

Development of a Turbocharged Direct Injection Downsizing Demonstrator Engine

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ABSTRACT

This paper describes the initial development of a 3 cylinder 1.2l technology demonstrator engine from MAHLE. The purpose of this highly turbocharged direct injection engine is to demonstrate production-ready technologies that enable low CO₂ emissions via downsizing by 50%.

Downsizing is one of the most proven paths to CO₂ emission reduction. By using careful design, a 2.4 l engine can be replaced by a 1.2l engine that has superior torque at all speeds and on-road fuel consumption benefits of 25 - 30%.

A two-stage turbocharging system has been developed for the engine to enable good transient response and the high torque levels at all engine speeds demanded by a downsizing approach. Several options were tested and the final system exceeds the 30bar peak BMEP target with stoichiometric fuelling. Indeed, $\lambda = 1.0$ fuelling is maintained over the majority of the full-load line and the 144kW peak power requirement is fulfilled at only 6000 rpm.

Test data are presented for both steady state and transient performance as well as vehicle fuel economy simulation results. Discussions on combustion system development and emissions performance status are included.

INTRODUCTION

Since the agreement within the EU to reduce average new car CO₂ emissions to 140g/km by 2008, fuel consumption improvement has been one of the main drivers for engine development within the automotive industry. Current status of the EU fleet indicates that significant work is still required to achieve this target, particularly in the field of gasoline spark ignition engines.

MAHLE have recently designed and built [1] an engine to demonstrate the application of aggressive downsizing to gasoline engines in order to improve operating efficiency. the engine was also designed to serve as a platform for addressing the technical challenges associated with downsizing, chiefly:

- A robust combustion system that allows a high compression ratio to maintain part load efficiency
- Good low speed torque and transient performance
- Real world fuel consumption benefits through a reduction in full load fuel enrichment
- Base engine robustness and durability



Figure 1: The downsized concept engine showing highly integrated design of intercooler, oil cooler/filter module and engine-mounted air cleaner.

These challenges require unique engine solutions to enable the significant efficiency improvements of an aggressively downsized engine to be realized [2,3,4]. In addition, technologies from across the business group were used to create highly integrated and low-cost sub-system designs.

This paper summarizes the development status of the new engine. Four of these 1.2l, 3-cylinder high specific output gasoline downsize engines have now been built and are undergoing testing in line with the MAHLE Powertrain Development Verification Plan (DVP).

The main purpose of the first phase of testing is to demonstrate the potential of the combustion system and boosting system to deliver the required performance levels. If this is successful, then the engine will go on to provide a research tool for the development of new technologies by the business group.

ENGINE DESIGN TARGETS

The design was focused on the use of technology which would be ultimately available for mass production. The targets were to produce an engine capable of replacing a typical 2.4l V6 PFI engine used in a C or D class European vehicle platform (circa 1600kg). The engine would be compliant with EU5 / ULEV2 emissions from conception.

The engine was initially conceived as a high technology demonstrator with a two-stage turbo-charging system. The two-stage system is shown schematically in Figure 2 and utilizes a small high pressure turbocharger in series with a larger low pressure turbocharger. A bypass for the high pressure turbine (effectively a full-flow wastegate) is provided to allow the worksplit to be varied between the two turbochargers as desired and for the high pressure unit to be completely bypassed at high power if required.

However, a variant was also designed and produced at a lower performance level. This derivative is considered an intermediate step to the twin turbo solution. Table 1 shows the basic performance and fuel economy targets for the twin turbo and single turbo engine derivatives.

Table 1 - Performance targets

Target Performance at Full Load		
Single turbo:		
Torque		
Maximum at n = 2,500 - 3,000 rpm	$T_{max} = 210 \text{ Nm}$	(BMEP = 22 bar)
Power Output		
Maximum at n = 6,000 rpm	$P_{max} = 90 \text{ kW}$	(Specific Power = 75 kW/l)
Twin turbo:		
Torque		
at n = 1,000 rpm	$M_d = 153 \text{ Nm}$	(BMEP = 16 bar)
Maximum at n = 2,500 - 3,000 rpm	$M_d = 286 \text{ Nm}$	(BMEP = 30 bar)
Power Output		
Maximum at n = 6,500 rpm	$P_{max} = 144 \text{ kW}$	(Specific Power = 120 kW/l)
Fuel Consumption at Part Load		
Optimum	$bsfc = 235 \text{ g/kWh}$	
at n = 2000 rpm / BMEP = 4 bar	$bsfc < 295 \text{ g/kWh}$	
Emissions		
EU5 / ULEV2 compliant		
Hydrocarbon emissions		
at n = 2000 rpm / BMEP = 2 bar	$< 5.4 \text{ g/kWh}$	
at n = 2000 rpm / BMEP = 4 bar	$< 4.4 \text{ g/kWh}$	

Figure 2 also shows the target torque curve for the two-stage turbo derivative.

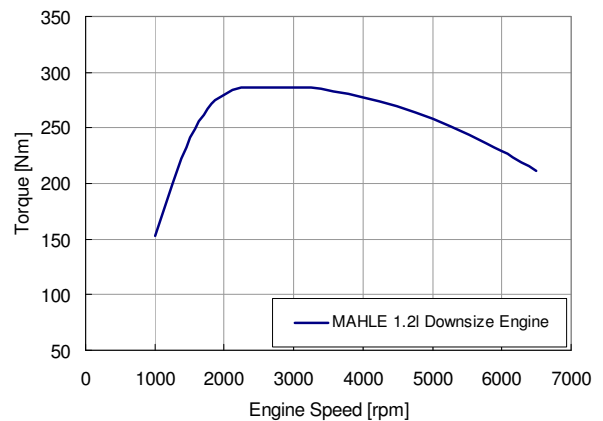
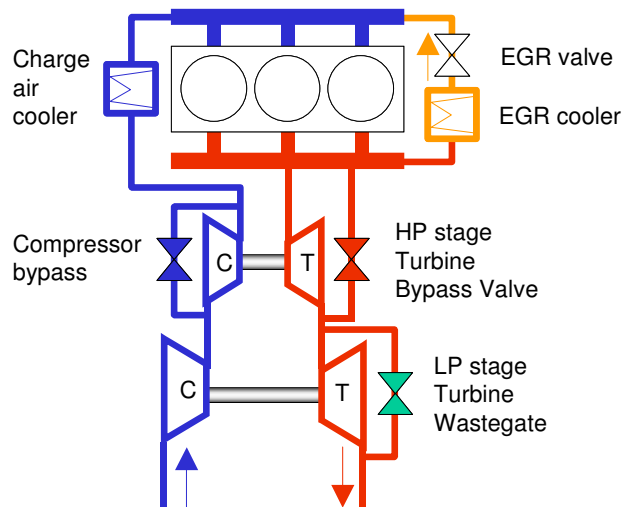


Figure 2: Two stage boost system schematic and target torque curve

ENGINE SPECIFICATION

Table 2 shows the most important mechanical design features and dimensions of the base engine and Figure 1 shows a general view of the complete twin turbocharged engine.

The key combustion technology features incorporated on the engine include centrally mounted piezo fuel injectors, high boost capability with an integrated, engine mounted intercooler, dual VVT, lightweight fabricated valves with sodium cooling for the exhaust, high energy ignition coils and carefully designed head cooling with high heat transfer coefficients on the exhaust side.

Table 2 – Base Engine Data

Main Data, Cylinder Block And Cranktrain	
Cylinder block material	A 356 (COSCAST)
Bore	83,0 mm
Stroke	73,9 mm
Stroke-bore ratio	0,89
Cylinder displacement	0,400 l
Cylindner number and arrangement	3 in line
Displacement	1.200 l
Bore spacing	91 mm
Conrod length	123 mm
Block height	189,5 mm
Compression ratio (max)	9,75
Firing order	1-3-2
Crankshaft main journal diameter	48 mm
Crankshaft pin journal diameter	48 mm
Cylinder Head and Valvetrain	
Cylinder head material	A 356 (COSCAST)
Combustion chamber configuration	4 valves/cylinder, pentroof
Valvetrain configuration	DOHC, roller finger follower
Fuel injection	Central GDI Piezo Injector
Valve head diameter (Inlet/Exhaust)	31,4 / 25,5 mm
Valve stem diameter (Inlet/Exhaust)	6,0 / 6,0 mm
Max. valve lift (Inlet/Exhaust)	11,0 / 11,0 mm
Valve angle (Inlet/Exhaust)	21,5 / 20,0 °
Overall Dimensions	
Length x Width x Height	438x675x706 mm
Dry weight - fully dressed	145 kg

HIGH LOAD COMBUSTION SYSTEM

Once the first engine was commissioned and the control system validated, a comprehensive series of experiments was undertaken in order to develop the combustion system, particularly in the area of high BMEP operation. Initial work with designed experiments was used to find settings for parameters with lower magnitude responses that would enable robust operation while investigating the higher magnitude responses and hardware effects. These parameters, such as fuel rail pressure, injector lift and injection phasing thus remain fixed to mapped values in the following discussions, unless specifically noted.

Spark timing in all cases is set to MBT or BLD. Air-fuel ratio is set to an exhaust lambda of 1.0 unless otherwise noted.

Figure 3 below shows an example of mapping a high magnitude response – in this case the influence of inlet cam timing on BMEP and combustion phasing at increasing manifold pressures. It can be seen in this case (3000 rpm) that the best BMEP is achieved at advanced inlet cam timing. The data presented in later figures have all been taken at values of inlet and exhaust camshaft timing that were optimized in this way.

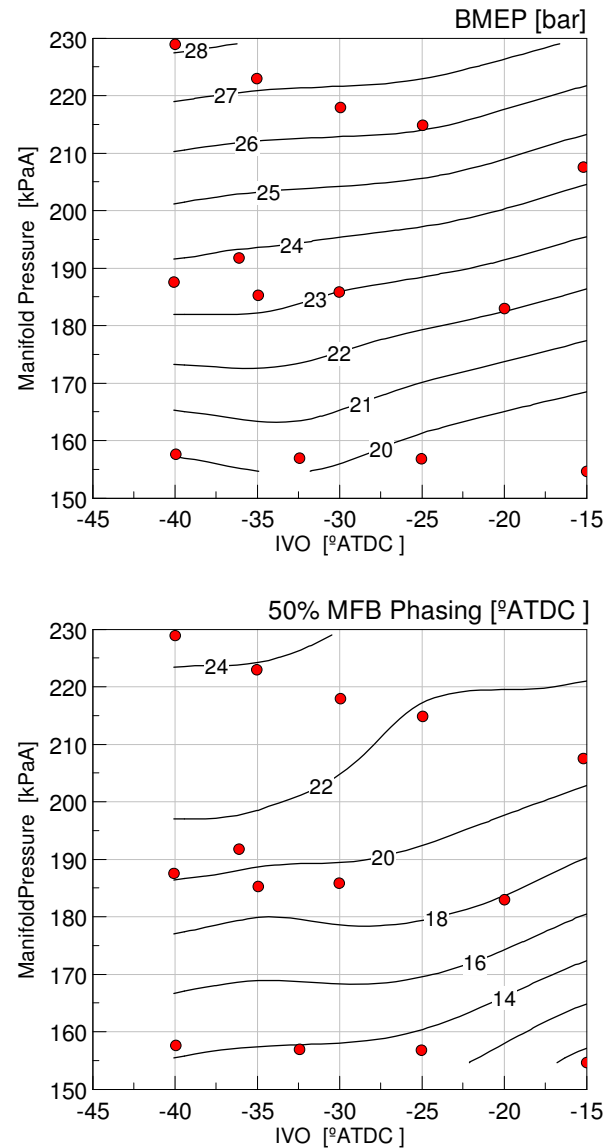


Figure 3: An example of camshaft position optimization for performance at 3000rpm. Results at BLD spark timing, $\lambda=1.0$.

Based on MAHLE Powertrain's considerable experience with turbocharged and GDI engines, a compression ratio in the range between 9:1 and 10:1 was expected to be suitable for this application and compression ratio selection was one of the earlier tasks undertaken.

A large amount of data was collected for this purpose, however the critical point in the analysis is expressed in Figure 4. As increasing levels of boost were applied to the engine there were three main factors that could limit the ultimate performance. The first was the cylinder pressure limit, set as 120 bar for average Pmax and 140 bar for maximum Pmax (although this was more flexibly applied!). The second was the turbocharger's ability to supply boost either as a shaft speed limit or turbine power limit. The third was the onset of unstable combustion.

Figure 4 shows some results from a boost pressure 'sweep' at 3000 rpm. It can be seen that for both compression ratios, the mean Pmax increases with increasing BMEP and that the lower compression ratio suffers generally higher Pmax levels. At 26 bar, however, it can be seen that the minimum Pmax drops to a very low level for the 9.7:1 CR case and this indicates the onset of combustion instability. Examination of the cylinder pressure logs show a number of misfires and very late burns with the final two logs in this sweep.

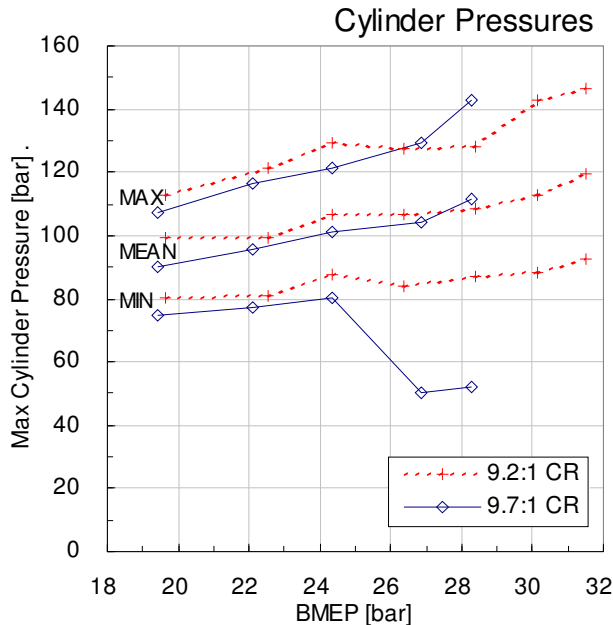


Figure 4: Comparison of two compression ratios at 3000rpm. Mean and limit peak cylinder pressures over a 300 cycle sample. BLD spark timing, $\lambda=1.0$.

Although not shown here, the max Pmax line was often also the indicator of instability. Pre-ignition events have been encountered at very high boost levels, particularly below 2000 rpm. These could easily achieve 220 bar cylinder pressure, however are rarely seen in groups of more than 10 at any time. The robustness of the engine design to these events has been excellent.

Figure 5 shows the burn angle data from the same two sweeps. The higher compression ratio exhibits slightly later combustion phasing (assessed via the 50% mass fraction burned timing) but generally faster burn times. The statistics at the highest BMEP point are suspect due to the number of late burns, but the overall combustion properties do not look significantly different apart from what would be expected due to the compression ratio difference.

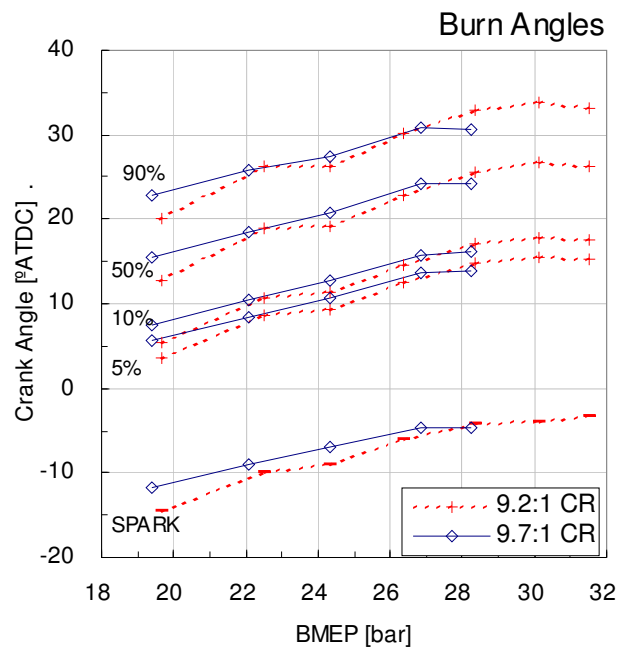


Figure 5: Comparison of two compression ratios at 3000rpm. Average mass fraction burned angles over a 300 cycle sample. BLD spark timing, $\lambda=1.0$.

More detailed investigations have revealed that the misfiring cycles are associated with a breakdown in spark current. Further work is planned in this area using an optical spark-plug to gather ignition and early burn data. We anticipate that an ignitability issue will be solvable with increased coil energy or plug configuration, however as we are running with 100mJ coils already, there may be limits to what is possible. In any case, the onset of instability provides a useful metric for us to compare different hardware levels.

As a higher compression ratio is desirable for fuel consumption and low-load stability reasons, some effort was taken to look at improving the high-load combustion at 9.7:1. One of the most effective parameters investigated was exhaust cam duration. Due to the very high cylinder pressures and long exhaust cam duration selected for peak power requirements, the exhaust valve can be opening at a very high cylinder pressure. At 3000rpm, 30 bar BMEP the cylinder pressure can be up to 15bar at EVO and this causes exhaust manifold pressure spikes up to 6 bar(abs) during blow-down. This was seen as potentially disruptive to other cylinders and overall gas exchange.

A shorter duration cam profile was investigated and the results are shown in Figure 6. It can be seen that the 240° cam does indeed delay the onset of misfire until greater than 28bar BMEP is achieved. However, Figure 7 shows that the effect at 6000rpm was unacceptable. The shorter cam causes the turbine temperature limit of 1025°C to be reached at only 22 bar BMEP, where the target BMEP is 24bar. The pre-turbine pressures are also 40kPa higher. The exercise proved a useful exercise for correlation of the GT-Power 1-D gas exchange modelling performed during the design phase.