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Honda Formula One Turbo-charged V-6 1.5L Engine

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Honda R&D Co.,Ltd.

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ABSTRACT

The "RA168E", a turbo-charged V-6 1.5-liter engine, was developed by Honda Motor Co., Ltd. for the 1988 Formula One Championship Race events.

Despite boost restrictions (2.5bar), the engine boasts a maximum power of 504 kw (685 ps), which is equivalent to 336 kw/ℓ (457 ps/ℓ). The development of improvements on the fuel consumption of this engine allowed the achievement of a minimum brake specific fuel consumption of 272 g/kwh (200 g/Psh).

This paper introduces major specifications, along with power output and fuel consumption characteristics of the RA168E racing engine. In addition, the effects of intake air temperature, boost, air-fuel ratio, fuel temperature and fuel ingredients on fuel efficiency and power output are presented.

SINCE 1983, HONDA has been supplying leading Formula One racing teams with dual turbo-charged V-6 1.5-liter engine. Although the engines were all identical in overall configuration, they were modified each year as the regulations were revised; therefore, respective engine code names were used for each modification, as shown in Table 1.

Honda engines have won 40 Grand Prixs in the last five years. As a result of these victories, Honda has been awarded the Constructor's Championship for the last three consecutive years. In accordance with the new regulations, beginning in 1989 only naturally-aspirated 3.5-liter engines will be allowed, and turbo-charged 1.5-liter engines are eliminated from participating in the Grand Prix races.

Stricter fuel restrictions have been gradually adopted, as shown in Table 1. In 1988, the restriction became particularly noticeable, with the boost limit being lowered to 2.5 bar and the maximum fuel capacity being restricted to 150 liters for the turbo-charged cars. Fuel restrictions lead to a reduction in total energy available during a race; therefore, indirectly limiting the maximum power of the engines. This description characterizes the 1988 Grand Prix racing season.

Honda's 1988 engine, the RA168E, was developed to simultaneously realize higher power and better fuel efficiency. The highest possible achievement of these two conflicting objectives resulted in an unprecedented 15 victories in 16 events in the 1988 World Championship races.

Few research papers have mentioned high-speed, high-boost engines(1-3)*; and there has been no research reported concerning the fuel efficiency of those engines. This paper introduces the major specifications and power characteristics, and presents study results relating to the fuel efficiency of the RA168E.

RA168E ENGINE SPECIFICATIONS

A highly rigid and compact cylinder block and bearing caps are cast from strong and stiff ductile cast iron. For reducing its weight, cylinder block thicknesses range between 2-3.5 mm, which also satisfies stiffness requirements. Aluminum alloy (Al-Si6Cu4) was adopted as the material for the cylinder head, while almost all remaining parts being cast from magnesium alloy (e.g., oil-sump and head cover). All this resulted in a compact, light-weight engine; only 146 kg fully-dressed with all equipment. This light-weight coupled with excellent power characteristics and reliability, made the RA168E

Table 1 Restrictions on turbo-charged cars and respective engine code names

Year	1985	1986	1987	1988
Boost pressure(bar. abs)	—	—	4.0	2.5
Fuel Capacity(ℓ)	220	195	195	150
Engine code name	RA165E	RA166E	RA167E	RA168E

*Numbers in parentheses designate references at end of paper.

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a highly competitive engine. The major engine specifications are given in Table 2, Figures 1 and 2 show the cross section and the external appearance of the RA168E, respectively.

The RA168E features a stroke of 50.8 mm and bore of 79 mm, (a stroke-bore ratio of 0.643), providing it with somewhat of a longer stroke when compared with ordinary racing engines. The valve angle is relatively narrow (32°), with the heads of the pistons set almost flat. This configuration allows a compact combustion chamber and a fairly high compression ratio of 9.4 to realize high power output and good fuel efficiency.

The camshafts are driven by the crankshaft via two idler gears. The valve operating mechanism, using rocker arms, allows high-load valve springs in the limited space between the valves. Through using this type of configuration, the reduction in the equivalent inertia weight of the mechanism has improved the revolution limit to an engine speed of more than 13,500 rpm.

Table 2 RA168E engine specifications

Engine Code name	168E
Layout	80°V6
Bore (mm)	79
Stroke (mm)	50.8
Displacement (cc)	1494
Compression Ratio	9.4
Weight (kg)	146
Fuel System	HONDA PGM FI
Ignition System	HONDA PGM IG(CDI)
Super Charger	Ceramic Turbo

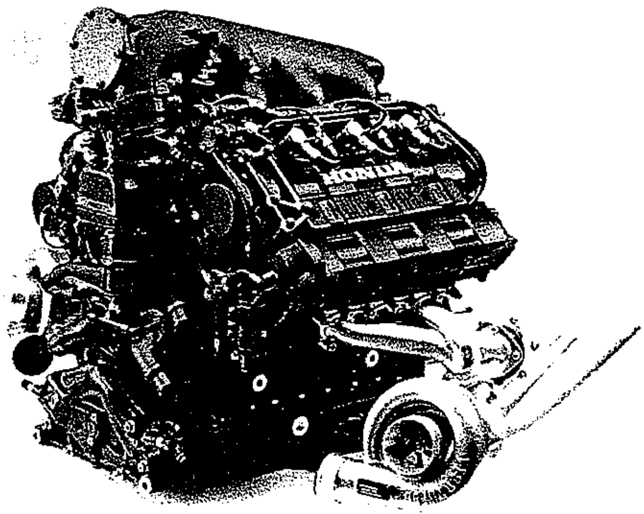

Fig. 2 RA168E external appearance

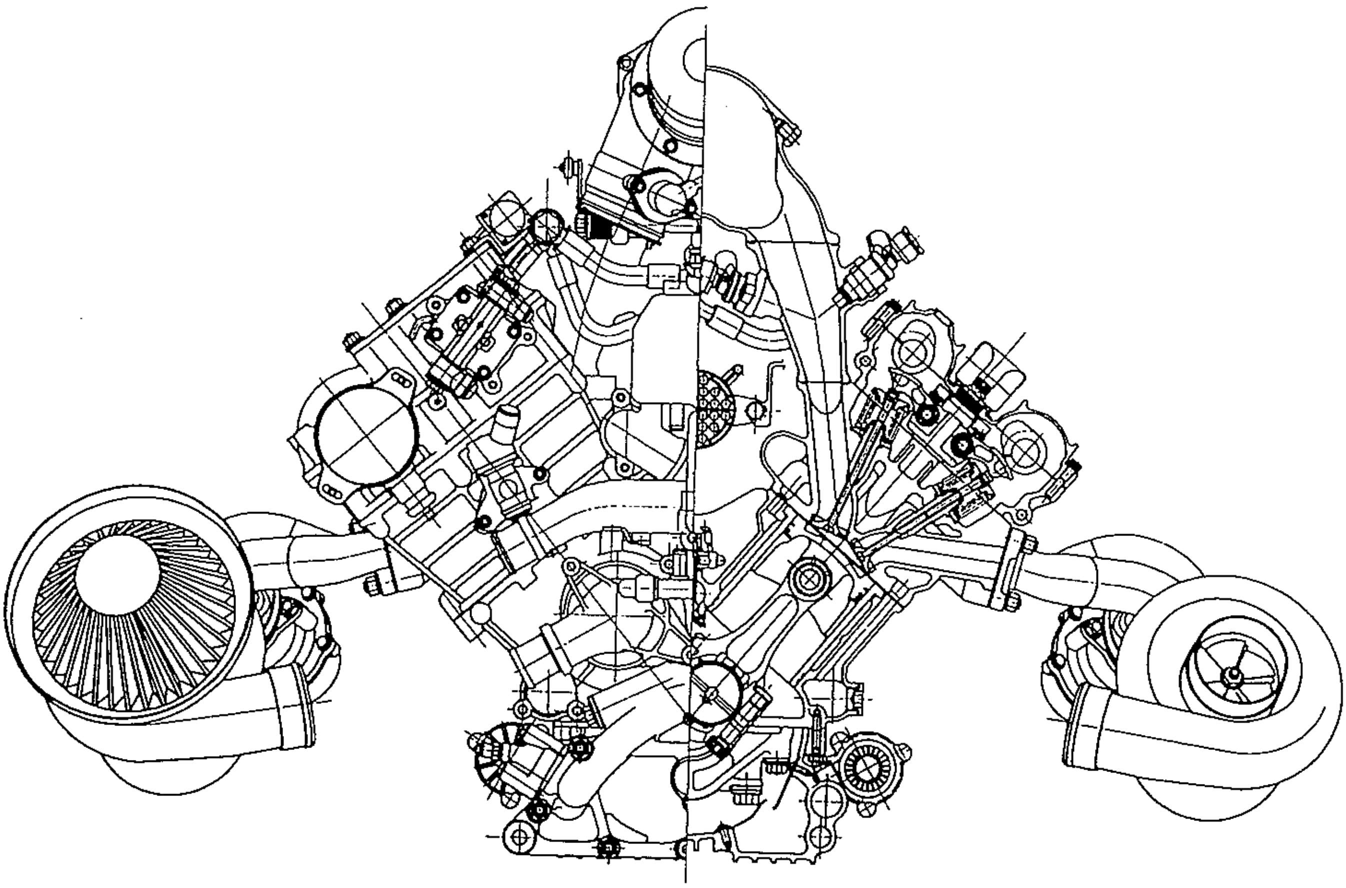

Fig. 1 RA168E cross section

A dry-sump lubrication system was adopted for the RA168E. Four independent scavenging pumps draw the lubricant through strainers at the corners of the oil-sump, and then feed it into an oil tank. With this layout, the scavenging pumps effectively circulate the lubricant, which tends to gather in the oil-sump due to longitudinal and lateral G-forces during acceleration, deceleration and cornering respectively.

Application of an appropriate cooling system is as important as that of the lubrication system for a racing engine. The RA168E has two water pumps which supply water from each side of the cylinder block. Water galleries are located on the outer walls of the cylinder block and the inner walls of the cylinder heads. Water travels from the cylinder block galleries, passing by the cylinder liners and through the cylinder heads, and flows into the cylinder head galleries. The lateral flow of water through the engine allows uniform thermal conditions throughout all cylinders.

We adopted a PGM-FI (Programmed Fuel Injection System), produced in-house, which sequencially injects fuel into six cylinders. Each cylinder is equipped with two injectors. Whether single- or double-injection is used depends on the amount of fuel required. In addition, the fuel line between the fuel pump and the injectors has a heat exchanger for pre-heating fuel by using water from the cooling system.

Dual turbo-chargers, code named "RX6D", are products of Ishikawajima-Harima Heavy Industries Co., Ltd. (IHI). They feature ceramic turbine wheel and ball bearings; drastically upgrading the transient properties by reducing the inertia moment and friction of revolving parts.

ENGINE PERFORMANCE

POWER OUTPUT AND TORQUE - For comparative data, we first present the performance of our RA167E engine. The RA167E was redesigned and upgraded to the prerequisites of the 4 bar boost restriction. Its maximum power was 742 kw (1,010 ps) at 12,000 rpm with a maximum torque of 664 Nm (67.7 kg-m). Power characteristics are shown in Figure 3; which were recorded under the following operating conditions: (1) Boost was 4 bar. A portion of the exhaust gas is emitted from the waste gate of the exhaust manihold. Turbine speed is controlled by adjusting the opening of the waste gate, thus controlling the boost at 4 bar. (2) Intake air temperature was 40°C. Hot intake air compressed by the turbo-chargers is cooled with air-cooled intercoolers. A portion of the air flowing into the intercoolers is expelled via by-pass valves. A temperature of 40°C is maintained through controlling the valve opening. (3) Equivalence ratio was 1.23. (4) Ignition timing was set at MBT or retarded at the knock limit. (5) Fuel with a toluene content of 84% was used. As far as this report is concerned, this fuel was generally used during bench dynamometer tests and race events (see Table 3).

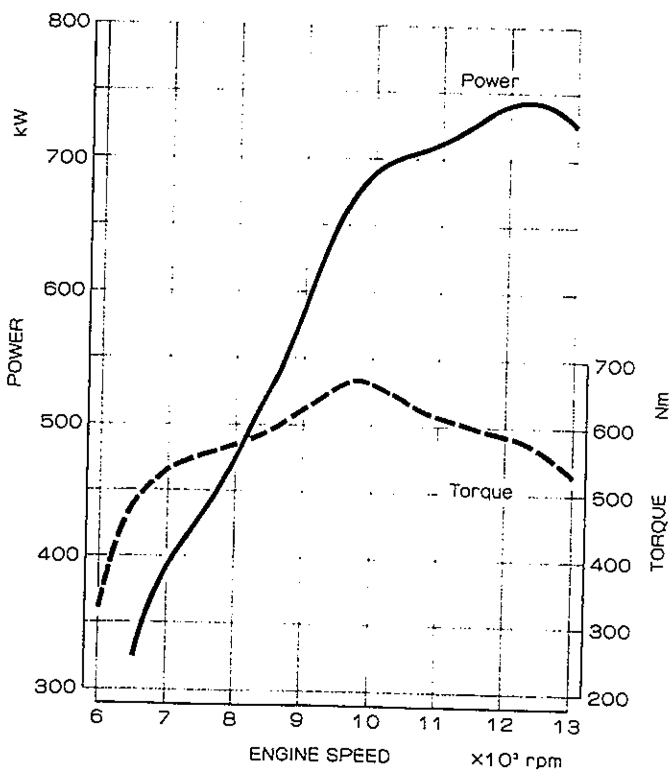

Fig. 3 RA167E power characteristics

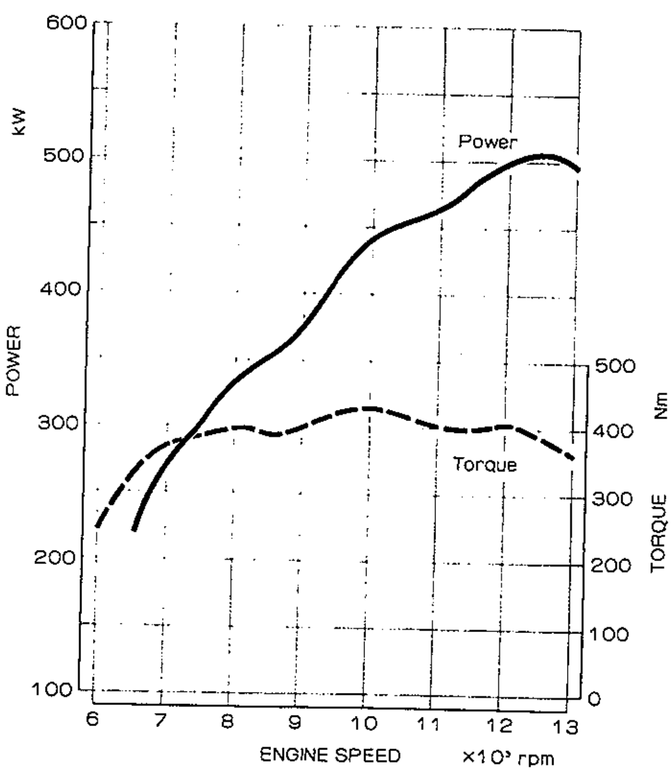

Fig. 4 RA168E power characteristics

Regulations were further revised in 1988 restricting the boost limit to 2.5 bar and fuel tank capacity to 150 liters. In order to generate higher power output, and improve fuel efficiency at the same time, the compression ratio of the RA168E was increased to 9.4, up from 7.4 of the RA167E. Figure 4 shows the power characteristics of the RA168E, which produces maximum power of 504 kw (685 ps) at 12,500 rpm and maximum toruque of 424 Nm (43.2 kg-m). Running conditions were set at;(1) boost of 2.5 bar, (2) intake air temperature of 40°C, (3) equivalence ratio of 1.15 and (4) ignition timing of MBT or spark retard at the knock limit. With a rich mixture of the equivalence ratio being 1.15 and a low intake air temperature of 40°C, the engine generates its maximum power. However, the fuel efficiency under these conditions was not satisfactory.

Under the conditions described above, it was sometimes obvious that it would be impossible to complete an entire race with such poor fuel efficiency. During a race specified running conditions were selected to improve fuel efficiency. Maximum power conditions were realized mainly in qualifying sessions, where sustained running was not necessary.

In terms of power per liter, the RA167E generated 495 kw/ℓ and the RA168E produces 336 kw/ℓ. As shown in Figure 5, taking boost pressure into consideration, comparison of the power characteristics shows that the RA168E is favorable, due to a higher compression ratio and lower friction.

CYLINDER PRESSURE DIAGRAM ANALYSIS - The RA168E generates 504 kw (685 ps) at 12,500 rpm. In the case of high-speed, high-boost engines, combustion often becomes unstable; in the worst case, leading to misfiring. This could pose a problem when attempting to manufacture engines which constantly generate high power. Concerning the RA168E, we increased the compression ratio, and modified the intake port configuration and the fuel injection system in order to stabilize combustion.

There have been few studies concerning cycle variations in a high-speed, high-boost engine. The cycle variation was analyzed by measuring cylinder pressure during maximum power generation.

A pressure transducer (product of Kistler) connected to a charge amplifier was used to measure the cylinder pressure. Samples were taken at each crank angle degree. A combustion analyzer (product of Ono Sokki) was adopted for the analysis.

Figure 6 is an averaged pressure diagram of 500 consecutive cylinder pressure measurements under the conditions of 12,500 rpm, boost of 2.5 bar, intake air temperature of 40°C, equivalence ratio of 1.15 and ignition timing of 35° B.T.D.C.

The maximum combustion pressure was 16.7 MPa, crank angle at which maximum pressure produced was 17° A.T.D.C., and indicated mean effective pressure (I.M.E.P.), was 3.8 MPa.

Figure 7 shows the frequency distribution of I.M.E.P. indicating that the standard deviation was 0.20 MPa against the average value of I.M.E.P. of 3.8 MPa, which revealed that fairly stable combustion was achieved.

FUEL EFFICIENCY

The main objectives when developing the RA168E to consume less fuel during a race were to improve fuel consumption characteristics and the development of high energy fuel (heavy specific weight fuel).

The throttle opening distribution during

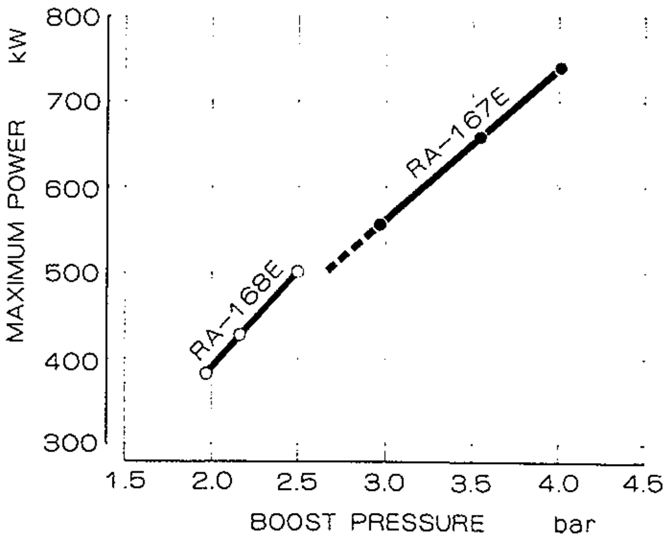

Fig. 5 Maximum power of RA168E and RA167E related to boost pressure

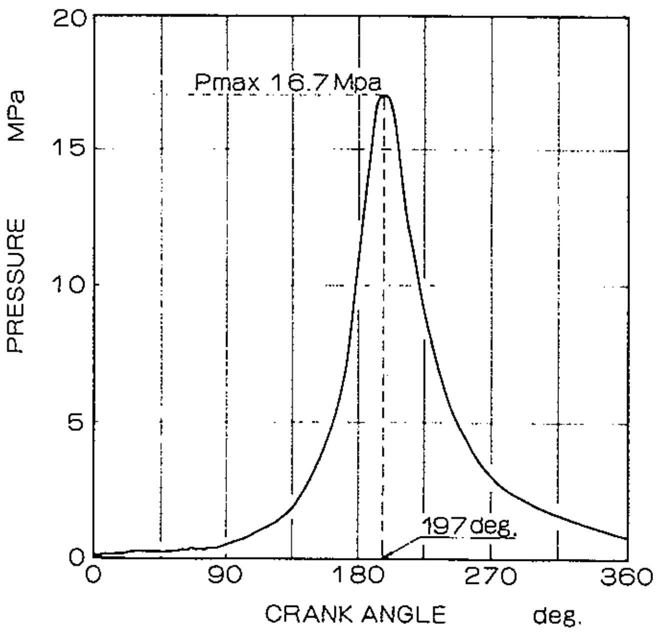

Fig. 6 Cylinder pressure diagram of RA168E under condition of maximum power generation

a race is shown in Figure 8 (1988 San Marino GP, with driver Alain Prost). Since it was revealed that full-throttle and closed-throttle dominate the throttle conditions during a race, we concentrated on improving the fuel consumption at full-throttle.

Figure 9 shows the RA168E performance when the major factors were set to minimize fuel consumption, indicating that minimum brake specific fuel consumption (B.S.F.C.) is 272 g/kwh (200 g/psh) at 12,000 rpm, and maximum power is 456 kw (620 ps) at 12,500 rpm. The operating factors of this condition are a boost of 2.5 bar, intake air temperature of 70°C, equivalence ratio of 1.02 and fuel temperature of 80°C.

Concerning circuits used for GP racing, straight stretches, number and type of corners and overall racing distances are not always the same. Thus, even if the engine's operating conditions remain the same, fuel consumption (km/ℓ) differs.

When favorable fuel consumption was observed at some circuit richer mixture setting and reduction in intake air temperature were possible. Though these conditions worsen B.S.F.C. of the engine itself, they can be selected as operating factors to generate higher power.

EFFECT OF INTAKE AIR TEMPERATURE - The intercooler system makes it possible to cool the intake air down to a temperature which is only 15°C higher than that of ambient air. This is achieved by closing the by-pass valves and directing all the air through intercoolers. As the intake air temperature increases, B.S.F.C. becomes better (see Fig. 10). After the intake air temperature exceeds 70°C, B.S.F.C. improvement tends to become saturated. Despite relatively bad volatility of the fuel used, a rise in the intake air temperature promotes vaporization. This B.S.F.C. improvement might be attributed to the well vaporized fuel.

However, as the temperature increases, there is a tendency to generate knocking, and the ignition timing must be retarded to avoid it. When the timing is retarded further from M.B.T., the results have an undesirable effect on B.S.F.C.

EFFECT OF BOOST PRESSURE - Higher boost has good effects on both power and B.S.F.C., as shown in Fig. 11. One of the prominent reasons is that, in proportion to the rise in boost, charging efficiency increases and provides higher indicated horse power while the engine's mechanical loss remains almost the same, resulting in an improvement of brake thermal efficiency.

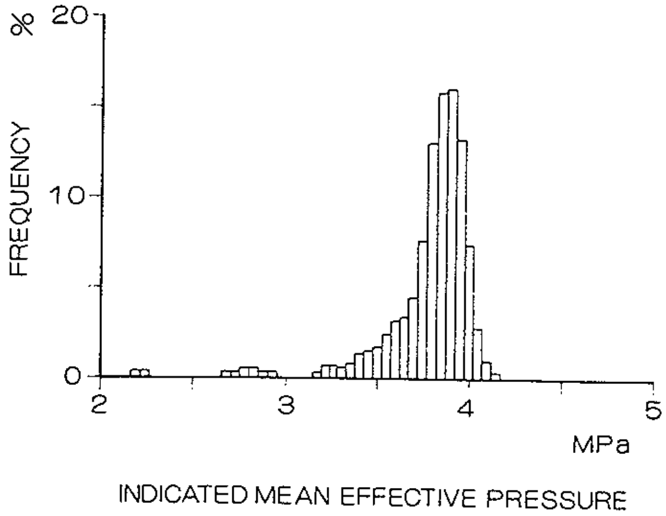

Fig. 7 I.M.E.P. frequency distribution

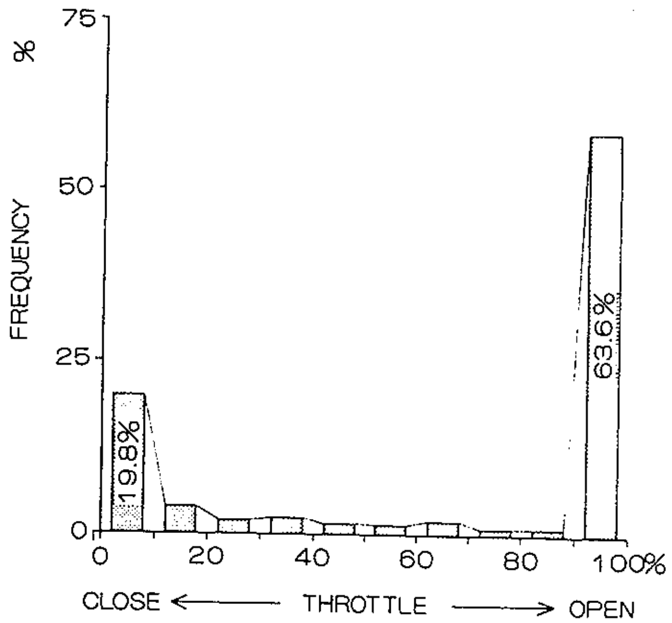

Fig. 8 Throttle opening distribution during a race event ('88 San Marino GP)

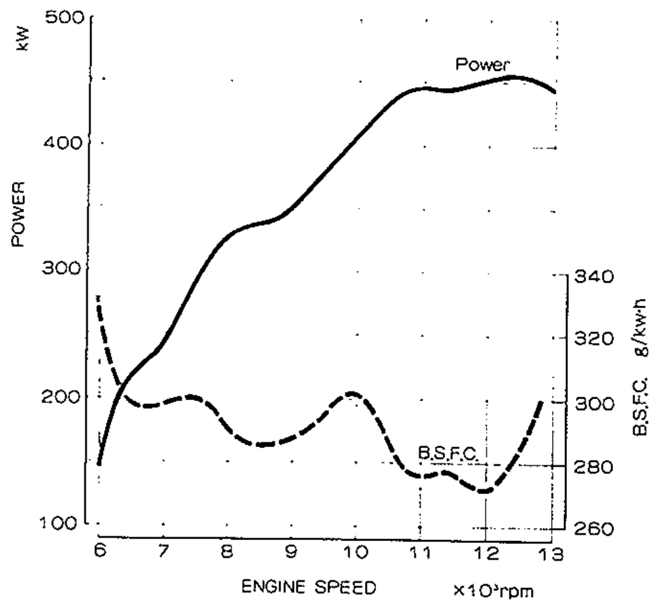

Fig. 9 RA168E performance at minimum fuel consumption

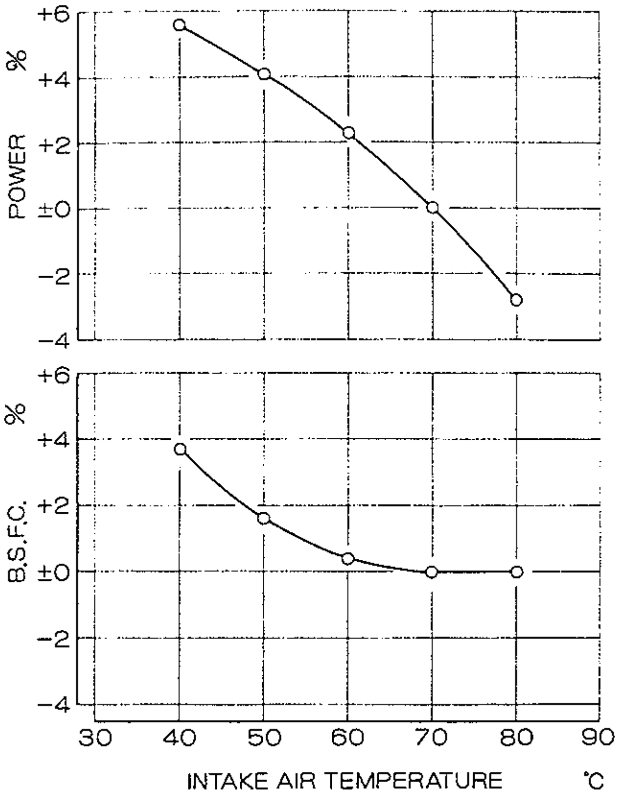

Fig. 10 Effect of intake air temperature on power and B.S.F.C.

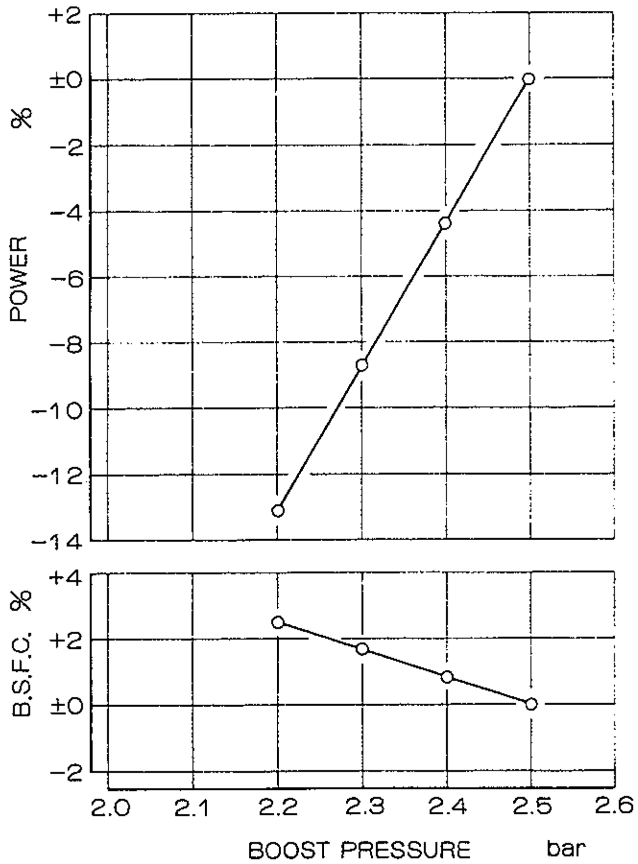

Fig. 11 Effect of boost pressure on power and B.S.F.C.

EFFECT OF AIR FUEL RATIO - Regarding air-fuel ratio, peak power is reached at an equivalence ratio of 1.15, and power gradually decreases as the ratio falls below this figure. The leaner the mixture becomes, the better the B.S.F.C., as shown in Fig. 12. However, with a ratio lower than 1.02, unsatisfactory transient response may appear, thus making the engine become insufficient for racing performance.

EFFECT OF PRE-HEATING FUEL - B.S.F.C. is improved through promotion of vaporization with higher intake air temperature. The same effect can be achieved using pre-heated fuel (i.e., fuel heated before injection). As the distillation characteristics shown in Table 3 reveal, the racing fuel for the RA168E is not easily vaporized at the ambient temperature because it does not contain low boiling-point ingredients. For this reason, the RA168E was equipped with a heat exchanger capable of pre-heating fuel using the water from the cooling system. After being fed by the fuel pump and then heated, it is distributed to the injectors. It is possible to heat the fuel to a temperature 15°C below that of the water.

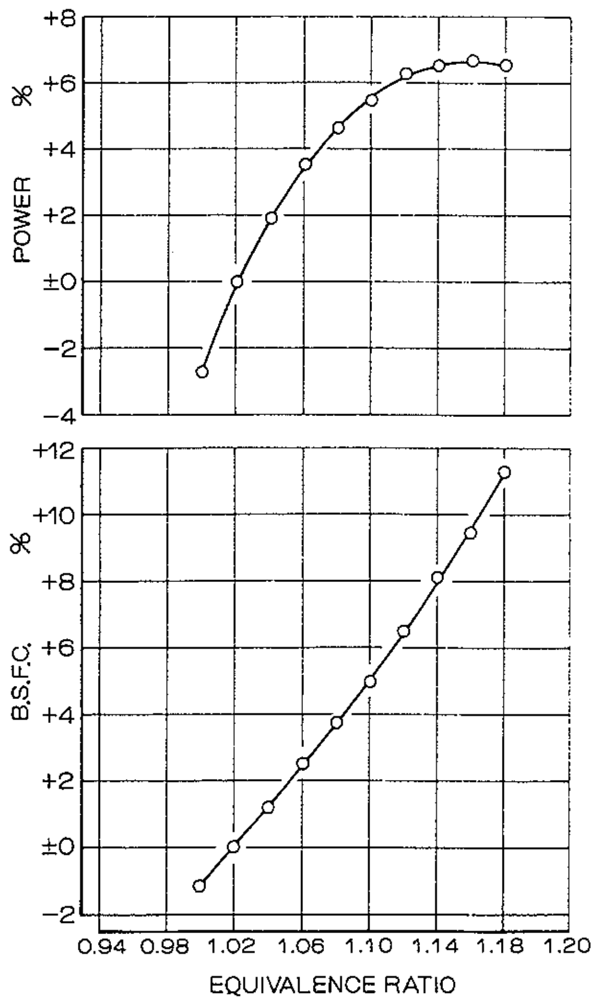

Fig. 12 Effect of equivalence ratio on power and B.S.F.C.

The amount of water which flows into the exchanger is controlled by a solenoid valve which maintains the desired fuel temperature.

Figure 13 shows the effect of fuel temperatures on the fuel efficiency. It is acknowledged that, although the effect is not significant, the fuel consumption is reduced as the fuel temperature rises.

A combination of operating factors to achieve the best B.S.F.C., while maintaining satisfactory performance, is an intake air temperature of 70°C, a boost of 2.5 bar, an equivalence ratio of 1.02 and fuel temperature of 80°C. With this combination, B.S.F.C. is 272 g/kwh (200 g/psh) at 12,000 rpm.

EFFECT OF TOLUENE CONTENT - The octane number of fuel for Formula One racing is limited to a maximum of RON 102. Adopting a higher compression ratio is expected to improve both power and B.S.F.C. However, at the same time, the possibility of knocking becomes higher. Therefore, the knocking properties of fuels determine maximum compression ratio. It sometimes appears that differences in fuel ingredients effect knocking properties, even though the RON of the fuels is the same. The development of a fuel with good knocking properties under high speed and boost conditions is essential for adopting a high compression ratio.

The tank capacity regulation limits fuel amount to 150 liters, and refueling during a race event is forbidden. In order to obtain a higher level of fuel energy for a race, a fuel largely calorific in capacity is needed (i.e., a dense fuel). Comparative evaluation of various fuels revealed that a fuel with high toluene content is most favorable to meet the requirements. Knocking properties and effects on B.S.F.C. of the test fuels (see Table 3) are shown in Figure 14.

Toluene content ratios were 30%, 60% and 84% for each test fuel. Appropriate amounts of normal heptane and isooctane were respectively mixed with toluene to achieve a RON of 102. As toluene has a heavier density when compared to paraffine fuels, a fuel containing much toluene has a larger calorific value in capacity (Cal/cc). Whether knocking occurred or not was observed through analyzing the output signal of a charge amplifier connected to a piezo-electric pressure transducer fixed in the place of the spark plug washer. Observations were conducted under the engine operating conditions of ; 12,000 rpm, 2.5 bar, intake air temperature of 70°C and an equivalence ratio of 1.15. The knock-limit ignition timing advances as the ratio of toluene increases in the fuel ingredients, resulting in better B.S.F.C. In addition, since the test fuels differ in density, "brake specific volumetric fuel consumption (B.S.V.F.C.)" (cc/kwh) was considered, and studies revealed that a fuel containing a higher ratio of toluene in the fuel ingredients proved most effective.

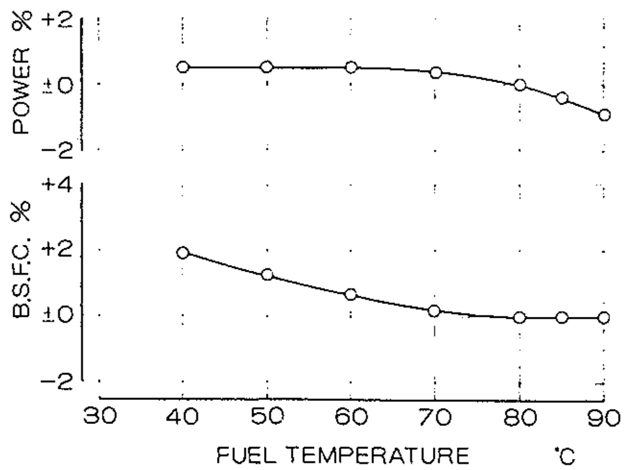

Fig. 13 Effect of fuel temperature on power and B.S.F.C.

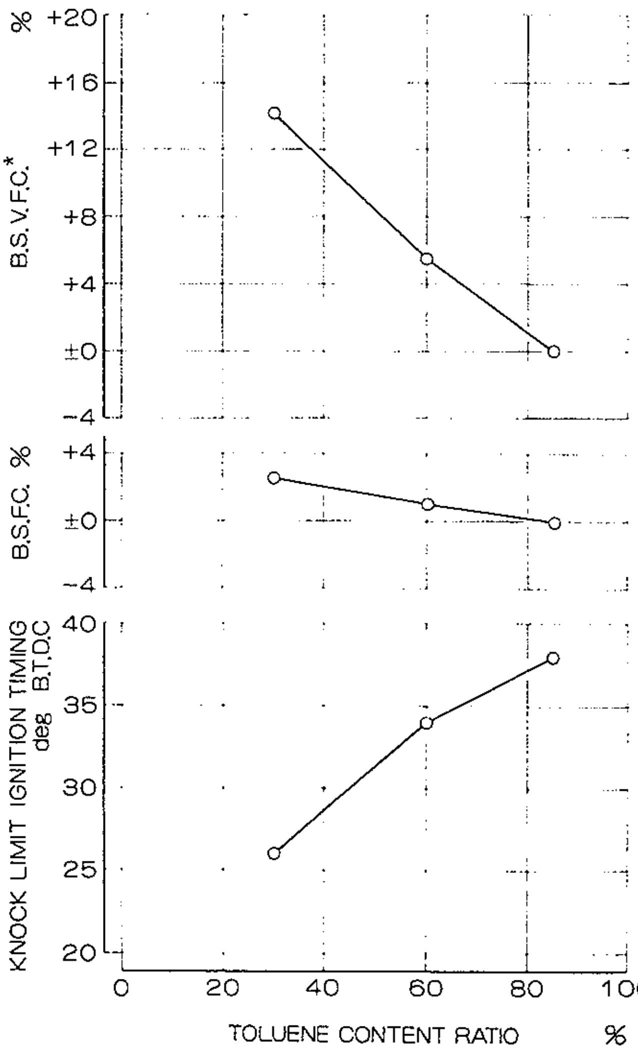

Fig. 14 Effect of toluene content ratio on knock limit ignition timing and fuel consumption

* B.S.V.F.C "brake specific volumetric fuel consumption" (cc/kwh)

Table 3 Test fuel specifications

		A	B	RACING FUEL *
Fuel ingredient	Toluene (%)	30	60	84
	n-Heptane (%)	4	9.5	16
	Isooctane (%)	66	30.5	0
Research Octane Number		101.6	101.9	101.8
Motor Octane Number		94.2	91.2	90.0
Density (at 15°C)		0.747	0.799	0.840
Net Calorific Value (Kcal/Kg)		10300	10015	9817
Stoichiometric Ratio		14.5	14.0	13.7
Reid Vapor Pressure (Kg/cm²)		0.154	0.141	0.120
Initial Boiling Point (°C)		96.0	98.5	100.0
10% (°C)		97.5	100.5	105.0
50% (°C)		98.5	102.0	106.0
90% (°C)		100.0	105.0	108.0
End Point (°C)		123.5	108.0	116.0

*This fuel is jointly developed by ELF FRANCE and HONDA for the special racing purpose.

SUMMARY

Honda has developed a Formula One turbo-charged V-6 1.5-liter engine, which was very successful in the 1988 Grand Prix racing season. Through the research and development on this engine, the results can be summarized as follows:

(1) Optimizing operating factors to produce the best results possible under the 2.5 bar restriction, the engine successfully produced a maximum power output of 504 kw (685 ps), which is equivalent to 336 kw/ℓ (457 ps/ℓ).

(2) A combination of operating factors was studied to minimize fuel consumption and the minimum brake specific fuel consumption was found to be 272 g/kwh (200 g/Psh) when the engine generated 456 kw (620 Ps).

(3) A high toluene content in fuel had a good effect on knocking properties and allowed advanced ignition timing, which resulted in improvement of brake specific fuel consumption. Toluene also had a good effect on improvement of brake volumetric specific fuel consumption because of the heaviness of its density.

ACKNOWLEDGEMENT

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REFERENCES

1. Chritian Hoefer, "Renault Gordini EF 1-erster Formel-1-Motor mit Abgasturbo-aufladung", Motortechnische Zeitschrift, 45(1984)10, pp. 415-418.

2. Prof.Dr.-lng.W.Rixmann, Esslingen, "Der BMW-Formel-2-Rennmotor M12/6", Automobil-technische Zeitschrift, 78(1976)4, pp. 147-149.

3. Dipl.-lng. Christian Hoefer, Fellbach, "Die Ferrari-Grand-Prix-Modelle", Automobil-technische, Zeitschrifrt, 78(1976)7/8, pp. 321-325.

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Turbocharging Engines for Racing and Passenger Cars

Hans Mezger
Dr. Ing. H.C. F. Porsche
Aktiengesellschaft

West Coast Meeting
Town & Country, San Diego
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Turbocharging Engines for Racing and Passenger Cars

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IN 1976, FOR THE FIRST TIME in the history of Le Mans, a car powered by a turbocharged engine won the 24 hour race. The winner, shown in fig. 1, was the 700 kg Porsche type 936 race car. Its engine was based on the aircooled six cylinder engine of the present production type 911. The crankcase, crankshaft, camshaft housing and the camshaft drive were identical parts from 911 series. The 2142 cm^3 (130 cu.in.) 1976 engine version delivered 382 kW (520 hp) at 8000 rpm, the boost pressure being 1.2 bar (17 psi).

At no other race event is the engine subject to such severe conditions as at Le Mans. Thus the 1976 Le Mans success of Porsche's 936 has not been just another race victory but represents a high-technology level in terms of function and durability, which will make exhaust gas turbocharging applicable to passenger car engines. The absolute necessity of improving the acceleration behaviour of turbocharged engines for racing use led to control systems, which are today's standard in turbocharging technology.

HISTORICAL DATA

The principle of turbocharging is known for more than 70 years. In 1905 a patent was given to Swiss Engineer Alfred Büchi for a four-stroke engine, equipped with a compressor at the intake side and a turbine at the exhaust side.

Fig. 2 is a drawing of the patent specification. As shown, engine, compressor and turbine are coupled by one shaft. Ten years later, in 1905, Büchi received another patent, characterizing turbocharging as it is used today: the compressor is driven by the exhaust gas driven turbine, both separated from the engine. First commercial application of turbocharging was made for ships in the late twenties, for railway engines in

ABSTRACT

The principle of exhaust gas turbocharging is known for more than 70 years. Only since a few years turbocharging is applied successfully to gasoline engines respectively to engines for passenger cars.

Mainly a poor throttle response and an unsatisfactory acceleration characteristic prevented turbocharging from general use.

The paper deals with the application of turbocharging for racing and passenger car engines. It is reported on the problems, improvements, and Porsche's experiences.

The future chances of turbocharging with regard to exhaust emissions and fuel economy are discussed.

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1933. During world war II turbocharged engines were used for aircraft. After the war turbocharging was applied to Diesel engines for trucks. In 1952 a race car equipped with a turbocharged Diesel engine was entered in the Indianapolis 500. In the middle of the sixties turbocharging was used for USAC respectively Indianapolis racing car engines, and also for light aircraft engines. In 1964 and 1965 a small number of turbocharged six cylinder engines and V 8 engines for passenger cars were produced in the United States.

No controls were in use until then. A quick engine response was either unnecessary or at least extremely unsatisfactory. Thus turbocharging was not in consideration for general use, since the passenger car requires an engine with quick throttle response.

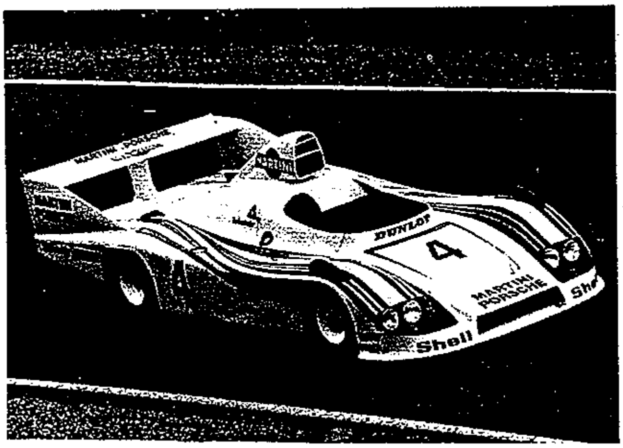


Fig. 1 - Porsche racing car type 936

PORSCHE'S APPROACH TO TURBOCHARGING

Porsche started turbocharging development in 1971, after FIA (Fédération Internationale de l' Automobile) had changed regulations for 1972 World Championship of Car Makes, outlawing the successful type 917 from racing thereby. Porsche then decided to participate in the popular North-American CanAm series from 1972. There was the choice between two power plants: an existing 16-cylinder engine and the reliable 12-cylinder 917 engine, both aircooled and normally aspirated. Finally, a decision was made in favour of a still to be developed turbocharged version of the 917 engine.

Up to that time, it was generally believed that an engine featuring exhaust gas turbocharging could be employed only on high speed tracks with little curves, such as the Indianapolis oval, for instance. Certainly, a throttle lag would make a race car uncompetitive on twisting race circuits, even if the expected performance should be reached. Thus, making the turbocharged engine responsive has been the point of main effort during the development of the 917 CanAm car.

Porsche's first turbocharged engine, the aircooled 12-cylinder type 917, is shown in fig. 3. In 1972, there was a 5 litre (305 cu.in.) engine delivering 735 kW (1000 hp). A 5.4 litre (328

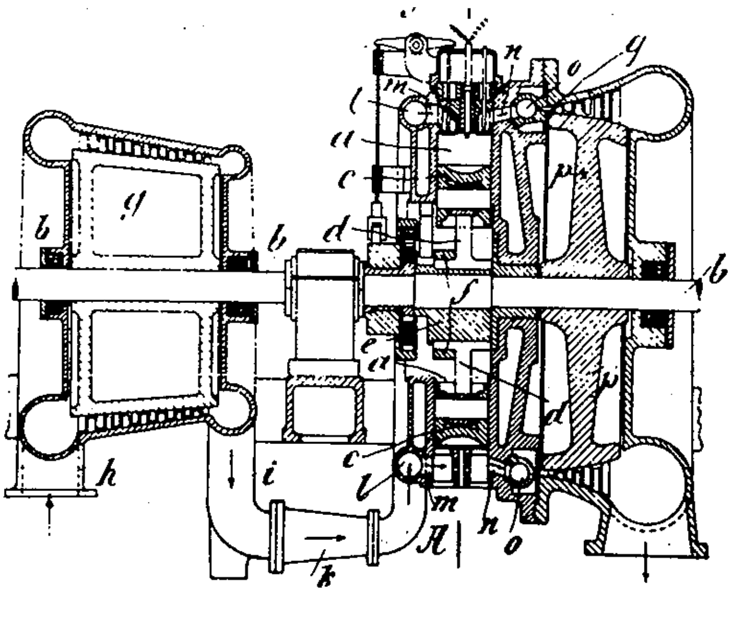


Fig. 2 - Büchi engine (patent specification 1905)

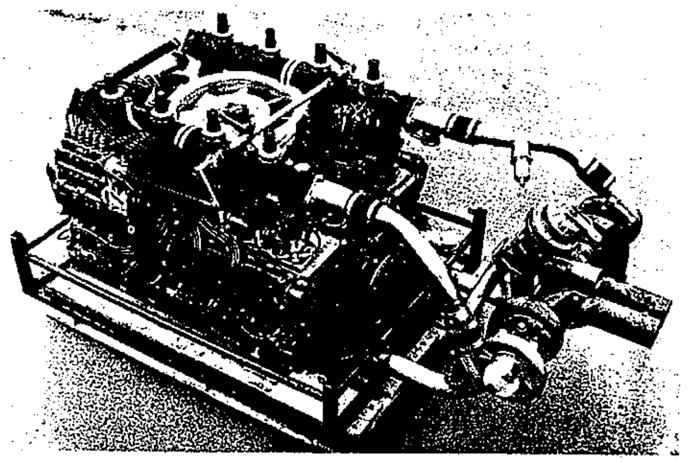


Fig. 3 - Turbocharged 12-cyl engine - Porsche type 917 (1100 hp)

cu.in.) version, developed for the 1973 CanAm season, had an output of 810 kW (1100 hp) at 7800 rpm and 1.4 bar (20 psi) boost pressure. The maximum torque was 1100 Nm (812 ft.lb.), total weight, including turbochargers, 285 kg.

The engine is equipped with two turbochargers. Each cylinder bank of the 12-cylinder flat engine, operating like a 6-cylinder with equal firing, feeds one turbocharger with exhaust gases, and is supplied by same turbocharger with boost. The boost air is led from the compressors via tubes to the two intake manifolds. Twelve throttles are engaged, located between the manifolds and the cylinder heads. The front ends of the manifold pipes are connected by a tube in order to equalize possible differences in boost pressure.

Although designed as a normally aspirated engine, delivering 463 kW (630 hp) in its 5 litre version, only a few changes to the original engine were required.

By exchanging two bevel gears in the cooling fan drive the fan speed was increased from 7400 rpm at nominal engine speed up to 9000 rpm in the turbocharged version, thus raising cooling air flow from 2400 litres up to 3100 litres per second. The normally aspirated engine is supplied with 3.8 litres of cooling air per second for each horse power, the turbocharged engine with 2.8 litres only.

The compression ratio, 10.5:1 in the normally aspirated version, had to be reduced to 6.5:1 in the turbocharged version.

In the normally aspirated engine the spark advance is controlled by the engine speed, and limited to 27 degrees before TDC. In the turbocharged version spark timing is fixed at 20 degrees before TDC.

The original 12.1 mm valve lift and the valve overlap was reduced by using the original 10.5 mm exhaust camshaft for both, intake and exhaust.

Heat problems with the exhaust valve guides were overcome by chrome plating the valve stems, increasing the valve running clearance, shortening the valve guides, and by improving the oil supply.

The turbochargers used for the 917 engine are provided with ball bearings. At the rated engine output they achieve a maximum speed of 90.000 rpm, delivering 0.55 kg boost air per second each at a temperature of 150 to 160 degrees centigrade. At full throttle the temperature of the exhaust gas is 1000 to 1100 degrees centigrade.

The plunger type injection pump, using engine speed and throttle position for fuel metering, received an additional control dependant on boost pressure.

TURBOCHARGER CONTROLS - In contrast to tuning and adjusting the naturally aspirated engine which can be carried out on the dynamometer, the optimum matching of turbocharging to an engine requires intensive road testing. The results of first 917 tests at the Weissach test circuit and the Hockenheim race track were not satisfactory. The drivers complained that the engine "would not accept the gas". Often a driver believed to have done a very fast lap only to be disappointed when learning the real lap time. Various turbine-compressor combinations were tried, and many different control devices were developed, tested and modified. Step by step the throttle response was improved. Finally in its first race the 917 CanAm car proved to be a car with excellent driveability. The performance of the 917 car in the CanAm series has shown that turbocharging is competitive. Fig. 4 shows the control diagram of the engine. Basic item of the control system is the boost pressure controlling wastegate. Located in the exhaust manifold upstream the turbine it will limit the manifold pressure to a required level. The wastegate valve is operated by a diaphragm. It will open when the control pressure has reached its critical level, thus bypassing the exhaust gases not required for maintaining the boost around the turbine. A standard wastegate is shown in fig. 5. In the 917 wastegate a screw was used in order to adjust the boost pressure limit by varying the spring load. Race drivers called it the "power screw". Very often it was used in order to gain higher power output. Increasing the boost pressure by 0.1 bar, the 917 engine performance went up by approximately 50 hp. In the 1973 CanAm 917 car the driver was enabled to control boost and performance during the race, fig. 4. The inlet of a manually operated pressure control valve, installed in the cockpit, is connected to the intake manifold (boost pressure line). The outlet is connected with the topside of the wastegate diaphragm housing (control pressure line), normally opened to the atmosphere. If the control pressure is zero, the boost pressure limit is fixed by the spring loading. Adjusting the control pressure to 0.1 bar, for instance, the boost pressure will be increased by 0.1 bar, too. This device has proved to be very helpful for daily use on the engine dynamometer.

There is another control used. As shown in fig. 4, a pressure relief butterfly valve is arranged in the intake manifold. Connected with the throttle valves by a linkage it will be opened when the throttles are closed completely, thus dropping the intake manifold pressure immediately. It helps to maintain the compressor speed during braking respectively cornering. The dropping of the manifold pressure will also maintain the braking behaviour of the engine, since no boost will pass the throttles. In addition, shocks and vibrations in the pressure line possibly causing a damage to the compressor wheel will be avoided.

The turbocharged 917 engine has been the most powerful road vehicle engine. It enabled the 917 car to accelerate from 0 to 100 km/h in 2.3 seconds.

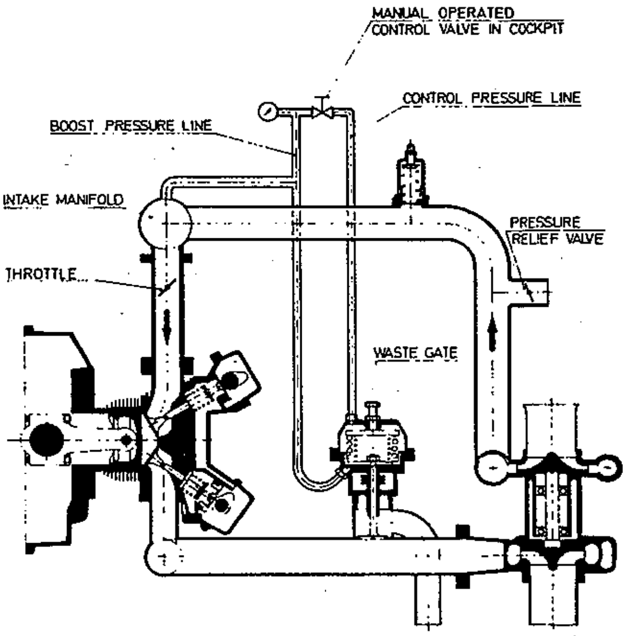


Fig. 4 - Turbocharging controls - racing engine type 917

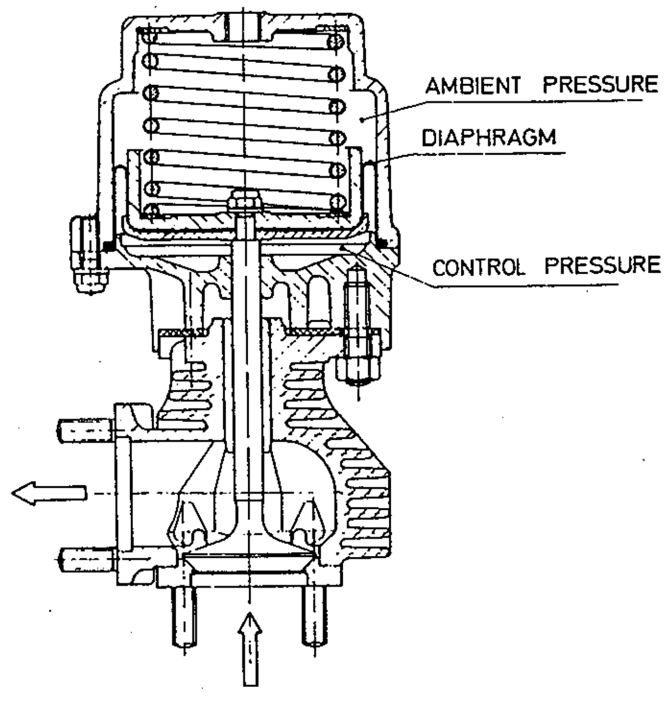


Fig. 5 - Boost pressure control valve (wastegate)

In 1972, the 917 won six out of nine races, and, in 1973, eight of eight races for the CanAm Championship. Then again, with the new CanAm regulations, the 917 was forced "out of the running".

A NEW ATTEMPT - From 1973 on, Porsche developed a turbocharged version of the production type 911 6-cylinder engine with a view to new regulations for the world championship of car makes, admitting production based race cars, only.

All experiences with regard to controls etc. gained from the 917 were transferred to the 6-cylinder engine. Already in 1974, a specially prepared 911 experimental car participated in prototype race events. Its turbocharged 2.1 litre engine delivered 345 kW (470 hp).

Experiences from turbocharging have shown that the performance of a turbocharged engine is limited rather by overheating the pistons, valves, cylinder head etc. than by mechanical difficulties. By providing the 6-cylinder engine with an air-to-air intercooler the engine components temperatures decreased, and the boost air temperatures were reduced from 150 degrees to approximately 75 degrees centigrade. Consequently, the power output could be raised up to 375 kW (510 hp).

The new world championship regulations, originally announced for 1975,did not become effective until 1976. Porsche entered a so-called production race car type 935, both, car and engine based on the production type 911. The turbocharged 2857 cm3 (174 cu.in.) engine delivered 441 kW (600 hp).

Another car, the type 936, participated in the sports car world championship events. It was powered by a turbocharged 2.1 litre engine, also based on production, as mentioned above. In 1976, both world championships and the 24 hour race of Le Mans were won. In 1977, both engines were equipped with a two-turbocharger system (fig. 6). An air-to-air intercooler was used in the type 936, and an air-to-water intercooler in the type 935.

The 936 sports car, in 1977 only entered in the Le Mans race won again. The 1977 world championship for car makes was won by the 935 production race car once more (fig. 7).

High performance engines, both turbocharged and naturally aspirated versions, normally are adjusted to high power output not to low fuel consumption. However, experiences have shown, that fuel economy of turbocharged race engines is superior to the fuel economy of naturally aspirated engines. The specific fuel consumption of a turbocharged 600 hp race engine, measured on the dyna-

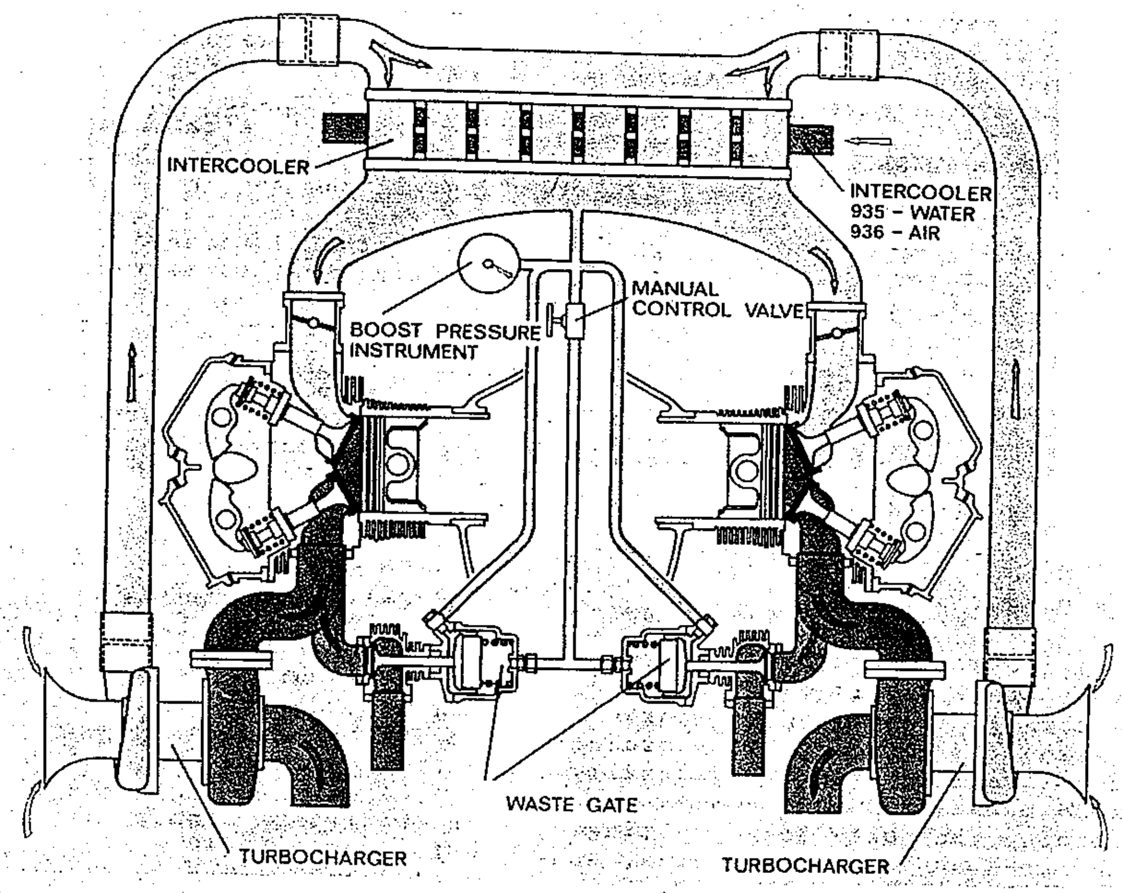


Fig. 6 – Turbocharging diagram – Porsche type 935 and type 936

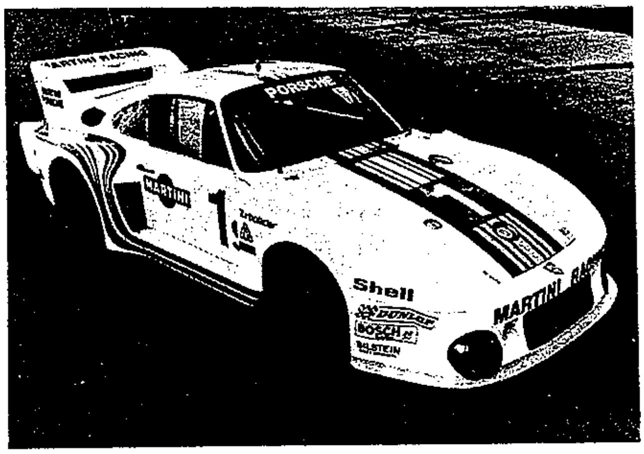


Fig. 7 - Porsche type 935 production race car

mometer, ranges from 292 g/kWh (215 g per hp and hour) to 320 g/kWh (235 g per hp and hour). Related to a race engine these values prove to be very low.

All Porsche factory cars entered into road racing since 1972, were equipped with turbocharged engines. A very successful participation has increased the confidence in turbocharging with regard to overall performance, driveability and reliability. It was inevitable that the idea arised of using turbocharging in Porsche production cars.

THE APPLICATION OF TURBOCHARGING IN SERIES

Early in 1975, Porsche started production of the 911 turbo, a high performance version of the well known 911 (fig. 8). The turbocharged aircooled 6-cylinder flat engine has the following data:

Displacement	2994 cm^3(183 cu.in.)
Bore	95 mm
Stroke	70.4 mm
Performance/Speed	191 kW (260 hp)/ 5500 rpm
Maximum Torque	343 Nm (253 ft.lb.)
Compression Ratio	6.5:1
Boost Pressure	0.8 bar (11.4 psi)

The development of the turbocharged engine, started in 1973, is characterized by transferring the knowledge on turbocharging gained from racing, to a production engine, uncompromisingly. Naturally, new problems arised and had to be solved, such as keeping the costs down, meeting current emission and noise standards, and finding space for additional components in the already filled up engine compartment.

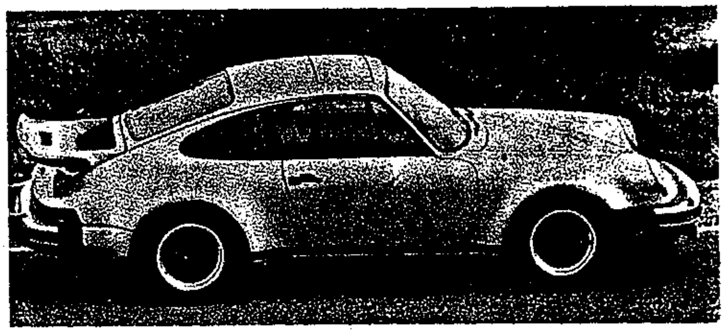
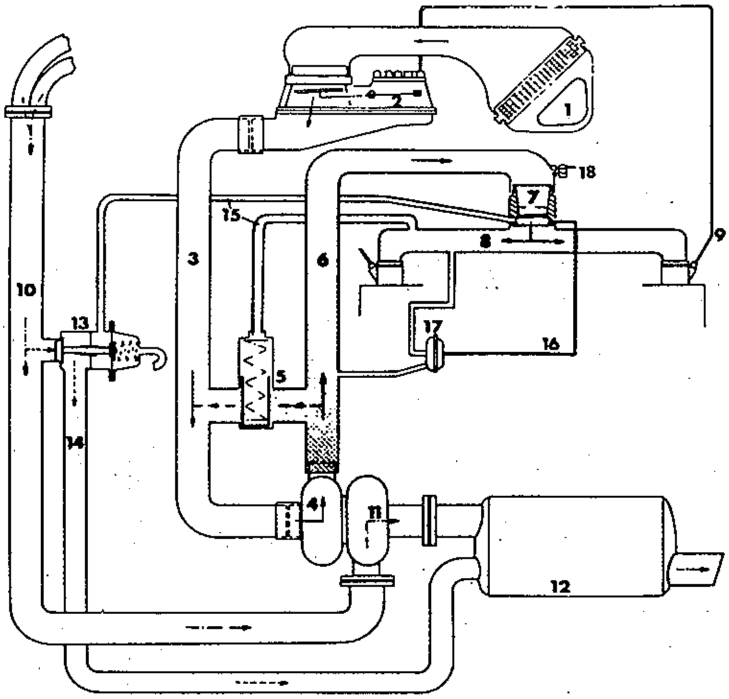


Fig. 8 - Porsche type 911 turbo 3 L

A characteristic mark of the engine is the combination of turbocharging with a fuel injection system named "K-Jetronic", which controls the intake air flow directly. An air sensor plate moved by the intake air flow actuates a fuel metering plunger. As shown in the diagram, fig. 9, the K-Jetronic unit (2 - fig. 9) and thus the air sensor is located conveniently upstream of the compressor (4-fig. 9). By that means the uncompressed intake air flow is measured, making an additional and possibly inaccurate boost pressure dependant fuel control unnecessary.

The intake air is aspirated by the compressor (4 - fig. 9) through the air cleaner (1 - fig. 9) and the air sensor (2 - fig. 9). After compressing, the air is led through the throttle valve (7 - fig. 9) and the intake manifold (8 - fig. 9) to the engine. At partial load and in that low-speed full load range in which the boost pressure has not yet reached its limit, the total exhaust gas volume will pass the turbine (11 - fig. 9).

At full throttle operation the maximum boost pressure of 0.8 bar (11.4 psi) is reached at an engine speed of 3000 rpm. The wastegate (13 - fig. 9) opens and part of the exhaust gas volume (14 - fig. 9) not required for maintaining the manifold pressure will flow directly to the muffler, bypassing the turbine. The diaphragm housing of the wastegate is connected with the throttle housing by a control line (15 - fig. 9). The wastegate valve is actuated by the manifold pressure. The wastegate used in the 911 engine is shown in fig. 5. If any fault should prevent the wastegate from opening, an overboost safety switch (18 - fig. 9), adjusted to 1.1 to 1.4 bar, will cut off the fuel pumps, thus avoiding an uncontrolled increase of boost and power.



1 = Air Cleaner
2 = Injection Unit
3 = Low Pressure Air Hose
4 = Turbo Charger (Compressor)
5 = Boost Pressure Relief Valve
6 = High Pressure Air Hose
7 = Throttle Valve Housing
8 = Manifold
9 = Fuel Line
10 = Exhaust Gas Collecting Tube
11 = Turbo Charger (Turbine)
12 = Muffler
13 = Wastegate (Boost Pressure Control Valve)
14 = Bypass Tube
15 = Control Lines
16 = Control Line to Aux. Air Valve for Overrun
17 = Vacuum Control Valve for Overrun
18 = Safety Switch for Boost Pressure

Fig. 9 – Turbocharging diagram – Porsche 911 turbo

In the Porsche racing engines a mechanically actuated pressure relief valve was in use, in order to drop the manifold pressure immediately, when the throttle is closed (fig. 4). The 911 engine is equipped with the same control device (5 - fig. 9). However, it is operated by the intake manifold pressure. In overrun condition, when the throttle is closed, it is opened by the manifold vacuum. By that means, as mentioned above, the braking performance and throttle response is improved, and damaging the compressor wheel is avoided. The boost air, passing the pressure relief valve, must not be released to the atmosphere. Since it is already registered by the K-Jetronic air sensor, it has to remain in the closed cycle, flowing back to the compressor intake side.

Fig. 10 is a cross section of the turbocharged 911 engine. On the left side the position of the wastegate is shown.

Some characteristic curves of the engine are shown in fig. 11. The characteristic of the boost pressure (p_2) is typical of a wastegate controlled turbocharged engine. As shown, the boost operated wastegate valve will open at an engine speed of 3000 rpm, then keeping the pressure constantly. At the nominal speed of 5500 rpm the boost temperature (t_2) is about 125 degrees centigrade. The exhaust manifold pressure (p_3), measured upstream the turbine, is approximately 1.1 bar at nominal speed. The charger delivers 0.24 kg/s boost (G_1) at maximum power.

During development of an emission control system for the US version of the 911 turbo, it was observed that secondary air injection does effect boost and performance characteristic positively. In fig. 12 is shown that there is a distinct increase of boost pressure, torque and power output in the low speed range up to 3000 rpm. By injecting air into the exhaust manifold the temperature of the exhaust gas went up by 80 to 90 degrees centigrade.

The diagram fig. 13 is the result of an experimental test on the engine dynamometer, imitating an acceleration process. From idling condition the throttle was opened suddenly, and the progress of the manifold pressure and the turbocharger speed was recorded.

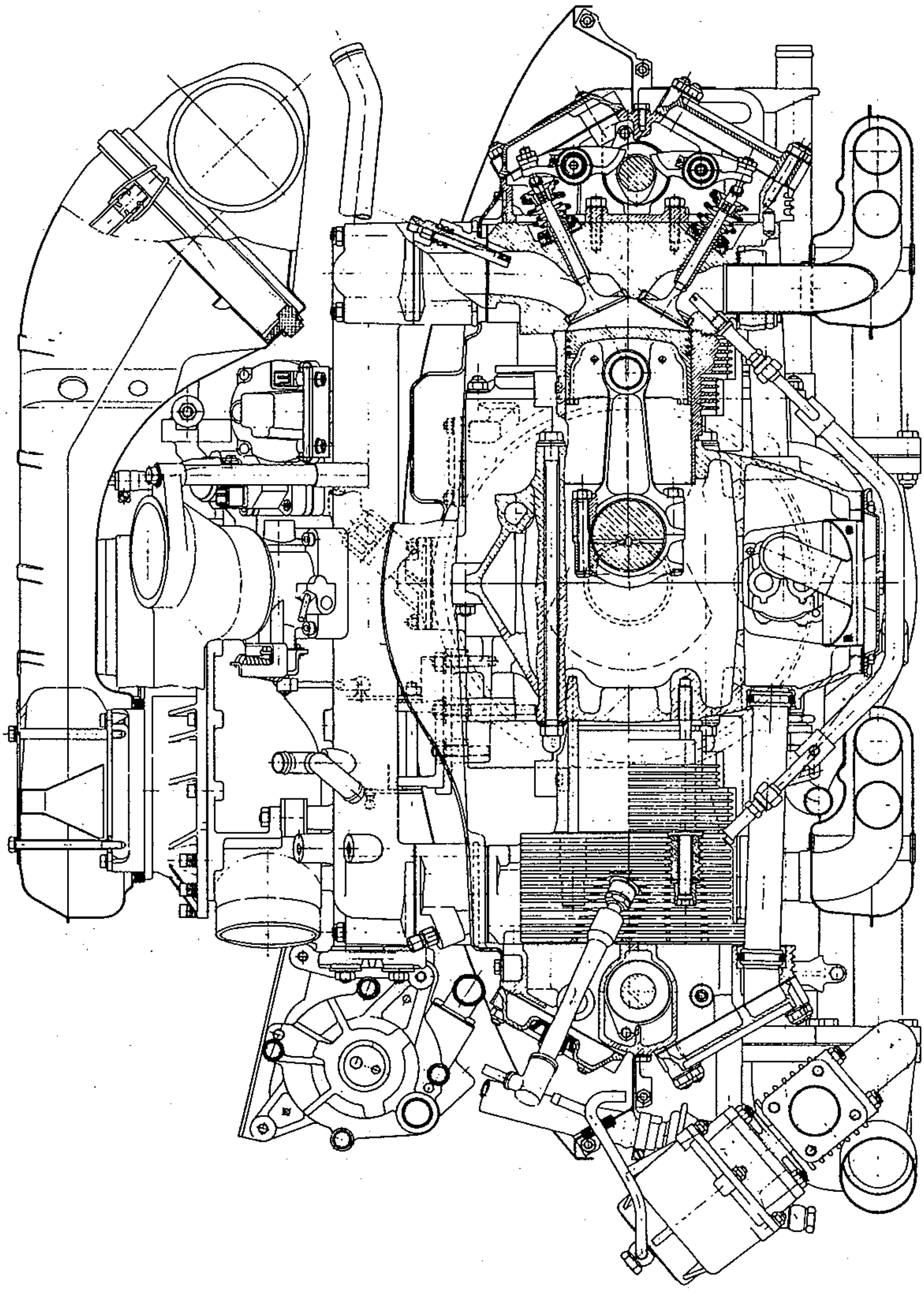


Fig. 10 - Cross-section - engine 911 turbo

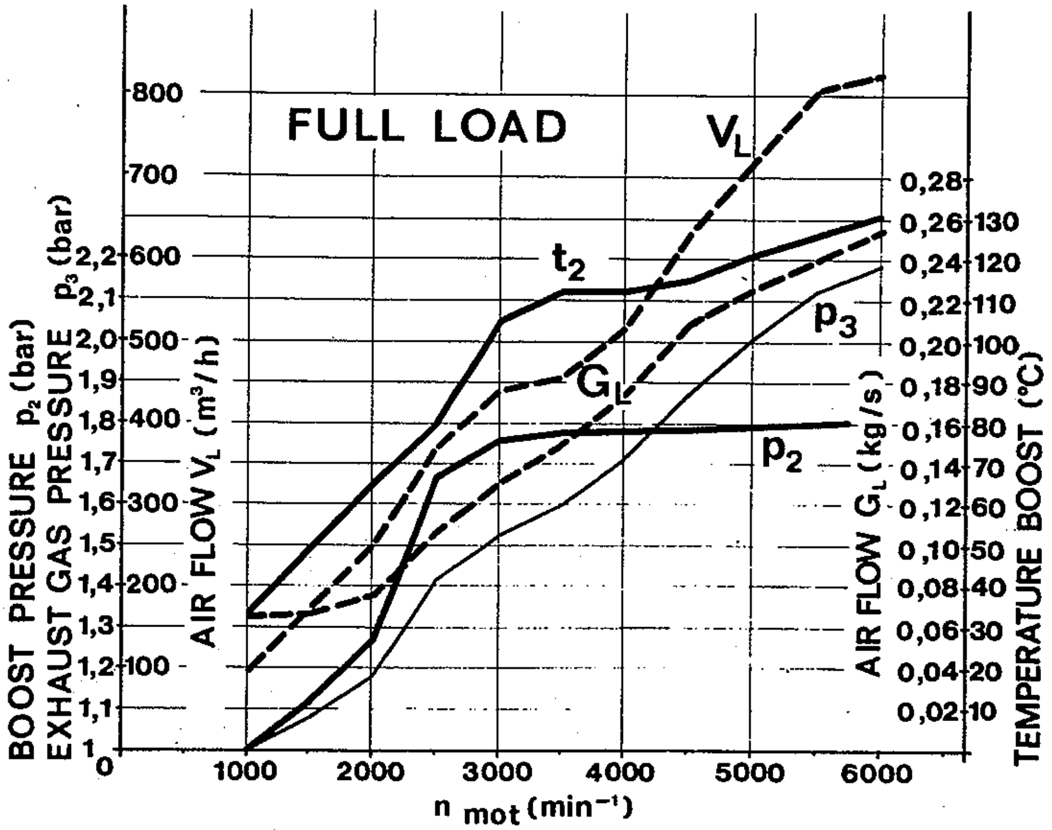


Fig. 11 - Turbocharger characteristics - engine 911 turbo

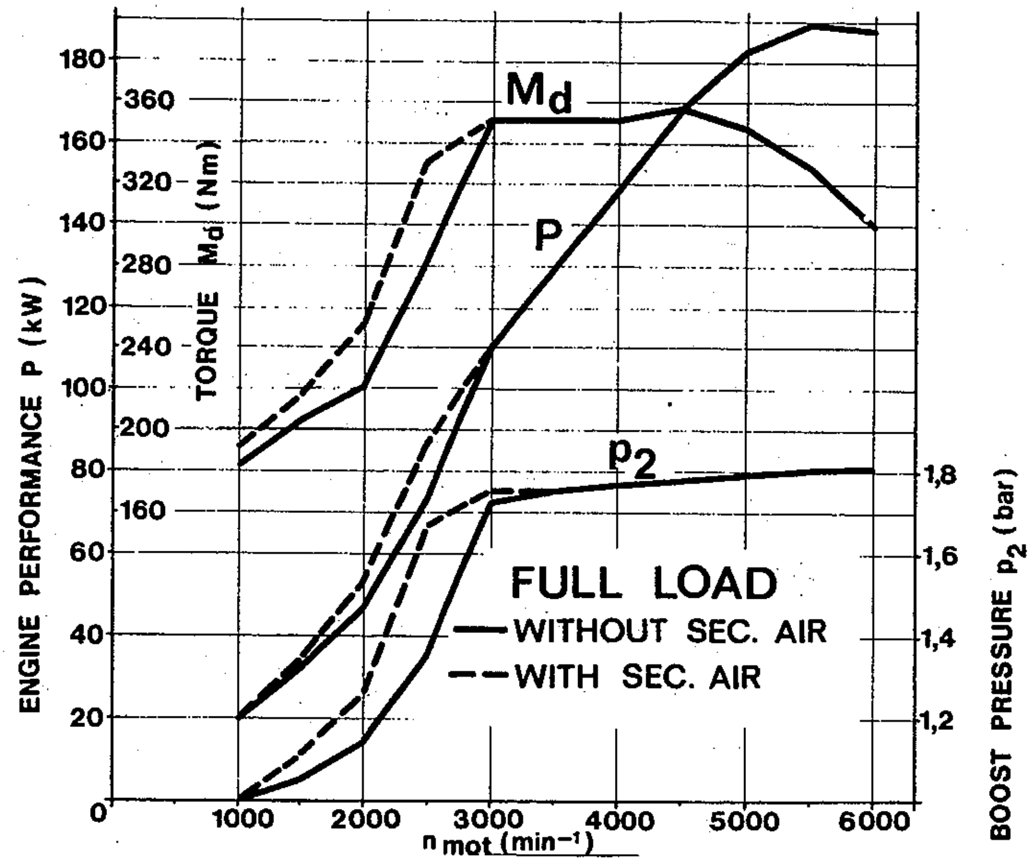


Fig. 12 - Effect of secondary air injection on boost pressure and engine performance

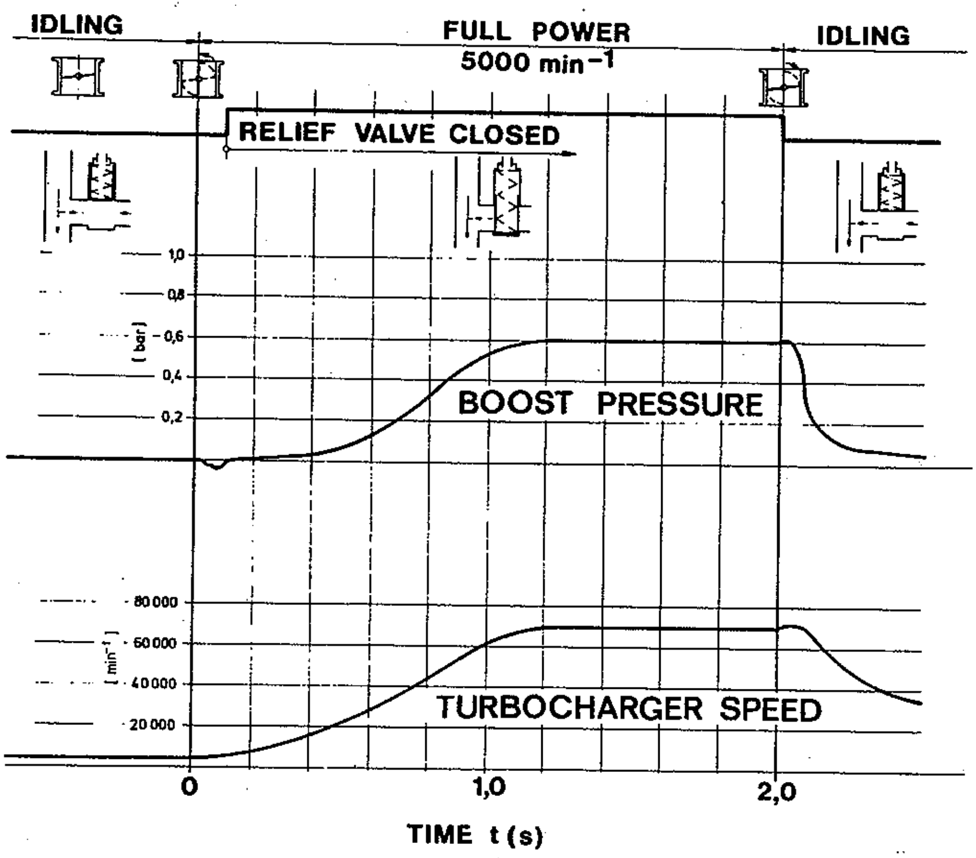


Fig. 13 - Acceleration test on engine dynamometer

Not to overspeed the engine, the dynamometer was prepared not to exceed 5000 rpm. The pressure relief valve closes automatically 0.1 second after opening the throttle. As shown in the diagram, there is a manifold underpressure for 0.1 second only. That is to say that after 0.1 second the engine will deliver more power than a normally aspirated engine.

Fig. 14 shows the compressor map of the 911 turbocharger, showing also the line of the boost pressure ratio at full load.

A scaled drawing of the engine is shown in fig. 15. The problem of placing the turbocharger was solved only by making the muffler smaller. The reduction of the muffler size was allowed by the fact, that, in general, turbochargers have noise reducing properties.

The diagram fig. 16 shows the valve lift curves of three different versions of the Porsche 6-cylinder 911 engine, a 2.7 litre (164 cu.in.) 154 kW (210 hp), a 3 litre (183 cu.in.) 147 kW (200 hp), and the turbocharged version. In the two normally aspirated engines the intake valve lift amounts to 11.6 mm and the exhaust valve lift to 10.3 mm. As shown in the diagram, the turbocharged engine requires a smaller valve lift than a normally aspirated engine. The valve lift in the 911 turbo is 9.7 mm for intake and 8.9 mm for exhaust. Also the valve opening period and the valve overlap of the turbocharged engine is smaller. The valve opening periods of the three compared engines are as follows:

	2.7 l	3 l	911 turbo	
Intake				
opens	64	24	22 deg	bTDC
closes	76	76	62	aBDC
Exhaust				
opens	64	66	52	bBDC
closes	44	26	20	aTDC
overlap	108	50	42	

Reducing the valve lift resulted in a reduction of the inertial forces and, consequently, to an improvement of the valve gear working conditions.

Fig. 17 is a photograph of the 3 litre turbocharged 911 engine, ready to install.

Turbocharging the 911 engine required only a few basic changes: The

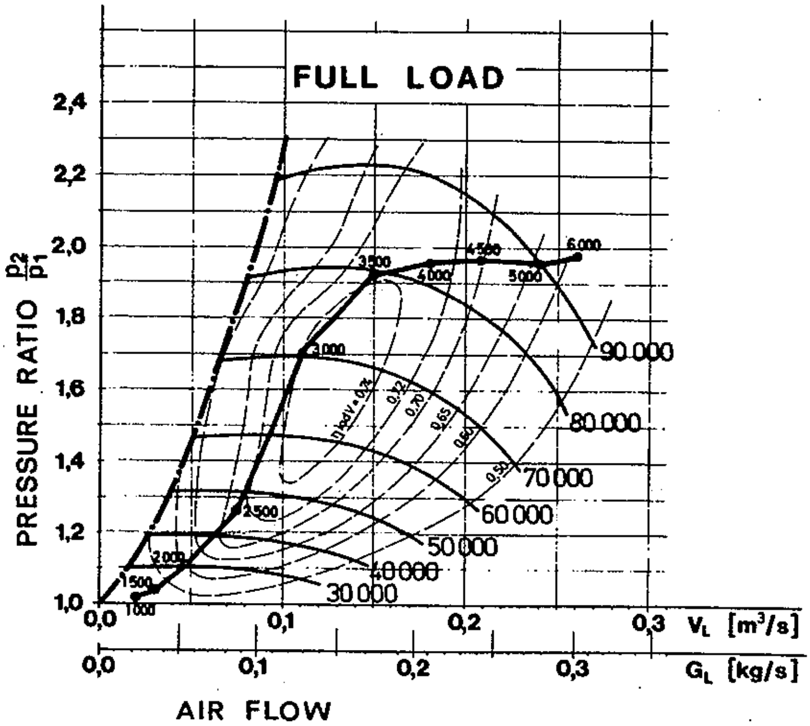


Fig. 14 - Compressor map - 3-L engine 911 turbo

compression was reduced from 8.5:1 to 6.5:1, thus new pistons were required. Due to the higher heat load sodium-filled exhaust valves are used. The performance of the air cooling fan was increased. The valve timing was changed.

Due to the low position of the turbocharger, shown in fig. 17, the oil, lubricating the turbocharger bearings, does not flow back by gravity. It is collected in a small tank and returned into the engine oil tank by a pump. The turbocharger requires 3 to 4 litres oil for lubrication.

At maximum power output the turbocharger speed is 90.000 rpm, and the power input of the compressor is 26 kW (35 hp). In a 500 hp race engine the compressor power input, provided by the exhaust gas, amounts to about 73 kW (100 hp).

The wastegate, working on high heat load conditions, is passed by about 22 per cent of the total exhaust gas volume at maximum performance of the 911 turbo engine.

The US version of the 911 turbo is equipped with two thermal reactors, secondary air injection, and exhaust gas recirculation. The 911 turbo is the first production car equipped with turbocharger which has received an emission certification from the EPA.

In 1978, the performance of the 911 engine was improved again. The displacement was increased and an air-to-air intercooler is used. By cooling the boost air it was possible to raise the compression ratio from 6.5:1 to 7:1. The data of the new 3.3 litre engine are as follows:

Displacement	3298 cm^3 (201 cu.in.)
Bore	97 mm
Stroke	74.4 mm
Performance	220 kW (300 hp)
Speed	5500 rpm
Performance US	195 kW (265 hp)
Maximum Torque	412 Nm (304 ft.lb.)
Compression Ratio	7:1
Boost pressure	0.8 bar

A photograph of the 3.3 litre turbocharged 911 engine is shown in fig. 18. By the use of an intercooler an improvement of fuel economy and an increase of the specific engine performance by approximately 5 per cent was achieved.

THE HISTORY OF THE 911 ENGINE - Since 1964, the Porsche type 911 is in production. The first type engine had a 2 litre displacement, and delivered 96 kW (130 hp). Today's top model 3.3 litre engine has a power output of 220 kW (300 hp), though severe noise and emission standards, not existing in 1964,

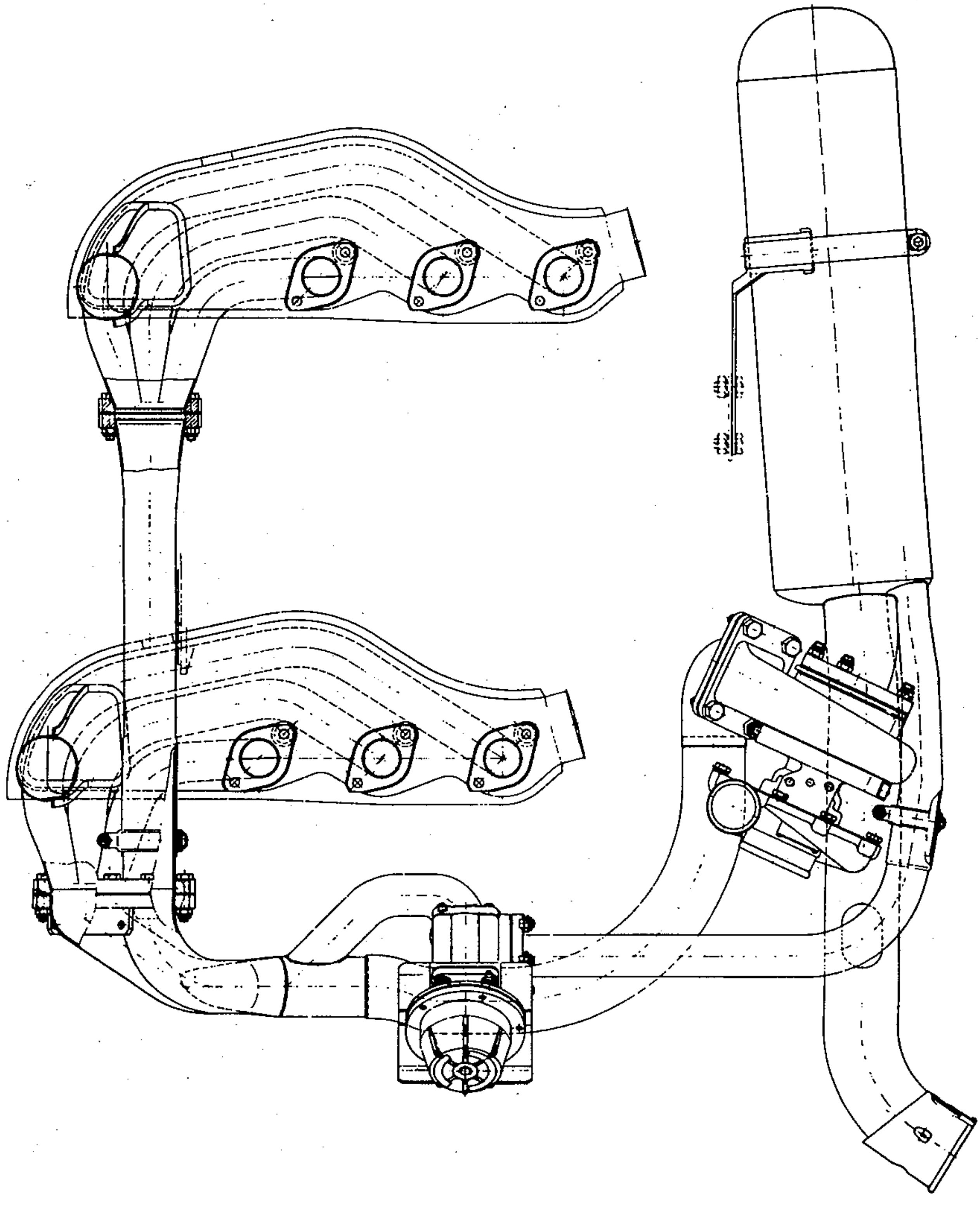


Fig. 15 - Exhaust system - engine 911 turbo

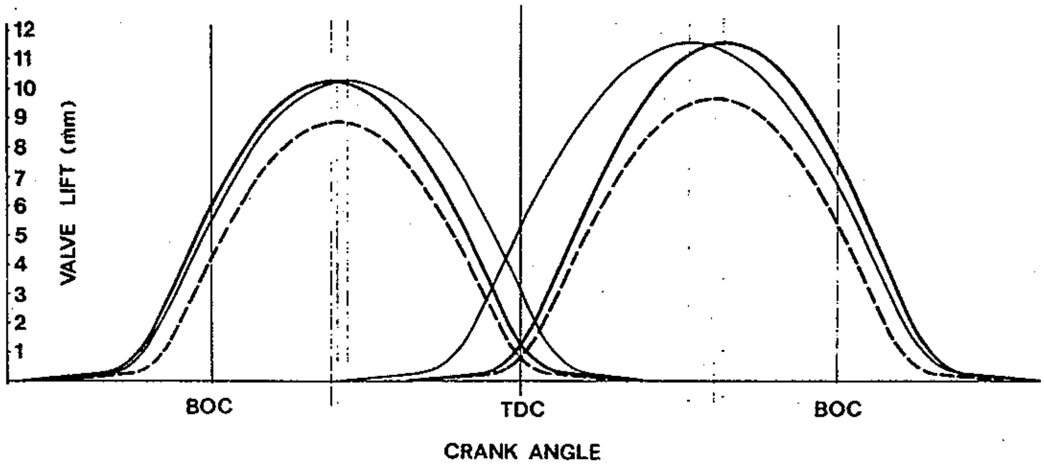


Fig. 16 - Valve lift diagram - Porsche type 911 engines

have to be met. The overall dimensions of the basic engine never were changed.

The application of turbocharging, only, allowed to produce this high performance road vehicle today, qualified for daily use.

In fig. 19 a graph of the historical evolution of the 911 engine is shown beginning with the first year of production, 1964. Caused by legal requirements the engine speed and the specific performance decreased since 1969. From 1972 on, the US emission standards forced Porsche to adapt the US type engines, shown by the dotted lines. Though the specific performance and the engine speed decreased, the power output of the naturally aspirated engines went up slightly until 1975, only by increasing the engine displacement.

The introduction of the 911 turbo, in 1975, marked by the figures 11, 12, 13 and 14, brought up both, the performance and the specific performance, considerably. At the same time the engine speed was reduced, thus effecting the 911 turbo to demonstrate qualified high performance.

TURBOCHARGING OUTLOOK

The chances of turbocharging in general application will depend on its qualification for future requirements.

ENGINE RESPONSE - Applying turbocharging to small engines has experienced that the engine response is in need of improvement. A limited improvement can be achieved by using the exhaust manifold pressure for wastegate control, instead of the intake manifold pressure. Fig. 20 shows the characteristic curves of the manifold pressure for uncontrolled, for intake manifold pressure controlled and for exhaust manifold pressure controlled turbocharging. In the diagram, fig. 21, the results gained from testing a 4-cylinder 2 litre engine are shown. The solid lines represent the torque and the intake manifold pressure, if the wastegate is controlled by the intake manifold pressure. Connecting the

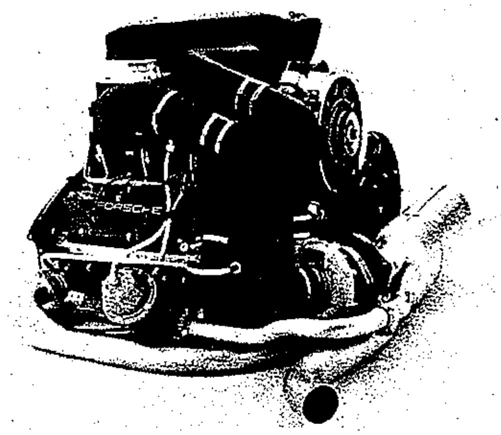


Fig. 17 - Porsche 3-L engine 911 turbo

wastegate with the exhaust manifold resulted in the dotted lines. A considerable increase of torque and intake manifold pressure is achieved. The use of the exhaust manifold pressure for wastegate control, as realized in the Saab turbo, needs careful attention to the knocking behaviour of the engine. Normally, a high boost peak can only be realized if the boost pressure at nominal speed is small.

EMISSION CONTROL - The experience made so far with turbocharged engines has shown that the measures required for emission control are generally the same as for normally aspirated engines. There is an advantage of the turbocharged: due to its smaller displacement at the same performance the exhaust gas volume in the idling range and in the range of low partial load is smaller than at the normally aspirated engine.

Normally, the geometric compression ratio of a turbocharged engine is lower than of a normally aspirated engine. With regard to the HC- and NO_x-emissions there are advantages for the turbocharged engine. Fig. 22 shows the influence of the compression ratio on the HC- and NO_x-emissions. The values shown in the diagram were obtained from tests on single cylinder test engine at partial load. At a compression ratio of 6.5:1 the HC-emissions are about half of the value measured at a ratio of 8.5:1. The NOx-emissions decrease to about 60 to 70 per cent at a corresponding compression ratio.

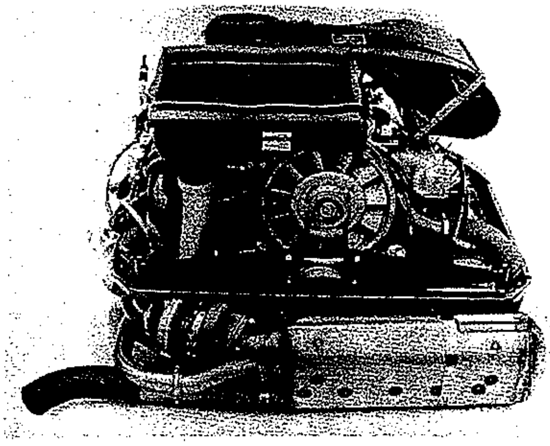


Fig. 18 - Porsche 3.3-L engine 911 turbo

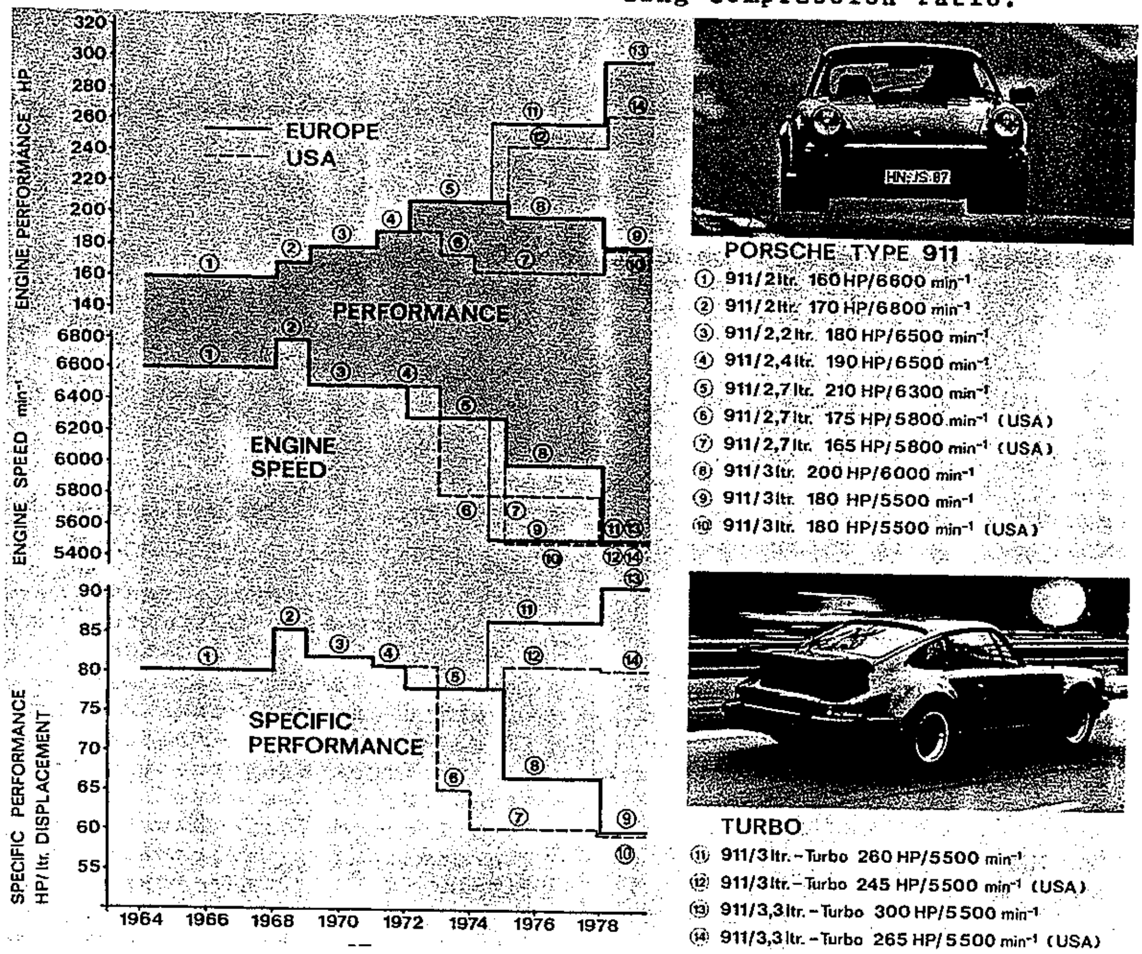


Fig. 19 - Evolution of the 911 engine performance

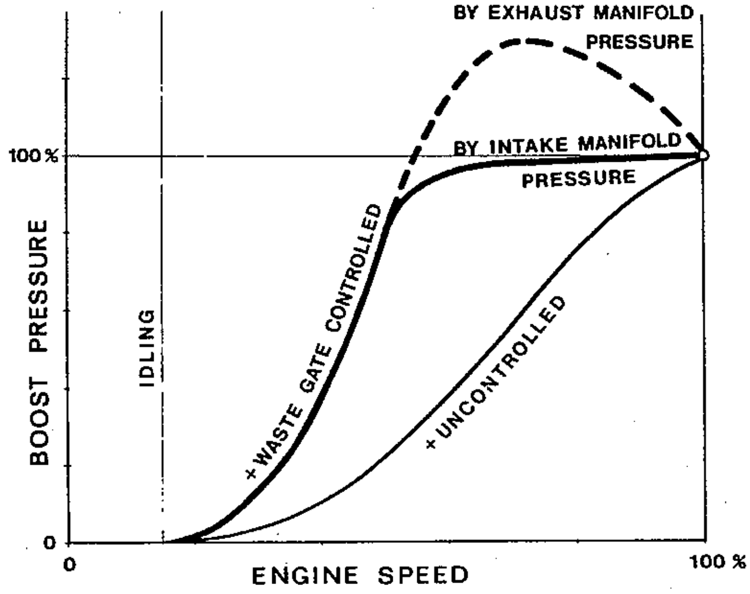


Fig. 20 - Boost pressure control characteristic

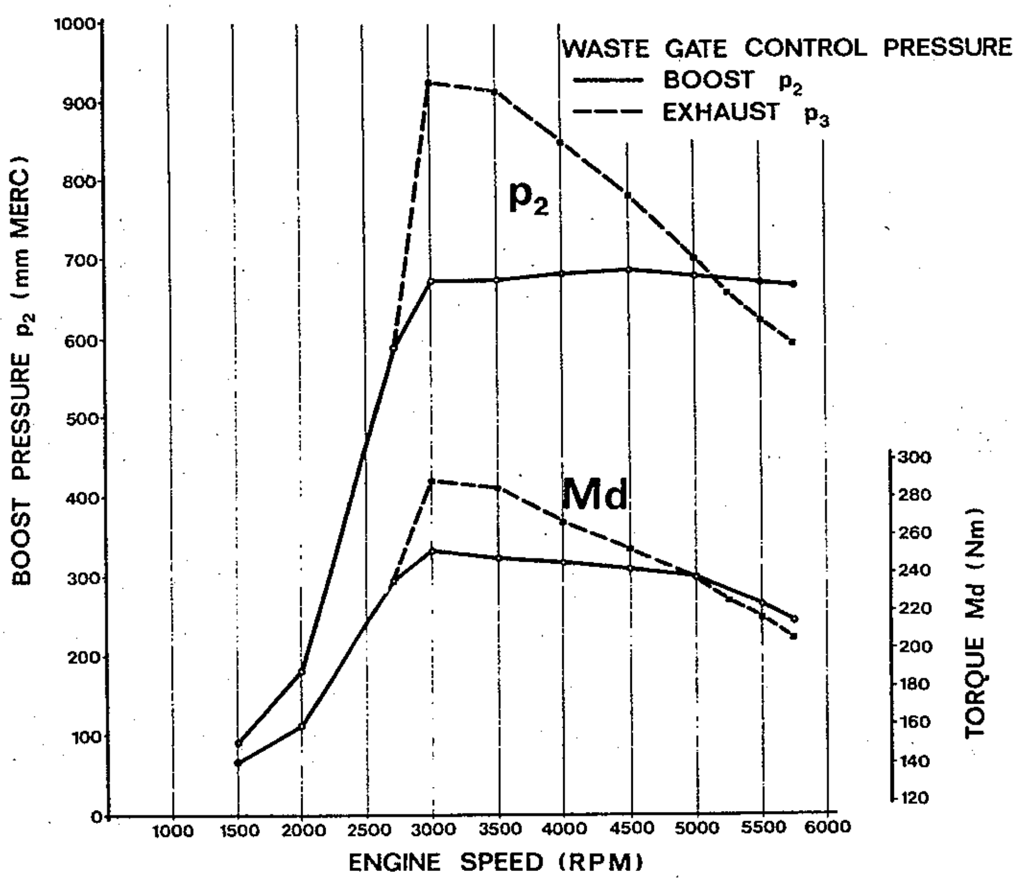


Fig. 21 - Boost pressure and torque 4-cyl 2-L engine

There are different possibilities in arranging the catalyst for turbocharged engines. Fig. 23 shows the catalyst placed close to the engine. This position has the advantage of a quick warm up of the catalyst. However, the turbine rotor can be damaged by small particles coming off the ceramic material, normally not affecting the catalyst function. A catalyst based on metallic substrate would probably allow a position upstream of the turbine.

Fig. 24 shows the catalyst arranged downstream of the turbine. In this position there is no danger of turbocharger damage. However, catalyst warm up is worse.

Fig. 25 shows another version of the downstream position. The exhaust gases passing the wastegate at full load will also bypass the catalyst, thus reducing the thermal load of the catalyst.

The same effect is reached by arranging the catalyst between the wastegate and the turbine.

FUEL ECONOMY - The fuel economy of a turbocharged engine is influenced by the same factors as the normally aspirated engine. Of great importance is the compression ratio. Unfortunately turbocharging normally makes a reduction of the geometric engine compression necessary. There is a relation between the boost pressure, the geometric and the effective compression ratio, shown in the diagram, fig. 26. It is shown, that a high boost pressure requires a low geometric compression ratio. That is to say, that a big increase of engine output, achieved by turbocharging, effects the compression ratio to be substantially reduced, thus causing poor fuel economy. As experienced fuel economy will be deteriorated at variable engine operations by about 3 to 6 per cent if the compression ratio is reduced by one point.

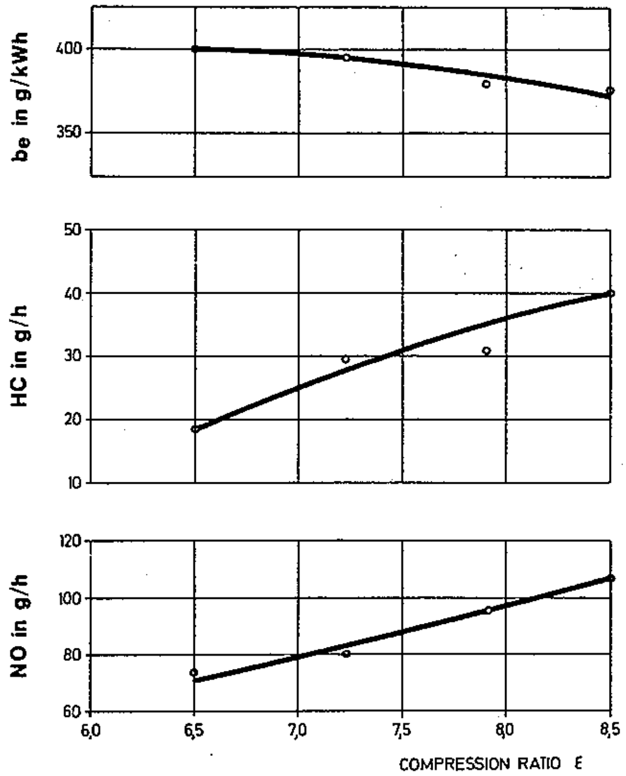


Fig. 22 - Influence of compression ratio on fuel consumption, HC and NO_x

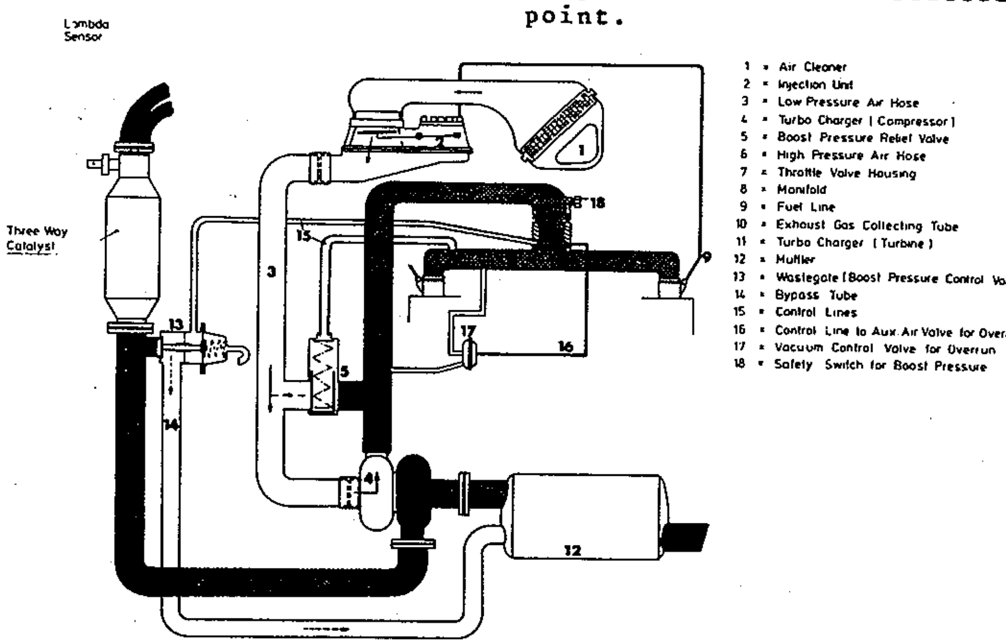


Fig. 23 - Catalyst position upstream of the turbine

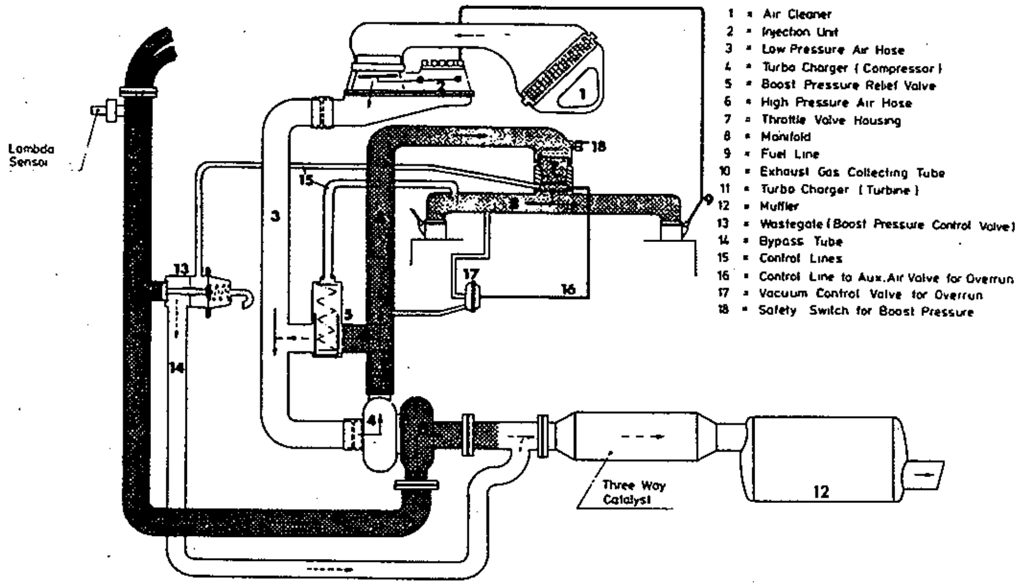


Fig. 24 – Catalyst position downstream of the turbine

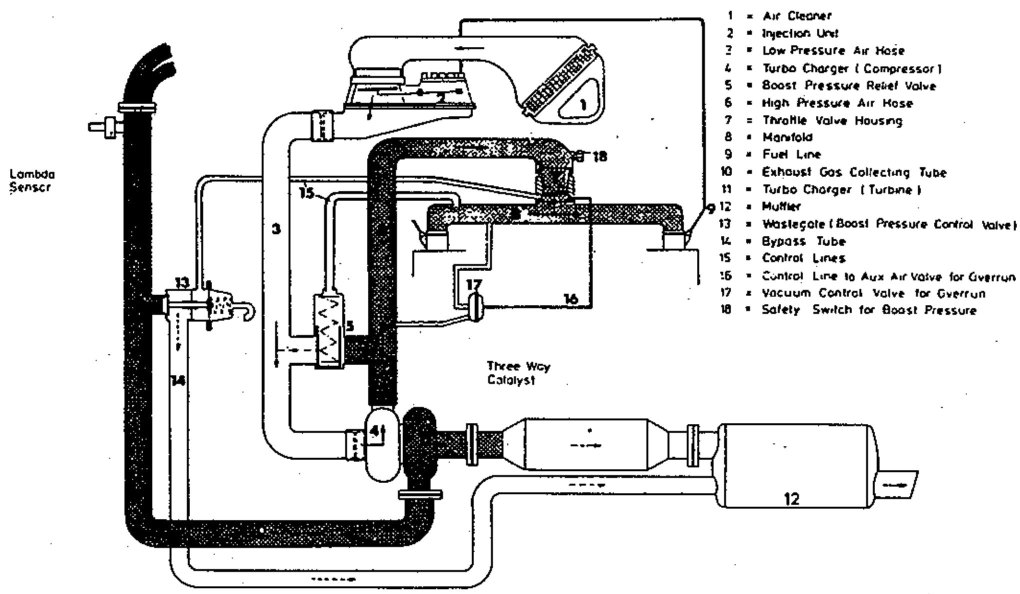


Fig. 25 – Catalyst position downstream of the turbine (bypass)

At high or at full load the specific fuel consumption of a turbocharged engine is superior to the consumption of a normally aspirated engine.
However, today's fuel economy regulations are judging engines and cars by its low-load consumption. Under consideration of comparable performance, the turbocharged engine is expected to achieve better fuel economy results than a naturally aspirated engine. At idling and at low load conditions the influence of the engine displacement is more effective on fuel economy than the influence of the compression ratio.

The diagram fig. 27 shows the fuel consumption of a V 8 engine at a low engine output of 10 kW (13.6 hp). The three lines represent three engine versions, differing only in bore and displacement: 3.9 litre, 4.5 litre and 5.0 litre.

Fig. 28 shows the fuel consumption of a 4.5 litre 8 cylinder engine, a 2.7 litre 6 cylinder engine, and a 2 litre 4 cylinder engine at low load output of 10 kW (13.6 hp). It is shown that a reduction of the engine displacement results in a significant decrease of fuel consumption at idling and partial load.

Fig. 29 shows the fuel consumption at idling related to the engine displacement. Under supposition that by the use of turbocharging the displacement of a 3 l engine, for instance, can be decreased to 2.2 - 2.4 litre, there is a 20 to 30 per cent improvement of consumption, even under consideration of a 10 per cent increase of consumption, caused by a reduced compression ratio.

An objective comparison with regard to emission control and fuel economy between a normally aspirating engine and a turbocharged engine is only possible under the supposition of the same driving performances.

The geometric compression ratio of a turbocharged engine has normally to be reduced compared to a normally aspirating engine, because of the limited effective compression ratio.

With regard to a good thermal efficiency the geometric compression ratio of the turbocharged engine should be kept as high as possible, thus limiting the performance improvement.

Corresponding to an increase in the specific engine performance of about 25 to 35 % the displacement of the turbocharged engine can be reduced. These measures will result in a significant improvement of fuel economy at partial and idling load conditions. Fuel economy can be improved in addition, if the number of cylinders and thus the engine respectively vehicle weight can be reduced.

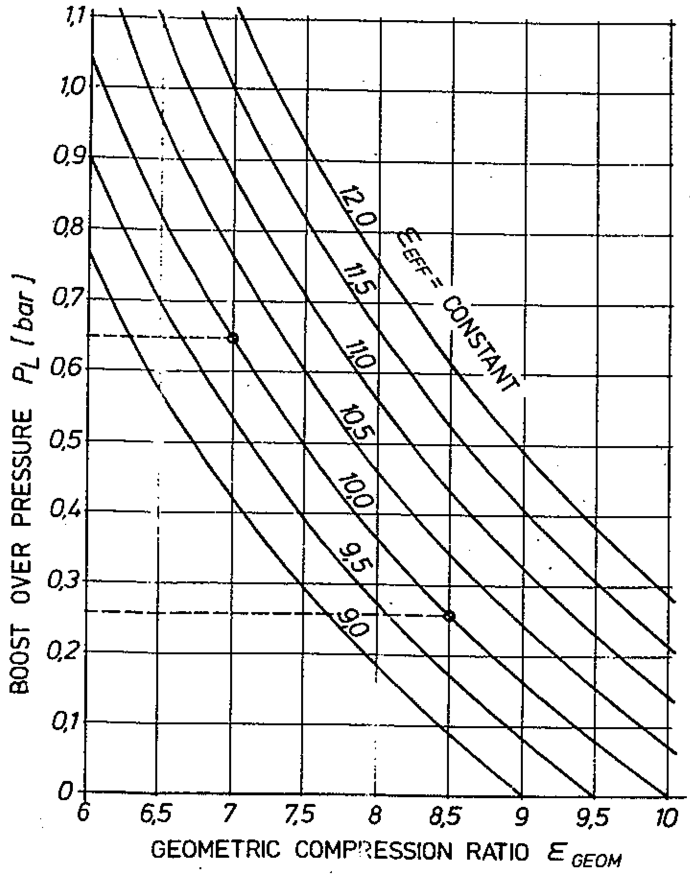


Fig. 26 - Relation between boost pressure, geometric, and effective compression ratio

SUMMARY

Exhaust gas turbocharging for passenger cars made a big progress in the seventies, after the typical throttle lag of turbocharged engines was overcome by the employment of turbocharger controls. Most efficient control device in today's turbocharger technology is a wastegate which bypasses exhaust gas around the turbine, thereby limiting boost to a desired level.

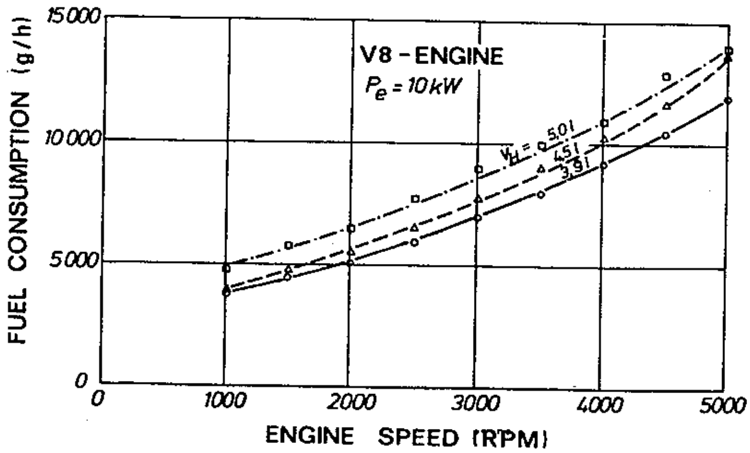


Fig. 27 - Influence of displacement on fuel consumption

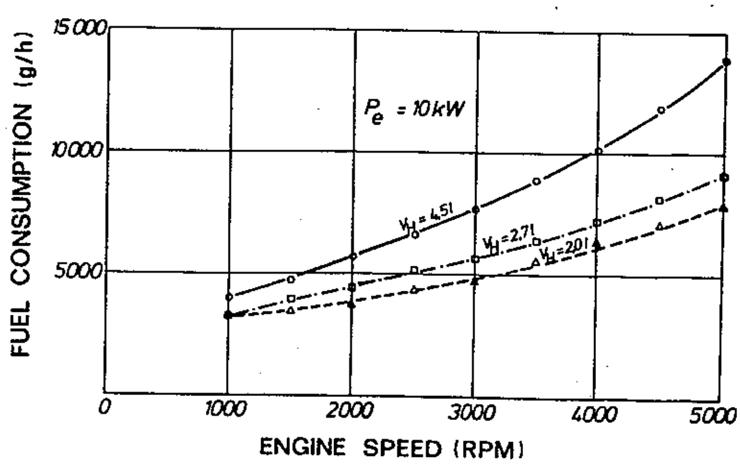


Fig. 28 - Fuel consumption at equal performance of 3 different engines

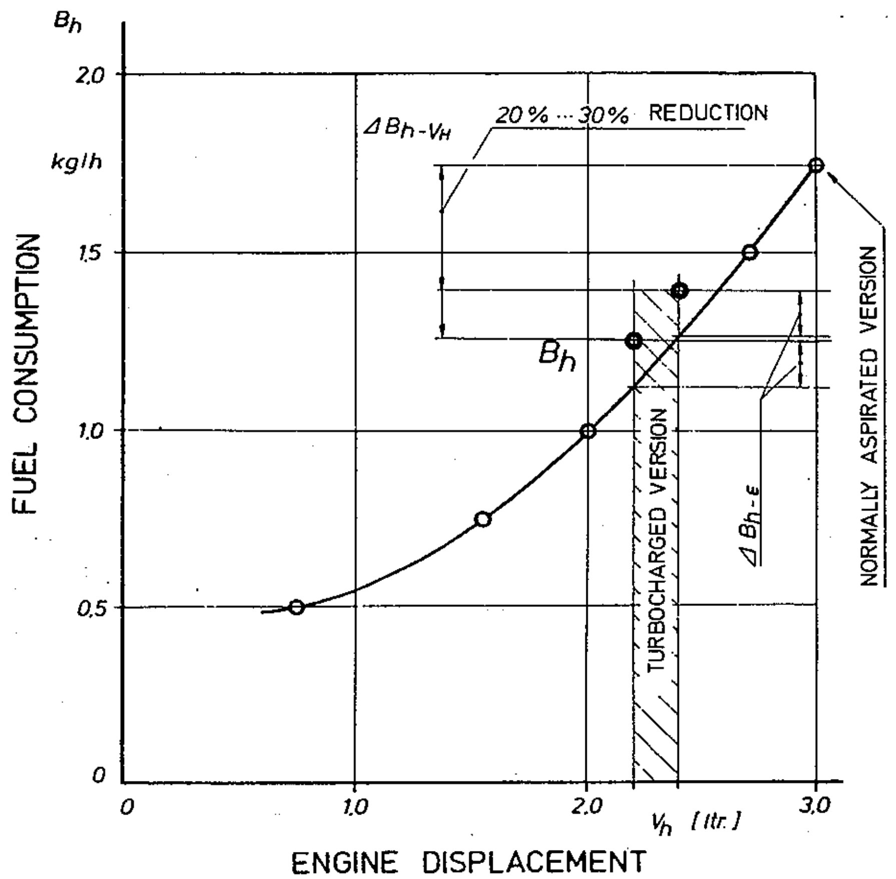


Fig. 29 - Fuel consumption at idling

Considerable merits in terms of performance, engine response and driveability of turbocharged engines were gained in racing development.

Turbocharged racing engines are delivering more than 300 HP per litre displacement. There is an existing production car having 270 HP respectively 82 HP per litre from a 3.3 litre turbocharged engine, still meeting current severe California emission standards HC 0.41, CO 9.0, NO_x 1.5 grams per mile.

Experiences have shown that the adaptation of turbocharging to existing engines will require only small modifications.

Turbocharging makes compensation of power losses, caused by environmental requirements, possible.

By using small turbocharged engines instead of normally aspirated large engines fuel economy can be improved.

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The Coventry Climax Racing Engine 1961 - 1965

Walter T. F. Hassan
Coventry Climax Engines Ltd. and Jaguar Cars Ltd.

SOCIETY OF AUTOMOTIVE ENGINEERS

Combined Powerplant
and Transportation Meeting
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THE PURPOSE OF this paper is to present information concerning the design and development of the 1961-1965 Coventry Climax racing engines but, to commence, it is of interest to ask why Coventry Climax took up motor racing. The answer is that it appeared to afford a means of publicizing our technical and manufacturing abilities to a wide and knowledgeable public, a proportion of whom could be potential customers.

It also provided a compelling incentive to improve our engineering knowledge, and to generate a sense of capability and achievement inside our own organization and amongst those associated with us.

In short, we sought prestige and our plan appears to have been reasonably successful.

HISTORY OF COMPANY'S PARTICIPATION IN MOTOR RACING

Coventry Climax was no newcomer to the field of competitive motoring. The founder of the company, Pelham Lee, father of our present managing director, started a company in Coventry before World War I and made the Coventry Simplex engines as well as a few motor cars; and many awards were won in their day.

Later, as Coventry Climax, they supplied a variety of engines to a large number of motor car manufacturers up to the mid-1930's, when the industry passed through a period of slump, and then turned to fire pump units and generator sets for defense purposes.

It was in 1950, when our defense authorities decided to modernize their fire fighting equipment, that we were given an opportunity to design an ultra lightweight fire pump using light alloy wherever possible, and it was the engine from this unit which attracted the interest of the racing fraternity.

In those days, very small lightweight racing cars were constructed, using 500 cc motorcycle engines, and our lightweight engines were fitted into these chassis with good effect.

In 1956 it was decided to build a new engine especially for use in the then current Formula II. This was a 4 cyl twin cam, type F.P.F., which was first shown at the 1956 British Motor Show and began to score successes during the following year. In 1958, the first Formula I Grand Prix race to be won with a Coventry Climax engine was achieved by Stirling Moss, in Rob Walker's Cooper, using one of these engines bored out to 1960 cc, racing against full 2500 cc cars in the Argentine Grand Prix. This was followed by victory

ABSTRACT

The paper discusses reasons for Coventry Climax's participation in motor racing and includes a history of events which led to the manufacture of the 1961-1965 racing engines.

Design philosophy is discussed; detailed design and development of the various engines is described, including the 1-1/2 liter V-8 with three combinations of bore and stroke and with two and four valves per cylinder, together with the 2 liter version of the V-8, and the 1-1/2 liter 16 cyl engine which was not developed in time to race. Notes on the special exhaust and inlet systems developed for these engines are included.

Lucas petrol injection, and transistorized ignition systems are also described.

The Appendix contains specifications of all the engines together with sectional arrangements and performance curves.

in the Monaco Grand Prix by Trintingnant, also in Rob Walker's car.

These results proved a turning point in racing car design. The combination of a small lightweight car, plus a light and powerful engine -- preferably placed towards the rear end -- made an astonishing change to the conception of a racing car, and the orthodox G.P. car became obsolete almost overnight.

Encouraged by this success, the engine was redesigned to provide a full 2-1/2 liters, and this enabled Jack Brabham in the Cooper car to win the World Championship for both 1959 and 1960.

The engine was further enlarged to 2.7 liters when Brabham and Cooper decided to go to Indianapolis for the 500 mile race in 1961 after the trial run the previous year with only 2-1/2 liters.

CHANGE OF FORMULA 1961

When the International Grand Prix formula was changed from 2500 cc to 1500 cc for the 1961 season, it was necessary for racing car manufacturers to have a more powerful 1500 cc engine than had previously been produced. The situation was complicated by the resistance shown by British contestants to the new formula, and it was not until the end of 1960 that it was finally ratified.

Although Coventry Climax had begun the design of a new V-8 engine, there was obviously too little time in which to develop it into race winning form for the coming year. A Mk. II F.P.F. 4 cyl was therefore produced as an interim measure. It gave considerably less maximum power than its Continental rivals, which had been in existence for some time but, nevertheless, it powered Lotus cars to victory on three occasions in 1961 -- Stirling Moss at Monaco and Nurburgring, and Innes Ireland at Riverside, U. S. A. (See Fig. 1.)

Some of these engines were also supplied to B.R.M. who were in the same position as ourselves, and had no engines ready of suitable size.

DESIGN PHILOSOPHY

Modern Grand Prix circuits vary considerably, some placing a premium upon acceleration, others upon maximum speed. However, if the use of engine performance is studied relative to the various circuits, it is evident that the time during which the car operates at maximum rpm and power is extremely small, compared with the time spent accelerating in order to reach this maximum, only to commence braking for the next bend. The most successful engine is, therefore, the one which combines a good maximum power with exceptional torque spread over at least 2500 rpm or, better still, 3500 rpm. Under these conditions the driver will have a far less fatiguing time, because gear changing will become less exacting and he will have more time to deal with the actual driving of the car, that is, steering, braking, and the like.

It has been our policy to produce engines possessing good torque characteristics in the middle speed range. To achieve this is impossible if valves and ports of unlimited size are used in order to achieve the highest maximum power regardless of other requirements.

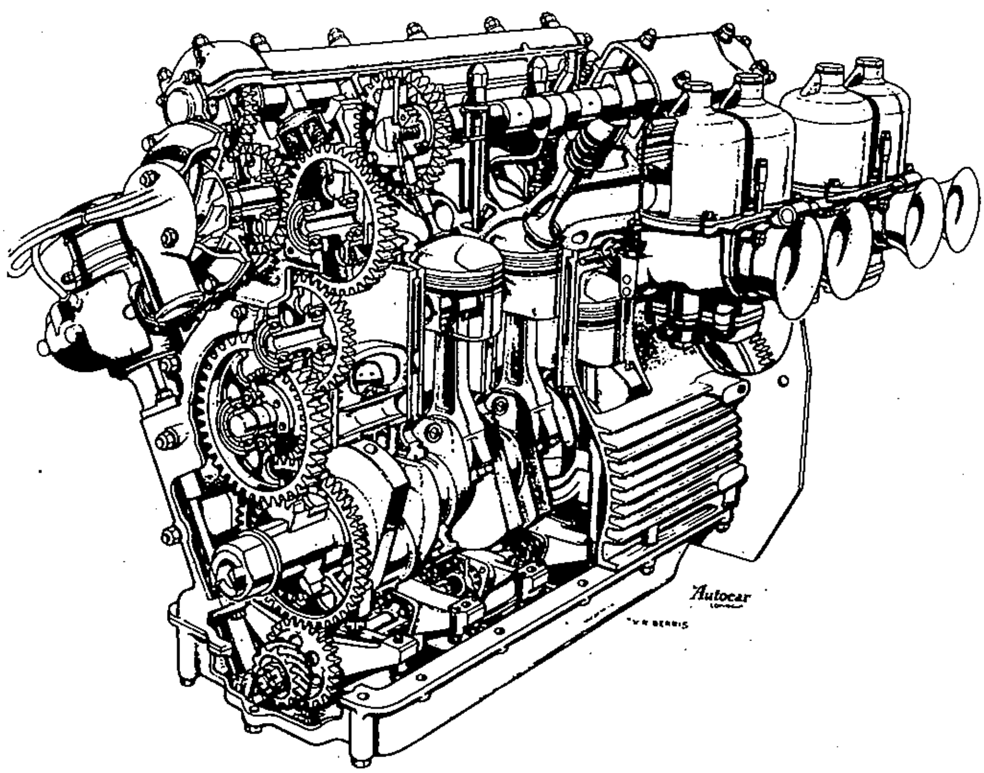

Fig. 1 - Cut-away drawing of F.P.F. 4 cyl engine

The Coventry Climax approach is, therefore, to incorporate the smallest valves and ports from which adequate power can be obtained, since this does allow maximum torque to be developed below the speed of maximum power, and a high compression ratio assists considerably in compensating for the slight lack of breathing at maximum speed. A comparative test showed a useful improvement in power above 8000 rpm for an increase of compression ratio from 10.3 to 11.9:1 (Fig. 2).

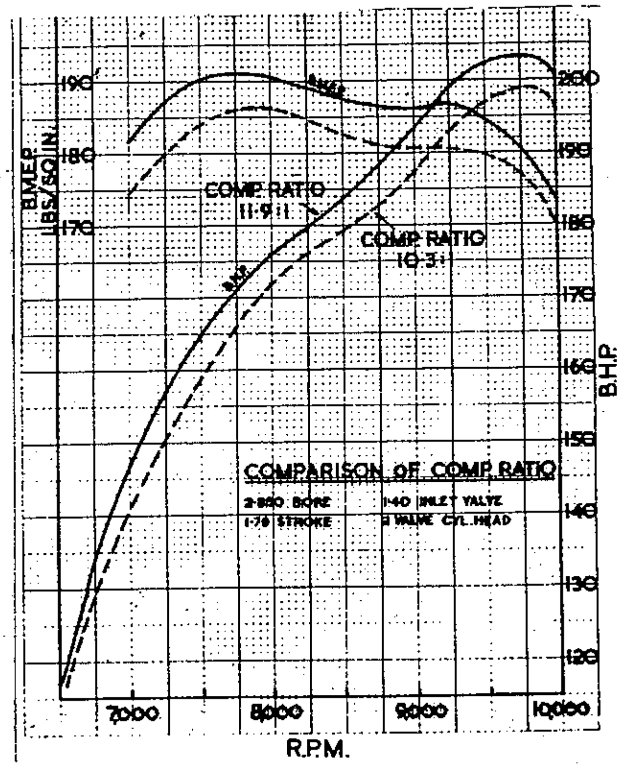

Fig. 2 - Effect on performance with change of compression ratio

BASIS OF DESIGN V-8 1500 CC

The F.P.F. 4 cyl engine (1956-1961) with a bore of 3.2 in. and stroke of 2.8 in. had a limited performance of 100 bhp/liter due to the speed limitations imposed by the valve gear and by the stress considerations of connecting rod and pistons. The engine developed 152 bhp at 7500 rpm, and when run up to 8000 rpm the tappets and connecting rod bolts occasionally gave trouble (Fig. 3).

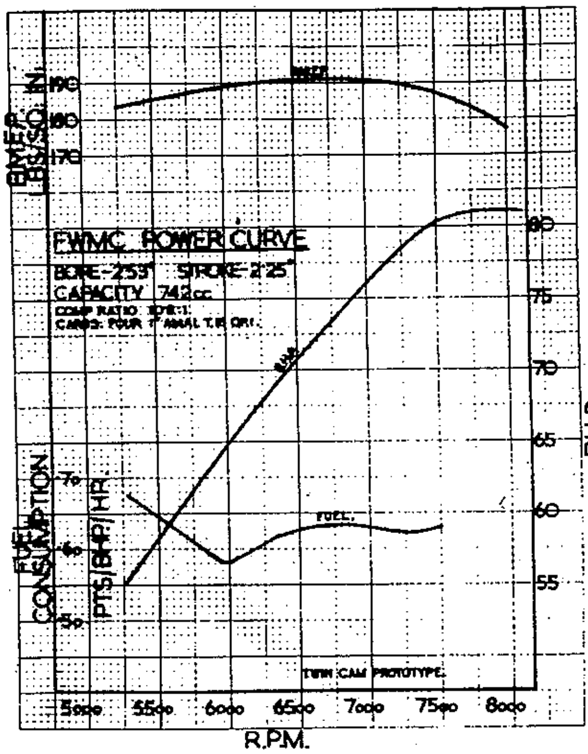

Fig. 4 - Power curve - F.W.M.C. 742 cc engine

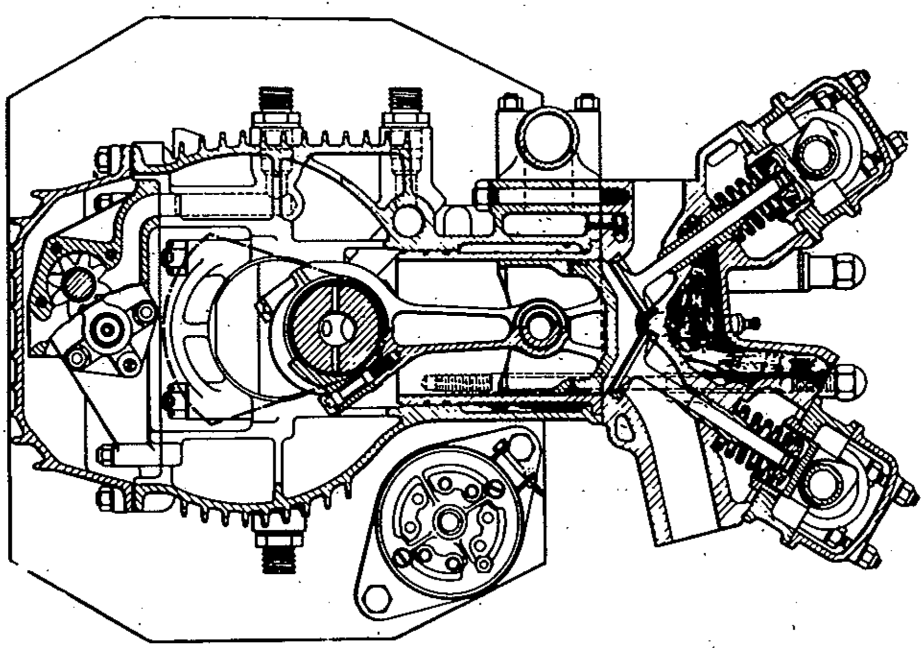

Fig. 3 - Cross-section of F.P.F. 4 cyl engine

In order to obtain a sufficient increase of power with reliability, it was obviously necessary to reduce the inertia of the reciprocating parts, this by reductions in dimensions.

A 90 deg V-8 cyl layout was the logical choice and, fortunately, we had a 4 cyl engine type FWMC of 742 cc capacity with a suitable stroke/bore ratio, on which we could test our new piston and port design.

This engine was basically that used in our small fire pump. When fitted with a twin camshaft hemispherical cylinder head with pistons to suit it gave 82.5 bhp at 8200 rpm which, at 1500 cc capacity, gave us 170 bhp as a satisfactory starting point. (See Figs. 4-6.)

Initially we did not think it wise to aim at too great an increase in rpm, and we also considered that a very much over-square bore to stroke ratio would involve a difficult and lengthy development period which, of course, we could not afford since the engines were required for the then current formula.

One major difficulty with a large bore to stroke ratio is the provision of a sufficiently high compression ratio with a smooth and compact combustion chamber. The provision of sufficient valve to piston clearance, using the large degree of valve overlap necessary for high speed operation, results in deep cavities in the piston crowns.

We therefore settled for a conservative peak crankshaft speed of 8500 rpm which, with a bore and stroke of 2.48 and 2.36 in., respectively, together with a connecting rod center distance of 4.2 in., gave a maximum piston acceleration of 97,800 fps^2, and a mean piston speed of 3340 ft/minute.

Earlier it was considered that piston acceleration should not exceed 100,000 fps^2, but it appears that modern piston rings will operate satisfactorily at substantially higher accelerations. The higher stresses in connecting rods and pistons are matters for design, in which the economic availability of forged pistons act as a bonus.

The design was deliberately simple and aimed at trouble free manufacture in extremely small quantities. A total of 20 engines was envisaged, and therefore "tooling up" was out of the question.

DESIGN FEATURES OF FWMV Mk. I - 1-1/2 Liter V-8

Initially the crankshaft was of conventional design, with five main bearings, and was made of EN24V steel (1-1/2% Ni-Cr-Mo). The front and rear crank pins were set at 90 deg to the center pair which were 180 deg apart. (Figs. 7-9).

The drilling was arranged so that the oil feed was continuous from end to end, thus assisting in the balance of feed to the big ends -- despite the possible variation in leak-off at the various bearings. The feed to the big end bearings was via holes drilled through both sides of the crankpin at 90 deg to the tdc position.

Later the 90 deg phasing was changed in favor of the 180 deg or "flat form" of crank and, to combat a wear problem, a change in material to nitrided steel EN40 was made.

The connecting rods, of 'I' section, were steel stampings in EN24V (1-1/2% Ni-Cr-Mo) and ran side by side in pairs. The caps were fastened with set screws, the split line being at 90 deg to the center line of the rod. The main and big end bearings were of Vandervell V.P.2 strip type lead-bronze indium plated.

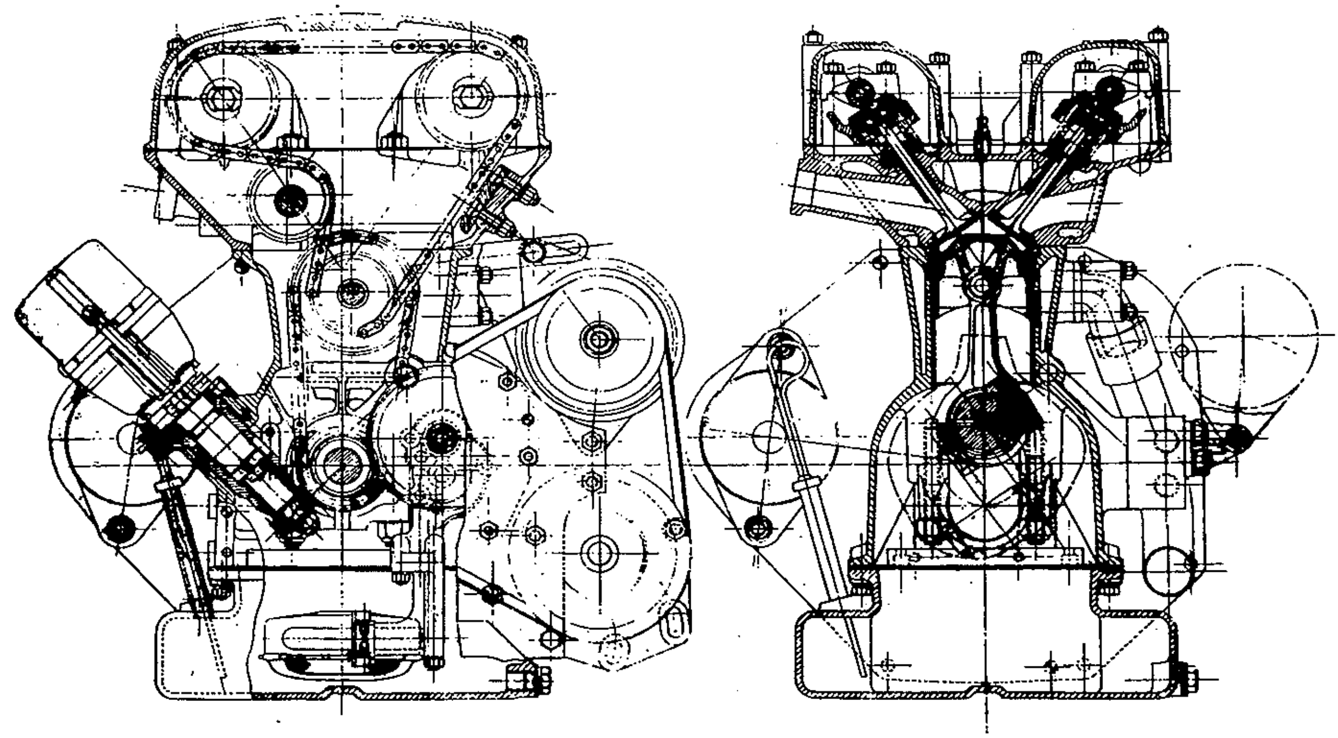

Fig. 5 - Cross-section - F.W.M.C. 742 cc engine

The cylinder block was a simple sand casting, using an open top construction to simplify the jacket coring. Provision for a jackshaft, running at half crankshaft speed, was made in the base of the " Vee" above the crankshaft, with a skew gear at the rear end to drive the ignition distributor.

The main bearing panels were substantial, with the studs set deeply into them by using a thread depth of twice the diameter. Steel main bearing caps, together with cross bolting, were used to increase the rigidity of the structure. Oil was fed to each main bearing from a large diameter gallery

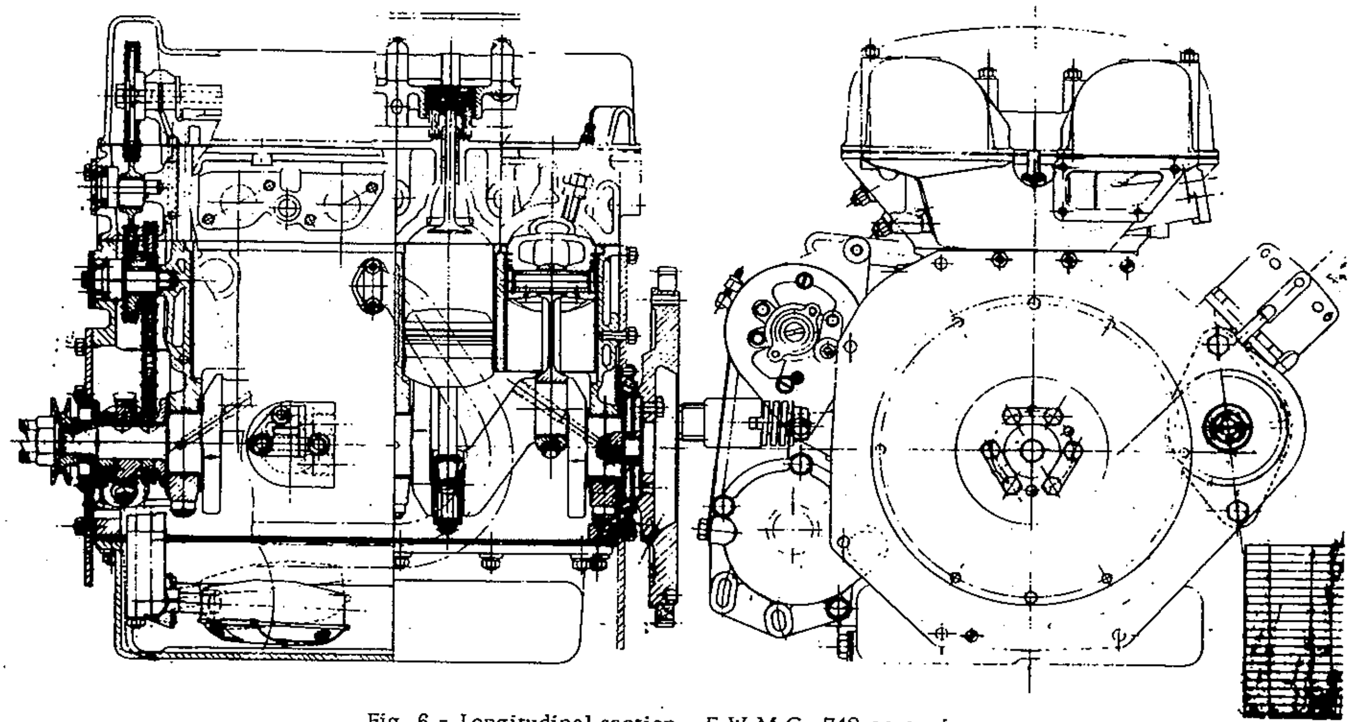

Fig. 6 - Longitudinal section - F.W.M.C. 742 cc engine

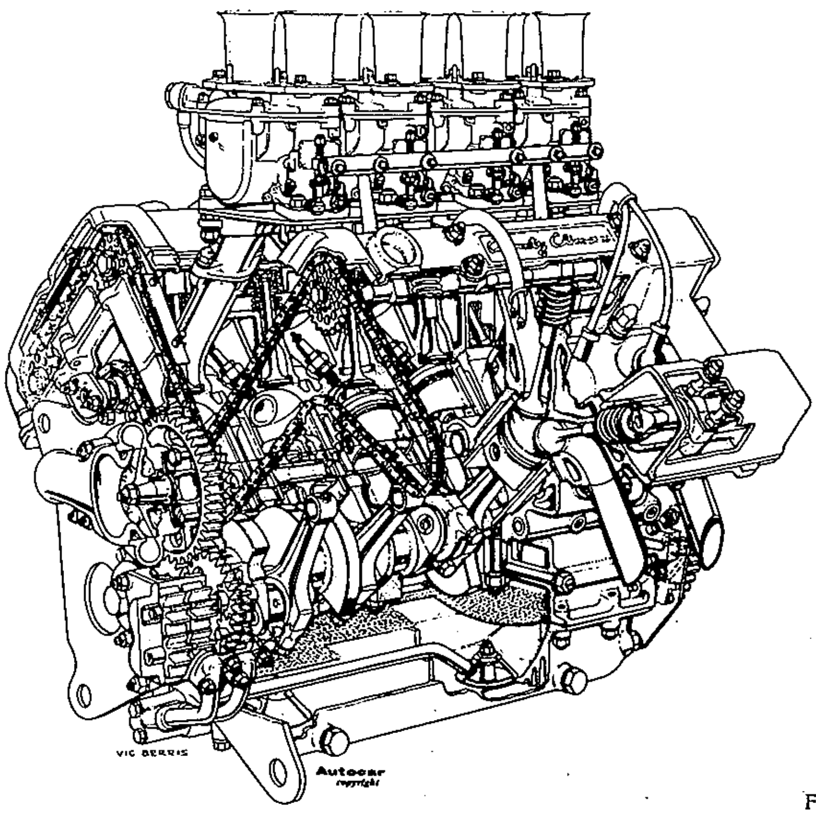

Fig. 7 - Cut-away drawing - 1.5 liter V-8

bored, from end to end, in the side of the crankcase.

The oil pumps were arranged in tandem, with the pressure pump set on the front main bearing panel behind the gear drive from the crank. The two separate scavenge pumps were located on the outside of the timing drive cover plate, and connected to the driven gear by means of an "Oldham" coupling. The pumps, which separately scavenged the front and rear of the sump, to accommodate both acceleration and braking conditions, had a combined capacity of twice the pressure pump. All the pumps were of the normal spur gear type.

The pressure pump fed the oil to a short drilling, via a

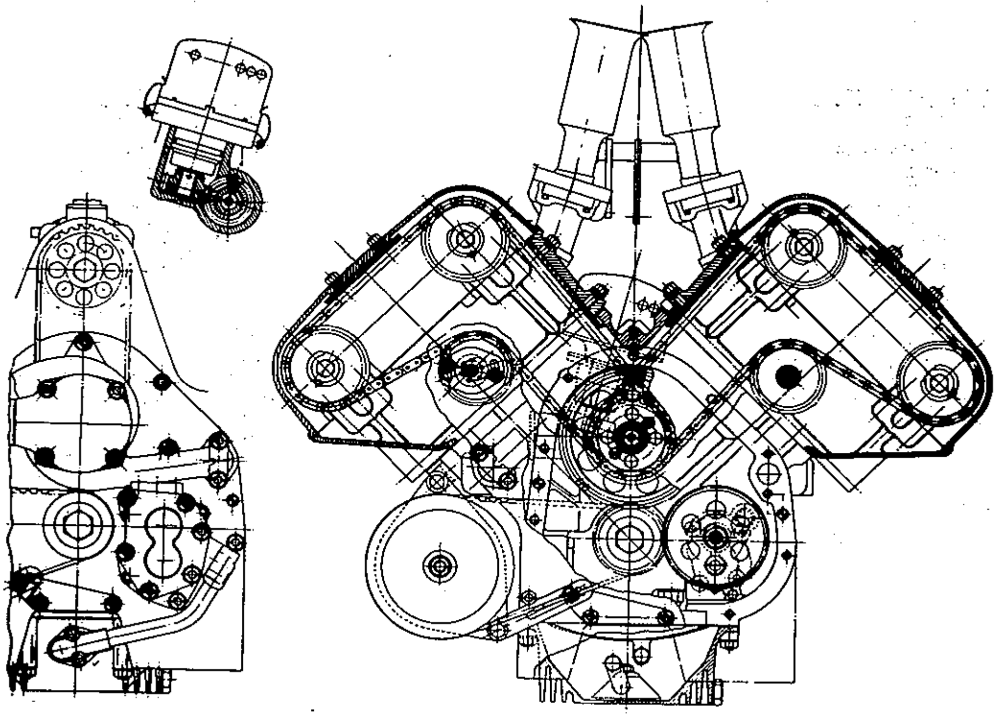

Fig. 8 - Cross-section - 1.5 liter V-8

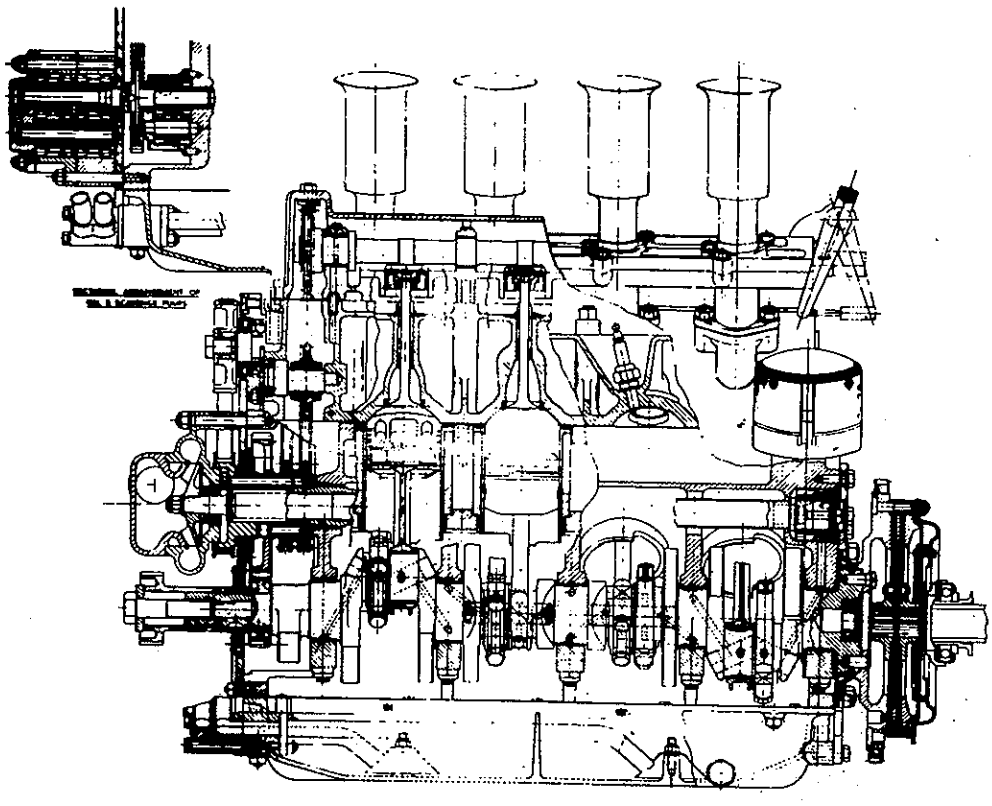

Fig. 9 - Longitudinal section - 1.5 liter V-8

pressure relief valve, and then to a cartridge-type oil filter which was bolted to a facing on the side of the crankcase. The filtered oil then passed to the gallery mentioned above.

The pistons were of conventional shape with a circular skirt. The three ring pack consisted of a top compression ring of the Dykes "L" pattern, a taper faced second, and the usual "U" section oil control ring -- all in normal high duty cast iron. The pistons were made by Brico in "Lo-ex" alloy and each was checked by X-ray for faults.

The cylinder liners were of the usual wet-type in cast iron, the lower end spigoting into the crankcase and sealed with O-rings. The lower flange was supported at the base of the water jacket, and the upper flange was sealed with a laminated steel sealing ring designed and made by Coopers Mechanical Joint Co. Ltd. This ring was originally conceived for Tony Vandervell's famous "Vanwall Special," and consisted of a pack of narrow rings, some flat and some with a corrugation of 0.010 in. thickness, all contained within a spun casing. They were made of stainless steel, and Nimonic material. The water joint was completed by an asbestos type jointing material (Figs. 10 and 11).

The timing drive consisted of a pair of straight spur gears of 1:2 ratio, driving from the nose of the crankshaft to a jackshaft situated in a tunnel in the Vee of the cylinder block,

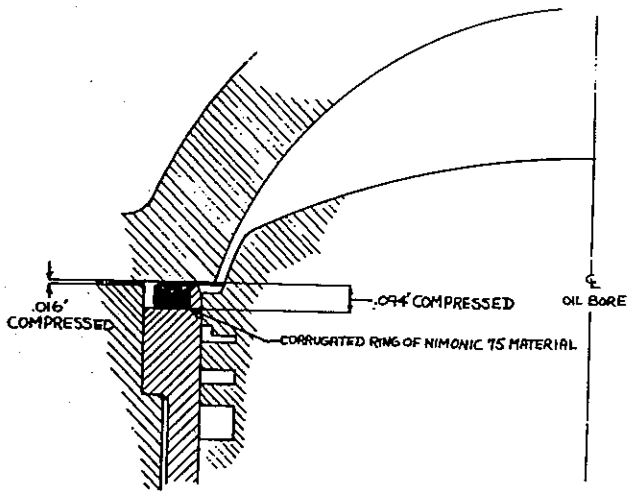

Fig. 10 - Installation of "Cooper" sealing ring

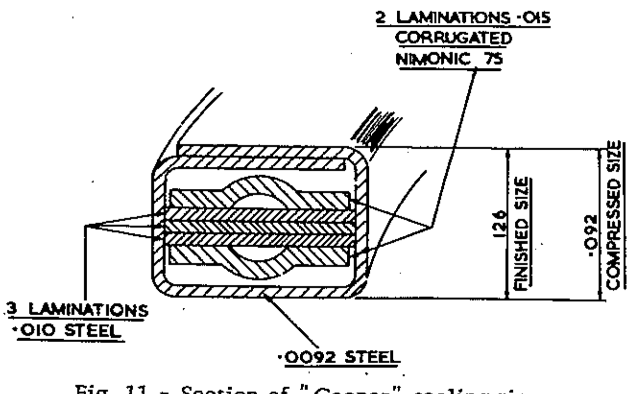

Fig. 11 - Section of "Cooper" sealing ring

from whence two separate chain drives were taken to each of the cylinder heads. The chains were of the single 3/8 in. pitch Reynolds roller type, each tensioned by an adjustable idler sprocket, one being situated in the tight strand on one side and in the slack strand on the other, so as to retain a symmetrical layout.

The cylinder heads were quite simple castings of the conventional hemispherical type, with one inlet and one exhaust valve set at 30 deg on either side of the center of the combustion chamber. The inlet ports were arranged on the inside of the Vee and sloped up quite steeply to the induction system.

The camshafts were carried in babbitt lined strip bearings and supported in castings which also served as tappet blocks.

The tappets were of simple bucket form and the valve clearance was adjusted by means of oil hardened steel shims of varying thickness. The tappets were originally of chilled cast iron and the camshafts of case hardened steel. Later, when the engine operating speeds rose, the materials were reversed. Currently, we use case hardened steel tappets in conjunction with cast iron camshafts with chilled cams. The tappets ran direct in the aluminium tappet blocks.

We used, successfully, exhaust and inlet valves in 21-4NS material, seating on Hydural bronze and austenitic steel respectively. Both inlet and exhaust valve seats were shrunk into the cylinder head.

The older 4 cyl engines were equipped with Nimonic 80 valves, and these suffered quite badly from seat scuffing and ridging so that it was difficult to maintain tune because of seat deterioration. The 21-4NS becomes workhardened with use, and this provides very favorable seat conditions, together with almost complete absence of valve failure.

LUCAS TRANSISTORIZED IGNITION EQUIPMENT

Lucas electronic ignition equipment was specified for these engines from the outset -- a bold venture for, in 1961, electronic ignition was an unknown quantity both in terms of reliability and behavior. The particular form adopted has already been described in some detail in an earlier paper.* Briefly, the equipment comprised of four components:

1. A magnetic variable reluctance pickup located between the rear engine plate and the flywheel.
2. A pulse shaping network based on a Schmitt Trigger circuit.
3. A spark generator using a novel circuit, which is basically a triggered high power blocking oscillator.
4. A high tension spark distributor.

All the electronic units used germanium transistors, and were encapsulated in a silica-filled epoxy resin to provide support against vibrational effects.

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*J. W. Sharpe, "Transistorized Ignition for High Speed Gasoline Engines." Paper 650498 presented at SAE Mid-Year Meeting, Chicago, May 1965.

Attached to the flywheel were four specially shaped steel trigger pieces, spaced at 90 deg intervals, which passed within 0.020-0.040 in. of the pickup poles. Each pass disturbed the magnetic field thus producing a voltage pulse at the pickup terminals. The special shape of the trigger pieces was required in order to compensate for time delays in the circuitry; otherwise an unacceptable degree of spark retard would occur with increase in speed.

The pickup pulses, of variable shape and amplitude depending on the speed, were fed to the pulse shaper. This converted the voltage signals into a current pulse having a shape and amplitude independent of speed. The output current pulse was used to trigger the spark generator (Fig. 12).

The particular features of the unit were:

1. It produced a spark almost instantly when commanded by the trigger pulse, the required power being drawn from the battery over a period of approximately 200 microsec.

2. Each spark had a constant value independent of speed. Minimum test values were 20kv on 50pf load and 13kv on 1 megohm + 50pf load at 800 sparks/sec.

3. The unit was capable of performing at extremely high sparking rates and figures of 3000 sparks/sec have been recorded although, at this rate, the components are being grossly overloaded.

4. The current pulse, taken from the battery at each spark, had a peak value of about 80 amp and large section battery cables were required to avoid excessive volt drop. The average current increases linearly with speed, being zero with a dead engine and 3-4 amp at maximum speed.

1962 was the first full racing season during which this equipment was used. With only works-supported cars using the few engines made during this period, it was a relatively simple job to produce and test the ignition systems under laboratory conditions. Very few failures occurred at this time, although it was found necessary to incorporate a diode to provide reverse polarity protection.

The best power transistors available were only rated at 15 amp and it was necessary to select those capable of handling the 80 amp peak current. This was done by measuring the ignition performance of the transistors in a dummy unit. As more engines became available during the 1963 and 1964 seasons, the scarcity of suitable transistors became a serious problem, and the standard of performance had to be reduced to enable sufficient units to be produced. It was not realized, however, that although the lower performance level was still adequate, the lower grade of transistors was not, and a strange form of transistor failure started to occur. The failure always occurred after switching off the engine on the starting grid, and with two or three "dead" cars on the line the start of the race became chaotic and fraught with danger. The rule concerning the dummy grid start was a direct result of this trouble.

The problem was eventually overcome by:

1. Specifying a 25 amp transistor which, fortunately, appeared on the market at that time.

2. By basing transistor selection on high current gain rather than on ignition performance.

Since then the system has proved extremely reliable with only the occasional random failure occurring.

INITIAL DEVELOPMENT PROBLEMS

Power Output - The early tests in 1960 were disappointing, the performance being much lower than was estimated. The trouble was eventually traced to a last minute change to the inlet port, as a result of suggestions from our consultant, which increased the rate of flow through the port. The change resulted in an increase of area adjacent to the valve guide.

Not until we sleeved the port of the original dimension was the estimated power achieved. It seems that the change of area just before the valve seating upset the ram condition, probably by constituting some form of pneumatic spring, which dissipated the dynamic energy of the column of intake gas in the port (Fig. 13).

Oil Churning - Another early difficulty concerned the churning of the oil in the crankcase by the crankshaft and connecting rods. A sizable gain in power was achieved by lowering the sump 2 in. Various forms of baffle were tried out and, eventually, a copper gauze screen, arranged to follow the path of the big ends, was found to overcome the

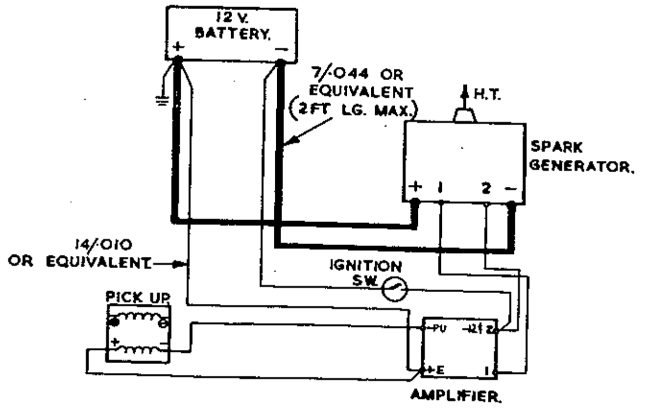


Fig. 12 - Circuit diagram of Lucas transistor ignition system

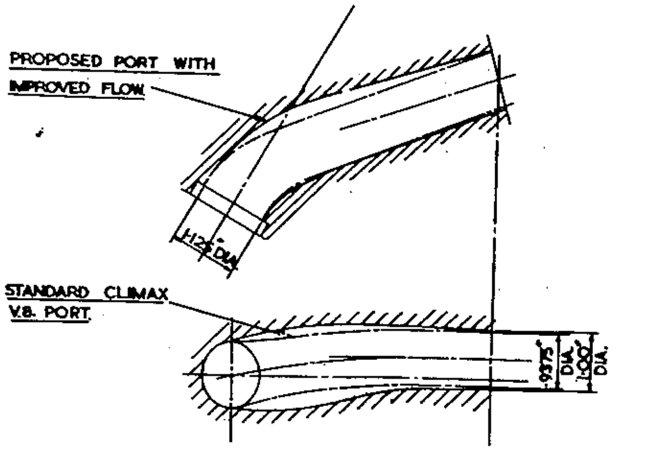


Fig. 13 - Comparison of inlet ports

problem successfully. It would appear that it acted as a non-return valve, and protected the oil from the windage effects of the rotating components. This screen enabled the base of the sump to be raised again to within 5/8 in. of the designed level (Fig. 14).

COOLING

The most difficult and serious trouble was the overheating experienced when the engines were first installed in the racing cars. It took three races -- the German, Italian, and U.S. Grand Prixs of 1961 -- to discover the real cause of the trouble; namely, the differential expansion of the cylinder liners and cylinder blocks as a result of the rather rapid changes in engine load and temperature.

The cast iron liner was seated on a flange 3 in. below the cylinder head seal and, therefore, the seal was subjected to a varying compression due to the differential expansion of 3 in. of cast iron and aluminium. This was probably accentuated by the fact that since the water temperature remained relatively constant, the height of the aluminium cylinder block also remained constant, whereas the cast iron cylinder liner was subject to temperature changes due to firing conditions, that is, throttle open, and throttle closed.

The sealing ring failed due to fatigue, and it was not until we rigged up the test bed with the car's actual sealed cooling system, and operated the throttle on a cycle of opening and closing, that the condition was achieved.

Once the cause was known, it was quickly and easily rectified by replacing the lower flange on the cylinder liner with an aluminium sleeve, sealing into the base of the cylinder block, and supporting the liner under the upper flange, so equalizing the expansion rates with that of aluminium (Fig. 15).

Incidentally, a deck was added to the top of each cylinder bank with the object of locating accurately the top of each cylinder liner, so that they were held truly perpendicular while the cylinder heads were assembled in them. No cooling troubles occurred after this modification.

EXHAUST SYSTEM

The V-8 posed a problem when a tuned exhaust system was considered. The tuned system is a "must" at this stage of the art, and a good system is worth many bhp as compared with a "hit and miss" layout such as was common a few years ago.

We had discovered that the original scheme used on the

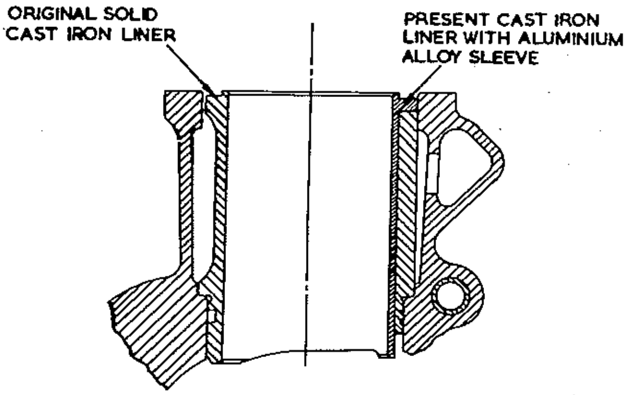


Fig. 15 - Comparison of cylinder liners

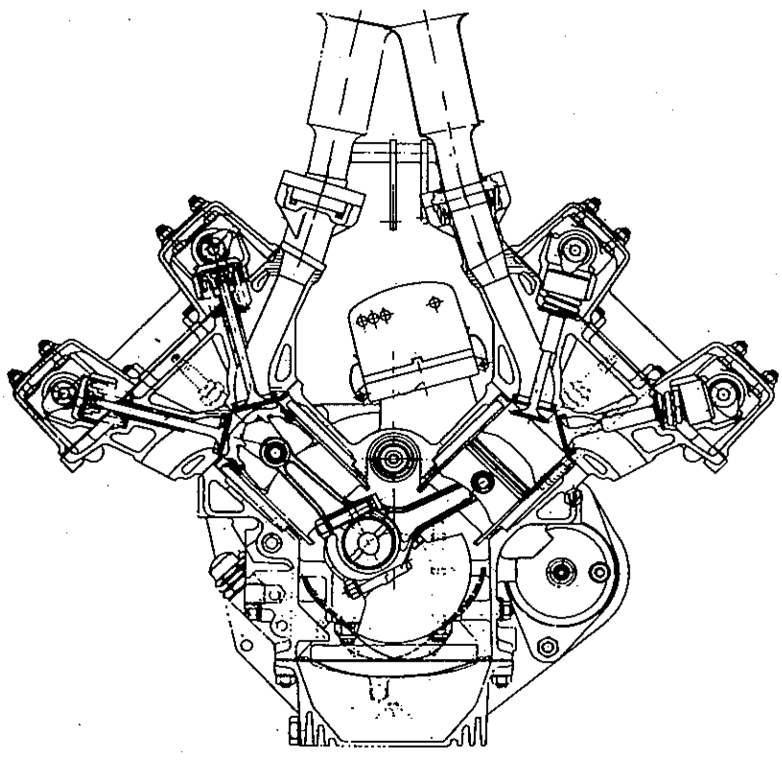


Fig. 14 - Cross-section of V-8 showing connecting rod path and oil antisurge baffle

4 cyl engines, where the two inner and the two outer exhaust ports were paired and then joined together nearer to the tail pipe, could be improved upon if all four were arranged to meet at a common junction, and thence into a divergent tail pipe or megaphone.

The original V-8 crankshaft was of the conventional 90 deg phased type, first popularized by Henry Ford, and later standardized by all V-8 manufacturers. This arrangement results in unavoidable and unequal phasing of the firing strokes, so that it was only possible on each bank to obtain the much desired condition of even exhaust pulses into common tail pipes by coupling the center pairs of cylinders on one bank to the outer pairs of the opposite bank, and vice versa.

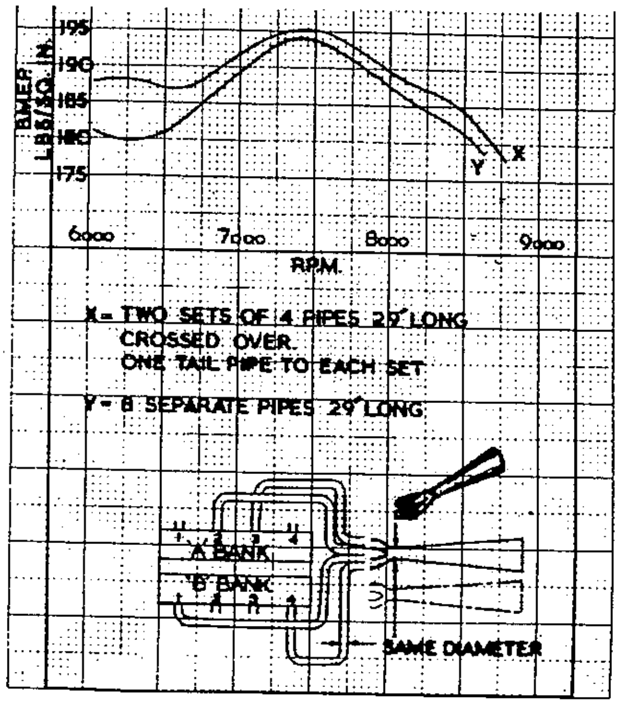


Fig. 16 - Exhaust systems for 90 deg phased crankshafts

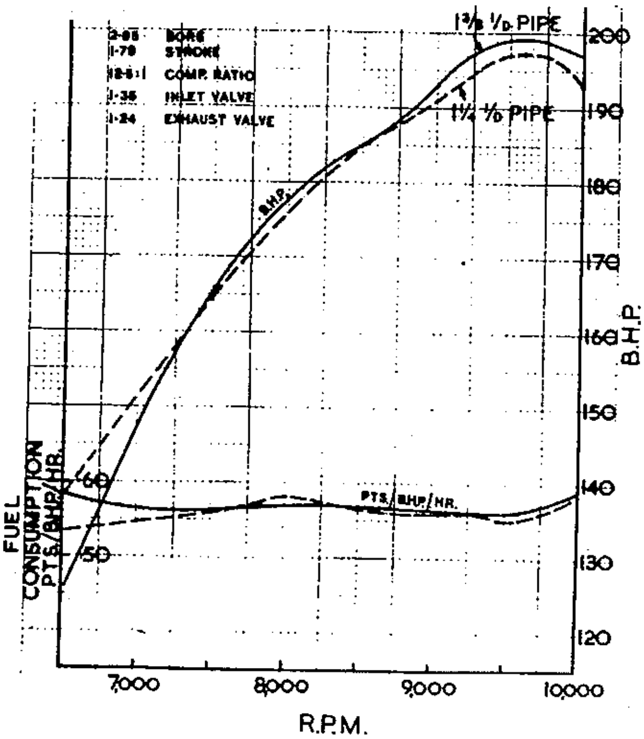


Fig. 17 - Effect on performance of change of exhaust pipe diameter

It was fortunate that the optimum primary pipe length was just sufficient for this crossing over to be possible, although it was necessary to utilize some exceedingly tight bends to do so. It does appear that tight bends are not detrimental to power output, as was popularly thought at the time. This complicated "cross-over" layout became known as "spaghetti" and upon its first appearance was received with some derision but subsequently, it has been used by most constructors -- including Ford at Indianapolis (Fig. 16).

When in 1964 our engine speeds and output had risen, it was found that an increase in exhaust pipe diameter was beneficial. Both primary pipes and the junction cluster was increased from 1-1/4 in. ID to 1-3/8 in. ID, as a result of which a loss occurred at 6500 rpm. However, at 8000 rpm a small gain was apparent and, at 10,000 rpm, the bmep held up and provided another small gain at this point (Fig. 17).

INFLUENCE OF EXHAUST SYSTEM ON CRANKSHAFT DESIGN

Ferguson wished to consider the use of our engine in his 4 wheel drive racing car and, since the engine was mounted at the front, it was impossible to accommodate the cross-over exhaust because of the driver's legs. It was obvious that the exhausts had to be arranged so that each bank of cylinders could use separate systems at either side (Fig. 18).

In order to achieve even firing down each bank, it was necessary to use a 180 deg phased crankshaft, but it was thought that a vibration problem would arise due to the increased secondary vibrations which, in theory, would result.

To our surprise the vibration of the engine was not noticeably different to that fitted with the 90 deg phased crankshaft. There was a small improvement in bearing loading, and a slight reduction in weight, due to the simplified system of balance weights.

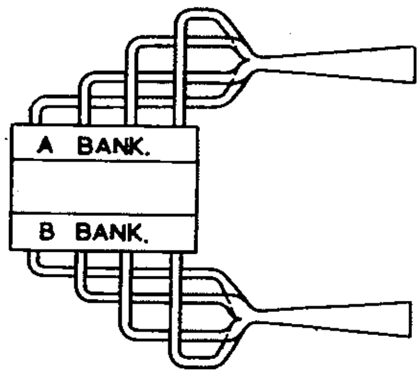


Fig. 18 - Exhaust system for 180 deg phased crankshaft

The exhaust system was much improved from the installation point of view, and the 180 deg crankshaft was adopted for all later engines.

GENERAL DEVELOPMENT

Our development followed two parallel courses -- one for improved power, and the other for reliability. Active participation in racing together with continuous development testing brought to light many weaknesses.

As noted previously our approach was conservative, the bore and stroke of the Mk. I 1961-62 engine being 2.48 × 2.36 in., respectively and, initially, we obtained 174 bhp at 8500 rpm. Later this was improved to 181 bhp.

Four Weber DCLN4 double choke downdraft carburetors, fitted with 32 mm choke tubes, were used for 1961 and 1962. These were not altogether suitable for the V-8, since they were installed across the Vee, the float chambers were considerably ahead of the jets and choke tubes, and this brought tuning difficulties especially when accelerating out of corners at part throttle. The over-richness under these conditions led to a tendency for tuners to weaken the overall mixture strength to a somewhat dangerous degree, a condition not overcome until fuel injection was fitted in 1963. (See Fig. 40.)

FUEL INJECTION

In 1963 the Lucas fuel injection system was fitted. This made it possible to achieve really clean acceleration out of a bend, and was a most worth-while modification. For some time it was not possible to achieve the same maximum power as that produced by carburetors but, eventually, an intake system was evolved which gave practically similar results. This consisted of slide throttles, with the discharge nozzles upstream -- the orifices being almost in line with the flared end of the true air intake. In operation this arrangement produced quite severe fuel blowback, and a secondary air intake of much larger size was added, surrounding and supporting the nozzle, thus completely overcoming the problem (Fig. 19).

The drive for the metering unit was arranged by moving the coolant pump forward, to enable a pulley to be mounted on the forward end of the half speed jackshaft, just behind the pump. A cog-toothed belt was used, and was quite satisfactory. A magnesium casting carried the metering unit low down between the cylinder heads, with the driving spindle and pulley projecting across the timing case.

Fig. 20 shows the system diagrammatically. A motor driven fuel pump provides a constant supply of petrol, at a substantial pressure, to the metering unit. Downstream of this unit is a relief valve which keeps the pressure, for metering, at 100 psi. At this pressure, vapor formation is effectively suppressed, and metering accuracy is maintained even at the high operating temperature reached by the unit as a result of its location in the center of the V of the engine.

Control over the fuel delivery is provided by a direct me-

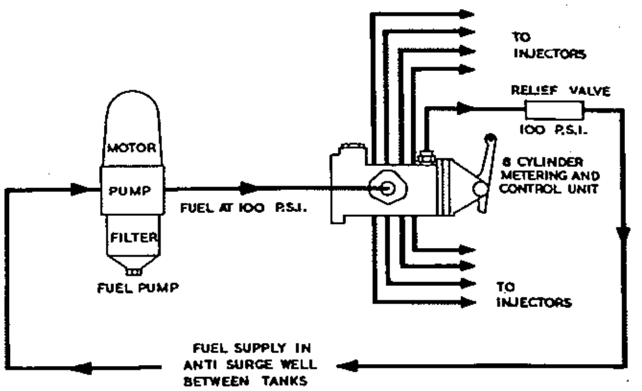
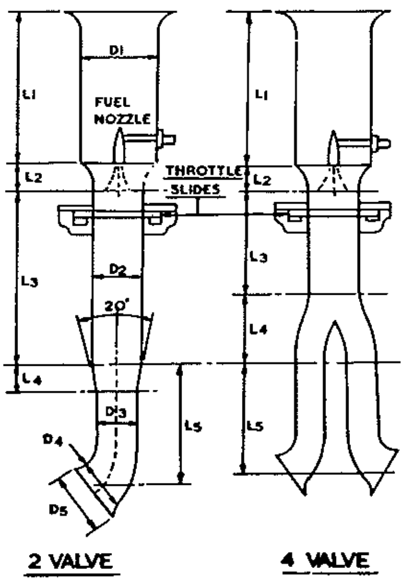


Fig. 20 - Diagrammatic arrangement of Lucas fuel injection system



NO OF CYLS	8				16
NO OF VALVES/CYL	2		4		2
DIAM OF INLET VALVE	1·35″	1·4″	1·04″	1·107″	1·07″
D1 AIRBELL	1·875″				
D2 IND STUB	1·25″				1·0″
D3 PORT	1·015″	1·03″	·75″	·783″	·8″
D4 THROAT	1·175″	1·22″	·895″	·946″	·95″
D5 MIN SEAT	1·266″	1·316″	·970″	1·037″	1·02″
L1	3·75″				3625″
L2	·375″				·625″
L3	4·125″		2·5″	2·69″	1·375″
L4	·625″		2·062″	2·156″	·50″
L5	4·25″		·315″		1·25″

Fig. 19 - Induction system for fuel injection

chanical linkage between the slide throttle plates, and a lever carrying a cam profile on the metering unit, so that for any throttle opening there is an appropriate fuel delivery to each cylinder.

Fig. 21 shows an 8 cyl metering unit with the lever control. The metering unit contains a twin-bored rotor, each bore containing a single shuttle which moves axially under fuel pressures between two stops; one fixed and the other adjustable by the cam profile. The rotor is driven at half engine speed by the splined drive shaft. The porting in the rotor and the sleeve in which it rotates are arranged so that fuel pressure from the supply pump drives the shuttles backwards and forward between the two stops. The distance traveled between the two stops determines the quantity of fuel discharged during each injection. This fuel is fed, through nylon pipes, to individual injectors mounted in the center of the bell-mouthed intake tubes. The system was developed to operate at speeds well over 10,000 engine rpm and, at this speed, each individual shuttle moves over 300 times a second.

Earlier units incorporated a separate oil pump as shown on Fig. 21, but the development of materials suitable for high speed running in petrol, have made it possible to achieve a satisfactory racing service from units, modified in detail design, and simplified by the omission of this oil pump.

Initially, the electrically driven, high pressure, fuel pump caused some trouble, due to overheating, when the unit was mounted at the rear of the power unit adjacent to the gearbox. The best position was found to be ahead of the radiator, right in the airflow, despite the lengthy fuel pipes required by this location.

Another problem was due to choking of the internal fuel fine filters, which would load up with dirt from the churns and funnels used by the teams, and resulted in a gradual but almost complete stoppage of fuel. In particular, this was the case with the small felt filter pad fitted in the casing of the metering unit. It was very difficult to diagnose because, when it dried off it was quite clear; only when it was wet and under pressure would it close up and so prevent the free flow of fuel. Being inaccessible, it was not easy to service, so another, much larger filter, which could be serviced easily, was fitted in the pipe line to the metering unit -- thus overcoming the trouble. Several races were lost the first two years before the matter was finally cleared up.

There is a fundamental difficulty in tailoring such a fuel control system to an engine in which induction ram plays an important part in obtaining high performance.

The fuel control arrangement provides a precise delivery of fuel, determined by the linked cam and throttle, and this is injected at every complete cycle, due to the positive half speed drive to the metering unit.

The control system is therefore based on air consumption being pro rata with rpm. In fact, this is not so for, due to ram effect, the air consumption at the point of maximum torque can exceed the theoretical capacity of the engine, whereas at speeds above and below this point the volumetric efficiency is reduced by varying amounts.

If the injection system is tuned to give a sufficiently rich mixture at maximum torque, it obviously goes richer still when the proportion of air induced is reduced. This enrichment is of little consequence at the lower speeds because acceleration is occurring in this range, but it does cause over-rich conditions at speeds of maximum power.

We struggled for some time with automatic leaning-off devices, aimed at correcting the mixture above maximum torque, but the problem was never satisfactorily solved and,

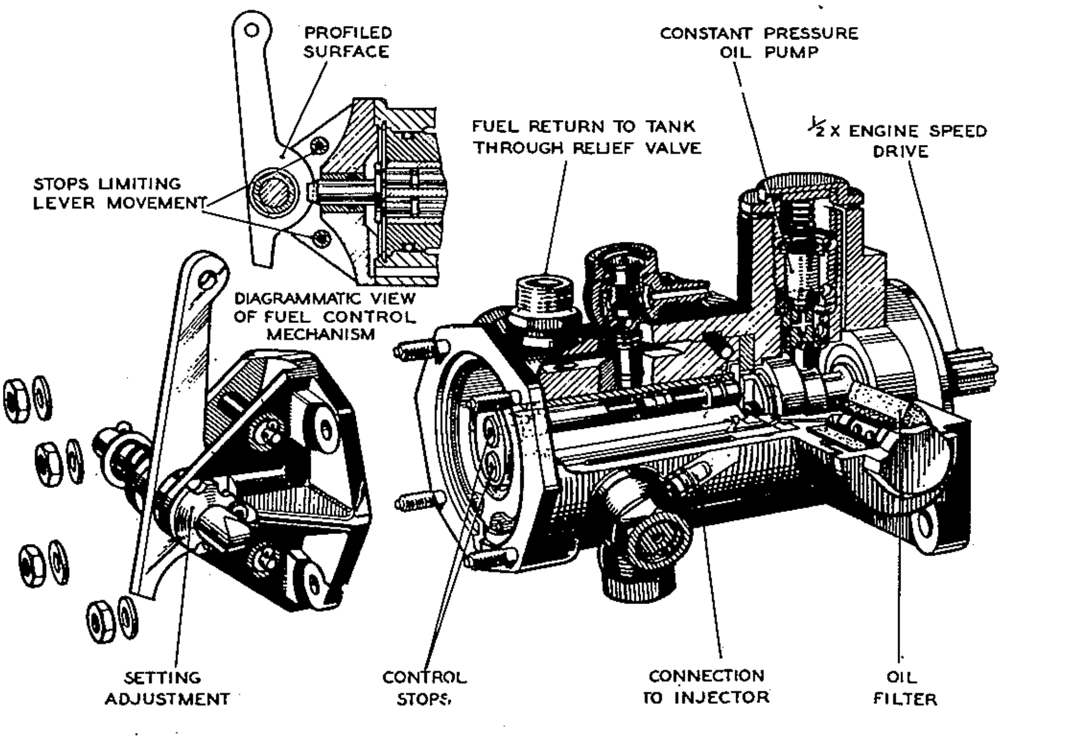


Fig. 21 - Lucas 8 cyl metering unit

in all races, we ran in a condition of a 3% loss of maximum power. This was regrettable but, perhaps, in view of the results we achieved in the races, it serves to show the relative unimportance of actual maximum power compared with middle range torque.

INCREASE OF OPERATION SPEEDS AND BHP

Using a given fuel and a naturally aspirated engine, the only manner in which more power can be acquired is to raise the crankshaft speed, assuming a reasonable optimum with regard to volumetric and mechanical efficiency.

Therefore, in 1963, we sought to raise the crankshaft speed and, in order to do this safely, we did not wish to exceed the piston speeds and maximum acceleration of the previous engine.

The stroke was reduced to 2.03 in. which gave us a mean piston speed of 3300 ft/minute at 9600 rpm, that is, 1100 rpm higher than in 1962, together with a maximum piston acceleration of 100,300 fps^2 In order to assist in keeping this figure as low as possible, the connecting rod center distance was increased from 4.2 to 5.1 in. giving a rod to stroke ratio of 2.5:1 -- that of the original engine being 1.78:1. This arrangement gave peak power at 9500 rpm in its earliest form and, eventually, at 9800 rpm.

In order to achieve an increase of power at the higher crankshaft speeds, the diameter of inlet valve was increased from 1.3 to 1.35 in. and the port dimensions were modified

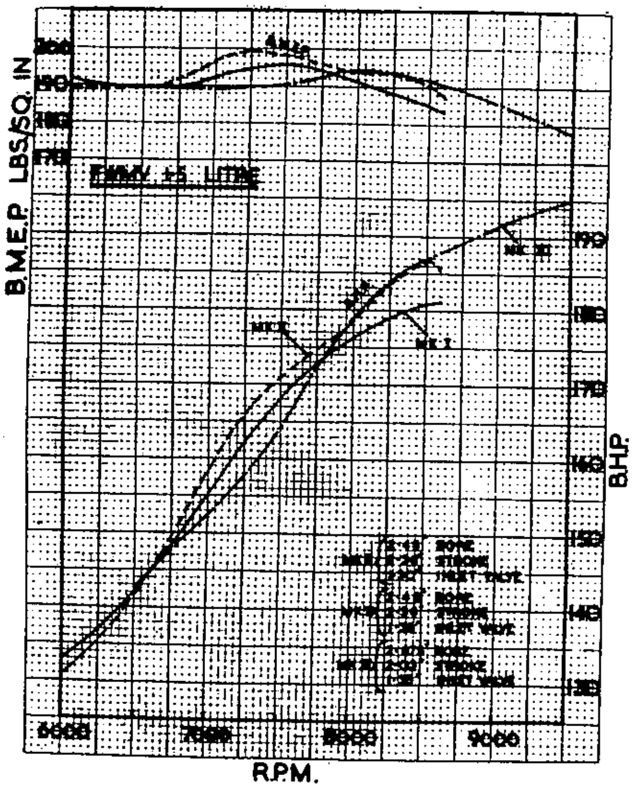


Fig. 22 - Performance of Mk.I, II, and III 1.5 liter V-8 engines

to suit. This resulted in 195 bhp at 9600 rpm, the maximum torque moving from 7600 rpm to 8000 rpm.

We modified a 2.36 in. stroke engine to the new inlet valve condition and this gave 186 bhp -- an increase of 5 bhp or 3.6%.

The gain due to (a) change of valve, and (b) change of stroke, may be roughly assessed by comparison of the following: (See Fig. 22.)

MK.I 1.3 in. inlet valve, 2.36 in. stroke = 181 bhp at 8500 rpm.

MK.II 1.35 in. inlet valve, 2.36 in. stroke = 187 bhp at 8500 rpm.

MK.III 1.35 in. inlet valve, 2.03 in. stroke = 195 bhp at 9500 rpm.

There were some small changes to the inlet system which gave minor improvements. The inlet port was enlarged from 1.015 to 1.025 in. diameter and was provided with a 20 deg included taper entry which reduced the effective length of the restricted diameter by approximately 1 in. The diameter of the large end of this tapered entry was 1.312 in. and this increase was extended to the throttle body diameter and ram pipe.

For 1964, a further shortening of the stroke was effected, primarily in aid of the four valves per cylinder engine which was designed with the idea of running up to perhaps 11,000 rpm. The four valve arrangement did not function too well in its early stages, and this is dealt with in a later part of the paper.

Having the necessary components on hand to assemble the ultrashort stroke as it was known, we felt obliged to test it out on existing two valve cylinder heads. The stroke was actually shortened to 1.79 in. and the bores increased to 2.85 in. This combination gave a further increase of rpm to 9750 using existing valve gear for a mean piston speed of only 2910 ft/minute.

Peak power was obtained at 9750 rpm, at which speed we considered the existing valve arrangement to be satisfactory. A further increase in inlet valve diameter to 1.4 in., together with a modified port, was made with some apprehension on our part for, while the engine was very willing to "rev," we were not really happy for the valve gear to run faster -- particularly with the larger and heavier valve.

However, this engine was very successful in Jim Clark's capable hands and gave an increase of power just exceeding the 200 bhp figure at 9750 rpm (Fig. 23).

We made several engines with 1.35 in. inlet valves and 1.79 in. stroke, using 4.2 in. center connecting rods which, even though they did not produce a worth-while improvement in performance, did give a lower and smaller profile, thus allowing a reduction in the cross-section of the bodywork at the engine bay. It is probable that the reduction in drag was worth more than 5-6 bhp at the higher road speeds.

FOUR VALVE DEVELOPMENTS

When it appeared necessary to increase our power beyond the 195 bhp obtained in 1963, we naturally thought in terms of more valves of smaller dimension and weight. Therefore a four valve design of conventional "pent-roof" form was put in hand.

The performance for the first 12 months was poor and never looked like even equalling that of the two valve engines. The valve and port dimensions were based on our previous work, and should have produced results.

The trouble appeared to be one of combustion, and was thought to be due to the effect of two equally flowing ports cancelling out any swirl which might have been present which, together with the shape of the pent roof piston and combustion chamber forming a barrier across the chamber, produced a peculiarly stagnant gas condition. Apparently, the squish areas on either side were insufficient to enliven matters.

The exhaust condition indicated a very slow combustion, and only by advancing the ignition point 10 deg earlier than on the two valve engine, did the engine commence to function.

We prepared two pairs of cylinder heads, one with valve and ports slightly larger than the other and, not unexpectedly, the smaller valved head proved better overall, although the maximum power obtained was rather greater from the large valve engine.

With the small valve engine, as run in 1965, the maximum power on the test bed was improved to 210 bph at 10,500 rpm. The higher rpm permitted slightly lower gear ratios to be used and this, together with a small improvement in the engine torque curve, produced enhanced acceleration.

The large valve engine gave 213 bhp at 10,500 rpm with a maximum bmep of 196.5 psi at 9000 rpm. This version was potentially better but, in Jack Brabham's car, did not perform very well. (See Fig. 24.)

The four valve camshaft drive caused difficulty and, whereas with the twin cam engine the single strand 3/8 in. pitch chains were trouble free, on the four valve engine they failed in a very short time. 1/2 in. pitch chains were tried, and these also failed. In order to overcome the trouble, a gear drive was devised and this worked quite well for a time, after which the gear carrier cracked the cylinder head walls to which it was attached (Figs. 25 and 26). It would seem that the trouble was due to the more pronounced torque peaks imposed by the double valve loading. For the higher operating speeds expected, the same valve spring loading was used as for the two valve engines.

The last difficulty was associated with the method employed to locate the gear carrier, which included bolting and the use of "Araldite" adhesive to provide suitable attachment lugs for the gear drive using the existing cylinder heads designed for chain drive (Fig. 27). This expedient was the result of a complete lack of time to obtain new cylinder head castings incorporating the necessary attachment bosses, and to machine them for the remaining races of the season.

We are now of the opinion that the valve angles chosen were too wide. It is considered that the combustion chamber would have been more efficient had the valves been ar-

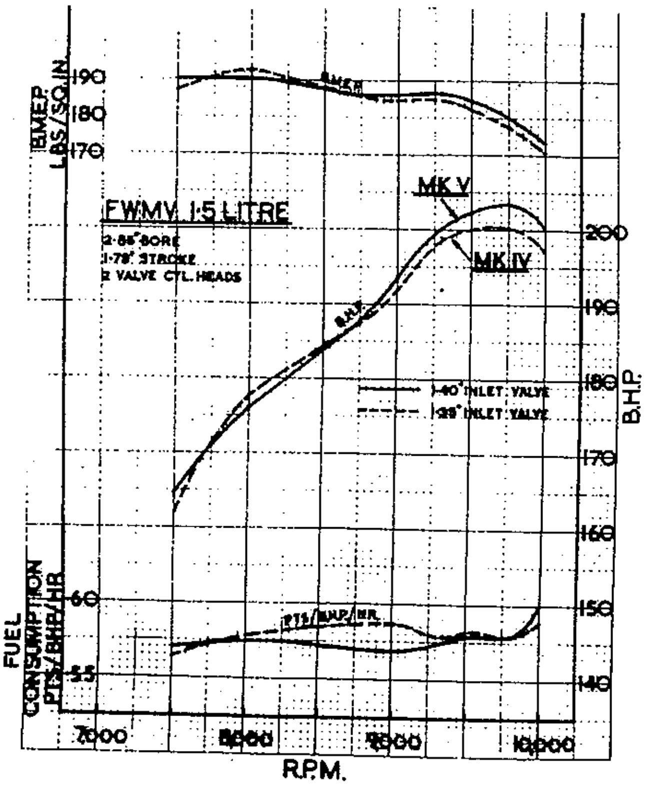


Fig. 23 - Performance of Mk. IV and V 1.5 liter V-8 engines

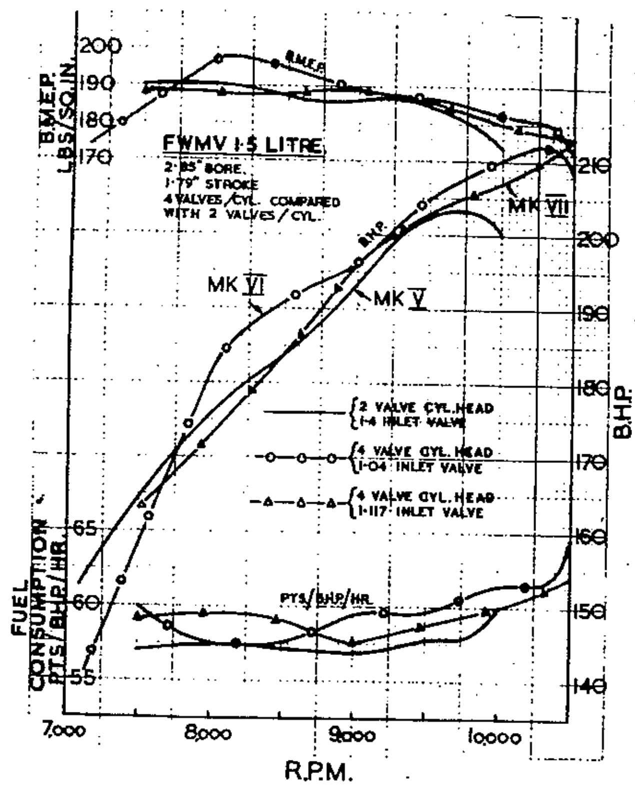


Fig. 24 - Performance of Mk. V, VI, and VII 1.5 liter four valve/ cylinder engines

ranged at a smaller included angle, thus enabling a flatter piston crown to be used. It does appear that swirl is important and, because it is gained only to the detriment of volumetric efficiency, it must not be spoilt by an exaggerated pent-roof to the piston.

One great gain with a four valve design, apart from the obvious reduction in valve gear inertia, is the improved airflow. It is noticeable that if two valves and ports are flowed into a cylinder, the combined airflow is considerably greater than the sum of identical valves and ports flowed separately. We have not decided why this should be, but it is probable that the energy loss is practically the same for one stream as for two streams flowing into one cylinder.

Figs. 28 and 29 show airflow measurement for single port and double port flow.

There remained a great deal of scope for improvement at the time the formula was changed, at which point our company decided to "opt-out" of racing, but this four valve unit undoubtedly assisted Jim Clark and his Lotus to win the last World Championship of the 1-1/2 liter formula.

2 LITER ENGINE

One of these engines was produced at the end of 1965, and was used by M. Spence to win the 1966 South African Grand Prix. It will probably be used by Lotus in the current Formula I races until the 3 liter B.R.M. engines become available.

The increased capacity was achieved by lengthening the stroke to 2.36 in., as used in 1962, and retaining the 2.85 in. bore more recently used on 1.5 liter engines. The longer stroke imposed a limit on rpm which was set at a maximum of 9000 rpm.

As sent to South Africa, it was fitted with the two valve cylinder heads, incorporating the normal 1.35 in. inlet valve, together with the induction system and ports as used on the 1.5 liter engine. 239 bhp was obtained at 8800 rpm. Subsequently, we prepared some cylinder heads and pistons to use the larger 1.4 in. diameter inlet valve, together with some increase in port dimension, and 244 bhp was produced at 8900 rpm.

In order to minimize the maximum piston acceleration

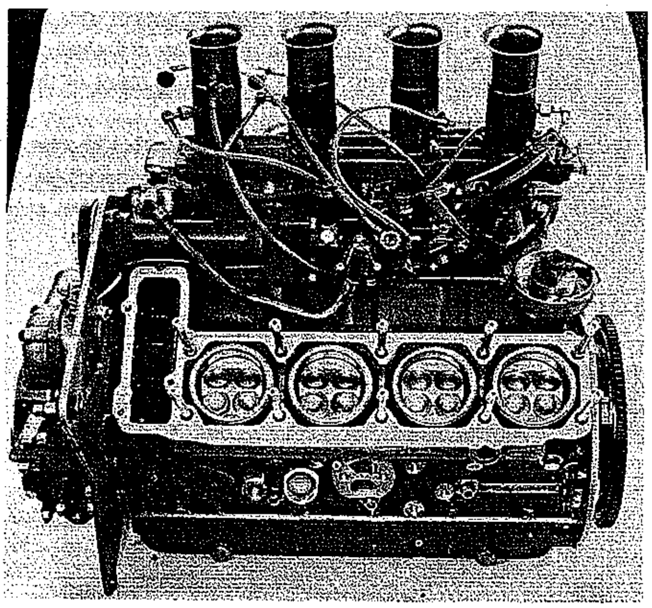


Fig. 25 - Four valve engine with pistons exposed

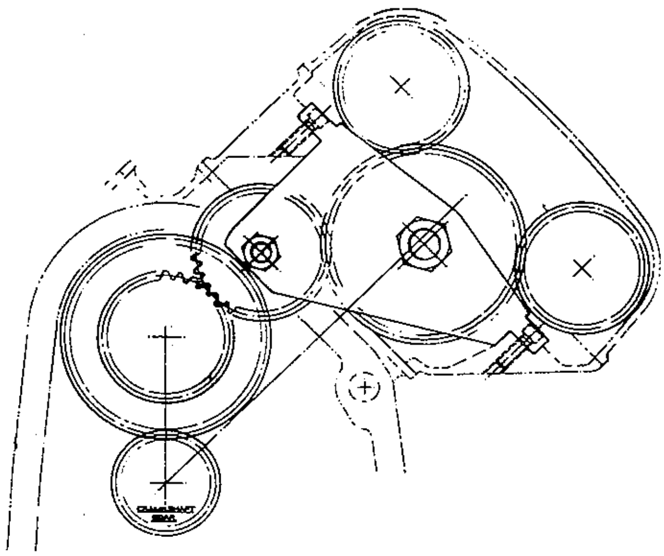


Fig. 27 - Gear drive for four valve engine

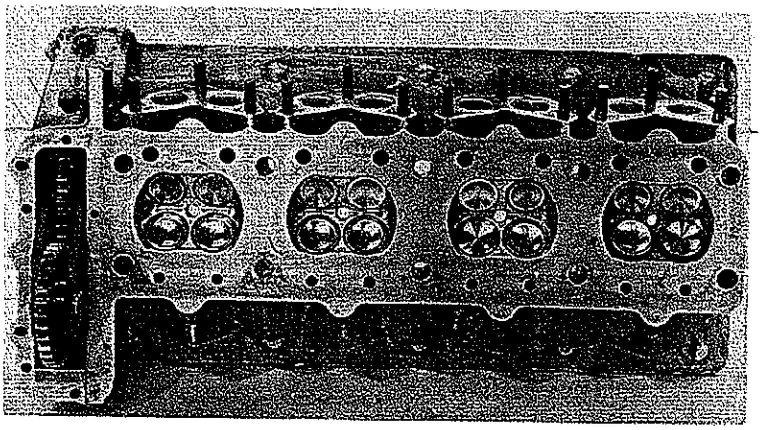


Fig. 26 - Four valve cylinder head and combustion chamber

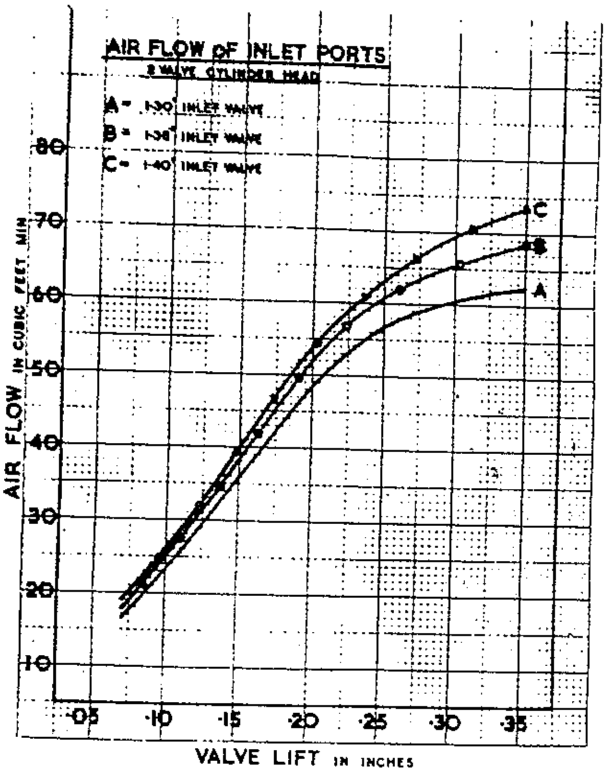


Fig. 28 - Airflow for two valve/cylinder

and, therefore, the inertia effects due to the use of the longer stroke, we reverted to the 5.1 in. center connecting rods instead of the 4.2 in. used latterly. Otherwise the engine was exactly similar to the 1-1/2 liter two valve engines.

The power is, of course, limited by the valve dimensions, as well as by the permissible rpm, and the gas mean velocities are extremely high. (See Table 1.) However, this has given the engine good torque in the middle range, and it will prove a useful "stop-gap" while the new engines are finalized by B.R.M. The performance of this engine, when fitted with 1.35 and 1.4 in. diameter inlet valves, together with that of the 1.5 liter engine using the 1.4 in. inlet valve and, for comparison, that of the F.P.F. 1960 cc 4 cyl, is shown in Fig. 30.

The comparison between the 1-1/2 liter and 2 liter engines, using the 1.4 in. valves, is interesting since they are of identical specification except for the change of stroke.

16 CYL ENGINE

This engine, although never raced, is considered, nevertheless, to be of interest as a design exercise and, therefore, has been included.

During 1963-1964, when the four valve engines were undergoing initial tests, the results were so disappointing that another line of attack was considered necessary, and the 16 cyl project was the result (Fig. 31).

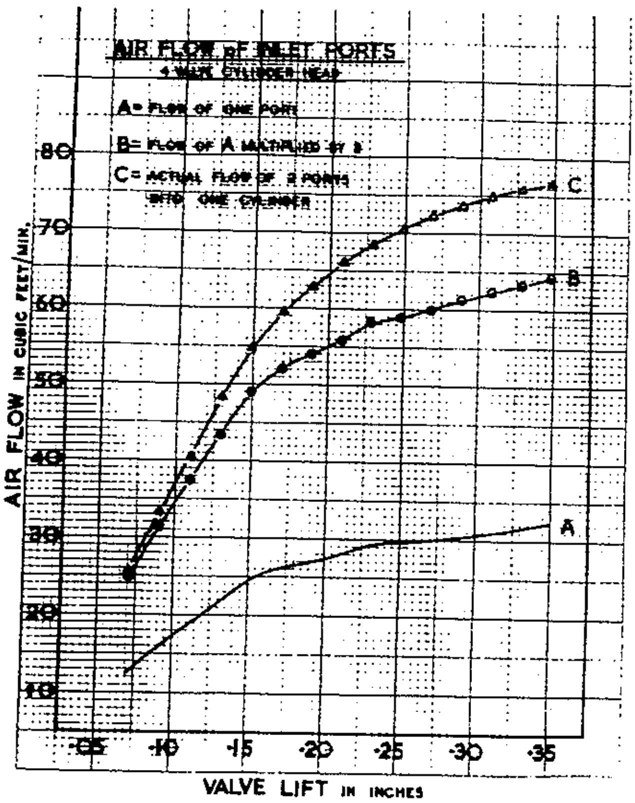


Fig. 29 - Airflow for four valve/cylinder

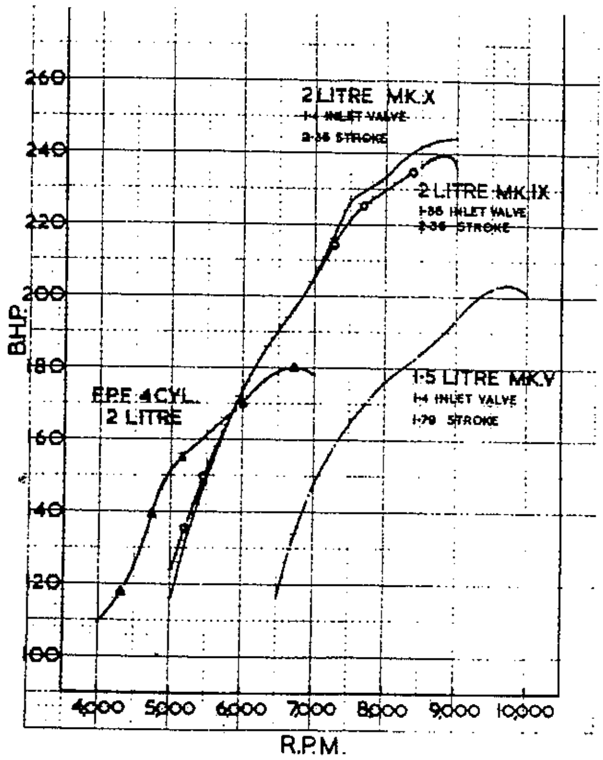


Fig. 30 - Performance compared Mk. V, IX, and X 2 liter V-8 engines

REASONS FOR ADOPTING FLAT 16 CYL DESIGN FOR 1965 1.5 LITER RACING ENGINES

The current 1-1/2 liter formula had been running since 1961 and several progressive changes made, whereby the performance improved from 150 bhp at 7500 rpm on the 4 cyl F.P.F. to 174 bhp initially on the V-8, and thence, in two stages, to just over 200 bph at 10,000 rpm.

At the end of 1963 it was apparent that Ferrari and B.R.M. were catching up, and that more than just a few bhp were required to retain the lead we had at that time. It was considered that the only worth-while improvement would be obtained by an increase in rpm, and two projects were initiated for this purpose.

The first was to decrease further the stroke of the current V-8 and to use the four valve arrangement in lieu of the two valve set-up. As mentioned previously, extremely disappointing initial results were obtained. The second was to

Table 1 - Inlet and Exhaust Gas Velocities, fps

Engine Type	Inlet at Max. bhp Port	Throat	Min. Seat	Inlet at Max. Torque Port	Throat	Min. Seat	Exhaust at Max. bhp Port	Throat	Min. Seat
Mk. I 2.48 × 2.36 in. 1.3 in. inlet valve	333	289.5	239.5	294	255.5	211.3	343.5	326.5	274.8
		At 8500 rpm			At 7500 rpm			At 8500 rpm	
Mk. II 2.48 × 2.36 in. 1.35 in. inlet valve	327	252.5	231.5	288.6	222.8	204.4	343.5	326.5	274.8
		At 8500 rpm			At 7500 rpm			At 8500 rpm	
Mk. III 2.675 × 2.03 in 1.35 in. inlet valve	370	285.5	262	307	237.4	218	388.5	370	310.5
		At 9600 rpm			At 8000 rpm			At 9600 rpm	
Mk. IV 2.850 × 1.79 in. 1.35 in. inlet valve	375	289.5	266	308	237.5	218	394	375	315
		At 9750 rpm			At 8000 rpm			At 9750 rpm	
Mk. V 2.850 × 1.79 in. 1.4 in. inlet valve	369.5	289.5	236.8	304	237.5	194.6	394	375	315
		At 9750 rpm			At 8000 rpm			At 9750 rpm	
Mk. VI 4 valve 1.04 in. inlet valve	370	282	237	305.5	233	196	281.4	366	299.5
		At 10,300 rpm			At 8500 rpm			At 10,300 rpm	
Mk. VII 4 valve 1.107 in. inlet valve	354	260	213	289	212.5	174.4	290.5	291.5	252
		At 10,700 rpm			At 8750 rpm			At 10,700 rpm	
Mk. VIII 2 liter 1.35 in. inlet valve	446	345	316.5	381	294	270	469	447	375
		At 8800 rpm			At 7500 rpm			At 8800 rpm	
Mk. IX 2 liter 1.4 in. inlet valve	415	337	281	350	284.5	237	474	452	380
		At 8900 rpm			At 7500 rpm			At 8900 rpm	
16 cyl 1-5 liter	378.5	289	258	284	216.5	193.5	314	338	307
		At 12,000 rpm			At 9000 rpm			At 12,000 rpm	

design the 16 cyl engine which we had hoped to have running for the latter part of 1964.

The 16 cyl engine was chosen as offering a greater potential in terms of rpm than the 8 cyl or a possible 12 cyl. In terms of piston speed, and using the stroke to bore ratio seemingly proved best from previous results -- that is, 0.76:1; the 8, 12, and 16 cyl gives the comparison in crank speed shown in Table 2.

The maximum piston acceleration is rather high at 13,000 rpm and we would have been satisfied, initially, with say, 12,000 rpm.

Piston area has been used by many as a criterion of performance and using this factor we obtain the following:

Nos. of Cylinders & Bore, in. 8 × 2.675, 12 × 2.33, 16 × 2.13.

Total Piston Area, sq in. 45, 52, 56.6.

If we take previous results we have achieved:

FPF 4 cyl, 3.22 in. bore, 150 bhp from 36.2 $in.^2$ = 4.6 bhp/sq in.

V-8 2.48 in. bore, 186 bhp from 38.6 $in.^2$ = 4.82 bhp/sq in.

V-8 2.675 in. bore, 202 bhp from 45 $in.^2$ = 4.5 bhp/sq in.

So that we could expect approximately 4.5 bhp/sq in. of piston area:

12 cyl with 52 sq in. × 4.5 = 230 bhp approximately.

16 cyl with 56.6 sq in. × 4.5 = 250 bhp approximately.

If we assess power on the score of rpm with similar bmep as currently obtained we arrive at:

200 bhp for 10,000, 220 for 11,000, 240 for 12,000 rpm, and so on.

These figures are obviously very optimistic but show the trend, and it is certain that for a given rpm, the 16 cyl will be slightly less stressed than the 12 cyl.

One other factor influenced our choice, and this was the design of the crankshaft. The form chosen was, in fact, two cranks exactly as we had previously used successfully in the V-8, both being shrunk onto a common gear, each shaft being phased at 90 deg to the other, and each being the simple flat crank as in an in-line 4 cyl. This formation had given us excellent bearing conditions together with very smooth running.

Previously, we had set out 12 cyl "Vee" and 16 cyl "Vee" engines, the latter with the conventional 135 deg angle between the banks. The 12 cyl was appreciably larger in cross-section, and the V-16, with the cylinders sloping upwards with power take-off below, set the center of gravity higher than we wished -- this shaft height being dictated by the existing form of transmission and the tire centers (Fig. 32).

We tried drooping the cylinders and this provided an excellent layout with very low center of gravity. Unfortunately, this scheme left no space for the lower frame tubes or for the exhaust system (Fig. 33).

A compromise was struck with the cylinders set horizontally, that is, a flat engine. With this layout we obtained a low center of gravity and a very small silhouette.

This scheme also allowed ample space, below the cylinders, for both exhausts and also for the lower chassis tubular members.

A further point in favor of the 16 cyl was that we obtained a firing sequence which made possible the close coupling of each of the series of 4 cyl, on either side of the en-

Table 2 - Comparison in Crank Speed

	8 Cyl	12 Cyl	16 Cyl
Mean Piston Speed	2.03 in. stroke	1.77 in. stroke	1.6 in. stroke
3500 ft/minute	10,340 rpm	11,870 rpm	13,100 rpm
Maximum Piston Acceleration, fps^2	123,900	142,400	156,800

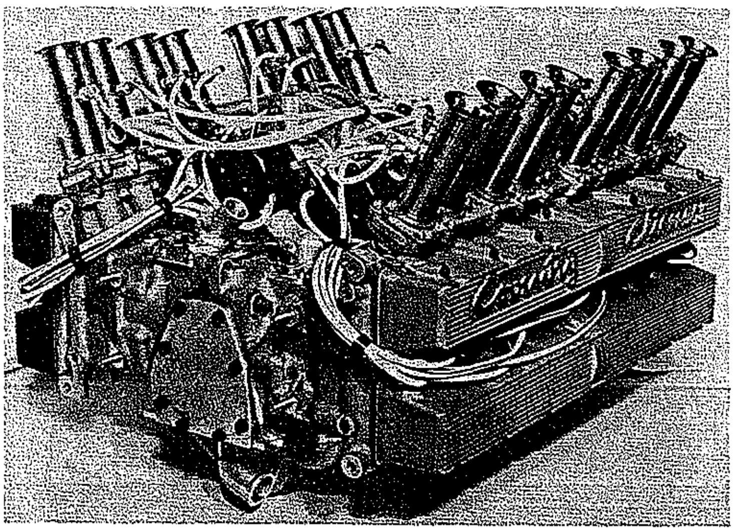


Fig. 31 - Sixteen cylinder engine

gine, so that we had four separate groups, each comprising a simple 4 cyl system. The correct coupling of exhaust ports makes possible a higher maximum power and also, as previously noted, a considerable spread of useful torque.

The separate power take-off, or drive shaft, solved several problems; namely:

1. The clutch speed is reduced and, for a crank speed of 12,000 rpm, the clutch turns at 9600 rpm so that existing transmission ratios are satisfactory.

2. The torsional vibration problem, with the rather long crankshaft, is minimized by taking the drive from the center.

3. The assembly of two simple crankshafts, by shrink fit with the drive gear, simplifies the manufacture of an otherwise difficult component.

CRANKSHAFT ASSEMBLY USING S.K.F. OIL INJECTION PROCESS

The S.K.F. process was used to join the two crankshafts together, and was originally developed by the S.K.F. Ball Bearing Co. of Sweden to facilitate the use of heavy roller races on marine crankshafts and rolling mills. In our case, the crankshafts were arranged with female tapers of approximately 1 in 50 engaging with the similarly tapered extensions of the power take-off gear. The dimensions were chosen to give a diametrical interference of 0.0035 in. with a pull-up distance of 0.175 in. Oil was injected through suitable drillings by means of a small hand pump, so that sufficient expansion occurred to allow the components to be pulled together by suitable clamping means. When the oil pressure was released, the components shrank together and the oil was forced away from the mating faces (Fig. 34).

The oil pressure required was around 15,600 psi, and the fits were calculated to withstand three times the maximum torque expected to occur at the joint. The joint could be broken and remade as often as required by simply injecting oil, when the components were automatically forced apart. Suitable stops were obviously required and, in our case, a lathe tail-stock sufficed. The joint was quite satisfactory in service, although we did find the interference became reduced after dismantling, obviously due to a small deficiency in hoop strength of the crankshaft ends. Probably the elastic limit was exceeded. We think the scheme offers an elegant solution to a normally difficult problem in a cheap and simple manner.

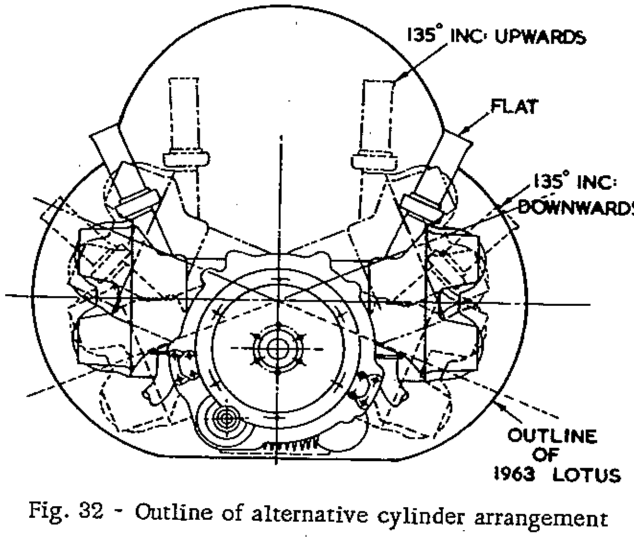


Fig. 32 - Outline of alternative cylinder arrangement

COVENTRY CLIMAX FLAT 16 CYL 1500 CC RACING ENGINE TYPE 'FWMW'

Bore -- 2.13 in.
Stroke -- 1.6 in.
Piston area -- 3.563 sq in.
Rpm at maximum power -- 11,000-12,000.
Piston acceleration at maximum power -- 110,500-131,400 fps^2
Piston mean speed at maximum power -- 3060-3200 ft/minute.
Power estimated -- 220 bhp plus.
(See Fig. 35.)

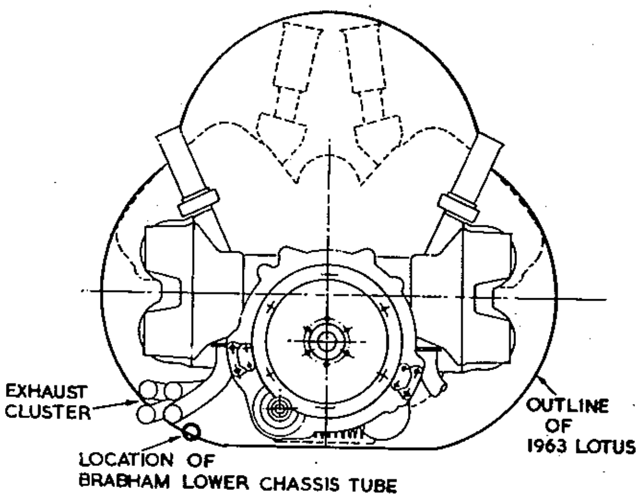


Fig. 33 - Outline of 16 cyl horizontal engine superimposed on V-8

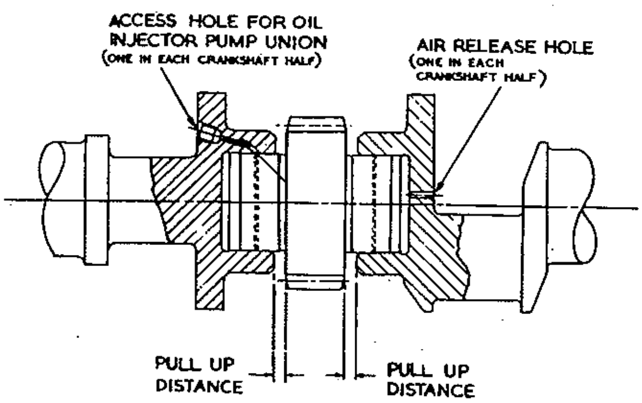


Fig. 34 - Assembly of 16 cyl crankshaft - oil injection method

The crank system consists of two separate cranks shrunk onto, and on either side of, a central spur gear from which the drive is taken for traction and auxiliaries. The cranks are phased at 90 deg to each other, each crank having the normal five bearings with four crank pins, Nos. 1 and 4 at 180 deg to Nos. 2 and 3, as with a normal 4 cyl in-line crankshaft. There is no flywheel on the crank system -- only a clutch carrier on the drive shaft, situated below the crankshaft, which is driven from the central crank gear. The crank arrangement results in two cylinders firing simultaneously and, therefore, the torque impulses are similar to a V-8 of similar capacity.

The arrangement was chosen in order to obtain the relatively simple exhaust layout, consisting of four separate 4 cyl systems, in each of which there are four firing strokes phased 180 deg apart, using the following firing sequence:

1 and 6, 11 and 16, 3 and 8, 9 and 14, 7 and 4, 13 and 10, 5 and 2, 15 and 12.

The cylinders are numbered from the front, left to right, so that the odd numbers are to the left and even numbers to the right looking down on the engine and to the rear of the car.

The crankcase is split vertically along the centerline of the crankshaft with the studs for the main bearings in one half, and the nuts on the exterior of the other (Fig. 36).

The auxiliary drive is by a straight-spur gear taken upwards from the central gear of 24 teeth to an idler gear of 32 teeth, which is carried in a housing, held by long studs, on the center of the top face of the crankcase. This housing is open at the sides, the facings coinciding with those for the cylinder heads so that the heads complete the closure.

The gear housing contains the 32 teeth central gear, which meshes with that on the crank, and also meshes with two 48 teeth gears set either side and slightly higher up. A 32 teeth gear is attached to each, to provide a compound drive to the gear trains in the cylinder heads.

Drives are taken from these pairs of gears to the Lucas alternator and the 8 cyl ignition distributor, mounted in tandem on the right hand side looking rearwards, and to another similar distributor on the left hand side. On the rear faces of the gear housing are attached two Lucas fuel injection 8 cyl metering units, also driven from these same gears.

The cylinder heads are arranged in pairs so that the gear train, consisting of a 38 teeth gear, meshes with the 32 teeth gear in the gear housing, and also with a 47 teeth gear set centrally between the pairs of cylinder heads (Fig. 37).

The 47 teeth gear meshes with two half width camshaft gears, which are mounted on the serrated ends of the camshafts provided separately for the front and rear cylinder heads. This arrangement is used to eliminate any slight misalignment which might exist between the front and rear heads. The choice of the numbers of teeth on the gears and serrations on the shafts provide a vernier for timing purposes. The gears are located by means of spring rings, and have 32 teeth.

The valves, two per cylinder, are operated by normal bucket-type tappets with the valve springs below. The clearance is adjusted in the usual manner by means of thimbles.

The scavenge pump is of the three-gear type, arranged to evacuate the front and rear of the engine separately. It is driven from a 20 teeth spur gear at the front of the main drive shaft, and is fastened to the crankcase walls by means

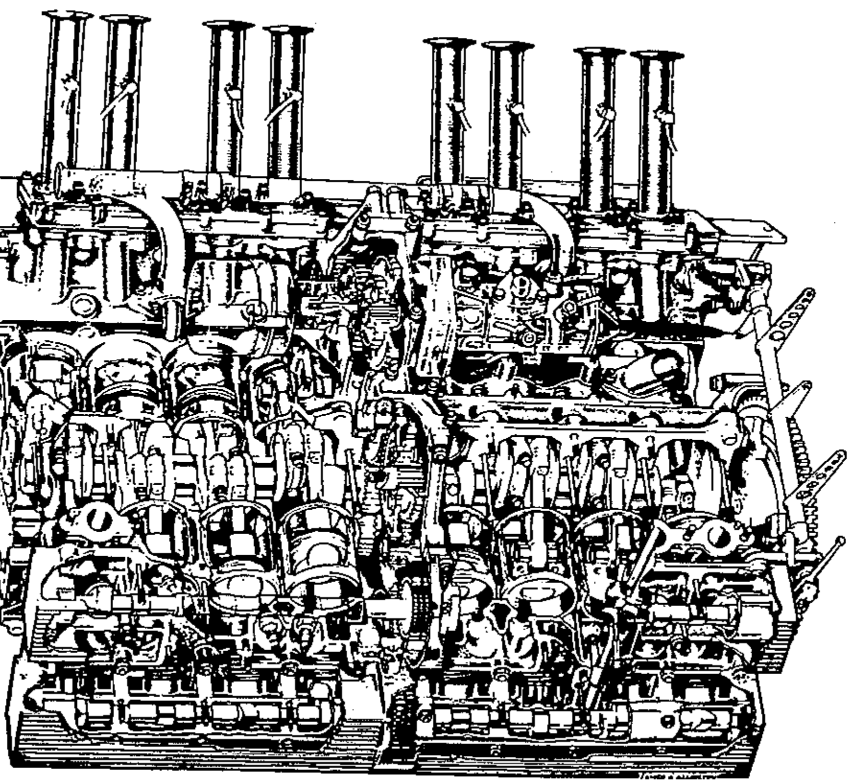


Fig. 35 - Cut-away drawing of 16 cyl

of side extensions which also convey the scavenged oil to drillings in the crankcase. The oil is picked up from the base of the sump via short extension pipes, equipped with O-rings, which seal into cover plates arranged to close the channels from the front and rear of the sump. The actual pickup openings are covered with coarse mesh wire gauze to prevent debris entering the pumps. The driven gear has 32 teeth.

A shaft extends from the scavenge pump central gear to the lubricating pressure pump which again bridges the crank-

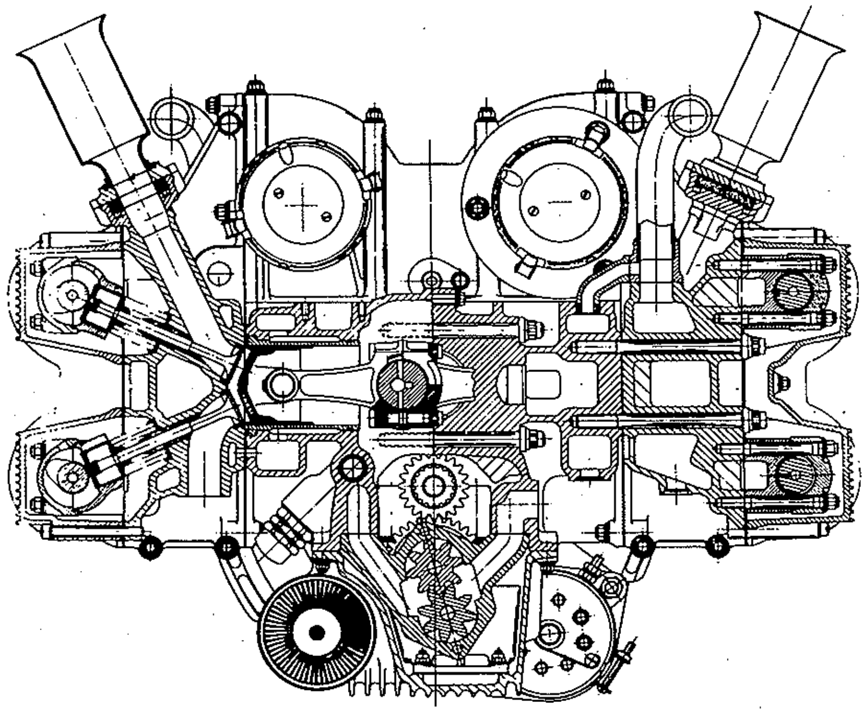


Fig. 36 - Cross-section of 16 cyl

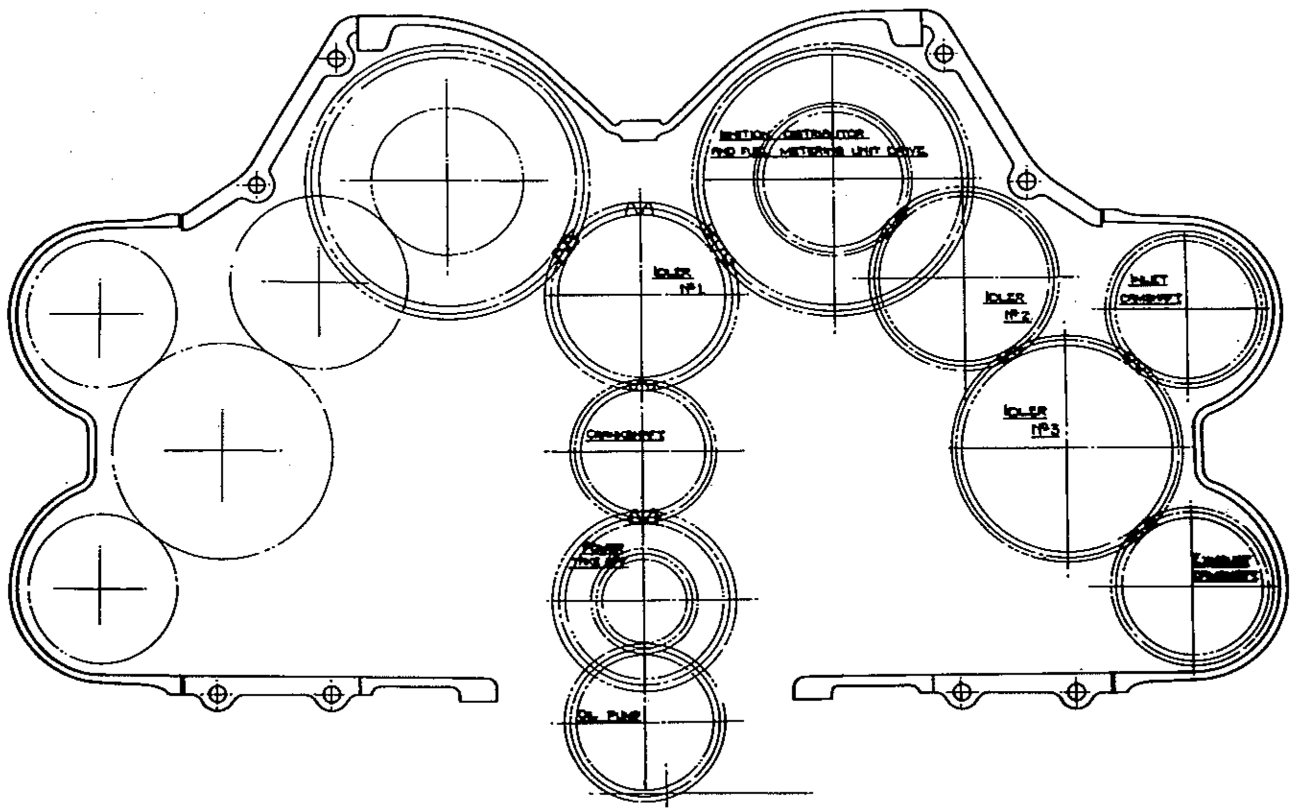


Fig. 37 - Arrangement of gear drive for 16 cyl engine

case walls, with hollow extensions carrying the oil from the oil tank connection and out to the drilling to the oil filter. The driven gear in the pump is set beneath the driver, forming a step-down drive to the coolant pump located on the front wall of the sump (Fig. 38).

The coolant pump is of the centrifugal type with a divided volute, each supplying half the total coolant flow to the galleries cast the full length of each cylinder block. The coolant is metered, by means of calibrated holes, to the cylinder heads near either side of each exhaust port, each hole having a separate O-ring seal. Coolant is directed out of each gallery, via small ports, into the jacket space around each cylinder liner, thence to a short upper gallery, and out through a small right angle connection to the outlet pipes from each of the cylinder heads.

The forged pistons are of slipper types, equipped with a Dykes type top ring, a normal second, and oil control rings. The gudgeon pins are retained by wire circlips with tags to prevent rotation, as developed by the Norton Motor Cycle Co.

The bearings are of the Vanvervell indium-plated V.P.2 steel-backed variety.

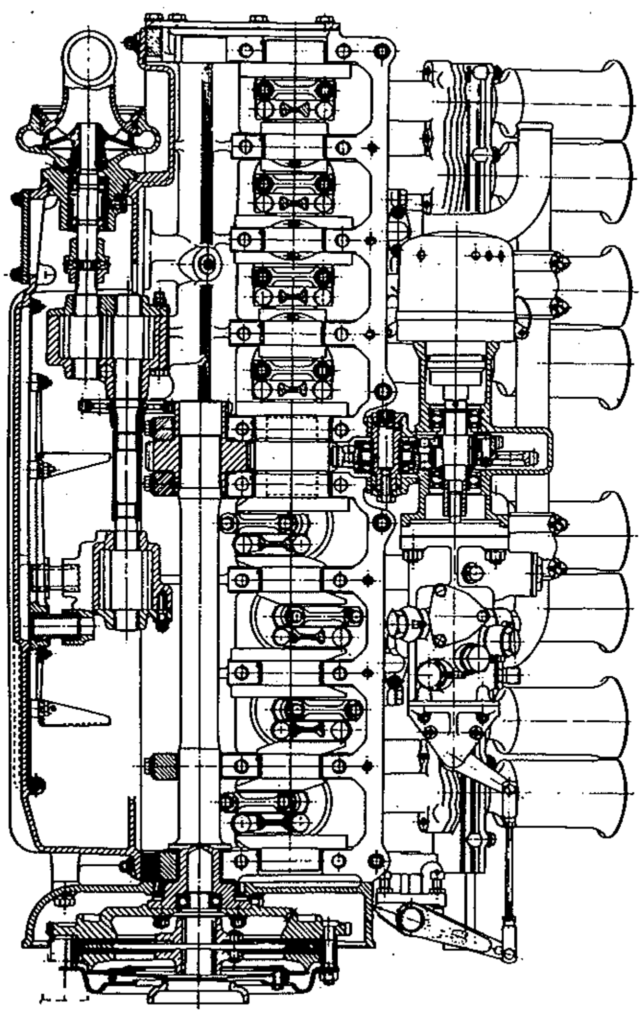


Fig. 38 - Longitudinal section of 16 cyl engine

The cylinder liners are cast iron of the wet type, and are supported by the top flange where the compression seal is effected by Cooper laminated metal seals. The lower ends spigot into the crankcase and are sealed with O-rings.

Ignition is by the Lucas transistorized system, with triggers on the clutch driving plate and pickup units on the flywheel housing.

Lucas type fuel injection is also used, employing two 8 cyl metering units, together with air intakes, and throttle slides running on roller bearings. The injectors are fitted at the end of the ram pipes, facing downstream, as on previous V-8 engines.

The engine weighs 10-15 lb more than the earlier V-8, due to the necessity of duplicating the fuel injection and ignition systems, there being no 16 cyl equipment available.

The grouping of the ancillaries above the crankcase brought a substantial saving in overall length, which worked out at only 1 in. more than the V-8 engines.

DEVELOPMENT OF 16 CYL ENGINE

The design commenced in 1963 but the first engine could not be assembled for test until the end of 1964 which, obviously, was all too late, as the formula had only one year -- 1965 -- to run.

However, optimism was high, and we hoped the engine would perform right away without any major snags, but this was not to be, as the following notes show:

The first difficulty arose immediately the engine was coupled to the dynamometer. It was started up, and ran at around 2000 rpm quite satisfactorily but would not run at any higher speed. While checks were made on mixture strength, fuel flow, and so on, taking in all about five minutes, smoke began to exude from the breathers and the quill shaft sheared within the crankcase.

The trouble was soon recognized as being due to a severe torsional period in the quill shaft, set up by the firing impulses at the low speed at which we were then running, acting through the torsionally flexible quill shaft with the high inertia of the dynamometer, propeller shaft, and clutch driving disc or flywheel firmly coupled at the other end.

Another quill shaft of larger diameter was made, and a new set of drive gears fitted, after some weeks' delay, when testing re-commenced.

Care was taken not to run the engine at speeds below 4000 rpm, which was calculated to be above the torsionally resonant frequency of the shaft.

No further trouble ensued from this particular item, but there was too much oil entrained in the timing gears. Much work was then carried out -- opening up the oil drain and space around the various gears, until it appeared we had eliminated the hp loss originally attributed to this oil churning.

At the same time we were having difficulty with the valve springs, and it was some months before we arrived at a satisfactory answer. In fact, this was not until we were able to get some springs specially wound by True Forged Pistons Inc. in the United States.

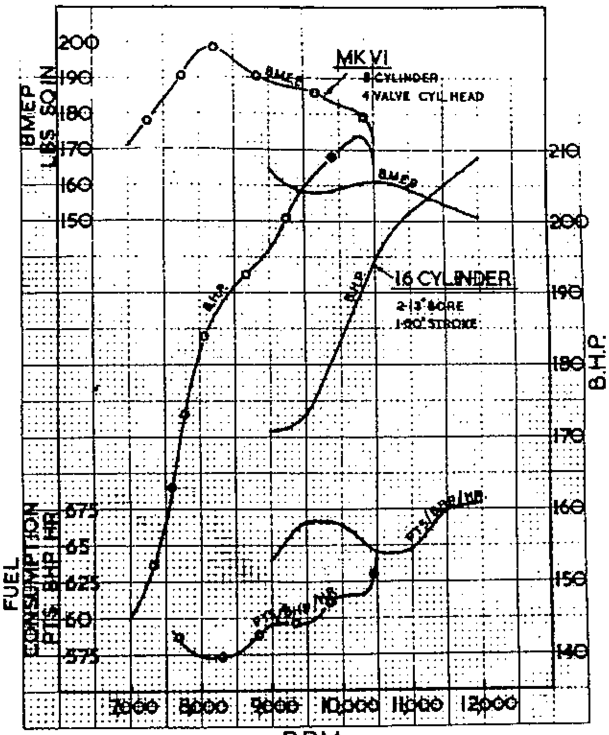


Fig. 39 - Performance of 16 cyl engine compared with Mk. VI

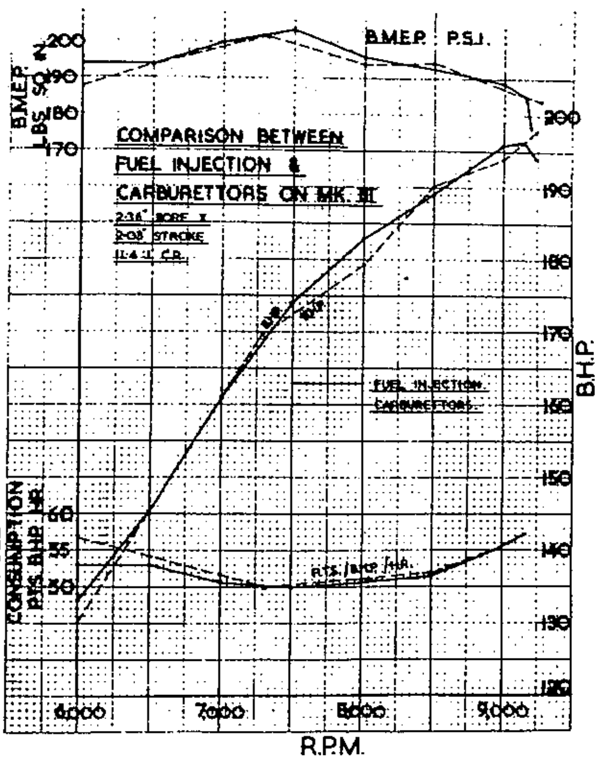


Fig. 40 - Comparison between fuel injection and carburetors

We did eventually obtain 209 bhp at 12,000 rpm from the engine, but the four valve V-8 engines were producing around 210 bhp at 10,500 rpm, and Jim Clark had, by this time, more or less got the Championship in his pocket so, regretfully, it was decided not to pursue the 16 cyl engine. In addition, it was not sufficiently better than the V-8 to warrant the cost of the work of running another car, even though the chassis had been nearly completed by Lotus for the purpose.

We carried out quite a number of tests in order to establish the reason for the engine's failure to achieve the estimated power of 220 bhp (Fig. 39). Morse tests showed the ihp to be as per estimate, but it appeared that frictional losses were greater than supposed.

While we were in oil drainage difficulties from the various drives, figures taken at 11,000 rpm showed an ihp of 295 for a bhp of 198 from which the frictional loss amounted to 97 hp and a mechanical efficiency of 67%, whereas at 9000 rpm the ihp was 219 for a bhp of 165 and frictional loss of 54 hp.

Similar tests on the four valve V-8 but at a lower speed show at 8000 rpm -- ihp to be 205.4 for a bhp of 163 and frictional loss of 42 hp = 79.5% mechanical efficiency.

The two valve V-8 also showed similar figures around 79% mechanical efficiency, both as 1.5 liters and also when the stroke was increased to provide 2 liters.

At 9000 rpm the 16 cyl showed a mechanical efficiency of 76%, which is fairly in line with the other engines.

We checked the power required for the camshafts and found these to need 8.5 bhp at 10,000 rpm, and unfortunately our rig could not cope with the extra 2000 rpm needed to cover 11,000 and 12,000 rpm.

However, checking on the four valve engine, we found that at 10,000 rpm, 6.2 hp was required, and at 11,000 rpm the power had risen to 7.7 hp, showing a steep increase of power required was occurring as the speed of 10,000 rpm was passed. It would seem possible for the 16 cyl to require perhaps 12-14 hp at 12,000 rpm.

The additional bearings in this engine probably impose the remaining frictional handicaps which prevented it from achieving the desired 220-230 bhp; a pity indeed we had not another year or so for development!

CONCLUSION

The Appendix includes as much data relevant to these engines as we have conveniently available at the present time.

ACKNOWLEDGMENTS

I feel it desirable to say that credit for the work described is attributable not to myself, or to any individual. The work is that of a team consisting not only of designers and engineers, but of the staff of other departments, without whose skill and enthusiasm the best of designs could never materialize.

I would, therefore, like to place on record my appreciation of the Coventry Climax team, and to say how pleasurable and rewarding has been my association with them.

I would thank my companies for permission to utilize their accumulation of experience for this paper; the Autocar and Messrs, Lucas for the provision of certain illustrations and data.

My thanks are also due in great measure for the encouragement given me by Leonard P. Lee, and for the ready assistance of my own staff in the preparation of this paper.

APPENDIX

Table 3 lists the major stages of development of the various engines discussed in this paper. Flow of lubricating oil and coolant water is shown in Table 4, and Table 5 shows the projected area of the bearings. Tables 6 and 7 give cam and valve data. Table 8 lists the material content of the racing engines.

SPECIFICATIONS OF 8 CYL ENGINES

1-1/2 LITER TYPE FWMV - YEAR 1961-1962. MK. I

8 cyl - 90 deg Vee
Bore - 2.48 in. × Stroke 2.36 in., or 63 mm × 60 mm approximate.

Table 4 - Coolant and Lubricating Oil Flow

Engine Type	Rpm	Imp. gpm	U. S. gpm
	Coolant Flow		
F.P.F. 4 cyl	7,000	30	36
F.W.M.V. 8 cyl	10,000	44	53.8
F.W.M.W. 16 cyl	12,000	48	57.6
	Lubricating Oil Flow		
F.P.F. 4 cyl	7,000	6.0	7.2
F.W.M.V. 8 cyl	10,000	5.1	6.1
F.W.M.W. 16 cyl	12,000	11.2	13.45

Table 3 - Major Stages of Development

Year	Mark	Variations			Max bhp	Rpm	Max Bmep, psi	Max Torque lb/ft	Rpm
1961-62	I	2.48 in. bore × 2.36 in. stroke 1.3 in. inlet valves. 10.4.1 CR.	Carburetors 4 pairs Weber		181	8,500	195.5	118	7500.
1962	II	As Mark I excepting 1.35 in. inlet valves	Type 38 D.C.N.L./4		186	8,500	197.8	119.3	7500.
1963	III	2.675 in. bore × 2.03 in. stroke 1.35 in. inlet valves 11.1 CR.			195	9,500	195	118	8000.
1964	IV	2.85 in. bore × 1.79 in. stroke 1.35 in. inlet valves 12.1 CR.			200	9,750	193	116.5	8000.
1964	V	As Mark IV excepting 1.4 in. inlet valve	Exhaust		203	9,750	190.5	115.2	8000.
1965	VI	4 valves/cyl 1.04 in. inlet valves 12.1 CR.	pipe diameter increased	Lucas Fuel Injection	212	10,300	197	119	8000.
1965	VII	As Mark VI excepting 1.107 in. inlet valves	1-1/4 in. ID to 1-3/8 in.		213	10,500	189	115	7000/9000
1965	Mk IX	2 liter 2.85 in. bore × 2.36 in. stroke 1.35 in. inlet valves	ID		239.7	8,800	194	155	7500.
1965	Mk X	As Mark IX excepting 1.4 in. inlet valves			244.2	8,900	198	158.2	7500
1965		16 cyl 1-1/2 liter 2.13 in. bore × 1.6 in. stroke			209	12,000	164.5	99.5	9000

Capacity - 1494 cc or 91.17 cu in.
Weight Dry with starter and dynamo - 270 lb.
Compression Ratio - 11.5:1.
Carburetors - 4 pairs Weber type 38 DCLN4.
Cylinder Liners - Cast iron - dry - slip fit.
Pistons - Cast Aluminium "Brico 201." Dykes "L" section top compression ring, plain second, and slotted oil control ring.
Gudgeon Pin - 5/8 in. diameter located by circlip.
Connecting Rods - I section forged nickel steel 4.2 in. centers. Horizontally split big end fastened with 3/8 in. diameter set screws.
Big End Bearings - Vandervell strip type lead bronze indium plated. 1-5/8 in. diameter × 3/4 in. wide.
Main Bearings - 2 in. diameter. Nos. 1, 3, and 5 being 0.850 in. wide while Nos. 2 and 4 are 0.725 in. wide.

Crankshaft - Forged steel nickel chrome molybdenum. 2 plane 90 deg big ends side by side. Balanced for part revolving weight plus balance weights in fly wheel and at the front end to counteract the rocking couple.

Timing Drive - Spur gear from crankshaft to half speed shaft which drives the coolant pump at the front end and the ignition distributor by a skew gear at the rear. Separate simple Reynolds roller chains to each cylinder head, each with twin camshafts running in 5 pressure lubricated white metal strip bearings, with an adjustable jockey sprocket set in front of each cylinder bank.

Cylinder Heads - Light alloy LM8.W.P. with austenitic valve seat inserts shrunk in. Valves at 60 deg included angle, inlet 1.3 in. head diameter and 1.237 in. exhaust, controlled by double coil springs and operated by case hardened steel tappets. Clearances adjusted by graduated thickness

Table 5 - Projected Bearing Areas and Loads Based on Assumed Cylinder Pressure or 1000 psi

Engine Type	Piston Area sq in.	Bearing Area, sq in.		Bearing Load, psi	
		Big End	Small End	Big End	Small End
F.P.F. 4 cyl	8.05	1.86	1.045	4320	7,700
F.W.M.V. 8 cyl	6.39	1.218	0.469	5250	13,600
F.W.M.W. 16 cyl	3.565	0.703	0.344	5075	10,360

Bearing Sizes

Engine Type	Main Bearing			Big End Brg.	Small End Brg.
	Front and Rear	Center	Inter.		
F.P.F. 4 cyl	2.5 in. dia. × 1.00 in.	2.5 in. dia. × 1.00 in.	2.5 in. dia. × 1.00 in.	2.125 in. dia. × 0.875 in.	0.938 in. dia. × 1.10 in.
F.W.M.V. 8 cyl	2.0 in. dia. × 0.812 in.	2.0 in. dia. × 0.812 in.	2.0 in. dia. × 0.625 in.	1.625 in. dia. × 0.750 in.	0.625 in. dia. × 0.750 in.
F.W.M.W. 16 cyl	1.75 in. dia. × 0.525 in.	2.187 in. dia. × 0.510 in.	1.75 in. dia. × 0.525 in.	1.312 in. dia. × 0.535 in.	0.625 in. dia. × 0.550 in.

Table 6 - Cam Details

Engine		Total Lift, in.	Ramp	Max. Velocity in./deg	Max. Acceleration in./deg/deg	
					Positive	Negative
V-8	2 valve cyl	0.360	0.010 in. at 0.00033 in./deg	0.008	0.6614×10^{-3}	0.1675×10^{-3}
	4 valve cyl	0.340 inlet 0.320 exh.	0.010 in. at 0.00033 in./deg	0.008	0.6614×10^{-3}	0.1675×10^{-3}
16 cyl		0.325	0.010 in. at 0.00033 in./deg	0.0075	0.6452×10^{-3}	0.1476×10^{-3}

steel thimbles. Valve material 21-4-NS for both inlet and exhaust.

Valve Timing -

Inlet opens	45 deg btdc
Inlet closes	65 deg abdc
Exhaust opens	65 deg bbdc
Exhaust closes	45 deg atdc

Table 7 - Valve Component Data

Weights of Valve Components, oz

Components	V-8 2 valves cyl	V-8 4 valves cyl	16 cyl
Inlet valve	2.062	1.642	1.5
Exhaust valve	1.875	1.562	1.5
Valve spring collar	0.937	0.5	0.437
Valve spring inner	0.093	0.093	0.093
Valve spring outer	1.312	1.25	1.187
Valve cotter	0.812	0.5	0.687
Valve tappet	2.03	1.142	1.0
Tappet shim	0.062	0.062	0.062

Valve Spring Loading, lb

Combined valve open	174	185	181
Combined valve closed	69	86	89

Valve Accelerations, fps^2

Positive	44,800	54,700	69,600
Negative	11,350	13,900	15,980
Engine rpm	9,500	10,500	12,000

Tappet Clearances Cold - 0.006 in. inlet; 0.012 in. exhaust.

Valve lift - 0.350 in.

Crankcase - Light alloy casting LM8W.P. extends below crankshaft to sump joint. Steel main bearing caps center and intermediate caps side bolted to crankcase walls.

Oil Pumps - Spur gear driven from crankshaft at 0.7 engine speed. Pressure pump located on front main bearing panel. Scavenge pumps mounted on front timing cover plate driven in tandem from main pressure pump by Oldham coupling.

Main pressure pump circulated 4.5 Imp Gal = 5.4 U.S. gpm at 8600 crankshaft rpm. Scavenge pumps each of similar capacity and scavenging front and rear of the engine separately.

Relief Valve - Tecalemit guided disc housed in main feed to gallery. Relieved oil fed back to the suction side of the pressure pump.

Coolant Pump - Centrifugal - impeller mounted on 1/2 speed jackshaft extension. Double outlet, one to each bank of cylinders. 33 Imp gpm or 40 U.S. gpm circulation.

Ignition - Lucas transistor system. 4 equally spaced triggers mounted on flywheel face. Pickup set in flywheel housing.

Firing Order - A1, A3, B3, A2, B2, B1, A4, B4.

Spark Plugs - 10 mm Lodge 10RL50 or Champion type G59R.

YEAR 1963 MK. II

A small number of engines were equipped with the larger inlet valve. 1.35 in. OD in lieu of the 1.3 in. valve of 1962; otherwise there was no other change. Carburetors still used.

YEAR 1963 MK. III

The majority of the 1962 engines were rebuilt with the following changes:

Bore - 2.675 in. × Stroke 2.03 in., or 72.39 mm × 45.47 mm.

Table 8 - Racing Engine Component Material and Weights

Description	V-8 Weight	V-8 Material	16 Cyl Weight	16 Cyl Material
Cylinder Block	46 lb 8 oz	LM.8WP (MOD)	48 lb 8 oz	LM.8WP (MOD)
Cylinder Head	20 lb 10 oz	LM.8WP	12 lb 8 oz	LM.8WP (MOD)
Sump	5 lb 6 oz	Electron 'C'	5 lb	Electron 'C'
Cambox Cover	4 lb 14 oz	Electron 'C'	6 lb	Electron 'C'
Water Pump Body	1 lb 6 oz	LM.4M	1 lb 6-1/2 oz	LM.4M
Oil Press. Pump Body	5 oz 9 drm	LM.4M	10 oz 2 drm	LM.4M
Piston	7 oz 10 drm	R.R.58	3 oz 15 drm	Mahle Forged
Tappet Block	2 lb 4 oz	LM.8WP	1 lb 14 oz	LM.8WP

Inlet Valve increased from 1.3 to 1.35 in. diameter: exhaust valve as before.

Lucas Fuel Injection adopted in lieu of carburetors.

Connecting Rods - Center distance increased from 4.2 to 5.1 in., giving a stroke/centers of 2.5:1. Bolts and nuts replacing the set screws previously used.

Crankshaft - Single plane 180 deg phased forged in EN40 nitriding steel.

Otherwise the specification remained similar to previous years.

YEAR 1964 MK. IV

Several engines were converted to conform to the proposed four valves/cylinder specification with a view to fitting four valve cylinder heads later if and when the development reached a satisfactory state. The current two valve cylinder heads were fitted together with pistons to suit the new bore and stroke of 2.875 × 1.79 in. These engines retained the chain drives to the camshafts as previously.

Camshafts modified to reduce acceleration.

Period extended from 296 to 302 deg.

0.350 in. lift.

MK. V

One Mk. IV engine converted to larger inlet valve 1.4 in. diameter with port to match. Apart from these changes the specification remained unaltered.

YEAR 1965 MK VI. FOUR VALVES PER CYLINDER

Two engines were built to the following specification during 1963 expecting to achieve results for 1964 racing. Reasonable success was not achieved however before the end of 1964, and these two engines only existed.

4 VALVE/CYL FWMV 1.5 LITER V-8

8 Cyl - 90 deg Vee

Bore - 2.85 in. × Stroke 1.79 in., or 72.39 mm × 45.47 mm

Capacity - 91.35 cu in. or 1496.36 cc

Weight Dry complete with starter and alternator - 298 lb

Compression Ratio - 12:1

Lucas Fuel Injection into the intakes.

Fuel - 100 Octane Premium Grade.

Performance - 210 bhp at 10,500 rpm

Rpm Limit - 11,000 rpm

Mean Piston Speed at 10,500 rpm - 3,135 ft/minute

Timing Drive - All spur gear drive. 1/2 speed drive crankshaft to jackshaft, and thence a separate gear train to each pair of camshafts, gears and bearings each contained in carrier brackets supported in the front compartment of each cylinder head.

Camshafts - C.I. with cams chill hardened. Each shaft running in 5 split steel backed babbit bearings pressure lubricated.

Tappets - Case hardened steel running direct to the cast aluminium housings which combine camshaft bearings with tappet guides in one simple component.

Crankshaft - Single plane 180 deg phased. Crank pins in nitrided alloy steel. Main journals, and crank pins 2 in. and 1-5/8 in. diameter, respectively. Bearings main and big end Vandervell strip VP.2 indium plated lead bronze. A small Holset torsional vibration damper is fitted on the front end.

Connecting Rods - 'I' section forged in nickel chrome steel 4.2 in. centers fitted with bolts, and nuts devoid of any locking device, and relying upon the friction to lock the fastening.

Pistons - Forged light alloy, slipper skirt, pent roof crown with valve clearance recesses. Dykes type top compression ring, taper face second, and scraper ring in high tensile cast iron. Circular wire circlips with tag to prevent rotation for gudgeon pin retention.

Gudgeon Pin - 5/8 in. diameter taper bored for lightening purposes.

Cylinder Liners - C. I. dry supported by top flange nesting on light alloy sleeves sealed at the lower ends by compression seal in the crankcase counterbore. Compression seal by Coopers patent joint ring, and coolant seal by Klingerite gasket.

Cylinder Block - As previous engines.

Cylinder Heads - LM.8 light alloy with pent roof combustion chamber. 10 mm plug in central position. Inlet valves 1.04 in. diameter of En59 set at 60 deg included angle to the exhaust valves 0.935 in. diameter. Alternatively larger valves of 1.107 and 1.043 in. diameter, respectively were also provided and tested of 21-4-NS steel seating on "shrunk-in" inserts of austenitic steel and hidurel bronze respectively.

Both inlet and exhaust ports run separately to the cylinder head face where both run into 'Y' pieces. On the inlet a throttle slide assembly sits directly on the 'Y' piece, and the air intake is located directly over the slide cover. Fuel injection nozzles are positioned inside the air bells so that the discharge takes place downstream, a small way above the throttle entry.

Firing Order - A1, B4, A2, B3, A4, B1, A3, B2.

Valve Timing -

Inlet opens	46 deg btdc	) Period 290 deg
Inlet closes	64 deg abdc	) Lift 0.330 in.
Exhaust opens	57 deg bbdc	) Period 280 deg
Exhaust closes	43 deg atdc	) Lift 0.310 in.

Tappet Clearance Cold - Inlet 0.006 in.; Exhaust 0.014 in.

Spark Plugs - 10 mm Champion type G.56.R.

Ignition Advance - 48 deg btdc

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Napier Nomad Aircraft Diesel Engine

Herbert Sammons and Ernest Chatterton,
D. Napier & Son, Ltd.

This paper was presented at the SAE Summer Meeting, Atlantic City, June 10, 1954.

THE Nomad engine has been designed and developed in England by Napier with the object of providing a powerplant which will enable the greatest possible operating economy to be achieved in air transport services, with particular reference to air freighting.

The special suitability of the engine for these purposes arises primarily from its low fuel consumption which remains below 0.35 lb per ehp per hr over the whole useful range of cruising speeds and altitudes, and at some conditions is as low as 0.326 lb per ehp per hr, this corresponding to a brake thermal efficiency of 42%.

The economic value of these low consumptions is emphasized by the use of relatively cheap diesel fuel, although the engine will run equally well on aviation kerosene or on wide-cut gasoline.

Although the engine operates on the diesel cycle, its specific output in terms of power developed per unit of cylinder volume is higher than that achieved

THE Nomad compounded diesel engine described in this paper has been developed by Napier during the past seven years.

Resulting from the combination of a diesel engine with a gas turbine, the engine gives lower fuel consumptions than can be obtained from any other type of powerplant and maintains these consumptions over a wide range of aircraft speeds and altitudes.

These characteristics, which are still further improved by the ability to use cheap fuels, produce an engine making possible extreme economy for air transport purposes and very long-range operations for either military or civil purposes.

The Authors

HERBERT SAMMONS is managing director of D. Napier & Son, Ltd., in Acton, England. In 1922 he was appointed chief designer of Petters Ltd., oil engine manufacturers, of Yeovil, England, where he later became chief engineer. Mr. Sammons was made manager of the Kadenacy Department of Sir W. G. Whitworth, Ltd., in 1934. He has been with D. Napier & Son, Ltd., since 1943.

ERNEST CHATTERTON is chief engineer of the Piston Engine Division of D. Napier & Son, Ltd. He received his degree in engineering from the University of London in 1922. Mr. Chatterton's association with Napier dates back to 1930 when he took charge of experimental design. He became assistant chief designer in 1938, chief development engineer in 1941, deputy chief engineer, and subsequently, in 1949, he was appointed to his present post.

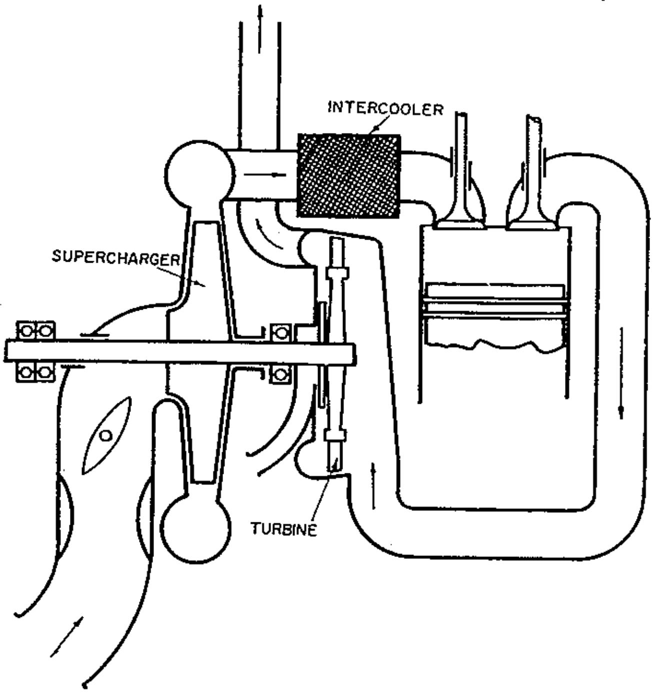

Fig. 1 – Exhaust-driven supercharger on Napier Lion

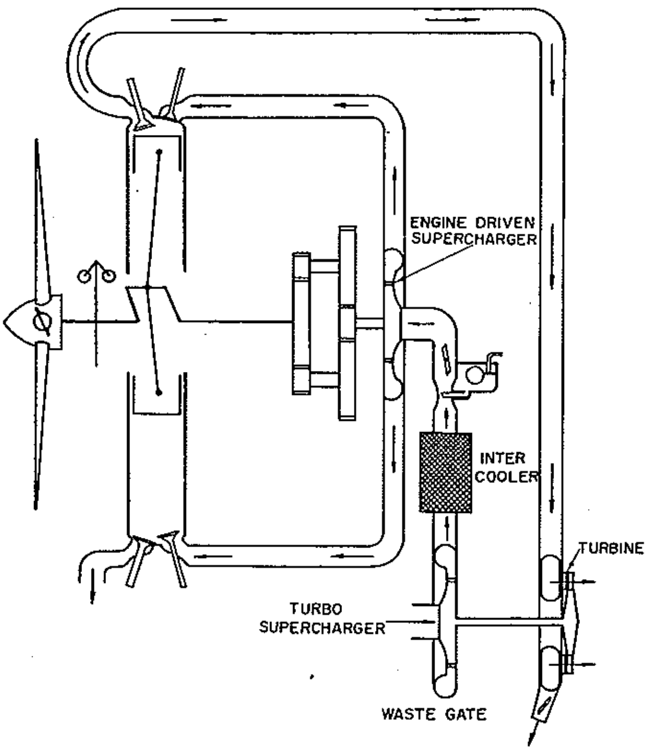

Fig. 2 – Arrangement of General Electric turboblower system

by the gasoline engine even in its most recent stages of development, the power output per cubic inch of cylinder volume under take-off conditions being 1.25 shp for the basic engine and 1.64 shp in its complete form with water injection. Due to these high outputs the specific weight of the engine can be reduced to below 1 lb per ehp.

Briefly then, the Nomad engine may be described as a diesel engine designed particularly for aircraft use, which weighs about 1 lb per hp and has a specific fuel consumption lying between 0.326 and 0.35 lb per ehp per hr over the whole of its useful operating range.

These unique performances are achieved by associating a piston engine with a gas turbine to form a "compound" engine, the operating cycle of which is devised so that each component can make its maximum contribution to the overall results.

The Piston Engine and the Gas Turbine in Partnership – The association of a gas turbine with a piston engine is by no means a recent conception. As early as 1924 a Napier Lion Series V engine was equipped with a turbine in the manner shown in Fig. 1, the exhaust gases being fed into the turbine to provide all the power to drive the engine supercharger.

At a later date during the 1939-1945 war, many engines were installed in operational aircraft fitted with "turbochargers" of the type shown in Fig. 2, in which an exhaust-activated turbine drives a subsidiary blower boosting the air intake of the normal engine supercharger.

While it may perhaps be said that the germ of the compounding principle is contained in these early examples, neither of them can be regarded as a true demonstration of compounding, for in each case the functional conditions of the engine are normal. No attempt is made to adjust these conditions to extract maximum energy from the combination, and the turbine is applied with the primary objective of extending the normal performances of the engine over a greater range of altitudes.

The extent to which this limited objective is achieved, as compared with a normal engine fitted with a two-speed mechanically driven blower, is illustrated in the power curves of Fig. 3, which show that the engine equipped with a turbine is able to maintain a selected cruising boost pressure and, therefore, its cruising power at a sensibly constant value over a wide altitude range.

This result serves to demonstrate the striking effects on engine performance which can be derived from the simplest possible association of a gas turbine with a normal piston engine.

A more recent powerplant which combines a piston engine with turbines is the Curtiss-Wright "Turbo-Compound" engine, which at the present time is installed in the Lockheed Super Constellation and other aircraft. This application differs from the two previously mentioned in that the power produced by the turbines is fed back into the crankshaft, and the engine may therefore be classified as "compounded."

In this case the 18 cylinders of the 4-stroke gasoline engine are divided into three groups of six, each group feeding a separate turbine geared into the crankshaft system. The basic engine is the C.18.CB Cyclone to which the turbines have been added with the object of recovering as much waste energy as possible from the exhaust gases without disturbance to the established cylinder operating conditions. To avoid any material back-pressure effects the turbines are therefore of the "blowdown" type, which are activated by the impulsive velocities in the exhaust ducts.

No reliable performance data on the Wright turbo-compound engine is available in England, but its extensive and growing use indicates that worthwhile improvements in performance are being achieved. It will, however, be demonstrated later that this type of association between a developed spark-ignition engine and gas turbines is subject to definite limitations which prevent the full virtues of the compounding principle from being realized.

If, however, a piston engine and a gas turbine are designed together in the original inception to produce a true "compound" design, the scope for improved performance becomes much greater, for in this case the cycle of operations can be manipulated to exploit the most valuable features of the piston engine and the turbine which can then be associated together to produce a powerplant possessing strongly defined performance characteristics, which cannot be achieved by any other type of prime mover.

It is the purpose of this paper to analyze the characteristics of the compound engine with particular reference to the Napier Nomad, which is the only existing example of the type, and to indi-

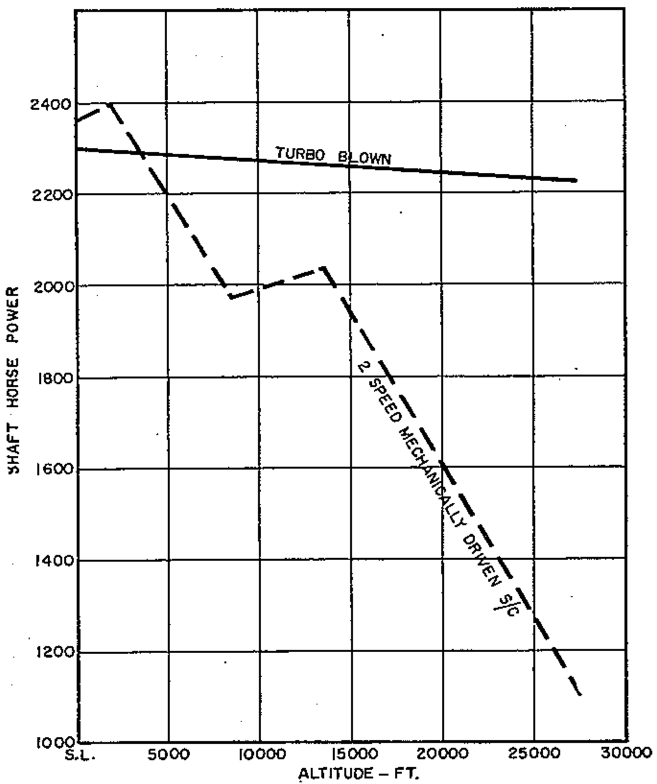

Fig. 3 – Altitude performance of turboblower engines

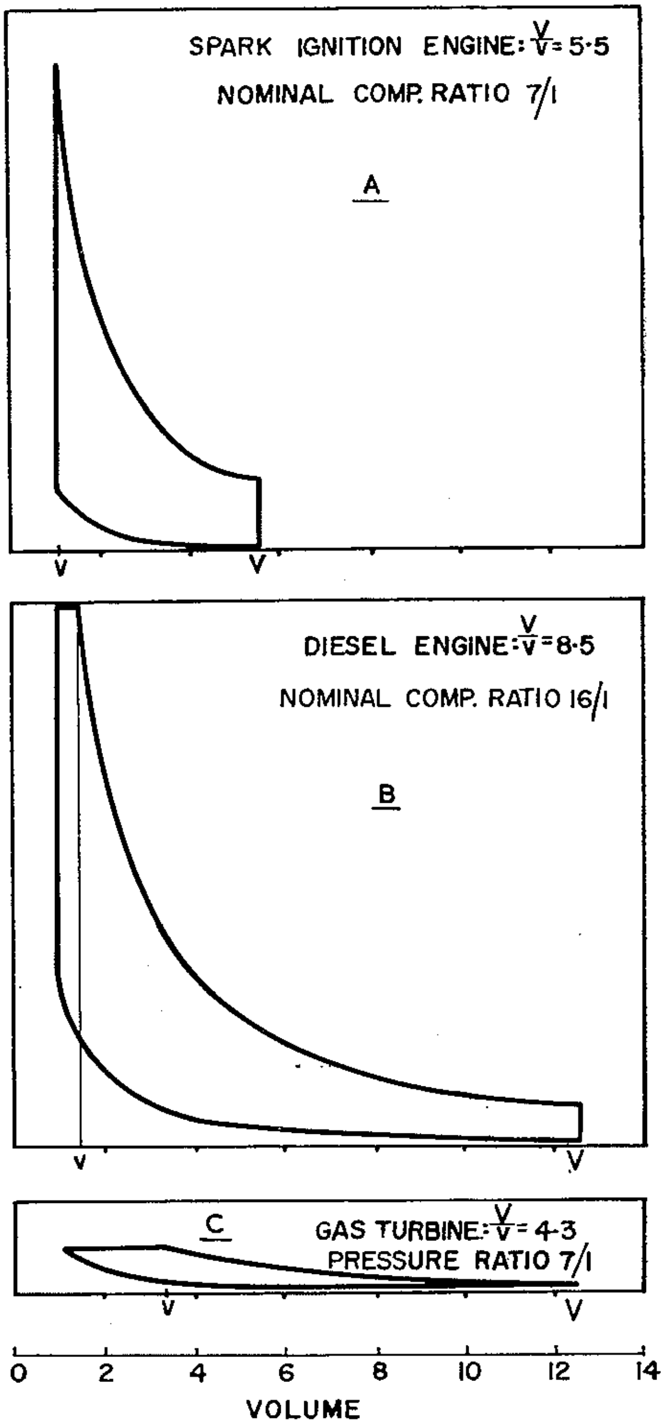

Fig. 4 – Typical indicator diagrams

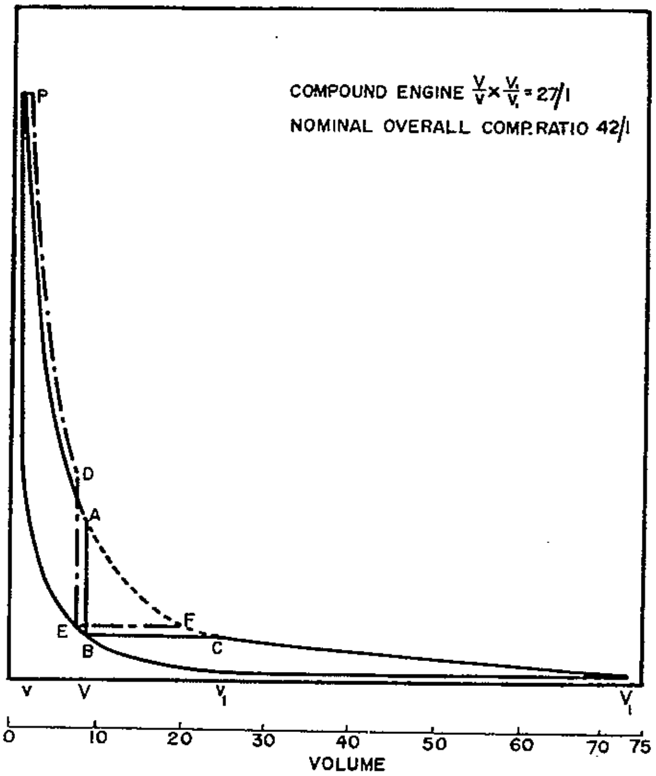

Fig. 5 – Compound indicator diagram

cate the contribution which such an engine can make in modern aviation.

How Compound Cycle Is Derived

If the major functional characteristics of the various types of engine are considered individually, it will be possible to indicate the factors involved in the derivation of the compound cycle and to assess the theoretical basis and practical limitations of this cycle in relation to others.

In broad terms, without too much respect for meticulous detail, the main relevant characteristics of different engines may be summarized as follows:

The Spark-Ignition Engine – The gasoline-burning engine, by its nature, consumes essentially all the air supplied to it, and the power developed by a given size of cylinder is almost entirely dependent on the rate at which it can inhale and reject air. Apart from changes in operational speed, the air quantity can be increased by supercharging, but the extent to which this is possible is limited by detonation and preignition difficulties.

An approximate "indicator diagram" for this type of engine is shown in Fig. 4A. Combustion takes place at constant volume, and the peak temperature of the cycle will be in the region of 4500 F. Basically the thermal efficiency of the cycle is governed by the volumetric ratio V/v, and this again is limited by the occurrence of detonation. Furthermore, if the supercharging pressure is increased to raise the maximum power, the compression and expansion ratios will, in general, require to be reduced to avoid detonation with consequent sacrifices in thermal efficiency over the cruising range of the engine.

It is, therefore, true to say that the destructive phenomenon of detonation is the limiting factor to both the power output and the thermal efficiency obtainable from this type of engine, and most of the spectacular power development demonstrated during the 1939-1945 war resulted directly from the introduction of fuels less prone to detonation coupled with the use of water injection and charge coolers which allowed the charge density to be increased and its temperature reduced, thus keeping within the detonation limits.

Typical performance figures achieved from spark-ignition engines in the more recent phases of their development are given in Table 1.

This type of engine is not amenable to manipulation of its cycle for compounding purposes, for any back pressure imposed in the exhaust system by a turbine will introduce practical difficulties, such as an increase in temperature of exhaust valves, and an increase in the mass and temperature of the residual gases which reduces the volumetric efficiency and raises the charge temperature, thus accentuating the already acute detonation difficulties. The best that can be done with this cycle is the arrangement already referred to as used in the turbo-compound in which a proportion of exhaust-heat recovery is achieved without disturbance to the normal pressure and temperature levels in the engine cylinder.

Although the foregoing remarks all relate to four-stroke engines, a similar situation exists if applied to the two-stroke gasoline-engine cycle.

The Diesel Engine – This engine also attempts to burn all the air which is supplied to it; but as in this case the mixing of fuel with the air has to take place in the cylinder within a maximum period of about 35 to 40 deg of crank rotation, it is not found possible in practice to utilize more than about 80% of the total air trapped in the combustion chamber. For a given maximum power output the cylinder dimensions will therefore need to be larger than in the spark-ignition engine. In the 2-stroke diesel engine a further loss of air occurs because of the necessity for effectively scavenging the burned gases from the cylinder, which implies a wastage of about 40% of the total air throughput which will escape from the exhaust ports or valves during the scavenging process. This air lost during scavenging is not entirely without value for it does act as an internal cooling agent and therefore reduces the thermal loadings on the pistons for a given amount of fuel burned.

As with the gasoline engine, the quantity of air trapped in the engine cylinders can be increased by supercharging, and, as the detonation limitation no longer exists, the degree of supercharge can be increased as dictated by the general balance of the engine design. It will be observed, however, that in the 2-stroke engine any degree of supercharge can only be effective providing that a corresponding resistance, or back pressure, is offered at the exhaust ports or valves.

The approximate indicator diagram for this cycle is shown in Fig. 4B. Here combustion takes place partly at constant volume and partly at constant pressure, the proportion of fuel burned at constant volume being to some extent controllable and determined by the limitation established by the designer for the maximum cylinder pressure.

Although not strictly correct, due to the portion of constant-pressure combustion, the thermal efficiency for present purposes may again be related to the expansion ratio V/v, which in this case is considerably higher than in the gasoline-engine case with resultant improvement in fuel consumption.

The flame temperature during combustion will also be in the region of 4500 F, but, due to defective air utilization, the heat of combustion is diluted to some extent and the maximum temperature spread over the total air in the combustion chamber will be reduced to about 3300 F.

The diesel engine has not been widely used in aviation, but one engine which saw considerable service before the 1939-1945 war was the Junkers Jumo engine, the results from which are also included in Table 1.

It will be apparent that the cycle of operations described, particularly when referred to the 2-stroke engine, is very much more adaptable for compounding purposes than the spark-ignition case, for the following possible variables exist:

(a) In the absence of detonation or preignition troubles the compression ratio and the degree of supercharge may be varied to suit design requirements.

(b) The maximum cylinder pressures are to some extent controllable by variation in the rate of fuel input.

(c) Back pressures in the exhaust system need not be an embarrassment and are in fact essential when high degrees of supercharge are to be used.

(d) An excess quantity of air, over and above that required for combustion, is passed through the engine, this serving as an internal cooling agent which also reduces the mean temperature of the exhaust gases.

It will be seen later that these possibilities may be turned to very good account in the compound engine, and it may be stated here that the 2-stroke diesel cycle forms the ideal basis for compounding.

The Gas Turbine – Discussion here is limited to the turboprop engine, as it is this arrangement which is of interest relative to the compound case.

The basic cycle on which this engine operates is shown in Fig. 4C. Combustion takes place at constant pressure, and the basic thermal efficiency is again governed by the volumetric expansion ratio which, in the present state of compressor and turbine development, is limited to a lower value than that employed in either the diesel or the spark-ignition motor. If such an engine burned all the air passed through it the temperature of combustion would again be in the region of 4500 F, and this is the continuously maintained temperature at which gases would enter the turbine. This temperature is prohibitive, and with available materials and design technique, the temperature at the inlet to the turbine is limited to about 1500 F. To achieve this reduction in temperature, air to

Table 1 – Comparative Performance of Piston Engines

Type of Engine	Actual Engine	Fuel	Maximum Shp at Take-Off	Maximum Shp per Cu In.	Maximum Shp per Sq In. of Piston Area	Specific Weight at Take-Off, lb per shp	Specific Fuel Consumption, Sea Level at 60% Take-Off Power, lb per shp per hr
Gasoline[a]	Napier Sabre VA	100/130 grade	2550	1.14	5.43	0.976	0.50
Gasoline[a]	Napier Sabre VII	100/130 grade + water injection	3050	1.37	6.5	0.840	0.50
Gasoline[a]	Wright Compound	115/145 grade + water injection	3700	1.11	6.97	0.92	0.400[b]
Two-stroke diesel	Junkers Jumo 205E	Diesel fuel	600	0.59	3.73	1.910	0.345
Compound diesel[a]	Napier Nomad	Diesel fuel	3046	1.22	9.0	1.175	0.359
Compound diesel[a]	Napier Nomad	Diesel fuel + water injection	3480	1.39	10.25	1.027	0.345
Compound diesel[a]	Napier Nomad	Diesel fuel + auxiliary combustion + water injection	3970	1.59	11.70	0.959	0.340

[a] Figures based on total shp.
[b] 51% power, auto-lean mixture.

the extent of about 3½ times the amount actually required for combustion is therefore passed through the engine to provide temperature dilution.

When such an arrangement is associated with a piston engine to form a compound system such high excess air factors become unnecessary, for the reduction in gas temperatures is achieved by virtue of the work done in the piston-engine cylinders and the compressor and turbine dimensions for a given rate of fuel consumed are thereby greatly reduced.

The objection may be raised that the figures here quoted for expansion ratio and turbine inlet temperature in the turboprop engine are lower than some which are projected at the present time. Although this is appreciated it does not affect present considerations relative to the compound cycle, and in any case it will be seen later that most of the developments which permit improved performances of the turboprop are also of direct application to the compound engine.

The Compound Engine – In the compound engine the cycle of operations is shared between a 2-stroke diesel engine and a turbine, exhaust gases from the engine being fed into the turbine. The back pressure imposed by the turbine on the engine cylinders establishes the lower pressure level of the operating cycle in the cylinders and also determines the degree of supercharge necessary to pass the required quantity of air through the engine.

The approximate form of the overall indicator diagram resulting from this combination is shown in Fig. 5, the division of work between the engine and the turbine being indicated by the line ABC, the difference in pressures between points A and B being that which exists between the pressure in the cylinder at the moment of exhaust release and the pressure at inlet to the turbine.

Considering first the characteristics of this complete diagram, it may be said that due to the high degree of compression which takes place outside the engine cylinders it is possible to use an overall compression ratio considerably higher than in the normal diesel engine. Furthermore, the expansion line in the turbine extends down to atmospheric pressure, and the result of these two factors combined is to produce a very high value for the overall expansion ratio. The actual expansion ratio which determines the limiting thermal efficiency, however, is lower than this due to the losses which occur at transfer from the diesel engine to the turbine, and its value may be derived in the following way.

It will be seen that the overall diagram is compounded of two separate ones, first that for the diesel engine which is similar in form to that of Fig. 4B, and second that for the compressor-turbine combination which is similar to that of Fig. 4C. The expansion ratio for the diesel cycle is V/v, and that for the turbine V_1/v_1, so that the actual expansion ratio for the complete cycle is given by the product $(V/v) \times (V_1/v_1)$.

Within practical limits, for a given overall diagram the value of expansion ratio given by the product $(V/v) \times (V_1/v_1)$ and therefore the limiting thermal efficiency of the cycle does not change materially with the position of line ABC indicating the division of work between the diesel engine and the turbine.

But the division of work does exercise considerable influence on the dimensions and weight of the combined unit for a given maximum power, and the selection of the various design constants affecting this point is one of the major problems in the early stages of compound engine design. Some indications of the nature of this problem may be obtained from a consideration of the following points.

Clearly the compound engine is subject to the same practical limitations as apply to other engines, so that the maximum cylinder pressure and the maximum turbine inlet temperature are reasonably well-defined points determined by practice. If now the compound engine operated on an ideal air/fuel ratio and burned all its air, the temperature at point P would be about 4500 F, and corresponding temperatures at various points down the expansion line would be defined. The maximum height for point C on the expansion line would therefore be fixed by the permissible turbine inlet temperature, and this would also establish the supercharging pressure.

But it has been already pointed out that the diesel engine does not consume all the air supplied to it and, furthermore, that in the two-stroke cycle the excess air passed through the cylinders during the scavenging period can be controlled by porting design.

It is, therefore, possible to design the cycle so that the amount of excess air is a relatively high proportion of the total mass flow through the engine, in other words, to operate the engine on a higher overall air/fuel ratio. This excess air will mix with the exhaust gases and reduce their temperature, so that to achieve the same limiting temperature at inlet to the turbine it now becomes possible to raise the position of point C on the expansion line to a new position F, the division of work now being indicated by line DEF. But this adjustment to the cycle has also raised the supercharging pressure so that a greater quantity of air will be trapped in the engine cylinder, thus permitting more fuel to be burned and a greater mean indicated pressure to be developed, as shown by the dotted expansion line.

The thermal loadings in the engine cylinders will set a limit to the extent of this adjustment; but, as a practical point, it may be said that the passage of excess air through the cylinders not only gives ideal scavenging, it also provides a considerable degree of internal cooling, any resulting increase in its temperature being subsequently recovered in the turbine and therefore retained in the cycle. By exploiting this factor it is possible in the compound-engine cylinder to achieve a rate of fuel burning which is much greater than in the normal diesel engine, with a corresponding reduction in the dimensions and weight of engine required for a given power. This, coupled with the reduced size of compressor and turbine required for a given rate of fuel burning, which has already been referred to in the section on The Gas Turbine, allows the size and specific weight of the compound powerplant to be reduced to values which, when coupled with its high thermal efficiency, render it more suitable than other powerplants for certain applications.

The determination of the appropriate division of work in any particular case, therefore, involves consideration of a range of different overall air/fuel ratios, each involving corresponding adjustments to the size and pressure ratio of the compressor and turbine and the dimensions of the engine cylinders derived from the thermal and mechanical loadings imposed. The lightest possible powerplant results when the best balance between these various factors is established in relation to the design targets.

Some characteristics of the compound engine as demonstrated by the Nomad are given in Table 1 to allow comparison with other types of engine.

Mechanical Arrangements for Compounding

For the purposes of this paper the compound engine is considered to consist of three major mechanical elements, first a 2-stroke diesel engine, second a gas turbine, and third a compressor. Investigators of compounding possibilities have sometimes proposed more complex arrangements involving a relative multiplicity of turbines, compressors, and heat exchangers, designed with the object of extracting maximum possible energy from the fuel burned. In practice it seems unlikely that such complexity would be justified, for the thermal and aerodynamic losses involved in the gas-transfer passages can exert serious influences on the actual results obtained. In any case, such arrangements are ruled out for aircraft purposes on the grounds of bulk and weight, and discussion is therefore limited to possible combinations of the three basic elements concerned. Of these components the piston engine and the turbine contribute to the total power output, but the compressor absorbs power and must be driven.

From the purely mechanical standpoint three possibilities exist and their applicability is considered.

1. *The compressor may be driven by the turbine – all output power being delivered by the piston engine (Fig. 6A).*

This arrangement constitutes in effect a 2-stroke diesel engine equipped with a turboblower. It demands that over the whole operating range the turbine must be in power equilibrium with the compressor. Such a condition is impossible to achieve in practice because, while at full power the turbine can produce power in excess of compressor requirements, at low outputs it cannot satisfy the compressor demands due to the small exhaust energy available from the engine. This arrangement is therefore not applicable.

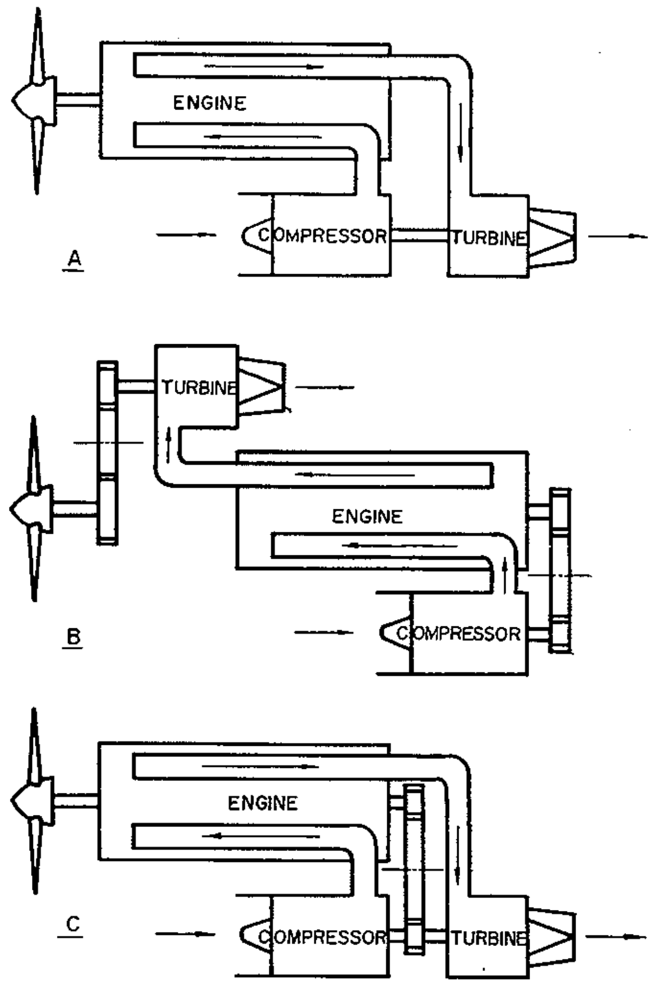

Fig. 6 – Mechanical arrangements for compounding

2. *The compressor may be driven by the piston engine – all output power being taken from the turbine (Fig. 6B).*

In this case the piston-engine-compressor combination operates purely as a gas generator for the turbine. Here the piston engine and compressor must remain in power equilibrium at all operating conditions, and the limitations arising from this requirement give rise to a number of disadvantages.

Firstly, at low power outputs the piston engine is prevented from making the contribution of which it is capable to the total power output, this resulting in very poor thermal efficiencies at part load conditions.

Secondly, due to the steep power law of the compressor against rpm the piston engine at cruising

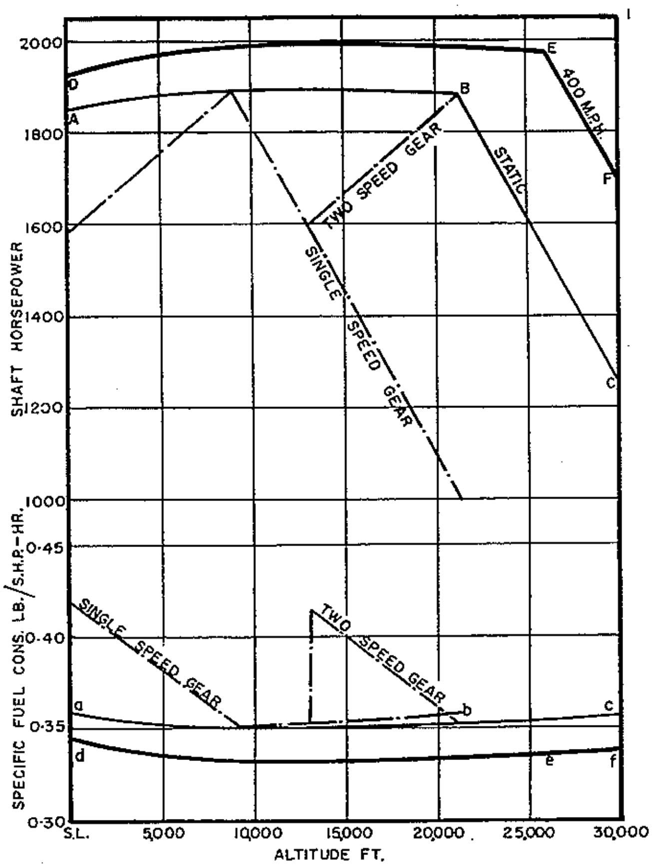

Fig. 7 – Altitude performance with fixed-gear and variable-gear superchargers

conditions requires operation at high piston speeds which results in greater wear and tear than necessary.

A further serious objection to this arrangement exists in relation to the powerplant weight, for it will be seen that two sets of high-ratio reduction gearing are required, one between the compressor and the engine crankshaft and the other between the turbine and the propeller shaft, and both of these are called on to transmit high powers.

3. *All three components may be coupled together to form a common mechanical system (Fig. 6C).*

In this system no requirement exists for power equilibrium between any two components, and this allows the greatest possible flexibility in the choice of operating conditions for best performances over the whole speed range. Furthermore, the lightest possible powerplant results for the compressor and turbine, which have sympathetic characteristics on a speed basis, can be connected solidly together, the reduction gearing connecting them into the engine system being called upon to transmit only the difference in the power produced by the turbine and that demanded by the compressor.

This is the basic mechanical system employed in the Nomad engine.

Altitude Performance – The powerplant layout suggested in Fig. 6C suffers from the defect common to all engines which depend upon a fixed-ratio gear-driven compressor in that its power falls off rapidly as altitude is increased, its power and fuel consumption curves for some part-load condition being of the form shown by the broken lines in Fig. 7. Some improvement may be obtained by the provision of a two-ratio gear between the engine and the turbocompressor set, in which case these results would be modified in the manner shown. A much more satisfying solution to the problem can be obtained if an "infinitely variable gear" is employed, in which case the power and consumption lines can be maintained at sensibly constant values over a wide range of altitudes, as shown by the full lines ABC and abc in the diagram. In fact, the employment of such a gear confers still greater benefits, for it permits the gear ratio at any flight condition to be adjusted to take full advantage of air intake "ram" resulting from forward speed so that at all times the compressor power absorption is kept to a minimum, thus enabling enhanced performances to be obtained, as shown by the lines DEF and def.

A further advantage of the variable gear is that it enables the compressor to be driven up to its maximum speed at any engine rpm, thus permitting greater maximum power altitudes to be achieved at low engine speeds.

Due to certain favorable factors which will be referred to later, it has been found possible to embody such an "infinitely variable gear" in the Nomad engine.

The Napier Nomad Engine

Having discussed briefly the basic problems involved in compounding, it is now possible to examine the practical solutions to these problems as demonstrated in the Napier Nomad and to consider the results obtained from this engine in relation to aircraft requirements.

Mechanical Details – The mechanical layout of the Nomad is shown in Fig. 8, and it will be seen that this follows the arrangement suggested in Fig. 6C, except that an "infinitely variable gear" has been interposed in the gear system between the turbocompressor set and the engine. This permits a range of gear ratios to be selected for any crankshaft speed.

The diesel engine has 12 cylinders in banks of six, horizontally opposed on a six-throw crankshaft, and the cylinder design is of the simplest possible 2-stroke type, using piston-controlled ports with no valve gear of any kind. The loop-scavenging system employed is that in which the incoming air is directed against the cylinder wall with an upward movement towards the combustion chamber, as shown in Fig. 9. This type of airflow not only gives effective scavenging of burned gases from the cylinder, but due to the well-organized flow path it permits high airflows with minimum aerodynamic losses.

The combustion chamber is hemispherical in form with a centrally disposed injector having one central orifice and five equally spaced radial orifices, the only unusual feature being that the radial sprays are directed at the combustion-chamber walls, experience having proved this to be beneficial.

The injector pumps, which are in blocks of six, are of normal "jerk-pump" design, specially developed to deal with the high outputs and speeds of operation required.

The pistons, which are illustrated in Fig. 10, have Y-alloy bodies but are fitted with austenitic steel tops designed to run at temperatures in the center of the crown between 1100 and 1300 F at full power. Oil cooling is provided behind the piston rings to ensure minimum temperatures at this point.

The unidirectional loadings on the connecting rods resulting from two-stroke operation permit the use of an unusual design, as shown in Fig. 11, both small- and big-end bearings being of the half-bearing or "slipper" type, with light straps as safety devices in the reverse-loading direction.

A further development of this type of bearing is applied at the small end. On this bearing high loadings are experienced accompanied by low rubbing velocities, and due to the unidirectional loadings no separation of the surfaces occurs on a normal bearing to permit the ingress of lubricating oil.

On the Nomad this difficulty has been overcome in the manner illustrated in Fig. 12.

The bearing is divided lengthwise into three sections comprising two outer bearings "X" and a center bearing "Y." The outer bearings are coaxial with each other but their centers are displaced transversely from the centerline of the connecting rod. The axis of the center bearing is similarly displaced on the opposite side of the connecting-rod axis. The three bearings therefore constitute, in effect, two bearings eccentric to one another, their eccentricity being equally spaced about the axis of the connecting rod. In practice the distance between centers of these two bearings is only 0.035 in., but the operation of the arrangement can be best understood if this dimension is greatly exaggerated as shown in B, C, and D of the diagram, from which it will be seen that relief of loading and separation between the surfaces is obtained through the rocking of the connecting rod.

The axial compressor has 12 stages and operates at a maximum pressure ratio of 8.25/1 with an air mass flow of 13 lb per sec, and to extend the operat-

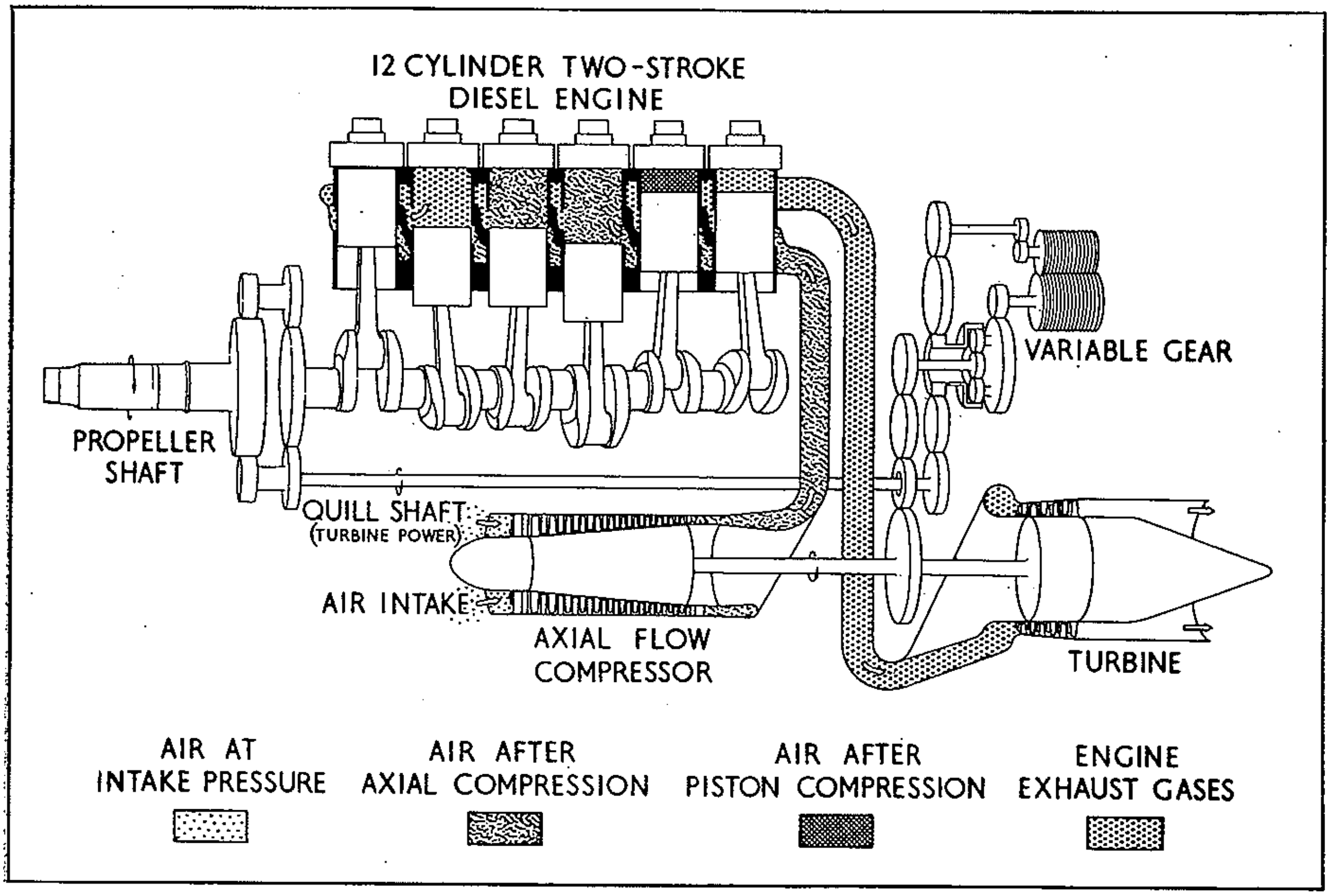

Fig. 8 – Diagram of Nomad engine

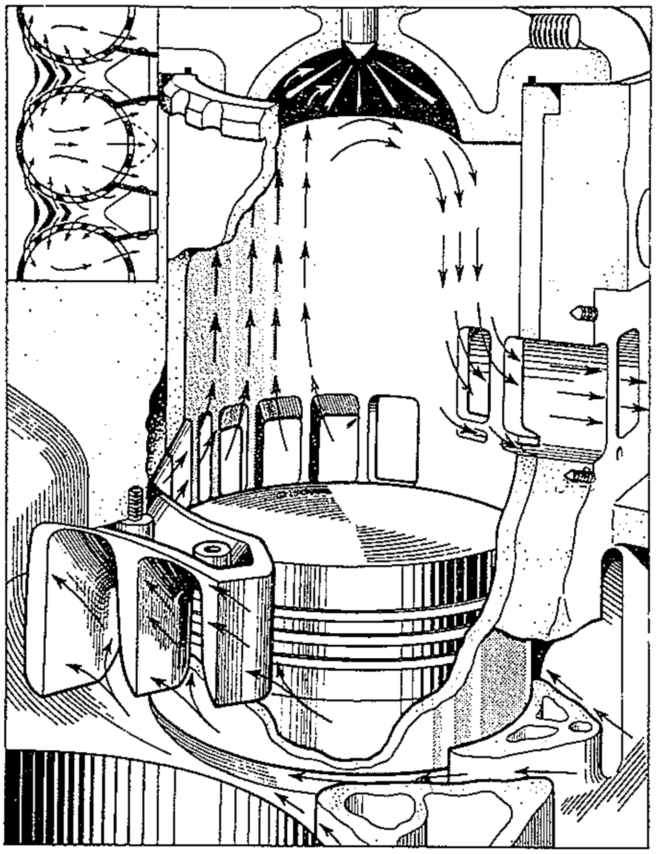

Fig. 9 – Arrangement of engine cylinder

ing range at low-speed end, adjustable inlet guide vanes are fitted.

The adiabatic efficiency of the compressor at take-off (sea-level static) is 85% and its peak efficiency 87.5%.

The three-stage turbine is mounted coaxially with the compressor, and its blading is designed to extract maximum energy from the gases, the residual jet thrusts being small. Its efficiency at take-off (sea-level static) is 84% and 86% at the altitude cruising rating.

Both compressor and turbine are connected into the rear gear casing by shafts having toothed couplings at each end to allow for expansion effects.

The Infinitely Variable Gear – This gear has been designed and developed by Napier to suit the particular requirements of the Nomad application, but the basic scheme was evolved by Dr. Joseph Beier and developed in Germany before the 1939-1945 war. It is an ingenious practical adaptation of the well-known scheme in which two conical members are end-loaded together, the gear ratio being varied by sliding one cone over the other. The actual details of the Beier version of this basic scheme are as shown in Fig. 13. Here a "pack" of discs with narrow conical rims are mounted on a central shaft and spring-loaded to trap between them a series of coned discs carried on a planetary shaft, which is arranged to swing about a fulcrum to obtain changes in gear ratio. By attaching a gear to the planetary shaft meshing with one at the fulcrum, it is possible to arrange a constant mesh gear system so that changes in gear ratio can be made while running. It will be clear that a number of planetary shafts can be similarly disposed around the center shaft, and in the Nomad three sets of planetary discs are used.

This type of gear would normally be classified as a "friction" device; but, in this particular application, the drag force between the discs is obtained by fluid shear of the oil film at the contact points, the end loads being adjusted in relation to the lubrication conditions to obtain the most efficient operation.

A great advantage of this device is that it transmits power through a multiplicity of contact points, the number of which can be varied experimentally to achieve most satisfactory performances.

To reduce the dimensions and weight of the variable gear to a minimum, it is connected into the gear system of the Nomad, as indicated in Fig. 14.

Here the speed-reducing gear for the turbocompressor set is of epicyclic form, the variable-speed device being connected across two members of the epicyclic train. This, in effect, produces two mechanical systems in parallel, only about 30% of the total power flow being transmitted through that part of the system in which the speed-varying device is fitted.

The arrangement of this complete system in the Nomad is shown in Fig. 15.

The Complete Engine – A longitudinal section through the engine is shown in Fig. 16, the layout following closely the diagrammatic arrangement of Fig. 8.

It will be observed that a torsional vibration damper of viscous-fluid type is attached to the free end of the crankshaft. This item is not necessary for any condition of normal operation but is provided to protect the engine in the event of one cylin-

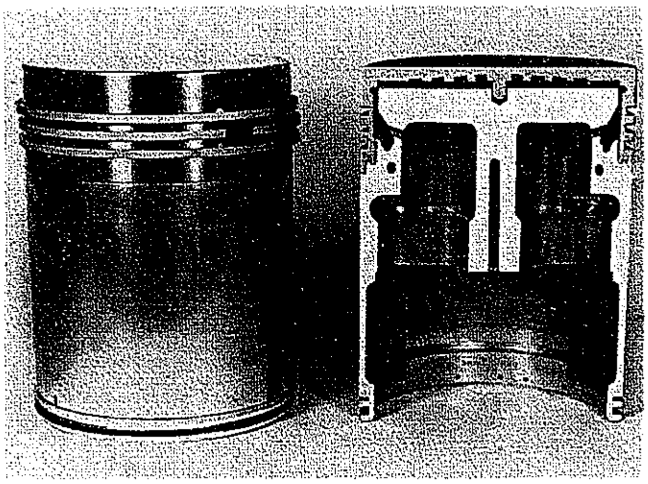

Fig. 10 – Arrangement of piston

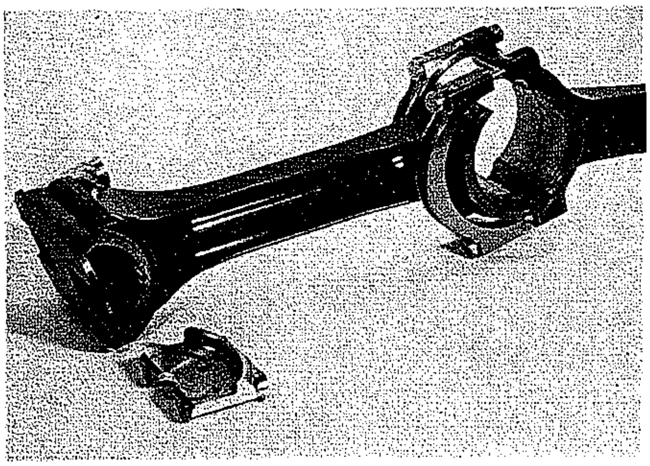

Fig. 11 – Arrangement of connecting rod

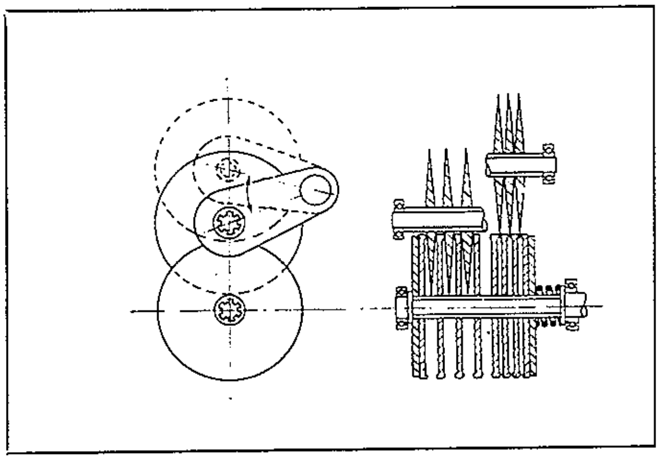

Fig. 13 – Diagram showing operation of variable gear

der "cutting out" due to seizure of a fuel injector pump.

Fig. 17 shows the engine in cross-section and indicates clearly the vertically divided magnesium crankcase and the aluminum cylinder blocks, the whole structure being bolted together by steel through-bolts. It will be seen that dry liners are employed, the liners themselves being of chromium-copper material chromium plated in the bores. The dimensions on this drawing show that the frontal area of the engine is small, in piston-engine terms, the shape being convenient for cowling purposes.

Fig. 18 shows a cutaway drawing from which the main features may be identified. The reduction gear is of conventional form using straight spur gears, two layshafts transmitting power from the engine crankshaft to the output gear, a further two providing the power connection between the output gear and the turbocompressor combination.

Fig. 19 is a photograph from one side and shows that the engine is suspended from four pickup points, two at the front end of the crankcase and two on the rear gear casing. This arrangement is convenient from the installation point of view and gives maximum accessibility, particularly to the compressor and turbine.

Starting is by electric motor, and as the compression ratio in the engine cylinders is insufficient to give self-ignition under these conditions, spark plugs are fitted in each cylinder head activated from a high energy system. These plugs are used only for starting.

A three-quarter front view is shown in Fig. 20.

Engine Operation and Control – The engine is equipped with a single lever control system which relates the variable factors together to produce a

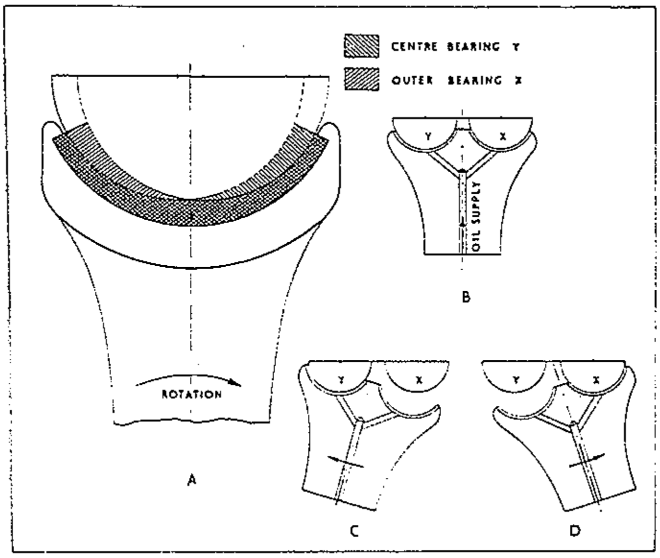

Fig. 12 – Connecting-rod small-end bearing

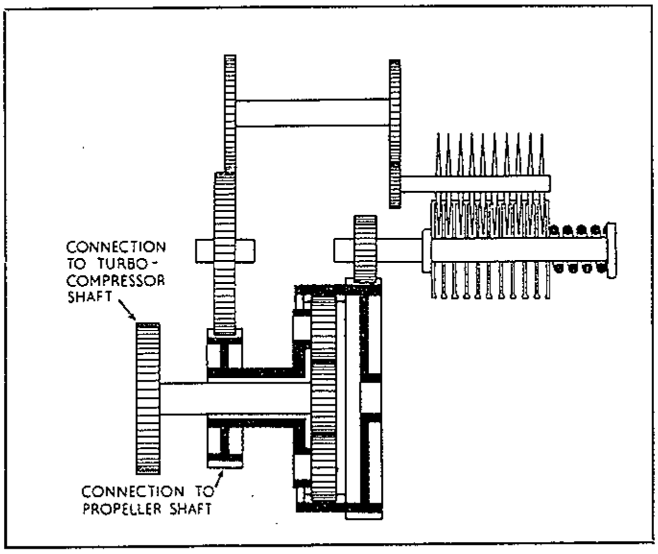

Fig. 14 – Connection of variable gear to epicyclic train

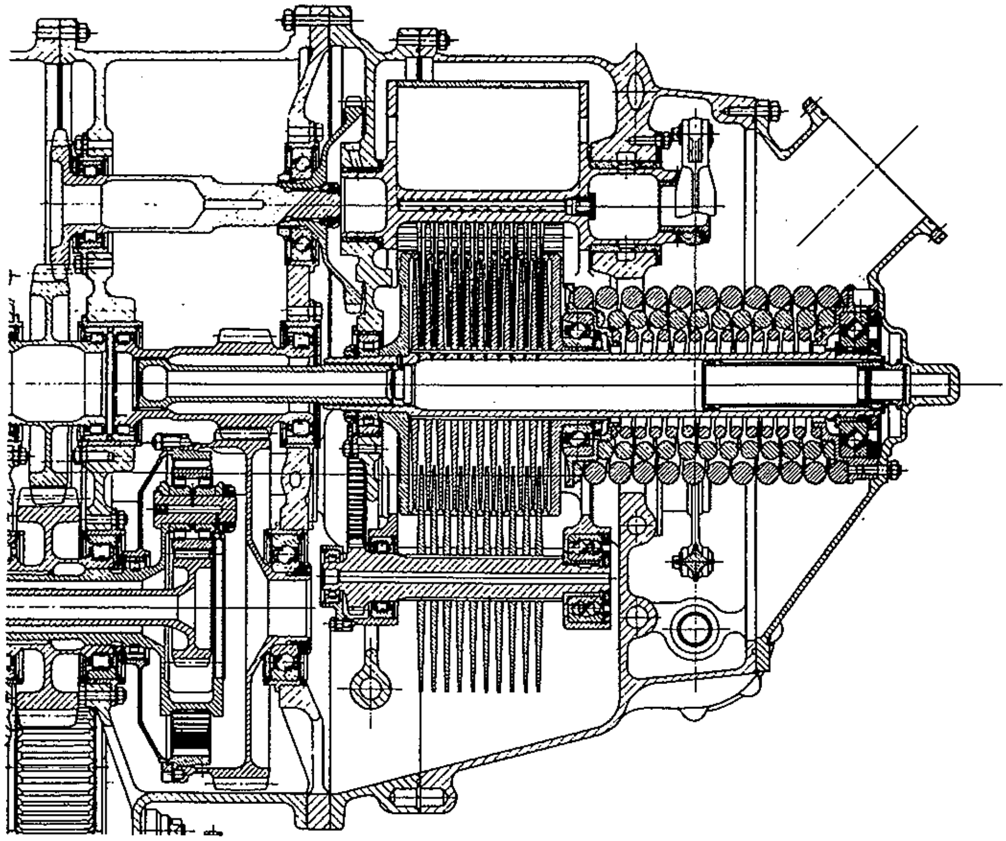

Fig. 15 – Variable gear as fitted to engine

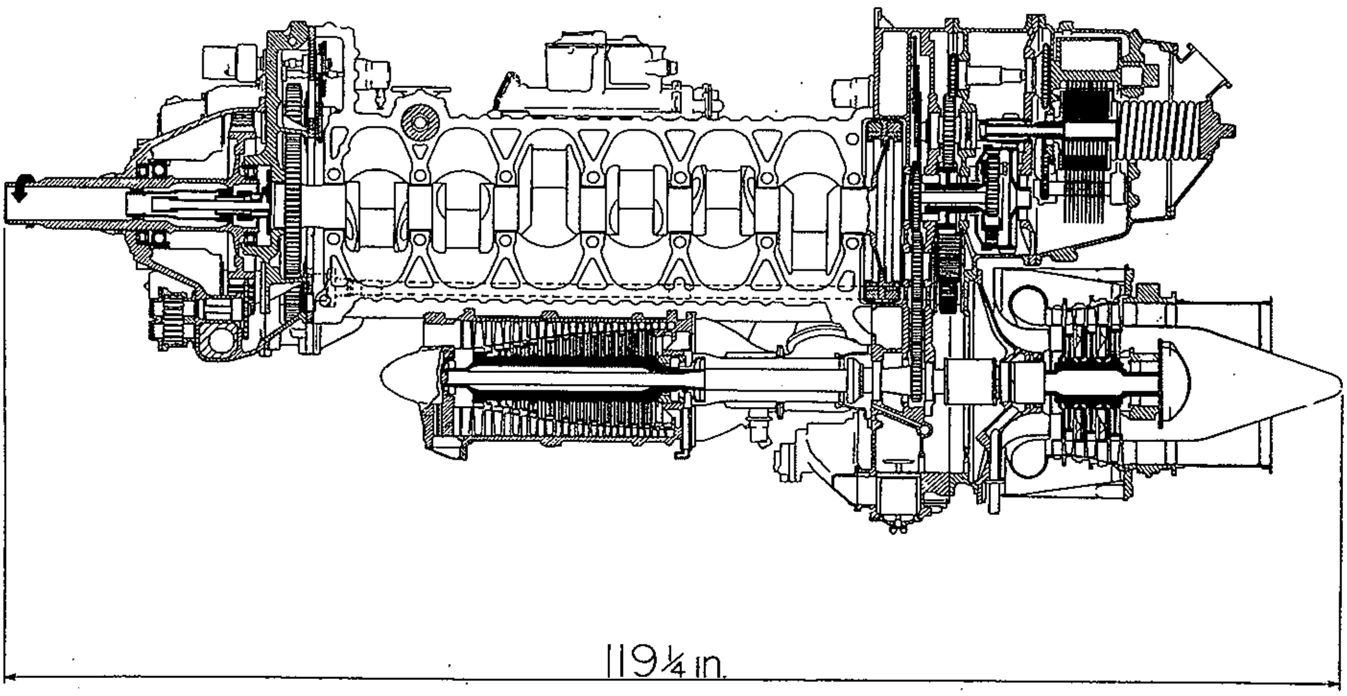

Fig. 16 – Longitudinal section of Nomad engine

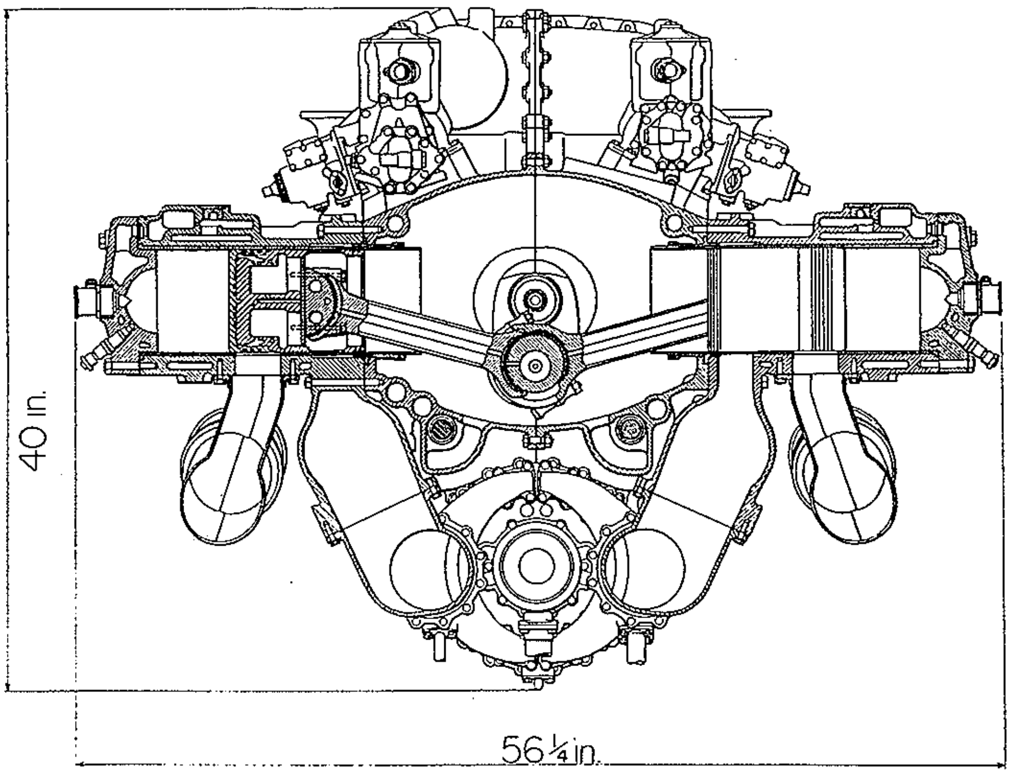

Fig. 17 – Cross-section of Nomad engine

predetermined power law against rpm. The speed function is controlled by a normal mechanically set constant-speed unit varying propeller pitch, each speed selection being mechanically related to a rack setting on the fuel pumps governing the fuel flow, and to a selected boost pressure. The boost pressure setting is applied to a normal type of variable-datum boost-control unit, the servo piston of which operates the setting of the infinitely variable gear, the ratio of which is thereby automatically adjusted to maintain the selected boost pressure. Any selected rpm, fuel flow, and boost pressure are in this way automatically maintained with increasing altitude until the maximum permitted rpm of the turbocompressor set is reached, this corresponding to the maximum power altitude for that particular engine condition. Above this point resetting mechanism is provided which reduces fuel flow in sympathy with decreasing boost. The engine is therefore capable of maintaining a constant selected boost and power with corresponding fuel consumptions over a wide range of altitudes which at 60% of take-off power extends from sea level up to 25,000 ft.

For optimum matching over the full power and altitude range of the engine, a variable-capacity compressor is necessary. This change in capacity is achieved by an automatic device which adjusts the incident angle of the compressor inlet guide vanes to a function of inlet pressure with a variable datum controlled by boost pressure.

Nomad Engine Performances

Brief reference was made in the section on The Compound Engine to some factors influencing the division of work between the diesel engine and the turbine. In the actual case this problem is more complex than was indicated, for there are other aspects of engine performance, such as altitude requirements and the fuel consumptions at part loads, which must also be considered. Taking all the design requirements into account and also taking advantage of considerable experimental data, the major operating factors used in the Nomad and the resulting performances are summarized in the following text.

Take-Off Performance of Basic Engine – The supercharging pressure used in the diesel engine for take-off is 89 psia, this corresponding to a volumetric compression ratio of 3.36/1. When this is multiplied by the actual compression ratio in the engine cylinders based on effective stroke (8/1) we arrive at an overall compression ratio of 27/1.

Allowing for air intake and ducting losses the

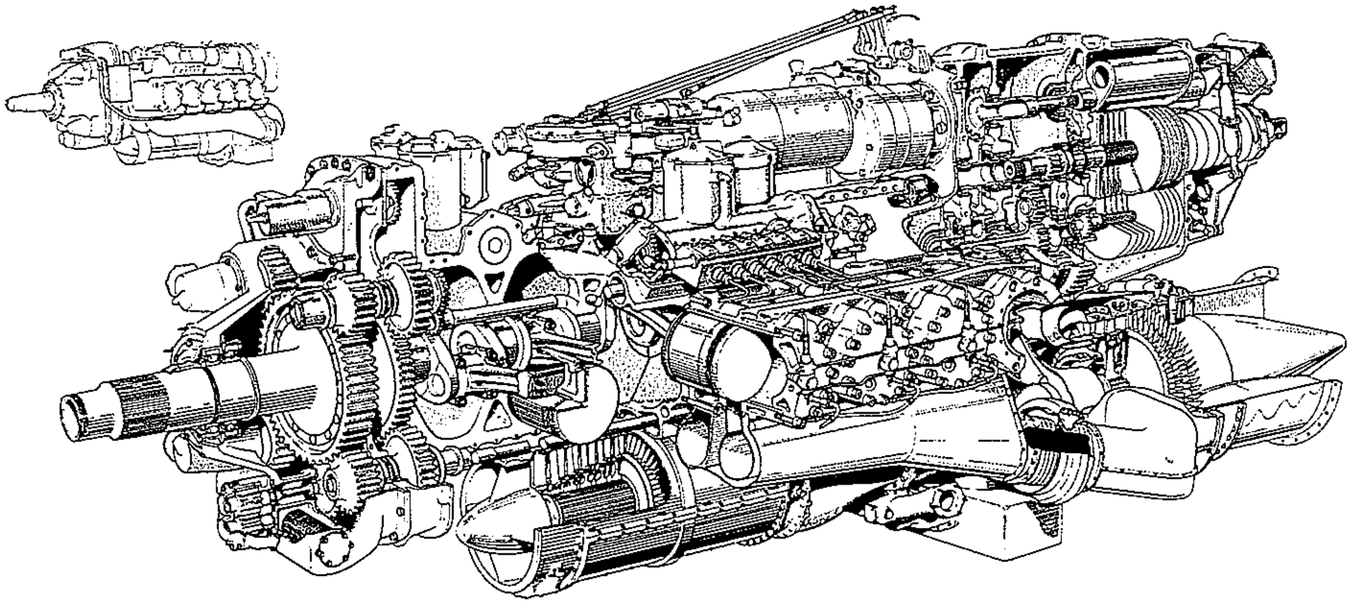

Fig. 18 – Cutaway view of engine

pressure ratio demanded from the compressor to provide the supercharging pressure of 89 psia at sea level is 6.28/1. The maximum pressure ratio of the compressor at its top speed is 8.25/1, and this permits the take-off boost to be maintained up to 7750 ft by variation in gear ratio.

The air/fuel ratio dictated by the cylinder performance requirements is 40/1, corresponding to an air mass flow of 13 lb per sec; and when operating with a maximum pressure limitation of 2200 psi a brake mean effective pressure of 205 psi is developed in the cylinders which have a bore of 6 in. and a total stroke of 7.375 in., giving a swept volume of 2502 cu in.

At the maximum crankshaft speed of 2050 rpm the piston speed is 2520 fpm, and the bhp developed by the diesel engine alone, neglecting any contribution from the turbine, is 2660, or 1.06 hp per cu in.

The ability of the engine to operate at such high outputs is an indication of the benefits derived from the excess air factor, for the recorded piston-ring temperatures at this power are no higher than those on a normal diesel engine developing less than half the power at equivalent piston speeds.

The pressure drop across the engine cylinders at take-off power is 13.0 psi, giving a turbine inlet pressure of 76 psia, and the temperature at the turbine inlet is 1222 F.

The pressure ratio across the turbine at sea level is 4.6/1, and this corresponds to a volumetric expansion ratio of 3.36/1 which when multiplied by the diesel expansion ratio of 8/1 gives an overall

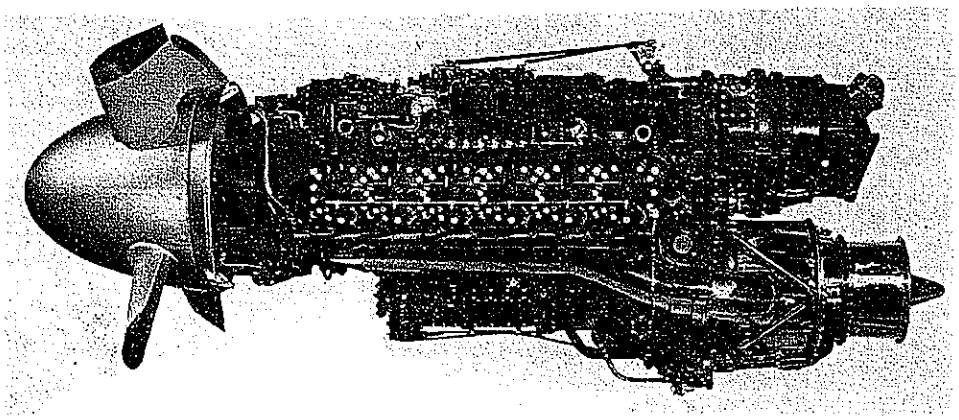

Fig. 19 – Side view of Nomad engine

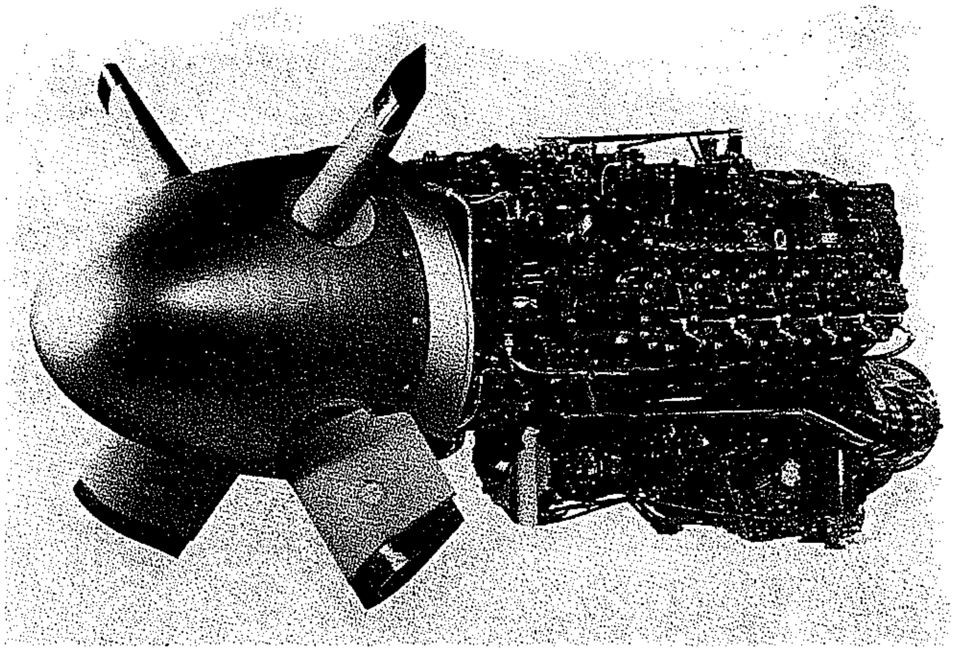

Fig. 20 – Three-quarter front view of Nomad engine

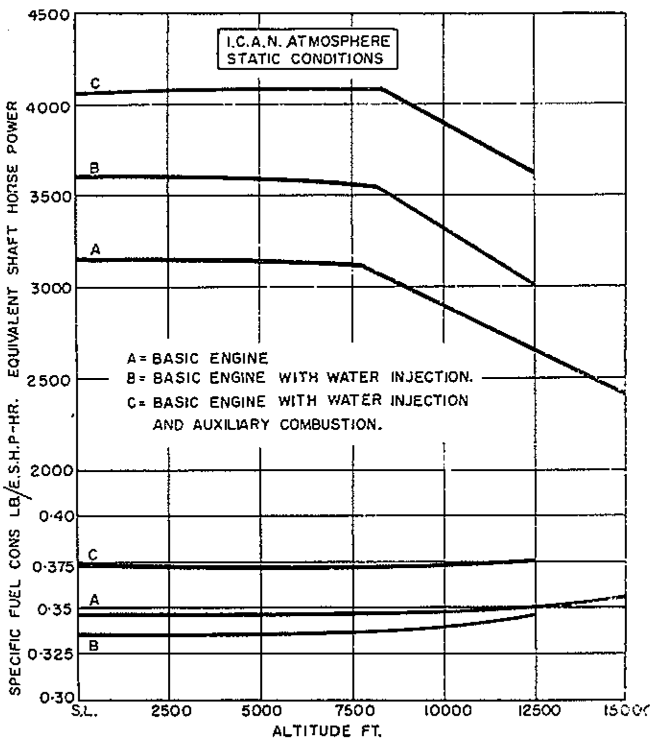

Fig. 21 – Take-off powers against altitude with fuel consumption

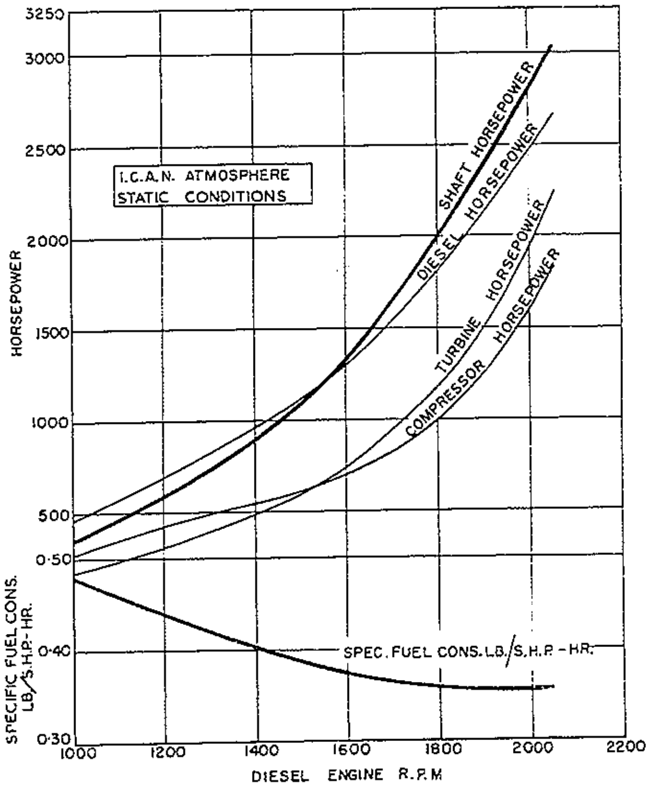

Fig. 22 – Sea-level static interconnection power curve

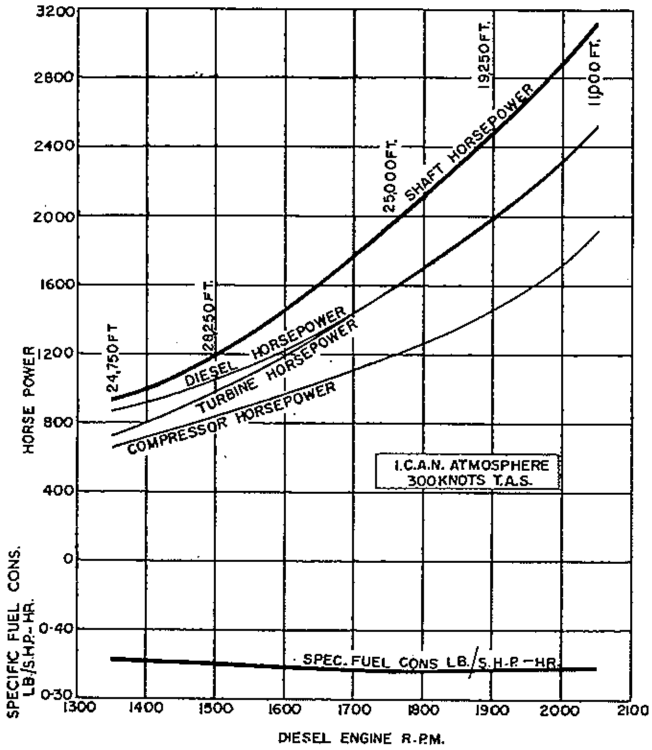

Fig. 23 – Altitude interconnection power curve

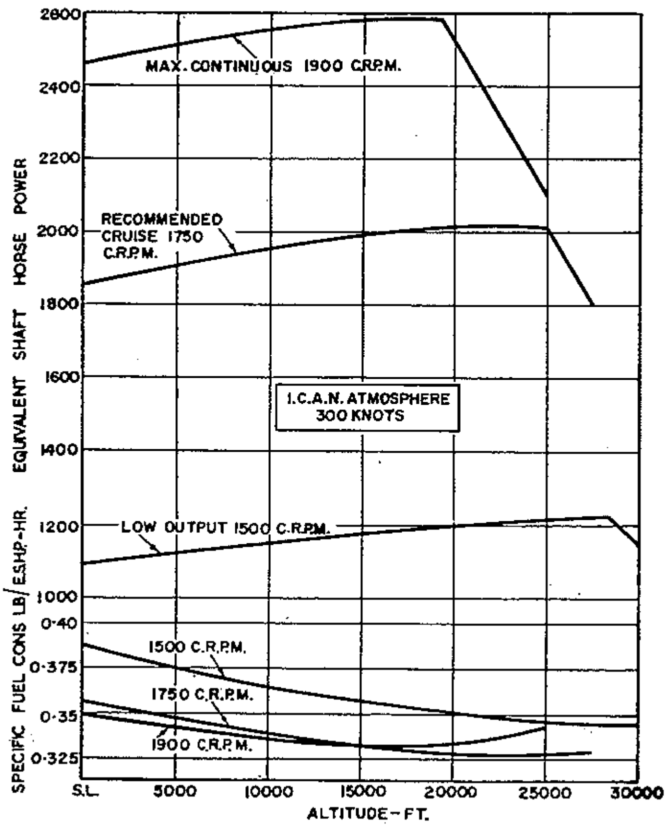

Fig. 24 – Altitude cruising curves with fuel consumptions

expansion ratio of 27/1.

The power developed by the turbine at sea level is 2250 hp of which 1840 hp is absorbed in driving the compressor, so that there is an excess of 410 hp, less a slight power loss of about 20 hp in the variable gear, as an addition to the diesel-engine output. The equivalent horsepower developed by the engine, taking into account the small amount of jet thrust available, becomes 3135 hp or 1.25 hp per cu in.

The net dry weight of the engine is 3580 lb, so that the specific weight is 1.175 lb per hp.

The specific fuel consumption based on the equivalent power is 0.345 lb per ehp per hr.

Take-Off Performance with Water Injection – With the high degree of supercharge used for take-off the charge air temperature is high, about 477 F at sea level. This could be reduced by aftercooling, but this would involve additional bulk and weight together with a serious loss of energy from the cycle. The charge temperature can be reduced most conveniently and its density increased without energy loss by the injection of water into the inlet manifolds, and when this is done a greater quantity of air can be packed into the engine cylinders and the power output still further increased. Using water injection in this way, the equivalent power of the engine is increased to 3580 hp (1.43 hp per cu in.) and the specific weight is reduced to 1.0 lb per hp. The specific fuel consumption is also reduced to 0.336 lb per ehp per hr.

Take-Off Performance with Auxiliary Combustion – A further possibility for increasing the maximum power output arises from the presence of excess air in the exhaust manifold between the engine and the turbine, for, at the expense of a slight increase in the turbine inlet temperature, a small quantity of additional fuel can be injected and burned at this point. This extra fuel is burned at low expansion ratio and therefore at low efficiency, but the effect on the overall specific consumption is small.

This system of auxiliary combustion can be used in association with water injection, and for a turbine inlet temperature of 1377 F the take-off power can be increased to 4100 hp (1.64 hp per cu in.), the specific weight being thereby reduced to 0.93 lb per ehp. The specific fuel consumption in this case is increased to 0.374 lb per ehp per hr.

The take-off performances against altitude for the three conditions referred to are illustrated in Fig. 21.

Cruising Performances at Altitude – The power law to which the engine is controlled over the speed range under sea-level static conditions is shown in Fig. 22, the shape of this curve being devised to maintain optimum thermal efficiency and altitude performance over as wide a range as possible and also to suit control requirement of the propeller.

On the same diagram are shown also the corresponding diesel, turbine, and compressor powers, and it will be observed that at 1500 rpm the turbine power exactly balances that absorbed by the compressor. At lower engine speeds the power developed by the turbine is below that demanded by the compressor, and this power deficiency has to be fed in from the diesel engine.

The engine rpm at which balance occurs between the turbine and compressor powers varies with altitude and forward speed, but over the useful cruising range the power transmitted through the infinitely variable gear is relatively small.

From the section on Engine Operation and Control it will be clear that as altitude is increased with the engine operating at any point on the interconnection curve, the boost pressure appropriate to that condition will be maintained by progressive adjustment of gear ratio until the limiting rpm of the turbocompressor set is reached, this representing the maximum power altitude for that particular operating condition. Bearing in mind the reduction in compressor work necessary to achieve the selected boost pressure due to forward speed and the improvements in compressor and turbine efficiency derived from increasing altitude, there results a continuous increase in power output until maximum power altitude is reached, with corresponding reductions in specific fuel consumption.

The power interconnection curve extended up to the maximum power altitudes at each value of rpm with a forward speed of 300 knots and the corresponding component powers are shown in Fig. 23. It will be observed that over the whole operating range at these conditions the specific fuel consumption remains sensibly constant between 0.34 and 0.36 lb per shp per hr.

Fig. 24 shows performances at varying altitudes with selected engine ratings, and again the constant values for both power and fuel consumptions are worthy of note. The best fuel economy of 0.326 lb per ehp per hr is achieved at the recommended cruising rating at 22,250 ft where a power of 2027 ehp is developed.

A complete heat balance for this operating point is depicted in Fig. 25, and from this it will be seen that a brake thermal efficiency of 42% is obtained, which is higher than that of any other aircraft engine.

Engine Weights – It is customary to compare weights of engines on the basis of the weight per horsepower produced under maximum power conditions.

This method of comparison is valid for some purposes but does not provide an accurate indication of the effects of the engine weight in flight for it fails to take into account the varying altitude performances of which the engines are capable.

This point is of importance in comparing the Nomad with other types of engine, for its ability to maintain constant power over a wide band of altitudes has the effect of reducing its effective specific weight under operating conditions.

In Fig. 26 a comparison is shown between the

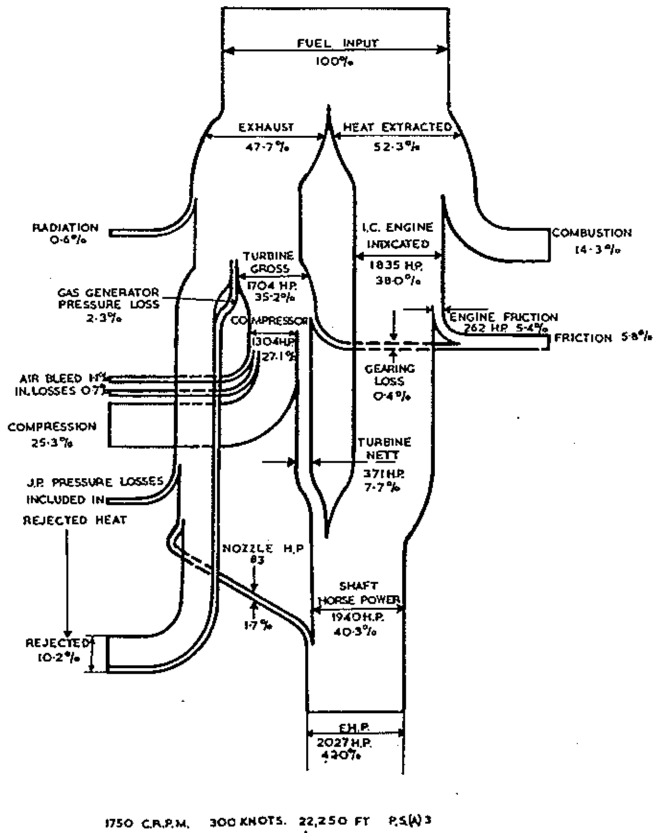

Fig. 25 – Heat balance diagram

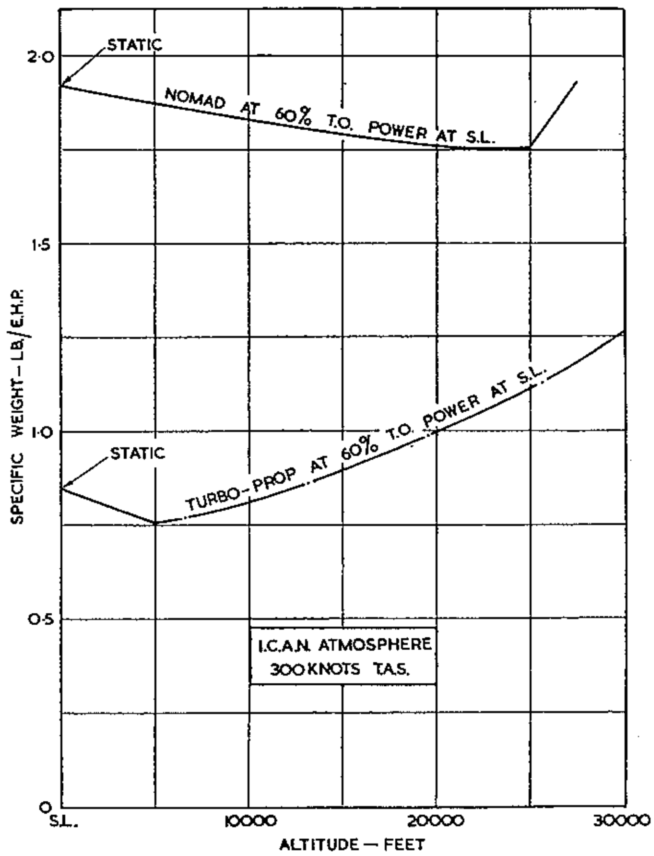

Fig. 26 – Specific weights at altitude

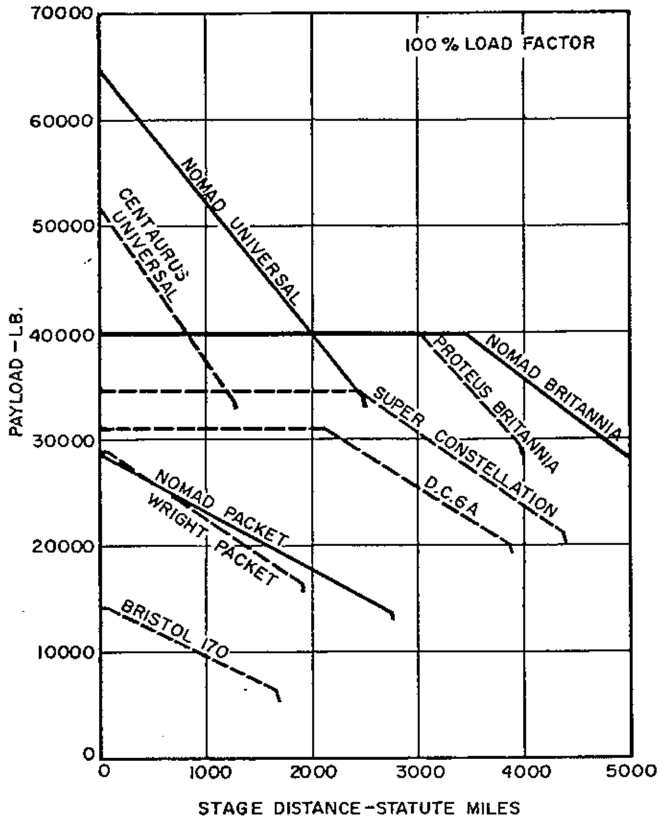

Fig. 27 – Variation in payload with stage distance

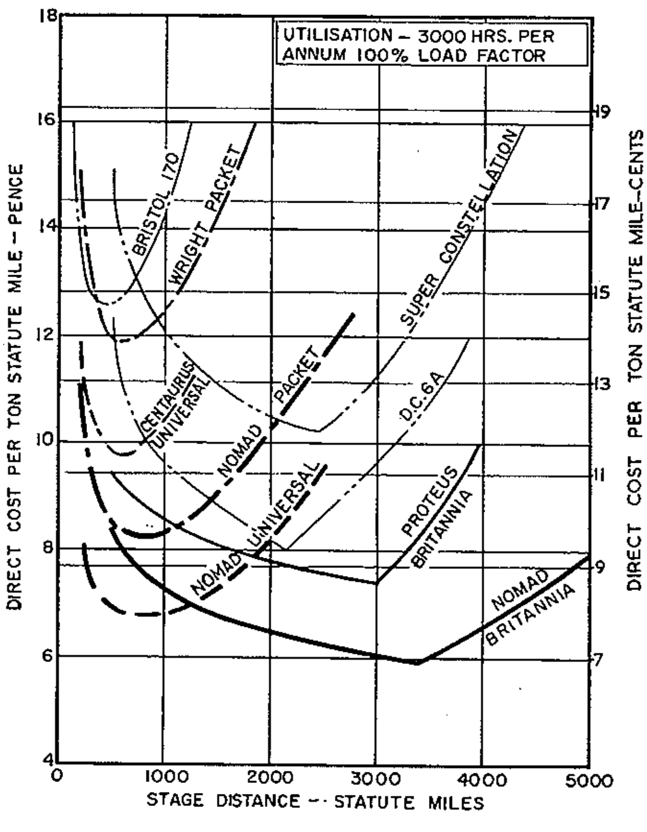

Fig. 28 – Improvement in operating costs per long ton-mile

specific weights under altitude power conditions between the Nomad with a specific weight of 1.17 lb per hp at take-off power and a turboprop engine weighing 0.5 lb per hp at take-off power. Both engines are considered to deliver a cruising power under sea-level static conditions of 60% of the take-off power, this increasing the specific weight figures to 1.9 and 0.83 lb per hp, respectively.

Due to the shapes of the power curves of the two engines against altitude the Nomad specific weight is reduced right up to the critical altitude, whereas the turboprop specific weight increases steadily as altitude is increased.

The difference between the two engines is therefore much less at 25,000 ft than at sea level. This factor must therefore be taken into account when aircraft performance comparisons are made.

General Characteristics – Apart from the performance figures which have been quoted there are two further characteristics which arise from the nature of the operating cycle and which assume some importance.

Firstly, there is extreme adaptability in the matter of fuels. Designed to run on diesel fuel for maximum economy, the engine will operate equally well on kerosene or wide-cut gasoline.

Secondly, the engine is insensitive to increases in ambient temperature, losing less than 2% of its take-off power for each 20 F increase in temperature.

Nomad Engine in Modern Aviation

Consideration of the performance figures which have been quoted demonstrates that the outstanding characteristics of the Nomad engine are its low fuel consumptions, which are maintained unimpaired over a wide range of powers, altitudes, and flight speeds, and its ability to use cheap fuels.

These two factors combine to provide great economy in operation; but apart from this the smaller quantity of fuel consumed, as compared with other engines for a flight of given duration, reduces the proportion of all-up weight of the aircraft absorbed by fuel storage, thus enabling the useful payload to be increased. Alternatively, the original payload may be flown over a greater distance.

There are specific aircraft applications in which these characteristics can display very definite operational advantages over other types of powerplant, and it is necessary to consider these in two categories, those applied to military purposes and those for civil use.

In the military field economy of operation is a secondary consideration; it is the ability to perform a defined operation more effectively which assumes greater importance. On this basis the Nomad engine constitutes the most suitable powerplant for maritime reconnaissance aircraft, which have to undertake prolonged periods of search at low altitudes and low flight speeds at considerable distances from base. Here the long-range propensities of the engine, coupled with the ability to employ any desired type of flight plan without significant change in the quantity of fuel consumed, enable maximum effectiveness of the military operation to be achieved with minimum total duration of flight.

A further military application in which the Nomad shows to advantage is in the tactical role of army support. In such cases the necessity for large apertures in the fuselage for troop and supply dropping makes pressurization difficult, if not impossible, and the ability to carry maximum payloads at altitudes below that at which pressurization becomes necessary is one which is derived directly from the operating characteristics of the engine.

It is in the civil field, however, and particularly in air freighting operations, that the Nomad can make its biggest contribution to aviation at the present time; and it was, in fact, for this purpose that the engine design was originally undertaken.

There can be little doubt that the greatest opportunity for expansion in the aviation industry in the immediate future lies in the development of air freighting services; there are tremendous potential markets in this field which are almost entirely unexplored, even in the United States where more progress has been made than elsewhere.

For freighting purposes economy is the vital factor, and it seems certain that satisfactory operating economy will be achieved only by aircraft designed primarily for freighting. The development of such aircraft is of vital economic importance at the present time, not only in relation to the business which can result from expansion in freighting activities, but also in potential sales of the aircraft concerned. There exist already two such British aircraft which are of noteworthy performance in this particular field, the Bristol Britannia for the longer ranges and the Blackburn Universal Freighter for the shorter routes.

With the cooperation of the aircraft companies concerned, design studies have been made by Napier to assess the effects on operating characteristics of these aircraft if the engines now fitted were replaced by Nomads. In addition, to cater for cases in which the volume of traffic would not justify the use of such large aircraft, a similar study was undertaken relative to a civil version of the Fairchild Packet.

The results of these studies are shown in Figs. 27 and 28. It is not possible here to describe in detail the methods by which these curves have been derived; it must suffice to state that they are based on the standard SBAC method of computation; but the fuel costs which have been used are given in Table 2, being average world prices excluding the United States.

For comparative purposes the characteristics of three other well-known aircraft, the Bristol 170 Freighter, the Lockheed Super-Constellation, and the Douglas DC-6A with their existing engines are also shown on the curves.

Table 2 – Fuel Costs

Aircraft	Engines	Fuel	Fuel Cost per Imperial Gallon	Fuel Cost converted to Cents[a]
Bristol Britannia	Proteus	Aviation turbine fuel	1/10	25.7
Blackburn Universal	Centaurus	Gasoline 100/130 grade	2/7¼	36.5
Fairchild Packet	Wright compound	Gasoline 115/145 grade	2/9	38.5
All three aircraft	Nomad	Diesel fuel	1/8	23.3

[a] Assuming $2.80 per £ sterling.

Fig. 27 shows the change in payload resulting from the fitting of Nomad engines, and Fig. 28 shows the improvement in direct operating costs per British ton-mile.

Examination of these results indicates in no uncertain manner the contribution which the Nomad engine can make to the problem of economical freighting over short or long ranges of flight.

Acknowledgments

The authors wish to express their thanks to the Board of Directors of D. Napier & Son, Ltd. for permission to publish this paper and to the Bristol Aeroplane Co., Ltd. and Blackburn and General Aircraft, Ltd. for their agreement to display the results shown in Figs. 27 and 28. They are also grateful to their colleagues at Napier, who gave assistance in the preparation of the paper.

DISCUSSION

Discusses Advantages of High-Pressure Turbocharging

– Rudolph Birmann
De Laval Steam Turbine Co.

THE individual components of the Nomad engine – that is, the loop-scavenging 2-stroke engine, the low-pressure turbine-driven compressor, and the variable-ratio turbine speed reduction gear – are outstanding developments for which the designers and the manufacturer deserve a great deal of credit. The integration of these three components into a successful aircraft powerplant represents an even greater accomplishment.

The performance figures given in the paper speak for themselves, actually proving that a compound engine of this type can have a definite place in the field of aircraft propulsion, primarily because of its extraordinarily low fuel consumption and its excellent power/weight ratio. The fuel consumption achieved is far lower than can be expected from the pure gas turbine for many years to come and lower than is possible with the piston type of engine alone. This achievement is made possible by compounding the piston engine with a turbo machine in such a manner that the reciprocating engine handles the high-pressure, high-temperature end of both compression and expansion and the turbo machine handles the low-pressure end. Such compounding takes advantage of the fact that the piston type of engine is the more suitable to cope with high pressures and high temperatures, whereas the turbo machine has a distinct weight, size, and performance advantage in the low-pressure range where the temperatures are comparatively moderate.

Because of the advantages to be derived from compounding the piston engine with the turbo machine, the compound engine has a promising future in the field of aircraft propulsion, and in the form of high-pressure turbocharging compounding it has already gone a long way toward keeping the reciprocating internal-combustion engine competitive with other prime movers in the fields of stationary, marine, and locomotive powerplants.

In all fields of application the future of the compound engine is assured; however, this is only if the engine layout is made as simple as possible to permit the attainment of best possible reliability and low production and maintenance cost. From this point of view it seems regrettable that the Nomad engine is perhaps more complicated than it need be. To reap the advantages of compounding it is not necessary to connect mechanically the exhaust-turbine-driven compressor to the engine crankshaft by means of a variable-ratio reduction gear. The much simpler alternative – that of all output power being delivered by the piston engine – could easily give equally good or even better results, but it is passed over by the authors with the statement that in practice it is impossible to achieve power equilibrium between compressor and turbine over the whole operating range. In reality there are an impressive number of turbocharged two-cycle marine engines in service, which contradicts the authors' misgivings regarding this much simpler compounding arrangement. The high-pressure turbocharging of diesel engines (employing compressor pressure ratios of 3/1) is rapidly coming into use, and judging from experience with heavy-duty stationary engines there is no reason why a similar compound arrangement of an even higher-pressure-ratio turbocharger and a reciprocating engine could not be evolved for aircraft application. As a matter of fact, the development of the Nomad engine has solved the most difficult problems involved – namely, the development of an exhaust-turbine-driven compressor capable of a combined efficiency of over 72% and suitable to produce a compressor pressure ratio of 8.25/1, and the development of a two-stroke engine designed to operate with inlet manifold pressures of 89.0 psia.

It is somewhat puzzling why, after having developed such outstanding components, these components were not combined in a simple, high-pressure-turbocharged engine – in other words, why the development of the Nomad engine was made much more difficult by the introduction of the complicated mechanical connection between the low-pressure turbo machinery and the high-pressure reciprocating engine. This question is justified in view of the outstanding performance already achieved by high-pressure turbocharger-engine combinations. Fuel consumptions as low as 0.306 lb per bhp per hr have been recorded, even though none of the test installations had the advantage of the high pressure ratios and high combined efficiencies of the Nomad turbocompressor. Turbocharged two-stroke engines, in which the turbocharger constitutes the sole source of air supply to the engine and has no mechanical connection of any kind thereto, are in everyday service.

It would be interesting to speculate on what the results would be if the reciprocating and turbo components of the Nomad engine were combined into a straightforward, high-pressure-turbocharged engine, in which the precompressed air was cooled in an intercooler. In such an arrangement the complicated variable-ratio reduction gearing (presently connecting the turbine to the engine crankshaft) would be eliminated and replaced by a simple intercooler. It should be brought out at this point that intercooling would not constitute a serious loss of energy from the cycle, as the authors indicate in the section on Take-Off Performance with Water Injection. As a matter of fact, the great merit of intercooling lies in the reduction of the negative work required for the compression in the engine cylinders – a reduction that can easily amount to much more than the 371 turbine horsepower which is delivered to the engine crankshaft through the reduction gearing. This reduction in the work of compression is the underlying reason for the sensitivity of the thermal efficiency of the diesel cycle to the charge temperature at the beginning of the compression stroke – the lower this temperature, the higher the thermal efficiency.

Straightforward turbocharging, with intercooling, maintaining the present pressure levels employed in the Napier Nomad engine and the turbine and compressor performance presently obtained, would result in numerous advantages over the present Nomad powerplant. It is estimated – on the basis of experience with high-pressure turbocharging of both 2- and 4-stroke engines – that these advantages would be as follows:

1. The entire engine layout would be more simple, correspondingly more reliable, lighter, and cheaper to manufacture.

2. The relationship of turbocharger and engine would not be dictated by considerations of the reduction gear, and the turbocharger could be so located that less frontal area and a more desirable configuration of the engine nacelle would be achieved, and perhaps even provisions for the recovery of pulsating exhaust energy could be accommodated.

3. The turbocharger, in lieu of producing net brake horsepower output, would produce a higher differential between intake and exhaust manifold pressures. This higher pressure differential would make it possible to pass the same amount of scavenging airflow through ports having a smaller area. In other words, the excessive height (63% of the stroke) of the intake and exhaust ports in the Nomad engine could be reduced and the ratio of effective stroke to total stroke made more favorable, thereby resulting in increased engine output and a better engine performance.

4. For the same net total output the thermal loading of the engine would be lowered by bringing about a lowering of the entire internal temperature level in the cylinders; and alternately, for the same thermal loading of the engine, the total output of the engine would be substantially increased.

5. Because of the reduction of the thermal loading (which can be brought about if the total output of the engine is maintained the same), the size, weight, and drag of the cooling radiator would be reduced sufficiently to make up for the weight and drag of the intercooler.

6. For the same total output, the specific fuel consumption would be reduced, probably to less than 0.3 lb per bhp per hr.

The authors deserve unstinted praise and congratulations for successfully going far beyond any previous development of compounding the reciprocating engine with the turbo machine. They are to be particularly commended for their pioneering work in selecting an extraordinarily high pressure level for the thermodynamic division between these two types of machinery, thereby pointing the way toward further development in high-pressure turbocharging of all types of internal-combustion engines.

Describes Double-Slipper Connecting-Rod Design

– Gregory Flynn, Jr.
Research Laboratories Division, GMC

THE Napier Nomad diesel engine is obviously a most interesting design and represents a remarkable engineering achievement. The paper certainly should prove stimulating to engine designers.

As one who is concerned primarily with the diesel engine I am carefully going to sidestep getting into any arguments between the turbojet, turboprop, and reciprocating-engine airplane drivers. Instead, I would like to confine my remarks to that part of the paper concerning the mechanical features of the diesel engine itself, which I found most interesting.

I have a few questions and some comments to make on various parts which perhaps would best be handled in the order in which they appear in the paper.

1. To accomplish the objectives of Napier as stated in the paper, the highly supercharged, compounded, 2-stroke diesel was, I believe, a wise choice. The compression ratios, pressures, and efficiencies are reminiscent of the work currently being done in France with the Pescara cycle which also uses a highly supercharged 2-stroke cylinder and a gas turbine.

2. The elimination of valve gear is always desirable, but I question the statement about "well-organized airflow path." When the piston is at bottom center the only air that gets "well organized" is that which follows the intake port walls. A good deal of air flows straight through and out the exhaust ports. It has been our experience from many studies of airflow in uniflow and loop-, or cross-

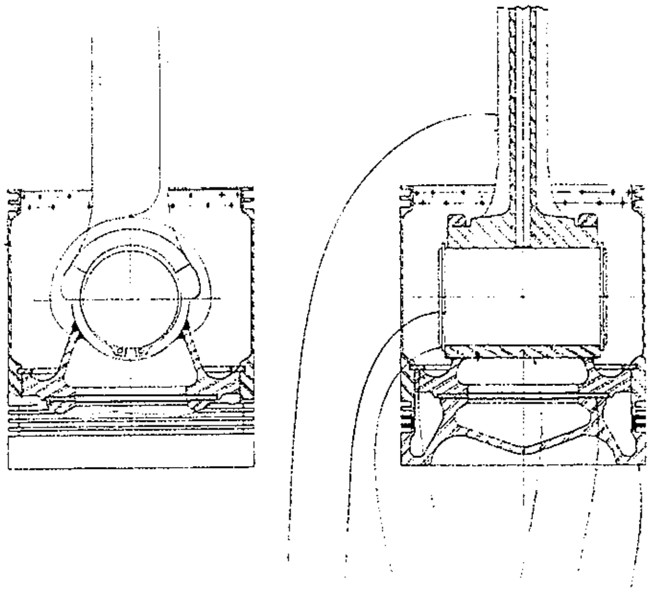

Fig. A – Double-slipper connecting rod

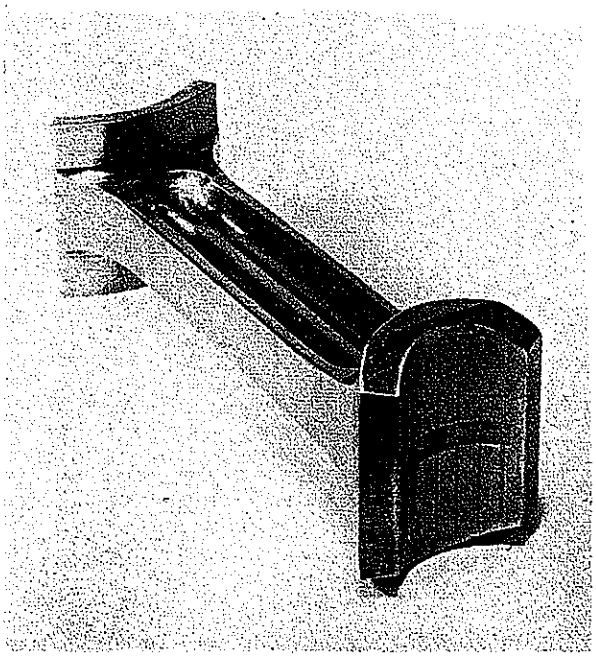

Fig. B – Finished connecting rod

Fig. C – Connecting rod assembled in carrier

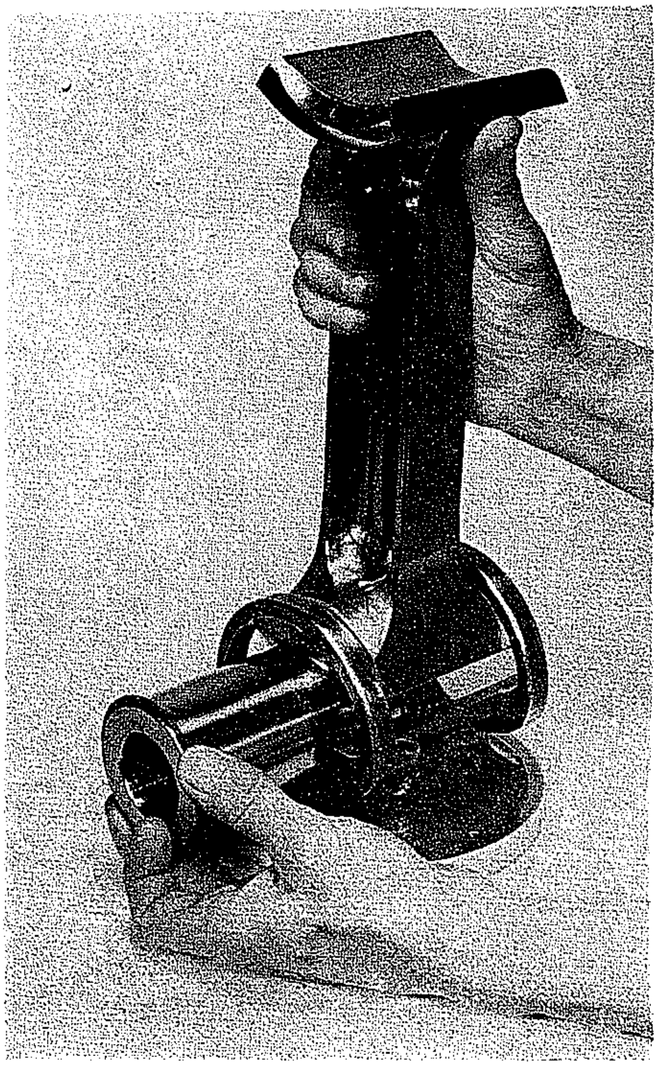

Fig. D – Wrist-pin assembly

scavenge, cylinders that the air doesn't always follow the pretty little arrows we designers like to put on the drawings and diagrams. Much of the air is short-circuited directly across the top of the piston. This is perhaps fortunate because some of this "unorganized" air performs a very useful function by cooling the top of the piston which would otherwise be in a very stagnant region if we were to believe the arrows denoting loop scavenge airflow. In any very high-output engine adequate piston cooling is mandatory.

The drawing in the paper shows a secondary row of ports or slots below the exhaust ports proper. I am curious to know if this is correct, and if so, what is the purpose of the slots?

3. With regard to the injection system, the statement is made that it has been found beneficial to "direct the spray at the walls of the combustion chamber." Because of the shape of the chamber this seems to be the only place to squirt fuel. However, there is a generous squish area on the cylinder head so that fuel will be injected into a region of extremely high-velocity airflow as the upward motion of the piston displaces air inward into the hemispherical chamber. Is not this the reason for the beneficial results obtained with the spray angles used and not because fuel is squirted at the chamber walls?

4. The piston design I find interesting since Napier found it necessary to employ steel heads. It has been our experience that for really high loads it is easier to make an all-steel piston which can be made lighter than those of cast aluminum. Adequate oil cooling, however, must be provided. I assume that all the cooling oil is put into the piston through the rod. Is that correct, and if so, how is it distributed in the piston? The drawing is not too clear in this respect. It would also be interesting to know how much oil is used.

The connecting rod is most intriguing. I do not consider the slipper-type crankpin bearing very unusual since we have been using this type of bearing in production diesel engines for many years. A slipper bearing rod permits a much smaller crankcase design than the more conventional articulated aircraft construction and would be the natural choice when space and weight are at a premium.

The wrist-pin bearing is another matter. As stated in the paper, high bearing loads, low rubbing velocities, and no separation of the surfaces is a problem that has plagued us for some time. The wrist-pin bearing is certainly a touchy spot in any two-stroke engine. We are constantly on the brink of trouble as we increase our engine outputs

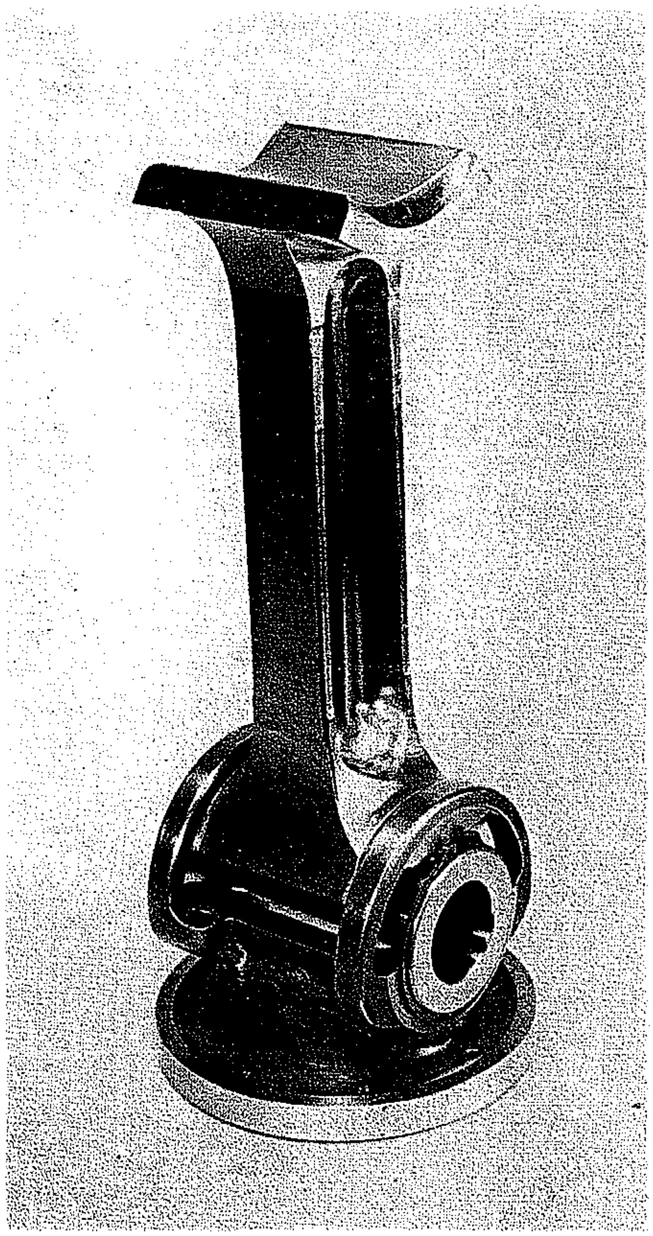

Fig. E – Complete rod, carrier, and wrist-pin assembly and bearing loads. In some cases it has been necessary to resort to very expensive bearing materials and caution in the choice of a lubricant to obtain satisfactory service.

Several years ago we designed a double-slipper rod for one of our high-output single-cylinder test engines. I thought we were being very smart to achieve a 35% increase in bearing area in the same piston, but it wasn't nearly as clever a trick as that employed by Napier.

Shown in Fig. A is a drawing of a double-slipper connecting-rod design. There were two main purposes for this arrangement. One was to alleviate wrist-pin loading and the other was the elimination of all threaded fastenings in the piston, which explains the choice of a floating piston with a rod, wrist pin, and carrier retained by means of a snap ring.

Fig. B is a photograph of the finished connecting rod. It was originally thought that the rod could be made perfectly symmetrical but because of the interference in the piston, it was necessary to use a reduced diameter on the wrist pin end.

Fig. C shows how the connecting rod is assembled in the carrier. First it is cocked and inserted, then it is straightened up and pulled against the hoops for the insertion of the wrist pin.

Fig. D shows the assembly of the wrist pin itself which holds the connecting rod and carrier assembly together and is retained by two snap rings at the outer ends of the pin.

Fig. E shows the complete rod, carrier, and wrist pin assembly ready to be installed in the piston.

Although this design has been successfully run for many hours of high-load operation, I was not able to sell it to one of our manufacturing divisions.

Everyone knows that to make a bearing work there must be a clearance space provided and a lubricant in the clearance. Increasing area or providing better material as is commonly done just takes the curse off a critical bearing. But providing the clearance so that it may fill with oil as the engineers of Napier have done is a most simple and straightforward engineering accomplishment.

It would be interesting to know what bearing materials Napier used in the connecting-rod bearings and how they are bonded to the rods.

Points Out Nomad's Unusual Combination of Constructions

– Marsden Ware
Packard Motor Car Co.

THIS engine brings to mind previous aircraft diesel engines, prominent among which were the Mercedes Benz, the Junkers Jumo, and the Packard radial diesel engine. All these engines had an impressive although limited service experience. Mercedes Benz engines made several Atlantic flights installed in the Hindenberg airship. The Junkers Jumo powered a considerable number of German fighters. The Packard diesel was used in a number of aircraft and established a nonrefueling record which held for a period of many years. With the performance that is being obtained with the Napier Nomad, those who have previously envisioned the diesel engine playing a prominent part in aircraft may yet find impressive evidence of real usage.

The Napier Nomad engine puts together a number of constructions which are unusual in their combination although not new individually. Among these are: compounding, 2-stroke with loop scavenging, direct fuel injection, compression ignition, high air/fuel ratio.

Unusual and noteworthy mechanical features include: variable-speed supercharger-turbine gearing, variable compressor inlet guides, piston-pin bearing and lubrication, piston construction.

While the Packard radial diesel engine was definitely in the specific weight classification of aircraft engines as of that day, the specific weight of the Napier Nomad is much less than any previous aircraft diesel engine. The Napier Nomad is quite definitely in the modern aircraft engine weight class. The Wright compound gasoline engine shows a specific weight of 1.1 lb per hp, while the Napier Nomad compound diesel engine shows a specific weight of 1.2 lb per hp.

The performance of the engine is quite impressive. The specific constructions used in the cylinder are evidently very important in obtaining the high cylinder performance. The 2-stroke valveless construction avoids questions of exhaust-valve capacity to handle the high heat load. Oil cooling and heat-resisting metals combine to enable the pistons to live with intense heat loading.

There is no end to the questions which may be raised concerning many details which are not included. A few of these are:

How much heat is carried off by oil cooling in the piston or how much oil is circulated through the piston?

What is the oil consumption?

Do you expect still greater specific power with greater air dilution?

What is the limit of air dilution?

What endurance and type test requirements have been met?

Is there a marine version of the engine, and if so, how does it compare with the aircraft engine?

It appears that the ordinary turbocharged diesel is not to be classed as a compound engine, although large increases in power and high expansion ratios are obtained with the turbocharger. Is it appropriate as inferred from the wording in the paper to confine the term "compounding" for actual mechanical power connection of both turbine and engine to the power take-off shaft?

Asks About Nomad's Performance and Reliability

– W. G. Lundquist
Wright Aeronautical Division, Curtiss-Wright Corp.

IN regard to the performance of the engine, the top question which comes to my mind concerns whether or not the experimental testing of the engine has confirmed or disagreed with the performance as projected analytically. I believe I am correct in assuming that the performance curves presented were derived analytically and they do not represent necessarily accomplished test data. To what extent have test data substantiated these curves and are such data available for publication? It would be very interesting to see the correlation.

In regard to the weight of the engine, I cannot make out whether or not the weight comparisons include the weight of the associated liquid cooling radiators, piping, and other installation requirements associated therewith including the weight of the coolant itself. In comparison, all installed weights of the Nomad engine compared with other engines would, of course, have to include such additional weights to give a correct overall comparison. Specifically then, the question is: Do the weights given in this paper include the complete weight of the coolant system and coolant?

Has it been determined what the engine's oil heat rejection is and what oil cooler requirements are necessary?

Obviously, in addition to the weight requirements of the cooling system and the oil cooling system, the frontal area requirements for the coolant and oil radiators will be of great importance in completing any installation comparison.

In regard to mechanical features, several questions occurred as I read the paper. In the first place, what has the experience been with this engine in regard to the loss of lubricating oil through the exhaust ports and the associated "coking" of these ports due to the leakage of lubricating oil? This has been one of the classical problems associated with the ported 2-stroke engine, and it would be interesting to know what has been experienced with the Nomad.

I note from the engine pictures that the cylinder heads are bolted on. Is the cylinder head to cylinder block joint a metal-to-metal design, or is a gasket used in this place? This is another trouble spot for diesel engines, and it would be interesting to know what experience has been obtained in the case of the Nomad on this item.

I think I would agree with the authors that the field of application for an engine of this type which is designed to have a very high thermal efficiency would be in the so-called air transport category as distinguished from military tactical operations. However, the trend in this field is definitely towards higher subsonic speeds, and in these areas the overall engine efficiency as defined by the miles per gallon yardstick is not a function of internal thermal efficiency of the engine alone but must include the effect on the flight efficiency of the weight and size factors involved in the engine design. For that reason, the more elaborate forms of compound engine have not been too promising in the past, and any consideration of their use must be predicated on the installed net overall efficiency including the induced and parasite drag characteristics of the installed powerplant. For these reasons, the piston-type engines encounter increasing competitive disadvantage as sonic speed is approached. It was for these reasons that we at Wright Aeronautical chose to make the turbo-compound engine the simplest practical form of compound engine which could be accomplished without increasing weight or other drag factors.

Another very important factor involved in the selection and use of an engine, particularly for commercial airlines, is the day-to-day on-schedule reliability which is demanded, and this factor is inherently tied in and controlled by mechanical simplicity or lack of simplicity as the case may be. We have learned very painfully throughout the years that mechanical complications in the basic engine design and the use of accessory components such as complicated fuel and power controls can be a prolific source of operational difficulties and schedule delays in airline operation. In other words, the probable time and cost to an airline operator involved in debugging an engine with such complications will be an extremely important consideration in the selection of the engine for transport use, particularly commercial airline operation. I am certain that this aspect of the Nomad engine has been considered carefully, and it would be interesting to hear what development progress in regard to reliability of external engine accessories has been achieved.

These factors did, in fact, control to a very large extent our thinking during the development of the Wright turbo-compound engine. The turbo-compound engine model currently operating on the airlines (and in United States military operations) was deliberately, therefore, kept as simple design-wise as could be and yet attain a definite beneficial utilization of the compound principle. The engine utilizes no accessory power control whatsoever, being operated exactly as an uncompounded piston engine. To as great an extent as possible, the compounding has been so devised as to add no additional mechanical or thermal load to the cylinders. Actual operating experience has confirmed that this deliberate emphasis on simplicity has been largely instrumental in the acceptance and success of the engine. We have in the past and we still are studying other more complicated types of compounding, and we may yet conclude that further complication may be warranted to achieve further improvements in performance. For that reason, the contribution represented by this paper on the Nomad engine is of great interest to us, and we feel the Nomad project represents a very important contribution to the development of piston aircraft engines. We would like, therefore, to commend the Napier company and others responsible for this project for their farsighted vision in undertaking the project, and we will look forward to the progress which can be made in the future in the field of compound piston aircraft engines.

Dislikes Nomad's Apparent Complexity

– F. W. Kolk
American Airlines

ALTHOUGH I am impressed by the low fuel consumption of the Nomad, I am concerned about the seeming complexity of the engine. The Nomad appears to combine the complexities of both aviation piston engines and gas turbines.

In a new airplane, utilization is very important, and if the plane is not available because of mechanical breakdown or maintenance, the operating cost figures can be very misleading. Utilization is a function of mechanical excellence and complexity. It is thus undesirable to increase the number of components in an engine.

Discusses Point in Theoretical Analysis

– L. C. Lichty
Yale University

THE compound engine developed by Napier is a very worth-while contribution. However, one point of interest in the theoretical analysis appears to be overlooked.

The area, ABCA, on the pressure-volume diagram (Fig. 5) is assumed to indicate the loss of work of the compound cycle because of the process of releasing the gases from the reciprocating engine at P_A to the turbine inlet at P_B by a thermodynamically irreversible process. As a result of this process, the entropy of the medium at point C would be greater than at point A, assuming an adiabatic process. This indicates a larger volume at point C than indicated in Fig. 5, and consequently the expansion curve for the turbine from the corrected location for C to turbine exhaust pressure would be higher than indicated. Thus, some of the loss of available energy caused by the irreversible release process is recovered in the turbine, and the loss of work would be less than indicated by the area ABCA.

For the foregoing reason, point F in Fig. 5 would not lie on the curve AFC, but would be on a similar curve above the curve PD, extended to intersect the turbine inlet-pressure line.

It can be shown that the correct value for V_C might be as much as 6% higher than the value indicated, depending on the pressure drop in the transfer process. Thus it appears the turbine work might be more than 6% higher than indicated in Fig. 5.

The foregoing discussion assumes the expansion processes are adiabatic isentropics. However, if they are actual expansion processes with heat transfer out of the medium, the recovery of some of the indicated loss of work is possible.

Authors' Closure To Discussion

MR. BIRMANN contends that mechanical connection of the turbine to the power output shaft is an unnecessary complication; however, we strongly disagree. Turboblowers are widely used in aircraft engines, and Napier is well aware of what can and cannot be done with them. For aircraft service, connection of the turbine to the output shaft is essential for best flexibility and performance. A turboblower system cannot be designed to produce the equivalent results.

In answer to Mr. Flynn's discussion, more air can be passed through the cylinder with less aerodynamic loss with the Nomad type of flow path. The development of the scavenge system of the engine was based on many thousands of hours of testing and research. The airflow and pressure losses are very much influenced by the directional flow control that can be effected.

The extra row of ports below the main exhaust ports in the cylinders eliminates the tendency for the bridges between ports to distort and cause piston rubbing. In addition, the small ports were found to improve engine performance.

A conical combustion chamber was first tried, but we found that, as the fuel spray angle was widened, better performance resulted. The hemispherical combustion-chamber form was thus a natural development. We do not claim that it is the best possible configuration, but for the Nomad it gives good results.

Regarding the piston design, oil is fed transversely to an annular chamber behind the piston rings. No oil is fed directly to the piston crown, although some of the oil does contact the crown undersurface.

We agree that the design of the connecting-rod bearings is not wholly new, but it does have a number of unusual features. Thin-walled bearings, steel backed with a lead-bronze lining, are used.

In answer to Mr. Ware's questions, the lubricating oil consumption of the Nomad is 8 pt per hr at cruising conditions and 16 pt per hr at full power. Only about half the air is consumed in the engine cylinders. The limitation to using more of the available air is the thermal load on the piston. We expect future development of the piston to make possible higher outputs from the engine.

We have no information on the limit of air dilution nor on the endurance qualities of the engine. There is no marine version of the Nomad engine.

Mr. Lundquist also raises some questions about performance data. All results given in this paper are actual test data, except the curves referring to flight conditions. The weights given are net dry weights and so do not include external separate equipment. This is in accord with the usual practice.

Lubricating oil is lost through the ports during idling conditions, producing a blue haze in the exhaust. There has never been any carbon buildup in the ports; this is probably a result of the temperatures involved. A copper ring is used as the cylinder-head sealing. This was the most satisfactory of several methods tried.

We do not regard the Nomad engine as being complex and think this criticism is not justified, as the Nomad is basically a very simple design. While the Nomad is more complicated than a simple low-powered piston engine, usually the only alternative high-output engines ever suggested are hypothetical engines that have not been built. All engines are very simple in the first design consideration stage; they become complex when built or when they are developed to working reliability.

The turbine and compressor of the Nomad are normal types of rotating machinery that have proved to be reliable. The piston part of the engine is similar to conventional engines and is simple and reliable. The fuel-injection system is basically more reliable than a spark-ignition system.

We agree entirely with the points raised by Professor Lichty.

There were a number of comparatively detailed points of this nature which arose during the preparation of the "indicator" diagrams used for illustrating the paper. A certain degree of simplification was regarded as desirable in order to avoid the necessity for somewhat lengthy explanations. The main purpose of the diagrams was, of course, to indicate the broad principles on which the compound engine operates.

An attempt was made to excuse such slight departures from the truth by the cunning introduction of the phrase, "in broad terms without much respect for meticulous detail."

ORAL DISCUSSION

Reported by Alan R. Schrader
USN Engineering Experiment Station

Carl F. Bachle, Continental Aviation and Engineering Corp.: How is it possible for the engine to operate with gasoline as a fuel?

Reply: The engine was developed to burn gasoline because this was a requirement of the British Ministry of Supply, which considered that the engine should be capable of using fuels available throughout the world. Actually, the engine is very insensitive to fuel quality. This is probably a result of the high degree of supercharging employed.

Mr. Bachle: What is the efficiency of the variable-speed gear?

Reply: The variable-speed drive efficiency ranges from 65 to 92%. For conditions where the efficiency is low, the amount of power transmitted is also low, so power losses are not serious.

Dale H. Brown, General Electric Co.: I note the use of variable compressor inlet guide vanes. What is the flow restriction characteristic of the diesel and the turbine that requires this degree of finesse? How much air does this compressor handle at sea-level take-off?

Reply: The compressor inlet variable guide vanes allow the operating range of the engine to be extended at the low-speed end.

Mr. Brown: Of the air that is delivered by the compressor, it appears that some must go right through the engine. How many pounds of air actually get trapped in the cylinder? Also, this does not look like as positive a scavenge system as the valve-in-head or opposed-piston engine. What is the ratio between the pounds of air trapped and the pounds of residuals left in the cylinder?

Reply: No information is available as to the proportion of air trapped in the cylinder. Data of this sort would be desirable, but we know of no reliable means of obtaining them.

Mr. Brown: Our experience indicates that one to two points of stage efficiency can be realized through use of a shroud on the turbine. What were your considerations in selecting and using an unshrouded wheel?

Reply: We are aware of the slight performance improvement made possible by shrouding the turbine. However, this knowledge was not available at the time of design, and it appears that redesign of the turbine would not be warranted for the small improvement gained.

Mr. Brown: While the turbine-compressor set appears small, it represents a tremendous inertia during starting and acceleration of the engine. What starting power was required? Did the compressor-turbine set require provision for slipping the drive during starting or other high-torque occasions?

Reply: No difficulty has been experienced due to inertia of the rotating parts.