ABSTRACT

Recently, some studies have shown high efficiencies using controlled auto-ignition by blending gasoline and diesel to a desired reactivity. This concept has been shown to give high efficiency and, because of the largely premixed charge, low emission levels. The origin of this high efficiency, however, has only partly been explained. Part of it was attributed to a lower temperature combustion, originating in lower heat losses. Another part of the gain was attributed to a faster, more Otto-like (i.e. constant volume) combustion.

Since the concept was mainly demonstrated on one single test setup so far, an experimental study has been performed to reproduce these results and gain more insight into their origin. Therefore one cylinder of a heavy duty test engine has been equipped with an intake port gasoline injection system, primarily to investigate the effects of the balance between the two fuels, and the timing of the diesel injection. Besides studying trends in the dual-fuel regime, this also allows to find best points to compare with conventional diesel combustion.

Results show that compared to more conventional combustion regimes, this dual-fuel concept can escape from the common NOX-smoke trade-off, reducing both to near-zero values. Although hydrocarbon emissions are somewhat increased, indicated efficiencies are significantly improved. The absolute efficiencies are not as high as reported in other work, but the increase does confirm the potential of the concept. The increase in indicated efficiency is shown to originate from a higher thermal efficiency, because short burn durations at high gasoline fractions enable for CA50 to be phased closer to TDC, without combustion occurring too much before TDC.

Pressure rise rates are as low as with conventional diesel combustion, when using the same Exhaust Gas Recirculation (EGR) percentage. Although the dual fuel concept has a much higher rate of heat release, this is phased better after TDC. A dedicated set of experiments has also shown that the late-cycle diesel injection is dominant in combustion phasing and that control has to be found in this diesel injections.

INTRODUCTION

Due to more stringent emission standards, engine development is forced to invest heavily in research for advanced combustion systems and exhaust after-treatment devices. The exotic materials used in after-treatment systems are pushing production costs; therefore reduction of in-cylinder emissions formation is preferred. Furthermore, rising fuel cost and a focus on reduction of greenhouse gases drive developments towards a better efficiency of the internal combustion engine.

Premixed Charge Compression Ignition (PCCI) is a combustion concept that is characterized by low temperature, partially premixed combustion using early injections, large ignition delays and high percentages of Exhaust Gas Recirculation (EGR). This concept promises intrinsically low emission levels and high thermal efficiencies. Common issues of PCCI combustion are its limited load range and lack of control of combustion phasing [1,2]. Tie Li and coworkers [3] found that for PCCI combustion at higher loads, the promotion of fuel-air mixing at relatively high intake oxygen concentration is necessary. They propose to use low
reactivity fuels, such as gasoline, and lower compression ratios, for example 12 instead of 17, to expand the operating load. Similarly, Kalghatgi and coworkers [1,2,4] showed that low reactive fuels such as gasoline elongate the mixing time of fuel with air and can be used for PCCI combustion at higher compression ratios.

In all of the aforementioned investigations, fuels were pre-blended before injection. Yet, auto-ignition characteristics in an engine depend heavily on the load. Therefore fuel reactivity should be controllable per cycle. Inagaki was the first who proposed a dual-fuel concept for this [5], which over recent years has been further developed by Reitz and coworkers [6]. This latter group is now leading in the field of dual fuel PCCI with use of a port gasoline injection system and early direct injection of diesel for in-cylinder fuel blending and combustion phasing control. Their Reactive Controlled Compression Ignition (RCCI) engine experiments have demonstrated control and versatility of dual fuel PCCI combustion with the proper fuel blend and injection timings.

In the University of Wisconsin (UW) RCCI tests [6,2], very low NOX and smoke emissions were shown, combined with extremely high efficiencies. Already in the first investigations, appropriate use of modeling was made, which has been extended later with more modeling [7,8] and optical engine tests [9]. However, up to now the concept has been mainly shown on Wisconsin’s SCOTe engine [6,7,8,9]. Another setup has been used in an independent-funded study [10], outside of the UW laboratory, though it was conducted in collaboration with UW staff. This study showed significantly lower efficiencies. In recent work, the concept was applied to a light-duty setup [11], in which high efficiencies were reproduced.

The purpose of the current investigation is to extend the application of the dual-fuel RCCI concept outside the Wisconsin labs, and to try to reproduce their results in a completely different experimental environment, essentially for the first time. Therefore a heavy duty test engine has been equipped with an intake port gasoline injection system, to investigate the effects of the balance between the two fuels, and the timing of the diesel injection. Besides studying trends in the dual-fuel regime, this also allows to find best points to compare with conventional diesel combustion.

The Wisconsin group have extended their work using multiple diesel injections [7], the use of alternative fuels [8] and strategies which rely on a single fuel, with and without ignition additive [12,13]. Although each of these extensions has offered advantages, in this first reproduction conventional diesel and gasoline are used, with one single injection for each of them. Varying injection timings are investigated, for three different port injected gasoline (PIG) percentages. Besides this variation of injection timing and gasoline percentages, a dedicated set of experiments is used to investigate the sensitivity of combustion phasing to the injected quantities of diesel and gasoline. To conclude, the best points found for the RCCI concept are compared with conventional diesel combustion.

**EXPERIMENTS**

**EXPERIMENTAL APPARATUS**

For this investigation at Eindhoven University of Technology a six-cylinder DAF engine, referred to as CYCLOPS, is used. Below a brief description of this set-up is given, with the changes made to it for the current investigation. For more information the reader is referred to a more detailed description [2].

The CYCLOPS is a dedicated engine test rig, see Table 1, based on a DAF XE 355 C engine. Cylinders 4 through 6 of this inline 6 cylinder HDDI engine operate under the stock DAF engine control unit and together with a water-cooled, eddy-current Schenck W450 dynamometer they are only used to control the crankshaft rotational speed of the test cylinder, i.e. cylinder 1. Apart for the mutual cam- and crankshaft and the lubrication and coolant circuits, this test cylinder operates autonomously from the propelling cylinders and uses stand alone air, EGR and fuel circuits for maximum flexibility.

**Table 1. CYCLOPS test setup specifications**

<table>
<thead>
<tr>
<th>Base Engine</th>
<th>6 cylinder HDDI diesel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cylinders</td>
<td>1 Test cylinder</td>
</tr>
<tr>
<td>Bore [mm]</td>
<td>130</td>
</tr>
<tr>
<td>Stroke [mm]</td>
<td>158</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>14.9 (original 17.0)</td>
</tr>
</tbody>
</table>

Fed by an air compressor, the intake air pressure of the test cylinder can be boosted up to 5 bar. Non-firing cylinders 2 and 3 function as EGR pump cylinders (see Figure 1), the purpose of which is to generate adequate EGR flow, even at 5 bar charge pressure and recirculation levels in excess of 70%. The EGR flow is cooled both up- and downstream of the pump cylinders. Several surge tanks, to dampen oscillations and to ensure adequate mixing of fresh air and EGR flows, and pressure relief valves, to guard for excessive pressure in the circuit, have been included in the design.

Direct injection of fuel into cylinder 1 is provided by a prototype common rail injector with a nozzle having 8 holes of 0.151 mm diameter and a cone angle of 153 degrees. Gasoline is added through port fuel injection. A Vialle28 injector is mounted in the intake manifold with an angle of 120 degrees resulting in an injected spray positioned on the intake valve. All steady state flows of gasoline, diesel, air and
EGR, are measured with Micromotion Coriolis mass flow meters.

For measuring gaseous exhaust emissions, a Horiba Mexa 7100 DEGR emission measurement system is used. Exhaust smoke level (in Filter Smoke Number or FSN units) is measured using an AVL 415 smoke-meter. All quasi steady-state engine data are recorded by means of an in-house data acquisition system (TUeDACS). A SMETEC Combi crank angle resolved data acquisition system is used to record and process crank angle resolved data.

MEASUREMENT MATRIX AND PROCEDURE

In the present investigations a strategy is explored to achieve RCCI combustion, using the fuels given in Table 2. In order to avoid high pressure rise rates (PRR), an EGR flow of 60 weight percent is used. A load of 11 bar IMEP is used, corresponding to ca. 45% of the rated power. For injecting small amounts of diesel, an injection pressure of 1000 bar is used. Gasoline injection is started just after intake valve open and after exhaust valve close (i.e. 300 deg bTDC) to spray directly into the cylinder and avoid possible blow-through of gasoline. The net fuel pressure of the port fuel injection system is set to ca. 3 bar by controlling the rotational speed of the fuel pump in the gasoline tank. For all measurements, the following conditions are kept constant for all measurements:

- 1200 rpm engine speed
- 11 bar gross IMEP load.
- 60 wt% heavily cooled EGR @ 300K.
- Intake pressure level: 2.0 bar.
- Exhaust pressure level approximately 1.15 bar.

<table>
<thead>
<tr>
<th>Properties</th>
<th>Diesel EN590</th>
<th>Gasoline RON 95</th>
</tr>
</thead>
<tbody>
<tr>
<td>T10</td>
<td>210 ºC</td>
<td>65 ºC</td>
</tr>
<tr>
<td>T50</td>
<td>268.5 ºC</td>
<td>115 ºC</td>
</tr>
<tr>
<td>T90</td>
<td>333.3 ºC</td>
<td>185 ºC</td>
</tr>
<tr>
<td>DCN</td>
<td>55.9</td>
<td>14.7</td>
</tr>
<tr>
<td>Density @300 K</td>
<td>825 kg/m3</td>
<td>753 kg/m3</td>
</tr>
<tr>
<td>LHV</td>
<td>41.54 MJ/kg</td>
<td>43.2 MJ/kg</td>
</tr>
</tbody>
</table>

Table 2. General fuel properties of diesel and gasoline. *LHV is the Lower Heating Value, and T10, T50 and T90 represent the 10, 50 and 90% distillation temperatures.*

A single early direct diesel injection strategy is investigated, combined with port gasoline injection. Combustion phasing is controlled by varying the amount of port injected gasoline and direct injected diesel. To investigate the influence of both the early diesel injection timing, and the gasoline percentage, the following measurement matrix is followed:

- Start of injector actuation (SOA) sweep −40 to −90 degrees aTDC, with 10 degree increments.
- 3 gasoline percentages 70-80-90, with the rest in one single diesel injection.

To calculate the performance, both flows, gasoline and diesel, are directly measured and added to compute one total fuel consumption. For calculating indicated efficiency the total injected mass is used. However, when referring to the thermal efficiency the actual converted fuel is used. Furthermore, all indicated specific values are based on the gross indicated work. For more information on the setup and the procedures and definitions used, the reader is referred to [2].

RESULTS AND DISCUSSION

COMBUSTION PHASING

When injecting diesel fuel early in the compression stroke, the mixing time is elongated, creating a partially premixed mixture of gasoline, diesel and air. When more time is available for the diesel fuel to mix with the premixed gasoline-air mixture, local lambda values become higher. Figure 2 shows injection timing sweeps of the diesel injection for three different gasoline percentages. As SOI is advanced, it is seen that start of combustion (CA5) is retarded. This can be explained by locally leaner and less reactive mixtures due to longer mixing times.
Furthermore, it was expected that when the gasoline percentage is increased for a constant diesel injection timing, start of combustion would also be retarded. From Figure 2 this appears to happen indeed. When the gasoline and diesel have enough time to mix, the local reactivity is dependent on the mixing ratio of the two components. In this case, the lower amount of diesel makes that the local reactivity is decreased and auto-ignition is delayed. Figure 3 displays measured in-cylinder pressure and heat release data for the three gasoline percentages at a constant start of injection of diesel. From this, the change from diesel to gasoline is more clearly observed to phase combustion later in the cycle by reducing local fuel reactivity. As will be shown below, the decrease in amount of diesel is dominant for this effect.

![Figure 2. Timing of Start of Combustion (CA5) for different gasoline fractions and diesel injection timings. 11 bar IMEP, 2 bar intake pressure, 60% EGR](image)

For completeness, in Figure 4 CA50 is given for the measurement points under investigation. Note the striking similarity with Figure 2, indicating a constant offset of about 4 deg CA between CA50 and CA5 (which in turn indicates a fairly constant burn duration). As this is generally accepted as a measure for the overall phasing of combustion, in the remainder of this paper data are presented versus CA50. This enables comparison of the different gasoline percentages, at equal combustion timing. The highest gasoline percentage is shown to phase combustion correctly after TDC for all injection timings. The lower percentages, however, require a very early diesel injection. This is necessary to reduce diesel stratification (and hence local reactivity) to delay auto-ignition. At these long mixing times and lower gasoline percentages, combustion phasing appears to be very sensitive to even a small change in diesel injection timing.

![Figure 4. Timing of CA50, for different gasoline fractions and varying diesel injection timings](image)

**EMISSIONS**

Longer mixing times create the possibility for partially premixed air-fuel mixtures to combust at lean and low temperature conditions. In the RCCI combustion concept, the majority of the injected fuel is premixed and thus is able to combust under such conditions. The advantage of lean low temperature combustion can be seen in Figure 5. As mixing time is increased, NO\textsubscript{x} levels decrease dramatically. Note that nearly all NO\textsubscript{x} emissions are below the Euro VI NO\textsubscript{x} emissions standards, which are 0.4 g/kWh.

Like for NO\textsubscript{x} emissions, smoke emissions are also known to benefit from locally less rich conditions. With the highly premixed gasoline, and long mixing times for diesel, the smoke levels experienced are very low, see Figure 6. For the highest gasoline percentage of 90%, all smoke levels are well below the Euro VI norm of 0.01 g/kWh. This already shows one of the promising aspects of the RCCI concept. In conventional diesel combustion (CDC), a NO\textsubscript{x}-smoke trade-off is generally experienced, i.e. a measure to reduce NO\textsubscript{x} will lead to higher smoke emissions and vice versa. Unlike CDC, the RCCI concept simultaneously shows low engine-out soot and NO\textsubscript{x} emissions.
Generally, when NO\textsubscript{x} and smoke emissions are decreased by lean low temperature combustion, combustion becomes less complete, which comes with a penalty in HC and CO emissions, see Figures 7 and 8. Two effects can be distinguished from Figure 7. First, as combustion is advanced for a fixed gasoline percentage, maximum cycle temperatures increase, leading to more complete combustion. Second, for a constant CA\textsubscript{50}, hydrocarbon emissions decrease with increasing gasoline percentage. This effect likely finds its origin in the mixing time associated with these points. For the lower gasoline percentages, injection has to be advanced far into the compression stroke to achieve conditions lean enough for combustion to be phased after TDC. Locally, however, this gives rise to over-leaning.

The same effects as discussed for hydrocarbon emissions can also be identified for CO emissions, see Figure 8. Again, for all gasoline percentages increasing CO emissions are found with retarding CA\textsubscript{50}. Also, for a constant CA\textsubscript{50}, lower carbon monoxide emissions are experienced at higher gasoline percentages. This shows that for combustion to be correctly phased after TDC, it is better to use higher gasoline percentages: an excessively long mixing time is not necessary then, which reduces the risk of over-leaning.

PERFORMANCE

In Figure 9, the indicated efficiency is presented. Commonly, the indicated efficiency increases when CA\textsubscript{50} is advanced from late timings to closer to TDC. However, unlike normally, for the measured points indicated efficiency even increases further when CA\textsubscript{50} is advanced to before TDC. This is hypothesized to have its origin in an increasingly
complete combustion. As discussed above, for longer mixing times and late CA50 phasing, the charge is overleaned and combustion is too lean for complete combustion. As injection is retarded, combustion is advanced and because of the reduced mixing time CO and HC emissions decrease.

MAXIMUM PRESSURE RISE RATE

As discussed above, increasing the gasoline percentage results in higher rates of heat release. As can be expected, Figure 11 shows that for a constant CA50 this results in slightly higher maximum pressure rise rates. However, the dominant parameter appears to be the combustion phasing. The higher gasoline percentage allows for CA50 to be phased correctly after TDC, therewith reducing maximum pressure rise rates significantly. Therefore, also for the highest gasoline fraction, the maximum pressure rise rate is not expected to be an issue.

To verify this assumption, in Figure 10 the thermal efficiency is shown. Instead of using the injected fuel mass, as is done for the indicated efficiency, only the actual converted fuel quantity is used. The thermal efficiency is thus corrected for any combustion loss, i.e. HC and CO emissions. This shows efficiency to be significantly flatter at CA50 around TDC. Furthermore, the effects of increasing gasoline percentage are more clear. For the highest gasoline percentage, thermal efficiency is significantly higher compared to the lower gasoline percentages, most likely because of the higher heat release rate. This higher combustion speed allows CA50 to be phased closer to TDC without combustion occurring too much before TDC. Because a more Otto-like, i.e. constant volume, combustion process is approached, the 90% gasoline points especially benefit from increased thermal efficiency.

SENSITIVITY TO INJECTED QUANTITY

One of the most important possibilities of RCCI is the control of SOC by controlling the in-cylinder mixture's reactivity. In conventional diesel combustion, start of injection is used as a control for start of combustion. In this case, when control is sought in the reactivity of the in-cylinder fuel, it is important to verify what the actual ignition trigger is. According to Reitz [11] global fuel reactivity can be controlled by adding diesel to the gasoline, creating a new in-cylinder fuel blend. However, the influence of adding diesel seems to have greater effect than the addition of gasoline, which indicates that local diesel might determine the timing of ignition, dependent on injection timing and amount of fuel. To assess the effect of the diesel injection on SOC, various gasoline fuel fractions are investigated. In Figure 12 one can schematically see the two experiments under investigation in this section. In the figures to follow, the baseline is indicated with a green diamond.
Figure 12. Diesel and gasoline fuel quantities for three different injection strategies.

First, the amount of gasoline is increased, indicated with a red square. It is found, see Figure 13, that when the gasoline fuel flow is increased, the change in SOC is far less significant than when the diesel fuel flow is increased, denoted with the purple triangles. As the diesel fuel’s reactivity differs more from the global reactivity, a change in diesel fuel quantity effects this reactivity more. Still, the global reactivity is only changed slightly. More importantly, the diesel spray will ensure a higher stratification and lower local lambda. It is also therefore, that this effect is less when injection timing is advanced, as this reduces stratification for all mixtures. From these results one can conclude that the diesel injection timing and quantity are the two most important parameters to control combustion phasing.

Figure 13. Start of Combustion (CA5) for three different injection strategies 2 bar intake pressure, 60wt% EGR, varying load.

DUAL-FUEL VS DIESEL-ONLY

To discover the full potential of RCCI combustion, the concept is compared to diesel-only Conventional Diesel Combustion (CDC). The experiments done for this comparison are at equal loads: 11 bar gross IMEP. To create a similar experiment, 60 wt% of the total mass flow is EGR for both combustion concepts. However, such a large amount of EGR is generally not used at CDC. Therefore also experiments with 20 wt% EGR are included. For the RCCI concept 90 wt% gasoline was used, which above showed to give the best results.

Figure 14. Nitric oxides emissions comparing three concept, logarithmic scale 11 bar IMEP, 2 bar intake pressure.

From Figure 14, on a logarithmic scale, one can see that the application of EGR is an effective method to reduce NOx emissions, independent of the concept used. However, the general drawback of the use of EGR is that the air fuel ratio is reduced. For conventional diesel combustion, this results in high smoke production, as can be seen in Figure 15, again on a logarithmic scale. Here lies the main advantage of partially premixed combustion concepts: they allows for simultaneous reduction of both nitric oxides and smoke emissions.

One of the drawbacks of the RCCI concept can be seen in Figure 16. Because of the highly premixed charge, and therefore hydrocarbons getting stuck in crevices, UHC emissions are significant. Also, long mixing times and globally lean mixtures can lead to overleaning. The resulting incomplete combustion gives rise to increased hydrocarbon emissions.

Figure 15. Smoke emissions comparing three concept, logarithmic scale 11 bar IMEP, 2 bar intake pressure.
Even despite the higher combustion losses for the RCCI concept, fuel consumption can be significantly reduced. The results in Figure 17 include the small effect of the heating value, which is higher for gasoline compared to diesel. This effect, however, is small compared to the advantages the RCCI concept displays, i.e. short burn durations and low combustion temperatures.

One other possible issue associated with higher heat release rates can be an unacceptably high pressure rise rate. However, as is shown in Figure 18, pressure rise rates are acceptable and even lower than for conventional diesel combustion, using low EGR rates. This can be attributed to (a combination of) correct phasing of combustion after TDC and the increased heat capacity and dilution effect of the recirculated exhaust gas.

**FINAL DISCUSSION**

One of the purposes of the experiments in this investigation has been to reproduce the RCCI results published by the University of Wisconsin's Engine Research Center. Under similar experimental conditions, also similar results were found for NOx and smoke emissions. Also HC and CO emissions are alike. However, in thermal efficiency significant differences are found. In the best measurement series the Wisconsin group reports a thermal efficiency of >55%, in contrast to these measurements where a maximum of 49% thermal efficiency is achieved. Even though absolute efficiencies are somewhat lower, the relative increase is significant. As also the baseline efficiency is significantly lower, differences should be found in the base engine.

One of the differences can be found in the amount of heat loss. Experiments in Wisconsin might have been performed at higher coolant temperatures, which can result in lower...
cylinder wall heat loss. Furthermore, the slight difference in engine speed may have a small influence, but mainly the hardware of the base engine is expected to contribute to differences in thermal efficiency. A second significant contribution to a difference in thermal efficiency can be the amount of blowby of the base engine. This blowby has to be further investigated if high absolute efficiencies are desired to be achieved in future experiments.

Another difference is the amount of EGR used. Reitz and coworkers use 40 wt%, compared to 60 wt% in the experiments performed in Eindhoven. An explanation for this could be the difference in fuel reactivity of the diesel fuel. The Cetane-Number of the US-diesel that was used by Reitz and coworkers is about 40, which should be compared to a CN of 55.9 of the EN590 diesel. Due to this lower reactivity of the US diesel, the diesel injection experiences a longer ignition delay and therefore lower EGR rates can be used. This suggests that future experiments should focus on reactivity of each of the two fuels used. Furthermore, a multiple injection strategy can be used for more control of combustion phasing.

**CONCLUSION AND OUTLOOK**

The purpose of the current investigation was to provide a broader demonstration of the dual-fuel RCCI, and to try to reproduce the results from the Wisconsin group in a completely different experimental environment, essentially for the first time. Therefore a heavy duty test engine has been equipped with an intake port gasoline injection system, to investigate primarily the effects of the balance between the two fuels, and the timing of the diesel injection. Besides studying trends in the dual-fuel regime, this has allowed us to find best points to compare with conventional diesel combustion.

Results show that compared to more conventional combustion regimes, this dual-fuel concept can escape from the common NOx-smoke trade-off, reducing both to near-zero values. Even though hydrocarbon emissions are somewhat increased, indicated efficiencies are significantly increased. The absolute efficiencies are not as high as reported in other work, but the increase as such confirms the potential of the RCCI concept. The increase in indicated efficiency is shown to originate from higher thermal efficiency, because short burn durations at high gasoline fractions enable for CA50 to be phased closer to TDC, without combustion occurring too much before TDC.

Pressure rise rates are found to be as low as with conventional diesel combustion, for the same EGR percentage of 60 wt%. Although the dual fuel concept has a much higher rate of heat release, this is phased better after TDC. A dedicated set of experiments has demonstrated that the late-cycle diesel injection is dominant in combustion phasing and that control has to be found in this single diesel injection.

In comparison to the RCCI results published by the University of Wisconsin's Engine Research Center, under similar experimental conditions, also similar results were found for NOx and smoke emissions. Also HC and CO emissions are alike. However, in absolute thermal efficiency some differences are found. In the best measurement series Reitz reports a thermal efficiency of >55%, in contrast to the present measurements where a maximum of 49% thermal efficiency is achieved. Although absolute efficiencies are slightly lower, the relative increase is comparable. As also the baseline efficiency is significantly lower, differences are expected to be found in the base engine or the diesel fuel, which should be further investigated.

Lower EGR rates are thought to be beneficial for increasing thermal efficiency and real-world applicability. One of the items of future research can therefore be to decrease EGR rates by investigating multiple injection strategies for more control of combustion phasing. Furthermore, a less reactive fuel can be used in such a multiple injection strategy. This less reactive fuel, for example US diesel, can be used to advance the second diesel injection and therefore create more mixing time for diesel-fuel to mix with the in-cylinder blend. This would allow for EGR rates to be reduced, which can boost efficiencies even further and make the concept more viable for real-world application.

**REFERENCES**


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