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A parametric investigation of diesel/methane dual-fuel combustion progression/stages in a heavy-duty optical engine



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HIGHLIGHTS

- Compression ignited heavy-duty optical engine operated in dual-fuel mode.
- Second derivative analysis, a novel method to identify dual-fuel combustion stages.
- Increasing pilot ratio or methane equivalence ratio increases locally fuel-rich zones.
- Overlapping of all three combustion stages at high-load (rich) conditions.
- Transition of HRR curve from multi-peak (M-shaped) to quasi-single peak (bell-shaped).

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Keywords: Dual-fuel Natural luminosity imaging Optical engine analysis Second derivative HRR analysis Combustion progression/stages

ABSTRACT

A single-cylinder heavy-duty optical engine is used to characterize dual-fuel (DF) combustion. In experiments, methane is applied as the main fuel while directly injected pilot diesel ignites the premixed methane-air mixture close to the top-dead center (TDC). In the present study, diesel-methane DF combustion is analyzed as a function of (1) the methane equivalence ratio, (2) initial charge temperature, and (3) the quantity of pilot diesel. Experiments are conducted at 1400 rpm and a load of 9-10 bar IMEP, and DF combustion is visualized in the engine through Bowditch-designed optical access. Meanwhile, a high-speed camera records temporally resolved natural luminosity (NL) color images of the combustion event. The results of the study suggest that DF combustion based on the apparent heat release rate (HRR) data consists of three overlapping combustion stages, where the level of overlap depends on mixture fractions of both pilot-diesel and methane in the in-cylinder charge. The stages are identified by analyzing the second derivative of HRR data. The study revealed that during the first stage, most of the pilot diesel burns in the premixed mode, and that the ignition delay time (IDT) directly influences the burnt charge mixture fraction of pilot diesel and entrained premixed methane-air mixture. In addition, the first-stage combustion is visualized as initial flame kernels originating from pilot-diesel sprays. IDT is found to be especially sensitive to the methane equivalence ratio and initial charge temperature. Furthermore, the concentration of methane and the quantity of pilot diesel in the charge distinctively influence combustion duration trends.

1. Introduction

Internal combustion engines (ICE) constitute a considerable portion of transportation and power generation sectors, consuming a vast amount of fossil fuels. ICEs alone consume about 70% of the world's daily crude oil demand [1–3], which is a major concern due to harmful emissions, such as nitrogen oxides (NO_x), particulate matter (soot) and carbon dioxide (CO₂) [4–7]. In recent years, research and development to mitigate engine-out emissions has adopted alternative fuels, shifting to promising combustion technologies [1,8,9]. The dual-fuel (DF)

combustion concept is a favorable strategy for adopting alternative fuels or natural gas (NG) into conventional diesel engines in order to achieve substantial reductions in emissions of CO_2 , NO_x and soot. DF engines using NG provide targeted performance with high efficiencies comparable to existing diesel engines [10–12]. Methane, which is the main constituent of NG, is a clean burning fuel and inherently produces low CO_2 due to its low carbon content. DF engines combine the attributes of compression-ignited (CI) and spark-ignited (SI) combustion processes by adopting two fuels of different reactivity. In DF engines, a premixed mixture of air and low-reactivity fuel (methane), which is

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Nomenclature

DF	dual fuel
TDC	top dead center
IMEP	gross indicated mean effective pressure
NL	natural luminosity
HRR	apparent heat release rate
AccQ	accumulative heat release
SDHRR	second derivative of heat release rate
IDT	ignition delay time
CC	combustion chamber
CW	clockwise
IE	injected energy
LHV	lower heating value
SOI	start of injection
CAD	crank angle degree
ATDC	after top dead center
AFR	air to fuel ratio
FA	flame area
Ι	averaged intensity of NL image
T _{TDC}	calculated isentropic temperature at TDC
fps	frames per second

difficult to auto-ignite by compression, is ignited by means of a highreactivity fuel (diesel) close to the top-dead center (TDC) [13,14]. DF engines operating under lean conditions produce lower NO_x and CO₂ emissions than conventional diesel engines [12]. In general, DF engines require very small quantities of pilot diesel in order to achieve low NO_x and soot emissions. However, this may produce ignition instabilities, leading to misfire, as well as significant emissions of unburned hydrocarbons (UHC) and carbon monoxide (CO) [15–22]. Despite the advantages of DF engines over conventional diesel engines regarding engine-out emissions and high efficiency, the operating window of DF engines is often limited by misfire [14,23] and knock [24–26] at low and high loads, respectively. Therefore, it is important to develop the DF combustion process further, especially considering the present limitations of the technology.

Recently, comprehensive numerical and computational fluid dynamics (CFD) simulations have gained considerable attention in DF engine research. Although numerical models and simulations can aid the development of DF combustion technologies, these models require experimental data for validation. Most numerical analyses [27-32] have only focused on the effect of operating parameters on DF combustion performance, engine-out emissions and optimization. These numerical studies include Reyonlds-averaged Navier-Stokes (RANS), zero-dimensional stochastic reactor model (0-D SRM), 1-D/ two-zone phenomenological models and 3D-CFD model based engine-scale simulations. Moreover, few investigations [33-37] discuss DF combustion fundamentally and explore ignition or the subsequent flame propagation process. It has been observed in 0-D numerical studies that the ignition of pilot-diesel is inhibited by the addition of methane, reducing the probabilities of pilot-fuel decomposition per unit time and leading to increased ignition delay time (IDT) [34]. Kahila et al. [35] investigated the DF combustion ignition process by means of large-eddy simulations (LES) in combination with a finite-rate chemistry model. They explained DF ignition as a volumetric process and a combination of three ignition stages: (1) low-temperature ignition, (2) high-temperature ignition, and (3) flame initiation in a premixed methane-air mixture. In addition, they found that methane retards pilot-fuel early decomposition by consuming OH radicals and initiating inhibiting reactions.

In addition to numerical studies, many researchers have previously investigated the DF combustion process experimentally in either singlecylinder or multi-cylinder engines [12,18,22,38–45]. Alla et al. [39]

I	first stage combustion
II	second stage combustion
III	third stage combustion
ṁ	mass flow rate
ϕ_{CH4}	methane equivalence ratio
φ _g	global equivalence ratio
P _R	pilot ratio
η_{th}	thermal efficiency
$I_{(\frac{r}{p},CAD)}$	spatio-temporal intensity
r	discretized radius of CC
R	radius of CC
$\theta_{\rm SOI}$	CAD at SOI
$\Theta_{\rm P}$	CAD where HRR magnitude becomes positive
$\theta_1, \theta_2, \theta_3$	CAD at start of a combustion stage
θ_{3a}, θ_{3b}	CAD at start of high-, low-intensity regime of third stage
	combustion
θ ₉₀	CAD where 90% of heat releases
Q _{id}	heat released during ignition delay time
Q_1, Q_2, Q_3	heat released during combustion stage
Q_{3a}, Q_{3b}	heat released during high-, low-intensity regime of third
	stage combustion

examined the effect of the pilot-diesel quantity on diesel-methane DF combustion performance under lean conditions in a single-cylinder research engine. The results show an increase in NO_x emissions due to high-temperature combustion and a reduction in CO and UHC emissions due to the improved combustion process using high pilot-diesel quantities. In addition, an increase in combustion duration and early onset of a knock has been reported at high loads with an increase in the pilot-diesel quantity. Papagiannakis et al. [38] investigated the impact of NG quantity on DF combustion at three engine loads: 40%, 60%, and 80%. The results showed that IDT and combustion duration increase with an increase in NG quantity, while emissions of UHC at low loads remain high. Karim et al. [13,14,46,47] investigated DF combustion comprehensively and revealed that IDT is sensitive to the effective mean temperature, type of gaseous fuel and its concentration in the charge. Moreover, DF combustion becomes unstable at low temperature with small pilot-diesel quantity and lead to long IDT.

Visualizing the combustion process inside the cylinder for advanced fundamental studies is challenging in full-metal engine experiments. Nevertheless, optical engines could offer support for the study of this shortcoming; however, few studies [48-54] have focused on optical diesel-methane DF combustion in the literature. Schlatter et al. [54] and Srna et al. [48] explored DF ignition optically in a rapid-compression-expansion machine (RCEM) based on OH* chemiluminescence and Schlieren photography with simultaneous detection of combustion species, such as CH₂O-PLIF and CH*. These studies found that IDT increases and flame propagation becomes prominent with an increase in methane quantity. The study of Srna et al. [48] further revealed that although methane prolongs high-temperature ignition of pilot fuel, it significantly prolongs low-temperature ignition (early CH₂O-formation). Dronniou et al. [49] characterized the DF combustion mode based on high-speed natural luminosity (NL) and single cycle OH* chemiluminescence images. They indicated a lean flammability limit for the premixed CNG mixture and reported flame propagation from the cylinder wall to the center of the combustion chamber.

Although prior literature has explored many aspects of DF combustion, only a few studies [49,55–59] have attempted to characterize DF combustion progression, modes and mechanisms after the start of combustion. Rochussen et al. [56] conducted DF combustion experiments in a single-cylinder full-metal engine and characterized DF combustion by identifying the combustion stages based on the first derivative of apparent heat-release-rate (HRR) data. Wang et al. [59] decouple the combustion processes of NG and diesel by comparing HRR curves. A small pilot-diesel quantity was noted to burn in premixed mode, whereas a sufficiently large pilot-diesel quantity enabled the mechanism to transit from premixed to diffusion mode.

As shown in the previous analyses, the operating limitations of DF combustion at low and high loads are still unresolved and our fundamental understanding of DF combustion remains inadequate. It is known that gaseous fuel significantly influences combustion. In addition, the physical and chemical interactions between pilot and gaseous fuel throughout the entire combustion process play an important role in affecting the overall combustion performance. However, optical studies of DF combustion progression and characterization of underlying combustion mechanisms after combustion start have been scarcely reported. The current study covers this DF research gap by investigating the progression of DF combustion. In this study, diesel-methane DF combustion is examined optically using a wide range of methane equivalence ratios, initial-charge temperature, and pilot-diesel ratios. The thermodynamic analysis and features of high-speed NL color images are used to characterize the combustion process. Primarily, this study aims to (1) identify the transition between DF combustion stages and justify the observed combustion mechanisms based on the features

Table 1

Test engine specifications.

Engine type	4 stroke, single cylinder optical engine Borosilicate window, $d_{optical access} = 65 \text{ mm}$					
Engine speed	1400	y optical access				
Displacement volume	1.4 L					
Bore \times Stroke	111 imes 145mm					
Effective compression ratio	13.5:1					
Intake Air Pressure	1.13 bar					
In cylinder swirl	2.7					
Pilot injection system	Bosch solenoid CNR13-20 common rail					
Injector no. of	$2 \times 0.138 \mathrm{mm}$ (Asymmetric)					
holes \times diameter						
Pilot injection pressure	870 bar					
Port fuel injection system	$2 \times \text{EG2000 gas injectors}$					
0 CAD	'Firing' TDC					
Valve timing		Inlet	Exhaust			
	Opening time	356 CAD	150 CAD			
		ATDC	ATDC			
	Closing time	-155 CAD	-340 CAD			
		ATDC	ATDC			

observed in NL images, (2) demonstrate the effect of operating conditions on DF combustion characteristics, and (3) investigate IDT and combustion duration.

2. Experimental setup and procedure

2.1. Engine test facility

In the present study, experiments are conducted in a heavy-duty optical engine based on an in-line six-cylinder AGCO-84 AW1 diesel engine that has been modified for optically accessed single-cylinder DF operation. The dimensions and technical specifications of the optical engine are summarized in Table 1. The engine is equipped with necessary parallel systems, such as engine heating, direct and port fuel injection, and an electro-hydraulic valve actuation (EHVA) system, as illustrated in Fig. 1. The EHVA system [60] consists of hydraulic pistons, proportional valves and position sensors for enabling the engine to perform variable valve timing and lifts. The intake-air system is specifically designed for DF operations where gaseous fuel is injected into the intake-air manifold by two gas injectors during the intake stroke. The intake-air system consists of an air compressor, PID control valve, air heater and a Rheonik-015 Coriolis mass flow meter with an accuracy of 0.5% for the flow rate. The air mass flow and temperature were digitally monitored and controlled by an engine control unit (ECU) based on the national instrument field-programmable-gate-array (NI-FPGA). All the systems and the engine speed controller are electronically instrumented with ECU and programmed to enable all parameters to be flexibly controlled.

The pilot diesel is injected by a modified standard nozzle, which is clogged for two asymmetric holes. The mass flow of the pilot-diesel is measured by conducting a separate series of tests in a single-cylinder full-metal research engine, a counterpart of the optical engine, for the same injector using similar injection parameters and TDC conditions as those investigated in the present study. The two sprays emerging from the nozzle are located in the left half of the combustion chamber (CC), as illustrated in a schematic field view of CC in Fig. 2. The adopted asymmetric spray orientation allows more distinct visualization of flame propagation in the methane-air mixture than could be provided by a fully symmetric multi-hole nozzle. Furthermore, Fig. 2 shows the actual dimension of the CC and depicts a portion of the CC, as visualized through the optical window at TDC along with the location of the inlet



Fig. 1. Schematic of the test engine setup.



Fig. 2. Schematic field view of the combustion chamber (CC) at 0-CAD (TDC). (Left): provides the dimensions and the visible field view of the CC. (Right): provides the actual location of pilot-diesel sprays, inlet and exhaust valves and pilot-diesel injector as seen through the camera.

and exhaust valves. There is a significant clockwise (CW) swirl inside the cylinder, which affects the mixing of the fuel and the oxidizer [61]. More details about the swirl number for the same cylinder head can be found in [62,63].

2.2. Operating method

The experiments are conducted in the optical engine at a load of 9-10 bar IMEP at 1400 rpm, in which 99.9% pure methane $(LHV_{methane} = 50 MJ/kg)$ is applied as the primary fuel and commercial diesel (EN590) (LHV_{diesel} = 43.1 MJ/kg) as a pilot fuel. In all investigated test points, methane is injected into the intake manifold by the gas injectors at 355 CAD ATDC, which is then ignited by pilot diesel, injected at $\theta_{SOI} = -$ 15 CAD ATDC (electronic). The actual sprays are detected to emerge from the nozzle at -11 CAD ATDC (actual). During all experiments, a skip fire protocol is adopted due to limitations stemming from the mechanical properties of the piston's optical window. Each combustion cycle is followed by six skip-fired cycles, and each test point is run for 150 cycles, of which 22 cycles are combustion cycles. The ambient conditions are maintained by pre-heating the cylinder head and liner through the water jacket at 353 K and continuously providing heated charge air. Due to cyclic variations observed in the combustion cycles, only the last 10 combustion cycles are computed for averaged in-cylinder pressure. A PiezoStar pressure sensor (Kistler type 6125C) is employed in combination with a signal amplifier (Kistler type 5011) to measure the in-cylinder pressure of a combustion event. The undesired noise from recorded raw-pressure data (unfiltered) is filtered by utilizing a low-pass Butterworth filter algorithm. An example of unfiltered and filtered pressure data are presented in Fig. 3.

The filtered pressure data sampled at every 0.2 CAD is used to



Fig. 3. An example of filtered pressure data obtained by filtering raw pressure data with low-pass Butterworth filter.

Table 2				
Specification	of the	image	acquisition	system.

Camera type	Photron FASTCAM APX-RS 250 K, High-speed color camera with Bayer filter
Objective-lens	Nikon AF-Nikkor 28–70 mm f/3.5
Frame rate	9000 fps
Shutter speed	1/30,000 s
Image resolution	512×512

calculate the apparent heat release rate (HRR)¹ and accumulative heat released (AccQ) during the combustion process using Eqs. (1) and (2), respectively [64]. The pilot ratio (P_R) and methane equivalence ratio (ϕ_{CH4}) in the charge are calculated by means of Eqs. (3) and (4), respectively.

$$HRR = \frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dP}{d\theta}$$
(1)

$$AccQ = \int \frac{dQ}{d\theta} d\theta$$
 (2)

$$P_{\rm R} = \frac{m_{\rm diesel} \times LHV_{\rm diesel}}{\dot{m}_{\rm methane} \times LHV_{\rm methane} + \dot{m}_{\rm diesel} \times LHV_{\rm diesel}}$$
(3)

$$\phi_{CH4} = \frac{(AFR_{CH4})_{stoichiometric}}{(\dot{m}_{air}/\dot{m}_{CH4})_{actual}}$$
(4)

A high-speed color camera (Photron Fastcam APX-RS 250 K) equipped with a Nikon AF-Nikkor 70 mm/3.5 objective lens records spatially and temporally resolved NL images. For all the test points under scrutiny, a shutter time of 1/30,000 s in combination with an aperture $f^{\#} = 3.5$ is adopted. The color images are recorded at a maximum field view resolution of 512×512 pixels at a frame rate of 9000 fps. The specification of the camera system is presented in Table 2, and spectral response curves of the RGB channels are presented in the Appendix A.

2.3. Test matrix

The experimental test parameters for CASE-A and CASE-B are outlined in Table 3. In CASE-A, four methane equivalence ratios (ϕ_{CH4}) are investigated at constant pilot injection duration (constant mass/cycle), where ϕ_{CH4} is obtained by varying the methane mass flow. Each ϕ_{CH4} is subjected to three different initial-charge temperatures (299 K, 315 K, and 325 K), which result in isentropic compression temperatures at TDC (T_{TDC}) as 744 K, 780 K, and 804 K, respectively. It should be noted that the real temperatures at TDC would be different from the aforementioned isentropic TDC temperatures, where the T_{TDC} s are calculated from a real motored pressure ratio. For the purpose of comparison, the

$$^{1}\gamma = \frac{CP}{CV} = 1.35$$
 for air.

Table 3

φ_{CH4}	φ_g	Diesel [mg/cycle]	P _R [%]	Methane [mg/cycle]		IE [kJ/cycle]		IMEP [bar]			η _{th} [%]				
CASE A	A: Study o	f impact of methane eq	uivalence rat	io at differe	nt T _{TDC}										
				744 K	780 K	804 K	744 K	780 K	804 K	744 K	780 K	804 K	744 K	780 K	804 K
0.57	0.71	9.52	17 ^a	40.46	40.46	40.94	2.44	2.44	2.46	8.86	8.7	8.72	50.8	50	49.8
0.63	0.74	9.52	15.5 ^a	42.93	45.22	47.12	2.56	2.67	2.76	9.43	9.84	9.65	51.6	51.6	49
0.72	0.83	9.52	14 ^a	49.98	49.98	52.36	2.90	2.90	3.0	10	9.54	9.9	48.3	46	46.2
0.85	0.98	9.52	12.2 ^a	58.55	58.31	60.69	3.33	3.33	3.44	10.1	10.0	10.3	42.5	42.1	42
CASE F	CASE B: Study of impact of pilot ratio at constant T														
	. otady o	j shipace oj plice ralio u		744 K											
0.55	0.66	9.52	16.5	41.88			2.51			8.72			48.65		
0.55	0.72	14.28	22.6	42.36			2.74			9.1			46.5		
0.55	0.75	16.66	25.6	42.12			2.83			9.41			46.55		
0.55	0.76	18.1	27.5	41.65			2.87			9.44			46		

Operating test parameters for the study of CASE-A and CASE-B, where T_{TDC}s, assuming isentropic compression, are used for the purpose of comparison.

^a Averaged P_R value.

Table 4

Measured uncertainties in the operating test parameters.

Parameters	Uncertainties
Engine Speed	1400 ± 3.88 rpm
Diesel flow rate	\pm 0.038 kg/h
Methane mass flow rate	± 0.43 kg/h
Air mass flow rate	± 0.29 kg/h
Intake air pressure	$1.13 \pm 0.025 \text{bar}$
Initial air temperature	± 0.8 K

T_{TDC}s will henceforth be used to represent charge temperatures. In CASE-B, four different pilot ratios (P_B) are obtained by varying the pilot injection duration at a constant methane equivalence ratio $(\phi_{CH4} = 0.55)$ and charge temperature ($T_{TDC} = 744$ K). In order to estimate the fluctuations and the level of accuracy in the measured values of experimental parameters, an uncertainty analysis is performed. Uncertainty in the measured value for a parameter is defined as the ratio of standard deviation from the measured value to the square root of a total number of measurements. The uncertainties in the parameters are presented in Table 4.

Standard Deviation(S) =
$$\sqrt{\frac{\sum_{i=1}^{N} (x_i - \bar{x})^2}{(N-1)}}$$
 (5)

Where,

 $\bar{\mathbf{x}} = \frac{\sum \mathbf{x}_i}{N},$ N = total number of measurements = 100



2.4. Image processing

The broadband NL color images are processed using Matlab algorithms for image cropping and temporal synchronization with engine CAD, and identifying reaction zone boundaries. In addition, the images are processed for an intensity distribution image and an overall spatiotemporal averaged intensity plot for cycle-to-cycle variability statistics. A background image is subtracted from each color image, and reaction zone edges are identified based on the maximum intensity gradient operator from the blue and red channels owing to the corresponding spectral sensitivity (illustrated in the Appendix A). An intensity distribution image provides a way to characterize a combustion event spatially at an instant based on registered pixel intensity counts. A Matlab algorithm allocates the recorded intensities from gravscale image into 12 logarithmically distributed bins in a range of 0-255 counts (for an 8-bit camera), which highlights low intensities (gas combustion) and spatially distinguishable intensity regions. An example of an intensity distribution image is illustrated in Fig. 4(A).

The spatio-temporal intensity $I_{(\frac{r}{R},CAD)}$ plot averaged over 10 individual combustion cycles is constructed by integrating the recorded intensity over a circumference of a circle of discretized radius (r_1, r_2, r_3) $r_3...r_n$) as presented in Fig. 4(B). It can be explained as $I_{(\frac{r}{R},CAD)} = \frac{\oint_{0}^{2\pi} I(\frac{r}{R},\alpha,CAD) d\alpha \times \max\left(\max_{0 \le \alpha \le 2\pi} I(\frac{r}{R},\alpha,CAD)\right)}{\oint_{0}^{2\pi} pix \times d\alpha}, \text{ where } r = \text{discretized}$ radius of the combustion chamber, R = radius of the combustion chamber, CAD = crank angle degree, α = an angle in polar coordinate



Fig. 4. (A): Intensity distribution of an individual image at an instant during a combustion event, (B): (left) illustration of the discretization of the combustion chamber radius (R) into a sequence of r_1 , r_2 , r_3 ..., r_n , (right) spatiotemporal intensity plot ($I_{(\frac{1}{T},CAD)}$) for the averaged intensity over 10 individual combustion cycles.



Fig. 5. Illustration of DF combustion conceptual model with overlapping combustion stage at high and low load conditions [14].

system, and Pix = projected pixel size.

2.5. DF combustion interpretation

DF combustion (burning of two fuels of different reactivity in the same combustion cycle) is a fundamentally complex process to analyze, as it depends on both the physical and chemical properties of pilot-fuel sprays, as well as the gas concentration in the premixed mixture. In diesel-methane DF combustion, pilot diesel is injected during the compression stroke close to the TDC, which auto-ignites and provides an energy deposit for the ignition of the premixed methane-air mixture. As a result, the characteristics of the heat release depict relatively complex physical and chemical interactions [14]. The process of DF combustion can be defined as a combination of three overlapping combustion stages, as proposed in Karim's DF combustion conceptual model [14] and illustrated in Fig. 5:

- 1. First-stage combustion (I) is due to pilot diesel and even some entrained premixed methane-air mixture combustion.
- 2. Second-stage combustion (II) is due to combustion of the premixed methane-air mixture in the immediate surrounding of the pilot diesel.
- 3. Third-stage combustion (III) is due to turbulent flame propagation in the remaining unburned diluted methane-air mixture.

Although DF combustion can be explained theoretically, it is challenging to identify and separate the overlapping combustion stages from experimental HRR data. Few studies [56-59] have characterized the DF combustion process based on combustion stages and modes, calling for further investigations. In the present study, a second derivative of HRR has been adopted to identify and separate the three overlapping combustion stages, where combustion mechanisms are justified by features of NL color images. The second derivative of HRR highlights the inflection points², which are rather difficult to identify by the first derivative and require further thresholding. Fig. 6 shows a typical DF combustion HRR profile along with the accumulated heat release (AccQ) and the second derivative of HRR (SDHRR), where the local maximum values of SDHRR are θ_1 , θ_2 , θ_{3a} and θ_{3b} , representing first-, second- and third-stage combustion, respectively. The trends of the flame area (FA) and the intensity (I) in combination with NL images are presented in Fig. 7. In order to systematically identify and isolate the combustion stages, a MATLAB algorithm is implemented in the SDHRR data. The philosophy and logic behind the selection of each combustion stage in combination with the features of NL images are presented in the following subsections.



Fig. 6. Characterization and identification of three overlapping combustion stages of a typical DF combustion representative case by the second derivative of HRR, where filtered cylinder pressure for the HRR is presented in Fig. 3. A detailed version of the plot with cylinder pressure and first derivative of HRR is provided in the Appendix A.

2.5.1. Definition of ignition delay time

The ignition delay time (IDT) starts from SOI and ends at the first peak of SDHRR (after HRR becomes positive). The IDT can be expressed as

$$IDT = \theta_1 - \theta_{SOI} \tag{7}$$

$$Q_{id} = \int_{\Theta_{CM}}^{\sigma_1} HRRd\theta$$
(8)

By analyzing SDHRR, the first local maxima (Θ_1) appears at 3 CAD ATDC, indicating the end of IDT and the start of first-stage combustion. Moreover, $Q_{id} = 32.9661$ J is 1.71% of the total heat released during the cycle, and it can be observed that for all the investigated test points, the Q_{id} always remains below 2%. Furthermore, the end of IDT is visualized as a faint flame in Fig. 7 at 3.11 CAD ATDC (NL image#2).

2.5.2. First-stage combustion (I)

First-stage combustion starts at θ_1 , which is achieved after the HRR magnitude becomes positive and ends at θ_2 . It can be expressed mathematically as follows:

$$\theta_1 = \theta|_{\max} \frac{d^2(\mathrm{HRR})}{d\theta^2}, \{\theta|\theta_p > \theta < \theta_2\}$$
(9)

 $^{^{2}}$ The combustion stage is considered to change when HRR increases at an instant with an increasing rate. Such transition points are identified as corresponding to those inflection points where concave up is the maximum.



Fig. 7. Characteristics of the NL images of a typical DF combustion case, where combustion stages are labelled as presented in Fig. 6 based on the second derivative of HRR. The numbering of the FA markers correspond to the numbering of NL images with edges (white edges are identified from the blue channel, and red edges from the red channel. The red edges depict locally fuel-rich zones due to pilot-diesel combustion). (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

 $\theta_{\rm P} = \text{CAD}$ where HRR magnitude becomes positive =

 $\theta|_{(|\mathrm{HRR}| > 0), \{\theta| \ \theta > \ \theta_{\mathrm{SOI}}\}}$ (10)

$$Q_1 = \int_{\theta_1}^{\theta_2} HRRd\theta \tag{11}$$

Auto-ignition of the pilot diesel is a dominant combustion mechanism during the first stage. Due to the long IDT observed in the investigated test points, most of the small pilot-diesel quantity burns in premixed mode (also reported in [59,65]) along with the entrained premixed methane-air mixture. First-stage combustion is identified as initial flame kernels, and by comparing the NL images in Fig. 7 at 3.11–5.90 CAD ATDC (NL image#2-4), it is apparent that the flame is in its early growth phase, and suggests a premixed combustion mechanism. The injected pilot diesel for the presented case has an energy of 414.12 J, whereas the heat released during the first-stage combustion is $Q_1 = 429.5 J$. This accessory amount of heat release indicates a combined combustion of the pilot diesel and entrained methane-air mixture during the first stage, where unburned pilot diesel from the first stage burns with locally fuel-rich zones (indicated by red edges) during the second stage.

2.5.3. Second-stage combustion (II)

Second-stage combustion is identified as the second peak in the SDHRR after the HRR becomes positive. The mathematical expression of the second stage can be given as

$$\Theta_{2} = \Theta|_{\max \frac{d^{2}(\text{HRR})}{d\Theta^{2}}, \{\Theta|\Theta_{1} < \Theta < \Theta_{3a}\}}$$
(12)

$$Q_2 = \int_{\theta_2}^{\theta_{3a}} HRRd\theta \tag{13}$$

In this stage, a simultaneous overlap is observed between locally

fuel-rich zones and the premixed flame-propagation combustion mechanism, where flame propagation occurs in the methane-air mixture within the immediate surroundings of the pilot-diesel. It is expected that flame propagation intensifies and the remaining unburnt pilotdiesel from the first stage completely burns during this stage, leading to a rapid combustion. By analyzing the SDHRR, second-stage combustion is estimated to start at 7.2 CAD ATDC for this case and can be visualized as a rapidly growing FA from the NL images at 5.90–7.29 CAD ATDC (NL image#4–5) in Fig. 7. The heat releasedQ₂ = 787.1 J at this stage is the highest of all the stages, as has also been reported in [14].

2.5.4. Third-Stage combustion (III)

Third-stage combustion is regarded as a combination of high- (θ_{3a}) and low- (θ_{3b}) intensity combustion regimes, which can be identified by implementing the following mathematical expression:

$$\theta_{3a} = \theta|_{\max \frac{d^2(\text{HRR})}{d\theta^2}, \{\theta|\theta_2 < \theta < \theta_{3b}\}} \theta_{3b} = \theta|_{\max \frac{d^2(\text{HRR})}{d\theta^2}, \{\theta|\theta > \theta_{3a}\}}$$
(14)

 θ_{90} = CAD where 90% of heat is released

$$Q_{3} = Q_{3a} + Q_{3b} = \int_{\theta_{3a}}^{\theta_{3b}} HRRd\theta + \int_{\theta_{3b}}^{\theta_{00}} HRRd\theta$$
(15)

Flame propagation in the remaining unburned methane-air mixture is a dominant combustion mechanism during the third stage. However, flame quenching and dilution during the low-intensity combustion regime θ_{3b} (far into the expansion stroke) may cause an ambiguous combustion mode. The third stage of combustion starts $at\theta_{3a} = 11.8$ CAD ATDC and ends at $\theta_{90} = 23.2$ CAD ATDC, whereas the low-intensity combustion regime starts $at\theta_{3b} = 16.6$ CAD ATDC. The heat releases slowly after θ_{3b} , and the magnitude is the lowest, as shown in Fig. 6. The high-intensity combustion regime is identified as sustained turbulent flame propagation from NL images at 12.86–15.65 CAD ATDC



Fig. 8. Averaged HRR and AccQ curves for the study of CASE-A (effect of ϕ_{CH4} and T_{TDC} on DF combustion). Combustion stages are labelled on the AccQ curves based on the second derivate of the respective HRR.

(NL image#9-11) in Fig. 7. The FA during the high-intensity combustion regime barely changes; however, the change in intensity indicates progression of the combustion. The FA and intensity starts to decrease after 17.04 CAD ATDC (NL image#12), indicating a shift to a low-intensity combustion regime.

In short, the SDHRR method robustly identifies the three overlapping DF combustion stages from a representative experimental case in overall accordance with Karim's conceptual model [14]. In addition, the features in recorded NL images provide a qualitative insight into DF combustion progression and underlying mechanisms at different stages. Section 3 evaluates the investigated experimental data and characterized it based on the SDHRR methodology demonstrated in Section 2.5.

3. Results and discussion

This section discusses the main results of the study, and it is divided into two subsections: CASE-A examines the effects of the methane equivalence ratio ($\varphi_{\rm CH4}$) in combination with initial-charge temperature ($T_{\rm TDC}$) and CASE-B the impact of the pilot ratio ($P_{\rm R}$) on dieselmethane DF combustion. Characteristics of heat release, such as IDT and combustion duration, along with the NL images are analyzed to identify the combustion process and highlight the underlying combustion mechanisms.

3.1. CASE-A: Effect of ϕ_{CH4} and T_{TDC}

Fig. 8 shows curves of averaged HRR and AccQ at different ϕ_{CH4} , ranging from lean to rich conditions, where each ϕ_{CH4} is subjected to different T_{TDC}, as listed in Table 3. In this research, HRR profiles with multiple peaks (M-shaped) are observed that vary in magnitude depending on the ϕ_{CH4} and P_R . Such profiles of HRR for DF combustion have been reported and characterized in several others studies [56–59]. For all the presented HRR data, the shape of the HRR curve is qualitatively prone to change from multiple peaks (M-shape) to a quasisingle peak (bell-shaped) with an increase in $\varphi_{\text{CH4}},$ while the combustion stages seem to remain intact. The HRR data are analyzed based on the second derivative method to deepen the study and comprehend the influence of increasing ϕ_{CH4} on DF heat release characteristics. The combustion stages as $\theta_1,\,\theta_2,\,\theta_{3a},\,\theta_{3b}$ along with θ_{90} have been marked on the respective AccQ curves for each ϕ_{CH4} case at different T_{TDC} . Furthermore, Fig. 9 depicts the fraction of heat released during each combustion stage, calculated from Eqs. (11), (13), and (15) for Q11, Q2, and Q₃, respectively.

Fig. 9 shows that the burnt charge mixture fraction decreases during the first stage (I) as the ϕ_{CH4} increases for each particular T_{TDC} -case, whereas it marginally increases with an increase in the T_{TDC} . The addition of methane has an inhibiting effect on pilot-diesel ignition [33–35,46–48,54], thereby causing a decrease in charge mixture



Fig. 9. Burnt charge mixture fraction during each combustion stage; first-stage (Q_1 /AccQ), Second-Stage (Q_2 /AccQ) and Third-Stage (Q_3 /AccQ), for the entire range of test points investigated in CASE-A at different ϕ_{CH4} and T_{TDC} .

reactivity and, in turn, a lower burnt mixture fraction. On the other hand, an increase in T_{TDC} helps increase the mixture reactivity [14,42,66] that leads to an increase in the burnt fraction. The NL images presented in Fig. 10 at θ_1 show the start of combustion as two distinct initial flame kernels originating from two pilot-diesel sprays, as located in Fig. 2. Although these flame kernels are located around the sprays, the site has moved due to the in-cylinder clockwise swirl. Autoignition of pilot diesel is a dominant combustion mechanism during the first stage, where the pilot diesel (along with the entrained methane-air mixture) is visualized as burning mostly as a premixed combustion in NL images for most of the experiments. The pilot-diesel burns in the premixed mode due to the short pilot injection duration, long IDT (as depicted in Fig. 12), and TDC temperature below 900 K, as also reported in [59,65]. Furthermore, the results show that at $\phi_{CH4} = 0.57$ (low-load condition), most of the heat releases during the first stage, and most of the diesel is considered to be consumed during this same stage. In contrast, at higher ϕ_{CH4} (high-load conditions), most of the heat is released during the second stage (II) as a high burnt fraction compared to other combustion stages (Fig. 9).

Fig. 9 also shows that the charge mixture burnt fraction increases during the second stage with an increase in both ϕ_{CH4} and T_{TDC} . However, the effect of T_{TDC} on the burnt fraction is insignificant. During this stage, the mean effective temperature of the mixture already increases beyond 900 K due to first-stage combustion, and premixed methane-air mixture ignition occurs in the surrounding of the pilot ignited plume, causing flame propagation in the premixed methane-air mixture. At higher ϕ_{CH4} , since a greater quantity of reactive premixed methane-air mixture is available around the pilot-diesel, the burnt fraction increases with an increase in ϕ_{CH4} . The increased temperature may improve the mixture reactivity and flame speed [14,66,67], with the results subsequently showing a marginal rise in burnt fraction with an increase in $T_{\text{TDC}}.$ The off trends in the T_{TDC} comparison data at a particular ϕ_{CH4} might be due to cycle-to-cycle variations in peak HRR magnitude and engine operating-parameter variations, for which an uncertainty analysis has been provided in Section 2.3. The second stage is visualized in Fig. 10 at θ_2 as a simultaneous combination of locally fuel-rich zones and premixed combustion mechanisms. It is assumed that orange-yellowish regions visualized around the periphery of the combustion chamber at θ_2 may result from a locally rich premixed mixture of pilot-diesel rather than diffusion combustion at high ϕ_{CH4} (lower O₂ concentration) and relatively low charge temperature.

Since increased temperature plays a significant role in enhancing the evaporation rate of pilot-diesel [46,66], no or fewer locally fuel-rich zones around the periphery are visualized due to sufficient availability of O₂ concentration for $\phi_{CH4} = 0.57 \& 0.63$ at higher temperatures. On the other hand, for $\phi_{CH4} = 0.72 \& 0.85$ even at elevated temperatures, locally fuel-rich zones are intense owing to insufficient O₂ concentration, leading to greater soot formation. Additionally, the fuel-rich zones at the center of the combustion chamber seem to result from injector dribbling [49,50,68]. More details about dribbling and flame



Фсн4 = 0.72

(caption on next page)

Fig. 10. Representative NL images and intensity distribution images immediately after the start of the combustion stages for the entire range of test points investigated in the study of CASE-A (effect of ϕ_{CH4} and T_{TDC} on DF combustion).

propagation in DF combustion can be found in our previous research [63].

The presence of locally fuel-rich zones during the second stage indicates an overlapping of the first and second stages [14]. At $\phi_{CH4} = 0.57$, most of the pilot-diesel is considered to be consumed during the first stage. However, at higher ϕ_{CH4} , pilot-diesel does not deplete completely during the first stage due to lower mixture reactivity, and the remaining unburnt pilot-diesel appears in the subsequent combustion stages. Qualitatively, the higher the amount of locally fuel-rich zones during the second stage, the higher the level of overlap between the stages. Therefore, at higher ϕ_{CH4} (high-load conditions), the HRR curve shape is prone to change to a quasi-single peak (bell-shaped) curve. Furthermore, an increase in T_{TDC} tends to increase the luminosity count of each combustion stage, as illustrated in the intensity distribution images for the corresponding NL-images presented in Fig. 10 for all the investigated test points.

During the third stage (III), the burnt charge mixture fraction increases marginally with an increase in ϕ_{CH4} , whereas it decreases with an increase in T_{TDC} , as shown in Fig. 9. The heat released from the remaining unburnt premixed methane-air mixture is caused by flame propagation during this stage (III). At higher ϕ_{CH4} , an additional amount of methane is available to burn in the third stage. Therefore, a marginal increase is observed in the burnt fraction. An increase in T_{TDC}

at a certain ϕ_{CH4} tends to burn an extra amount of methane during the first and second stages. This causes a decrease in the quantity of remaining unburned methane during the third stage, which results in a lower burnt fraction for the charge mixture. The results show that the burnt fraction of the mixture during the third stage is a function of a remaining quantity of unburned methane-air mixture. The start of the third stage at θ_{3a} , shown in Fig. 10, is visualized as a combustion chamber engulfed with premixed flames. For all the presented test points, pilot diesel is entirely consumed by the end of the second stage. However, at $\phi_{CH4} = 0.85$, pilot-diesel burns slowly, and locally fuelrich zones due to pilot diesel appear during the third stage, thus indicating an overlap between all three stages. This implies that at high load and high temperature conditions, a single peak (bell-shaped) HRR curve can be achieved. Furthermore, an increase in T_{TDC} causes rapid combustion due to long IDT and an increase in combustion luminosity, as illustrated in the intensity distribution images shown in Fig. 10.

Based on the recorded intensity of a DF combustion event, Fig. 11 demonstrates an overall spatio-temporal progression of NL with different r/ R ratios as a $I_{(\frac{r}{R},CAD)}$ plot for the CASE-A. The spatio-temporal averaged intensity ($I_{(\frac{r}{R},CAD)}$) is calculated as described in Section 2.4, where the integrated intensity at a r/R ratio for 10 images from 10 combustion cycles are averaged at a single CAD. The plot provides overall statistics based on intensity evolution. It is apparent that intensity close to the periphery



Fig. 11. Spatio-temporal intensity plot for CASE-A at $\phi_{CH4} = 0.57 \& 0.85$ and at $T_{TDC} = 744 K \& 804 K$. It is seen that at $\phi_{CH4} = 0.85$, the locally fuel-rich zones around the periphery of the combustion chamber are more intense than those observed at $\phi_{CH4} = 0.57$.



Fig. 12. (Left) IDT trend for the CASE-A study, (right) CD trend for the CASE-A study.

region of the combustion chamber increases with an increase in $\varphi_{\rm CH4}$ due to locally fuel-rich zones of pilot diesel. In addition, the luminosity of these locally fuel-rich zones increases with an increase in $T_{\rm TDC}$, indicating high-temperature soot incandescence. Furthermore, the relative extension of the intensity at $T_{\rm TDC}=804\,{\rm K}$ in a late combustion cycle is due to sustained flame propagation and the presence of locally fuel-rich zones. The high intensity regions at the center are due to dribbling from the injector, as described earlier in this section.

Ignition Delay Time and Combustion Duration (CASE-A)

IDT is calculated using the SDHRR method explained in Section 2.5.1, and the IDT trends observed for CASE-A are summarized in Fig. 12(left). The IDT trends show that with an increase in ϕ_{CH4} , the IDT increases due to the increasing inhibiting effect of methane on pilotdiesel ignition [33-35,46-48,54]. This trend can also be visualized in NL images at θ_1 as an initial flame kernel for different ϕ_{CH4} . Furthermore, IDT trends show a decrease in IDT with an increase in T_{TDC} owing to the increased Arrhenius rate of the pre-ignition chemical process as well as the physical processes in the enhanced spray [46,66,69]. It is also evident from NL images at θ_1 for different T_{TDC} that the onset of ignition occurs earlier when the temperature increases. IDT is an essential characteristic of DF combustion and seems to influence significantly the burnt fraction of the charge during the first stage (I). An explanation for the off-trend apparent in the IDT trend at $\phi_{CH4} = 0.72$ probably lies in the combustion chemistry (NTC behavior [70]), which would be a point of interest for future studies.

Fig. 12(right) shows a trend in overall combustion duration (CD₉₀) between θ_1 and θ_{90} for the entire range of experiments presented in CASE-A. As expected, CD increases with an increase in ϕ_{CH4} . However, CD decreases at $\phi_{CH4} = 0.85$, which might be attributed to the presence of locally fuel-rich zones in the second and third stages. These pilot-diesel zones widen in the entire combustion chamber, burn slowly until the premixed methane-air mixture burns rapidly towards the end of the second stage and during the high-intensity third stage (θ_{3a}), as can be seen from the NL images in Fig. 10. In addition, the methane-air mixture in the low-intensity third stage (θ_{3b}) is no longer diluted and burns with a comparatively high flame propagation speed. Furthermore, the CD trend shows an expected decrease in the CD with an increase in T_{TDC} .

3.2. CASE-B: Impact of P_R

This section examines the effect of pilot ratio (P_R) on DF combustion. Fig. 13 shows averaged HRR and AccQ curves for CASE-B (as tabulated in Table 3) at different P_R investigated at constant $\phi_{CH4} = 0.55$ (low-load condition) and T_{TDC} = 744 K. For the presented HRR data,

the shape of the HRR curves seems to change to a quasi-single peak (bell-shaped) as the P_R increases in the charge mixture. The HRR data are analyzed based on the SDHRR method as presented in Section 2.5, and the combustion stages have been marked as θ_1 , θ_2 , θ_{3a} , θ_{3b} along with θ_{90} on respective AccQ curves at different P_R . In addition, Fig. 14 illustrates the fraction of heat released during each combustion stage, calculated from Eqs. (11), (13), and (15) for Q_1 , Q_2 , and Q_3 , respectively.

Fig. 14 shows that the burnt charge mixture fraction increases during the first stage (I) with an increase in P_R . The increase in pilotdiesel quantity within the charge accumulates into the high reactivity zones. This accumulation causes a greater entrainment of the premixed methane-air mixture due to improved pilot injection characteristics, leading to increased penetration and spray volume [71]. In addition, this provides a greater multitude of ignition centers and a larger reaction zone [14,39]. Therefore, more heat is released at ignition, and the burnt charge mixture fraction increases with an increase in P_R during the first stage. The NL and intensity distribution images presented at θ_1 in Fig. 15 illustrate the start of combustion as two initial flame kernels where premixed combustion constitutes the dominant combustion mechanism. Even though CASE-B is investigated at $\phi_{CH4} = 0.55$ (lean



Fig. 13. Averaged HRR and AccQ curves for the study of CASE-B (Impact of $P_{\rm R}$ on DF combustion) at $\varphi_{\rm CH4}$ = 0.57 and $T_{\rm TDC}$ = 744 K. The combustion stages are labelled on the AccQ curves based on the second derivate of the respective HRR.



Fig. 14. Burnt charge mixture fraction during each combustion stage for the entire range of test points investigated in CASE-B at $\phi_{CH4} = 0.57$ and $T_{TDC} = 744$ K.

condition), pilot-diesel has been visualized to burn as locally fuel-rich zones at the periphery of the combustion chamber towards the end of the first stage with an increase in P_R. It is observed that these locally fuel-rich zones are more prominent and luminous at $P_{\rm R} = 25.6\%$ and $P_{\rm R} = 27.5\%$ than at lower $P_{\rm R}$ test points. At $P_{\rm R} = 27.5\%$, pilot injection duration is sufficiently long to allow pilot spray driven combustion to be recognized around 4 CAD ATDC in NL image videos. This implies that the pilot-diesel injected during the end part of the injection burns as diffusion combustion rather than premixed rich conditions. Occurrence of the diffusion combustion may lead to a comparatively smaller entrainment of the premixed methane-air mixture and a lower burnt fraction during the first stage, while increasing the burnt fraction during the second stage (II). Similar results have also been reported by Rochussen et al. [56]. The results show that the size of the initial flame kernel (as illustrated in Fig. 15) increases with an increase in P_B due to rapid combustion. In addition to this, the burnt charge mixture fraction increases during the first stage as long as pilot diesel undergoes no diffusion combustion.

Fig. 14 depicts the way in which the burnt charge mixture fraction decreases with an increase in P_R during both the second (II) and third (III) stages, since more heat tends to be released in the first stage, thereby decreasing the quantity of the remaining unburnt premixed

methane-air mixture available in stages II and III. However, as can be seen from NL images (Fig. 15) at θ_2 and θ_{3a} , combustion appears to be a simultaneous combination of locally fuel-rich zones and premixed combustion with an increasing P_R . At $P_R = 16.5\%$, pilot-diesel is regarded as fully consumed during the first stage with no locally fuel-rich zones being visualized during any stage. In contrast, at higher P_R test points, locally fuel-rich zones appear in subsequent combustion stages and cause an overlap between two or even all combustion stages. At $P_R = 27.5\%$, locally fuel-rich zones start to appear towards the end of the first stage and keep appearing in the subsequent combustion stages as well as in the late combustion cycle. An increase in P_R at a constant ϕ_{CH4} results in an overlap between the stages, and at high P_B, a quasisingle peak (bell-shaped) HRR may be achieved. It is evident from the NL and intensity distribution images in Fig. 15 at θ_2 and θ_{3a} that with an increase in P_R, combustion becomes rapid and more luminous due to the presence of locally fuel-rich zones, suggesting soot incandesce at high combustion temperatures [72]. Furthermore, increased P_B helps sustain the flame propagation that envelopes the entire combustion chamber even under lean conditions.

Fig. 16 illustrates the spatio-temporal averaged intensity $I_{(\frac{R}{R},CAD)}$ plot for CASE-B, which clearly shows that there is a weaker flame propagation at $P_R = 16.5\%$, and the center of the combustion chamber



PR = 25.6 %

Pr = 27.5 %

Fig. 15. Representative NL images and intensity distribution images immediately after the start of the combustion stages for the entire range of test points investigated in the study of CASE-B at $\phi_{CH4} = 0.57$ and $T_{TDC} = 744$ K.

is devoid of any intense flame propagation. In addition, flame does not sustain during late combustion cycle, which implies a lean flammability limit of premixed methane-air mixture under these operating conditions. Nevertheless, it is apparent that with an increase in P_R , the intensity could increase and flame propagation could be sustained under lean conditions. Furthermore, the intensity extends into the late

combustion cycle due to the presence of locally fuel-rich zones of pilot diesel.

Ignition Delay Time and Combustion Duration (CASE-B)

The investigated IDT trend outlined in Fig. 17(left) shows an insignificant decrease in IDT with an increase in P_R under lean conditions.



Fig. 16. Spatio-temporal intensity plot for CASE-B at $\phi_{CH4} = 0.57$ and $T_{TDC} = 744$ K. It is clearly seen that the flame propagation sustains steadily at $P_R = 27.5\%$ as compared to $P_R = 16.5\%$.



Fig. 17. (Left) IDT trend for the CASE-B study, (right) CD trend for the CASE-B study. (CD_{end} trends are illustrated only in CASE-B because in CASE-A study, the CD trends for both CD_{90} and CD_{3b} show however similar trends with different magnitudes).

This indicates that the additional quantity of pilot diesel injected beyond the point of ignition will not considerably help in ignition. Nevertheless, the additional pilot-diesel improves overall combustion by introducing rapid combustion and sustained flame propagation in the entire combustion chamber. In recent research [73], we have observed a significant increase in the IDT at an extremely low P_R . This points to the need for adopting a critical P_R to permit ignition and stable combustion. It is worth noting that we are not actually close to the limit of no-ignition in the present study.

Fig. 17(right) summarizes the trends of combustion duration (CD) observed in the CASE-B. CD_{90} represents an overall combustion duration between θ_1 and θ_{90} , whereas CD_{3b} represents combustion duration between θ_1 and θ_{3b} . The CD_{90} trend shows that CD increases with an increase in P_R , which is in line with the literature [39] but contrary to our expectations, as an increase in P_R leads to rapid combustion. Nevertheless, CD_{3b} shows a decreasing trend with an increase in P_R . The trends of CD_{90} and CD_{3b} point towards a diluted premixed methane-air mixture combustion and the presence of pilot-diesel sooty flames between θ_{3b} and θ_{90} that burns slowly until it quenches, causing an increase in CD_{90} with an increase in P_R .

4. Conclusions

In this study, diesel-methane dual-fuel (DF) combustion was investigated in a single-cylinder optical engine at 1400 rpm and a load of 9–10 bar IMEP for a wide range of methane equivalence ratios (ϕ_{CH4}), initial-charge temperatures (T_{TDC}), and pilot ratios (P_R). The combustion stages were identified and analyzed by adopting the second derivative of the apparent heat release rate (SDHRR) methodology. Simultaneously, the spatial and temporal features of natural luminosity (NL) color images were used to characterize the diesel-methane DF combustion process and underlying mechanisms. The main findings of this study are summarized as follows:

 The DF combustion process is identified as a combination of three stages, which are separated by the SDHRR. The first stage (I) is visualized as initial flame kernels, during which most of the pilot diesel is burning. The second stage (II) and the third stage (III) are characterized by premixed combustion of the methane-air mixture. However, Stage II usually receives a higher contribution from the pilot-diesel combustion associated with more locally fuel-rich zones than does Stage III. In addition, based on optical observations, Stage III is subdivided by the SDHRR into two regimes, namely high-intensity and low-intensity.

- 2) HRR profiles are prone to change from multiple peaks (M-shaped) to a quasi-single peak (bell-shaped) with an increase in either ϕ_{CH4} (methane quantity) or P_R (pilot quantity).
- 3) During Stage I, the burnt charge mixture fraction decreases with an increase in ϕ_{CH4} , whereas an increase in burnt charge mixture fraction is observed with an increase in either P_R or T_{TDC} . At low-load under lean conditions ($\phi_{CH4} = 0.57$), most of the heat is released during the first stage.
- 4) During Stage II, the burnt charge mixture fraction increases with an increase in ϕ_{CH4} and T_{TDC} , but decreases with an increase in P_R .
- 5) The burnt charge mixture fraction during Stage III is mainly a function of P_R and T_{TDC} . Furthermore, overlapping of the three stages is observed for high ϕ_{CH4} or P_R (extended pilot-diesel burning).
- 6) The ignition delay time (IDT) is found to be a function of ϕ_{CH4} , T_{TDC} and P_R . IDT increases with an increase in ϕ_{CH4} and decreases with an increase in either T_{TDC} or P_R . However, the effect of P_R on IDT is observed to be minimal. IDT is more sensitive to ϕ_{CH4} and T_{TDC} than to P_R . IDT has a considerable influence on the heat released during Stage I.
- 7) DF combustion duration (CD) decreases with an increase in T_{TDC} and shows an increasing trend with an increase in either ϕ_{CH4} or P_R . CD is observed to increase with an increase in P_R due to slow combustion processes during the low-intensity regime of Stage III.

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Appendix A



Spectral response curve of RGB channels of Photron FASTCAM APX-RS 250K, High-speed color camera.



A detailed version of Fig. 6 with cylinder pressure and first derivative of HRR. Characterization and identification of three overlapping combustion stages of a typical DF combustion representative-case by the second derivative of HRR.

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