

The Lotus Range Extender Engine

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ABSTRACT

The paper discusses the concept, specification and performance of a new, dedicated range extender engine for plug-in series hybrid vehicles conceived and designed by Lotus Engineering. This has been undertaken as part of a consortium project called Limo Green, part-funded by the UK government.

The Lotus Range Extender engine has been conceived from the outset specifically as an engine for a plug-in series hybrid vehicle, therefore being free of some of the constraints placed on engines which have to mate to conventional, stepped mechanical transmissions. The paper starts by defining the philosophical difference between an engine for range extension and an engine for a full series hybrid vehicle, a distinction which is important with regard to how much power each type must produce. As part of this, the advantages of the sparkignition engine over the diesel are outlined. The rationale leading to the adoption of an in-line 3-cylinder configuration and specification is then discussed, followed by a description of the principal design characteristics of the engine; it also points out any significant differences from what might be expected were a 'conventional' engine to be designed.

The use of 'monoblock' construction which integrates the cylinder block with the cylinder head and also with a water-cooled exhaust manifold is described. An indication of the likely parts reduction possible with this approach is made, together with a brief discussion of the pros and cons of the approach and which other engine types this architecture might suit.

Engine performance on 95 RON unleaded gasoline is detailed, together with observations on why the engine has been additionally protected to operate on ethanol and methanol.

The paper also includes a brief discussion of other engine concepts which might suit the purpose of range extension, but which (compared to the more conventional reciprocating spark-ignition engine embodied by this approach) will necessarily require greater development even if they may offer the potential for further improved efficiency, package volume and/or NVH characteristics.

INTRODUCTION

The desire to reduce anthropogenic CO₂ emissions to combat global warming and also to improve the energy security of individual nations is driving a move towards increasing electrification of the vehicle [1,2]. Since most forms of renewable energy are gathered via electrical means (wind, solar, wave etc.) and nuclear power provides electricity which has low fossil CO₂ impact, the desire to increase the use of such electricity in general and in transportation in particular is understandable¹. However, current battery technology is both heavy and expensive, the reason for which is alluded to in [Figure 1](#), showing the gravimetric and volumetric energy densities of various fuels in their stored state on a vehicle, i.e. including the fuel storage system.

While hydrogen is considered by some to be a potential energy vector for renewable energy, its well-to-tank (WTT) energetic efficiency is very poor in comparison to electricity due to the requirement to change the form of the energy and

¹In this paper the term 'renewable energy' will henceforth refer to the true renewables - wind, solar, wave etc. - as well as nuclear energy which, in its conventional form, is still technically a depletable.

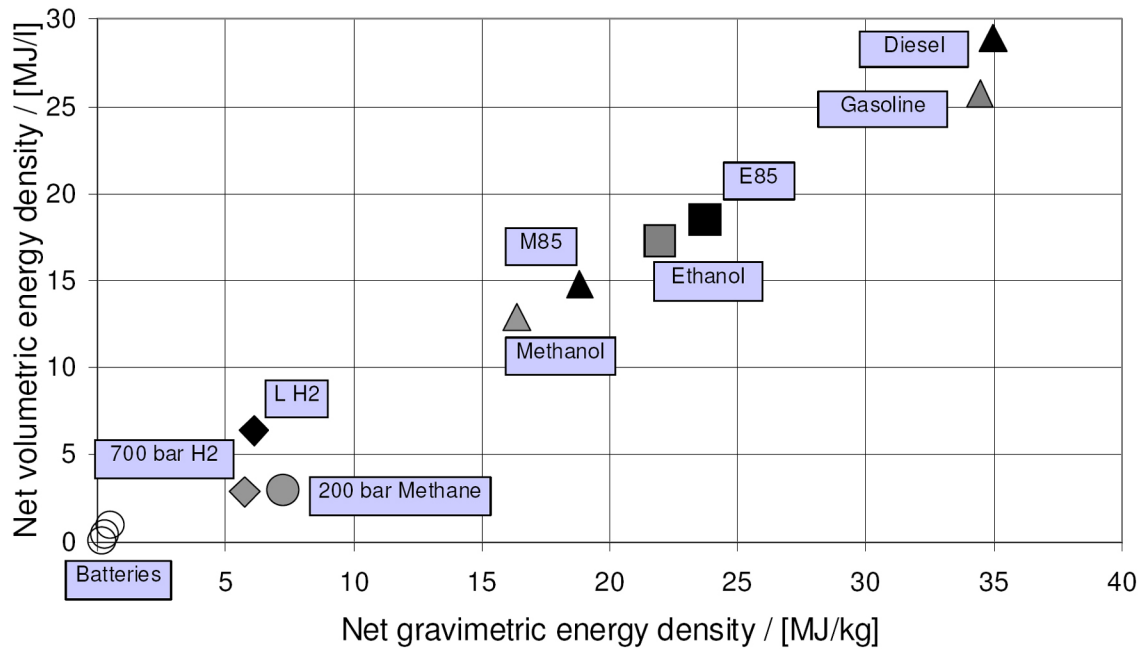


Fig. 1. Gravimetric and volumetric energy density of various fuels and their storage systems, calculated using information on storage systems in actual production and research vehicles. In this figure M85 and E85 represent 85% by volume methanol and ethanol in admixture with gasoline, respectively, LH2 is liquid hydrogen, and the pressures associated with hydrogen and methane are the system storage pressures

the inherent losses that this brings. In terms of WTT efficiency the electrical route far exceeds the efficiency of hydrogen [3,4,5], and also permits the ready introduction of biomass into the transport energy sector through co-firing in fossil-fuel based power stations (note that, if carbon sequestration is also used in such power stations, the use of biomass also leads to a net reduction in CO₂ in the atmosphere). The tank-to-wheel (TTW) efficiency of electricity far exceeds that of any combustion-engine-based powertrain, due partly to the constraint of the Carnot cycle efficiency on all heat engines. As a consequence the overall well-to-wheel (WTW) efficiency of electrical vehicles is very high, where:

$$\text{WTW} = \text{WTT} \times \text{TTW} \quad (\text{Eqn 1})$$

Thus in broad terms reducing greenhouse gas (GHG) emissions or improving energy security would both best be served by a switch to electricity as the energy carrier for all forms of power, since the laws of thermodynamics favour this approach. However, the realities of economics provide a serious challenge for electrical vehicles.

This is because whereas stationary energy consumers do not require a significant energy storage capability, autonomous

vehicles do, due to the requirement that they carry their own energy supply².

The range of current light-duty internal combustion engine-powered vehicles is generally between 300 and 500 miles (480 and 800 km), depending on tank size, fuel, duty cycle etc. Such fuel systems are very cheap and also light (see Figure 1), and form a very important part of why these vehicles are affordable and therefore successful. Even with the increased TTW efficiency of electric vehicles, 1 kWh of electrical energy will only drive a light duty vehicle about 4 miles (6.4 km) on a legislated drive cycle such as the New European Drive Cycle (NEDC); in 'real-world' conditions, the situation is somewhat worse - 3 miles per kWh (4.8 km per kWh) is a more realistic figure. Hence a large battery is required. Unfortunately electrical power storage is significantly more expensive than liquid energy storage. Leaving aside the potentially serious materials security issue of lithium availability, this expense is because current lithium-ion batteries often use extremely rare elements in their chemistries, such as cobalt. While the automotive industry has since its beginning had great success in reducing costs through economies of scale, the reality of this historical state of affairs is that it could only be achieved using cheap, widely-available materials such as iron and aluminium. Even platinum usage for catalytic converters has been reduced

²The efficiency of electric rail traction is often used as an argument for electrifying the automobile as soon as possible. However, there is a crucial difference between trains and road-going vehicles: the electric train does not have to carry its energy with it. Surprisingly, this fact is sometimes forgotten when the argument is made.

through engineering application and process improvement. When a truly rare element is required, and the chemistry precludes its removal below a certain minimum, the economics of mass production (which many politicians and the public appear to believe will bring the cost down) will ensure that instead of getting cheaper these batteries will become more expensive.

While some process improvements will offset the increased cost of the batteries, it is difficult to foresee an assembled battery cost of much less than \$300/kWh. Thus, the sobering reality is that every mile of vehicle range will cost in the region of \$100, and hence a battery giving 100 miles of range will cost ~\$10,000. Note that for all the high cost of the battery, this range is still significantly less than the customer is presently used to with liquid fuels and this is all before the issue of recharging times is addressed.

Consequently the customer will have to either pay a great deal more than he has been used to for his personal mobility or he will have to get used to significantly reduced vehicle range and increased stress due to 'range anxiety'.

In order to address these shortcomings while still reaping most of the benefits of full electrification, the concept of a range-extended electric vehicle has been suggested. Here, what is in reality a series hybrid is envisaged but which has an electrical storage capability sufficient for most daily drives, and which also carries another powerplant which starts when the battery state-of-charge (SOC) falls below a certain level and which thus acts as a 'range extender'. In theory this should entirely alleviate the issue of range anxiety for the customer while still reducing his day-to-day running costs dramatically (through the ability to drive most journeys using stored electric energy only). Simultaneously the battery storage capability can be significantly reduced and part of the concomitant saving in mass and cost used to support the fitment of the range extender to the vehicle. The concept thus has a major attractiveness for all the stakeholders in the future mobility scenario: governments (since it encourages electricity use to the benefit of GHG reduction and energy security) customers (since it will offer reduced running costs, at least until governments revise their transportation energy taxation strategies, which is only to be expected should EV take-up become significant) and OEMs (since the battery costs reduce and because, with the increasing emphasis on tailpipe CO₂ as the basis for taxation of their products, increasing responsibility for the TTW CO₂ impact of the fuel is passed to those who actually benefit from its supply, i.e. the fuel and energy suppliers).

The remainder of this paper will discuss a dedicated range extender engine that has been designed by Lotus as its

contribution to a Jaguar Cars-led consortium, 'Limo Green', which is seeking to realize a series hybrid vehicle capable of 120 g/km of CO₂ on the European drive cycle in charge-sustaining mode, and which comprises Jaguar, Lotus, MIRA and Caparo Vehicle Technologies. This consortium is part-funded by the UK government through its Technology Strategy Board.

The paper starts with a broad discussion of the definition of series hybrids and range extenders, then defining what a range extender engine is from Lotus's perspective. It then discusses some of the various types and technologies available for range extenders and the reasons for the choice of a dedicated reciprocating sparkignition (SI) engine for the Lotus Range Extender (LRE). There is then the determination of the required power output range for such a unit. The process leading to the 3-cylinder configuration chosen is then discussed, followed by description of its detail design (and how this differs from a conventional engine), before its performance on gasoline is presented. The paper concludes with an outline of how the 3-cylinder engine described in the main body can itself be grown into a family of dedicated units with different power outputs³.

DEFINITION OF SERIES HYBRID TRANSMISSIONS, RANGE EXTENDERS AND ENGINES FOR FULL SERIES HYBRID VEHICLES

The concept of the series hybrid vehicle is not new: Ferdinand Porsche famously created what many people think of as the first series hybrid vehicle, 'Semper Vivus', in 1900, which was produced as the 'Mixte Hybrid' by Lohner from 1901 to 1905 [7]. In an echo of modern vehicle engineering challenges, in order to overcome mass, cost and energy storage limitations Porsche effectively added an internal combustion engine and generator to a battery electric vehicle (BEV), naming it 'Semper Vivus' (or 'always alive') because the use of a combustion engine meant that the vehicle would not be rendered inoperable by a discharged battery. Interestingly, the Mixte Hybrid also employed a 'frictionless' drive train with electric motors in the wheel hubs, and while it had no significant plug-in capability, its general series hybrid transmission technology has been widely used away from the automotive world in diesel-electric trains and some military vehicles.

The two defining characteristics of any series hybrid vehicle are that there is no mechanical link between the power unit and the wheels and that there is an on-board energy store to act as a buffer and smooth out the load demand on the engine. While the distinction of no mechanical connection is

³ A broad description of the engine described in detail in this paper was presented at the SAE 2010 World Congress as an oral-only presentation [6].

necessary for a combustion engine, since a fuel cell only produces electrical power and heat it has to employ a series transmission by default, which can be hybridized to a greater or lesser degree by the provision of electrical storage capability⁴.

Having defined what constitutes a series hybrid configuration, there then needs to be a distinction made between a range-extended plug-in series hybrid vehicle (PSHV) and a full series (FSHV) hybrid vehicle. While the latter may have a degree of plug-in capability perhaps as great as what a PSHV may have, the distinction made by the authors in this paper is that a FSHV is capable of achieving the maximum vehicle speed (V_{max}) on the engine alone, whereas the PSHV cannot: its maximum vehicle speed is achieved only by using a degree of energy storage system power from the hybrid system, and it has a second and necessarily lower maximum speed, the 'charge-sustaining V_{max} ' which it can achieve on whatever power unit it is employing for range extension. Consequently, between the two V_{max} values, the battery is always depleting and only below the 'charge-sustaining V_{max} ' is there the potential to generate some excess power to charge the battery.

An engine for a FSHV would thus be termed a prime mover, while that for a PSHV might be termed an auxiliary power unit, despite the fact that it is capable of functioning as the prime mover at or below the charge-sustaining V_{max} .

Once operation on the range extender is necessary (following depletion of the battery to a point where it cannot supplement engine power), there are also implications for vehicle acceleration performance, dependent on the battery SOC. This and the desired charge-sustaining V_{max} of the vehicle are thus of importance when considering the power output requirement for the range extender in a PSHV versus the minimum performance requirements for the vehicle. The magnitude of this power requirement will be discussed in a later section.

ENGINE CONCEPTS FOR RANGE EXTENDERS

The removal of the mechanical link between the engine and the wheels permits some reassessment of the ideal type of engine for the vehicle. Many different types of powertrain are currently being considered for range extension, and, due to the markedly different requirements provided by the removal of the fixed link between engine and wheels, some concepts which have hitherto struggled in the marketplace due to the necessity of using a stepped mechanical transmission are again being considered. Some of these concepts will be reviewed here as background to the choice eventually made

for the LRE. Many of them are also eminently suitable to the role of prime movers for FSHVs.

THE 2-STROKE CYCLE VERSUS THE 4-STROKE CYCLE

In the automotive world the 4-stroke, or Otto, cycle engine has achieved near 100% penetration for spark-ignition (SI) engines, despite the disadvantage of throttling loss that it suffers at part load (through the need to regulate the amount of air entering a cylinder which has its expansion ratio fixed by the desire to achieve as high an efficiency as possible). This throttling loss has in turn driven the current downsizing trend in 4-stroke engines, where a small, pressure-charged engine replaces a larger naturally-aspirated one. Reduction of throttling loss is also the prime motivation behind stratified operation, helps explain some of the benefits of homogeneous charge compression ignition (HCCI) in 4-stroke engines, and indeed provides some of the benefits of hybridization in general. The 2-stroke cycle, however, does not suffer from throttling loss in the same manner, but unfortunately loses some of its charge air to the exhaust under some situations, leading to direct fuel carryover in the case of external mixture preparation.

The problems of direct fuel carryover can be eliminated by direct injection, but this leads to excess air operation. Although the remaining hydrocarbon (HC) and carbon monoxide (CO) emissions can be oxidized in a simple oxidation catalyst, unfortunately this means that oxides of nitrogen (NO_x) cannot easily be controlled. Consequently, unless it can be made to operate in near-full-time HCCI, a 2-stroke engine would probably have to employ a NO_x trap. These are expensive and require regeneration strategies where the engine operates stoichiometrically, in turn reducing fuel economy. While significant breakthroughs are being made to realize a 2-stroke engine with all of the theoretical advantages over its 4-stroke equivalents [8,9], in order to provide a cost-effective package to the series hybrid market the stoichiometric 4-stroke engine has some significant advantages, chief amongst which are ease of manufacture, simplicity of after-treatment and the minimization of throttling loss if the engine is made to operate at full load. It is also self-scavenging, which is a theoretical advantage over a 2-stroke externally-scavenged engine, and indeed its pumping losses are similar to a 2-stroke engine at full load, because it is not functioning as an excess-air engine. There are therefore strong arguments to retain the 4-stroke cycle in a range extender engine.

⁴Note that a series hybrid transmission utilizing hydraulics can also easily be provided with any prime mover providing a mechanical power output such as a typical engine, but obviously not so readily by a fuel cell.

THE DIESEL ENGINE

A diesel engine may be extremely efficient when mated to a stepped transmission but it cannot automatically be assumed it will also be a good choice for a series hybrid vehicle.

The emissions characteristics of the diesel engine are such that in the areas of the operating map where it shows maximum thermal efficiency the well-known NO_x versus particulate matter (PM) trade-off is not available at a level which will permit emissions compliance: consequently the engine will likely be making large amounts of both pollutants. Therefore meeting emissions limits necessitates two things: a very expensive after-treatment system for both NO_x and PM (which would probably need to be considerably larger and consequently more expensive than is presently the case for light-duty diesels with stepped transmissions) together with more-frequent periods of regeneration, which by necessity will be in significantly less efficient areas of operation. As a result a diesel engine in a series hybrid application is thought likely to be a more expensive, bulky and heavy solution than a stoichiometric SI solution, and which may not actually realize as high an efficiency. Therefore, while the diesel may suit a parallel hybrid architecture, it was discounted for the LRE concept.

THE WANKEL ENGINE

The Wankel engine, long the underdog to the conventional reciprocating engine, has some significant advantages as a constrained-operating-locus device in series hybrids. Chief amongst these are that it is extremely compact and light, has excellent NVH (due to its inherent perfect balance, if not its noise signature) and is potentially cheap to manufacture. To this must be added that (in light of previous observations) in its classic 2:3 lobe form, it operates on the 4-stroke cycle, and so its exhaust emissions can be controlled by a simple three-way catalyst (TWC). One of the minor disadvantages of the Wankel in stepped transmission engines is that the height of the eccentric shaft is greater than that of a typical reciprocating engine, but when it is to be connected to a generator this then becomes an advantage. In fact, there is no reason to assume that a Wankel range extender could not have a smaller envelope than the generator to which it is attached.

As a consequence of these advantages, the Wankel is an extremely attractive concept as a range extender, and is once again being investigated by the industry [10,11]; the engine reported in [11] is actually descended from air-cooled rotor technology used in motorcycles and karts [12]. However, the

fact that the Wankel has been neglected for so long by the majority of the industry necessarily means that it has had neither the development time nor the manufacturing investment that the reciprocating engine has enjoyed. Also, the combustion chamber shape is long and thin in the Wankel engine and, while this can be improved by adopting a low 'K factor'^{5,6}, for geometric reasons relating to the value of K this approach means that the engine cannot easily support a compression ratio (CR) higher than 10:1 without the combustion space simultaneously becoming a very poor shape for SI flame travel [13]. A low value of K also makes oil cooling more difficult with an eccentric shaft sufficiently strong in bending, meaning that air-cooling of the rotor becomes more of a necessity - this is the case in the Wankel engines reported in [11] and [12], and this in turn brings lubrication and bearing issues best solved with roller-element bearings. From the literature, the rotor of the engine in [11] appears to be oil-cooled and this would suggest a conservative K factor of approximately 7, limiting the achievable CR. Since it was intended to protect whatever engine concept was chosen for the LRE for more-optimized operation on alcohol fuel such a limitation was considered a disadvantage. Compounding or turbocharging are options to mitigate this and compounded Wankel engines have previously been developed [14], but this approach significantly increases complexity (although one could conceive of compounding arrangements which could significantly improve efficiency).

In concluding this section, it is very possible that the Wankel will emerge as an engine of choice for series hybrid vehicles⁷ but in the short term, given current OEM scepticism about the Wankel engine, the lack of a broad Tier-1 base for its component supply and the desire to bring a product to market quickly and cost-effectively, it has been decided that a reciprocating 4-stroke SI engine currently makes more sense as a range extender.

THE GAS TURBINE

The gas turbine could be developed as a generator set for a series hybrid, and may yet prove successful in this role in the longer term. However, if the mass and complexity of a heat exchanger is to be avoided, a high-pressure cycle with concomitantly high turbine inlet temperature would be required - say, 1300-1400°C. While the maximum permissible turbine inlet temperature of turbochargers is increasing all the time, it is currently at a level of approximately 1050°C and consequently somewhat below what is required to achieve a reasonable specific fuel

⁵In the geometry of the Wankel engine, K is defined as the generating radius (R) divided by eccentricity (e).

⁶The approximate swept volume of each chamber of a Wankel engine is approximately equal to $3\sqrt{3} \times R \times e \times B$, where B is the width of the trochoid housing.

⁷If the Wankel did emerge as the long-term choice for range extender engines, the irony which started with the 4-rotor Mazda R26B engine winning the Le Mans 24 Hour Race during the last year of the World Sportscar Championship 'fuel consumption formula' would be complete: in that application it demonstrated better fuel consumption and reliability than the reciprocating engines it was competing against, achieved through its operation in a constrained area of its map [15].

consumption (SFC) at maximum power in a gas turbine. To increase this value while maintaining acceptable emissions (principally those of NO_x) will require significant effort in the fields of metallurgy and combustion control in order to avoid the formation of locally-rich areas in the combustion chamber. While the authors believe it to be possible to achieve this, the current situation means that the gas turbine was discounted for the LRE concept as too experimental. The authors do acknowledge that the gas turbine has some very attractive attributes (NVH and mass, for instance) but cost-effectiveness linked to efficiency is unlikely to be one of them in the short term.

FUEL CELL ENGINE

The fuel cell is seen by some as the long-term powertrain for light-duty applications. Whether this happens or not, and which of the possible configurations emerges as the dominant one, is linked to the choice of fuel.

While a reformer can be used to provide a hydrogen stream from a practical liquid hydrocarbon fuel for use in a low-temperature fuel cell, this significantly increases cost and most fuel cell advocates now favour the use of molecular hydrogen directly as a fuel. However, molecular hydrogen is unlikely to become the main energy vector for transportation in the near-to-medium term for many reasons, if ever [3,4,5]. It is possible to synthesize practical liquid fuels which can be used in low-temperature fuel cells such as the direct methanol fuel cell (DMFC), but for direct use of a pure hydrocarbon fuel a high temperature device such as a solid oxide fuel cell (SOFC) is required. SOFCs can theoretically be significantly cheaper than hydrogen-consuming proton exchange membrane (PEM) fuel cells because they contain significantly less precious metal, but due to their higher operating temperatures they do require an extended switching-on time and are still sensitive to fuel quality relative to the internal combustion engine. Like all fuel cells for automotive use, they are unlikely to be cost-effective in the near- or mid-term, although in combination with a gas turbine cycle they may approach some kind of ideal. However, as a consequence of near-term realities, the fuel cell in general was discounted at this time.

LRE: CONCEPT CHOICE

The conclusion drawn from the above reasoning was that while it is possible for either a 2-stroke or Wankel engine to be developed as a range extender, either would take time and effort relative to a simple reciprocating engine operating on the 4-stroke cycle, which can therefore be introduced far more rapidly. Like the Wankel, the gas turbine is very attractive from an NVH viewpoint, but will similarly require significant effort to realize. In its PEM form the fuel cell is considered limited by both its requirement to operate on a fuel which is unlikely to be widely available in the mid-term and also by its very high material cost.

As a consequence, it was decided that the initial incarnation of the LRE was to be a spark-ignition reciprocating engine operating on the 4-stroke Otto cycle, but with a heavily constrained area of operation. The magnitude of this is discussed in the next section. However, in this first incarnation it could adopt various architectural technologies themselves made possible as a result of its focused performance specification. These technologies could in turn reduce its complexity, mass and cost relative to that of a conventional engine (which itself must operate over a full engine map due to the demands of driving through a stepped transmission).

POWER OUTPUT CONSIDERATIONS

The amount of power a vehicle requires to complete a drive cycle is a function of several factors, including vehicle mass, drag and friction. The series hybrid can theoretically provide an ideal solution since it can use its energy storage system to smooth out the vehicle loads and therefore only require the engine to operate at a single condition which can be made very efficient (and which needs to be so in order to overcome the losses in the transmission). In this respect, like downsizing, stratified direct injection or Miller cycle operation, hybridization can be considered a means to mitigate the throttling losses inherent in the Otto cycle.

In order to establish some limits of power output, first consider a vehicle operating on 95 RON gasoline and which emits 120 gCO₂/km on the new European drive cycle (NEDC). This drive cycle is 11.028 km long and takes 1180 seconds. Assuming that a litre of 95 RON gasoline has a lower heating value of 42.7 MJ/kg and emits 3.1724 kg of CO₂ per kg of fuel [16], this implies that the vehicle consumes 1.615 MJ/km with an average rate of fuel energy consumption is 15.10 kW. Further assuming a thermal efficiency on the NEDC drive cycle of approximately 20%, this implies a useful continuous power of just 3 kW from the engine averaged, when over the complete drive cycle.

However, the requirements of the real world are different to those of any given drive cycle (where neither V_{max} nor gradeability of the vehicle are considered). In real-world driving conditions some researchers have shown that the average power output for vehicles with stepped transmissions are still often extremely low - even when driven in a sporting manner the average power output is often in the region of only 15 kW, even in hilly conditions and at high momentary loads [17]. Recently Bassett et al. [18] have shown this to be the case in a study of fleet vehicle usage in the UK and have also shown that the Artemis cycle has the best match to real-world usage patterns.

Analysis was therefore undertaken to establish the necessary power output of a practical series hybrid vehicle, dependent on V_{max} and gradeability for conditions of extended

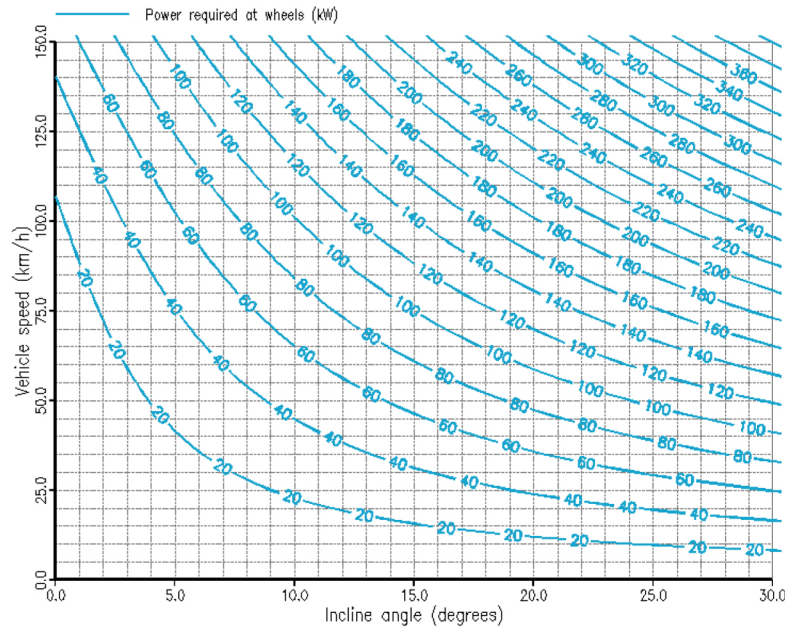


Fig. 2. Lines of constant wheel power plotted on speed versus gradeability map for a vehicle mass of 1650 kg

climbing with a depleted battery. This was assessed for a vehicle mass of 1650 kg and a typical drag coefficient and rolling friction for a practical vehicle. While heavier than a typical C- or D-segment car, this mass was utilized because of the unknown mass of the battery pack. Towing capability was not considered at this point. The results of this assessment are shown graphically in [Figure 2](#).

A reasonable target V_{max} in a charge-sustaining mode (i.e. analogous to cruise performance with fully-depleted batteries) was assumed to be 70 mph (120 km/h). This speed would represent the minimum change to a customer's driving habits on a long journey when the battery is fully depleted. Coupled with realistic electrical system efficiencies of 90% for the generator and motors, the results of this analysis suggested that a peak engine power of 35 kW would be needed to achieve the target 70 mph (120 km/h) vehicle cruise speed. This is in line with the results of Bassett et al., who found that charge-sustaining V_{max} is the biggest driver on range extender engine power requirement [18]. Additionally, for the real world, electrical 'hotel loads' (typically comprising heating or cooling loads and those associated with infotainment systems) need to be taken into account and these were anticipated to be up to 3 kW for a luxury vehicle. Anything that could be done to the vehicle to reduce friction and drag losses (together with minimizing hotel loads) could thus be expected to improve V_{max} , while mass reduction could be expected to improve acceleration

performance. It should also be remembered that the vehicle may well be capable of higher than the charge-sustaining V_{max} , but that in that case the batteries would be continually depleting and eventually the vehicle speed would reduce to that which the engine and electric transmission could support.

At the same time, in order not to cycle the battery continuously, it was considered advantageous to have a reduced power condition for the engine, which would preferably also have a high efficiency. Operation at this condition would have the further benefit of improving NVH in the vehicle under prolonged low-speed operation, but its efficiency would still have to be high enough in order not to compromise the primary reason for employing a series hybrid transmission in the real world⁸. From the drive cycle considerations outlined above, it was decided that 15 kW would be the target 'lower charging point' with good efficiency. While this is considerably greater than the continuous 3 kW that a drive cycle would require (yielding 120 gCO₂/km on gasoline at 20% thermal efficiency - see above), the hotel loads in a typical vehicle would tend to increase the power draw and 15 kW could therefore be expected to cover most driving conditions up to an average speed of 30 mph (48 km/h), i.e. typical urban and extra urban conditions. In the real world or on different drive cycles the difference between the average power and this lower charging point would similarly be expected to reduce.

⁸It is important to make this distinction. The primary attraction of the PHEV for the manufacturer is to get credits for that portion of the anticipated life the vehicle operates with zero tailpipe CO₂. While the authors of [18] shows that on the NEDC the reported tailpipe CO₂ is relatively insensitive to range extender efficiency when a large battery capacity is used, in the real world the customer will be dissatisfied if the 'engine-on' efficiency is poor, since this condition is likely to arise frequently with a small battery and the long recharge times forced by the limitations of typical domestic electricity supply. This challenge is at the root of picking the right combination of battery capacity, charge rate and engine size/cost/efficiency trade-off.

In order not to suffer from pumping losses to the same degree as a typical SI engine operating on the Otto cycle, this lower charging point was determined to be obtained primarily through reducing the speed of the engine, while keeping it at near-to-full load (see below). Necessarily this then implies a range of power that the engine must be capable of providing. From this it is interesting to note that, in common with the area of power production, in series hybrid vehicles it becomes necessary for the powertrain engineer to adopt the metric of power output when considering SFC and emissions, etc. The conventional automotive engineering approach of engine maps based on engine speed (x-axis) and load (y-axis) is no longer valid when there is no fixed link between engine and wheels and a much-constrained operating regime is permissible.

From the above, the power range of the LRE engine was therefore specified to be 15-38 kW.

ENGINE CONFIGURATION

GENERAL

This section will describe the decision process adopted once the desire to produce a cost-effective, lightweight and efficient 4-stroke SI engine with a maximum power output of 38 kW was taken. In order to fulfill this brief one critical observation was made. Nearly all modern SI engine attain their peak efficiency at near to full load at approximately maximum torque conditions - say, 3000-3500 rpm. The operating mode is nearly always homogeneous; spray-guided direct injection combustion systems can yield high efficiency at low load conditions relative to that which homogeneous systems can deliver, but the overall best efficiency of engines utilizing this approach is normally still when they operate in homogenous mixture mode [19,20]. Because of this homogeneous combustion situation and the removal of the need to minimize throttling losses through downsizing, a naturally-aspirated, port fuel injection (PFI) solution was decided upon for the initial design. At the best thermal efficiency condition of such an SI engine, a value of 35% is typical at a BMEP of 10-11 bar. This was considered the target condition, with the background desire for low cost.

It is worth explaining why a value of 35% thermal efficiency was considered acceptable when other researchers target higher values. While it had been a very early aim to create an engine with extremely good fuel consumption, a vehicle system-based assessment of the generally-desirable attributes of range extender engines (beyond being immediately acceptable to the marketplace) actually prioritises them thus:

- Lightness (of the complete package, i.e. including package volume) - this is the most important attribute since the range extender engine is always carried by the vehicle; even though

it should ideally operate infrequently, it contributes to electric energy consumption even when switched off.

- Cost - the range extender system is fundamentally an on-cost to a respecified BEV and so to be attractive it must maximize its cost advantage over the proportion of the battery pack that it is effectively replacing.
- Efficiency - because of the primary and secondary attributes, it is only important that the engine be 'efficient enough', i.e. efficiency should not overly compromise the first two considerations, although it will have to satisfy the customer in real-world usage⁹.

It is assumed that NVH can be addressed both within the design process and vehicle installation, and anyway, from the above survey of range extender types, a reciprocating 4-stroke SI engine had already been selected.

Nevertheless, it was generally considered desirable to maximize efficiency where it was possible to do so without compromising the other, more desirable attributes. This was a mantra within the design process.

Additionally, it was also desired that the engine operate at Lambda 1 (i.e. at the chemically-correct air-fuel ratio) throughout its operating map so that a simple TWC could be adopted for emissions control. This in turn meant that, in order to meet or better the efficiency target, a good combustion system would be required in order to support a high CR and operation on 95 RON fuel at Lambda 1 too. Ideally then the bore should be small (to permit a short flame path and hence reduced combustion period to the benefit of knock suppression), the spark plug should be as close as possible to the centre of the combustion chamber and a high degree of turbulence should be provided for the mixture. On top of this, the finished design should ideally have no overlap between successive exhaust valve events, in order to minimize hot exhaust residual contamination of incoming charge due to blow-down interactions between the cylinders. Finally, the exhaust back pressure should be kept as low as possible too, to reduce both pumping work and residual retention.

Thus the power target of 38 kW at approximately 3500 rpm implied a swept volume of 1.2 litres, with a BMEP of 10.8 bar. In order to provide some protection for later versions to make slightly more power, an absolute engine speed of 4000 rpm was specified. Since there is no possibility of the driver over-revving the engine with a missed gearshift, this speed limit was enforced rigorously with regard to reducing friction in the bearings and the valve train. The low specific power output (approximately 42 bhp/l or 31 kW/l) in turn meant that the valve area necessary would be small so that a small bore area could be specified and that the oil pump and

⁹ Again, see observations made in the previous footnote.

coolant pump capacities could be reduced accordingly. The following sections will discuss how these considerations helped to drive the final configuration chosen and hence the detailed design, after some initial consideration of the number of cylinders necessary for the engine.

CONSIDERATION OF CYLINDER NUMBER AND BORE SIZE

As mentioned above, the original concept design of the range extender engine was focused entirely on the goal of achieving a good maximum thermal efficiency (notionally 35% or above) at a single speed and load operating point with 95 RON at Lambda 1. It was envisaged that this approach would allow for much simplicity in the base engine design, which was in turn synergistic with the primary goals of low mass and reduced BOM cost.

In pursuit of this, it was initially decided to specify a combustion chamber with minimum surface area to minimize energy losses through heat rejection to the cylinder head and coolant. Consequently a spherical segment was chosen as the combustion chamber design. Into this would be positioned two large valves (again to cut down on the surface area ratio between the aluminium combustion chamber and the steel valve head to further reduce heat rejection). However, as these valves were originally to be arranged opposite to each other (in a manner similar to normal pent-roof designs) with the valve heads tangential to the combustion chamber surface, the valve head diameter could not be so large as to limit the spherical radius of the combustion chamber. Likewise, as a monoblock design had already been selected regardless of cylinder layout (see later section on this), there was an additional limitation on the valve angle and head diameter, as the valve seats would have to be machined through the cylinder bore. Similarly the valves themselves would have to be assembled from the same direction. Finally, this compromise was further complicated by the desire to design an engine with a small bore-to-stroke ratio in order to keep the combustion chamber volume to an absolute minimum.

A further complication was that it was desired that the easily-configured CR range of the engine should be within the range of 9:1 to 14:1, in order to produce later variants more-optimised towards operation on alcohol fuels (see above).

At the outset of the project it was not known what the bore and stroke of the engine should be, or indeed which arrangement of cylinders would be most appropriate to the final goals. The total displacement suggested either a two- or three-cylinder unit. The decision to minimise parts count and the choice of monoblock design ruled out a V arrangement

for the two-cylinder option. An inline two-cylinder had the theoretical friction advantage over a three-cylinder of having one fewer cylinder's worth of bearing surfaces (main and crank bearings plus the piston and rings in the cylinder bore itself). It would also have lower valvetrain friction for similar reasons.

However, because the engine was to be even-firing for a good noise signature, a 4-stroke, 2-cylinder crankshaft would generate a large vertical first- and second-order out of balance forces by virtue of having both crankpins at the same angle (a so-called '360° crankshaft'). In a parallel 2-cylinder engine with such a 360° crankshaft layout, there are two principal mechanisms to counter primary out-of-balance forces. The first is a conventional counter-rotating balance shaft system, and the second a 'pendulum balancer' (or reciprocating balance mass), which also counters secondary forces. However, because a pendulum balancer has to oppose the forces generated by two pistons, there could be a significant increase in engine mass. Also, driving the pendulum would necessitate a third throw on the crankshaft 180° from (and probably between) the crankpins. The bending forces between the adjacent crank throws would require an additional main bearing, thereby partially cancelling out the frictional gains of the 2-cylinder design. The pendulum oscillating mass itself would be connected to the crankshaft using a connecting rod. The mass would not have to reciprocate in a cylinder bore and would instead oscillate on another lever perpendicular to its connecting rod. This latter lever would be pivoted in the crank case at the end opposite to the bob-weight. It should be noted that similar designs have been used successfully in small engines by Ducati, Yamaha and latterly BMW [21].

The 2-cylinder engine would also have a much better ratio of total combustion chamber volume to surface area. However, to keep the highly-thermally-conductive aluminium surfaces of the combustion chamber roof and piston crown to a minimum compared to the cylinder bores and steel valves, a small bore would be required. This would then force the adoption of a very long stroke, since the entire engine displacement would be split between only two cylinders. The extra-long stroke also exacerbates the primary and secondary balance problems, as well as making the engine very tall and thus more difficult to package; this effectively made a pendulum balancer impossible to adopt, meaning that a conventional balance shaft would have to be used instead¹⁰, and the issue of mass and stress in the crankcase would become more pertinent (since unlike the pendulum balancer, a balance shaft does not resolve the out-of-balance forces directly at the crankshaft).

¹⁰Note that a pendulum balancer can also be configured at 90° to the cylinder axis, as in the Ducati Supermono engine, but if configured in that manner it cannot balance secondary forces, themselves of significant magnitude in the case of a large-swept-volume parallel twin as considered here.

In contrast, a 3-cylinder engine would have a more sensible bore/stroke ratio for the same displacement, resulting in more manageable overall engine dimensions. An even-firing 3-cylinder generates no out-of-balance forces, only primary and secondary couples. The primary couple would normally be cancelled out using an engine-speed, contra-rotating balance shaft but this would add mass and another set of bearings. However, as the engine would not be coupled directly to the drivetrain, the constraints on the engine mount system were much less severe, leading to the possibility that a balance shaft could be omitted from the 3-cylinder engine. It was felt that very soft engine mounts could be employed spaced further than normal from the engine's centre of gravity, thereby reducing the actual forces generated at the mounting points by the engine's out-of-balance couples. The inherently longer engine created by the inline 3-cylinder layout also naturally spaces the engine mounts wider apart. By contrast, the considerable out of balance forces of the 360°-crankshaft 2-cylinder would be transferred directly into the engine mounts regardless of mounting strategy and such an engine would have to be balanced by some mechanism.

PARAMETRIC COMPARISON OF 2- AND 3-CYLINDER OPTIONS

Despite the apparent advantages of a 3-cylinder, the exact choice was still unclear since the adoption of such a configuration went against the philosophy of minimum parts count and maximum volume / surface area ratio in the combustion chamber. A parametric study was therefore undertaken to compare key characteristics of both engine layouts over a range of bore sizes from 65mm to 88mm. These characteristics were:

- Combustion chamber volume to surface area ratio (initially considering CRs of 9:1, 10:1 and 11:1)
- Friction
- Overall engine package
- Forces at the engine mounts (in consideration of NVH)
- Combustion chamber shape and valve packaging versus monoblock construction
- Physical dimensions (package volume)

A parametric model of each of the two engine configurations was created and driven by the chosen bore/stroke combination. Assumptions about the design of surrounding components allowed the overall engine dimensions to be estimated. A simplified single-cylinder monoblock model consisting of the cylinder bore, combustion chamber and valves was created in CATIA v5 to check the combustion chamber layout generated by parametric model. Lotus Engineering Software's own Concept Cranktrain and Engine Simulation (LES) software was used to determine the out-of-balance forces or couples for each configuration, and estimate

friction losses, respectively. This section will detail the results of this investigation with regard to the characteristics list above.

Combustion chamber volume to surface area ratio

Figure 3 shows a plot of combustion chamber volume-to-surface area ratio for different engine configurations and CRs, all at 10° after top dead centre (ATDC). In general the 2-cylinder has a higher volume-to-surface area ratio than the equivalent 3-cylinder for the same bore size, as would be expected. However, there is a crossover within the range of bore sizes and CRs considered. For example, a 3-cylinder with a 65 mm bore and a 10:1 CR is as good as than a two-cylinder with an 88 mm bore and an 11:1 CR. This is an important consideration when the propensity to knock with larger bores is remembered: the advantage of the 2-cylinder in terms of volume-to-surface area ratio could easily be lost with any requirement to reduce the CR or retard the ignition due to the knock limit.

Friction

The friction for each engine configuration was estimated by LES and Concept Cranktrain using the basic parameters of bore, stroke and number of cylinders. A Concept Cranktrain model of each crankshaft configuration, and an LES friction estimator model of the piston were used to determine power losses through friction.

Due to its inherently long-stroke design and consequently higher mean piston speed, the 2-cylinder exhibits much higher piston friction than the 3-cylinder, but the twin's three-bearing cranktrain generates less friction than the triple's four-bearing design. However, the addition of the necessary primary balance shaft or pendulum balancer to the twin would necessarily offset this bearing friction improvement, but in fact high piston friction makes the total losses in the balanced 2-cylinder much higher than those in the unbalanced 3-cylinder, especially for small bore sizes (see Figure 4), even assuming a best-case balance shaft friction power of less than 0.5 kW. The subject of balance shaft friction will be returned to when discussing the results from development testing later in this paper.

It was envisaged that the optimized engine would have a long stroke and a small bore, however, the LES model had already demonstrated that a long-stroke engine creates higher friction. Therefore in attempting to optimize the engine, we may inadvertently have done the opposite by increasing friction; thus some analysis of the potential configurations was conducted versus a baseline 'traditional' configuration. This comparison showed that for all of the bore sizes considered, it is less likely that the 2-cylinder engine will demonstrate the required net increase in power to overcome its friction demerit unless the bore is very large. The same could be said of the 3-cylinder, but because of the smaller volume per

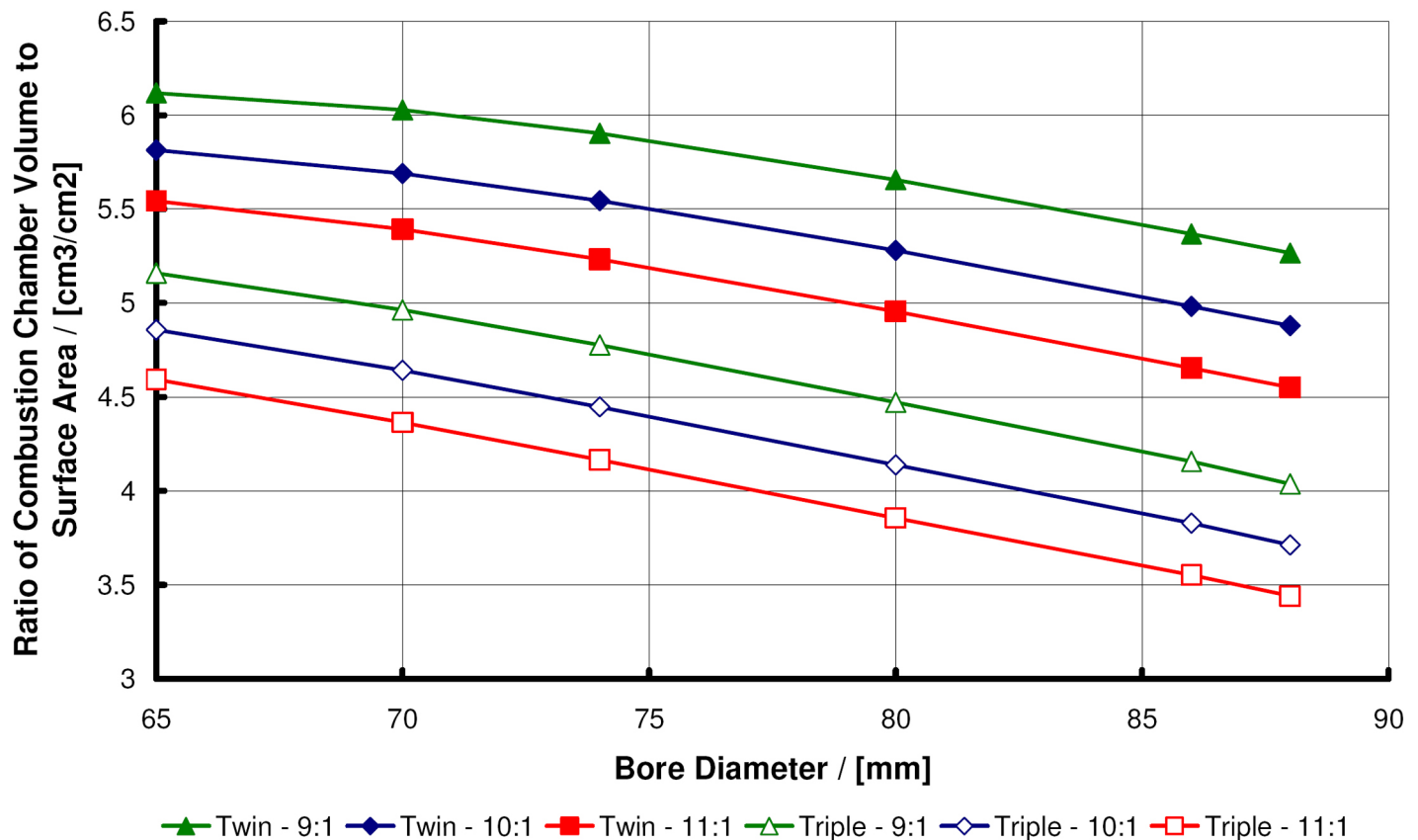


Fig. 3. Ratio of combustion chamber volume-to-surface area at 10° atdc for potential engine configurations at different compression ratios

cylinder, the trend was reduced as the stroke increases. This assessment therefore favoured the 3-cylinder configuration.

Forces at the engine mounts (NVH)

As discussed above, both engine configurations have inherently different out-of-balance characteristics when configured for even firing which itself was considered necessary for best aural NVH. The 2-cylinder configuration has vertical primary and secondary out-of-balance forces which are reacted directly into the engine mounts. By contrast, the 3-cylinder has both primary and secondary out of balance couples, which are resolved into forces at the mounts by virtue of the distance between the engine's centre of gravity and the mounts themselves.

For the 2-cylinder layout, assuming the engine is mounted at both ends, the out of balance forces will be reacted equally through both engine mounts:

$$\text{Primary force at each mount} = \text{Primary force} / 2 \quad (\text{Eqn 2})$$

For the range of bore sizes considered, the primary forces at the engine mounts range from 2.8kN to 4.5kN

For the three-cylinder with the same mounting configuration, the out of balance moment will be reacted equally through both engine mounts:

$$\text{Primary force at each mount} = \text{primary moment} / 0.5 \times \text{distance between mounts} \quad (\text{Eqn 3})$$

In this case the primary forces range from 1240 N to 1736 kN, which Figure 5 shows is approximately 35% lower than an equivalent 2-cylinder for the same bore size. In addition, the primary engine mount forces for the 3-cylinder configuration were all lower than 2-cylinder configuration for the complete range of bore sizes considered. The increase in engine length forced by the extra cylinder is advantageous in that it increases the moment arm length between the centre of rotation and the engine mounts. In the case of the 3-cylinder, from equation 3 increasing the span of the mounts will always reduce the forces, whereas this approach is not available for the 2-cylinder with the noise-signature-favoured 360° crankshaft.

Combustion chamber shape and valve packaging versus monoblock construction

As already mentioned, the decision to adopt a monoblock construction in order to reduce the pins count through the