Analysis of Dual-Fuel CNG-Diesel Combustion Modes Towards High Efficiency and Low Emissions at Part Load

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ANALYSIS OF DUAL-FUEL CNG-DIESEL COMBUSTION MODES TOWARDS HIGH EFFICIENCY AND LOW EMISSIONS AT PART LOAD

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KEYWORDS – CNG, dual-fuel, RCCI, pilot-ignition, energy balance

ABSTRACT

Previous research carried out by the authors described the role of dual-fuel combustion efficiency on overall engine efficiency at high diesel substitution ratio. At low loads, high diesel substitution ratios result in excessive total unburned hydrocarbon emissions due to bulk flame quenching, and consequently lower substitution ratios in combination with intake air heating, throttling and increased turbulence should be adopted (or even full diesel operation) in order to meet emission legislation targets. At those operating points where combustion efficiency is not a limiting factor, engine operation can be really flexible and different options can be implemented. Conventional dual-fuel (CDF) pilot-ignition and reactivity controlled compression ignition (RCCI) combustion have been evaluated and compared based on emissions, heat transfer and efficiency. The results presented in this paper show that CDF is a really robust combustion mode but efficiency and NOx emissions are not optimal when substitution ratio is not maximized. Diesel injection close to TDC promotes fuel rich areas and NOx is formed in near-stoichiometric regions surrounding the diesel jets, so diesel fuel amounts should be kept to a minimum in order to reduce NO_X emissions and heat losses. However, excessively low fuel quantities turn out in combustion phasing control and ignition quality problems and consequently high hydrocarbon emissions. Efficiency at low substitution ratio is reduced due to the effect of fuel trapped in crevices and excessive combustion temperatures, resulting in high heat losses through combustion chamber walls. For these reasons, RCCI combustion mode allows a more efficient use of CNG at high substitution ratios than CDF because of high reactivity fuel dispersed over the entire combustion chamber, including squish volume, while NO_X emissions are sharply reduced compared to less premixed combustion modes.

INTRODUCTION

Conventional spark ignition CNG engines are limited in terms of efficiency because they are operated at low compression ratio. For this reason, pilot-ignition dual-fuel CNG-diesel engines have potential for greater efficiency because of the higher compression ratio available on production compression ignition engines nowadays. Instead of using a spark plug, natural gas is ignited via a short diesel injection close to TDC position. That event triggers the main combustion event and controls combustion phasing. This concept (conventional dual-fuel, CDF) can be seen as a hybrid between diffusion combustion (compression ignition engines) and premixed combustion (spark ignition engines). The total amount of each fuel used will define which type of combustion is predominant. The concept of substitution ratio (SR) used in this paper is defined as:

$$SR\ (\%) = \frac{\dot{m}_{CNG} \cdot LHV_{CNG}}{\dot{m}_{CNG} \cdot LHV_{CNG} + \dot{m}_{diesel} \cdot LHV_{diesel}} \times 100$$

Previous studies performed by the authors [1] proved that combustion efficiency plays a dominant role at low load operation. Flame quenching is the main source of hydrocarbon emissions at low loads [2, 3, 4]. Operation at high substitution ratio requires simultaneous control of intake temperature and combustion phasing in order to generate high enough combustion temperatures to achieve low total unburned hydrocarbon emissions (TUHC). In particular, methane emissions are the main concern because it has 25 times higher global warming potential than carbon dioxide [5]. The introduction of exhaust gas recirculation (EGR) makes these controls even more important [6]. Oxidation of methane in the aftertreatment system requires high exhaust temperatures and this condition is hardly reached at low loads [7, 8].

However, at those points where combustion efficiency is not the main limiting factor, dual-fuel combustion is really flexible in terms of possible combustion strategy adopted. Different modes, depending on the degree of premixing and quantity of diesel fuel, can be implemented. Partially premixed compression ignition (PPCI) and reactivity controlled compression ignition (RCCI) were evaluated by the authors at low loads [9]. At mid loads, heat transfer effects are expected to have a higher specific weight on overall engine efficiency since thermal stress on the combustion chamber is higher than at low load operation. For that reason, in this paper an investigation about the effect of diesel premixing on overall engine characteristics is presented. Firstly, the transition between CDF and RCCI, through PPCI, is presented and analyzed. Diesel injection timing and EGR are the main control factors modified during experiments. Secondly, deeper analysis of the effect of substitution ratio (via diesel fuel injection quantity) and EGR on CDF combustion process is described.

EXPERIMENTAL SETUP

The experiments have been carried out on a 4-cylinder Volvo D4 2 liter diesel engine, modified for dual-fuel operation. The original turbochargers have been removed and replaced by a new exhaust manifold and backpressure valve, in order to have full control over intake and exhaust pressures. The CNG injection system consists of 4 Keihin 73cc CNG injectors, which inject the CNG directly in the intake ports via an adaptor plate placed between the ports and the original intake manifold. Figure 1 shows the intake manifold and the adaptor plate used. Gas pressure is fixed to 5 bar (abs). CNG is only injected while the intake valve is open in order to minimize cylinder-to-cylinder variations. The engine has no aftertreatment system. Table 2 contains most relevant engine and fuel specifications.

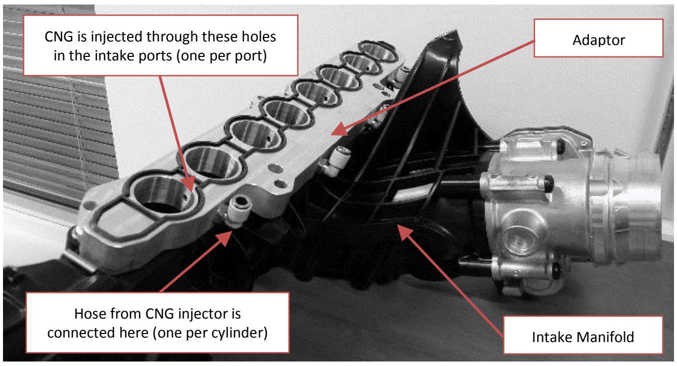

Figure 1: Adaptor plate for CNG injectors attached to intake manifold. Each injector supplies CNG to two different ports (one injector per cylinder)

The engine is controlled via a LabVIEW based program developed at the Division of Combustion Engines at Lund University. High resolution data (every 0.2 CAD) is sampled via a NI PXI-FPGA 7854R. Slow signals are logged via a NI PXI-6221 multifunction data acquisition card while temperatures are measured with a NI PXI-4353 card. Diesel direct injectors, high pressure diesel pump and diesel common rail dump valve are controlled using NI 9751 Direct Injector Driver Module kits. CNG injectors are controlled via an “in-house built” peak and hold driver.

Engine type	Volvo D4
Displacement [cm^3]	1969
N˚ of Cylinders	4
Compression ratio	15.8
Bore x Stroke [mm]	82 x 93.2
Valves	4
Diesel fuel system	Denso G4S (direct injection)
CNG fuel system	Keihin CNG 73cc (port injection)
Max Power [diesel, kW]	133
Max Torque [diesel, Nm]	400

	CNG		Diesel
Type [mol-% for CNG composition]	CH_4 C_2H_6 C_3H_8 C_4H_{10} CO_2 Others	88.8% 6.02% 2.45% 0.99% 1.11% 0.63%	MK1
Lower Heating Value [MJ/kg]	48.5		45.7
H/C ratio	3.70		2.33
Molecular Weight [g/mol]	18.4		170
AFR_{STOICH}	16.9		15.2

Table 1: Engine and Fuel specifications

In-cylinder pressure has been acquired using four AVL GH14P pressures sensors, using the existing standard glow plug bores. In-cylinder pressures and torque are measured every 0.2 CAD and 150 cycles were saved and averaged for every operating point under steady state conditions. Exhaust emissions (TUHC, CH_4, CO, CO_2, O_2, NO_X and CO_2 intake) were measured by an AVL AMA i60 Exhaust Measurement system and soot values were acquired using an MSSplus AVL Micro Soot Sensor. Brake values were calculated using a torque sensor HBM T40. System pressures are captured using Keller slow pressure sensors. Air and CNG flows are measured using Bronkhorst F-106CI and F106AI respectively (direct thermal mass flow measurement). Diesel flow was measured using a Sartorius MSE balance.

Heat transfer in the combustion chambers was measured by acquisition of different temperatures on the different oil and coolant galleries surrounding the combustion chambers. Results presented here described only piston oil and combustion chamber coolant heat losses. EGR cooling circuit was modified to be independent from main engine cooling path and oil heat losses do not include lubrication of other auxiliaries.

RESULTS

<u>Transition from CDF to RCCI mode</u>

In this section the effects of diesel fuel pilot injection timing on overall combustion characteristics are presented and discussed. RCCI combustion requires prior long mixing of both fuels and low enough combustion temperature to achieve simultaneous reduction of NO_X and soot [8]. For that reason, different EGR levels have been evaluated and their impact on the overall process of reaching RCCI conditions is presented here. Starting from CDF combustion type, diesel injection timings was advanced until excessive diesel dilution resulted in combustion instabilities. During the process to setting conditions for RCCI conditions, 90% and 85% substitution ratios with 18% EGR were tested. However, these conditions did not allowed diesel injection targets out of piston bowl. Diesel fuel became too diluted in the combustion chamber and its autoignition was not transferred into self-sustained flame propagation, so unstable conditions made engine running impossible. For that reason, substitution ratio was reduced to 70% via higher diesel flow, promoting more powerful ignition sources in the combustion chamber. The engine has been operated at Lambda-CNG=2 and 35°C intake temperature for all points presented in this section. Previous studies performed by the authors [1] pointed out that these settings allowed higher efficiency than stoichiometric operation without excessive penalties on exhaust emissions and maintaining enough exhaust energy for efficient methane aftertreatment. Main engine operating conditions are presented in Table 2.

Diesel Common Rail Pressure [bar]	1250
CNG Common Rail Pressure [bar]	4.6
Intake Manifold Pressure [bar]	1.40
Exhaust Manifold Pressure [bar]	1.55
Coolant Temperature [°C]	85
Engine Speed [RPM]	1500
Engine Load [bar $IMEP_G$]	10

Table 2: Common settings for all points analyzed in this section

Under the conditions described above, diesel injection timing has been varied, promoting a transition from CDF, through PPCI and finally RCCI conditions. As it can be observed in Figure 2, if diesel ignition timing is advanced early enough in the cycle, despite a slight increase in TUHC, NOx emissions can be substantially reduced and combustion efficiency increased due to diesel fuel dispersed over the entire combustion chamber, including the squish volume. Methane emissions tend to represent a lower portion of TUHC for early timings than for typical CDF. When diesel is only injected into the piston bowl, methane emissions represent around 75% of TUHC, while for early injection timings their weight goes down to 55% (case EGR=17%).

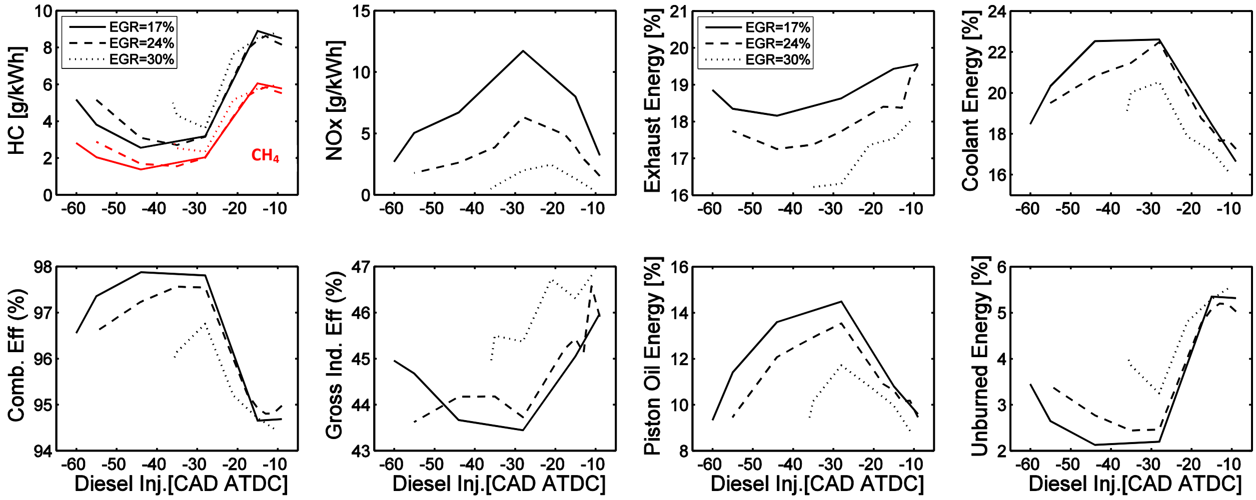

Figure 2: Emissions, efficiencies and energy balance for SR=70% transition from CDF towards RCCI combustion at different dilution levels

Although the highest EGR levels translate into the highest gross indicated efficiency and the lowest NO_X emissions levels, maximum diesel fuel premixing with the air-CNG mixture is limited by combustion stability. With lower EGR ratio, sustained flame propagation can remain even if diesel fuel is further diluted. This makes possible low temperature combustion, maintaining reasonable high combustion efficiency over a wider diesel injection window.

In general, advanced diesel injection timings increase heat losses and decrease overall efficiency. However, if high enough premixing of diesel fuel is achieved (in this case, -60 CAD ATDC diesel injection timing), heat losses in the combustion chamber (coolant, piston oil, etc.) can be highly reduced, resulting in overall high gross indicated efficiency again. In the upcoming sections, this operating point is considered representative of RCCI combustion.

If CDF and RCCI are compared in terms of energy balance, unburned fuel is reduced in the RCCI case since CNG fuel trapped in crevices is now probably oxidized, when before combustion phasing and turbulence optimization were the only way to reduce crevice effect. Secondly, energy transferred to engine coolant is higher, probably linked to pre-reactions taking place during the end of the compression stroke that do not appear during CDF. This phenomena can be observed in the heat release comparison presented in Figure 3. Finally, no significant trend appears between the different combustion modes in terms of piston oil and exhaust energy. These pre-reactions are probably linked to the lower RCCI gross indicated efficiency compared to the CDF case shown in Figure 2, since they represent heat release not transformed into mechanical work. For that reason, combustion phasing in RCCI combustion plays an essential role for achieving high efficiency, which combined with difficulties to control combustion phasing in chemical kinetics driven combustion, makes controllability of this combustion mode really challenging. On the other hand, it offers significant benefits in terms of methane emissions, which in combination with high enough exhaust energy for efficient aftertreatment (exhaust temperature is over 400°C for all cases presented here) and low NO_x levels, makes this concept suitable for low emissions targets.

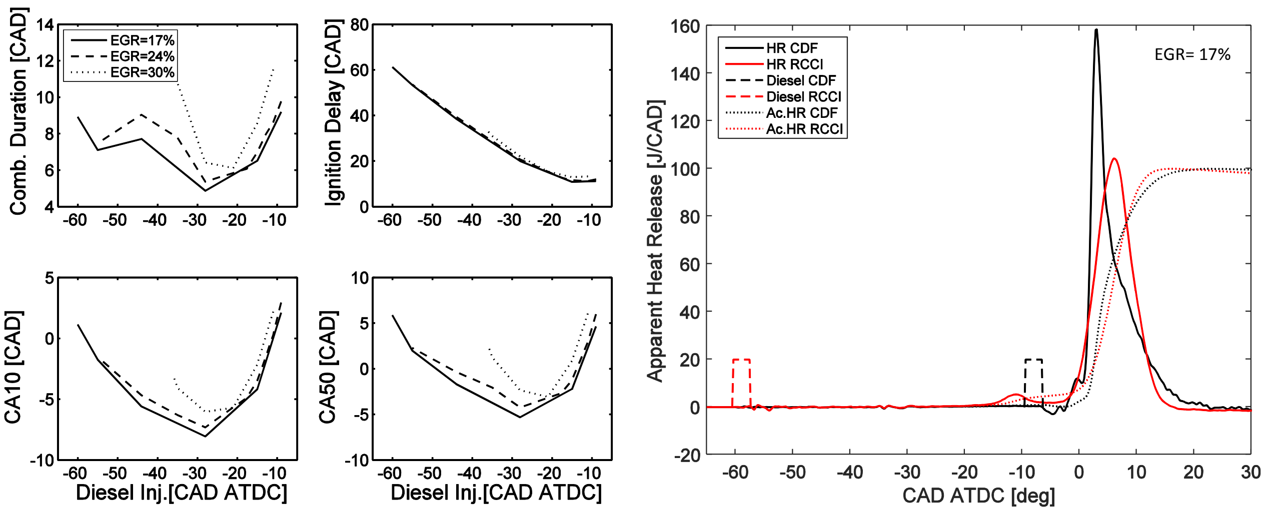

Figure 3: Heat release characteristics for SR=70% comparison between CDF and RCCI combustion

Pilot-ignition dual-fuel combustion stays as a really robust combustion mode, with lower requirements in terms of controllability and valid for high efficiency CNG engines. It is important to highlight that CDF seems, according to these results, more energy efficient than RCCI. Basically, CDF combustion is usually short and centered just after TDC, so effective expansion rate is maximized (high compression engine). Considering that diesel fuel contributes only to 30% of total energy input, although CDF combustion starts at the end of the compression stroke, the temperature raise linked to the diesel combustion is probably significantly lower than in conventional diesel combustion at same loads. For all these reasons, further investigations have been performed in order to evaluate the effect of substitution ratio on the emissions, energy balance and overall efficiency of the engine operated under CDF combustion.

Towards High Substitution Ratio CDF

In this section the effects of pilot diesel fuel quantity on the ignition of a fixed air-CNG mixture are presented and discussed. Dilution via EGR has been also analyzed in order to identify characteristics and limitations of this strategy. Lambda-CNG=2 and 35°C are maintained here too. One of the most important benefits of CDF is its simplicity in terms of combustion phasing control. For high efficiency targets, combustion phasing can be controlled via diesel injection timing, until ignition delay is so long that premixed combustion dominates. On the other hand, for low NO_X/soot emission targets, diesel plays still an important role on overall emissions [1]. For this reason, substitution ratio should be maximized, in order to reduce combustion temperatures. However, excessively low diesel fuel quantities can turn out in poor ignitability and flammability of the main air-CNG mixture, specifically for high EGR dilution levels, turning out in high TUHC emissions. These aspects are presented in Figure 4 and Figure 5.

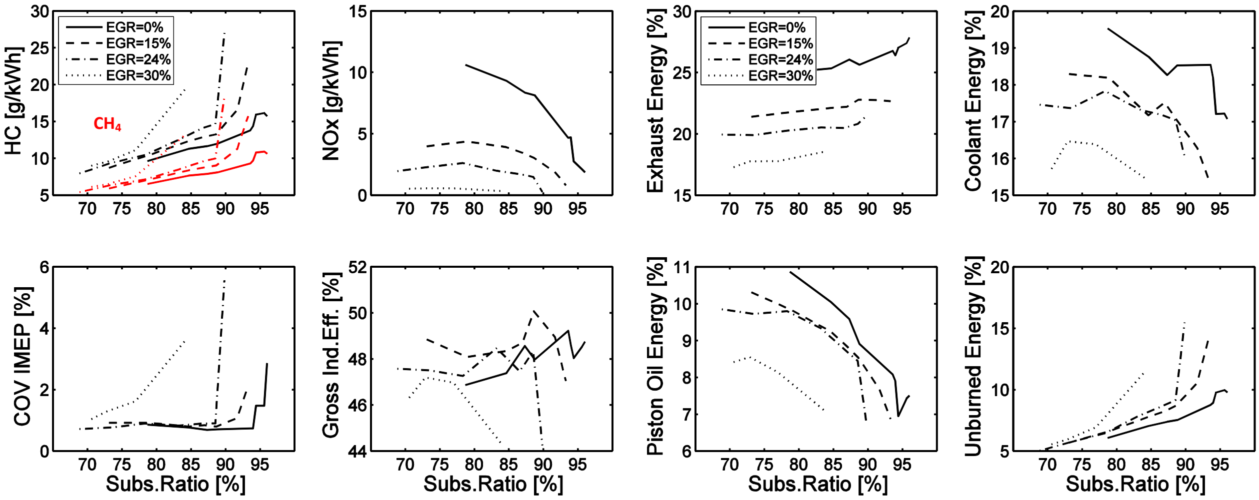

Figure 4: Emissions, efficiencies and energy balance for transition from CDF 70 to 95% SR at different dilution levels

Several characteristics should be highlighted from Figure 4. First, methane emissions account for 67 to 68% of total hydrocarbon emissions for all cases. This means that for the highest substitution ratio cases, this parameter does not fully explain TUHC emissions (89% methane content CNG used). Therefore, diesel fuel oxidation still seems to play an important role on overall hydrocarbon emissions, even over 90% SR. Heat losses percentage sharply decrease for high substitution ratio cases and at the same time combustion efficiency deteriorates. This shows the important role of ignition quality in pilot-ignition dual-fuel engines.

EGR dilution highly affects engine operating range. For the 30% EGR cases, substitution ratios over 85% results in poor combustion stability and misfiring events. As expected, high dilution results in significantly lower NO_X emissions. However, if high substitution ratio operation is desired, lower dilution levels will be necessary. Under these conditions, combustion efficiency quickly deteriorates when diesel fuel quantity is reduced. Pilot-ignition dual-fuel engines are usually really robust in terms of combustion stability because of diesel multi-hole injectors installed (shorter flame distances compared to spark ignition CNG engines). This translate in really low COV_{IMEP} values until misfiring, when COV increases rapidly, as it is observed in Figure 4 bottom left plot.

In terms of gross indicated efficiency, 15% EGR turns out to be optimal due to significantly lower heat losses. Despite higher combustion efficiency, non-diluted mixtures have significantly higher heat losses and exhaust energy which consequently reduce engine efficiency compared to the 15% EGR case, where despite slightly lower combustion efficiency, overall efficiency increases. Gross indicated efficiency remains over 48% (50% peak) until combustion stability deteriorates.

Figure 5 summarizes the effects of reduced diesel fuel quantity on the heat release process. As was introduced before, combustion phasing was only controlled via diesel injection timing (5 CAD ATDC desired). However, diesel timing cannot be the only control action when ignition delay is higher than 14 CAD if constant CA50 is desired.

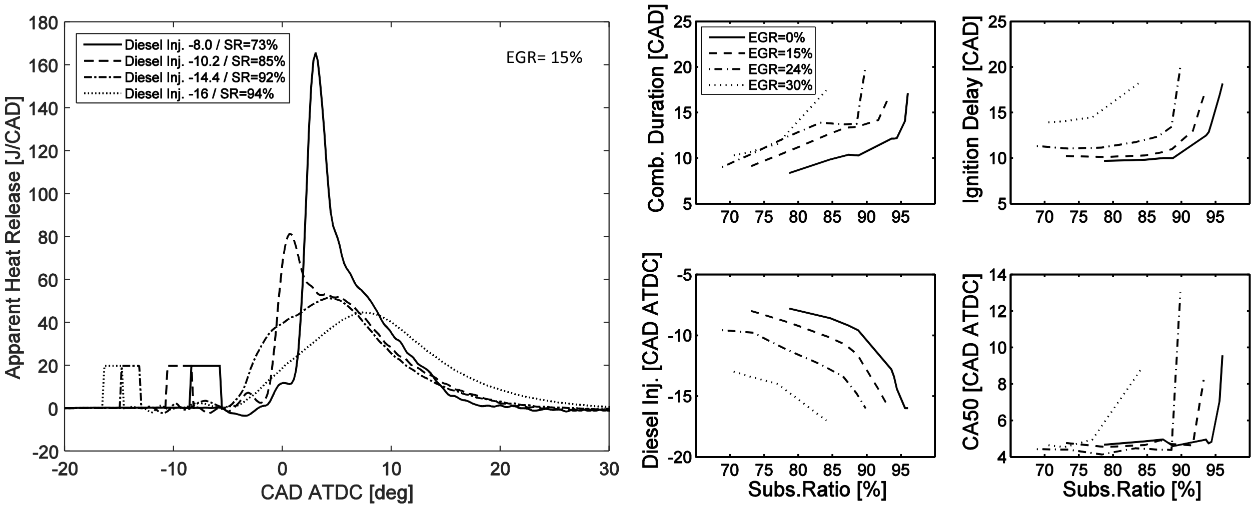

Figure 5: Heat release characteristics for different diesel injection timing [left] and EGR levels [right]

When diesel injection is advanced, combustion is not controlled any longer only by diesel ignition after injection. Contrarily, combustion starts to be mainly driven by the degree of premixing between both fuels, resulting in lower combustion robustness and higher TUHC. In order to apply effective combustion phasing control, other parameters should be selected in combination with diesel injection timing. For these experiments, injection timings before -15 CAD ATDC resulted in too long ignition delay. This situation represents the turning point between pilot-ignition dual-fuel combustion, where diesel injection in CI engines is used as the spark plug for SI engines, and dual-fuel premixed combustion modes, where approximately *1:1* relation is stablished between diesel injection timing and ignition delay (kinetically driven combustion, see Figure 3-ignition delay plot).

It is important to highlight that the maximum substitution ratio achieved during the CDF experiments has been 96.5% (using standard production diesel injectors). High turbulence (swirl) and higher intake temperatures could increase further this limit by improving diesel ignition and enhancing flame propagation. However, when using really short diesel injection durations, instabilities linked to injector non-linear behavior and shot-to-shot variations increase the need of more accurate and faster cylinder-to-cylinder and fuel metering control.

CONCLUSION

In this paper, investigations about the effects of diesel injection timing and substitution ratio (SR) on engine efficiency, heat losses and exhaust emissions in a production light duty diesel engine operated in dual-fuel CNG-diesel mode are presented. Experiments were performed under steady-state conditions at 10 bar $IMEP_G$ approximately. Different dilution levels via EGR, diesel injection timings and diesel fuel quantities were tested and analyzed. The main goal was to further understand heat transfer in conventional dual-fuel (CDF) combustion versus high substitution ratio reactivity controlled compression ignition (RCCI) combustion and how it is affected by different operating parameters. High SR was set as requirement in order to maximize CNG CO_2 reduction potential.

Main conclusions drawn by the authors from this study are:

- RCCI combustion has potential for equally low nitrogen oxides and reduced methane emissions from compared to CDF combustion. Substitution ratio should be maximized in order to keep NO_X emissions controlled in CDF mode.
- However, dilution via EGR limits maximum substitution ratio achievable. At the same time, appropriate diesel fuel quantity is necessary to maintain high combustion efficiency and low heat losses simultaneously without combustion quality deterioration.
- Engine efficiency can be maximized via EGR and substitution ratio simultaneous optimization, while keeping low nitrogen oxides emissions. Hydrocarbon emissions remains as an issue in dual-fuel combustion, but high enough temperature at mid-load is present in the exhaust for efficiency aftertreatment of methane. Diesel fuel dispersed over the entire combustion chamber (squish volume and topland included) helps reducing the specific weight of methane on overall hydrocarbon emissions.
- Finally, substitution ratio affects controllability of CDF combustion phasing via diesel injection timing. Under high SR conditions (>90%), ignition delay is excessively long so control strategies involving intake temperature and swirl control should be implemented for correct combustion phasing and consequently overall engine efficiency.

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