

Challenges for Increased Efficiency through Gasoline Engine Downsizing

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ABSTRACT

In order to achieve the required future CO₂ reduction targets, significant further development of both gasoline and diesel engines is required. One of the main methods to achieve this with the gasoline engine in the short to medium term is through the application of engine downsizing, which has resulted in numerous downsized engines already being brought to production. It is, however, considered that there is still significant further CO₂ reduction potential through continued development of this technology.

This paper considers the future development of gasoline engine downsizing in the short to medium term and the various technologies that can be applied to further increase the efficiency of operation. As such this paper covers, among other areas, fundamental engine layout and design, alternative boosting systems, methods of increasing part load efficiency and vehicle modelling, and uses analysis tools and engine test results to show the benefits achievable.

INTRODUCTION

It is well known that reducing carbon dioxide (CO₂) emissions has been the main driver in passenger vehicle powertrain technology over recent years, and it is expected that this is going to continue. In recent times this has largely been driven by evolving emissions legislation and taxation strategies, however increasing public awareness about the impact of global warming and the rising cost of crude oil can also be seen to be

shaping consumer choices for low fuel consumption vehicles.

European Union (EU) fleet fuel economy data has shown that over the past 14 years there has been a continual reduction in drive cycle CO₂ emissions, largely through developments in diesel engine technology and increased diesel sales [1]. In addition in the UK there has been an increase in sales of smaller vehicles, with inherently better fuel economy. In order to meet future global emissions goals, in the short to medium term it will be necessary for significant further development of the gasoline engine. Recent studies into the development of future powertrains and the potential improvements in fuel economy that can be achieved report that spark ignition engines have more potential improvement than compression ignition engines and predict that the relative advantage of diesel over gasoline engines is going to be reduced [2-4].

In recent years a range of different technologies have been developed for gasoline engine fuel economy improvement, with the most significant being:

- Gasoline direct injection (GDI) (Homogenous and stratified lean)
- Exhaust gas recirculation (EGR)
- Variable valvetrains (camshaft phasing, profile switching, variable lift and duration systems)
- Controlled auto ignition (CAI) or homogeneous charge compression ignition (HCCI)
- Gasoline engine downsizing
- Friction reduction

Of these, gasoline engine downsizing has been shown to have the most immediate potential for fuel economy improvement. This has led to numerous turbocharged engines being introduced into production, initially in Europe, but now also in the US, with claims of drive cycle fuel economy improvement between 10 and 17% [5-7]. Significant research work is continuing in this area with the aim of further fuel economy improvements through either direct efficiency improvements or increased degrees of engine downsizing [8-13]. Figure 1 shows the performance levels achieved by GDI and port fuel injection (PFI) downsize engines currently in production. In comparison the current status of the MAHLE Downsize Demonstrator engine [14] is shown with an additional curve showing further development potential as an indication of what is achievable with increased levels of downsizing.

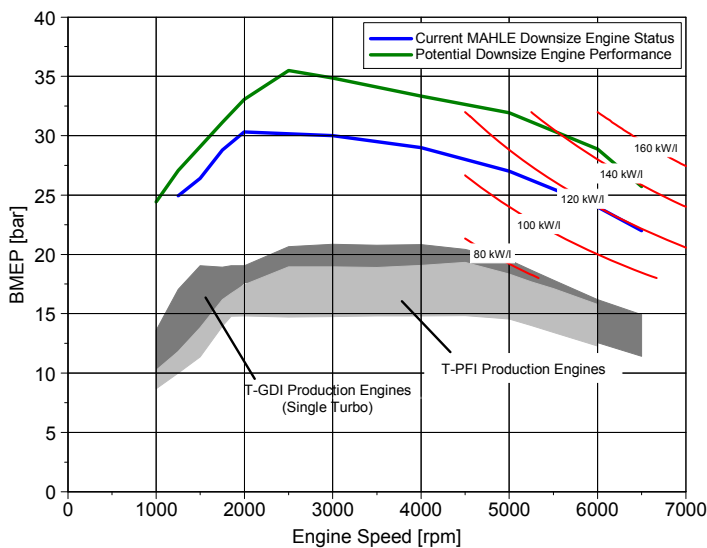


Figure 1: Gasoline downsize engine full load performance

The main technical challenges to being able to realize the further development of downsized engines are:

- Combustion limitations – Increased propensity to knock leads to reduced compression ratio and retarded spark timing, hence lower efficiency
- Steady state low speed torque – With increased downsizing, low speed BMEP requirement increases to maintain acceptable performance
- Transient performance – Transient response needs to be maintained with increased low speed torque requirement
- Engine geometry/layout – As engine capacity falls below ~1 litre, bore size and/or cylinder number will require re-optimization
- Part load fuel economy – As downsizing continues the fuel economy gains inherently reduce, measures will need to be taken to mitigate this

This paper looks at the development of gasoline engine downsizing in the short to medium term with regard to more widespread application of the technology and to the possibility of further efficiency improvements.

COMBUSTION SYSTEM

DIRECT INJECTION

As engine specific output is increased the issue of knocking combustion takes on greater significance, requiring the compression ratio to be reduced in order to maintain acceptable combustion phasing at all loads. As the compression ratio is reduced so is the indicated efficiency at part load leading to an increase in part load brake specific fuel consumption (BSFC). In order to achieve the best improvement in fuel economy the compression ratio needs to be maintained as high as possible.

This requirement has led to the widespread introduction of direct injection to turbocharged engines. Injecting the fuel directly into the combustion chamber, as opposed to the port, reduces the charge temperature in cylinder through the latent heat of vaporisation of the fuel. This has the combined effect of both increasing charge density, and hence volumetric efficiency, and also moving the knock limit of the combustion system. As a consequence it is typically expected that the compression ratio can be increased by at least 1 for an equivalent specific power output, with the associated increase in efficiency (figure 2).

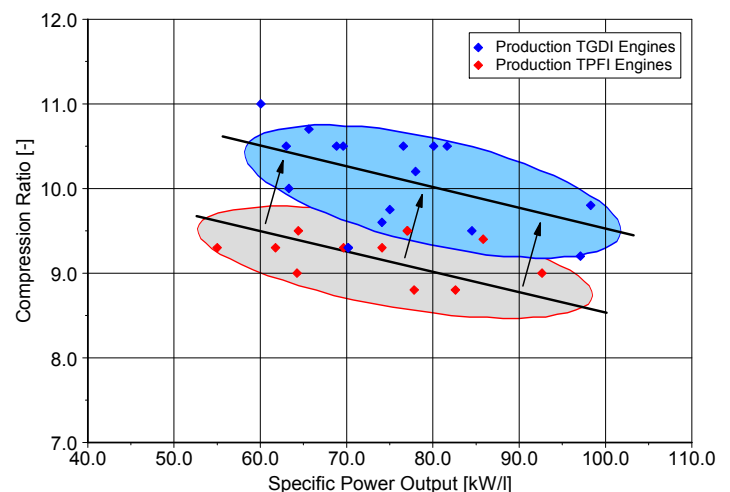


Figure 2: Compression ratio against specific power output for production turbocharged engines

Additional advantages are realised on a turbocharged direct injection engine through the synergy with variable camshaft phasing [8, 15]. Due to the positive pressure ratio across the engine that occurs at low speed on a turbocharged engine, significant blow-through will occur

when large amounts of valve overlap are used. On a port fuel injected (PFI) engine this will result in an unburned air/fuel mixture in the exhaust, causing both significant engine out hydrocarbon emissions, possible catalyst damage and increased fuel consumption. By injecting after the exhaust valve has closed direct injection engines are not limited in this way, and so can utilise high levels of overlap to achieve complete scavenging of the combustion chamber. This further improves the knock limit at low speed / high load and moves the operating point to a more favorable position on the compressor map improving low speed torque.

The combined advantage that can be achieved through these technologies can clearly be seen in figure 1, with the increase in low speed torque of the GDI over the PFI production engines, whilst simultaneously increasing the specific power output.

Until recently the majority of production DI combustion systems have been first generation side direct injection. There is now however a gradual move to second-generation central direct injection systems (so-called 'Spray Guided' or SGDI). Figure 3 shows a typical view of the two different combustion system layouts.

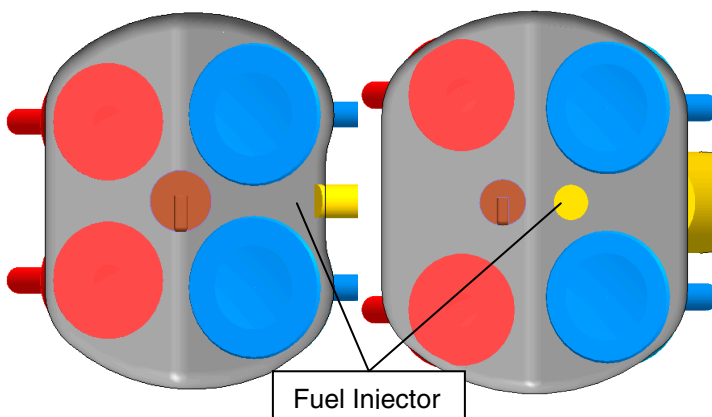


Figure 3: Typical side and central DI combustion system layouts

The main benefits of the SGDI combustion system are largely possible due to the location of the injector next to the spark plug. The close proximity of the injector and spark plug allows more accurate air fuel mixture control, within the combustion chamber and at the spark plug, through the variation of injection timing. Further benefits are achievable when multiple injection events are used. SGDI combustion systems can be used with both solenoid multi-hole and outward opening piezo injectors.

The same strategy of split injection with SGDI can be further utilized to improve the knock limit at full load, and it is also an enabler to run stratified lean.

Figure 4 shows results from a chilled fluids cold start spark retard (CSSRe) test, that is used to assess the performance of a combustion system for catalyst light-off. For this testing the engine is run at 1350rpm, 1 bar BMEP and with a combustion stability limit of 0.25bar standard deviation of IMEP (SDIMEP). By utilizing a split injection strategy with an early injection event to provide a homogeneous lean mixture within the combustion chamber, and then a late injection event to provide a rich mixture at the plug at the time of ignition, very high levels of heat flux can be achieved. This increased heat flux can be utilized to reduce the time to achieve catalyst light-off, or to offset the increased thermal inertia in the exhaust with additional turbochargers.

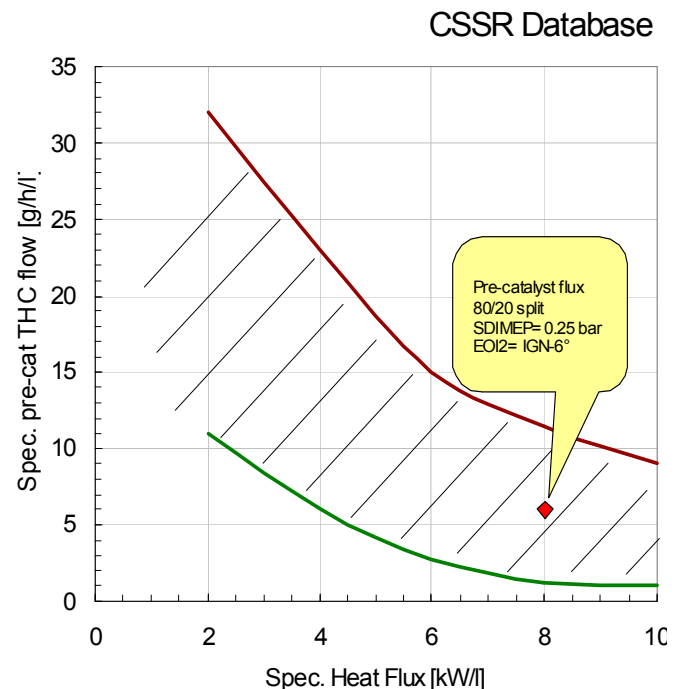


Figure 4: CSSRe capability

COMBUSTION CHAMBER LAYOUT

As can be seen in figure 3, packaging both the injector and the spark plug in the centre of the combustion chamber will influence the combustion chamber layout. The critical requirement is to maintain sufficient cooling around both the spark plug and the injector tip. If sufficient cooling is not maintained then the spark plug heat range will need to be changed, which could have an influence on cold start capability. In addition, if the injector tip temperature is not controlled then there is the danger of coking degrading performance over time.

As the level of downsizing is increased, then ensuring adequate cylinder head cooling becomes more critical as pressures and temperatures in the combustion

chamber increase. Figure 5 shows a section through a high specific output SGDI cylinder head, where the cooling jacket has been designed to completely surround the plug and injector for optimum cooling.

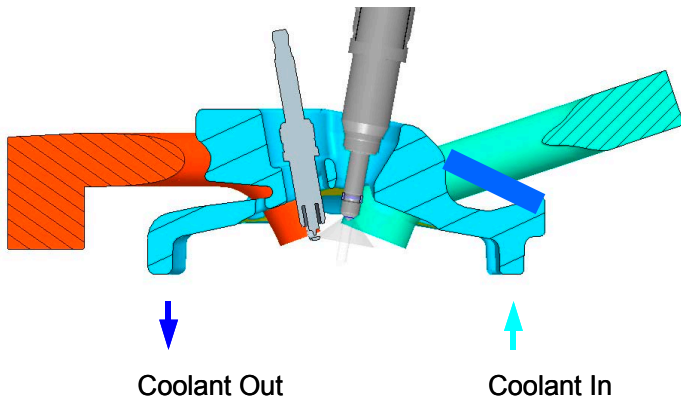


Figure 5: SGDI cooling jacket

For larger capacity engines (bore sizes > 80mm) SGDI combustion system packaging can typically be achieved without significantly compromising the gas exchange and hence performance of the engine. However, as there is a continued trend towards more efficient, smaller capacity engines bore sizes could reduce and the influence of packaging a central injector will increase (an alternative would be to reduce the number of cylinders, although this is less desirable for reasons of NVH and transient response).

In order to investigate the potential of reducing bore size, an analytical study has been carried out for a 800cc, 3-cylinder engine to show the effect of bore size and injector location on full load performance. The 1D-cycle simulation code GT-Power has been used to show the influence as valve sizes change with bore diameter. Target performance was set to achieve in excess of 21bar BMEP with a specific power output of 85 kW/l.

Typical side and central injection combustion systems were designed with valve sizes optimized for the given bore size. The central injection system was designed to ensure sufficient cooling around both the injector and spark plug (as figure 5), with standard cast wall and minimum water jacket thicknesses.

In the study the bore diameter was swept from 83 to 58mm and a simulation was carried out at the peak power speed (6000rpm). Figure 6 shows a schematic of the combustion chamber with 83 and 58mm bore diameter for both the side and central injector location. It is clear to see from these schematics the influence of the injector position on valve position and sizing, particularly as the bore size is reduced. Figure 7 shows

the differences in valve sizes with bore diameter, which shows clearly the offset between the combustion chamber configurations.

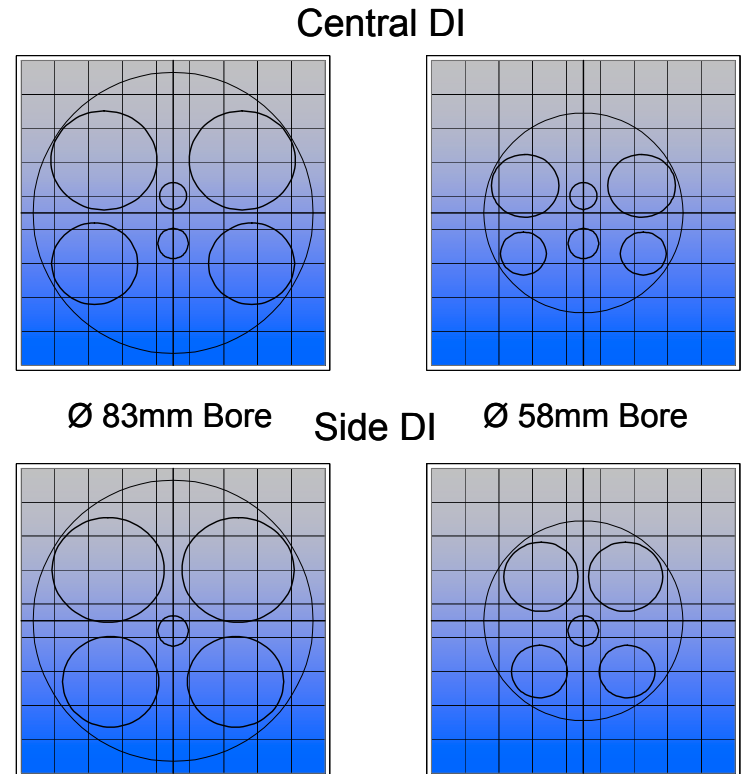


Figure 6: Combustion chamber schematics with side and central injector location

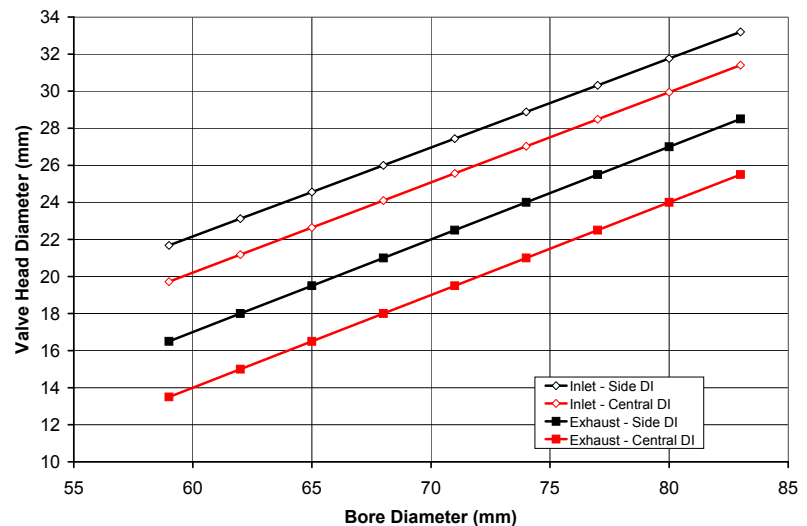


Figure 7: Valve size diameter against bore diameter

GT-Power analysis was carried out at the peak power engine speed (6000rpm) using the combustion chamber geometries above. At each of the speed sites a sweep of BMEP was carried out. Figures 8 and 9 show the contour plots of specific power, Exhaust

compressor pressure ratio and wastegate diameter against bore diameter and BMEP, for both injector locations.

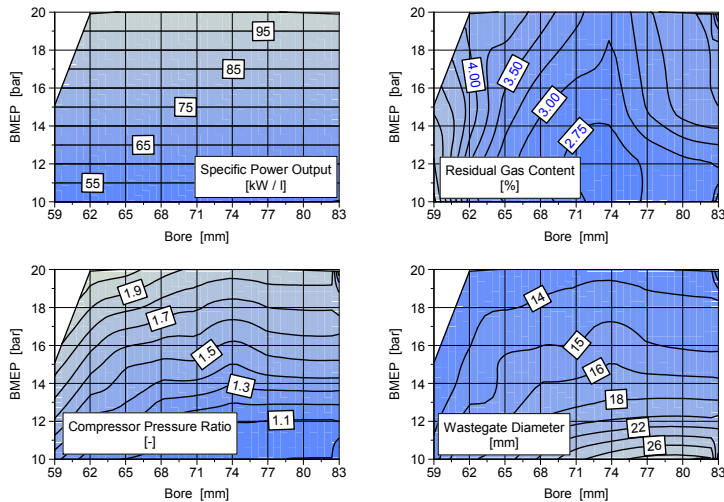


Figure 8: Effect of bore diameter on engine performance at 6000rpm with side injection location

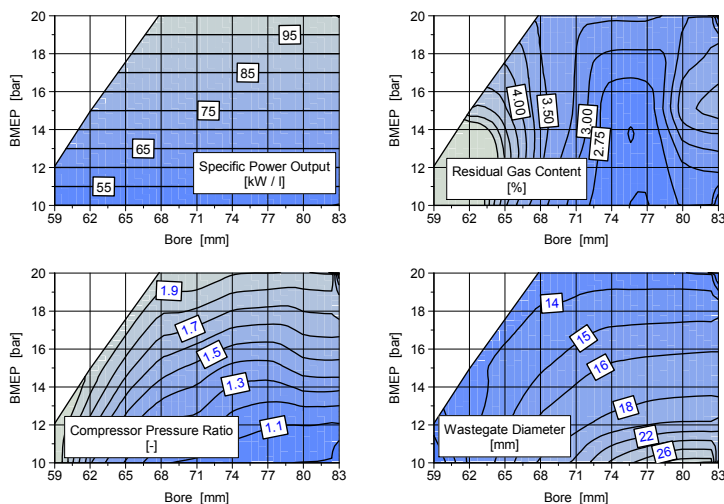


Figure 9: Effect of bore diameter on engine performance at 6000rpm with central direct injection

Comparing figures 8 and 9 shows the increasing influence of the valve size restriction as bore diameter is reduced. With the central direction injection system the point at which the full load performance can no longer be achieved occurs earlier, limiting the absolute specific output of the engine. This is also reflected in the higher compressor pressure ratio that is required in order to overcome the increased restriction, which is also manifested in an increase in exhaust residuals as pre-turbine pressure also increases. This would have the knock-on effect of increasing the knock sensitivity of the engine and hence possibly limiting the compression ratio that can be achieved as the bore size is reduced.

However, even in this scenario it can be seen that it is possible to achieve 85 kW/l with the central GDI combustion system down to 66mm bore diameter, which would give an under-square bore/stroke ratio of 0.85. If the bore diameter were increased to give a square bore/stroke ratio, this would reduce the restriction of the engine and allow more efficient operation at high speed.

Although the performance can be achieved, this study highlights the need to carefully consider the combustion chamber design, particularly with small bore diameters, if the advantages of a central direct injection combustion system are to be realized. If the engine were also intended for naturally aspirated operation, the increased restriction would limit the peak performance achievable.

BOOSTING SYSTEMS

Both steady state low speed torque and transient response are critical if any level of engine downsizing is going to be considered acceptable by the customer [16-20]. Currently single turbocharger systems are the predominant boosting system for downsized gasoline engines, with the VW twincharger the exception. Diesel engine technology is moving to 2-stage turbocharger systems as the specific power output (degree of downsizing) is pushed ever higher.

By utilizing the complimentary technologies of turbocharging, GDI and VCT (as previously discussed) significant improvements in single turbocharger low speed torque and transient response have already been achieved. This has enabled acceptable performance with a single turbocharger to be achieved with specific power outputs up to 80kW/l. The performance of the single turbocharger is, however, always going to be restricted by the availability of exhaust energy at low engine speeds and also the compressor map width. Therefore, in order for greater degrees of engine downsizing to be considered acceptable to the customer, additional boosting technology is required to address the issues of steady state low speed torque and transient response.

Figure 10 shows schematics of 4 different engine boosting systems.

- A) Single Turbocharger – Current standard boosting system with a single compressor and turbine. In order to improve the performance the technology has developed to twin scroll, variable flow turbines, variable geometry turbines etc; however they all reach a limit in performance.
- B) 2-Stage Turbocharger – Utilises a small, high pressure (HP) turbo for low speed in series with a large low pressure (LP) turbo for high speed.

Compressor and turbine by-pass valves are required for the HP turbo, to prevent overspeed / restriction as mass flow is increased.

C) Supercharger & Turbocharger – Utilising a supercharger (typically roots type) in series with a larger turbocharger. Supercharger is used at low engine speed and then bypassed as speed increases and turbo has spooled up. Bypass valve and drive system with clutch are required for the supercharger.

D) Electric Supercharger & Turbocharger – Combines an electrically driven compressor installed up- or downstream of a large turbocharger. The electric supercharger is used at low engine speeds in series with the main turbocharger and then bypassed once the main turbocharger has spooled up.

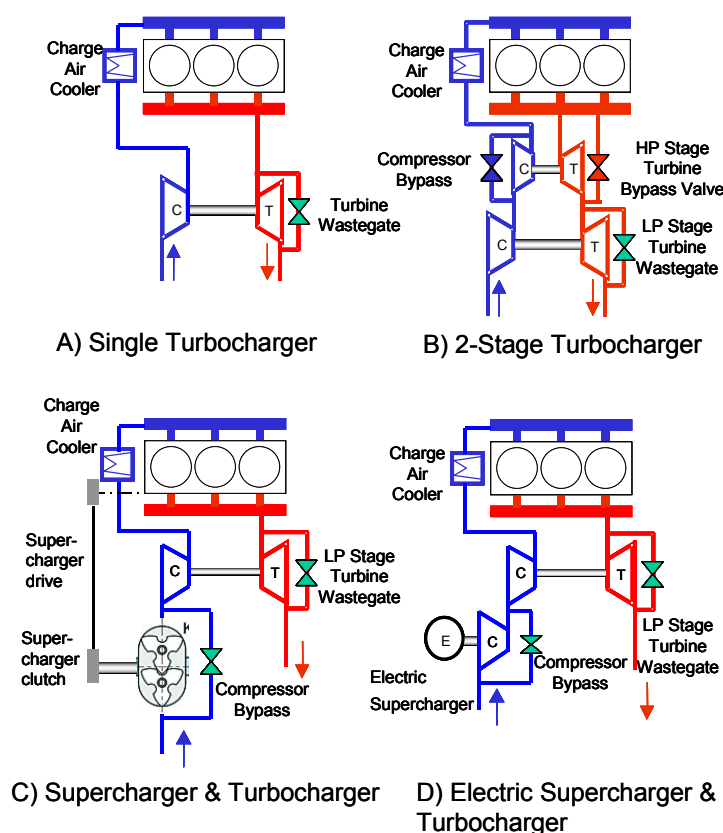


Figure 10: Boosting system schematics

In order to assess the relative performance of different boosting systems under both steady state and transient conditions, testing has been carried out with the MAHLE Downsizing engine. The engine has been developed to achieve full load performance targets of 30 bar BMEP and a specific power output of 120 kW/l.

As part of the engine boosting system development a range of different turbochargers have been tested in

both single and 2-stage configurations, to enable the targets to be achieved. For electric supercharger comparison testing Controlled Power Technologies (CPT) kindly agreed to supply their 2nd Generation 12V VTES (Variable Torque Enhancement System) Electric Supercharger. This uses a switched reluctance direct drive motor to power a centrifugal compressor. The VTES unit was installed upstream, in series with the low-pressure turbocharger that had been selected for the 2-Stage turbocharger configuration. It should be noted that the VTES compressor had not been optimized for this engine but had been carried over from a previous application.

Figure 11 shows the steady state BMEP curves for the LP turbo only, 2-stage turbocharger and VTES & LP turbo combined. With the LP turbo only it can be seen that the low speed BMEP falls significantly below the target, with the 30 bar BMEP target not being achieved until 3000rpm. The 2-stage turbocharger system can be seen to exceed the original low speed BMEP targets that had been set for the engine, achieving in excess of 23 bar BMEP at 1000rpm and reaching 30 bar BMEP by 2000rpm. In effect the addition of the HP turbo in series has moved the run-up line forward by 1500rpm. The addition of the VTES unit can also be seen to significantly improve the low speed BMEP, although not quite able to achieve the original target curve. Under the steady state conditions the low speed torque is increased by greater than 55%.

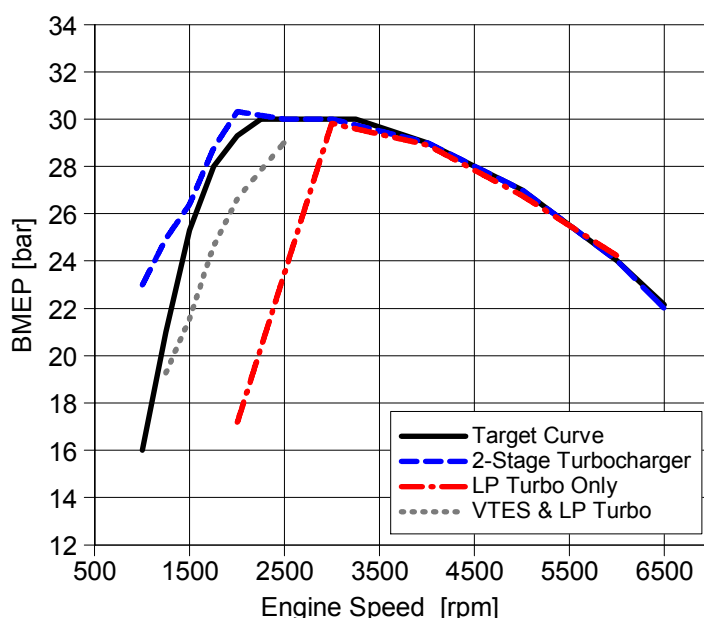


Figure 11: Steady state full load curves with different boosting systems

Figure 12 shows a comparison of the torque curves against a production 2.0l TGD engine and 2.4l (twice the capacity of the 1.2l downsizing engine).