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USE OF ADAPTIVE INJECTION STRATEGIES TO INCREASE THE FULL LOAD LIMIT OF RCCI OPERATION

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ABSTRACT
Dual-fuel combustion using port-injection of low reactivity fuel combined with direct injection of a higher reactivity fuel, otherwise known as Reactivity Controlled Compression Ignition (RCCI), has been shown as a method to achieve low-temperature combustion with moderate peak pressure rise rates, low engine-out soot and NO_x emissions, and high indicated thermal efficiency. A key requirement for extending to high-load operation is moderating the reactivity of the premixed charge prior to the diesel injection. One way to accomplish this is to use a very low reactivity fuel such as natural gas. In this work, experimental testing was conducted on a 13L multi-cylinder heavy-duty diesel engine modified to operate using RCCI combustion with port injection of natural gas and direct injection of diesel fuel.

Engine testing was conducted at an engine speed of 1200 RPM over a wide variety of loads and injection conditions. The impact on dual-fuel engine performance and emissions with respect to varying the fuel injection parameters is quantified within this study.

The injection strategies used in the work were found to affect the combustion process in similar ways to both conventional diesel combustion and RCCI combustion for phasing control and emissions performance. As the load is increased, the port fuel injection quantity was reduced to keep peak cylinder pressure and maximum pressure rise rate under the imposed limits. Overall, the peak load using the new injection strategy was shown to reach 22 bar BMEP with a peak brake thermal efficiency of 48%.

INTRODUCTION
Strict new standards for CO_2 emissions and fuel consumption of heavy-duty engines and vehicles [1], coupled with the business interests of heavy-duty vehicle operators to reduce total cost of ownership, drive an increased push for high-efficiency heavy-duty motive power. Stringent emissions standards also mandate low system-out (including engine and aftertreatment) levels of gaseous (including NO_x and NMHC) and particulate emissions [2]. Reflecting the need for higher efficiency heavy-duty engines, key objectives of the Department of Energy (DOE) SuperTruck program include the demonstration of a heavy-duty engine with 50% brake thermal efficiency and establishment of a pathway to 55% brake thermal efficiency [3]. Many previously developed low-temperature diesel combustion strategies produce low engine-out emissions but frequently at the penalty of lower engine efficiency. However, increasingly effective aftertreatment systems for heavy-duty diesel engines have enabled higher overall engine efficiency [4] – combustion concepts designed to produce low engine-out emissions must therefore also deliver comparable efficiencies to remain competitive. Dual-fuel combustion concepts were developed as a method of addressing the limitations of earlier low-temperature combustion strategies while providing high engine efficiency.

By introducing two fuels of differing reactivity into the combustion chamber separately, overall fuel reactivity levels can be adjusted with operating conditions. Further, separating the fuel delivery, port-injecting the low reactivity fuel and direct injecting the high reactivity fuel, has been shown to create in-cylinder reactivity stratification, slowing combustion and decreasing the pressure rise rate, a primary load-limiting factor for many HCCI-type combustion strategies [5,6]. Dual-fuel strategies using gasoline and diesel have shown potential for high efficiency, both quantified by indicated thermal efficiency measurements on single-cylinder engines [5,6] and by brake thermal efficiency measurements on multi-cylinder engines [7,8,9]. By using a low compression ratio, it has been shown that a dual-fuel concept can cover the entire operating range of a heavy-duty engine [10]; efficiency with the low compression ratio, however, may not offer advantages over current single-fuel conventional diesel engines.

Different variations on the dual-fuel concept have been explored, with key differences being the delivery conditions of the diesel injection. The Reactivity Controlled Compression Ignition (RCCI) concept uses very early injection timing with relatively low injection pressure – start of combustion and subsequent combustion phasing is largely kinetically driven [5,6]. A similar concept, denoted as 'CM2' has also been reported in literature [8,11,12]. However, by using a more conventionally timed, high pressure, single diesel injection, greater control over the combustion phasing can be achieved and, depending on the thermal boundary conditions, potentially an expansion of the upper load limit of the operating range can be accomplished [7,8,9].

Despite all of this work on kinetically controlled RCCI, it has been typically limited to the low to mid load range. Thus, there is still a need to reach full load, i.e. 20+ bar BMEP. Recently, work [13-17] has looked at combining both kinetically and mixing controlled combustion strategies as a method to increase the peak load capability of advanced combustion modes. Such strategies, called Adaptive Injection Strategies (AIS), have used a mixed mode or even a two stage combustion strategy where a fraction of the fuel is injected early for premixed auto-ignition combustion followed by a late, mixing controlled combustion event to reach the target load without increasing the maximum pressure rise rate (MPRR). Practically, methods to achieve this have been to use either two fuels and two direct injections or one fuel and two direct injectors [13-17]. In these two modes, results (at mid load conditions) achieved low NO_x and reasonable efficiencies, but could have excessive PM, which was very sensitive to the cylinder conditions at the SOI, namely the temperature and O_2 concentration. The goals for this work are to improve on these past results by using two fuels with PFI and a single direct injection in order to realize full load operation with RCCI.

NOMENCLATURE

AFR	Air Fuel Ratio
AHRR	Apparent Heat Release Rate
AIS	Adaptive Injection Strategies
AKI	Anti Knock Index
ATDC	After Top Dead Center
BMEP	Brake Mean Effective Pressure
BTE	Brake Thermal Efficiency
CA	Crank Angle
CDC	Conventional Diesel Combustion
CN	Cetane Number
CO	Carbon Monoxide
DDFS	Dual Direct Fuel Stratification
DI	Direct-Injection
DOE	Department of Energy
EGR	Exhaust Gas Recirculation
EGT	Exhaust Gas Temperature
EPA	Environmental Protection Agency
FSN	Filter Smoke Number
FTE	Friction Thermal Efficiency
GMEP	Gross Mean Effective Pressure
GTE	Gross Thermal Efficiency
HC	Hydrocarbon
HCCI	Homogeneous Charge Compression Ignition
HRR	Heat Release Rate
IMEP	Indicated Mean Effective Pressure
LHV	Lower Heating Value
LTC	Low Temperature Combustion
MAF	Mass Air Flow
MON	Motor Octane Number
MHRR	Maximum Heat Release Rate
MPRR	Maximum Pressure Rise Rate
NMEP	Net Mean Effective Pressure
NMHC	Non Methane Hydrocarbons
NO_x	Oxides of Nitrogen
NTE	Net Thermal Efficiency
OEM	Original Equipment Manufacturer
PCP	Peak Cylinder Pressure
PFI	Port-Fuel-Injection
PHI	Equivalence Ratio
PM	Particulate Matter
PMEP	Pumping Mean Effective Pressure
PON	Pump Octane Number
PPM	Parts Per Million
PRR	Pressure Rise Rate
PTE	Pumping Thermal Efficiency
RCCI	Reactivity Controlled Compression Ignition
RON	Research Octane Number
SCR	Selective Catalytic Reduction
SI	Spark Ignition
SOI	Start of Injection
TDC	Top Dead Center
US	United States
VGT	Variable Geometry Turbine
WHR	Waste Heat Recovery

EXPERIMENTAL CONFIGURATION

Instrumentation

Low speed measurements of engine operating parameters, including temperatures, pressures, emissions, and fuel flow are logged at 10 Hz. High speed combustion data were taken using commercially available software, which was also used to post process the pressure data.

Crank-angle resolved measurements of cylinder pressure are collected using Kistler 6125B transducers in all six cylinders at 0.1 crank angle degree resolution for 300 engine cycles. A 3 kHz low-pass filter is used to suppress significant cylinder pressure oscillation due to chamber resonance. In all the results, pressure data from cylinder number 2 were chosen as the most representative cylinder of the overall engine operation.

Gaseous emissions are measured using a standard 5-gas analyzer (Horiba MEXA-7100DEGR), which includes CO_2 measurements in the intake for EGR calculation. An AVL 415 Smoke Meter is used for soot emissions measurements.

Fuel flow rates, for both natural gas and diesel fuel, are measured with Micro Motion CMFS010 Coriolis-type flowmeters. The airflow rate was measured using a laminar flow element.
The test engine used in this study is a MY2010 Navistar MaxxForce 13 six-cylinder heavy-duty diesel engine modified to reach peak thermal efficiency, as per the goals of the SuperTruck program. A schematic of the test engine configuration is shown in Figure 1, along with basic engine specifications in Table 1.

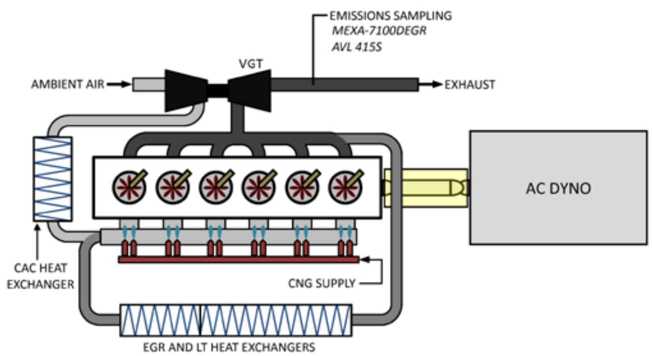

Figure 1. Schematic of test engine configuration.

The SuperTruck engine features an air system configured for high thermal efficiency, utilizing a single stage turbocharger and a dual-pass EGR cooler with high and low temperature stages. Prototype pistons feature a higher than stock compression ratio. A 2500 bar capable common-rail diesel fuel injection system is used with prototype injectors.

Table 1 Engine Specifications.

	Navistar 13L DF
CYLINDERS[-]	6
BORE [mm]	126
STROKE [mm]	166
DISPLACEMENT [L]	12.4
GEOMETRIC CR[-]	>17
DI INJECTION PRESS. [bar]	2500
PFI INJECTION PRESS. [bar]	6

Several alterations to the production engine were made to enable and enhance low-temperature dual-fuel combustion. A prototype variable geometry (VGT) turbocharger was implemented to decrease pumping losses compared to the OEM two stage system. A modified intake manifold is used which incorporates two port fuel injection (PFI) injectors per cylinder. The natural gas fuel is injected at 6 bar above intake manifold pressure directed at the intake runner of each cylinder.

Stock EGR air cooling systems are maintained. A building process water system is used for cooling heat exchangers in the high and low temperature engine coolant loops, and to directly cool the high pressure charge air cooler. Cooling systems were tuned using Navistar internal metrics to mimic heat rejection available in a production vehicle. Intake and exhaust restrictions were likewise adjusted to simulate vehicle air filtration systems along with exhaust aftertreatment and muffler restrictions. Constant restrictions were used across all tested conditions.
Start of injection timing, both as a reported value and used in computation of ignition delay, is determined from ECU command and not directly measured from injector current.

Test Fuels

The dual-fuel operating strategy employed in this study uses two fuels concurrently: a low-reactivity fuel (natural gas) along with a high-reactivity fuel (diesel). The natural gas used in this study was supplied by pipeline, and had a lower heating value of 47.14 MJ/kg, while the diesel was an ultra-low sulfur (ULSD) certification diesel fuel with typical diesel fuel properties and a cetane number (CN) of 44. Relevant properties of the diesel fuel are listed in Table 2.

Table 2. Diesel Fuel Properties.

	Diesel
DENSITY [KG/L]	0.851
NET HV [MJ/KG]	42.8
RON [-]	-
MON [-]	-
CN [-]	44
AROMATICS [%]	28
OLEFINS [%]	1
SATURATES [%]	71
H:C RATIO	1.88
SULFUR [PPM]	10
T_{10} [°C]	214
T_{50} [°C]	259
T_{90} [°C]	317

Operating Conditions and Limits

To constrain the results to more realistic operating conditions, limits were placed on the emissions and engine operation. Peak cylinder pressure (PCP) is limited to 250 bar, while maximum rate of pressure rise (for each cylinder, measured on an unfiltered cylinder pressure signal) is limited to less than 15 bar/deg.
Combustion noise was targeted to be below 90 dBa. This limit is established as an outer boundary based on engine hardware

durability and from the USCAR Advanced Combustion & Emissions Control (ACEC) working group [18].

Since maximum thermal efficiency is the target for the SuperTruck project, engine-out NO_x emissions targets were relaxed to 15 g/kW-hr. This value was chosen based on the assumption of the use of a selective catalytic reduction (SCR) system with 97% conversion efficiency in order to meet EPA 2010 emissions standards for heavy-duty on-road engines. Soot emissions are not explicitly limited, but were kept below 1 FSN. Likewise, explicit limits are not placed on CO or HC emissions, but were typically minimized to increase thermal efficiency. Combustion stability was maintained at each operating point, with a COV of engine-averaged IMEP less than 2% for all tested conditions.

RCCI Operating Strategy

To achieve dual fuel combustion, the low-reactivity natural gas was introduced via port injection and the high-reactivity diesel fuel was introduced via direct injection into the cylinder during the compression stroke. A schematic of typical port and direct injection timings is included in Figure 2.

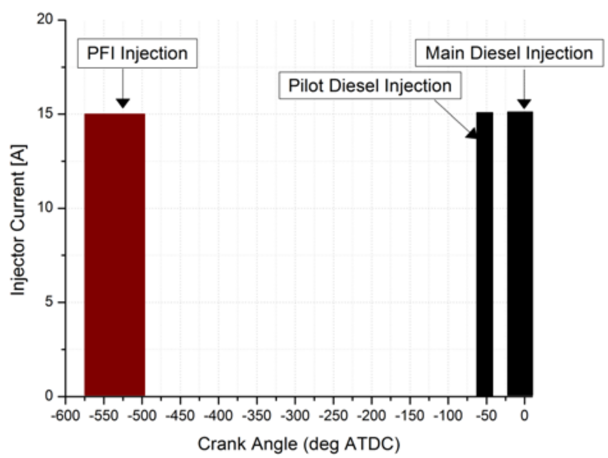

Figure 2. Representative injection strategy with PFI injection during intake stroke and the dual direct diesel injections.

The single natural gas port fuel injection used a constant start of injection timing of 570°BTDC for all test cases. Single and double pulses were used for the diesel injection, with the pilot injection phased early in the compression stroke (-55°ATDC) and the main injection near top dead center (0-15°ATDC). The double injection strategy is used to around 15 bar BMEP where the pilot injection is shut off as the premixed charge is reactive enough to ignite without it and only the main direct injection was then used to deliver the diesel fuel.

Again, this strategy differs from past results and more closely follows [13-17], which were designed to reach high loads while keeping noise and PCP within acceptable limits. The late injection also offers control of the combustion phasing using SOI timing. Cooled exhaust gas recirculation (EGR) was used for some of the tested operating conditions, varying from 0-42%. To minimize efficiency losses from excessive pumping work, the EGR level was controlled using VGT vane and EGR valve position to offer the maximum intake pressure at the desired EGR rate for each operating condition while keeping the PMEP below 0.5 bar.

RESULTS

As stated in the introduction, the work involves exploring various injection parameters at a baseline operating condition for the best thermal efficiency and emissions performance. This baseline operating condition was derived from work by Wissink [17]. From these parameter sweeps, a load sweep was then performed at a constant engine speed of 1200 rpm. Figure 3 shows the cylinder pressure and AHRR for cylinder 2 and Table 3 gives the specific operating parameters of the baseline condition.

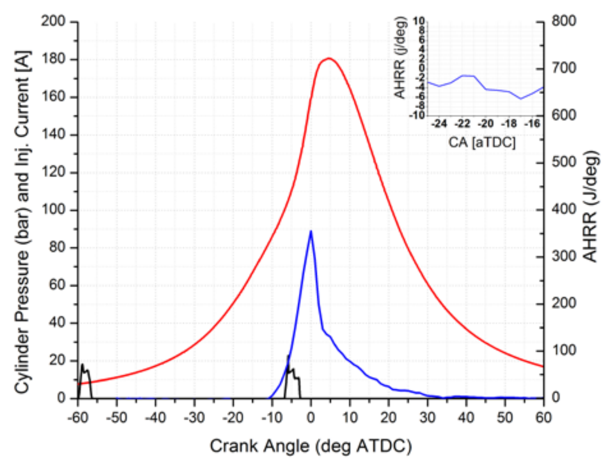

Figure 3 Baseline operating condition cylinder pressure and apparent heat release rate

The baseline condition was chosen as it allows for a wide variety of operation conditions without reaching the physical limits of the engine and is in the middle of the engine load range.

Table 3 Baseline Operating Condition Parameters

BMEP [BAR]	10
SPEED [RPM]	1200
EGR [%]	36
PHI [-]	0.60
P_{INT} [BAR]	2.0
T_{INT} [°C]	45.5
PFI FRACTION [%]	66
RAIL PRESSURE [BAR]	1770
PILOT/MAIN SOI [°ATDC]	-58/-5
PILOT FRACTION [-]	0.22
PFI MASS [MG/INJ]	73.5
PILOT MASS [MG/INJ]	3.3
MAIN MASS [MG/INJ]	14.75
CA50 [DEG ATDC]	0.8
MPRR [BAR/DEG]	12.7
COMB. NOISE [DBA]	86

PFI Mass Fraction

One of the main benefits of RCCI is the ability to modify the global fuel reactivity to control combustion phasing. One way to do that is to vary the ratio of the port injected fuel to the direct injected fuel mass. In this test, the baseline condition PFI mass fraction is varied from 53-77%.

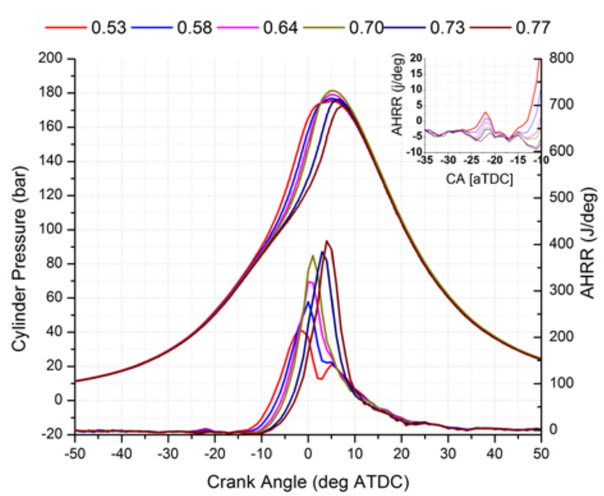

Figure 4 PFI mass fraction sweep cylinder pressure and apparent heat release rate

Figure 4 shows that increased PFI fraction delays combustion phasing. Due to the late main injection timing, the combustion phasing is less sensitive to PFI fraction as [5,6], but the trend with delayed phasing is the same. As the PFI fraction is decreased from 77%, it is also possible to see the start of combustion advance and eventually have to two peaks where the natural gas and diesel apparent heat release rate (AHRR) separate at 53% PFI fraction. Figure 5 shows that, as the PFI fraction increased, HC and CO emissions increased due to the lower temperatures seen during the expansion stroke. This might also explain the PM increase, as the PM oxidation rate slowed. NO_x emissions were seen to slightly decrease as the PFI mass fraction increased, due to the higher quantity of premixed fuel.

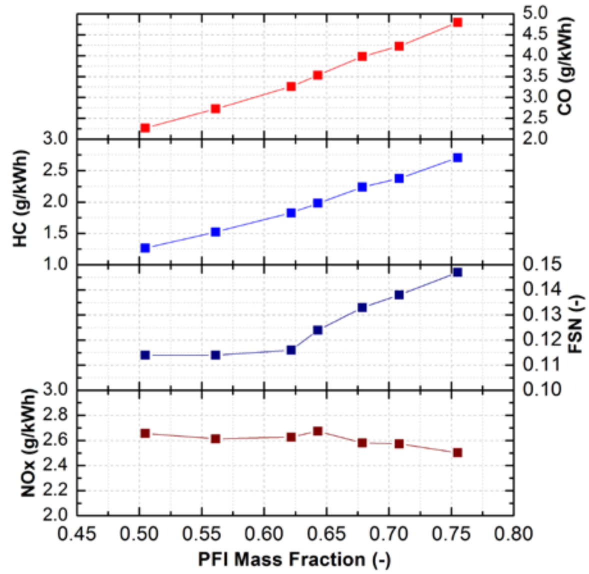

Figure 5 Emissions performance for the PFI mass fraction sweep

Brake thermal efficiency (BTE) peak occurred with slightly more PFI fuel than the baseline case of 66% due to the slightly later CA50 of 3°ATDC as shown in Figure 6. MPRR and combustion noise are seen to have a minimum value between 55 and 60% PFI fraction. This minimum occurs because as the PFI fraction is lower than 55% the start of combustion is too advanced and as the PFI fraction is higher than 60% the combustion had a higher maximum AHRR.

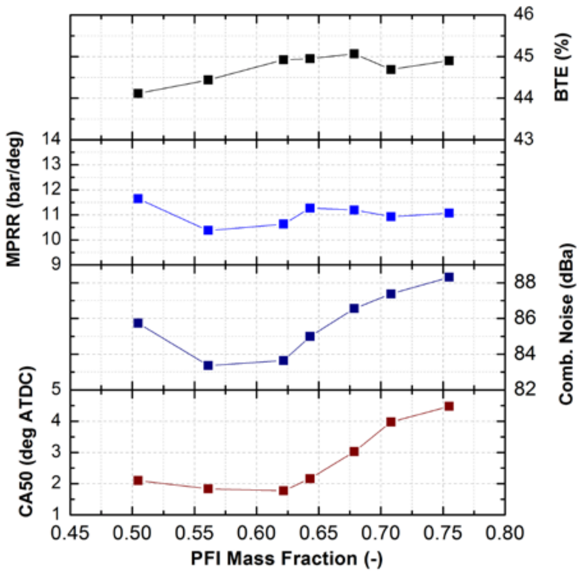

Figure 6 Thermal efficiency and combustion performance for the PFI mass fraction sweep

Pilot SOI

Next, tests varied the pilot SOI timing from -78 to -38°ATDC. Similar to previous RCCI results, earlier injections were beneficial in reducing NO_x [5,6]. As the injections advanced and the charge became more premixed, NO_x and PM decreased while HC and CO were nearly constant, as shown in Figure 7. The peak BTE was found at -68°ATDC, slightly earlier than the baseline case of -58°ATDC as shown in Figure 8.

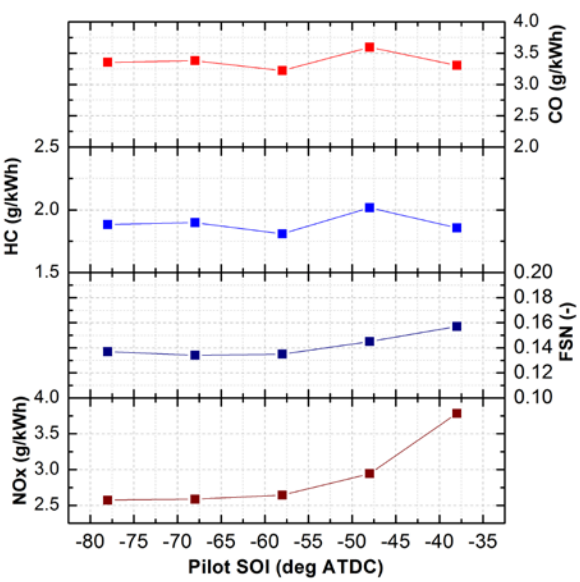

Figure 7 Emissions performance for the pilot SOI timing sweep

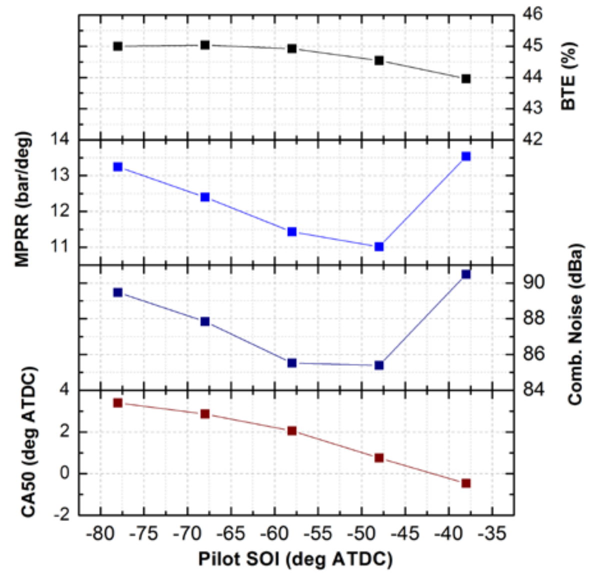

Figure 8 Thermal efficiency and combustion performance for the pilot SOI timing sweep

Similar to the PFI fraction sweep, the minimum noise and MPRR occurred around -58 and -48°ATDC. At -38°ATDC, the premixed portion of the combustion event can be seen to start advancing and separates from the later direct injected diesel fuel, as shown in Figure 9. Then as the pilot SOI is advanced to -78°ATDC, the combustion phasing was delayed, but the maximum AHRR increases, thus increasing combustion noise.

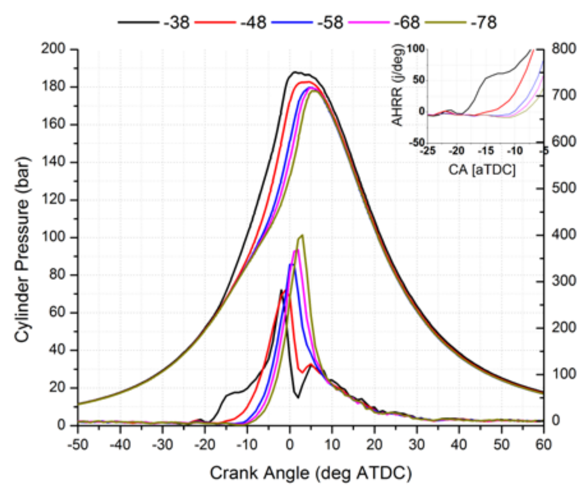

Figure 9 Pilot SOI timing sweep cylinder pressure and apparent heat release rate

Main SOI

Different to other RCCI injection strategies [5,6], the near TDC main injection offers direct control of CA50 and increases the stability of combustion [19] due to lower variation in the local equivalence ratios (and thus ignition delays) at the time of ignition. Figure 10 shows the cylinder pressure and AHRR for the main SOI timing sweep.

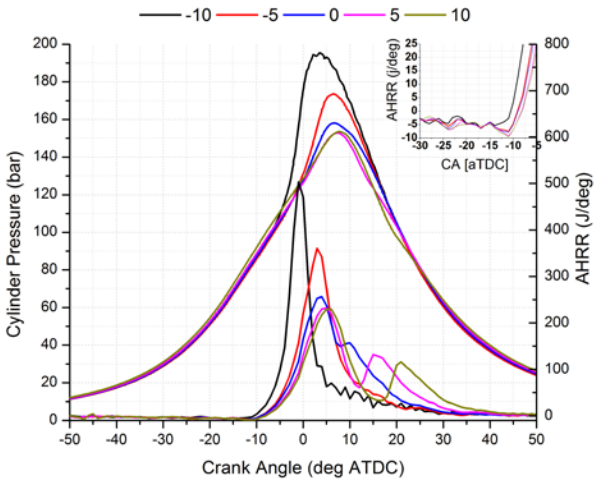

Figure 10 Main SOI timing sweep cylinder pressure and apparent heat release rate

Figure 10 and Figure 11 show that the combustion phasing is controlled by the main SOI timing. Figure 10 shows that as SOI was advanced, the maximum AHRR increases and as it was delayed, the combustion events associated with the premixed (PFI and pilot injections) and main direct injection start to become separate as the AHRR separates, lowering thermal efficiency. Peak BTE was found at the baseline timing of -5°ATDC, which was a compromise of where the combustion noise was too high and where the phasing was too late and the efficiency was poor.

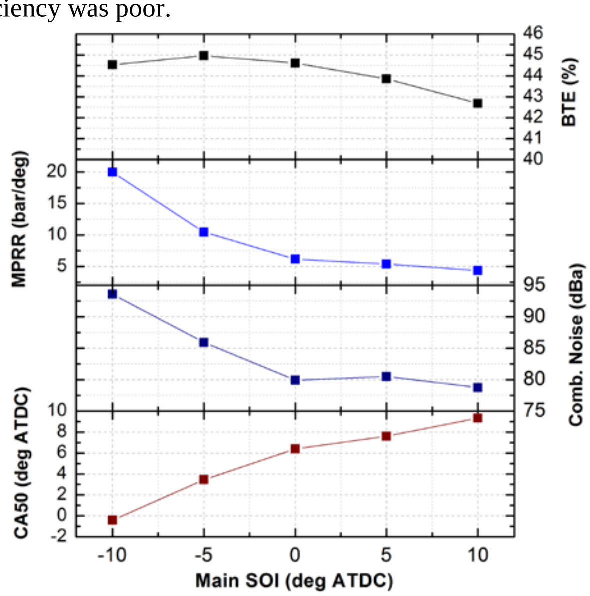

Figure 11 Thermal efficiency and combustion performance for the Main SOI sweep

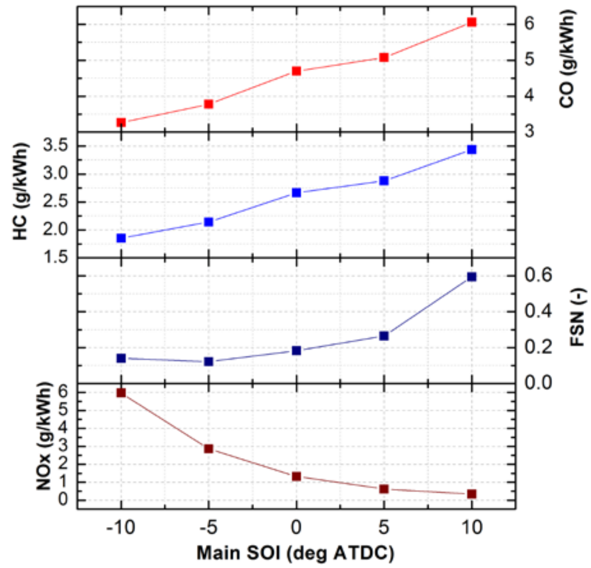

Figure 12 Emissions performance for the Main SOI timing sweep

Figure 12 shows that at the later main SOI timings, there is a recognizable NO_x/PM trade-off, as in conventional diesel combustion (CDC). At the late timings, the HC and CO increase due to the lower temperatures seen during the expansion stroke, which could also explain the increased PM as the oxidation rate slows with decreased temperatures.

Pilot Fraction

As with PFI fraction sweep, the direct injection split fraction effects the mixture preparation and emissions performance. Injected fuel quantities and splits were calculated using the injector calibration in the OEM engine controller. Figure 13 shows that increased amounts of pilot fuel advances combustion before TDC, separating the premixed and diffusion combustion AHRR.

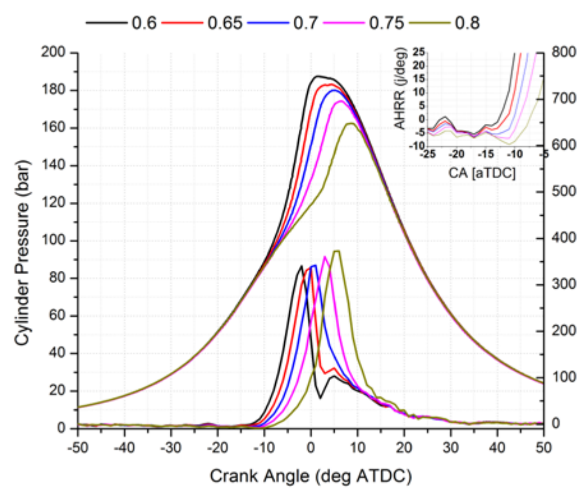

Figure 13 Pilot fraction timing sweep cylinder pressure and apparent heat release rate

Figure 14 shows HC and CO increase as the premixed charge leans the reactivity stratification increases. NO_x was not affected. PM reduces slightly with less pilot fuel, possibly from lower PM formation due to the lower cylinder temperatures.

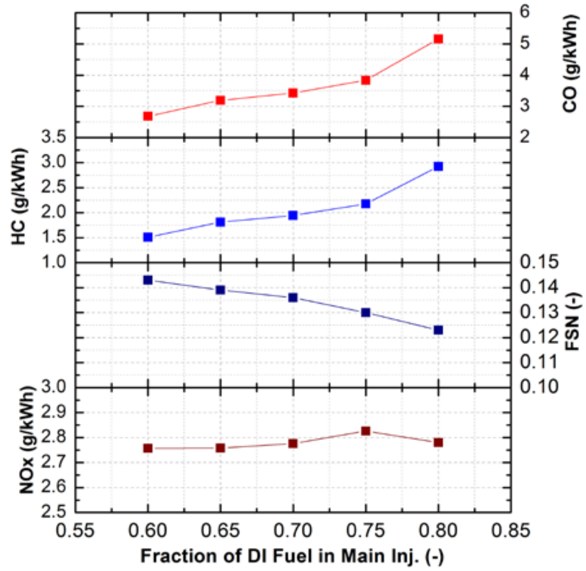

Figure 14 Emissions performance for the pilot fraction timing sweep

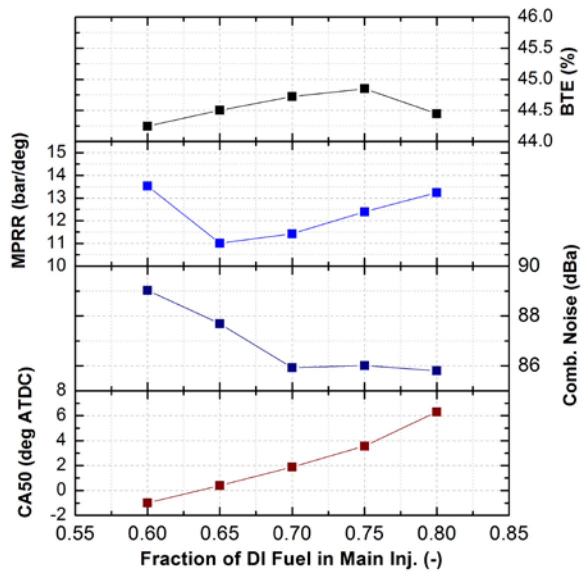

Figure 15 Thermal efficiency and combustion performance for the pilot fraction sweep

In Figure 15, combustion noise and MPRR performance can be seen. Again, the MPRR is at a minimum before the premixed combustion becomes too advanced and where the AHRR was too high. CA50 is delayed with less fuel in the pilot injection, similar to past RCCI results. The peak BTE was found at baseline condition of 75% of the fuel injected during the main injection, which is a compromise between the early and late combustion phasings.

EGR

As with other combustion strategies, EGR was initially used for NO_x and combustion phasing control. Because of this, the combustion phasing was kept constant with changes to the PFI mass fraction during the EGR sweep. Figure 16 shows the cylinder pressure and AHRR for the EGR sweep.

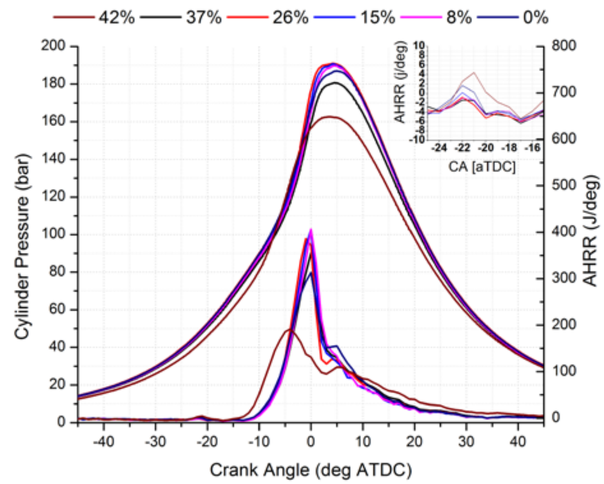

Figure 16 EGR sweep cylinder pressure and apparent heat release rate

Figure 17 shows that HC and CO increase with less EGR and the resulting increased amount of PFI fuel shown in Figure 18. There was a large PM/NO_x trade-off after the baseline point of 36% EGR.

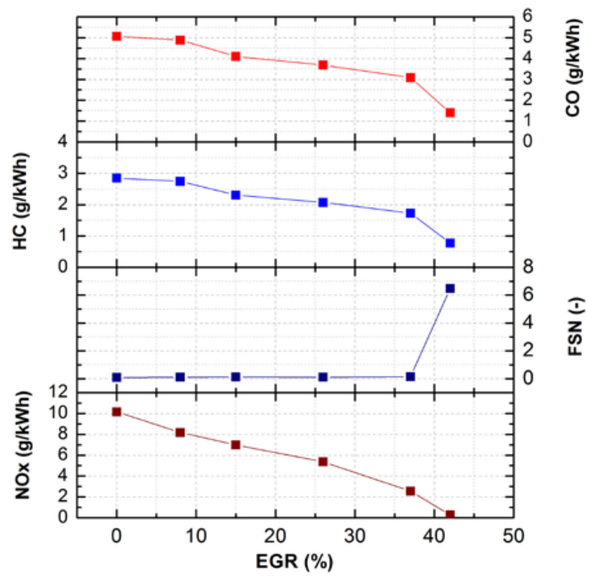

Figure 17 Emissions performance for the EGR timing sweep

Here, it is thought that the balance between PM formation and oxidation is very sensitive to the cylinder temperature and O_2 concentration. As the O_2 concentration drops beyond 36% EGR, the PM exponentially increases. Figure 17 shows the rapid increase in soot from 0.5 to 6 FSN with less than 10%

EGR change. Figure 18 shows the peak BTE and PFI fraction were found at 0% EGR due to the reduced PMEP. Due to the changing PFI fraction, CA50 was kept constant within +/- 0.5 deg, but the 42% point was running poorly and the target CA50 could not be recovered with only changes to CA50. Similarly, combustion noise and MPRR follow the CA50 trend.

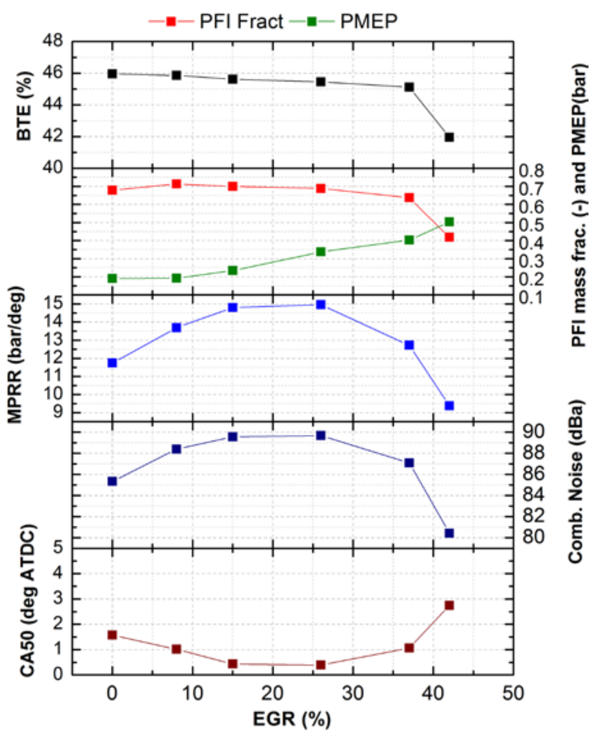

Figure 18 Thermal efficiency and combustion performance for the EGR sweep

Rail Pressure

Similar to other RCCI results, the common rail pressure has been shown to have a significant impact on the combustion event. However, Figure 19 shows there were only slight changes to the combustion phasing and the peak AHRR.

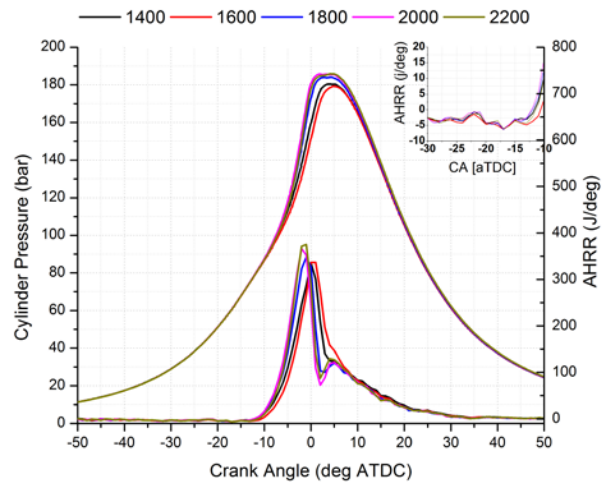

Figure 19 Rail pressure timing sweep cylinder pressure and apparent heat release rate

Figure 20 shows the BTE and combustion metrics from the rail pressure sweep. As the rail pressure was lowered, the combustion phasing delayed and reduced the combustion noise and MPRR. Since the combustion phasing did not change drastically, the BTE was unaffected by the change in rail pressure.

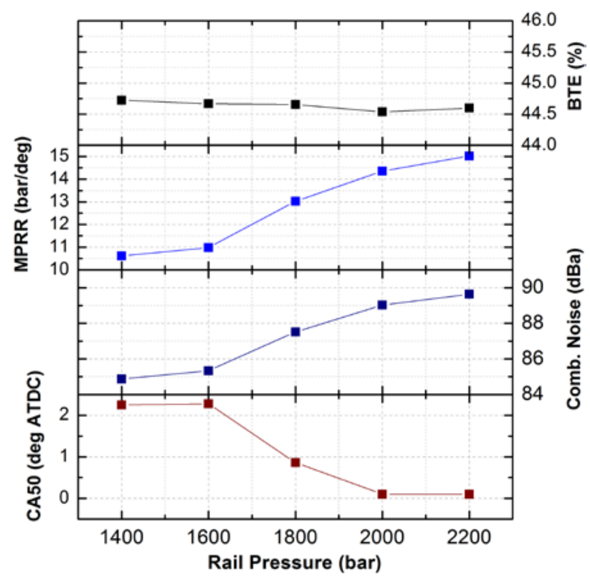

Figure 20 Thermal efficiency and combustion performance for the rail pressure sweep

Figure 21 shows the emissions from the rail pressure sweep. Increased rail pressure advanced phasing and increased the local temperature and equivalence ratios which lead to increased NO_x emissions. As the NO_x increased, the PM decreased, indicating that there was mixing controlled combustion occurring. The HC and CO emissions increase as the NO_x drops. This is caused by the lower temperature and local equivalence ratio, which then cause the PM oxidation rate to slow.

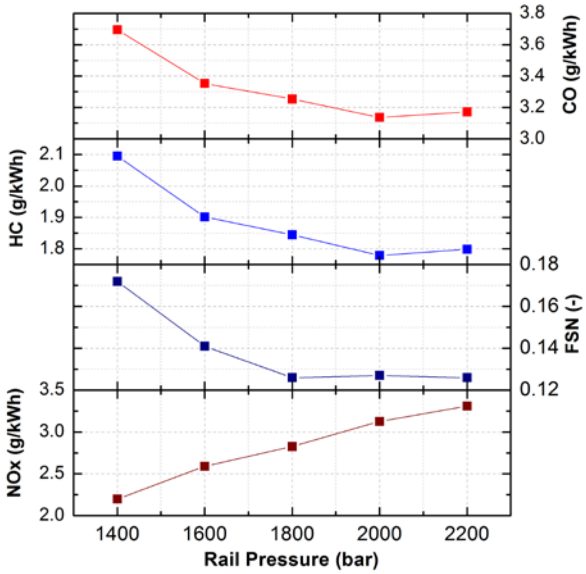

Figure 21 Emissions performance for the rail pressure sweep

Lower rail pressure resulted in lower NO_x with little impact on BTE. This is useful at higher loads to reduce NO_x in order to lessen the impact on urea usage in the SCR system. More work is needed to find the optimum combination of NO_x, urea and BTE for this package. However, care needs to be taken when using EGR as the PM can increase rapidly with low rail pressure, especially at higher loads and temperatures.

Load

Finally, using the best practices developed based on the parameter sweeps, a load sweep was conducted from 5 to 22 bar BMEP. Table 4 gives the operating parameters for the load sweep. Operating conditions for each load were based on the optimum baseline conditions, without EGR. The EGR sweep showed to offer the highest thermal efficiency without EGR. Not using EGR is also helpful to maximize the amount of heat energy in the exhaust where it can be recovered in a WHR system, a potential pathway to meet the SuperTruck project goals for 55% BTE. Additionally, the double direct injection strategy was not employed beyond 10 bar BMEP as the early injection of high reactivity fuel was not needed with the high compression ratio at these elevated loads.

Table 4 Load Sweep Engine Operating Parameters

BMEP [BAR]	5	10	15	20	22
EGR [-]	0	0	0	0	0
PHI [-]	0.36	0.44	0.51	0.59	0.62
P_{INT} [BAR]	1.37	2.0	2.6	2.84	3.0
T_{INT} [°C]	29	39.5	44.5	49	51
PFI FRACTION [%]	73.5	66	54	36	29
RAIL PRESS. [BAR]	1270	1770	1600	1600	1750
PILOT/MAIN SOI [°ATDC]	-58/-5	-58/-5	-/-3	-/-3	-/-4
PILOT FRACTION [-]	0.71	0.22	0	0	0
CA50 [°ATDC]	-0.1	0.8	9.4	8.9	10.1
MPRR [BAR/DEG]	7.5	12.7	13.7	14.7	12.2
COMB. NOISE [DBA]	82	86	91	90.7	88

Figure 22 shows the changes in AHRR where the pilot injection is shut off at 15 bar BMEP. Low loads have a more typical RCCI-like AHRR with early phasing for maximum combustion efficiency. The AHRR then delays to reduce noise, with the pilot injection dropping out at 15 bar. As load further increases, the main injection fuel amount increases, which can be seen in the two stage AHRR above 20 bar BMEP.

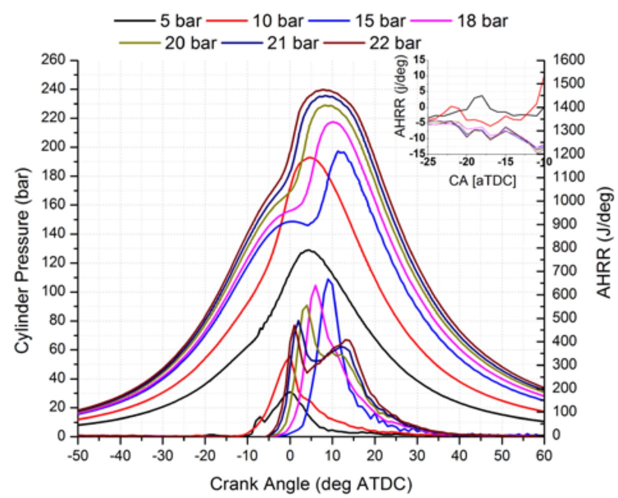

Figure 22 Load sweep cylinder pressure and apparent heat release rate

As load increases, there was an increased amount of mixing controlled combustion. This higher temperature combustion lowered the PM, HC and CO, but also increased NO_x emissions, as shown in Figure 23. The PFI mass fraction also drops with load to keep PCP and MPRR within the 250 bar and 15 bar/deg limits, respectively.

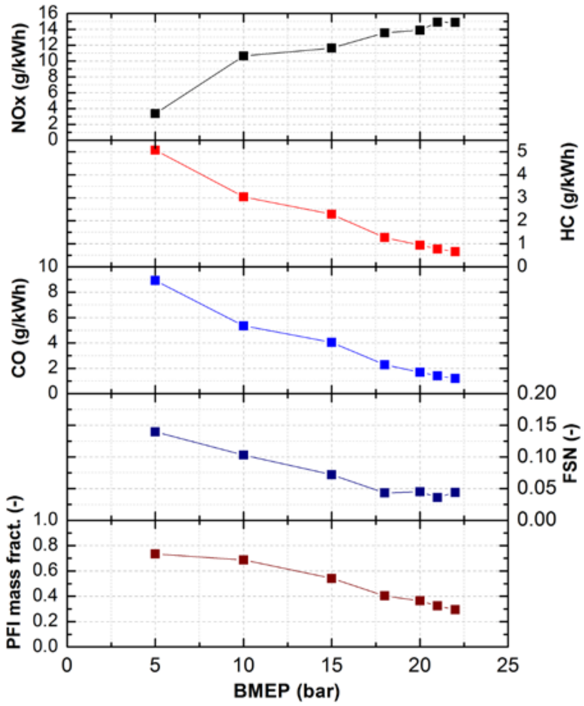

Figure 23 Thermal efficeincy and combustion performance for the load sweep

Figure 24 shows that the combustion noise was less than 92 dBa across the load sweep. Like the PFI mass fraction sweep, CA50 is delayed with load to keep noise and PCP under the limits. Also shown in Figure 24 are the ACEC target guidelines for light-duty combustion noise [18]. While there is further optimization necessary to exactly match the guidelines, the current work shows that it is possible to meet these light-duty noise targets with high thermal efficiency on a heavy-duty engine. Also note that low noise does not follow the expectations for low MPRR, as the high boost and compression ratio increase, the base motoring MPRR increases.

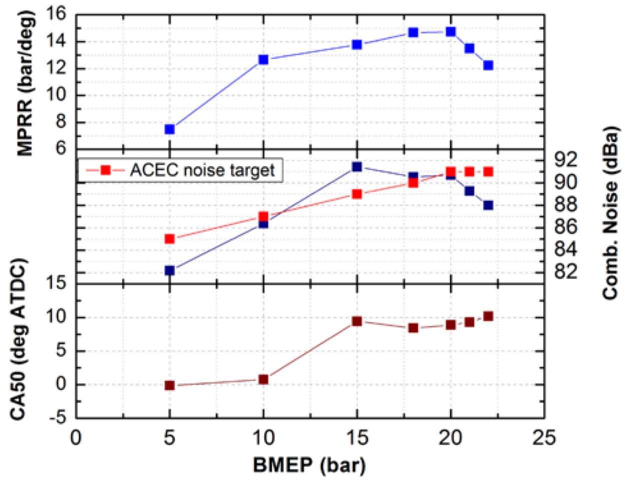

Figure 24 Combustion performance for the load sweep

Figure 25 shows the efficiencies of the load sweep. As one would expect, the pumping thermal efficiency and friction thermal efficiency (PTE and FTE) decrease with load, as they become a smaller fraction of the total load. PTE is defined as the PMEP divided by the FuelMEP and FTE is defined as the FMEP divided by the FuelMEP. Gross thermal efficiency (GTE) and Net thermal efficiency (NTE) peak at 10 bar and drop with increasing load as the charge becomes richer because the intake pressure is reduced to limit PCP. Peak BTE was found at 20 bar BMEP where the GTE/FTE tradeoff is the best. Future work is planned to increase the GTE at 20 bar to the 52% GTE seen at 10 bar in order to meet 50% BTE.

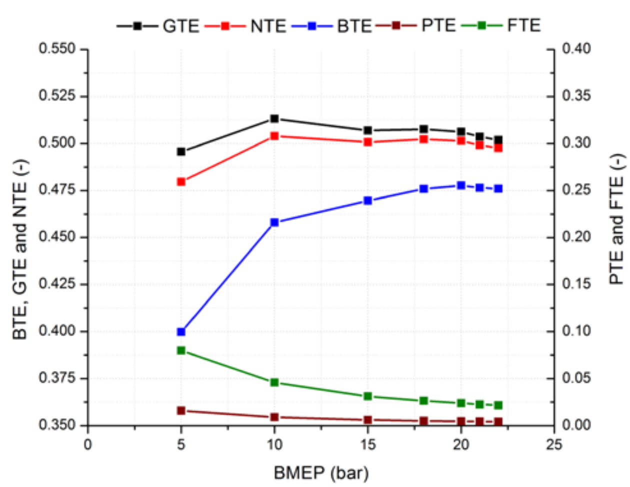

Figure 25 Efficiency performance for the load sweep

Finally, Figure 26 shows that as load increases, heat transfer losses drop and exhaust losses (as defined in [17]) slightly increase. Both of which are good for waste heat recovery (WHR) systems as the energy in the exhaust system has more exergy than the energy in the cooling system. Combustion efficiency was also seen to increase with load.

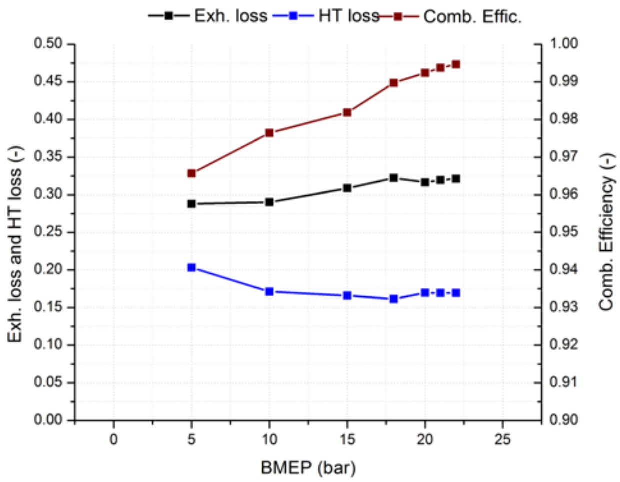

Figure 26 Efficiency and losses for the load timing sweep

DISCUSSION

In order to reach high loads with RCCI at reasonable noise levels, a different operating strategy was needed. By using the strategies proposed in the literature [13-17], a combination of RCCI and CDC modes provided high efficiency combustion with low combustion noise. Note that the current results are an initial investigation into this two stage combustion mode, more work is needed to fully explore the complex combustion interactions and to reach the full potential for thermal efficiency and emissions.

Additionally, the current results were pushed towards high load operation as heavy-duty emissions are weighted towards high load operation. Improved low-load operation is still possible. With the high geometric compression ratio and relaxed NO_x emissions targets, there is room for substantial improvements to the combustion efficiency and thermal efficiency at low-load conditions more typically seen in light-duty vehicles.

One of the enabling technologies for this combustion method is the availability of high efficiency SCR systems, which allow for increased BTE at expense of higher engine-out NO_x. Additionally, relaxing NO_x standards can help many other issues seen with LTC, such as increasing the exhaust enthalpy for WHR, and enabling transient operation (the late main injections help increase combustion stability). The higher combustion temperatures also help lower PM, HC and CO emissions and offer lower combustion noise while maintaining high thermal efficiency.

Finally, due to the high compression ratio, there was reduced PFI mass fraction at high loads, in order to lower PCP and combustion noise. One method to increase the PFI fuel usage is shown in [17], where the low reactivity fuel is direct injected in place of the main diesel injection. Having such engine hardware could increase the low reactivity fuel fraction to almost 99%.

CONCLUSIONS

The use of AIS was applied to RCCI combustion and investigated as a method to allow full load operation in a heavy-duty, multi-cylinder, dual-fuel engine using natural gas and ULSD fuels. Using a baseline engine operating condition, different operating parameters were swept to find their sensitivity on the combustion process. From the results of these tests, the following conclusions are drawn:

Direct injection parameters showed to have sensitivities similar to both CDC and RCCI combustion. Combinations of both combustion mode trends for emissions and performance indicated the mixed combustion mode.

The use of 0% EGR allowed for increased thermal efficiency with reductions in most emissions, except NO_x. However, not using EGR requires a high efficiency SCR system to meet current NO_x emissions targets.

Finally, peak BTE of 48% was demonstrated at 20 bar BMEP with 0% EGR and minimal pumping and friction losses.

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