A STUDY ON FLEXIBLE DUAL-FUEL AND FLEXI COMBUSTION MODE ENGINE TO MITIGATE NO, SOOT AND UNBURNED EMISSIONS

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Abstract

In the present study, an existing commercial light-duty automotive diesel engine is modified to a flexible dual-fuel engine (FDFE). The FDFE operates with different low and high reactivity dual fuel combinations under low temperature combustion (LTC) mode using combined multipoint fuel injection and common rail direct injection systems. The FDFE can smoothly transit between LTC and conventional diesel combustion (CDC) mode. FDFE combines SI and CI benefits and stands as a potential internal combustion engine for future hybrid electric options. In this study, the modified engine was operated in flexi fuel mode with methanol/diesel, methanol/biodiesel, methanol/dimethyl ether (DME), methanol/polyoxymethylene dimethyl ether (PODE), (methanol + Isobutanol blends)/diesel and (methanol+PODE blends)/diesel in LTC strategy at a different speed and torque conditions. This approach improved the brake thermal efficiency by 8%, decreased NO and soot emissions by more than 90% compared to CDC mode. The improvement in brake thermal efficiency reduced CO_2 emissions compared to CDC mode. In the FDFE engine, combustion phasing and fuel energy input are maintained as same as in CDC mode to investigate the dual-fuel effects in LTC mode over a neat diesel mode. Experimental study with energy and exergy analysis was carried out to assess the technical suitability of the FDFE as compared to the conventional diesel engine. The results proved that without relying on the after- treatment systems and fossil fuels, it is possible to reduce the NO, soot, unburnt hydrocarbon, carbon monoxide and CO_2 emissions from the diesel engine, paving the way for extending the life of the diesel engine.

Highlights

1. Flexible dual-fuel engine could be a potential option for future mobility.
2. Through flexible dual-fuel engine, current emission norms can be achieved without relying on the after-treatment systems.
3. Different fossil and renewable fuel combinations are used for experimentation.
4. Dimethyl ether and Polyoxymethylene dimethyl ether provide better thermal efficiency and near zero emissions.

Keywords

Combustion efficiency; CO_2 mitigation; Flexible Dual Fuel Engine; Low Carbon Fuels; LTC; NOx; Soot.

Abbreviations

BD	Biodiesel
BMEP	Brake Mean Effective Pressure
BTE	Brake Thermal Efficiency
CA10	Crank Angle at Which 10% of Total Energy Released
CA5	Crank Angle at Which 5% of Total Energy Released
CA50	Combustion Phasing
CD	Combustion Duration
CDC	Conventional Diesel Combustion
CE	Combustion Efficiency
CI	Compression Ignition
CO	Carbon Monoxide
CO_2	Carbon Dioxide
COV_{IMEP}	Coefficient Of Variation of Indicated Mean Effective Pressure
CRDI	Common Rail Direct Injection
DME	Dimethyl Ether
ECU	Engine Control Unit
EGR	Exhaust Gas Recirculation
ERC	Engine Research Centre
ESC	European Steady State Cycle

Evs	Electric Vehicles
FDFE	Flexible Dual Fuel Engine
FEI	Fuel Energy Input
GHG	Greenhouse Gas
HC	Hydrocarbon
HCCI	Homogeneous Charge Compression Ignition
HRF	High Reactivity Fuel
ICE	Internal Combustion Engine
ID	Ignition Delay
IMEP	Indicated Mean Effective Pressure
LPG	Liquified Petroleum Gas
LRF	Low Reactivity Fuel
LTC	Low Temperature Combustion
M	Methanol
MK	Modulated Kinetics Combustion
MPFI	Multipoint Fuel Injection
NO	Nitric Oxide
NO_2	Nitrogen Dioxide
NO_x	Oxides of Nitrogen
PCCI	Premixed Charge Compression Ignition
PHEVs	Plug-In Hybrid Electric Vehicles
PM	Particulates
PODE	Polyoxymethylene Dimethyl Ether
RCCI	Reactivity Controlled Compression Ignition
RoPR	Rate of Pressure Rise
SI	Spark Ignition
TC-CRDI	Turbocharged Common Rail Direct Injection
UPCR	Delphi Unit Pump Common Rail System
WHSC	World Harmonized Steady-State Cycle

1. Introduction

It is known that engine manufacturers are under constant pressure in most countries due to stringent emission norms. The government worldwide started bringing their standards on par with global standards. Additionally, fuel efficiency norms are also in place, in which the engine manufacturers must increase their fuel efficiency by 30% or more between 2021 and 2030 [1]. It is predicted that a combination of the internal combustion engine (ICE), mild hybrids, and less than 10% electrification, an ICE, plug-in hybrid electric vehicles (PHEVs) and electric vehicles (EVs), and a group consisting of EVs, and PHEVs can meet CO_2 emission targets of 100 g/km CO_2, below 100 g/km CO_2, and 50 g/km CO_2 respectively [2].

The ICEs mostly run on fossil fuels, burning around 3000 million tonnes of oil equivalent each year [3], accounting for nearly 10% of global greenhouse gas (GHG) emissions. One of the main goals for engine researchers and manufacturers is to improve fuel economy and reduce pollutants. As a result, numerous alternatives to ICE, such as electric drives, have been proposed to reduce pollutants and fuel consumption. Considering the rapid innovations and disruption in ICE, existing ICE built-in fueling infrastructures, and current/post-economic conditions across the globe due to the COVID-19 crisis demands a solution to extend the life of ICE for mobility applications. As a result, to de-fossilize and limit engine exhaust emissions, it is critical to focus on developing high-efficiency flexi fuel engines using low-carbon fuels. Adopting modern LTC methods in ICEs allows for increased efficiency and flexible fuel options to reduce emissions and compete with electric propulsion systems.

To meet the EURO VI emission norms (In India, it is BS-VI equivalent to EURO VI norms), diesel-powered ICE's have undergone a rapid change over the years with the help of technologies such as; (i) electronically controlled flexible common rail direct injection (CRDI) [4], (ii) variable geometry turbochargers (VGT) [5], (iii) exhaust gas recirculation (EGR) [6], (iv) multi-valving, (v) variable valve timing (VVT) [7], and (vi) various after treatment systems (Diesel oxidation catalysts (DoC), Diesel particulate filter (DPF), Selective catalytic reduction (SCR)) [8-11 Diesel-powered engines/vehicles are more dependent on after-treatment technologies with complex control strategies to reduce the diesel particulates (PM) and oxides of nitrogen (NOx) to comply for EURO VI / BS VI norms. As an alternate to this, many researchers have worked on an alternate combustion technique known as low-temperature combustion (LTC) to reduce the PM and NOx emissions simultaneously [12]. This demands a leaner homogeneous air-fuel mixture formation and compression ignition in diesel engines. Many researchers, based on their approach of air-fuel mixture formation, coined different names for LTC, such as homogeneous charge compression ignition (HCCI) combustion [13,14], premixed charge compression ignition (PCCI) combustion [15,16] and modulated kinetics combustion (MK) concepts [17]. In diesel HCCI, air-fuel mixture formation occurs in the intake event itself by using the fuel vaporizer technologies [18]. In PCCI well advanced direct fuel injection timing is performed [19], and in MK concept retarded direct injection of fuel with higher EGR dilution and swirl ratio was used to achieve LTC operation in diesel engines [20].

Although each method can reduce NOx and PM at the same time, it faces challenges such as limited operating range, wall wetting / fuel accumulation in crevice volumes (resulting in increased unburned emissions at higher magnitudes), combustion chamber modifications, poor combustion control at high loads, and combustion efficiency. The LTC mode is fuel sensitive and hence, single fuel with high cetane or octane cannot provide better control over combustion [21]. Research work carried out by Kalghatgi et al., [22], Dec et al., [23] and Bessonette et al., [24] revealed that LTC fuel requirements are different from the conventional diesel combustion (CDC) mode and found that for achieving better combustion control in LTC mode, the fuel used should have both auto-ignition quality (cetane) and auto-ignition resistance quality (octane) according to in-cylinder thermodynamic conditions to assist auto-ignition and control combustion for engine loads and speeds.

Based on these observations, a group of researchers led by Professor RD Reitz of University of Wisconsin formulated an idea of mixing two different reactivity fuels through in-cylinder blending to meet the fuel requirements of LTC mode. This method is called reactivity controlled compression ignition (RCCI) combustion [25-29]. In the RCCI technique, two fuels with low and high reactivity characteristics are supplied to the engine through the port (at the intake) and direct injection system (compression event of the engine), respectively. The low to high reactivity fuel ratio may be in the range of 0.5 to 0.9 [30] between low to high load conditions to create a well premixed fuel-air mixture with varied reactivity gradient across the cylinder. Better volatility of low reactivity fuel and port injection leads to better mixing due to higher turbulence in the port during the intake process. Compared to the CDC mode, a well-developed direct injection of high-reactivity fuel over the compressed low-reactivity fuel-air combination during the compression event gives a longer ignition delay. This is mainly owing to the low reactivity fuel's auto-ignition resistance and the advanced injection of high reactivity fuel. The engine cylinder is filled with a well-premixed low/high reactivity fuel-air mixture with varying reactivity gradients and ignited due to the longer ignition delay. In comparison to CDC and other proposed LTC modes, this strategy provided greater combustion control, a larger reduction in engine emissions, and higher thermal efficiency. Despite these advantages, the RCCI method has numerous unsolved issues, such as greater HC and CO emissions and decreased combustion efficiency, especially at lower loads due to poor thermodynamic conditions, reactivity gradient and reactivity stratification [31].

The RCCI studies were conducted with conventional fuels and alternative fuels on both stationary and automotive diesel engines by many researchers [32]. Internal combustion engines and their associated energy conversion processes are currently being studied to reduce pollutant emissions while maintaining or improving fuel economy to reduce greenhouse gas (GHG) emissions[33]. In recent years de-fossilization has been aimed seriously in all energy conversion sectors along with the supportive government policies to bring EVs to replace the ICEs phase by phase [34,35]. This depicts the scenario that, while ICE will continue to play an important role in the near to medium-term future, it is critical to participate in the development of ICEs while they are in use. As a result, flexi fuel engines using low carbon fuels (LCF) are an attractive approach since they can be integrated into commercially available engines to minimise CO_2 emissions [36-38]. In this context, a study on flexi dual-fuel RCCI operation with low carbon fuels on the modified light-duty automotive diesel engine is attempted and to the best of knowledge, a research work on this area is not found in the literature.

Literature reported flexible-fuel vehicle engines operated with gasoline + alcohol blends under spark ignition combustion mode [39]. Whereas this study is attempted to use gasoline and diesel-like renewable low carbon fuels such as Methanol (M), Jatropha Biodiesel (BD), Dimethyl Ether (DME) and Polyoxymethylene dimethyl ether (PODE) in RCCI combustion mode to demonstrate flexi dual fuel operation with reduced NOx, Soot, HC, CO, and CO_2 emissions with better brake thermal and combustion efficiencies than the CDC mode. Hence, this approach paves a way to use alternative renewable low carbon fuels and show how to de-fossilize diesel engines. Especially alcohol, DME, PODE, and biodiesel fuels in the existing diesel engines with less retrofit costs with an ability to meet the stringent emission norms. The concerns associated with alcohol, DME, PODE, and biodiesel fuels during flexi fuel operation, such as calorific value, viscosity, and ignition quality, can be addressed through effective fuel management to produce the best output from the engine. The efficient use of low-carbon renewable fuels in diesel engines can reduce the cost of oil imports while also lowering CO_2 emissions. The CO_2 can be minimised through the natural recycling process because these fuels can be made from renewable resources such as plant seeds, biomass, etc. In flexi fuel and flexi combustion mode, a commercial, light-duty, 1.5L, 3 cylinder turbocharged, CRDI diesel engine is adequately tuned to use any low to high Cetane dual fuel combinations. The existing commercial light duty diesel engine was modified into flexible dual fuel engine. In the present study dual fuel control was developed to manage two-fuel injection systems (i.e., low reactivity fuel and high reactivity fuel) with single ECU. During the experiments every time fuels were changed manually, and the necessary control maps (fuel injection, fuel pressure, fuel mass, throttle position, torque-speed, EGR, Boost pressure, coolant, and oil settings etc.,) were tuned for individual dual fuel combinations (i.e., Methanol/Diesel, Methanol/Biodiesel, Methanol/DME, Methanol/PODE etc.,) for better performance and emissions.

In this study, to investigate the fuel effects under similar combustion conditions the same base diesel engine fuel energy input (FEI = mass of fuel x LHV of fuel) and combustion phasing (CA50) obtained at each test points were maintained during the dual fuel operation. This method minimised the number parameter sweeps during the experiments for achieving better performance and emissions. Accordingly, control parameters (LRF mass fraction, HRF injection timing, number of injections, mass of injection, injection pressure and EGR) were tuned for better performance , emissions and compared with base engine.

Since there are no limitations from the hardware part of the electroni control unit (ECU) with the collected data and individually tuned operating maps/tables, in future the study, development of flexi dual fuel algorithms with automatic switch over to different maps/table once the fuel combinations identified will be perfomed. In the present work both fuel combinations and algorithms were chosen manually during the engine operation. The use of low carbon content and oxygen enriched fuels like methanol, Isobutanol, biodiesel, DME, and PODE compared to neat diesel in dual fuel combustion mode would further reduce the unburnt emissions and CO_2 emissions from the FDFE.

2. Experimental setup and methodology

2.1 Experimental setup

A turbocharged, 3 cylinders, 1.5 litre commercial light-duty CRDI diesel engine with hot EGR was retrofitted with; (i) electronically controlled multi-point fuel injection (MPFI) system, (ii) cooled EGR circulation, (iii) dimethyl ether fuel supply system, and (iv) dual-fuel ECU. The detailed schematic diagram is shown in Fig. 1 and the specifications of the test engine are shown in Table 1. The engine was coupled to the ECB 200 (Dynalec make) eddy current dynamometer. The existing diesel fuel supply line was modified with a valve control to allow required fuel (Diesel/Biodiesel/PODE/DME) for facilitating flexible-fuel operation. The intake manifold of the engine was modified to mount three port fuel injectors for the supply of high octane number (low reactivity) fuels and connected to the port fuel injection system. Multi-point fuel injection system consists of the low reactivity fuel tank with electric feed pump, fuel filter, coriolis fuel mass flow meter, pressure regulators, distributor block and port fuel injectors. Except for DME, all other fuels like diesel, biodiesel and PODE were supplied using the existing fuel supply system of the test engine. Test engine system has Delphi unit pump common rail system (UPCR), a well-proven green strategy common-rail technology for small to medium diesel engines.

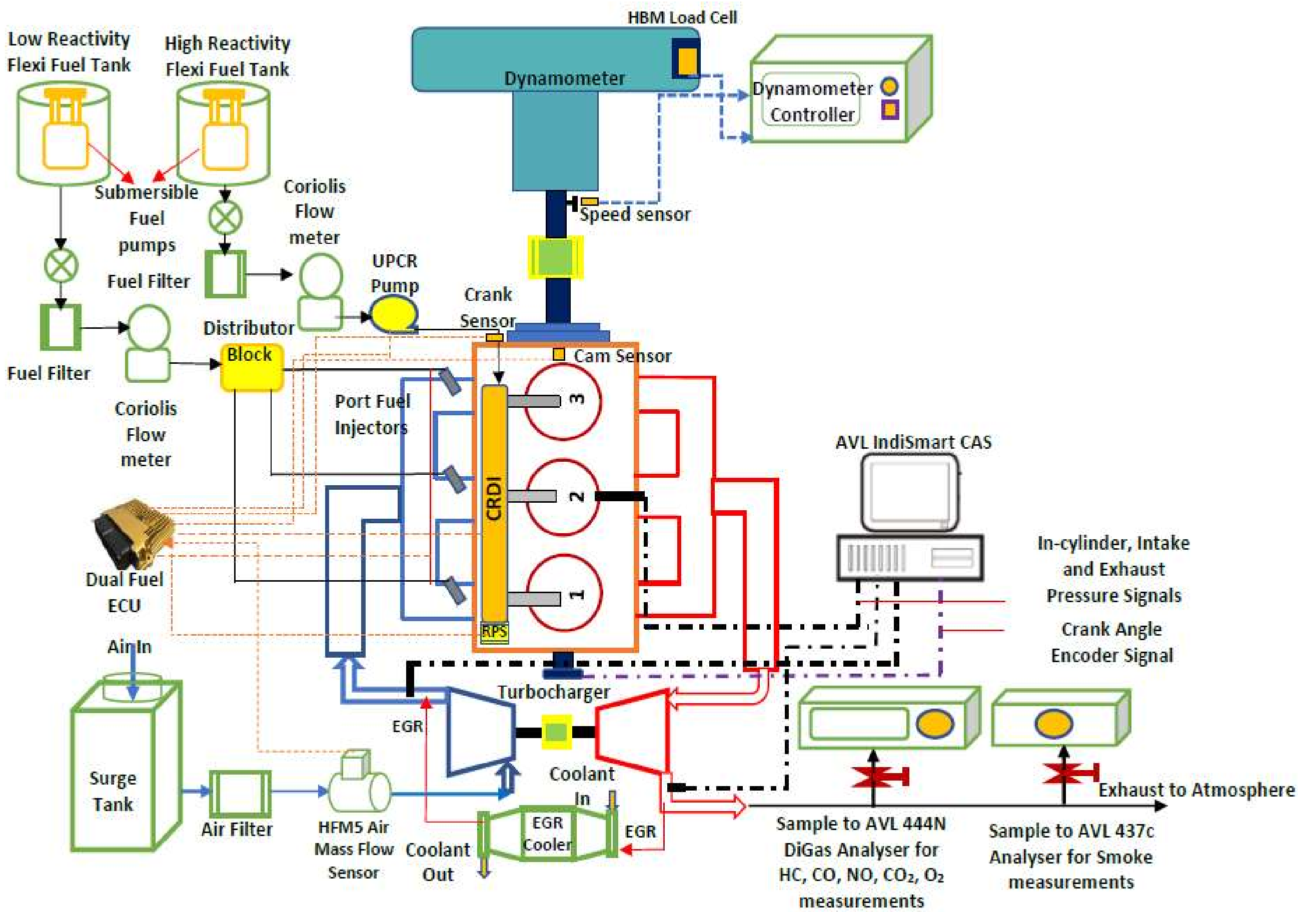

Fig. 1 Experimental setup

Table 1. Specification of the test engine

Displacement [L]	1.478
Number of cylinders	3
Compression ratio	17.2
Bore * Stroke [mm]	80 * 98
Connecting rod length [mm]	148
Engine power	42.51 kW @ 3000 rpm
Engine Torque	157.5 N.m @ 1600 – 2400 rpm
Valve actions	
Exhaust valve close [° bTDC]	-21
Intake valve open [° bTDC]	19
Intake valve close [° bBDC]	-53
Exhaust valve open [° bBDC]	69

Fast solenoid diesel injectors and a common rail, a program-tailored engine control module (ECM), a durable unit fuel pump with an inlet metering valve, and an efficient, low-cost fuel filter are all features of the Delphi diesel UPCR System. In addition, lightweight UPCR devices directly contribute to CO_2 reduction. A separate fuel delivery system was developed and combined with the OEM engine UPCR system for the DME fuel system (Fig. 1). The DME supply system consists of a DME cylinder, pressure regulator, mixing chamber, two LPG pumps, and a heat exchanger. They facilitate the onward and return supply of fuel and exchange of heat to cool the return fuel. In this study, DME is direct-injected in liquid condition with the help of a stock engine UPCR system. The same unit pump and high-pressure solenoid injectors are used for DME injection. To keep the DME in the liquid state, DME was stored above atmospheric pressure 5 bar. Hence two LPG pumps were used; one is to pressurize the inducted DME via mixing chamber from the cylinder to the supply side; and another is to pressurize the return DME from the UPCR. Stainless steel and Teflon materials were used for fuel supply system and as sealing material due to the poor lubricity and viscosity properties of DME as compared to diesel and biodiesel fuels. The DME supply system was integrated with the test engine UPCR system for direct injection of DME.

The test engine stock ECU, which can control single fuel, was replaced by a dual fuel ECU with features to handle two fuels (low and high reactivity fuel combinations), which can control injection pressure, injection mass, injection timing and the number of injections. These modifications allowed flexible dual operation with an advanced low-temperature combustion strategy called reactivity controlled compression ignition combustion. To measure the modified engine combustion pressure concerning engine crank angle position, the cylinder head of the test engine and crankshaft end were facilitated to mount combustion pressure sensor with a range of 0-300 bar and crank angle encoder with a resolution of 0.1°CA. The details of various measuring instruments used in the experimental test rig are given in Table 2. For proper combustion pressure measurements, it is necessary to measure the intake and exhaust charge pressure of the charge for referencing. Hence, on both the intake and exhaust sides, AVL low pressure (0-10 bar range) sensors were mounted. These sensors outputs were fed to the AVL IndiSmart advanced combustion analyser to analyse and collect the combustion parameters such as pressure-crank angle data, actual pressure-volume data, mean effective pressure data, rate of pressure rise data, energy release rate data, total energy release data, CA5, CA10, CA50 and CA90 data on a crank angle and cycle basis.

Table 2. Technical specification of instruments used in the experiments

Measurement Parameter	Name of the equipment used	Make and model	Accuracy/Sensitivity
Speed and Torque	Eddy current dynamometer with controller	Dynalec ECB 200	Torque: 1% of FSR Speed: ± 1 rpm
In cylinder Pressure	Pressure transducer	Kistler 6056A	-20.0 pC/bar
Intake pressure	Air-cooled low-pressure sensor	AVL LP11DA	934 mv/bar
Exhaust pressure	Water-cooled low-pressure sensor	AVL LP11DA	934 mv/bar
Crank Angle	Optical Encoder	Kistler 2613	0.1 deg
Combustion Parameter	DAQ System	AVL IndiSmart 612	***
High reactivity fuel flow Diesel/PODE/Biodiesel DME	Coriolis flow and density meter Weighing balance	Emerson make / Elite Series-CMF010M ESSAE-10	0.10% of the rate 0.1% of FS
Low reactivity fuel flow Methanol and its blends	Coriolis flowmeter	Emerson make / Elite Series-CMF010M	0.10% of the rate
Temperature	Thermocouples & digital indicator	K type	±1 ° C
Pressure (oil, fuel)	Pressure gauge	Wika Make	0.1 bar
Emissions			
CO CO_2 NO HC O_2	AVL Digas Analyser	AVL 444N	0-10% ± 0.02%abs 0-16% ±0.3% abs ± 3% rel. ± 5ppm 0-4000 ppm ± 8 ppm 3%rel. 0.02% abs 1% rel.
Smoke	AVL Smoke Meter	AVL 437C	0.1 ms-1 / 0.1% of opacity
Airflow	Mass airflow Sensor	Bosch – HFM 5	0.1% FS

Emissions such as HC, CO, CO_2, and NO were measured using AVL 444N DiGas analyser and smoke was measured using AVL437c smoke meter. The measured raw data are converted into g/kWh using standard available formulae. Low and high reactivity fuel mass flow rate was measured using a Coriolis mass flow meter. The airflow rate was measured using Bosch make hot film mass airflow sensor. Engine oil, coolant, air, and exhaust gas temperatures were measured using oil and coolant temperature sensors and k-type thermocouples. The engine sensors such as CAM, Crank, Boost pressure, temperatures, rail pressure and actuators were connected to the dual- fuel ECU via a proper wiring harness.

2.2. Test fuels

Commercially available diesel fuel was used as a base fuel, which was procured from local fuel stations. Low carbon content and renewable fuels were used in this study. Futher, previous LTC studies referred as Diesel Methanol Duel Fuel clearly indicated that methnaol is the best fuel to obtain better performance and lower emissions [40-44]. Analytical grade alcohol fuels methanol and Isobutanol were procured from M/s Alpha Chemika, Mumbai, with 99.5% purity. Ether fuels DME and PODE were procured from M/s Proton gas, Mumbai. The fuel properties were measured at the fuel characterisation laboratory of Anna university (Table 3) and few properties obtained from the literature [40,41]. The blend fuels such as M+PODE10 (denoted as (M+PODE)) and M+IB20 (denoted as (M+IB)) were prepared on a volume basis. M+PODE10 denotes the blend of 10% PODE and 90% methanol. Similarly, M+IB20 means a blend of 20% Isobutanol and 80% methanol.

Table 3. Properties of the test fuel [45,46]

Properties	**Methanol**	**Isobutanol**	**Diesel**	**DME**	**PODE**
Molecular Formula	CH_3OH	C_4H_9OH	$C_{12}H_{24}$	CH_3-O-CH_3	$CH_3O - [CH_2O]_n - CH_3$
Carbon content [mass %]	**37.5**	**65**	86	**52.2**	**44.2**
Oxygen content [mass %]	**50**	**21.5**	0	**34.8**	**46.9**
Hydrogen content [mass %]	12.5	13.5	14	13	8.9
Viscosity [mm^2/s] at 40° C	0.59	4.5	2.8	<0.1	1.1
Density [kg/m^3] at 25° C	790	810	840	667	1047
Lower heating value [MJ/kg]	19.7	25.6	42.5	27.6	20.9
Cetane number [-]	<5	-	48	>55	78
Octane number [-]	110	106	-	-	-
Latent heat [kJ/kg]	1100	683	300	467	359
Auto-ignition temperature [K]	733	688	523	508	511

2.3 Experimental procedure

The engine experiments were performed using world harmonized steady-state cycle (WHSC) test points. WHSC test points contain a total of 13 test points, among these, 2 high load points were not experimented due to safety reasons. Further, 2 idle points were not considered because the thermal efficiency of the engine is equal to zero and brake specific emission are not predictable. Generally, combustion phasing, i.e., CA50, is used as a parameter for combustion control in advanced combustion engines to maintain stable combustion and high thermal efficiency.

Table 4 Base engine data at different WHSC test points

Speed Rpm	Load Nm	BMEP bar	CA50 ° CA aTDC	FEI kW
1350	27	2.3	4	16
1600	31	2.6	5	20
1850	34	2.9	6	25
2100	36	3.1	7	30
1600	68	5.8	7	36
1850	77	6.5	10	50
1600	128	10.9	10	61
1850	100	8.5	14	56
2100	101	8.6	16	62

The relevant CA50 data for the stock engine was collected from the baseline trials in conventional diesel combustion mode using open ECU and validated with stock ECU because the stock engine was built for improved thermal efficiency and lower emissions. CA50 and fuel energy input (FEI) were kept constant at all WHSC test points during the RCCI experiments to compare the performance and emission improvements between conventional diesel combustion mode and flexi dual- fuel RCCI combustion mode to shed light on the differences between CDC and RCCI combustion modes. As a result, in RCCI mode, the brake mean effective pressure (BMEP) varies depending on the operating strategy. The CA50, fuel energy input (FEI), and brake mean effective pressure data from the stock engine's WHSC test points are shown in Table 4. Combustion phasing (CA50) of RCCI combustion predominantly depends on three operating parameters: low reactivity fuel energy ratio, direct injection timing of high reactivity fuel and exhaust gas recirculation.

Other operating parameters, such as high and low reactivity fuel injection pressures, and low reactivity fuel injection time, were kept constant due to their marginal impact on CA50 [47]. In this study, low reactivity fuel methanol was fed through the intake port at 360° CA bTDC and 4 bar. At the low, mid, and high load areas of the WHSC test points, high reactivity fuels (diesel, biodiesel, and DME) were injected at 30 MPa, 50 MPa, and 82 MPa, respectively. To begin, experimental parametric analysis was conducted to determine the best operating state for each of the WHSC test points. M+PODE/D dual- fuel RCCI tests were only performed at some WHSC cycle points due to a fuel amount limitation.

Table 5. Uncertainty values of the measured and deduced parameters

Measured/Deduced Parameter	Uncertainty in ±%
Speed (rpm)	0.1
Torque (Nm)	0.5
Peak In-cylinder pressure (bar)	0.3
Methanol mass flow rate (kg/h)	1.0
Diesel mass flow rate (kg/h)	1.0
Hydrocarbon (g/kWh)	1.3
Carbon monoxide (g/kWh)	2.5
Carbon di oxide (g/kWh)	1.4
Nitric oxide (g/kWh)	1.5
Soot (g/kWh)	3.4
Brake power (kW)	1.0
Indicated thermal efficiency (%)	1.3
Combustion efficiency (%)	1.6

All measurements were repeated three times, with the average result used to calculate and plot performance and emission characteristics. 100 consecutive test point data was used to derive the combustion parameters in the instance of combustion analysis. Table 5 displays the results of the uncertainty analysis using the typical approach, and Table 6 shows the detailed operating conditions of the tests.

3. Results and discussion

The findings are compared with CDC mode to investigate the benefits of flexible dual- fuel engines running in RCCI combustion mode. Table 7 shows the combustion, performance, and emission data of conventional diesel combustion.

Table 6. operating conditions

(Constant : P_{PFI} – 4 bar, SOI_{PFI} - 360° CA bTDC)

BMEP*	Speed	Torque	FEI	CA50	P_{boost}	EGR	MER	P_{DI}	SOI_{DI}						
									CDC	M/D	M/DME	M/BD	M/PODE	M+IB/D	M+PODE/D
bar	rpm	Nm	kW	° CA aTDC	bar	%	%	Bar	° CA bTDC						
2.3	1350	27	16	4	1.13	20	60	300	9.3	37.6	38.3	35.0	-	32.1	31.2
3.4	1500	40	24	5	1.25	20	60	300	-	-	-	-	35.2	-	-
2.6	1600	31	20	5	1.43	20	60	300	10.0	39.0	39.7	36.4	-	31.3	30.8
2.9	1850	34	25	6	1.55	20	60	300	11.1	40.1	40.8	37.4	-	29.4	28.6
3.1	2100	36	30	7	1.61	20	60	300	12.5	41.0	41.6	38.3	-	28.5	28.1
5.8	1600	68	36	7	1.94	20	70	500	7.2	35.0	36.3	32.9	-	-	-
6.5	2100	77	50	10	2.27	20	70	500	8.1	36.1	36.5	31.6	-	-	-
10.9	1600	128	62	10	2.23	40	80	820	6.4	31.0	32.0	28.6	-	-	-
8.5	1850	100	56	14	2.34	40	80	820	5.1	31.5	32.4	29.1	-	-	-
8.6	2100	101	61	16	2.47	40	80	820	10.0	39.0	39.7	36.4	-	-	-

Table 7. Combustion, performance, and emission characteristics of CDC at different test points

Speed	Load	BMEP*	Max.P	CD	ID	RoPR	IMEP	COV_{IMEP}	BTE	CE	NO	HC	CO	Soot
rpm	Nm	Bar	bar	° CA	° CA	bar/° CA	bar	%	%	%	g/kWh	g/kWh	g/kWh	g/kWh
2.3	1350	27	70.7	15.6	9.8	5.5	3.7	0.8	23.9	99.5	5.09	0.40	0.85	0.14
3.4	1500	40	75.5	17.3	8.1	5.2	4.8	1.2	27.8	99.5	4.54	1.13	1.43	0.56
2.6	1600	31	71.2	16.6	10.2	5.4	4.3	1.0	25.9	99.6	4.50	0.39	1.17	0.19
2.9	1850	34	73.8	18.2	10.8	5.2	4.7	1.1	26.3	99.5	4.35	0.91	1.30	0.33
3.1	2100	36	74.1	19.6	11.4	5.1	5.0	1.3	27.1	99.5	4.24	1.23	1.47	0.57
5.8	1600	68	84.7	18.4	9.0	4.7	8.5	0.9	31.6	99.8	7.44	0.18	0.40	0.46
6.5	2100	77	89.1	20.2	9.4	4.4	9.4	1.0	33.8	99.8	6.57	0.23	0.38	0.74
10.9	1600	128	99.9	23.5	7.9	3.5	15.2	0.7	35.2	99.8	6.73	0.07	1.13	0.67
8.5	1850	100	82.9	22.2	7.7	3.6	11.5	0.8	34.6	99.9	6.52	0.13	0.19	0.79
8.6	2100	101	93.4	23.8	8.1	3.5	12.5	1.0	35.4	99.9	5.82	0.10	0.11	0.85

3.1. Combustion characteristics

A prolonged ignition delay is important to form a well-premixed air-fuel mixture inside the cylinder. In the present investigation, the ignition delay is derived from the cumulative heat release data. It is defined as the crank angle difference between the start of high reactivity fuel injection timing ($\Theta_{SOI,\ HRF}$) and the 5% of total heat release inside the cylinder (Θ_{SOC}). Fig. 2 shows the comparison of ignition delay of different dual fuel combinations in RCCI combustion and CDC modes. Approximately 24°CA longer ignition delay is observed for all the dual fuel combinations in RCCI combustion exhibited an when compared to CDC operation.

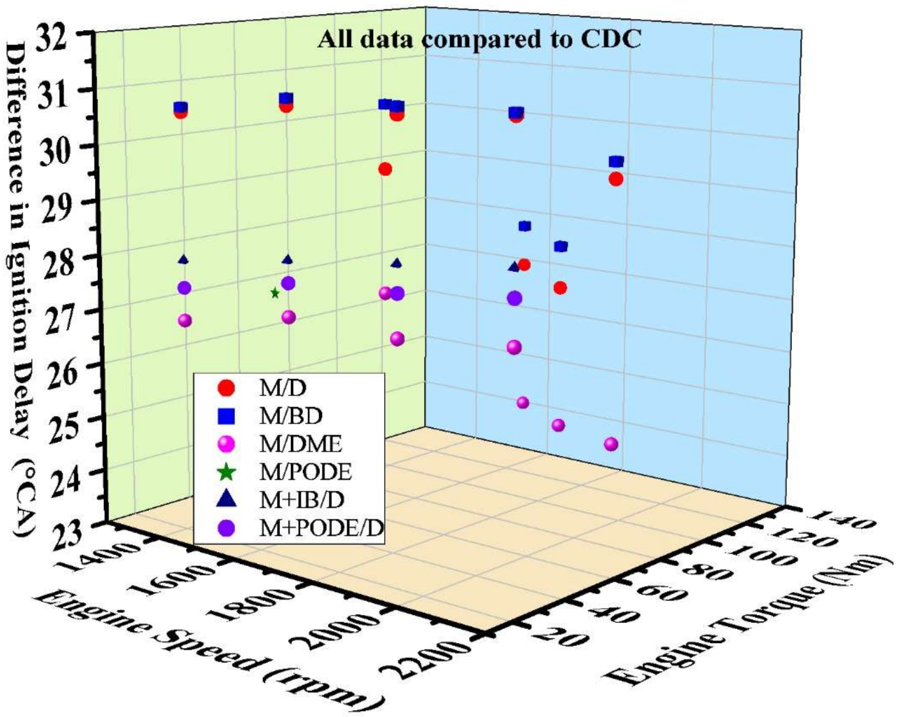

Fig. 2 Absolute change in ignition delay of FDFE operation in RCCI combustion compared to CDC mode

Due to premixed low reactivity fuel-air mixture internal energy available inside the cylinder is low which is not sufficient to vaporize the high reactivity fuel. Hence, the ignition delay is prolonged in the case of RCCI combustion compared to CDC mode. Among the tested dual fuel combination in RCCI combustion, ignition delay is in the order of M/DME < M/PODE < (M+PODE)/D < (M+IB)/D < M/D < M/BD. (Table 7 and Fig. 2) It is worth noting that ignition delay in RCCI combustion is influenced by a variety of physical and chemical features in both low and high reactivity fuels. Even with premixed methanol and air combination inside the cylinder, DME fuel has a higher cetane number, lower boiling point, and viscosity, allowing for better air-fuel mixing and shorter ignition delay. On the other hand, PODE fuel has a greater cetane number and ignitibility characteristics, resulting in a shorter ignition delay. Furthermore, due to improved reactivity of the low reactivity fuels (i.e., M+PODE and M+IB), the ignition delay is minimised when blending high Octane fuel with high Cetane fuel. Compared to all fuel combinations, M/BD RCCI combustion exhibited a longer ignition delay due to its higher viscosity and molecular mass of the biodiesel fuel. The ignition delay of RCCI combustion gradually reduces with an increase in engine load due to better in-cylinder conditions.

Fig. 3 shows the absolute difference in combustion duration of different dual- fuel combinations in RCCI combustion compared to CDC mode. Compared to the CDC mode, RCCI combustion resulted in 2 to 6.5°CA shorter combustion duration. This could be because the charge inside the cylinder is more premixed than in CDC mode. The experimental results indicated that the combustion duration of tested dual fuel combinations has the combustion duration in the following order M/D < M+PODE/D < M+IB/D < M/PODE < M/BD < M/DME. Due to more premixed fuel and less mass of fuel inside the cylinder, M/D RCCI combustion exhibited a shorter combustion duration than M/BD RCCI combustion.

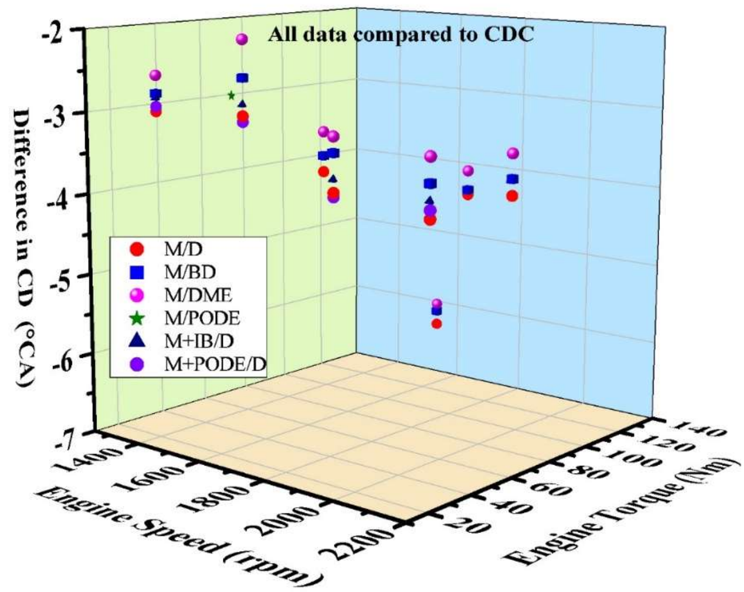

Fig. 3 The absolute difference in CD of Flexi fuelled RCCI combustion compared to CDC

The rate of pressure rise (RoPR) is a useful indicator of engine smoothness and an important parameter in engine safety. In general, more efficiency is achieved by burning for a shorter time, thereby resulting in a higher RoPR. Increased RoPR weakens engines over time, resulting in increased combustion noise. As a result, keeping RoPR below acceptable limits is critical when introducing novel combustion techniques. The % decrease in RoPR compared to CDC is shown in Figure 4.

In comparison to CDC combustion, RCCI combustion had a higher RoPR. RoPR is high, especially at high loads (Nearly 60 percent higher than CDC). In RCCI mode approximately 20 to 30% increase in RoPR is observed at low to medium loads.

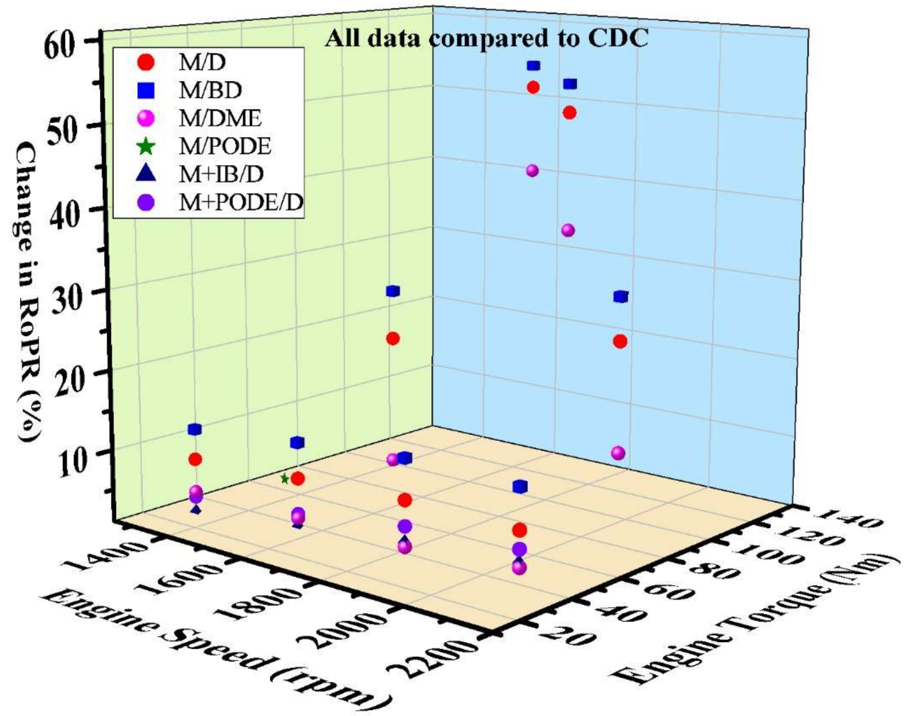

Fig. 4 Percentage change in RoPR rise of different dual-fuel combinations in RCCI combustion compared to CDC

Among the different dual-fuel combinations investigated under RCCI combustion, M/BD fuelled RCCI combustion exhibited 15% higher RoPR at low loads due to longer ignition delay. Reactivity improved low reactivity fuel (i.e., M+PODE and M+IB) combinations exhibited marginally higher RoPR than M/DME and M/PODE fuelled RCCI combustion. From this, it is clear that the vaporization and mixing of high reactivity fuels have more impact on the start of combustion and premixed fuel formation than dual- fuel cetane number (i.e., reactivity) inside the cylinder.

Table 8. IMEP and COV_{IMEP} data for different fuel combinations of RCCI combustion with CDC

Fuel Combination and Combustion Mode	IMEP (bar)	COV_{IMEP} (%)
Diesel and CDC	4.95	1.3
M/D and RCCI	5.12	5
M/BD and RCCI	5.18	4.3
M/DME and RCCI	5.13	6.1
M/PODE and RCCI	5	4.3
M+PODE/D	5.21	4.1
M+IB/D	5.2	3.7

The indicated mean effective pressure value and it's COV is provided in the Table 8. When comparing the CDC to the reactivity regulated compression ignition mode, the indicated mean effective pressure of the flexible dual-fuel engine is higher. This pattern concurs with the experiments conducted by Reitz et al. [27] and Anand et al. [48]. (M+PODE)/D > (M+IB)/D > M/BD > M/D = M/DME and M/PODE are the sequences observed. It's worth noting that reactivity enhanced low reactivity fuels had a higher indicated mean effective pressure than RCCI combustion with pure methanol. Furthermore, when comparing RCCI and CDC combustion, RCCI combustion had a larger fluctuation in indicated mean effective pressure. The deviations, however, are within the acceptable range.

3.2. Performance characteristics

The percentage difference in brake thermal efficiency between various dual-fuel RCCI combustion and CDC is shown in Fig. 5. The RCCI combustion resulted in a nearly 2 to 8% increase in brake thermal efficiency than CDC mode. This could be attributed to (i) lower heat transfer losses and (ii) improved engine IMEP net (due to reduced compression work and increased expansion work). Reitz et al. [25] reported a similar trend in their recent publication. Furthermore, due to the improved in-cylinder thermodynamic conditions, BTE increases as engine load and speed increase. Due to the shorter combustion period, reactivity improved low reactivity fuels generate higher BTE than neat methanol fuelled RCCI combustion. At low loads, fuel effects on BTE in RCCI combustion are significantly observed and it is concealed with increasing engine load.

The term combustion efficiency (CE) refers to the efficiency of complete combustion. The increasing unburned emissions imply a reduced combustion efficiency. The combustion efficiency of a commercial diesel engine is approximately 99 percent. Fig. 6 shows the difference in combustion efficiency between flexi dual-fuel RCCI combustion and CDC combustion. The combustion efficiency of dual-fuel RCCI combustion is about 2 to 9% lower than the CDC. Due to poor thermal conditions inside the cylinder, a considerable loss in CE (5% to 9%) is observed at low loads. As the fuel's inherent oxygen enhances total combustion inside the cylinder, oxygenated high reactivity fueled RCCI combustion has a higher combustion efficiency than other fuel combinations. Furthermore, it is interesting to note that reactivity improved methanol fuelled (i.e., (M+PODE) and (M+IB)) RCCI combustion provides significant improvement in combustion efficiency than neat methanol fuel.

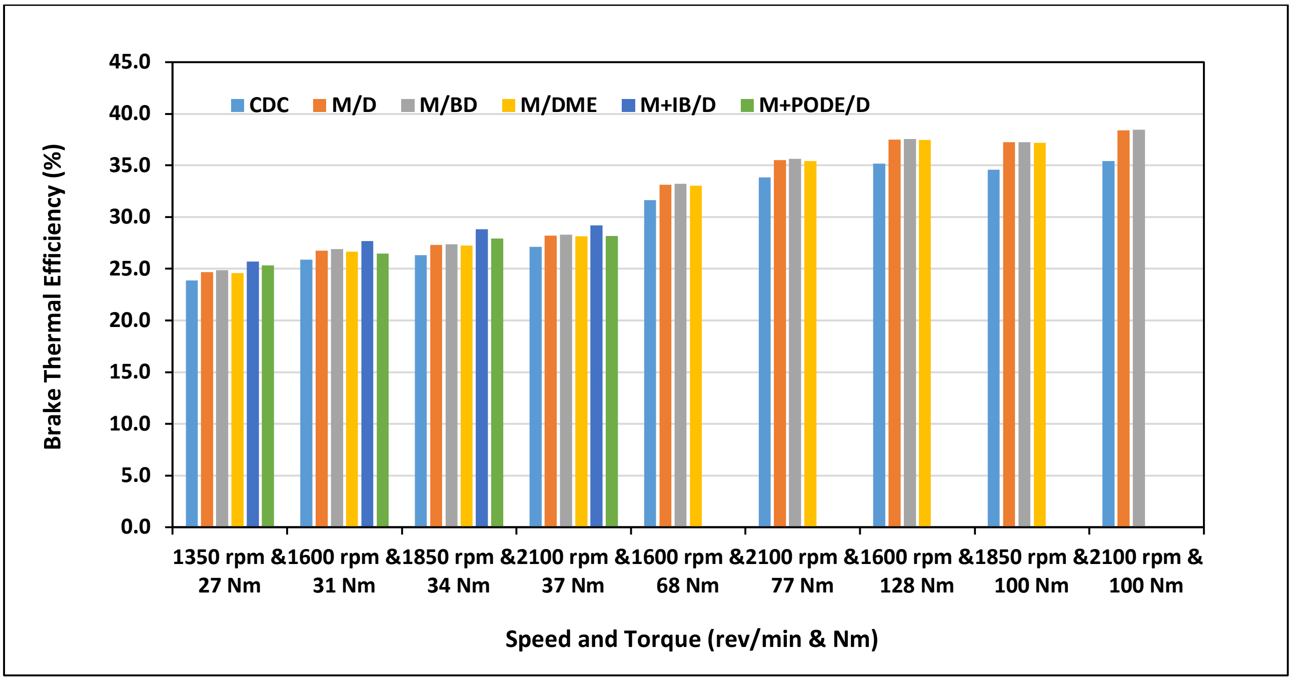

Fig. 5 Brake thermal efficiency of Flexi fuel RCCI combustion compared to CDC mode

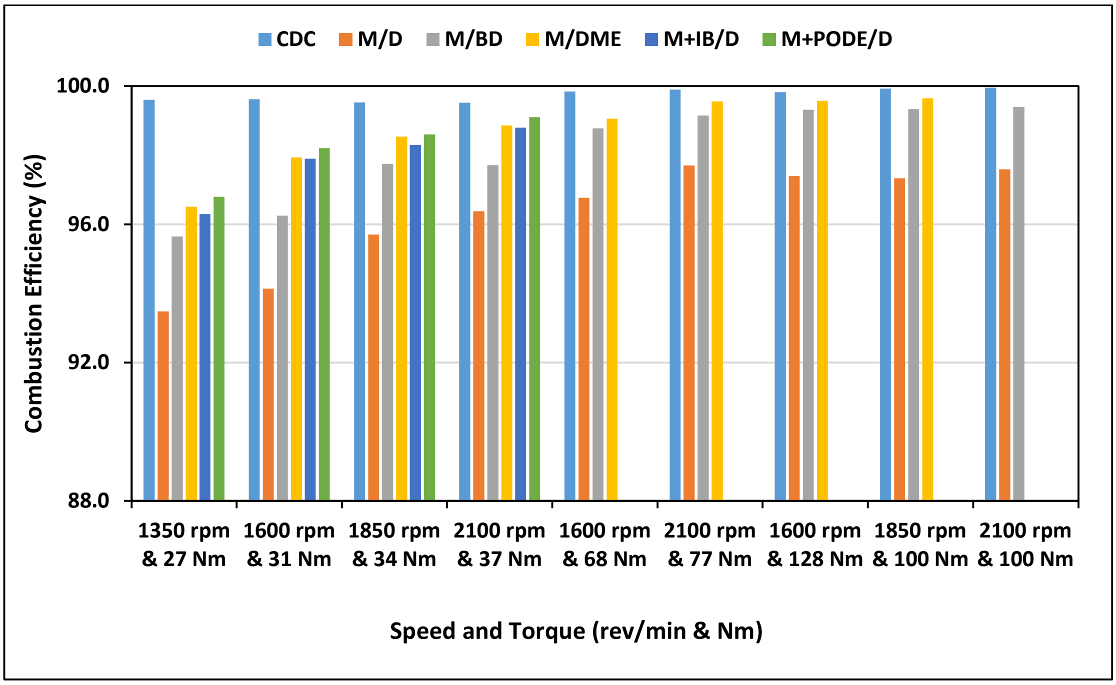

Fig. 6 Combustion efficiency of Flexi fuel RCCI combustion compared to CDC mode

3.3. Emission characteristics

In most cases, the formation of nitrogen oxides occurs inside the cylinder due to greater in-cylinder temperatures, oxygen availability, and nitrogen exposure time in a high-temperature environment. Nitric oxide (NO) molecules and a tiny amount of nitrogen dioxide make up the majority of NOx emissions. Fig. 7 illustrates the percentage change in nitric oxide emission concentration from dual- fuel RCCI combustion. Compared to CDC combustion, NO emissions in RCCI mode are reduced by nearly 81 to 93 percent. Compared to low loads, high loads show a more significant reduction in NO emissions. The M/D RCCI combustion had the highest NO emission decrease among the different dual-fuel combinations of RCCI combustion, followed by M/DME, M/PODE, M+IB/D, M+PODE/D, and M/BD RCCI combustion.

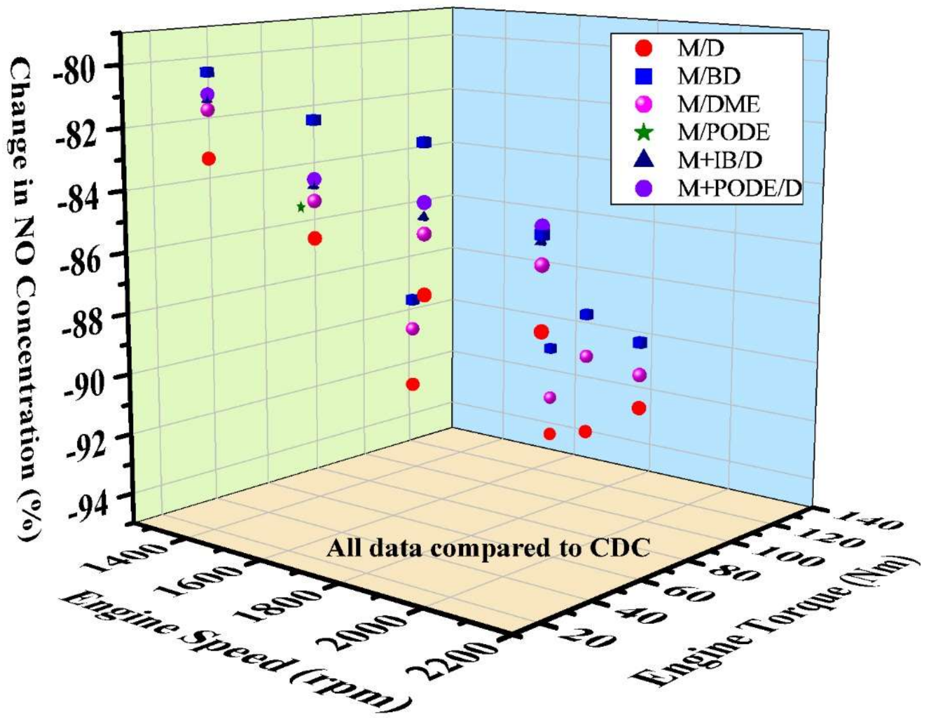

Fig. 7 Percentage difference in NO concentrations of Flexi fuel RCCI combustion compared to CDC

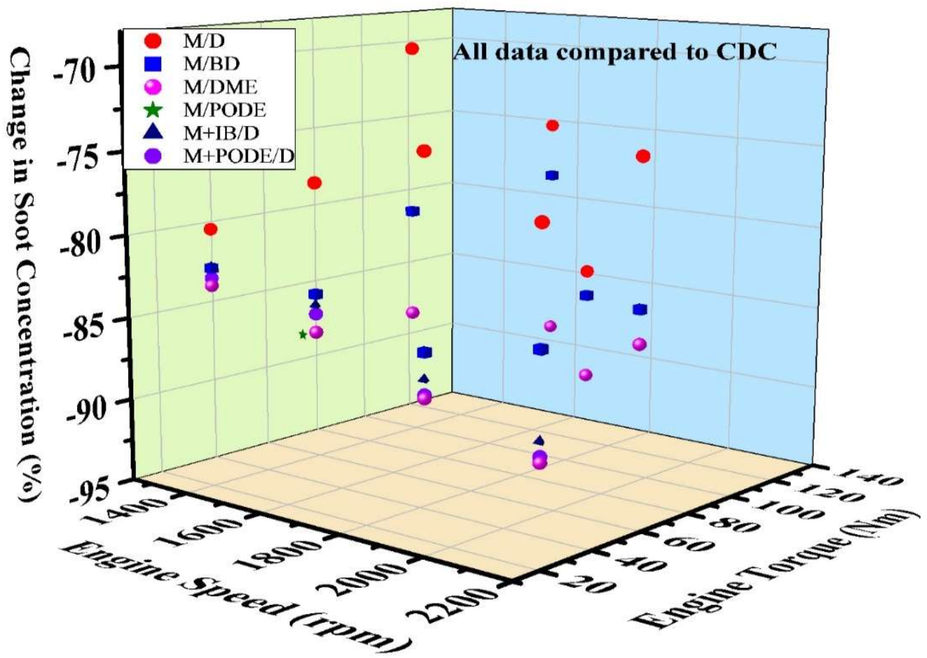

Fig. 8 Percentage difference in Soot concentrations of Flexi fuel RCCI combustion compared to CDC

The oxidation of high reactivity fuel is the primary cause of soot production in dual- fuel RCCI combustion. Due to increased oxidation inside the cylinder, oxygenated high reactivity fuelled RCCI combustion showed a higher reduction in the soot emission. In RCCI combustion, a soot reduction of 78 - 95 percent is observed than CDC mode. In comparison to diesel-fueled RCCI combustion, oxygenated and reactivity improved methanol fuel mixture fuelled RCCI combustion revealed a significant reduction in soot emission (Fig. 8). Despite the fact that biodiesel and ether fuel (DME and PODE) contain more oxygen, DME and PODE have a higher soot reduction due to the absence of the C-C bond, lower boiling temperature, and lower carbon content. Furthermore, due to an increased global equivalents ratio inside the cylinder and lower homogeneity of the air-fuel mixture, soot emission slowly increased with the increasing engine load and speed.

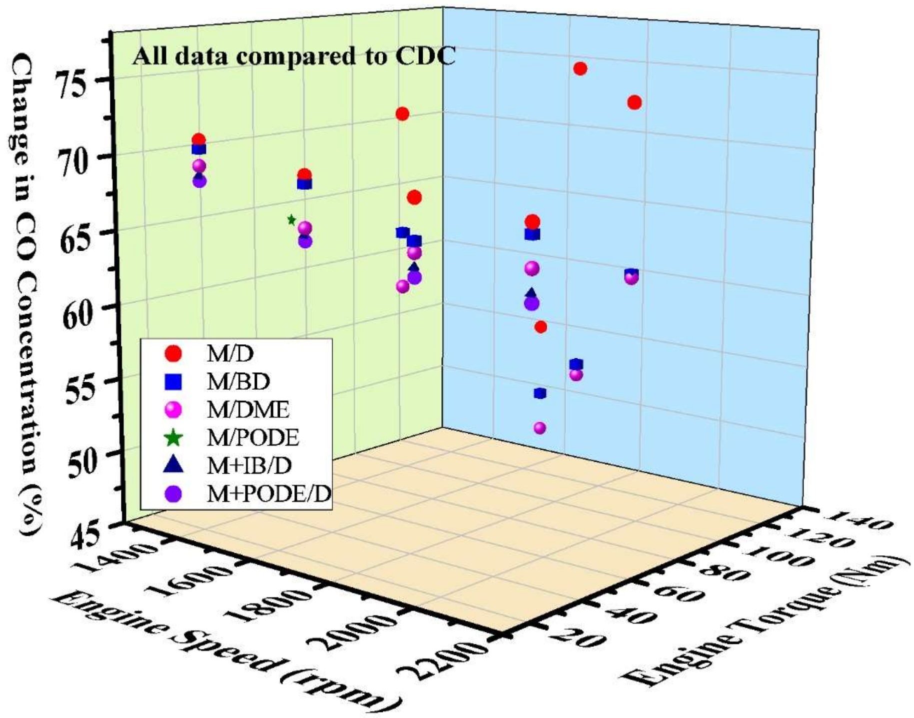

Fig. 9 Percentage difference in CO concentrations of Flexi fuel RCCI combustion compared to CDC

The CO emissions are formed in IC engines due to decreasing in-cylinder temperature and oxygen concentration. It is observed that CO emission is much higher in dual-fuel RCCI combustion than in CDC, (Fig. 9). This is owing to a lean homogeneous air-fuel mixture and a lower peak bulk gas temperature. When RCCI combustion is compared to CDC combustion, CO emissions increase by 58 - 75 percent on average. The increased CO emission in RCCI combustion could be due to the following factors: (i) reduced oxidation of hydrocarbon fuel due to lower in-cylinder temperature caused by higher latent heat of vaporisation of methanol; and (ii) lower oxygen content inside the cylinder due to the EGR (dilution effect) and port-injected methanol replacing pure oxygen. It was found that M/DME and M/PODE RCCI combustion produced less CO than M/D and M/BD RCCI Combustion. This may be attributed to the reason that DME and PODE have a lower boiling point, a wider spray angle inside the cylinder and they do not have fuel-rich zones in the combustion chamber, thereby resulting in fewer CO emissions. In addition, the fuel's low C/H ratio, absence of C–C bonds, and high oxygen content should result in faster and more effective oxidation of intermediate species, resulting in cleaner combustion than other high reactivity fuels. Approximately 20% reduction in CO emission is observed when DME and PODE is used as a high reactivity fuel compared to diesel and biodiesel. In RCCI combustion, relatively higher CO emission is observed at low load conditions when compared to high load conditions due to lower in-cylinder temperature and over lean air-fuel mixture.

The generation of HC emissions in compression ignition engines is generally due to fuel in the flame out region and tail spray. In RCCI combustion, more fuel (i.e., low reactivity fuel) is fed into the intake port/manifold, which increases contact with the crevices/ piston surface area and causes unfavorable in-cylinder thermodynamic conditions, which impacted the oxidation process and raised HC levels (Fig. 10). At high loads, the HC emission is lower due to a higher in-cylinder temperature, which may improve fuel oxidation in crevices and piston surfaces. At low, medium, and high loads almost 20, 10 and 7 times higher hydrocarbon emission is observed in RCCI operation than CDC mode. The HC emission of M/D RCCI is higher than that of M/DME, M/PODE, and M/BD dual- fuel RCCI combustion.

The higher cetane number and better reactivity gradient inside the cylinder help reduce the HC emission while using DME, PODE and Biodiesel as high reactivity fuels in the RCCI combustion compared to Diesel as a high reactivity fuel. Ether fuel exhibited a lower HC emission between ether and ester fuelled RCCI combustion. This may be due to a wider spray pattern and smaller droplet size created by the higher volatility and lower molecular weight of ether fuel (DME/PODE). The high reactivity fuel was appropriately blended with a premixed air-methanol mixture owing to the larger spray. More ignition sites are available inside the cylinder as a result of the wider spray, which improves the oxidation of fuel trapped in crevices and piston surfaces and decreases HC emission. DME also has a low boiling temperature and a high vapour pressure, resulting in enhanced fuel atomization, mixture formation, and neat to full combustion with lesser hydrocarbon emissions.

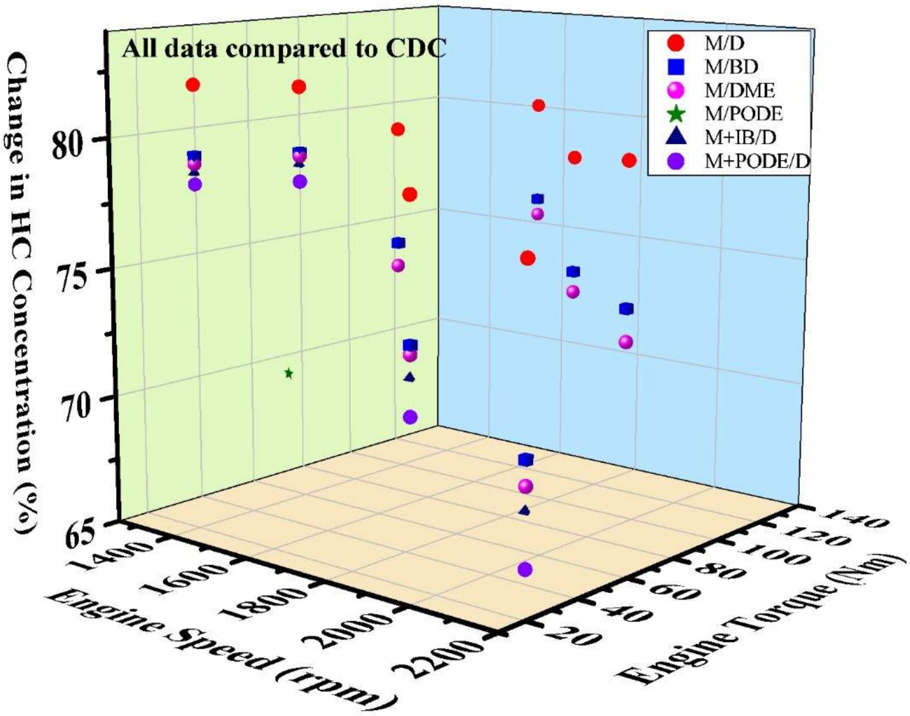

Fig 10. Percentage difference in HC concentrations of Flexi fuel RCCI combustion compared to CDC

3.4. Energy analysis

Understanding the system's energy and exergy distribution is critical to improving the system's overall efficiency. In general, the first law of thermodynamics is used to analyse the energy distribution (Quantity analysis) of the system. In contrast, the second law is used to analyse the exergy distribution (Quality analysis) [49]. Energy and exergy analysis were carried out in the current study using the procedure described in the literature [50]. In comparison to CDC, Fig. 11 depicts the energy distribution of various dual fuel combinations in RCCI combustion. When RCCI combustion is compared to CDC combustion, it is found that the fuel energy transformed into useable power is nearly 3 to 4% higher. In addition, reactivity improved low reactivity fuel-based RCCI combustion converts more energy into useful power. Due to lower combustion temperature, the energy released through exhaust and coolant significantly reduced in RCCI combustion compared to CDC mode.

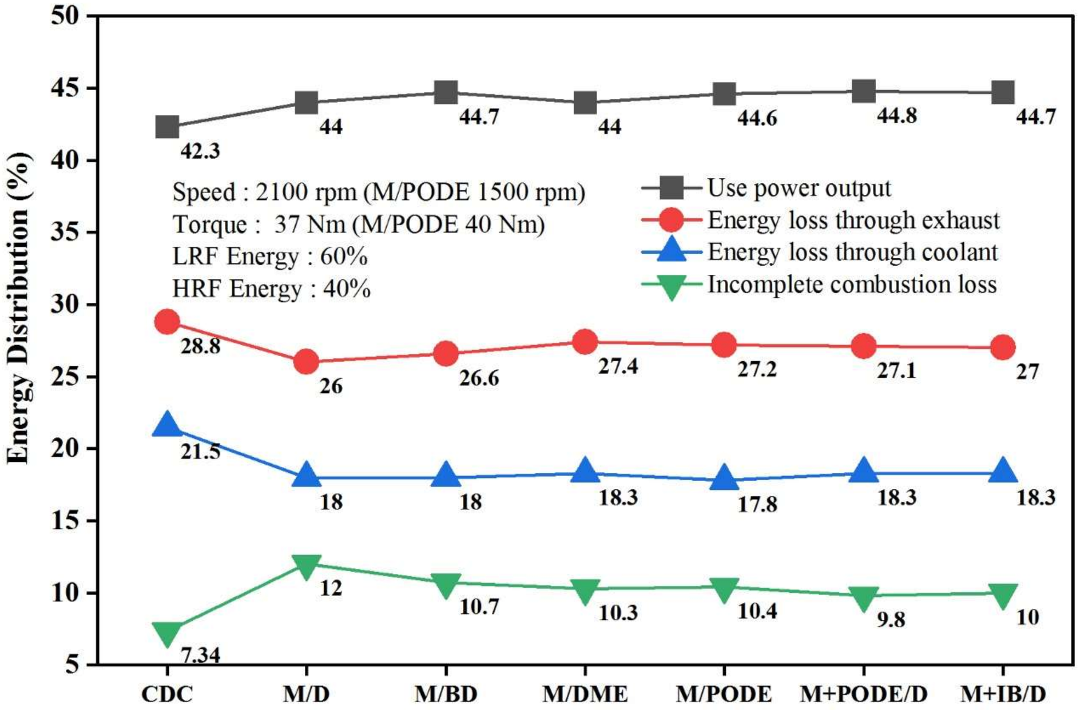

Fig. 11 Energy distribution of flexi fuel RCCI combustion along with CDC

The energy distribution for CDC and different dual- fuel combinations of RCCI combustion is shown in Fig. 11. Dual fuel RCCI combustion resulted in a larger incomplete combustion loss than CDC due to higher HC and CO emissions. However, partial combustion loss is smaller when utilising oxygenated high reactivity fuel (i.e., DME, PODE, and biodiesel) than when using non-oxygenated high reactivity fuel. At M/D RCCI combustion, a maximum of 12% incomplete combustion loss is observed. Incomplete combustion losses of almost 10.7%, 10.3%, and 10.4% were observed in M/BD, M/DME, and M/PODE RCCI combustion, respectively. Due to the longer combustion time and higher peak in-cylinder temperature, more energy is lost through exhaust and coolant in CDC combustion than in RCCI combustion. When flexi dual- fuel RCCI combustion was compared to non-oxygenated high reactivity fuel-based RCCI combustion, oxygenated high reactivity fuel RCCI, combustion had a higher energy loss through coolant and exhaust. Furthermore, compared to non-oxygenated fuel combinations, stated thermal efficiency is the highest with oxygenated fuel combinations.

In comparison to CDC mode, Fig. 12 displays the exergy distribution of flexi dual- fuel RCCI combustion. Exergy study shows that in the case of dual- fuel RCCI combustion, the magnitude of exergy loss in coolant and exhaust is lower than in the case of CDC combustion. The explanation for this could be because dual- fuel RCCI combustion has a lower boundary layer temperature, resulting in less interaction between the high-temperature flame and the cylinder surfaces [51]. Exergy destruction is lower in dual-fuel RCCI than in CDC, indicating that dual- fuel RCCI mode has greater energy utilisation. Exergy research reveals that incomplete combustion at RCCI combustion contains about 10% high-quality energy. If this high-quality energy can be recovered by fine-tuning the operational settings, the energy utilisation can be improved, even more, increasing the useable power production. It was also shown that in dual-fuel RCCIs, an average of 26% of low-quality energy was accessible in the exhaust and coolant, which could be recovered using waste heat recovery methods to enhance total energy conversion efficiency.

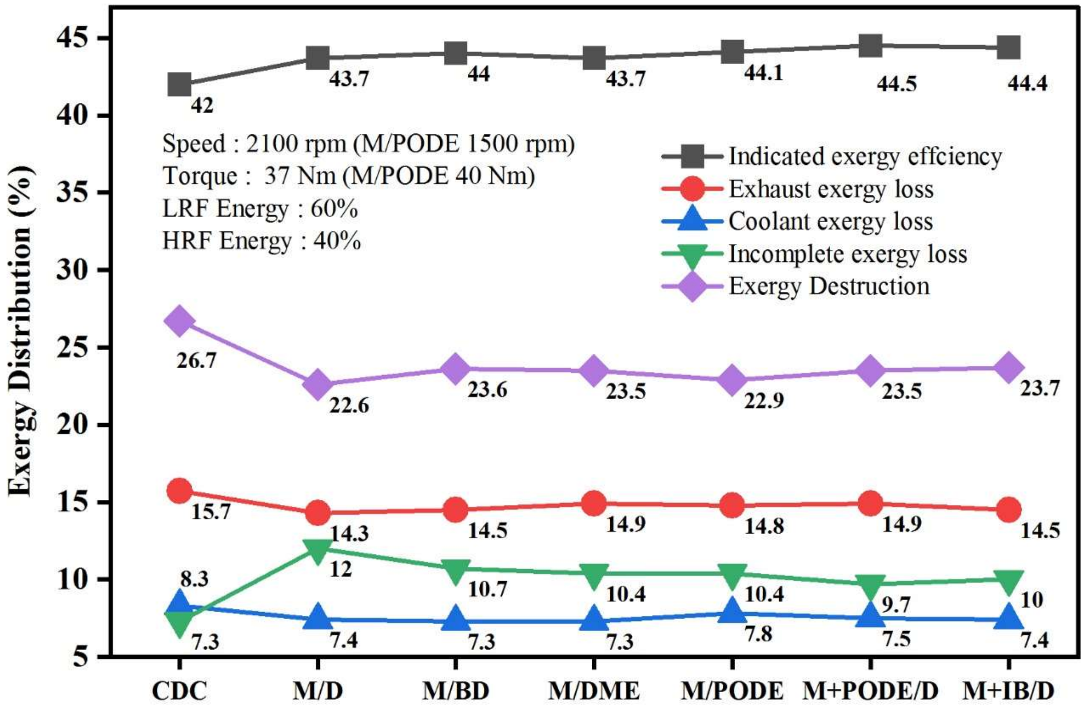

Fig 12. Exergy distribution of flexi fuel RCCI combustion compared with CDC mode

4. Conclusions

The experiment was carried out on a modified flexi dual fuel reactivity-controlled compression ignition combustion engine at constant combustion phasing (CA50) and fuel energy input as per World

Harmonized Steady State Cycle (WHSC) points (at various speeds and loads). Using several oxygenated biofuels, this study proved the flexi dual- fuel flexibility and mode switching capabilities and addressed challenges such as reduced CE, increased RoPR, and cyclic variability. The important findings from the experiment are given below.

- Longer ignition delay is observed in the case of RCCI combustion when compared to CDC mode. Among the different dual-fuel combinations, ID is observed to be in the following order of M/DME< M/PODE < M/D< M/BD.
- A shorter combustion duration is observed in RCCI combustion compared to the CDC mode. The combustion duration is found to be in the order of M/D < M/BD < M/PODE < M/DME < CDC.
- An improvement in brake thermal efficiency of about 8% is seen in RCCI combustion mode when compared to CDC mode. From the tested flexi dual fuel combinations, M/BD RCCI combustion has higher brake thermal efficiency than M/D, M/PODE and M/DME.
- 86% reduction in NO emission is observed in RCCI mode compared to CDC mode. Oxygenated dual fuel combinations (M/PODE and M/DME) have resulted in marginally higher NO emissions than M/D RCCI combustion.
- Nearly 90% reduction in soot emission is observed in RCCI combustion as compared to CDC mode. Furthermore oxygenated fuelled RCCI combustion provides lower soot (almost zero) compared to non-oxygenated fuelled RCCI combustion.
- The RoPR is higher in RCCI combustion than CDC mode owing to the occurrence of additional premixed fuel.
- The lower combustion efficiency is observed in RCCI combustion due to lean homogenous mixture and lower in-cylinder temperature as compared to CDC. However, the use of oxygenated high cetane fuel as a high reactivity fuel improved the combustion efficiency compared to non-oxygenated high reactivity fuel.
- The cyclic variability of RCCI combustion is higher compared to the CDC mode. The variations are in the order of CDC< M/BD< M/PODE< M/D < M/DME.
- Energy and exergy analysis indicated that in the case of flexi dual-fuel RCCI combustion mode, the utilisation of fuel energy (i.e., Indicated power) is increased with an increase in methanol

energy ratio due to enhanced homogeneity of the fuel-air mixture, which ultimately leads to clean and complete combustion.

- The energy lost through coolant and exhaust is decreased with increasing methanol energy ratio due to shorter combustion duration and lower in-cylinder temperature.

Overall, this study concludes that implementation of flexi dual fuel and flexi combustion mode would be a viable option for achieving improved thermal efficiency, low emissions, and de-fossilization. The flexi dual fuel and flexi combustion mode engine is also a good choice for hybrid EVs and it may extend the life of the diesel engine in use. Further research on control systems and fine-tuning the engine for low carbon fuels is our future scope and research direction.

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550