Advanced Automotive Power Systems, Part 2: A Diesel for a Subcompact Cars*

P. Hofbauer and K. Sator
Volkswagenwerk AG

QUITE PROBABLY that pioneering genius of engine design, Rudolf Diesel (1858–1913) would disclaim any connection between his engine (Fig. 1) and the VW DESIGN (Fig. 2). We even debated whether it would be proper to call the new Volkswagen engine by the name of Diesel at all. But all doubt vanishes if you take a closer look, for this engine features such things as

1. Fuel injection into an enclosed chamber;
2. Heterogeneous mixture (stratified charge) and
3. Compression ignition;
4. Fuel volume control; power output controlled by fuel volume.

Taken together, these features describe a Diesel engine all over the world.

However, with the exception of these features there is hardly anything to connect today's Diesel engines and the engine designed by Rudolf Diesel. Besides, VW's passenger car Diesel engine incorporates new features even compared to present-day Diesel engines, especially as far as its basic concept is concerned.

AVAILABILITY OF DIESEL FUEL

According to some optimistic estimates, the proportion of passenger cars produced and equipped with Diesel engines might rise as high as 25% of the total within the next 10 years.

This means that before 1990 hardly more than 10% of all cars will be Diesel cars and that only 25% of all fuel required will be Diesel fuel as late as the Year 2000.

The petroleum industry maintains (3,4) that a changeover as slow as this will cause no problems, even if the final percentage should go up to the maximum given above.

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ABSTRACT

System analysis studies have shown that Diesel engines can be an alternative to spark ignition engines even in subcompact cars. According to some optimistic estimates, the share of Diesel engines in the total market may reach 25%, so that Diesel engines might well supplement the existing range of spark ignition engines.

During the period from 1970 to 1990, the significance of all individual objectives whose demands are not favorable to the Diesel will decrease, whereas the significance of objectives favoring the Diesel will increase.

It was the objective of the VW Passenger Car Diesel Engine Project to alleviate considerably all the ill-famed Diesel disadvantages, such as noise, smell, smoke, and slow acceleration, the latter being due to the low horsepower-to-weight ratio. On the other hand, we intended to preserve for subcompact car use the classical Diesel advantages, such as excellent fuel economy, long service life, low incidence of malfunctions, and favorable emissions.

We aimend especially for compliance with US emission regulations in addition to meeting requirements worldwide. Compared to a spark ignition engine of the same power output, we intended to improve the fuel economy of the Diesel by 30 to 50%, and its service life by more than 50%.

Another objective was to keep Diesel production cost low by using the largest possible number of spark ignition engine parts for Diesel production.

Of all Diesel operating principles, the swirl chamber design was found to comply best with the manifold requirements connected with a subcompact car. By virtue of extensive development work on the spark ignition engine parts, we ensured that both Otto and Diesel engines can be manufactured on the same transfer line.

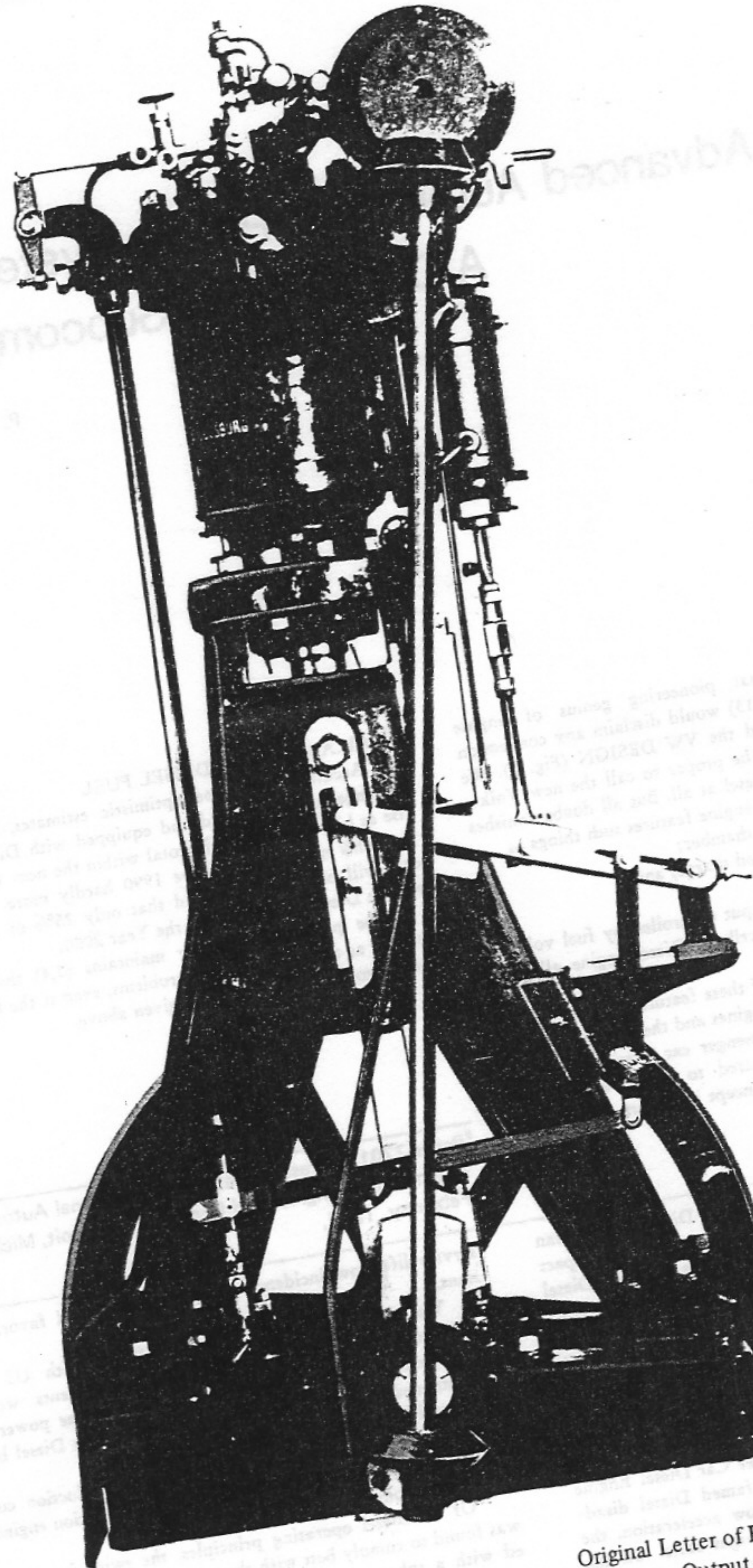


Fig. 1 - First Working Diesel Engine. After 30 Months of Development (During Which Rudolf Diesel had to Abandon Most of the Ideas Contained in his Original Letter of Patent) and After Repeated Modification the Power Output of the Engine was Sufficient at the End of 1895, Bore 220 mm, Stroke 440 mm

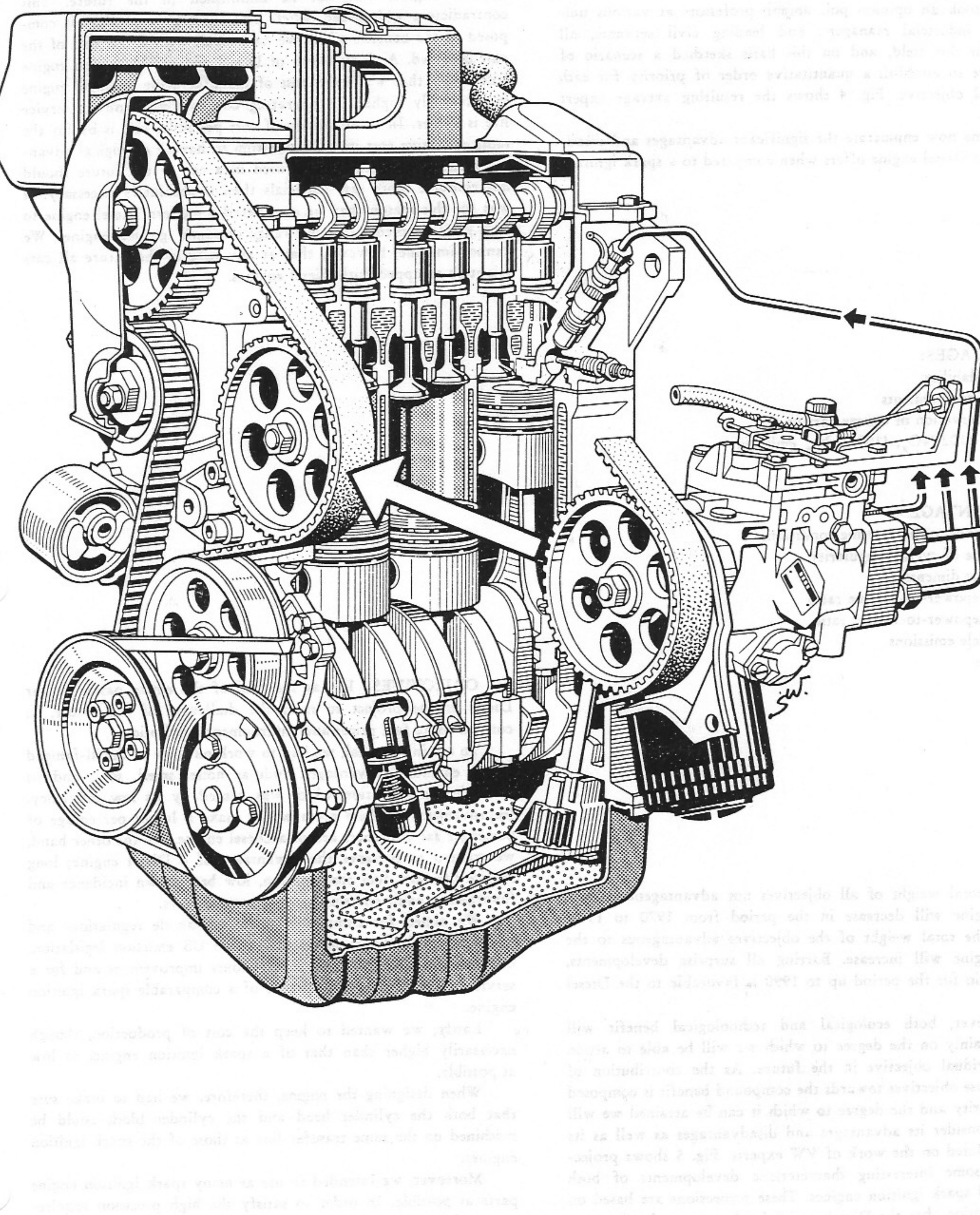


Fig. 2 – VW-Diesel Engine

WHY DIESEL ENGINES FOR PASSENGER CARS?

Let me first deal with the question, why should Diesel engines be used in passenger cars weighing up to 1,000 kg? Volkswagen's Research and Development Department has performed a systems analysis (1,2) based on the objectives for passenger car engines (Fig. 3).

We took an opinion poll among professors at various universities, industrial managers, and leading civil servants, all experts in this field, and on this basis sketched a scenario of the future to establish a quantitative order of priority for each individual objective. Fig. 4 shows the resulting average expert prognosis.

Let me now enumerate the significant advantages and disadvantages a Diesel engine offers when compared to a spark ignition engine.

ADVANTAGES:
1200 Availability
1500 Fuel requirements
3100 Conservation of energy carrier
4101 CO, 4102 NO_x, 4103 C_nH_m emission

DISADVANTAGES:
1300 Behavior under extreme conditions
1401 Degree of cyclic irregularity
2100 Engine dimensions
2200 Horsepower-to-volume ratio
2300 Horsepower-to-weight ratio
4103 Particle emissions
4200 Noise

The total weight of all objectives not advantageous to the Diesel engine will decrease in the period from 1970 to 1990, whereas the total weight of the objectives advantageous to the Diesel engine will increase. Barring all surprise developments, the scenario for the period up to 1990 is favorable to the Diesel engine.

However, both ecological and technological benefit will depend mainly on the degree to which we will be able to attain each individual objective in the future. As the contribution of each of these objectives towards the compound benefit is composed of its priority and the degree to which it can be attained we will have to consider its advantages and disadvantages as well as its priority. Based on the work of VW experts, Fig. 5 shows projections of some interesting characteristic developments of both Diesel and spark ignition engines. These projections are based on the assumption that the Diesel engine holds a great development potential. We are sure that, as it has been true in the past, this assumption will continue to hold good as far as power plants, working procedures, and even subcontracted parts are concerned.

The ecological and technological benefit of each objective is thus composed of its priority and its characteristics.

Fig. 6 shows the results of our systems analysis.

Given the input data shown in figure 4 the advantages of a Diesel engine over today's standard power plant, the spark ignition engine, will not be diminished in the future. This contradicts a widespread belief. The value of a product is composed of the benefits which we have dealt with so far and of the cost involved. A Comparison of Diesel and spark ignition engine cost shows that while the cost of manufacturing a Diesel engine is necessarily higher, its operating cost is lower and its service life is longer. In a calculation of cost per mile, fuel is by far the most expensive cost item. In addition to having ecological advantages any power plant to be used in a car of the future should not consume more raw materials than is absolutely necessary. It was for this reason that we considered a modern Diesel engine to be a good supplement to our range of spark ignition engines. We cannot imagine, however, that at any time in the future all cars might be equipped with Diesel engines.

OBJECTIVES It was the goal of the entire passenger car Diesel engine project to meet the challenges of the future, to combat rising fuel prices and environmental pollution.

To attain this goal, we had to work on some of the ill-famend Diesel engine disadvantages, such as noise, smell, soot, and its dead slow acceleration which was caused by its low efficiency. Improvements in these areas would make a larger percentage of passenger car customers accept a Diesel engine. On the other hand, we wanted to preserve the advantages of a Diesel engine; long service life, low fuel consumption, low breakdown incidence and favorable exhaust emission rates in compact cars.

We aimed for compliance with worldwide regulations and especially with the requirements of the US emission legislation, but also for a 30 to 50% Fuel Economy improvement and for a service life 50% longer than that of a comparable spark ignition engine.

Lastly, we wanted to keep the cost of production, though necessarily higher than that of a spark ignition engine, as low as possible.

When designing the engine, therefore, we had to make sure that both the cylinder head and the cylinder block could be machined on the same transfer line as those of the spark ignition engines.

Moreover, we intended to use as many spark ignition engine parts as possible. In order to satisfy the high precision requirements of Diesel engine design selected parts from normal production are matched.

This paper describes the extent to which we were able to attain these objectives.

1000 FUNCTIONAL QUALITY	1100 CONTROL RE-QUIREMENTS (OPERATING COMFORT)	1101 FLEXIBILITY (NUMBER OF GEARS)
		1102 TIME PERFORMANCE
		1103 ENGINE BRAKING EFFECT
	1200 AVAILABILITY (BREAKDOWN SUSCEP-TIBILITY)	1201 RELIABILITY
		1202 MAINTENANCE REQUIREMENT
		1203 SERVICE LIFE
		1204 OPERATION RANGE (HEAT VALUE)
	1300 PERFORMANCE UNDER EXTREME CONDITIONS	1301 STARTABILITY
		1302 PERFORMANCE LOSS AT HIGH ALTITUDE
		1303 ENERGY AVAILABLE FOR HEATING
	1400 VIBRATION LEVEL (OSCILLATION BEHAV.)	1401 UNIFORMITY
		1402 MASS BALANCE
	1500 FUEL REQUIREMENTS	1501 MULTI-FUEL CAPABILITY
		1502 OPERATING DANGER
2000 INSTALLATION SUITABILITY	2100 DIMENSIONS (SHAPE)	2101 LENGTH
		2102 HEIGHT
		2103 WIDTH
	2200 POWER TO VOLUME RATIO	
	2300 POWER/WEIGHT	
	2400 SUITABILITY FOR MASS PRODUCTION	
3000 RAW MATERIAL CONSERVATION	3100 ENERGY	3101 OIL
		3102 NATURAL GAS
		3103 COAL
	3200 MATERIAL	3201 ALUMINIUM
		3202 CHROMIUM
		3203 COBALT
		3204 COPPER
		3205 IRON
		3206 LEAD
		3207 MANGANESE
		3208 MOLYBDENUM
		3209 NICKEL
		3210 PLATINUM GROUP
		3211 MAGNESIUM
		3212 TUNGSTEN
		3213 TITANIUM
		3214 VANADIUM
		3215 ZINC
		3216 TIN
4000 ENVIRONMENTAL PROTECTION	4100 POLLUTANTS	4101 CO
		4102 NO_x
		4103 C_nH_m
		4104 PARTICULATES
	4200 NOISE	
	4300 WASTE	

Fig. 3 – Objective System

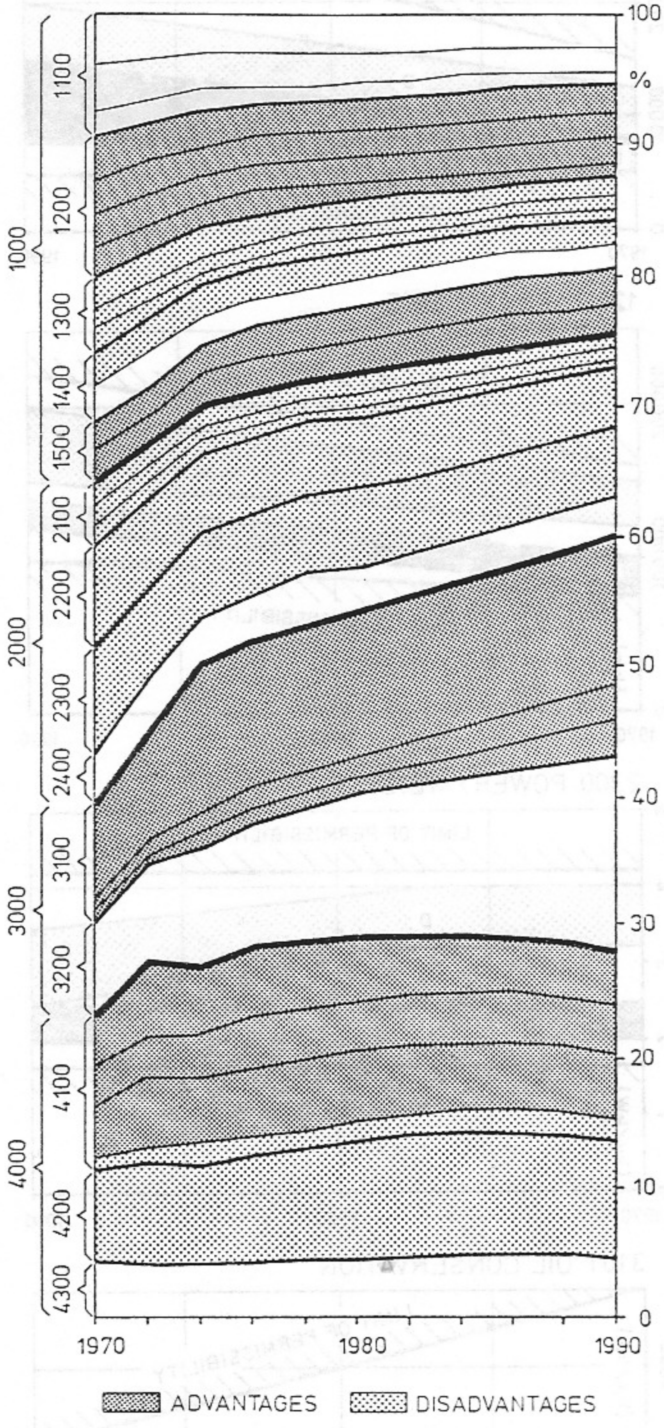


Fig. 4 – Weighting Functions for Different Power Plants, Subcompact Car Application, Average Expert Prognosis. Advantages and Disadvantages of a Diesel Engine when Compared to a Spark Ignition Engine

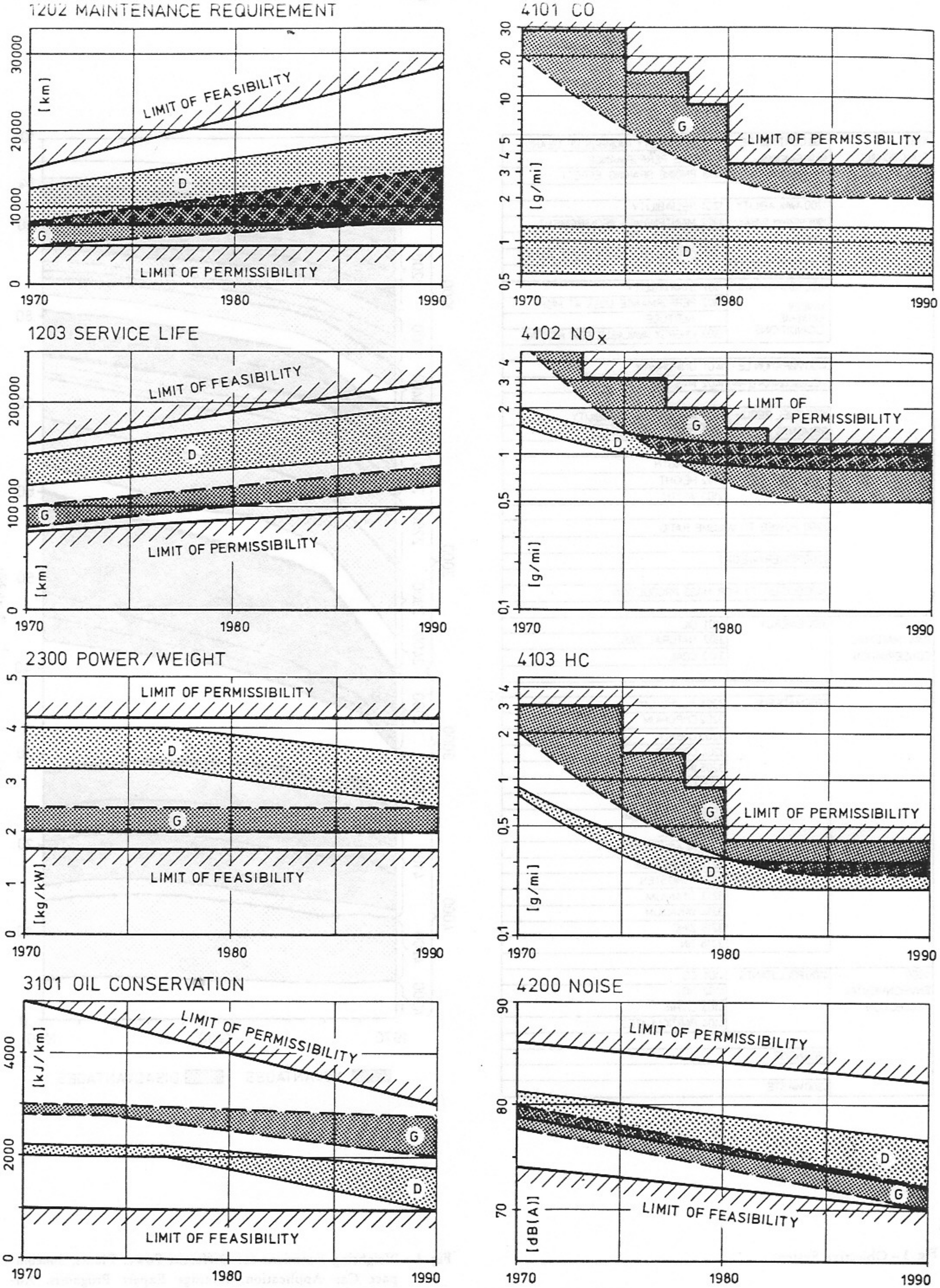


Fig. 5 – Projections of 8 Selected Objectives

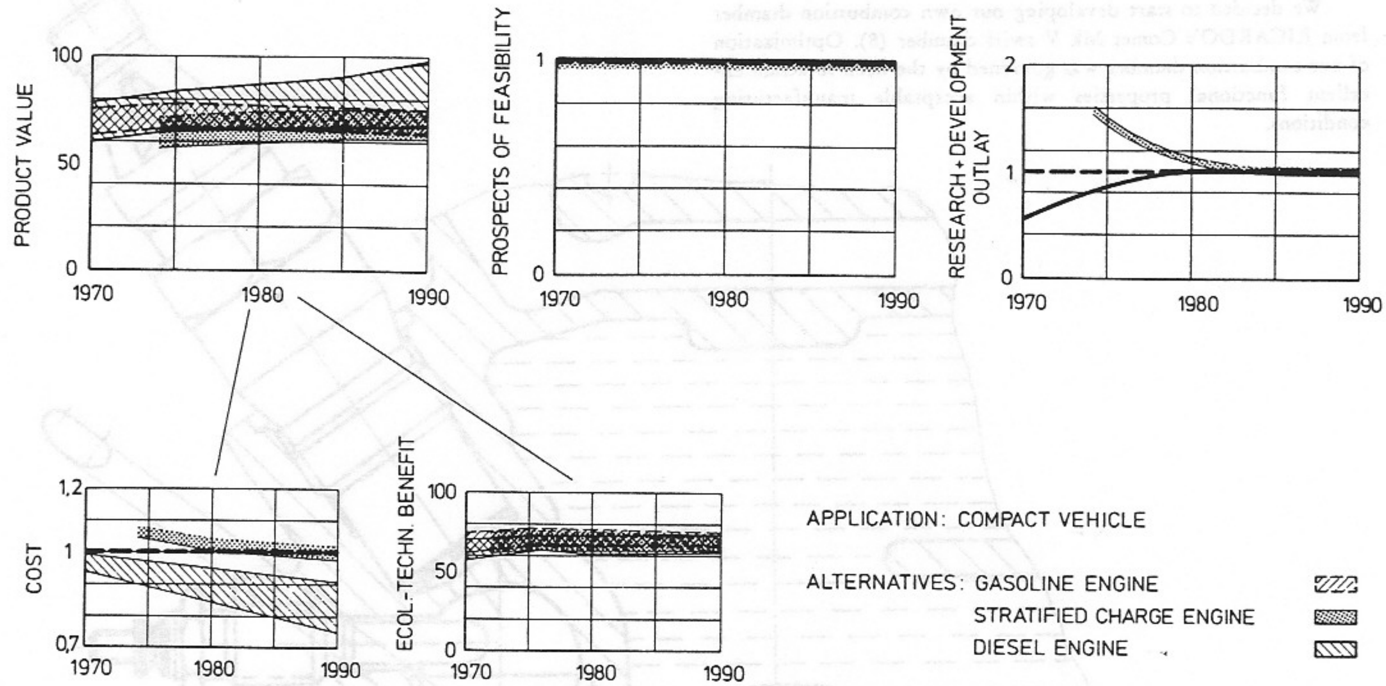


Fig. 6 – Decision Basis

CONCEPT AND DESIGN OF COMPONENTS

WORKING PROCEDURE

COMBUSTION SYSTEM. In the initial stage of this Diesel engine project our engine research staff was confronted by the question, of the existing Diesel concepts (Fig. 7) Which would be most appropriate to a Volkswagen Diesel engine? The one basic decision influencing the engine concept was between DIRECT INJECTION and SEPARATE WORKING CHAMBERS (5).

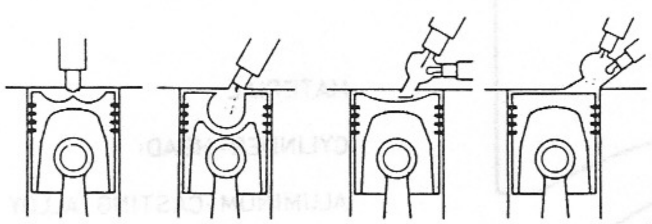


Fig. 7 – Diesel Combustion Systems from left to right: Direct Injection, MAN-M-System, Prechamber, Swirlchamber

We decided in favor of separate working chambers, as a study on engines with a displacement of less than 400 cm³ per cylinder and subcompact cars of an inertia weight of less than 2,500 lbs had produced the following findings.

Provided that separate working chambers are used, the requirements of the US Emission Legislation ($CO < 3.4$; $HO < 0.41$; $NO_x < 1.5$) can be met; if direct injection is used, meeting emission levels becomes extremely risky.

If all US emission regulations are left out of consideration the fuel consumption of a Diesel engine can be reduced by 10 to 15% by using direct injection. Under US conditions, however, the importance of this advantage over separate-chamber engines is reduced to a mere slight improvement in fuel consumption, which can be evened out easily, inexpensively and without risk by simple gearbox modifications, such as adding a fifth gear.

Especially if the MAN-M concept is used, the idling noise of a Diesel engine can be reduced to the level of a spark ignition engine, but the separate-working-chamber concept generates less noise under driving conditions, which we believe is of more importance.

Cold-start provisions are extensive in DI engines. We were especially concerned with the loads acting on the power plant as we intended to develop VW's Diesel engine from a production spark ignition engine. As separate chambers and especially swirl chambers permit higher rated engine speeds, engines of this concept will yield the same power when operated at lower mean effective pressures. Moreover, the very fact that the combustion chamber is divided contributes substantially towards reducing peak pressures.

We decided in favor of the swirl chamber concept rather than the PRE-CHAMBER CONCEPT because our priorities in conceiving a modern Volkswagen were: Satisfactory fuel consumption under all operating conditions, sufficient power output, and perfect black-smoke emission behavoir. All these objectives are more easily attained by applying the swirl-chamber concept. Lastly, the fuel economy of a swirl-chamber engine is superior to that of pre-chamber engines, especially at high speeds (6).

Only swirl chambers will permit rated engine speeds in excess of 83 U/sec or 5,000 rpm and a specific power output of 34 BHP per liter (25 kw).

Finally, tests at high altitudes prove conclusively that prechamber engines are far more delicate as far as controlling black-smoke emission is concerned than swirl chamber engines. Having found at a rather early stage in our research that with the sole exception of its idling noise, a swirl-chamber Diesel engine can be suppressed to the noise level of a spark ignition engine, we immediately decided in favor of this concept. (Fig. 8) (7).

We decided to start developing our own combustion chamber from RICARDO's Comet Mk V swirl chamber (8). Optimization of our combustion chamber was governed by the need to attain excellent functional properties within acceptable manufacturing conditions.

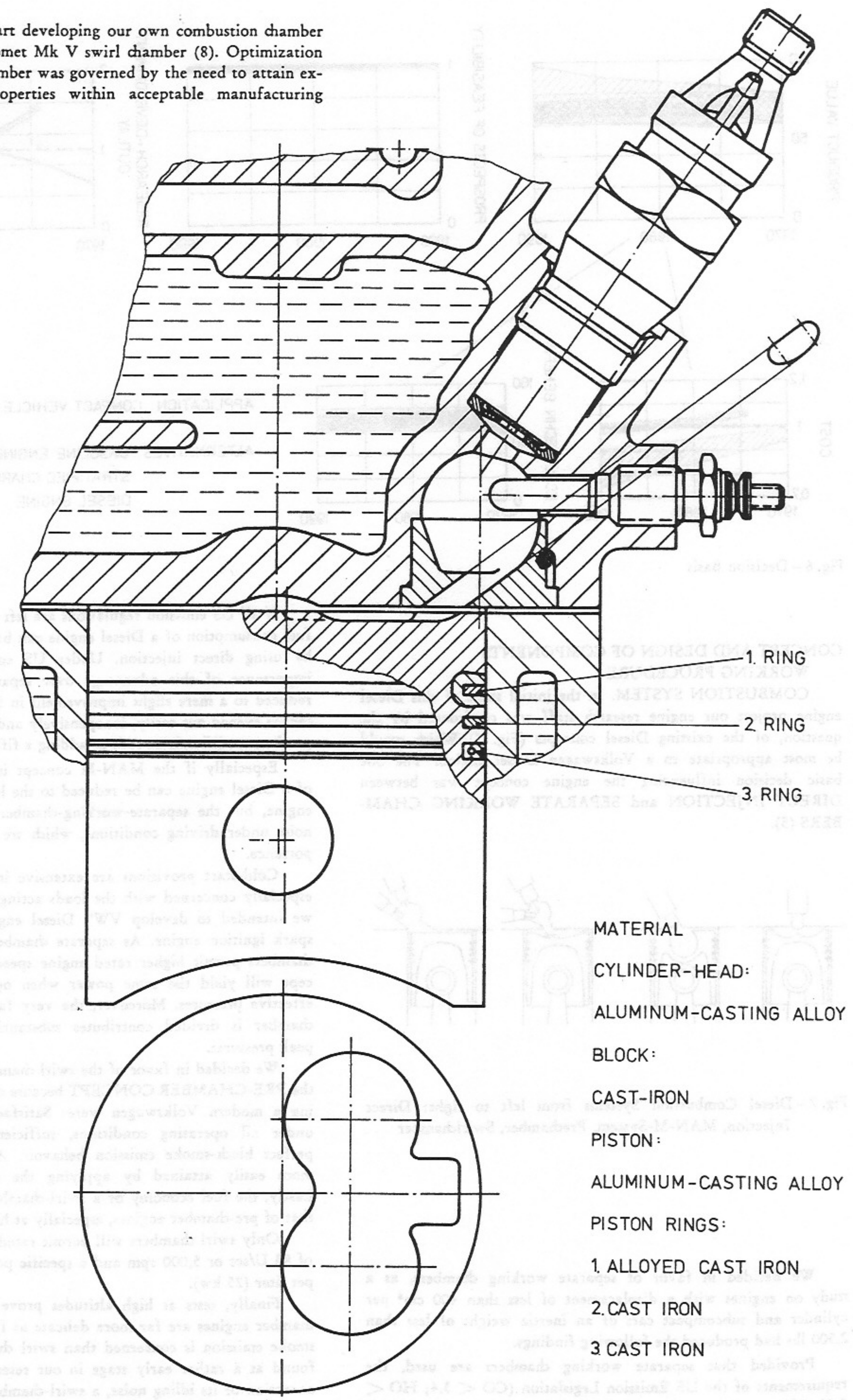


Fig. 8 – VW-Diesel Engine Combustion Chamber Configuration

If the displacement of a cylinder is small, exact adherence to the COMPRESSION RATIO gains outstanding significance while becoming extremely difficult. All passenger car Diesel engines are troubled by problems concerning coldstart behavior, warm-up engine noise, and the idling noise produced by both cold and warm engines.

To solve these problems in a satisfactory manner we had to adopt a mean compression ratio of $\varepsilon = 23 : 1$.

The disadvantages of a compression ratio as high as this are that it precludes attaining optimum fuel economy, and that it makes it difficult to solve the manufacturing problems caused by a high compression ratio coupled with a small compressed volume.

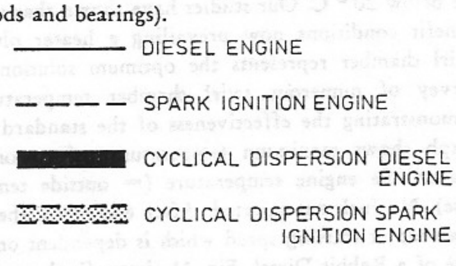
The compression ratios which permit attaining optimum fuel consumption in small swirl-chamber engines range between $\varepsilon = 16$ and $\varepsilon = 18$. However, at the present stage of technological development it is impossible to produce cold-start and idling characteristics which are satisfactory to a passenger car customer at engine ompression ratios of less than $\varepsilon = 20$. Any developments aimed at improving the fuel consumption of small swirl-chamber engines should be directed towards optimizing the compression ratio.

Lowering the compression ratio would mean that manufacturing would become much easier as well. If we could manufacture our Diesel engine components to the same tolerances applying to spark ignition engines, whose compressed volume is more than 3 times as large, compression tolerance would increase to $\Delta\varepsilon = 8$.

At this point, the effect of an enormous advantage makes itself felt: Volkswagen Diesel engines use parts which are also used in the spark ignition engine which is being produced in large numbers. We are in a position to select close-tolerance parts to be used for the Diesel engines from the total amount of, for instance, crankshafts and connecting rods produced. Additional measures ensuring close adherence to the set compression ratio are high-precision machining of cylinder blocks and pistons for height and compression distance, classification of all components and the use of 3 different thicknesses of cylinder head gaskets to compensate for differences in the extent to which pistons project from the cylinder block.

We came as close as possible to having a true constant-pressure cycle in order to keep the strain on the power plant and the noise level as low as possible while retaining satisfactory fuel economy properties. We used both swirl chamber geometry and injection characteristics to ensure that peak pressures would exceed the final compression pressure by only a small margin. In Fig. 9, during the OPERATING CYCLE three full-load and one partial-load points on the engine map are used to illustrate the extent to which we attained this goal. These indicator diagrams also show the extent of cyclical dispersion measured in the pressure peaks which are supposed to fluctuate around the final compression pressure. For purposes of comparison, pressure readings taken from a 1.6-liter spark ignition engine of 110 BHP have been entered into this chart as well.

The 1.5 l 50 BHP Diesel engine and the 1.6 l 110 BHP spark ignition engine use the same internal engine components (crankshaft, pistons, conrods and bearings).



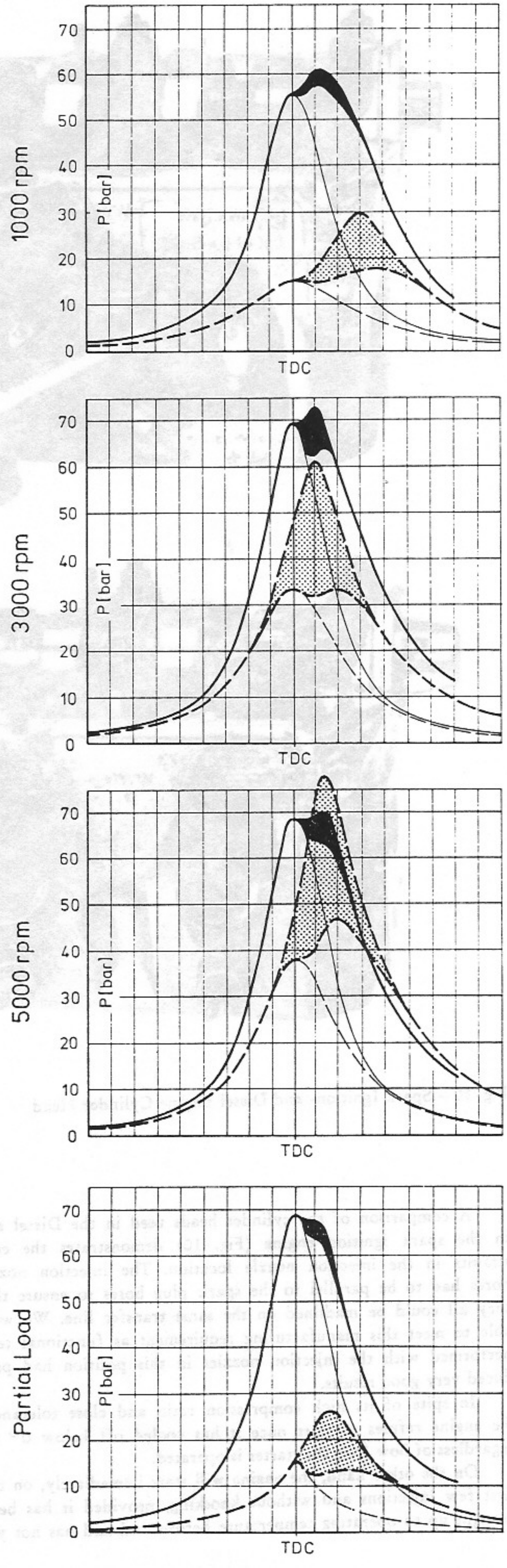


Fig. 9 – Pressure Comparison of Spark Ignition and Diesel Using the Same Internal Engine Components

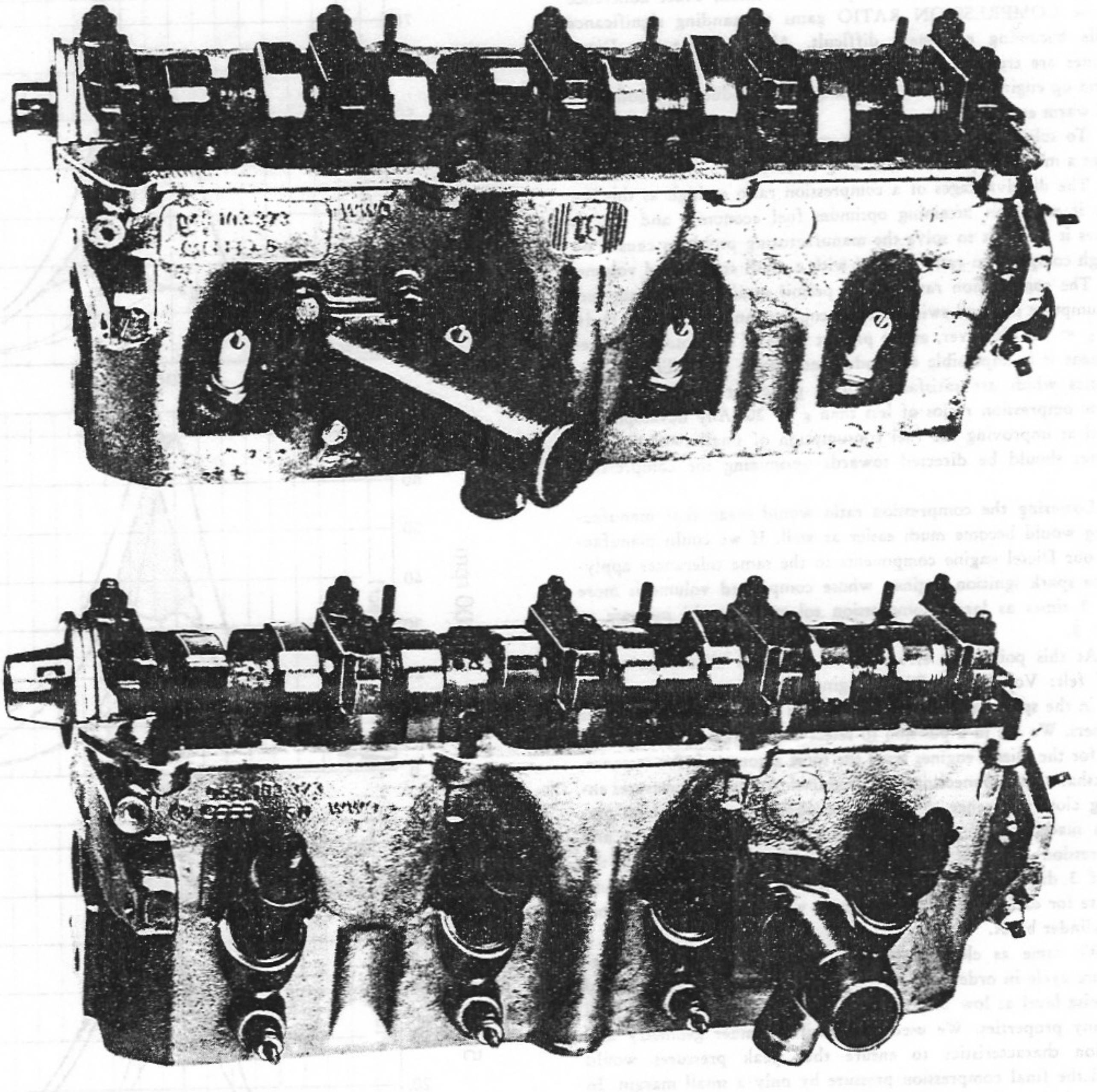


Fig. 10 – Spark Ignition- and Diesel Engine Cylinder Head

A comparison of the cylinder heads used in the Diesel and in the spark ignition engine (Fig. 10) demonstrates the constraints in the injection nozzle location. The injection nozzle bores had to be parallel to the spark plug bores to ensure that they all could be machined on the same transfer line. We were able to meet this manufacturing requirement as functional tests performed with the injection nozzles in this position had produced very good results.

In spite of its high compression ratio and close tolerances the engine refuses to start once it has cooled off below 0 ° C, regardless of how long the starter is operated.

On the other hand, the engine will start immediately, on the first few injections and without knocking, provided it has been brought up to operating temperature beforehand and has not yet cooled down again below 40° C, a process which may take several hours even at low ambient temperatures. However, a customer who has bought a passenger car will not tolerate the inconvenience of starting without a STARTING AID with the engine below 20 ° C. Our studies have shown that under the cost and benefit conditions now prevailing a heater plug located in the swirl chamber represents the optimum solution. Fig. 11 gives a survey of numerous swirl chamber temperature measurements demonstrating the effectiveness of the standard heater plug. The graph shows maximum temperatures after compression plotted against the engine temperature (= outside temperature, in this case). No fuel was injected. After 6 seconds, the temperature was measured at starting speed which is dependent on engine temperature of a Rabbit Diesel. Fig. 11 shows final compression temperatures recorded both with and without pre-heating.

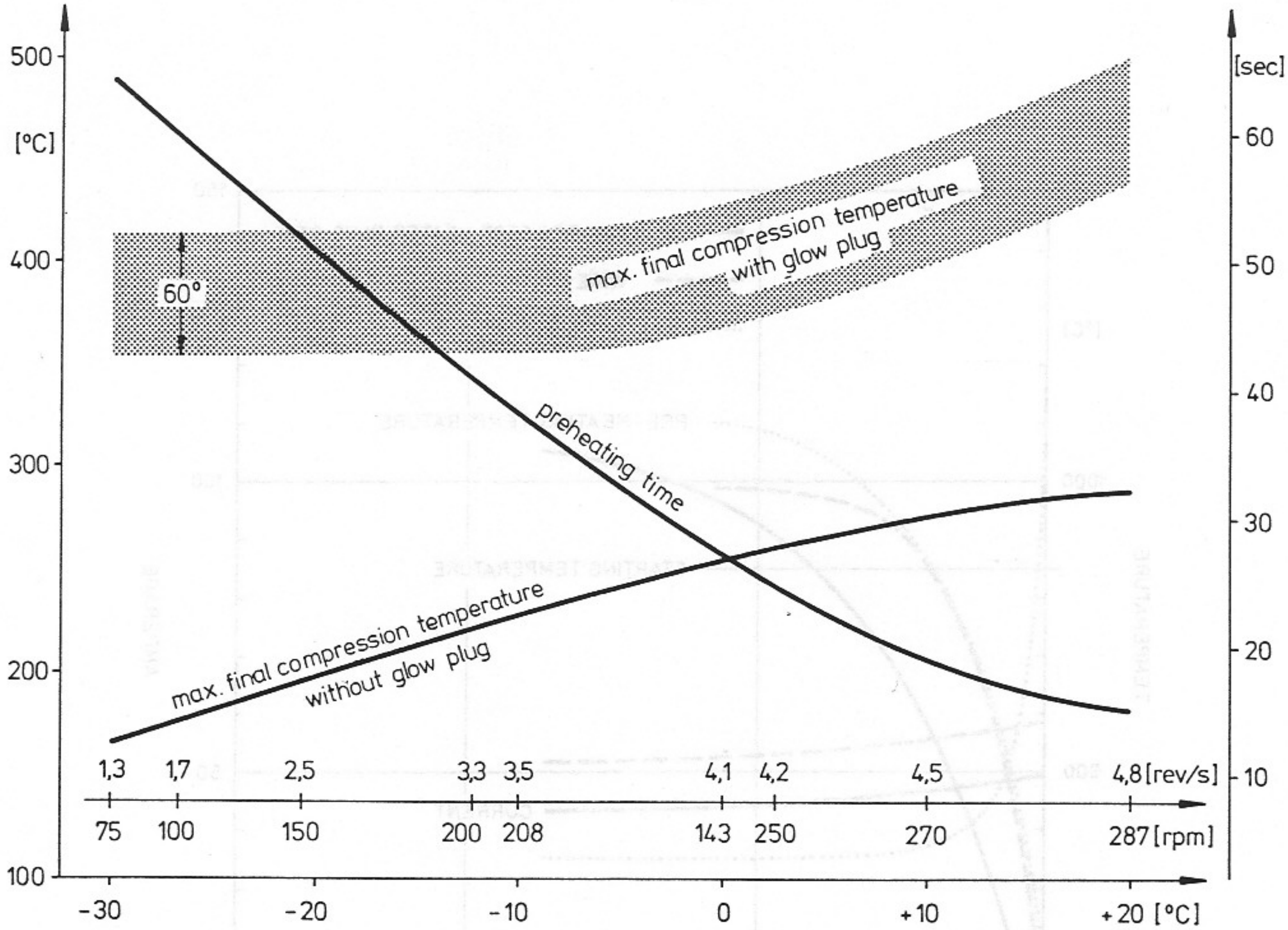


Fig. 11 – Swirl Chamber Temperature at Starting

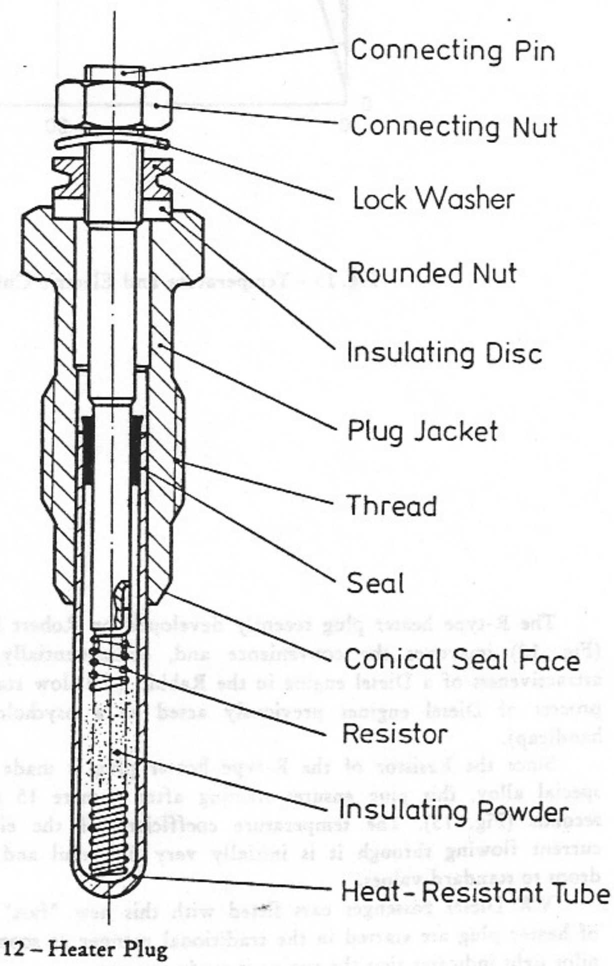


Fig. 12 – Heater Plug

We also studied what engineering changes could be made to raise the final compression temperature, which is greatly affected by losses of mass and temperature.

We investigated the following alternatives (9):

- Further increase in the compression ratio;
- Advancing the closing time of the inlet valve even further;
- Using gas-leak proof piston rings;
- Using ceramic materials.

Within the limits of feasibility, these and other engineering measures brought insufficient improvement. Consequentially, it became necessary to have a starting aid, and the only solutions which seemed feasible in a passenger car were the flame starting device or the heater plug.

In a separate-chamber engine the advantage of the heater plug lies in the fact that it produces heat close to the jet of fuel injected. The wide temperature range shown in Fig. 11 demonstrates that temperature distribution in the swirl chambers is not homogeneous. However, ignition is assured as the jet of fuel is dispersed by swirling and will ignite as soon as it touches the heater plug, if not earlier.

Cost and operational safety reasons prompted us to decide in favor of the HEATER PLUG concept.

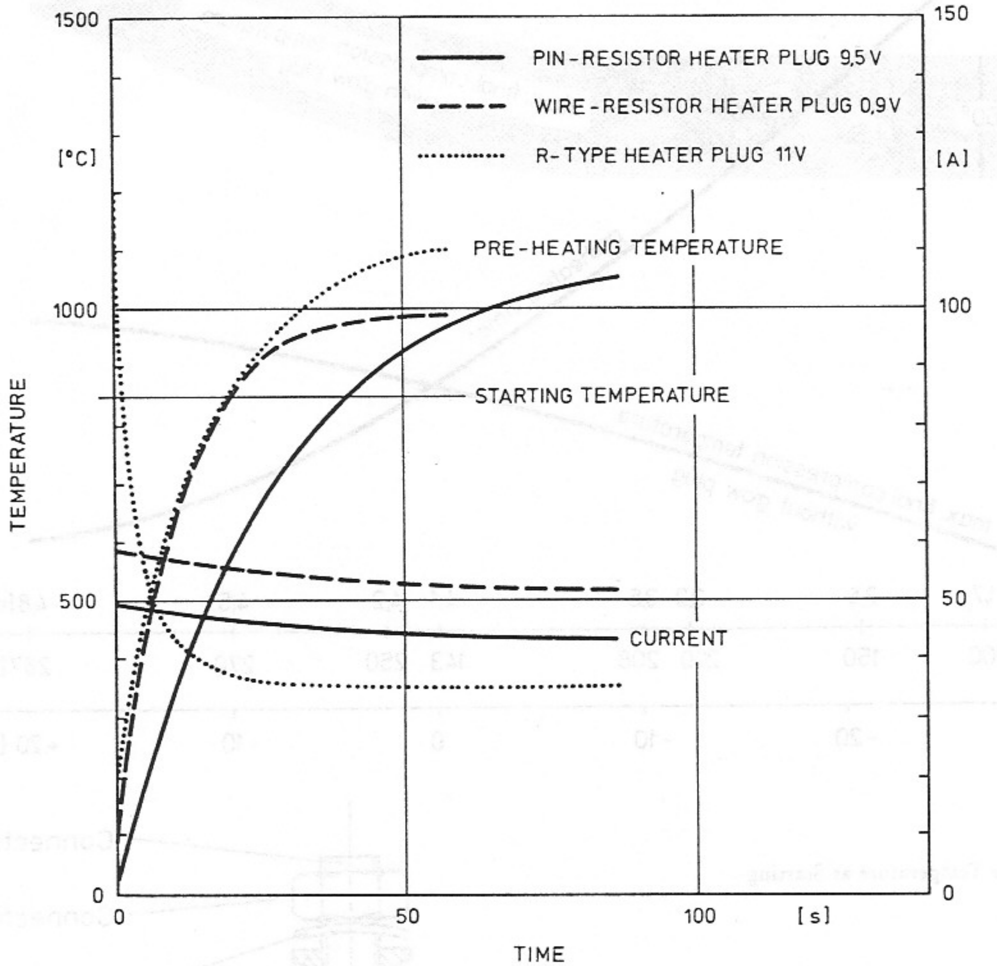


Fig. 13 – Temperature and Electric Current Flow for Differant Types of Heater Plugs

The R-type heater plug recently developed by Robert Bosch (Fig. 12) improves the convenience and, consequentially, the attractiveness of a Diesel engine in the Rabbit, (the slow starting process of Diesel engines previously acted as a psychological handicap).

Since the Resistor of the R-type heater plug is made of a special alloy, this plug ensures starting after a mere 15 to 20 seconds (Fig. 13). The temperature coefficient of the electric current flowing through it is initially very powerful and then drops to standard values.

VW Diesel passenger cars fitted with this new "fast" type of heater plug are started in the traditional manner as soon as a pilot light indicates that the engine is ready to start.

The pilot light will light up as soon as the key is turned into the preheating position and the engine may be started as soon as the pilot light has gone out. The time actually required for preheating depends on the cylinder head temperature. Preheating time is controlled automatically to ensure optimum starting conditions, i.e. minimum emission of blue smoke and good driveability from the first moment. Although it is possible to start the engine before the pilot light has gone out, the user will have to reckon with a poor driveability in this case.

If the engine is still at operating temperature the pilot light will not go on and the engine may be started immediately.

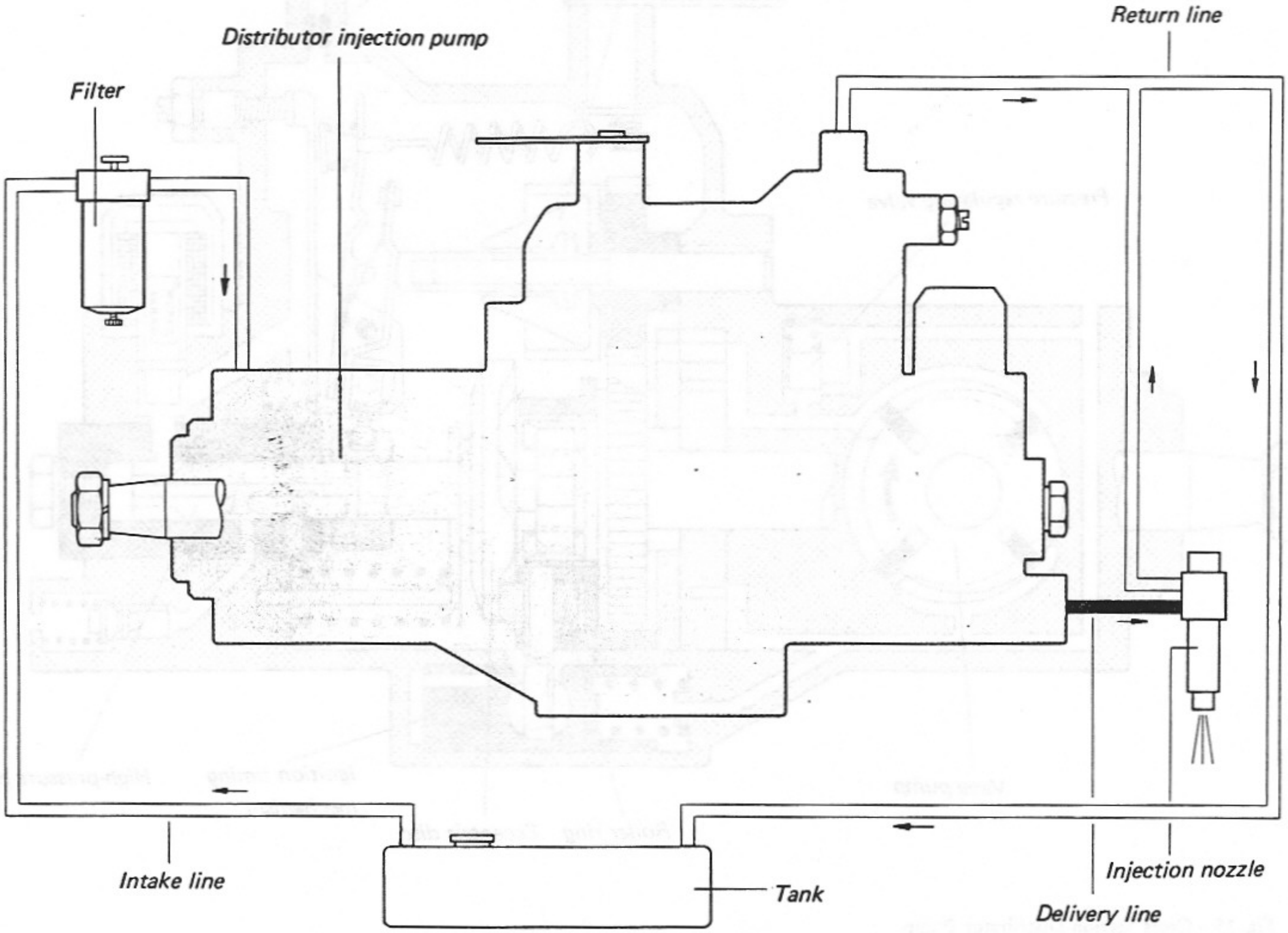


Fig. 14 – Fuel Supply System

FUEL INJECTION SYSTEM

VW's 1.5 Liter Diesel engines are equipped with distributor pumps and throttle pintle injectors. Compared to in-line pumps, distributor pumps are lighter, smaller, and cheaper to manufacture. Lastly, they can be operated at higher speeds.

Fig. 14 shows the FUEL SUPPLY SYSTEM. The fuel is drawn from the tank through a filter and distributed to the injection nozzles in accordance with the firing sequence. Excess fuel, fuel gases from the injection pump, and fuel which has by-passed seals is returned to the tank via a return line.

The Diesel Rabbit does not require a separate fuel pump. Fuel is transported by a vane-type pump integrated into the distributor pump.

If the tank has run dry it is not necessary to use a hand pump to bleed the air from the fuel supply system because the above-mentioned vane-type pump is capable of transporting fuel at starting speeds and of bleeding the fuel supply system automatically.

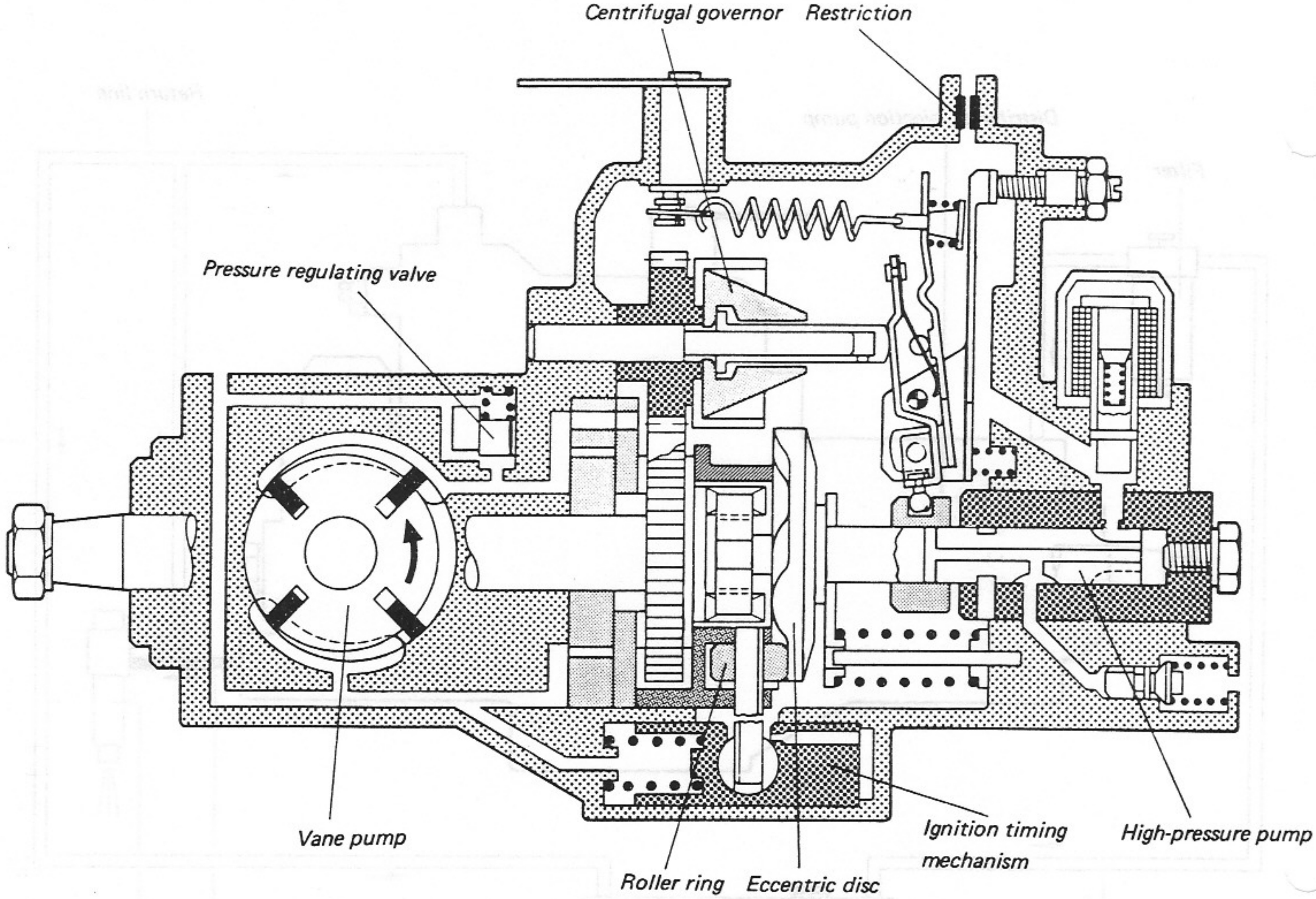


Fig. 15 – Cross Section Distributor Pump

Fig. 15 illustrates the design of the DISTRIBUTOR PUMP (10). The design of Robert Bosch VE pumps is based on that of the well known VA pumps which have been in production for a number of years. As a unit, the distributor pump contains a high-pressure injection pump, a low-pressure fuel pump, an all speed governor, and an injection timer. VW uses a variation of this pump which is fitted with an electric on-off switch. High pressure is generated by a rotating plunger which distributes fuel to the cylinders.

The vane-type fuel pump, the roller ring, the distributor plunger, and the all speed governor are all powered by the injection pump shaft. Unlike the former VA types the speed of VE pumps is governed mechanically and not hydraulically. An injection timer varies the injection timing as required by the engine speed, increasing the fuel pressure inside the pump casing as soon as the engine speed goes up. This activates the spring-loaded plunger of the injection timer, which in turn rotates the roller ring by means of a pin (Fig. 16). This has the effect of displacing its point of contact with the disc contrary to the pump's direction of rotation and of advancing the timing of injection.

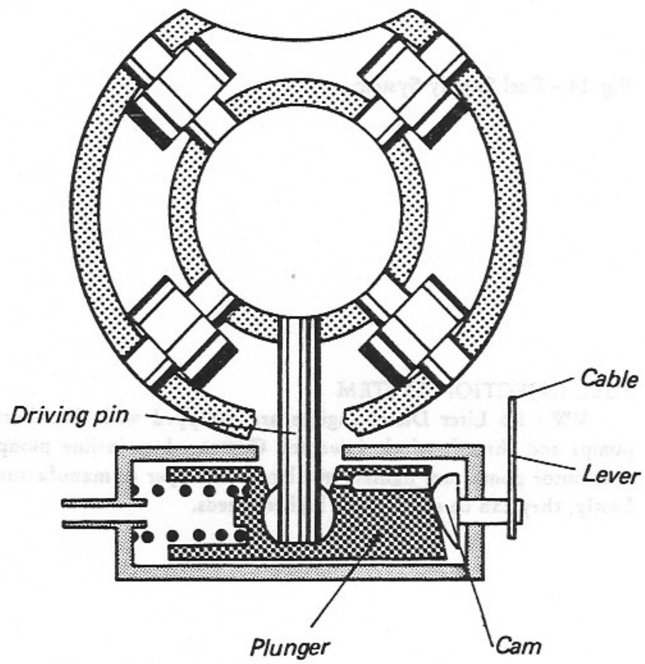


Fig. 16 – Injection Timing

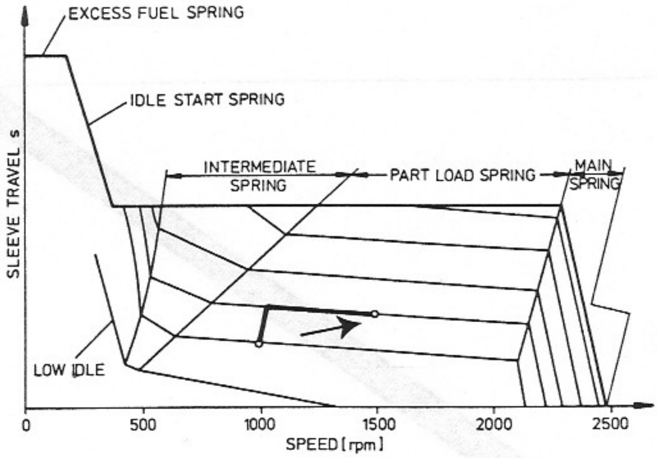


Fig. 17 – Operation of the Car Application Speed Governor

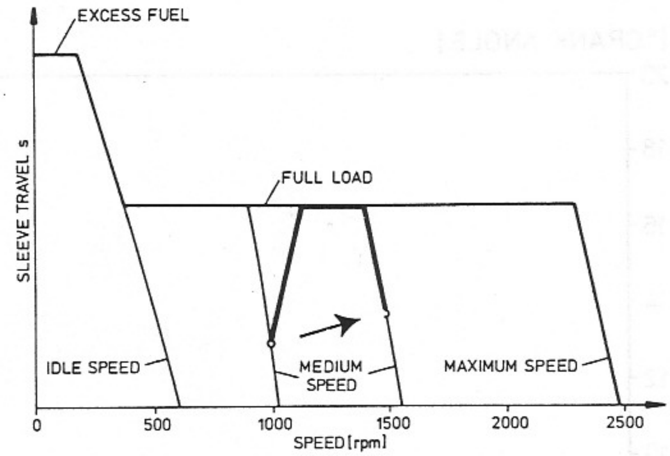


Fig. 19 – Operation of the All Speed Governor

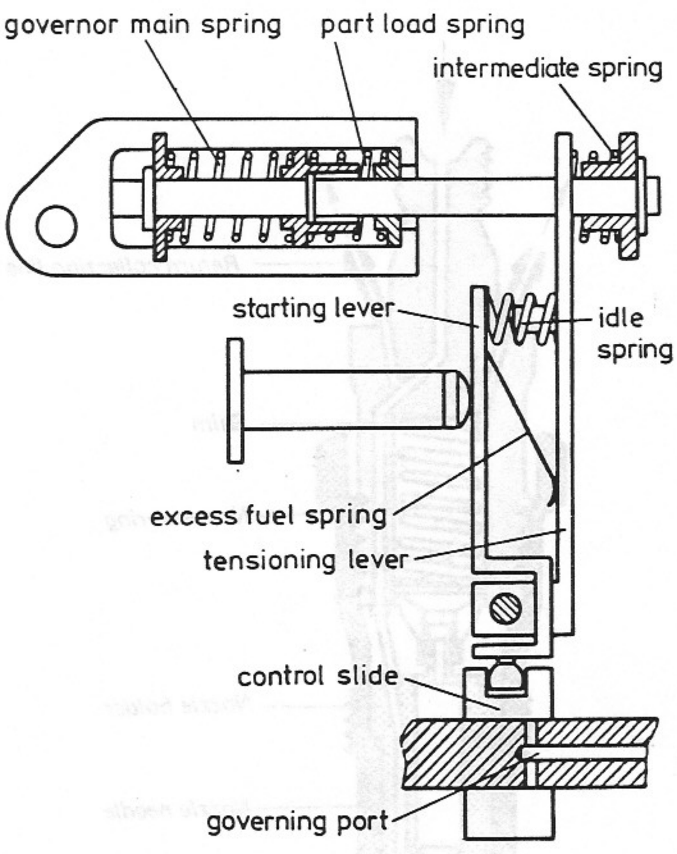


Fig. 18 – Structure of the Car Application Speed Governor

We took care to adjust the injection pump to the requirements of the VW Diesel engine in two respects:

- Adapting the governor to the requirements of a subsompact car;
- Cold-start and warm-up behavior.

When we adapted the VW Diesel engine to the vehicle we ran into several problems connected with the speed governor. Originally, we intended to use an all speed governor which, however, because of the Rabbit Diesel's high HP/IW ratio, produced an acceleration jerk which was unacceptable for good driveability (Fig. 19).

Therefore, we developed what we call a CAR APPLICATION SPEED GOVERNOR which has given the Diesel an acceleration behavior similar to that of a well-groomed spark ignition engine.

The low-speed range of this modified part-load governor has been expanded. It also governs maximum engine speed. When operating at low speeds, this governor ensures smooth load absorption on starting, thus making the maximum possible use of the advantages of its design.

Fig. 17 shows the operation of our car application governor over the entire speed range, covering

- starting
- idling
- part-load (speed range not affected by the governor and controlled directly by the accelerator)
- full load (cutoff speed).

Fig. 18 illustrates both the position and the function of the governor. Its operating range is determined by the type of springs used as well as by the geometry of its levers. Figs. 17 and 19 show the functional difference between a car application speed governor and an all speed governor entered into the acceleration mode of an engine map.

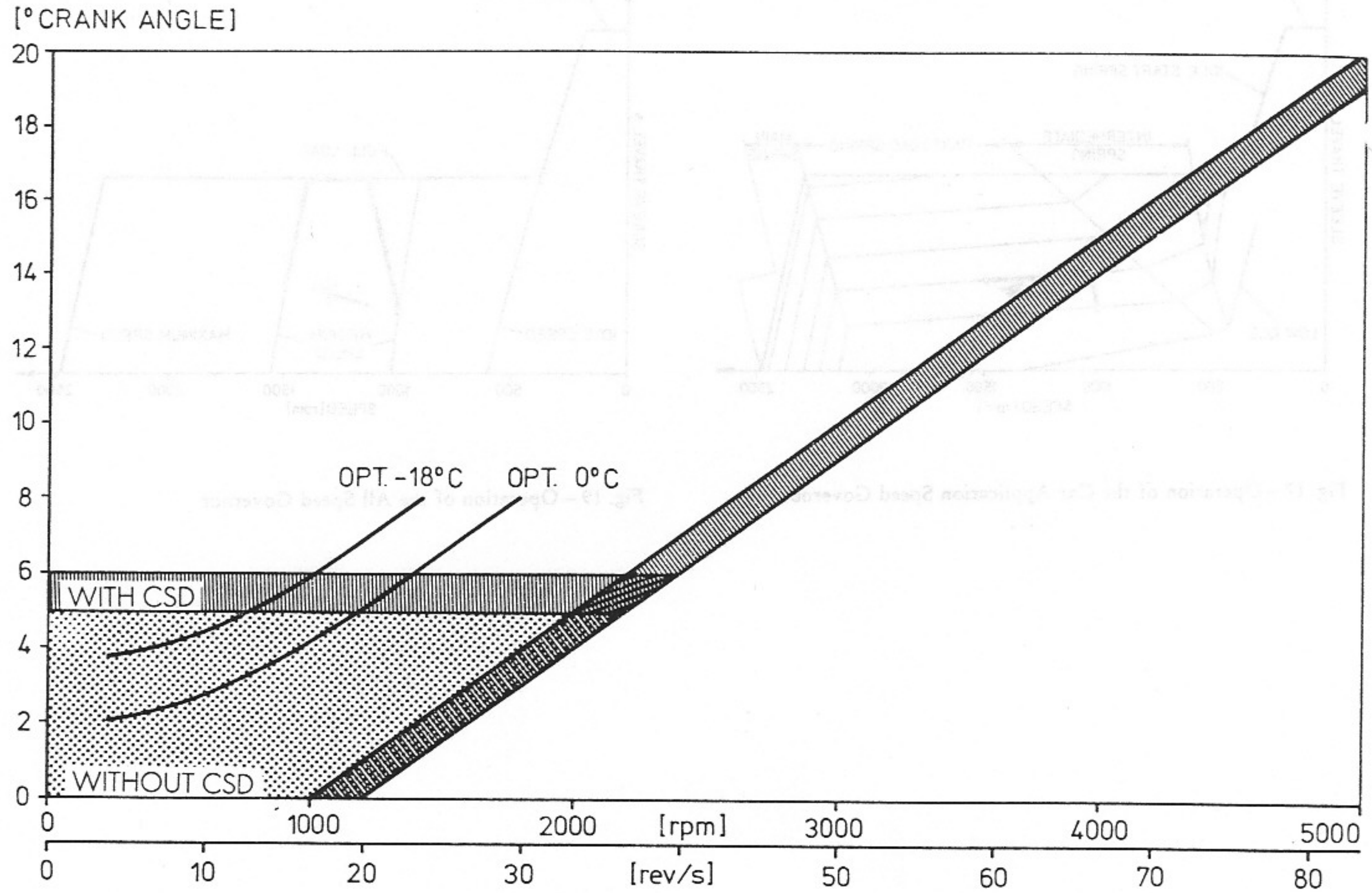


Fig. 20 – Injection Timing Adjustment with CSD

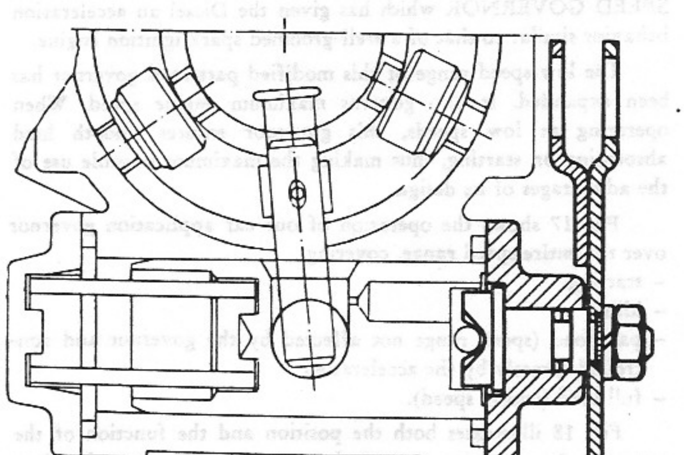


Fig. 21 – Structure of the CSD

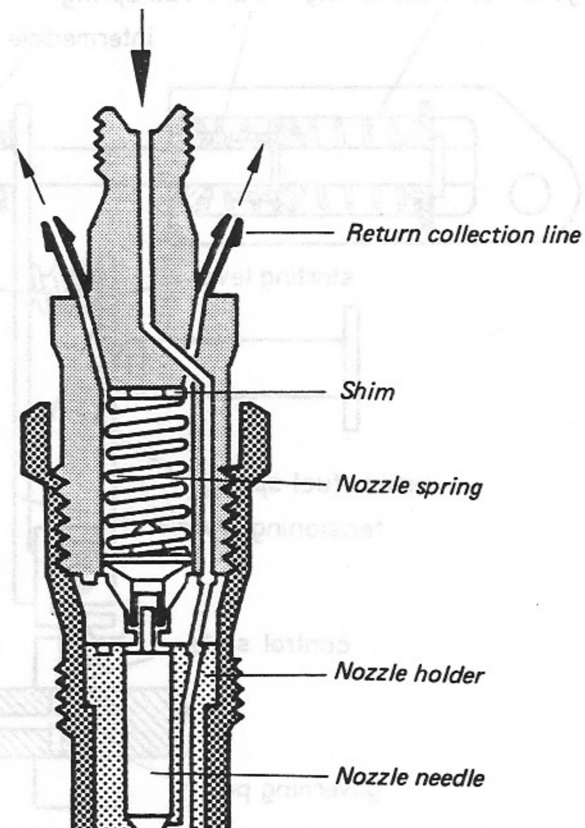


Fig. 22 – Cross Section of a Pintle Nozzle

Another problem was presented by the vehicle's driveability and its blue-smoke emissions under cold-start conditions at temperatures below 0 ° C. In spite of its failings during the warm-up stage the engine started immediately, even at low temperatures.

It is the function of the COLD-START DEVICE (CSD) to advance the time of ignition while the engine is still cold so as to guarantee proper combustion. Once the CSD button is pushed back in after the warm-up phase, injection timing is retarded again. Resetting the CSD button will cut down the engine idle noise appreciably. The curves on Fig. 20 represent ignition timing adjustments both with and without CSD. Moreover, the same diagram contains a curve indicating the injection timer settings required for optimum warm-up (minimum blue smoke). As soon as engine speeds go beyond 2,000 rpm injection timing is not affected any more by the CSD, so that combustion at higher speeds more or less follows a constant-pressure cycle. As a matter of principle, a CSD button, unlike a choke, cannot be mishandled. Fig. 21 shows the design of the CSD. The CSD lever rotates and turns through a pin fitted into the plunger of the injection timer, thus advancing the time of injection.

To get the best possible INJECTION NOZZLE we traded off spray penetration, injection angle, fuel dispersion, nozzle geometry, injection pressure, and swirl chamber geometry. Our Diesel engine uses DNOSD 193 nozzles made by Bosch in KDA SD 27/4 sockets. Fuel is injected at 130 bar pressure. Fig. 22 shows a cross-section of a throttling-pintle nozzle.

To optimize combustion, the injection system and combustion chamber must be compatible with each other. The INJECTION CHARACTERISTIC, or beginning of injection and fuel delivery curve, which is determinded by the type of injection system used, together with the movement of the air in the combustion chamber determines both carburation and combustion. Therefore, we adapted injection pumps, nozzles, nozzle sockets, and injection lines to the requirements of the engine in question.

Because we are using a distributor pump we are in a position to adjust the rate of injection by modifying the cam lift, the cam geometry, the cam configuration, the length of the injection lines, and the reverse flow damping. Moreover, we can adjust the beginning of injection by means of the injection timer.

The volume of fuel displaced by the injection pump plunger is not free of distortion when injected through the nozzle. There is a delay amounting to several °NW between the geometric beginning of delivery and the beginning of injection at the nozzle itself.

Fig. 23 is a comparison between the geometric feed rate and the rate of injection at the rated pump speed of N = 2,500 rpm, which corresponds to 5,000 engine rpm.

Reading from top to bottom, those four diagrams show

1. Geometric feed rate computed from plunger lift (H) and plunger speed (c);
2. Pressure curve at the injection nozzle inlet side;
3. Pintle lift;
4. Actual rate of injection.

Running through the injection lines at the speed of sound, the feed wave front generates sufficient pressure at the nozzle to lift the pintle. The required quantity of fuel is then injected.

All figures use the same time scale. Moreover, the scale quantity is the same for the figure showing the feed rate (top) and the figure showing the rate of injection (bottom). To facilitate comparison, however, we pictured the beginning of the nozzle pressure rise and the beginning of the plunger feed (at about 10° NW) as occuring at the same time. Any retardation of injection timing which might be required because of the engine speed must be controlled by the ignition timer.

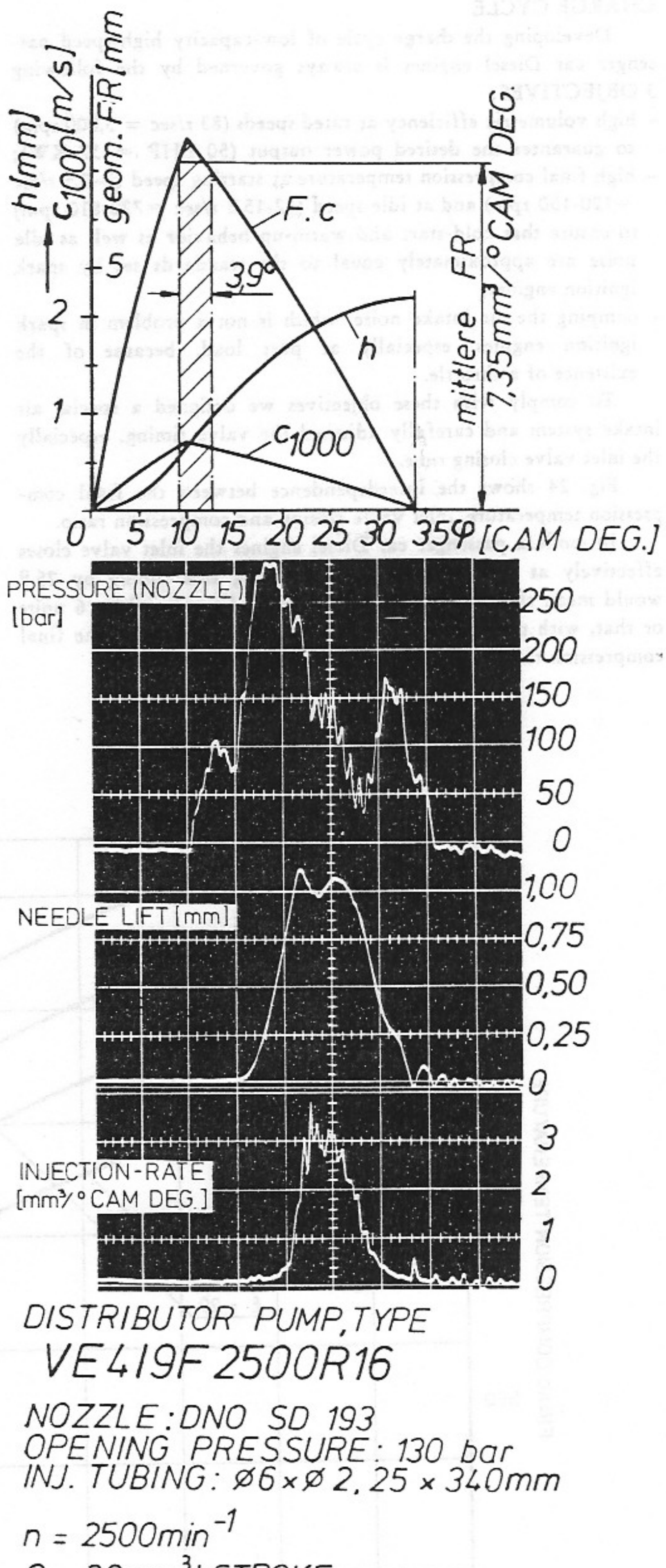


Fig. 23 – Comparison Feed Rate and Injection Rate

CHARGE CYCLE

Developing the charge cycle of low-capacity high-speed passenger car Diesel engines is always governed by the following 3 OBJECTIVES:

- high volumetric efficiency at rated speeds (83 r/sec = 5,000 rpm) to guarantee the desired power output (50 BHP = 37 KW);
- high final compression temperature at starting speed (2–2.5 r/sec = 120-150 rpm) and at idle speed (12-15.5 r/sec = 720-810 rpm) to ensure that cold-start and warm-up behavior as well as idle noise are approximately equal to the standards set by spark ignition engines;
- damping the air intake noise, which is not a problem in spark ignition engines, especially at part load, because of the existence of a throttle.

To comply with these objectives we designed a special air intake system and carefully adjusted the valve timing, especially the inlet valve closing time.

Fig. 24 shows the interdependence between the final compression temperature, inlet valve closing and compression ratio.

In modern passenger car Diesel engines the inlet valve closes effectively at 90 ° after BDC. Advancing this timing by 25 ° would mean that either compression could be reduced by 6 units or that, with the compression ratio remaining the same, the final compression temperature could be increased by 40 ° C.

Therefore, an effort is made to advance the inlet valve closing time as much as possible when optimizing the inlet cam geometry of a passenger car Diesel engine. On the other hand, delivery at rated speed must be sufficient to ensure producing the desired rated power output. Fig. 25 shows the restrictions imposed by the kinematic conditions of the power plant. In this low-capacity engine, we minimized the gap between piston and valve so as to be able to use nearly 50 % of the dead space for the swirl chamber. For this reason, the extent of the lift face of the inlet cam is limited by the piston travel.

So we had to deal with a typical trade-off problem: We had to manoeuvre between a narrow cam (as early as possible inlet valve closure during starting and idle and due to kinematic requirements, inlet valve opening as late as possible) and a broad cam (to ensure maximum possible power output).

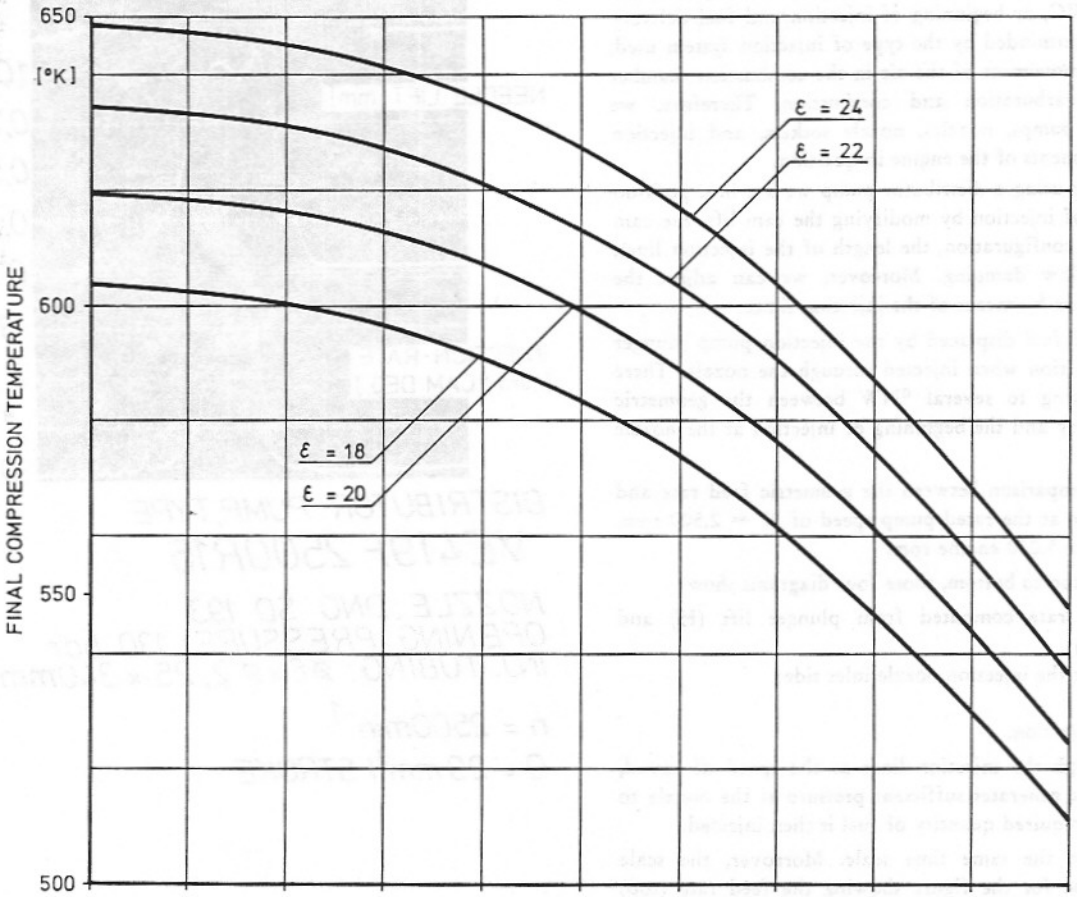


Fig. 24 – Final Compression Temperature as a Function of Inlet Valve Closing Time, Environmental Temperature 20° C

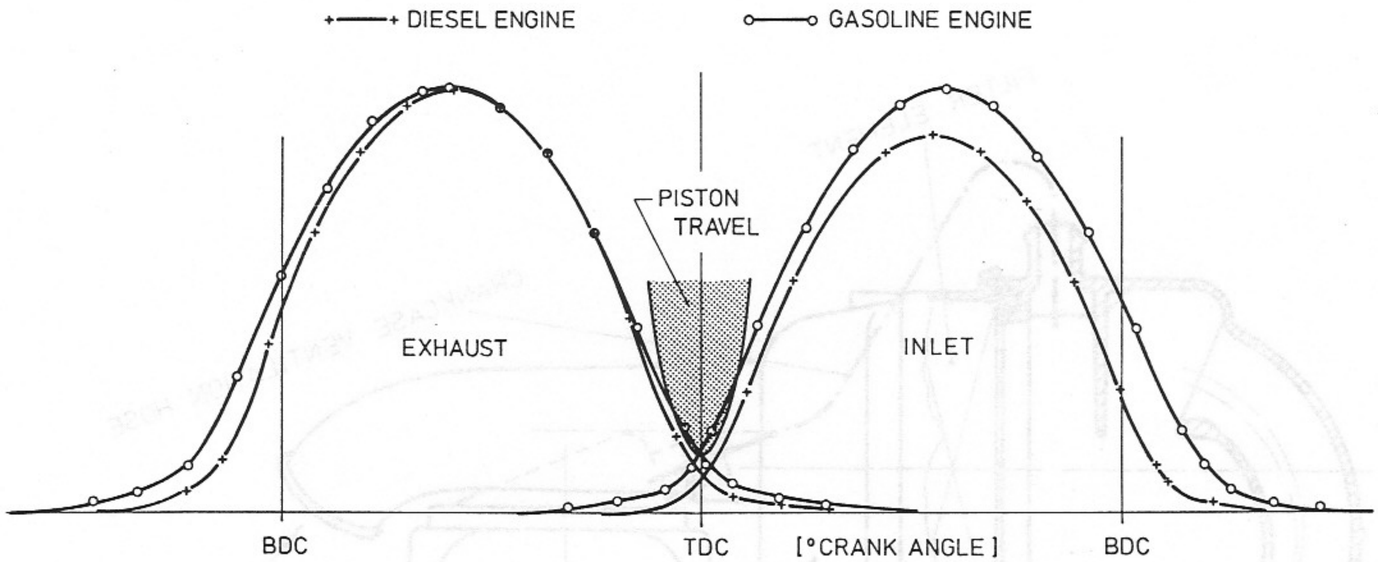


Fig. 25 – Valve Travel Curve

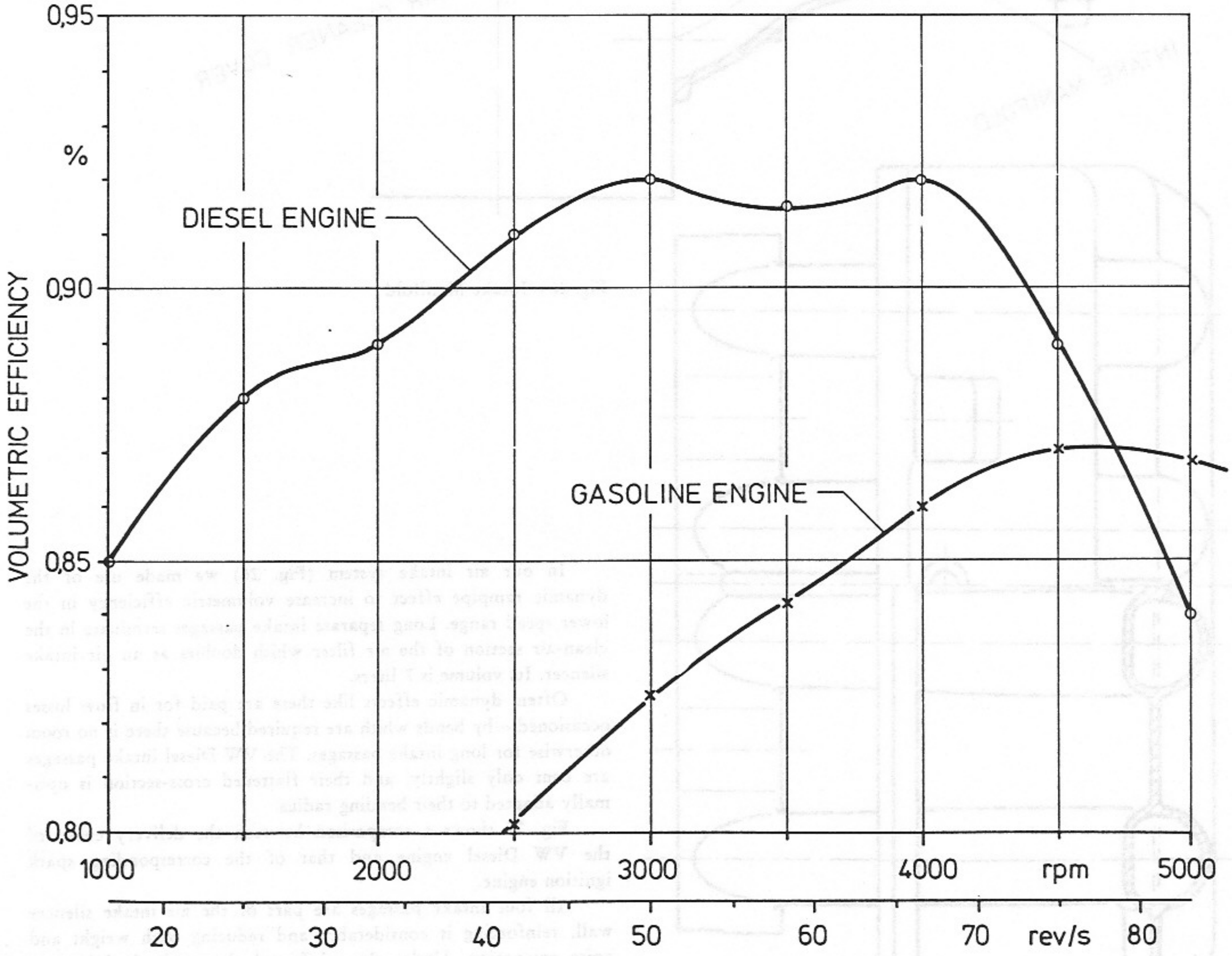


Fig. 27 – Volumetric Efficiency

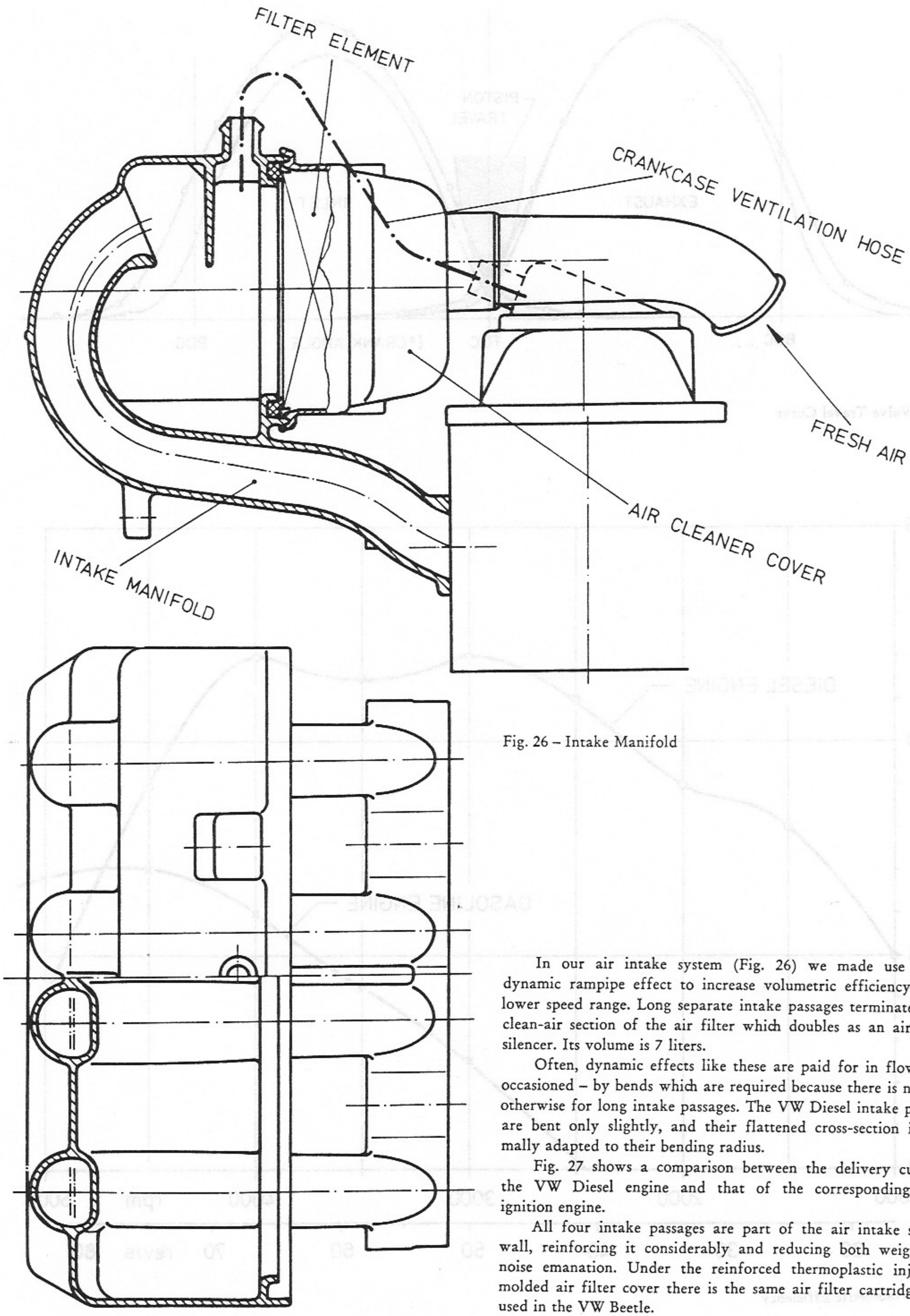


Fig. 26 – Intake Manifold

In our air intake system (Fig. 26) we made use of the dynamic rampipe effect to increase volumetric efficiency in the lower speed range. Long separate intake passages terminate in the clean-air section of the air filter which doubles as an air intake silencer. Its volume is 7 liters.

Often, dynamic effects like these are paid for in flow losses occasioned – by bends which are required because there is no room otherwise for long intake passages. The VW Diesel intake passages are bent only slightly, and their flattened cross-section is optimally adapted to their bending radius.

Fig. 27 shows a comparison between the delivery curve of the VW Diesel engine and that of the corresponding spark ignition engine.

All four intake passages are part of the air intake silencer wall, reinforcing it considerably and reducing both weight and noise emanation. Under the reinforced thermoplastic injection-molded air filter cover there is the same air filter cartridge as is used in the VW Beetle.

POWER PLANT

CYLINDER HEAD (FIG. 28)

Our Rabbit Diesel engine was developed from the 1.5 liter spark ignition engine. (11). Theoretical studies showed that this robust power plant is capable of sustaining much strain. Our theoretical findings were confirmed by the results of another research project in the course of which this basic power plant was brought up to 160 BHP by means of exhaust gas turbocharging.

The cylinder head, which is made of aluminum, is rather high and stiff. As all inlet and outlet ports are located on one side of the cylinder head the swirl chambers could be accommodated on the other side. Of course, we had to modify the cylinder head water jacket quite a bit to ensure increased reinforcement and to adapt the water flow to the requirements of a spark ignition engine.

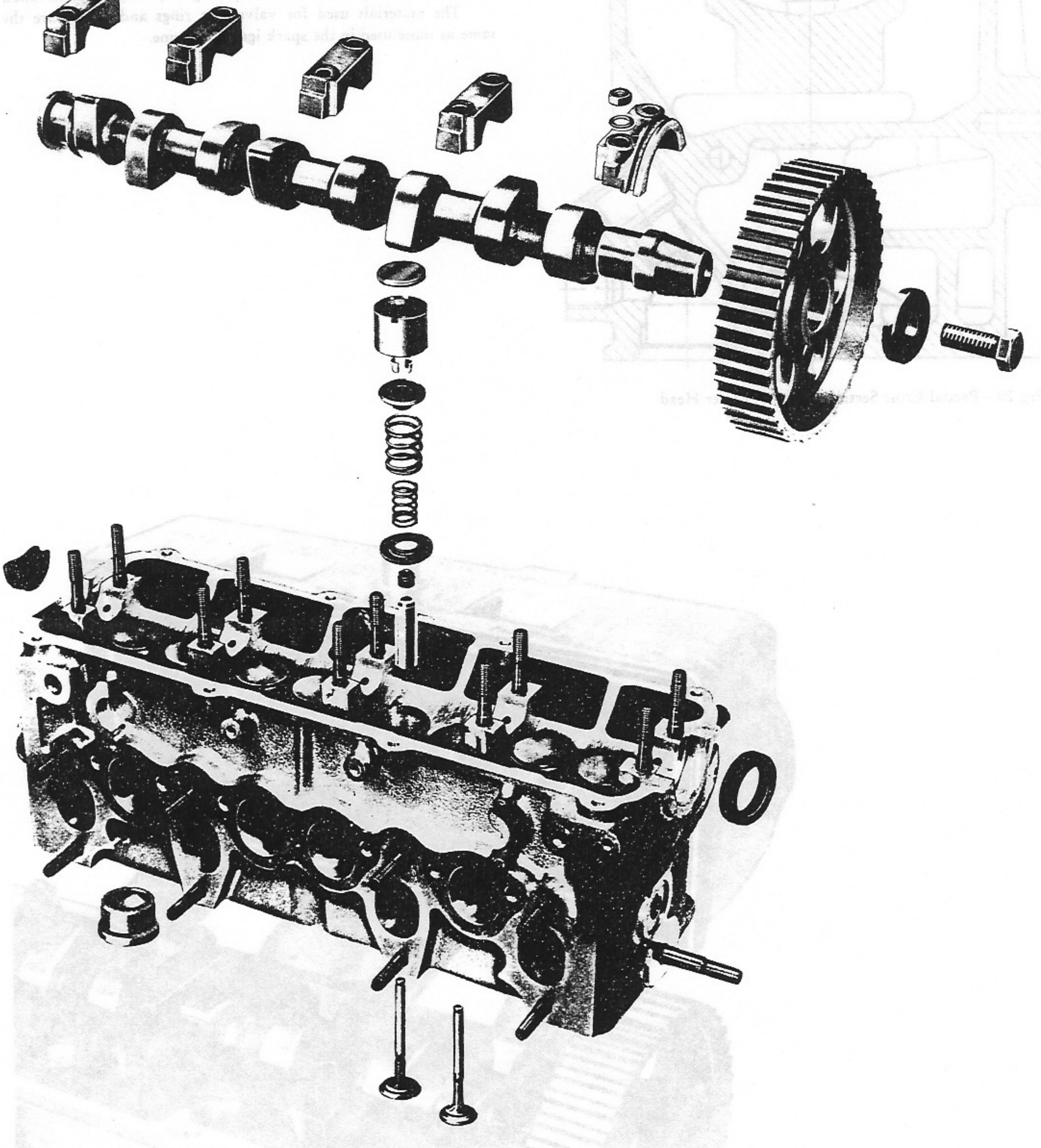


Fig. 28 – Cylinder Head

The wall thickness in the area of the head gasket was heavily reinforced.

To keep thermal overloading from causing cracks between the valve seats, ribs were cast into the cylinder head to direct 75 % of the coolant between the inlet and outlet ports towards the swirl chambers (Fig. 29). This is not the case in the spark ignition engine.

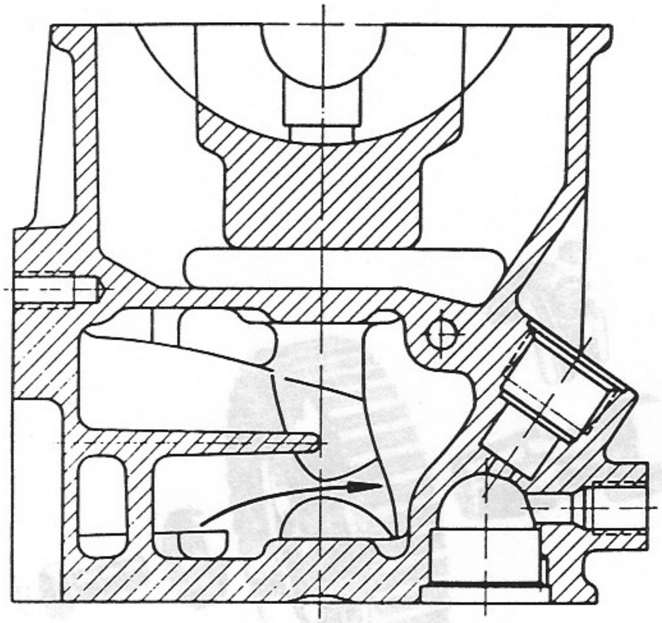


Fig. 29 – Partial Cross Section of the Cylinder Head

We were able to use a cylinder head of nearly the same configuration as that of the spark ignition engine, so that the same mold can be used for casting both. Moreover, both can be machined on the same transfer line.

Diesel engines are always troubled by prematurely wearing inlet valve seat rings. By controling the blow-by carefully we were able to tackle this problem so successfully that in VW Diesel engines valve lash checks are required only at 15,000 km intervals. Valve adjustments are required even more rarely. A rib cast into the air filter casing distributes the blow-by uniformly (Fig. 30). Oil droplets are received by the lower part of the filter, and conducted directly into the intake passages by way of 1-mm holes.

The materials used for valve seat rings and valves are the same as those used in the spark ignition engine.

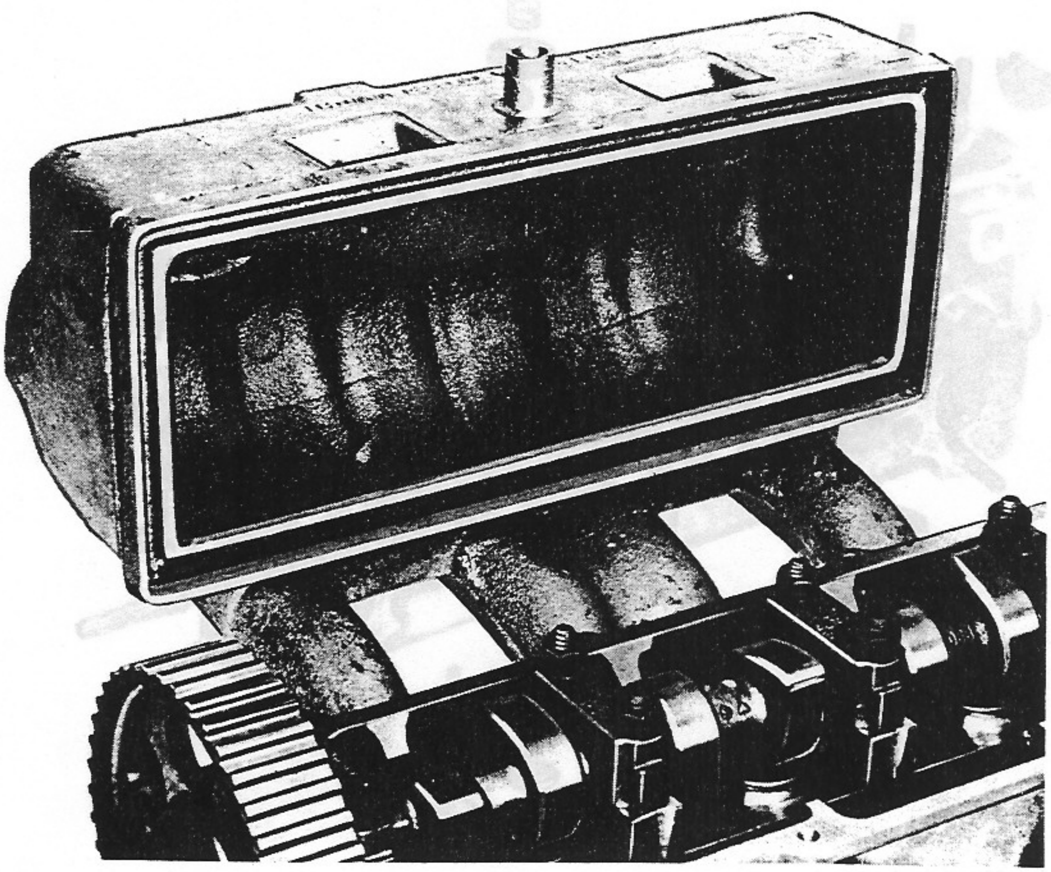


Fig. 30 – View of Air Filter Housing

CYLINDER HEAD GASKET (Fig. 31)

In order to keep close compression ratio tolerances and to ensure that under present conditions of manufacture the gap between piston and cylinder head is kept at a minimum while avoiding any collision between valves and pistons, three different thicknesses of cylinder head gaskets (1.2, 1.3 and 1.4 mm) are used in producing the Diesel engines. Fig. 32 shows the statistical distribution of compression ratios in production.

Exhaustive preliminary tests were carried out before subjecting the gasket to any large-scale testing. We determined statistically the pressure distribtuions occurring at various tightening torques. The torque setting finally decided upon was 90 Nm. That this torque is really the best was demonstrated by further experiments. Increasing the tightening torque to 100 Nm will improve the static distribution of pressure but will decrease the resistance of the assembly to extreme thermal shocks.

The dynamic setting behavior of released gaskets was determined by means of the lead-block method. We found that with the engine operating, gaskets will set with a variations of 0.03 mm due to differences in measurement locations.

Tolerances can be closely adhered to by using gaskets of a thickness compatible with the prevalent degree of piston protrusion.

Proper functioning of the gasket was least insured in the area around the outlet ports (low pressure). This condition was relieved by partially extending the flange (Fig. 31).

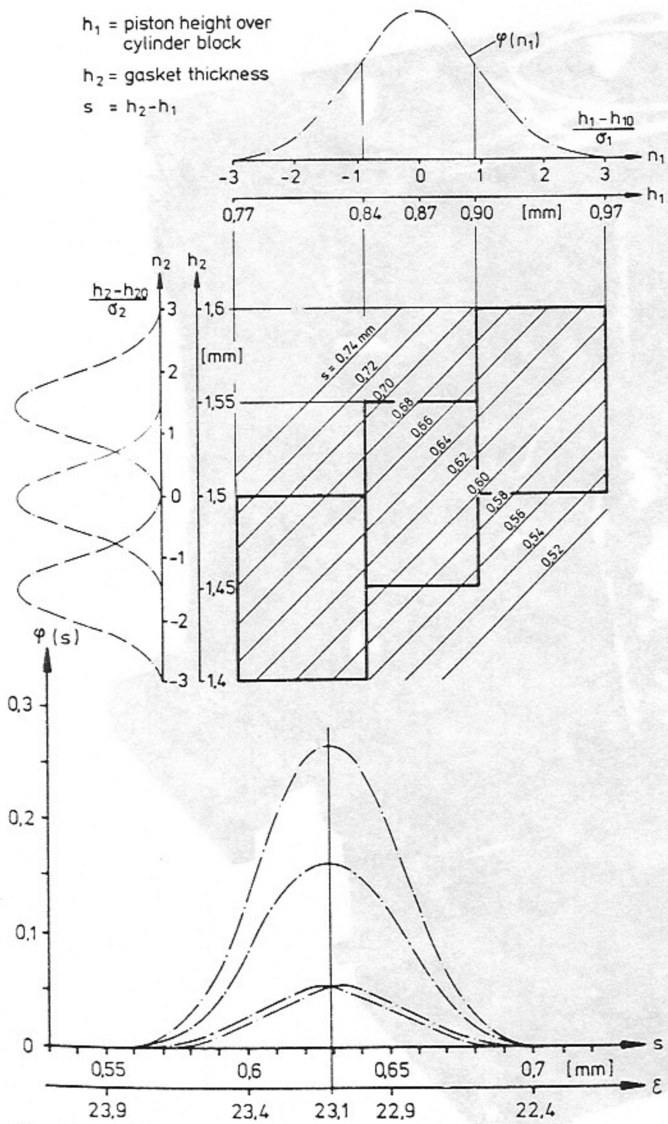


Fig. 32 – Distribution Compression Ratio

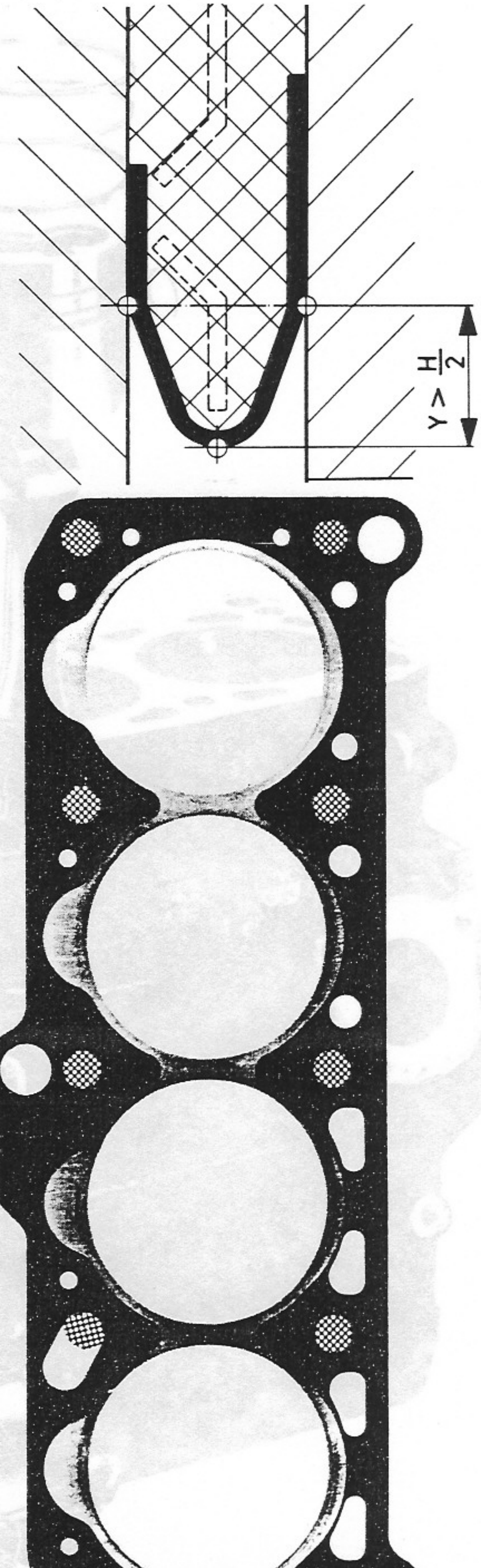


Fig. 31 – Cylinder Head Gasket

CYLINDER BLOCK (Fig. 33)

The cylinder block was also developed from that of the 1.5 liter spark ignition engine (11) the capacity of which has meanwhile been increased to 1.6 liters by expanding the bore from 76.5 mm to 79.5 mm.

To ensure the long service life generally expected of Diesel engines we decided to use the 1.5 liter cylinder block. A 76.5 mm bore allows the flow of coolant between the cylinder barrels, and the distance between them is sufficiently large to ensure the efficiency of the cylinder head gasket.

The seal face was reinforced by modifying the water jacket. This did not cause any heat dissipation problems in connection with the upper piston ring as we enlarged the piston heat dam as well.

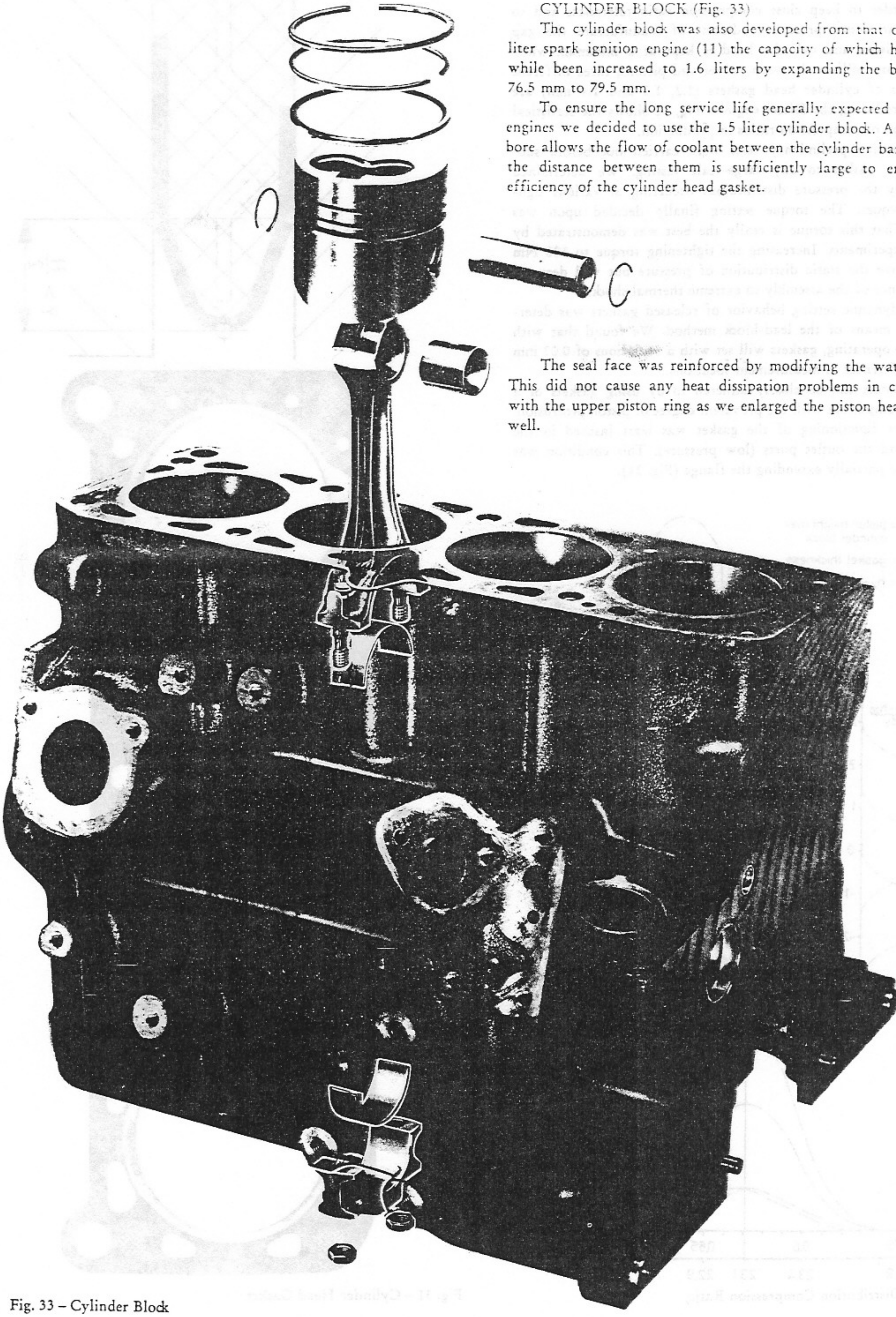


Fig. 33 – Cylinder Block

AUTOTHERMIC PISTON (Fig. 41)

The following requirements were to be met in developing the piston:

- Long service life up to modern passenger car Diesel engine standards;
- Low noise;
- Uniform low oil consumption, minimal blow-by, no seizure.

In recent years, developments in high-speed Diesel engine pistons have shown that their service life greatly depends on the wear of the upper ring and groove. Under favorable conditions, pistons made of one metal only may run for up to 100,000 km, but they are not up to the requirements of heavy duty or a longer service life. For this reason, experiments to develop a ring carrier piston began relatively early. These pistons wear much more slowly because they are equipped with cast-in Ni-Resist ring carriers in which the upper ring groove is punched. This also reduces wear on the piston ring flanks. An Alfin process is used to fuse ring carrier and piston. For details, see Figs. 34 and 35.

Figs. 36 and 37 show the structure of ring carrier (NI-Resist) and piston (Mahle 124).

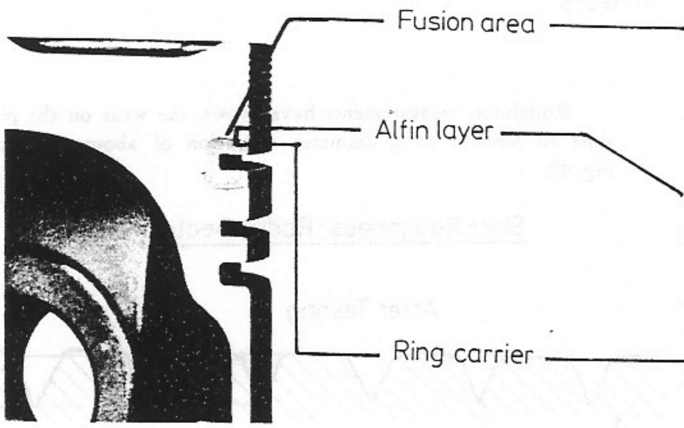


Fig. 34 Cross Section of Ring Carrier

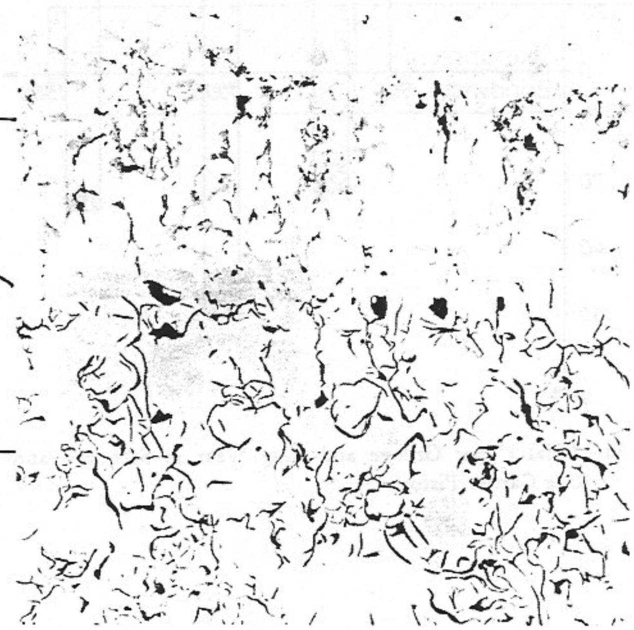


Fig. 35 – Metallic Structure Showing the Fusion of the Ring Carrier (Magn. 100)

Fig. 36 Structure of Ring Carrier Material (Ni-Resist) Magn. 200

Fig. 37 – Structure of Piston Alloy (Mahle 124, Al Si Cu Ni Mg) Magn. 100

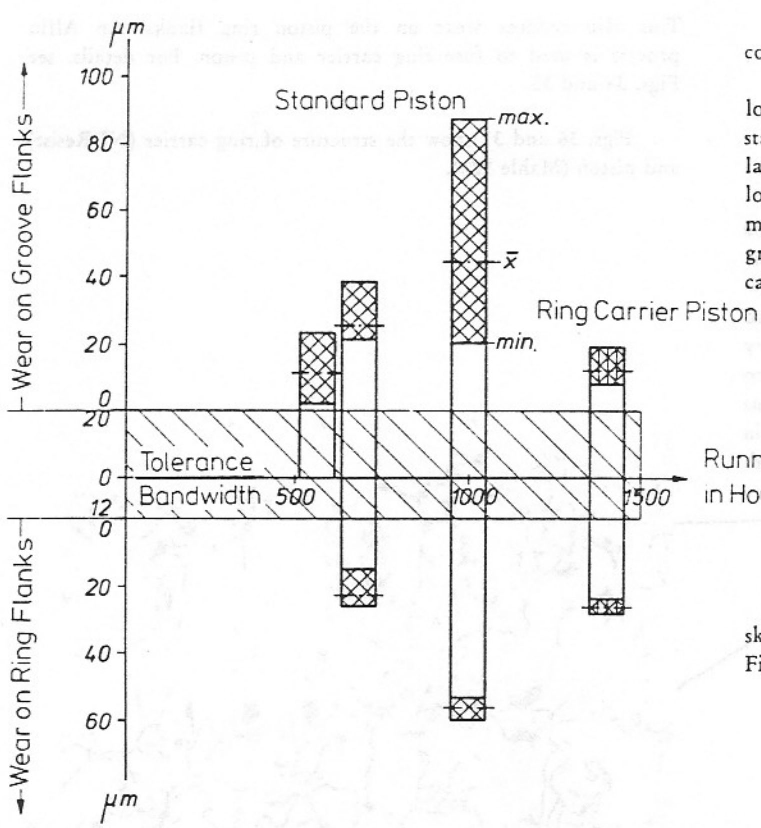


Fig. 38 – Axial Upper Groove and Ring Wear in Standard and Ring Carrier Pistons

Experiments comparing ring-carrier and standard pistons confirmed that this is a development in the right direction.

Fig. 38 shows wear measured in the course of the first few long-term tests at the upper rings groove. Rings and grooves of standard pistons will show wear by about 0.1 mm after bench tests lasting no more than 500 or 1,000 hours. Operating under the same load, ring carrier pistons show wear amounting only to approximately 12 micron at the upper and lower flank of the upper groove after 1.500 hours (see Fig. 39). Axial ring wear, in this case, amounts to about 25 micron.

Roughness measurements have shown the wear on the piston skirt to amount to a diameter reduction of about 20 micron, Fig. 40.

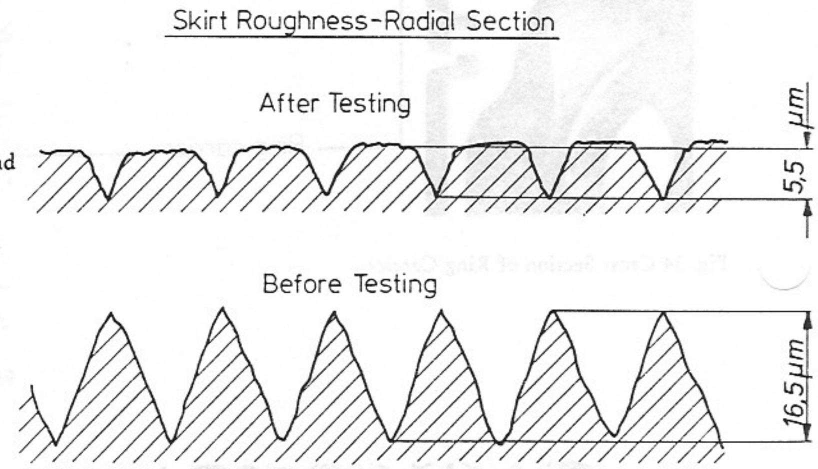


Fig. 40 – Piston Skirt Roughness before and after Testing

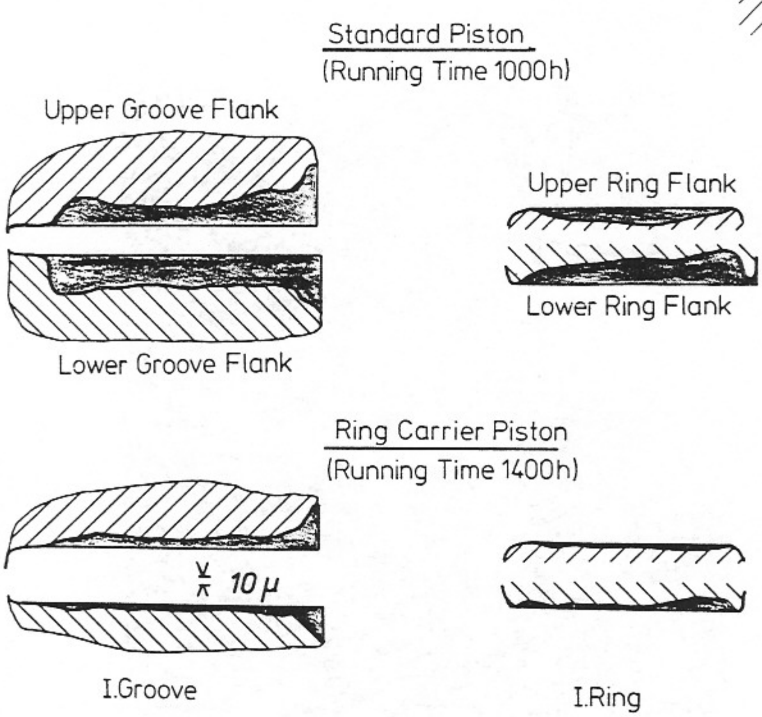


Fig. 39 – Wear of Grooves and Ring Flanks

To reduce the noise level Mahle has developed steel-strutted pistons for Diesel engine use featuring

- low clearence as well as
- favorable deformation behavior.

All Mahle autothermic ring carrier pistons used by us are steel-strutted pistons without a thermal slot between skirt and ring section. Installation clearance is low, which means high warp resistance as well as a uniform and uninterrupted flow of heat from piston top to skirt.

Fig. 42 shows the expansion of the piston under heat. The effect of the struts reacting to pressure in the skirt ($\alpha K = 16.10^{-6}$) is such that more than 50% of the expansion caused by heating piston and cylinder is nullified. Therefore, the engine will work on very low clearances through its entire load range. The clearance of smooth-skirted pistons is designed for maximum power only, so that they are larger than is really required by a cold engine and under partial load.

Fig. 43 shows the contours of the piston together with some installation clearances at various levels. The interrupted line in the region of the skirt indicates the clearance required by a smooth-skirted piston.

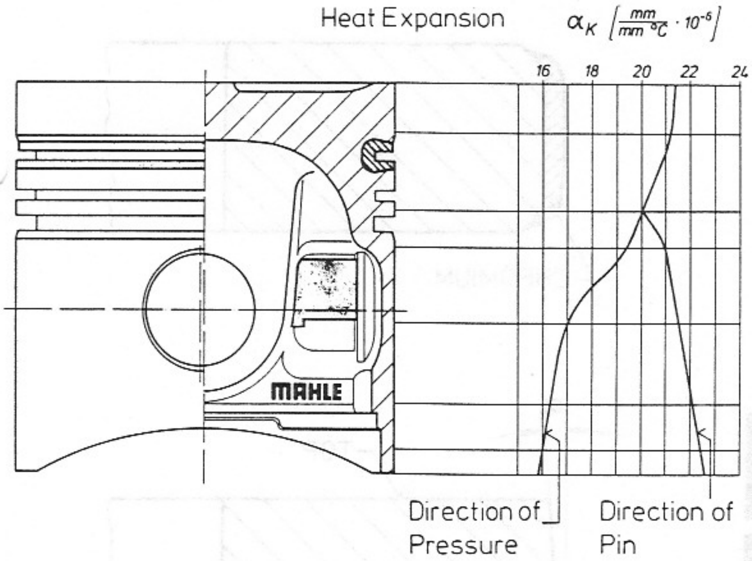


Fig. 42 – Self-Regulating Effect of Autothermic Piston

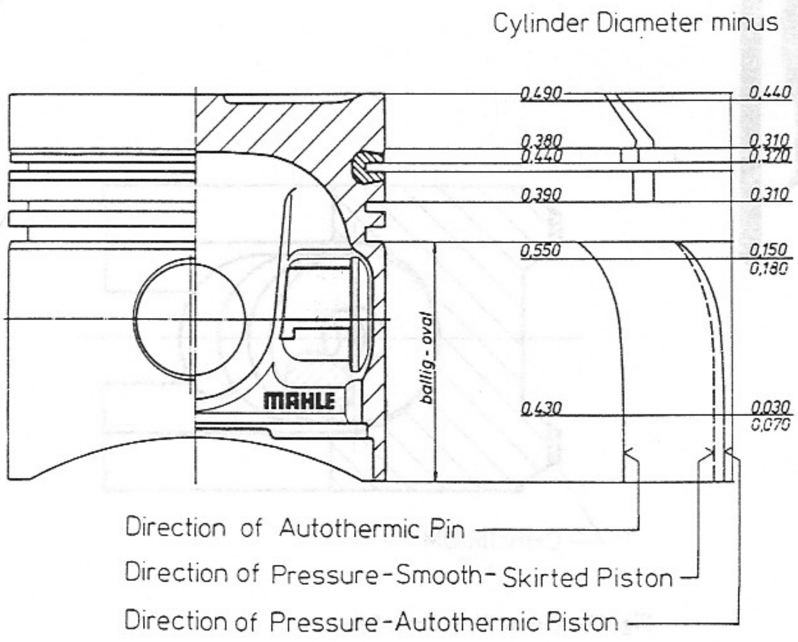


Fig. 43 – Configuration, Clearance, and Ovality of the Autothermic Piston

Skirt deformation is one of the major factors influencing the long-term behavior and service life of a piston. A skirt must be sufficiently elastic to yield to deformations caused by differences in loads and temperatures. However, to ensure a low risk of seizure a certain amount of stiffness is needed to avoid plastic skirt deformation. Bench test results were favorable:

No deformation at the skirt below the oil ring groove; at the nominal diameter level (barrel), deformation amounted to approximately 20–30 micron According to our experience, road test results will be lower.

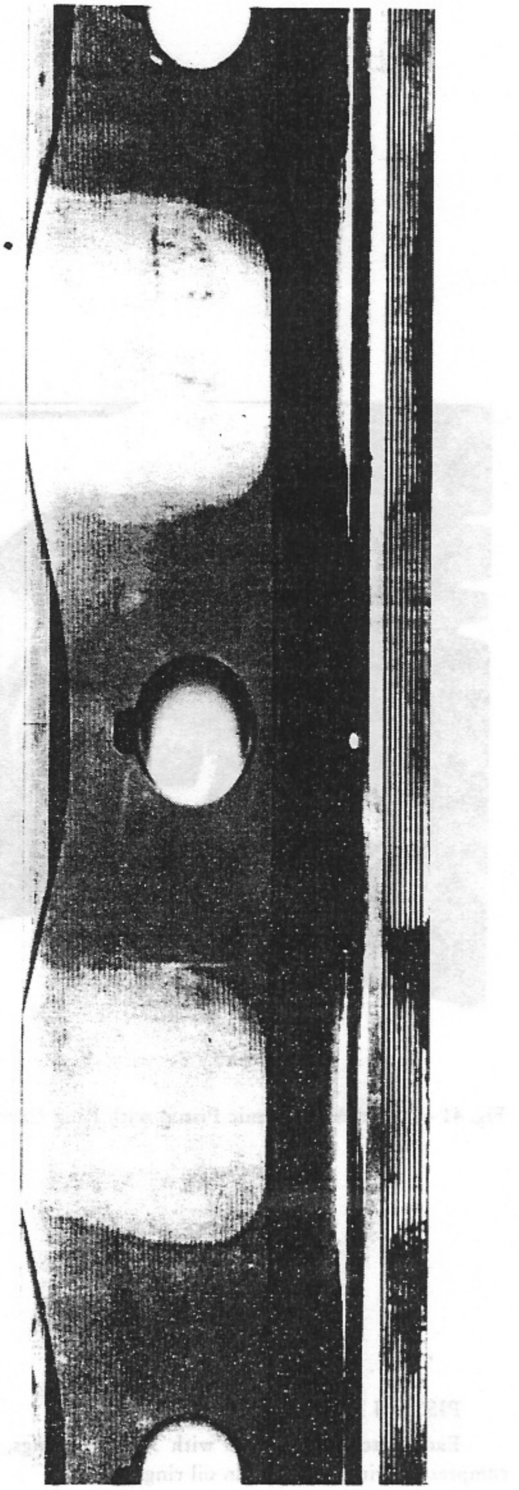


Fig. 44 – Piston Wear (360°) after a 1.400 Hour Bench Test.

OIL CONSUMPTION AND BLOW-BY

An optimum relationship between piston, piston rings and cylinder was found at a relatively early stage in the history of piston development. Long-term tests proved this configuration to be the best (see Fig. 44).

Fig. 44 – Piston Wear (360°) after a 1.400 Hour Bench Test.

Because in the long run the development of oil consumption and blow-by depends on piston wear and ring mobility, the heat dam clearance was designed to prevent serious carbonization in the piston ring grooves. Moreover, the fact that there is a sealing flange above the first groove considerably reduces the thermal load acting on the upper piston ring.

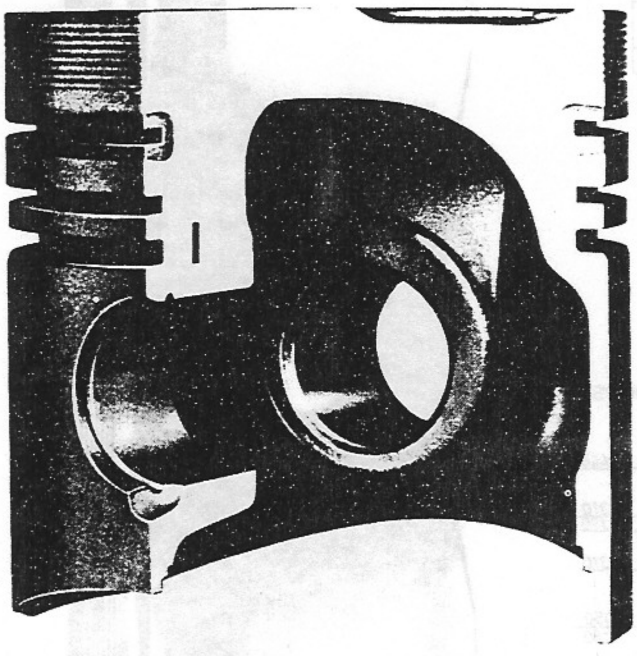


Fig. 41 – Mahle Autothermic Piston with Ring Carrier

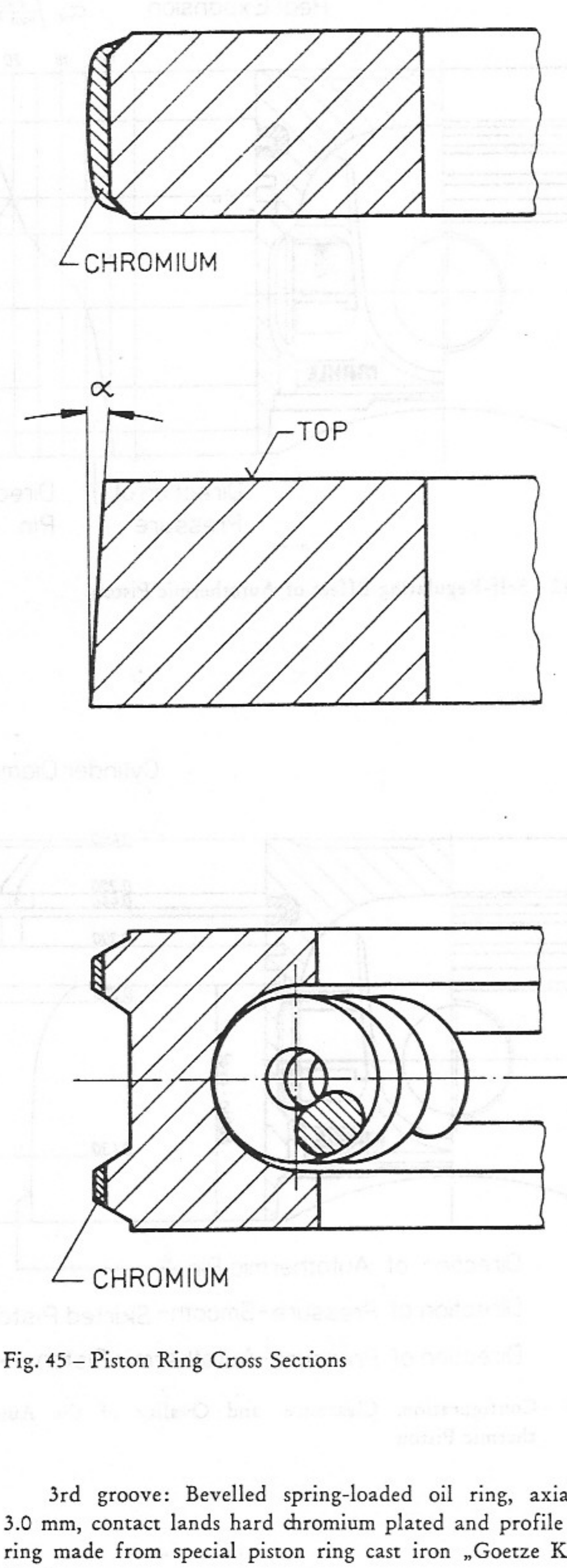


Fig. 45 – Piston Ring Cross Sections

PISTON RING EQUIPMENT (Fig. 45)

Each piston is equipped with 3 Goetze rings, 2 of which are compression rings and 1 is an oil ring.

1st groove: Plain compression ring, axial width 1.75 mm running surface hard chromium plated and barrel lapped; ring made from high strength spheroidal graphite cast iron type „Goetze KV1".

The hard chromium plated running surface of the ring ensures low wear over a long operating period and the barrel lapped running surface provides good compatibility by improving lubrication.

2nd groove: Taper faced compression ring axial width 2.0 mm. Taper angle of the running surface $\alpha = 45' \pm 15'$, ring entirely phosphated and made from special piston ring cast iron „Goetze K1" standard.

Due to its tapered running surface this ring not only provides adequate gas sealing but also aids in controlling the oil consumption of the engine.

3rd groove: Bevelled spring-loaded oil ring, axial width 3.0 mm, contact lands hard chromium plated and profile ground; ring made from special piston ring cast iron „Goetze K1" standard.

With its small axial width and the corresponding small cross section this ring is very flexible. Due to the rear-mounted coil spring it ensures good conformability and effective oil scraping action. The hard chromium plated contact lands are highly wear-resistant. Their small width, which is obtained by using small machining tolerances, as well as their defined shape, which i obtained by profile grinding, are maintained during long running periods.

Given the design of pistons and piston rings as demonstrated in this chapter, the average oil consumption is 0.5 g/PSh, whereas blow-by amounts to 0.4 – 0.5 % of the theoretical air throughput.

TOOTHED BELT DRIVE (Fig. 46)

Camshaft, injection pump and oil pump are driven by a Pirelli ISORAN toothed belt. The toothed surface of the belt drives the camshaft and the injection pump, while the oil pump is friction driven by the back of the belt. The use of a toothed belt was instrumental in accomplishing the design of a Diesel engine that combines the traditional advantages of a Diesel engine with those of the spark ignition engine.

The advantages offered by this type of drive system have been proven in spark ignition engines with small and medium displacement. The drive is not very elastic, does not need any lubrication, is very light, rugged, compact, and has a low noise level. Maintenance is easy and economical.

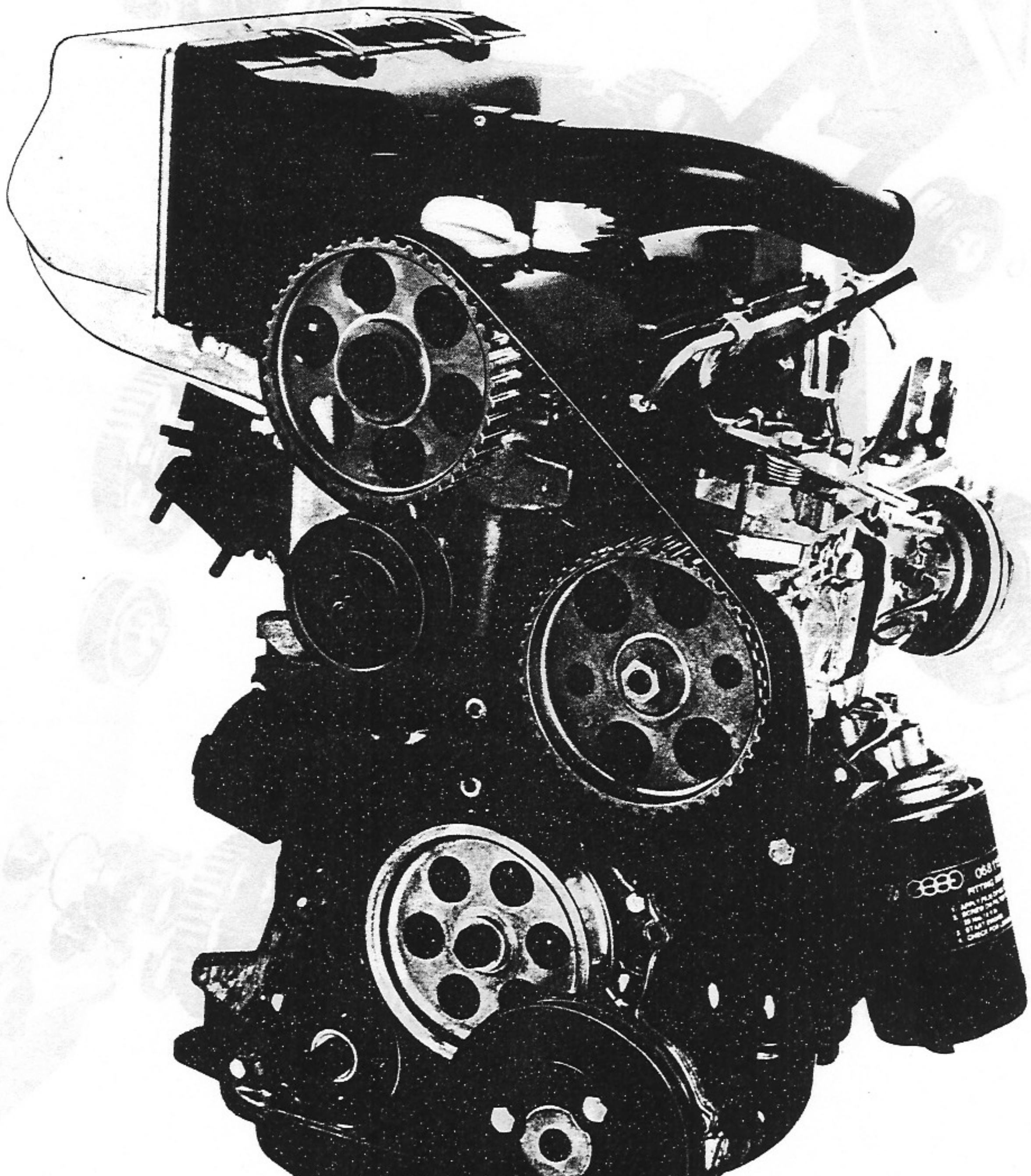


Fig. 46 – Toothed Belt Drive

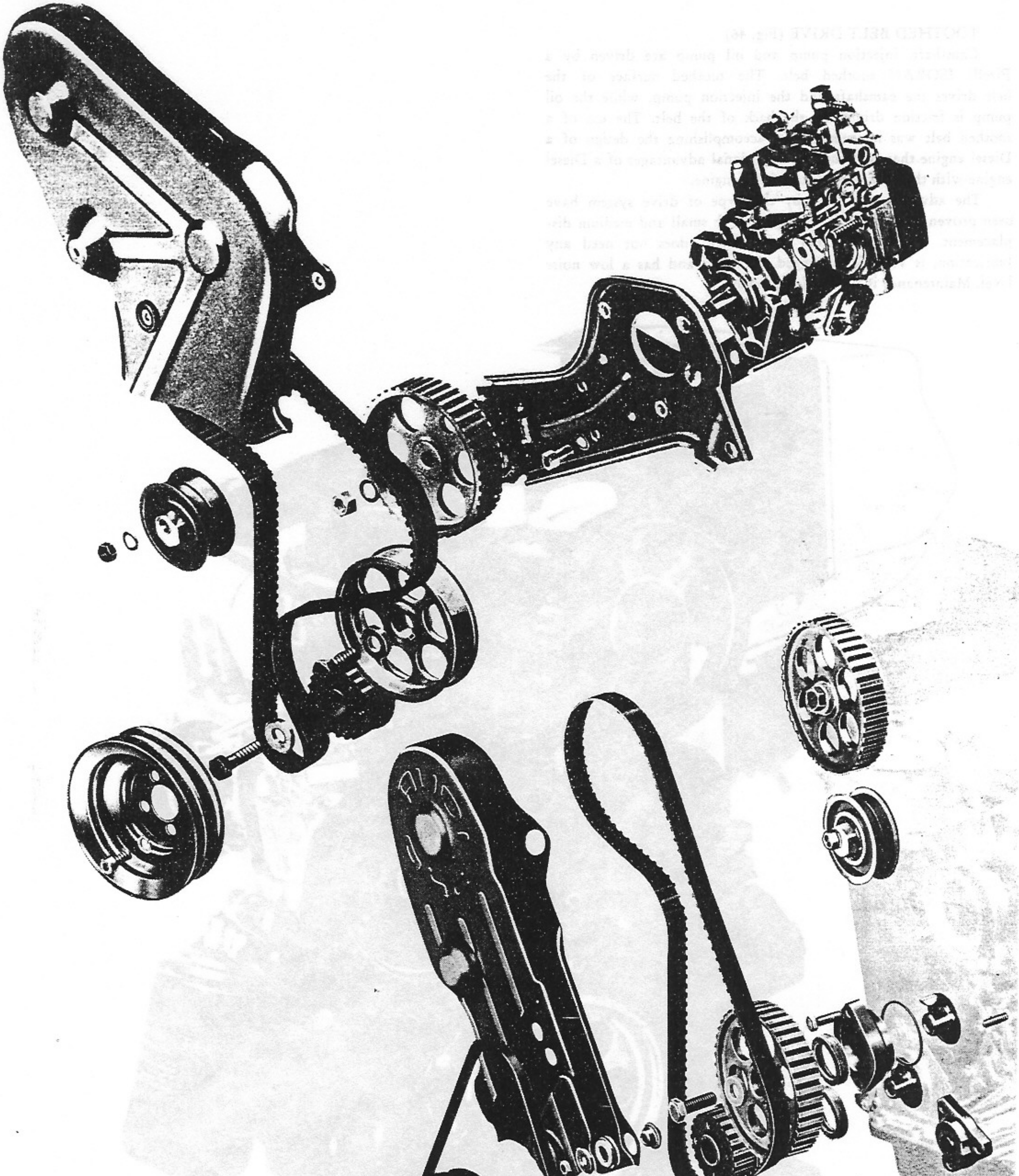


Fig. 47 – Arrangement of Toothed Belt Drive: VW Diesel- Spark Ignition Engine

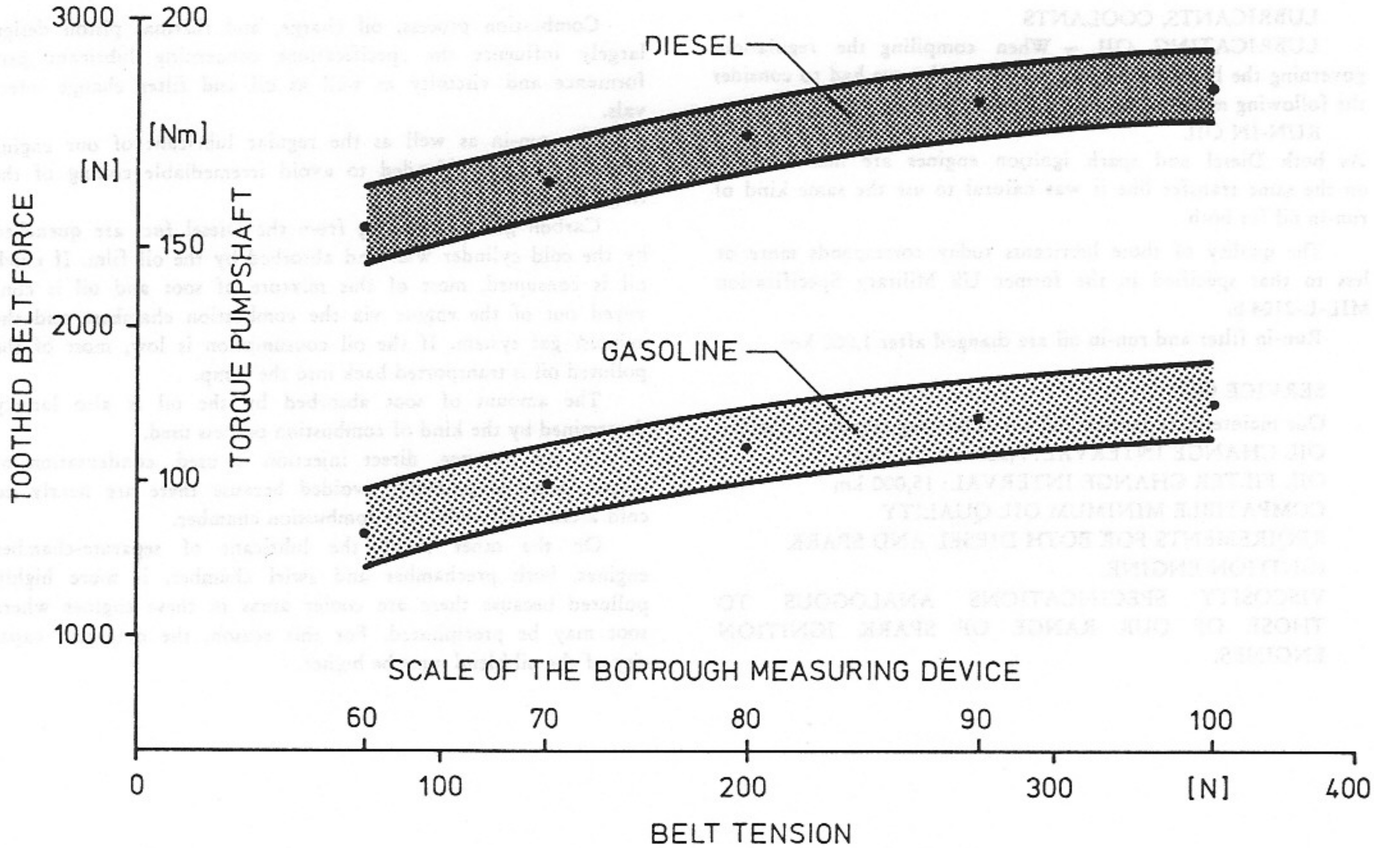


Fig. 48 – Jumping of the Toothed Belt as a Function of Belt Tension

		R	RH
CODE	width-pitch ratio b_b/P_b	0,33	0,46
	height-pitch ratio h_b/P_b	0,19	0,24

Our research work was aimed at eliminating any chance of slippage.

We succeeded in this by taking the following steps (fig. 47):

- As the camshaft pulley is extremely critical we increased the angle around it from 135° to 215° by designing for the intermediate shaft to be driven by the back oft the belt.
- The little space remaining above the camshaft pulley is blocked by a wedge.
- The entire toothed-belt drive is protected by metal covers.
- We used a wider belt – 25 mm instead of 3/4 in.
- Tooth geometry was changed from R to RH.
- The entire structure of the belt was improved.

As is shown in Fig. 48, these improvements permit transferring 100 % more torque. The type of toothed belt used in VW Diesel engines is made by Pirelli (ISORAN RH) (Fig. 49). Its structure proved to be best suited to our requirements. In addition to the ISORAN R type usually fitted by the automobile industry, this type of belt was developed specifically for heavy-duty purposes.

Most of the strain on the toothed belt is caused by peak torques originating from the fuel injection pump (6 mkp max.).

The ISORAN RH belt differs from the R type belts (Fig. 49) in regard to its profile configuration, material, and structure. Its teeth are 3/8" (9.525 mm) apart, similar to the distance between the teeth of the R-type belts. This enabled us to design belt drives featuring pulleys and axial distances similar to those of spark ignition engines. The teeth of the RH-type belt are stronger and higher. Its structural characteristics are:

- Fibre glass core
- Teeth and belt body of synthetic rubber of great shore hardness
- Double tooth protection by reinforced nylon fabric and a dry, self-lubricating medium on the surface.

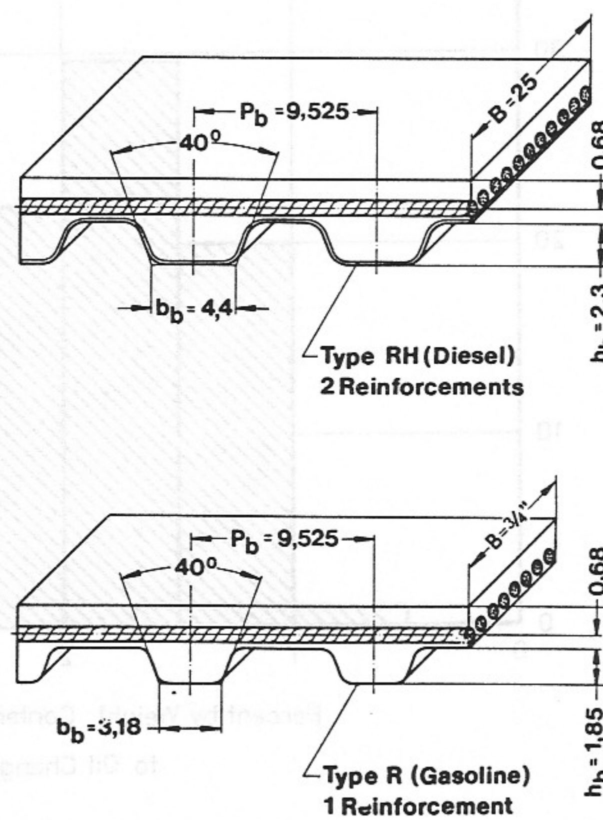


Fig. 49 – Structure of Toothed Belt

LUBRICANTS, COOLANTS

LUBRICATING OIL – When compiling the regulations governing the lubrication of our Diesel engine we had to consider the following manufacturing and servicing requirements:

RUN-IN OIL

As both Diesel and spark ignition engines are manufactured on the same transfer line it was natural to use the same kind of run-in oil for both.

The quality of those lubricants today corresponds more or less to that specified in the former US Military Specification MIL-L-2104 b.

Run-in filter and run-in oil are changed after 1,000 km.

SERVICE OIL

Our maintenance service had the following requests to make:
OIL CHANGE INTERVAL: 7,500 km (3 liters)
OIL FILTER CHANGE INTERVAL: 15,000 km
COMPATIBLE MINIMUM OIL QUALITY REQIREMENTS FOR BOTH DIESEL AND SPARK IGNITION ENGINE.
VISCOSITY SPECIFICATIONS ANALOGOUS TO THOSE OF OUR RANGE OF SPARK IGNITION ENGINES.

Combustion process, oil charge, and thermal piston design largely influence the specifications concerning lubricant performance and viscosity as well as oil and filter change intervals.

The run-in as well as the regular lubricant of our engine had to be highly blended to avoid irremediable coking of the ring grooves.

Carbon gases emanating from the Diesel fuel are quenched by the cold cylinder wall and absorbed by the oil film. If much oil is consumed, most of this mixture of soot and oil is conveyed out of the engine via the combustion chambers and the exhaust gas system. If the oil consumption is low, most of the polluted oil is transported back into the sump.

The amount of soot absorbed by the oil is also largely determined by the kind of combustion process used.

If, for instance, direct injection is used, condensation of carbon gases is generally avoided because there are nearly no cold areas at all within the combustion chamber.

On the other hand, the lubricant of separate-chamber engines, both prechamber and swirl chamber, is more highly polluted because there are cooler areas in these engines where soot may be precipitated. For this reason, the detergent capacity of the oil blend must be higher.

Number of Oil Samples										
	0	1,0	20,8	31,2	22,9	10,4	6,2	6,2	1,0	Perc.
	0	1,0	21,8	53,	75,	86,3	92,5	98,7	100	% cum.

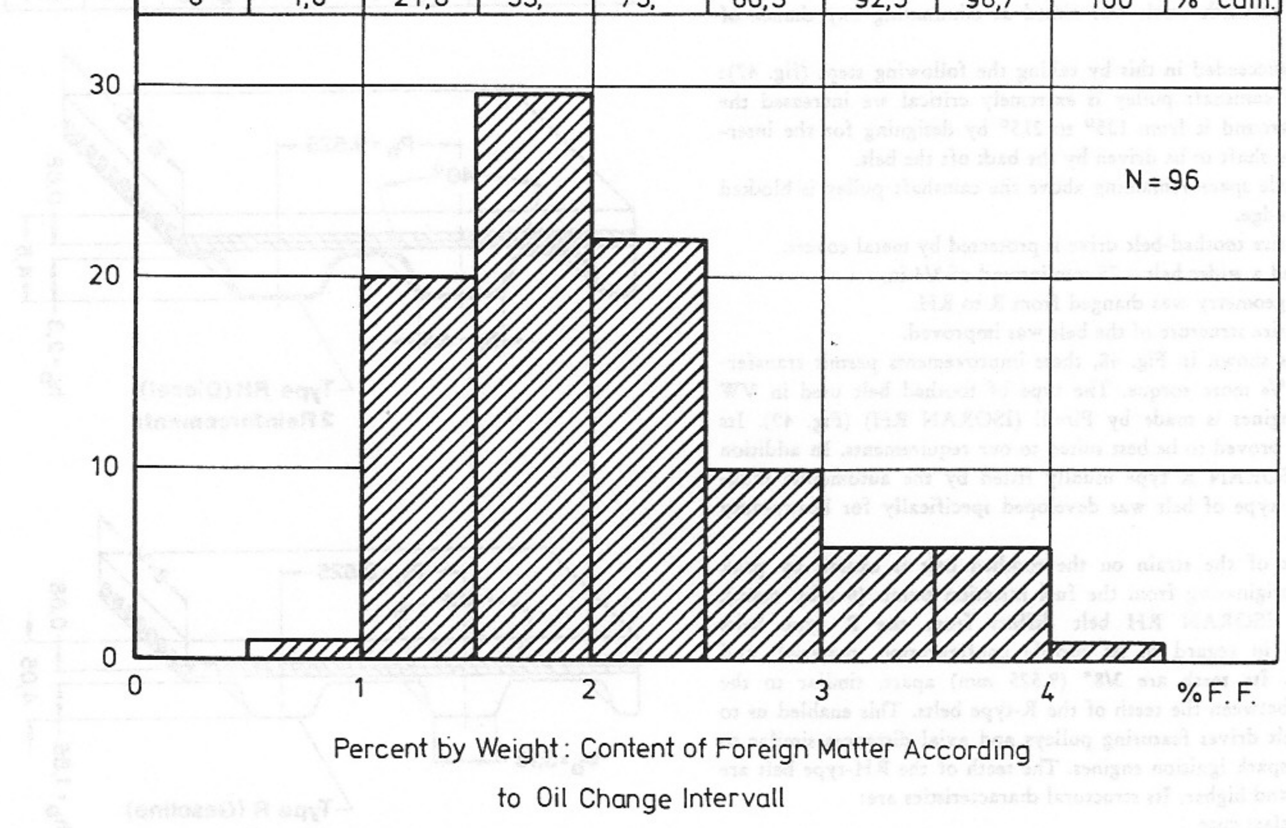


Fig. 50 – Oil Pollution in Vehicles Operated under ,Road Conditions'

We conducted 3 engine test runs to determine the oil change intervals for the VW Diesel engine. We used 3 kinds of lubricant. All of which involved the same oil base but·different additives in accordance with their respective API classes.

Lubricant data:

I	SAE 30	API-CC	MIL-L-2104 b
II	SAE 30	API-SE/CC	MIL-L-46152
III	SAE 30	API-SE/CD	MIL-L-2104 C

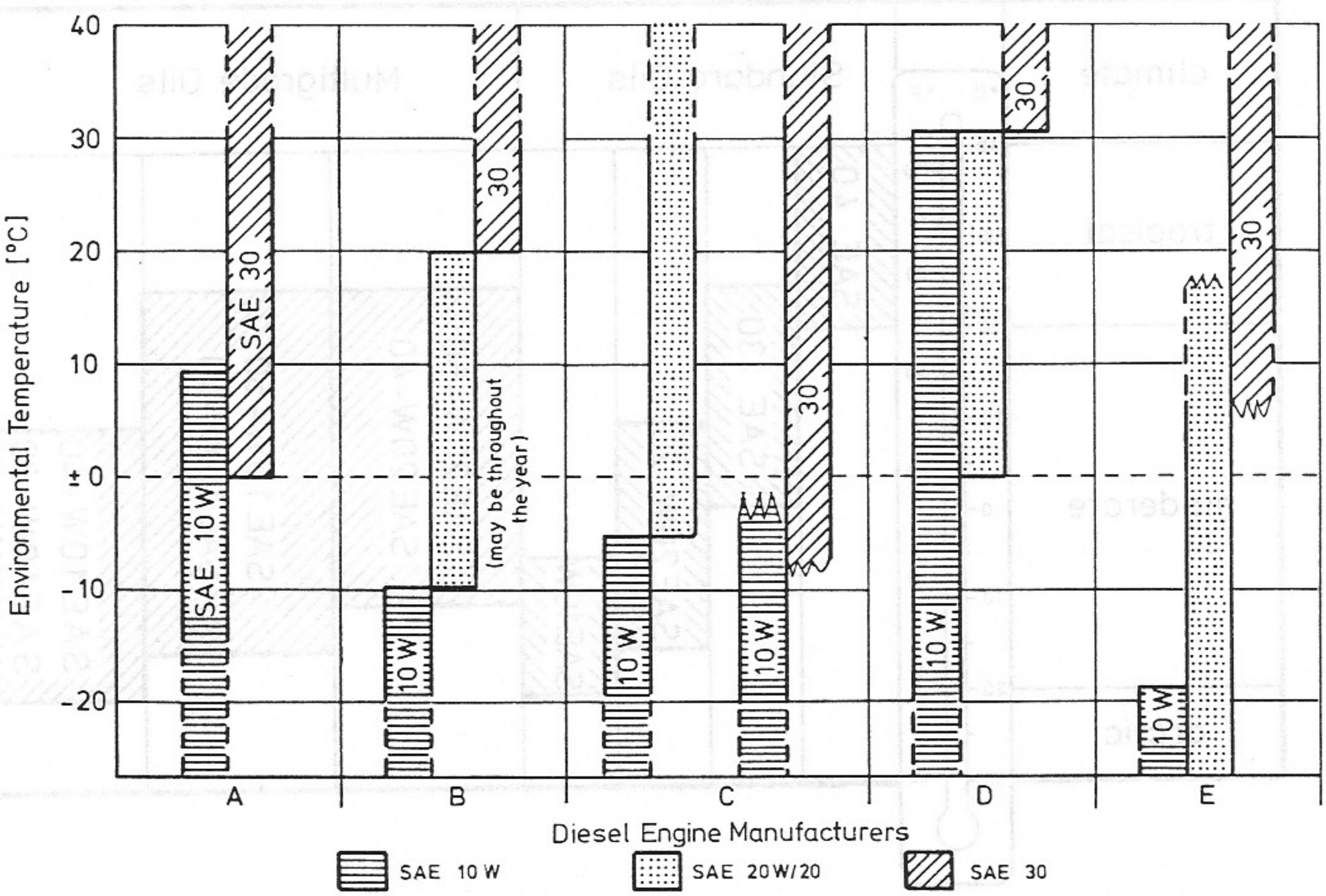
– 10 hours' running in under variable loads; oil change; filter change
– 50 hours' running under full load; sump temperature 120° C.

In the course of these tests we found the probable oil change interval (minimum oil quality API-CC) to be 7,500 km. This assumption was confirmed by the results of a 3000-vehicle fleet test (Fig. 50). Again, we used 3 kinds of lubricant staggered according to API criteria.

One of the major results of this fleet test is the extraordinary lack of general oil pollution, the maximum being 1.5 to 2%/7,500 km, which means that especially the engine's cold start behavior will not be significantly affected.

After API-SE/CC, the mean viscosity increase – computed from 68 samples – of a SAE 20 w–50 lubricant amounted to 17.2% (temperature 210°).

During the entire fleet test we found the TBN reserve to be sufficient; iron abrasion, which, after all, is the criterion of wear, remained below 80 ppm.



Viscosity Specifications for Diesel Engine Manufacturers in Germany for High Speed Diesel Engines

Fig. 51 – Viscosities Recommended from German Diesel Manufactures for Passenger Car Application

Fig.s 51 und 52 show the specifications finally decided upon after our tests had produced excellent results in regards to soot precipitation and oil viscosity change. In these specifications, we deviate from the general run of European low-capacity Diesel engine manufacturers.

This shows that under the climatic conditions prevailing in Europe and the US a multigrade lubricant based on a SAE W 15 oil will guarantee satisfactory cold start behavior.

The performance of the engine locates it somewhere in the lower 1 H CATERPILLAR range (upper ring groove temperature 240° C, sump temperature 130° C under full load). We are therefore in a position, to satisfy one of the major demands of our service organisation, namely, to standardize workshop regulations for Diesel and spark ignition engines as far as possible.

- The same lubricant is used for both Diesel and spark ignition engines (API-SE/CC = MIL-L-46 152, whose requirements are met by commercial SAE qualities 15w–40, 15w–50, and 20w–50).
- Low oil pollution in actual use enabled us to set the oil change interval at 7,500 km and the oil filter change interval at 15,000 km, which is analogous to spark ignition engines.
- The strain imposed on the oil through viscosity increase due to soot is relatively low, and the vehicle's cold-start properties have been adapted to that fact. For this reason, we formulated oil viscosity specifications (temperature limits given in the owner's manuals) that are the same as those for the spark ignition engine.

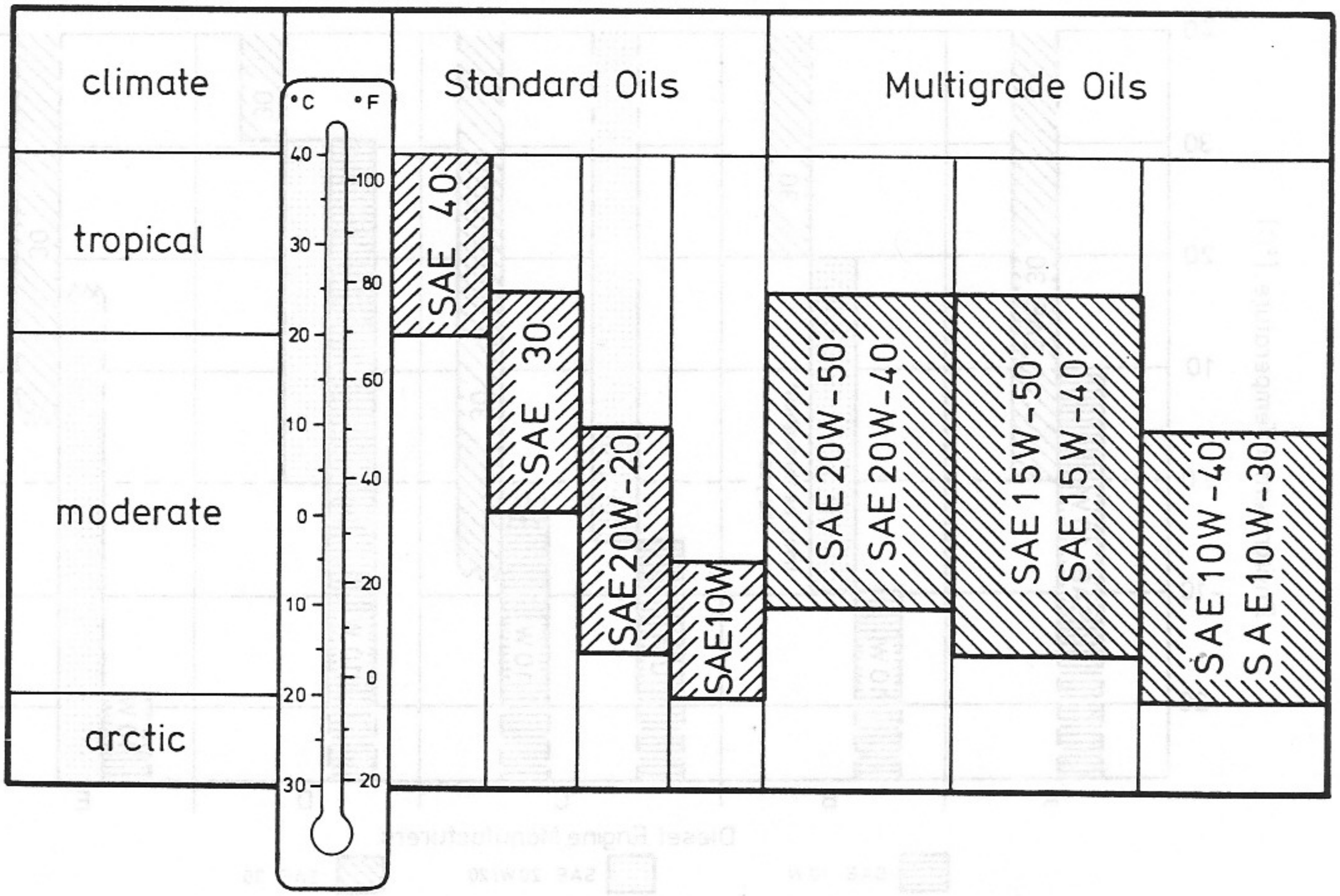


Fig. 52 – Viscosities Recommended for the VW Diesel Engine

COOLANT

Our tests have shown that the coolants commonly used in spark ignition engines may be used in Diesel engines as well. We were unable to find any negative changes in the coolant used after the engine had run for 100,000 km. Both pH-index and alkalinity still conformed to our requirements even after this mileage.

Coolant samples drawn from several vehicles of different mileages were subjected to anti-corrosion tests according to ASTM D 1384-70. The results of these tests show that even after high mileage the anti-corrosion effect still meets our requirements. Therefore, we have no reservations about using radiators made of aluminum.

To establish whether any changes in the coolants had occurred, coolant samples were drawn from several Diesel-powered vehicles of varying mileage. These samples were tested for coolant additive concentration (density test), pH-index, and alkalinity, the latter being measured according to ASTM D 1121 for 100 ml of a 10% coolant solution.

Quite independent of mileage, the coolant additive concentration level remained at between 40 and 50% by volume.

While decreasing slightly after relatively low mileage the pH-index in all vehicles remained somewhere in this region for the entire duration of the test.

Many metals, aluminum among them, are amphoteric, which means that they will corrode and dissolve in an acid as well as in an alkaline environment (low and high pH-indexes). Every corrosion inhibitor will work only within a limited pH index range. Once these relatively narrow limits are transgressed, a corrosion inhibitor may become entirely ineffective and may even help to increase corrosion.

The results of our modified ASTM D 1384-70 tests show that the anti-corrosion properties of the coolant additives are fully up to our requirements after a mileage of 75,000 km.

The corrosion rate of all metals used in the ASTM test shows a mere 10% increase compared to a newly charged radiator, the only exception being lead, which showed a 30% increase in its corrosion rate after 75,000 im. (See Fig. 53).

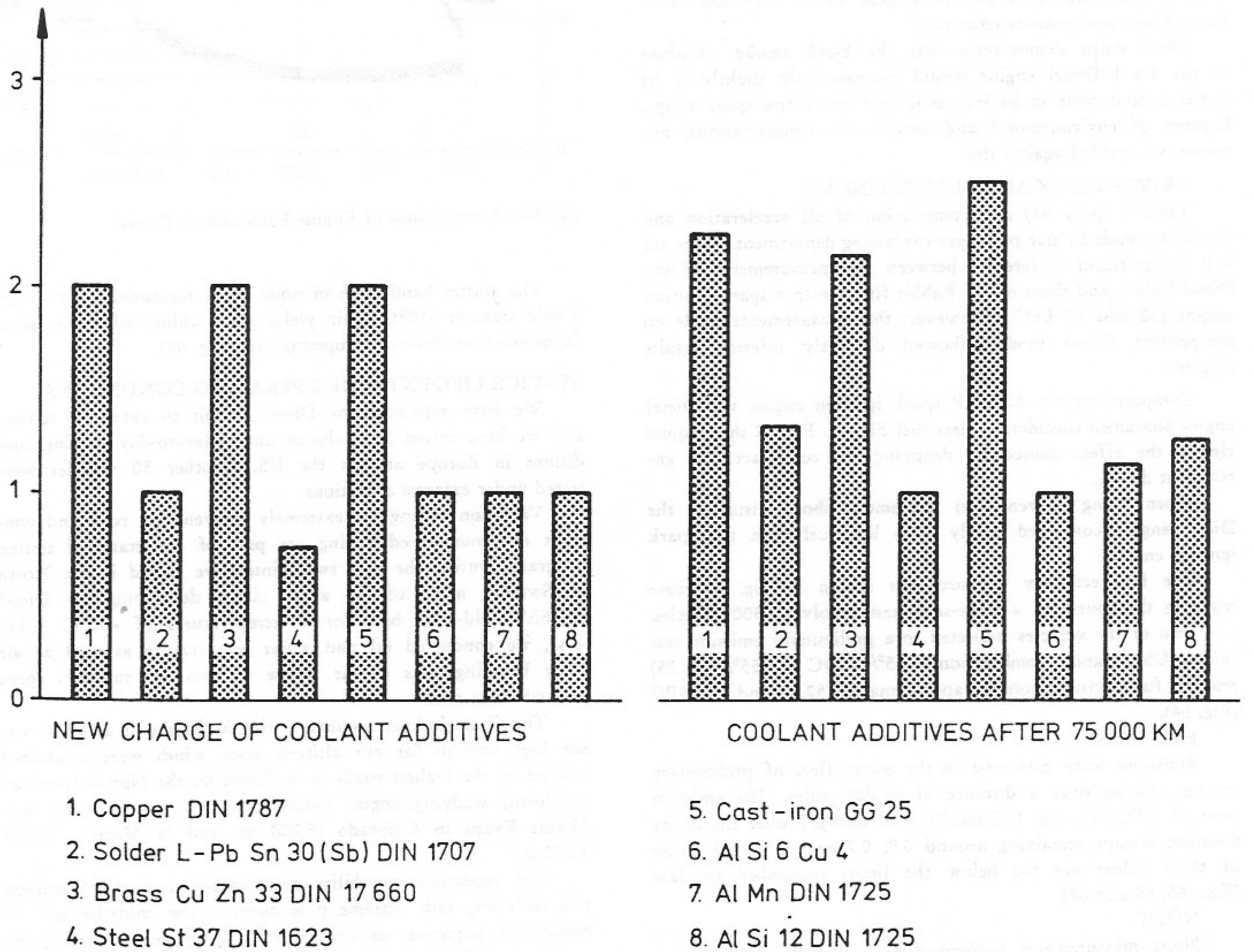


Fig. 53 – Changes in Coolant Additives after 75.000 km

EXPERIMENTAL FINDINGS

ENGINE MAPS

ENGINE Delivery CURVES – Fig. 54 shows the scatter bandwidth of delivery curves taken from 50 1.5 liter Diesel production engines. The mean values resulting from these curves are as follows:

Maximum power output/engine speed
36.6 kW / 49.8 BHP / 5,000 rpm

Blackening rate at maximum power output
3.4 Bosch numbers

Maximum torque/engine speed
84 Nm (8.4 mkp)/3,000 rpm

Minimum fuel consumption/engine speed
280 g/kWh (206 g/BHPh)/2,600 rpm

Fuel consumption at maximum power output
330 g/kWh (243 g/BHPh).

ENGINE MAPS – The fuel consumption engine map is shown in Fig. 55. At engine speeds of up to 2,500 rpm and at three-quarter load the minimum specific rate of fuel consumption attained was 250 g/kWh (185 g/BHPh).

Fig. 56 shows the soot emissions pertaining to this map. These emissions vary with the load and the engine speed; they are lowest at partial load.

Fig.s 57, 58 and 59 show the HC, CO, and NO maps. The tractive resistance curve in fourth gear shows very low emissions of soot and noxious substances.

These maps demonstrate that the black smoke emissions of the 1.5 l Diesel engine would increase only slightly if its power output were to be increased over the entire speed range. Because of environmental and service life considerations, however, we decided against this.

DRIVEABILITY AND FUEL ECONOMY

Table 1 (page 41) is a compilation of all acceleration and flexibility made by our passenger car testing department. There are only insignificant differences between the measurements of the Diesel Rabbit and those of the Rabbit fitted with a spark ignition engine (50 and 55 BHP). However, the measurements made on competitive Diesel models showed distinctly inferior results (Fig. 60).

Compared to the 50 BHP spark ignition engine the Diesel engine consumed considerably less fuel Fig. 61. Fig. 62 shows quite clearly the effect caused by dropping the cold-start fuel enrichment device.

When being driven over extremely short distances the Diesel engine consumed nearly 50% less fuel than the spark ignition engine.

The fuel economy measurements shown in Fig. 63 were made in the course of a large-scale test involving 300 vehicles.

Two of the vehicles subjected to a preliminary emission test (HDC, US 75, and a combination of 45% HDC and 55% US 75) reached fuel consumptions of approximately 52.38 and 45 MPG (Fig. 64).

EMISSIONS

Emissions were measured in the entire fleet of preliminary testing vehicles over a distance of 50,000 miles. The emission rates of NO, CO, and HC hardly ever changed over the entire distance, always remaining around 0.9, 0.7 and 0.3 g/mi; some of these values are far below the limits prescribed by law (Fig.s 65, 66 and 67).

NOISE

Noise measurements performed on 2 vehicles have shown that the ISO pass-by noise of the Diesel Rabbit is approximately 80 dB (A). Its interior noise level (driving as well as coasting) measured over the entire range of partial load is either well below or at the lower limit of the scatter bands of the noise levels measured in competing Diesel vehicles (Fig. 68).

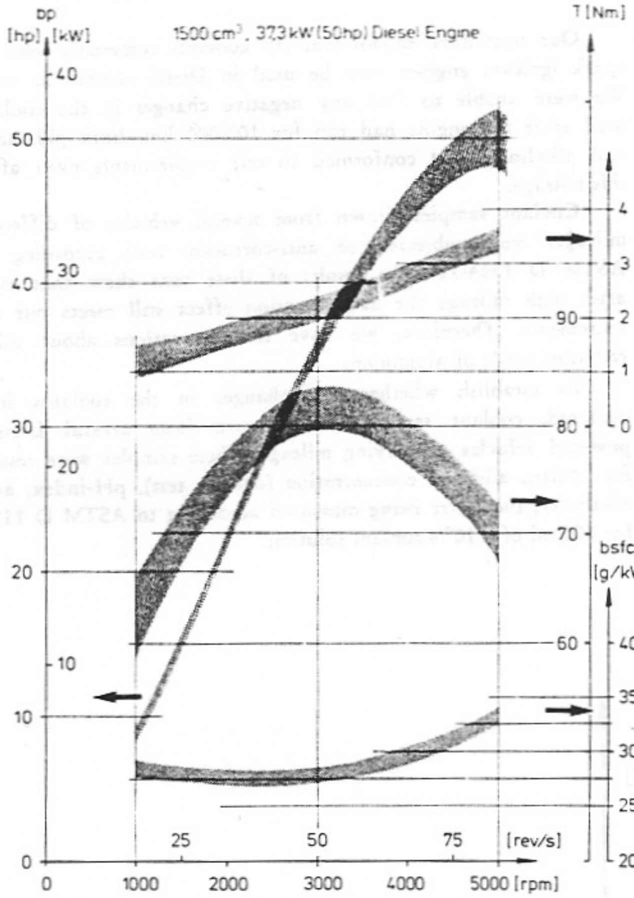


Fig. 54 – Scatterbands of Engine Performance Curves

The scatter bandwidth of noise levels measured at idle after a cold start at −10°C again yields mean values which are more favorable than those of competing cars (Fig. 69).

SERVICE LIFE/EXTREME OPERATING CONDITIONS

We have subjected the Diesel Rabbit to extensive testing, and we have tested 300 vehicles under day-to-day driving conditions in Europe and in the US. Another 50 vehicles were tested under extreme conditions.

Vibration testing on extremely uneven test runs and constant maximum-speed testing are part of our standard testing program. During the past two winters we stayed in the North of Sweden, north of the arctic circle, developing the Diesel Rabbit's cold-start behavior at temperatures of −30°C. Likewise, we conducted oil and water temperature as well as air filter blocking tests in the course of two hot summers spent in the Sahara desert.

The Grossglockner mountain with its height of 2,500 m was not high enough for our altitude tests, which were conducted instead on the highest roads to be found on the North American continent, studying engine behavior at extreme altitudes near Mount Evans in Colorado (5,000 m) and in Mexico (2,000–3,200 m).

Our repeatible durability tests extended over 15 million km, including such extreme procedures as the 'midwife test' or continuous sequences of cold starts, which means starting the engine at −20° C, waiting until it runs smoothly, switching off and cooling it down to −20° C again without ever allowing it to warm up.

Endurance bench tests, run under full load during 83% of the time, showed that the service life of the Rabbit Diesel engine is twice as long as that of the spark ignition engine.

GOLF Diesel

Acceleration

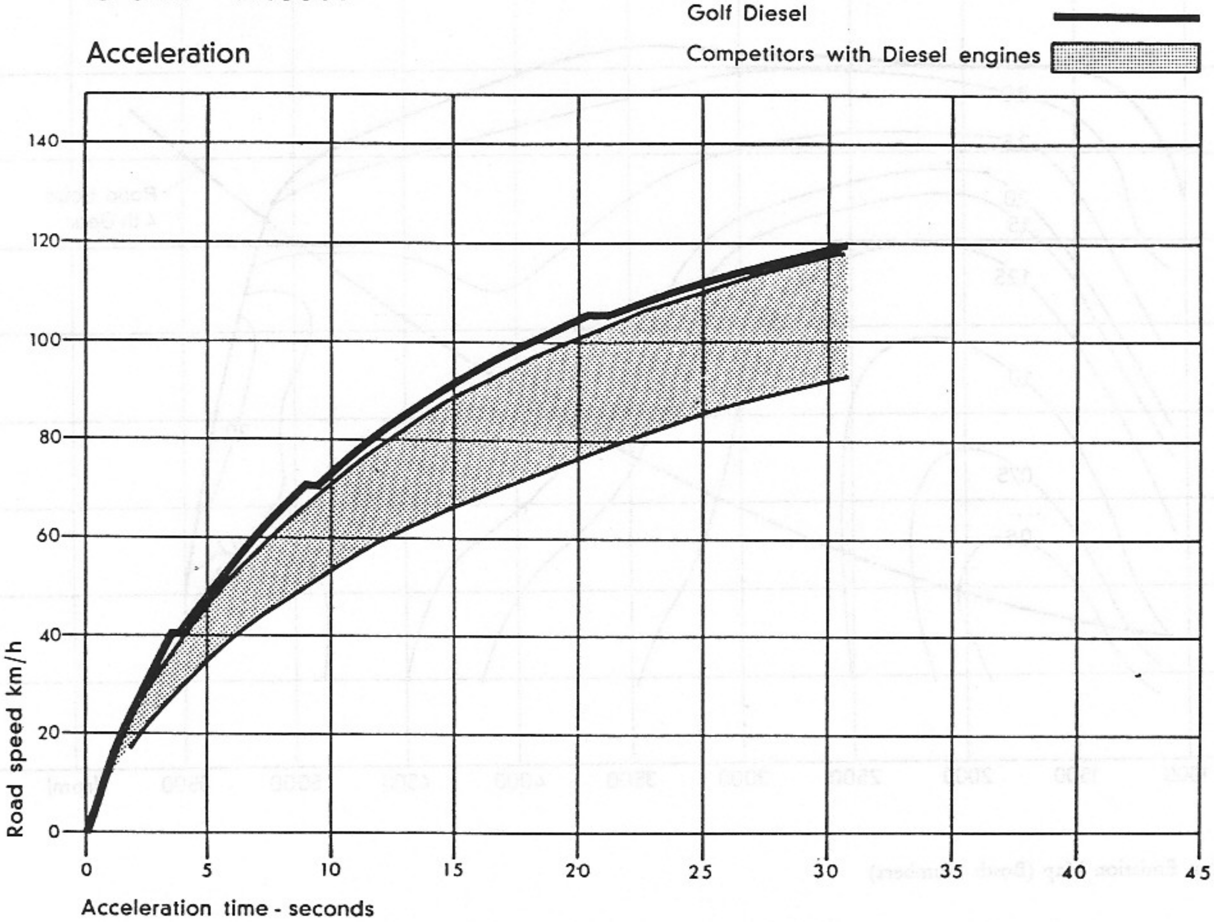


Fig. 60 – Acceleration

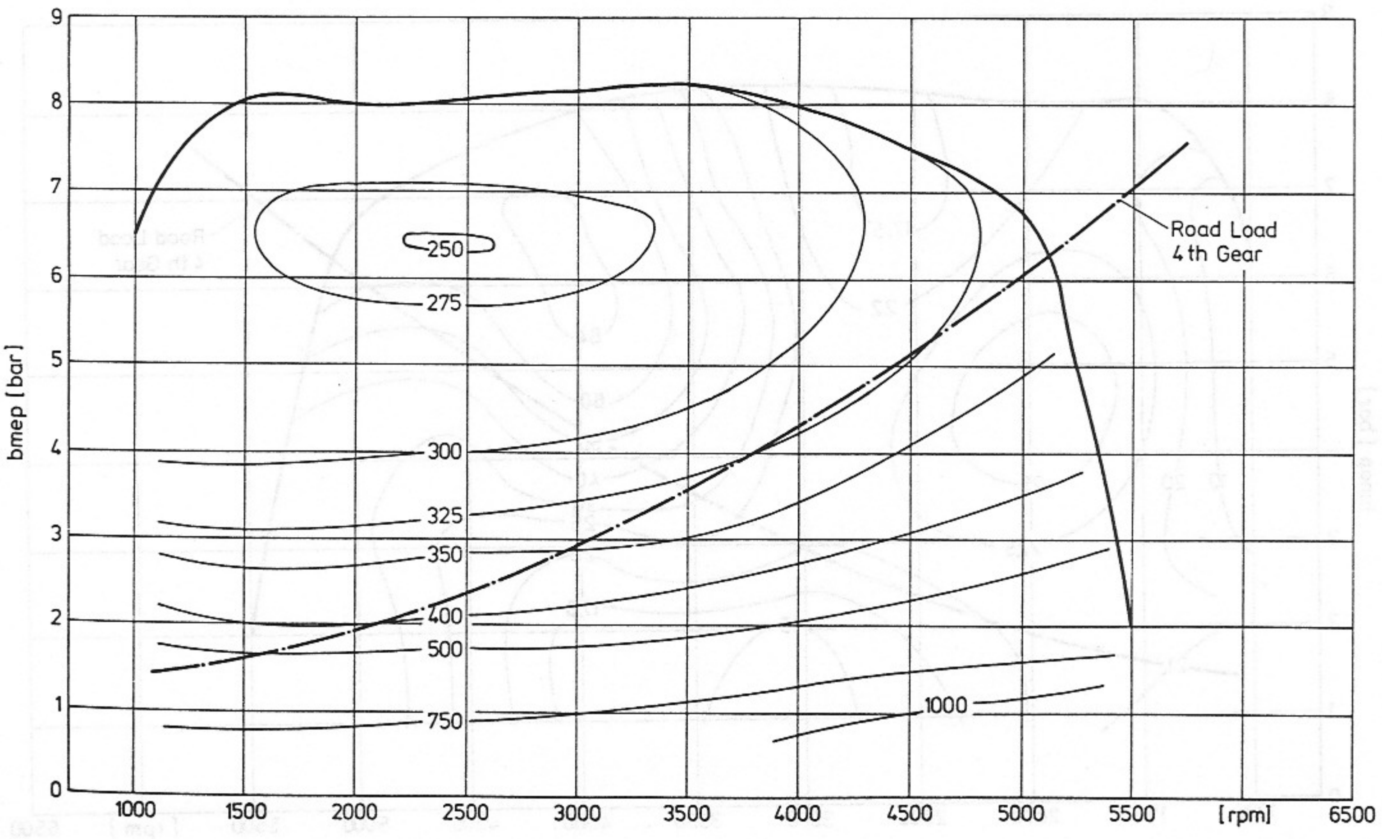


Fig. 55 – Fuel Economy Map (G/KWH)

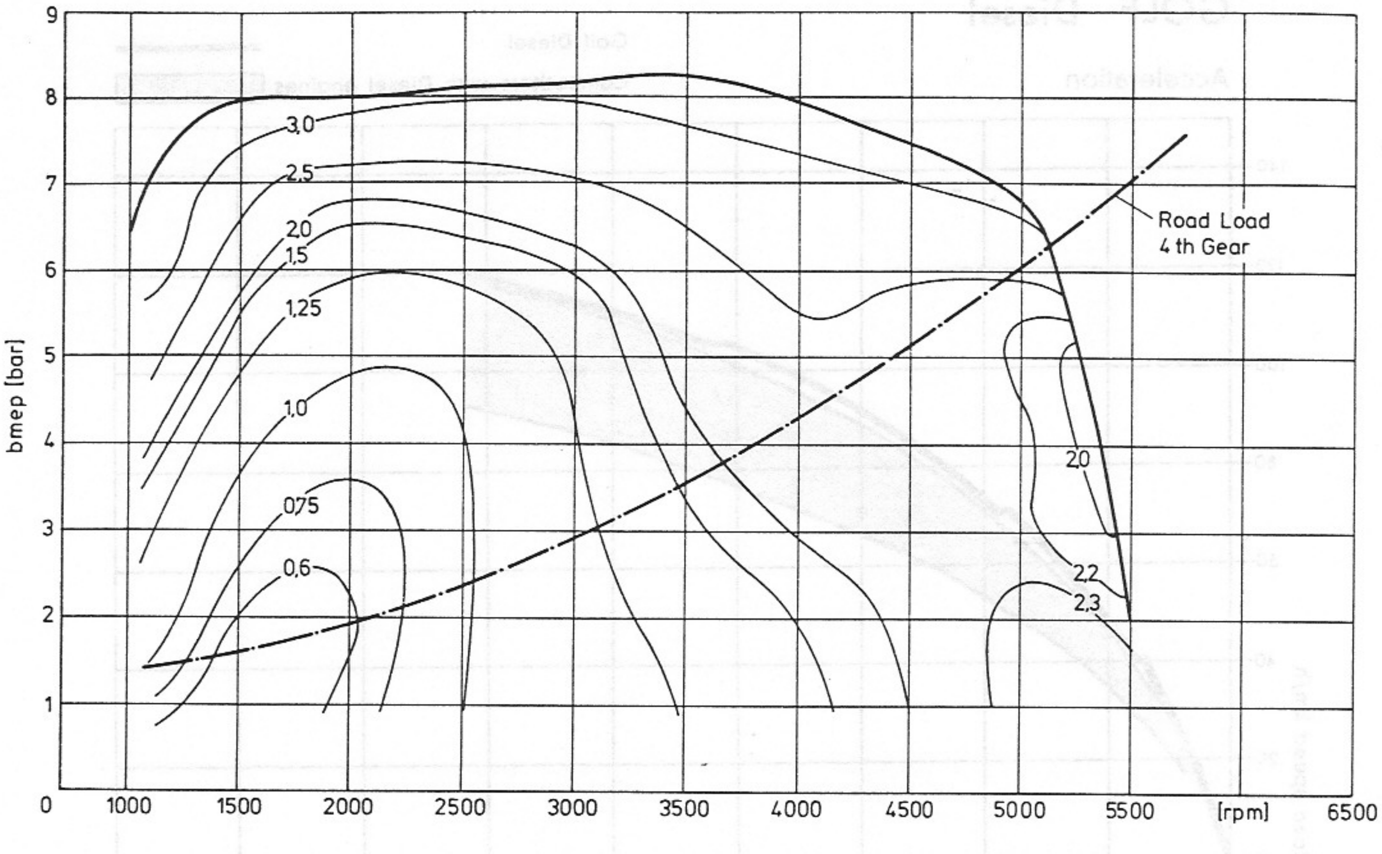


Fig. 56 – Soot Emission Map (Bosch Numbers)

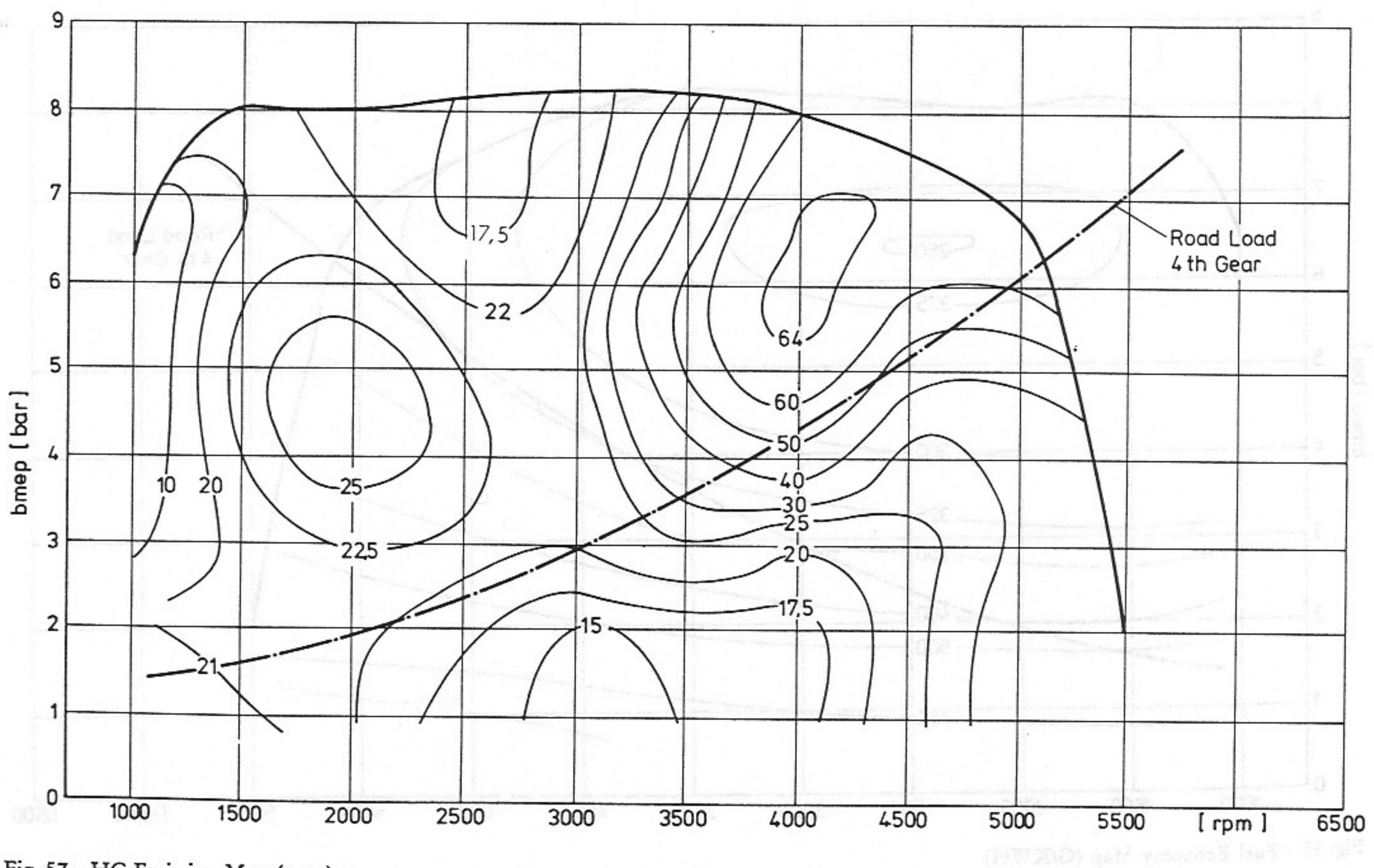


Fig. 57 – HC Emission Map (ppm)

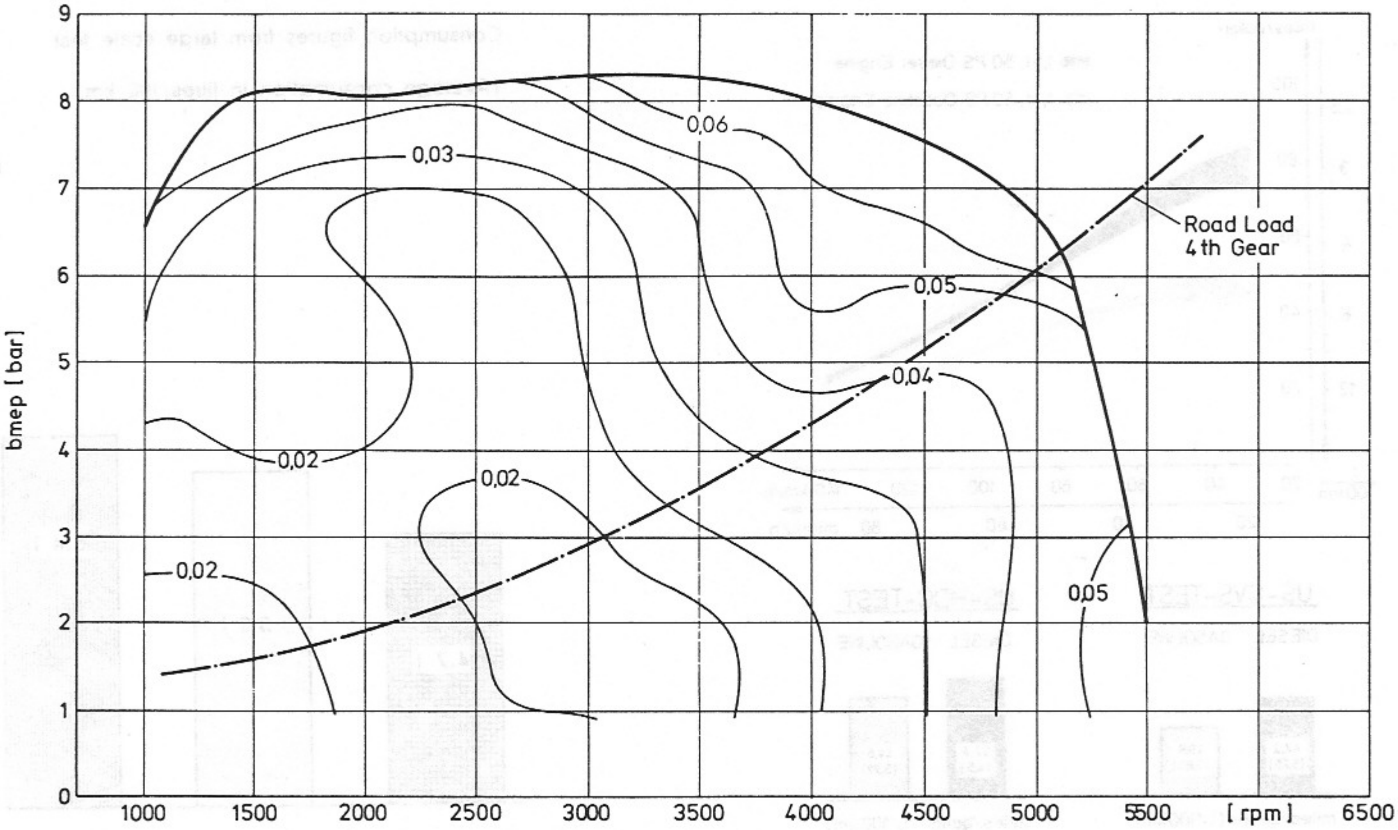


Fig. 58 – CO Emission Map (Vol%)

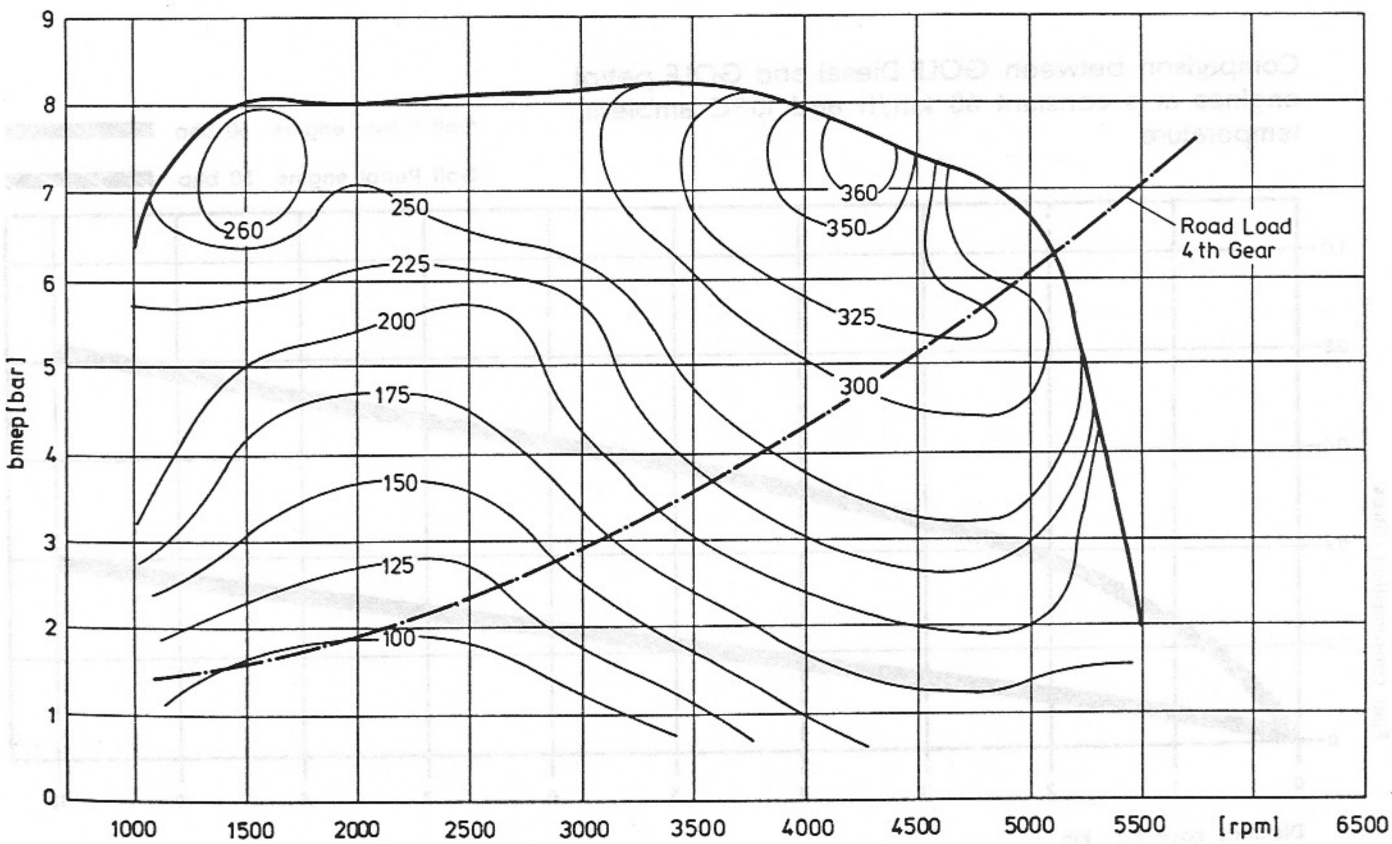


Fig. 59 – NO Emission Map (ppm)

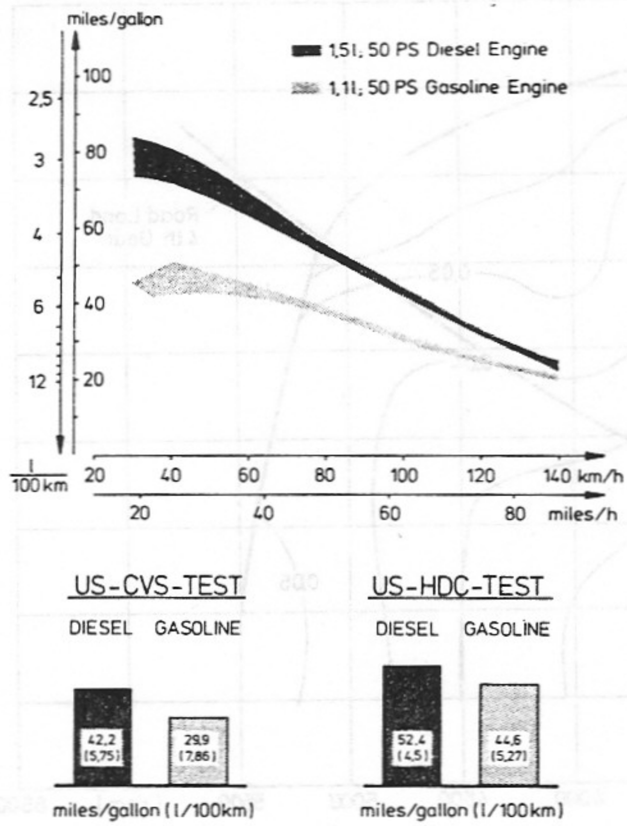


Fig. 61 – Consumption under Partial Load and During EPA Cycle

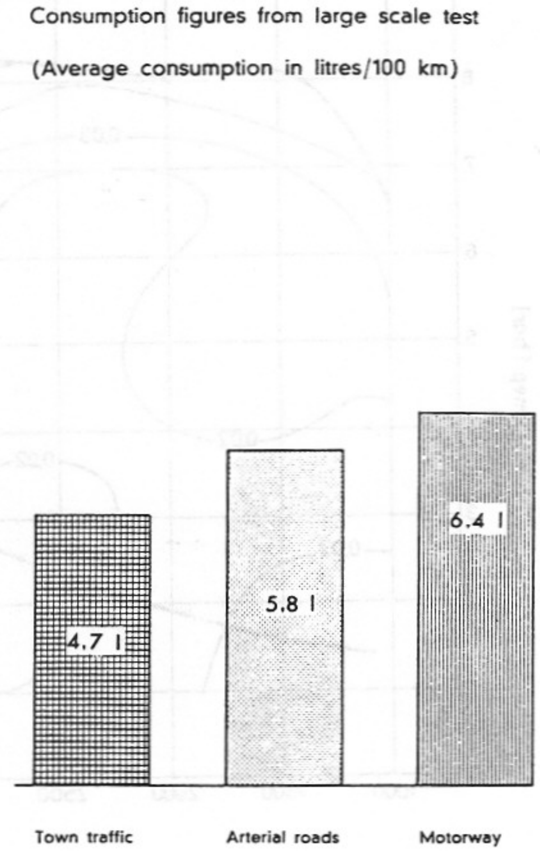


Fig. 63 – Average Fuel Consumption of Fleet Test in Germany

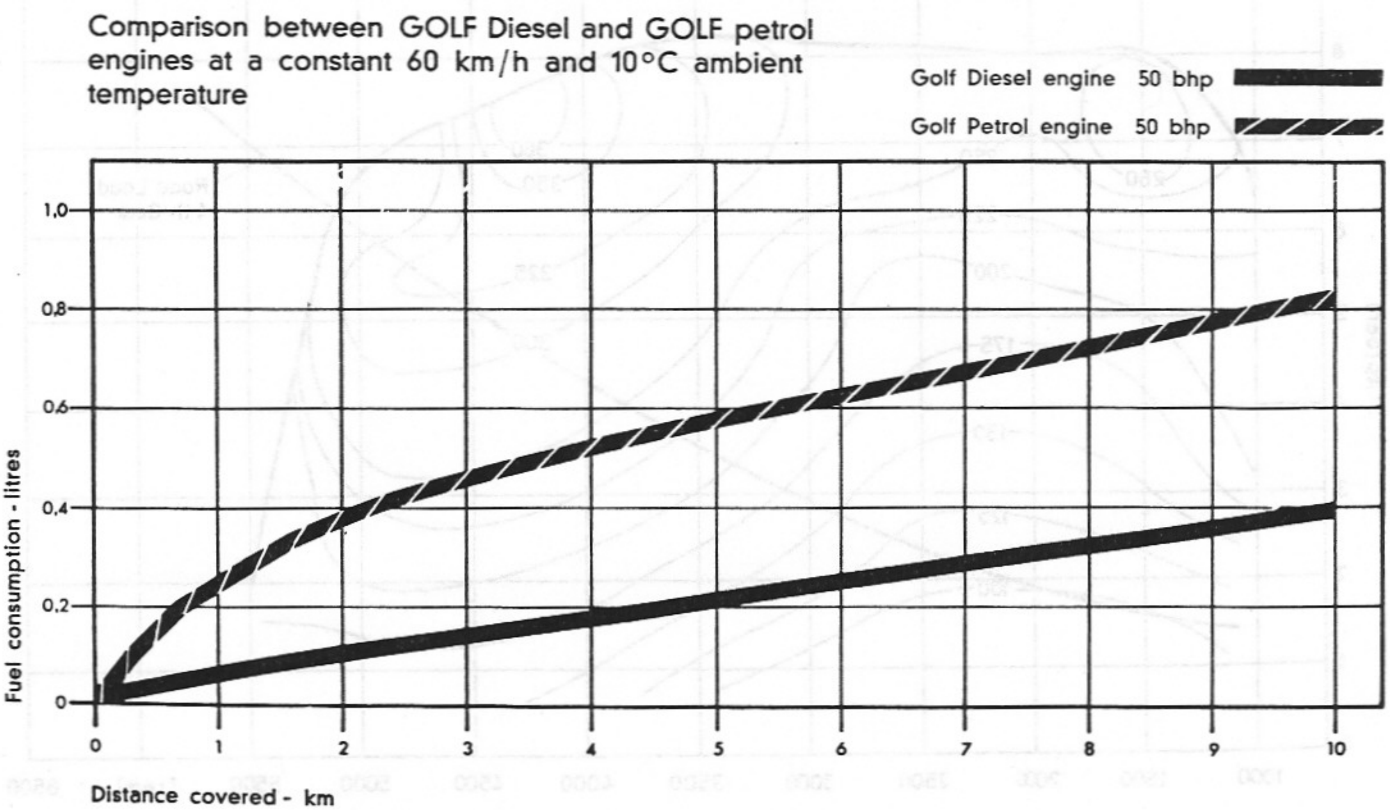


Fig. 62 – Fuel Consumption After Cold Starting

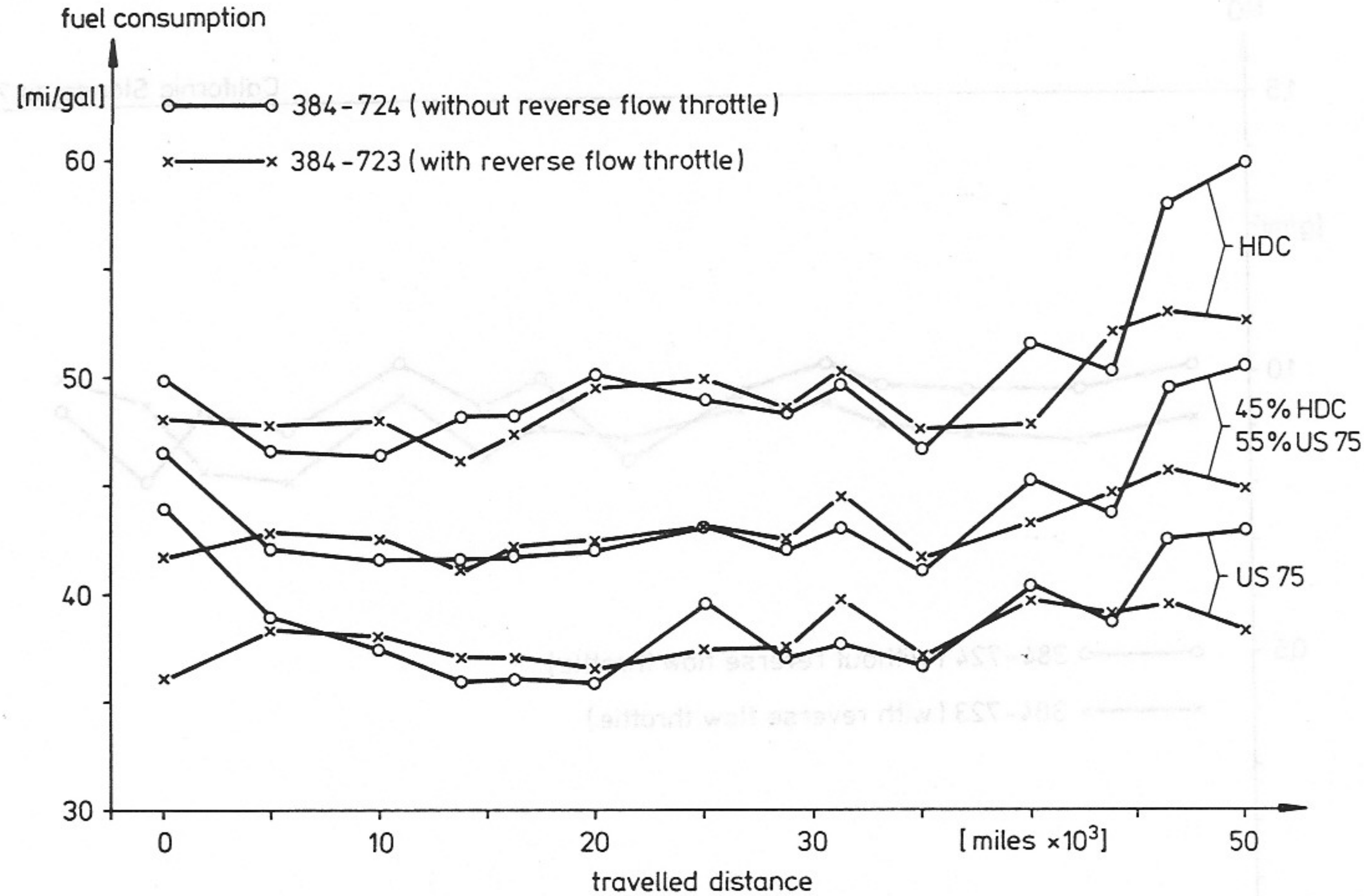


Fig. 64 – Average Fuel Consumption of the Prototype Emission Test Fleet

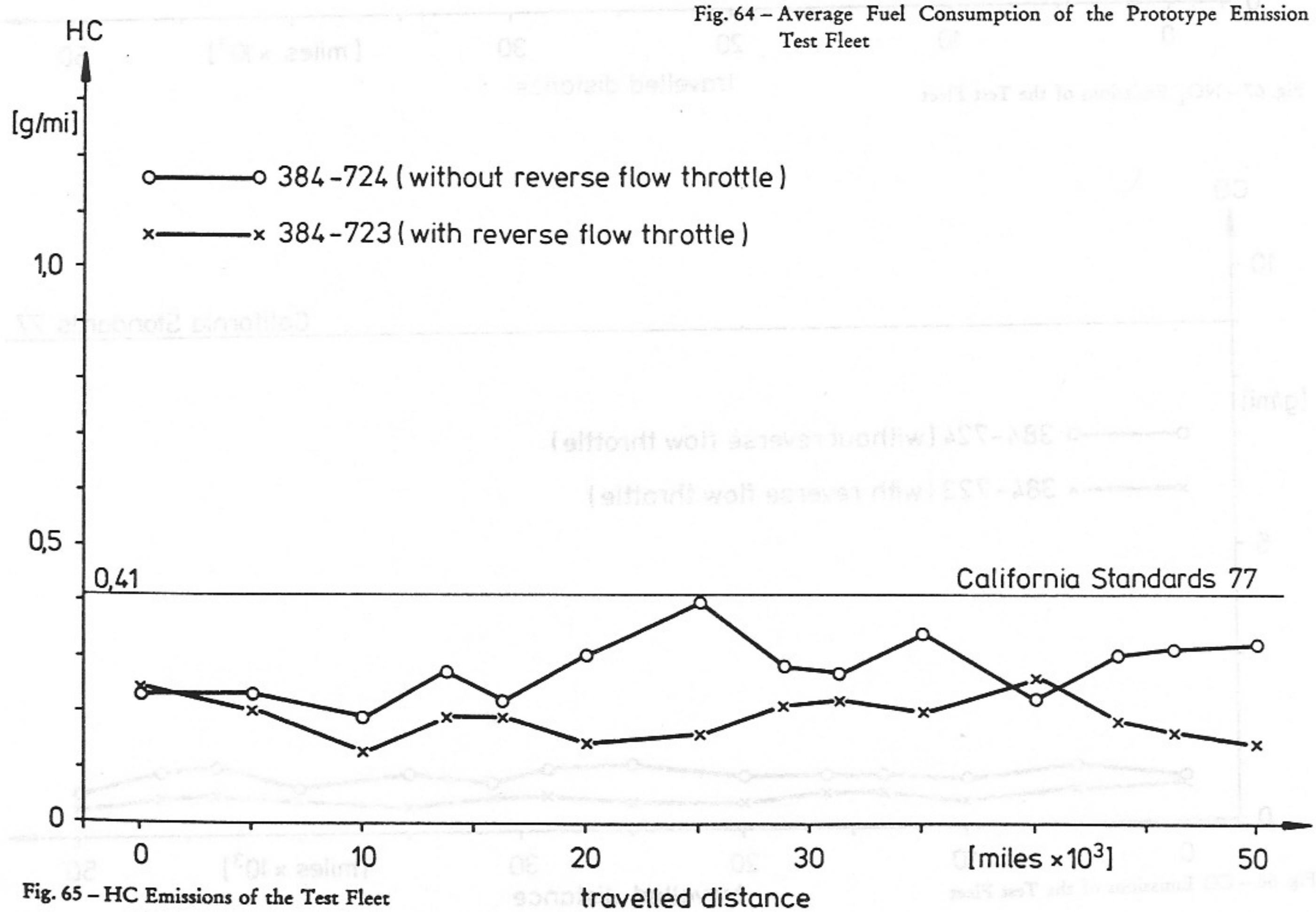


Fig. 65 – HC Emissions of the Test Fleet

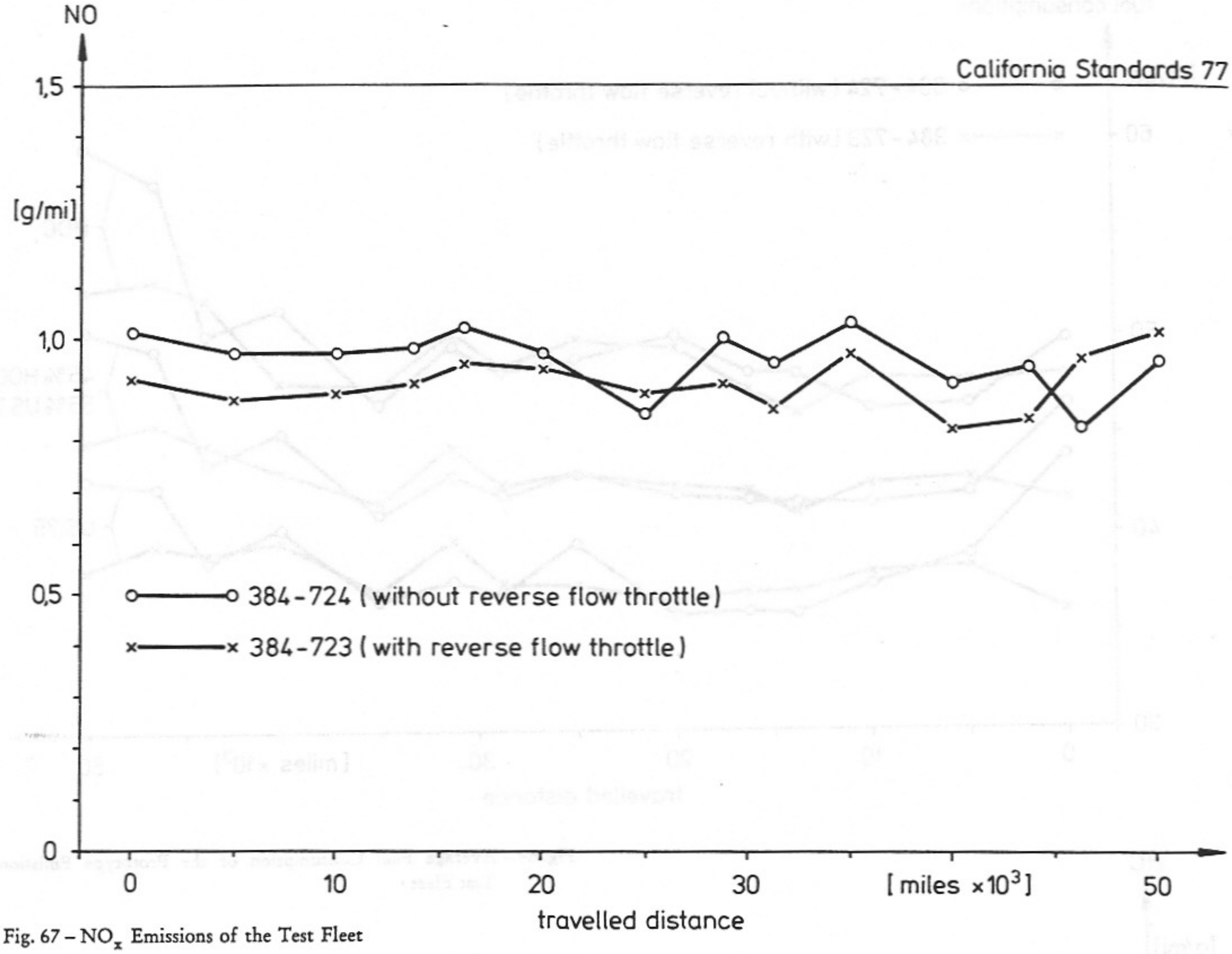


Fig. 67 – NO_x Emissions of the Test Fleet

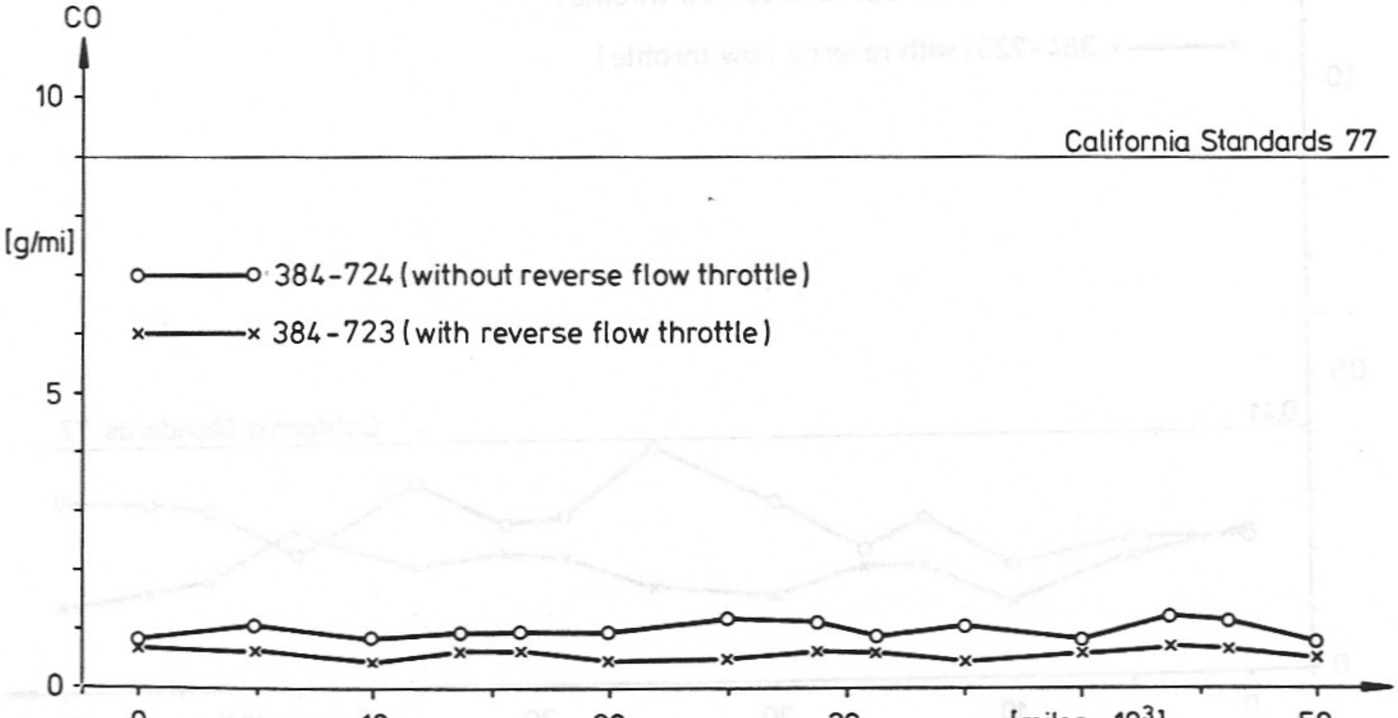


Fig. 66 – CO Emissions of the Test Fleet

Acceleration:	
0–100 km/h	18,2 sek
0– 80 km/h	11,5 sek
0– 50 mi/h	
Flexibility at 4. Gear:	
40– 80 km/h	16,8 sek
40–100 km/h	26,8 sek
Max. Speed:	
V_{max}	140,5 km/h
	87,8 mph

Table 1 Performance Data

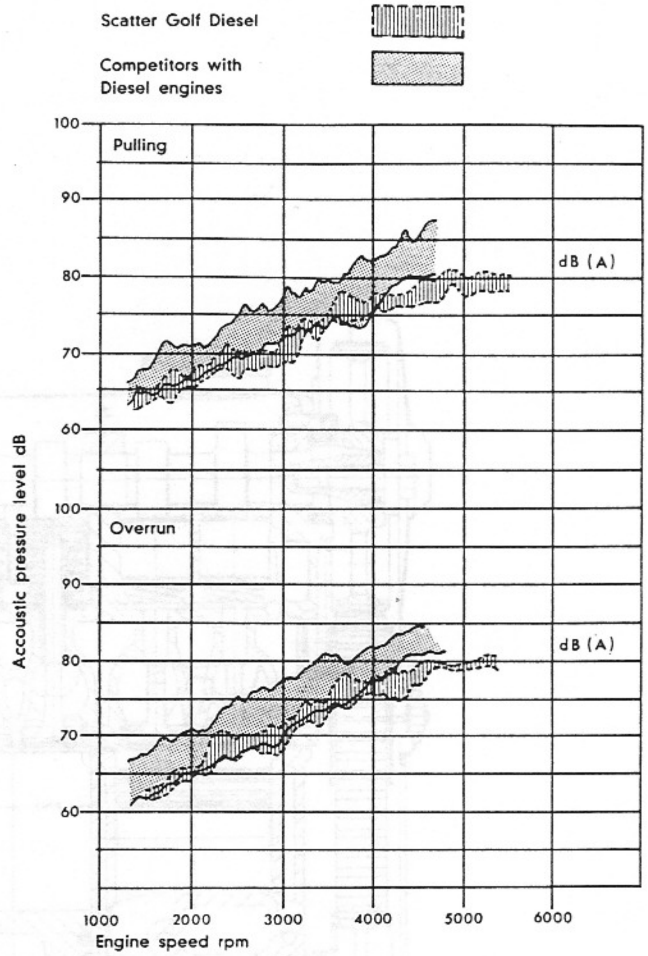


Fig. 68 – Interior Noise Level Measured at Partical Load

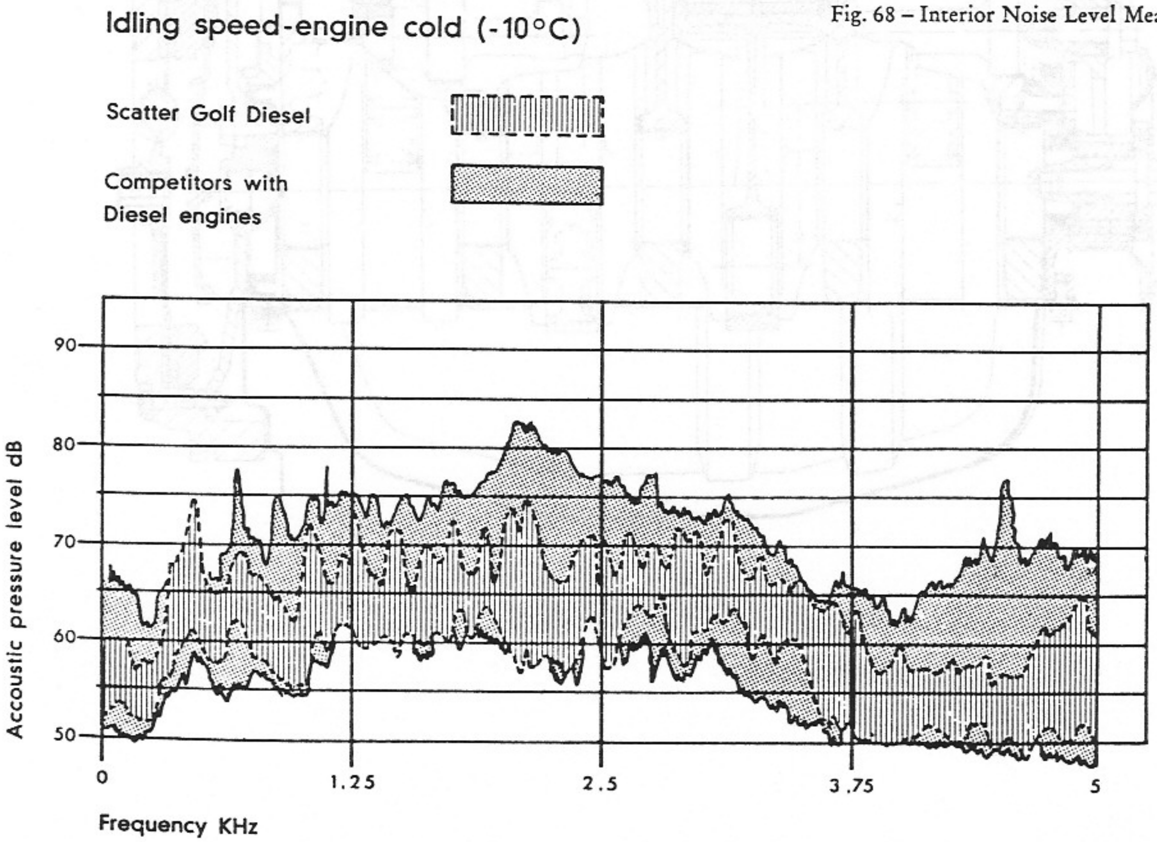


Fig. 69 – Exterior Noise Level Measured After Cold Starting

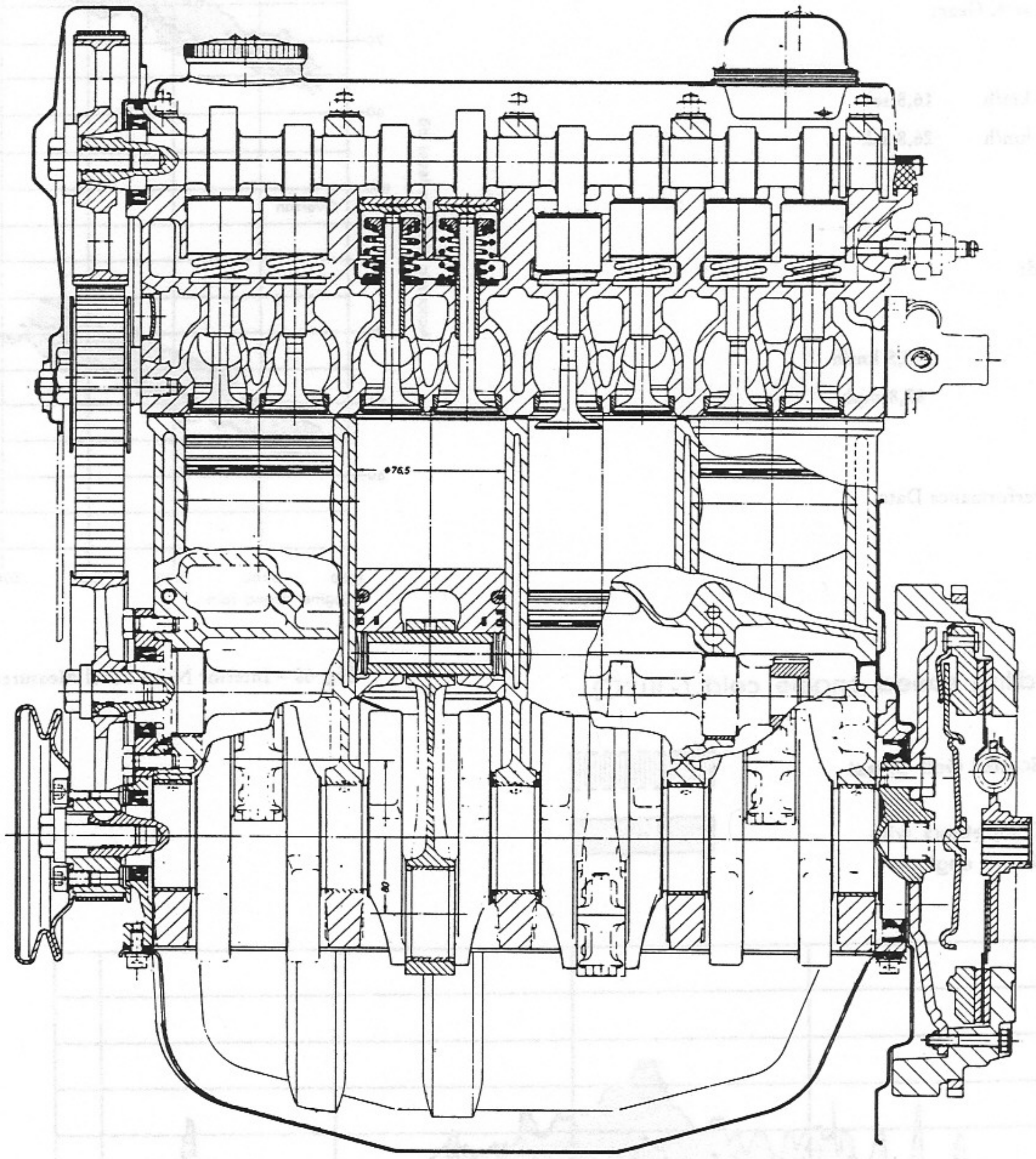
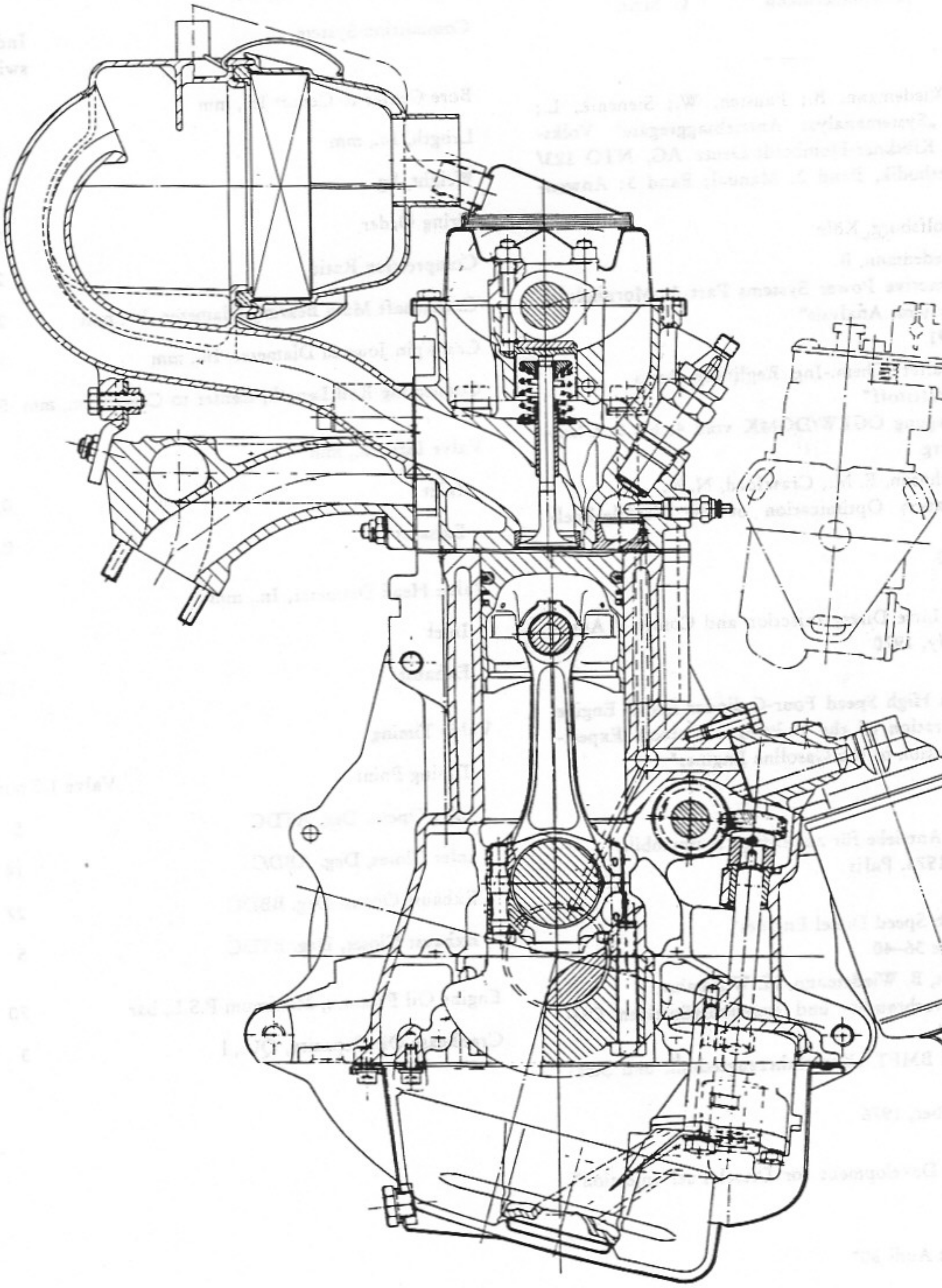


Fig. 70 – Main Cross-Sectional Views of the VW Diesel



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P. Hofbauer
P. Deja
W. Kiegeland
W. Kurpiers
C. Schwarz
B. Wiedemann
M. Willmann
K. Brockmüller
E. Behrend
W. Heimermann
R. Strauß
K. Zimmermann
K. Sator
K. Barnert
D. Gümmer
I. Löhner
G. Stein

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APPENDIX
GENERAL SPECIFICATIONS

Bore, In., mm	3.012	76,5
Stroke, In., mm	3.149	80
Ratio, Stroke: Bore		1.05:1
Displacement, Cu. In., cm³	90	1471
Combustion System	Indirect injection swirl Chamber	
Bore Center to Center In., mm	3.46	88
Length, In., mm	20.2	512
Weight, kg	120	
Firing Order	1–3–4–2	
Compression Ratio	23:1	
Crankshaft Main Bearing Diameter, In., mm	2.1	54
Crankpin Journal Diameter, In., mm	1.8	46
Connecting Rod Length, Center to Center, In., mm	5.35	136
Valve Lift, In., mm		
Inlet	0.32	8.0
Exhaust	0.35	9.0
Valve Head Diameter, In., mm		
Inlet	1.34	34
Exhaust	1.22	31
Valve Timing		
Timing Point	Valve 1.0 mm Off Seat	
Inlet Opens, Deg. ATDC	5	
Inlet Closes, Deg. ABDC	14	
Exhaust Opens, Deg. BBDC	27	
Exhaust Closes, Deg. BTDC	5	
Engine Oil Pressure, Maximum P.S.I., bar	70	4.8
Crankcase Oil Capacity, QU., l	3	3.5