

Ultra Boost for Economy: Extending the Limits of Extreme Engine Downsizing

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ABSTRACT

The paper discusses the concept, design and final results from the 'Ultra Boost for Economy' collaborative project, which was part-funded by the Technology Strategy Board, the UK's innovation agency. The project comprised industry- and academia-wide expertise to demonstrate that it is possible to reduce engine capacity by 60% and still achieve the torque curve of a modern, large-capacity naturally-aspirated engine, while encompassing the attributes necessary to employ such a concept in premium vehicles.

In addition to achieving the torque curve of the Jaguar Land Rover naturally-aspirated 5.0 litre V8 engine (which included generating 25 bar BMEP at 1000 rpm), the main project target was to show that such a downsized engine could, in itself, provide a major proportion of a route towards a 35% reduction in vehicle tailpipe CO₂ on the New European Drive Cycle, together with some vehicle-based modifications and the assumption of stop-start technology being used instead of hybridization. In order to do this vehicle modelling was employed to set part-load operating points representative of a target vehicle and to provide weighting factors for those points. The engine was sized by using the fuel consumption improvement targets and a series of specification steps designed to ensure that the required full-load performance and driveability could be achieved.

The engine was designed in parallel with 1-D modelling which helped to combine the various technology packages of the project, including the specification of an advanced charging system and the provision of the necessary variability in the valvetrain system. An advanced intake port was designed in order to ensure the necessary flow rate and the charge motion to provide fuel mixing and help suppress knock, and was subjected to a full transient CFD analysis. A new engine management system was provided which necessarily had to be capable of controlling many functions, including a supercharger engagement clutch and full bypass system, direct injection system, port-fuel injection system, separately-switchable cam profiles for the intake and exhaust valves and wide-range fast-acting camshaft phasing devices.

Testing of the engine was split into two phases. The first used a test bed Combustion Air Handling Unit to enable development of the combustion system without the complication of a new charging system being fitted to the engine. To set boundary conditions during this part of the programme, heavy reliance was placed on the 1-D simulation. The second phase tested the full engine.

The ramifications of realizing the engine design from a V8 basis in terms of residual friction versus the fuel consumption results achieved are also discussed. The final improvement in vehicle fuel economy is demonstrated using a proprietary fuel consumption code, and is presented for the New European Drive Cycle, the FTP-75 cycle and a 120 km/h (75 mph) cruise condition.

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INTRODUCTION

In terms of light-duty passenger car fuel consumption, anything that can reduce light-load losses in the 4-stroke cycle engine is especially important on several of the common legislative drive cycles such as the New European Drive Cycle (NEDC) and proposed harmonized World Light-Duty Test Cycle (WLTC). Many routes have been and are being investigated with respect to reducing fuel consumption, most of which in the case of Otto (spark-ignition 4-stroke) engines target the reduction of throttling losses inherent in the cycle. Hybridization, homogeneous charge compression ignition, variable valve timing, exhaust gas recirculation and dilute combustion (including lean operation and with EGR) all provide benefits in this area to some degree, together with other effects. Of these, lean operation (either stratified or homogeneous) offers distinct thermodynamic advantages, but it presents challenges for exhaust after treatment (EAT). This is because of the associated emission of oxides of nitrogen carried in exhaust gas which contains excess oxygen. The EAT required is also very sensitive to fuel quality (specifically sulphur content), currently making it a region-specific technology.

Of all the technologies recently investigated, engine 'downsizing' is especially beneficial in reducing throttling losses and thus has rapidly become a megatrend in the automotive industry. While primarily associated with spark-ignition (SI) engines, it also gives benefits in compression ignition (CI) engines where, while the throttling loss reduction may be absent, the improvement in mechanical efficiency that results from operating the engine at increased load is significant because of the increased friction that diesel engines carry versus their gasoline counterparts. As the present work will show, friction reduction (as well as reduced thermal losses) is a very important aspect of extreme downsizing.

The attractions to the OEM of gasoline engines include that their technology is very cost-effective to produce versus diesel engines (especially when the costs of the EAT system are included), that there are still significant efficiency gains to be made due to the losses associated with the four-stroke cycle

[1], and that pursuing the technology does not entail the writing-off of existing production facilities (as would be required by a quantum shift to electric or fuel-cell vehicles, for instance). As such, SI engine downsizing has become firmly established as the primary route for the automotive industry to improve vehicle fuel consumption, providing as it does an affordable solution to the twin issues of reducing tailpipe CO₂ and improving fuel economy while also providing improved driveability from engines.

Commercial View

Most downsized gasoline engines offered to the market place at present appear to have a 'downsizing factor' of about 35-40%. The downsizing factor is defined here to be

$$DF = \frac{V_{Swept_{NA}} - V_{Swept_{Downsized}}}{V_{Swept_{NA}}},$$

Eqn 1

where DF is the downsizing factor, $V_{Swept_{NA}}$ is the swept volume of a naturally-aspirated engine of a given power output and $V_{Swept_{Downsized}}$ is the swept volume of a similarly-powerful downsized alternative.

On a commercial level, it may be possible to eliminate entire engine families from an OEM's range, for example by replacing 6-cylinder engines with 4-cylinder ones; the same could be said of 8- versus 6-cylinder configurations. Obviously, in the case of downsizing from a Vee-configuration engine to an in-line one, there are potentially significant further bill of materials (BOM) and manufacturing cost savings too. These savings can help to offset the additive technologies required to recover the power output from an engine which usually has a significantly smaller swept volume, because clearly some means of increasing specific output has to be found to retain installed power in a vehicle.

Operation on the 2-stroke cycle would bring all of the above benefits and more, but is not currently favoured by the industry because of the amount of investment historically made in the production of poppet valve 4-stroke engines. This has not stopped some researchers from investigating this operating cycle again, even proposing a 2-stroke/4-stroke switching approach with a poppet valve architecture suitable for manufacture down existing transfer lines [2], but until a practical means of instantly and safely changing camshaft speed by a factor of two is created, it is difficult to see such a concept progress to production.

As a consequence, the most pragmatic approach taken to date has been to pressure charge the 4-stroke engine, normally by turbocharging it, thus recovering some of the waste thermal energy in the exhaust. A key enabler for this approach was the development and application of gasoline direct-injection (DI) systems, which allowed the use of the latent heat of vaporization of the fuel to be beneficially employed to allow an increase in compression ratio (CR). A further development was the adoption of camshaft phasing devices which, together with the DI system's ability to delay fuel introduction until after exhaust valve closing (EVC), permitted large amounts of valve overlap to be used without combusting short-circuited fuel in the exhaust path. In turn this permitted large amounts of scavenging air to be passed through the combustion chamber, simultaneously shifting the operating point of the turbocharger compressor (to the benefit of driveability) and internally cooling the chamber (to the benefit of flushing out residual gases and trapped charge temperature reduction, both extending the knock limit of the engine) [3]. Under some conditions secondary oxidation of partially-combusted exhaust products in the exhaust could also potentially safely increase the enthalpy available to the turbine and reduce chemical energy losses. These approaches potentially allow further increases in CR for the turbocharged DI spark-ignition engine (DISI) or a higher DF to be used, again to the benefit of part-load fuel economy.

Various supporting technologies have been developed and productionized such as the adoption of pulse-divided exhaust manifolds in 4- and even 8-cylinder engines [4,5], the use of integrated exhaust manifolds [6], combinations of the two [7], or adoption of 3-cylinder groups with or without some of the other features [8,9]. Furthermore, combined super- and turbocharging systems have also been in production for some time, allowing positive pressure to be applied to the intake all the time and desensitizing the engine to the pulse overlap effects (particularly an issue in four or more cylinder groups) [10]. Other interesting downsizing approaches such as Divided Exhaust Period or Valve-Event Modulated Boost hold promise in improving the efficiency of the Otto cycle through pumping work reduction, scavenging and related combustion improvement [11,12,13].

One early successful downsized engine with a DF of approximately 40% provided a reduction in fuel consumption on the New European Drive Cycle of 20% [14]¹. A single research engine has been shown with a DF of 50% [16]. The authors believe that no engines have been developed with a higher DF. Consequently the collaborative project 'Ultra Boost for Economy' (or 'Ultraboost') was formed with the major tasks of specifying, designing, building and operating an engine with a minimum of 60% downsizing factor, this being defined by the torque curve of an existing high-feature, high-performance naturally-aspirated engine. Through the results obtained it was intended to establish what the limiting factor to downsizing at this level was and whether there would be further benefits in downsizing further. The paper outlines how this was achieved.

BACKGROUND TO THE PROJECT

Project Targets

As mentioned, the project was to demonstrate the feasibility of 35% reduction in fuel consumption through extreme downsizing. Specifically, this was to be relative to a 2010 model year (MY) Range Rover fitted with the Jaguar Land Rover (JLR) AJ133 naturally-aspirated (NA) V8, which was a current production vehicle when the project was started. The developed concept was to be modelled in a 2013 MY Range Rover. Although hybridization is not a feature of the change in vehicle specification, because of the technical changes made to the vehicle line since, stop-start, vehicle mass reduction and the adoption of a transmission with a greater number of ratios were included as assumed technology upgrades. The effects of these will be discussed in further detail later in this paper.

From the outset, and in order to have a robust baseline for comparison, the general project target of achieving the same torque curve of the naturally-aspirated (NA) version of the 5.0 litre Jaguar Land Rover AJ133 V8 engine was adopted. Although transmission matching can disguise the typically reduced low-speed torque of downsized engines when they are used in passenger cars, high torque at low engine speeds is still a significant advantage for off-road driving and so it was considered desirable to retain this attribute for this research project. Furthermore, driveability comparable with the JLR V6 diesel engine was to be provided for the same reason. From previous studies [15], a DF of 60% would be expected to yield 23% of the 35% improvement required, with stop-start functionality coupled to the other vehicle changes being expected to supply the other 12%. Thus an outline specification of a 2.0 litre pressure-charged DISI engine was created as a first iteration at the outset to the project.

1. Note that this result is *not* independent of vehicle effects: a six-speed gearbox was also adopted with the 1.4 litre Twincharger engine to help provide the reduction in fuel consumption. From work previously conducted by JLR [15], a DF of 40% would in itself be expected to yield a fuel economy improvement of about 12% of the reported 20%. Note that the 12% value is considered to be only due to downsizing the engine swept volume and the friction reduction due to down-cylindering, and does not include vehicle and transmission modifications taking the total to 20%.

Figure 1 shows that adopting this swept volume would require operation at up to 32.4 bar brake mean effective pressure (BMEP). Consideration of a realistic mean peak cylinder pressure (PCP) in the region of 130-135 bar was also included in the engine sizing decision. Figure 1 also illustrates one extremely challenging target for the project in matching the torque curve of the original 5.0 litre V8 NA engine: the new 2.0 litre engine has to produce 25.1 bar BMEP at 1000 rpm, where preignition and other abnormal combustion phenomena would normally be expected to be severe limitations. Taken together, these targets were considered extremely aggressive and beyond anything achieved up to that point in time.

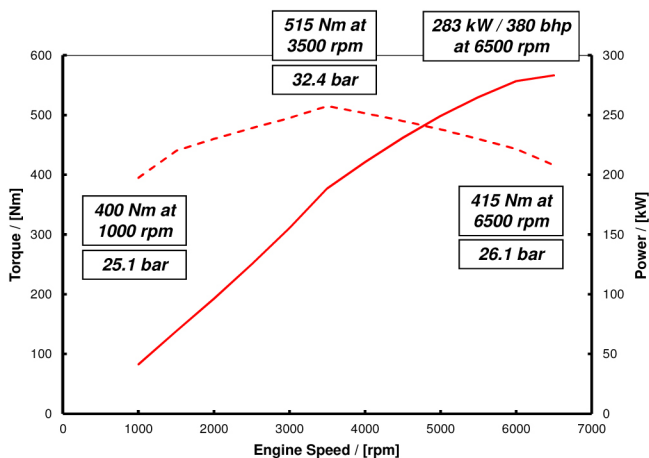


Fig. 1. Target power and torque curves and selected associated BMEPs for a 2.0 litre engine

Project Partners

Ultraboost was a collaborative project part-funded by the Technology Strategy Board, the UK's innovation agency. The project partners were Jaguar Land Rover, Lotus Engineering, GE Precision Engineering, CD-adapco, Shell Global Solutions (UK), the University of Bath, Imperial College London and the University of Leeds. It started in September 2010 and had a duration of three years.

The structure of the project has been described in detail elsewhere in an earlier publication [17]. Nevertheless, it is useful to recap the members and their responsibilities before discussing its phases.

Jaguar Land Rover (JLR) was the lead partner, with responsibility for engine build, general procurement, engine-mounted charging system integration and overall project management.

GE Precision Engineering provided engine design and machining capabilities as well as background knowledge on the design of high-specific-output racing engines.

Lotus Engineering provided dedicated engine management systems (EMS), 1-D modelling and know-how on pressure-charged engines, and support for engine testing.

CD-adapco supported the design process with steady-state and transient CFD analysis primarily in order to support intake port design.

Shell Global Solutions provided knowledge of autoignition phenomena, base fuel for engine development (including full fuel analysis), detailed lubricant analysis to aid in interpretation of engine wear, and designed and interpreted a test matrix of fuel formulations in a thorough programme to screen engine response to various fuel properties, results of which will be reported elsewhere (see later).

The University of Bath conducted all of the testing, having dedicated boosting and cooled exhaust gas recirculation (EGR) rigs which were used for the initial Phase 2 testing of the demonstrator engine (see below).

Imperial College London specified the charging system components, with support from both JLR and Lotus, and tested them in order to characterize them accurately so that the 1-D model was as robust as possible.

The University of Leeds developed their autoignition model to assist with the 1-D modelling process.

Phases of the Ultraboost Project

The project was split into several parts, which are shown schematically in Figure 2. This has been discussed in detail elsewhere but is reprised here in order to provide context when discussing the engine results later in the paper.

In Phase 1, a production AJ133 NA engine was commissioned on the test bed at the University of Bath using the Denso EMS which was then used in production. This part of the programme provided baseline data for the V8 engine in terms of performance, fuel economy and transient response, while also facilitating the shaking down of the test facility at Bath.

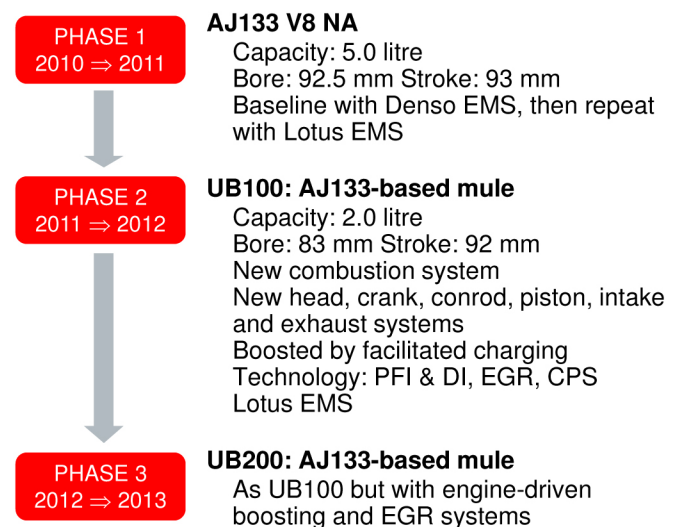


Fig. 2. The three phases of the Ultraboost project

The Denso EMS was then replaced by the Lotus system, which was shown to be capable of controlling the engine and giving exactly the same performance at full and part load, including matching the steady-state fuel consumption of the production engine and Denso EMS combination to an average of 0.6% lower (see later). This phase therefore set the fuel consumption benchmarks for the project's downsized engine design and proved the capability of the Lotus EMS when controlling a direct injection engine with twin electrohydraulic cam phasers and two high-pressure fuel pumps, while at the same time utilizing multiple-injection and multiple-ignition strategies.

In parallel with the V8 engine test work, during Phase 1 the first iteration of Ultraboost engine was specified, designed and procured. This was to be known as UB100. In order to do this, the pooled knowledge of all the parties was used, resulting in a current industry best-practice high-BMEP engine with some additional novel features. It was first tested in Phase 2, the test programme of which utilized a test bed combustion air handling unit (CAHU) and a specially-designed EGR supply rig. Phase 2 was primarily intended to prove out the efficacy (or otherwise) of the newly-developed combustion system. The testing portion of this phase would also permit fuel testing to be undertaken without the complicating effects of an engine-driven charging system, although this important subsystem would also be specified, modelled, procured and validated in a parallel work stream within Phase 2.

Because the UB100 engine would not have a self-driven charging system, in order to provide relevant results throughout testing this engine it was operated with boundary conditions determined by the detailed 1-D model (itself used to help the design process and the specification of the charging system to be used later). Thus, in areas where the supercharger would have to operate, any supercharger drive torque which the engine would ordinarily have to supply would be part of the necessary target torque for the Phase 2 engine to produce at the flywheel, i.e. the brake torque would have to be greater than the values shown in [Figure 1](#) by a previously-determined amount. Where the turbocharger would be operating by itself, just the pressure and temperature boundary conditions would be sufficient to establish whether the engine was capable of meeting the targets with the eventual engine-mounted charging system in place. As reported later in this paper, the turbocharger (but not the supercharger) would in fact be tested on the UB100 engine in the latter stages of Phase 2, to allow engine-based verification of the turbocharger run-up line as input to the choice of supercharger drive ratio for the Phase 3 engine.

Phase 3 was intended to comprise any necessary redesign of the UB100 engine coupled with mounting the engine-driven charging system. The engine was then to be known as UB200.

Ultimately, the level of achievement of the project targets was to be demonstrated by a combination of direct measurement of engine performance on the dynamometer (power, torque, fuel

consumption etc.) and modelling (by the application of gathered minimap fuel consumption data to a proprietary vehicle performance modelling tool).

ULTRABOOST ENGINE DESIGN

Major Technologies Employed on the Core UB100 Engine

In order to save time, the Phase 1 engine was designed using the cylinder block and main bearings of the donor Jaguar Land Rover V8 engine, with Bank B (the left hand bank) blanked off. This approach has been discussed in a previous publication [17], and was chosen since the general engine structure and main bearings were already known to be capable of handling the stresses imposed by the power and torque targets; essentially, these components are used in the supercharged version of the AJ133 engine which has much higher ratings than the NA variant [18].

A flat-plane crankshaft was used, which also reduced the engine stroke, while a siamesed liner pack was employed to reduce the bore diameter. The UB100 engine also used the standard water, oil and high pressure fuel pumps, main bearing shells and fuel rails of the AJ133, plus other parts as suited to the task. For general design information on the donor AJ133 engine, including its use of a sump-mounted auxiliary shaft to drive the fuel pumps (rather than using the camshafts), see [18].

In order to minimize flame travel while at the same time providing sufficiently large throat diameters for gas exchange, the 1-D model was used to determine the minimum practical bore size. This was established to be 83 mm, which, with a stroke of 92 mm, gave a swept volume of 1991 cc. The compression ratio was set at 9.0:1 with a substantially flat-topped piston. A study of the possible combustion chamber geometries led to an included valve angle of 45°, giving an open chamber for minimum quenching and heat loss (returned to later in the Discussion). Bulk air motion and turbulence was to be provided by a new design of intake port, the performance of which is detailed later.

The only compromise associated with adopting the approach of using the AJ133 cylinder block was that the cylinder head bolt spacing would have to be the same as that of the donor V8, which has a much bigger bore of 92.5 mm and an associated cylinder centre distance of 100 mm [18]. A completely new cylinder head was designed with these criteria in mind which incorporated a second-generation close-spaced DI combustion system, cam profile switching (CPS) on both the inlet and exhaust cams and the new inlet ports. CFD was used to verify the cooling capability of the existing block water jacket, liner pack and head; as a result deflector plates were added to the cylinder block jacket in order to provide robust cooling of the structure.

The incorporation of CPS tappets on the inlet side was in order to allow investigation of a degree of Miller-cycle operation at both low- and mid-load, as already used in the naturally-aspirated AJ133. Their use on the exhaust side was in order to minimize exhaust pulse interaction at low engine speeds, which is an issue for linked cylinder groups comprising more than three cylinders operating with conventional exhaust periods, as is the case here [3,19]. The 1-D model was developed in concert with this design process, and was used to help guide it since it was also being used to help specify the charging system, thus assisting in the important choice between a turbocharger or supercharger for the high-pressure stage of the compound charging system arrangement [17,20].

The cylinder head also adopted the dual continuously-variable camshaft phasing (DCVCP) devices of the AJ133; its chain drive system, albeit slightly modified, was also carried over. The DCVCP system necessitated the addition of 'thumper cams' at the rear of the head in order to provide an uneven torque input to drive the fast-acting phasers to move the camshafts. This thumper cam system comprised a follower with a spring and twin cam lobes on the shaft, minimizing parasitic losses, but nevertheless they would have slightly increased the friction of the engine, a subject which will be returned to in a later section.

A photograph of the assembled cylinder head with tri-lobe camshafts for the CPS tappets *in situ* is shown in Figure 3. The two-lobe thumper cams can be seen at the back of the head.



Fig. 3. Cylinder head assembly. Note the fully-machined inlet ports and the twin-lobe 'thumper cams' at the back of the head, providing uneven drive torque for the fast-acting camshaft phasing devices

Another feature of the combustion chamber and cylinder head configuration was the potential to fit a second spark plug under the intake ports. This was facilitated by two things: first, the fact that in order to achieve high tumble rates DISI engine ports have generally become 'raised' above their port-fuel injection (PFI) predecessors, and second, the adoption of central DI meant that the injector was not located under the inlet ports. The ability to put a second ignition point under the intake

valves, where knock can frequently occur, was considered to be advantageous but was unfortunately not tested within the project due to time constraints.

Simple log inlet and exhaust manifolds were designed for the initial testing of the UB100 engine. An exhaust back pressure (EBP) valve was used to set the requisite manifold pressure to mimic pre-turbine conditions in the engine as determined by the 1-D model.

Specific Design Details of the Core UB100 Engine

The core UB100 engine, which was also the basis of the later UB200 build, was designed using a holistic design approach. Attention to detail was of particular importance given the operating conditions and the thermal loading that such a highly-boosted engine would be subjected to. Each system, sub-system and component was critiqued and where necessary optimized with respect to its form, fit and function: no unnecessary risks were taken, safety factors were kept sensible and reliability was of key importance. Initial 1-D cylinder pressure data was deemed pessimistic as combustion system performance was anticipated to be more favourable, and hence the stress analysis boundary conditions applied were based on higher loads which, given the eventual reliability of the engine, proved prudent. Component fatigue life constraint was based on 32 million cycles using measured S-N data. Further details of some specific components are reported below.

Cranktrain

The piston was made from heat-treated Al 2618A aluminium alloy and designed to mitigate potential knock or pre-ignition while being optimized for performance: mass, component life, friction, oil carry-over, blow-by and manufacture. The ring pack was also specified to minimize oil carry-over, friction and top ring groove wear. The gudgeon pin was made from GKHW heat-treated steel, and was nitrided and DLC-coated to prevent galling.



Fig. 4. Piston and connecting rod assembly



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The connecting rod was 300M heat-treated steel with a steel bush and used MJ8 high tensile bolts. The rods turned a GKHW heat-treated steel crankshaft which was also nitrided.

A photograph of the piston and connecting rod assembly is shown in [Figure 4](#).

Crankcase and Liner Pack

The crankcase was an aluminium pressure die casting (from the AJ133 V8 donor engine). The open-deck arrangement was modified to receive a single-piece liner pack and seal arrangement. O-ring tracks were machined for fluid sealing within the block, as discernible in [Figure 5](#). The liner pack itself was a single-piece design optimized for cooling and stiffness and made from Al 2618A heat-treated billet. Nickel-ceramic bore coating was applied. Twin grooves for O-ring seals were machined into the lower portions of the protruding liners which were then spigotted into the block. [Figure 6](#) shows a photograph of the liner pack, in which the O-ring grooves at the bottom can just be discerned.

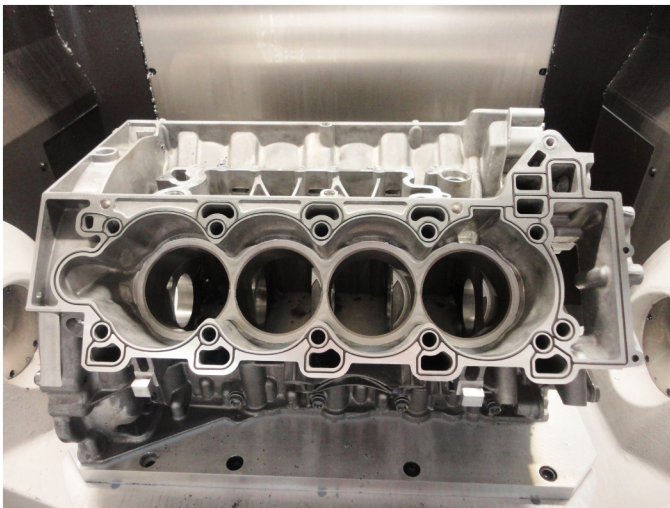


Fig. 5. Modified crankcase (A Bank only). Note O-ring grooves for fluid sealing



Fig. 6. Liner pack

Instead of a conventional cylinder head gasket, gas-energized rings (or "Wills rings") were used for combustion sealing. The groove for these can be seen on each cylinder in [Figure 6](#).

Cylinder Head Assembly

The cylinder head was made from an Al A356 heat-treated sand casting. The combustion chamber, inlet and exhaust ports were designed to promote advantageous in-cylinder charge motion and flame propagation; the intake port is discussed in detail later. The ports and combustion chamber were fully machined.

The DI injector and spark plugs were packaged in a similar 'asymmetric' arrangement to AJ133 [18], and the coolant jacket was optimized to achieve target heat transfer coefficients (HTCs) with minimal pressure drop.

Camshafts, including the thumper cam arrangement for the cam phasers, and the CPS tappet system were also packaged into the head. The inlet valves were stainless steel and the exhaust valves were sodium-filled and made from Ni80A steel. The valve guide and seat material was chosen based on wear and heat transfer properties. The cam cover was a structural design with integral front and rear cam caps.

Water-Cooled Exhaust Manifold

Initially, a tubular steel manifold was used, but it was decided to design and procure a water-cooled exhaust manifold (WCEM) for reasons which will be discussed later. This was tested during part of Phase 2 and throughout Phase 3. It was an Al A356 heat treated sand casting. The manifold was pulse-divided and designed to help achieve sufficient cooling for turbocharger protection. This component will be discussed in greater detail later in the section related to UB200 design.

Intake Port Design and Performance

In order to achieve the necessary air motion and mixture preparation in DISI engines there has been a general evolution towards high-tumble intake ports; this has only been made possible by the simultaneous adoption of pressure charging to overcome the flow loss generally associated with this move. It is worth noting that under-port placement of the injector has had a symbiotic relationship with the developmental direction of the general port configuration of DISI engines, but nevertheless the situation has arisen that reduced flow rate is seen as a worthwhile trade-off to achieve high tumble rates (and hence improved mixture preparation and charge cooling). Obviously, any loss of flow capability can be expected to manifest itself in increased charge cycle (pumping) work, and so a desire for Ultraboost was to achieve a balance of flow and tumble considered to be significantly beyond the current state of the art. This was especially important given the high BMEP rates and specific power targeted by the project. This section briefly discusses how this was achieved and compares the performance of the adopted port with a current production turbocharged DISI engine benchmark.

Initially, a target was agreed upon based on the JLR engine database and the knowledge of the other partners. Several ports were then designed which fitted the cylinder head package. With these ports designed, CD-adapco then brought

their capabilities to bear in two distinct parts of the process: a first simulation period where the steady-state flow characteristics were determined using the STAR-CCM+ software package, and a second stage where full transient in-cylinder simulations were carried out using the STAR-CD and es-ice software packages. Figure 7 shows a port in the test rig model, including the intake runner.

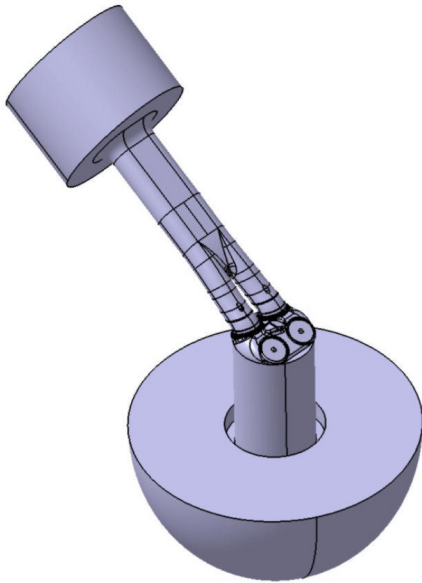


Fig. 7. Intake port and runner within the air flow CFD model

During the first part of this process 20 ports were designed and their steady-state results used to filter the designs. This enabled five port configurations to be carried forward to the second analysis stage, from which a single port was chosen and then machined into the first UB100 cylinder head; this port, known as IP18, is shown in Figure 8, together with the exhaust port. The other available heads were held back from machining should any changes found to be necessary from the first engine test results in Phase 2.

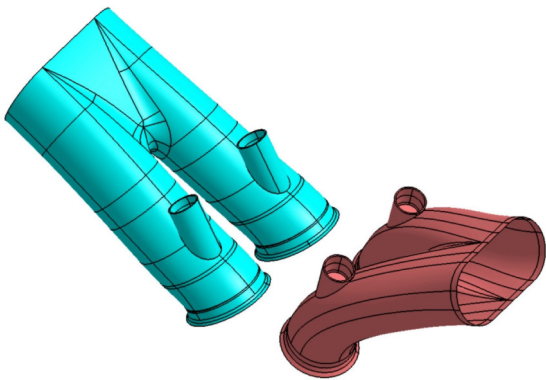


Fig. 8. Final intake ('IP18') and exhaust ports

The ports were flow tested on Lotus Engineering's cylinder head air flow bench. These results were compared to data from the BMW N20 2.0 litre I4 engine which had also been measured on the same flow rig. Although the N20 engine is rated at a BMEP level significantly below that which Ultraboost was targeting, it

was still considered to be the state-of-the-art in terms of specific power and BMEP at that time, coupled with the fact that it had a central DI combustion system employing a multi-hole solenoid injector [4]. The results of this flow bench testing are shown in Figures 9, 10, 11 and are discussed below.

Figure 9 presents the flow capability of the inlet port in comparison with the BMW engine. The Ultraboost flow at the maximum valve lift of 10.5 mm is 182 CFM, and that for the BMW at a similar lift is 139 CFM, representing an increase of 30.9%. Since the Ultraboost intake port has a throat diameter of 28.4 mm versus 27.6 mm for the N20, this is comfortably greater than the 5.9% increase in throat area, illustrating the advantage of the Ultraboost port design. (Note that in this comparison the valve stems have not been taken into account in determining the difference in throat areas.)

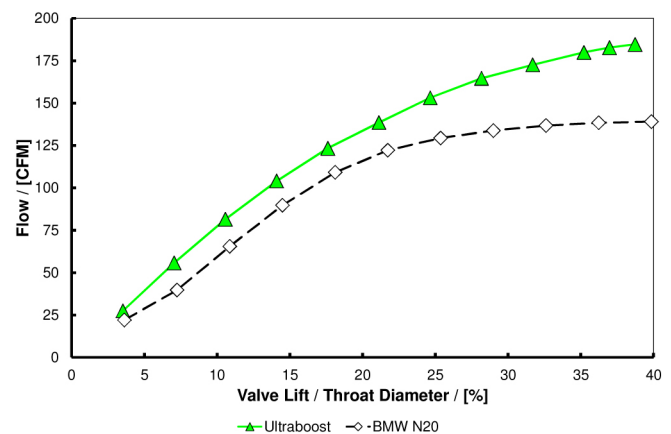


Fig. 9. Intake port flow comparison for the Ultraboost and BMW N20 engines

Figure 10 shows the flow coefficients, with Ultraboost having 0.633 and the BMW 0.520 at the same 10.5 mm valve lift condition. From this it can be seen that the port flow performance of Ultraboost in comparison to the N20 is extremely good, despite the Ultraboost engine having a smaller bore than the BMW (83 mm v 84 mm).

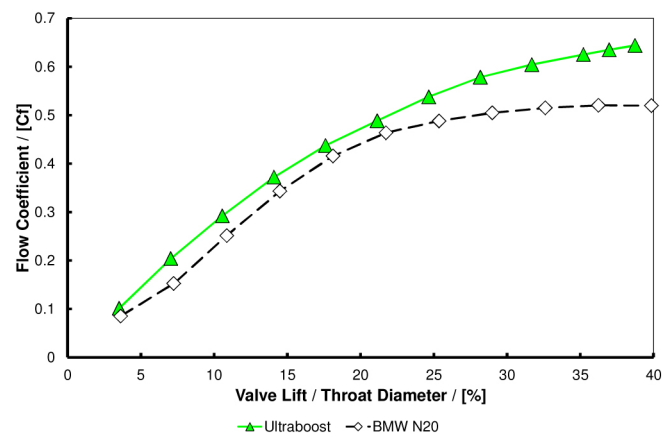


Fig. 10. Intake port flow coefficient comparison for the Ultraboost and BMW N20 engines

A comparison of non-dimensional tumble number is made in Figure 11. The N20 offers significantly higher tumble at low lift, but it must be borne in mind that it employs valve shrouding in order to increase tumble in this area of the curve, a specific requirement because of its use of Valvetronic mechanically-variable valve train [4]. The adoption of this form of valve train makes it especially important to generate sufficient tumble at low valve lifts, since valve lift and duration are the primary means of controlling load while minimizing throttling loss. As a consequence Valvetronic only utilizes the high lift region during high load operation, and so a compromise here is presumably considered acceptable for the N20 engine.

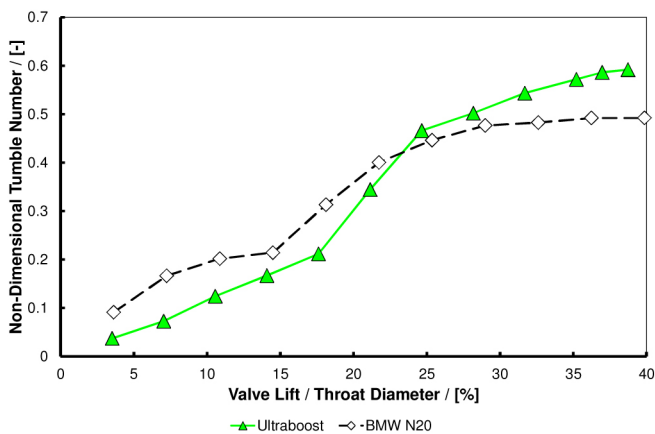


Fig. 11. Intake port non-dimensional tumble number comparison for the Ultraboost and BMW N20 engines

In comparison, Ultraboost was only ever to be fitted with two-step CPS tappets and so high outright tumble rates were considered to be more important, even for part-load operation; greater in-cylinder air motion would be the result albeit at the expense of increased throttling loss. The use of shrouding in the BMW is reflected in the values for the tumble ratio for the two ports, with Ultraboost having 1.626 and the BMW 1.868.

The fact that the adopted port shows high tumble performance throughout the majority of the effective high-lift cam profile - from 7 mm to 10.5 mm - was considered a success, especially when paired with the high flow coefficient. As a consequence of the air flow rig testing and the engine test results gathered during Phase 2, the IP18 intake port was not changed at all throughout the project.

UB100 Engine Specification

The general configuration of the UB100 engine, as initially built for testing, is given in Table 1. The undersquare nature of the engine is readily apparent. CAD images of the engine in UB100 specification are shown in Figures 12 and 13.

Table 1. Ultraboost UB100 engine specification

General architecture	4-cylinder in-line with 4 valves per cylinder and double overhead camshafts
Construction	All-aluminium AJ133 cylinder block converted to single-bank operation on the A Bank (right-hand side) Siamesed liner pack to facilitate reduced bore diameter Dedicated cylinder head
Firing order	1-3-4-2
Bore	83 mm
Stroke	92 mm
Swept volume	1991 cc
Combustion system	Pent-roof combustion chamber with asymmetric central direct injection and spark plug High-tumble intake ports Auxiliary port-fuel injection Possible second spark plug position in an under-intake-port location
Compression ratio	9.0:1
Valve gear	Chain-driven double overhead camshafts with fast-acting dual continuously-variable camshaft phasers (DCVCP) Cam profile switching (CPS) tappets on inlet and exhaust
Engine management system	Lotus 'T6' ECU

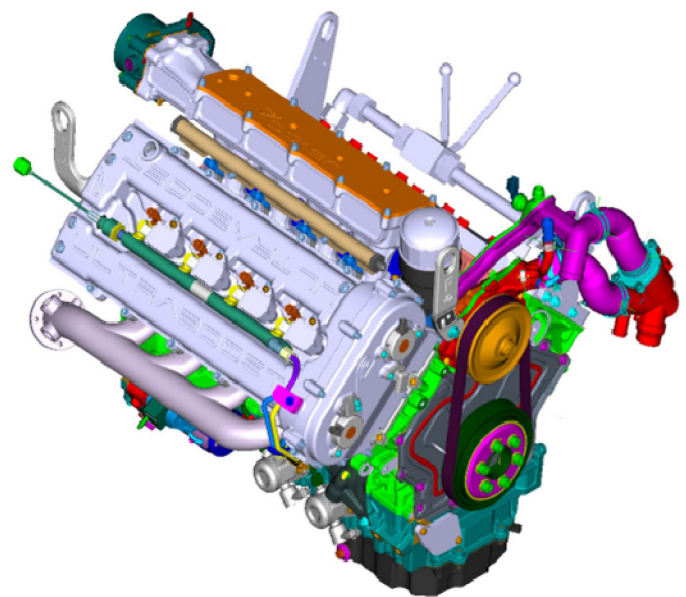


Fig. 12. CAD image of front three-quarter of assembled UB100 engine, as originally tested; note coolant bypass pipe for the absent B Bank cylinder head

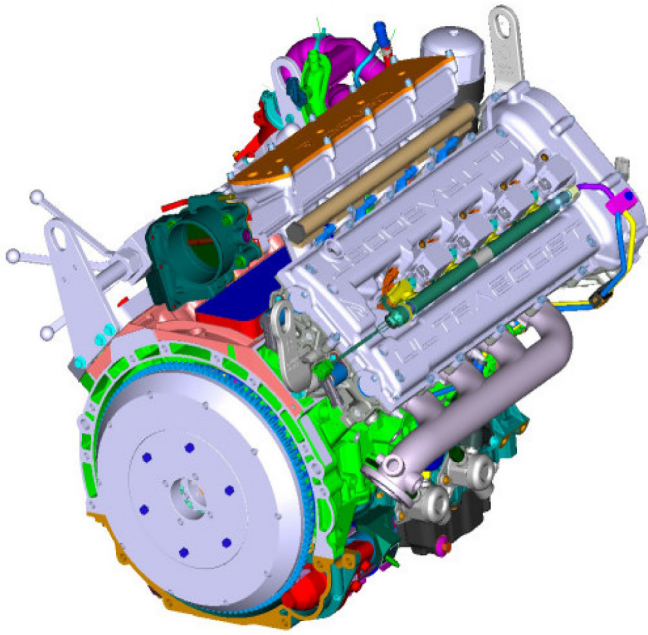


Fig. 13. CAD image of rear three-quarter of assembled UB100 engine, as originally tested with log-type exhaust manifold

Major Additional Technologies Employed on the UB200 Engine

The chief differentiator between the UB100 and UB200 hardware was the fitment of the engine-driven charging and associated low-pressure cooled EGR systems to the core engine. During Phase 2 the 1-D model was extensively used to help specify this, with several different technologies and technology combinations being investigated. This work was described in detail in a previous paper [20] but, in summary, a series system was adopted using a Garrett GT30 turbocharger as a low-pressure stage and an Eaton R410 supercharger as a high-pressure stage, with chargecooling after each stage.

The supercharger was clutched so that it could safely be driven at a high ratio to the crankshaft at low engine speeds, in order to compound the boost pressure available from the turbocharger, in turn enabling a match for that device more favourable for maximum power. The supercharger also provided improved driveability from low engine speeds and mass air flow rates. Figures 14 and 15 show CAD images of the UB200-level build; in these the UB100 core engine can be discerned by comparison to Figures 12 and 13.

This system has certain similarities to that employed on the Volkswagen Group's 'Twincharger' engines [10,14], except that for the Ultraboot engine the configuration was fundamentally different insofar as the turbocharger was the first stage in the charge air path. This configuration was chosen for several reasons, including the ability to interpose an intercooler between the charging stages in addition to another within the plenum.

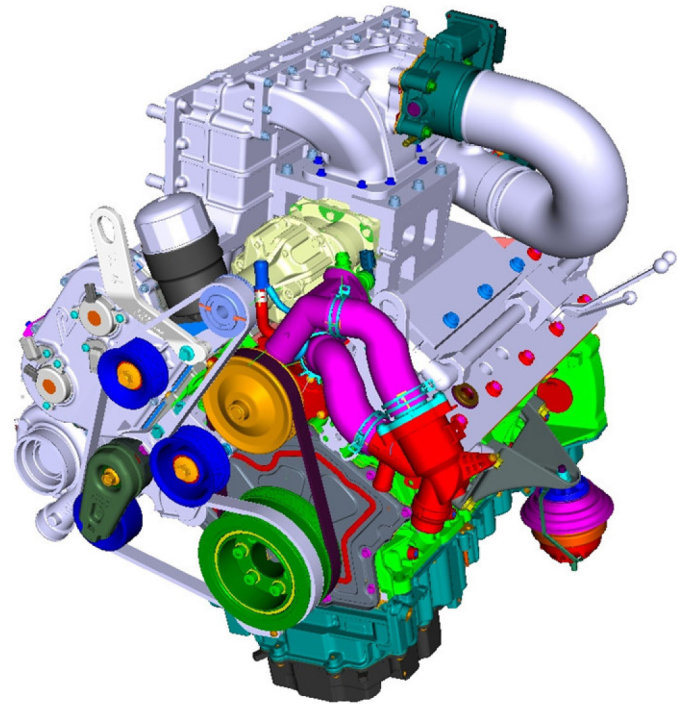


Fig. 14. CAD image of front three-quarter of assembled UB200 engine, showing supercharger and drive belt

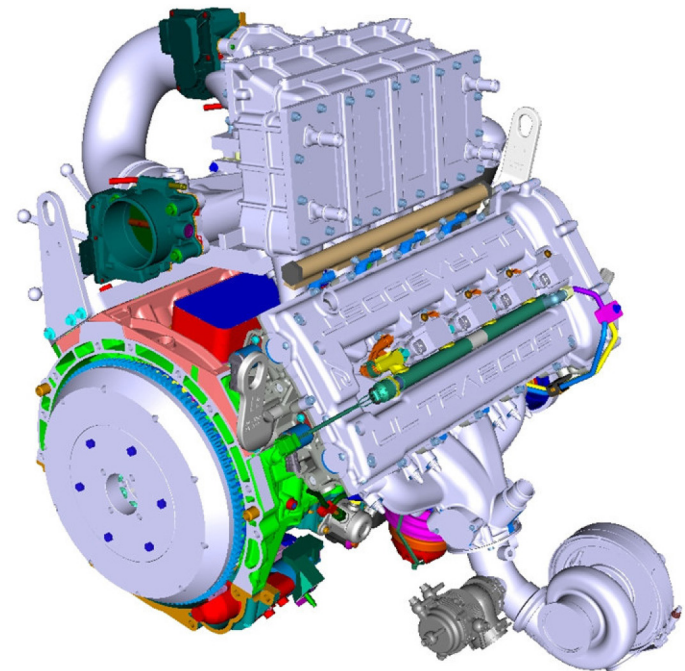


Fig. 15. CAD image of rear three-quarter of assembled UB200 engine, showing water-cooled exhaust manifold and plenum chamber with integrated chargecooler

When the supercharger was declutched the turbocharger effectively blew through two chargecoolers in series, ensuring high chargecooler system effectiveness overall. Conceptually, the Ultraboot system was very similar to the charging system employed by the Lancia Delta S4 rally car [21], except that in that application the supercharger was not declutched.