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Experimental investigation of low octave fuel composition effects on load range capacity in Partially Premixed Combustion (PPC)

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Abstract
Manente and coworkers [1] discovered that a fuel in the gasoline boiling range with 70 Octane Number (ON70) is applicable for Partially Premixed Combustion (PPC) over the complete load range without additional engine adaptations. Using such a fuel in combination with an appropriate dilution strategy was shown to simultaneously achieve low emissions and high efficiency without any drawback in combustion control. In the present paper the load range applicability of ON70 fuels in PPC is further investigated. To do so, four fuels in the ON70 range are selected, with different chemical and physical properties. The goal of testing these fuels is to achieve low emissions and high efficiency over the complete load range of PPC.

Introduction
More stringent emission regulations all over the world force researchers to develop new technologies to reduce the emissions of combustion engines. One way is the development of new engine after treatment systems. However, these systems are expensive and reduce fuel economy. Therefore, current research opts for another path; a new combustion concept in combination with the selection of an optimal fuel to meet current emissions regulations.

In Homogeneous Charge Compression Ignition (HCCI) combustion, fuel is injected earlier in the cycle than in conventional Compressions Ignition (CI) engines, in order to obtain enlongated mixing time for reduced soot emissions. Besides that, Exhaust Gas Recirculation (EGR) is used to suppress combustion temperature, which in its turn reduces Nitrogen Oxides (NOx) emissions. However, HCCI suffers from control and Pressure Rise Rate (PRR) problems. Therefore, the operational load window of HCCI remains limited even when a proper control strategy is developed [2].

Next, to overcome the limitations of HCCI, several other combustion concepts are introduced, one of which is Partially Premixed Combustion (PPC). In essence, this is a combination of conventional CI and HCCI combustion. To increase the mixing time compared to conventional combustion, both concepts use an early injection strategy. However, in PPC the injection is not phased as early as in HCCI combustion. As a result, the mixture in the cylinder is (mildly) stratified and not completely homogeneous like in HCCI.

Kalghatgi [3, 4] obtained low soot and NOx emissions simultaneously with high efficiency using gasoline in combination with PPC combustion. After optimizing the dilution strategy (air and EGR) and testing a variety of fuels in the gasoline octane number range, Manente and coworkers [5, 6] obtained high thermal efficiencies up to 56%. The authors were able to achieve these efficiencies simultaneously with low NOx, Carbon Monoxide (CO) and Unburned Hydro Carbon (UHC) emissions using multiple injection gasoline PPC. Furthermore, the PRRs were acceptable and soot emissions were significantly reduced.

Finally, Manente [1] performed a series of tests using fuels with octane numbers in the range from diesel to gasoline to check the load range capacity of fuels in PPC combustion. Fuels with low Research Octane Number (RON) performed stable throughout the complete load range of PPC. However, it resulted in the high soot and NOx emissions known from conventional CI combustion. Fuels in the RON 95 range performed well in emission reduction; unfortunately they were only applicable for higher loads. It was shown that a fuel in the boiling range of gasoline with an octane number in the range of 70 is the best candidate to be able to run the complete operational load window in PPC mode. Combining this fuel with a proper dilution strategy, it is possible to obtain high thermal efficiency over the complete load range in PPC operation.

Specific Objectives
As mentioned before, Manente discovered that fuels in the RON range of 70 are best applicable over the full load range of PPC. This will be investigated in more detail in this research. Four fuels were selected that all have RON 70, but different boiling range, aromatic and bio-derived content. All with the aim to obtain low emissions and high efficiency, which were reported in earlier PPC research. The lower load limit is generally restricted by combustion efficiency and stability targets, whereas the maximum load capability is confined by excessive soot formation and engine hardware limitations. Furthermore, NOx emissions should be acceptable over the complete load range to prove applicability of the fuel in PPC mode.

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Experimental Setup

The test-setup used, referred to as CYCLOPS, utilizes an in-line 12.6 liter six cylinder DAF XE355c engine. The cylinder bore and stroke are 130 and 158 mm respectively. Furthermore, compression ratio is lowered from 15.7:1 to 17.0:1.

Cylinder 1, or the so-called test cylinder, is used for combustion concept testing and can operate autonomously from the other cylinders, apart from the lubrication system, cam- and crank shaft. Cylinders 2 and 3 are used to pump EGR from the exhaust of the test cylinder, through a cooler, back to the intake manifold. The cooled exhaust gas is mixed with fresh air, supplied by an air compressor, before it is fed back to the intake of cylinder 1. The remaining cylinders are used to propel the engine, even when the test cylinder is not fired. These propelling cylinders are controlled by the stock DAF electronic control unit (ECU) and regulate the engine speed in combination with a Schenk W450 dynamometer.

The test cylinder uses a standalone injection system which consists of a Resato HPU 200-625-2 high pressure fuel pump, common rail and prototype Delphi injector. The high pressure fuel pump is of the double-acting air-driven type and can deliver fuel pressures up to 4200 bar. The injector has 8 holes of 151 µm with a spray angle of 153 degrees; and it is suitable for fuel pressures up to 2500 bar. The mass flow of fuel in this system is measured using a MicroMotion Coriolis mass flow meter.

Emissions of the test cylinder are measured using a Horiba Mexa 7100 DEGR exhaust analysis system. Exhaust gas concentrations of O₂, CO₂, NOₓ, UHC and CO are measured online. Soot emissions (in Filter Smoke Number) are monitored using an AVL 415S smoke meter.

Two data acquisition systems are used; fast changing parameters are measured using a SMETEC Combi crank angle resolved system. This system records the in-cylinder pressure (via an AVL GU12c uncooled transducer), common rail fuel pressure, intake air pressure and injector current. Secondly, slow changing parameters, like inlet air and exhaust gas temperature, are recorded via an in-house developed data acquisition system called TUeDACS.

Fuel Characteristics

For this research four fuels are tested, their specifications are presented in Table 1. All fuels have RON in the range of 70, with the following nomenclature:

- G (Gasoline): Refinery stream with gasoline boiling range
- D (Diesel): Refinery stream with diesel boiling range
- DLA (Diesel Low aromatics): Diesel boiling range with no aromatic content
- DBC (Diesel Bio-Content): Diesel boiling range with 30 vol% Fatty Acid Methyl Esters (FAME)

### Table 1: Overview of fuel specifications: C, H/C and O/C represent the average carbon number, Hydrogen/Carbon- and Oxygen/Carbon atomic ratio, respectively. Furthermore, Lower Heating Value (LHV), stoichiometric Air/Fuel (A/F) ratio and boiling range are given for all fuels.

<table>
<thead>
<tr>
<th></th>
<th>RON</th>
<th>C</th>
<th>H/C</th>
<th>O/C</th>
<th>LHV [MJ/kg]</th>
<th>A/F</th>
<th>Boiling Range [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>G</td>
<td>70.1</td>
<td>9.69</td>
<td>2.19</td>
<td>0.00</td>
<td>44.82</td>
<td>15.01</td>
<td>40-184</td>
</tr>
<tr>
<td>D</td>
<td>71.1</td>
<td>9.71</td>
<td>1.66</td>
<td>0.00</td>
<td>41.18</td>
<td>14.63</td>
<td>179-334</td>
</tr>
<tr>
<td>DLA</td>
<td>69.7</td>
<td>10.50</td>
<td>2.19</td>
<td>0.00</td>
<td>44.39</td>
<td>15.00</td>
<td>187-213</td>
</tr>
<tr>
<td>DBC</td>
<td>69.5</td>
<td>12.06</td>
<td>2.10</td>
<td>0.04</td>
<td>41.75</td>
<td>14.20</td>
<td>187-366</td>
</tr>
</tbody>
</table>

Measurement Matrix

Five load points will be investigated at different combustion phasings (CA50), taking the hardware limitations into account.

- **Load (±0.3):** 4, 8, 12, 16 and 20 bar IMEPg
- **CA50 (±0.3):** 0, 2.5, 5, 7.5 and 10 °CA aTDC. CA50 is varied to obtain maximum gross indicated efficiency at each load point.

At each of these operation points the following conditions are set:

- **Engine speed:** (1250±5) RPM
- **Fuel pressure:** 600 bar for the lowest load point of 4 bar IMEP. For the subsequent load points the fuel pressure will be set equal to 1000, 1400, 1800 and 2200 bar respectively.
- **Air excess ratio (λ):** (1.5±0.1) except for the 4 bar load point, here inlet pressure will be set to atmospheric pressure (1 bar) since a realistic turbocharger can not give a significant boost pressure at low load.
- **EGR mass flow:** as guideline (50±5) wt-% EGR is used. However, this might not be satisfactory at low loads due to stability problems. The minimum amount of EGR at these points is set by combustion stability and the EURO VI NOₓ emission levels of 0.4 g/kWh [7].

The limits of ISHC (Indicated Specific HydroCarbon) emissions at low load and ISPM (Indicated Specific Particulate Matter) emissions at high load are set by EURO VI emissions legislations [7]. This emission legislation limits ISHC emissions at low load to a maximum of 0.13 g/kWh. At high load ISPM is the limiting factor, which is set equal to 0.01 g/kWh.

Data Analysis and Definitions

When comparing experimental results of engines, it is common to determine load with respect to the power measured at the crankshaft, i.e. brake power. However, due to design of the CYCLOPS it is impossible to measure the power of the test cylinder at its crankshaft. Therefore, all values are given with respect to the gross indicated mean effective pressure (IMEPg).
The crank angle position where 10 percent of the fuels energy is converted is referred to as CA10. In a similar way, CA50 and CA90 can be defined. CA10 and CA90 are defined as Start of Combustion (SOC) and End of Combustion (EOC), respectively. Analysis of injector current data showed that at 1250 RPM there is a constant delay of 4 °CA between Start of Actuation (SOA) of the injector and Start of Injection (SOI). Injection duration is set during operation, adding up the injection duration to SOI, End of Injection (EOI) can be found. Using these definitions other combustion parameters are defined:

- Start of injection: SOA+4 [°CA aTDC]
- Burn duration (BD): EOC-SOC [°CA]
- Ignition delay (ID): SOC-SOI [°CA]
- Mixing Period (MP): SOC-EOI [°CA]

Furthermore, specific emissions are calculated from their specific concentrations in the exhaust gas flow by using their molar weights. NOx is treated as NO2, in accordance to the European legislations. PM emissions are computed via an empirical relation by Christian [8].

Results and Discussions

At the 4 bar IMEP load points it was not always possible to meet the requirements for λ and EGR wt-%, due to combustion stability problems. Furthermore, the desired fuel pressure at the 20 bar IMEP load points was not feasible due to the low viscosity of the fuels which imposed high injector leakage. The settings of these control parameters at 4 and 20 bar IMEP can be found in Table 2.

<table>
<thead>
<tr>
<th>Table 2: Exceptions from measurement matrix</th>
</tr>
</thead>
<tbody>
<tr>
<td>G</td>
</tr>
<tr>
<td>D</td>
</tr>
<tr>
<td>DLA</td>
</tr>
<tr>
<td>DBC</td>
</tr>
</tbody>
</table>

Efficiency

At every load point, combustion phasing (CA50) is varied to obtain the combustion phasing with the highest gross indicated efficiency (ηgross), see Figure 1. As can be seen, up to 16 bar IMEP the general trend is an increase in ηgross as load is increased. This effect is expected due to the increase in BD, which reduces the heat transfer losses to the cylinder walls. Nevertheless, ηgross starts to decrease when load is increased even further, to 20 bar IMEP. At high load, it was not possible to phase CA50 earlier than 7.5 °CA aTDC due to hardware limitations of the set-up; maximum incylinder pressure was reached. As a result, in-cylinder temperature was lower and more heat was lost to the exhaust stroke.

Fuel D shows the highest gross indicated efficiency over the complete load range. The maximum obtained is 0.51 at 16 bar IMEP. For this fuel also the shortest BD (see Figure 2) and largest ID (see Figure 3) are reported, which imply premixed combustion. This combustion regime is preferable to obtain high gross indicated efficiency, since heat loss is then minimized. These observations in BD and ID can be certified due to the high aromatic content of this fuel. Aromatics are stable chemical structures which require a high energy to break down; therefore ID is shown to be high. However, when the mixture ignites, all fuel molecules react simultaneously which results in short BD.

Fuel DBC contains long carbon chains (C = 12.06) compared to fuel D (C = 9.71) and no aromatics. Therefore, ID and BD are worse compared to fuel D, since long alkanes are more reactive than aromatics. So, ηgross is lower for fuel DBC. Furthermore, oxygenate content does not seem to have an influence on ηgross.

Like in DBC, only alkanes can be found in fuels G and DLA. Therefore, BD is longer and as a consequence ηgross is lower. Fuel DLA shows the lowest gross indicated efficiency for IMEP ≥ 8 bar. An explanation for this observation is not found yet, possibly the relative long chain length (C = 10.50) in combination with the absence of aromatics play a roll.
Unfortunately, none of these solutions will be practical. On the other hand, fuel DLA and DBC meet the EURO VI ISHC legislation at IMEP ≥ 12 bar. These fuels also report the longest ID. Long BD indicates that mixture-controlled combustion takes place, the peak temperature of this combustion type is considered to be higher than for premixed combustion. High combustion temperatures are beneficial for ISHC (and ISCO) emission reduction. Only fuel DLA and DBC meet the EURO VI legislation at IMEP ≤ 8 bar. In the near future an inlet air heater would be beneficial. In the near future an inlet air heater will be installed to reduce ISHC and ISCO emissions for IMEP ≤ 8 bar.

Figure 3: Ignition delay (ID) versus IMEPg

It is well known that combustion efficiency (η_comb) increases when load increases, see Figure 4. At higher loads more fuel is injected, as a result the global in-cylinder temperature is higher, which reduces ISHC and Indicated Specific CO or ISCO emissions (not shown in this paper). The trends in η_comb are inversely proportional to ISHC emissions, which will be discussed more extensively later on.

Figure 4: η_comb versus IMEPg

Emissions

As mentioned before, lower and higher load limit are respectively set by excessive ISHC and ISPM emissions. To investigate the load range capacity of the fuels these emissions are depicted in Figure 5 and 6.

At low loads (< 8 bar IMEP) ISHC emissions are high due to low global in-cylinder temperature, which imposes long ID. This gives fuel sufficient time to get in crevices of the engine. Figure 3 shows that the fuel with the shortest ID at low load, fuel DBC, gives the lowest ISHC emissions. Besides, it was expected that fuels with a high boiling range may suffer from ignition problems because of low global in-cylinder temperatures imposed by increased heat losses at low load. Fuels D and DBC have the highest boiling ranges, but no ignition problems were reported. At 4 bar IMEP, a higher global in-cylinder temperature is concluded to be sufficiently high to prevent ignition problems. However, a higher global in-cylinder temperature at low load would be beneficial. In the near future an inlet air heater would be beneficial. In the near future an inlet air heater will be installed to reduce ISHC and ISCO emissions for IMEP ≤ 8 bar.

Figure 5: ISHC emissions versus IMEPg

Figure 6 shows ISPM emissions versus load for all investigated fuels. Soot emissions of fuel D were expected to be higher compared to fuel G, since the high aromatic content of fuel D indicates a high concentration of soot precursors (which aromatics are considered to be). On the other hand, fuel DLA is expected to produce less soot compared to fuel D, since it contains no aromatics (like GTL). Fuel DBC has a significant oxygenate content; which could show an influence on soot emissions. Oxygenates are known to reduce soot emissions, because fuel carbon atoms molecularly bound to an oxygen atom are not likely to form soot. Unfortunately, none of these mentioned expectations were proven by the measurements. Most of
all, it was expected that soot emissions would increase at high load, since injection duration is increased to meet the required load. Therefore, the mixture ignites before the injection event is over (i.e. negative mixing period), which imposes mixture controlled combustion with local rich spots. Fuel G and D show the expected increase with load, however most striking is the maximum in soot emissions at mid load. Moreover, fuels DLA and DBC show major extrema at mid load. ISPM emission extrema at mid load make it impossible to verify the aforementioned hypotheses.

These extrema in can possibly be explained by soot formation and oxidation. At 4 bar IMEP the mixing period (MP) is long, see Figure 9, which indicates for premixed combustion. As discussed before, the temperature of premixed combustion is lower. Up to 12 bar IMEP more fuel is injected; the mixing period is shortened and in-cylinder temperature increases, as a consequence soot is formed. However, the temperature is not sufficiently high to oxidize the formed soot. At 12 bar IMEP a transition takes place, the MP becomes negative, which suggests mixture controlled combustion taking place. This combustion regime is known for higher peak combustion temperatures, therefore the formed soot is oxidized, as a result ISPM emissions decrease. At 20 bar IMEP CA50 is phased late (>7.5 ºCA aTDC), induced by hardware limitations of the fuel system. Due to injector leakage imposed by fuel’s low viscosity, the desired fuel flow at the desired fuel pressure could not be obtained. At this lower fuel pressure, the fuel will not mix as easily and locally the mixture will remain rich with elevated soot emissions as result.

Furthermore, there appears to be a relation between the increased soot emissions at the mid load and BD, see Figure 8 and 2. When BD exceeds 24 ºCA, ISPM emissions tend to increase rapidly. Fuels G and D only show minor extrema, however fuels DLA and DBC show major soot emissions at mid load due to BD > 26 ºCA. As already discussed; long BD indicates for mixture-controlled combustion instead of the, for soot emissions preferable, premixed combustion.

Possibly, a relation between fuel injection pressure and BD can be found, which is a measure for evaporation. To investigate this hypothesis, fuel pressure is varied at constant load and ISPM emissions are monitored, see Figure 9. The figure of the so called fuel pressure sensitivity shows ISPM emissions to increase exponentially below a certain injection pressure. Especially, fuel DBC seems to be very fuel pressure sensitive since soot emissions already increase exponentially below 1400 bar injection pressure and no benefit of oxygenate concentration is reported. For this research a certain injection pressure is selected for every load point (see measurement matrix), which is the same for every fuel, to compare the fuels in a fair way. However, not all fuels have the same viscosity and volatility. Probably, an increase in fuel pressure can enhance evaporation and prevent the soot extrema at mid load for fuels DLA and DBC in particular.

Nevertheless, EURO VI legislation for soot is met at 8 bar IMEP for fuels D and DLA as well as for fuel G at 16 bar IMEP.
Finally, Figure 10 shows the Indicated Specific NOx (ISNOx) emissions. At 4 bar IMEP, NOx emissions are high due to the low set EGR wt-% (see Table 3). At low load, fuel DBC is most beneficial, which is probably an effect of the obtained dilution induced by moderate λ and EGR wt-% (see Table 3). Introducing more EGR at low load (> 50 wt-%), as long as combustion is stable, can possibly solve this problem. At higher loads (> 4 bar IMEP), the conventional NOx-soot trade-off is manifested: high global in-cylinder temperatures reduce soot emissions, but cause NOx emissions to elevate. Fuels DLA and DBC with the highest ISPM emissions have the lowest ISNOx emissions, close to EURO VI legislation level (0.4 g/kWh). Fuels G and D have the lowest ISPM levels, but the EURO VI legislation is not met. Again, introducing more EGR, up to 55 wt-%, can most probably reduce NOx emissions below EURO VI legislation for IMEP > 8 bar.

![Figure 10: ISNOx versus IMEPg](image)

**Conclusions**

Gross indicated efficiency has shown to be the highest for fuel D with a maximum of 51% at 16 gIMEP. It is considered that this high $\eta_{\text{gross}}$ is obtained due to the high aromatic content, since it induces a long ignition delay and short burn duration. Aromatics streams are now considered as waste in refineries; however they seem beneficial to increase engine efficiency this will be investigated in more detail in future research. Considering the difference in carbon chain length between the other fuels a difference in $\eta_{\text{gross}}$ is reported, however the absence of aromatics yields lower $\eta_{\text{gross}}$ over the load range for all of them.

Combustion efficiency increases with load, as expected. Fuel DBC shows the highest $\eta_{\text{comb}}$ at low load due the long BD and short ID. In the near future, an inlet air heater will be installed to increase $\eta_{\text{comb}}$ and decrease ISHC along with ISCO emissions at low load.

All fuels show maxima in soot emissions at mid load, which are induced by the premixed and mixture controlled combustion regimes which are related to the soot formation and oxidation processes. Furthermore, long BDs (> 24 °CA) were reported, which also indicate for mixture controlled combustion. Possibly, an increase in fuel pressure at mid load may elongate the mixing period and shorten burn duration of mainly DLA and DBC, to reduce soot emissions. For now, fuels G and D show the best ISPM reducing potential; however EURO VI soot legislation is not met over the complete load range.

Due to high local temperatures NOx emissions are high at low load for every fuel. Introducing more EGR at low load, as long as combustion remains stable, may solve the problem. For IMEP > 8 bar a small increase in EGR wt-% up to 55% can reduce NOx emissions below EURO VI NOx legislation.

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**References**


