Combustion phasing controllability with dual fuel injection timings

Combustion Phasing Controllability with Dual Fuel Injection Timings

2012-01-1575
Published
09/10/2012

C.A.J. Leermakers, L.M.T. Somers and B.H. Johansson
Eindhoven University of Technology

doi:10.4271/2012-01-1575

ABSTRACT

Reactivity controlled compression ignition through in-cylinder blending gasoline and diesel to a desired reactivity has previously been shown to give low emission levels and a clear simultaneous efficiency advantage. To determine the possible viability of the concept for on-road application, the control space of injection parameters with respect to combustion phasing is presented. Four injection strategies have been investigated, and for each the respective combustion phasing response is presented. Combustion efficiency is shown to be greatly affected by both the injection-timing and injection-strategy. All injection strategies are shown to break with the common soot-NOx trade-off, with both smoke and NOx emissions being near or even below upcoming legislated levels. Lastly, pressure rise rates are comparable with conventional combustion regimes with the same phasing. The pressure rise rates are effectively suppressed by the high dilution rates used.

INTRODUCTION

Recently, studies [1,2,3,4,5] have shown high efficiencies using Reactivity Controlled Compression Ignition (RCCI) through in-cylinder blending gasoline and diesel to a desired reactivity. This concept has been shown to give low emission levels, because of the largely premixed charge, combined with a simultaneous efficiency advantage. Part of the efficiency gain has been attributed to a lower combustion temperature, giving lower heat losses. Previous results have also shown that short burn durations at high gasoline fractions enable an optimized combustion phasing.

To determine the possible viability of the concept for on-road application, controllability is of vital importance. While the authors recognize that the impact of control parameters can be highly load and speed dependent, in this paper one operating point has been chosen. The authors present a determination of the control space of injection parameters with respect to combustion phasing. Furthermore, the effects of these parameters on emissions and efficiencies are also determined. The experiments have been performed on a heavy duty test engine [6], equipped with an intake port fuel injection (PFI) system and a common-rail direct injection (DI) system [7].

For the given speed/load point, three types of combustion phasing control can be distinguished: First, the balance between the high (DI) and low reactive fuels (PFI), i.e. gasoline and diesel respectively, is of vital importance. This first order effect has been extensively studied in [7] to enable viable single-cylinder tests of the concept.

Secondly, the timing of the DI timings, one early and one late, can be controlled. This gives some control on the level of stratification, both in mixture strength and reactivity. Especially the timing of the second injection can be used to phase combustion correctly. However, the timing of the first injection also has a significant effect on this phasing.

Last, the ratio of these two diesel injections can be controlled. Again, this has its influence on the in-cylinder stratification of the charge. Also for this third type of control the sensitivity will be shown.

Below, first the setup of the experiments is given. A brief description of the experimental apparatus is presented, with the modifications implemented for the present investigation. After this, the measurement procedure is given, with all constants and variables considering the different injection strategies. In the results section, the focus is on the combustion phasing response, but also emission levels, efficiencies and pressure rise rates are discussed.

EXPERIMENTS

Experimental Apparatus

For this investigation a six-cylinder DAF engine, referred to as CYCLOPS, is used. For more information on the setup the reader is referred to a detailed description [6], of which this subsection is a short summary.

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 from 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

Base Engine	6 cylinder HDDI diesel
Cylinders	1 Test cylinder
Bore [mm]	130
Stroke [mm]	158
Compression ratio	14.9 (original 17.0)

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 with 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.

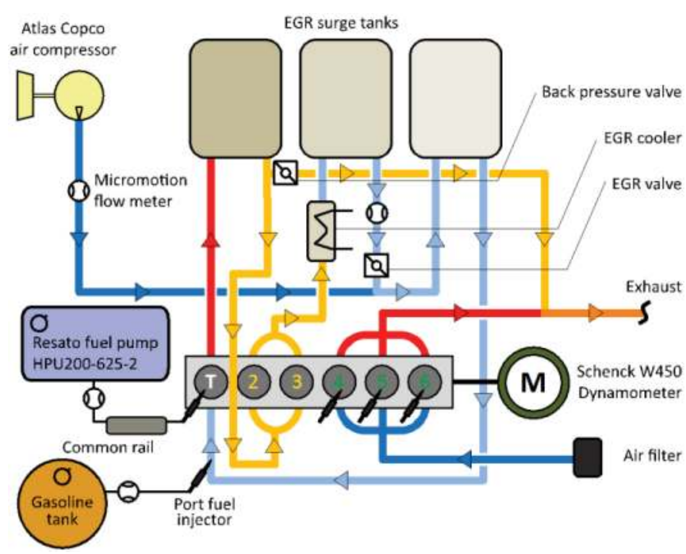

Figure 1. Schematic of CYCLOPS experimental setup: a modified DAF engine using separate fuel, air and EGR systems for one dedicated test cylinder

Measurement Matrix and Procedure

In the present investigation different injection strategies, using the fuels from Table 2, are explored to determine their respective control effects on combustion phasing and emissions. A single target load of 11 bar gross IMEP is used, corresponding to ca. 40% of the engine's rated torque. The fuel mass flow is kept constant, and small variations in load (originating for varying efficiencies) are allowed accordingly. The engine speed is set to 1200 rpm, which is typical for a heavy duty vehicle during highway cruising. An EGR flow of 60% weight percent is used, both to limit pressure rise rates, as found in [7] and to have nitrogen oxides emissions below Euro VI levels [8, 9].

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

Properties	Diesel EN590	Gasoline RON 95
T10	210° C	65° C
T50	268.5° C	115° C
T90	333.3° C	185° C
DCN	55.9	14.7
Density @300 K	825 kg/m3	753 kg/m3
LHV	41.54 MJ/kg	43.2 MJ/kg

The latter two references have stated that a combustion temperature higher than 1500K is necessary to promote the reactions from CO to CO2, and that on the other hand it is important to be below 2000K to avoid thermal NOx formation (Zeldovich' mechanism). Apart from an EGR weight percentage of around 50%, this also implies the use of a global lambda value of around 1.5. To achieve such an air excess ratio, intake pressure is set to 2.0 bar absolute.

The recirculated exhaust gas is heavily cooled using cold process water, to ca 300K and exhaust pressure was constant at 1.15 bar absolute. While the authors acknowledge that this value does not represent real-life conditions and has its effects on charge temperatures, EGR composition and the amount of internal EGR, a more realistic value was not possible for the setup at the time of the experiments.

In [7] it was shown that unburned HC emissions can be limited by injecting a rich enough, well-timed gasoline mixture. Therefore 80 wt% of the injected mass is gasoline, of which the 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 PFI system is set to approximately 3 bar by controlling the rotational speed of the fuel pump in the gasoline tank.

Summarized, for all measurements the following conditions are kept constant:

- 1200 rpm (σ=0.44 rpm),
- 11 bar target gross IMEP, by keeping fuel flow constant at 1.23 g/s (σ=0.022 g/s),
- 2 bar absolute intake pressure (σ<0.005 bar),
- 1.13 bar absolute exhaust pressure (σ=0.014 bar),
- 62wt% heavily cooled EGR (σ=0.9wt%),
- 306 K intake temperature (σ=2.4 K),
- Port injected gasoline of 80% of injected mass, timed before IVC.

For these constant load, speed and ambient conditions, which result in a lambda value of 1.60 (σ=0.03), four diesel injection strategies are investigated. All injection timings are referred to by their Start of Actuation (SOA). From logged injector current and fuel rail pressure, the difference between start of actuation and start of injection for the present engine speed is estimated at 4 degree crank angle.

First, as a benchmark, 20 wt% of injected mass is injected in a single diesel injection. For injecting such small amounts, a diesel injection of 1000 bar is used. This is the minimal pressure to have stable operation of the injector, using a 500 microsecond (=3.6CAD) actuation duration. For this single diesel injection the start of injector actuation (SOA) is swept from −40 to −90 degrees aTDC, with 10 degree increments. The rest, i.e. 80% of the fuel is port injected.

In the second and third strategies the 20 wt% diesel is equally divided over two injections, one early and one late. To enable stable operation of the injector, the injection pressure has be lowered to 500 bar, to enable a sufficiently long actuation duration. It will be shown that this pressure reduction in itself has major implication for e.g. emission levels, but unless another injector nozzle is used, this is the only way to apply such small injections.

In the second strategy the late injection is fixed at −10 degrees aTDC, with an early injection variation from −40 to −90, with 10 degree increments. In the third strategy the early injection is fixed at −70 degrees aTDC, with a late injection variation from −25 to −5, with 5 degree increments. Both these strategies are graphically shown in Figure 2.

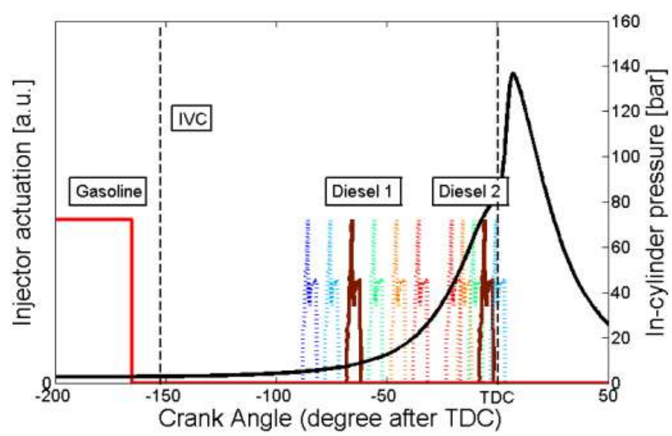

Figure 2. Dual diesel injection strategies with 50-50 distribution with exemplary cylinder pressure trace. Port injected gasoline of 80% of injected mass timed before IVC. Two diesel injections, with 10% injected mass each. The solid brown injector actuation signals represent the fixed values, while colored dashed actuation signals represent the variations to these values.

The fourth strategy is derived from the third one, with the early injection fixed at −70 degrees aTDC, with a late injection variation from 25 to 5, with 5 degree decrements. Now, again with 500 bar injection pressure, the early-late balance is shifted to 70:30, to test the influence of this division. The four injection strategies can be summarized as Table 3:

Table 3. Injection strategies used.

Code	PFI	DI1	DI2	Description
80-20-00-D1	80	20	00	DI1 timing variation, 1000 bar rail
80-10-10-D1	80	10	10	DI1 timing variation, 500 bar rail
80-10-10-D2	80	10	10	DI2 timing variation, 500 bar rail
80-14-06-D2	80	14	06	DI2 timing variation, 500 bar rail

Specific fuel consumption and emission levels are all based on the gross indicated work. The SMETEC Combi system logs cylinder pressure at 0.1 °CA increments for 50 consecutive cycles and processes this to the average and standard deviation of important combustion parameters, such as CA10, CA50 and IMEP. For more information on the setup and the procedures and definitions used, the reader is referred to [6].

RESULTS AND DISCUSSION

The results are presented and discussed below in six subsections. First, and most important, the combustion phasing effect is presented and compared for each of the four injection strategies. In the subsequent subsections also the effects on UHC and CO emissions, fuel consumption, PM and NOx emissions and maximum pressure rise rate will be compared. Finally, a short summary is given, mainly considering the combustion phasing effects.

Combustion Phasing Control

The principal argument for testing the four injection strategies is to determine their respective control spaces with respect to combustion phasing. First, this effect is shown for the single injection strategy. Secondly, this is compared to an 50:50 early injection variation. After a comparison of the respective effects of the early and late injection in this 50:50 strategy, this late injection variation is also compared to a similar variation in a 70:30 strategy.

Single Early Injection

In Figure 3, the response of CA50 is shown for a single early injection. The first thing one can notice is that advancing this injection timing gives more time for the diesel to mix with both air and gasoline, resulting in a leaner and less reactive local mixture when approaching top dead center. Through this locally leaner and less reactive mixture, advancing injection timing results in retarded combustion.

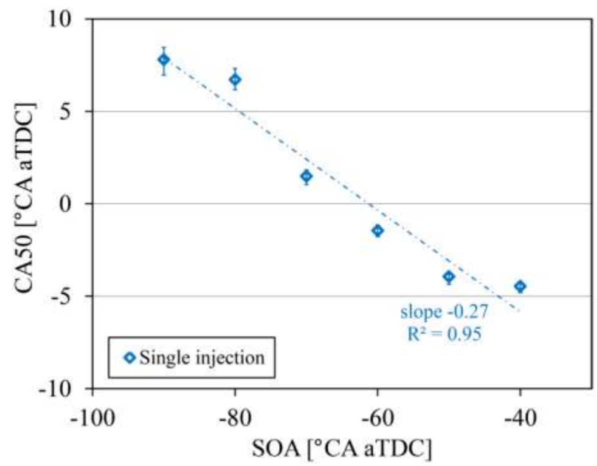

Figure 3. Timing of CA50, for varying single early diesel injection timings. Marker and vertical errors depict the mean and standard deviation, respectively, of 50 measured cycles per operating point. The given slope and coefficient of determination (R^2) are based on a linear regression fit.

A first order fit of the measured points gives a quantification of this negative slope, the sensitivity of the ignition delay defined as

$$s_{ID} = \frac{\partial CA50}{\partial SOA}$$

takes a value S_{ID}=−0.27. This negative and relatively low value means that the bandwidth of the control might be too low, as very large changes in injection timing will be necessary to have a significant impact on combustion phasing.

Furthermore, the coefficient of determination is relatively low at 0.95, which shows that the strength of the linear association is not very high. For later injections, this is expected as the sign should change, as will be shown later. At the injection timings of −50 and −40 a reduction of the negative slope can already be seen.

80-10-10-D1 (First Injection Variation)

The second injection strategy under investigation consists of two equally divided diesel injections. The D2 is fixed at −10 degrees after TDC, while the first (D1) is swept from −40 to −90. From the cylinder pressure and heat release traces, as shown in Figure 4, the clear effect of advancing this first injection can be seen.

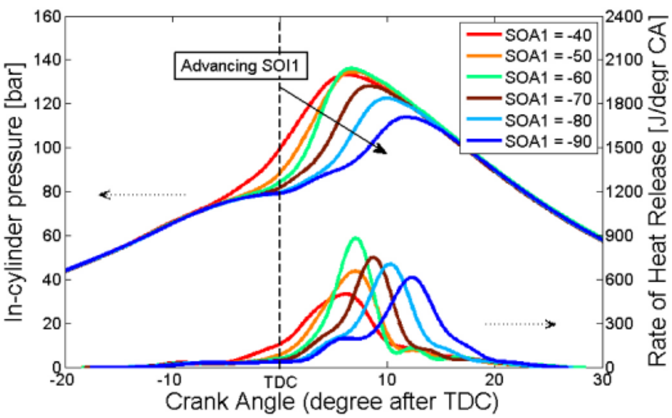

Figure 4. Single shot in-cylinder pressure and aROHR for a varying injection timing of the first diesel injection in a 50:50 strategy. Start of the second diesel injection is fixed at −10 degrees after TDC. Colors as in Figure 2.

Like found for the single early injection strategy of the 80-20-00-D1 case, again an advancing injection results in a retarded combustion. By giving the injected diesel more time to mix with both air and gasoline, the mixture is locally leaner and less reactive, respectively, by the time the piston approaches top dead center.

Not only does Figure 4 show that the phasing of CA50 is retarded for advancing injection timing, but also the shape (width and height) of the apparent rate of heat release is affected. Despite this observation, in the remainder of this investigation CA50 will be used as the main control variable.

In Figure 5, the response of the 80-10-10-D1 strategy is compared to that of the single injection strategy. The coefficient of determination is comparable for the two strategies, and so is the fact that the slope is negative. However, as expected, because the mass in the early injection is halved, the slope is also significantly reduced; S_{ID}=−0.1. In fact it hardly shows any response to the timing of injection. Therefore it is barely usable as a control parameter.

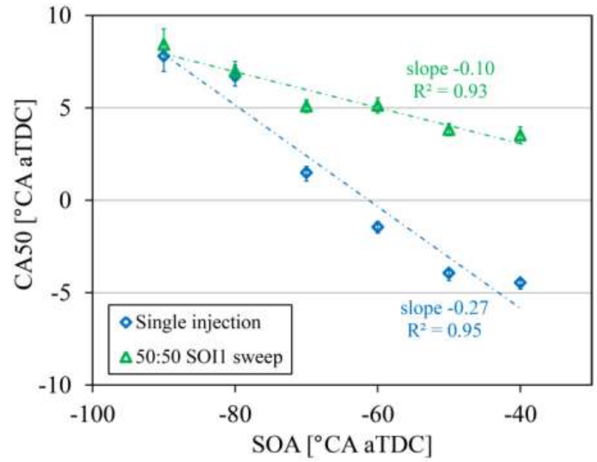

Figure 5. Timing of CA50, for varying single early diesel injection timings and a variation of the timing of the first injection in a 50:50 dual injection strategy.

80-10-10-D2 (Second Injection Variation)

In the third injection strategy (80-10-10-D2), the early diesel injection is fixed at −70 degrees after TDC and the second, late injection is varied to control combustion phasing. Again, in Figure 6 cylinder pressure and heat release traces are given, but now for the late injection timing variation. One can directly note that the injection timing response on CA50 is reversed, as will be further discussed below.

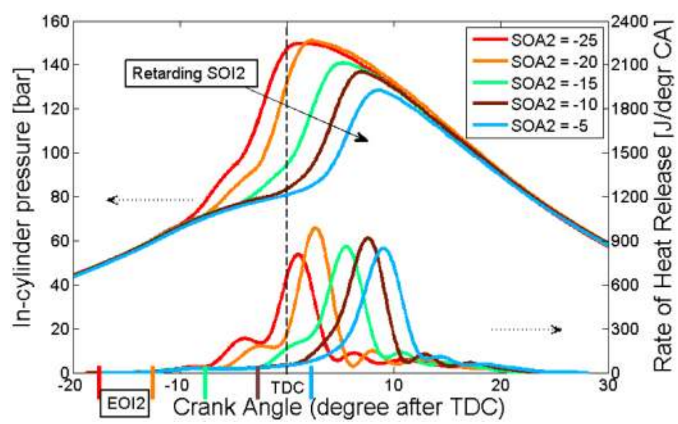

Figure 6. Single shot in-cylinder pressure and aROHR for a varying injection timing of the ***second*** ***diesel injection in a 50:50 strategy. End of this second injection is indicated on the x-axis. Start of the first diesel injection is fixed at −70 degrees after TDC. Colors as in*** Figure 2.

When the rate of heat release shapes are compared for the different late injection timings both the overall shape and the maximum values are similar. Furthermore, for all timings, injection is finished before the corresponding heat release reaches a significant level.

Even though the combustion event does not overlap with the injection event, this late injection timing variation offers a much higher level of controllability. The sensitivity now becomes as can be observed in (Figure 7) S_{ID} = 0.45 and has changed sign with respect to the 80-10-10-D1 strategy where −0.1 is found. For such a small injection, and still without any overlap with the combustion event, it is fairly high. Still is it not unity as one would find for a fully conventional injection.

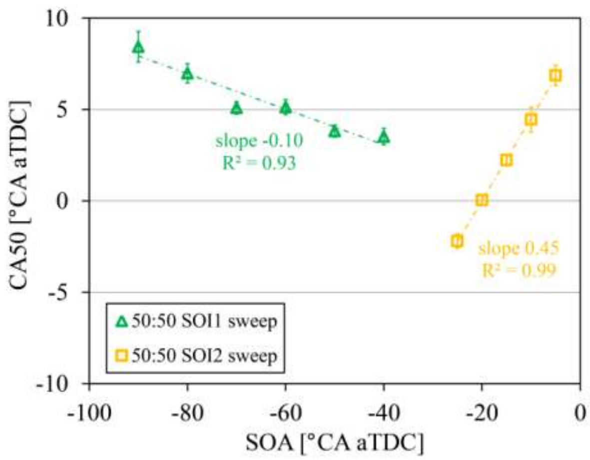

Figure 7. Timing of CA50, comparing variations of the timings of the first (D1) and second injection (D2), respectively, in a 80-10-10 dual injection strategy.

80-14-6-D2 (Second Injection Variation)

As a fourth, and last, injection strategy the amount of mass in first and second diesel injection is no longer kept equal. In this strategy again the late injection is varied. In Figure 8, the response of this strategy is compared to the late injection timing variation of the equally divided strategy.

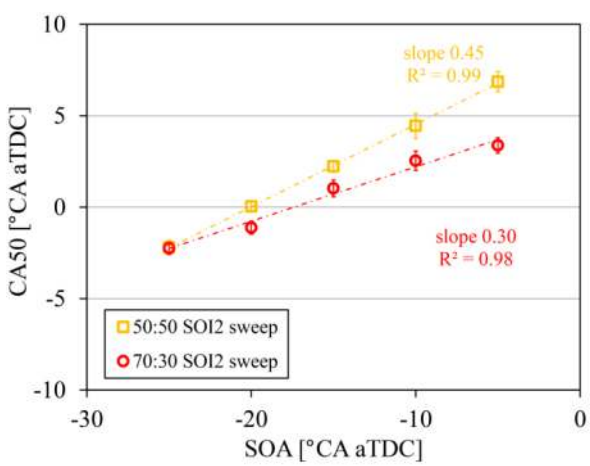

Figure 8. Timing of CA50, comparing variations of the timing of the second injection, in a 50:50 and 70:30 injection strategy, respectively.

The strategy with the reduced late injection still has a quite linear response on combustion phasing. However, the slope of this response has decreased, largely proportional to the amount of fuel injected in this late injection. Although this somewhat lower slope might still be enough for stable and robust control, the higher response of the 80-10-10-D2 strategy is preferred as it is able to shift CA50 over a broader range.

Summary

The combustion phasing response of all four injection strategies is combined in Figure 9. As discussed above, the single and early injections have an inverse effect on combustion phasing. Furthermore the linear association of the response is not very strong and in a double injection strategy, the response of CA50 on a variation of the first injection is (very) weak.

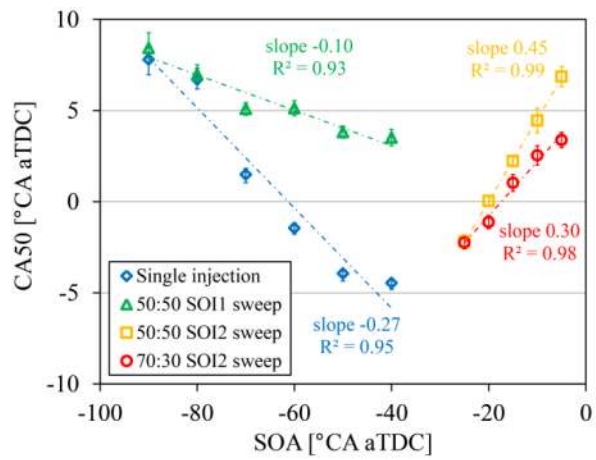

Figure 9. Timing of CA50, for injection timing variations in 4 different injection strategies.

The response to a variation in the late injection has a positive S_{ID}, which is nearly exactly linear and has a larger value compared to the early injection variation strategies. Furthermore, above it was shown that the more fuel is admitted in the second injection, the larger $|S_{ID}|$ is. Therefore 80-10-10-D2 strategy was found to be most favorable with respect to combustion phasing response.

UHC and CO Emissions

Not only do the respective injection strategies have their effects on the phasing on combustion, also emissions might differ significantly. In Figure 10 first the unburned hydrocarbon emissions are shown for the four strategies.

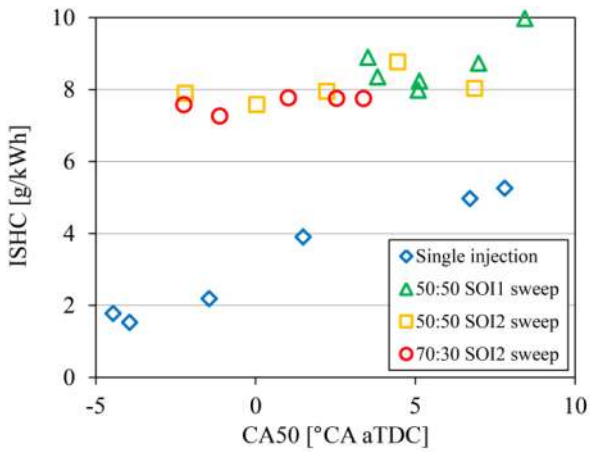

Figure 10. Unburned hydrocarbon emissions vs. CA50 for injection timing variations in 4 different injection strategies.

The single injection strategy differs most from the other three and shows an effect of the combustion phasing. Apparently an early combustion phasing yields higher temperatures, high enough for combustion to be as complete as possible. But even at the earliest combustion phasing, with the highest maximum temperatures, combustion efficiency is relatively low. The combustion efficiency is computed from the unburned hydrocarbons and carbon monoxide emissions, and shown in Figure 11. As most unburned hydrocarbons are thought to be gasoline, the heating value of the unburned hydrocarbons is assumed to be that of gasoline.

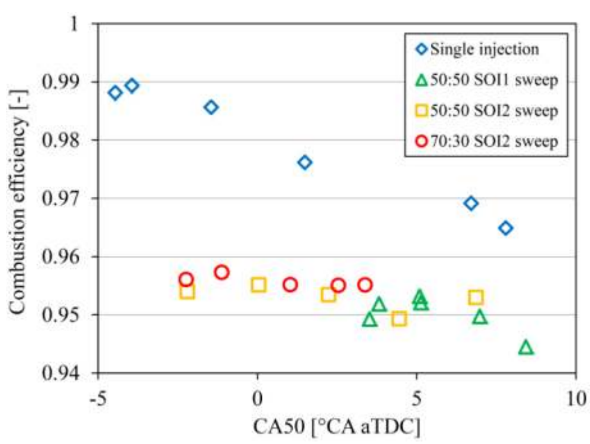

Figure 11. Combustion efficiency, based on UHC and CO emissions, vs. CA50 for injection timing variations in 4 different injection strategies.

For this combustion concept, the unburned hydrocarbon emissions from the single injection strategy are generally thought to originate from the premixed charge trapped in crevice volumes. Remarkably the results for the double injection strategies are poorer (Figure 10 and 11). For these double injections, the injection pressure had to be lowered to 500 bar, and together with the very short injection, this appears to be dramatic for the completeness of combustion. One hypothesis is that the length of the diesel spray penetration is too short to ignite all of the premixed gasoline-air mixture. Another hypothesis is that bad vaporization and mixing, originating in the lower injection pressure, might hamper the ignition of the premixed charge.

The same observations hold for carbon monoxide emissions, as shown in Figure 12. Again, for the single injection strategy the levels decrease with advancing combustion, but even at the earliest phasing remain high. Also, the double injection strategies all give unacceptably high values.

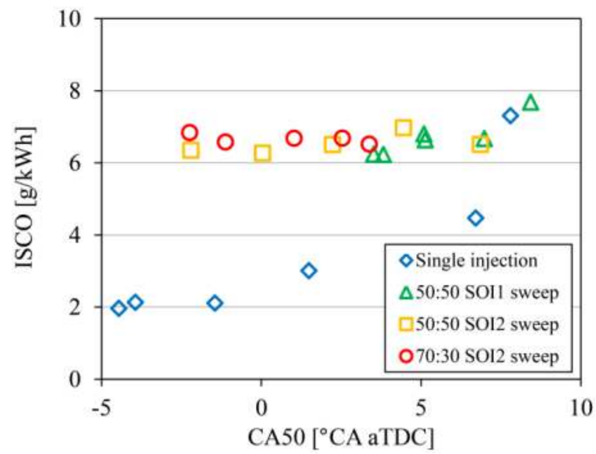

Figure 12. Carbon Monoxide emissions vs. CA50 for injection timing variations in 4 different injection strategies.

Both for CO and HC emissions, the different double injection strategies show very little differences. However, all of their values are nowhere near the upcoming, or even current, legislated limits. The exact origin and possible solutions for this issue should be further research to enable the viability of such injection strategies.

Efficiency

As can be expected, the previously discussed trends for carbon monoxide and unburned hydrocarbon emissions have a significant effect on the gross indicated fuel efficiency, as shown in Figure 13.

For the single injection strategy, at early phasings a reasonable efficiency is achieved, while at later phasings the unburned hydrocarbon and carbon monoxide emissions take their toll and decrease efficiency. As such, the high efficiencies at early phasings are not caused by the thermodynamic process, but through more complete combustion. This is confirmed by taking the combustion efficiency into account to compute the thermal efficiency (Figure 14).

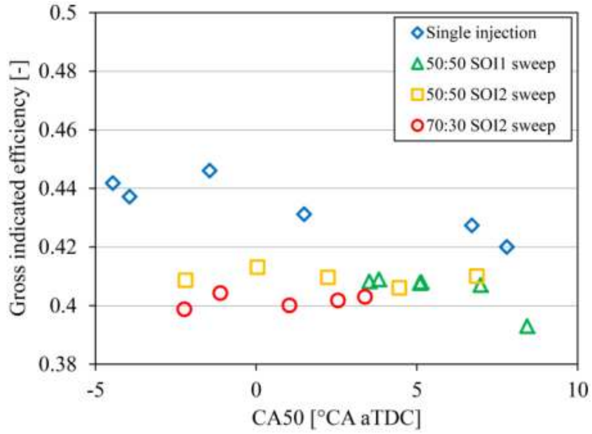

Figure 13. Gross indicated efficiency vs. CA50 for injection timing variations in 4 different injection strategies.

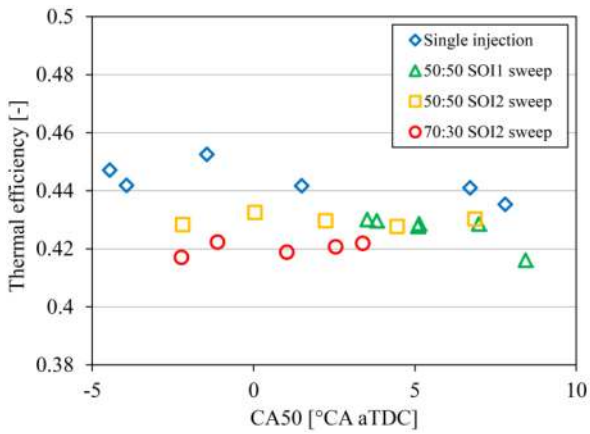

Figure 14. Thermal efficiency vs. CA50 for injection timing variations in 4 different injection strategies.

All double injection strategies have an indicated efficiency of about 10 percent lower than the single injection strategy. Other tests, as referred to in the introduction, have shown that the dual-fuel concept, even with double injections, is possible of producing very high efficiencies, and thus low fuel consumption. However, because of the low completeness of combustion, this is not achieved in the present investigation. Furthermore, because of the even lower combustion efficiency, the double injection strategy adds up to 10 percent to fuel consumption compared to the single injection strategy.

NOx and Smoke Emissions

One of the most important reasons of developing a dual fuel concept is to reduce the engine-out nitrogen oxides and smoke emissions. In this way, expensive exhaust gas after treatment systems do not have to be used (as much), while upcoming legislation levels can still be met. In Figure 15, first the nitrogen oxides emissions are shown for the four injection strategies.

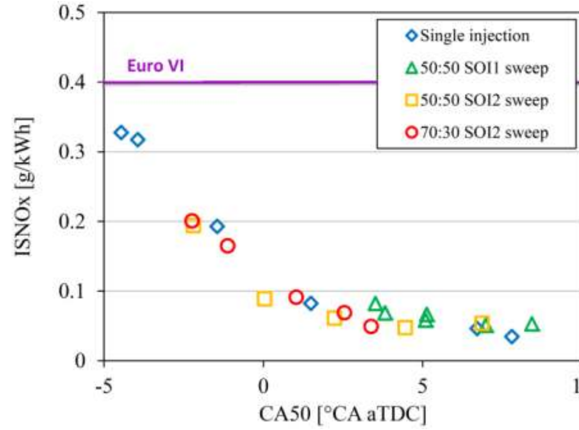

Figure 15. Nitrogen oxides emissions vs. CA50 for injection timing variations in 4 different injection strategies. Euro VI level depicted by purple line.

One can clearly see the effect of combustion phasing on nitrogen oxides emissions. With advancing combustion peak temperatures rise considerably and when combustion is advanced before TDC the charge remains at high temperature for a longer time. Because of the constant dilution level and oxygen concentration this results in increased nitrogen oxides emissions. The inverse effect was seen above in a single injection strategy for the completeness of combustion, which benefits from the increased temperatures. However, because of the high dilution level applied in this concept, for most, or all, of the measured points NOx emissions are still well below the current and upcoming legislated emission levels.

The highly premixed nature of the dual-fuel combustion concept is known to reduce smoke emissions significantly compared to conventional diesel spray combustion. In Figure 16, the smoke emissions are compared for the four injection strategies.

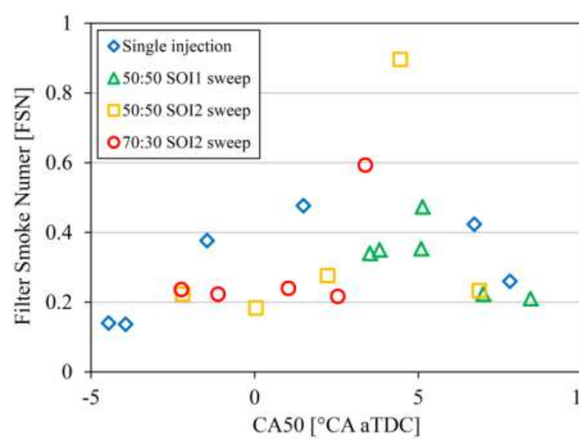

Figure 16. Smoke emissions (in Bosch Filter Smoke Number) vs. CA50 for injection timing variations in 4 different injection strategies.

No clear differences can be found between the strategies, as all levels are very low, for many of the measured points near or below upcoming legislated levels.

It is general practice to plot smoke emissions versus nitrogen oxides emissions to see how different strategies behave with respect to the common NOx-soot trade off. The latter is commonly experienced in conventional diesel combustion, where a measure to decrease one, leads to an increase in the other. From Figure 17 it can be seen that the present injection strategies largely escape from this trade-off, with both smoke and NOx emissions being near zero.

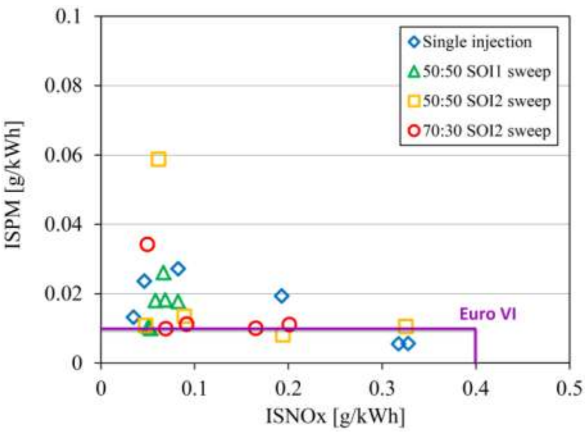

Figure 17. Smoke emissions (in g/kWh) vs. nitrogen oxides for injection timing variations in 4 different injection strategies. Euro VI emission levels depicted by purple box.

For both smoke and NOx emissions, therefore the chosen injection strategy does not have a big impact. For smoke, also the combustion phasing has a minor effect, whereas for nitrogen oxides the emission levels increase with advancing combustion, but remain reasonably low.

Maximum Pressure Rise Rate

The last parameter which is compared for the injection strategies is the maximum pressure rise rate. Absolute levels of the maximum pressure rise rate depend amongst other things on the calculation method (i.e. forward, backward or central differentiation), and the sampling rate. The effect of the latter is shown in Figure 18, where for one injection strategy the maximum pressure rise rates computed with full resolution (i.e. 0.1 degCA) are compared to lower sampling rates.

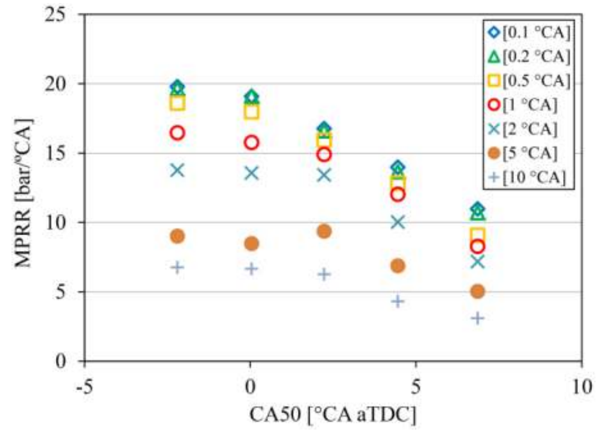

Figure 18. Maximum pressure rise rate vs. CA50, for variation of the second injection and computed with 7 different frame sizes.

Furthermore, to verify that the acquired maximum pressure rise rate is sampled at an appropriate timing, the crank angle at which this maximum pressure rate occurs is plotted versus CA50, see Figure 19.

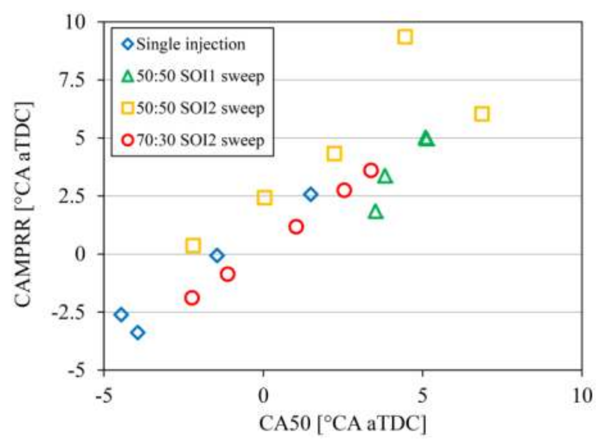

Figure 19. The phasing of the maximum pressure rise rate vs. CA50 for injection timing variations in 4 different injection strategies. The pressure signal is used at full resolution.

Largely premixed combustion can lead to unacceptably high pressure rise rates and the chosen injection strategy might have an effect on this. From Figure 20, it shows that the maximum pressure rise rate is largely independent from the chosen injection strategy, but mainly depends on the resulting combustion phasing.

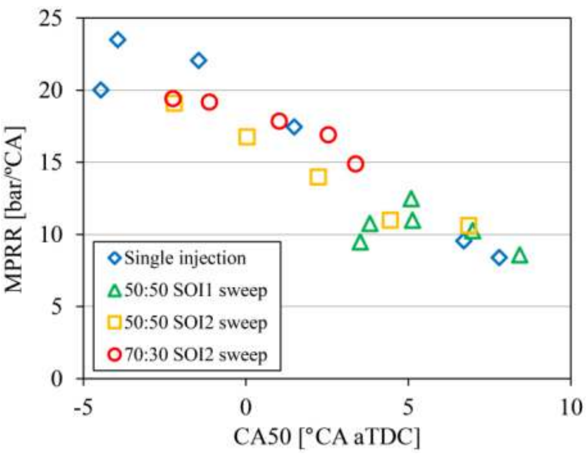

Figure 20. Maximum pressure rise rate vs. CA50 for injection timing variations in 4 different injection strategies. The pressure signal is used at full resolution.

For all strategies pressure rise rates are comparable with conventional combustion regimes with same combustion phasing. The pressure rise rates are efficiently suppressed by the high dilution rates used. To keep the pressure rise rate at an acceptable level the chosen injection strategy is thus not that important. However, CA50 is, so it is desired to have an injection strategy that offers a wide range of control. With such an injection strategy, combustion phasing can be shifted such that the maximum pressure rise rate always stays below acceptable levels.

CONCLUSIONS

Reactivity Controlled Compression Ignition through in-cylinder blending gasoline and diesel to a desired reactivity has previously been shown to give low emission levels, combined with a simultaneous efficiency advantage. To determine the possible viability of the concept for on-road application, a determination of the control space of injection parameters with respect to combustion phasing has been presented. The four injection strategies under investigation can be described as:

1. Single diesel injection; early diesel injection timing variation

2. 50:50: SOI1 variation; diesel injections divided 50:50, vary first

3. 50:50: SOI2 variation; diesel injections divided 50:50, vary second

4. 70:30: SOI2 variation; diesel injections divided 70:30, vary second

The following points were noted:

• A variation in the timing of the first or second diesel injection has an opposite effect on combustion phasing. The response is reasonably linear, but the sensitivity S_{ID} of the first injection is weak.

• The sensitivity S_{ID} of the late injections is positive and larger in absolute value compared to the early injections variation strategy. Furthermore is the sensitivity correlates with the amount injected in the second injection. As such the 80-10-10 is most favorable.

• For the single injection strategy, an early combustion phasing is necessary to have temperatures high enough for combustion to be as complete as possible. Even at the earliest combustion phasing, with the highest maximum temperatures, combustion efficiency is relatively low, caused by crevice volumes.

• All three double injection strategies give very poor combustion efficiency. For these double injections, injection pressure had to be lowered to 500 bar, and together with the very short injection this results in a low combustion efficiency. One hypothesis is that the length of the diesel spray penetration is too short to ignite all of the premixed gasoline-air mixture. Another hypothesis is that bad vaporization and mixing might hamper the ignition of the premixed charge.

• The dual-fuel concept, even with double injections, is known to be possible of producing very high efficiencies. Because of the low combustion efficiency, this is not achieved in the present investigation.

• Because of the high dilution level and largely premixed mixture, all present injection strategies break with the common soot-NOx trade-off, with both smoke and NOx emissions being near or below upcoming legislated levels.

• For all strategies pressure rise rates are comparable with conventional combustion regimes with same combustion phasing. The pressure rise rates are efficiently suppressed by the high dilution rates used. The maximum pressure rise rate is largely independent from the chosen injection strategy, but mainly depends on the resulting combustion phasing. Therefore it is desired to have an injection strategy that offers a wide range of control.

REFERENCES

1. Inagaki, K., Fuyuto, T., Nishikawa, K., Nakakita, K. et al., "Dual-Fuel PCI Combustion Controlled by In-Cylinder Stratification of Ignitability," SAE Technical Paper 2006-01-0028, 2006, doi:10.4271/2006-01-0028.

2. Kokjohn, S., Hanson, R., Splitter, D., and Reitz, R., "Experiments and Modeling of Dual-Fuel HCCI and PCCI Combustion Using In-Cylinder Fuel Blending," *SAE Int. J. Engines* 2(2):24-39, 2010, doi:10.4271/2009-01-2647.

3. Hanson, R., Kokjohn, S., Splitter, D., and Reitz, R., "An Experimental Investigation of Fuel Reactivity Controlled PCCI Combustion in a Heavy-Duty Engine," *SAE Int. J. Engines* 3(1):700-716, 2010, doi:10.4271/2010-01-0864.

4. Splitter, D., Hanson, R., Kokjohn, S., and Reitz, R., "Reactivity Controlled Compression Ignition (RCCI) Heavy-

Duty Engine Operation at Mid-and High-Loads with Conventional and Alternative Fuels," SAE Technical Paper 2011-01-0363, 2011, doi:10.4271/2011-01-0363.

5. Splitter, D., Kokjohn, S., Rein, K., Hanson, R. et al., "An Optical Investigation of Ignition Processes in Fuel Reactivity Controlled PCCI Combustion," *SAE Int. J. Engines* 3(1): 142-162, 2010, doi:10.4271/2010-01-0345.

6. Leermakers, C., Luijten, C., Somers, L., Kalghatgi, G. et al., "Experimental Study of Fuel Composition Impact on PCCI Combustion in a Heavy-Duty Diesel Engine," SAE Technical Paper 2011-01-1351, 2011, doi: 10.4271/2011-01-1351.

7. Leermakers, C., Van den Berge, B., Luijten, C., Somers, L. et al., "Gasoline-Diesel Dual Fuel: Effect of Injection Timing and Fuel Balance," SAE Technical Paper 2011-01-2437, 2011, doi:10.4271/2011-01-2437.

8. Manente, V., Johansson, B., Tunestal, P., and Cannella, W., "Influence of Inlet Pressure, EGR, Combustion Phasing, Speed and Pilot Ratio on High Load Gasoline Partially Premixed Combustion," SAE Technical Paper 2010-01-1471, 2010, doi:10.4271/2010-01-1471.

9. Manente, V., Tunestal, P., Johansson, B., and Cannella, W., "Effects of Ethanol and Different Type of Gasoline Fuels on Partially Premixed Combustion from Low to High Load," SAE Technical Paper 2010-01-0871, 2010, doi: 10.4271/2010-01-0871.

CONTACT INFORMATION

C.A.J. Leermakers
Combustion Technology
Department of Mechanical Engineering
Eindhoven University of Technology
P.O. Box 513, Gem-Z 3.136
5600 MB Eindhoven
The Netherlands
T +31 40 247 2393
F +31 40 243 3445
C.A.J.Leermakers@tue.nl
www.combustion.tue.nl

ACKNOWLEDGMENTS

This project was funded by the Dutch Technology Foundation STW (the technical sciences division of NWO) and the Technology Programme of the Ministry of Economic Affairs. DAF Trucks N.V., Shell Global Solutions, Avantium Chemicals B.V. and Delphi are also acknowledged for their contributions to the project. The authors kindly appreciate the support of the technicians of the Eindhoven Combustion Technology group: Bart van Pinxten, Hans van Griensven, Gerard van Hout and Theo de Groot.

The Engineering Meetings Board has approved this paper for publication. It has successfully completed SAE's peer review process under the supervision of the session organizer. This process requires a minimum of three (3) reviews by industry experts.

ISSN 0148-7191

Positions and opinions advanced in this paper are those of the author(s) and not necessarily those of SAE. The author is solely responsible for the content of the paper.

SAE Customer Service:
Tel: 877-606-7323 (inside USA and Canada)
Tel: 724-776-4970 (outside USA)
Fax: 724-776-0790
Email: CustomerService@sae.org
SAE Web Address: http://www.sae.org
Printed in USA