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Optimizing engine efficiency by balancing dilution, heat release rate and combustion phasing

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Abstract

In this investigation highly diluted engine tests have been conducted to study effects of dilution, heat release rate and combustion phasing on both efficiency and emissions in the Partially Premixed Combustion regime. It has been reported that over the complete phasing range an increase of more than two percent points in indicated efficiency at high load could be expected, with an astonishing 4 to 5 percent points for the low load case. The origins of these increases were sought and found to be caused by improved heat release shapes, combustion efficiencies or heat losses or a combination of these. On the other hand, soot emissions are greatly reduced due to the excess oxygen available, but nitrogen oxide emissions are found to increase: both in concentration and power specific units.

Introduction

The new EURO VI emission legislation has forced automotive industry to come up with advanced after treatment solutions such as particulate filters and selective catalyst reduction systems. These systems, however, introduce a significant penalty in overall efficiency and consequently, fuel consumption. Advanced combustion concepts, e.g. Homogenous Charge Compression Ignition (HCCI) and Partially Premixed Combustion (PPC), have been introduced to reduce engine out emissions and decrease the need for after treatment. Several research groups [1-4] found low emission and high efficiency results at various load points using PPC like circumstances. Lund University [5] showed that, if the proper fuel reactivity is selected, the complete load range could be covered. This important observation has led to further research on maximizing efficiency using those fuels. It has been found that increased gas densities could improve efficiency via the mechanism of reduced heat loss through both the walls and exhaust [6, 7]. All in all, together with large amounts of recirculated exhaust gas (EGR), both emissions and efficiencies are greatly affected with respect to conventional diesel combustion.

Specific objectives

High dilution, defined in terms of the so-called gas-fuel-ratio (GF; see Equation (1)), was shown to improve efficiency.

\[ GF = \frac{m_{\text{air}} + m_{\text{EGR}}}{m_{\text{fuel}}} \]  

Since air requirement depends on fuel composition, the authors have decided to use the gas excess ratio, denoted with \( \Lambda \), which is analogous to the air excess ratio (see Equation (2)).

\[ \Lambda = \frac{GF}{AFR_{\text{stoich}}} \]  

Nowadays, production engines run with retarded combustion phasing as a compromise between wall and exhaust heat losses. For such highly-diluted premixed combustion, this trade-off is affected. Because of the lowered heat losses in highly diluted premixed combustion, it is argued that combustion may start at top dead center (TDC), or even before that. As less heat is lost and as close to TDC the negative work is negligible, the effective expansion ratio can benefit from early combustion phasing. Secondly, relatively more air determines physical properties, such as heat capacity. Consequently, the ratio of specific heats (\( \kappa \)) will not change that significantly after combustion. This implies a more effective expansion cycle using massive dilution, independent of fuel choice. Experiments have been conducted to study the exact position of the optimal heat release and the effect of dilution levels. The purpose of this investigation is to determine the trade-off between dilution, combustion phasing and heat release shape on efficiency.

Experimental apparatus

Measurements are conducted on a dedicated test rig, based on a heavy duty DAF XE355c engine [7, 8]. Only cylinder 1 of this engine will be used for combustion concept and fuel testing. Cylinders 2 and 3 function as EGR pumps whereas the remaining three operate under the stock engine control unit together with a Schenck W450 engine brake. Figure 1 and Table 1 provide an overview of engine and test bed geometry.
Table 1: Engine specifications

<table>
<thead>
<tr>
<th>Base Engine</th>
<th>DAF XE355c HDDI</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore [mm]</td>
<td>130</td>
</tr>
<tr>
<td>Stroke [mm]</td>
<td>158</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>15.7 (original 17.0)</td>
</tr>
</tbody>
</table>

Fuel selection and measurement matrix

During the investigations, the engine is solely run on Primary Reference Fuel 70 (PRF70: 70% iso-octane and 30% n-heptane). This fuel has been selected for its reactivity and ease of blending. A lubricity additive (Infineum R655) has been added to ensure sufficient lubrication. Furthermore, PRF70 can easily be used for modeling purposes and additional optical engine tests.

Prior to the measurements, a target measurement matrix is constructed with three main variables. First of all, the midpoint of combustion (CA50) is varied from −6 °CA to 16 °CA at two different dilution levels, or GF ratios. Subsequently, the heat release shape will be altered by changing the fuel pressure according to the combinations shown in Table 2.

Table 2: Heat release shape combinations; the numbers represent fuel pressure setpoints in bar

<table>
<thead>
<tr>
<th>Low dilution</th>
<th>High dilution</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low load</td>
<td>600</td>
</tr>
<tr>
<td>High load</td>
<td>1200</td>
</tr>
</tbody>
</table>

All experiments are done at (1250±5) rpm, a typical engine speed for highway cruising of a heavy-duty transport vehicle. Furthermore, an EGR flow of (50±5) wt-% is used and the pressure difference between exhaust and inlet is kept at a constant (0.3±0.1) bar. Although this pressure difference is set ‘arbitrarily’, the exhaust enthalpy might be insufficient to drive currently available commercial turbochargers.

A couple of words can be spent on the accuracy of both gas excess ratio and load. First of all, the gas excess ratio is known to change as certain, hardly controllable, working pressures increase or decrease with changing combustion phasing. Secondly, load will be specified by a target load. However, significant deviations of this target load are measured as a result of changing efficiency.

Results and discussion

The main goal of this investigation is to find the trade-offs between dilution, heat release rate and combustion phasing. Although an efficiency increase is expected, sources of loss are compared, i.e. exhaust- and wall heat losses, and incomplete combustion phenomena. First of all, the efficiency results will be presented after which a detailed analysis will be done.

In Figure 2 and 3 the resulting gross indicated efficiencies are plotted. Note that highly diluted conditions are depicted by open markers. Furthermore, the steep rise when advancing combustion for the low fuel pressure, highly diluted CA50 sweep is caused by an additional increase in dilution (from λ ~ 4 to 4.5). It is clear that the shape of the efficiency – phasing trade off is affected by varying dilution. It appears that at high load and high dilution maximum efficiency has not been obtained as a consequence of hardware limitations (peak firing pressure). That is why combustion could not be advanced any further.

Combustion effects and engine out emissions can be studied in cylinder 1 as soon as the engine has reached its operating temperature. Therefore, cylinder 1 can be operated fully autonomously apart from the mutual crank- and camshaft, oil- and coolant systems.

Intake air can be boosted by an external compressor. By using the aforementioned EGR pumps (see Figure 1) an EGR flow up to 70 wt-% is possible, which is cooled both up- and downstream of the pumping cylinders.

Furthermore, fuel pressure is provided by the Resato HPU200-625-2 pump, which can deliver pressures up to 4200 bar. The common rail, as depicted in Figure 1, is mimicked by a fuel accumulator close to the injector (~0.2 m). Currently, a Delphi prototype injector, having 8 holes of 151 µm and an umbrella angle of 153 degrees, is used. All relevant flows, like for fresh air, EGR and fuel supply, are measured with Coriolis mass flow meters.

Other parameters (fuel pressure, injector current and in-cylinder pressure) are recorded on a crank angle resolved basis (0.1 °CA). The high speed data acquisition is done by a SMETEC Combi system, which records and processes cylinder pressure (uncooled AVL GU12c pressure sensor).

Finally, sophisticated Horiba emission analysis equipment is used to measure the exhaust concentrations of unburnt species, nitrogen oxides, oxygen and carbon monoxide. Furthermore, this system (MEKA 7100 DEGR) is capable of calculating the EGR percentage by measuring the CO₂ concentration in the intake runner, however this has not been applied yet. Particulate matter emissions are determined via the filter paper method by an AVL 415S smoke meter.

Figure 1: Overview of test bed
efficiency with higher injection pressures is expected to be better visible in the heat release shapes. The larger the first, premixed peak is, the better the thermodynamic cycle will be. Figure 4 tries to explain this effect by comparing the heat release rates for the low load case at CA50 of (10±1) °CA. A small window, first order Savitzly-Golay filter is applied to smoothen the traces.

As can be seen, the high gas densities influence the rate of combustion. A transition to mixing controlled combustion is observed at high dilution levels, which is expected to be caused by the reduced spray penetration as a consequence of the reduced discharge pressure described by Siebers [9].

Due to the intrinsic longer injection durations for lower fuel pressures, the ignition dwell or mixing period (CA10 – EOI) is significantly reduced as is observed in Figure 5. Furthermore, as the appropriate circumstances for ignition are met in an earlier stage for highly diluted conditions, ignition dwell decreases with dilution.

Figure 2: Gross indicated efficiency at 8 bar target IMEP

Figure 3: Gross indicated efficiency at 16 bar target IMEP

Both ignition delay (Figure 4) and burn duration decrease simultaneously with increased gas densities. This phenomenon is thought to be caused by the increased probability of collisions. A reduced burn duration could also be beneficial for increased thermodynamic efficiency.

However, due to the intrinsic method of calculating burn duration (CA90 – CA10), the effect of increased efficiency with higher injection pressures is expected to be better visible in the heat release shapes. The larger the first, premixed peak is, the better the thermodynamic cycle will be. Figure 4 tries to explain this effect by comparing the heat release rates for the low load case at CA50 of (10±1) °CA. A small window, first order Savitzly-Golay filter is applied to smoothen the traces.

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Figure 4: Ignition delays for the low load case

Figure 6: Combustion mode shift from premixed to mixing controlled with increased dilution
Subsequently, an attempt is made to quantify all loss terms by comparing all efficiencies. Note that from Figure 4 onwards only the lower load case will be evaluated for brevity since the mechanisms are expected to be equal.

Since some (minor) part of the energy will be lost by means of incomplete combustion, the combustion efficiency is shown in Figure 7. The extra amount of available fresh air is beneficial for the outburn of HC and CO emissions. Via a second mechanism (wall impingement protection by high gas densities) flame quenching is inhibited. Independent of fuel pressure combustion is rather complete for the highly diluted case. The more conventional points in terms of dilution and fuel pressure, however, show a specific trade-off with respect to combustion phasing. It is argued that for lower injection pressures the injection event should preferably be phased close to top dead center to improve vaporization. Only for extremely late timings a significant reduction in combustion efficiency is observed since in-cylinder temperatures have dropped tremendously. On the other hand, high fuel pressures help atomization and therefore, no efficiency losses for early timings are reported here. Nevertheless, the decrease in combustion completeness is more pronounced for retarded injection events, which is attributed to improper fuel spray-piston alignment. Apparently, the current combination of a DAF XE390c piston with a relative narrow cone angle nozzle is not optimal for conventional timings.

Consequently, exhaust losses are studied. In order to do so, the enthalpy in the exhaust gas is calculated. For ease of calculations, the specific heat at constant pressure is based on a constant exhaust gas composition of 10 vol-% CO₂, 5 vol-% O₂ and the remaining part N₂ and H₂O. Equation (3) represents the exhaust enthalpy calculation.

\[
\hat{H} = \dot{m}_{\text{exhaust}} \cdot c_p \left( T_{\text{exhaust}} \right) \cdot \left( T_{\text{exhaust}} - T_{\text{ambient}} \right)
\]

Furthermore, since no direct coolant losses can be measured, wall heat losses are estimated via the subtraction of all known loss terms.

![Figure 7: Combustion efficiency for the lower load case](image)

![Figure 8: Exhaust losses](image)

![Figure 9: Estimated wall heat losses](image)

In summary, it has been shown that losses and consequently fuel consumption can be lowered with this newly presented concept. Another question arises: how will emissions be affected by the increased gas densities?

As a consequence of the massive dilution, fuel sprays do not penetrate as far as with ‘conventional’ gas densities [9]. Especially with less reactive fuels, this is expected to be of particular importance for obtaining complete combustion. It is another reason why excess air is helpful in exploiting all fuel’s potential.

When the unburnt exhaust species are investigated, the reduction in CO is obvious (see Figure 10). Hydrocarbon emissions are found to be relatively independent on gas fuel ratio; some reduction, however, is obtained. The authors hypothesize that CO emissions are reduced via the reduced probability of flame quenching near the cylinder wall. Hydrocarbon
emissions, on the other hand, are not that greatly affected, since spray-piston alignment does not change significantly.

Particulate matter emissions increase with the changed heat release shape, as had already been observed in another paper by the authors [10]. The locally richer areas, however, seem to be of minor influence on soot emissions. On the contrary, the main factor reducing particulates is found to be the air excess ratio and not heat release shape. It can be concluded that the premixed fraction method, which correlates the premixed peak with smoke emissions, can only be used with more or less comparable conditions and not with largely varying oxygen content.

Figure 12 clearly depicts the EURO VI compliant combustion in terms of soot for all high fuel pressure points and part of the highly diluted low fuel pressure points. The more conventional settings, i.e. low fuel pressure and dilution, are not even EURO I compliant (0.4 g/kWh).

Conclusions

In this investigation it was shown that high dilution is the key in approaching the theoretical thermodynamic efficiency. First of all, as has clearly been shown, gross indicated efficiencies increase with increased dilution. Apparently, the high gas densities affect the insulation layer near the cylinder wall, which greatly reduce wall heat loss. An increase of 4 to 5 percent points is observed throughout almost the complete combustion phasing range for the lower load case. On the other hand, the higher load case is intrinsically more efficient and that is why the efficiency benefit is limited to 2 percent points. On top of that, combustion could not be phased before 0 – 2 °CA aTDC as a consequence of hardware limitations (either peak firing pressure or maximum pressure rise rates). Furthermore, high loads are air limited and that leaves less margin for improvement by dilution. The authors believe that this newly presented concept could contribute to elevating low end efficiencies.

A disadvantage is, unfortunately, the introduction of oscillations in the combustion chamber which might break the thermal insulation layer. It has also been observed that combustion changes to a mixing controlled type, which is mainly due to a shorter ignition delay.
Although the results from this investigation show potential, it will demand for sophisticated turbochargers. A lot of energy has to be extracted from the exhaust enthalpy. Fortunately, an unexpected benefit was found in the shift of heat loss to the exhaust. This concludes the double acting principle of highly diluted combustion concepts: the reduction in wall heat loss contributes both in overall efficiency and available exhaust enthalpy increase.

On the emission side massive dilution was found to lower both HC, CO and soot emissions. Nitrogen oxide emissions (NOx), on the other hand, have increased due to the confinement of the sprays in the center of the combustion chamber. These globally richer areas elevate NOx levels as a consequence of higher flame temperatures. The authors propose to study the quality of the dilution, i.e. the composition of the dilution used (varying EGR rates).

Future work will mainly consist of constructing a denser efficiency-load-combustion phasing trade-off, with special attention for the appropriate dilution strategy in terms of O2 concentration and intake pressure. Moreover, similar experiments could be conducted with conventional diesel (ENS90) to check direct implementation in nowadays engines. Followed by advanced optical experiments, for example fuel tracer laser induced fluorescence; the authors will hopefully manage to fully understand the underlying mechanisms of efficiency increase. As a final step, all these results will be used as validation for upcoming modeling efforts.

Acknowledgements
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