere geometrical sizing down of existing types of engines would not be satisfactory in the long run, new ways were found in the design and economic manufacture of such engines. The small engine has now reached a stage of operational maturity where no more major advances are foreseeable while adhering to the principle of the reciprocating engine; therefore, this seems to be a suitable time to present a summary of what has been achieved in two-stroke efficiency.

A comprehensive survey of the present status of the small internal combustion engine would exceed the scope of this paper in view of the variety of designs available and the vastness of actual and possible applications. Thus, by limiting this paper to essentials, really new information will not be presented to the expert, especially since no theoretical discussions are offered. Instead, the main purpose of this paper will be to properly portray the high-performance level of the apparently commonplace but, in its versatility, rather complex 50 cc utility engine. This piston displacement, stipulated by statutory specifications, is preferably used in European two-wheeler manufacture and is represented by various types of vehicles, such as small motorcycles of 50 cc displacement, with and without speed limitations. Moreover, there is a special 50 cc class in cross-country and road racing.

The 50 cc engine is just as popular in industrial applications. This type of engine probably owes its development not only to market considerations but, in particular, to the similarity and close relationship of its assembly and thus adaptability to existing manufacturing programs.

PRESENT-DAY EFFICIENCY OF 50 CC TWO-STROKE ENGINE

Variegated motorization demands led to various developments of small engines. Power requirements called for sev-

ABSTRACT

The paper deals with the present-day efficiency of small 50 cc, air cooled, two-stroke engines, showing the essential performance characteristics of utility engines and higher output engines. The problems arising when output is increased by modification of the cylinders and exhaust systems are discussed, as well as the degrees of accuracy required in the manufacture of power determining assemblies. Some of the essential technical considerations involved in engine designing are also discussed, and some of the studies carried out to prove the suitability of the exhaust system, combustion-chamber configuration, spark plug assembly, piston rings, and cylinder material are described.

eral designs of the given cylinder volume in order to achieve necessary outputs. When the existing standard design was not suited to the motorization requirements, new designs were developed. The number of solutions found is legion; therefore a classification of what has been achieved within this cc class may be given as follows:

1. Utility Engine - This includes moped engines and all-purpose (industrial) engines.
2. Higher Output Engine - This includes small motorcycle engines and single-purpose (industrial) engines.
3. Maximum Output Engine - This includes cross-country sports engines, and road racing engines.

This division into three major groups has been made by reason of the original demands placed on each group. With regard to engine output, which is not limited to technical engine characteristics, the original demand must also be considered as a basis of engine design. Vastly different engine designs, such as moped and industrial engines, may therefore be representative of one and the same "output" class. Table 1 lists the performance characteristics of the various engine types discussed in the sections that follow.

50 CC UTILITY ENGINE - For high efficiency and economy of two-stroke engines, exact and low-loss scavenging is of particular importance. To approach this demand, costly expedients, such as uniflow scavenging by means of double pistons or loop scavenging by means of rotary inlet valves, are prohibitive for utility engines of the 50 cc class. Therefore, port scavenging and, in some cases, inlet diaphragms are used. When properly coordinated and harmonized, these control units will achieve results which are at least equivalent to those of the more complicated four-stroke engine with regard to torque and output. Common criteria used for utility engines, based on the original demands of the ultimate users, are shown in Table 2 along with the designers own demands.

Moped Engine - This type of prime mover is applied in mopeds (bicycles with auxiliary engine, and small motorcycles), a popular vehicle for local short-distance travel and commuter traffic (about 1,000,000 are newly licensed in Europe every year). This type of engine (Fig. 1) is characterized by the additional demands listed in Table 3.

All-Purpose (Industrial) Engine - This type of engine (Fig. 2) is applied mainly in farming, forestry, industry, and sm

Table 2 - Original Demands on Utility Engines

Users' Demands	
Purchase Costs	< 10 DM/kg
Operating Costs	< 1 DM/hr
Repair Costs	General overhaul = 1/2 of purchase price
Service Life	> 20,000 km = 1000 hr
Operation	Foolproof
Power-to-Weight Ratio	< 5 kg/hp
Fuel Consumption	< 400 g/hp/hr
No oil containing exhaust fumes.	
Designers' Demands	
Torque Characteristics	Buffalo type
Oil Addition to Fuel	< 4%
Servicing Intervals	> 3000 km = > 100 hr
Simplification of engine types	
Simplification of maintenance, servicing, and repair	

Table 1 - Engine Performance Characteristics

	Moped Engine	All-Purpose (Industrial) Engine	Small Motorcycle Engine	Single-Purpose (Industrial) Engine	Cross-Country Sports Engine	Racing Engi
Output, hp (DIN)	0.8-2.6	1.8	5.2	2.5	8	12-13
at rpm	(3500-5000)	(4500)	(7400)	(6000)	(10,000)	(13,000)
Specific Output, hp/liter	16-52	36	104	50	160	260
Weight-to-Hp Ratio, kp/hp	12.5-3.8	5.0	3	1.6	2.25	< 1
Mean Effective Pressure, kp/cm^2	2.1-4.7	3.6	6.3	3.7	7.2	9
Practically Useful Crankshaft Speed, rpm	1500-6000	2000-4500	2000-8500	2500-7000	4000-11,000	10,000-14,
Mean Piston Speed, m/sec	4.6-7	6.3	10.8	8.6	14.6	--
Compression Ratio	1:6-1:9	1:7	1:9	1:9	1:10	1:14
Stroke-to-Bore Ratio	1.1	1.1	1.15	0.88	1.15	--
Specific Fuel Consumption, g/hp/hr	350-400	420	350	450	320 Super	600
Noise Level, db	68-75	76	78	78		
Carburetor Diameter, mm	8-12	14	17	15	20	20

trades. Its wide range of application requires a high degree of universal suitability and adaptability. Additional demands upon such engines are shown in Table 4.

50 CC HIGHER OUTPUT ENGINE - Through systematic research and development, the output of the modern two-stroke engine has increased 450% during the last 10 years without any essential sacrifice in performance. Thus, as a result of good engine design, both specific fuel consumption and service life have remained essentially unchanged.

In vehicular engines of this type, development was mainly concentrated on increased output, whereas in stationary engines emphasis was given to extremely lightweight construction in addition to a more modest increase in output. The demand for increased output induced the final user to accept some sacrifices with regard to service life and maintenance requirements, as well as purchasing and operating costs. The essential original demands made on high-output engines have remained as shown in Table 5.

Small Motorcycle Engine - This type of engine (Fig. 3) is applied as a prime mover for small motorcycles, a type of vehicle which is not subject to any speed limits. The minimum age to obtain a driver's license is 16 years. It is regarded as the stage in motorization which precedes the purchase of a four-wheel vehicle. Additional demands for this type of engine are shown in Table 6.

The increased output of small motorcycles was obtained by favorable port configurations in the cylinder (in some cases, three transfer ports), favorable combustion-chamber design (spherical head with compression edge), and harmonization of the back pressure and intake line. Unfortunately, not all technical possibilities (rpm, exhaust system) can be fully utilized because transport law restrictions must be observed.

Single-Purpose (Industrial) Engine - This type of engine has special uses, such as for chain saws, portable sprayers, percussion drilling equipment, vibration plates, boat engines, starting engines, and other complicated applications for which ruggedness, simplicity and, above all, weight advantages are required. Additional demands on this type of engine (Fig. 4) are shown in Table 7. As the space-saving exhaust system of stationary engines usually dispenses with an exhaust pipe, unfortunately it is not possible to make use of gas-dynamic effects in this area. Therefore, specific output is below that of the motorcycle engine.

MAXIMUM OUTPUT ENGINE - Maximum output, heavy-duty engines are frequently regarded as a criterion to indicate

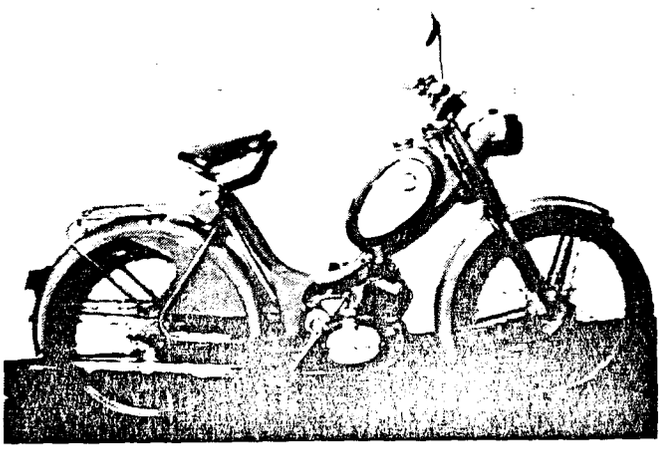

Fig. 1 - Moped equipped with Sachs 50/ Automatik engine

Table 3 - Additional Demands on Moped Engines

Statutory Specifications (Europe)	
Piston Displacement	50 cc, exempted from motor vehicle tax. Some models require no driver's license, are exempted from road admission license, and are governed by certain age limits
Maximum Speed	Speed limits vary in European countries from 25, 30, and 40 to 50 km/hr
Maximum Output	0.8 DIN hp (Switzerland) 1.0 DIN hp (Sweden)
Sound Level	70 db (Switzerland) 75 db (other European countries)
Radio interference suppression statutory	
CO Content (statutory regulations to be issued)	4% during idling

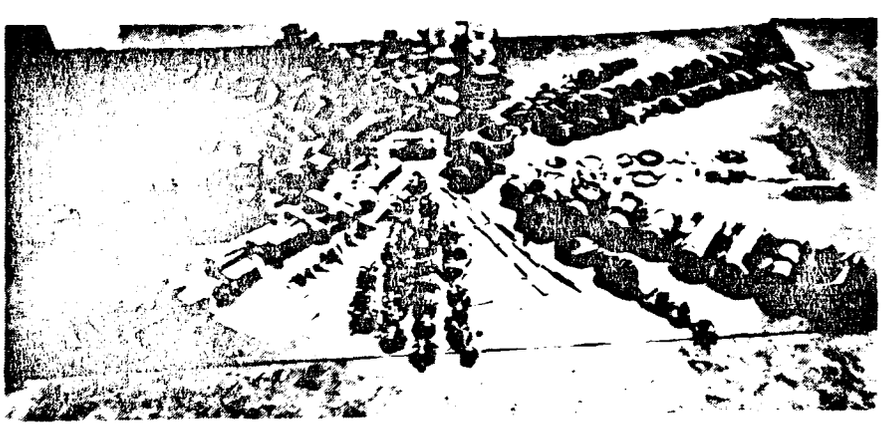

Fig. 2 - Variable system Sachs-Stamo 50

Table 4 - Additional Demands on All-Purpose (Industrial) Engines

Statutory Requirements	
Sound Level	80 db
Final Users Demands	
Resistance to and Compatibility with	Engine location or mounting, dust, humidity, types of lubricants
Adaptability	Variation of assemblies: carburetor, exhaust system, air cleaners, unit assembly system, and adaptability to extreme altitudes and climates
Reversibility	Change direction of rotation
Multifuel Operation	Should be suitable for kerosene and gasoline
Possibility of Extension	For instance, addition of ±2.5% speed control, 1:45 reduction gear, or flywheel clutch

Table 5 - Original Demands on High-Output Engines

Buyers Demands	
Output	> 50 hp/liter
Service Life	> 10,000 km = 500 hr
Power-to-Weight Ratio	< 3 kg/hp
Fuel Consumption	< 450 g/hp/hr
Own Demands	
Torque Characteristics	Ability to start at 2000 rpm
Oil Content in Fuel	< 4%
Should be simple to repair	

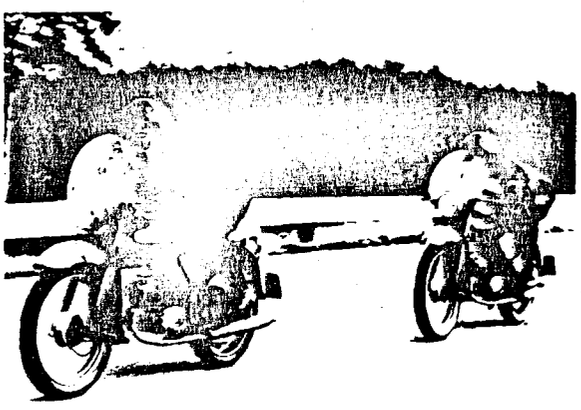

Fig. 3 - Sachs 50 S small motorcycles with five-speed gearbox during 40,000 km nonstop race (total speed averaged: 75 km/hr)

the technical and organizing efficiency of an engine manufacturing plant. Depending on the market situation, maximum output engines are used for all kinds of motorsports events. However, the main value of this type of engine is probably reflected by its significance as a study and research

Table 6 - Additional Demands on Small Motorcycle Engines

Statutory Requirements (Europe)	
Cylinder Capacity	50 cc, exempt from motor vehicle tax, less rigid requirements for driver's license
Maximum Speed	Unlimited
Maximum Output	Unlimited
Radio interference suppression statutory	
CO Content (statutory regulations to be issued)	4% at idling
Sound Level	78 db/75 db (Switzerland)
Buyers Demands	
Output	100 hp/liter
Repair Possibility	Simple, Cheap
Own Demand	
Service Life	> 20,000 km/hr

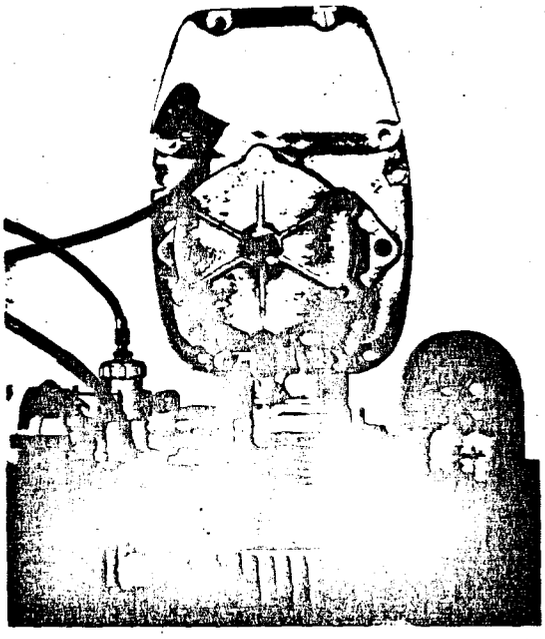

Fig. 4 - Sachs-Stamo 51 (2.5 hp output; 7000 rpm engine speed; 1.6 kg/hp power-to-weight ratio)

object, and thus as a promoter of technical progress. In spite of their apparent relationship, there is some dissimilarity between cross-country sports engines and racing engines with respect to the original demands made on them (Table 8).

Due to widely differing interests in various types of motor sports, as evidenced by the diversity of the events carried out, a diverging development of maximum output engines has taken place. The cross-country sports engine developed from the small motorcycle engine remained closely related to series produced models. The knowledge resulting from its development increased the utility value of series production engines. Thus, the highly developed racing engine has a large indirect influence on series production.

Cross-Country Racing Engines - These are used in cross-country events, six-day trials, moto-cross (national championship). The fierce competition between the various makes leads to certain specific additional demands (Table 9). In view of these additional demands, cylinder volume is not utilized to attain the theoretically possible output of such an engine. Since the vehicles are not subject to official limitations, noise suppressing equipment (intake and exhaust mufflers) can be designed to develop maximum output. Unlike series produced engines, these are made and put together by selected, skilled personnel; an example of a cross-country motorcycle is shown in Fig. 5.

Road Racing Engines - These are applied in national and world road racing championship events. Additional demands include: > 240 hp/liter output and < 1 kg/hp lightweight construction.

The high standard of development of the 50 cc two-stroke engine is evident by the achievements of the racing engine. They are of special construction, mostly using a rotary slide valve and were originally fitted to provide a favorable asymmetric inlet timing characteristic, that is, shifting of a larger amount of crankangle to the induction period. Nowadays the main advantage is considered to be the hydrodynamically more favorable arrangement of the inlet port below the piston, resulting in a more favorable location of the transfer ports (at least three). Since the present standard with regard to air induction, and thus cylinder charging, in the speed range up to 10,000 rpm can hardly be exceeded, further increases in engine speed might be required.

The designs for this type of engine are not uniform. Single-cylinder and two-cylinder engines are being built, some with horizontal cylinders and others with vertical cylinders.

Table 7 - Additional Demands on Single-Purpose (Industrial) Motors

Statutory Requirements	
Sound Level	< 80 db
Buyers Demands	
Stability	Compatibility with forces equalling 30 times acceleration due to gravity
Engine Speed	8000 rpm in continuous operation
Ambient Temperatures	-40 C to + 50 C
For Special Applications	Must be free of obnoxious exhaust fumes, should be explosion-proof
Own Demands	
Service Life	> 1000 hr
Output	> 60 hp/liter

Table 8 - Original Demands on Maximum Output Engines

Output	140 hp/liter
Service Life	50-100 hr
Weight-to-Horsepower Ratio	2 kg/hp
Speed Range	As large as possible
Resistance	To ambient influences
Accessibility	To all controls

Table 9 - Additional Demands on Cross-Country Racing Engines

Cooling	Should be adequate also at slow forward speeds, with mud-caked cooling ribs and housing
Air Cleaner	Should not be affected by sand, dust, water
Sealing	In housing, carburetor, feed lines
Disturbances	Should be unaffected by excess mechanical or thermal strain, such as impact of stones, falls, rough handling, or overheating due to mud-caking
Torque Range	Ability to start at as low engine rpm as possible

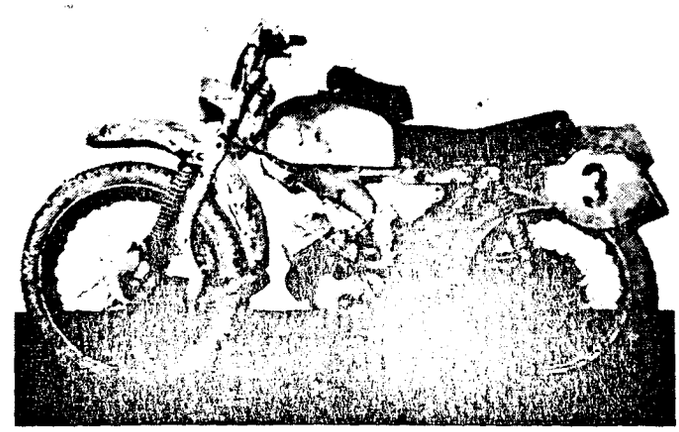

Fig. 5 - Cross-country motorcycle Hercules with Sachs 50 GS (6-speed gearbox; 100 km/hr maximum speed)

Some use simple rotary valves and others two-plate rotary valves. There are single-carburetor and multicarburetor engines; mechanical double interrupter ignition as compared with transistorized ignition; air cooling as compared with fluid cooling; large-end bearings with metal cages and those with steel cages; varying numbers of piston rings, ring configurations, ring materials, and sealing materials; and gearboxes ranging from 6-12 speeds. All these variations are subject to constant change in accordance with the latest technical standards.

SUMMARY - An attempt has been made to portray the 50 cc two-stroke engine in the light of present-day technical standards. The major portion of those in current production, and thus of the greater significance, is unquestionably the utility engine, which is faced by a multitude of different demands and requirements. The fact that these universal properties could largely be incorporated into the utility engine, thus limiting its production to a few types, deserves special recognition. A brief outline has been given of what can be achieved through technical progress in the manufacture of special types of engines.

FACTORS GOVERNING MODERN SINGLE-CYLINDER, TWO-STROKE ENGINE PERFORMANCE

One of the requirements for economic production of large series engines is to derive various performance factors from as few basic engine types as possible. Therefore, it is important to know how to obtain many performance varieties through as little engine modification as possible as well as the smallest possible expenditure for manufacture, assembly, and accuracy requirements. In the following sections, some possibilities of variation to influence engine performance are dealt with, and the influences of various methods on accuracy are described.

Performance can always be influenced by:

1. Changing the cylinder capacity. An illustration is a three-combination engine allowing for either of three different cylinder capacities (50, 75, and 100 cc) to be installed on the same integral engine gearbox block (for example, Zweirad-Union of Nuremberg). In this case, the piston displacement is a variable while the rpm and mep are constants.

2. Maintaining the same cylinder capacity but changing the mep and rpm by altering the timing (piston displacement = constant, rpm and mep = variables); changing the mep alone by modifying the exhaust system (piston displacement and rpm = constants, mep = variable); and changing the mep and rpm by altering the timing and modifying the exhaust system (piston displacement = constant, rpm and mep = variables).*

INFLUENCING ENGINE PERFORMANCE WHILE MAINTAINING SAME CYLINDER CAPACITY - The nonuniform European transportation and traffic laws specify various construction features (for example, maximum output, vehicle weight, maximum speed, and so on) for bicycles with auxiliary engines (mopeds and mokicks fall into the same category).

A large number of vehicles are equipped with Sachs moped engines conforming to the respective output and/or speed requirements. For reasons of economic production, a uniform type of engine should be manufactured. This has re-

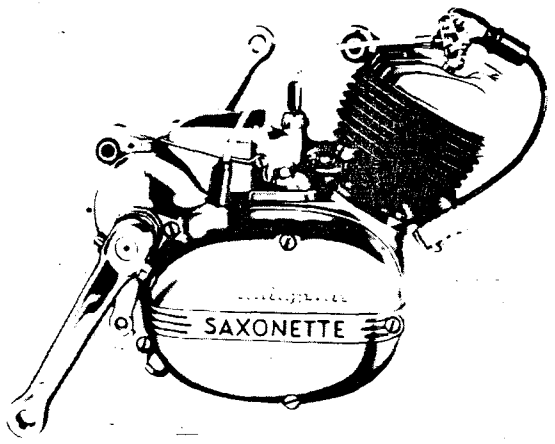

Fig. 6 - Sachs 50/M basic engine

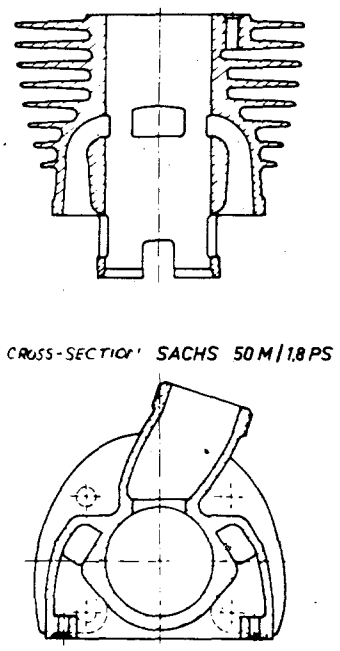

Fig. 7 - Longitudinal and cross-section views of Sachs 50 M/1.8 hp cylinder

*Examples: Engine manufacturing programs of Fichtel & Sachs AG, Schweinfurt Kreidler-Fahrzeugbau, Kornwestheim Zundapp-Werke, Munich.

sulted in endeavors to make an existing basic type of engine adaptable only by modification of one assembly. Fig. 6 shows a Sachs 50 basic engine.

The assembly lending itself readily to modification appeared to be the cylinder because the pre-assembled integral engine and gearbox blocks can be completed simply in the final assembly by one or the other type of cylinder. In addition, the exhaust system does not pass the engine assembly line at all.

Influencing Mean Effective Pressure and Crankshaft Speed by Altering the Timing - Fig. 7 shows a longitudinal and cross section view of the Sachs 50 M/1.8 hp basic cylinder. Fig. 8 shows the inlet diaphragm. By altering the cross-sectional areas, routing, and configurations of the exhaust port and transfer ports, output can be varied by a factor of about 3.

Figs. 9-12 show the port measurements of the four different performance types currently produced in series: 0.8 hp at 3300 rpm; 1.3 hp at 3250 rpm; 1.8 hp at 4500 rpm; and 2.4 hp at 6400 rpm.

Figs. 13 and 14 show associated characteristic curves for

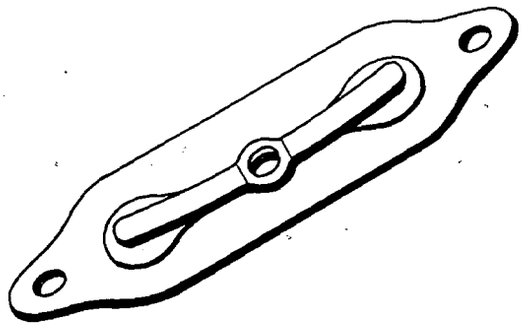

Fig. 8 - Inlet diaphragm of Sachs 50 M/1.8 hp cylinder

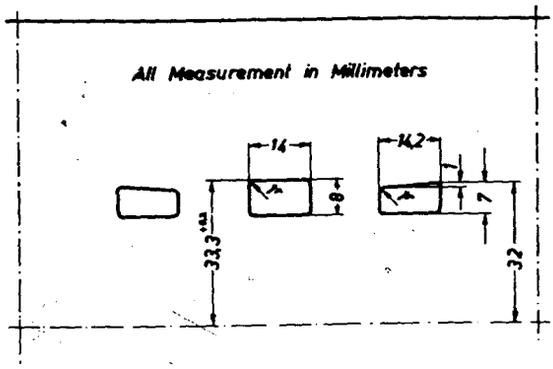

Fig. 9 - Port measurements of CH - 0.8 hp cylinder

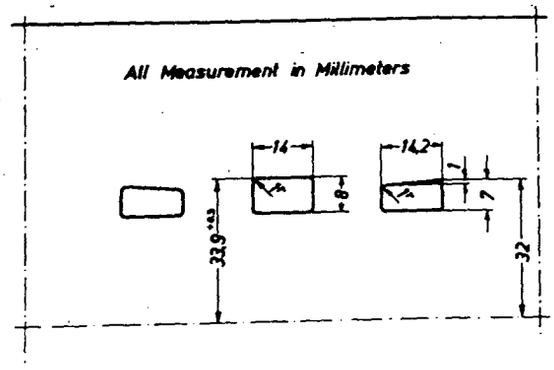

Fig. 10 - Port measurements of MC - 1.3 hp cylinder

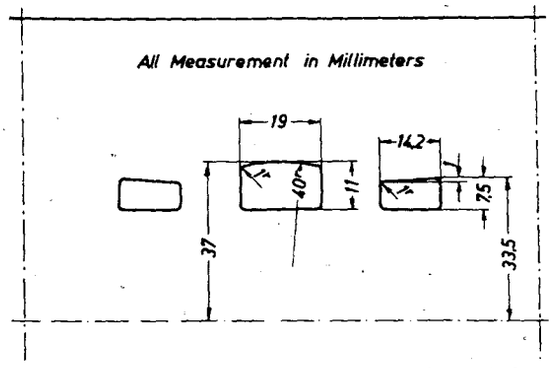

Fig. 11 - Port measurements of NL - 1.8 hp cylinder

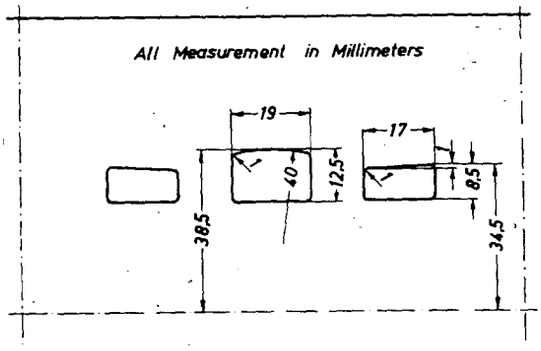

Fig. 12 - Port measurements of 2.4 hp cylinder

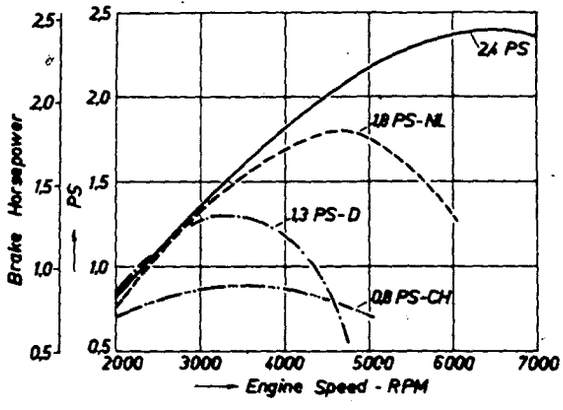

Fig. 13 - Characteristic curves for hp depicted in Figs. 9-12

hp and mep, which are largely influenced by altering the timing. Although the maximum torque of the engines subject to a specified low rated output must be decreased considerably, it can be shifted to the lower rpm range in such a way that it is fully available for acceleration and pull. Pressure tests have shown that in the lower rpm range, the exhaust port built as a throttle smooths the reflection waves set up by the exhaust stroke so that the change in charges is not disturbed. In the upper rpm range, however, there is no boosting effect by the reflection waves so that the mep decreases rapidly. The larger exhaust cross section of the higher output engines causes the waves reflected by the exhaust system to boost the cylinder in the upper rpm range with fresh gas that has already been sucked off. The larger time cross sections favorable to the exchange of charges insure that the mep does not also decrease too much in the lower rpm range.

It should be added that, in all four cylinder types, the intake side is almost the same with respect to configuration and cross-sectional area of the intake port. The induction manifold leads directly into the transfer ports. The carburetor and intake diaphragm are completely identical. All have the same exhaust pipe diameter and length. However, for the cylinder of the type corresponding to Fig. 12, an exhaust muffler of larger diameter (otherwise identical) is used. With the series produced exhaust muffler, the mep characteristic is somewhat less favorable.

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 (rigid control diagram).

Influencing Mean Effective Pressure Through Modification of Exhaust System - Further variable factors for influencing engine performance relate to the exhaust pipe length and exhaust system. In Fig. 15, four exhaust pipe combinations are shown. For the exhaust pipe shown at the bottom, the exhaust system (which happened to be available for other engine types) had to be additionally modified. Corresponding characteristics for hp/rpm and mep/rpm are shown in Figs. 16 and 17, respectively.

The characteristics for the mean effective pressure (Fig.

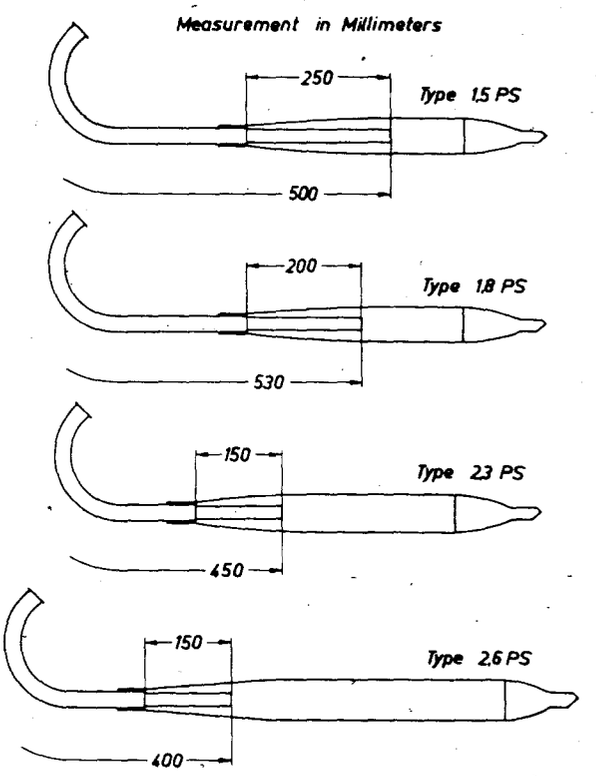

Fig. 15 - Exhaust pipes and mufflers for 1.5, 1.8, 2.3, and 2.6 hp cylinders

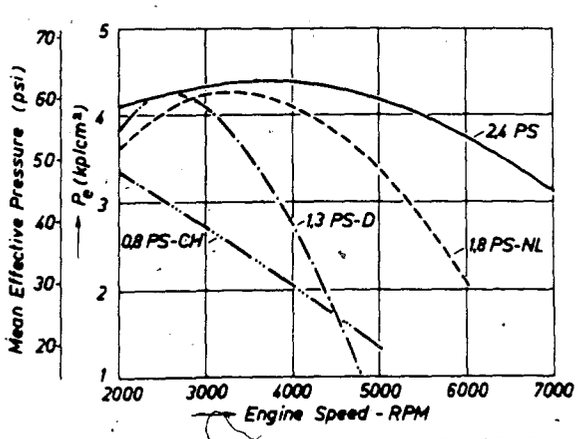

Fig. 14 - Characteristic curves for mep depicted in Figs. 9-12

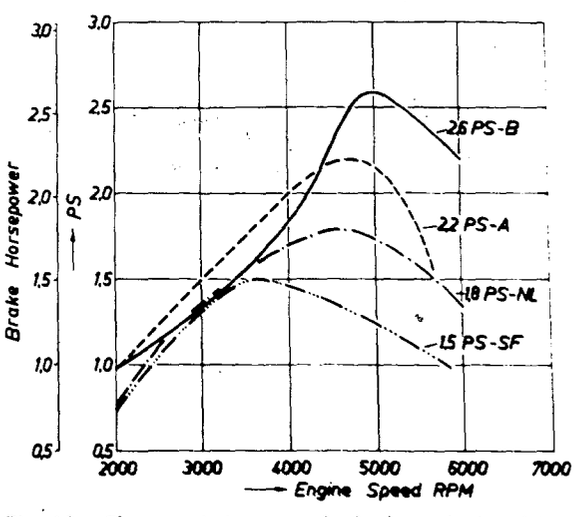

Fig. 16 - Characteristic curves for hp/rpm depicted in Fig. 15

17) show that maximum torque is shifted to the higher rpm range, and increases by shortening the length of exhaust pipe projecting into the diffusor. As is known from pressure gagings, the influence of the more or less suppressed reflection waves makes itself felt here. With a shorter length of pipe, they are dampened less and in this manner a noticeable boost is caused at higher rpm. At lower rpm, however, there is some disturbance during the exchange of charges so that the cylinder charge is lower. Longer lengths of pipe introduced into the diffusor reverse the effect due to a higher degree of dampening of the reflection waves. These pronounced effects have been utilized for quite some time now, together with the inlet diaphragm control for controlling output and limiting rpm. Fig. 17 shows that maximum rpm has not been influenced in spite of the shift in maximum torque. This is essential for the (limited) maximum road speed.

Although in the case of higher output designs flush introduction of the exhaust pipe end would achieve the desired torque in the upper rpm range, the controlled slope effect (sharp drop in the torque curve) following the peak torque would be less pronounced. Therefore, an expedient was found by combining the influence on the reflection waves by means of the exhaust pipe with the shift in torque, thus enlarging the volume of the exhaust muffler body. By doing so, an undesirable deflection occurred. This was tolerated, however, in view of the desired peak performance.

Examples show that an existing engine unit can be adapted to various performance stages by simple means at the vehicle manufacturing plant. For exact harmonization of performance, and (above all) stabilization of the torque characteristic, minor changes in the compression ratios and carburetor adjustments may be additionally made. But the influences achieved in this way are of a secondary nature and may be omitted here.

Influencing Mep and Rpm Through Combined Timing Change and Modification of Exhaust System - An extreme example of how power reserves may be activated in a given engine design is the Sachs 50/4 LKH. This type of engine is based on the same basic engine (Fig. 6) as the engine types previously described. However, it already belongs in the engine class for small motorcycles and is not subject to limitations with regard to maximum output and maximum road speed. Fig. 18 shows the port cross sections, and Fig. 19 a characteristic curve. In this connection (refer to Figs. 9 and 14) the engine may be regarded as the lowest output engine in this series of engine types.

Apart from influencing the timing, attempts were made in a special study of the exhaust system to utilize the pressure wave originated by the exhaust stroke for an increase of the air induction and the qualitative degree of scavenging efficiency. For this purpose, the knowledge gained from studies of combustion-chamber configurations, piston and ring designs, and induction air harmonization was put to good use. These are further discussed.

INFLUENCING SINGLE-CYLINDER, HIGH-EFFICIENCY, TWO-STROKE ENGINE PERFORMANCE

Prior to the development of a high-efficiency engine, the performance determining factors must be clearly recognized and utilized to an optimum degree. As the influence of the individual factors is largely known through the literature, a

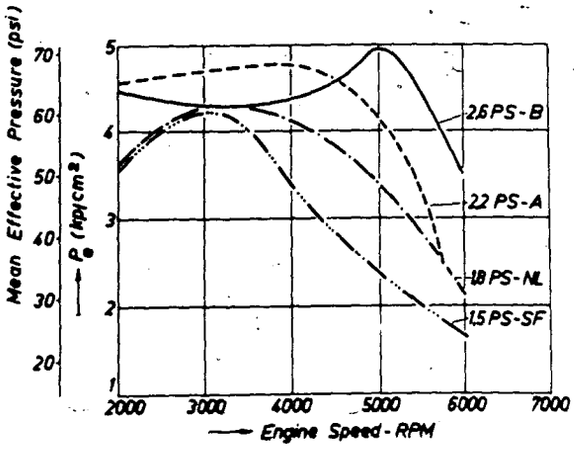

Fig. 17 - Characteristic curves for mep/rpm depicted in Fig. 15

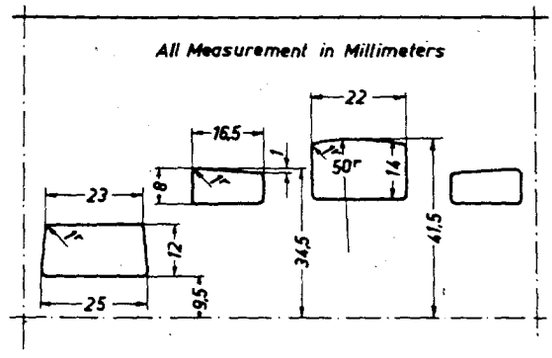

Fig. 18 - Port measurements of Sachs 50/4 LKH

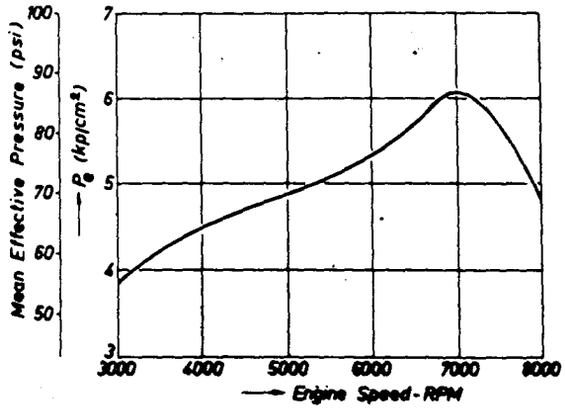

Fig. 19 - Characteristic curve for mep of Sachs 50/4 LKH

detailed discussion may be omitted here. This section will attempt to clarify the following questions:

1. How are scavenging and charging influenced by maintaining tolerances in the manufacture of the guiding edges; by determining tolerances for the exhaust pipe length; by dimensional accuracy of the casting cores; and by exhaust system configuration.

2. How is the combustion process influenced by combustion space configuration and modification of the spark plug assembly.

3. How can combustion pressure be utilized through practical tests with piston rings and cylinder roundness.

FACTORS INFLUENCING SCAVENGING AND CHARGING

Maintaining Tolerances in Manufacture of Guiding Edges in Cylinder - Soon after the design conception of the Sachs 50 S engine (Fig. 20) with an effective output of 5.2 hp at 7400 rpm, a decision had to be made on the manufacturing tolerances to be maintained in the machining of the guiding edges at the upper edge of the exhaust port for series production purposes. The indicator diagram plotted for the effective power is shown in Fig. 21 from recordings of the combustion and charging pressures at 3700 rpm. The mep characteristic of this engine, recorded during test performance, is shown in Fig. 22.

When, for instance, the dependence of the maximum combustion pressure from load at various pre-exhaust sizes is examined at 7400 rpm, the influence exerted by this factor is clearly recognizable. The difference in size was 0.3 mm measured toward the raised portion of the upper edge of the port (Fig. 23).

Fig. 24 shows the dependence of the maximum combustion pressure from the best timing value and also the deviations made in increments of 0.25 mm (7400 rpm). Thus, it is shown that a positive machining tolerance, that is, toward the raised portion of the exhaust port in the grey cast cylinder, is admissible within a maximum range of 0.3 mm.

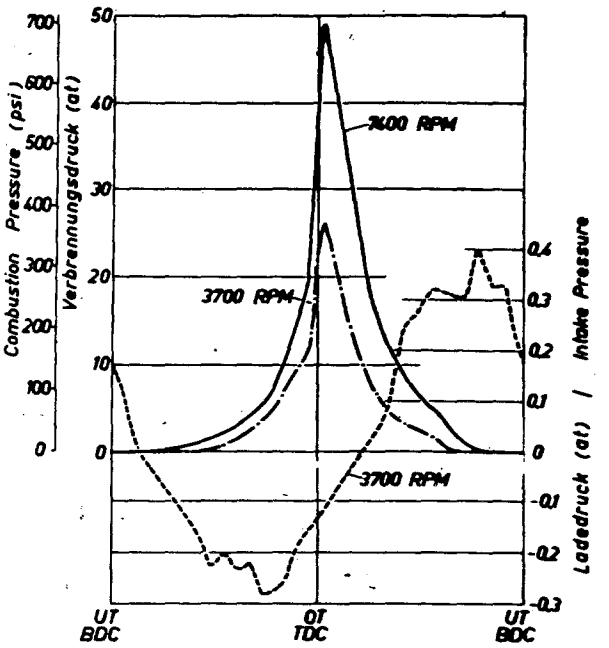

Fig. 21 - Sachs 50 S combustion pressure characteristics at 7400 and 3700 rpm plus charging pressure characteristic at 3700 rpm

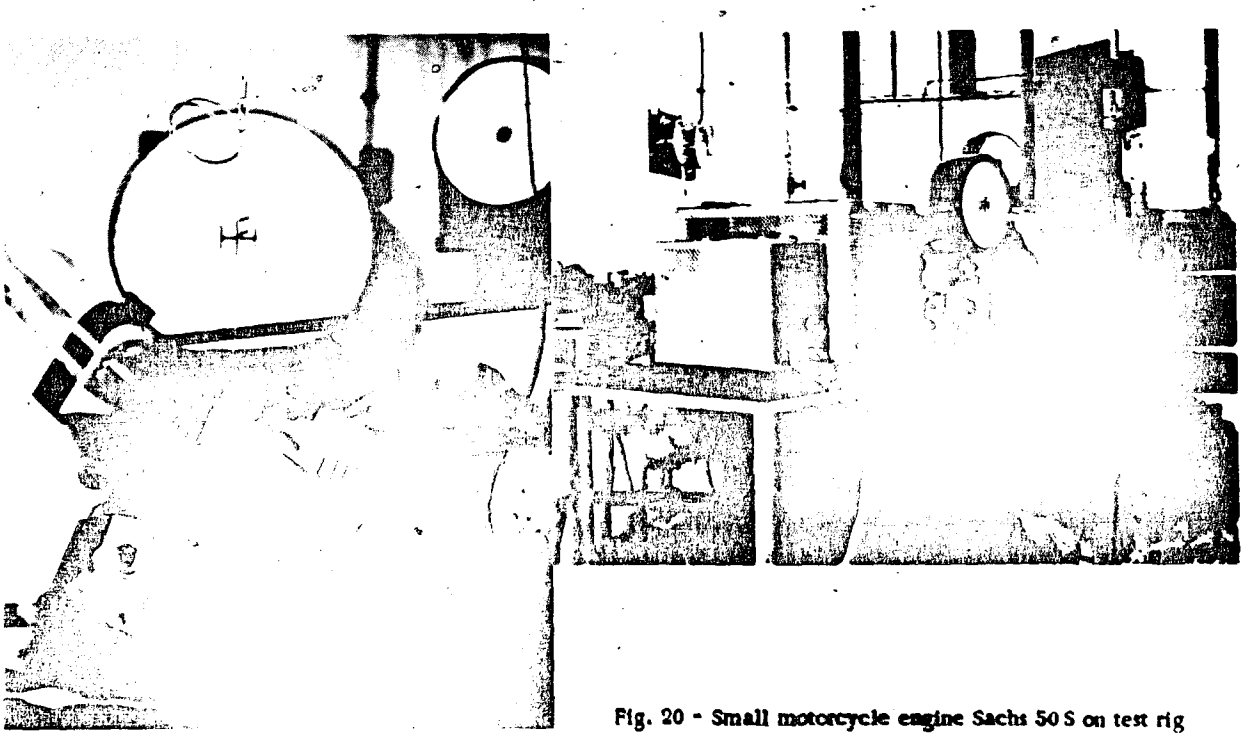

Fig. 20 - Small motorcycle engine Sachs 50 S on test rig

For lower speeds, analogous measurements were made. As expected, the pressure loss was higher by a factor of 2.

Admissible Tolerance of Exhaust Pipe Length - As manufacturers of separately supplied engines, we try to eliminate faulty assembly work at the customers' plants by supplying complete induction and exhaust systems adapted to the respective engines. The various chassis design principles require different exhaust pipe shapes and lengths, because the aim is to rigidly attach a considerable dampening volume at the frame. This makes a uniform exhaust pipe design impossible. We examined the length tolerances that might be acceptable without essential power sacrifices. Fig. 25 shows the mep characteristic as a function of exhaust pipe length and rpm. For each crankshaft speed (7500, 5500, and 3500 rpm), there is a pronounced peak torque. As expected, these peaks are not related to one single exhaust pipe length. For this reason, such manipulations are undesirable with well-harmonized two-stroke engines. Finally, 25 mm was regarded as a permissible length deviation for the exhaust pipe.

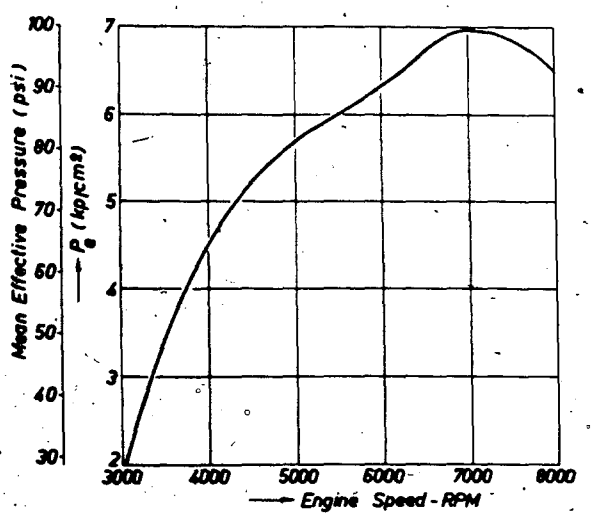

Fig. 22 - Sachs 50 S mep characteristic recorded during test performance

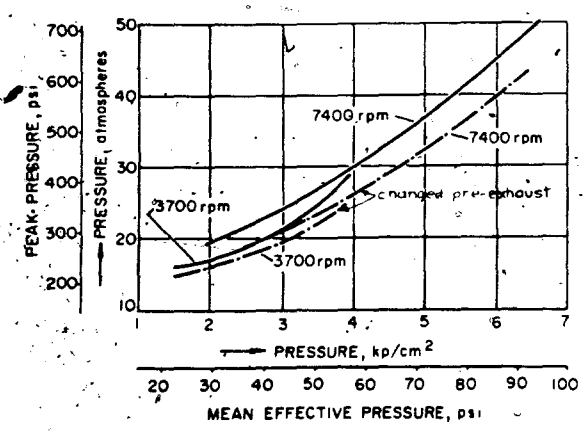

Fig. 23 - Dependence of maximum combustion pressure from load at various pre-exhaust sizes at 7400 rpm

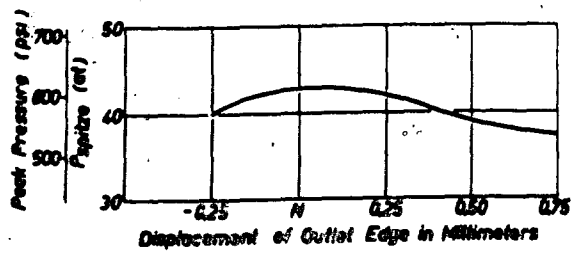

Fig. 24 - Combustion pressure at 7400 rpm depending on outlet edge

Influence of Small Casting Inaccuracies on Engine Characteristics - The importance of exact symmetrically dimensioned transfer ports for the scavenging of remaining gases and the fresh gas charge, and thus for combustion, is shown in Fig. 26. A slight difference in the height of the transfer ports, as may occur when the casting core is negligently put together, causes deviations in engine characteristics to the extent shown in Fig. 26. While tractive power remains practically unchanged, there is a pronounced power deficiency in the speed range from 4500 rpm onward. Starting at this speed range, the degree of delivery is almost constantly 4% below the reference curve. At the same time, there is a drop in the exhaust temperature.

By quality control measures, such displacements of the scavenging ports and other casting inaccuracies which tend to prevent the formation of exact scavenging currents must be eliminated. Fig. 27 shows the port cross sections of the two reference cylinders. In Fig. 28 the arrangement of the scavenging ports can be seen in longitudinal and cross section views of the cylinder. It may be seen that the flow of gases in the ports has been largely adapted to the requirements of fluid dynamics applicable to modern flow controlled engines, among which the present-day, high-efficiency, two-stroker may be counted.

Influence by Exhaust System Configuration - In addition to the achievable maximum output, it is important to take

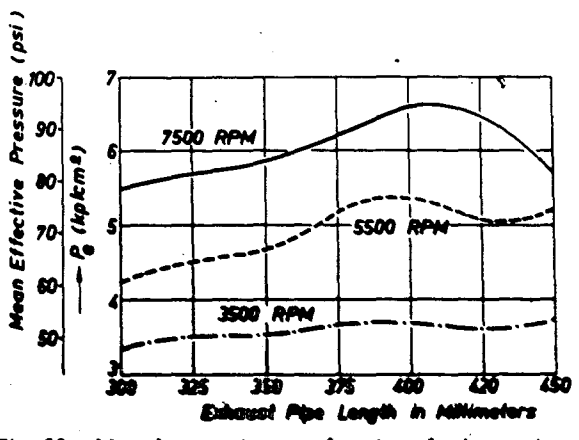

Fig. 25 - Mep characteristics as function of exhaust pipe length and rpm (3500, 5500, 7500)

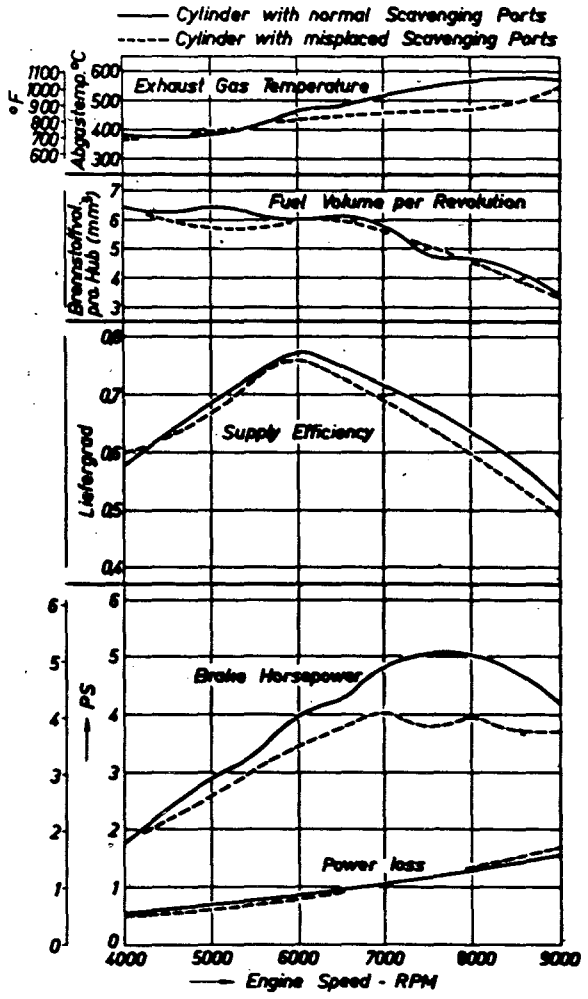

Fig. 26 - Characteristics caused by normal and misplaced scavenging ports

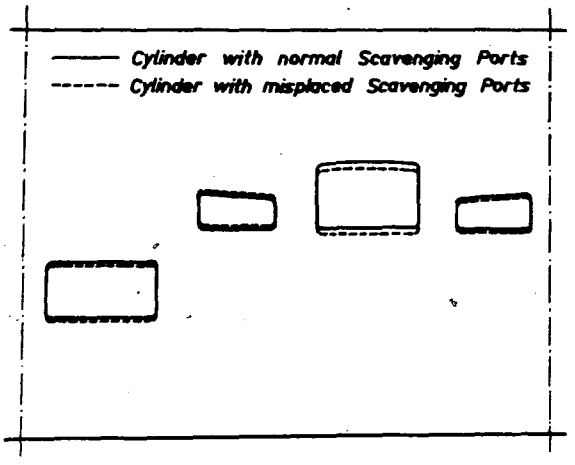

Fig. 27 - Comparison of port measurements of cylinders with normal and inaccurate cast scavenging ports

into consideration the torque characteristic for engine elasticity. These properties are largely determined by the entire exhaust system. For this reason, various modifications in the power influencing factors were examined in a series-built exhaust assembly. The following were studied: configuration of the diffuser; location of the reflection wall, and thus the length of the first expansion chamber; configuration of the reflection wall; and configuration of the muffler end piece. The measurements included only one influencing factor at a time in the exhaust system completely equipped with muffling chamber.

1. Configuration of the diffuser: In order to determine the influence exerted by the exhaust taper or, more precisely, the angle of taper, various muffler configurations with otherwise unchanged external dimensions were adopted. Three such examples are shown in Fig. 29.

From Fig. 30 it may be seen that the taper angle not only influences the displacement of the maximum mep but also the maximum value. With a smaller angle, the maximum mep is shifted to below its value and location. The optimum angle (found by small increments) is about 6 deg. Presumably, there is no separation of the flow at this angle, so that marginal eddies and throttling are avoided. In order to largely eliminate the influence of the adjoining cylin-

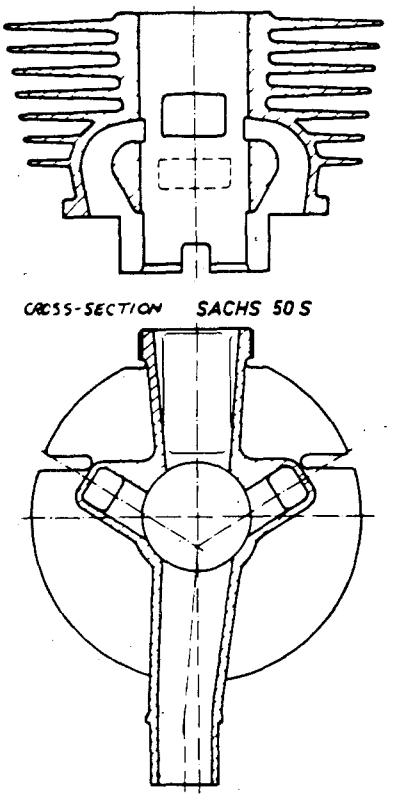

Fig. 28 - Longitudinal and cross-section views of Sachs 50 S cylinder

drical shell, and thus the chamber volume, the baffle plate was located by increments of 50 mm behind the diffuser so that the adjoining chamber size could be regarded as being constant. Since this would have changed the location of the reflection wall in respect to the exhaust opening in the cylinder, a compensation for length was made by means of the exhaust pipe.

2. Location of the reflection wall: The reflection wave, which is positively reflected at the end wall of the first expansion chamber, may have a detrimental effect when it backfires through fully opened ports into the cylinder and is propagated via the transfer port down into the crankcase. With the aid of the reflection wall, the character of the reflection wave can be influenced at the appropriate engine speed. Therefore, every attempt should be made, by proper location and configuration of the reflection wall, to harmonize the moment of arrival, time dwell, and energy of the reflection wave in such a way that it will arrive at the exhaust port at the most favorable moment and at the rated maximum power rpm, but will not interfere too much at other engine speeds. Fig. 31 shows the locations of the reflection wall which were examined.

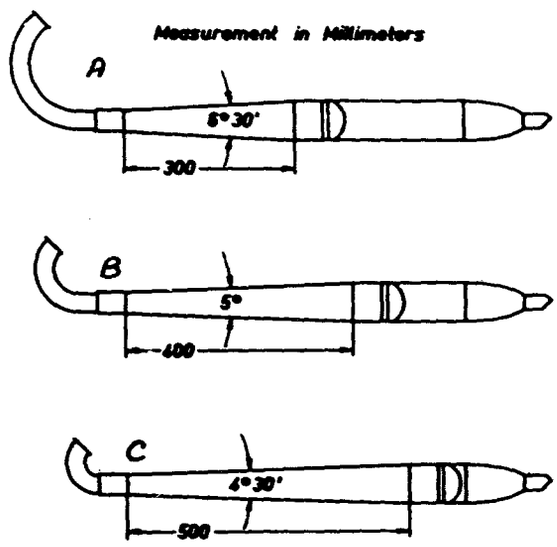

Fig. 29 - Exhaust mufflers with various diffuser aperture angles

By a change in baffle plate location, the arrival time of the reflection wave (similar to the change in length of the exhaust pipe) is influenced. With the concurrently effected change in the length and volume of the expansion chamber, the time dwell of the reflection wave, and thus its characteristic, is changed. The combined effect resulting from these two influences is shown in Fig. 32. Maximum torque is greatly shifted with respect to its value.

3. Configuration of the reflection wall: A semisphere with variously shaped off-flow openings was examined. The shape of the semisphere had been found to be an optimum solution in preceding tests. The question was to what extent the energy of the arriving pressure wave must be reflected in order to achieve an optimum output. For this purpose, the number of bores in the semisphere were reduced (Fig. 33).

From the curves plotted in Fig. 34 it may be seen that, in comparison with the configuration shown at the top of Fig. 33, an increase in the mep may be obtained by increasing the reflection (center configuration of Fig. 33). This increase occurs particularly in the upper rpm range. At further throttling (bottom of Fig. 33), that is, even greater reflection of the pressure wave, this effect is reduced. Thus, it is

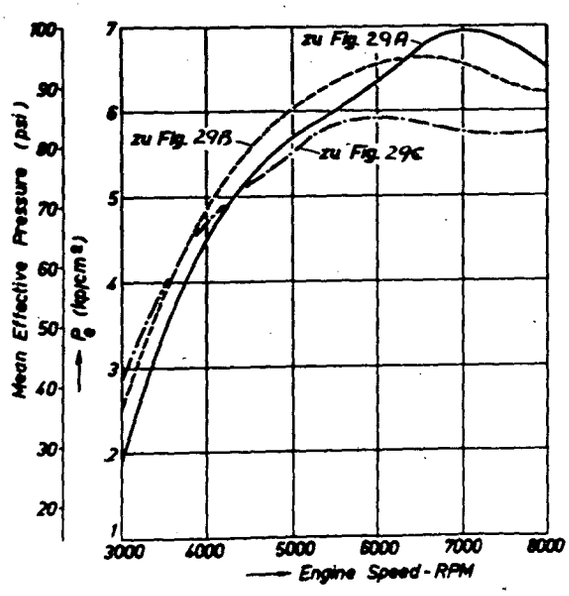

Fig. 30 - Characteristic mep for exhaust mufflers depicted in Fig. 29

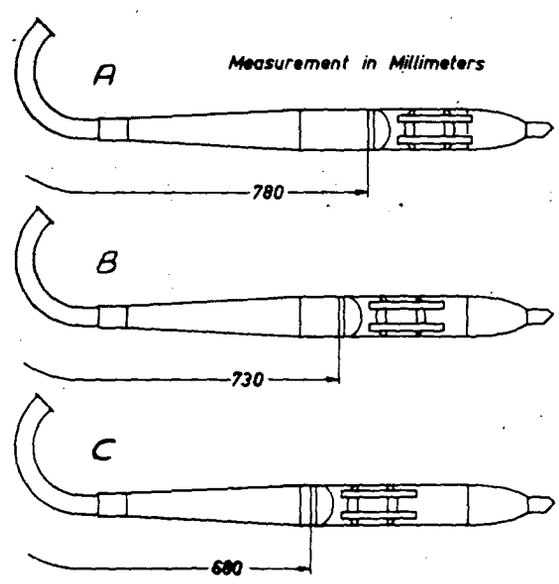

Fig. 31 - Exhaust mufflers with various diff

shown that a too intensively oscillating exhaust system may cause disturbances in the gas exchange. A tapered reflection wall, for instance, may be sufficient to largely equalize undesirable effects because it has reflection properties that differ from those of the shape examined. It is therefore used more frequently.

4. Configuration of the exhaust tail piece: In order to examine the influence of the configuration of the silencing member lined up behind the reflection wall, various tail pieces of the muffler were measured. The interior assembly was otherwise unchanged (Fig. 35). The influence expected did not make itself felt (Fig. 36). Moreover, it could be proved by the oscillogram that the characteristic of the resonant waves remained practically unchanged. In this connection, it would appear that the size of the off-flow openings from the first chamber, and particularly the uniflow effect of the silencing member, has some effect by dampening repercussions from the adjoining chamber lined up behind the first chamber.

On the other hand, it was known that when the silencing

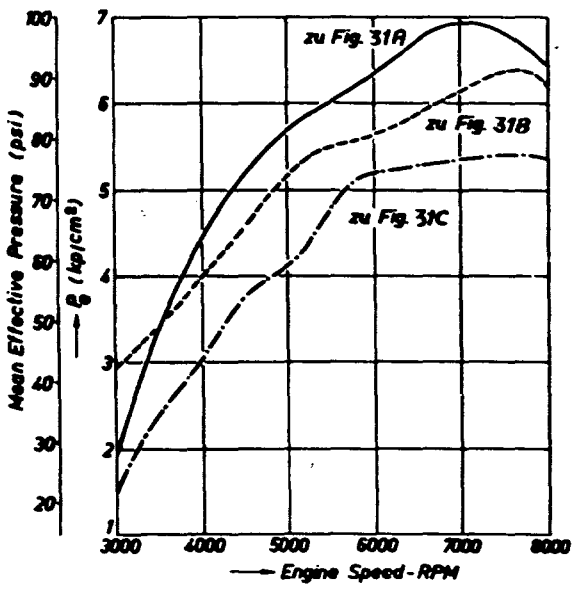

Fig. 32 - Characteristic mep for exhaust mufflers depicted in Fig. 31

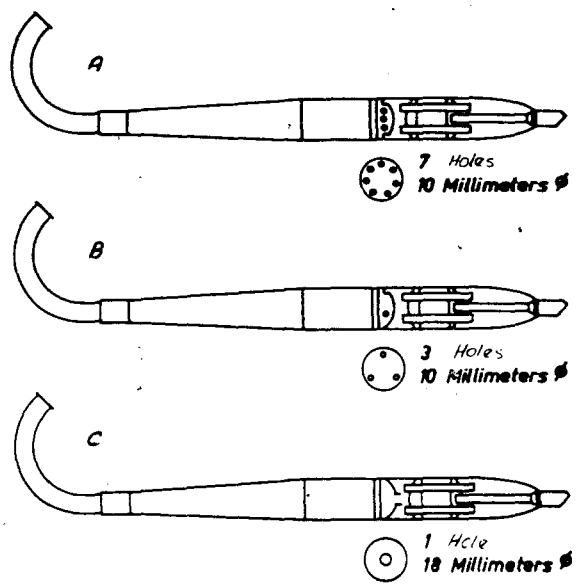

Fig. 33 - Exhaust mufflers with various configurations of reflection wall

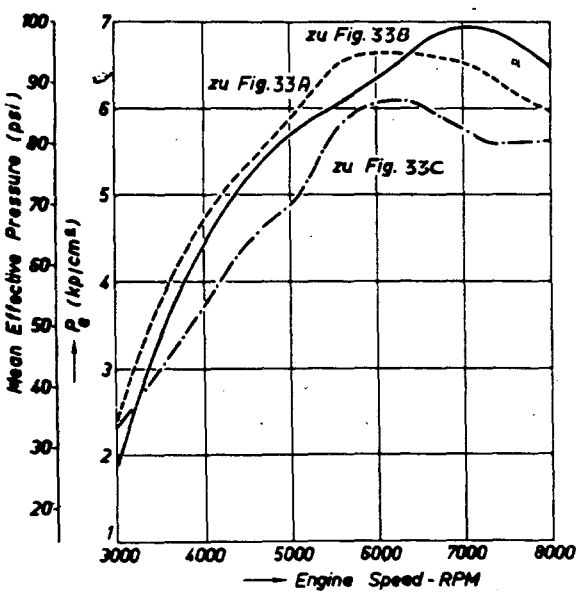

Fig. 34 - Characteristic mep for exhaust mufflers depicted in Fig. 33

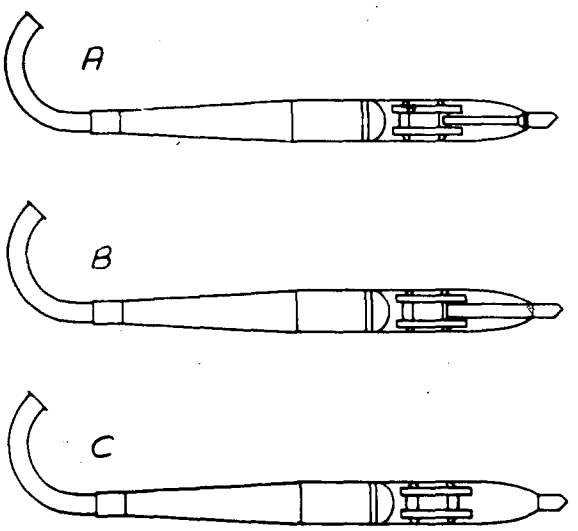

Fig. 35 - Exhaust mufflers with various configurations of exhaust tail piece

member (sports engines) was omitted and reflection walls with larger off-flow openings were used, a gain in power can be obtained. Here it was proved that the second chamber, in interdependence with the tail piece, is able to suppress persistent oscillations of the reflection wave, and thus disturbances in the flow of exhaust gases from the cylinder.

In summary; the designing and dimensioning of the exhaust system is still largely a matter of practical tests. The extract given above of our experiments is not intended as a prerequisite for increasing engine power, but is offered as a survey of the influence exerted by certain factors.

FACTORS INFLUENCING COMBUSTION PROCESS

Configuration of Combustion Space - Consider an example to show how the maximum combustion pressure can be influenced by the combustion space configuration at an unchanged compression volume, and thus unchanged compression ratio. The combustion space is shaped hemispherically. What is changed are its radius and particularly the compression zone. Fig. 37, shows the configurations, and Fig. 38 the maximum pressure at 7500 and 3750 rpm.

The two less pronounced compression zones shown in Fig. 37 (center and bottom) render a reduced peak pressure. As is known from the combustion pressure characteristic, the pressure increase in such cases is less steep because the combustion process is delayed. Furthermore, there is an interdependence between combustion space configuration and scavenging. This is to say that for every power determining factor, whether scavenging pressure, location of the scavenge flow, scavenging system, or its combination, the optimum combustion space configuration must always be found experimentally.

Spark Plug Assembly - Various demands determine the position of the spark plug. The two-stroke charge (mixture of gasoline plus oil) should be fired as centrally as possible and thus contribute to the formation of a concentrically pro agated flame front. On the other hand, the spark plug elec trodes should be thoroughly flushed by fresh gas for cooling purposes and to avoid bridging. However, for reasons of spa and accessibility, this ideal position cannot always be realized.

In the configuration examined, the position of the spark plug was not central. In Fig. 39, four radial spark plug arrangements are shown, and in Fig. 40 three axial spark plu positions are shown. Fig. 41 shows the influence of these measures on maximum combustion pressure.

The pressure characteristic reveals that not only is the position of the spark plug axis relative to the scavenging fl

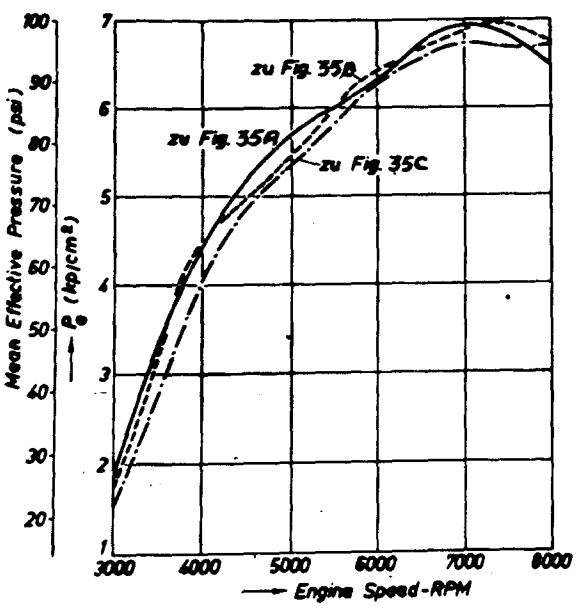

Fig. 36 - Characteristic mep for exhaust mufflers depicted in Fig. 35

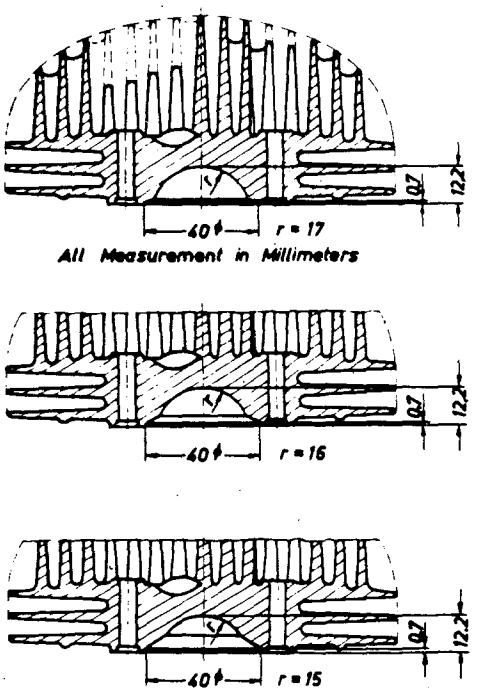

Fig. 37 - Combustion space configurations with various compression zones

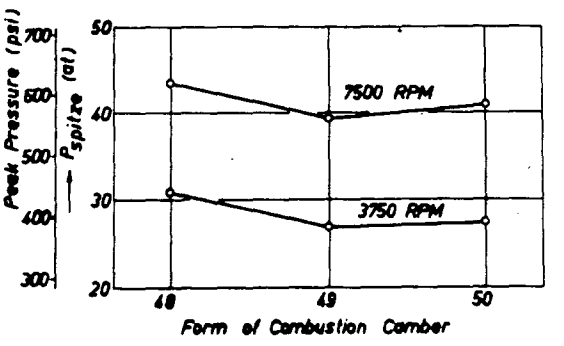

Fig. 38 - Peak pressure characteristic at 7500 and 3750 rp for combustion space configurations depicted in Fig. 37

decisive, but also the immersion depth of the electrodes into the combustion space. The optimum solutions found at 7400 rpm were: symmetrical position relative to the scavenging flow and electrodes projecting by about 3 mm. At low engine speeds, these effects are hardly noticeable.

OPTIMUM UTILIZATION OF COMBUSTION PRESSURE

Practical Tests with Piston Rings - The effects of the number, position, height, and material of piston rings, type of ring gap closure, type of grinding, axial and radial play on the piston temperature, sealing, friction and wear -- all these are generally known. But as a point of interest, we attempted to prove that mep might have a possible influence on small piston displacements and high engine speeds. Thus, the following examinations that were conducted will be dealt with here: influence of the ring position and ring shape; influence of the ring gap (closure); and influence of the number of rings.

Since it was expected that the individual influences woul be small, special provision had to be made for power measurements in order to obtain a high degree of accuracy of the information thus obtained. Altogether, two pistons were used, identical in weight, bottom thickness, shaft diameter, piston play, and chemical analysis. Both had been artificially aged under the same conditions and showed the same performance when operated in a cylinder on the test rig. The homogeneity of the texture had been established by measuring the external diameters at different temperatures. From the 25 pistons selected for testing, two were found which met all these requirements. In order to remove any stress or strain, the test cylinder had been operated at full load for 100 hr and was then reground. Measurements were made at 15 minute intervals at thermal equilibrium. Ambient conditions were kept constant. Also controlled were the amount and temperature of the coolant; cylinder temperature; fuel con-

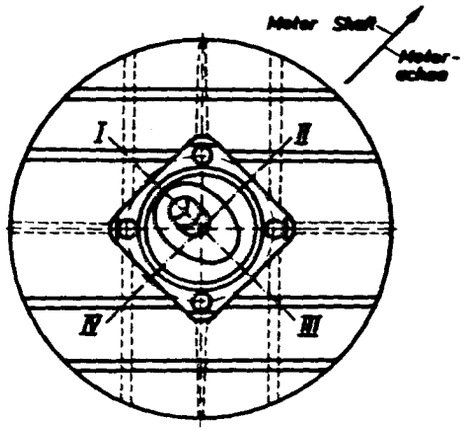

Fig. 39 - Different spark plug positions radial varied

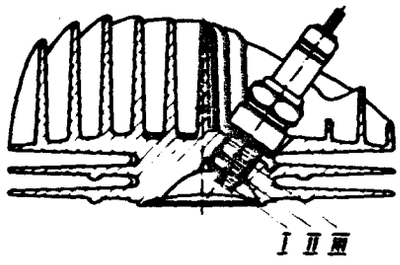

Fig. 40 - Different spark plug positions axial varied

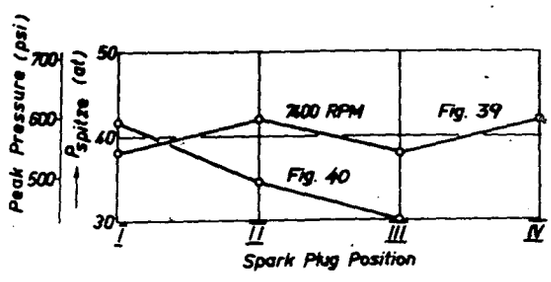

Fig. 41 - Peak pressure characteristic for spark plug positions depicted in Figs. 39 and 40

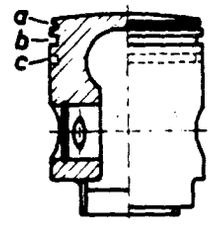

Fig. 42 - Ring groove positions at piston (a - L-ring groove; b - upper rectangular groove; c - lower rectangular groove projected)

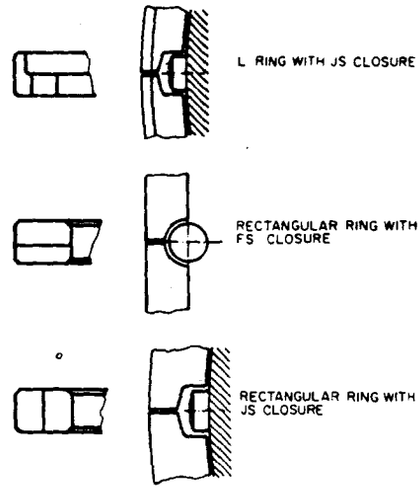

Fig. 43 - Piston rings and ring gap closures

sumption; induced air volume; ignition timing; and deposits on the piston head and in the exhaust system.

The checked rings were new and had not been subject to wear during the measuring operations. Maximum cylinder wear was 5 microns. The measurements made served only to verify the results which had been previously obtained from numerous test series. The sequence of the ring combinations had to be chosen in such a way that by subsequent cutting of a new ring groove, or by altering an existing groove, a new modification could always be tried out (Figs. 42 and 43). For measurement 2 (Fig. 44), the existing groove of the 1.5 mm rectangular ring was unringed as for measurements 4-10 (Figs. 45 and 46), the groove of the L-ring in the piston head. However, control measurements 14 (Fig. 48) and 3 (Fig. 44) showed that in the interesting rpm ranges the influences of the empty grooves were within the range of unavoidable measuring mistakes.

The following types of piston rings were examined: rectangular ring JS-38/34.5 × 1.5 fz DIN 24910, referred to as JS-ring 1.5 mm; rectangular ring JS-38/35 × 2 fz nonstandard, referred to as JS-ring 2 mm; rectangular ring FS-38/35 × 2 fz nonstandard, referred to as FS-ring 2 mm; and L-ring JS-38 ϕ, nonstandard, referred to as L-ring.

1. Influence of piston ring location and ring shape: In Fig. 44, the mep characteristic has been plotted for the piston fitted with JS-ring 1.5 mm (measurement 1) and an L-ring (measurement 2). When measurement 3 was made, the groove cut in the meantime for the L-ring was empty. From the curve plotted for this control measurement, it may be seen that the empty L-ring groove hardly had any influence on the test result.

Apparently as a result of its better sealing performance, the L-ring provides a torque increase from 6000 rpm upward. Although it is subject to extreme thermal stress due to its exposed position, heat conduction to the cylinder wall appears to be satisfactory owing to the large matting surface Rectangular rings fitted in this position will fail. The normally arranged rectangular ring (measurement 1), although not greatly subject to the thrust of the combustion gases, attains the sealing performance of the L-ring only when fitted in pairs, but then it is inferior to the L-ring because of higher friction.

2. Influence of the piston ring height: In Fig. 45, JS-ring 1.5 mm (measurement 4) is compared to JS-ring 2 mm (measurement 5). Measurement 6 shows the mep characteristic for FS-ring 2 mm, which differs additionally by the type of ring gap closure. Measurement 6 is therefore excluded from this discussion, because here only rings of the same design but of differing axial heights are to be compared.

At about the same total tension, the ring with the lower axial height better conforms to the cylinder liner as a result of its higher specific expanding pressure. It would appear that it can better follow irregularities in the cylinder diameter and fit more snugly into any unevenness. Friction influence appears to be improbable here because, as mentioned above, the piston ring tension was about equal for both rings.

3. Influence of the piston ring gap (type of closure): Measurements 5 and 6 (Fig. 45) indicated that the FS closure would have to be regarded as less efficient. Further confirmation is given by the measurements shown in Fig. 46, conducted with JS-ring 2 mm (measurement 7) and FS-ring 2 mm (measurement 9). Measurement 8 was made with two JS-rings 2 mm, and measurement 10 with two FS-rings 2 mm.

The influence of the ring gap closure on sealing performance is quite clear both for pistons with one ring and with two rings. Apart from some inconstancy below 5000 rpm, the trend in the mep characteristic shows an outspoken gain

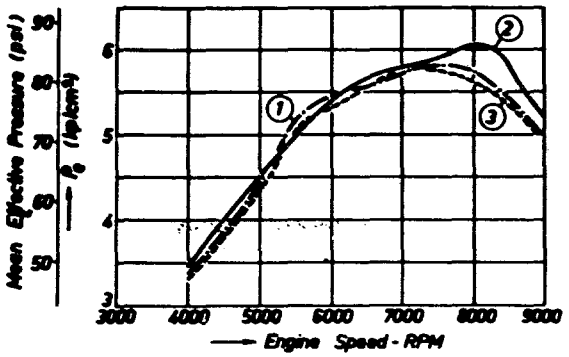

Fig. 44 - Mep characteristic with various piston ring locations

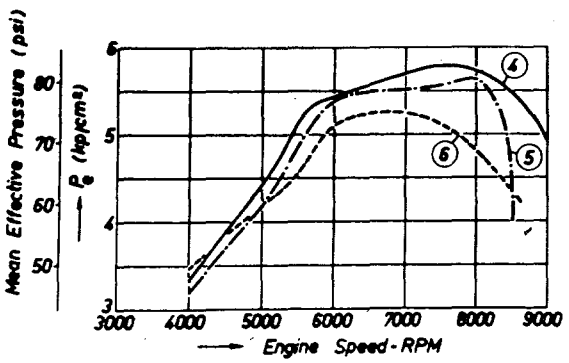

Fig. 45 - Mep characteristic for various piston ring heights

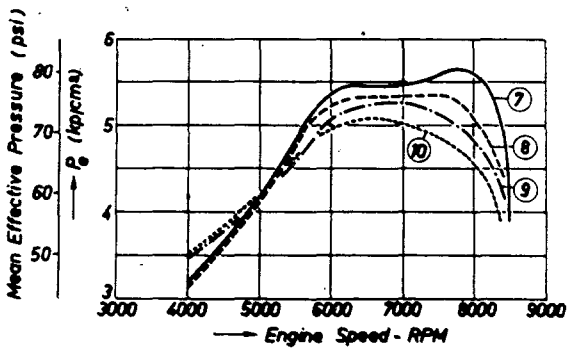

Fig. 46 - Mep characteristic for various types of ring gap closure

in torque in the upper speed range. This superiority of the JS-ring closure, which can only be explained by the smaller gap loss, is offset by greater assembly difficulties and a higher susceptibility to wear. Nevertheless, the usefulness and fitness of these rings for further use could be proved after 40,000 km nonstop trials. Efficient air cleaning is, of course, indispensable for such results.

4. Influence of the number of piston rings: Fig. 47 shows the mep characteristics for one L-ring (measurement 11) and two L-rings (measurement 12). Doubling the number of rings must necessarily show up in higher friction. With the two-piston ring systems having inferior sealing properties, the increased friction losses take up the gain in pressure (compare measurement 8 with 7, and 10 with 9 in Fig. 46). The mep for two piston rings (regardless of the type of ring gap closure) is below that of single-ring systems over the entire rpm range. On the other hand, measurements 11 and 12 (Fig. 47) show that the good sealing properties of the two L-rings admit for effects of increased friction only from 8000 rpm upward.

The mep loss shown to occur with two-ring systems cannot be explained by disturbances of the heat flux in the piston (Fig. 48). Below the L-ring, grooves of 1 mm depth (measurement 14) and 2 mm (measurement 15) were cut. In comparison with the mep obtained with an L-ring (measurement 13), artificially provided heat throttles had no influence.

Examination of Contour Stability of Cylinders - Deformations of the piston and cylinder as a result of nonuniform temperature distributions, differing coefficients of thermal expansion, and material stresses and strains caused by the addition of assembly and operating strains influence the piston play and render the sealing functions of the piston rings more difficult. The thermal and mechanical deformations of the piston can be largely compensated for by the proper selection of its material, by complicated machining, and by integral casting of sheet steel strips. Perfect cylinder roundness, particularly with air coolant, depends on the non-deforming properties, good strain and stress and heat distribution and unvaried cooling. And, apart from design and construction measures, the cylinder material plays a decisive role.

It appeared to be of interest to examine the kind and extent of deformation, that is, the deviation from the ideal cylinder shape as dependent alone on the material, and then to compare the results with the stabilizing influence by enlarging the ribs. In view of the overlapping thermal stresses, pre-stresses, and operating stresses, and the changes at var-

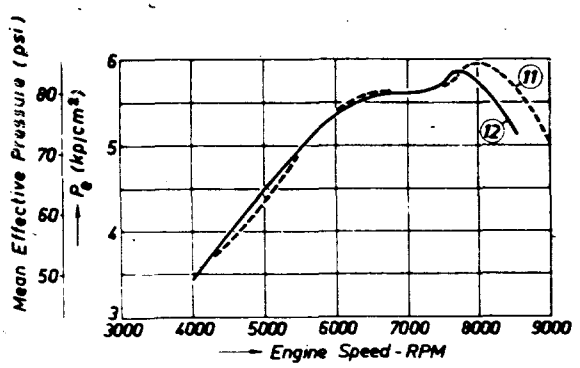

Fig. 47 - Mep characteristic for various numbers of piston rings

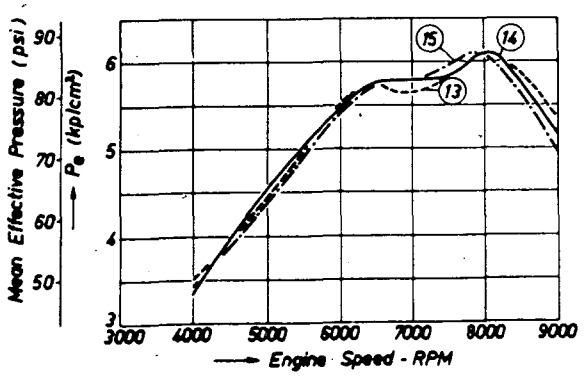

Fig. 48 - Mep characteristic with artificially provided heat throttles

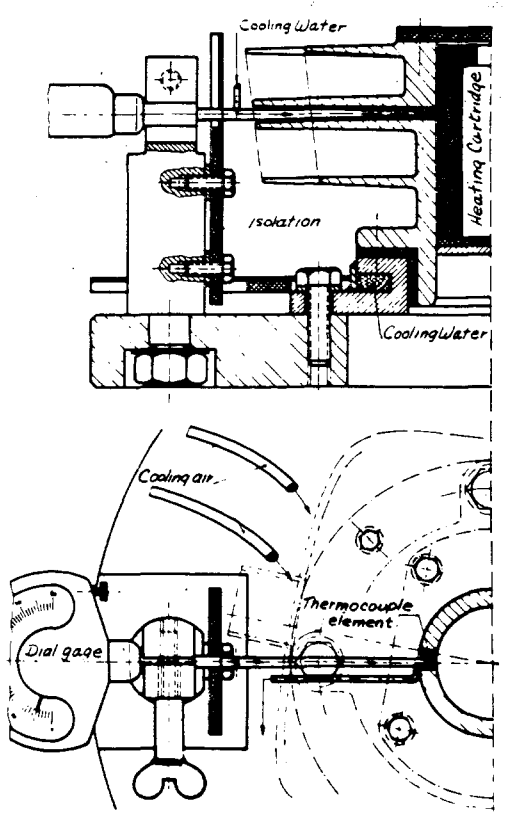

Fig. 49 - Device for measuring cylinder deformation

ious operating conditions, it was necessary to assess the changes in shape under static condition. For this purpose, a measuring device was used which could ascertain the deformation of the cylinder and the influence of thermal expansion at one level. The measuring device was made independent of the ambient temperature by connecting a coolant circulation (Fig. 49). During the first few test series, the cylinder deformations occurring at true-to-life temperature distribution found in a trial run were transferred to the cartridge heated test specimen by well-batched cooling airstreams from 10 nozzles.

Fig. 50 shows the deformations of two equally sized cylinders of grey cast iron (GG 3) and aluminum (GAL Si 7, Cu 3) at the temperature distribution corresponding to that of the grey cast cylinder. The resultant deformation of the aluminum cylinder was unexpectedly large, so that in a further test series the cylinder distortions of both test specimens were ascertained at the same amount of heat input. This

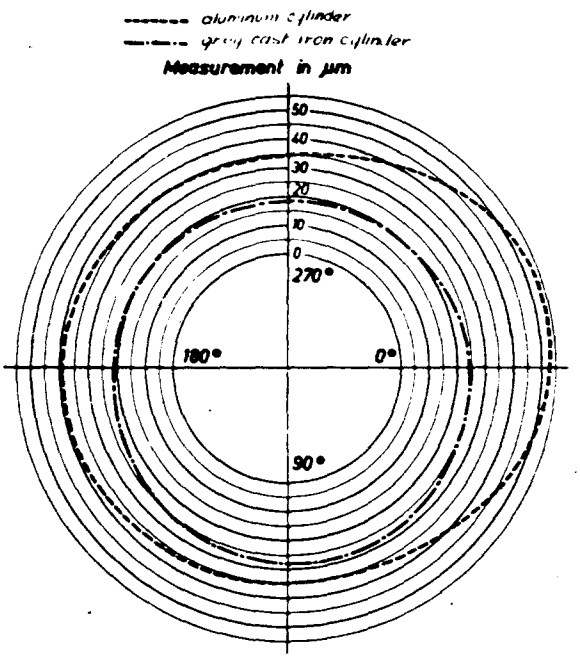

Fig. 50 - Cylinder deformation of heated aluminum and grey cast cylinders at same temperature distribution

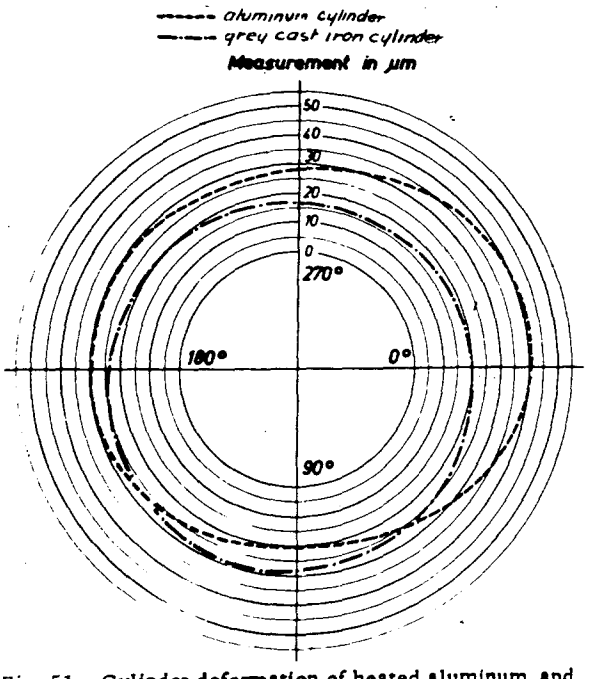

Fig. 51 - Cylinder deformation of heated aluminum and grey cast cylinders at same heat input

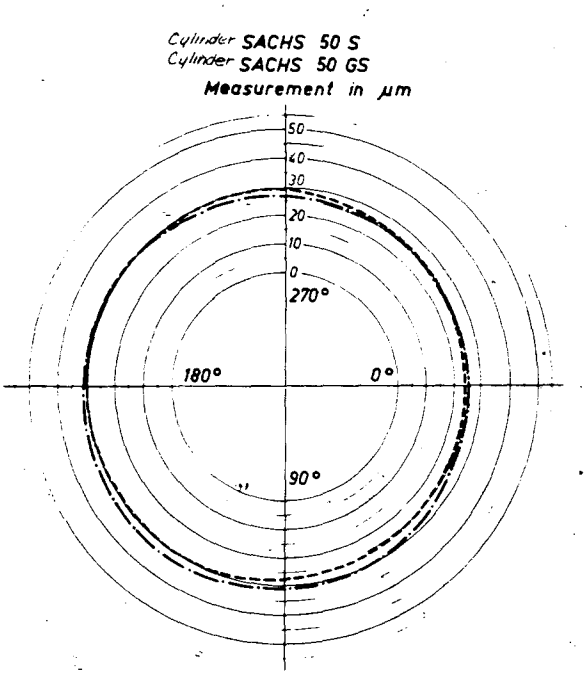

Fig. 52 - Cylinder deformation of two heated, grey cast cylinders with different cooling vanes depicted in Fig. 53

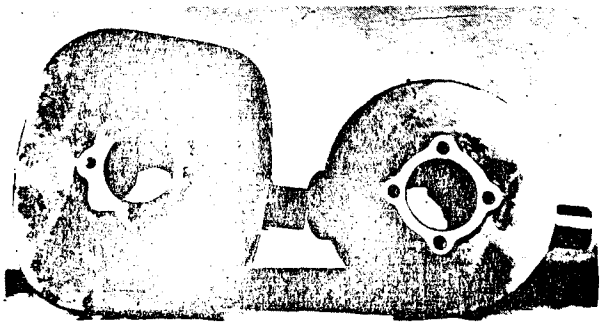

Fig. 53 - Cylinder models used for test depicted in Fig. 52 (Right-cylinder from small motorcycle Sachs 50 S engine; Left - cylinder from cross-country Sachs 50 GS sports engine)

was based on the consideration that due to its better heat conductivity the aluminum cylinder will have a 30 deg lower temperature level in actual operation. As may be seen from Fig. 51, the influence of the lower cylinder temperature on deformation is noticeable in the case of aluminum.

As a point of interest, the extent of the deformation influences which may arise in grey cast cylinders through construction measures was ascertained. For this purpose, cylinders of different rib size were subjected to test conditions corresponding to the temperature distribution of the smaller ribbed cylinder. Fig. 52 shows that the configuration stability of the grey cast cylinder can be influenced to a much lesser extent only by enlarging the ribs. Finally, it must be mentioned that cooling air conditions were chosen for this test as they arise in a motorcycle. As is known, these are not favorable because the cylinder is located in lee of the front-wheel mudguard. This, in particular, is the reason why such relatively large one-sided deformations occurred in the aluminum cylinder during the experiments described above.

Without doubt, the aluminum cylinder allows for the better form retention by efficient cooling air distribution as available through fan cooling and air guiding systems. This may not influence the form retention of good grey cast cylinders, however. The cylinder models that were used in the test depicted in Fig. 52 are shown in Fig. 53.

SUMMARY

Attempts to influence the power of small two-stroke engines have not yet come to an end. Many methods and procedures for increasing engine output have been developed and followed to date. It would be beyond the scope of this paper to try listing the major part played by the power determining factors involved and their treatment. Rather, this paper attempted to show some of the problems associated with the two-stroke engine, and to report on experiment induced effects on series produced engines. The successful development of efficient two-stroke engines, which includes a satisfactory power output, will never result from the accumulated successes of individually applied measures but from the proper selection and combination of optimum factors.