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DOWNSIZING OF A NATURALLY ASPIRATED ENGINE TO TURBOCHARGED GASOLINE DIRECT INJECTION VARIABLE VALVE TIMING ENGINE

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ABSTRACT

The need to reduce CO_2 emissions and fuel consumption has become the biggest aim of Original Equipment Manufacturers (OEM's) over the past decade. Different strategies are being sought after to comply with stringent emission regulations being enforced on the global automotive market. Engine downsizing is proving to be the most cost-effective method of reducing emissions and fuel consumption. This paper presents 1-D CFD simulation using Ricardo WAVE of 30% downsizing on a 2.0 litre naturally aspirated spark ignition engine to a 1.4 litre turbocharged gasoline direct injection variable valve timing engine. Performance improvement has been achieved by using turbocharging and variable valve technology. Turbocharging a spark ignition engine increases the intake temperature and pressure, causing the mixture to detonate. To prevent engine mixture detonation, direct injection has been used, which also allows an increase in the compression ratio. The downsized engine has been optimised for maximum volumetric efficiency, maximum brake mean effective pressure and knock elimination. Valve opening timing, their durations and lifts have been optimised to improve volumetric efficiency. Fuel-air ratio and start of injection timing have also been optimised to improve the maximum brake mean effective pressure. The results indicated 74% improvement in torque, 102% improvement in power and 24% reduction in brake specific fuel consumption (bsfc).

Key words: downsizing, turbocharging, direct injection, variable valve timing, engine optimisation, 1-D engine simulation

1. INTRODUCTION

The growing emphasis in automotive development is the reduction in the greenhouse gases through exhaust flow, primarily carbon dioxide (CO_2). Legislations and increasing public awareness about global warming are leading vehicle manufacturers to reduce their carbon footprint from well to wheel. OEM's are making efforts to reduce greenhouse emissions and fuel consumption by calling for development in different areas, including vehicle dynamics, weight reduction and engine management systems, though most of the emission and fuel consumption reduction is seen through increasing the efficiency of powertrain systems. The most effective strategy to improve the efficiency of powertrain is engine downsizing [1, 2, 3, 4].

During most of the driving conditions, the conventional spark ignition engine runs at part load and a low engine speed application which leads to poor engine brake thermal efficiency. Throttling generates pumping losses which are quite high at part loads, and reduces the brake thermal efficiency. Reducing the engine displacement makes the engine operate at a higher load, thereby increasing the mechanical and brake thermal efficiency and reduces fuel consumption by limiting the pumping and friction losses and even heat transfer energy losses due to decreased total surface area [4, 5, 6].

Turbocharging permits larger specific power densities and allows partial recovery of exhaust energy. This is achieved by increasing the amount of air inducted per unit time mainly to burn a greater amount of fuel in a given engine [5, 7, 8, 9]. The downside of turbocharging a spark ignition engine is the increase in intake temperature and pressure of the engine. The increased intake pressure and temperature results in greater tendency of the mixture to detonate or pre-ignite [5, 7]. For this reason, turbocharged (TC) spark ignition engines employ lower compression ratios, which results in lower thermal efficiencies in direct comparison to similarly sized naturally aspirated engines. Thus turbocharged gasoline engines in turn result in having greater fuel consumption [3, 7, 10]. Rich mixtures can also be used to control knocking, which further increases the specific fuel consumption [5, 7].

Another limitation of turbocharged engine is the transient response, limited by the inertia of the turbine, compressor and the turbine shaft. Even though the steady state torque curve of a turbocharged engine can be made equal or larger than similar naturally aspirated engine, the driver can perceive the difference in the transient response. This can be overcome by either running the engine at higher engine speed, which can further decrease the fuel economy, or by the need to use multi-stage turbocharging [11].

Also, for a turbocharged port fuel injection engine, the valve overlap needs to be confined to a very small range otherwise scavenging of burnt gasses would also be accompanied by the loss of large quantities of fuel-air mixture due to higher intake pressures using turbocharging. Valve overlap is thus strictly avoided for turbocharged gasoline engines [5].

To prevent the fore mentioned knock problem, direct injection (DI) technology has been introduced to gasoline engines, which can also allow the increase of the compression ratio by about 10% [5]. Injecting the fuel directly into the combustion chamber reduces the charge temperature in the cylinder due to fuel vapourisation, taking up the latent heat of vapourisation from surrounding compressed air mass. The reduction in intake air temperature helps to prevent knocking, and increases both the charge density and the volumetric efficiency [5, 10].

Since in direct injection engines, fuel delivery is independent of air intake as only air is inducted in the intake stroke, better scavenging is possible in comparison to conventional port fuel injection engines [4, 5]. This makes it possible to phase the intake valve opening and exhaust valve closure overlap, reducing the residual burned gases concentration. For turbocharged engines, at high engine speeds, the valve overlap period should be reduced to take into account the balance between exhaust back pressure and intake pressure becoming negative, increasing the risk of back flow resulting in poor scavenging. Variable valve timing (VVT) can result in specific fuel consumption benefits of up to 4% at low and medium engine speeds and up to 7% at high engine speeds [5].

The aim of this work was to downsize an existing 2.0 litre naturally aspirated port fuel injection engine, with equivalent or more torque and power characteristics of the base engine using turbocharging and variable valve timing. Therefore the torque and power densities of the downsized engine would be higher than the base engine.

2. SIMULATION AND DOWNSIZING METHODOLOGY

Ricardo WAVE software package has been used for steady state and transient analysis. WAVE is a 1-D engine and gas dynamics simulation software package which enables performance simulations on wide variety of intake, combustion and exhaust system configurations [12].

The 2.0 litre engine was geometrically downsized to 1.6, 1.4 and 1.2 litre engines to study the engine performance reduction for each engine. This was required to select the engine with least cubic capacity that would match the performance of the 2.0 litre base engine after incorporating the turbocharger, direct injection and variable valve technology. Benefits of having a smaller engine cubic capacity have already been discussed in the previous section.

The bore and stroke have been reduced to decrease the engine cubic capacity. 5 engine configurations for each 1.6, 1.4 and 1.2 litre engines with varying bore to stroke ratio were created to select the engine with the best performance respectively. The performances for these engines have been compared for torque, power and bsfc. To design an engine which would directly replace the existing 2.0 litre engine, the longitudinal length of the downsized engine was kept as less than or equal to the 2.0 litre base engine. Therefore, the summation of bore and cylinder centre to centre distance has been kept less than or equal to the base engine. From each of the 5 configurations for 1.6, 1.4 and 1.2 litre engines, the configuration with the best performance was selected, which gave final 3 configurations for the downsized engines. Bore to stroke ratio of 1.5 for 1.6 litre engine and 1.6 for 1.4 and 1.2 litre engines was found out to be optimum with respect to the engine performance and the longitudinal length of the engine.

Section 3 will discuss the reasoning for selection of the 1.4 litre engine for conversion to turbocharged direct injection engine with variable valve timing.

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3. SIMULATION RESULTS

Figures 1 and 2 show the engine performance characteristics against the engine speed for the 2.0 litre base engine and the selected downsized engines. The maximum torque and power reductions due to downsizing are 22%, 34% and 45% for the 1.6, 1.4 and 1.2 litre engines respectively.

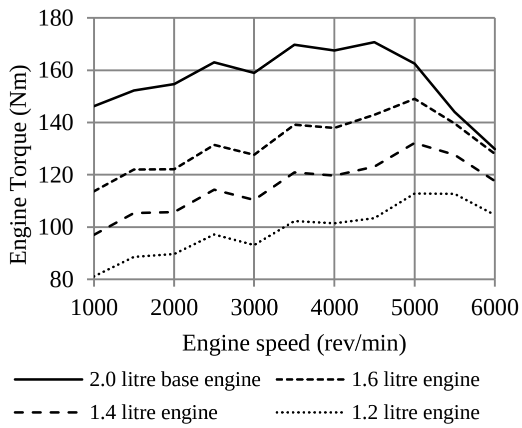

Figure 1. Engine torque against engine speed

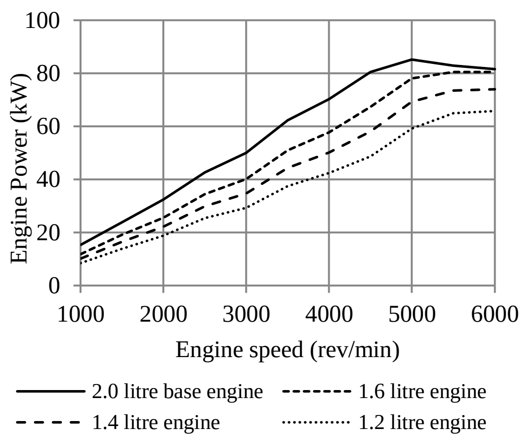

Figure 2. Engine power against engine speed

The individual curves clearly indicate a reduction in torque and power at various engine speeds, most noticeable at 3000 rev/min, and was speculated as the limitation of the mathematical model on WAVE, which causes an increase in the back flow through the intake valve reducing the volumetric efficiency, and hence the engine performance.

Figure 3 shows the variation of the backflow through the intake valve during the intake and exhaust valve overlap period against the engine speed for the 2.0 litre base engine.

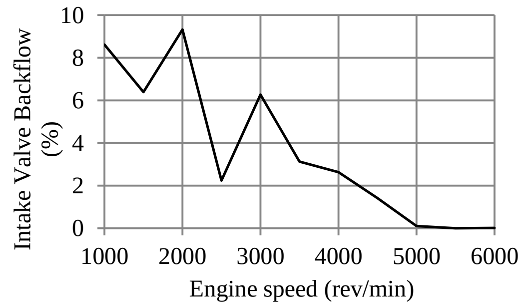

Figure 3. Engine intake valve back flow against engine speed

The peaks represented in Figure 3 correspond to the engine speeds at which the performance drops for the 2.0 litre base engine and the downsized engines shown in Figure 1 and Figure 2. During the optimization of the downsized engine, efforts have been made to minimise the intake valve back flow by altering the intake and exhaust valve overlap durations for various engine speeds.

Figure 4 represents the variation of engine brake specific fuel consumption against the engine speed for the 2.0 litre base engine and the downsized engines. The maximum improvement in bsfc achieved is 11%, 13% and 14% for the 1.6, 1.4 and 1.2 litre engines respectively, above the engine speed of 2000 rev/min.

The results clearly show the improvement in bsfc achieved by downsizing due to an increase in mechanical and thermal efficiency, which support the use of downsized engines to reduce the CO_2 emissions.

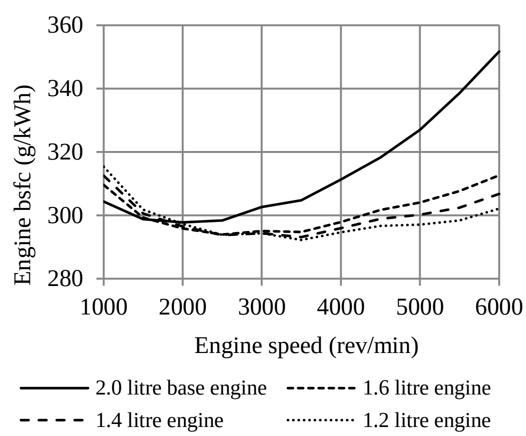

Figure 4. Engine bsfc against engine speed

Figure 4 also indicates that for engine speeds below 2500 rev/min, the downsized engines are showing an increase in bsfc with decrease in the engine cubic capacity. This could be due to the

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increasing effects of the surface to volume ratio which leads to an increase in the heat loss to the cooling system and therefore reduces the thermal efficiency and bsfc.

The engine torque, power and bsfc plots indicate an improvement for most of the engine speeds for the downsized engines compared to the 2.0 litre base engine. Inferring from the results of the above simulations, it was speculated that incorporating the turbocharger, direct injection and variable valve timing technology would suit 1.4 litre engine to match the performance with the 2.0 litre base engine. The 1.6 litre engine was dropped for conversion and optimisation, as the use of smaller cubic capacity engine was possible. The 1.2 litre engine was dropped for conversion and optimisation as the use of turbocharging, direct injection and variable valve timing technology might not be sufficient to match the performance compared to the 2.0 litre base engine. Therefore the 1.4 litre engine was selected.

Garrett GT2056 turbocharger maps, in-cylinder injectors and parameterisation of valve timing, their durations and lifts were incorporated into the 1.4 litre engine model and the compression ratio increase to 10 was done as part of the conversion process on WAVE.

The 1.4 litre downsized engine prior to optimization is called as *Source Engine* henceforth.

4. ENGINE OPTIMISATION

Since a spark ignition is quantitatively governed, the air flow, that is, volumetric efficiency has been optimised first, and then combustion and knock optimisation have been performed [13]. Further sub-sections in this paper discuss the parameters selected for optimisation of volumetric efficiency, combustion and knock.

4.1. Volumetric efficiency optimisation

Volumetric efficiency has been optimised by controlling the intake valve opening, the intake valve scaling and the exhaust valve scaling. Valve scaling is a multiplication factor which allows the valve duration to be stretched or shrunk and the valve lift to be increased or decreased simultaneously.

Parametric study results for the intake valve opening against engine speed are shown in figure 5. Using the sweep constants in WAVE, different values of the intake valve opening have been used to run the simulation and obtain the volumetric efficiency results for different engine speeds. The outcome is a 3-D plot as shown in Figure 5.

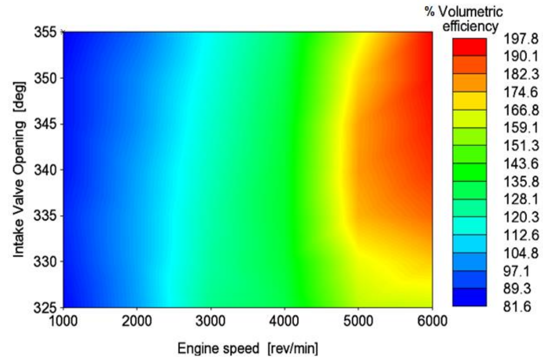

Figure 5. Effect of intake valve opening on volumetric efficiency

For each engine speed, the corresponding intake valve opening value has been selected corresponding to the maximum volumetric efficiency. The outcome of this study is the varying intake valve opening timings for different engine speeds. The graph clearly indicates that with decreasing engine speeds, an earlier intake valve opening is desired for higher volumetric efficiency and with increasing engine speed late intake valve opening is desired. This is due to the low incoming air velocity through the intake valve at low engine speeds, requiring an early valve opening to induct the same amount of air compared to late intake valve opening at high engine speeds.

Figure 6 represents a similar parametric study done for intake valve scaling against different engine speeds. The effect of intake valve scaling for maximum volumetric efficiency has been studied for each engine speed. The graph indicates that with the increase in engine speed, a higher intake valve lift and its duration are favourable for higher volumetric efficiency and with decreasing engine speeds, a lower intake valve lift and its duration are favourable to achieve higher volumetric efficiency.

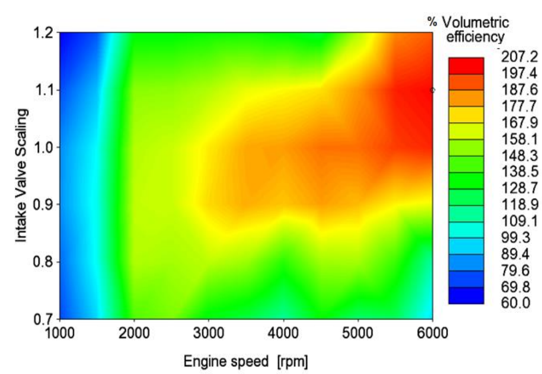

Figure 6. Effect of intake valve scaling on volumetric efficiency

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The values of intake valve scaling corresponding to the maximum volumetric efficiency have been incorporated in the downsized engine.

Figure 7 shows the effect of exhaust valve scaling against engine speed on the volumetric efficiency. The graph indicates that with the increase in engine speed, a higher exhaust valve lift and its duration is beneficial for volumetric efficiency and at lower engine speeds, a lower exhaust valve lift and duration benefits the volumetric efficiency.

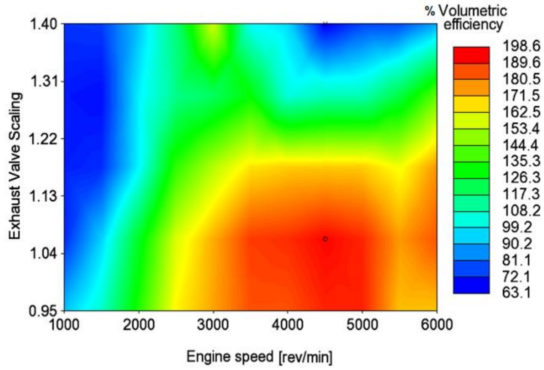

Figure 7. Effect of exhaust valve scaling on volumetric efficiency

Similar to the other parametric studies, values of exhaust valve scaling for different engine speeds corresponding to maximum volumetric efficiency have been selected and incorporated for optimisation.

A parametric study on the effect of exhaust valve opening on volumetric efficiency was also considered, but exhaust valve opening did not show an effect on the volumetric efficiency, because the air gets inducted into the cylinder only after the exhaust valve is about to close.

The parametric values extracted from the above studies for maximum volumetric efficiency have been used for optimisation. Figure 8 shows the variation of the volumetric efficiency against the engine speed for the optimised engine compared to the source engine, indicating an improvement over most of the engine speed range. This graph also shows the minimisation of the intake valve back flow leading to an improvement in the volumetric efficiency.

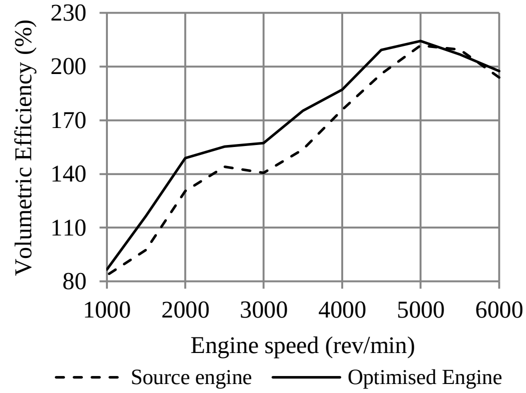

Figure 8. Volumetric efficiency against engine speed

4.2. Combustion optimisation

Combustion has been optimised by controlling the start of injection and fuel-air ratio to achieve a condition known as maximum braking torque point.

Figure 9 shows the parametric study of start of injection before top dead centre (btdc) against engine speed on brake mean effective pressure. The graph indicates an early start of injection leads to an increase in bmep for all engine speeds, due to the fuel getting more time to vaporise before the start of combustion and forms a better homogeneous mixture compared to late start of injection.

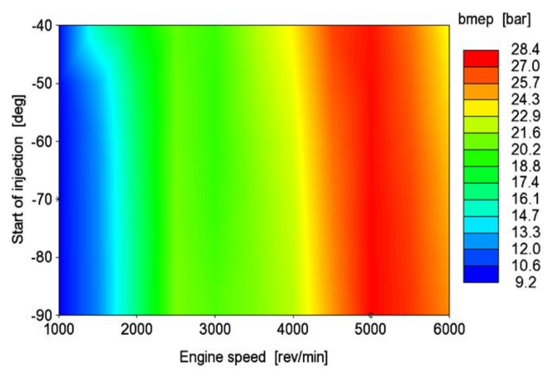

Figure 9. Effect of start of injection on bmep

From the graph it is evident that a 90° btdc start of injection is beneficial for achieving higher bmep, beyond which the bmep only marginally increases.

Figure 10 shows the effect of fuel-air ratio on bmep against engine speed. It is evident that with the increase in engine speed, a rich mixture, that is, a mixture with higher fuel-air ratio is beneficial for achieving higher bmep.

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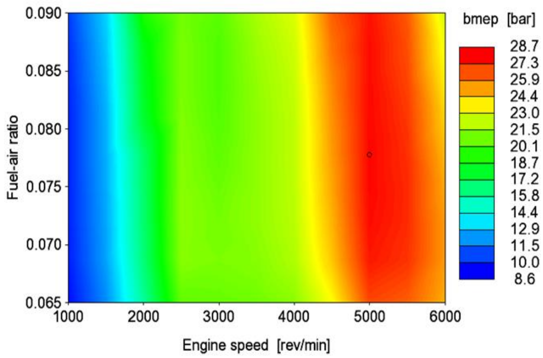

Figure 10. Effect of fuel-air ratio on bmep

At low engine speeds, that is 1000-2000 rev/min, a stoichiometric air-fuel ratio is maintained, and at high engine speeds, that is 4500-6000 rev/min, the air-fuel ratio is about 12.5.

The effect of fuel-air ratio on exhaust gas temperature has also been investigated to control the exhaust gas temperature. Figure 11 shows the parametric study results of fuel-air ratio on exhaust gas temperature against engine speed.

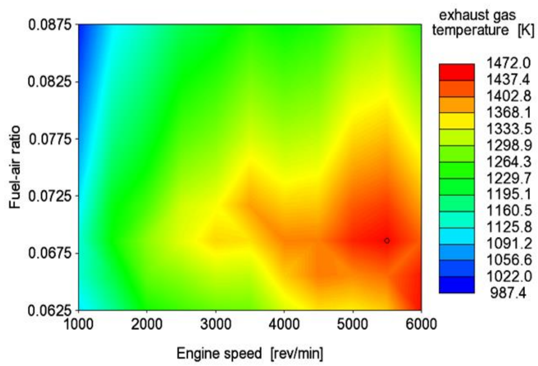

Figure 11. Effect of fuel-air ratio on exhaust gas temperature

Rich mixtures are required at medium to high engine speeds, that is 3000-6000 rev/min, to keep the exhaust temperatures below 1400 K to protect the turbocharger in accordance with the specifications of the turbocharger manufacturer [14].

From the parametric studies done for combustion, the optimised values for start of injection and fuel-air ratio were incorporated in the downsized engine. Figure 12 shows the variation of the overall improvement in bmep achieved by combustion optimisation against engine speed.

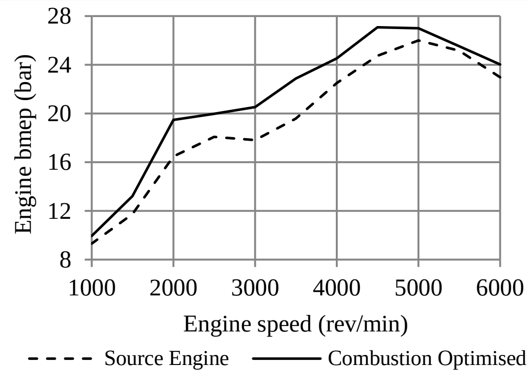

Figure 12. Brake mean effective pressure against engine speed

A maximum of 18% improvement in bmep is achieved for the combustion optimised engine compared to the source engine.

4.3. Knock optimisation

Figure 13 shows the variation of the knock intensity for the source engine against engine speed. Knock intensity is represented as a normalized fraction of the fuel remaining at the knock event.

The engine shows high levels of knock at engine speed ranges of 2000-2500 and 4000-6000 rev/min. The trend should be that an increase in engine speed should lead to decrease in knock which is contrary to the results shown in Figure 13.

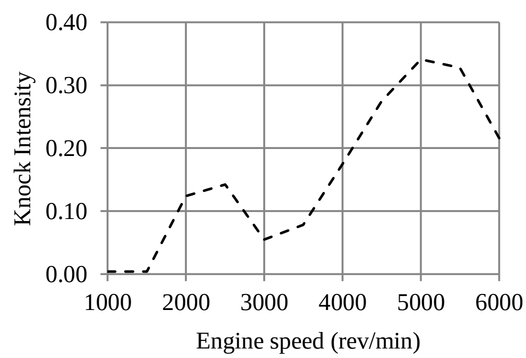

Figure 13. Knock intensity against engine speed

It is speculated that due to the use of simple 1-D Wiebe model to simulate the combustion, the swirl generated inside the combustion chamber is not taken into account. A further investigation using a 3-D combustion model on the software package can be considered.

Figure 14 shows the effect of the 50% burn point on the knock intensity against engine speed. The 50% burn point is represented as degrees btdc. The graph shows that with advancing the start of

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combustion, knock intensity increases for all engine speed ranges. It is also evident from the graph the limitation of the 1-D combustion model used which displays high knock occurrences for the engine speed range of 4000-6000 rev/min, discussed previously.

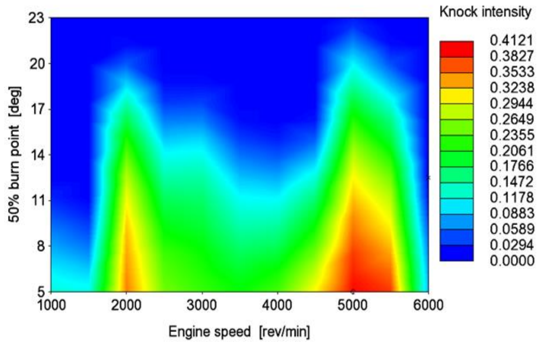

Figure 14. Effect of 50% burn point on knock intensity

To eliminate the engine knock, the 50% burn point, and hence the start of combustion was retarded for the engine speed ranges with knock event as specified before. The values for the retardation have been incorporated in the knock optimised engine and a graph of peak cylinder pressures has been plotted for the source engine and the knock optimised engine against engine speed, as shown in Figure 15.

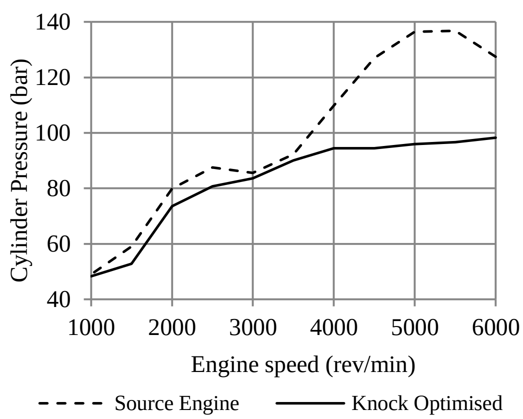

Figure 15. Peak firing pressure against engine speed

The graph clearly shows the reduction in peak cylinder pressures achieved for the knock optimised engine compared to the source engine, most noticeable at an engine speed range of 4000-6000 rev/min. The maximum cylinder pressure at 6000 rev/min is 98.2 bar.

Table 1 shows the specifications of the 1.4 litre turbocharged gasoline direct injection variable valve timing (TC GDI VVT) engine. The engine configuration is inline-4 over square, that is with the bore to stroke ratio greater than 1.

Table 1. 1.4 litre engine specifications

Bore	89.3 mm
Stroke	55.8 mm
Bore/ Stroke	1.6
Connecting Rod length	95.0 mm
Connecting Rod Length/ Crank radius	3.41
Total displacement	1398 cc

5. STEADY STATE SIMULATION

5.1. Wide Open Throttle performance

Figure 16 shows the variation of the engine torque against the engine speed for the 1.4 litre engine compared to the 2.0 litre base engine. *Turbolag* can be seen at 1000-1500 rev/min where the boost pressure is low and the engine is not able to develop high brake mean effective pressures. The turbocharger develops a boost pressure of 2.5 bar with wastegate fully open at 4500 rev/min, producing a peak torque of 296.5 Nm at that engine speed. A maximum of 43% improvement in torque is seen at an engine speed 2000-3000 rev/min compared to the 2.0 litre base engine, and a maximum of 102% improvement at a rated engine speed of 6000 rev/min.

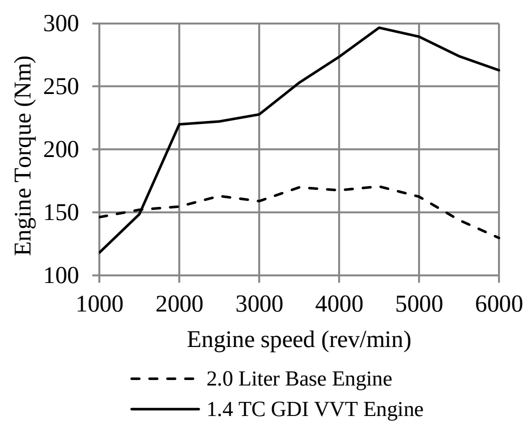

Figure 16. Engine torque against engine speed

From the torque characteristic curve, the suggested duty cycle of the engine should be to accelerate the vehicle from rest to cruise speed from an engine speed of 2000 to 5000 rev/min for maximum torque output and cruise at engine speed of 2000 to 3000 rev/min for a near constant torque delivery of 220-230 Nm.

Figure 17 shows the variation of the engine power against the engine speed for the 1.4 litre engine compared to the 2.0 litre base engine. The improvement achieved for the 1.4 litre engine is similar to the torque characteristic results compared to the 2.0 litre base engine.

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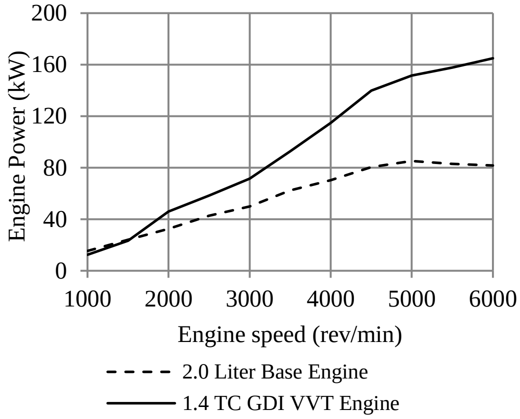

Figure 17. Engine power against engine speed

Figure 18 shows the variation of the brake specific fuel consumption against engine speed for the 1.4 litre engine compared to the 2.0 litre engine. The graph clearly shows significant improvement in specific fuel consumption achieved over all of the engine speeds. A minimum of 19% improvement is achieved throughout the engine speed range, with a maximum improvement of 24% at an engine speed of 2000 rev/min, at which a minimum bsfc of 227.6 g/kWh is observed.

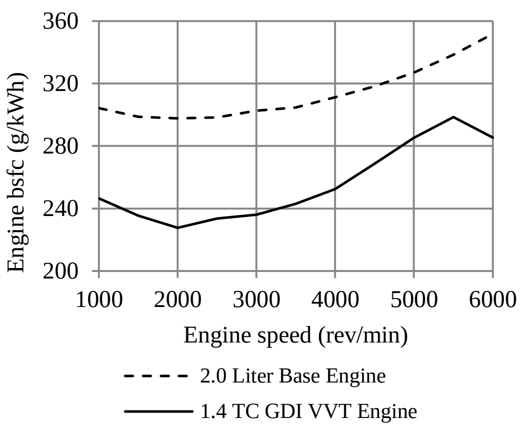

Figure 18. Engine bsfc against engine speed

A downsizing of 30% in cubic capacity has led to improvement in 24% engine brake specific fuel consumption.

Figure 19 shows the variation of the engine brake thermal efficiency against engine speed for the 1.4 litre engine compared to the 2.0 litre base engine. The maximum brake thermal efficiency achieved is 37% at an engine speed of 5000 rev/min compared to 28% for the 2.0 litre base engine at the same engine speed.

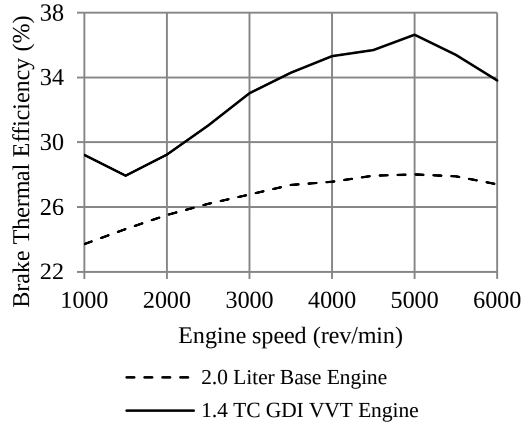

Figure 19. Engine performance against engine speed

Evident from the results, there is a significant improvement in the thermal efficiency for the downsized engine, which supports the idea of downsizing engine cubic capacity as an effective means of reducing fuel consumption and improving engine thermal efficiency.

The maximum average air intake temperature is 352 K, which suggests an improvement in the intercooler performance is required to keep the intake temperatures in the range of 330 K [15]. High intake temperatures will also reduce the volumetric efficiency.

Steady state simulation results indicate acceptable performance characteristics for the downsized engine. With downsizing and turbocharging, power density of 118 kW/L and torque density of 212 Nm/litre have been achieved. The peak performance densities (as of 2012) are for Mitsubishi Lancer Evolution FQ-400 at 151 kW/L and 263 Nm/L for power and torque respectively [16]. The newly developed Ford EcoBoost engine with similar turbocharged direct injection variable valve timing technology offers 134 kW/litre power, and therefore the results of the simulation are acceptable [17].

5.2. Part Throttle performance

Figure 20 shows the part throttle performance of the engine against engine speed as a measure of brake mean effective pressure and brake specific fuel consumption. Minimum bsfc of 227.6 g/kWh is observed for engine speeds from 1700 to 2800 rev/min with 90 to 95% throttle. Below 2 bar bmep, the bsfc values are extremely high because the throttle being at a minimum position, the engine is being made to run at high engine speeds, which signifies a condition of wheels driving the engine. In this case, the engine works as a compression brake, which WAVE Build code is unable to comprehend,

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and displays negative values for bsfc. This is the reason blue coloured regions can be seen below 2 bar bmep. This region should be considered as high bsfc values, signified by red colour code in the surrounding regions. The author's speculate that transient analysis using a comprehensive driveline model would resolve this limitation of the software package.

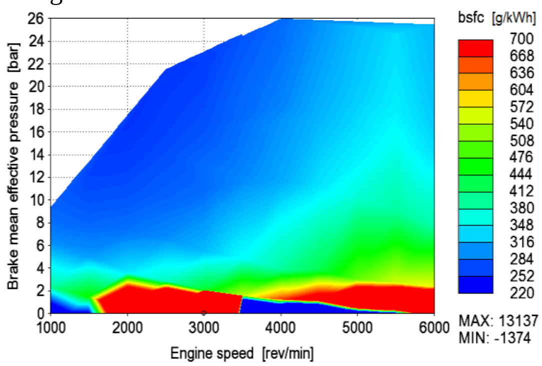

Figure 20. Engine part throttle performance map

6. TRANSIENT SIMULATION

As a case study, a transient analysis based on New European Driving Cycle (NEDC) using simplified 2013 Ford Focus driveline parameters was used to extract results for comparison of fuel economy between the 2.0 litre base engine and the 1.4 litre downsized engine. The gear ratios and the final drive ratio have been converted into vehicle speeds based on the engine speed, and the vehicle speeds for NEDC in turn have been converted to engine speeds according to the engine duty cycle suggested in the previous section. The following assumptions and simplifications were made to run the transient simulation:

1. The engine idle engine speed is 1000 rev/min (during the time when vehicle is at rest).
2. The vehicle operates in wide open throttle mode for the engine.
3. The engine speed changes during gear changes are considered momentarily and their effect is considered negligible.
4. The vehicle loads, static and dynamic resistances are ignored.
5. The driveline model (gear ratios and final drive ratios) for both engines are the same.

Similar simplifications were made on the 2.0 litre base engine, hence a direct comparison between the simulations is considered as a fair assumption; however WAVE code allows only constant parameter values during transient analysis rather than variable, the direct consequence of which is that the results are compared between the 2.0 litre base engine and the 1.4 litre source engine rather than the optimised engine.

Figure 21 shows the variation of the fuel economy comparison against time for the 1.4 litre engine and the 2.0 litre base engine.

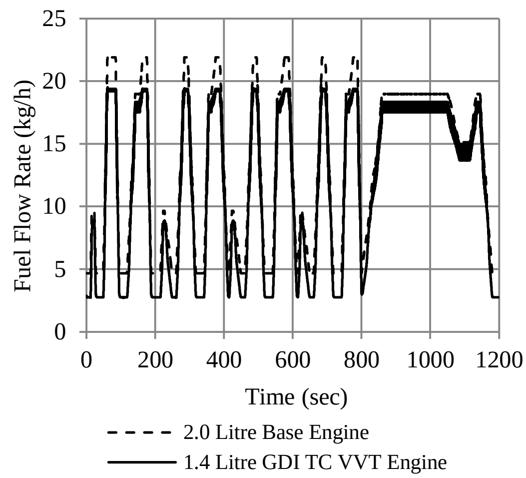

Figure 21. Fuel flow rate against engine speed

The fuel economy improvement achieved in terms of average fuel flow rate is 9.3% for the 1.4 litre engine compared to the 2.0 litre base engine.

Table 2 presents an important outcome of the transient simulation, as the overall performance improvement achieved in terms of average power, torque, bsfc and fuel flow rate for the 1.4 litre engine compared to the 2.0 litre base engine.

Table 2. Transient analysis performance comparison

	Avg. power (kW)	Avg. torque (Nm)	Avg. bsfc (g/kWh)	Avg. fuel flow rate (kg/h)
2.0 litre	51.5	161.6	308	15.9
1.4 litre TC GDI VVT	63.3	192.0	230	14.5
Improved	23.0%	18.8%	25.2%	9.3%

The transient simulation represents more closely the actual results for the vehicle, rather than the engine performance, which were discussed in the steady state simulation results. With a direct replacement of the 2.0 litre engine with the 1.4 litre engine, the average improvement for the same vehicle would be 23% for torque, 18.8% for power and 9.3% for the fuel consumption.

7. CONCLUSIONS

The 2.0 litre base engine has been successfully downsized and optimised to 1.4 litre turbocharged gasoline direct injection engine with variable valve timing, with the achievement of 30% downsizing.

1. A peak torque of 296.5 Nm has been achieved with an improvement of 74% compared to the base engine at an engine speed of 4500 rev/min. Low end torque can be further improved by using variable geometry turbocharger.
2. A peak power of 165 kW has been achieved with an improvement of 102% compared to the base engine at the rated engine speed of 6000 rev/min.
3. A minimum bsfc of 227.6 g/kWh has been achieved with an improvement of 24% compared to the base engine at and engine speed of 2000 rev/min.
4. Late intake and exhaust valve closing showed to improve volumetric efficiency with increasing engine speed. Early intake valve opening and closing showed to improve volumetric efficiency with decreasing engine speeds.
5. Stoichiometric air fuel ratios at low engine speeds and rich mixtures at high engine speeds improve brake mean effective pressure. Qualitative Governing can be further investigated with the use of Exhaust Gas Recirculation (EGR) to eliminate the throttle body.
6. Rich mixtures at high engine speeds are also required to limit exhaust gas temperatures to protect the turbocharger. Use of 4-2-1 exhaust system design to further reduce exhaust gas temperatures can be studied, which will optimise the exhaust flow.
7. An early start of injection is desired for all engine speed ranges; however injector and spray optimisation can be performed using 3-D combustion modeling.
8. The transient analysis performed on New European Driving Cycle (NEDC) showed a 9.3% improvement in overall fuel economy compared to the 2.0 litre base engine. Similar case studies can be performed using other internationally recognised driving cycles.

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