Synergies of Cooled External EGR, Water Injection, Miller Valve Events and Cylinder Deactivation for the Improvement of Fuel Economy on a Turbocharged-GDI Engine; Part 1, Engine Simulation

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Abstract

As CO2 legislation tightens, the next generation of turbocharged gasoline engines must meet stricter emissions targets combined with increased fuel efficiency standards. Promising technologies under consideration are: Miller Cycle via late intake valve closing (LIVC), low pressure loop cooled exhaust gas recirculation (LPL EGR), port water injection (PWI), and cylinder deactivation (CDA). While these efficiency improving options are well-understood individually, in this study we directly compare them to each other on the same engine at a range of operating conditions and over a range of compression ratios (CR). For this purpose we undertake a comprehensive simulation of the above technology options using a GT-Power model of the engine with a kinetics based knock combustion sub-model to optimize the fuel efficiency, taking into account the total in-cylinder dilution effects, due to internal and external EGR, on the combustion. Based on a carefully designed design of experiments (DOE) our results indicate a potential CO2 improvement of up to 7% at part load conditions compared to the base engine without the above mentioned technologies while limiting the loss of high load performance using the different technologies in combination.

Introduction

The four-stroke Otto cycle gasoline engine is still the dominant form of propulsion for the majority of passenger cars worldwide after more than 140 years since its inception [1]. Its longevity and continued development is a remarkable achievement considering the many challenges from alternative combustion and propulsion concepts such as diesel engines and battery electric vehicles. In fact, it is recently experiencing a resurgence as a more balanced approach for both CO2 and criteria pollutant emissions. Moreover, even in electrified powertrains, the importance of the gasoline engine has not diminished as it is a reliable and readily available power plant that can charge the battery and extend the electric range of many vehicles.

A key development that dramatically improved the performance envelope of the internal combustion engine has been exhaust gas turbocharging [2,3,4]. Using some of the waste energy in the exhaust to drive more air into the cylinders has proven to be an effective enhancement to the conventional naturally aspirated engine. With turbocharging, a smaller engine can propel the same vehicle at equal or even increased levels of performance while still satisfying emission and drivability requirements. This led to the current trend of engine downsizing or small size efficiency. However, typical downsized engines were also found to be less than ideal regarding CO2 efficiency in real world applications when compared to certification test figures [5]. Understanding and removing some of the obstacles like knock or abnormal combustion which leads to efficiency compromises, is therefore an important goal in realizing the true benefits of downsizing.

For naturally aspirated engines, some manufacturers have been applying the Atkinson cycle principle of extended expansion, in order to increase the thermodynamic efficiency of the four-stroke engine [6]. Using a standard valve-train but with longer duration intake cams allows one to use a higher geometric compression ratio (gCR), which is beneficial from a thermodynamic cycle efficiency perspective over the indicated loop [Eq. 1]. Having the intake valve open for longer
than 180° leads to a rejection of some of the inducted air, that effectively reduces the final gas compression pressure and temperature and consequently helps reduce the risks of auto-ignition at higher engine loads.

\[
\eta = 1 - \frac{1}{CR^{\gamma-1}} \quad \text{Equation 1.}
\]

The downside of the Atkinson cycle is however a reduction in the specific power of the engine. Thus, it can be viewed as a counter-trend to downsizing, resulting in larger and heavier engines.

In view of the two countering trends, it is natural to consider applying turbocharging to an over-expanded cycle engine. The downsizing and efficiency aspects can then be combined to reach high specific output while, at the same time, preserving the efficiency benefits of the Atkinson cycle. We should mention here that an alternative approach is the Miller cycle which was originally based on the idea of rejecting some of the inducted cylinder air in a supercharged engine [7], but nowadays it is more closely associated with EIVC (early intake valve closing) or short duration intakes.

One missing component after combining boosting with over-expansion is the combustion aspects of higher CR and its higher end of compression pressures and temperatures. It should be noted that both boosting and over-expansion by valve duration lead to abnormal combustion events at high load that can make the combination infeasible. In this paper we propose two methods to address this issue. At moderate loads we propose cooled external EGR as an effective solution for knock mitigation where it can be used to replace some of the variable valve timing (VVT) driven hot internal EGR to push up the knock boundary [8]. While effective at increasing engine efficiency, we find cooled EGR as performance lowering at high loads, and difficult to significantly lower the charge temperatures using reasonably sized EGR coolers. Therefore we also evaluated individual port water injection at load levels where knocking is driven primarily by thermal effects. Water is a fluid with a very high heat of vaporization and therefore an effective cooling medium to extend the limits of performance without upsizing the engine [9].

A similar approach to our study was used by [10] to investigate the potential BSFC benefits of combining intake variable valve lift (VVL), variable compression ratio (VCR), external EGR and port water injection using a 1-d simulation model and a genetic algorithm based optimizer. Three load points at a single engine speed were used to represent a WLTP (Worldwide harmonized Light vehicles Test Procedure) duty cycle and the optimizer always selected an EIVC solution, which could have been a local minimum. Potential improvements of BSFC between 5 and 9% are predicted with different mechanisms leading to the benefit at each load level, while the performance limit impacts of the different technologies are not covered. In [11], intake valve closure (IVC) was swept from early to late and the bi-modal optima of the BSFC risked finding one branch solution only when using a blind search method.

In our study, we explore the combination of downsized boosting, over-expansion with long duration intake valve events, and charge dilution in a systematic way to understand the ultimate limits of performance as well as efficiency gains, including all interactions and synergies. For the purpose we adopted a modern direct injection turbocharged engine from Hyundai Motor Company as our baseline and developed a detailed GT-Power simulation model of the same engine. After validating the model with actual engine test data, we modified the engine as we describe below and carried out an exploration of the entire calibration space at key corners of operation. The usefulness of exploring in simulation is obvious when considering the time and cost involved in actual engine testing. In our case we had a parallel engine build and test program, but we also chose to simulate the same operating points with the model, in order to provide guidance and a deeper understanding of the overall engine behavior. The results of the engine testing are described in a companion paper [12].

### Engine Model Description

The base engine selected for this study is a four cylinder spark-ignited, turbocharged, direct injection engine typically found in compact to mid-sized vehicles currently in the marketplace. The main engine parameters are shown in Table 1. The engine is equipped with dual continuously variable valve timing as well as an electrically actuated waste-gate valve in the turbine.

The base engine was modeled in GT-Power using a medium fidelity approach for the level of discretization (Figure 1). This choice allowed fast run times while retaining a good accuracy, which proves useful when the DOE calls for a large number of experiments. The main purpose of the simulation model was to verify the feasibility of the different combination concepts at varying operating conditions and serve as a test guidance tool.

To capture the effects of variable valve timings and LPL EGR, as a first step we developed an empirical correlation of the ignition timings to our best estimates of total burned residual fraction as influenced by the valve overlap period and average delta pressures across the cylinders. As a second step, we then applied the kinetic knock model in GT-Power to represent the knock limited behavior at high loads. It should be noted that the resulting combustion model was therefore semi-predictive, in that the center-of-combustion and burn-durations were varying as functions of the ignition timing, ignition delay and the total residual level, but the burn rates

#### Table 1 Main engine parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of cylinders</td>
<td>4</td>
</tr>
<tr>
<td>Displaced volume [L]</td>
<td>1.4</td>
</tr>
<tr>
<td>Number of valves</td>
<td>4</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>10</td>
</tr>
<tr>
<td>Rated power [kW]@RPM</td>
<td>104@6000</td>
</tr>
<tr>
<td>Low end torque [Nm]@RPM</td>
<td>244@1500-3200</td>
</tr>
<tr>
<td>Fuel injection system</td>
<td>Direct</td>
</tr>
<tr>
<td>Boosting system</td>
<td>eWG single-scroll</td>
</tr>
<tr>
<td>Valve duration - intake [crank\°]</td>
<td>248</td>
</tr>
<tr>
<td>Valve duration - exhaust [crank\°]</td>
<td>232</td>
</tr>
</tbody>
</table>
were imposed per the custom empirical correlation model developed specifically for this study.

Because GT-Power does not capture cycle-to-cycle variations, we also developed a correlation between combustion stability and the total in-cylinder residual to hold the maximum dilution below a chosen limit value. The LPL EGR has a high performance cooler followed by a controlling EGR valve, and the flow is introduced just before the turbo-compressor via a T-junction. The fresh air and EGR mix is then compressed and cooled in a charge air cooler. In our model the charge air cooling is scheduled by engine speeds and loads to provide a flow dependent intake manifold temperature that is typical in actual vehicle operation. The turbocharger in the model was kept the same as in the baseline engine without modifications to improve the match given the hardware changes from the baseline.

Port water injection was modelled using the InjPulseConn object with an imposed pulse width that was calculated based on the desired amount of water mass per injection event. The injection timing was then synchronized to the crankshaft by a calibratable offset based on the intake valve opening event. Using the incompressible liquid fluid template set to fully vaporized, the model handled the enthalpy of the ensuing mixture using GT-Power’s built-in thermodynamic properties.

As a final step, the engine model was calibrated to produce results that match the general trends in similar type of engines as well as the dynamometer test results from an actual prototype engine configured exactly as in the model. Instead of trying to achieve a tight match to the real engine values at every operating point, it was decided to preserve the predictive capability of the model by focusing the matching to four key operating points and allowing larger deviations in other operating conditions. Each operating point is defined by an engine speed and BMEP pair and further constrained by engine parameters complying with limits defined and shown in Table 3. The constraints ensure on one hand that a fair comparison of the simulation results engine with the default calibration. The results were produced by setting the model and test engine to the same actuator positions, or control settings, as described in Table 2. The reference cam phase values can be found in the companion paper (Part 2) in Figure 7. As can be seen in Figure 2, the model load and specific fuel consumption values closely match the observed actual engine test values.

DOE Test Design

After verifying that the simulation model gives sensible results and correlated well at the four key operating points for the baseline engine configuration, we laid out a DOE test to explore the proposed technology options at significant corners of typical engine operation. The operating points were selected to assess distinct areas of the engine map with different boundary conditions to take into account the impacts on both the fuel consumption and performance limits. Each operating point is defined by an engine speed and BMEP pair and further constrained by engine parameters complying with limits defined and shown in Table 3. The constraints ensure on one hand that a fair comparison of the simulation results

### Table 2: Engine test data used for model correlation and setting

<table>
<thead>
<tr>
<th>Engine RPM</th>
<th>BMEP (bar)</th>
<th>BSFC</th>
<th>Intake Cam Phase</th>
<th>Exhaust Cam Phase</th>
<th>Main Throttle % Open</th>
<th>Turbine WG</th>
<th>CASO</th>
<th>COV IMEP</th>
</tr>
</thead>
<tbody>
<tr>
<td>1500</td>
<td>4.0</td>
<td>100</td>
<td>2.1</td>
<td>20.5</td>
<td>11.9</td>
<td>100.0</td>
<td>8.4</td>
<td>0.9</td>
</tr>
<tr>
<td>1500</td>
<td>22.6</td>
<td>110.8</td>
<td>-25.0</td>
<td>5.9</td>
<td>99.9</td>
<td>6.7</td>
<td>34.6</td>
<td>1.2</td>
</tr>
<tr>
<td>2000</td>
<td>16.0</td>
<td>94</td>
<td>-21.0</td>
<td>1.2</td>
<td>99.9</td>
<td>10.6</td>
<td>28.1</td>
<td>1.1</td>
</tr>
<tr>
<td>5501</td>
<td>18.2</td>
<td>137.4</td>
<td>23.9</td>
<td>-18.0</td>
<td>99.9</td>
<td>19.7</td>
<td>28.5</td>
<td>1.2</td>
</tr>
</tbody>
</table>
can be delivered and on the other hand that a safe and robust engine operation is guaranteed in the actual test engine. Simulation results which did not comply with the constraints were eliminated in the downstream evaluation process.

A 1500 RPM, 4 bar BMEP point was selected to represent a low speed low load condition where a typical passenger car may spend a large portion of its time. At this point, the engine combustion stability is affected negatively by increasing amounts of dilution and the addition of external EGR could lead to NVH issues [13]. Therefore in order to avoid any degradation in vehicle drivability the COV of IMEP was held below a target value 3.0%. This value corresponded to a total in-cylinder burned residual gas fraction of 26% in our combustion system as correlated to actual engine calibration. The above correlation was found by matching an actual cam-sweep simulation system as correlated to actual engine calibration. The turbocharger also typically operates at the limit in this condition and the maximum turbine inlet pressure, turbine inlet temperature and compressor-turbine shaft wheel speed were kept below their critical values.

To assess comprehensively all the technology variants, a DOE plan has been developed. An overview of the test design is shown in Figure 3. The base engine configuration with CR 10:1, and 248 degrees duration intake cam and without external EGR was evaluated at 3 levels of external dilution, 5%, 9% and 13%. In order to capture the best combined effect of internal and external dilutions, the intake and exhaust valve timings were swept over their authority ranges, at the 3 discrete levels of EGR. The valve timing sweeps were conducted sequentially from the corner of minimal valve overlap, corresponding to minimum dilution, towards the corner of maximal valve overlap, or maximum dilution. Each sweep contains 198 evaluation points. Additionally, longer duration intake cams of 280 and 300 degrees were evaluated over the same range of conditions.

Finally, 5500 RPM and 17 bar BMEP was included to maintain the rated power of the base engine with the addition of fuel saving technologies. At this key point the boundary conditions under observation are the thermal limits of the engine hardware on the hot gas side. To protect the exhaust system components from thermal stress under high speed and high load conditions the base engine configuration uses fuel enrichment to reduce the exhaust gas temperature. Instead of fuel enrichment, we applied water injection for the same objective. The turbocharger also typically operates at the limit in this condition and the maximum turbine inlet pressure, turbine inlet temperature and compressor-turbine shaft wheel speed were kept below their critical values.

Table 3 shows 198 evaluation points. Additionally, longer duration intake cam of 280 and 300 degrees were evaluated for each one of the key operating points selected for the study. Figure 3 does not show the 9 additional configurations that were evaluated for water injection, which replaced EGR at the rated power point. The adopted test procedure enabled the identification of the most eligible engine configurations in

<table>
<thead>
<tr>
<th>Operating Point</th>
<th>Constraints</th>
</tr>
</thead>
<tbody>
<tr>
<td>1500 RPM, 4 bar</td>
<td>Exh O2 Content ≤ 1.5 %</td>
</tr>
<tr>
<td></td>
<td>Burnt Mass Fraction at Combustion Start ≤ 26 %</td>
</tr>
<tr>
<td>1500 RPM, 15 bar</td>
<td>O2 Content ≤ 1.5 %</td>
</tr>
<tr>
<td></td>
<td>CA50 ≤ 36° ATDC</td>
</tr>
<tr>
<td></td>
<td>Knock Integral ≤ 0.8</td>
</tr>
<tr>
<td>2000 RPM, 16 bar</td>
<td>O2 Content ≤ 1.5 %</td>
</tr>
<tr>
<td></td>
<td>CA50 ≤ 36° ATDC</td>
</tr>
<tr>
<td></td>
<td>Knock Integral ≤ 0.8</td>
</tr>
<tr>
<td>5500 RPM, 17 bar</td>
<td>O2 Content ≤ 1.5 %</td>
</tr>
<tr>
<td></td>
<td>CA50 ≤ 36° ATDC</td>
</tr>
<tr>
<td></td>
<td>Knock Integral ≤ 0.8</td>
</tr>
<tr>
<td></td>
<td>P_{in} Turbine ≤ 3 bar</td>
</tr>
<tr>
<td></td>
<td>T_{in} Turbine ≤ 900 °C</td>
</tr>
<tr>
<td></td>
<td>N_{turbine} ≤ 200,000 RPM</td>
</tr>
</tbody>
</table>
terms of fuel saving potential and illustrated the governing physical drivers of the engine efficiency gains as well the performance loss at the corners of the operational map.

Results

To handle the large dataset created by the extensive DOE evaluations, we designed a workflow to facilitate the evaluation process of the different engine configurations for a reliable and objective comparison. A main focus of the evaluation process was to be able to narrow down to a practicable operational space and provide a realistic picture of the impact on fuel consumption of the different engine configurations. After presenting global depictions of the potential BSFC benefits and possible steady-state performance we also present summary charts to simplify the relative rankings of the different technology components.

Low Speed Low Load Case: 1500 RPM 4 Bar BMEP

Figure 4 shows the contours of BSFC for the 1500 RPM 4 bar BMEP case for the base engine configuration with no new components, as the IVO and EVC timings are swept, over their allowable phasing authority range. The colored range is the region where the engine satisfies the imposed operational limits as shown in Table 3. The blank space is the region where one or more constraints were not satisfied in the cam phasing map. It can be seen that BSFC is largely determined by the amount of valve overlap in the baseline results. A calibration with some safety margin could place the valve timings at the blue square mark, inside the yellow band, which could achieve a baseline BSFC which we adopt as 100 as reference scale in percentage in the rest of the paper.

Figure 5 is a matrix of plots where Figure 4 is repeated for each configuration as the CR and intake valve durations are changed while the EGR, IVO and EVC are swept. The top left sub-plot is a replica of Figure 4 for reference. Similar depiction is used in the rest of the paper.

A summary chart of the fuel consumption reductions with each technology component is compiled in Figure 6. The sizes of the bubbles are scaled to the relative efficiency gains, and it can be seen here that each component contributes to the improvement, while the relative benefits vary. In order to break down the relative contributions, we show one possible path in Figure 7 where CR is increased first, and then a longer duration intake cam is added, and then EGR is finally added, for a total benefit of 7.2% in BSFC. Other orderings are possible to reach this final benefit and could vary by component availability and architectural plans for an engine family.
The 1500 RPM, 4 bar BMEP point always had MBT timing for CA50, or optimal placement of the combustion center, and this was because the combustion is occurring at a low pressure and temperature. The observed variance in BSFC is primarily caused by the pumping loop losses as the valve timings and engine configurations are varied. The chart in Figure 8 captures the values for PMEP. The best BSFC configuration corresponds to the maximum or most positive value of ~7 kPa, in the top right corner, which was achieved by an aggressive placement of the operating point, illustrated by the blue square in the bottom right Figure 5 subplot. This suggests that de-throttling measures like cylinder de-activation (CDA) are effective measures for fuel efficiency at low loads. We do not include in this paper but the companion test paper [12, 14] show the benefits of CDA.

All of the technologies evaluated in this paper, except for water injection, provide significant fuel economy advantages at light and medium loads but the high end-of-compression pressure-temperatures caused by increased CR can lead to higher knock propensity at increased loads. To understand the tradeoffs, the 1500 RPM 22 bar BMEP point was simulated and the results are analyzed with the same workflow.

Low Speed High Load Case: 1500 RPM WOT

Figure 9 shows the BMEP contour plot of the base engine configuration. As the focus at this point was to maximize the low-end-torque, the contour and the bubble charts are showing engine BMEP rather than BSFC as in the previous key point. The baseline engine configuration has a narrow band of darkest red where it achieves 22 bar BMEP. The maximum overlap region is truncated because the simulation does not satisfy either the O2 constraint or the combustion constraint for knock, and shown blank in the figure. At most valve timing combinations, the engine makes less than the target output because either the cylinder cannot hold the required amount of fresh air, or the combustion becomes very retarded.

The trends for low-end-torque for all the other configurations are compared in Figure 10. Only four configurations are able to achieve the target load. As EGR rate increases, more design points are able to satisfy the constraints but the maximum achievable load decreases. Conversely, as CR is increased fewer points satisfy the constraints with the maximum achievable loads decreasing, as well. The switch to longer intake valve opening durations follows a similar trend to EGR rate and results in a decrease in the maximum achievable low-end-torque.

We summarize the maximum load achievable for each configuration in the bubble chart in Figure 11. It can be seen that only three other configurations besides the base configuration can reach 22 bar BMEP. The contour grids in Figure 10 illustrate a gradual shift of the valve timings towards the lower left in the coordinate system, which corresponds to an advanced EVC and advanced IVO, in order to achieve maximum torque output. All other configurations do not reach the target BMEP with a downward trend with higher EGR rate, higher CR and longer intake valve opening durations. The percentage drops for each configuration are shown in Figure 12.

One reason the target load is not achieved is the lack of enough fresh air in the cylinder. This can be seen by the summary bubble chart in Figure 13 where it shows how an increasing portion of the in-cylinder mass is inert in the configurations achieving lower BMEP. Increased burned gas residual fractions help to reduce knock propensity but hinders the amount of fresh mixture in the cylinder needed to achieve...
load. The reduction in fresh mixture is displayed by the trapped volumetric efficiency in Figure 14. This suggests improved turbocharger efficiency is a key requirement for high performance when components that improve fuel efficiency are introduced into the engine system.

**Mid Speed High Load Case:**

**2000 RPM 16 Bar BMEP**

The 2000 RPM 16 bar BMEP point is a case where some of the factors in the previous two key points occur simultaneously. We show the BSFC variation at this point in Figure 15 as the IVO and EVC are swept for the baseline case. There are two regions of low BSFC in this space so a choice has to be made on which region to operate in. For minimizing the BSFC, the minimum overlap corner is the best choice, and it is marked by a green circle. The other region colored dark green and toward the maximum overlap corner has higher BSFC and it could be preferred if smaller valve timing adjustment is desired when moving in the low speed high load corner of the engine map.
For the other engine configurations, Figure 16 shows the contours of achievable BSFC at 2000 RPM 16 bar BMEP. The top left corners in each plot are blank due to \( \text{O}_2 \) constraint violations when valve overlap leads to too much scavenging. At retarded IVO values, the cylinder is not holding enough fresh mixture at IVC to support the high load. It can be also seen that applying more EGR, higher CR or longer intake valve duration cam in general reduces the valid operating range of the engine. In all configurations, knock is a limiting factor and ignition timings were adjusted by a dedicated controller to the same borderline knock value. A small green circle is placed at the locations where the engine could operate for best efficiency within the combustion constrains.

Figure 17 summarizes the comparative BSFC benefits for each engine configuration. In this key point, a major factor in the BSFC result is the combustion phasing and thus we show the CA50 values in Figure 18. The general trend is that a more advanced CA50 is possible when dilution level is increased. The trend is then confounded when the thermal effects of large amounts of burned gas residuals create a knock risk that calls for a retardation of the ignition timing. The bubble chart in Figure 17 shows that increasing the CR decreases the BSFC benefit in almost every case, while a longer duration intake cam improves the BSFC. Note also that with longer duration cams the valid operating range narrows (Figure 16). While multiple factors interrelate to create a complex landscape, a maximum BSFC benefit of 5.8% can be achieved in 5 configurations compared to the baseline.

When running in the boosted region with a large valve overlap, the turbine energy levels are increased and the pressure balance across the engine tilts towards a higher intake pressure. This effect is illustrated in Figure 19 where the pump loop work sign is reversed from the usual. It means the turbocharger is adding positive work back to the piston by efficient
waste heat recovery. This phenomenon is another factor that makes the optimization quite complex in this key point, where the nonlinear natures of the turbocharger and engine flow interact.

One possible path to achieving a 5.8% BSFC benefit is shown in Figure 20. In this case, adopting a 280° duration for the intake cam has the advantage of moderate timing shifts while keeping enough valid operating range for safe engine operation.

High Speed High Load Case: 5500 RPM 17 Bar BMEP

The final key point in our study is the 5500 RPM 17 bar BMEP case which is important because it represents the rated power of the engine. In Figure 21 we show the contours of BSFC for the baseline configuration that achieves target power, while the blank regions represent where the configuration does not satisfy boundary conditions. Fuel enrichment cooling is being used to protect hardware elements in the exhaust system because the temperatures are elevated.

At this key point, the maximum EGR rate that could make target load was 5% for the base CR and intake duration. All other configurations could not comply with the limits. Therefore, we ran only the 0% and 5% EGR rate conditions and replaced the 9% and 13% EGR rates with water injection at the ports of each cylinder during the valve overlap event. The amount of water injected was 20% of the fuel mass per event. Figure 22 is the matrix of plot showing the results with a small green circle for the feasible configurations that reach the targeted engine power and red × where no feasible configurations could be achieved. The dark blue slices mean that with water injection, the target power can be achieved at a
very low BSFC, using a common scale from Figure 21. This is because water is a very effective cooling medium and in our simulation, 20% of the fuel mass was injected as water in each case to achieve unitary lambda levels (Figure 23). Besides the large BSFC reductions as seen in Figure 24, injecting water allowed a fairly advanced CA50 values which further help with the efficiency (Figure 25). This could be an important factor for RDE compliant engines in the near future that still meets the traditional vehicle propulsion power levels.

Conclusions

In this paper we have shown the possible gains in CO2 efficiency in a modern turbocharged engine when higher CRs, longer duration intake cams, cooled external EGR and water injection are combined. The largest CO2 benefit occurs at a low speed low load operating point where using the highest CR, with longest cam for largest internal EGR and with 13% external EGR rate conditions combine to provide up to 7.2% gain in fuel efficiency. The simulation results also indicate significant CO2 benefit, of up to 5.8%, is possible at a medium speed and high load (2000 RPM 16 bar BMEP) operating point where multiple performance limits come into play simultaneously, such as O2, risk of misfire and knock.

The improvements in efficiency are to be contrasted with the loss of top end performance. The peak torque achievable at low engine speed is shown to be reduced when higher CR, longer cams or increasing amounts of EGR are used. Moreover, at the rated power point, it is shown how the baseline configuration is the only one with an operation region sufficiently large for feasible calibration. One exception to this is water injection, which is shown to be very effective for rated power even with high CRs.

When efficiency and performance goals are considered together, the choice of technology options to incorporate becomes more limited and the size of the feasible space for operational reliability is an important factor. By selectively choosing a few operating points and simulating comprehensively all the tuning parameters in a modern combustion engine, we balanced the requirements for both efficiency and performance of the next CO2 improving technology options for the reliable and proven four-stroke Otto cycle engine.

References


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Acknowledgments
We acknowledge Daniel Jakiela, David Roth, Joseph Ciaravino and Roberto Rastelli for their assistance and input in the preparation of this work and paper.

Definitions/Abbreviations
ATDC - After Top Dead Center
BMEP - Brake Mean Effective Pressure
BSFC - Brake Specific Fuel Consumption
CA50 - Crank Angle 50 Percent Burned
CDA - Cylinder De-Activation
CO₂ - Carbon Dioxide
COV - Coefficient of Variation
CR - Compression Ratio
DOE - Design of Experiments
EIVC - Early Intake Valve Closure
EGR - Exhaust Gas Recirculation
EVC - Exhaust Valve Closure
gCR - Geometric Compression Ratio
GDI - Gasoline Direct Injection
IMEP - Indicated Mean Effective Pressure
InjPulseConn - Sequential Injector with Imposed Pulse Width
IVC - Intake Valve Closure
IVO - Intake Valve Opening
LIVC - Late Intake Valve Closure
LET - Low End Torque
LPL - Low Pressure Loop
MBT - Minimum Best Timing
NTurbin - Turbocharger Wheel Speed
NVH - Noise Vibration Hardness
P_in Turbine - Turbine Inlet Pressure
PMEP - Pumping Mean Effective Pressure
PWI - Port Water Injection
RDE - Real Driving Emissions
T_in Turbine - Turbine Inlet Temperature
VVT - Variable Valve Timing
VVL - Variable Valve Lift
VCR - Variable Compression Ratio
WG - Waste Gate
WLTP - Worldwide Light-duty Testing Procedure