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Volvo's MEP and PCP Engines: Combining Environmental Benefit with High Performance

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ABSTRACT

In two research programs, Volvo has investigated high performance turbocharged versions based on the new 3-litre inline six-cylinder naturally aspirated engine. Power and torque targets were 180 kW and 385 Nm respectively, with a wide usable torque range.

The MEP-(Methanol Environment Performance)-project was linked to alternative fuel studies and focused on methanol (M85) and Flexible Fuel Vehicle-(FFV)-development.

With alternative fuels , it is important to investigate not only the emissions and fuel efficiency, but also the performance potential, in particular when used in turbocharged engines. The MEP-engine could be reduced to 2.5 litre displacement, due to the good specific performance with M85 fuel. Higher charge pressures could be used compared to gasoline. An M85 turbocharged high performance engine must be designed for higher peak combustion pressures.

For the MEP-engine a series-sequential turbo system was developed, resulting in particularly good low end torque characteristics.

The PCP-(Power Concept Project)-program was focused on the development of a high torque, wide operating range engine version, with high fuel efficiency and low emissions, running on unleaded 95 RON gasoline.

The PCP version retained the 3-litre swept volume, and a parallel-(bi-turbo)-twinturbocharging system was developed for good overall efficiency.

Both the PCP and MEP engines easily attained 200 kW and 400 Nm. In practical use, both engines can be compared with much larger N/A versions in regard of usable transient performance.

The PCP-engine had nearly the same fuel efficiency as the N/A reference engine, showing only 2-3% increased fuel consumption in cycle testing.

Better efficiency and considerably more power could therefore be obtained with the MEP version, compared to the reference engine.

INTRODUCTION

Volvo Car Corporation has extensively investigated the areas of alternative fuels and supercharging technology. Exhaust gas turbocharging of both spark-ignition and diesel engines has been used for a number of years. This forced aspiration principle has proved successful both on ordinary passenger cars and on Group A competition vehicles, and has been subjected to continuous further development and refinement over the years.

Current engines, exemplified by the Volvo 940 2-litre 4V-turbo engine, have reached a very advanced state of development.

Novel systems, including boost and exhaust temperature control, have been developed and applied in series production. Despite this, there is still a lot more to be gained from improved turbocharging technology.

In the area of alternative fuels, comprehensive work has earlier been focused on methanol fuels, both in M100 and FFV-form. The possibility that M85 fuels might become an alternative fuel makes it worth while to investigate not only its emissions and efficiency possibilities, but also the performance capabilities, in particular together with turbocharging. This could result in smaller, more fuel efficient power units.

One of the most important areas of development for current engines with high specific output and torque, is the torque development at low engine speeds for a greater flexibility.

OBJECTIVE

Target performance data for the PCP and MEP engines were set at 180 kW of power and 385 Nm of torque. This represents a typical level for performance engines in the upper range of vehicles.

The baseline engine is the new VOLVO 3-litre 4-valve engine with maximum output of 150 kW and 267 Nm, corresponding to 51.3 kW/litre and 91.4 Nm/litre

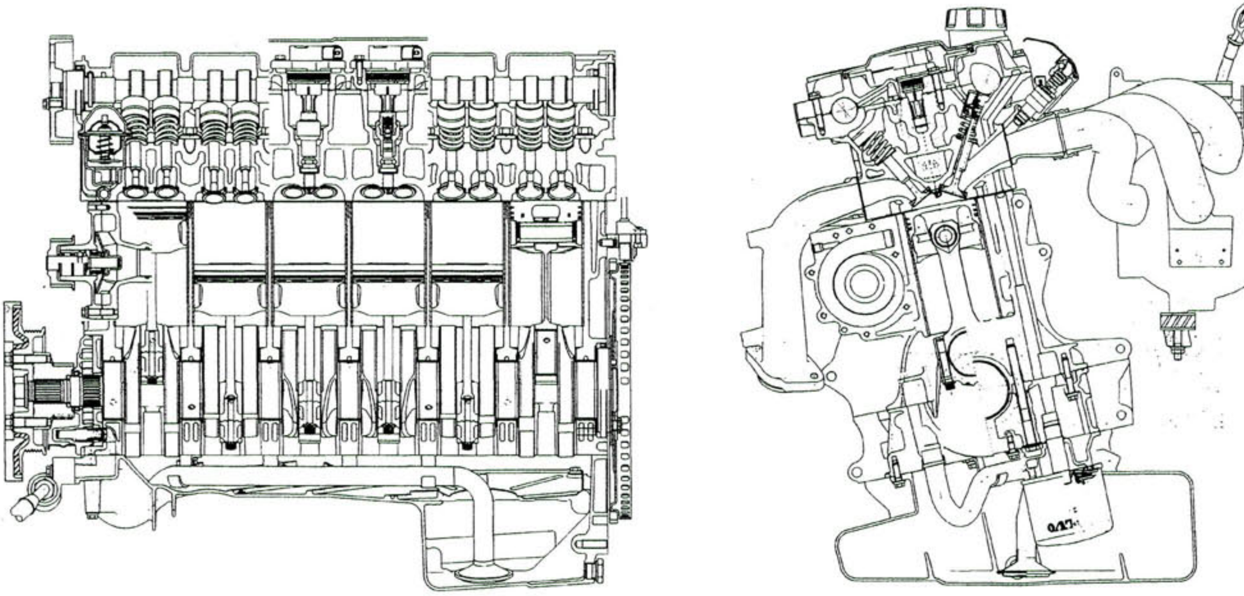

Fig 1 - Longitudinal and cross-section of the B6304F engine

respectively. See fig 1, appendix A, and (1)*.

If the targeted performance should be attained from this engine in N/A-form, the only possibility to reach the power target would mean a significant increase in engine speed. This would require a redesign of the engine and result in difficult compromises in regard of mechanical stresses, and force the use of low weight materials in order to avoid a considerable increase in vibration and noise levels.

All in all, development in this direction does not readily fall in line with the characteristics of an upper range high performance (N/A) engine.

With the use of supercharging, there are much better possibilities to retain the speed range and also to reach the targeted torque level.

The most effective technology for performance increase is the use of exhaust turbo supercharging. Since the mid seventies the use of turbochargers has expanded from a selected few high performance options from specialist car companies to a regular performance option in almost all manufacturers' production line-up.

Continuous development of the charging systems has meant that practical response time has been significantly improved, and with smaller electronically regulated wastegates, the basic problem with slow response at low engine speeds has been reduced to an acceptable level.

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* Numbers in parantheses designate references at end of paper

The fuel characteristics and quality has also slowly changed during the same time frame. The availability of higher grades in regard of octane number, has made the use of higher compression ratios possible. Together with recalibrated turbomatching, smaller turbochargers with less inertia and more optimized gas exhange and engine management systems, more performance has been achieved for a given fuel quality. The development of effective knock detecting systems and novel exhaust temperature sensing systems has made truly flexible octane fuel quality use possible.

With the recent interest in alternative fuels, exhaust emissions have been of major importance. In practical use the performance potential of a novel fuel is also of great interest, in particular if the specific output possibilities are different and could influence the engine sizing and other main engine parameters with significant importance for the fuel economy in a fully optimized vehicle.

In this study comparison was to be made of different fuels (gasoline and methanol), different compression ratios and possible boost pressures, for selected combinations of fuel mixtures (Gasoline to Methanol/M85).

SINGLE CYLINDER STUDY

A comprehensive single cylinder study preceeded the multicylinder development program. In the single cylinder rig boosted conditions could be created by appropriate intake and exhaust manifold conditions. A temperature controlled air charging system was arranged with a corresponding backpressurized exhaust manifold. This way turbocharged conditions could be realistically duplicated.

A pressure transducer in the cylinder head was used for pressure recording as well as for continuous heat release analysis.

The test engine had a cylinder head of 4-valve type. Further details of the experimental set up are given in appendix B.

TEST RESULTS SINGLE CYLINDER ENGINE - All tests were run with boosting parameters according to table 1.

N=1200, 2000, 2400, 3000, RPM N=4500, 5000 RPM MBT OR 3° CA BEFORE DBL		λ (WMMP) and λ=1 λ (WMMP) FULL LOAD
BOOST PRESSURE (BAR)	**CHARGE-AIR * TEMPERATURE (°C)**	**EXHAUST GAS BACK PRESSURE (BAR)**
1.0	30	1.020
1.2	40	1.210
1.4	47	1.420
1.6	55	1.630
1.8	62	1.840
2.0	68	2.050

*60% Charge air cooling assumed

Table 1 - Boosting parameters for single cylinder tests. WMMP (Weakest Mixture Maximum Power) settings.

Boost pressure up to 200 kpa (absolute) were tested in the speed range of 1000 - 5000 rpm. Combustion characteristics, pressure level, brake and indicated fuel consumption together with exhaust emissions and exhaust temperature were recorded. Knock was detected via the pressure traces and at threshold audible level.

Performance was compared with a target torque curve, corresponding to a level compatible with performance characteristics on a larger displacement naturally aspirated engine.

Boost pressures were tested in steps of 20 kpa from N/A conditions. Fuelling, ignition and timing were selected for optimum torque or with a margin of three degrees to the knock borderline.

Results for gasoline and methanol M85 are presented in figs. 2 and 3.

With fuel as a parameter it can be seen that with methanol it is possible to boost more than 100 kpa without abnormal knocking, also with a compression ratio of 10.8:1. The limitations of performance are rather set by other parameters like maximum acceptable combustion pressures, maximum pressure gradients and ignition and fuel systems. The higher octane rating

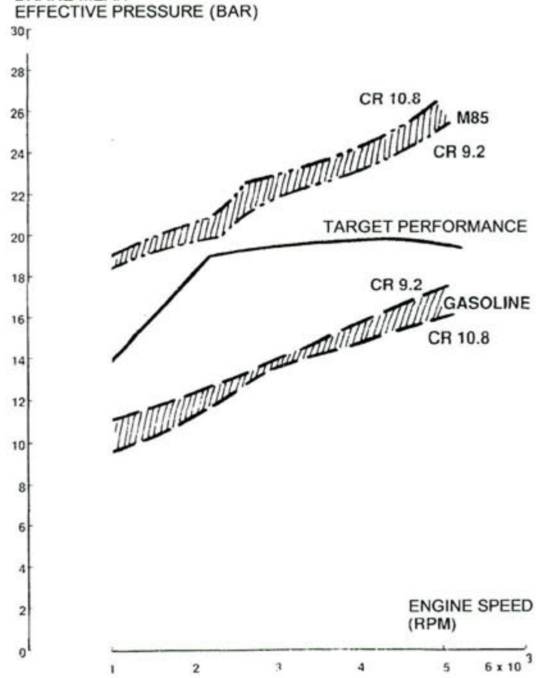


Fig 2 - Single cyl. BMEP as a function of compression ration (CR) and fuel quality.

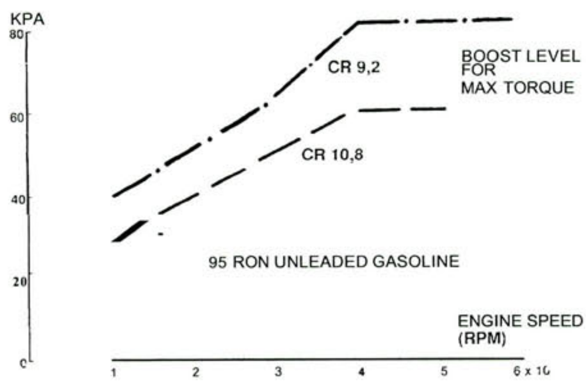


Fig 3 - Charge pressure for max torque at two different CR, 95 RON gasoline.

of methanol is of major importance, together with the cooler compression phase and combustion, leading to cooler residual exhaust gas and better volumetric efficiency.

With gasoline as a fuel the possible boost levels are much more restricted and more linked to what ignition retard and exhaust temperature can be accepted. Also a lower basic compression ratio is necessary if a performance level approaching the set target should be achieved. Figs. 4 and 5.

Interestingly, a similar comparision of the methanol-gasoline mixtures indicates a strong influence of the M85 characteristics. Significant better performance can be achieved with a methanol percentage as low as 30-35%. Fig 6. RON/MON data for the methanol/-gasoline mixtures are given in table 2.

MIXTURE RATIO (M85%)	RON	MON
0	95	85
25	100	86.2
50	102	86.8
75	104	87.2
100	106	87.6

Table 2 - RON/MON for different gasoline-M85 (methanol-gasoline) mixtures.

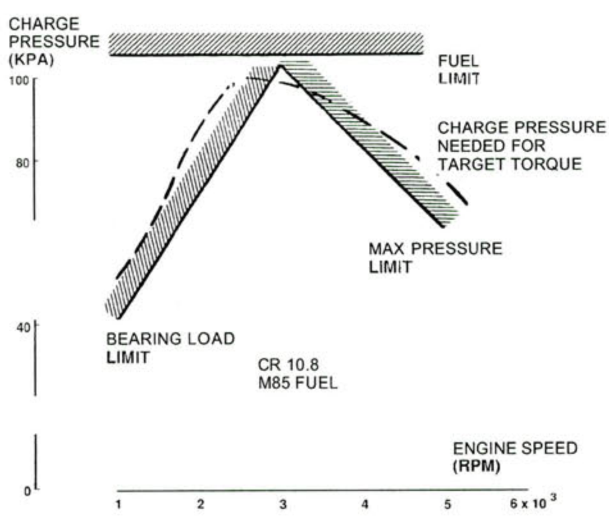


Fig 4 - Charge pressure envelope at CR 10.8, at engine and fuel limits

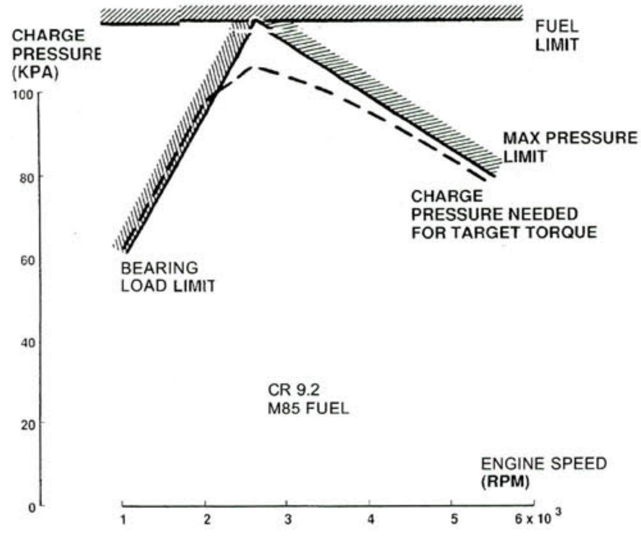


Fig 5 - Charge pressure envelope at CR 9.2, at engine and fuel limits.

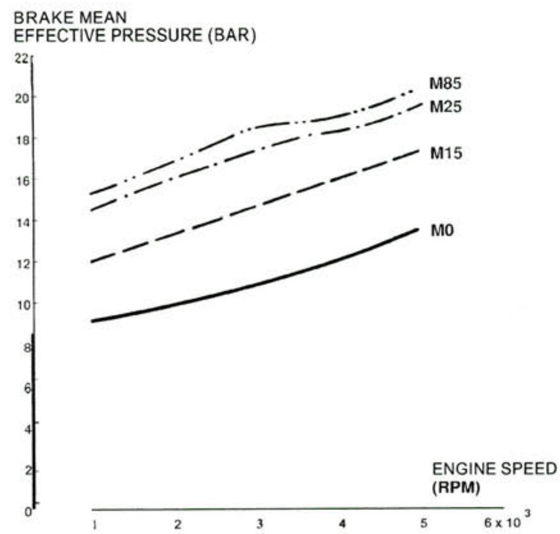


Fig 6 - Single cyl BMEP as a function of methanol-gasoline mixture.

Based on this study the following conclusions can be made:

- With methanol M85 more specific performance can be achieved.
- A higher compression ratio can be used.
- A high charge pressure can be used at all engine speeds which means low end boosting systems can be utilized.
- For a given performance, a smaller engine can be used compared to a boosted gasoline engine.

- With an engine developed for flexible fuel use, more performance will be available with the methanol fuel and will be noticeably higher also at less methanol percentage.
- An optimized boosted FFV engine should be more efficient compared to a similarly performing gasoline version due to a possibly smaller displacement and higher compression ratio.

The single cylinder study gave a solid base for further multicylinder work, giving input to the characteristics of suitable charging systems, engine sizing, compression ratio and charging pressure. The recorded combustion pressures also gave input to the necessary modifications to piston, ring package, connecting rod and crankshaft.

The ignition system requirements were also studied. Due to the wide operating range (cold start and idle with methanol or gasoline fuel, to full load on methanol fuel) a turbocharged FFV engine needs spark plugs with an extremely wide temperature range. There are currently no spark plugs available which can fully span this wide operating range, and further development of FFV ignition systems is required.

BOOSTING SYSTEMS WITH REGARD TO FUELS

Results from the single cylinder tests have shown that the methanol fuelled MEP engine could be boosted higher, especially at low engine speeds. This created an interesting challenge to develop a charging system with a broad operating range to fully utilize the advantages of methanol fuel.

In the PCP project the task was somewhat different. The goal was to create a supercharged version of Volvo's new inline six-cylinder engine with a displacement of 3 litres. The engine should be a fuel efficient alternative to a larger (4-4.5 litres) naturally aspirated (N/A) engine. Hence, it was clear that the normal power lag associated with turbocharged engines must be reduced to a very low level. The project therefore started with an extensive survey of possible charging methods. Both mechanically driven superchargers and turbocharging were considered.

ANALYSED CHARGING SYSTEMS - Single exhaust turbocharaer - Using a single turbocharger is probably the simplest and most cost effective way of rising the engine output. Matching the turbocharger is, however, always a compromise. At low speeds and low loads the exhaust flow and energy is low, and to get boost pressure a small turbine flow area is necessary.

Selecting a small turbine flow area means that at high engine speeds a lot of the exhaust flow must be dumped through the wastegate, resulting in low turbocharger efficiency, high exhaust back pressure and high fuel consumption.

Today most turbocharged engines have a sufficient broad operating range. A more severe problem is to minimize the so called turbo lag. At low loads there is little available exhaust energy to accelerate the turbocharger at a sudden load increase, resulting in poor driveability.

The single turbocharger system is used as a reference case from which we try to widen the operating range and decrease the lag time for load changes.

Improvement of the standard turbocharger can be made by using ceramic turbine wheels for lower moment of inertia (lower turbo lag) and by using ball bearings to decrease bearing friction.

Sinqle exhaust turbocharaer with variable turbine nozzle - With this system it is possible to get both a small turbine flow area at low engine speeds and for turbo acceleration during transients, and a large turbine flow area for better efficiency at high engine speeds. It is possible to get a very fast boost pressure built up but at thecost of high back pressure during the turbo acceleration. For the best result careful consideration must be made for the control sf the turbine nozzle

Both more conventional radial inflow turbines with a variable nozzle ring and axial turbines were analysed. The advantage of the axial flow turbine is a lower inertia and normally a higher efficiency at actual specific turbine speeds. The drawback of an axial turbine is a more complex mechanism for the nozzle.

The nozzle mechanism is the weak part of a turbocharger with a variable turbine nozzle; it is sensitive and expensive. Using a simple mechanism usually results in poor turbine efficiency with little gain compared to a turbine with a wastegate.

Sinqle turbocharaer with hvdraulic assist - A small hydraulic turbine (normally of pelton type) can be placed in the bearing housing, and power can be added by injecting oil on the turbine when the exhaust: flow is low and during turbo acceleration. Drawbacks are low overall efficiency for the hydraulic system and the high cost.

Two turbos in parallel (Bi-turbo) - Connecting one turbocharger to a group of three cylinders on an inline six-cylinder engine is a very favourable way of turbocharging. Such a system makes efficient use of the exhaust pulse energy. Only one cylinder is open to the exhaust manifold at a time, and there is no interference between cylinders. The manifolds can also be made very compact so that the pressure rises quickly during the blow-down. At low engine speeds this means that a large part of the turbine power is generated during and immediately after biow-down, utilizing pulse energy that otherwise would have been lost as a pressure loss across the exhaust valve. During the valve overlap the pressure in the exhaust manifold is almost the same as downstream of the turbine, fig 7. The result on engine performance is that boost pressure can be created at low engine speeds even with a relatively large turbine flow area, resulting in a broad operating range. The boost pressure can also be lower for a certain torque level because the engine virtually does not feel the back pressure at low engine speeds. The lower boost pressure together with favourable scavenging with a low amount of residual gases reduces the engine knock level.

Turbo lag is also reduced with a bi-turbo system. The moment of inertia decreases faster than the flow range with a smaller turbocharger. The flow is approximately proportional to √d, and the moment of inertia to d^5. Taking into account the higher speed of the small turbocharger the decrease of inertia is about 30% for a bi-turbo system (4).

A bi-turbo system requires a wide compressor map, the compressor can in fact set the limit for the operating range of the system.

The bi-turbo system was selected for the PCP engine. It is ideally suited for an inline six-cylinder engine and gives many advantages over the single turbocharger system with little additional complexity.

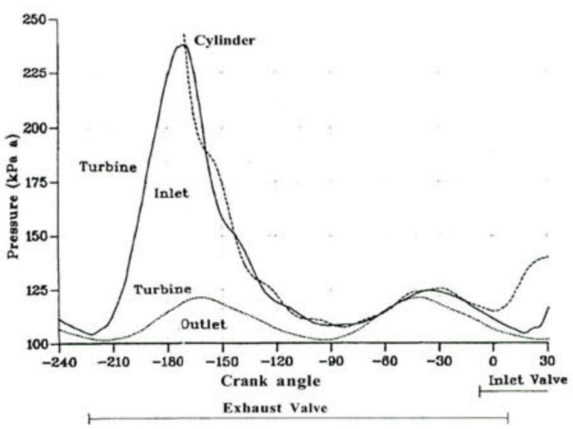


Fig 7 - PCP Exhaust pulse at 30 rps ICES simulation.

Sequential turbocharaina with turbos in parallel - This system has been used in limitied applications, like in the Porsche 959 engine (2), (3). Running at high engine speeds it works as the bi-turbo system, but at low engine speeds one turbocharger is shut off. All six cylinders must be connected to a common exhaust manifold. A valve, called turbine cut in valve (TCV), is placed upstream (alternatively downstream) one of the turbines. When running on one turbocharger this valve is used to control the boost pressure and it is fully opened at high engine speeds. The boost pressure is then controlled by a wastegate. On the compressor side the air from the second turbocharger is ducted through a nonreturn valve when both are running. During single turbo operation the air from the second turbo is ducted through a bleed valve back to the air filter to avoid compressor surge.

The advantage of this system is that only one small turbocharger is used at low engine speeds and two at high engine speeds. The result is a very broad operating range and quick response, as only one turbocharger needs to be accelerated at low engine speeds when available exhaust flow is low. The problem is the difficulty to achieve a smooth transition from operation of one turbo to two. The second turbocharger starts spinning up due to the flow through the TCV valve, but not fast enough during fast engine transients, as on first gear. To avoid overspeed of the first turbo, the second turbo must be cut in before it is up to full speed, resulting in a dip in boost pressure and torque. At slower engine transients on high gears, a smooth transition can be made with a well matched turbo and control system.

Sequential turbocharaina with turbos in series - This system makes use of one small and one large turbocharger in a series sequential arrangement. The primary charger is directly close coupled to the exhaust manifold with very short and effective flow paths from the cylinders. The exhaust gas from the primary turbocharger is then ducted to the secondary turbocharger turbine inlet. There is also a large by-pass channel with a control valve from the manifold to the secondary turbine (6).

Intake air is always routed through the secondary large turbocharger compressor on the air side. At low flows when the large turbocharger cannot give the required boost pressure, the air is ducted through the primary small turbocharger compressor. At low engine speeds and during transients, the major pressure build up is in the small primary turbocharger, and the pressure is then controlled by the turbine by-pass valve. When the secondary turbocharger spins up, the primary turbocharger goes down to idle, and the air bypasses the compressor through a nonreturn valve.

The system principle is shown in fig 8.

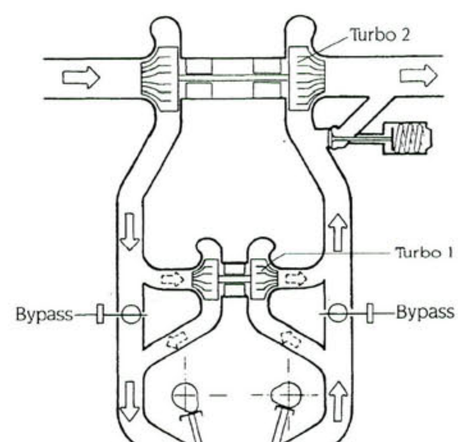


Fig 8 - Series Sequential Turbocharger System Principle.

The advantage of the system is a very wide operating range, quick response and the transition from the small to the large charger is much smoother compared to the sequential parallel system. The drawbacks are some pressure losses on both the air side across the nonreturn valve and across the turbine priority (bypass) valve on the exhaust side. There is also, as in the sequential system described above, a need for a sophisticated control system.

This system has been used in the MEP engine.

Mechanically driven superchargers - A survey of available mechanically driven superchargers has been carried out in the PCP project. The advantage is that turbo lag is eliminated. One problem is noise from most displacementtype of compressors. Mechanically driven turbo compressors do not have a noise drawback but need a continuously variable transmission, Installation may also be a problem. The charger is normally positioned at the front of the engine where a modern engine already has an alternator and often a steering servo pump and an AC compressor. An advantage with respect to installation is that only the air side ductings are affected.

COMPUTER SIMULATION - In both the MEP and the PCP project extensive computer simulation has been used. The MEP project has used simulations for design and initial tuning of the control system. The PCP project started with a theoretical study of all systems described above. The following set-ups were taken:

- To carry out matching calculation to select charger sizes and to give stationary performance
- To test load transients at constant engine speeds
- To check vehicle acceleration 0-100 km/h for practical response
- To carry out gas exhange simulation for baseline engine tuning
- To make control system simulations

All computations except the gas exchange simulation have been made with a simplified engine model. In a complete gas exchange simulation, the conditions (pressure, temperature, velocity and mixture) in different parts of the engine are calculated at every crank angle increment. This is, however, time consuming when the main purpose is to study the turbo transients. A simplified engine model has therefore been developed. This model describes the physics of the engine at 5 points, and it is calibrated against known engine performance. The model reacts in an approximately correct manner for changes in inlet and outlet conditions and for changes in ignition timing, fuel-air mixture and type of fuel. The model was first calibrated against known N/A engine performance.

A difficulty is that the model does not handle pulses in the exhaust manifold. To simulate the effect of pulses, a factor has to be applied to the turbine efficiency. The level of the pulse factor is dependent on the installation and operating conditions, and it has to be selected based on experience or from results obtained by gas exchange simulations.

To simulate the turbocharger, models must be used for the turbine and compressor behaviour under highly offdesign conditions. The models must work even at idling conditions and when the compressor acts as a restriction for the air flow (pressure ratio below 1.0).

The simulation program used (ICES) allows modelling of any type of turbo system. The initial matching of the turbo system and tuning of a control system can be done using the fast 5-point model. The 5-point model can later be changed to a filling and emptying model or the full gas exchange model. The engine and turbo system model can also be linked to a transmission and vehicle model.

The MEP system model has also been implimented in the ACSL system and in a real time simulation computer. The latter can in fact be directly coupled to the control system hardware for tests.

Dynamic real time computer simulations were carried out, basically to investigate the control system in regard of minimizing boost and torque fluctuations during the transfer of boosting from the small primary turbo unit to the larger secondary turbocharger.

SIMULATION RESULTS - Practical response is calculated as a vehicle acceleration from 0-100 km/h. The vehicle is a VOLVO 760 with an automatic transmission, see appendix E. Results from the PCP and MEP simulation are shown in figs. 9 to 19. For ranking of different turbo systems simulations have been done both with perfect wheel to ground grip and with a limited friction coefficient together with traction control.

Vehicle acceleration with perfect grip is shown in figs. 9 and 10, and boost pressure and torque in figs. 11 and 12. Boost pressure build up is faster for the MEP engine but engine torque is lower during the transient than for the PCP engine. This is due to the smaller displacement and in some extent to the higher back pressure for the MEP engine. It is also highly influenced by the control algorithms and assumed efficiencies. Time to reach 40 km/h and 100 km/h is shown in table 3, and it can be seen that the PCP starts a bit faster. When the series sequential system is matched to the PCP engine, acceleration time is reduced about 5% during the early phase of acceleration.

In the simulation model there is no engine torque reduction during gear shifts. This explains the difference between test results and the calculated 0-100 km/h time for the PCP.

No test results are available for the MEP with a comparable transmission.

As can be seen, there is still some lag in the build up of boost pressure. This is, however, hardly noticed in practice. The vehicle acceleration reaches a high level in less than 0.5 seconds.

The turbocharger speeds for the MEP are shown in fig 13. After 2 seconds all boost pressure is generated by the large secondary turbo.

In figs. 14 and 15 the steady-state torque curve is plotted together with dynamic torque during the acceleration from a standing start. Low speed steady state torque is not reached because the engine is accelerating.

Operating conditions for the compressors, at steady-state and during acceleration transients, are shown in figs. 16 and 17 for the PCP and in figs. 18 and 19 for the MEP engine.

Case	0-40 km/h	0-100 km/h
MEP calculated, perfect grip	2.14	6.32
PCP calculated, perfect grip	2.10	6.25
MEP calculated, traction control	2.34	6.58
PCP calculated, traction control	2.25	6.50
PCP test results	2.20	6.80

Table 3 - Vehicle acceleration time (in seconds) with automatic transmission.

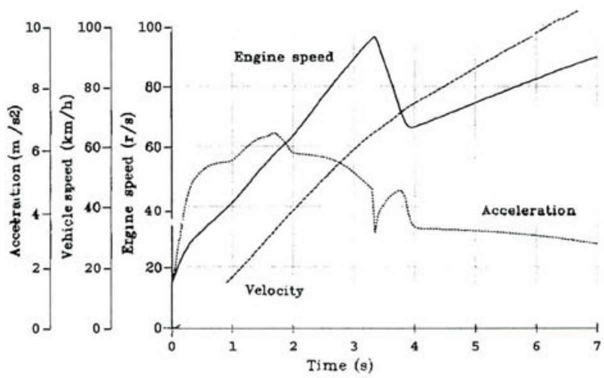


Fig 9 - PCP Vehicle acceleration with perfect grip.

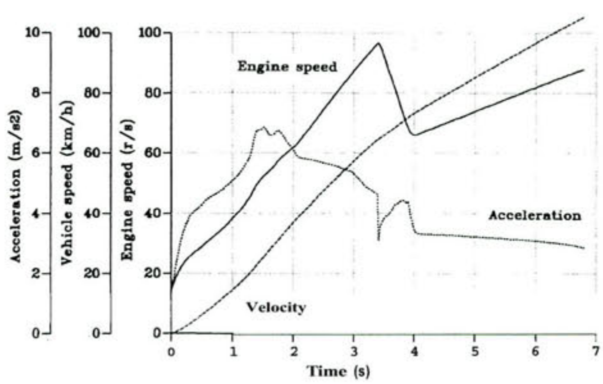


Fig 10 - MEP Vehicle acceleration with perfect grip.

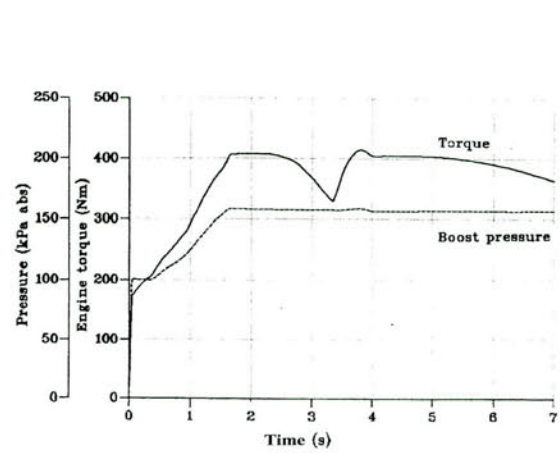


Fig 11 - PCP Boost pressure and torque during vehicle acceleration.

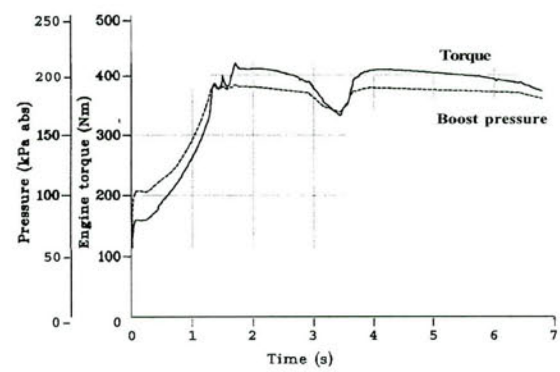


Fig 12 - MEP Boost pressure and torque during vehicle acceleration.

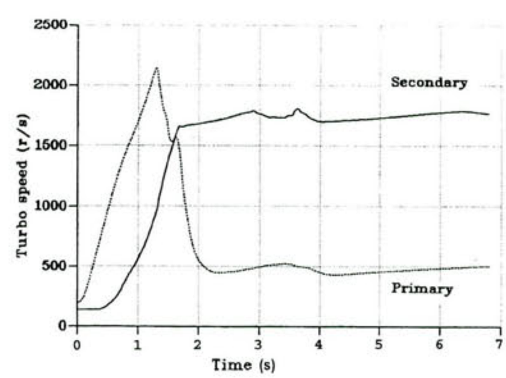


Fig 13 - MEP Turbocharger speeds during vehicle acceleration.

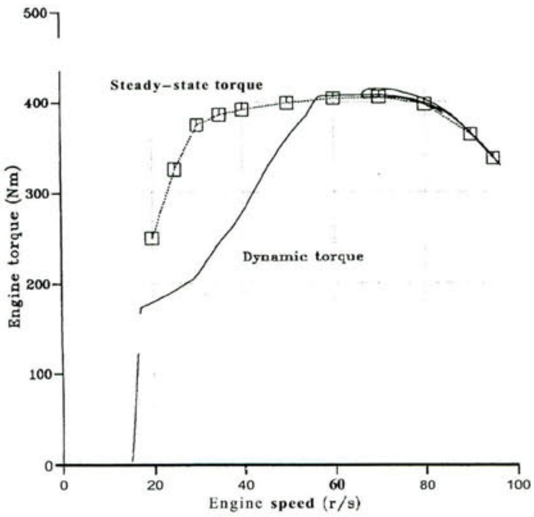


Fig 14 - PCP Dynamic torque during vehicle acceleration.

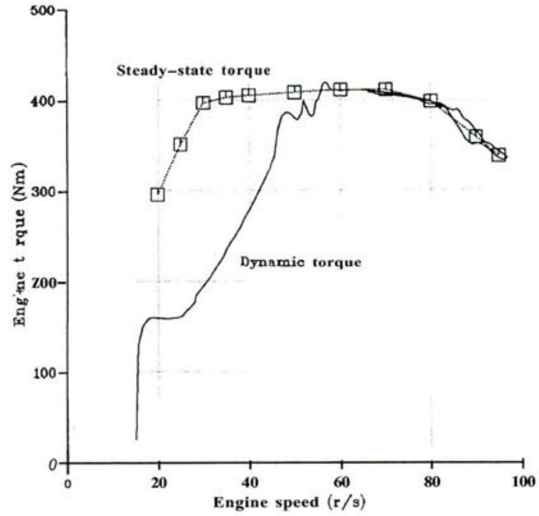


Fig 15 - MEP Dynamic torque during vehicle acceleration.

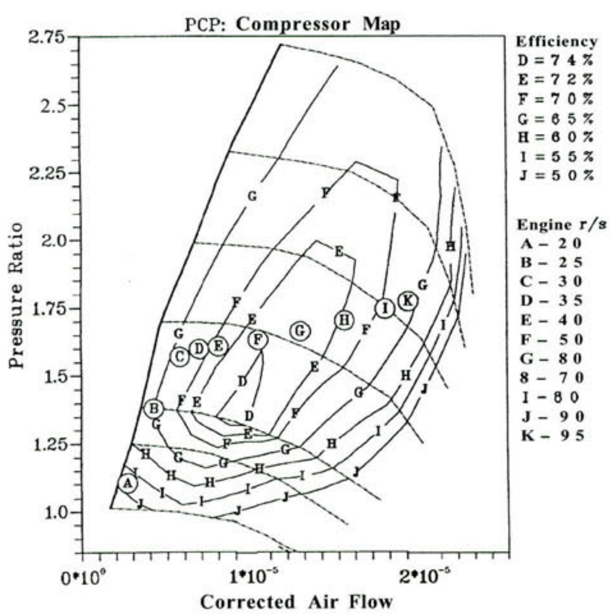


Fig 16 - PCP Full load operation points. ICES simulation.

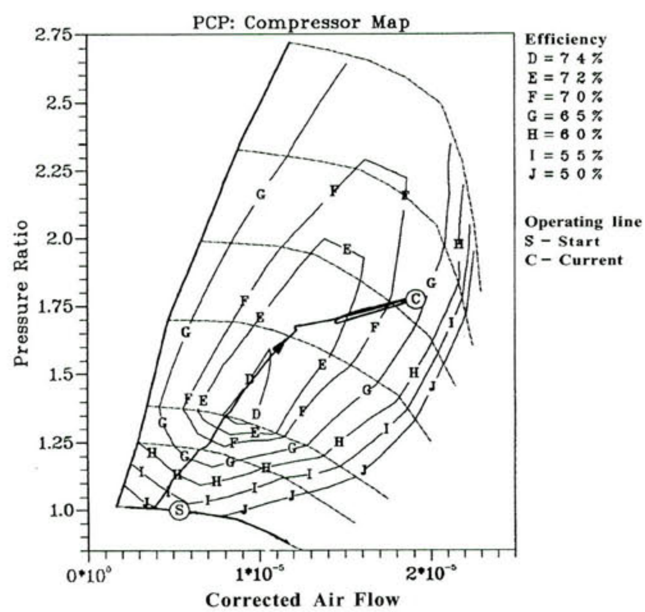


Fig 17 - PCP Transient compressor operation.

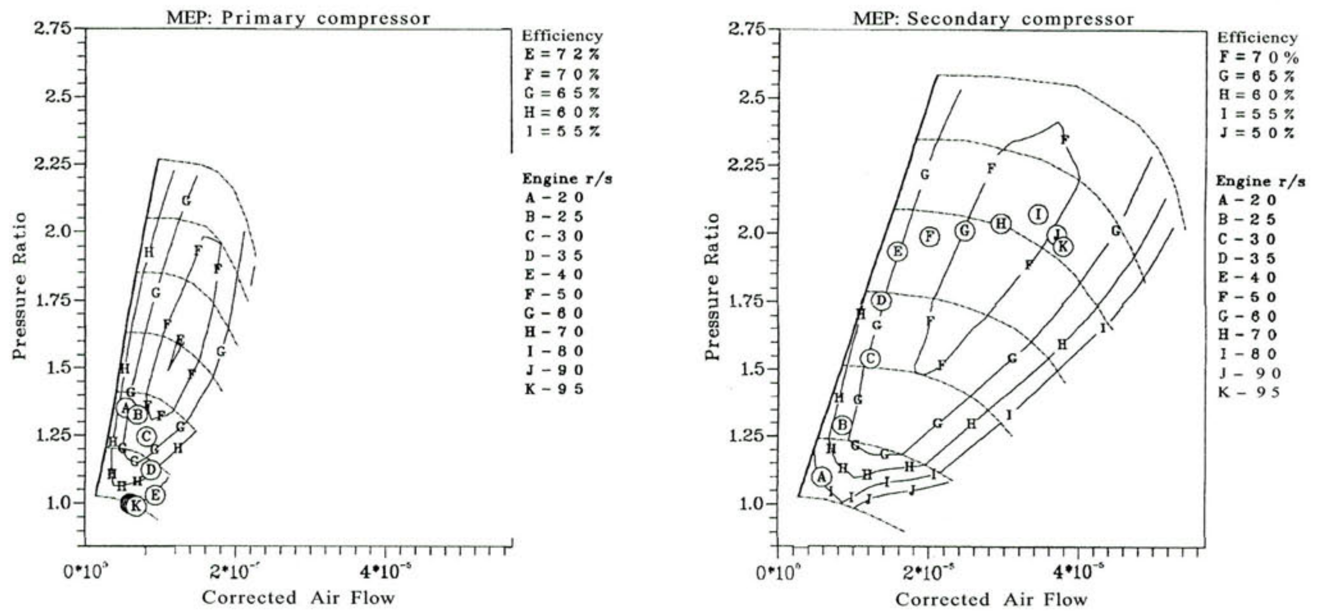


Fig 18 - MEP Full load operation points. ICES simulation.

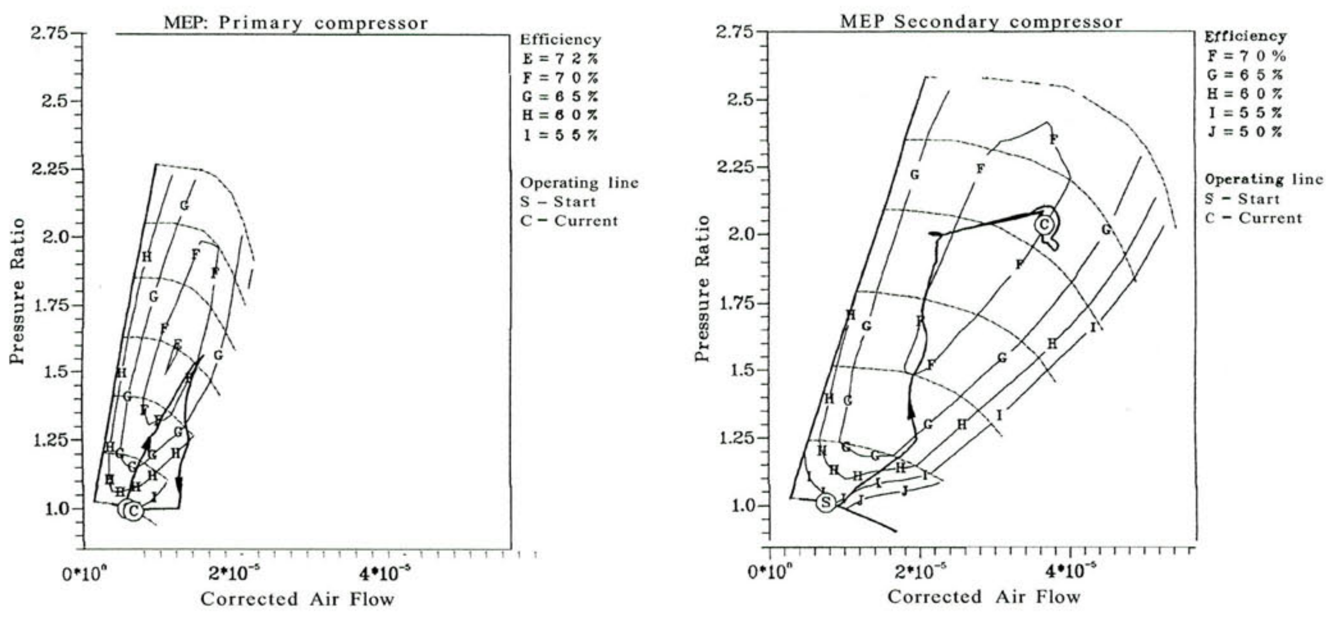


Fig. 19 - MEP Transient compressor operation.

REFERENCE ENGINE

The VOLVO 3-litre six-cylinder engine with the designation B6304F is an all aluminium, twin overhead camshaft 4-valve engine, designed and adapted for the VOLVO 960 top-of-the-line model.

The particular attention to low weight and refined structural design with a full bedplate makes this engine very suitable for turbocharging. The very effective gas exchange system allows the use of camprofiles with lift and duration suitable for good low end torque characteristics. The total power train weight is still relatively low even with the addition of two turbochargers, due to the low weight of the aluminium cylinder block.

The aluminium block with cast in grey iron cylinder liners was of special interest for a methanol engine version. The particular cold start and warm up wear and the influence of corrosive exhaust products might be resolved by alternative cylinder liner materials. If developed, they could easily be implemented in the engine block design.

Finally, the 4-valve cylinder head design is very suitable for methanol fuel usage, due to the better cooling of the valves with less risk for preignition.

Details of the reference engine are given in appendix A and in (1).

MEP ENGINE LAYOUT AND DESIGN

The main difficulty in the layout and design phase was to retain good flow paths and favourable turbine locations, within the restraints of the width and height of the engine packaging in the vehicle.

The advantage of the inline-6 concept was put to good use by routing exhaust manifold outlets in two directions, upwards to the small primary turbo and downwards via a turbine valve to the larger main turbo.

This also resulted in favourable flow paths for the exhaust flow from the small unit to the large one, and similarly for the charge air pipes with the non return air valve installed on the lower forward side of the engine.

Engine width could be retained within the limits acceptable for the 960 vehicle. The complete engine package can be studied in figs. 20 and 21.

The turbochargers are of Garrett T2 and T3 type. Details are given in appendix C. The turbine and air valves were unique developments for the MEP engine. Exhaust manifolding was cast in Ni-Resist material.

Air filter and air meter were uprated for a low pressure drop at maximum air flow, similarly the charge air cooler and the non return air valve were developed to give acceptable flow losses at full power conditions.

The fuel system is based on the Bosch Motronic 1.8 group sequential injection system. The very wide dynamic fuel flow range (from gasoline idle to full power M85 methanol fuelling) was handled by twin injectors per cylinder, with only the injector nearest the valve operating when gasoline is used, and at low fuelling conditions with M85 fuel.

This required a modified intake manifold. The injector installation can be seen in fig 22.

The exhaust system was designed for low flow losses with two parallel TWC catalysts. Metal substrate was chosen for minimum dimensions for a given pressure drop, but also for fast light off with M85 fuel.

The base engine was redesigned with a new engine block with a smaller bore (81 mm) for added stiffness at high combustion pressures, as expected from initial single cylinder engine tests.

The high mean effective pressures possible with M85 fuel, means a smaller displacement was possible for the target performance.

This was achieved with a shorter stroke (80 mm) forged crankshaft. A new connecting rod/piston assembly capable of accepting combustion pressures

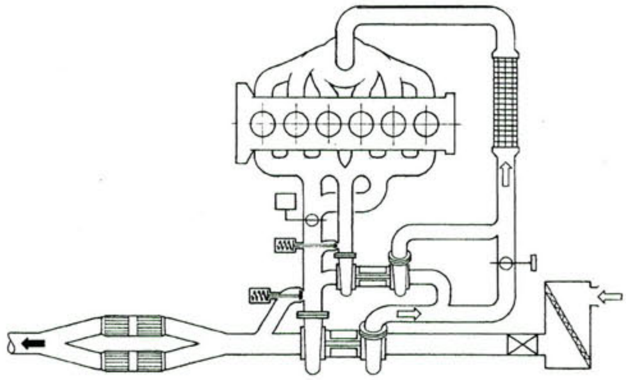


Fig 20 - Principle layout Series Sequential Turbocharger system MEP-engine.

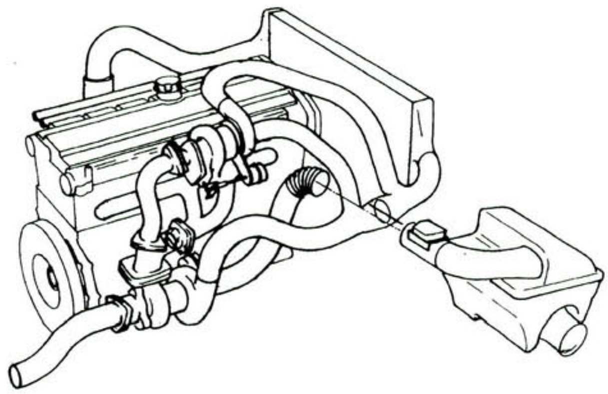


Fig 21 - Turbocharger installation MEP-engine.

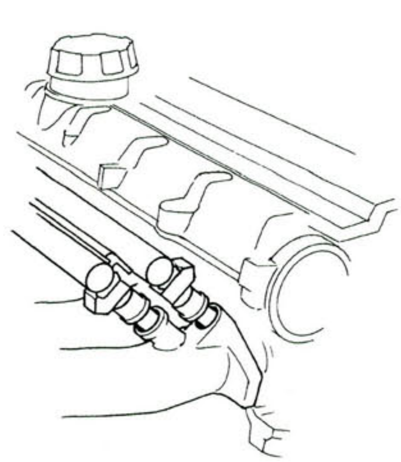


Fig 22 - Dual fuel injector installation details.

up to 120-130 bar was developed. The piston/-connecting rod assy can be compared with the reference engine components in fig 23.

The already generously dimensioned water and oil pumps were retained from the N/A engine.

A new cylinder head gasket for the smaller bore was used.

The oil and water cooling circuitry was modified to include the lubrication and cooling of the two turbos.

Oil drain from the turbos was brought to a modified oil sump.

An enlarged crankcase ventilation system was designed to handle the increased blow by flow due to high cylinder pressures.

CHARGE AIR CONTROL SYSTEM - For the charge pressure control, integral wastegate valves for the two turbines are used complemented with additional turbine valve and non return air valve. See fig 24.

Wastegate opening is controlled via two separate threeway electromagnetic valves. The turbine valve is controlled by an electromechanic servo, connected to

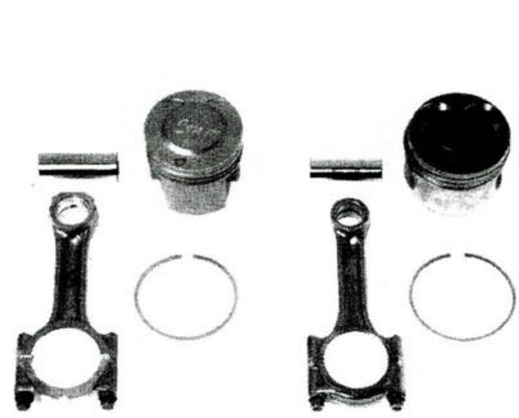
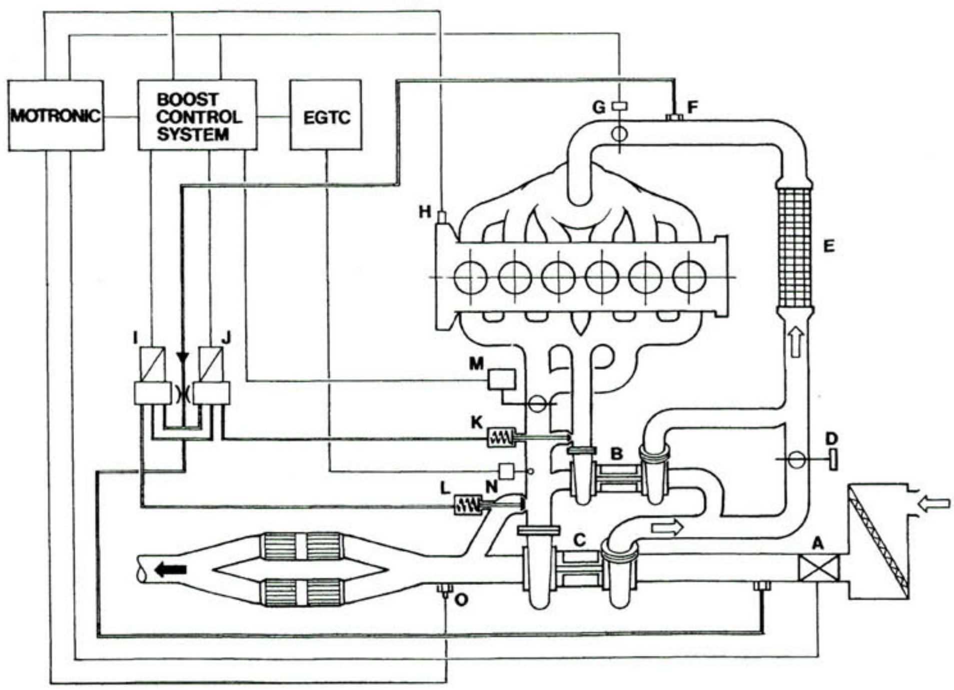


Fig 23 - Comparison MEP and reference engine, piston and connection rod assembly.



A. Air Meter
B. Primary Turbo (T/C 1)
C. Secondary Turbo (T/C 2)
D. Air Valve
E. Charge Air Cooler
F. Charge-Pressure Transducer
G. Throttle Sensor
H. Engine Speed Sensor
I. Control Valve (T/C 2)
J. Control Valve (T/C 1)
K. Wastegate (T/C 1)
L. Wastegate (T/C 2)
M. Turbine Valve Actuator
N. Exhaust temperature Sensor
O. Lambda Sensor

Fig 24 - Control system MEP-engine.

the valve by a push/pull rod system. The air valve is of differential pressure type and so designed that it automatically moves to an open position when the secondary charging system is operational.

The control system evolved based on input from computer simulations, engine rig testing and finally vehicle testing. The control algorithms are of necessity rather complex due to the different response from the primary and secondary turbochargers. Due to the different flow losses depending on which turbocharger is operating, the boost level is different for a constant torque output.

The final control program was capable of regulating the charge system without noticeable torque fluctuation during the transfer of turbocharger operating regimes.

In order to accurately monitor the output torque a torque sensor was fitted to the input shaft of a manual transmission. This made instantaneous readouts of the actual engine torque possible and was extremely useful in the final fine tuning of the control system.

TEST RESULTS MEP-ENGINE - The MEP-engine development tests closely followed the single cylinder test results.

With M85 fuel, MBT ignition could be used, resulting in rather high combustion peak pressure and low exhaust temperatures. Intake pressures, up to two bar, absolute, could be used also with the initial 9,5:1 compression ratio. However, when run on gasoline, the engine was severely knock limited also at reduced boost pressures. There was firm evidence the knock characteristics had to be improved.

The combustion system was modified, by lowering the CR to 9.1:1, and by increasing the squish height. This resulted in less turbulence and reduced temperatures of the cylinder head surfaces.

The piston design and ring pack specification was modified for better blow-by and oil control.

A reduction in the exhaust system back pressure reduced the negative effects of hot residual gas fraction, and with these modifications it was possible to obtain an acceptable performance curve with the MEP-engine also when run on gasoline.

Steady state turbo matching resulted in full load pressure ratios according to figs. 25 and 26.

Transients were simulated in rig tests to calibrate the wastegate and turbine valve settings. It was found that the primary (small) turbocharger had to give approx 10 kpa higher boost pressure for a given torque, compared to the larger secondary turbocharger operating alone.

Specific air flow was reduced when running on methanol M85. This is quite important, since all flow paths and valves can be dimensioned smaller than for a corresponding gasoline version with equal power output. The reason for this is a combination of improved combustion efficiency, the better charging efficiency due to lowered in cylinder temperatures and the fuel itself with some oxygen content.

The relative air flow can be seen in table 4.

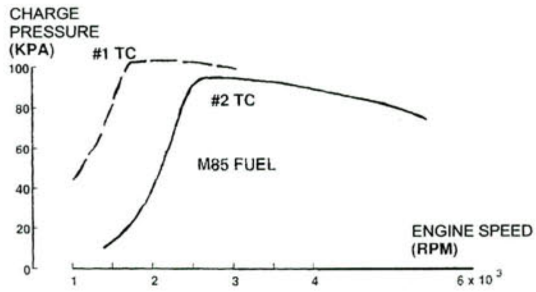


Fig 25 - Steady state charge pressure at full load, M85 fuel.

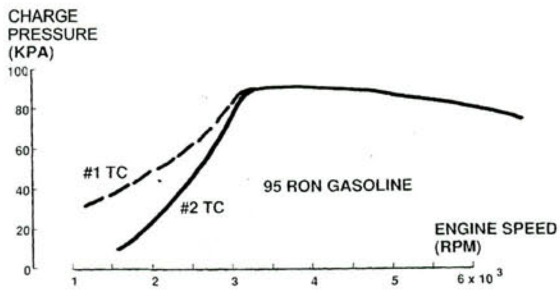


Fig 26 - Steady state charge pressure att full load, 95 RON gasoline.

REL. AIR FLOW AT EQUAL POWER M85 VERSUS 95 RON GASOLINE

SPEED (RPM)	BMEP (BAR)	REL. AIR FLOW (%)	ENGINE TYPE
2000	11.5	90	SINGLE CYL.
4500	9.0	92	SINGLE CYL.
2400	14.2	93	MULTI CYL.
2700	14.7	95	MULTI CYL.
3000	15.8	94	MULTI CYL.
Rel. Air Flow		93	

Table 4 - Relative air flow at a given load, for M85 fuel and 95 RON gasoline

After calibration, the engine was tested in a 760 vehicle, specification in appendix E. Chassis dynamometer tests indicated that the turbine transfer valve control worked with very little torque fluctuation, and subsequent tests with the vehicle also confirmed this satisfactorily.

One transient run is shown in figs 27 and 28. It can be seen that the torque can be kept nearly constant during the turbine valve opening and follows the programmed set value very closely during the acceleration run.

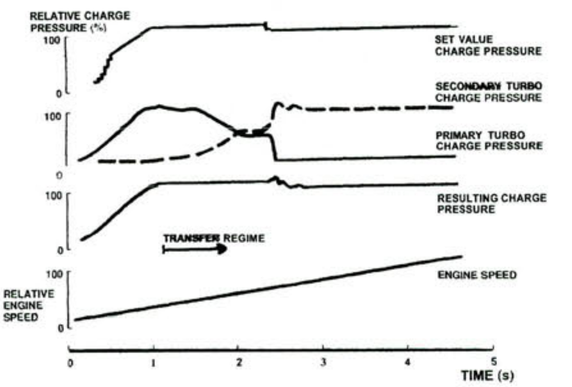


Fig 27 - Turbocharger operation during a transient run Boost pressure versus time.

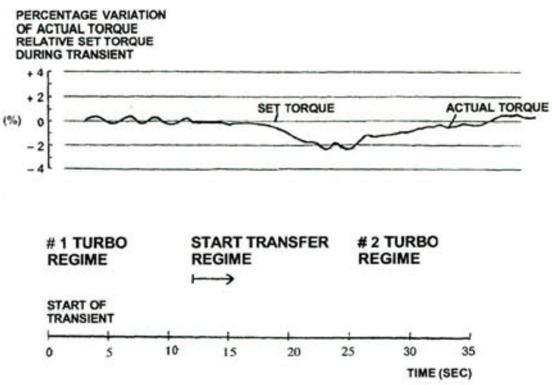


Fig 28 - Variation of actual torque relative set torque during a transient run.

Further vehicle tests revealed some weaknesses in the initial turbine valve design, and a longer sleeve and increased valve shaft diameter was introduced. With optimized tolerances, this very compact turbine valve design gave satisfactorily operation also in prolonged usage.

Different spark plug specifications were used for M85 and gasoline, in order to ensure safe engine operation. Modified ignition systems, which should minimize the FFV-drawbacks, have to be developed.

Furthermore, the FFV-operation requires the boost pressure to be adjusted according to M85 or methanol gasoline mixture operation. From earlier single cylinder tests, it was found that near M85 performance characteristics were reached already at above M35 mixtures. This indicates a simplified boost control could be satisfactory for a developed engine.

This could also simplify the wastegate and turbine valve actuator design and specification.

Finally, the knock control strategy was modified at FFV/M85 operation. To avoid excessive residual gas temperatures and thereby increasing cylinder gas charge temperatures, knock algorithms were biased towards boost reduction and rather less use of ignition retard. The background for this strategy is further described in the combustion characteristics section.

Full load performance with M85 fuel and gasoline are shown in fig 29. Comparison is made with the base N/A engine.

The mean effective pressures are compared in fig 30.

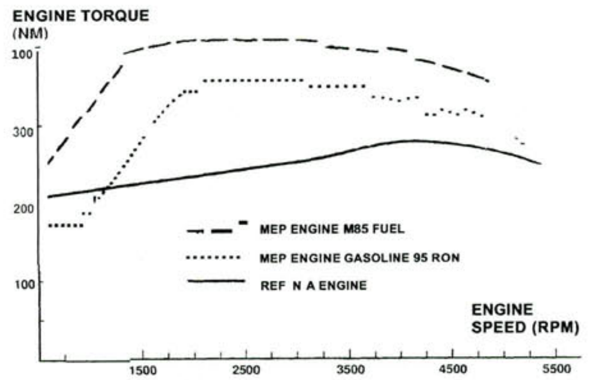


Fig 29 - Engine torque MEP-engine as a function of fuel quality, compared to reference N/A-engine.

COMBUSTION CHARACTERISTICS - Cylinder pressure curves were taken in both single cylinder and multicylinder tests.

Heat release analyses were made with an AVL Combustion Analyzer. MBT ignition and (weaker), WMMP mixture calibration settings contributed to faster heat release, with higher pressure gradients and high maximum combustion pressures in cylinder, compared to gasoline. Figs. 31 and 32.

Engine design limitations, given by maximum bearing loads at low speeds, maximum peak pressures in regard of gasket, and engine block design and, lastly, maximum acceptable rate of pressure rise for noise, means that the performance capability of the M85 fuel cannot be fully utilized without some redesign of the engine.

For maximum high load efficiency a turbocharged M85 FFV engine must be designed accordingly. The cooler exhaust temperature (800-850°C versus 920-940°C for gasoline) is favourable for turbocharger durability. Fig 33.

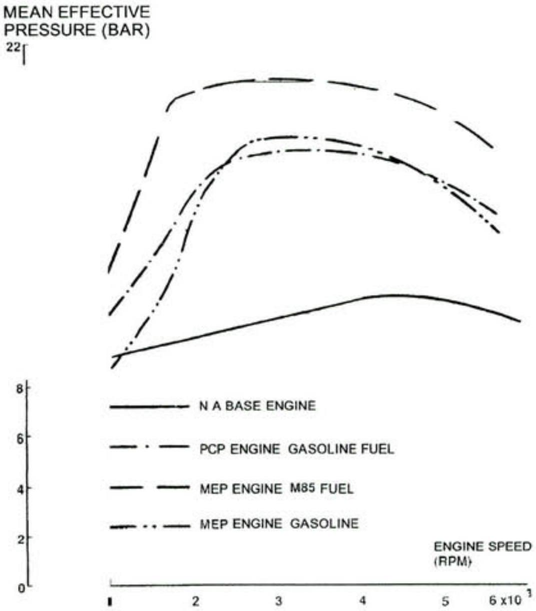


Fig 30 - BMEP, MEP-engine on M85 and 95 RON gasoline, compared to PCP and reference N/A-engine.

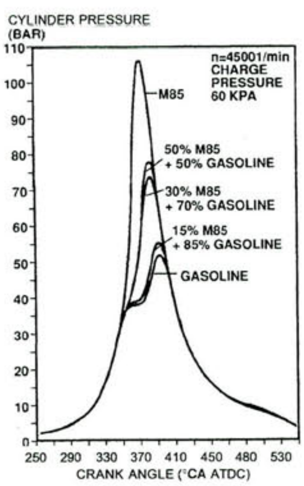


Fig 31 - Cylinder pressure as a function of methanol/-gasoline mixtures.

However, the knocking sensitivity and characteristics indicated M85 reacts differently compared to gasoline at retarded ignition settings. The hotter residual gas fraction heats up the gas charge and the knock margins are quickly reduced, and, eventually, it is possible to get into an abnormal combustion regime at over-retarded conditions.

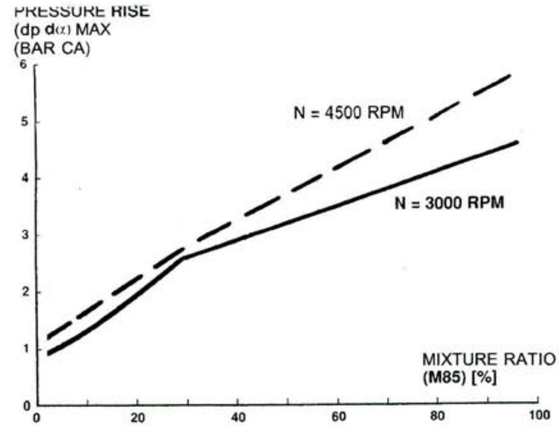


Fig 32 - Pressure gradient as a function of methanol/-gasoline mixtures.

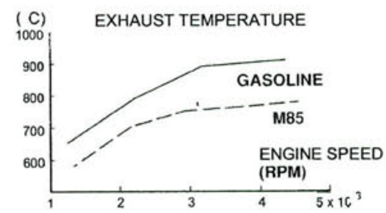


Fig 33 - Exhaust temperature at full load versus fuel quality.

There is also evidence a M85 fuelled high performance turbo engine must be designed with a good cooling system capacity. Under severe operating conditions and at high ambient temperatures, coolant temperatures increase must be minimized to retain low cylinder gas temperatures. The engine calibration should include a reduced charge pressure level at severe ambient conditions, to minimize the risks for abnormal combustion and to retain the knock margin.

From the combustion point of view, an M85 turbo charged engine should be designed with the fuel in mind.

The engine size can be reduced by approximately 15%, if the basic design can cope with higher combustion pressures and a higher rate of pressure rise, without leading to worse noise, vibration and harshness characteristics.

EFFICIENCY - The M85-calibrated MEP engine shows an advantage in efficiency due to the following factors:

- The smaller displacement means less relative mechanical losses at a given absolute load.
- The combustion efficiency is slightly better due to the higher CR and faster burn with the M85 fuel, and the possibility to use MBT ignition in

the full operating range of the engine. Due to lower exhaust temperatures, the mixture need not be enrichened at a rate corresponding to a gasoline version.

- The comparison between PCP and MEP engine versions also includes the different turbocharging systems. The wider range of the series-sequential turbocharger system means the compressor efficiency at high charging conditions can be improved over the speed range, compared to the parallel bi-turbo charge system. Due to unavoidable flow losses for the turbine and air valves the practical difference is somewhat reduced.

Brake efficiencies are shown in fig 34 for the reference N/A engine, the MEP and PCP versions. Particularly at higher speeds, the MEP engine has a better efficiency.

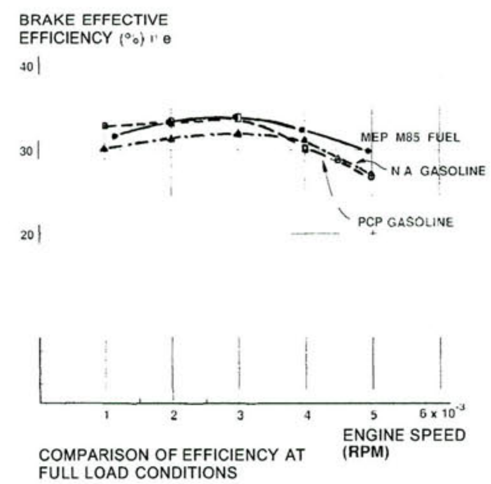


Fig 34 - Brake effective efficiency comparison, full load, MEP, PCP and reference N/A-engine.

Overall improvement potential for the MEP engine is approx 10% compared to the PCP version. Since the PCP engine shows a practical fuel efficiency almost equal to the reference N/A version, it can be safe to predict the MEP engine could have a better efficiency than the base reference N/A engine. This underlines the potential of the M85 fuel, if available in the future, to lead to considerably more efficient engines, compared to current large N/A engines running on gasoline.

EMISSIONS - Single cylinder tests indicated the base emissions of HC and NOx can be approximately halved with M85 operation compared to gasoline. Figs. 35 and 36.

The emissions reduction at FFV-operation shows the reduction is relatively proportional to the methanol percentage. However, there is a rather noticeable reduction already at an M30-M35 operation. The cooler

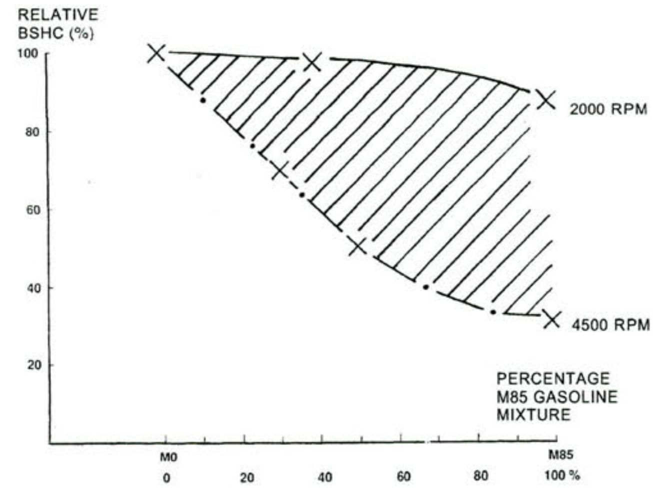


Fig 35 - Relative BSHC versus gasoline/methanol mixture.

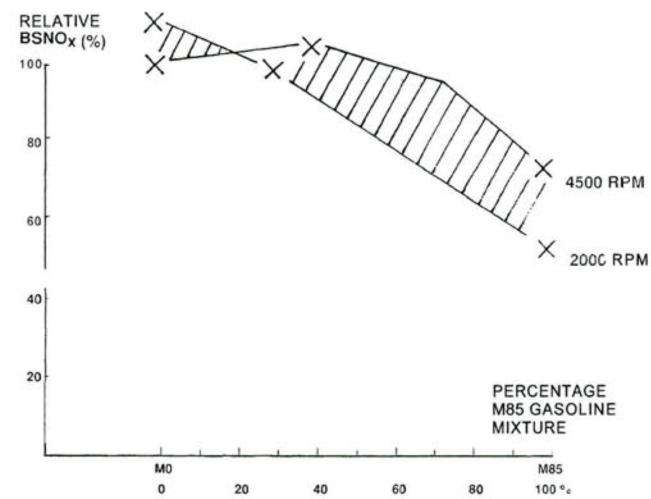


Fig 36 - Relative BSNOx versus gasoline/methanol mixture.

combustion with M85 is the main reason for lowered NOx, while the HC emissions are rather more linked to the engine speed and the M85 fuel combustion characteristics.

The better efficiency of the smaller MEP engine also means the CO_2 emissions could be lowered by more than 10%.

For the required low tailpipe emissions, a TWC-system is needed. Due to the cooler exhaust temperature the catalysts should be positioned somewhat closer to the engine.

The good turbocharger efficiency and the possibility to operate with stochiometric mixture and MBT-ignition also at high power outputs, means the M85 engine can retain very low emissions levels also when the full performance is utilized. This is of particular advantage for a high performance engine.

However, the M85 fuel also results in a slower warm-up and later light-off for the catalyst. This means the basic emissions advantage is somewhat reduced.

For the very strict emissions standards, expected in the future, similar exhaust aftertreatment methods will be required as for gasoline engines.

Formaldehyde emissions are a problem for all methanol fuelled engines. The turbocharged M85 MEP engine shows some advantage compared to high compression N/A engines, since the lower CR-means exhaust temperature are higher under early start and warm-up conditions. The relative specific load is higher, and this means better combustion conditions for less formaldehyde emissions.

However, for good control of formaldehyde emissions, actively heated precatalysts might be necessary. VOLVO developments in this area on FFV-N/A engines have shown favourable results in this respect.

The MEP engine, when run on gasoline, retains some of the above advantages, primarily in regard of CO_2- and HC-emissions.

Final positioning of catalysts must also cope with full power use with gasoline. The main concern is how much performance reduction can be accepted when using gasoline instead of M85 fuel.

A final engine design and specification would be linked to what role an M85 fuel might have in the future.

PCP-LAYOUT AND INSTALLATION

GENERAL - The PCP engine is based on the 3 litre N/A engine with a parallel turbo installation, utilizing very compact exhaust manifolds per each pair of three cylinder groups.

Two turbochargers of the Mitsubishi TDO4L type are fitted to manifolds of approx 0.45 litre volume. This corresponds to a relative volume ratio of 0.31. The ideal 240° between the exhaust pulses and low inertia turbochargers (0.166×10^{-4} kg m^2) facilitates boost at low speed and good response.

The exhaust manifolds are, due to the higher exhaust temperature, manufactured of Ni-Resist with 36% nickel contents. Compared to standard high silicon material these have improved strength and lower expansion at elevated temperatures.

As the B6304 engine is inclined only 12" towards the exhaust side, the two turbochargers can be installed without conflicting restrictions. The location of the rear turbocharger required the design of a new right hand engine bracket. The two chargers are connected to the engines lubrication system, and the drain is piped to the cylinder block. The chargers are also connected to the cooling system.

The two charge air connections are joined into one just before the intercooler. This is of the air to air type, and it is placed in front of the engine radiator. The frontal area is as large as the space available would allow. The connections are 70 mm in diameter, and at maximum speed the temperature drop is around 70° with a low 6.9 kpa pressure drop.

From the intercooler the air is routed to the inlet manifold via smoothly curved ducting. The manifold is the same as the N/A engine with one modification. The noise insulating rubber connectors had to be changed to a type which could withstand the higher boost.

An inlet filter with increased flow capacity and a modified exhaust system are fitted. The modification of the exhaust system consists of a catalytic converter with metal substrate, which gives lower back pressure with the same outer dimensions.

Other changes of the base engine are low compression pistons and sodium cooled exhaust valves. The camshafts have, compared to the N/A engine, shorter duration and lower lift, which contributes to good low speed performance, a high level of combustion stability and reduced emissions.

To match the increased heat rejection due to the higher power level, modifications were made to the cooling system. Both the engine radiator and the oil cooler are replaced by more efficient types.

As a supplementary to the standard transmission cooler an air to oil cooler has been added in the extreme front of the car.

ENGINE CONTROL SYSTEMS

Enaine control - The engine is controlled by the same advanced electronic control system as the N/A engine. This is the Bosch Motronic 1.8 with integrated functions for fuel, ignition, automatic transmission, climate unit and diagnosis, (1).

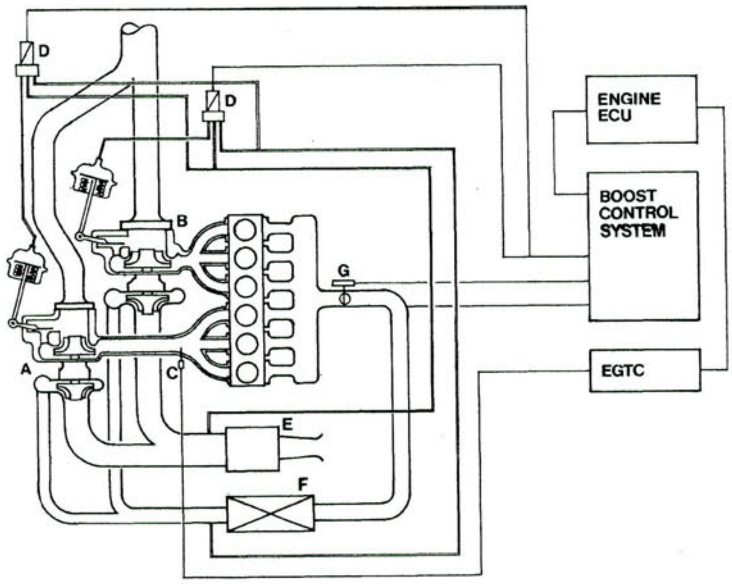
Some modifications have been made, for example the system has an input for fuel enrichment at high exhaust temperatures. Injectors with increased flow capacity are also fitted.

Charae air control - The charge air control is designed specifically for the bi-turbo system, see fig 37. It is a development of the system used in the series VOLVO B204 FT/GT engine. Each turbocharger has an integral wastegate which is controlled by separate soleniod valves. The control signals are generated in the boost control unit. The boost which is controlled, is sensed upstream the throttle by an absolute pressure sensor built into the control unit. The desired boost is dependent on throttle angle and engine speed.

The following advantages are obtained with this system:

- Improved efficiency at part load due to lower throttle losses
- The system works with absolute pressure which gives constant maximum power independent of atmospheric pressure
- Adaptive boost control to reduce the influence of the tolerances of the different components which reduces variations in power level in production.
- Optimum turbo response because of the system's fast and accurate control

Exhaust Gas Temperature Control. EGTC - This is a unique Volvo developed control system, also used in the series production of B204 FT/GT engine, which



A. Front Turbocharger
B. Rear Turbocharger
C. Thermocouple EGTC
D. Turbo Control Valve
E. Air Filter
F. Intercooler
G. Throttle Potentiometer

Fig 37 - Engine charge air control system PCP-engine.

feeds extra fuel only under conditions as:

- Longer periods of high speed/high output conditions
- When using full power under hot climate conditions
- When a low octane fuel is used

Turbocharged engines are often calibrated with particular attention to maximum exhaust temperatures. Since a fixed calibration normally has to cope with the worst conditions (low octane fuel and high ambient temperatures), excessive fuel enrichment often has to be used in order to keep exhaust temperatures low under all conditions.

The VOLVO EGTC control system only uses enrichment when it is necessary, and by this calibration strategy the fuel consumption can be reduced in practical vehicle use.

There is also a considerable "lag" in the rise of exhaust temperature during typical acceleration runs-up to full power, and a stabilized exhaust temperature is reached only after 30-90 seconds (5).

Thus, a significant reduction in fuel consumption is possible, since fuel enrichment is not necessary during typical overtaking conditions. Even when accelerating up to top speed, no extra enrichment is needed, only being added if very high speed, high power usage is continued. Practical fuel efficiency can therefore be improved by 10-20% at higher speeds.

PCP-TEST RESULTS

Performance - The target was set at 180 kW and 385 Nm and theoretical analysis showed a required boost pressure of 155-160 kpa.

Early tests confirmed that the target could easily be achieved. A boost pressure of 157 kpa from 1800 rpm and up was selected and this was then maintained throughout the project.

The initial compression ratio 9.2:1 was changed for 8.6:1 giving a better high speed performance characteristic.

A number of different intake manifolds were tested, but the N/A engine manifold proved to be the best trade off between torque and power.

The final torque curve is shown in fig 38.

The boost builds up with short delay. However, in practice the small delay which of course exists is effectively masked by the torque converter characteristic.

Possible solutions to decrease the delay still further, follow the trend to even smaller turbocharger units and/or ceramic turbine rotors.

Efficiency - Turbocharged engines with increased exhaust backpressure and lower compression normally have a higher fuel consumption than an N/A engine with the same cylinder volume. Due to the favourable way of charging the PCP engine, as previously dis-

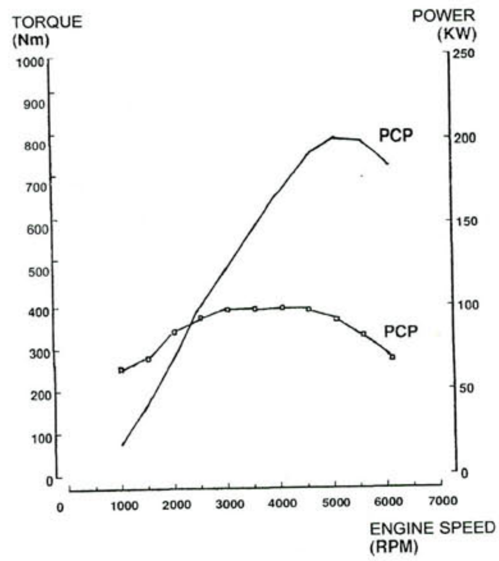


Fig 38 Power and torque curve, PCP-engine

cussed, the penalty is only in the order of 2-3% measured at normal constant speed or in an FTP 75 test cycle.

Another reason is the favourable torque characteristic making it possible to run at low engine speed for a given vehicle speed, even considerably lower than the value chosen for the test cars.

Compared to a modern high performance N/A engine of approx 4.5 litre, giving the same torque, the PCP engine has some 10-15% lower fuel consumption at 120 km/h. Only at very high speeds the N/A engine has an advantage over the PCP engine, see fig 39.

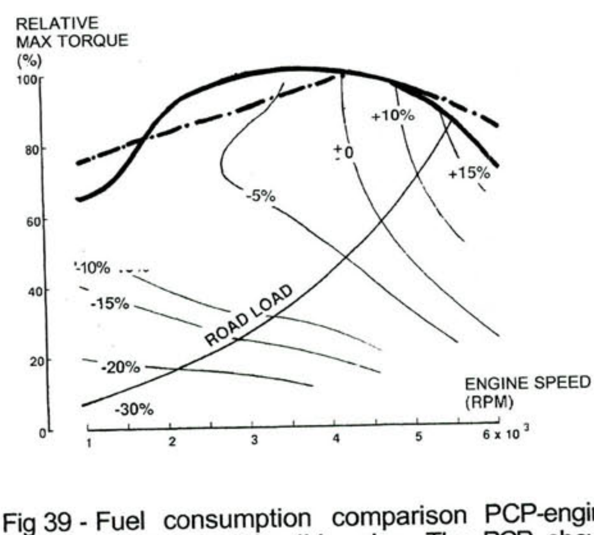


Fig 39 - Fuel consumption comparison PCP-engine versus an N/A 4V-engine. The PCP shows overall better efficiency.

Emissions - The turbocharged PCP engine has a somewhat lower exhaust gas temperature than the reference N/A engine. This has a negative effect on the light-off time of the catalyst.

Using a catalyst with metal substrate and thin-wall austenitic stainless steel exhaust pipes, light-off is achieved within the same time period as the N/A engine.

The car equipped with a PCP-engine and automatic transmission fulfills the US-federal regulations without any extra treatment except the TWC. The small valve overlap also contributes to a low emission level.

PRACTICAL ASPECTS, POTENTIAL APPLICATION

The PCP and MEP projects have similar performance characteristics, but the main difference is the fuel used.

The PCP bi-turbo gasoline engine can be considered a viable near term alternative to N/A engines with larger swept volume (and more cylinders).

An inline 6-engine with carefully optimized bi-turbo boost system results in good transient performance capabilities, and the application of such an engine version falls more in the technical/commercial consideration area.

The MEP engine is very much linked to if M85 will become an alternative fuel in the future, and if so at what level of availability. This decides to a high degree the overall compromise between M85 and gasoline optimization and in the end the absolute performance and fuel efficiency of the concept.

Some unresolved problems are still linked to the M85/FFV concept, such as cold starting, cold climate running and engine wear, ignition system specification for a high performance MEP/FFV engine etc. Further FFV-development is necessary to solve these problems.

Currently an MEP engine version can not be defined in a scheduled time frame.

CONCLUSIONS

A 3 litre PCP-engine with bi-turbo (parallel) charging system run on gasoline and a 2.5 litre MEP engine with (series-) sequential turbocharging system running on methanol (M85) fuel and with FFV-capability, have been developed and evaluated in test bed and in vehicle application. Figs. 40, 41, 42 and 43.

Both engine versions were based on the VOLVO B6304F 6-cylinder engine, and achieved a maximum power of 200 kW and a torque maximum of 400 Nm.

The application of the turbocharging systems was simplified by the inline cylinder configuration. Both the bi-turbo and the series-sequential turbo systems could be designed within the under hood space constraints of the VOLVO 960-vehicle.

The MEP engine shows a typical fuel efficiency advantage of 10% compared to the larger PCP engine. This fuel efficiency is further improved at high power out-puts.

With the MEP engine, the fuel efficiency could subsequently be improved by 7-8% compared to the reference N/A engine. This is based on the fact that the PCP engine only lost 2-3% in fuel efficiency, although the performance was increased by more than 35%.

Transient response and practical performance was demonstrated and found satisfactory, with good driveability also in comparison to larger, equally powered N/A engines. Both engine versions are very fuel efficient in this perspective.

Methanol (M85) fuel was found to have good high performance capabilities, but for full utilization the engine design must be capable of handling high maximum cylinder pressure as well as high rates of pressure increase. The MEP engine was developed to handle peak pressures of 120 bar and a $dp/d\alpha$ of up to 4.5 bar/CA.

Particular attention had to be given to avoid excessive ignition retard when running on M85 fuel. This can result in abnormal combustion due to increased residual gas heating up the intake gas charge.

Base HC and NOx emissions can be reduced by approximately 50% when M85 fuel is used. A low NOx level is also achieved due to the higher compressor efficiency with sequential charging, resulting in lower charge air temperatures and lower exhaust back pressures.

Fully electronic boost control systems were developed for both the bi-turbo system and the series-sequential systems. Novel turbine priority valve and air valve designs were developed for the MEP engine.

There are still unresolved problem areas when using M85 methanol fuel. Further development must be directed to cold starting, engine wear, formaldehyde emissions in cold start and warming up conditions. Novel ignition system and spark plug designs for a wider heat range must also be developed, if successful FFV-operation shall be possible.

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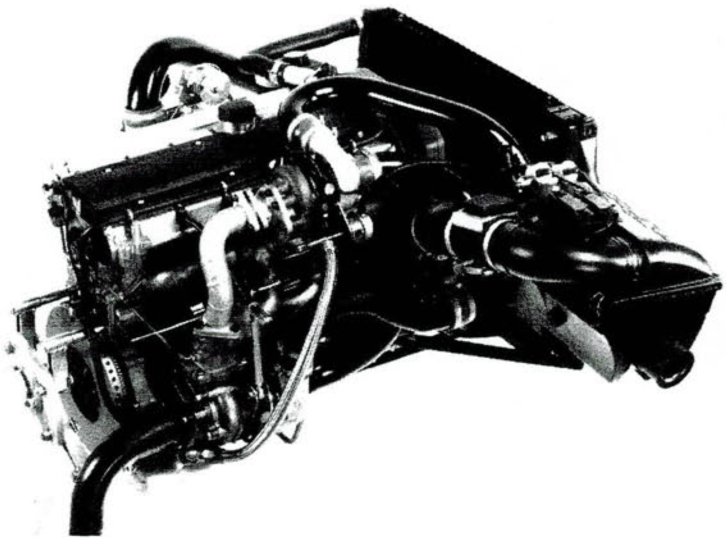

Fig 40 - The MEP engine

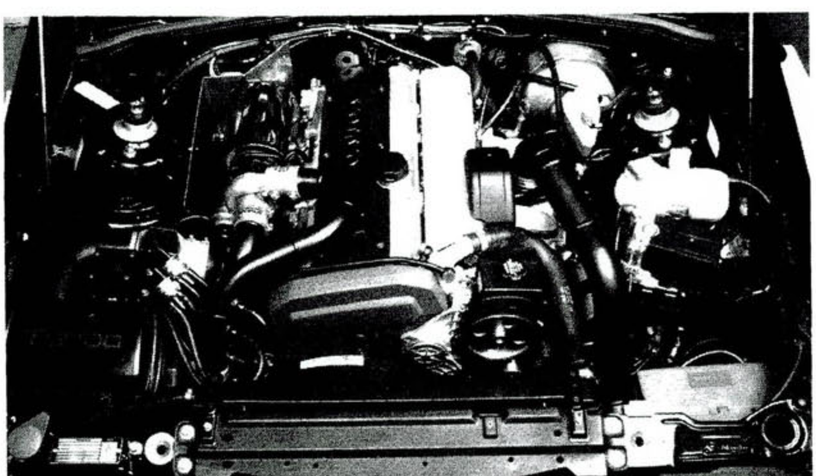

Fig 41 - The MEP engine vehicle installation

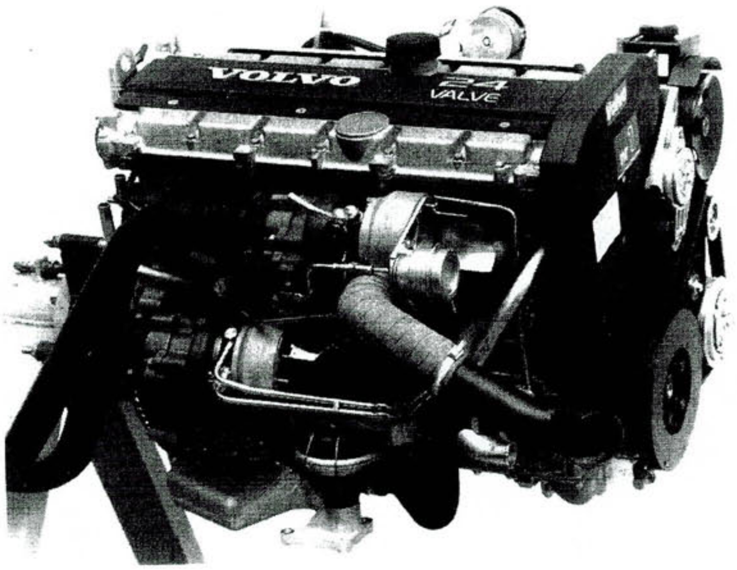

Fig 42 - The PCP engine

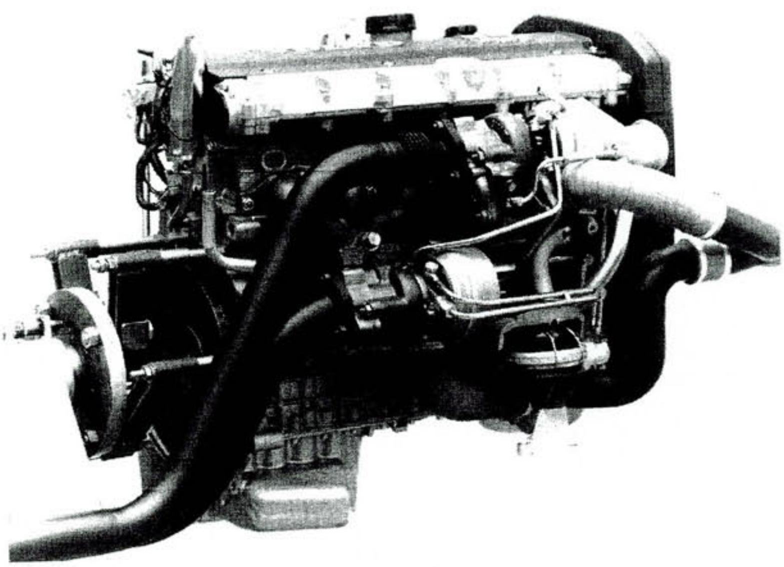

Fig 43 - The PCP engine

APPENDIX A - TECHNICAL DATA REFERENCE ENGINE

Engine Type:	B6304F 6-cylinder all aluminium block with cast in iron liners. DOHC, 24-valves (29° valve inclination), combustion chamber of pent roof type.
Bore:	83 mm
Stroke:	90 mm
Stroke To Bore:	1.08
Swept Volume:	2922 cm^3
Compression Ratio:	10.7:1
Valve dia intake:	31 mm
Valve dia exhaust:	27 mm
Fuel System:	Motronic 1.8 electronic fuel injection, microprocessor controlled, with two knock sensors and diagnosis system.
Fuel Quality:	95 RON unleaded gasoline.
Exhaust Emission System:	Threeway catalyst system with heated oxygen sensor.
Ignition System:	Direct ignition type, distributorless. Each spark plug is fired by an individual coil per cylinder. Fully electronic, microprocessor controlled system.
Maximum Power:	150 kW (204 Hp) at 600 rpm.
Maximum Torque:	267 Nm at 4300 rpm.
Engine weight:	182 kg
Maximum BMEP:	11.5 bar

APPENDIX B - SINGLE CYLINDER ENGINE DATA

Engine type:	AVL-based, single cylinder engine.
Cylinder head:	4-valve research unit, in aluminium.
Bore:	83 mm
Stroke:	79.5 mm
Displacement:	430 cm^3
Compressor ratio:	9.2:1 and 10.8:1
Charge of system:	Independent boost pressure and air temperature control.
Exhaust system:	Separate exhaust back pressure control.
Fuel injection:	Lab system controller with Bosch sequential injectors.
Ignition system:	Lab system/Bosch TSZ.
Spark plug:	Bosch type - nos 2 and 3 for M85, (CR 10.8 and CR 9.2 respectively). no 5 for 95 RON gasoline.

APPENDIX C - MEP ENGINE DATA

(Specification differences as compared to the reference VOLVO engine)

Type:	6-cylinder inline, two series sequential turbo chargers.
Displacement:	2472 cm^3
Bore:	81 mm
Stroke:	80 mm
Stroke to bore:	0.99:1
Compression ratio:	9.1:1
Boost pressure:	160-200 kpa abs, depending on fuel.
Turbo control:	NIRA system with Bosch components.
Fuel quality:	M85 (85% Methanol/15% unleaded gasoline) but FFV-capable to 100% 95 RON unleaded gasoline.
Fuel sensor:	Optical type.
Turbochargers:	
Primary unit:	Garrett T2; Compressor 0.48A/R 50; Turbine 0.47A/R 57.
Secondary unit:	Garrett T3; Compressor 69 mm/50; Turbine 0.63A/R 68.
Turbine priority valve:	Flat type, with electric servo operation.
Air priority valve:	Differential pressure type with selfregulating function.
Intercooler:	Air to air type.
Catalytic converter:	Two metal substrate catalysts, in parallel for low flow losses.
Exhaust system:	Low pressure loss 070 system.

Maximum power:	202 kW at 5450 rpm
Maximum torque:	406 Nm at 3000 rpm
	330 Nm at 1500 rpm
Maximum BMEP:	20.6 bar

Corresponding data for 95 RON unleaded gasoline are as follows:

Maximum power:	165 kW at 5300 rpm
Maximum torque:	350 Nm at 2800 rpm
	200 Nm at 1500 rpm

APPENDIX D - PCP ENGINE DATA

Type:	6-cylinder inline, two parallel turbochargers.
Valve mechanism:	DOHC, 4-valve
Displacement:	2922 cm^3
Bore:	83 mm
Stroke:	90 mm
Stroke/bore:	1.08
Compression ratio:	8.6
Boost pressure:	157 kpa abs
Valve timing	
Intake opens BTDC:	8^0
Intake closes ABDC:	44^0
Exhaust opens BBDC:	44°
Exhaust closes ATDC:	8°
Valve lift, intake and exhaust:	8 mm
Fuel ignition system:	Bosch Motronic 1.8 (semi sequential).
Turbo control system:	Fully electronic system.
Fuel quality:	95 RON unleaded gasoline.
Turbochargers:	Mitsubishi TDO4L
Compressor:	07B
Turbine housing:	6 cm^2
Exhaust temperature control:	VOLVO/NIRA-EGTC-system.
Intercooler:	Air to air type.
Catalytic converter:	Metal substrate 177.8 x 114.3 mm.
Max power:	205 kW at 5100 rpm.
Max torque:	414 Nm at 4000 rpm.
Max BMEP effective pressure:	17.8 bar.

APPENDIX E - TEST VEHICLE DATA

VEHICLE

Type:	VOLVO 760
Curbweight:	1560 kg
Wheelbase:	277 cm
Track f/r:	148/153 cm
Length:	485 cm
Width:	176 cm
Height:	141 cm
Wheels:	7.0 - 16"
Tires:	205/55 ZR16

TRANSMISSION

Automatic transmission with lock-up on 2:nd, 3:rd and 4:th gear.
Three shift modes, electronically controlled by engine management system.

Converter:	156 K
Ratios 1st:	2.804:1
2nd:	1.532:1
3rd:	1.000:1
4th:	0.754:1

Final ratio 3.54:1
Dana Trac-Loc Limited slip differential.
Stability control with throttle retard and fuel cut-off.

VEHICLE PERFORMANCE PEP ENGINE

0-40 km/h	2.2 s
0-100 km/h	6.8 s
Top speed	232 km/h