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HISTORICAL BACKGROUND

IN 1936, DAIMLER-BENZ INTRODUCED the first Diesel passenger car model 260 D with a 4-cylinder 2.6 liter engine. Today's success of the Mercedes-Benz diesel cars was founded in 1948, when the model 170 D was put into production as a fuel-efficient alternative to the corresponding gasoline-powered vehicle. The next important step was the modern engine type OM 621 with overhead camshaft in the 190 D, in 1958. This engine was the basis for today's Diesel engine production program (1, 2, 3)[+]. Its last stage of development is represented by the OM 616 with 2.4 liter displacement and a bore of 91 mm which was obtained by means of the well-proven patented cooling arrangement between the cylinders.

The 5-cylinder engine type OM 617 introduced in the 300 D in 1974 was a further milestone in Daimler-Benz Diesel passenger car development: It was the first engine with 5 cylinders installed in a passenger car. Its design was based upon the 4-cylinder engines featuring identical bore and stroke as the 2.4 liter engine. At the time, Daimler-Benz decided upon the 5-cylinder engine because this - for the given cylinder dimensions - represented the best compromise in terms of weight, bulk and production cost, as compared to a 6-cylinder engine, for the vehicle in question. In addition, careful engine balancing and engine suspension refinements together with other measures resulted in a vibrational behavior very close to a 6-cylinder engine (4).

[+]Numbers in parantheses designate References at end of paper.

ABSTRACT

The engineering target was a suitable Diesel engine for the largest available luxury sedan. After consideration of various alternatives including the design of a completely new 6-cylinder engine it was decided to equip the existing naturally aspirated 5-cylinder engine presently installed in the smaller sedan and the coupe with a turbocharger. Development time and cost as well as investment could thus be minimized.

This paper describes in detail the complete engine and particularly the modifications required to cope with the higher thermal and mechanical loads, the installation and adaptation of the turbocharger, the adaptation of the fuel injection pump, and the refinements of the combustion process. This work, together with suitable turbocharger characteristics due to wastegate control on the turbine side, resulted in an attractive power plant with very good stationary and transient operational behavior. Finally, engine and vehicle performance data including noise, exhaust emissions and fuel economy are given.

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TYP		ENGINE	1.PRODUCTION YEAR	DISPLACEMENT cm^3	STROKE mm	BORE mm	MAX.OUTPUT AT SPEED (DIN) kW	1/min	TORQUE AT SPEED Nm	1/min	NUMBER OF CYLINDERS	DRY ENGINE WEIGHT KG
	260D	OM138	1936	2545	100	90	33	3300			4	
	170D	OM636	1949	1697	100	73,5	28	3200	96	2000	4	
	180D	OM636	1953	1767	100	75	31	3500	101	2000	4	
	190D	OM621	1958	1897	83,6	85	37	4000	108	2200	4	
	200D	OM621	1965	1988	83,6	87	40	4200	113	2400	4	
	200D	OM615	1967	1988	83,6	87	40	4200	113	2400	4	
	220D	OM615	1967	2197	92,4	87	44	4200	126	2400	4	
	240D	OM616	1973	2404	92,4	91	48	4200	137	2400	4	
	300D	OM617	1974	3005	92,4	91	59	4000	172	2400	5	
	200D	OM615	1976									195
	220D	OM615	1976									197
	240D	OM616	1976			AS ABOVE						197
	300D	OM617	1976									229
	300CD	OM617	1977									229
	300SD	OM 617 A	1978	2998	92,4	90,9	85	4200	235	2400	5 TURBO-CHARGED	244

Fig. 1 - Historical survey of Mercedes-Benz Diesel cars

The newly developed 300 SD with the turbocharged 5-cylinder Diesel engine type OM 617 A, as top of the Diesel car line, is the final achievement in the historical process of the evolution of the Mercedes-Benz (MB) Diesel passenger car. Thus, dieselization of the MB passenger car fleet is one important approach toward complying with the requirements of the Energy Act.

Fig. 1 is a historical survey of the main engine data of MB light duty diesels. The rate of progress in engine performance, rated speed and displacement is depicted in Fig. 2. Originally, the desired increase of performance was achieved by means of raising rated engine speed and displacement. With engine OM 616, installed in the 240 D, individual cylinder displacement reached a limit which represents an excellent compromise as regards combustion, efficiency and operating smoothness. The final two steps of the increase in performance were those of adding a cylinder, thus producing the naturally aspirated (NA) five-cylinder engine type OM 617, and, now, the utilization of turbocharging (TC), engine type OM 617 A.

DEVELOPMENT REVIEW

In the beginning of 1976, in spite of all uncertainties then existing as regards the future of Diesel cars in terms of the NO_x Standard, the decision was taken to develop a Diesel engine suitable for installation into the larger S-type body. The object was to offer an extremely fuel-efficient vehicle in this size and weight class. The general layout of the TC-engine OM 617 A can be seen in Figs. 3, 4, 5, and 6. Preliminary tests, including experiences ob-

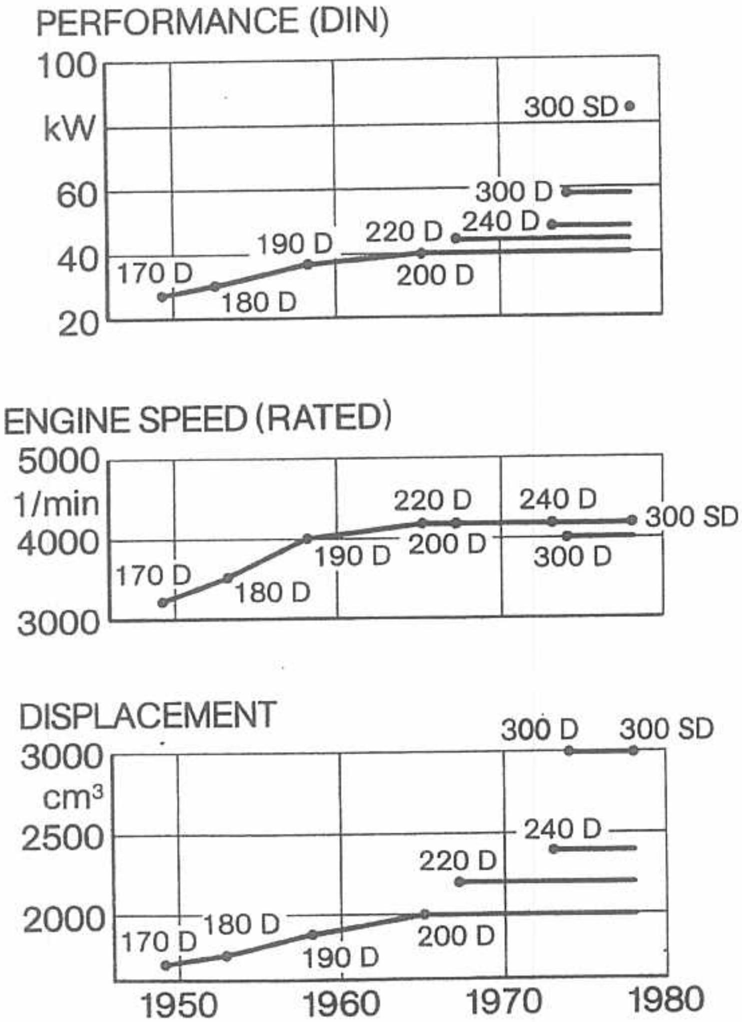

Fig. 2 - Progress in development of LDV Diesel engines

tained on very large Diesel engines, established the suitability of the MB prechamber combustion system for purposes of turbocharging. The mechanical behavior of prototypes was equally good. Thus, the decision for mass production of the engine could be taken. In 1977, 16 class and 3 world records were won, e.g. 10,000 miles at an average speed of 251.798 km/h (156 mph), with a pre-prototype-engine having an output of about 140 kW (190 hp) installed into an advanced version of the experimental C 111 vehicle (5).

On the basis of preliminary tests it was possible to list and describe the necessary changes to be performed to the NA-engine right at the outset of the development of the TC-version. Interestingly, this first job list did not have to be changed in the course of the work so that pilot production for a large-scale fleet test could begin in early 1977. Series production for the USA began during the early months of 1978. As compared to the NA-version, the OM 617 A displays an increase in performance of 43 % with only 7 % added weight.

COMBUSTION SYSTEM

Selection of a suitable combustion system for a high-speed Diesel engine in passenger cars must be based upon the following criteria:

- exhaust emissions
- smoothness of operation (noise)
- specific performance
- fuel consumption
- durability
- cold start capability
- suitability for turbocharging.

While other manufacturers of Diesel passenger cars have selected other combustion systems, Daimler-Benz has always regarded the prechamber combustion system as the best compromise because it displays better features in most of these criteria. In recent years, the criteria of exhaust emissions, smoothness of operation and suitability for turbocharging have verified the prechamber design as a modern combustion system.

EXHAUST EMISSIONS - Aside from the completely uncritical CO emissions - typical for all Diesels - the prechamber exhibits the lowest hydrocarbon emissions, while its NO_x emissions - identical vehicle and basic engine design assumed - in comparison to the swirl-chamber system are marginally higher. However, this slight advantage of the swirl chamber is by no means sufficient to be of decisive importance in complying with future emission standards for NO_x.

OPERATIONAL SMOOTHNESS - Over a wide part of the operating range, a prechamber system permits to reach almost ideally the desired goal of combustion at constant pressure, which is of prime importance for the generation of noise and - particularly in case of a TC-engine - for the mechanical load upon a number of engine components.

Fig. 7 displays by means of the pressure/time-traces in the main chamber of a warm engine the excellently smooth operation at low idle which is of particular interest for the acceptancy of Diesel engines, as well as at maximum torque and maximum power.

The adaptation of the injection system to the increased fuel quantity for a TC-engine in case of the prechamber permits a further prolongation of the duration of injection which is of advantage as regards NO_x emissions, smoke emissions, und unchanged quality of idle, with only a very small trade-off in specific fuel consumption as compared to optimum values, while the emissions of the other compounds do not increase.

At the same time, practically no higher pressure levels result by the initiation of combustion, so that the maximum pressure in the

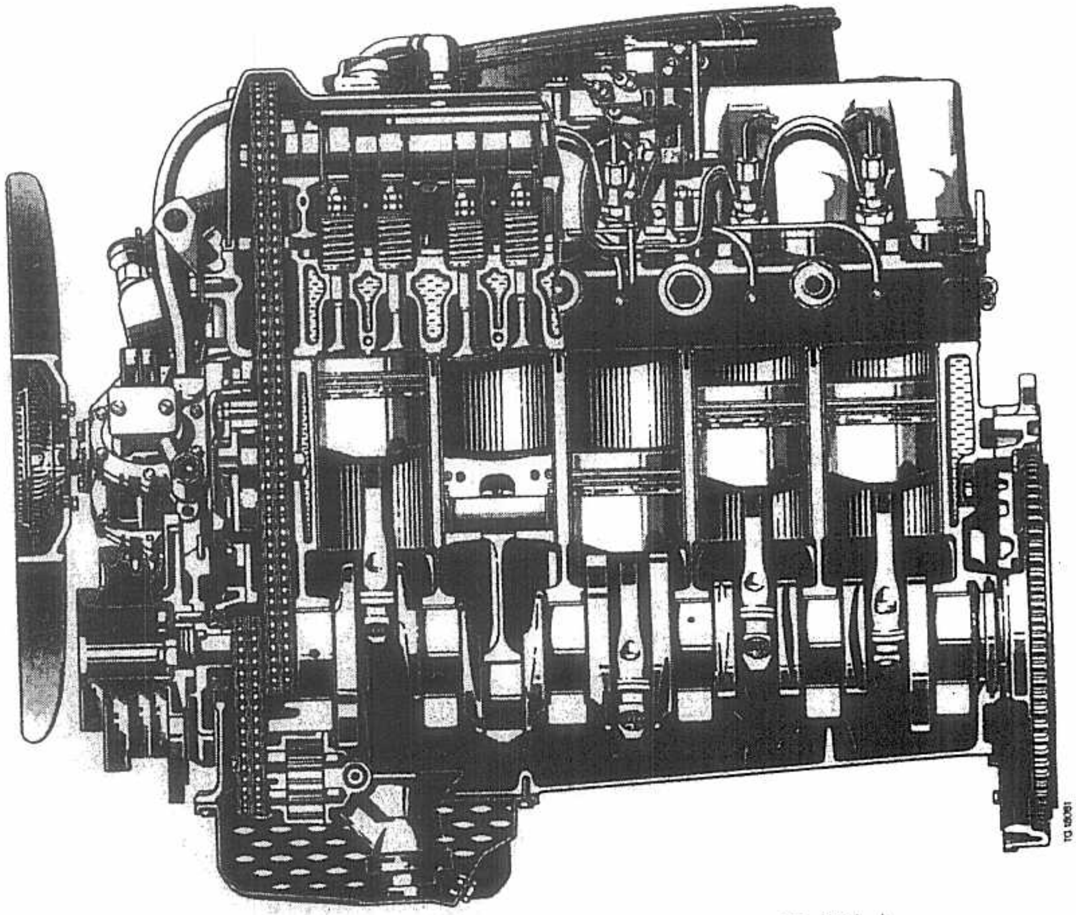

Fig. 3 - Longitudinal section of engine OM 617 A

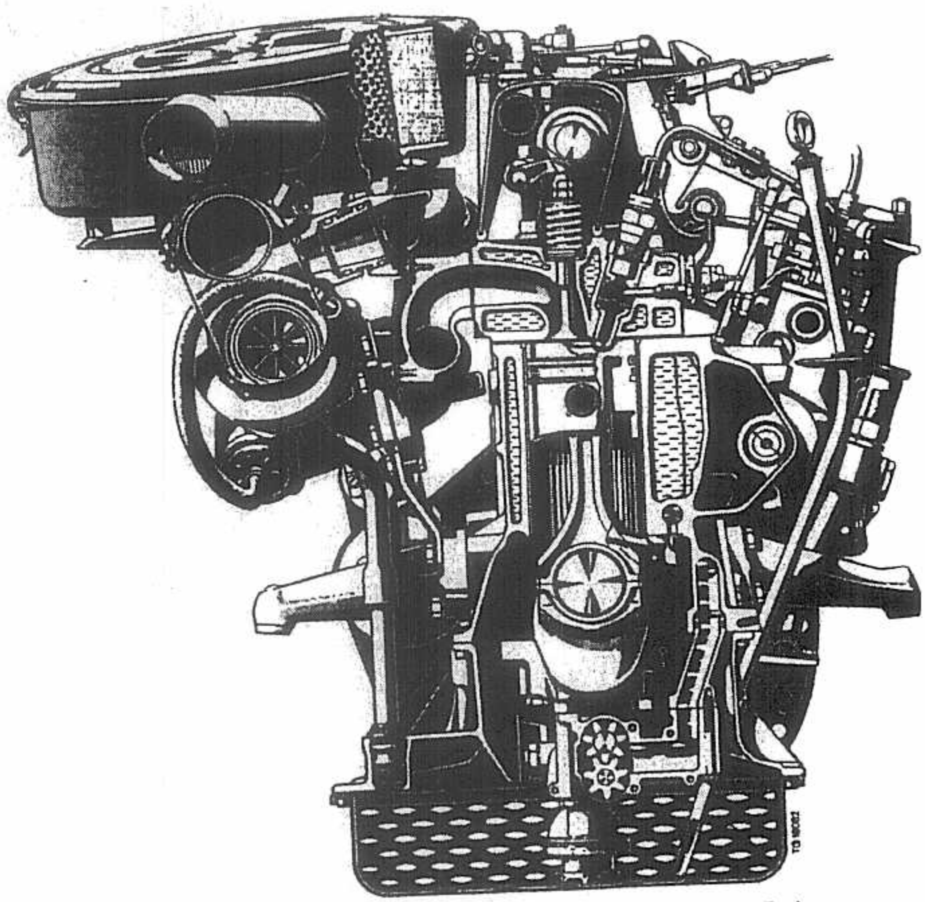

Fig. 4 - Cross section of engine OM 617 A

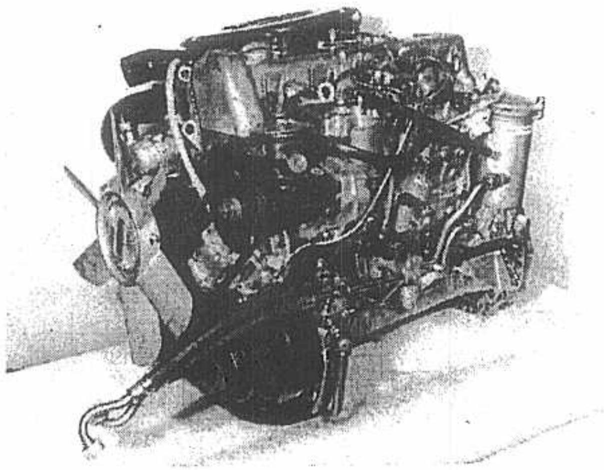

Fig. 5 - View on injection pump side

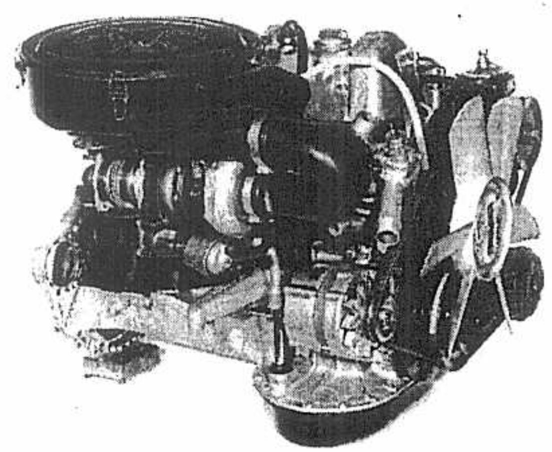

Fig. 6 - View on turbocharger side

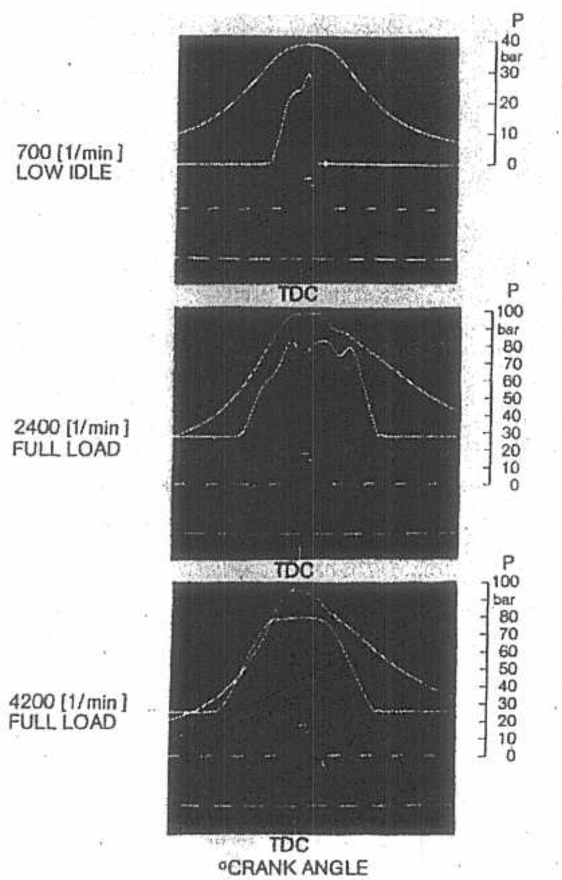

Fig. 7 - Traces of main chamber pressure and injection nozzle needle lift

main chamber is mainly defined by the level of boost pressure and compression ratio. Idle quality and behavior during the warm-up phase do not permit a reduction of the compression ratio without other extensive efforts. Therefore, the compression ratio for engine OM 617 A was fixed at 21.5, similar to the NA-engine.

SUITABILITY FOR TURBOCHARGING - In case of a TC-engine utilizing a turbocharger with boost control the maximum pressure in the main chamber is not higher than 95 bar while the NA-engine displays a peak level of 65 bar. Due to the lower compression ratio as compared to swirl-chamber Diesel engines, turbocharging in connection with the prechamber combustion system, therefore, results in the lowest possible peak pressure levels.

ENGINE MECHANICS

CRANKSHAFT - The higher ignition pressure in the TC-engine leads to intolerably high stresses in the inductively hardened crankshaft of the NA-engine. The crankshaft, therefore, is now bath-nitrided. Nitrogen diffusion into the peripheral material creates an increase in hardness and doubles the fatigue strength under continuously alternating loads. Additionally, the tempering effect during forging ensures the necessary lack of distortion, which is important because subsequent alignment could result in cracks occurring in the crank radii. The tests showed extremely good operational properties of the bearing journals, which are lapped after nitriding.

MAIN BEARINGS - Main bearings are identical to those in the NA-engine. The steel base material is coated with a bonding layer of Al-foil and the actual bearing surface is of Al Sn 20. The sleeves of the axial thrust bearings with their integral thrust collar, however, displayed fatigue fractures. Accordingly, they were replaced by a bearing with separate thrust plates, Fig. 8. The upper sleeve halves of the main bearings are without a location stud, while those on the bearing cover are provided with two studs which ensure their correct position. Additionally, these punched studs serve as a protection against rotation.

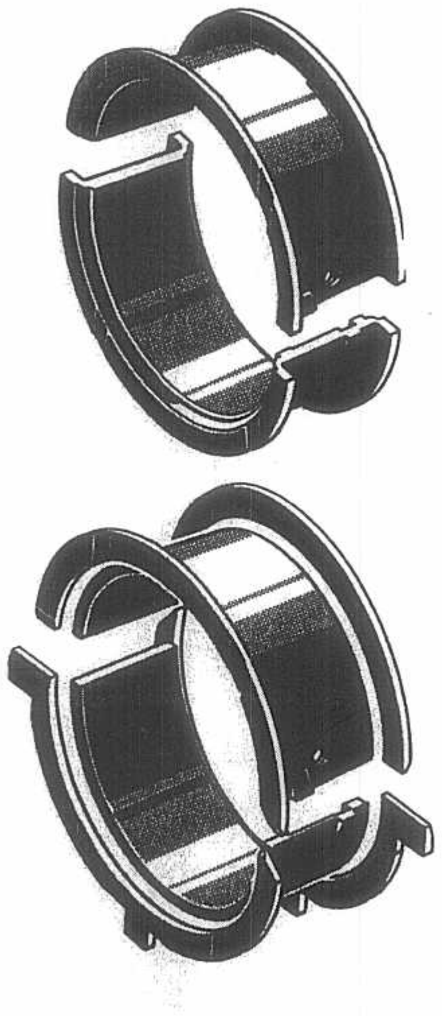

Fig. 8 - Axial thrust bearing of NA-engine (integral, top) and of TC-engine (loose plates, bottom)

This bearing design was not sufficient for the connecting rod bearings which exhibited damage in the Al Sn 20 coating. It required quite extensive testing to define the final form of this bearing: the steel carrier is coated with an Al-bonding layer and a harder Al Sn 6 alloy with lower Sn content on top. This top layer has, however, very bad break-in features, so that an additional break-in coating had to be provided. Before this break-in layer is deposited galvanically, a flash of nickel of 2 to 4 μm is provided as a bond.

Fig. 9 is a photo of the connecting rod bearings of a TC-engine after 100.000 miles of operation: The left set of bearing shells clearly pictures the break-in procedure in the center portions. In the brighter regions, the break-in procedure has been concluded, and the nickel coating shows.

PISTON - During the first very basic tests in 1973, increased piston temperature caused ring scuffing, piston abrasion, and piston seizures. The first prototype TC-engines were, therefore, equipped with a fixed cooling oil jet directed at the interior of the piston crown. With this initial arrangement, the temperatures behind the first piston ring groove of 270° C and close to the edge of the piston recess of 340° C, both at an engine speed of 4500 rpm, were excessive. Only the design of a piston with integrated cooling pipes managed to reduce these temperatures to the level occurring in the NA-engine, 225° C and 275° C respectively, enabling perfect engine operation, as shown in Fig. 10.

The cooling piston for the TC-engine was developed in close cooperation with the piston vendors with the aid of photoelastic models, engine-external measurements, and dynamic load tests on the hydraulic pulse tester with a total number of load alternations of up to 3×10^7. During these tests, piston stresses due to combustion pressure were simulated by hydraulic means. Aside from the stresses in the piston crown and piston pin region, the forced directed at the piston skirt were also simulated by means of an inclined auxiliary connecting rod (6).

Fig. 11 shows the location of the internal oil pipe, as it was optimized regarding heat transfer, castability, and strength. As can be seen, the piston is also equipped with an alfin-bonded integral ring support for the upper piston ring against wear. Oil inlet and oil outlet were optimized together with the form of the jet and its design, and the quantity of oil, the filling-efficiency, and the shaker effect due to the oscillating piston. The cooling pipe is cast by inserting a salt core into the mold, supported via the oil inlet and outlet. After casting, the salt core is melted out.

The internal contour of the piston was very careful optimized with regard to deformation, rigidity, and susceptibility to cracks. Particularly the inside contour of the lower edge of the piston skirt in the region of the oil jet required extensive testing before its final form could be defined.

The grinding pattern of the piston, as described in Fig. 12, was designed so that the contact surface appeared perfect both in a cold and a warm engine. The solid line shows the profile of the piston in the plane of motion of the connecting rod, the dashed line in the plane of the connecting pin. Special seizure tests helped to ensure that even under extreme conditions (e.g. during overtemperature or sudden full-load operating of a cold engine) no seizures or material smearing occurred. At the same time, the generation of noise had to be observed carefully. In a lengthy tuning process it was possible to obtain good results with a clearance of 50 μm in the lower part of the skirt.

The top land of the piston is contoured in order to adapt the thermal distortion of the land region to that of the cylinders. Above the ring carrier, the top land was extended as far as possible toward the cylinders in order to re-

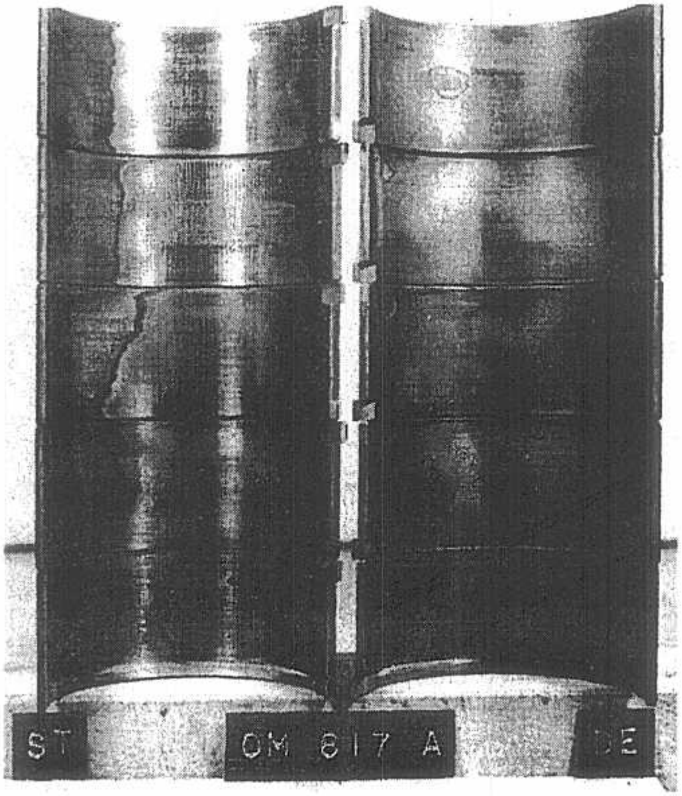

Fig. 9 - Connecting rod bearings after 100 000 miles

duce the heat flow into the first ring groove. The ring carrier which is utilized in order to reduce axial groove wear is recessed as compared to the top land so as to avoid its contact with the cylinder surface during operation. The contour of the piston below the oil scraper ring is chamfered in order to guarantee a uniform distribution of lubricant. The skirt itself is convex/oval, and graphite coated.

In the original piston crown, cracks occurred at the edge of the deep recess which houses the lower portion of the prechamber insert in TDC position. These disappeared completely when this edge was rounded-off and adapted to the contour of the new prechamber insert.

The piston is equipped with the following rings: The top ring, 3 x 3.8 mm, is of rectangular cross section, chamfered in the inside, and made from heat-treated nodular iron; the second compression ring is of similar design with dimensions 2.0 x 3.8 mm, and the surface of both rings carries a special plasma-sprayed Mo-base coating. The oil scraper ring is of tapered design with the dimensions 4.0 x 3.2 mm. It is assisted by a flexible spring tubing with tighter coiling at the joint and a teflon jacket to reduce wear.

Fig. 13 shows a circumferential photo of a piston's running surface after about 100,000 miles of operation in a severe vehicle durability test with maximum pay load and maximum speeds on the "Autobahn", mixed country-road and city driving.

The wrist pin bearing was designed for higher loads by increasing the diameter by 2 mm to 28 mm. Also, its axial length in the piston was extended by 4 mm, shortening it in the connecting rod correspondingly.

CYLINDER HEAD AND CRANKCASE - In Diesel engines, the most critical region in the cylinder head is that of the bridge between the intake and the exhaust valve. To reduce the thermal load in this region, Daimler-Benz introduced a special coolant passage, starting with engine type OM 616 in 1974. The excellent results obtained with this design were verified on the TC-engine: The lower face of the cylinder head does not provide any problems, same as in the NA-engine. Only the machining process of the bore housing the prechamber and the glow-plug seat had to be adapted to the new operational conditions, Fig. 14.

In order to limit wear of the valve seats to the usual values the valves were adapted to the higher combustion pressures by providing increased thickness at the edges of the valve head resulting in improved rigidity. Additionally, the exhaust valve is cooled with sodium, and the valve stem seal, therefore, is more temperature-resistant.

Design changes of the crankcase are mainly due to a change in the lubrication system. The oil pump drive originally located at the side of the engine was dropped. Instead, the crankcase was extended axially to provide added space for the additional second chain driving the new and enlarged oil pump. This was achieved by providing a cover around the front crank shaft bearing which houses the shaft seal, as is depicted by Fig. 15.

From the oil pump, the lubricant is passed through a bent pipe into the crankcase. Additional flanges at the pressure oil line carry the oil jets for piston cooling which are located by means of dowels. During engine assembly, their spraying direction is checked individually. With the oil collecting recesses in the lower edge of the piston skirt and the carefully tuned jet diameters it is possible to accurately meter the correct amount of lubricant. This arrangement is illustrated by Fig. 14.

Obviously, the TC-engine also brings about higher requirements for the cylinder head gasket. Fig. 16 reveals its form which corresponds to that of the NA-engine. The composition of the elastic interior material with its special impregnation is a new development. This impregnation cures during engine operation, and markedly improves the long-term compression characteristics of the seal.

Of particular advantage for the thermal and mechanical loads exerted upon the cylinder head

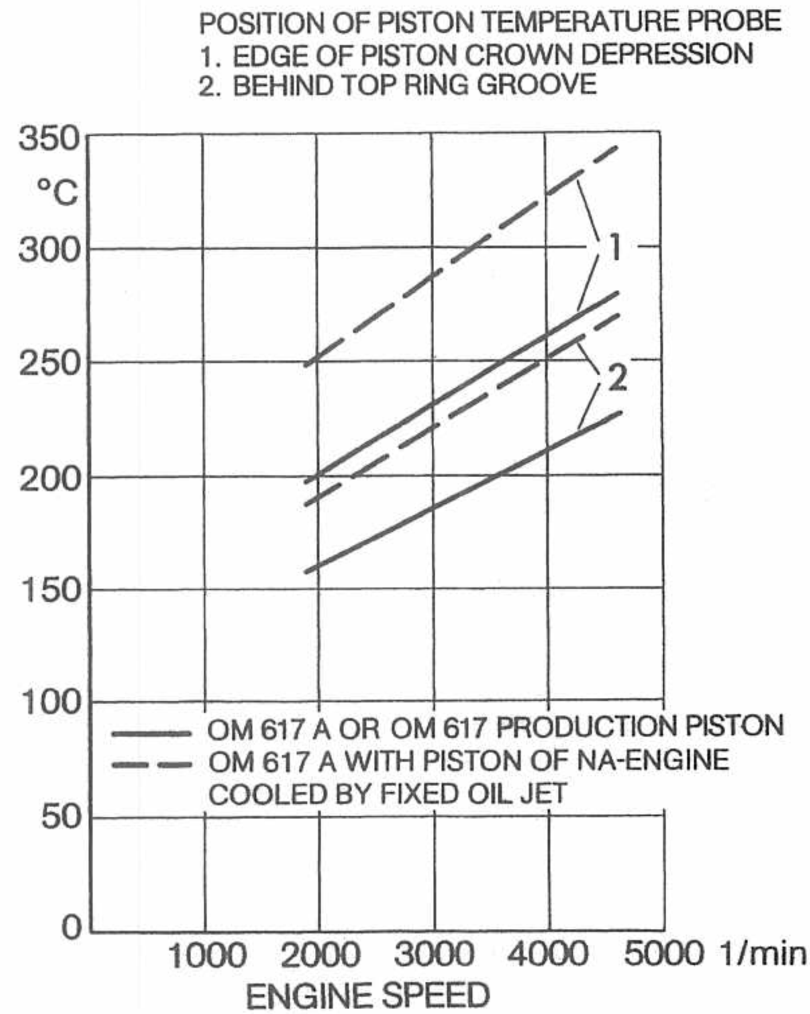

Fig. 10 - Comparison of piston temperatures at full load

seal is the prechamber combustion system in which the seal is not in contact with the hot auxiliary chamber as can be seen by looking back at Fig. 14. The coolant slots between the cylinders which are open toward the gasket maintain low gasket temperatures between the cylinders, Fig. 17.

Six cylinder head bolts around the cylinder circumference (shaded in Fig. 16) create the necessary initial stress in the gasket and guarantee a favorable distribution of the load.

Under these favorable conditions the metallic skirt of the seal developed in fatigue tests can well serve its purpose of sealing gas and water, even under extended operation in the TC-engine, Fig. 18.

ADAPTATION OF PRECHAMBER

One of the important parameters in designing the TC-engine for the same durability as a NA-version is the temperature level in the lower part of the prechamber insert, the so-called burner. Actually, the bottom of the burner is one of the engine components subject to the highest thermal load, and it requires very careful design and, equally, choice of material.

The tests have revealed that the increased thermal load due to turbocharging can completely be compensated by an improved dissipation of the heat into the cylinder head. The contact area between the burner and the cylinder head was enlarged through a 2 mm larger burner diameter and an extension in the axial direction. Together with a reduced area of the lower burner tip these measures resulted in maximum burner temperatures of around 900° C and thus did not exceed the level observed in the NA-engine, Fig. 19. With this, identical burner material (Nimonic 80 A) can be used in the TC-engine.

Further operational reliability is achieved by means of electrolytic burring of the inner

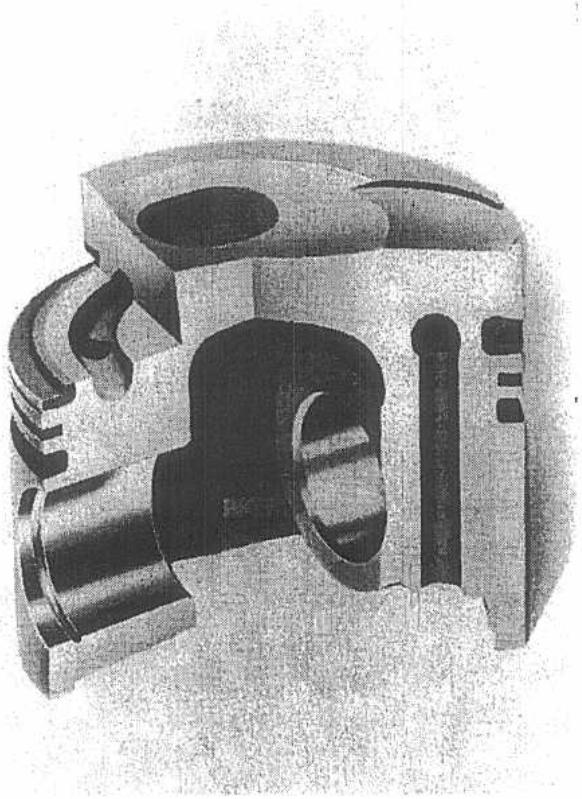

Fig. 11 - Piston with integral cooling channel and cast-in ring carrier

edges of the transfer passages, which can be seen in Fig. 20. Extensive tests toward optimization of the geometry of these transfer passages and the volume of the prechamber resulted in an unsymmetrical design with different passage diameters and an additional passage in the bottom of the prechamber, thus ensuring uniform combustion throughout various regions of the main combustion chamber in spite of the unsymmetrical location of the prechamber. With the modifications of the burner described above the volume of the prechamber was only slightly increased.

Thus, the prechamber was adapted to the conditions in the TC-engine. The results obtained verify the excellent suitability of this combustion system for turbocharged operation.

GLOW PLUGS

Divided-chamber Diesel engines permit the installation of the required starting aid for coldstart: Glow plugs are located in the individual prechambers.

The most economical solution, utilized on the NA-engine, and providing sufficient durability is that of hot-wire-coils connected in series. The higher requirement of the TC-engine was met with quick-glow pin-plugs connected in parallel. Both plug types show identical temperatures during the first 20 seconds of heat-up, after this the pin plug is hotter.

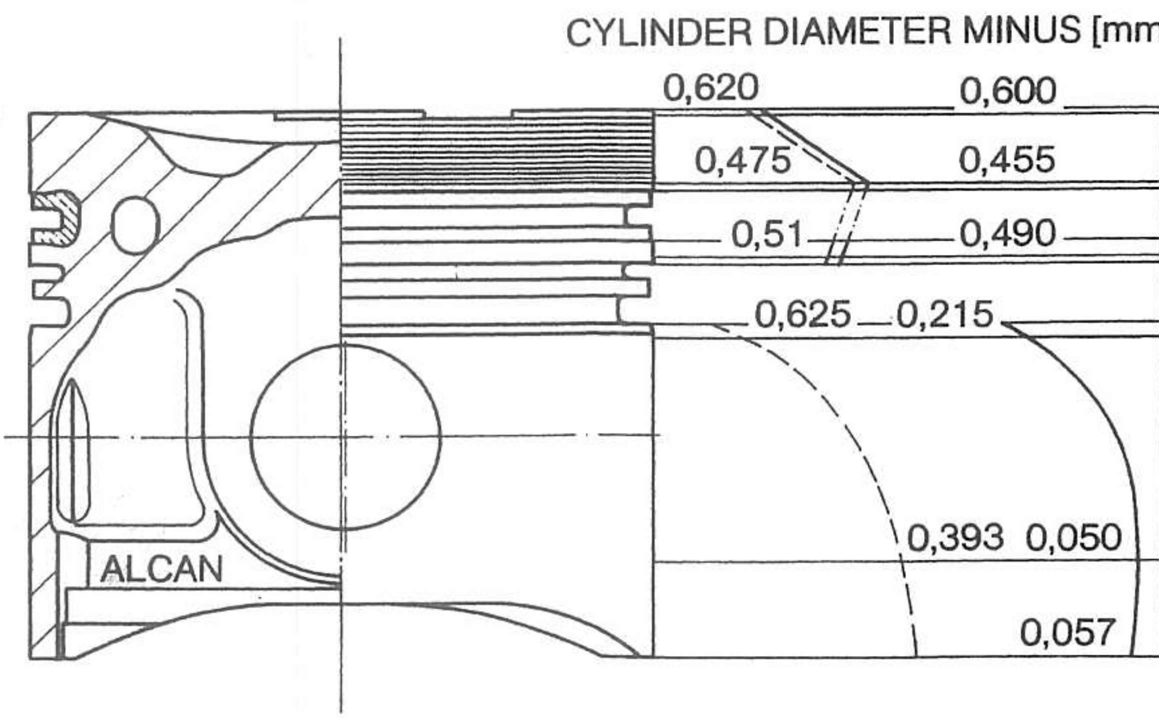

Fig. 12 - Piston dimensioning

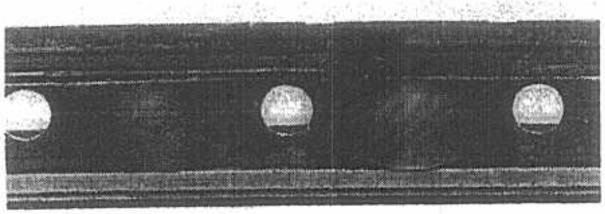

Fig. 13 - Circumferential view of piston running surface after 100 000 miles

In case of the parallel connection, however, the driver would not notice a possible failure of one of the pin plugs, other than at extremely low temperatures. An electrical circuit was, therefore, developed which - after a successful start of the engine - leads to a flashing display of the glow plug indicator light in the dash, if one of the plugs is not energized, thus reminding of an occasional repair.

In all other respects the key-operated electrical starting system corresponds to that of the NA-engine. As Fig. 21 displays, it features the same coolant temperature sensor for a quick start with the warm engine.

LUBRICATION SYSTEM

Compared to the NA-engine, the TC-engine has two additional lubricant consumers: the oil jets for cooling of the pistons, and the exhaust turbocharger. The general layout of the lubricating system can be seen in Fig. 22.

The gear drive of the oil pump of the NA-engine could no longer handle the roughly double quantity of oil in a satisfactory fashion. The most extensive engine design change was therefore conducted at the oil pump and the oil pump drive. After investigations of various pump types and pump drives, a chain-driven gear-type pump located in the oil sump was selected, similar to that installed in MB V8 gasoline engines. A duplex roller chain (Fig. 15) displayed superior wear behavior. The aluminum housing of the pump is attached to the front crankshaft bearing support. The oil pressure control valve limits the oil pressure to 8 bar (note: all pressure values in this paper are absolute pressures) and contains an integral bypass back into the suction side of the pump. A spring-loaded plastic chain tensioner guarantees smooth chain operation.

The lubricant volume in the engine sump had to be adapted to the larger flow rate, for which the lower sheet steel oil pan of the 6-cylinder Otto-engines rendered itself. The upper part of the oil pan from pressure-die-cast aluminum had to be adapted to the new design. It is provided with a connector for the oil return flow from the turbocharger, while a sump cover is cast-in. This guarantees uniform conditions of lubrication for all cylinders.

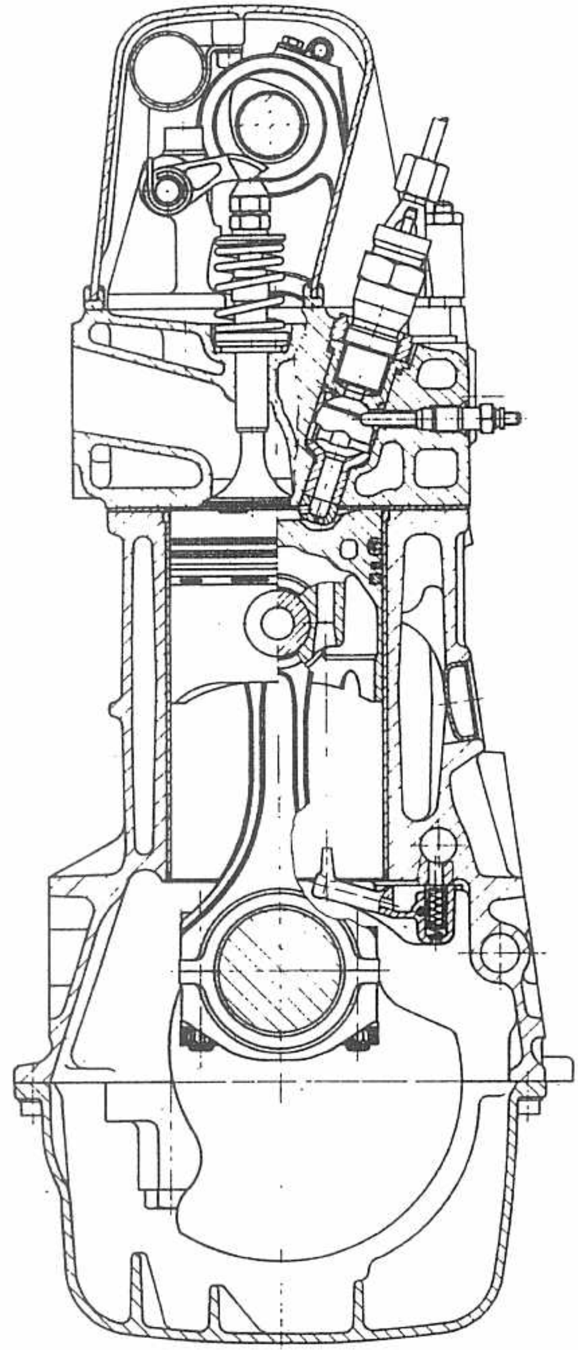

Fig. 14 - Engine cross section showing layout of combustion chamber and piston cooling arrangement

During the development work, a new oil filter for all passenger car Diesel engines was designed which facilitates a replacement of the combined full flow-/partial flow cartridge in the vehicle from the top, Fig. 23. Upon opening the filter cover the oil in the filter runs into the sump. From there it can be sucked out through the dip stick guide.
A one-way valve safeguards against complete drainage of the filter and the lubricating pas-

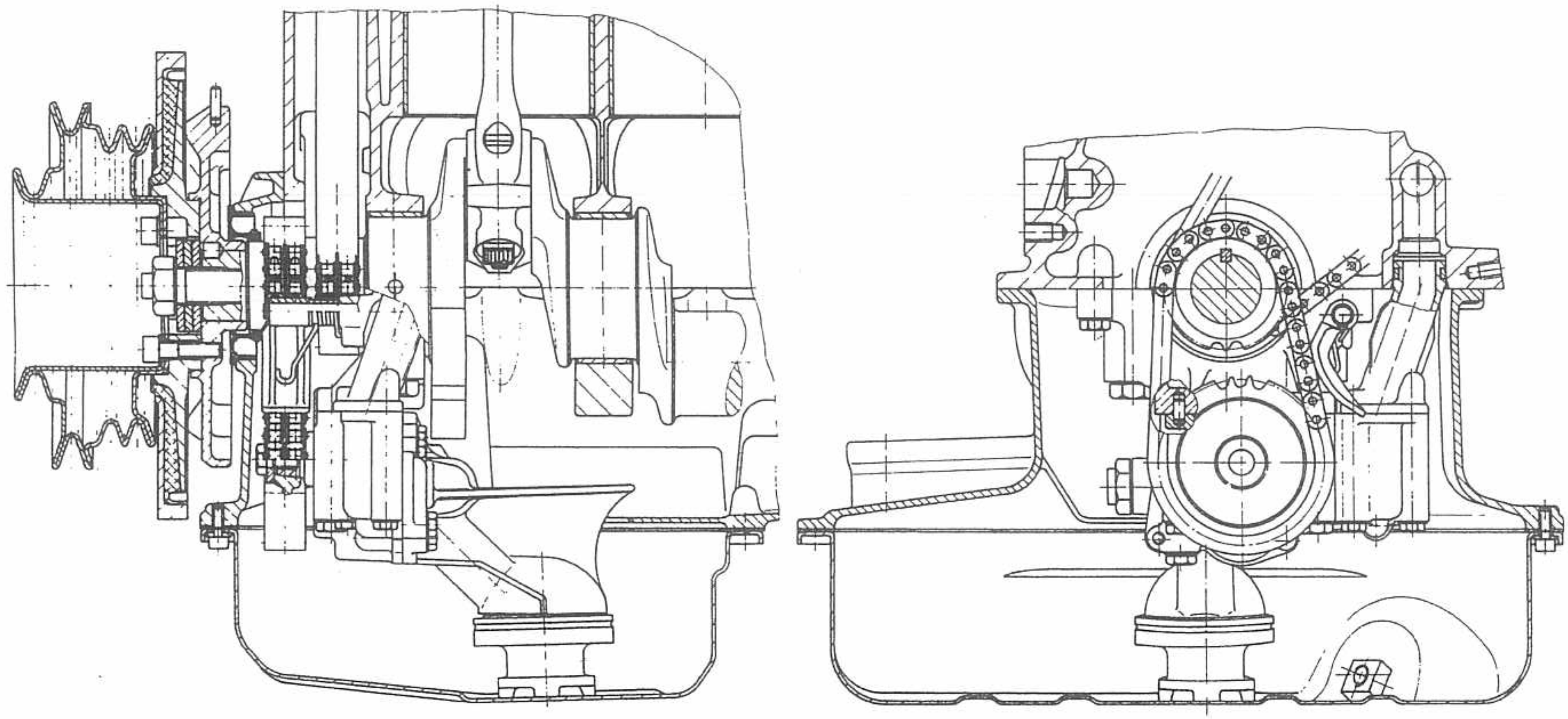

Fig. 15 - Crankshaft front end with oil pump

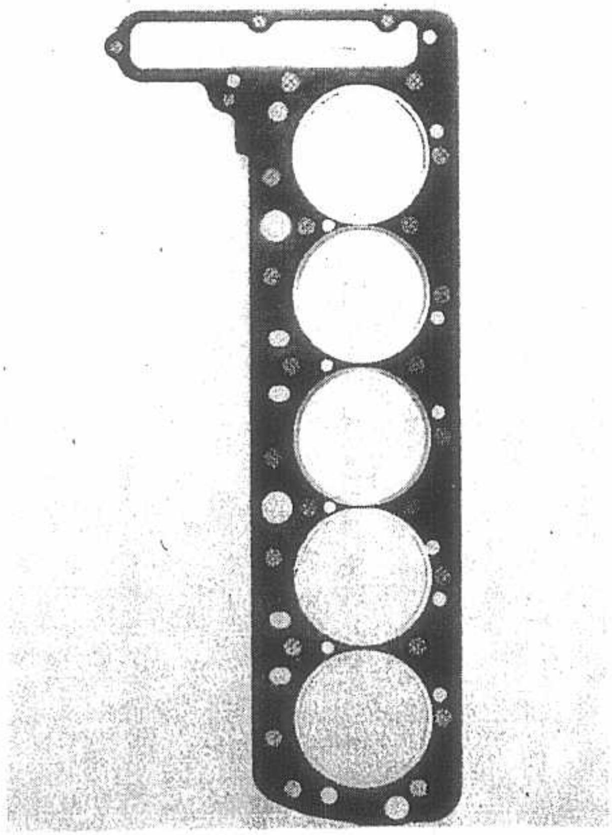

Fig. 16 - Cylinder head gasket

sage for the turbocharger. A 90° C thermostatic valve integrated into the filter can divert the lubricant into an oil cooler which is located next to the radiator behind the radiator grille. Fig. 24 shows this condition. A by-pass parallel to the oil cooler reduces the pressure drop across it.

The filter insert itself was designed as a combined cartridge. The full-flow paper cartridge was matched to the higher oil flow rates. The partial-flow filter consists of a cotton packing, and the size of an orifice in the return flow line to the crankcase restricts the flow rate through it. With this design a high dwell time is achieved and even the smallest particles can be deposited.

If plugging occurs in the filter as a result of omitted maintenance, a differential pressure valve ensures that the engine is fed with oil, Fig. 25. The lubricating oil for the turbocharger is branched-off downstream of the full-flow filter. Its mesh size of 25 μm results in the high quality of filtering required by the high-speed plain bearing of the turbocharger shaft.

The oil jets for the pistons are made of pressure-die-cast aluminum and are attached to the pressure line in the crankcase. Valves contained in the oil jets close these below 2.5 bar pressure. This provides for sufficient oil pressure also during idling conditions, when cooling of the pistons is not required.

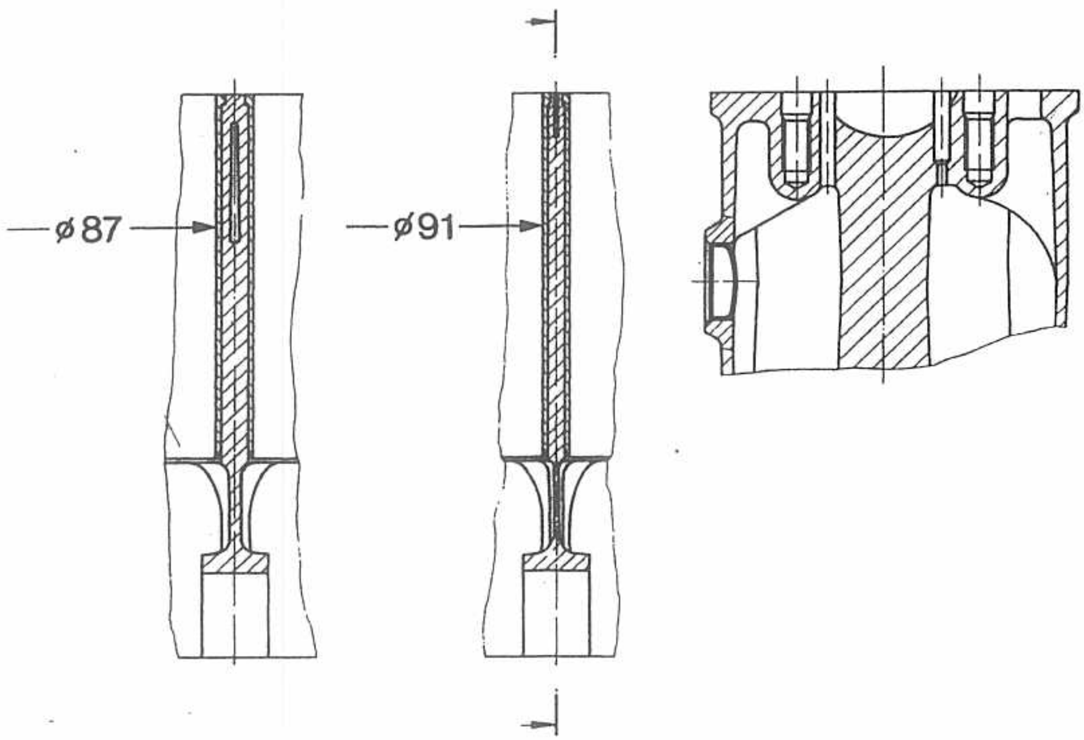

Fig. 17 - Coolant slot between cylinders

The pressure line in the crankcase also directly feeds the main bearings of the crankshaft and - via a 180° groove in the bearing and a T-form bore - the connecting rod bearings. At the front end of the pressure line, the oil supply for the sprocket bearing, the vacuum pump, the injection pump and the chain tensioner is branched off. The feed line for the overhead camshaft is also located here. The oil quantity for the camshaft is controlled by means of a metering groove in the first bearing. An oil pipe running parallel to the camshaft feeds the oil to the remaining three camshaft bearings and contains oil jets for lubrication of the cams, in the usual fashion. The layout can be seen in Fig. 22.

With these measures it was possible to release oils from the SE-CC quality level. The oil change interval - generally shorter for Diesel engines - could be extended to 4.000 miles. With this period, a sufficient safety margin is guaranteed before the filter clogs or the oil deteriorates excessively.

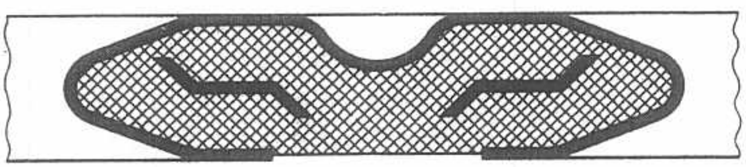

Fig. 18 - Schematic cross section of cylinder head gasket

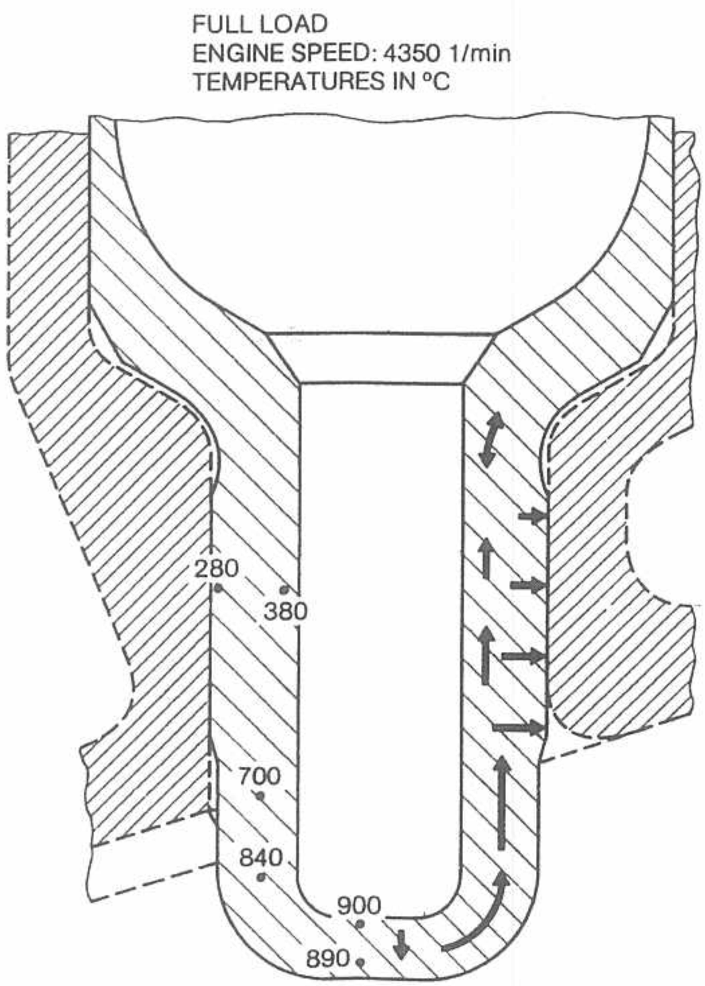

Fig. 19 - Maximum temperatures of lower prechamber and heat flow pattern

FUEL INJECTION SYSTEM

INJECTION PUMP - Daimler-Benz, for its passenger car Diesel engines, has always utilized in-line injection pumps by Robert Bosch. Their design is partly responsible for the success and the reputation of MB Diesel cars. In the course of the development of the TC 5-cylinder Diesel engine, their far-reaching adaptability to new requirements by simple means has, once again, shown.

The basic model of the injection pump for the new OM 617 A engine is that of the NA-engine, with the usual advantages of such high degree of commonality. This 5-cylinder in-line injection pump with mechanical governor was optimized as regards load and speed dependent injection characteristics, so as to obtain best results in performance, fuel consumption, and, moreover, exhaust emissions and smoothness of operation.

The US version of this injection pump features reverse flow damping valves for the elimination of erratic late injections, reducing hydrocarbons, and an automatic altitude correction device varying maximum fuel flow dependent upon barometric pressure, thus limiting smoke emissions. The prechamber combustion system and the high precision injection system result in a consistency of combustion over long periods of time typical for MB-Diesel engines. All of these

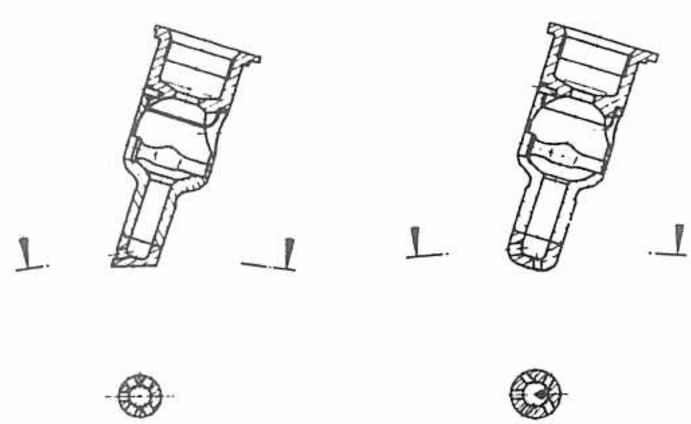

Fig. 20 - Prechamber configuration of NA-engine (left) and TC-engine (right)

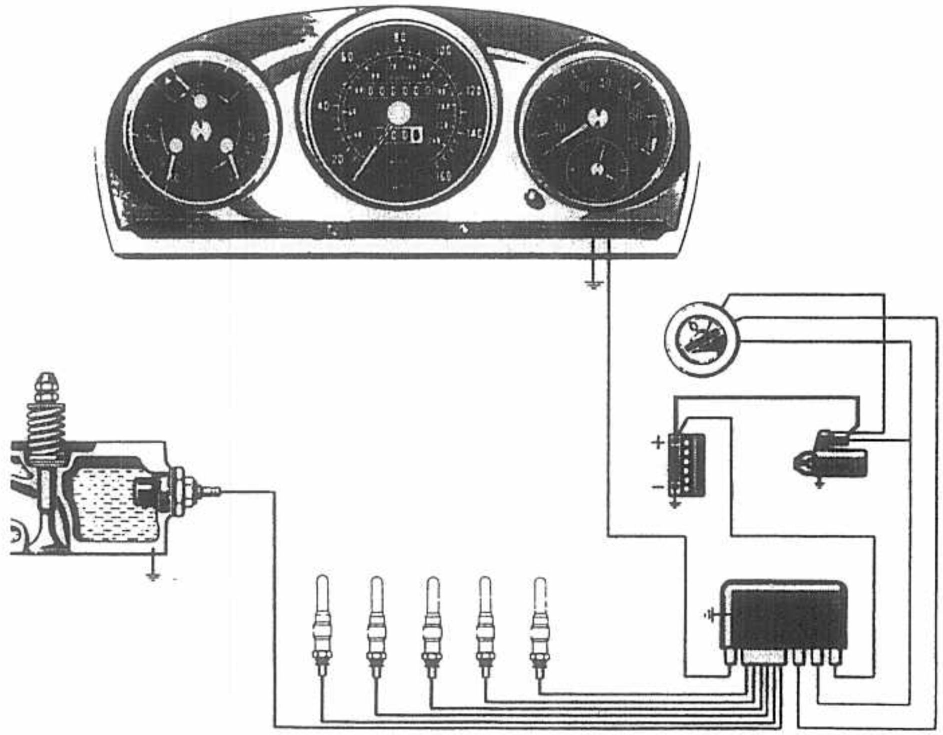

Fig. 21 - 300 SD starting system schematic

measures guarantee a constant and extremely low level of exhaust emissions at any altitude. Thus Diesel engines are by no means - as some say - uncontrolled in terms of emissions: From the early development days Diesels have required a high degree of technical precision in order to obtain their operational characteristics.

Emission tests conducted during the development of turbocharging have verified the expected further decrease of HC- and CO-emissions. A NO_x-Standard of 2.0 gpm poses difficulties for the vehicle in question, if the injection rate, i.e. the injection quantity per time, were raised proportionally to the increase in injection quantity over that of the NA-engine. This would additionally result in a deterioration of operational smoothness, particularly during idle and after cold start.

The decisive advantage of the prechamber combustion system proved to be that in conjunction with turbocharging it enabled a certain prolongation of the duration of injection with only very slight trade-offs in specific fuel consumption. With this very feature the engine's excellent characteristics regarding exhaust emissions and smoothness of operation were maintained.

In matching the increased injection quantity to the boost pressure of the turbocharger, an aneroid measuring the absolute manifold pressure and controlling the quantity of injected fuel was developed. It ensures low smoke emis-

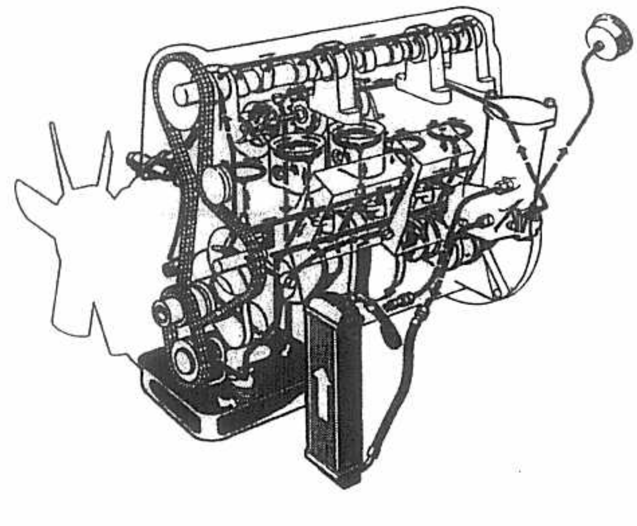

Fig. 22 - Lubrication system

sions under stationary and transient conditions, both at sea-level and at high altitudes. As Fig. 26 depicts, this absolute pressure sensing device called ALDA acts upon the mechanical governor at the same place where the automatic altitude correction device (AAC) of the NA-engine is located.

The fuel-feed characteristics of the plungers were modified in correspondence with the higher performance, with particular emphasis on the desired load response characteristics

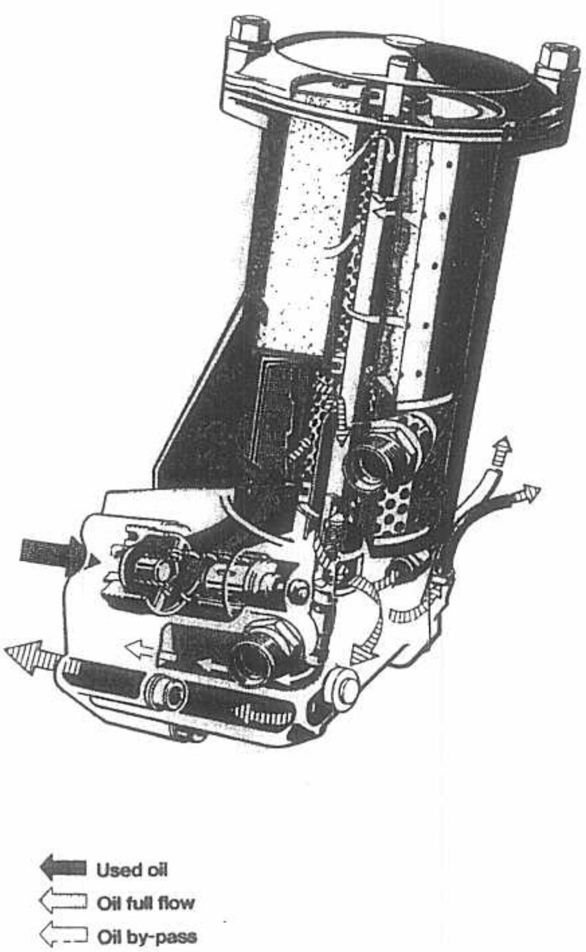

Fig. 23 - Oil filter flow pattern at low temperature

during idle. The behavior of the governor was adapted to these changed plunger characteristics.

The kinematic and hydraulic characteristics of the injection system in conjunction with a corresponding lay-out of the aneroid controls the necessary quantity of fuel without boost pressure both at sea-level and at high altitudes to the corresponding values of the NA-engine. From this baseline, the fuel quantity is increased as the boost pressure rises up to a certain maximum value, so that under transient conditions the same air excess ratio as in the NA-engine is obtained resulting in similar smoke characteristics. A further increase in boost does not result in a corresponding increase in fuel quantity, so that in a thermodynamically ideal fashion a higher air excess ratio is maintained and controlled, which has an additional positive effect on smoke.

Fig. 27 displays this relationship in the form of a fuel rack control map with and without boost, at various altitudes, as a function of engine speed. It was thus possible to develop a system which permits optimum utilization of

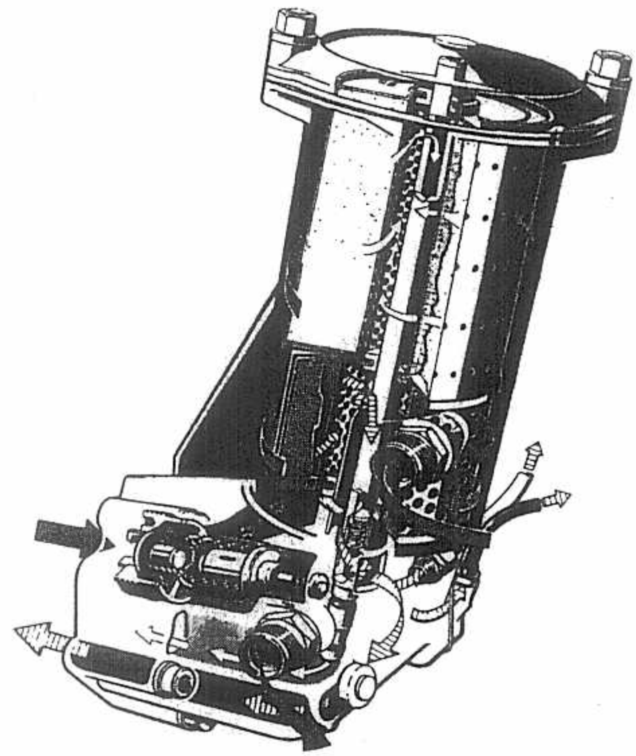

Fig. 24 - Oil filter flow pattern at high temperature

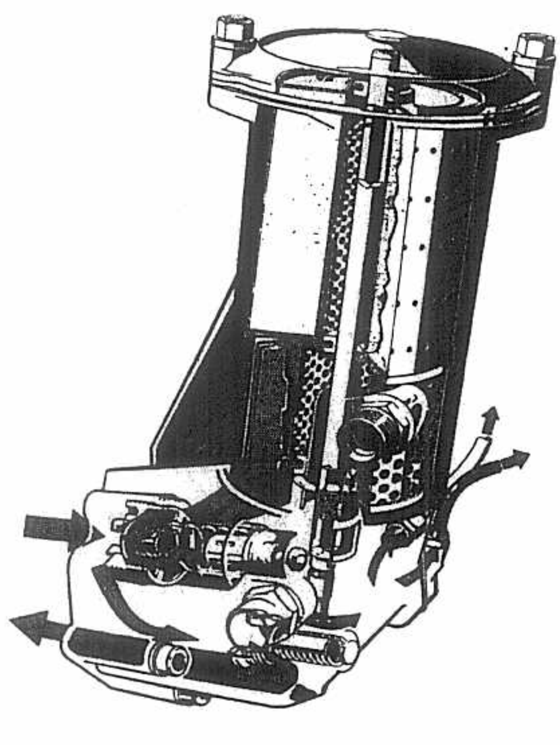

Fig. 25 - By-pass flow pattern in case of contaminated oil filter

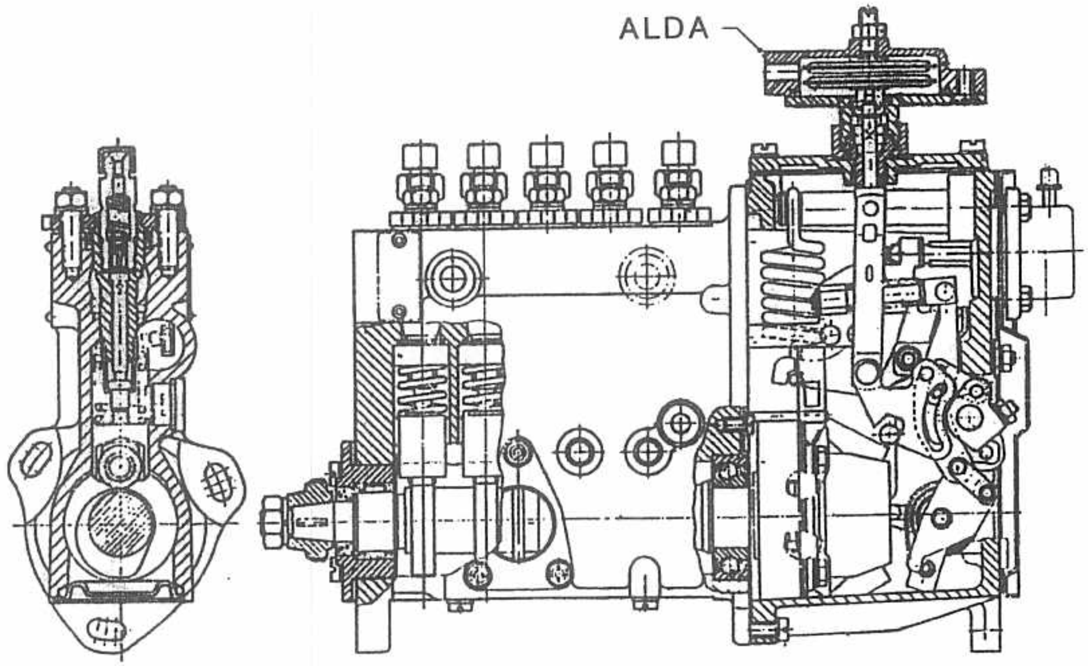

Fig. 26 - Bosch fuel injection pump with absolute pressure sensing system (ALDA)

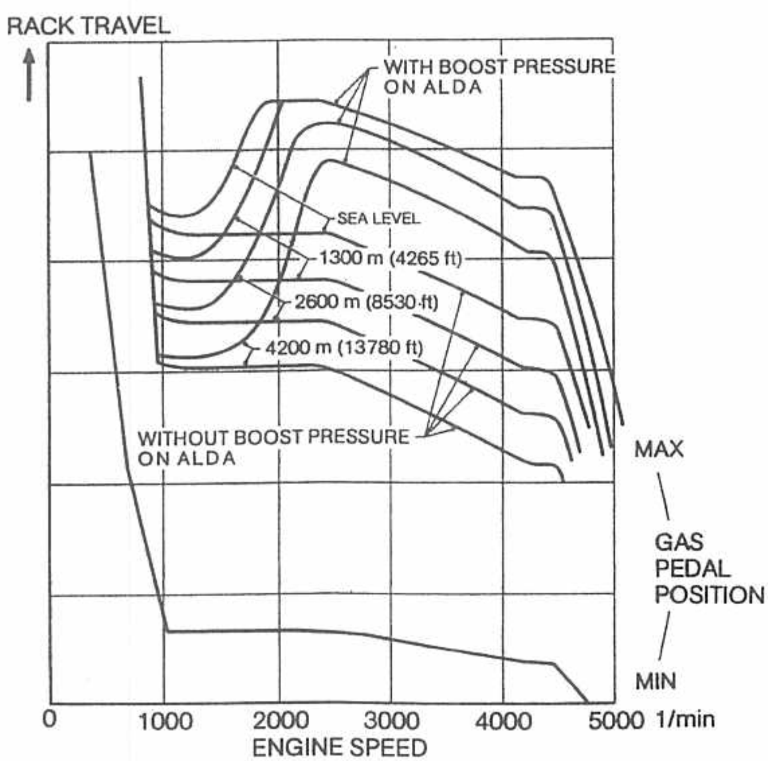

Fig. 27 - Operational characteristics of the mechanical governor with absolute pressure sensing system (ALDA)

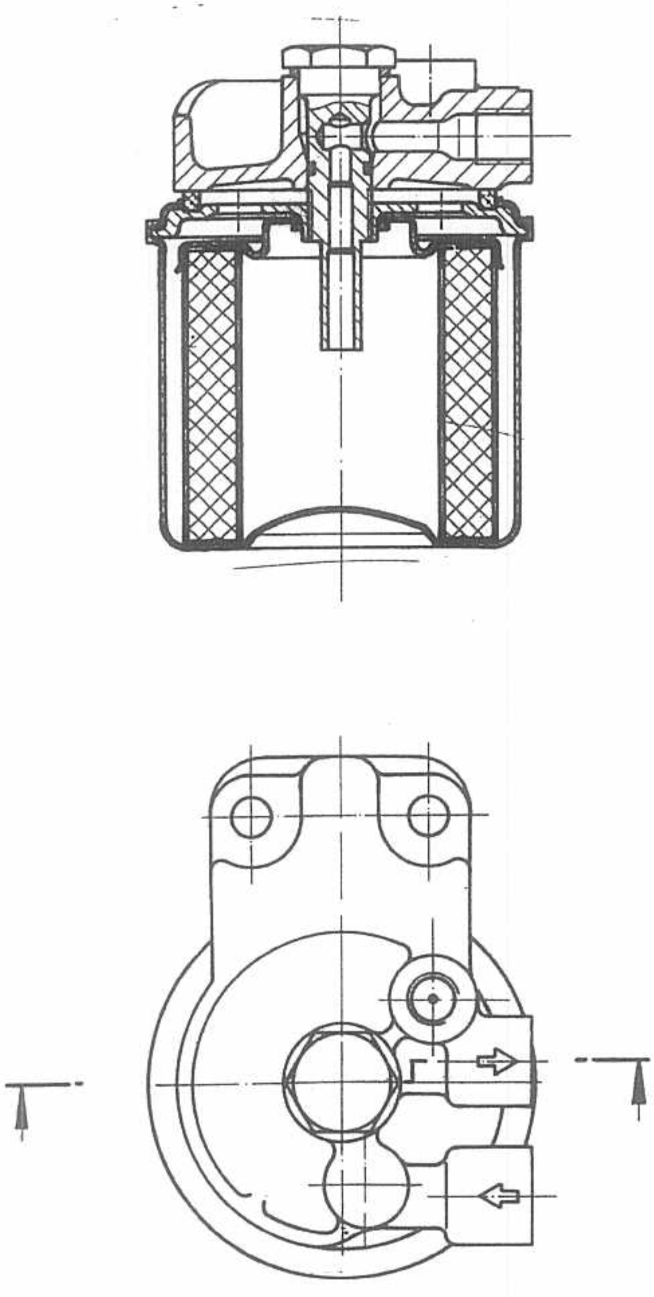

Fig. 28 - Fuel filter

the acceleration capability at all modes of operation, at all altitudes, without unacceptable smoke and gaseous exhaust emissions. Turbocharging can be regarded as yet another technology for reducing emissions of Diesel engines, provided that the necessary design precautions are taken. The cost implications are obvious.

FUEL SUPPLY SYSTEM - The injection pump drives the positive-displacement fuel pump which feeds the fuel via the fuel filter located at the highest point in the system into the injection pump. Pump inlet pressure and fuel flow rate are additionally controlled by means of a return-flow valve. The influence of filter contamination upon fuel feed is, therefore, low as compared with fuel systems in which the filter is located upstream of the fuel pump. As a result, excellent filter durability and low maintenance requirements are exhibited.

An important advantage of the in-line injection pump is its fairly low requirement for filtering, because it is not sensitive to water contained in the fuel, and thus does not require a water separator demanding frequent maintenance.

Fig. 28 depicts the fuel filter developed by Daimler-Benz. A vent bore in the support continuously returns fuel and possibly contained air bubbles back to the fuel tank. Since the central tapping point is located low, the fuel system is insensitive even to larger air quantities such as occur with an almost empty fuel tank, and the cartridge can be changed easily with only one screw. During this process, hardly any fuel is lost, because the support above the cartridge contains only a minimal volume of fuel.

INJECTION PUMP DRIVE - All injection pumps of Mercedes-Benz Diesel engines are driven by means of a timing chain via the injection timing control unit and an auxiliary shaft, as is depicted by Fig. 29. In case of the NA-engines, the speed-proportional timing unit is subjected to the drive moment of the oil pump in addition to that of the injection pump. In spite of the higher drive moments of the injection pump on the TC-engine the total load on this drive is reduced due to relocation of the oil pump. The timing mechanism could, therefore, be retained, only the angle of adjustment and the speed response characteristics had to be adapted.

The auxiliary shaft for the injection pump also drives (and lubricates) the vacuum pump required for the vehicle braking system. For the higher weight of the 300 SD vehicle, the performance of the membrane-type standard pump was not sufficient. Additional vacuum consumers further increased the performance requirements. Therefore, a new twin-membrane pump was developed from the well-proven standard components, one half of which primarily serves the braking system and the other one the reamining servo functions. In case of a leakage in the vehicle vacuum system, a valve separates it from the vacuum branch for the brakes.

INJECTION NOZZLES - In order to cope with the higher combustion chamber pressures, the pintle nozzles were modified for a higher injection pressure of 140 bar. Thermal and mechanical loads occurring due to turbocharging are altogether only marginally higher and could be handled without problems.

An important step in the development of nozzles was the introduction of a longitudinal bore in the center of the pintle, which is connected to a transverse bore, as seen in Fig. 30. This so-called CHIP nozzle (CHIP = Central Hole in Pintle) ensures a much more stable flow of the injected fuel at low needle lifts, i.e. in the early injection phase, because this injec-

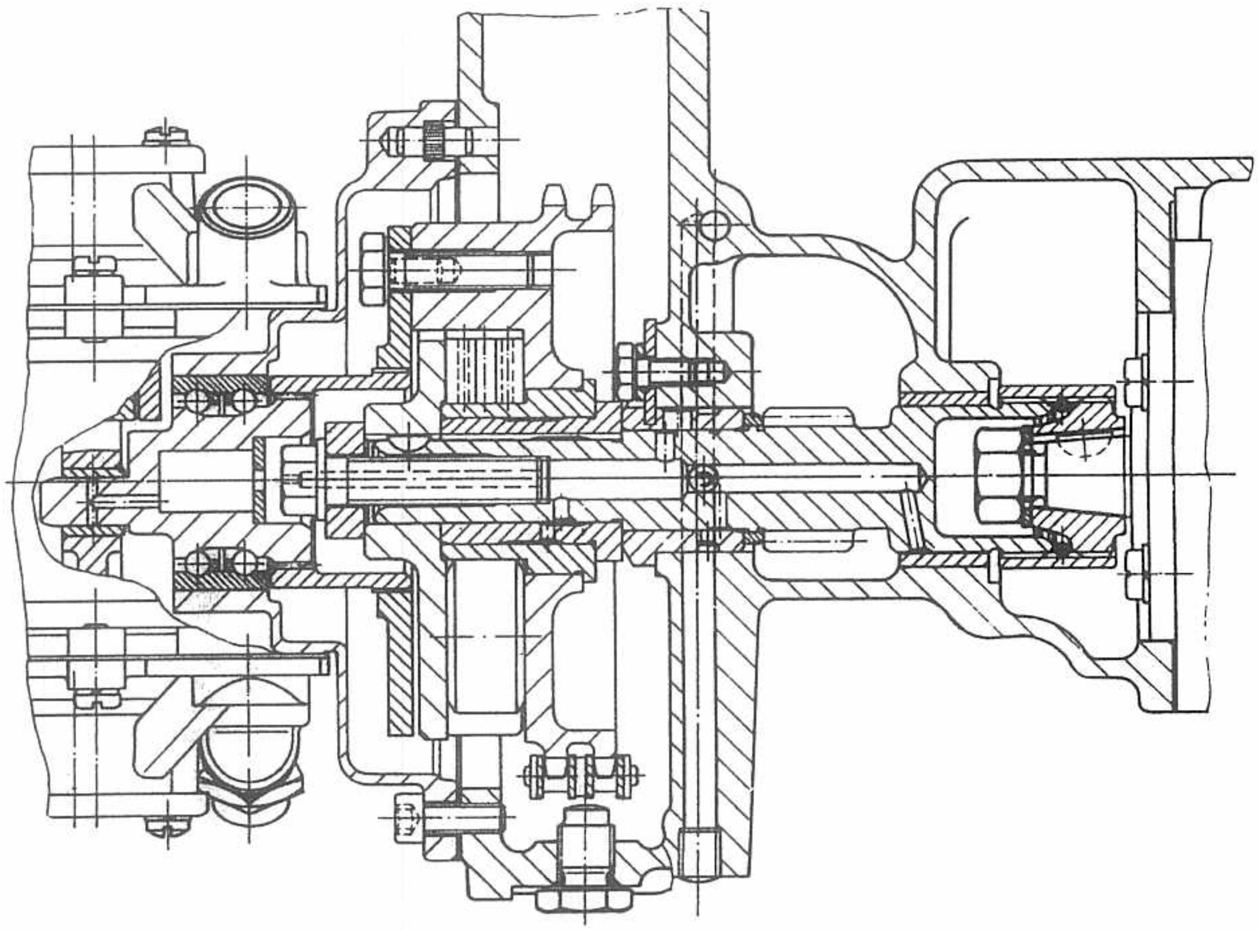

Fig. 29 - Injection pump drive (right) with injection timing control unit (center) and vacuum pump drive (left)

tion geometry is less prone to coking than the original design with only the usual annular slot around the pintle.

The CHIP nozzle thus helps to eliminate the pinging noises after cold start which are caused by a completely clogged-up annular slot. The small central bore requires a high degree of manufacturing precision.

ENGINE OVERLOAD PROTECTION - The design features of the injection system permit the design of a simple system for overload protection. In case of a malfunctioning wastegate control of the turbocharger, boost would increase sizably, without any significant change in the behavior of the engine. After an extended period of time engine damages could occur. Thus an overload protection becomes a necessity, if a reliability of 100 % is to be guaranteed, because malfunctioning of the wastegate cannot be completely excluded.

In case of a significant but still uncritical excess of the normal maximum boost a pressure switch sensing the boost pressure, as shown in Fig. 31, activates a three-way valve located in the connecting line to the aneroid of the ALDA, venting this line to atmosphere to that

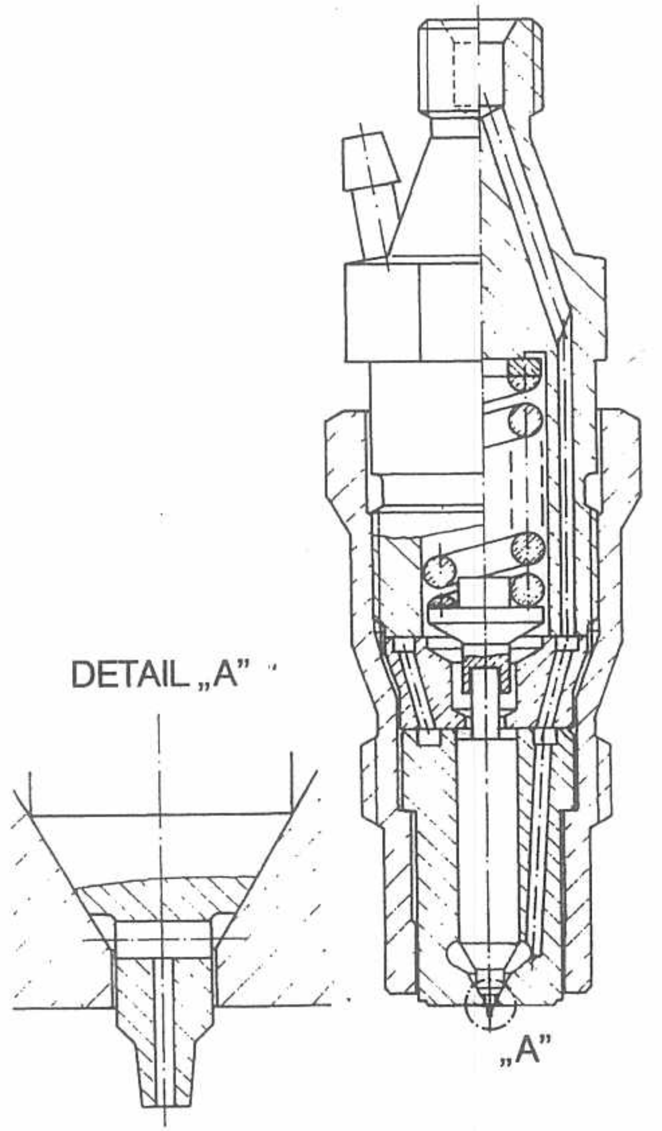

Fig. 30 - Bosch injection nozzle with central hole in pintle (CHIP-nozzle)

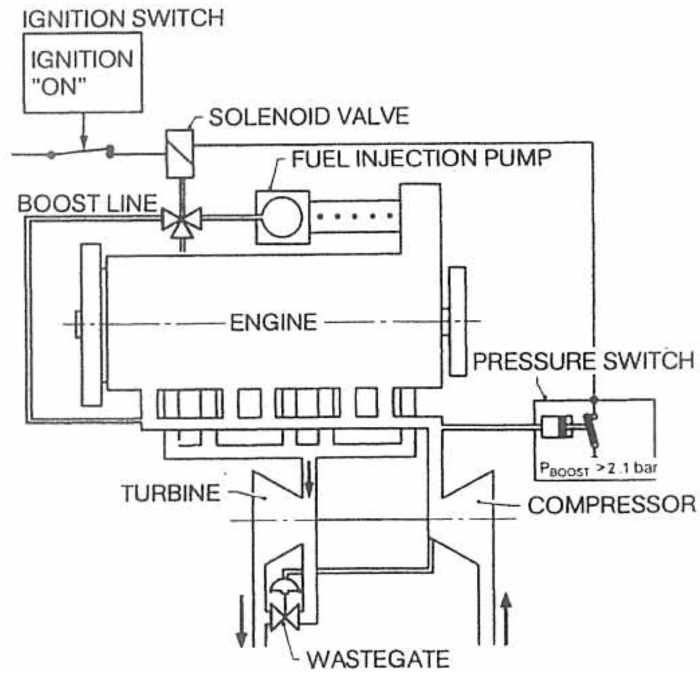

Fig. 31 - Functional principle of engine overload protection system

the injection quantity is reduced. Since the driving performance of the vehicle is then clearly lower but still absolutely safe the driver is thus encouraged to seek engine service.

TURBOCHARGER

Tests with conventional turbochargers used on heavy duty Diesel engines revealed that a relatively large turbine housing has to be selected if acceptable maximum boost pressures are to be achieved in the upper speed range. Due to the much wider speed range of passenger car Diesel engines, however, the resulting insufficient boost pressures in the lower speed range even under stationary operating conditions and slow turbine and engine response to load change are by no means satisfactory.

Prior to commencement of the final engine development an agreement for a suitable turbocharger unit was reached with Garrett AiResearch Industrial Div. They had achieved considerable progress in the following areas:

- improvement of compressor efficiency in the relevant, wide flow ranges, among other things through utilization of backward curved blades,
- application of high-quality manufacturing technologies for economical production of small turbochargers with sufficient efficiency,
- reduction of the moment of inertia of rotating masses utilizing small wheel diameters in order to improve transient behavior, and
- technical command of a wastegate control for the boost pressure on the turbine side.

These features were of decisive importance in providing the operational characteristics of the TC-engine.

LAYOUT OF TURBOCHARGER - The adaptation of such a turbocharger, as seen in Fig. 32, permits use of a turbine housing of about half the traditional size, so that maximum boost is achieved at an engine speed of 2000 rpm. At higher engine speeds, boost is maintained at a constant pressure level by the exhaust gas turbine by-pass valve, i.e. the wastegate. The level of boost as a function of engine speed can be influenced

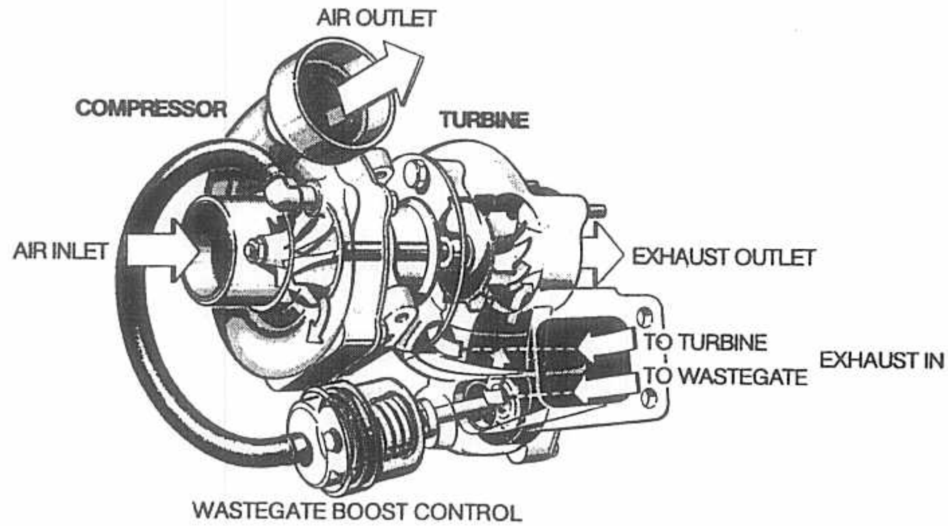

Fig. 32 - Garrett turbocharger with integral wastegate control

through corresponding tuning of the wastegate valve lift characteristics and of the by-pass channel.

The wide range of good operating efficiency of the compressor permits a favorable match to the corresponding engine air flow rates, so that under most operating conditions, i.e. at engine speeds above 1600 rpm, compressor efficiencies are between 65 to 74 %, as is indicated in Fig. 33.

The lower limit in the selection of the turbine housing size is given by the continuously increasing pumping losses at higher engine speeds. For passenger car engines, good specific fuel consumption is particularly important in the medium speed range, and it appears that the suitable turbine size has to be selected considering optimum performance at medium speeds and loads.

The advantages of this specific type of control are the effectiveness of the turbocharger at low engine speeds, the limitation of the boost pressure, and, with it, also of the boost temperature. Peak engine combustion pressures and the increase in thermal and mechanical loads are thus limited. Transient response is considerably improved as compared to an uncontrolled turbocharger. Response in fact is practically immediate at medium and higher engine speeds, where boost is already available at partial loads.

Command of a by-pass control at the turbine side of the turbocharger requires specific precautions because the control components are located in the proximity of the hot turbine housing. The diaphragm which is subjected to the boost is scavenged and cooled on its low pressure side by means of a defined flow of air

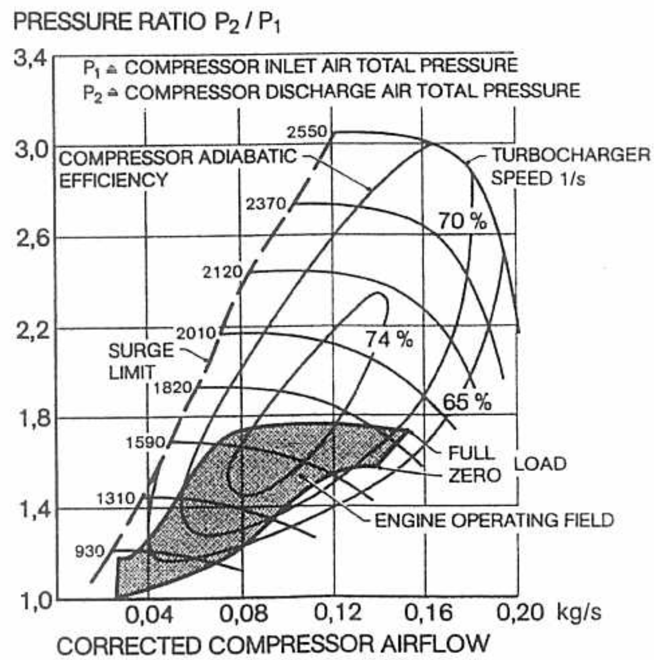

Fig. 33 - Engine operating field shown within compressor characteristics

guided through the valve stem. Air and exhaust gas leaking along the valve stem as a result of the prevailing pressure differences are vented downstream of the turbine by means of a respective venting channel, as seen in Fig. 34.

INSTALLATION OF THE TURBOCHARGER - The arrangement and location of the turbocharger were designed under consideration of the Mercedes-Benz vehicles. In terms of space, weight, and function a favorable solution was found because

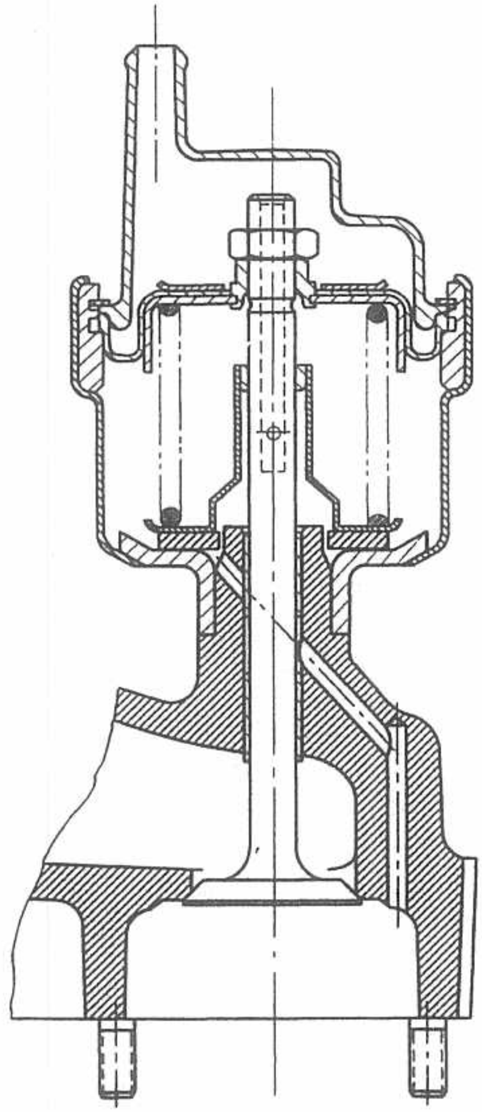

Fig. 34 - Wastegate with exhaust gas vent and air cooling

the inlet and exhaust ports of the engine are located on the same side of the cylinder head, as seen in Fig. 35.

Due to the odd number of cylinders and the utilization of the wastegate control only a single-scroll turbine housing could be used. Tests showed that a single exhaust pipe leading into the turbine housing presented no disadvantages as compared to a dual exhaust. In fact, the utilized design facilitates a volume between exhaust valves and turbine as small as possible, which is advantageous in terms of response.

The turbocharger is rigidly flanged to about the center of the exhaust manifold. Sealing is performed with a temperature-resistant steel seal with creases. Form and material of the housings carrying exhaust gas are optimized with regard to low deformation and durability. This was obtained with Niresist material. The temperature sensitive wastegate was positioned in a cool section of the engine compartment, as far down as possible.

All other connections are flexible. With this concept, no undesirable stresses due to assembly tolerances and displacements on account of different thermal expansions can occur. This arrangement is optimal even in terms of noise, because sound cannot be conducted between the components.

Air and exhaust gas flows in the following fashion: Ambient air from outside the engine compartment is aspirated in the region of the head lights of the vehicle. The intake funnel is connected to the air cleaner by means of a corrugated plastic tube which can account for tolerances and engine deflections. The paper-cartridge air cleaner, the size of which was adapted to the higher flow rates, is attached

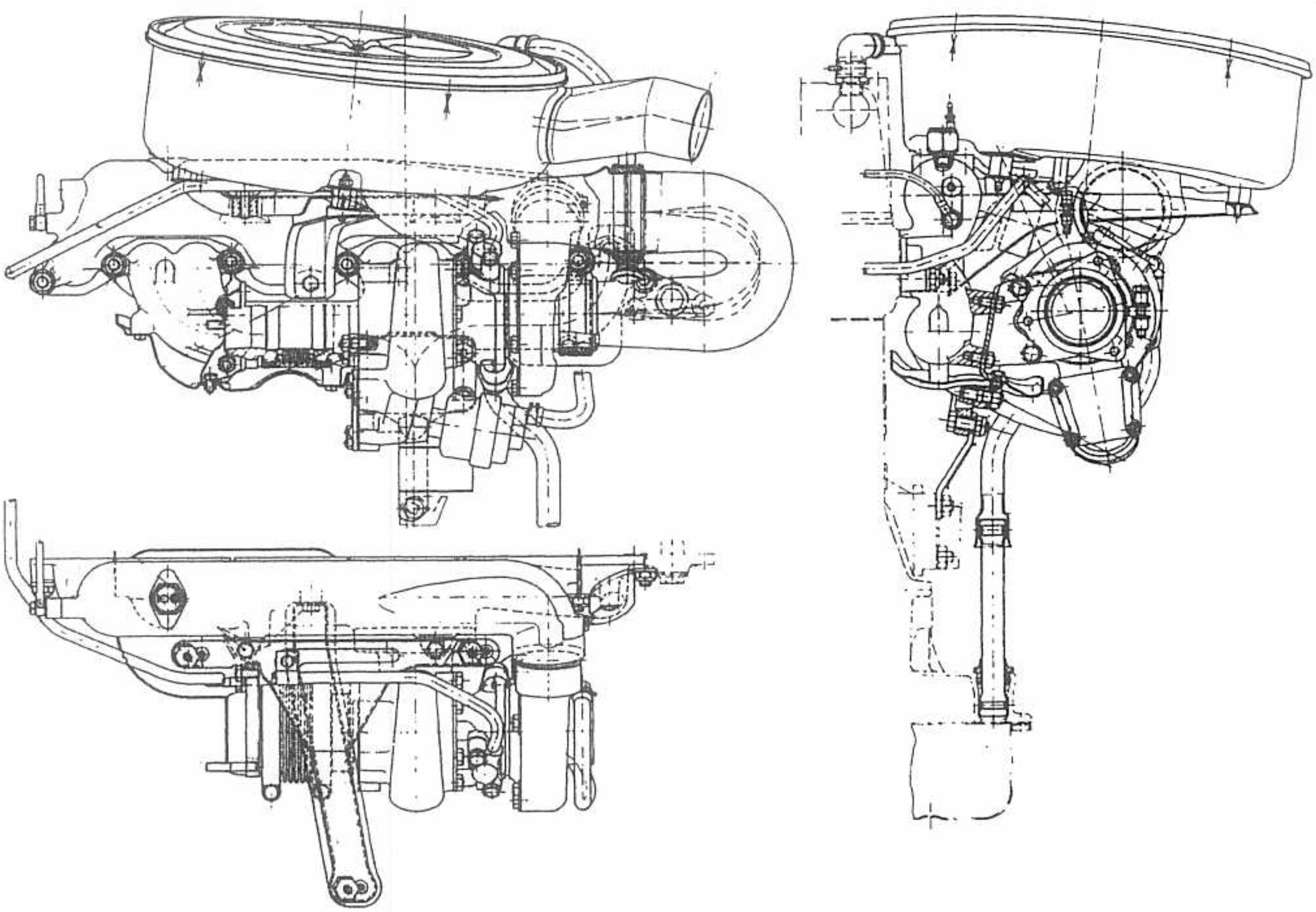

Fig. 35 - Turbocharger installation

to a sheet steel support by means of three rubber buffers. Following the cleaner, the air is guided through a funnel-shaped duct.

For its connection with the inlet of the compressor, a flexible plastic pipe is utilized which narrows toward the compressor like a diffuser. It carries temperature-resistant rubber rings at both ends for perfect sealing. The vacuum pump is connected to the exit of the air cleaner and, a little further downstream, the crankcase ventilation gases are fed in. Both of these pipes are sealed and supported by means of rubber. Downstream of the compressor, the compressed hot air together with the ventilation gases are forced through a diffuser-shaped aluminum connector supported in large O-rings into the boost manifold, from where it is fed into the individual cylinders.

Inside the turbine housing, the by-pass of the wastegate control is branched off and refed into the main flow. The front end of the vehicle exhaust system is attached to the exhaust manifold by means of a flange. This flange and the turbine housing are connected by means of a corrugated metallic pipe with an internal heat-resistant pipe for improved flow and temperature-shielding purposes.

Similar care was used in designing the lubricating lines for the turbocharger, because in case of breakage a complete engine failure would have to be expected on account of the high oil flow rate. Due to the high internal and external temperatures rubber hoses would not be suitable. Instead, steel pipes are applied. The oil supply line to the turbocharger is so designed that it can account for any movement of the turbocharger within its own elasticity. Since the return-flow pipe with its larger diameter had to be as short and vertical as possible, this design principle could not be used there. It was, therefore, separated into two pieces, the lower one of which is supported by two O-rings of large volume.

ENGINE ACCESSORIES

One of the important advantages of the Diesel engine as compared to the Otto engine is the stability of the combustion system over extended mileages and the resulting low maintenance re-

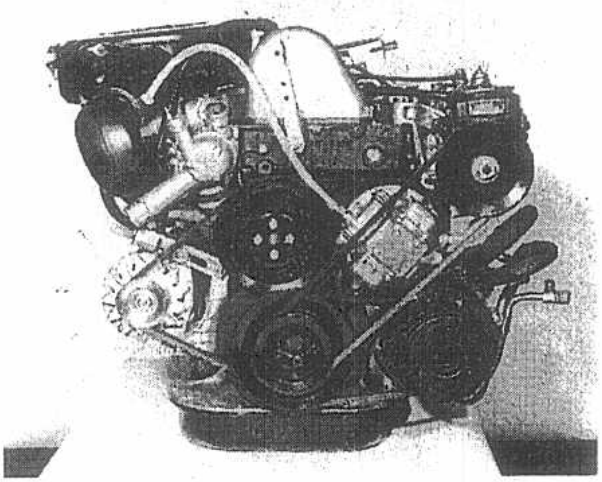

Fig. 36 - View on engine front (fan removed)

quirements. To further improve these positive features for the benefit of the customer, development efforts were directed towards minimizing and simplifying other necessary maintenances. Among these, for example, were the V-belts driving the accessories.

ALTERNATOR - As shown in Fig. 36, all MB Diesel engines have a triangular V-belt drive for alternator, water pump, and fan. In recent years, this belt drive was continuously burdened by increasing alternator loads. For optimum generation of electricity, the drive ratio should be as high as possible. If this is realized by means of a small pulley diameter on the alternator, durability is further reduced.

Both theoretical calculations and tests have shown that durability of the V-belts is mainly influenced by the smallest pulley diameter and by the tension in the belt. Based upon these criteria the accessory drive was, therefore, designed anew. The effective pulley diameter of the alternator was increased from 66 to 71 mm. A larger angle of grip on the crankshaft pulley could be obtained by raising the alternator. Together with the largest possible diameter of the crankshaft pulley and under consideration of the necessary fan drive ratio of 0.9, this resulted in an alternator drive ratio of 2.17. Vehicle durability tests verified the expected increase in reliability based upon this layout, utilizing two belts of 9.5 mm width.

A/C-COMPRESSOR - The former A/C-compressor model YORK A 210 had to give way to the turbocharger. At the same time, the slightly larger interior volume of the vehicle 300 SD also required increased cooling performance. Additionally, durability of the respective belt drive was to be increased.

Selection of a compressor model DELCO R4 which suited the installation conditions and which can operate at 1.4 times the engine speed resulted in a weight savings of 4.5 kg. The layout can also be seen in Fig. 36.

Instead of providing a tensioning pulley which would reduce the respective angles of grip, the compressor itself can now be swivelled. The attached coolant pipes follow the swivelling motion. Measurements of slip and durability tests, again, verified good performance and longevity for the new design.

POWER STEERING PUMP - The power steering pump with its integrated hydraulic fluid container had to be relocated axially only to match the new operationg plane of the belt drive. With its 180° angle of grip and a large distance between the axes this drive meets all requirements.

In general, great emphasis was laid upon good accessibility and operational simplicity. An important principle here is the separation of the functions of belt-tensioning and accessory attachment. Thereby the respective operations can be conducted independently from one another, while maintaining a defined tension in the belts.

ENGINE COOLING SYSTEM

The cooling system of all MB Diesel engines is equipped with a thermostat located at the entrance of the engine. During the warm-up phase of the engine, when the radiator is by-passed and particularly when the engine is shut-off frequently, this design avoids a possible opening of the thermostat on account of a temperature rise after the engine has been stopped, in which case cold coolant from the radiator could flow into the engine. This would result in a corresponding cooling of the exhaust gas contained in the cylinders to below their point of condensation, so that the sulfur contained in the fuel would lead to the formation of aggressive acids which markedly increase cylinder wear.

Due to the given relatively high position of the thermostat special precautions for the ventilation of the pipe from the bottom of the radiator to the thermostat during initial filling of the system have to be taken, since otherwise not enough coolant is supplied, and the coolant circuit would break down, see Fig. 36. While hitherto during the filling operation a vent screw at the top of the thermostat had to be opened, the housing of the thermostat of the OM 617 A has an integrated venting passage which makes filling simple and fool-proof, Fig. 37. The exact form of this passage was determined in tests. Although it bypasses the thermostatic valve when this is closed, the corresponding flow rate is so low that its influence on engine warm-up is not measurable.

ENGINE AND VEHICLE DATA

ENGINE DATA - Fig. 38 depicts the comparison of engine power and engine torque between the NA-engine and the TC-version. Maximum torque

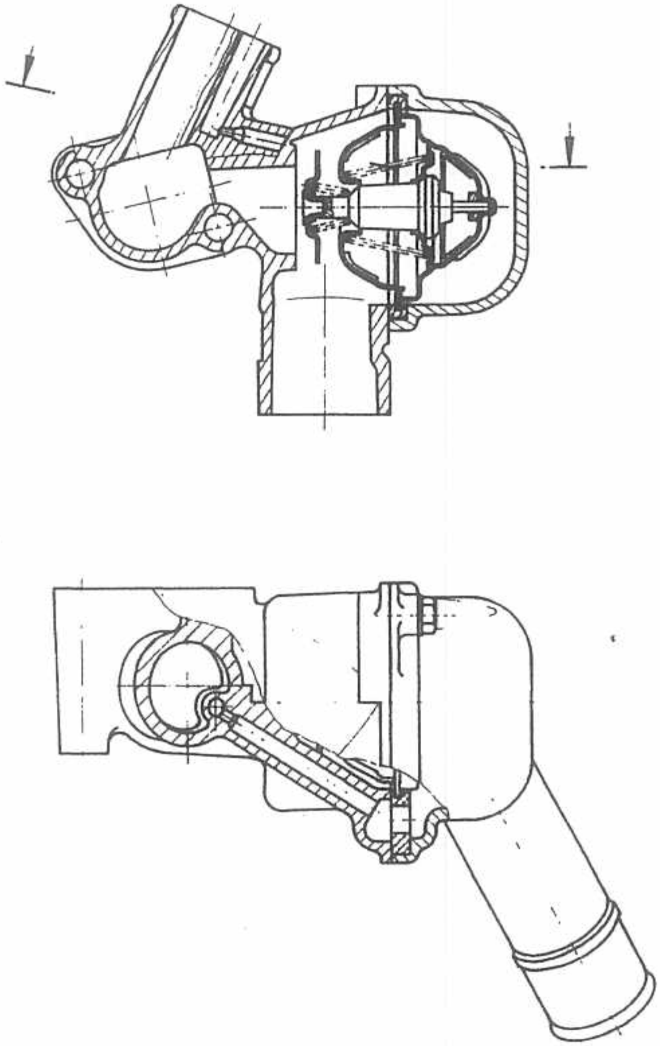

Fig. 37 - Coolant thermostat

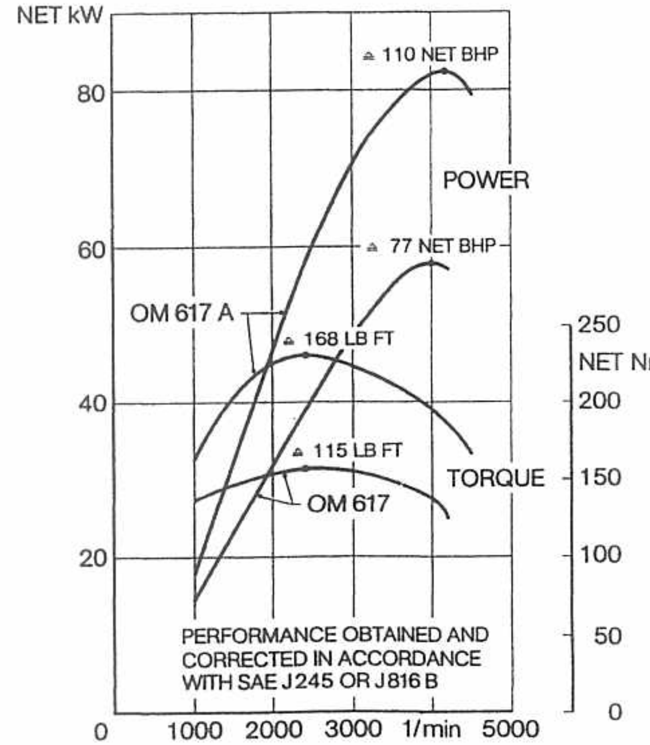

Fig. 38 - Engine performance comparing NA-engine OM 617 with TC-engine OM 617 A

of the OM 617 A is 46 % higher and occurs at the same engine speed of 2400 rpm, the performance is increased by 43 %. The precise tuning of the turbocharger with its wastegate control is demonstrated by the torque characteristics.

Fig. 39 plots the main engine- and turbocharger-data under full load operation:

- Low exhaust opacity is achieved by means of tuning the boost pressure to a thermodynamically favorable air excess ratio.
- The maximum BMEP of 10 bar at 2400 rpm engine speed with a specific fuel consumption of 260 g/kWh are both good values for a Diesel engine of this size.
- The maximum boost pressure amounts to 1.7 bar, while the compressor discharge air temperature does not go beyond 100^{o} C which is favorable in terms of mechanical and thermal stresses as well as NO_x emissions.

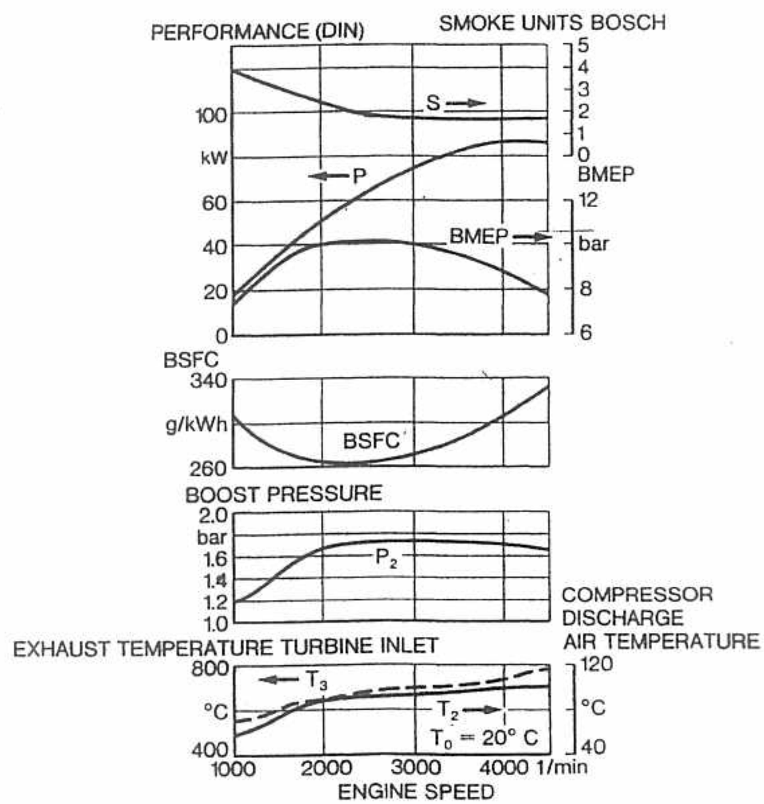

Fig. 39 - Engine and turbocharger characteristics at full load

- Gas temperatures at the turbine inlet reach a maximum of less than 800° C, a value uncritical for the usual turbine wheel materials.
- The fuel consumption map, shown in Fig. 40, with its best value at 245 g/kWh, and its wide extension of lines representing constant consumption over a broad operating range is particularly favorable. This was achieved by means of the described careful adaptation of the turbocharger.

A differential fuel consumption map for the NA- and the TC-engine, Fig. 41, reveals that the turbocharged engine has a fuel consumption better by a maximum of over 20 g/kWh. This is mainly due to improvement of the thermodynamic efficiency, higher mechanical efficiency, and combustion improvements achieved through design refinements of the prechamber.

VEHICLE DATA - As Fig. 42 depicts, considerable progress has been achieved in terms of the time and distance required for vehicle acceleration from 0 to 60 mph, from the 240 D with its 4-cylinder NA-engine to the 300 D with its 5-cylinder NA-engine and the 300 SD with its 5-cylinder TC-engine.

The comparison of vehicle passing capability shows similar results and is given here for a vehicle acceleration from 30 to 60 mph, with the automatic transmission kept in direct gear.

Fig. 43 depicts the considerable progress in vehicle acceleration comparing all Diesel cars ever sold in the USA.

TRANSIENT BEHAVIOR OF TURBOCHARGER - Upon maximum acceleration from a standing start with a vehicle with maximum pay-load transient behavior of the turbocharger is shown in Fig. 44. The maximum boost pressure is reached after only about 3 seconds. This is a large improvement as compared to an uncontrolled turbocharger with equal maximum boost pressure.

When the vehicle is accelerated from other speeds, response of the turbocharger is much faster or practically immediate, since a certain boost pressure level is already maintained prior to the acceleration, as was shown in Fig. 33.

Apart from this, almost the maximum torque is already available at lower boost pressures due to the described design of the injection system.

FUEL ECONOMY - Fuel economy at constant vehicle speeds and level road, shown in Fig. 45 for the three Diesel vehicles 240 D/300 D/300 SD, displays the reputed superior economy of Diesel vehicles, particularly at low vehicle speeds.

Compared to the 240 D, the fuel economy of the 300 D is only slightly inferior, since the greater number of cylinders for identical vehicle weight results in a lower BMEP, and thus in higher specific fuel consumption. This effect is minimized by installing a rear axle ratio of 3.46 as compared to 3.69 of the 240 D.

The advantage of turbocharging is exemplified by the fuel economy of the 300 SD since the

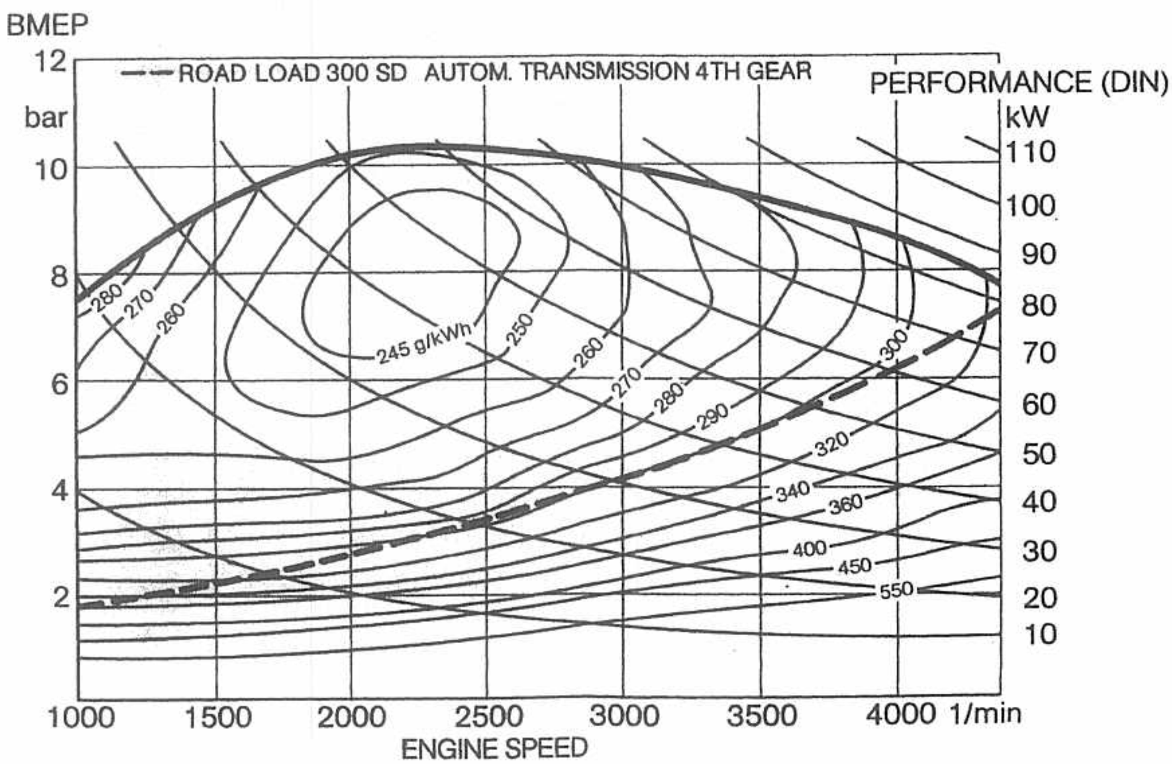

Fig. 40 - Fuel consumption map

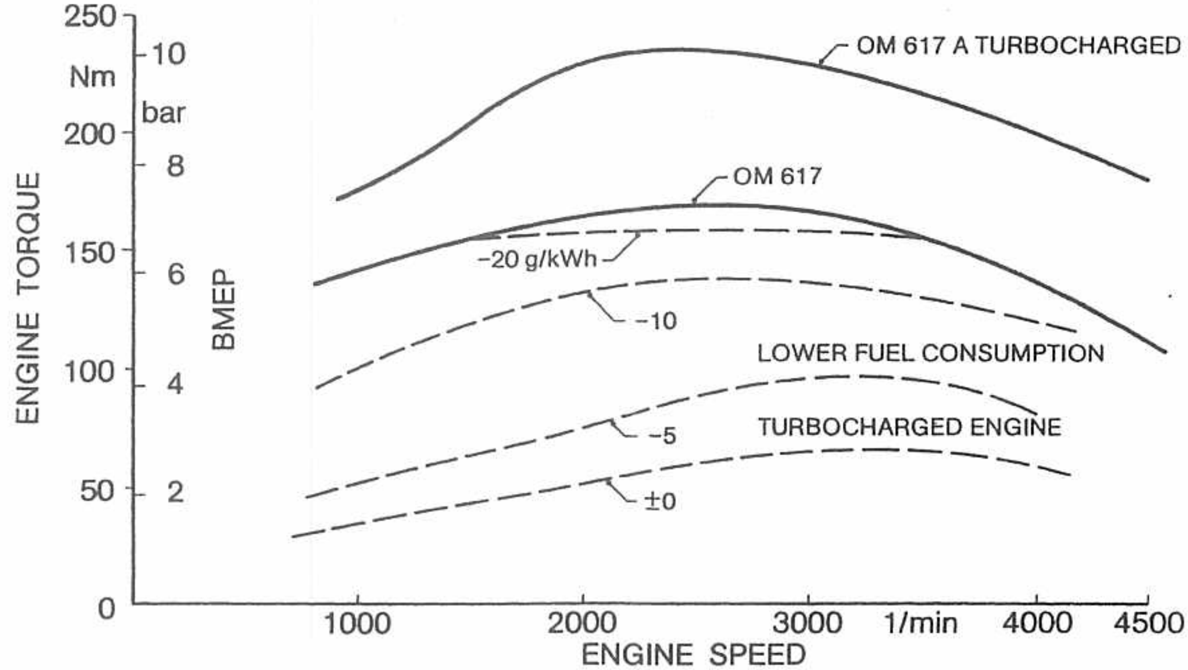

Fig. 41 - Differential fuel consumption map showing consumption differences between NA- and TC-engine

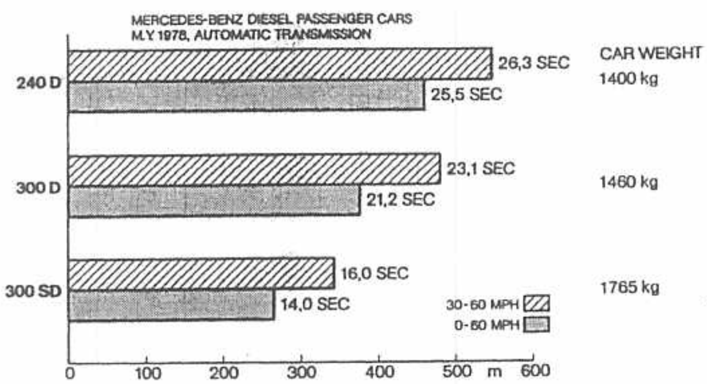

Fig. 42 - Vehicle accelerating capability (0 - 60 mph and 30 - 60 mph)

engine in connection with a rear axle ratio of 3.06 is operated in a range of better specific fuel consumption. This advantage becomes particularly apparent if a 300 SD with an NA-engine of equivalent performance is considered, the fuel economy of which would lie lower than that of the 300 D.

EXHAUST EMISSIONS - A comparison of the exhaust emissions and fuel economy of the 300 D with the 300 SD, Fig. 46, shows that the HC- and CO-emissions of the turbocharged vehicle are considerably lower, while the 1978 certification for a NO_x-Standard of 2.0 gpm was just possible. This verifies that the argument of a NO_x waiver

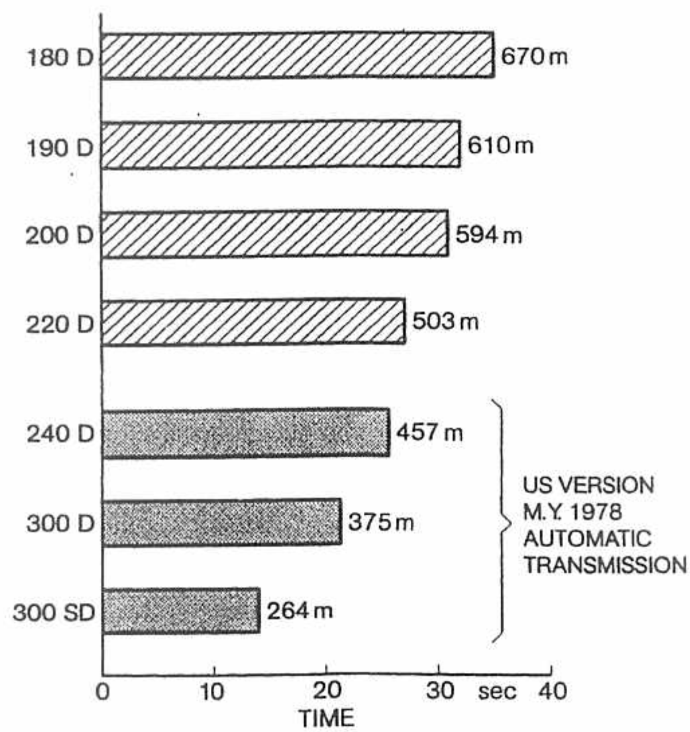

Fig. 43 - Historical survey of acceleration times and distances of Diesel cars (0 - 60 mph)

for Diesel cars carried forward with particular emphasis of the extremely low and consistent HC- and CO-emissions was viable.

The combined fuel economy of the 300 SD family-size sedan is 26 MPG, an improvement of 6 % over the 300 D.

While Mercedes-Benz Diesels equipped with NA-engines already today comply with the Statutory HC- and CO-Standards, and TC-engines do so easily, a TC-Diesel engine of identical displacement has a greater problem in meeting the NO_x-Standard. NO_x emissions of Diesel cars mainly depend upon vehicle weight. Future NO_x-Standards, however, cannot be met by merely reducing vehicle weight. New technologies have to be developed. Their influence on the characteristics, operating features, and lastly reputation of the Diesel passenger car is, at the time, not quite clear.

NOISE LEVEL - For a critical examination of the interior noise level of an automobile a lot of factors have to be considered, and it is by no means satisfactory to measure overall-noise level in dB(A).

One of the important criteria is the pattern of the frequency bands. Fig. 47 shows the octave band spectrum from 125 to 1000 Hz at three frequently used vehicle speeds, for various vehicles:

30 mph = speed limit in European cities
55 mph = speed limit on US-highways
80 mph = speed limit on many European highways.

It is interesting to see that, especially at high car speeds, there is no significant difference between Mercedes-Benz Diesels and a 4-cylinder gasoline engine with similar power. At all speeds, octave band levels decrease uniformly and moderately with higher frequencies. Also, there are no dominating frequency peaks at any vehicle or engine speed.

One of the reasons for this excellent noise behavior is the extremely low noise radiation of the engine. The combustion system, many refinements and a long experience in development of Diesels for passenger cars are responsible for this success.

A special problem of the 5-cylinder-engine is the beat of 2nd and 2.5th order, which on the new TC-engine could be solved perfectly by very careful tuning even of engine-external components.

GENERAL OUTLOOK

Since the energy crisis in 1973 the importance of Diesel passenger cars due to their superior fuel economy has increased worldwide, as a result of which the share of Mercedes-Benz passenger cars equipped with Diesel engines as well as their absolute number has risen clearly, as shown in Fig. 48.

Even more important is the fact that the Diesel sales share of Mercedes-Benz vehicles in the USA has now reached a level similar to that in Europe. It is to be hoped that the Diesel acceptancy rate will increase further. At this moment, Diesel passenger cars offer a great number of advantages, especially those of altogether low and consistent exhaust emissions, extreme dependability, good driveability, and unparalleled fuel efficiency, particularly in a family-size sedan.

Further development work remains to be done to ensure the future of this power plant. However, whether a sizable dieselization of the US-passenger car fleet will take place or not, also depends on regulatory conditions, which should be based upon a comprehensive understanding of

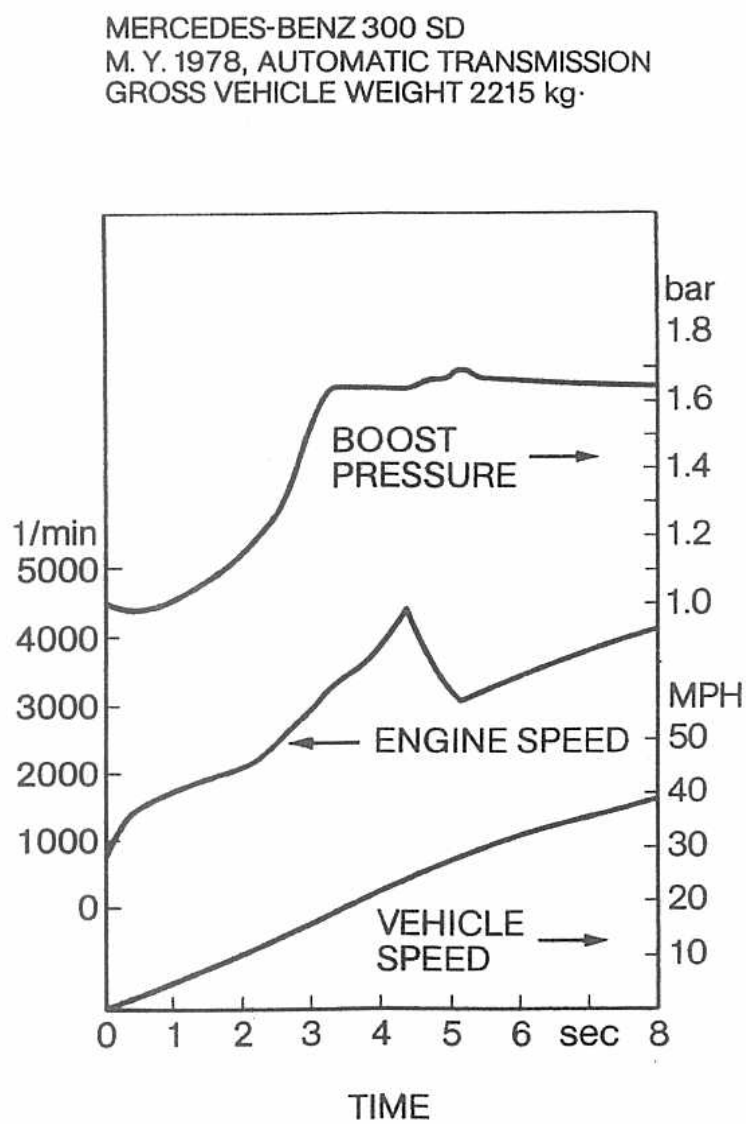

Fig. 44 - Transient behavior: full load acceleration from low speed idle

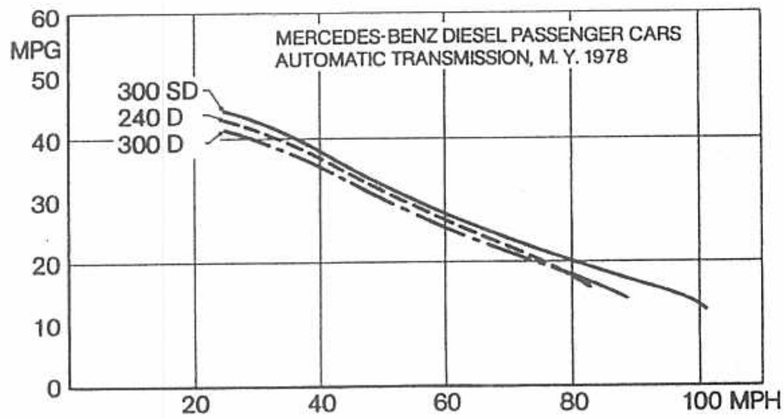

Fig. 45 - Comparison of fuel economy at road load

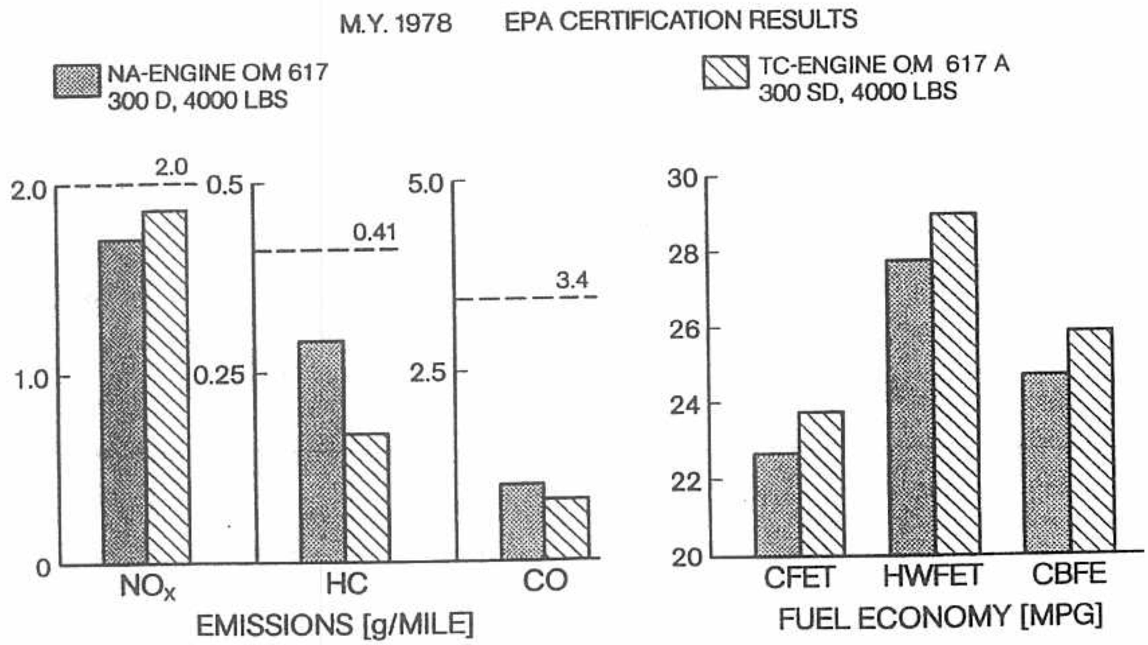

Fig. 46 - Comparison of emissions and fuel economy of 300 D with NA-engine OM 617 versus 300 SD with TC-engine OM 617 A

the Diesel's capabilities. The key to this is the unique combustion process.

The turbocharged 5-cylinder Diesel engine model OM 617 A as the most powerful of the Mercedes-Benz Diesel engine line and as the first turbocharged Diesel engine to be installed in a production passenger car is considered as another milestone in Diesel engine development. Rather than installing it into the body of the 240 D/300 D or that of the 300 CD it was decided to utilize the extra performance gained by means of turbocharging to propel the higher weight of the larger sedan, so that now three of four different body types are available in a Diesel version.

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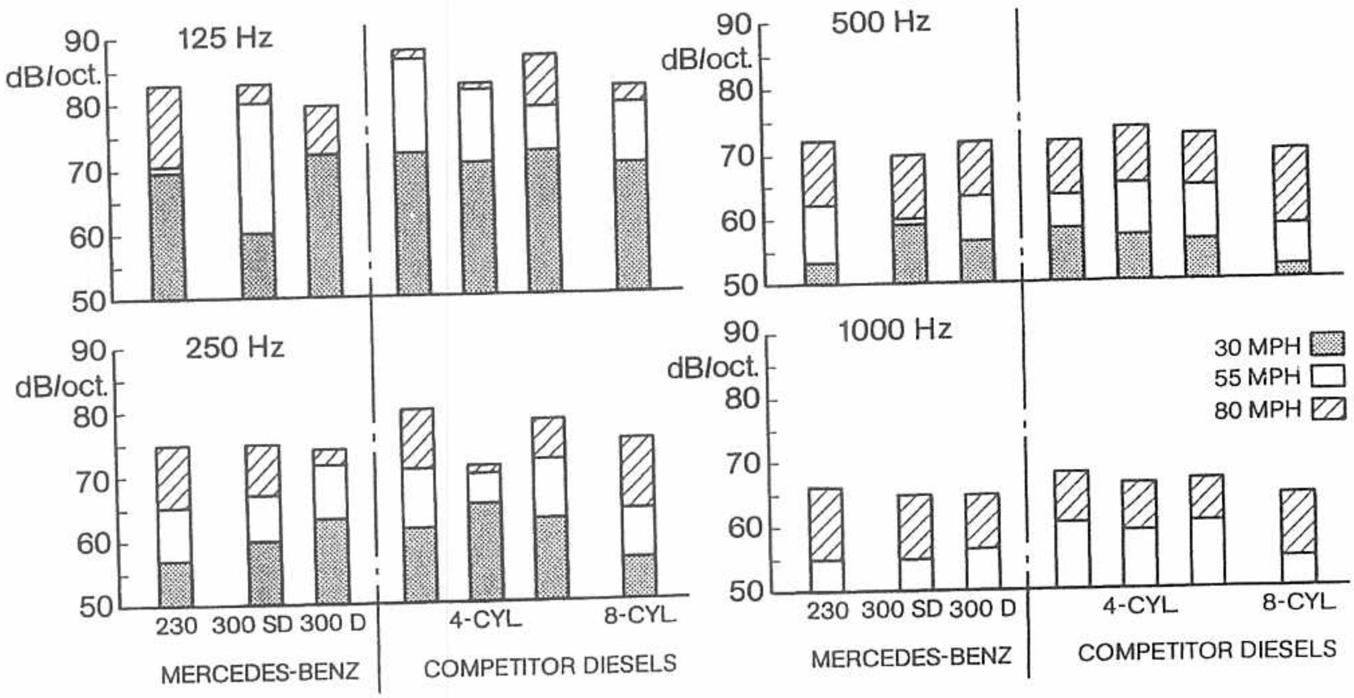

Fig. 47 - Interior noise level (octave band spectra) of various Diesel passenger cars

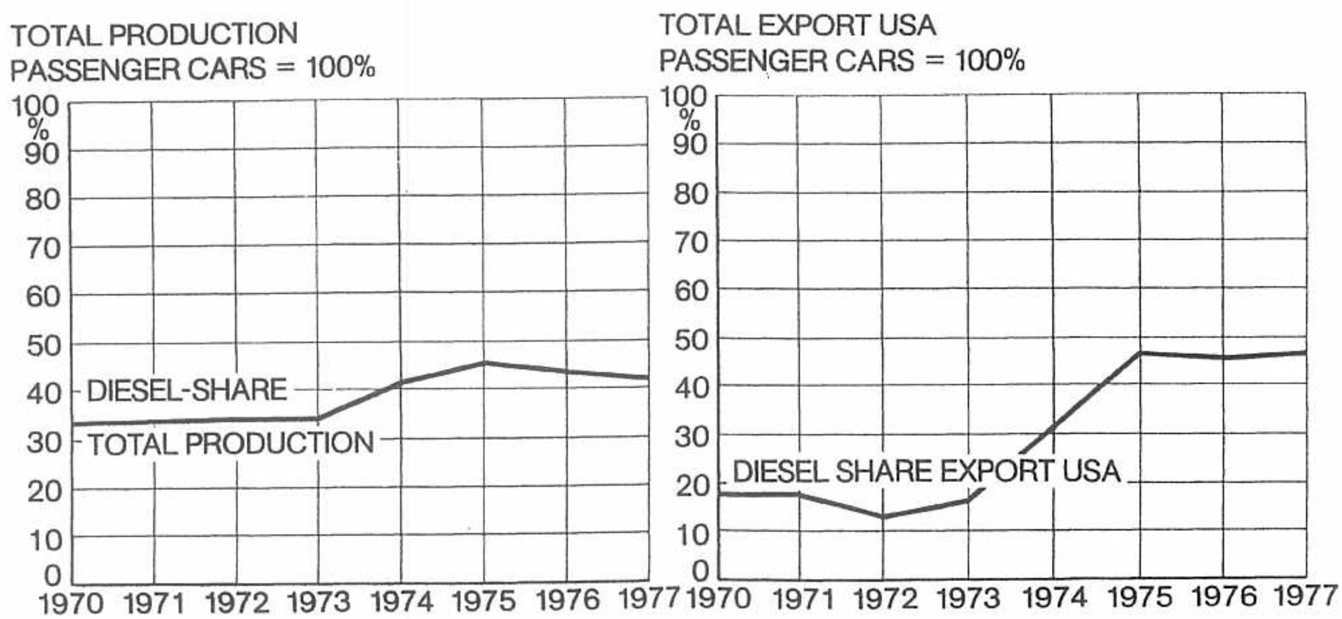

Fig. 48 - Mercedes-Benz Diesel share of passenger car production and exports to the USA

5-cylinder Diesel engine, and particularly the contribution of J. Friedlein and F. Thoma to this paper. They also thank P.v.Manteuffel for his assistance in preparing the English text.

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