Injection timing effects on partially premixed diesel-methane dual fuel low temperature combustion

E. Scott Guerry, Mostafa S. Raihan, Kalyan K. Srinivasan, Sundar R. Krishnan¹, Aamir Sohail

*Department of Mechanical Engineering, Mississippi State University,
Mississippi State, MS 39762, USA*

ABSTRACT

Dual fuel low temperature combustion (LTC) strategies are attractive for future internal combustion engines due to their promise of very low engine-out emissions of oxides of nitrogen (NO_x) and particulate matter. In the present work, experimental results for diesel-ignited methane dual fuel LTC on a compression ignition single cylinder research engine (SCRE) are presented. Methane was fumigated into the intake manifold and diesel injection was used to initiate combustion. The engine was operated at a constant speed of 1500 rev/min, and diesel injection pressure was fixed at 500 bar. The start of injection (SOI) of diesel fuel was varied from 260° to 360° (i.e., TDC) to quantify its impact on engine performance and engine-out, indicated-specific emissions of NO_x (ISNO_x), carbon monoxide (ISCO), and unburned hydrocarbons (ISHC), and smoke emissions. The SOI sweeps were performed at different *net* indicated mean effective pressures (IMEPs) of 4.1 and 12.1 bar. Intake manifold pressure and methane percent energy substitution (PES) were fixed at 1.5 bar and 80%, respectively, for 4.1 bar IMEP and at 1.8 bar and 95%, respectively, for 12.1 bar IMEP. For all loads, when SOI was advanced, the longer ignition delays caused the separation between the fuel injection and the combustion events to increase. This was accompanied by a change in the shape of the AHRR curve from a distinct two-stage profile to a smooth, single-stage (almost Gaussian) profile.

¹ Corresponding Author; E-mail: krishnan@me.msstate.edu, T: +1-662-325-1544, F: +1-662-325-7223, Mechanical Engineering Department, Mississippi State University, Mississippi State, MS 39762

Advancing SOI to 300° and beyond yielded minimal engine-out ISNO_x emissions (~ 0.15 g/kWh) at 4.1 bar IMEP and ~ 1.3-1.5 g/kWh at 12.1 bar IMEP). Smoke emissions were negligible (< 0.05 FSN) for all loads and all SOIs. Very high ISHC and ISCO emissions were observed for near-TDC SOI at all loads. The lowest ISHC and ISCO levels occurred for SOIs near 310° and for more advanced SOIs, both emissions increased. High pressure rise rates and the tendency to knock prevented engine operation at intermediate SOIs between 310° and 340° for 12.1 bar IMEP. On the other hand, for both 4.1 bar and 12.1 bar IMEPs, high coefficient of variation of IMEP (> 5%) caused unstable engine operation for SOIs advanced beyond 280°.

Keywords. Injection timing; low temperature combustion; dual fuel; natural gas; NO_x emissions

ABBREVIATIONS

AHRR	Apparent heat release rate [J/deg]
CA 10-90	Crank angle duration between locations of 10% and 90% cumulative heat release [crank angle degrees or CAD]
CA5	Crank angle at which 5% of cumulative heat release occurs [CAD]
CA50	Crank angle at which 50% of cumulative heat release occurs [CAD]
CAD	Crank angle degrees
CO	Carbon monoxide [g/kWh]
COV IMEP	Coefficient of variation of IMEP [%]
EOI	End of injection [CAD]
FCE	Fuel conversion efficiency [%]
HC	Unburned hydrocarbons [g/kWh]

46	HCHO	Formaldehyde [ppm]
47	ID	Ignition delay [CAD]
48	IFCE	Indicated fuel conversion efficiency [%]
49	IMEP	Indicated mean effective pressure [bar]
50	LTC	Low temperature combustion
51	LTHR	Low temperature heat release [J/deg]
52	MPRR	Maximum pressure rise rate [bar/deg]
53	NO _x	Oxides of nitrogen [g/kWh]
54	PES	Percent energy substitution (of gaseous fuel) [%]
55	Q _{LHV}	Mass-averaged lower heating value [kJ/kg]
56	SCRE	Single cylinder research engine
57	SOC	Start of combustion [CAD]
58	SOI	Start of injection of diesel fuel [CAD]
59	TDC	Top dead center
60		

1. INTRODUCTION

Due to environmental and economic factors as well as increasingly strict emissions standards, internal combustion engine design and development has been the topic of much research in recent years. Compression ignition (CI) engines are known for exhibiting higher efficiency and reliability compared to spark ignition (SI) engines. However, CI engines produce high concentrations of oxides of nitrogen (NO_x) and particulate matter (PM) or soot emissions, which are regulated emissions; consequently NO_x and PM reductions are a major focus in CI engine research [1]. Expensive aftertreatment systems have been designed and implemented to meet CI engine exhaust emissions standards. Research shows that it is challenging to simultaneously reduce engine-out NO_x and PM emissions without sacrificing efficiency; however, meeting this challenge will be helpful in reducing the load on or even potentially eliminating some of the aftertreatment systems.

Dual fuel engines have been the subject of considerable research and could be practical alternatives to the widely used diesel and gasoline engines of today. Dual fuel engines need two fuels to produce power, and the combustion process is significantly different than that exhibited by conventional diesel combustion. The primary fuel (usually methane or natural gas) is fumigated a sufficient distance upstream of the intake manifold to allow for mixing with the intake air. The methane-air mixture is introduced into the cylinder during the intake stroke. The secondary (pilot) diesel fuel is directly injected into the cylinder during the compression stroke and compression-ignites, which in turn ignites the surrounding methane-air mixture. One advantage of dual fuel engines is that, when utilizing stock diesel injection systems, they also provide the flexibility to revert to straight diesel mode when necessary. Key roles in the combustion process are played by the relative percentage of methane and diesel in the fuel

mixture, diesel start of injection (SOI), and availability of air [2, 3]. Karim described the dual fuel combustion process as occurring in three distinct stages [4]. After the period between diesel SOI and onset of ignition, known as ignition delay, the diesel spray releases energy and initiates the combustion process. Then, the surrounding methane-air mixture ignites. Finally, the combustion process continues by flame propagation through the remaining lean methane-air mixture.

There are some inherent problems affecting dual fuel engine performance and emissions that need to be addressed. Although dual fuel engines prove advantageous over conventional diesel engines through higher fuel conversion efficiency (FCE) at full load and lower emissions, they tend to exhibit knock at high loads and produce higher unburned hydrocarbon (HC) and carbon monoxide (CO) emissions at low loads [5, 6]. Dual fuel combustion has been studied at advanced SOIs, where homogeneous charge compression ignition (HCCI) combustion can lead to unwanted pressure waves and knock [7]. The HCCI combustion process is characterized by the mixing of fuel and air well before the start of combustion, followed by ignition of the premixed charge as a result of temperature increase in the cylinder during the compression stroke [8, 9]. In the experiments that will be described in the present paper, methane is mixed with air in the engine intake, and at very advanced diesel SOI, there is sufficient time for diesel to mix with the already premixed methane-air charge. Combustion characteristics exhibit “HCCI-like” behavior at these very advanced SOI and could be considered diesel-assisted methane HCCI.

Natural gas, of which methane is the primary component, is an abundant and popular gaseous fuel used in dual fuel engines [5, 10-14]. With diesel-methane dual fuel combustion, unburned methane constitutes a very high fraction of the total HC emissions. Current United States Environmental Protection Agency (EPA) standards regulate the amount of non-methane

hydrocarbon (NMHC) emissions, but future regulations may include limits on methane emissions. Consequently, it is imperative to reduce total HC as well as methane emissions from dual fuel engines.

The focus of the present study is to systematically characterize the effects of diesel SOI on diesel-ignited methane dual fuel low temperature combustion (LTC) over a range of engine loads. Several researchers have examined the importance of diesel SOI on dual fuel combustion. For example, Abd Alla et al. [15] observed lower HC and CO emissions and higher FCE with advancing SOI, but these benefits were also accompanied by higher NO_x emissions. However, their study was limited to an SOI range of 325-330 absolute crank angle degrees (CAD) after gas exchange top dead center (TDC). A similar study with biodiesel-ignited natural gas combustion performed by Ryu [16] showed reduced smoke and NO_x values with advancing SOI; however, this study was also limited to an SOI range 337-349 CAD. Sayin and Canaksi [17] also studied diesel-ignited methane dual fuel combustion in a naturally aspirated single-cylinder CI engine but again within a limited SOI range (327-339 CAD). Polk et al. [18] and Gibson et al. [19] studied diesel-ignited methane and propane dual fuel combustion in a 1.9-liter turbocharged direct-injected engine, which utilized the stock engine control unit (ECU). The performance during those experiments benefited from boosted intake conditions. However, it was not possible to vary diesel SOI or diesel injection pressure due to the use of a mechanical diesel injection system and the stock ECU, which limited the maximum methane percent energy substitution (PES). Lower NO_x emissions and improved FCEs were demonstrated at advanced SOIs by Singh et al. [20] over a wider SOI range (300-345 CAD); however, the HC emissions were high at very early SOIs.

Liu et al. [21] studied the effects of pilot fuel quantity in a diesel-ignited natural gas dual fuel engine and found that dual fuel operation resulted in higher CO emissions. They also observed an average of 30 percent reduction in NO_x emissions as well as increased HC emissions, 90 percent of which were unburned methane. Carlucci et al. [22] presented experimental data from CNG-diesel dual fuel operation to characterize the effects of CNG substitution, diesel injection pressure, and diesel quantity at various engine loads and speeds. They observed higher CO and HC emissions as well as lower NO_x and PM emissions and concluded that the emissions trends were less sensitive to pilot quantity at higher engine speeds. Abd Alla et al. [23] observed reduced HC and CO emissions as the amount of pilot fuel was increased but NO_x emissions were higher and the tendency to knock increased with increasing pilot quantity at high loads. These observations are also corroborated by Krishnan et al. [24] who observed a clear trade-off between HC and NO_x emissions in dual fuel combustion. Papagiannakis et al. [25-27] analyzed diesel-ignited natural gas dual fuel combustion at various natural gas substitutions. They observed an increase in CO and HC emissions as well as a decrease in NO_x and PM emissions as the percentage of natural gas substitution was increased. In another study, Papagiannakis et al. [28] observed improved FCE and reduced CO emissions but higher NO_x and PM emissions with increasing SOI advancement and pilot fuel quantity.

In addition to conventional diesel fuel, other high-cetane pilot fuels have also been explored as ignition sources for dual fuel combustion. For example, Imran et al. [29] compared natural gas engine performance and emissions with diesel and rapeseed methyl ester as pilot fuels and observed slight differences in FCE and emissions. Shoemaker et al. [30] also evaluated biodiesel as a pilot ignition source for methane and propane in a 1.9-liter turbocharged direct-injected engine (with the stock ECU) and concluded that for both biodiesel-ignited methane and

propane dual fuel combustion, NO_x and smoke emissions can be reduced at high methane or propane substitutions with significant increase of CO and HC emissions. They also confirmed that that maximum methane and propane substitution with biodiesel pilot ignition was limited by the onset of partial misfire at low loads (brake mean effective pressure (BMEP) of 2.5 bar) and by the onset of engine knock at higher loads (10 bar BMEP). Park et al. [31] explored biodiesel ignition of bioethanol and gasoline in dual fuel combustion mode and showed lower NO_x and soot (again at the expense of HC and CO) compared to single fuel combustion. They also observed that while premixing gasoline with intake air reduces HC compared to bioethanol premixing, the latter resulted in longer ignition delays, lower peak combustion pressures, and slightly higher IMEP.

The dual fuel LTC strategy presented in this study involves a methane-air mixture that is ignited using diesel pilot sprays. This strategy is similar to the idea that in-cylinder stratification of fuel reactivity between the low-cetane primary fuel and high-cetane pilot fuel can be used to control partially premixed dual fuel combustion [32-37]. Other factors that influence dual fuel combustion characteristics include pilot injection timing, partial pressure of O_2 (availability of O_2) as well as presence of diluents (which affects the specific heat of the in-cylinder mixture). Diluents and in-cylinder mixture composition alter end-of-compression temperatures, thereby affecting the start of combustion (SOC). It is possible to separate the SOI from the SOC by advancing the pilot injection. The separation between SOI and SOC helps to achieve LTC, which results in very low engine-out NO_x and soot emissions. This was demonstrated in the advanced low pilot ignited natural gas (ALPING) dual fuel LTC concept [32, 38-41], which utilized SOIs as early as 300 CAD (i.e., 60 CAD BTDC), to achieve good FCEs and very low NO_x emissions ($< 0.25 \text{ g/kWh}$). However, these benefits were accompanied by high HC and CO

emissions [32, 39]. Hot exhaust gas recirculation (EGR) and intake charge heating were later used to achieve nearly 70 percent reduction in HC emissions and improvements in FCE without sacrificing the NO_x benefits of ALPING dual fuel LTC [40-41]. Kusaka et al. [42] also registered lower HC and NO_x emissions and higher FCE with the use of EGR and heated intake air in diesel-ignited natural gas dual fuel combustion. However, they noted that excessive EGR (> 50 percent) can lead to deteriorated combustion because of lack of oxygen in the intake charge.

The experimental results presented in this work were performed on a single cylinder research engine (SCRE) without EGR. The same SCRE has been used in previous research [43-45] to characterize diesel-ignited gasoline and diesel-ignited methane LTC over a range of diesel injection pressures, intake boost pressures, and PES of gasoline or methane at a constant load of about 5.2 bar IMEP.

2. OBJECTIVE

The objective of the present work is to systematically investigate the effects of diesel SOI (260°-360°) on the performance and emissions characteristics of diesel-ignited methane dual fuel LTC in the SCRE at two representative engine loads, (i.e., net indicated mean effective pressure (IMEP) of 4.1 bar and 12.1 bar) and at a constant engine speed of 1500 rev/min.

3. EXPERIMENTAL SETUP

Figure 1 shows a schematic of the experimental setup and Table 1 lists the relevant engine specifications. The SCRE was coupled to a 250 hp AC regenerative engine dynamometer and a Dyne Systems Inter-Loc V controller, which provided torque and speed measurements and

speed control. Engine load was controlled by adjusting the overall fueling rates of diesel and methane to achieve the desired level of methane PES. A Bosch CP3 common-rail diesel injection pump and a solenoid-actuated injector were used to achieve direct injection of diesel fuel. A Drivven (National Instruments) stand-alone diesel injection driver coupled with CALVIEW software was used to control crank angle-resolved diesel injection. An electronic needle valve (Hanbay, Inc., Model MCM-050AB) was used to control the methane flow rate into the engine intake manifold.

3.1 Data Acquisition

The data acquired in these experiments include: (a) engine synchronous (crank-resolved data such as cylinder pressure and needle lift), and (b) engine-asynchronous (i.e., steady state data such as temperatures, mass flow rates, manifold pressures, etc.). Intake, exhaust, coolant, and oil temperatures were measured using Type-K thermocouples. Coolant and oil inlet temperatures were maintained at 80°C for all experiments. Gaseous and exhaust emissions were measured downstream of the exhaust manifold using an emissions sampling trolley and an EGAS 2M six-gas emissions bench manufactured by Altech Environment S.A. An AVL SESAM i60 FT Fourier Transform Infrared (FTIR) analyzer was used for additional speciation of the gaseous exhaust emissions. Smoke was measured in filter smoke number (FSN) units using an AVL 415S variable sampling smoke meter. An engine exhaust particle sizer (EEPS) spectrometer was also available to measure particle number concentrations and size distributions in the exhaust stream.

LEGEND:

1 – Single Cylinder Engine	10 – Engine Oil Temperature Conditioner
2 – AC Regenerative Dynamometer	11 – Six-Gas Emissions Bench
3 – Coriolis Flow Meter	12 – AVL FTIR
4 – Coriolis Flow Meter	13 – EEPS
5 – Diesel Supply Tank	14 – Smoke Meter
6 – Diesel Level Tank	15 – Intake Tank
7 – Methane Supply Tank	16 – Exhaust Tank
8 – Sonic Flow Orifice	17 – Data Acquisition System
9 – Engine Water Temperature Conditioner	

Figure 1. Experimental setup of the single cylinder research engine

220 **Table 1.** Engine Specifications

Parameter	Value
Engine Type	RSi-130 DV11 single cylinder research engine, compression-ignition
Bore $\times$ Stroke	128 mm $\times$ 142 mm
Connecting rod length	228 mm
Displaced Volume	1827 cm ³
Compression ratio (nominal)	17.1 : 1
Valve train system	4-valve, OHV
Intake valve open [*]	32°
Intake valve close [*]	198°
Exhaust valve open [*]	531°
Exhaust valve close [*]	14°
Diesel fuel injection system	CP3 Bosch common-rail
Methane fuel system	Fumigation into intake manifold
Maximum speed	1900/min

221 ^{*}Note: All valve timings are in absolute crank angles (i.e., 0° $\equiv$ gas exchange top dead center).

222 In-cylinder pressure was measured using a Kistler model 6052C pressure sensor and a

223 Kistler 5010B type charge amplifier. Needle lift was measured using a hall-effect-based needle

224 lift sensor (Wolff Instruments). The in-cylinder pressure and needle lift sensors were phased

225 with respect to crank angle using an incremental shaft encoder (BEI Model XH25D-SS-3600-

226 ABZC-28V/V-SM18) with 0.1 crank angle degree (CAD) resolution (i.e., 3600 pulses per

227 revolution) coupled to the engine crankshaft. For the sake of brevity, absolute crank angle

228 convention will be used in this paper; i.e., 0° refers to gas exchange TDC while 360° refers to

229 compression TDC. An external air compressor (Atlas Copco Model GA75) and a heatless

desiccant air dryer (Atlas Copco Model CD 250) were employed to provide compressed air to the engine intake manifold. Intake air mass flow rate was measured using a calibrated sonic orifice (FlowMaxx Model SN16–SA–235) coupled with measurements of upstream pressure and temperature. To ensure choked flow across the sonic orifice, a minimum inlet-to-outlet pressure ratio of 1.2 was maintained across the orifice using a manual pressure regulator located upstream of the orifice. Intake air temperature was maintained between 27°C and 32°C (constant for a given IMEP) using heating strips mounted in the intake surge tank and a Chromalox temperature controller. Two coriolis mass flow meters (Micro Motion Model CMF025M319N2BAEZZZ) with 0.56 percent and 0.35 percent accuracies (of reading) were used to measure methane (primary) and diesel (secondary) mass flow rates, respectively. Important measured experimental parameters and their accuracies are summarized in Table 2.

Table 2. Accuracies of Experimental Measurements

Data Type	Make/Model	Type	Unit	Accuracy
Temperature	Omega	K-type	°C	±0.75% of reading
Air mass flow	Flowmaxx	Sonic orifice	kg/h	±0.1% of reading
Diesel mass flow	Micro Motion	Coriolis	kg/h	0.35% of reading
Methane mass flow	Micro Motion	Coriolis	kg/h	0.56% of reading
Smoke	AVL 415S	Filter	FSN	0.001 of reading
NO _x and NO	ESA EGAS 2M	CLD	ppm	<1% of full scale
THC	ESA EGAS 2M	FID	ppm	<0.5% of full scale
CO	ESA EGAS 2M	NDIR	%	<1% of full scale
CO ₂	ESA EGAS 2M	NDIR	%	<1% of full scale
Cylinder Pressure	Kistler 6052C	Piezoelectric	bar	±0.005 of reading
Pressures (intake, exhaust, etc.)	Setra Model 209	Capacitance	psig	±0.25% of full scale

3.2 Experimental Procedure

The objective of this study was to investigate the effects of SOI on the performance and emissions characteristics of diesel-methane dual fuel LTC in the SCRE. To achieve this objective, the engine was operated at two constant loads (4.1 and 12.1 bar net IMEPs) and a constant speed of 1500 rev/min while the SOI was varied. Additional experiments were also performed at 6.5 bar IMEP and 1500 rev/min but these results are not discussed here for the sake of brevity; the interested reader is referred to Ref. [46] for a detailed discussion of the 6.5 bar IMEP results. Previous experimental results from the same SCRE [45] were used to determine “optimal” values of other operating parameters such as diesel injection pressure, intake manifold pressure, and methane PES for each SOI sweep. From the previous experiments [45] and considering the wide range of SOIs investigated in this study, the diesel injection pressure was set to 500 bar for all of the experiments to avoid excessively high peak cylinder pressures and maximum pressure rise rates (MPRR) and to yield reasonably low ISNO_x and smoke emissions and good indicated fuel conversion efficiencies (IFCE).

The parameters of each experiment are summarized in the experimental test matrix given in Table 3. While all experiments presented in this paper were performed at a constant engine speed of 1500 rev/min, it should be noted that the trends at other speeds may be qualitatively similar for different SOIs and IMEPs. For the 4.1 bar IMEP SOI sweep, the intake pressure and methane PES were fixed at 1.5 bar and 80 percent, respectively. The intake pressure was chosen to avoid misfire at advanced SOIs. The PES was limited by excessive ISHC and ISCO emissions as well as high COV IMEP at advanced SOIs. The SOI range for these experiments was 260°-360°. For the SOI sweep at 12.1 bar IMEP, the intake pressure and methane PES were fixed at 1.8 bar and 95 percent, respectively. The PES was increased to avoid high in-cylinder

pressures and MPRRs at advanced SOIs but the maximum possible PES was also limited by diesel injection pressure control issues. These experiments were performed with a standard common-rail diesel injection system. Therefore, it was difficult to maintain constant injection pressures for very small pilot quantities (for methane PES greater than 95 percent). The 12.1 bar IMEP experiments were performed at SOIs of 340°-350° and 260°-310°. The SOI of 360° was omitted due to excessive ISHC emissions. The SOIs of 320° and 330° were omitted due to excessive in-cylinder pressures and high MPRR.

Table 3. Experimental Test Matrix

Engine Load (IMEP)	SOI	Constant Parameters
4.1 bar	260° - 360°	P _{in} = 1.5 bar PES = 80%
12.1 bar	260° - 310° 340° - 350°	P _{in} = 1.8 bar PES = 95%
For all experiments: engine speed = 1500 rev/min, diesel injection pressure = 500 bar		

To streamline the discussion, the combustion parameters used in this paper such as equivalence ratio (Φ), percent energy substitution (PES), ignition delay (ID), combustion efficiency (η_c), apparent heat release rate (AHRR), and specific heat ratio (γ) are defined below:

$$PES = \frac{\dot{m}_g LHV_g}{\dot{m}_d LHV_d + \dot{m}_g LHV_g} \times 100\% \quad (1)$$

$$\Phi = \frac{\left(\frac{A}{F}\right)_{st-tot}}{\left(\frac{\dot{m}_a}{\dot{m}_d + \dot{m}_g}\right)} \quad (2)$$

$$ID = CA5 - SOI \quad (3)$$

$$\eta_c = 1 - \frac{\sum_i x_i Q_{LHV_i}}{\left(\frac{\dot{m}_f}{\dot{m}_f + \dot{m}_a} \right) Q_{LHV_f}} \quad (4)$$

$$AHRR(\theta) = \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dP}{d\theta} \quad (5)$$

$$\gamma(T) = 1.338 - 6 \times 10^{-5} T + 1 \times 10^{-8} T^2 \quad (6)$$

In Equations 1 and 2, $\dot{m}$ refers to the mass flow rates of diesel (subscript d), gaseous methane fuel (subscript g), and air (subscript a), and LHV refers to the corresponding fuel lower heating values. Stoichiometric air-fuel ratio $(A/F)_{st-tot}$ is defined as the stoichiometric air required for complete oxidation of both diesel and methane into CO_2 and H_2O . Therefore, $(A/F)_{st-tot}$ is dependent on the PES of methane. The start of combustion (SOC) is defined as CA5, or the crank angle at which 5 percent of cumulative heat release occurs. Combustion duration (CA 10-90) is defined as the crank angle duration between the locations of 10 percent and 90 percent cumulative heat release.

Combustion efficiency is usually calculated based on the exhaust concentrations of CO, HC, H_2 and PM emissions and their respective lower heating values [47]. In this study, only the measured CO and HC concentrations and the H_2 mass fractions estimated from stoichiometry calculations are used in the combustion efficiency calculations. In Equation 4, which defines combustion efficiency (η_c), x_i are the mass fractions of HC, CO, and H_2 , Q_{LHV_i} are the lower heating values of these species, and subscripts “ f ” and “ a ” denote fuel and air, respectively. Following Heywood [47], the lower heating values for CO and H_2 are assumed to be 10.1 MJ/kg and 120 MJ/kg, respectively. The concentrations of various HC components in the exhaust are not known from the EGAS 2M emissions bench measurements, but since the LHV of HC is quite

similar to that of the fuel, Heywood recommends using the LHV of the fuel for HC in the combustion efficiency calculations. However, since two fuels are used in dual fuel combustion, the exhaust HC can theoretically originate from both fuels; therefore, the combined mass-fraction-weighted LHV of “the fuel mixture” (i.e., mass-fraction-weighted average of 42.6 MJ/kg for diesel and 50 MJ/kg for methane) is used for the exhaust HC. The relative mass fractions of diesel and methane change with PES, and consequently, the estimated lower heating value of the exhaust HC ranges from 48.5 MJ/kg (for 80 PES) to 49.6 MJ/kg (for 95 PES).

Apparent heat release rate (AHRR) is calculated using Equation 5 [47]. The ratio of specific heats (γ) was determined using Equation 6 based on global in-cylinder bulk temperature (T) [48]. As mentioned before, cylinder pressure (P) was measured at 0.1 CAD resolution. By utilizing the measured in-cylinder pressure, the instantaneous volume, and the estimated mass trapped at IVC in the ideal gas equation of state, the bulk temperature was calculated at each crank angle. The rest of the variables in Equation 5 were computed in the following manner. The rate of change of volume with respect to crank angle ($dV/d\theta$) was calculated using engine geometry and piston position. The instantaneous volume (V) was determined using known engine geometry values such as compression ratio, bore, stroke, and connecting rod length coupled with engine crankshaft position measurements. The rate of change of pressure with respect to crank angle ($dP/d\theta$) was calculated using crank angle-resolved cylinder pressure data.

4. RESULTS AND DISCUSSION

4.1 Results for 4.1 bar IMEP

4.1.1 Apparent Heat Release Rate and Cylinder Pressure

Figures 2 and 3 show the cylinder pressure and AHRR histories for various SOIs at 4.1 bar IMEP. Peak AHRR first advances as SOI is advanced from between 360° to 330° and then retards as SOI is advanced beyond 320° . As SOI is advanced from 330° to 320° , the shape of the AHRR curve and combustion characteristics change significantly. For this reason and for easier reading of results, cylinder pressure profiles and AHRR are plotted in two different SOI groups.

For SOIs between 360° and 330° , the peak cylinder pressures increase as SOI is advanced. Moreover, the AHRR curves exhibit two distinct stages of heat release and no significant low temperature heat release (LTHR) peak. The absence of LTHR at retarded SOIs can be explained as follows. For the more retarded SOIs, there is little or no separation between the end of injection (EOI) and SOC. Figure 4 shows the crank angle duration between EOI and SOC (SOC-EOI) and ignition delay for each SOI. Furthermore, at these retarded SOIs, diesel is injected at relatively higher in-cylinder temperatures (see Fig. 11(a)). Previous studies have found that LTHR is more likely to occur at temperatures below 850 K [9]. The relatively higher temperatures at retarded SOIs could possibly suppress LTHR and lead directly to high temperature ignition.

As SOI is advanced earlier in the compression stroke, the separation between EOI and SOC increases and the AHRR curves exhibit a single-stage (almost Gaussian) profile. For SOIs advanced to 320° and beyond, the in-cylinder temperatures are relatively lower when injection begins, and consequently, LTHR is clearly evident around 340° . At 320° SOI, EOI occurs at

330° and the main combustion event starts at 347°, which corresponds to 17 CAD separation between EOI and SOC. As SOI is advanced to 300°, LTHR is still present as the separation increases from 17 CAD to approximately 52 CAD. As SOI is advanced further from 300° to 260°, the magnitude of heat release increases and the peak AHRR is phased after TDC. Accompanying these AHRR trends, the peak cylinder pressures decrease monotonically and occur later in the expansion process with SOI advancement from 320° to 260°. Furthermore, as SOI is advanced from 300° to 260°, the magnitude of LTHR decreases and a “well-mixed” combustion of diesel-methane is observed. The decreasing magnitude of LTHR may be the result of diesel being “over-mixed” with the methane-air charge. Although the in-cylinder temperatures are relatively low and may not suppress LTHR, the increased time available for mixing could lead to leaner local mixtures, thus inhibiting heat release until hot ignition.

Figure 2. (a) Cylinder pressure histories and (b) crank angle-resolved heat release rates at late SOI at 4.1 bar IMEP, 80 PES, N=1500 rev/min, $P_{in}=1.5$ bar

Figure 3. (a) Cylinder pressure histories and (b) crank angle-resolved heat release rates at early SOI at 4.1 bar IMEP, 80 PES, $N=1500$ rev/min, $P_{in}=1.5$ bar

4.1.2 Ignition Delay, Maximum Pressure Rise Rate, and Rate of Combustion

Figure 4 shows the SOC-EOI and ignition delay (ID) versus SOI. From the ID and SOC-EOI trends, it is evident that both of these variables remain nearly invariant between 340° and 360° but increase linearly for SOIs earlier than 340° . While the “physical component” (e.g., diesel evaporation and mixing) of the ID may contribute more for retarded SOIs, the “chemical component” of the ID (i.e., preignition reaction rates) likely dominates for more advanced SOIs since there is sufficient time for diesel evaporation and effective mixing with the surrounding methane-air mixture. Figure 5 shows the COV IMEP and MPRR trends at different SOIs. As SOI is advanced beyond 310° , a sharp increase in COV IMEP is observed, possibly due to the onset of misfire for mixtures that are too lean and homogeneous. For SOIs between 320° and

350°, the highest MPRRs are observed, which correlates with the earliest onset of ignition (well before TDC) and combustion phasing (CA50) close to or before TDC as shown in Fig. 6.

Figure 6 shows the SOC (i.e., CA5), CA50, and combustion duration (CA10-90) for each SOI. A decrease of CA10-90 is observed from about 27 degrees to about 15 degrees when SOI is advanced from 360° to 310° and then remains between 15 and 17 degrees for SOIs between 300° and 260°. A relatively invariant CA10-90 at early SOIs might be attributed to well-mixed, homogeneous combustion. Also, CA50 shifts from after TDC to before TDC when SOI is advanced from 360° to 330° and then shifts back to after TDC as SOI is advanced further. This trend is linked to the SOC trend at those conditions. As SOI is advanced from 360° to 330°, the SOC also occurs earlier in the compression stroke (before TDC); however, for SOIs more advanced than 320°, the combustion becomes more “homogeneous” and starts closer to TDC, leading to more delayed combustion phasing (CA50) but nearly constant CA10-90.

Figure 4. SOC-EOI and Ignition Delay at various SOIs at 4.1 bar IMEP, 80 PES, N=1500 rev/min, $P_{in}=1.5$ bar

Figure 5. COV IMEP and MPRR at various SOIs at 4.1 bar IMEP, 80 PES, N=1500 rev/min, $P_{in}=1.5$ bar

Figure 6. CA5, CA50 and CA10-90 at various SOIs at 4.1 bar IMEP, 80 PES, N=1500 rev/min, P_{in} =1.5 bar

4.1.3 Fuel Conversion Efficiency and Combustion Efficiency

Figure 7 shows the indicated fuel conversion efficiency (IFCE) and combustion efficiency trends at different SOIs. Combustion efficiency decreases as SOI is advanced from 360° to 340°, then increases, reaching a peak value of about 75 percent at 300° SOI. The combustion efficiency behavior with SOI is related to the HC and CO emissions trends presented later. In general, the higher the HC and CO emissions, the lower the combustion efficiency and vice versa. The IFCE increases from about 21 percent at 360° SOI to about 36 percent at 300° SOI but decreases with further SOI advancement. This increase in IFCE may be attributed to the phasing of CA50 closer to TDC. As SOI is advanced from 300° to 260° IFCE decreases, which is likely due to the combined effect of CA50 moving away from TDC and decreasing combustion efficiency.

Figure 7. Combustion and net indicated fuel conversion efficiencies at various SOIs at 4.1 bar IMEP, 80 PES, N=1500 rev/min, $P_{in}=1.5$ bar

4.1.4 Emissions

Figure 8 shows $ISNO_x$ and smoke trends for various SOIs. As SOI is advanced from 360° to 340°, the $ISNO_x$ emissions increase sharply from about 3.5 g/kWh to about 11.5 g/kWh. However, when SOI is advanced further to 310°, the $ISNO_x$ values decrease significantly to around 0.15 g/kWh and remain low until 260° SOI. As SOI is advanced to 310° and beyond, the diesel spray has increasingly longer time available for mixing with the lean methane-air mixture before the onset of ignition. Therefore, the in-cylinder fuel-air mixture becomes locally leaner and more homogeneous during the combustion process, which leads to lower local temperatures and reduced $ISNO_x$ emissions. For reference, the premixed methane-air equivalence ratios as well as the overall equivalence ratios including diesel, methane, and air at each SOI are shown in Figure 9. It is clear from this figure that over the range of SOIs examined at 4.1 bar IMEP, the premixed methane-air equivalence ratio was very lean, varying between 0.2 and 0.35. Also, as

evident from Fig. 8, engine-out smoke emissions are extremely low (< 0.04 FSN) for the duration of the SOI sweep. Since bulk of the fuel energy input (~ 80 PES) arises from the lean, premixed methane-air mixture, the smoke emissions remain very low for all SOIs. In this regard, it is important to note that the simultaneous reduction of $ISNO_x$ and smoke emissions at and beyond 310° is indicative of the occurrence of LTC at these SOIs. Similar trends were obtained in Ref. [44] for diesel-gasoline dual fuel LTC on the same SCRE used in the present study.

Figure 8. $ISNO_x$ and smoke emissions at various SOIs at 4.1 bar IMEP, 80 PES, $N=1500$ rev/min, $P_{in}=1.5$ bar

Figure 9. Equivalence ratio for methane-air and diesel-methane-air mixtures at various SOIs at 4.1 bar IMEP, 80 PES, N=1500 rev/min, $P_{in}=1.5$ bar

Figure 10 shows the ISCO and ISHC emissions between 360° and 260° SOIs. A decrease in ISCO emissions is observed from about 30 g/kWh at 360° to 15 g/kWh at 320° followed by an increase to 70 g/kWh at 260°. A decrease in ISHC emissions is observed from 120 g/kWh at 360° to about 45 g/kWh at 300° followed by an increase to nearly 80 g/kWh at 260°. The percentage of methane present in the total ISHC emissions was likely very high for all SOIs. However, the actual percentage could not be quantified from the data acquired because the methane emissions exceeded the available range of the AVL FTIR analyzer.

Carbon monoxide (CO) is an intermediate stable species formed during the oxidation process of any hydrocarbon fuel. After initial breakdown of the fuel molecule leading to CO formation, CO is oxidized to carbon dioxide (CO₂) mostly with hydroxyl (OH) radicals that are available in high concentrations in high temperature, fuel-lean regions within the combustion chamber. The CO oxidation reaction ($CO + OH \rightarrow CO_2 + H$) is a very important exothermic

reaction in hydrocarbon combustion [49]. It occurs following the oxidation of the original fuel and the intermediate hydrocarbon fragments and is relatively slow at temperatures below 1100 K [50]. Therefore, under engine combustion conditions, the CO oxidation reaction requires high combustion temperatures (> 1500 K for HCCI engine conditions, according to Ref. [51]) and the availability of OH radicals. For straight diesel operation at all BMEPs, the CO emissions are extremely low (not shown here) compared to the corresponding values for dual fuel LTC, irrespective of SOIs. In conventional diesel combustion, CO is formed within the diesel jet periphery and is oxidized almost completely within the hot diffusion flame, where OH radicals are abundant and high local temperatures accelerate CO oxidation. However, with dual fuel LTC, CO may be formed both within the diesel jet periphery (for late SOIs when the jet does not completely disintegrate before ignition) as well as in the lean premixed methane-air mixture, which oxidizes following diesel ignition. Even if the jet remains intact for late SOIs, it is likely that the CO formed within the diesel jet is largely oxidized in the jet periphery. However, most of the CO formed in the lean, premixed methane-air mixture may remain unoxidized until the exhaust valve opens for the following possible reasons: (1) reduced local availability of OH radicals due to relatively low local temperatures, (2) inadequate mixing between high temperature and low temperature regions, and (3) insufficient time available for CO oxidation due to shorter combustion durations; i.e., CO oxidation freezes quickly after combustion ends.

The major sources of unburned hydrocarbons (HC) for dual fuel LTC (with methane fumigation in the intake manifold) are crevices and bulk quenching of the burning diesel-methane-air mixture. The in-cylinder global temperature has a significant influence on both HC and CO oxidation rates. For retarded SOIs between 360° and 340° as well as for more advanced SOIs between 290° and 260° , the peak in-cylinder bulk temperatures are relatively lower, which

will inhibit CO and HC oxidation. At retarded SOIs, CA50 is phased well after TDC, so crevice hydrocarbons are not oxidized, leading to higher ISHC emissions. At very advanced SOIs between 300° and 260°, increasing trends are observed both in ISHC and ISCO. As SOI is advanced, diesel is injected into progressively less dense methane-air mixtures, leading to longer diesel spray penetration and increasing the potential for spray impingement on the cylinder walls or piston. Therefore, a significant amount of diesel fuel can escape into relatively cooler boundary layers in the cylinder periphery and not participate in the ignition of the premixed methane-air mixture. The lack of availability of more reactive diesel fuel increasingly impedes combustion of the lean premixed methane-air mixture, which results in higher unburned fuel and partially oxidized species manifesting as increased ISHC and ISCO emissions.

For SOIs between 330° and 320°, ISCO emissions are fairly low. This may be attributed to the CA50 being phased closer to TDC, which can lead to higher in-cylinder bulk temperatures that promote CO oxidation. To test this hypothesis, in-cylinder bulk temperatures for different SOIs are shown in Fig. 11. As SOI is advanced from 360° to 330°, the peak bulk temperature increases from about 1000 K to more than 1200 K and also occurs earlier; however, upon further SOI advancement the peak bulk temperature decreases up to 300° SOI. For SOIs ranging from 300° to 260°, peak bulk temperatures are around 1200 K. Since the peak bulk temperatures did not exceed 1500 K for any of the SOIs at 4.1 bar IMEP, ISCO emissions were inevitable. Nevertheless, the overall bulk temperatures during the combustion process are lower for retarded SOIs compared to advanced SOIs (between 320° and 260°). As SOI is advanced beyond 320°, ISCO emissions exhibit a steeper increase than ISHC emissions; on the other hand, as SOI is retarded from 320°, the ISHC emissions increase sharply while the increase in ISCO is less dramatic. These trends can be explained by recognizing the potential competition (tradeoff)

between HC and CO oxidation. As discussed above, while the primary product of fuel oxidation is CO, the oxidation of CO to CO₂ occurs later in the combustion process. Since HC oxidation is much faster than CO oxidation, the latter does not start until all of the fuel and intermediate hydrocarbons are consumed. Consequently, CO-to-CO₂ conversion is inhibited at lower bulk temperatures, leading to higher ISCO emissions. Similarly, for SOIs that yield very low bulk temperatures (e.g., 360° SOI), even HC oxidation (especially for the HC trapped in crevices and caused due to bulk quenching) may be inhibited, resulting in lower CO formation and extremely high ISHC emissions. A combination of these factors and the associated tradeoffs cause the trends exhibited in Fig. 10.

In addition to the criteria pollutant emissions discussed above, CO₂ emissions are also important, considering their global warming potential. Another species of interest when methane is used as the low-cetane fuel in dual fuel LTC is formaldehyde (HCHO). It is well known that HCHO is an important reaction intermediate formed both in methane-air flames [49]. Moreover, HCHO may also be formed due to partial HC oxidization in the exhaust manifolds of natural gas engines [52]. Although HCHO emissions are not currently regulated as criteria pollutants from engines in the United States, they still pose an environmental and human health concern. Therefore, it is desirable to quantify HCHO emissions in dual fuel engines. Figure 12 shows indicated specific CO₂ (ISCO₂) and HCHO emissions at different SOIs. The minimum HCHO emissions were observed at 320° SOI. When SOI is retarded from 320° to 360° or advanced from 320° to 260°, a significant increase is observed in HCHO emissions, especially for the advanced SOIs (from about 100 parts per million (ppm) at 320° SOI to about 350 ppm at 260°). Interestingly, the increase in HCHO emissions from 320° to 260° SOI closely parallels the sharp increase in ISCO emissions within the same SOI range. Without chemical kinetic simulations, it

514 is difficult to speculate on the precise source of the HCHO emissions at different SOIs but it is
 515 evident that HCHO in the exhaust of diesel-methane dual fuel LTC may be left over
 516 (unconsumed HCHO) from preignition reactions or be formed due to partial oxidation of
 517 unburned methane in the exhaust manifold. By comparison, the ISCO₂ trends are slightly more
 518 complicated. In general, the lower the ISHC and ISCO emissions, the higher the ISCO₂
 519 emissions and vice versa. For example, for SOIs between 310° and 330°, the highest ISCO₂
 520 emissions are observed, and not coincidentally, these SOIs also yield the lowest ISCO emissions
 521 and relatively low ISHC emissions as well.

Figure 10. ISHC and ISCO emissions at various SOIs at 4.1 bar IMEP, 80 PES, N=1500 rev/min, P_{in}=1.5 bar

Figure 11. Bulk temperature schedules at (a) late SOIs and (b) advanced SOIs at 4.1 bar IMEP, 80 PES, $N=1500$ rev/min, $P_{in}=1.5$ bar

Figure 12. ISCO₂ and HCHO emissions at various SOIs at 4.1 bar IMEP, 80 PES, N=1500 rev/min, P_{in}=1.5 bar

4.2 Results for 12.1 bar IMEP

The engine was operated at 12.1 bar IMEP, 1500 rev/min and 95 PES while diesel pilot SOI was varied from 350° to 260°. The diesel injection pressure and intake boost pressure were maintained at 500 bar and 1.8 bar, respectively. Compared to 4.1 bar IMEP, a higher PES (95%) was required to avoid high peak in-cylinder pressures and MPRRs at advanced SOIs. The 12.1 bar IMEP experiments were performed at 350°-340° and 310°-260° SOIs. The SOI of 360° was omitted due to excessive ISHC emissions. The SOIs of 330° and 320° were omitted due to excessive peak in-cylinder pressures and MPRRs.

4.2.1 Apparent Heat Release Rate and Cylinder Pressure

Figures 13 and 14 show the cylinder pressure and AHRR histories for various SOIs. For retarded SOIs of 350° and 340°, the AHRR curves exhibit two stage combustion, with most of the heat release occurring in the second stage due to the higher PES for 12.1 bar IMEP. Again, a significant change in the shape of the AHRR curve from two-stage to single-stage heat release is observed as SOI is advanced from 340° to 310°. Furthermore, the peak AHRR magnitudes decrease and combustion is phased away from TDC for SOIs advanced beyond 310°. This is likely due to the increased level of premixing of diesel with the methane-air mixture, which reduces overall combustion rates. Also, unlike the 4.1 bar IMEP AHRR results, LTHR is not present for any SOI at 12.1 bar IMEP. This may be attributed to significantly higher in-cylinder bulk temperatures, which would prevent the occurrence of LTHR.

Figure 13. (a) Cylinder pressure histories and (b) crank angle-resolved heat release rates at late SOI at 12.1 bar IMEP, 95 PES, $N=1500$ rev/min, $P_{in}=1.8$ bar

Figure 14. (a) Cylinder pressure histories and (b) crank angle-resolved heat release rates at early SOI at 12.1 bar IMEP, 95 PES, $N=1500$ rev/min, $P_{in}=1.8$ bar

4.2.2 Ignition Delay, Maximum Pressure Rise Rate, and Rate of Combustion

Figure 15 shows SOC-EOI and ignition delay trends while Figure 16 shows the COV IMEP and MPRR at different SOIs for 12.1 bar IMEP operation. The SOC-EOI and ignition delay trends are quite similar to the trends obtained at 4.1 bar IMEP. However, as mentioned before, 330° and 320° SOIs resulted in high MPRRs; so, data were not obtained at those SOIs. As SOI is advanced from 350° to 340°, SOC-EOI and ignition delay are relatively invariant; however, beyond 310°, both parameters increased monotonically. Also, as SOI is advanced beyond 310°, a sharp increase in COV IMEP is observed. This trend is comparable to the 4.1 bar IMEP results. Since PES is higher for this SOI sweep, the quantity of diesel required to achieve 12.1 bar IMEP may be small enough to lead to overmixing and unstable operation at advanced SOIs. For SOIs between 310° and 340°, the highest MPRRs are observed. Again, 320° and 330° SOIs were omitted due to excessive MPRRs. By comparison, as SOI is advanced beyond 310°, the MPRR decreased to very low values (< 4 bar/CAD) due to delayed combustion phasing.

Figure 17 shows that CA10-90 decreases from about 23 CAD to about 11 CAD when SOI is advanced from 350° to 310° and then increases to about 20 CAD for 270° SOI. This trend is quite different from the relatively constant CA10-90 observed at advanced SOIs at 4.1 bar IMEP. The increase in CA10-90 at very advanced SOIs may be attributed to the higher methane PES, which leads to slower overall combustion. The CA50 shifts from after TDC to very close to TDC when SOI is advanced from 350° to 310° and then shifts after TDC as SOI is advanced further. This combustion phasing trend partially explains the MPRR and the COV IMEP trends shown in Figure 16. For SOIs earlier than 310°, combustion is more homogeneous and CA5 occurs closer to TDC, leading to relatively low MPRRs but high COV IMEPs.

Figure 15. SOC-EOI and Ignition delay at various SOIs at 12.1 bar IMEP, 95 PES, N=1500 rev/min, P_{in} =1.8 bar

Figure 16. COV IMEP and MPRR at various SOIs at 12.1 bar IMEP, 95 PES, N=1500 rev/min, P_{in} =1.8 bar

Figure 17. CA5, CA50 and CA10-90 at various SOIs at 12.1 bar IMEP, 95 PES, N=1500 rev/min, P_{in} =1.8 bar

4.2.3 Fuel Conversion Efficiency and Combustion Efficiency

Figure 18 shows IFCE and combustion efficiency between 350° and 260° SOI at 12.1 bar IMEP. Combustion efficiency increases as SOI is advanced from 350° to 310° reaching a peak value of about 92 percent at 310° SOI. This increase can be attributed to more conducive (near TDC) CA50 and shorter CA10-90 at the earlier SOIs; however, for more advanced SOIs up to 260°, combustion efficiency decreases due to delayed CA50 and longer CA10-90. The IFCE and combustion efficiency trends are similar. The IFCE increases from about 42 percent at 350° SOI to almost 45 percent at 310° SOI but decreases for more advanced SOIs. The initial increase in IFCE may be attributed to improved combustion efficiencies, CA50 phased closer to TDC, and shorter CA10-90. As SOI is advanced from 310° to 260°, the IFCE decreases due to lower combustion efficiency, delayed CA50, longer CA10-90, and higher COV IMEP.

Figure 18. Combustion and indicated fuel conversion efficiencies at various SOIs at 12.1 bar IMEP, 95 PES, $N=1500$ rev/min, $P_{in}=1.8$ bar

4.2.4 Emissions

Figure 19 shows $ISNO_x$ and smoke trends for different SOIs at 12.1 bar IMEP. Smoke emissions are less than 0.05 FSN for all SOIs due to locally lean conditions and a high methane PES of 95%. As SOI is advanced from 350° to 340° , the $ISNO_x$ emissions increase sharply from about 4.6 g/kWh to about 13.6 g/kWh. However, when SOI is advanced further, $ISNO_x$ decreases and remains between 1.3 and 1.5 g/kWh beyond 280° SOI. The decreasing $ISNO_x$ trend with SOI advancement beyond 310° is a consequence of longer separation between EOI and SOC, which leads to longer time available for mixing of diesel with the lean methane-air mixture. Longer mixing times, in turn, result in locally lean and increasingly homogeneous combusting mixtures, which yield lower local temperatures and lower $ISNO_x$ emissions. Despite the decrease in $ISNO_x$ emissions at advanced SOIs, the overall magnitude of $ISNO_x$ is much higher at 12.1 bar IMEP compared to lower loads. This can be explained by the relatively higher

local equivalence ratios and higher local in-cylinder temperatures at this IMEP (indirectly evident from the higher bulk temperatures shown in Figure 22).

Figure 19. ISNO_x and smoke emissions at various SOIs at 12.1 bar IMEP, 95 PES, N=1500 rev/min, P_{in}=1.8 bar

Figure 20 shows ISHC and ISCO emissions trends between 350° and 260° SOIs. Again, similar to 6.5 bar IMEP, both ISCO and ISHC emissions trends closely follow each other. The ISCO emissions decrease from about 5 g/kWh at 350° to 2.5 g/kWh at 310° followed by an increase to 3.5 g/kWh at 260°. A similar decrease in ISHC emissions is observed from 22 g/kWh at 350° to about 13.4 g/kWh at 310° followed by an increase to about 21 g/kWh at 260°. The percentage of methane present in the total HC emissions is likely very high for all SOIs, even though the actual percentage could not be quantified because exhaust methane concentrations exceeded the range of the AVL FTIR analyzer.

As mentioned earlier, temperature has a significant effect on HC and CO oxidation. For SOIs from 310° to 260°, peak bulk temperatures are around 1600 K. The significantly higher temperatures will increase HC and CO oxidation rates, contributing to the overall lower ISHC and ISCO emissions compared to lower loads. Again, since crevices are the major source of HC emissions in fumigated dual fuel LTC, for very early and very late SOIs, HC emissions are relatively high. For these SOIs, the CA50 is phased well after TDC and crevice HC are not oxidized, leading to higher ISHC emissions. As SOI is advanced from 350° and 310°, SOC (i.e., CA5) occurs around 350° while CA50 occurs exactly at 360°. The combination of an earlier SOC and a heat release profile centered at TDC with higher peak AHRR leads to a simultaneous reduction in both ISHC and ISCO emissions. At very advanced SOIs between 300° and 260°, ISHC and ISCO emissions exhibit a slightly increasing trend, which may be attributed to overmixed combustion and increasing engine instability as evident from the increasing COV IMEP trend shown in Figure 16. It is also interesting to note the tradeoff between ISNO_x emissions on the one hand, and ISHC and ISCO emissions on the other, for these advanced SOIs at 12.1 bar IMEP. While ISNO_x emissions decrease significantly with SOI advancement from 300° to 260°, the ISHC and ISCO emissions increase. Similar tradeoffs were also observed for advanced SOIs at high loads (~ 12 bar BMEP) by Krishnan et al. [32]. In this regard, temperature-controlled EGR may be helpful in improving engine stability, ISHC, and ISCO emissions without adversely affecting ISNO_x emissions at advanced SOIs [40-41].

Figure 21 shows ISCO₂ and HCHO emissions trends between 350° and 260° SOIs for 12.1 bar IMEP. The HCHO emissions remained relatively invariant (~55 ppm) between 340° and 290° SOIs. However, when SOI is retarded beyond 340° or advanced beyond 290°, the HCHO emissions increased; this increase was slightly higher for the late SOIs (about 100 ppm at

659 350° SOI compared to about 90 ppm at 260°). In general, following the trends from 4.1 bar
 660 IMEP, the magnitude of the HCHO emissions at 12.1 bar IMEP is even lower. Since HCHO
 661 emissions may be left over from preignition reactions or be formed due to partial methane
 662 oxidation in the exhaust manifold, and since both in-cylinder bulk temperatures and exhaust gas
 663 temperatures are higher at 12.1 bar IMEP, the HCHO emissions are lower. The ISCO₂
 664 emissions, however, did not show a clear trend, except that the overall behavior was somewhat
 665 similar to the ISHC and ISCO trends.

666
 667 **Figure 20.** ISHC and ISCO emissions at various SOIs at 12.1 bar IMEP, 95 PES,
 668 N=1500 rev/min, P_{in}=1.8 bar

Figure 21. ISCO₂ and HCHO emissions at various SOIs at 12.1 bar IMEP, 95 PES, N=1500 rev/min, P_{in}=1.8 bar

Figure 22. Bulk temperature schedules at (a) late SOIs and (b) advanced SOIs at 12.1 bar IMEP, 95 PES, $N=1500$ rev/min, $P_{in}=1.8$ bar

5. CONCLUSIONS

Diesel-ignited methane dual fuel low temperature combustion (LTC) experiments were performed on a single-cylinder research engine (SCRE). Methane was fumigated upstream of the intake manifold and diesel was injected during the compression stroke with a common rail injection system. The engine was operated over a range of engine loads at a constant speed of 1500 rev/min; however, results were presented for two representative loads of 4.1 bar and 12.1 bar *net* IMEP. For all experiments, diesel injection pressure was fixed at 500 bar. Intake manifold pressure and methane percent energy substitution (PES) were fixed at 1.5 bar and 80%, respectively for 4.1 bar IMEP and at 1.8 bar and 95%, respectively, for 12.1 bar IMEP. Diesel start of injection (SOI) was varied between 360° (compression TDC) and 260° to characterize its

impact on engine performance and engine-out ISNO_x, ISHC, ISCO, and smoke emissions. The results obtained in this study lead to the following conclusions:

- For both loads, when SOI was advanced, the separation between EOI and SOC increased, the ignition delay increased, the AHRR curve changed from a distinct two-stage profile to a smooth single-stage (almost Gaussian) profile, and the peak cylinder pressure increased up to 320° SOI and subsequently decreased.
- The IFCE increased when SOI was advanced to about 300° for both loads, likely due to a combination of higher combustion efficiencies, more conducive CA50 (phased near TDC), and shorter combustion durations (CA 10-90).
- At 4.1 bar IMEP, advancing SOI to 300° reduced ISNO_x emissions to less than 0.15 g/kWh. At 12.1 bar IMEP, further SOI advancement to 280° was required to achieve low ISNO_x emissions; however, the minimum possible ISNO_x levels at 12.1 bar IMEP were still much higher (~1.3-1.5 g/kWh) compared to 4.1 bar IMEP.
- Engine-out smoke emissions were extremely low (< 0.05 FSN) for both loads and all SOIs examined in this study, likely because the high methane PES (80-95%) ensured that most of the combustion process occurred in the lean premixed methane-air mixture, preventing soot formation.
- Very high ISHC and ISCO emissions were observed for retarded (near-TDC) SOIs at all loads. The lowest ISHC and ISCO levels were observed for SOIs near 310° and for more advanced SOIs, both ISHC and ISCO emissions increased. In general, the ISHC (~ 40-120 g/kWh) and ISCO (~ 15-70 g/kWh) emissions at 4.1

bar IMEP were much higher compared to 12.1 bar IMEP due to relatively low in-cylinder bulk temperatures.

- Extremely high maximum pressure rise rates and the tendency to knock prevented engine operation at intermediate SOIs between 310° and 340° for 12.1 bar IMEP. On the other hand, for both 4.1 bar and 12.1 bar IMEPs, high COV IMEP (> 5%) caused unstable engine operation for SOIs advanced beyond 280°.

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