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Ultra Boost for Economy: realizing a 60% downsized engine concept

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ABSTRACT

The paper discusses Ultra Boost for Economy, a collaborative project part-funded by the Technology Strategy Board, the UK's innovation agency. 'Ultraboost' combines 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 large 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 5.0 litre V8 engine, the main project target was to show that such a downsized engine could in itself provide a viable route to a 35% reduction in vehicle tailpipe CO_2, with the target drive cycle being the New European Drive Cycle. 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 these points. The engine was sized by using the fuel consumption improvement targets while a series of specification steps, designed to ensure that the required full-load performance and driveability could be achieved, was followed. The intake port in particular was the subject of much effort, and data is presented showing its performance versus a current state-of-the-art production design.

The use of a test-cell-based charging system, while the engine-mounted charging system was being developed and characterized in parallel, is discussed. This approach allowed development of the base engine and combustion system without the complicating effects of the charging system performance coming into play. Finally, data is presented comparing the performance of the engine in this guise with that when the engine-driven turbocharger was used, showing that the peak torque and power targets have already been met.

1 INTRODUCTION

1.1 Spark-ignition engine downsizing

Spark-ignition (SI) engine downsizing is now established as a 'megatrend' in the automotive industry, providing as it does an affordable solution to the twin issues of reducing tailpipe CO_2 emissions and improving fuel economy while providing improved driveability from gasoline engines. The 'downsizing factor' is here defined to be

$$DF = \frac{V_{Swept_{NA}} - V_{Swept_{Downsized}}}{V_{Swept_{NA}}}, \qquad \text{Eqn 1}$$

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.

To the OEM the attractions of a downsizing strategy include that gasoline engine technology is very cost-effective to produce versus diesel engines (especially when the costs of the exhaust after treatment (EAT) system are included), that there are still significant efficiency gains to be made due to the losses associated with the 4-stroke Otto cycle, and that pursuing the technology does not entail investing in completely new production facilities (as would be required by a quantum shift to electric or fuel-cell vehicles, for example).

The advantages of downsizing a 4-stroke spark-ignition (SI) engine stem chiefly from shifting the operating points used in the engine map for any given flywheel torque, so that the throttle is wider-open to the benefit of reduced pumping losses. At the same time, the mechanical efficiency increases, this being defined as

$$\eta_{Mech.} = \frac{BMEP}{IMEP}, \qquad \text{Eqn 2}$$

where $\eta_{Mech.}$ is the mechanical efficiency, BMEP is the brake mean effective pressure and IMEP is the indicated mean effective pressure [1]. Thermal losses also improve and, in the case of downsizing and 'decylindering' from a Vee-configuration engine to an in-line one, crevice volume losses can be markedly reduced and there are potentially significant bill of materials (BOM) and manufacturing cost savings, too.

These savings can help to offset the additive technologies required to recover the power output, because some means of increasing specific output has to be provided to retain installed power in a vehicle. This is normally done by pressure charging the engine, with turbocharging generally being favoured because it allows some exhaust gas energy recovery. There are significant synergies with other commonplace technologies such as direct injection (DI) and camshaft phasing devices, too [2].

To date production downsized engines have generally been configured with a *DF* in the region of approximately 40%, with one research engine shown with this value at 50% [3]. Consequently the Ultraboost project was formed with the major tasks of specifying, designing, building and operating an engine with a minimum of 60% downsizing factor. Through the results obtained it was intended to establish whether 60% is a practical limit for the approach or whether there would be benefit in further downsizing, and that such a downsized engine could in itself provide a route to a 35% reduction in vehicle tailpipe CO_2 (importantly, without the use of hybridization other than a Stop/Start system).

Consequently a primary aim of the project was to achieve the power and torque curves of the Jaguar Land Rover 5.0 litre AJ133 naturally-aspirated V8 engine with a pressure-charged engine of approximately 2.0 litre capacity. These curves are reproduced in Figure 1, together with the associated BMEP values required from the downsized engine at peak torque, peak power and 1000 rpm. The CO_2 emissions and fuel consumption improvement was to be demonstrated by using dynamometer measurements and vehicle modelling, with the target drive cycle being the New European Drive Cycle (NEDC).

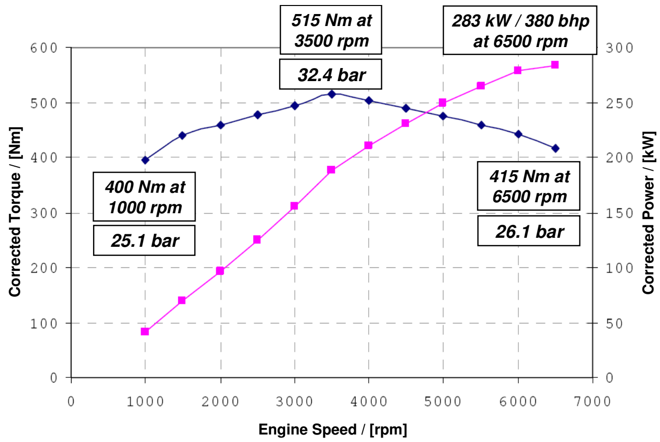

Figure 1: Target power and torque curves and selected associated BMEPs for a 2.0 litre engine

1.2 Ultraboost project partners

The Ultraboost project comprised eight partners, Jaguar Land Rover (JLR), GE Precision Engineering, Lotus Engineering, CD-adapco, Shell, the University of Bath, Imperial College London and the University of Leeds. It started in September 2010 with a duration of three years.

JLR is the lead partner, with responsibility for engine build, general procurement, engine-mounted charging system integration and project management. GE Precision provided engine design and machining capabilities as well as background knowledge on the design of high-specific-output racing engines. Lotus Engineering provided a dedicated engine management system (EMS), 1-D modelling and know-how on pressure-charged engines, and support for engine testing. All engine testing was to be conducted at the University of Bath, where dedicated boosting and cooled exhaust gas recirculation (EGR) rigs were used for initial testing of the demonstrator engine. CD-adapco supported the design process with steady-state and transient CFD analysis primarily in order to support intake port design, which is discussed in detail below. Shell provided test fuels and autoignition know-how. Imperial College specified the charging system components, with support from both JLR and Lotus, and tested them in order accurately to characterize them so that the 1-D model was as robust as possible. Finally, the University of Leeds developed their autoignition model to assist with the 1-D modelling process. This project structure was reviewed in an earlier publication [4], where some of the background detail to the establishment of the projects targets was also discussed.

1.2 Phases of the Ultraboost project

The project was split into several parts. In Phase 1, a production JLR 5.0 litre AJ133 V8 engine was commissioned on the test bed at the University of Bath using the Denso engine management system (EMS) then used for production. This was then replaced by the Lotus EMS, which was demonstrated 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 ≤ 0.5%. 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 many high-technology features, including multiple-injection strategies.

In parallel with the Phase 1 engine test work, Phase 2 specified, designed and procured the core Ultraboost engine (known as UB100). 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. The Phase 2 test programme utilized a test bed combustion air handling unit (CAHU) and a specially-designed EGR pump rig. It was primarily intended to prove out the efficacy of the newly-developed combustion system. The testing portion of this phase also permitted fuel testing to be undertaken without the complicating effects of an engine-driven charging system, although this important subsystem would also be specified, modeled, procured and validated in a parallel work stream within this phase.

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.

The present paper discusses some of the engine-specific technologies configured and tested in Phase 2; the results of the fuels testing and of the Phase 3 engine will be reported separately in later publications.

Ultimately, the level of achievement of the project targets will be demonstrated by a combination of direct measurement (power, torque, driveability etc.) and modelling (by the application of gathered minimap fuel consumption data to a vehicle performance model, this being necessary since the baseline AJ133 engine is no longer fitted to the target vehicle).

2 ENGINE DESIGN

2.1 Derivation of engine swept volume

At the start of the project the actual swept volume was unconstrained. In order to establish this parameter, vehicle modelling was employed to set part load operating points representative of the target vehicle and to provide weighting factors for these points. The engine swept volume was then determined 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; these were informed by previous work undertaken by JLR [5].

The engine was then designed in conjunction with 1-D modelling which helped to combine the various technology packages of the project. These included an advanced charging system (discussed in a previous paper [6]) and a valvetrain system with the necessary variability to deliver target performance. The modelling also helped to determine the flow characteristics required of the intake port. Ultimately this had stretch targets set for it to ensure the necessary charge motion for fuel mixing and to help suppress knock, and was subjected to a full transient CFD analysis. This is discussed later.

In Phases 2 and 3 of the project the 1-D model was also used to guide testing, primarily to set intake and exhaust system boundary conditions to make them representative of what could be expected of the real charging system. It was also used to calculate the extra torque that the core engine would have to produce for the results to be representative of the combined engine and charging system. It was also used to help to explain trends in the results.

2.2 General engine specification

From this preliminary work the engine was specified as shown in Table 1. The undersquare nature of the engine is readily apparent; this helps to shorten the flame travel to the benefit of knock and to reduce thermal losses. It also possibly benefits preignition, the causes of which are believed to include oil being ejected from the piston top land, and reducing the bore diameter directly reduces the top land area [7,8]. Effectively, the engine is one bank[1] of a heavily-modified AJ133 V8, with a new bore and stroke, a flat-plane crankshaft and attendant firing order. This approach was taken because the bearings and scantlings of the AJ133 engine would easily be capable of handling the performance. A CAD image of the UB100 engine, fitted with the original log-type exhaust manifold, is shown in Figure 2.

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
Bore	83 mm
Stroke	92 mm
Swept volume	1991 cc
Firing order	1-3-4-2
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

The engine management system was configured to be capable of controlling the many functions on the engine as detailed in Table 1 and ultimately also the selected charging system components, including the supercharger clutch and bypass system [6].

The engine has been designed to withstand peak cylinder pressures of 130 bar, with known further countermeasures should it be considered advantageous to increase this to a greater level (for instance, when investigating high-octane fuels). The aluminium alloy piston itself is safe to 145 bar for the sort of duty cycle a research engine is used for.

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[1] *The active bank is the A Bank (on the right-hand side of the engine).*

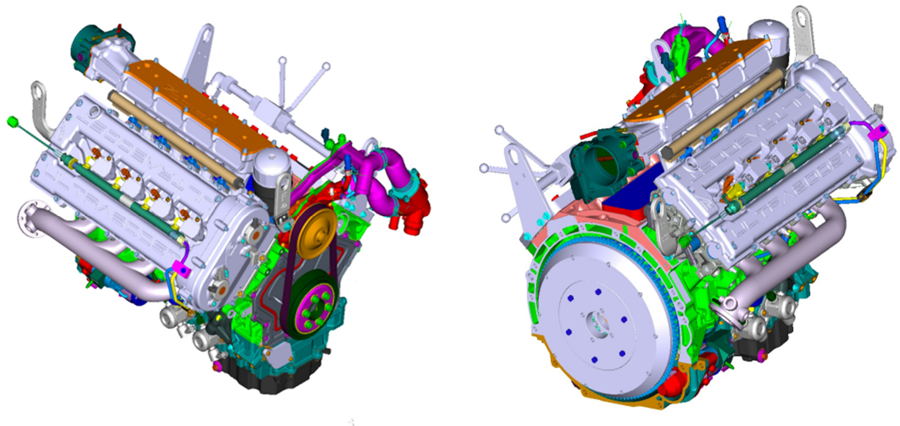

Fig. 2: CAD images of assembled UB100 engine, as originally tested with a log-type exhaust manifold; note coolant bypass pipe for the absent B Bank cylinder head

2.3 Intake port design and flow-rig performance

In order to achieve the necessary air motion and mixture preparation in DISI engines there has been a general evolution of 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 had a symbiotic relationship with this evolution of the general port configuration of DISI engines, but nevertheless the situation has arisen that flow rate is seen as a worthwhile trade for tumble (and hence improved mixture preparation and charge cooling). Obviously, any loss in flow capability can be expected to manifest itself in increased charge cycle (pumping) work, and so a prime 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.

While it is accepted that it is of primary importance to have high charge motion near to top dead centre (TDC) when the spark is initiated, high tumble has another function earlier in the cycle as a means to homogenize the air, fuel, residuals, oil droplets and temperature as fully as possible. Near to TDC piston geometry has an important effect with regard to the bulk flow breakdown and the generation of microturbulence, but during the intake stroke the importance of its geometry gradually lessens towards bottom dead centre (BDC). Thus intra-cycle CFD should be employed to determine the best overall engine geometry but the air flow rig can be used as a good differentiator early in the port development process. This section briefly discusses how this process was followed within the project and compares the performance of the adopted port with a current production turbocharged DISI engine benchmark.

Initially, a target was agreed upon based upon 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 stages of the process: a first calculation stage where the steady-state flow characteristics were determined, and a second one where full transient calculations were carried out.

During the first part of this process many ports were schemed. From these, 20 were designed and analyzed under steady-state conditions. Filtering led to five being chosen and carried forward to the second transient analysis stage. Finally one port design was selected and machined into the first UB100 cylinder head, with the other available heads being held back from machining should it be found necessary to implement any changes as a result of engine performance testing.

After the design had been created and the first head machined, the ports were flow tested on Lotus Engineering's cylinder head air flow rig. These results were compared to data from the BMW N20 2.0 litre I4 engine which had also been measured on the same 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 current state of the art in terms of specific power, BMEP and the fact that it had a central DI combustion system employing a multi-hole solenoid injector [9]. The results of this flow rig testing are shown in Figures 3 to 5 and are discussed below.

Figure 3 presents the outright 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. Figure 4 shows the related 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 1 mm smaller bore diameter. Part of this increased flow will be due to the 5.9% larger throat area of Ultraboost, but this does not in itself account for the fact that the Ultraboost port flows nearly 30% more air than that of the N20 at 10.5 mm valve lift.

A comparison of non-dimensional tumble number is made in Figure 5. The N20 offers significantly higher tumble at low lift; however it employs valve shrouding in order to increase tumble in that area of the curve, a specific requirement because of its use of Valvetronic mechanically-variable valve train [9]. The adoption of this form of valve train makes it especially important to generate high tumble at low valve lifts, where 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 at low load is presumably considered acceptable for the N20 engine.

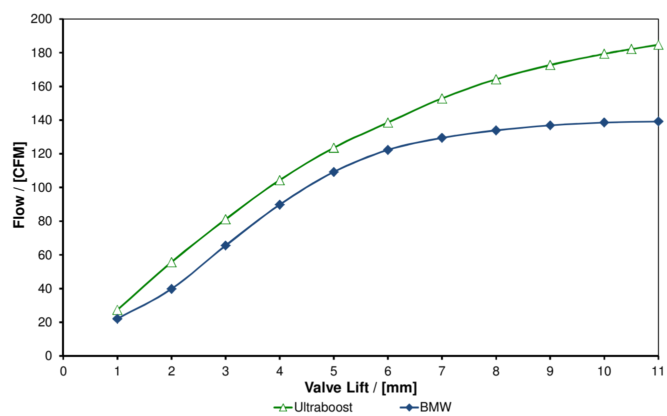

Fig. 3: Inlet port flow comparison for the Ultraboost Phase 2 and the BMW N20 cylinder heads

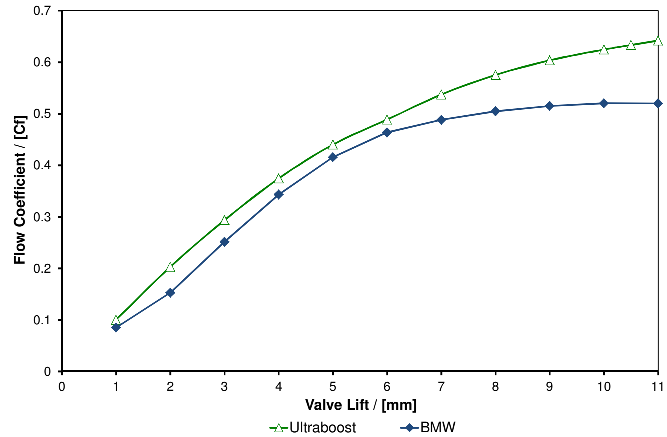

Fig. 4: Inlet port flow coefficient comparison for the Ultraboost Phase 2 and the BMW N20 cylinder heads

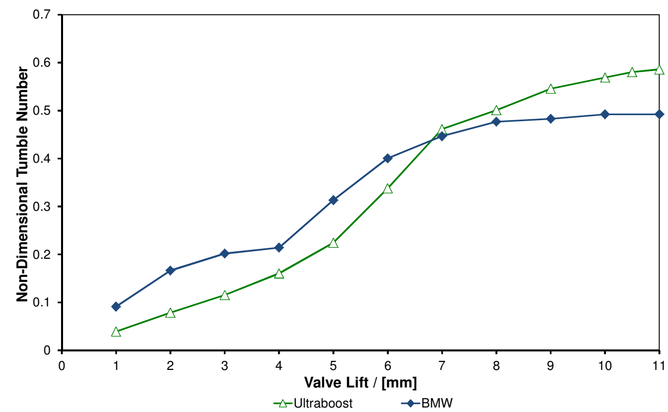

Fig. 5: Inlet port non-dimensional tumble number comparison for the Ultraboost Phase 2 cylinder head and the BMW N20

Conversely, Ultraboost was only ever to be fitted with two-step CPS tappets and so the achievement of high outright tumble rates was considered to be paramount, even for part-load operation (where greater in-cylinder air motion would still result albeit at the expense of relatively higher throttling loss). The use of valve shrouding in the N20 is reflected in the values for the tumble ratio for the two ports, Ultraboost giving 1.626 and the BMW 1.868.

The fact that the Ultraboost port gives high tumble 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. This was also borne out by the fact that the port has not had to be changed since it was finalized; the engine is extremely knock tolerant and does not suffer from preignition, both of which would be expected to benefit from extremely good homogenization of the charge at the point of ignition, as discussed earlier.

2.4 Water-cooled exhaust manifold

The integrated exhaust manifold (IEM) is becoming a common technology for production engines [10,11], and is particularly advantageous for turbocharged units since it allows the removal of a large degree of component protection over-fuelling at high load [2,12]. Unfortunately, because of the bore pitch and cylinder head bolt spacing necessarily inherited from the AJ133 engine, it was not feasible to design an IEM into the Ultraboost cylinder head (shown in the left hand side of Figure 6). There was, however, an interest in investigating a water-cooled exhaust manifold (WCEM) from the point of view of assessing the full-load heat rejection. At the same time, the new WCEM permitted a more advantageous geometry than the original log manifold, and mitigated the fact that the original's outlet geometry was restrictive. It also permitted the provision of a flow splitter which could separate all the cylinders completely, pulse divide numbers 1 and 4 from 2 and 3, or permit full mixing (all at the entry to the turbine). This is shown *in situ* in the right hand side of Figure 6.

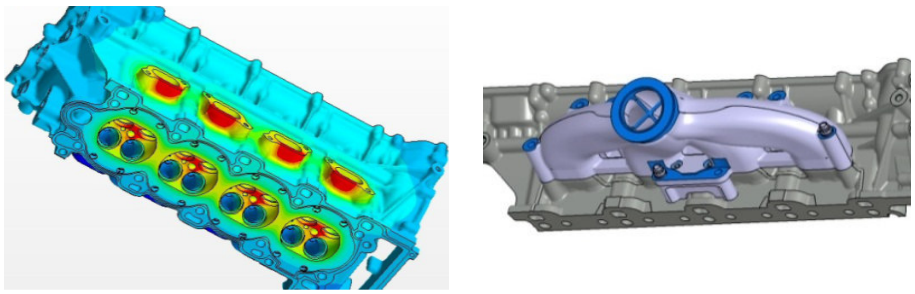

Fig. 6: Ultraboost cylinder head showing large bore pitch and cylinder head bolt spacing inherited from the AJ133 engine (left) and the water-cooled exhaust manifold with enlarged outlet area and flow splitter *in situ* (right)

3 ENGINE TESTING AND RESULTS

In the present work, the results quoted have been gathered mostly with the CAHU system, the GT-Power model being used to apply boundary conditions so that the brake results are representative of those to be expected when the engine and charging system are combined during Phase 3 of the project. Thus, in the area where the supercharger would operate, its drive torque as determined using the 1-D model was added to the values shown in Figure 1 to give the target brake torque. Where the turbocharger would operate by itself, just the pressure and temperature boundary conditions were sufficient to establish whether the engine was capable of meeting the torque targets with the eventual engine-mounted charging system in place.

All results reported here were gathered using commercially-available 95 RON fuel supplied by Shell; it complied with EN228 and had 5% ethanol content by volume. Other fuels will be tested as part of the project and reported in later publications.

Testing to date has shown that the engine can generate the performance required to achieve the target torque curve. Furthermore, among other investigations, specific tests have been carried out in the areas of intake temperature (to show the combustion system's sensitivity to this parameter) and PFI/DI split ratio. The engine showed no particular sensitivity to air intake temperature, being capable of delivering target performance at up to 80°C (the design target is 35°C), demonstrating a very robust combustion system and justifying the effort expended on the intake port design. This is further supported by the engine's response to PFI/DI split ratio, shown in Figure 7, where 100% DI fuelling gave the most performance; in fact the performance of the engine is broadly constant down to and including 70% of the total fuel load being supplied by the DI system. This result is attributed to optimum air-fuel mixing and the maximum use of the latent heat of the fuel being ensured by the very high tumble flow, while PFI operation not only removes most (but not all) of this effect but also displaces more oxygen (13).

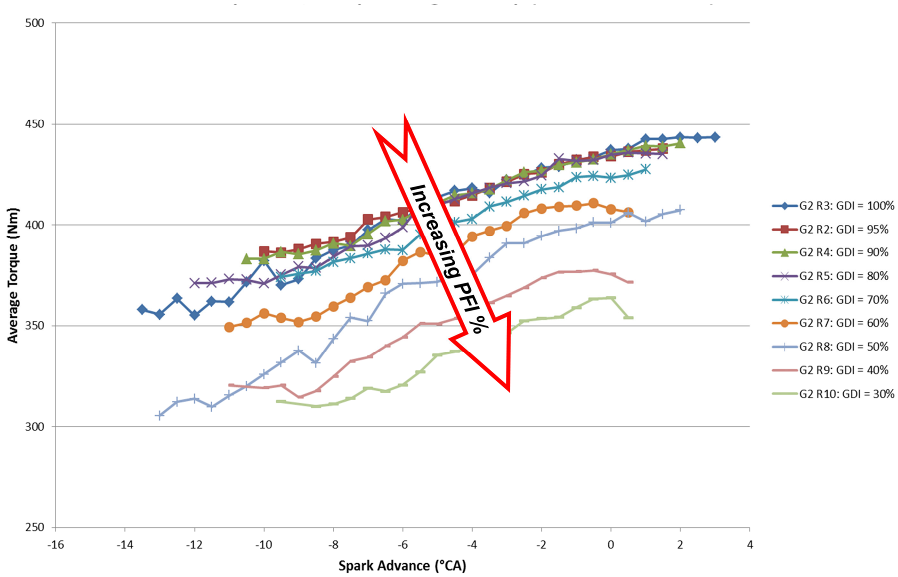

Fig. 7: Results of DI/PFI split loop at 2000 rpm and constant intake pressure of 2.2 bar Abs. Percentage of total fuelling supplied by direct injection shown in key. Conducted at constant intake/exhaust valve maximum opening points (MOPs) of 88° ATDC / 96° BTDC

Results using the CAHU and with the engine in the configuration shown in Figure 2, i.e. with the log-type exhaust manifold, are shown in Figure 8. Here it can be seen that up to 4000 rpm the UB100 engine exceeded the target torque by the equivalent of the predicted supercharger drive torque of approximately 48 Nm, but that its performance started to dip thereafter. This was despite the intake manifold conditions supplied by the CAHU being exactly as called for by the 1-D model. Investigation revealed that this was due to the exit area of the log manifold being too small for the exhaust gas mass flow, causing it to choke. This situation was an artifact of not considering the waste gate flow in the original specification of the log manifold, and so for that reason the design and procurement of the WCEM described above was accelerated, since it had the correct sizing.

In order to alleviate the problem of manifold restriction, the WCEM was fitted and the engine tested again with the intake pressure and temperature boundary conditions supplied by the CAHU as determined by the 1-D model. However, performance was again limited, this time by peak cylinder pressure (PCP) in Cylinder 2. Examination of the individual cylinder pressure traces and those for the exhaust and intake manifolds showed that a wave dynamic effect was causing Cylinder 2 to generate more BMEP than the others, eventually reaching the PCP limit prematurely in that cylinder. To circumvent this issue, it was decided to conduct an early test with the selected Honeywell GT30 turbocharger [6] instead of using the CAHU. This test also allowed engine-based verification of the turbocharger run-up line as an input to the choice of supercharger pulley ratio for the next-phase UB200 engine. The result of this test is shown in Figure 9, where it can be seen that the engine achieves the full-load torque curve from 3000 rpm onwards and has thus has technically delivered both the maximum torque and power targets.

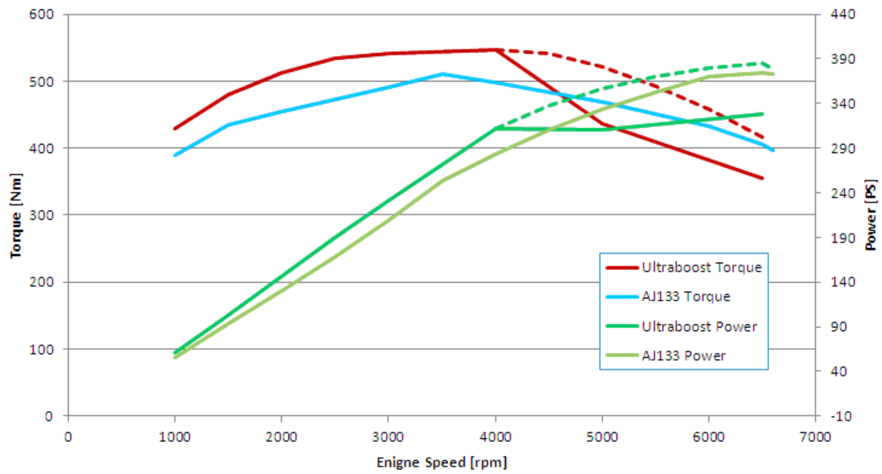

Fig. 8: UB100 performance versus AJ133 target torque curve. Intake and exhaust conditions were derived from the 1-D model. Required supercharger drive torque is approximately 48 Nm up to 3500 rpm. Original log-type exhaust manifold fitted; the dip in performance from 4000 rpm onwards is discussed in the text

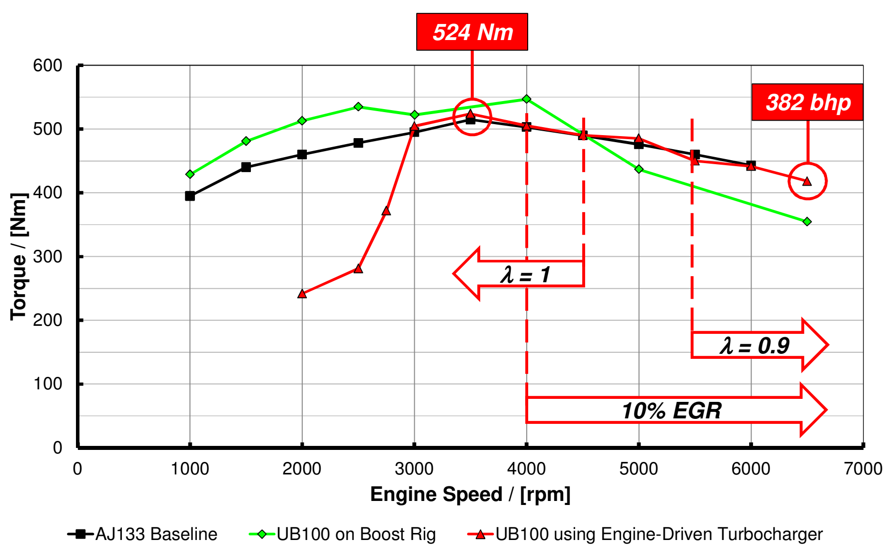

Fig. 9: Engine performance with selected GT30 turbocharger (supercharger not fitted). Relative air-fuel ratio (λ) and EGR rate for the curve with the engine-driven turbocharger also shown

The results shown in Figure 9 were obtained with 10% EGR from 4000 rpm onwards, and slight enrichment to λ=0.9 above 5500 rpm. This was very much an exploratory test using the UB100 build specification and future work with the final UB200 configuration will concentrate on removing and hopefully eliminating enrichment throughout more of the speed-load range.

Even when operated at a gross BMEP of 28 bar at 1000 rpm the engine does not suffer from preignition. Even after subtracting supercharger drive torque, this is in excess of the project target output at that speed. The reasons for this absence of preignition are the subject of further investigation, since to the degree found it appears to be outside of the experience of the engine research and development community within the context of engine downsizing.

A single surface preignition failure has been experienced to date, when the WCEM was first fitted and the engine operated using the CAHU and without the turbocharger; however, this situation was complicated by the fact that the second side-mounted spark plug was fitted to the cylinder head in use (but inoperable at the time). In this case it is surmised that the removal of enrichment fuelling and EGR enabled by the high heat removal of the WCEM drove the exhaust port temperature to a point where the electrodes of the second spark plug overheated, followed by the exhaust valves. As such, this is not considered to have been a low-speed preignition (LSPI)-type event leading to superknock.

More extensive cooled EGR testing has been also carried out and this is the subject of ongoing research. As part of this, pre- and post-catalyst EGR take-offs have been investigated, because there are conflicting views in the literature as to which is the more beneficial [14,15]. From the work to date, it is thought that such conflicting evidence may be confused by the use of high- or low-pressure EGR loops, which are in themselves known to affect turbocharged engine performance in different ways, regardless of whether the EGR is catalyzed or not [2,16]. Results of this investigation will be published at a later date.

A full optimization of part-load fuel economy, based on a 15-point speed-load minimap, will be performed using UB200, but tests to date (again using the 1-D model to set boundary conditions and to guide control parameters such as camshaft timing) show every indication that the 35% fuel economy target can be met in-vehicle.

4 CONCLUSIONS

The Ultraboost project is a major collaborative engine research project part-funded by the Technology Strategy Board, the UK's innovation agency. It is led by Jaguar Land Rover with industry- and academia-wide support. It seeks to realize a concept engine downsized by 60% (from a naturally-aspirated 5.0 litre V8 engine baseline) utilizing production technologies and with attributes suitable for deployment in premium saloons and SUVs. To that end it employs direct injection, independent cam profile switching and camshaft phasing for the intake and exhaust sides of the engine and an advanced charging system including two stages of charge air cooling. It also utilizes low-pressure cooled EGR and is fitted with a water-cooled exhaust manifold. Overall, it is designed to withstand peak cylinder pressures of 130 bar.

Thus far it has been demonstrated that the power and torque targets can be met with the selected turbocharger and, when operated using a facilitated charging system, that it can deliver the gross BMEP necessary to meet the low-end torque target.

With this level of downsizing preliminary fuel economy tests show that 35% fuel economy improvement in the target vehicle should be achievable.

5 ACKNOWLEDGEMENTS

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ABBREVIATIONS

ATDC	After top dead centre
BDC	Bottom dead centre
BMEP	Brake mean effective pressure
BTDC	Before top dead centre
CAHU	Combustion air handling unit
CPS	Cam profile switching
DCVCP	Dual continuously-variable camshaft phasing
DF	Downsizing factor
DI	Direct injection
EAT	Exhaust after treatment
EGR	Exhaust gas recirculation
IEM	Integrated exhaust manifold
IMEP	Indicated mean effective pressure
JLR	Jaguar Land Rover
MOP	Maximum opening point
NA	Naturally-aspirated
NEDC	New European Drive Cycle
PCP	Peak cylinder pressure
SI	Spark-ignition
TDC	Top dead centre
UB	Ultraboost (Ultra Boost for Economy)
V_{Swept}	Swept volume
WCEM	Water-cooled exhaust manifold
$\eta_{Mech.}$	Mechanical efficiency