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Experimental Investigation on Effects of Fuel Injection Timings on Dimethyl Ether (DME) Energy Share Improvement and Emission Reduction in a Dual-Fuel CRDI Compression Ignition Engine

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Abstract

The impacts of different fuel injection timings (base (15°BTDC) and retarded (12°BTDC)) on a single cylinder common rail direct injection (CRDI) compression ignition (CI) engine under dual fuel mode (DME-diesel) was examined at the maximum torque with the engine speed of 2200 rpm. The diesel (liquid) fuel was directly injected into the engine's cylinder at the end of the compression stroke and Dimethyl ether (gas) fuel was injected into the intake system of the engine during the suction stroke. The maximum DME energy share (DME ES) with base injection timing (15°BTDC) under controlled auto-ignition (CAI) is limited to 52%. Delayed injection timing prevents knock occurrence to a certain extent due to the lower peak combustion pressure and temperature. The DME energy share (DME ES) without knocking is extended to 58% with delayed injection timing (12°BTDC). Three stages of combustion (low-temperature region, high-temperature region and diffusion combustion region) were observed under DME-diesel mode. The low-temperature region (LTR) occurs at the temperature range from 700–900 K whereas high-temperature region (HTR) was at the temperature beyond 1000 K. The cyclic variations with the coefficient of variation (COV) of the important parameters (peak in-cylinder pressure, peak rate of pressure rise and indicated mean effective pressure) were analyzed. NO_x emission decreased along with zero smoke emission. However, the other emissions such as CO, HC increased marginally with retarded timing. The notable point emerged from this study is that the retarded injection timing is a promising strategy for enhancement of DME energy share with reduced emissions.

Keywords: CRDI engine; Controlled auto-ignition; Dimethyl Ether; DME energy share; Knocking; Retarded injection timing.

Research Article

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1. Introduction

Diesel Compression ignition (CI) engines, due to their higher fuel conversion efficiency, torque capacity and reliability, are majorly used for industrial applications, public transportation systems, constructional work, agricultural equipment and heavy machinery. However, due to the heterogeneous mixture of air-fuel charge during combustion, the diesel engines emit a higher level of NO_x and Smoke emissions and have an adverse impact on human health. In order to improve energy security by reducing the dependency on fossil fuels and reducing emissions to meet the stringent emission regimes, researchers are moving towards the utilization of alternative fuels, which shall be renewable, recycled and environmentally friendly. Due to the higher cetane number (CN > 55), lower self-ignition temperature (235°C), higher oxygen content (34.8% by mass) and no C-C bond, Dimethyl Ether (DME) fuel is considered to be the most promising, renewable or clean fuel which can be suitable for compression ignition engines [1,2]. Researchers found that diesel engines can easily be retrofitted to run on DME without any major modifications to the existing engine. The pressurized DME cylinder contains the fuel in liquid form, and when it comes in contact with the atmospheric condition, it can quickly evaporate into the gaseous state. The fuel delivery systsem should be pressurized to keep the DME in a liquid state to utilize it to 100% (neat DME) in the CI engine, and partial substitution of DME (gaseous form) along with diesel is possible under dual fuel (DME-diesel) mode by minor modification in the suction manifold [3,4,5,6]. The widely available feedstock such as coal, natural gas, biomass resources (combination of sewage waste, animal waste, municipal solid waste, industrial waste, agricultural crops & residues and forest residues) can be used to produce DME fuel [7].

Khunaphan et al. conducted experimental research on a DME-diesel fueled CI engine operating at 3600 rpm under various loads (25 percent, 50 percent, and 75 percent). In comparison to diesel mode, the test findings showed that increasing the DME energy share (DME ES) enhanced engine performance at greater loads. Smoke emissions decreased to zero, but NO_x, CO, and HC emissions increased as temperature increased [8]. Theinnoi et al. investigated the experimental examination on a CRDI engine in DME-diesel mode at 1500 rpm and changing loading conditions at various DME ES percentages (10, 30, 50, 70 and 90 percent). They reported that the engine's brake thermal efficiency (BTE) increased slightly in DME-diesel mode. Smoke emission decreased to zero level; however, NO_x, CO, and HC emissions were negatively impacted in DME-diesel mode [9]. Other researchers obtained similar kinds of results; Ying et al. [10], Wang et al. [11] investigated the experimental studies on diesel engines in DME-diesel mode. The engine's improvement in BTE with zero smoke emission level and the adverse effect on CO and HC emissions at higher DME ES was found by Wang et al., who carried out the experimental study on diesel engine in DME-diesel mode at 2100 rpm [12]. Researchers reported that achieving controlled auto-ignition (CAI) with diesel-assisted ignition is difficult since DME ignites faster than diesel due to its greater cetane number. Without the risk of uncontrolled auto-ignition (UAI), a diesel engine can be run with controlled auto-ignition (CAI) up to a certain energy share of DME.

Fuel injection timing is considered one of the important strategies to improve CI engines' performance, combustion, and emission. Mani et al. [13], Cenk et al. [14] performed the experimental work on performance, combustion and emission characteristics in a CI engine using oxygenated alternative fuels, waste plastic oil and diesel-ethanol blends under different fuel injection timing. They reported that the exhaust emissions from the engine vary intensively with the change of fuel injection timing. Kulandaivel et al. [15] investigated the impact of retarded fuel injection timings (23°, 18° and 13°BTDC) on CRDI CI engines fueled with waste high-density polyethylene (WHDPE oil) blended with diesel (D70H30) at various BMEP. The results showed that the maximum combustion pressure decreases the duration of combustion gets shorter with retarded fuel injection timing. The engine's performance deteriorates with an increase in smoke, HC and CO emissions, whereas NO_x emission decreased significantly with retarded fuel injection timing. Ozturk et al. [16] examined the influence of retarded injection timing (28°, 26° and 24°BTDC) on combustion and emissions of CI engines fueled with diesel blended with canola biodiesel (D90C10) at 2200 rpm and varied loads operating conditions. The results revealed that the maximum combustion pressure and HRR decreased with retarded injection timing. The emission of NO_x reduced drastically, whereas CO, HC and smoke emissions caused slight increments with retarded injection timing. Gnanasekaran et al. [17] carried out the experimental investigations on single cylinder CI engine fueled with fish oil biodiesel to investigate the influence of injection timings (21°, 24° and 27°BTDC) on combustion, performance, and emission characteristics at 1500 rpm at different loads. It was observed from the results that the emissions of NO_x, HC and CO were reduced with a decrease in maximum in-cylinder pressure and combustion duration, whereas the performance of the engine was found higher with retarding fuel injection timing.

The above literature survey shows that DME fuel can become an alternative fuel for CI engines to reduce exhaust emissions with a positive effect on BTE. Many research studies have been done on the influence of injection timing with different alternative fuels, but no information is available on the impact of fuel injection timing on performance, combustion and emissions of DME fueled engines under DME-diesel mode. Thus, this present work is aimed to study the performance, combustion and emission characteristics of compression ignition engine equipped with a common rail direct injection (CRDI) system at 2200 rpm and at varied load conditions (20%, 40%, 60%, 80% and 100%) at standard fuel injection timing (15°BTDC) with an injection pressure of 1000 bar and further this study is extended to study the effect of retarding fuel injection timing (12°BTDC) on knock mitigation which leads to enhancement of DME energy share (DME ES) at 2200 rpm and 100% load conditions with same injection pressure under DME-diesel dual fuel mode.

2. Experimental details and methodology

2.1 DME fueled engine setup details

Fig. 1 shows the schematic layout of a single-cylinder, four-stroke common rail direct injection (CRDI) water-cooled compression ignition (CI) engine test rig. The conventional CRDI engine was modified into DME-diesel mode with minor modifications in the inlet system of the engine. The dynamometer was coupled with the output shaft of the engine. The gaseous DME fuel (main fuel) was inducted through the intake system during suction stroke, and the ULSD diesel (pilot fuel) was injected directly into the combustion chamber by CRDI electronically controlled injector at the end of compression stroke into the engine's cylinder. The technical specifications of the DME fueled CRDI research engine are given in Table 1. A newly developed electronic control unit (ECU) driver was used to control the entire operation of the CRDI engine with diesel and DME fuel injection timings. The diesel and DME flow rates were measured using Coriolis-based mass flow meters, whereas air flow rate was recorded using a thermal-based flow meter. Table 2 listed the properties of the fuels (Diesel and DME) used in the experimental study. The piezoelectric pressure transducer was used for recording the in-cylinder pressure data. An optical sensor was used to record the crank angle in degree. The smoke discharge from the exhaust of the engine was measured by using a Smoke Opacity meter with an uncertainty of ±1.85%, whereas NDIR based AVL Di-gas analyzer, was used to record the exhaust discharge such as carbon monoxide, unburned hydrocarbon and oxides of nitrogen with an uncertainty of ±1.36%, ±1.4% and ±1.4% respectively.

Table 1. Technical specifications of DME fueled CRDI engine

Make and model	Mahindra Jeeto Engine
Engine details	Common rail direct injection (CRDI)
Number of cylinders	One
Bore and Stroke	0.093 m and 0.092 m
Displacement volume	625 cc
Engine's compression ratio (CR)	19:1
Power output (rated)	11/8.2 11 hp (8.2 kW) at 3000 rpm
Maximum torque	36 Nm @ 1800 rpm
DME start of injection (suction stroke)	20 deg CA ATDC
Diesel main injection (compression stroke)	15 deg CA BTDC and 12 deg CA BTDC
Diesel fuel injection pressure	1000 bar

Table 2. Properties of fuels (DME and diesel) used in experimental tests

Characteristics	Diesel	DME
Cetane Number (CN)	50-55	55-60
Lower heating value (LHV) MJ/kg	43.2	28.4
Fuel density (kg/m^3)	855	1.98
Boiling point (°C)	126-400	-24.9
wt% oxygen	0	34.8
wt% carbon	86	52.2
wt% hydrogen	14	13

The error analysis of the measured parameters can be estimated based on the root mean square method by using Eq. (1).

$$Y_R = \left[\left(\frac{\partial U}{\partial x_1} y_1\right)^2 + \left(\frac{\partial U}{\partial x_2} y_2\right)^2 + \cdots + \left(\frac{\partial U}{\partial x_n} y_n\right)^2\right]^{1/2} \quad (1)$$

Where, Y_R denotes the total value of uncertainty; U implies the given function; x_1, x_2, …. x_n indicates the independently measured variables in function; y_1, y_2, …. y_n shows the uncertainty values of independently measured variables.

2.2 Experimental test procedure

The common rail direct injection (CRDI) compression ignition (CI) engine was used for the experimental tests under the dual-fuel (DME-diesel) at the speed of 2200 rpm with varied load conditions (20%, 40%, 60%, 80% and 100%) respectively. The experimental tests were performed under two phases. During the first phase, the CRDI engine was set to run under diesel and DME-diesel dual fuel mode (induction of DME during suction stroke in the intake system and diesel injection during the end of the compression stroke in the engine's cylinder) at 2200 rpm, varied load conditions, standard fuel injection timing (15°BTDC) and injection pressure of 1000 bar. The second phase implies the tests under diesel and DME-diesel dual fuel mode substituting the diesel percentage by the amount of DME percentage at 2200 rpm, 100% load, retarded fuel injection timing (12°BTDC) and injection pressure of 1000 bar. Further, the results of both the phases were compared respectively. Further retardation in the fuel injection timing leads to the engine's instability, resulting in incomplete combustion with deteriorated performance

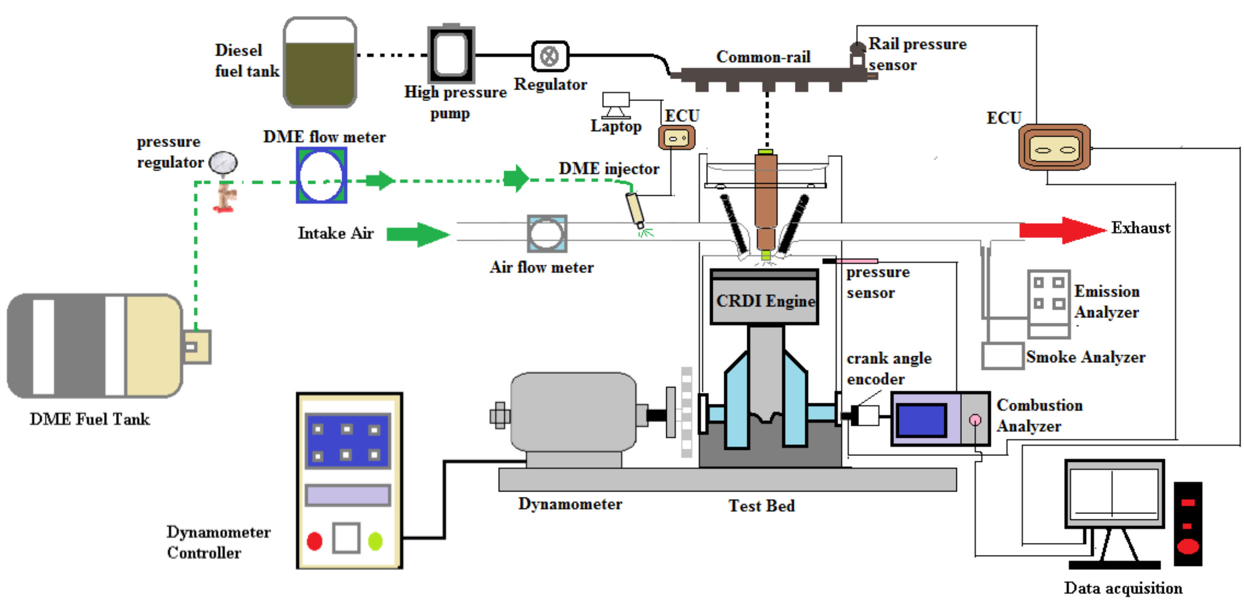

Fig. 1. Schematic layout of the CRDI engine fueled with DME

2.3 Data reduction

The DME energy share (ES_{DME}) can be calculated using Eq. (2).

$$ES_{DME} = \left[\frac{(m_{g_{DME}} * C.V._{DME})}{(m_{l_D} * C.V._D)+(m_{g_{DME}} * C.V._{DME})}\right] X\ 100 \quad (2)$$

Where, m_{l_D} denotes the mass flow rate of diesel (pilot fuel) in a liquid state (g/s), $m_{g_{DME}}$ indicates the flow rate of DME (main fuel) in gaseous state (g/s), $C.V._D$ denotes the lower calorific value of diesel (43.6 MJ/kg), $C.V._{DME}$ indicates the lower calorific value of DME (28.4 MJ/kg), respectively.

The brake thermal efficiency (η), brake specific energy consumption (BSEC) and volumetric efficiency (η_v) of the CI engine under DME-diesel dual mode can be analyzed by using Eqs. (3), (4) and (5).

$$\eta = \frac{BP\ (kW)}{(m_{l_D} * C.V._D)+(m_{g_{DME}} * C.V._{DME})}\ X\ 100 \quad (3)$$

$$BSEC = \frac{(m_{l_D} * C.V._D)+(m_{g_{DME}} * C.V._{DME})}{BP\ (kW)}(\text{MJ/kWh}) \quad (4)$$

$$\eta_v = \frac{2\ m_a}{\rho_a * V_s * N} * 100 \quad (5)$$

Where, m_a shows the mass of air (g/s), ρ_a denotes the density of incoming air (kg/m^3), V_s indicates the swept volume of the engine (m^3), N means the speed of the engine respectively.

The percentage degree of maximum DME ES (%) can be estimated by using Eq. (6).

$$percentage\ deg.of\ max.DME\ ES = \frac{Knock\ limited\ max.DME\ ES\ (\%)}{Possible\ max.DME\ ES\ (\%)} \quad (6)$$

The combustion parameters under the two phases of experiments were analyzed by applying the concept of the first law of thermodynamics using Eqs. (7), (8) and (9).

$$\delta Q_n = \delta Q_{overall} - \delta Q_{wall} = dU + \delta W \quad (7)$$

$$\frac{dQ_n}{d\theta} = \left(\frac{dQ_{overall}}{d\theta} - \frac{dQ_{wall}}{d\theta}\right) = \left[\left(\frac{\left(\frac{C_p}{C_v}\right)}{\left(\frac{C_p}{C_v}\right)-1}\ p\frac{dV}{d\theta}\right) + \left(\frac{1}{\left(\frac{C_p}{C_v}\right)-1}\ V\ \frac{dp}{d\theta}\right)\right] \quad (8)$$

$$Q_{cumulative} = Q_\theta + Q_{\theta-1} \quad (9)$$

Where, dQ_n denotes the net rate of heat release (J/°CA); (C_p/C_v) indicates the ratio of specific heat (unitless); $dQ_{overall}$ implies the comprehensive heat release rate by combustion; dQ_{wall} indicates heat lost to walls of engine's combustion chamber; $Q_{cumulative}$ denotes the cumulative heat release rate (J); θ is the crank angle (degree); p shows the combustion pressure at a given crank angle (Pa), and V denotes the cylinder volume (m^3).

The coefficient of variation (COV) of peak combustion pressure (p_{max}) and indicated mean effective pressure (IMEP) could be analyzed by using Eqs. (10), (11) and (12).

$$COV_{pmax} = (\sigma_{pmax} * 100)/(\mu_{pmax}) \quad (10)$$

$$COV_{IMEP} = (\sigma_{IMEP} * 100)/(\mu_{IMEP}) \quad (11)$$

$$\sigma = \left[\frac{1}{N}\ \sum_{i=1}^{N}(x_i - \mu_x)^2\right]^{1/2} \quad (12)$$

Where, σ denotes the standard of deviation of parameters, μ denotes the mean value of parameters, N indicates the total number of cycles, i specifies the cycle number, and x represents the parameters (p_{max} and IMEP), respectively.

3. Results and Discussion

Fig. 2 shows the influence of DME Energy Share (DME ES) on the performance of the CRDI engine under DME-diesel mode at 2200 rpm with different loads at standard fuel injection timing (15 deg BTDC). The BTE and BSEC of the engine were analyzed by using Eqs. (3) and (4) respectively. The figure indicates that the BTE of the engine increased significantly with an increase in DME energy share (DME ES) at each load, and beyond a certain percentage of DME ES, BTE decreased. At 100% load, the highest BTE of the engine with DME addition was found to be higher by 9.50% compared to diesel mode. Similarly, at 80%, 60%, 40% and 20% loads, the efficiency with DME-diesel mode was 11.2%, 11.2%, 13.0% and 21.2% respectively higher than diesel mode. The homogeneous mixture of DME fuel and air, and reduction in BSEC lead to rapid combustion (due to higher cetane number of DME fuel (> 55) and latent heat of vaporization), which increased the maximum combustion pressure. As a result, BTE of the CRDI engine increased with DME-diesel mode up to a maximum of 52% of DME ES at 100% load. Similarly, the maximum DME ES at 80%, 60%, 40% and 20% load conditions were found to be 54%, 55%, 58% and 63% respectively. To achieve the combustion with controlled auto-ignition (CAI) mode, it is necessary to keep the DME fuel-air mixture beyond the flammability limit, i.e. 3.5 to 18.5% by volume. Otherwise, uncontrolled auto-ignition (UAI) mode leads to occur, if the charge is found to be within the flammability limit. The UAI mode of operation was observed beyond 52% of DME ES resulted in knock occurrence, and high ROPR leads to a decrease in BTE (loss of useful power) of the CRDI engine. Similarly, knocking phenomena were observed beyond 54%, 55%, 58% and 63% of DME ES at 80%, 60%, 40% and 20% loads, respectively.

The BSEC of CRDI engine for DME-diesel and diesel operations was observed as 12.06 MJ/kWh (at maximum DME ES) and 13.21 MJ/kWh at 100% load condition. The BSEC of engine with DME-diesel mode decreased by 8.70%, 10.07%, 10.09%, 11.46% and 17.48% (at maximum DME ES) at 100%, 80%, 60%, 40% and 20% loads as compared with diesel mode respectively. The increase in the level of homogeneity (uniform mixing of DME-air charge) leads to better combustion, resulting in a reduction of BSEC of the engine with an increase in DME ES. Further investigation was carried out to analyze the percentage degree of maximum DME ES (%), which is illustrated in Table 3.

Fig. 3 demonstrates the variation of the maximum rate of pressure rise (ROPR) with respect to DME ES at 2200 rpm, different loads and standard fuel injection timing (15°BTDC). It can be observed from the figure that at maximum load condition, ROPR decreased from 5.08 bar/°CA (diesel fuel) to 3.1 bar/°CA (optimum 52% DME ES). This specifies the favorable engine stability conditions with the smooth engine running and less combustion noise under CAI mode. Further increase in the DME ES beyond 52%, ROPR increased to 8.06 bar/°CA showing the UAI mode with knock occurrence. Similar trends have been observed with other load conditions. Fig. 3 displays the influence of DME ES on equivalence ratio (ϕ). It is observed that at 52% of optimum DME ES at 100% load, the equivalence ratio (ϕ) found equivalent to 1.0, which specifies the stoichiometric combustion. Further, an increase in the percentage of DME ES beyond 52%, the equivalence ratio is observed with greater than 1.0 and exhibits incomplete combustion due to uncontrolled auto-ignition (UAI) phenomena results in knocking occurrence and higher ROPR. A similar kind of results was observed at other load conditions, respectively.

Fig. 4 demonstrates the emission analysis at 2200 rpm with different loads and standard fuel injection timing (15°BTDC) under DME-diesel mode. Emissions of NO_x and CO increased marginally with an increase in the percentage of DME ES, but it is still lower than diesel fuel. The formation of NO_x emission is primarily dependent upon the combustion temperature, residence time, and nitrogen and oxygen concentration. At 100% load, NO_x emission decreased from 1.65 g/kWh (diesel) to 1.38 g/kWh (52% optimum DME ES), which could be due to dilution of oxygen (the gases formed during LTR and HTR combustion of DME fuel acts as an internal EGR) by DME fuel.

It can also be noted that the emission of NO_x increased slightly with an increase in the percentage of DME ES from 1.35 g/kWh to 1.40 g/kWh under controlled auto-ignition combustion mode due to the rise in the combustion temperature, which shows a more dominant nature under dual fuel mode than the dilution effect. The formation of NO_x (NO and NOx) through the Zeldovich mechanism (during the combustion process) is given in Eqs. (13-15) [18];

Equations	Temperature Range-K	
$N_2 + O \rightarrow N + NO$	2000-5000	(13)
$N + O_2 \rightarrow O + NO$	300-3000	(14)
$N + OH \rightarrow H + NO$	300-2500	(15)

At 100% load, CO emission reduced from 7.75 g/kWh (diesel) to 2.77 g/kWh (52% DME ES) due to an increase in the combustion temperature results in the oxidation of CO to CO_2. However, CO emission further increased moderately from 0.85 g/kWh to 2.80 g/kWh with an increase in the percentage of DME ES due to the incomplete combustion of the trapped mixture of DME and air present around the piston bowl and crevice of the cylinder. This trapped mixture slows down the oxidation reaction and results in the formation of CO.

Fig. 4 illustrates the increase in HC emission from 0.01 g/kWh (diesel) to 0.09 g/kWh (52% optimum DME ES), which could be due to the trapped DME fuel mixture inside the cylinder's piston bowl and crevice. It results in a heterogeneous distribution of fuel mixture inside the engine's chamber, leading to incomplete combustion. The smoke emission reduced drastically to zero levels, as shown in Fig. 4, from diesel (30.2% opacity) to DME-diesel mode (52% DME ES) at 2200 rpm and 100% load condition. The better homogeneity of DME-air mixture due to higher diffusivity and absence of C-C bond in DME structure results in a reduction of smoke emission to zero level.

Table 3. Percentage degree of maximum DME ES at 2200 rpm and different loads

Speed (rpm)	Load (%)	Knock limited maximum DME ES (%) [A]	Knock at DME ES (%) [B]	Possible maximum DME ES (%) [C]	Percentage degree of maximum DME ES (%) [D = A/C]
2200	100	52.0	56.0	80.0	65.00
	80	54.0	57.0	80.0	67.50
	60	55.0	58.0	80.0	68.75
	40	58.0	62.0	80.0	72.50
	20	63.0	65.0	80.0	78.75

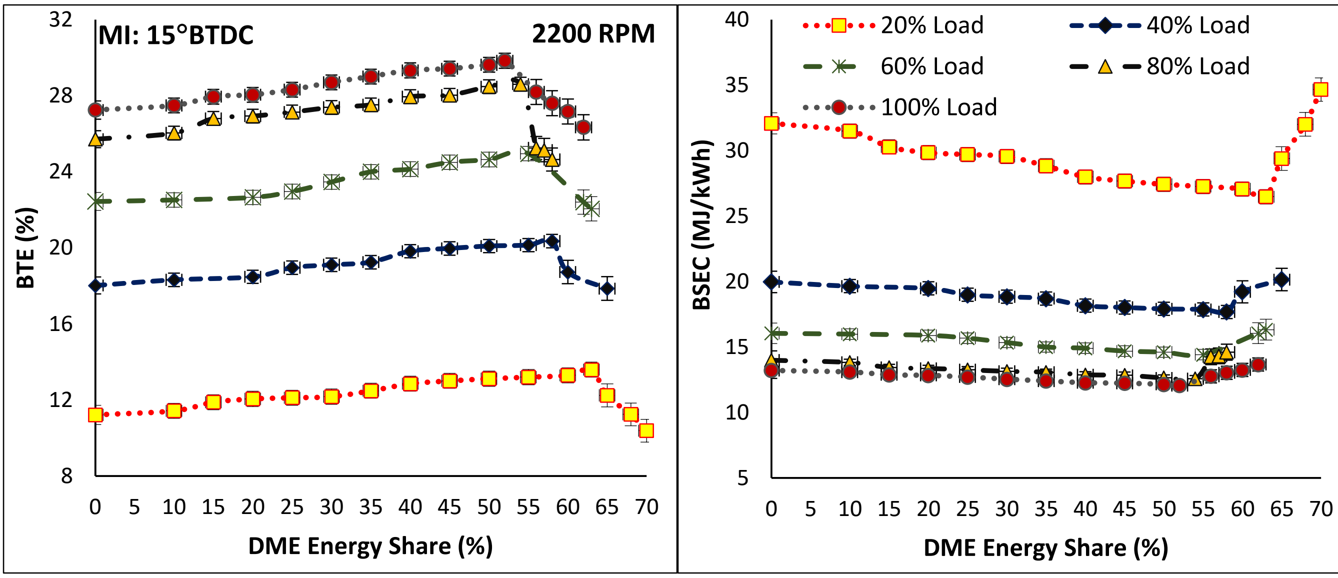

Fig. 2. Performance of engine at 2200 rpm, different loads and 15°BTDC injection timing

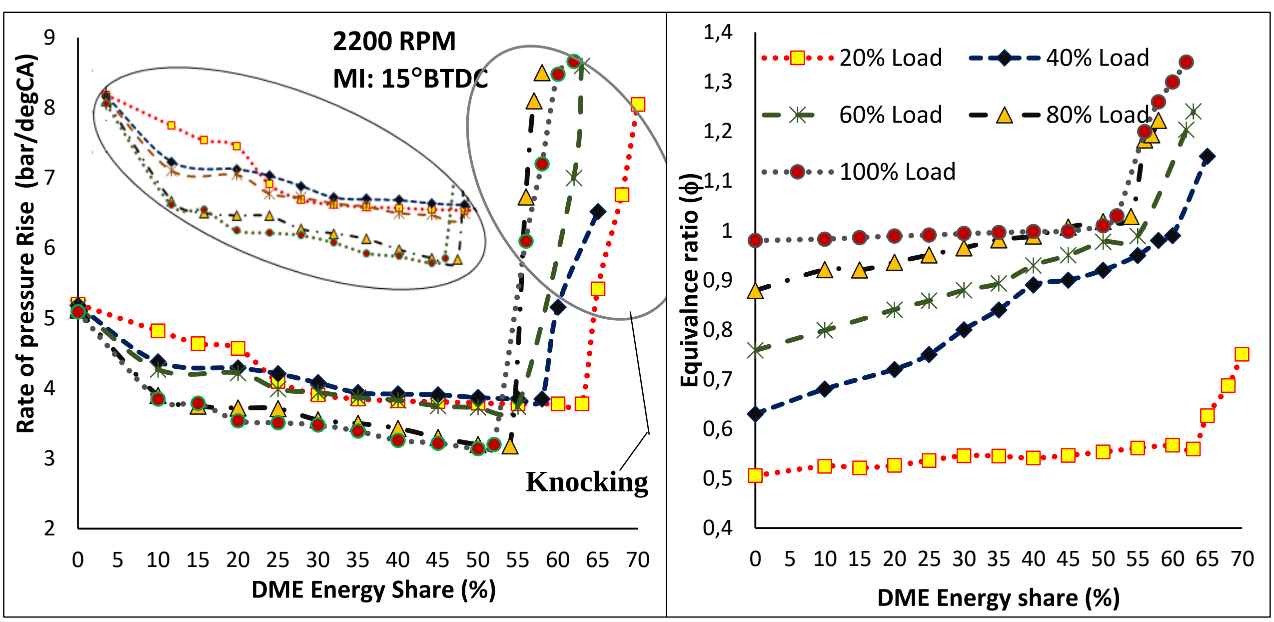

Fig. 3. Influence of DME ES on ROPR and equivalence ratio at 15°BTDC injection timing

The impact of DME ES on combustion pressure under DME-diesel mode at 2200 rpm, 100% load and 15°BTDC fuel injection timing are shown in Fig 5. The experimental results showed that as load increases, more fuel gets consumed, which subsequently leads to an increase in maximum combustion pressure and temperature in the engine's combustion chamber. It was observed that peak in-cylinder pressure with DME-diesel mode was 91.8 bar higher than with base diesel mode of 90 bar. The DME fuel, due to its higher CN (55-60) and lower self-ignition temperature (508 K), gets ignition prior to diesel and initiates the combustion process for the DME-diesel system in the form of HCCI (Homogeneous Charge Compression Ignition) mode resulting in homogeneous mixture formation between air and DME fuel during the period from suction stroke to compression stroke (prior to diesel injection) and, leads to complete oxidation and better combustion of mixing resulting in the advancement of position and, increased in maximum combustion pressure and temperature significantly. The maximum DME ES in DME-diesel mode is limited up to 52% at

2200 rpm, 100% load. Beyond 52% DME ES, knocking phenomena occur leads to uncontrolled auto-ignition (UAI).

The variation in the plots of heat release rate (HRR) – theta (θ) and cumulative heat release (CHR)-theta (θ) with an increase in the percentage of DME ES at 2200 rpm and 100% load are shown in Figs 6 and 7. Due to the higher cetane number (55-60) of DME fuel, it gets ignites first (due to lower self-ignition temperature (508K) than diesel (588K)) and then initiates the ignition of diesel leads to an increase in combustion pressure and combustion temperature under DME-diesel mode. The rate of heat release (HRR) shows three stages of combustion, i.e. low-temperature reaction (LTR-cool flame) region, high-temperature reaction (HTR) region and diffusion-controlled CI combustion. The DME-diesel mode shows the combustion similar to HCCI combustion and exhibits the LTR-cool flame and HTR regions. When the temperature is in the range of 700-900 K, the LTR region promotes the ignition by breaking the C-H bond of DME, which leads to the formation of CH_3O-CH_2O_2 with subsequent intermolecular isomerization. Further, when the temperature reaches above 1200 K, the mixture gets ignited in the HTR region. In this region, a large amount of energy gets released with the formation of H_2O_2, CO_2 and CO. The diffusion combustion is dominant for diesel fuel. The LTR and HTR combustion phase becomes dominant under DME-diesel mode due to the availability of adequate time for mixing of charge (from suction stroke to prior injection of diesel at the end of compression stroke) and better ignition quality of gaseous fuel (DME). It is observed from Fig. 6 that the LTR-cool flame region occurs at a crank angle between 25 to 20°CA BTDC and HTR region takes place at a crank angle between 10 to 5°CA BTDC. Similar kinds of results were reported by Wang et al. [19], Kim et al. [20] and Zhang et al. [21] regarding three staged heat release analysis under DME-diesel mode in CI engine.

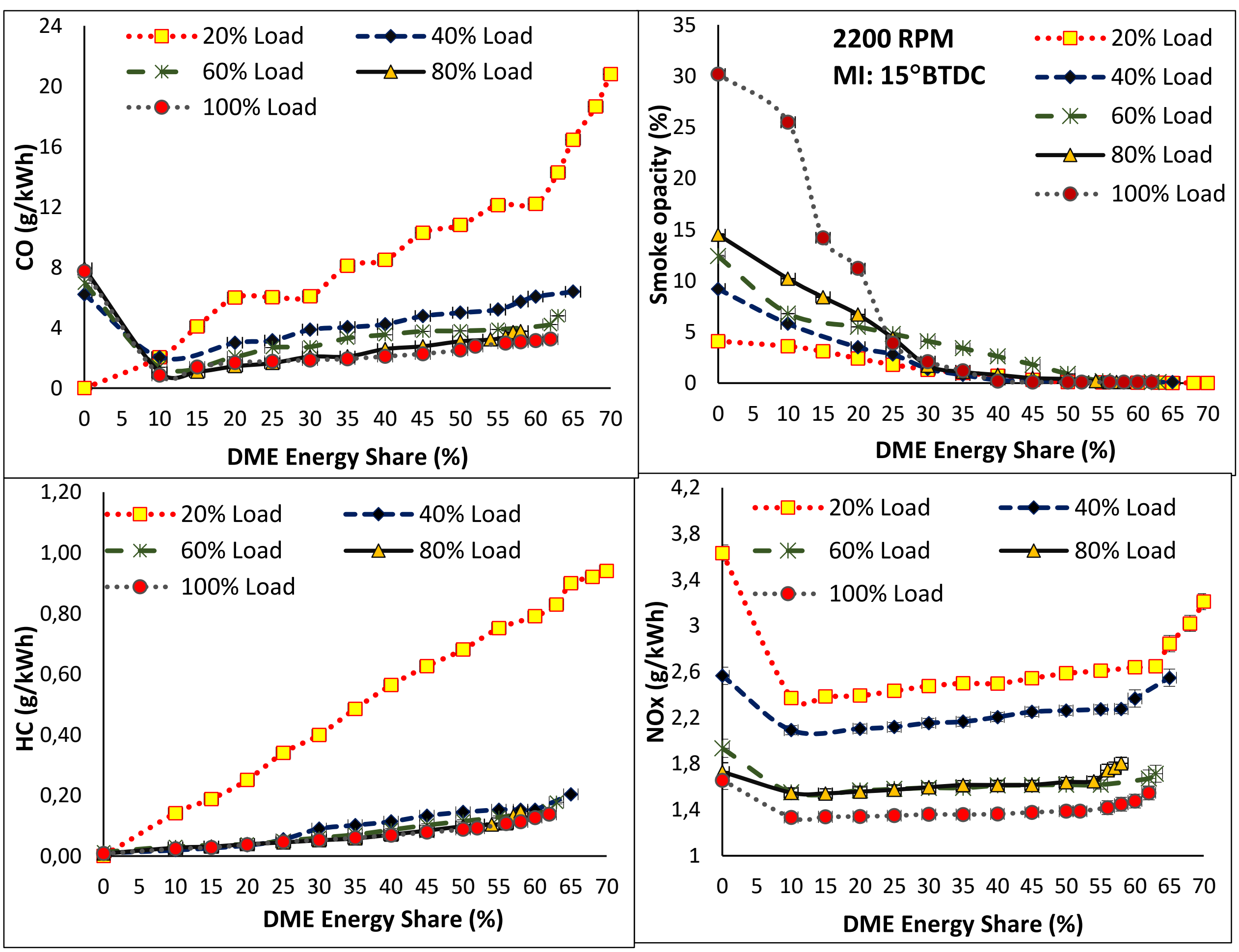

Fig. 4. Emissions analysis at 15°BTDC under DME-diesel mode at 2200 rpm & different loads

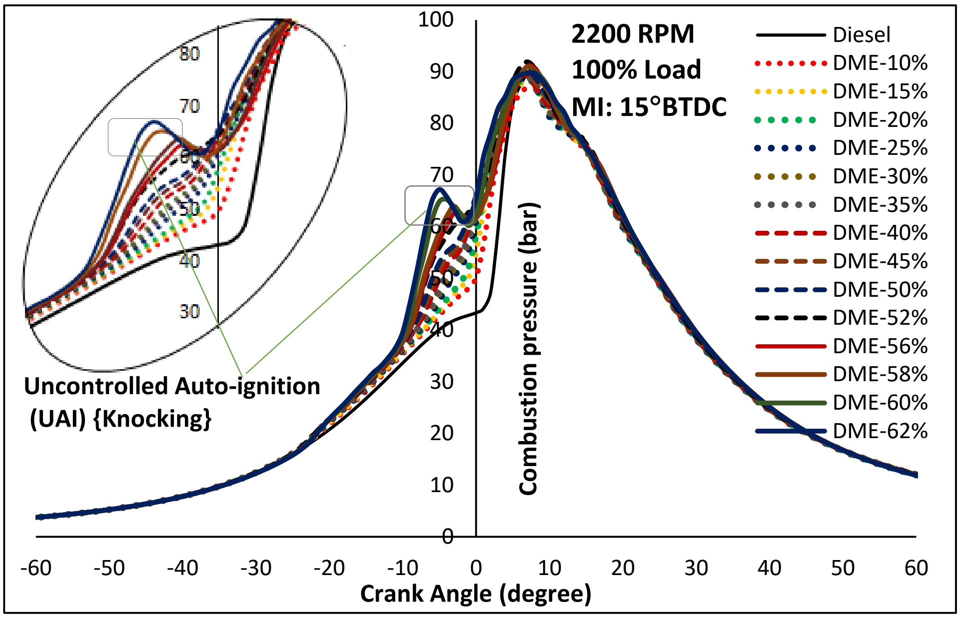

Fig. 5. Analysis of combustion pressure at 15 °BTDC under DME-diesel mode

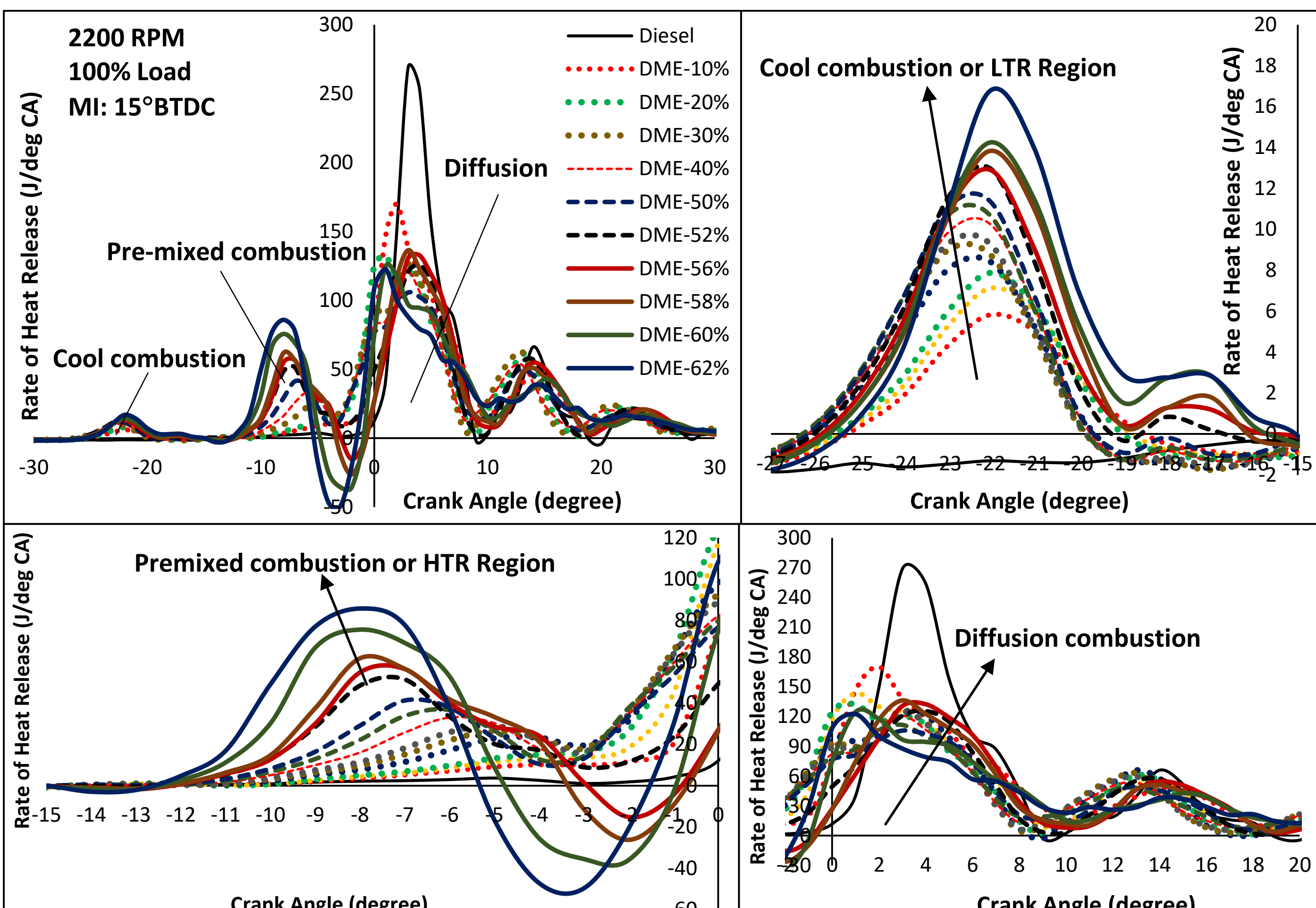

Fig. 6. Impact of DME ES on heat release at 15°BTDC under DME-diesel mode

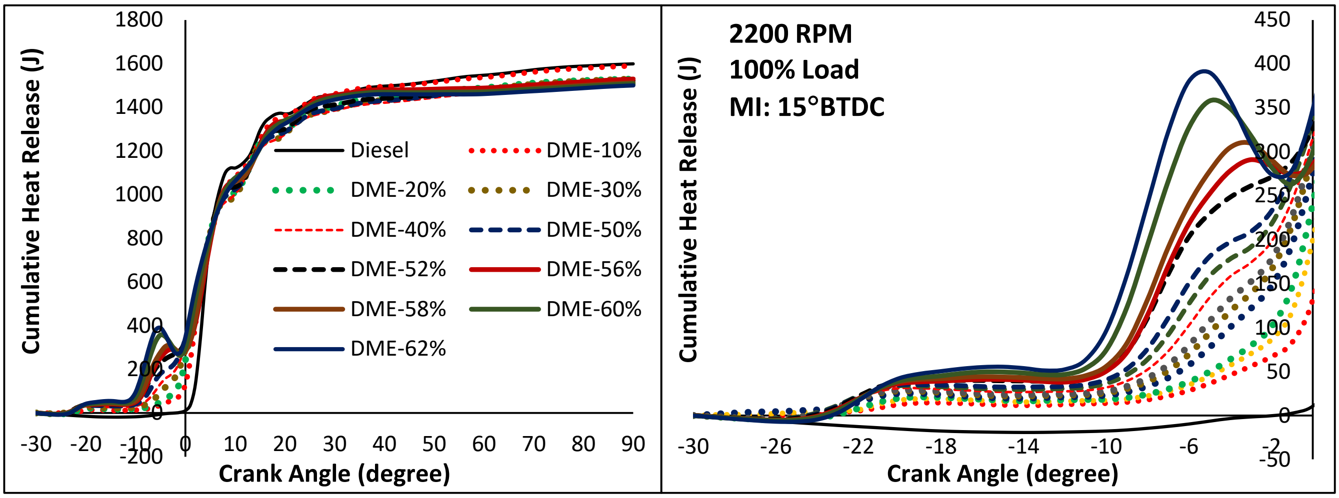

Fig. 7. Influence of DME ES on cumulative heat release at 15°BTDC under dual fuel mode

Fig. 8 shows the variation of brake thermal efficiency (BTE) with increasing DME ES at different injection timing and maximum load conditions. The BTE of the CRDI engine from diesel mode to limited energy share of DME under DME-diesel mode at 2200 rpm and 100% load varies from 27.25% to 29.84% (52% DME ES), 27.01% to 29.90% (58% DME ES) at 15°BTDC and 12°BTDC respectively. The BTE deteriorated marginally when the fuel injection timing was delayed from 15°BTDC to 12°BTDC.

For 12°BTDC injection timing, BTE decreased marginally by 0.88% with diesel fuel at 100% load compared to standard fuel injection timing (15 °BTDC). This is maybe due to the shorter ignition delay (17 deg CA with 15°BTDC to 14 deg CA with 12°BTDC), and shorter combustion duration (30 deg CA with 15°BTDC to 25 deg CA with 12°BTDC) leads to incomplete combustion (due to non-uniform mixing of air-fuel charge), thus diminishing the BTE of the engine. Further, with the induction of DME percentage under dual-fuel, BTE increased and achieved the highest efficiency of 29.90% at 58% of DME ES due to the uniform mixing of charge (higher cetane number and wider flammability limits of DME fuel) leads to rapid combustion. Hence, at 12 °BTDC, BTE increased uniformly with DME-diesel mode upto 58% of maximum DME ES at 100% load. Beyond 58% DME ES, the knock that occurred results in a further decrease in BTE of the engine due to the rate of pressure rise and loss of power.

Similarly, BSEC increased when the injection timing is delayed from 15°BTDC to 12°BTDC. This could be due to the late start of fuel injection leading to poor mixing of charge and shifting the combustion phase away from the top dead centre, leading to more fuel consumption and incomplete combustion. Further, with the injection of DME under dual fuel mode, BSEC was reduced from 13.32 MJ/kWh to 12.02 MJ/kWh (at 58% DME ES) due to the uniform mixing of gaseous DME fuel with air leads to rapid and better combustion at 12°BTDC injection timing. Beyond 58% of DME ES, knocking tendency tends to occur results in more consumption of fuel leads to increase in BSEC of the engine. Fig. 8 also demonstrates that as DME ES increases, the engine's volumetric efficiency (ηv) significantly decreases with both injection timings, which may be due to an increase in the percentage of DME fuel induction (in gaseous form) during suction stroke. Fig. 8 displays the variation of DME ES with different injection timings (15°BTDC and 12°BTDC) under DME-diesel mode at 2200 rpm and 100% load. Retarded injection timing could mitigate knock to a certain extent due to the lower in-cylinder pressure rise and temperature. The DME ES increased from 52% with 15°BTDC to 58% with retarded 12°BTDC injection timing. The lower self-ignition temperature and higher cetane number of DME fuel lead to shorter ignition delay (17 deg CA with diesel to 6 deg CA with 52% DME ES at 15°BTDC, and from 14 deg CA with diesel to 5 deg CA with 58% DME ES at 12°BTDC), which further leads to lower combustion pressure rise rates and temperature results in enhancement in DME ES at retarded injection timing. It was found from the study that CI engine under DME-diesel mode is limited to a certain amount of DME ES due to more probability of uncontrolled auto-ignition and knock problems.

The maximum rate of pressure rise (ROPR) shows a similar trend (Fig. 9) with both fuel injection timing, i.e. 15°BTDC and 12°BTDC at 2200 rpm and maximum load conditions. ROPR decreased from the diesel mode to DME-diesel mode up to a certain percentage of DME ES, showing the controlled auto-ignition combustion (smooth running condition). Beyond a certain percentage of DME ES, ROPR increased over the standard value of 6 bar/°CA, leads to uncontrolled auto-ignition results in knocking phenomena. Similarly, the variation in equivalence ratio (ϕ) shows a similar trend (Fig. 9) with both fuel injection timings (15°BTDC and 12°BTDC) at maximum load conditions. The equivalence ratio was observed to be less than or equal to 1.0 indicates the stoichiometric combustion. However, beyond a certain percentage of DME ES, an equivalence ratio greater than 1.0 indicates the uncontrolled auto-ignition leads to knocking occurrence with incomplete combustion.

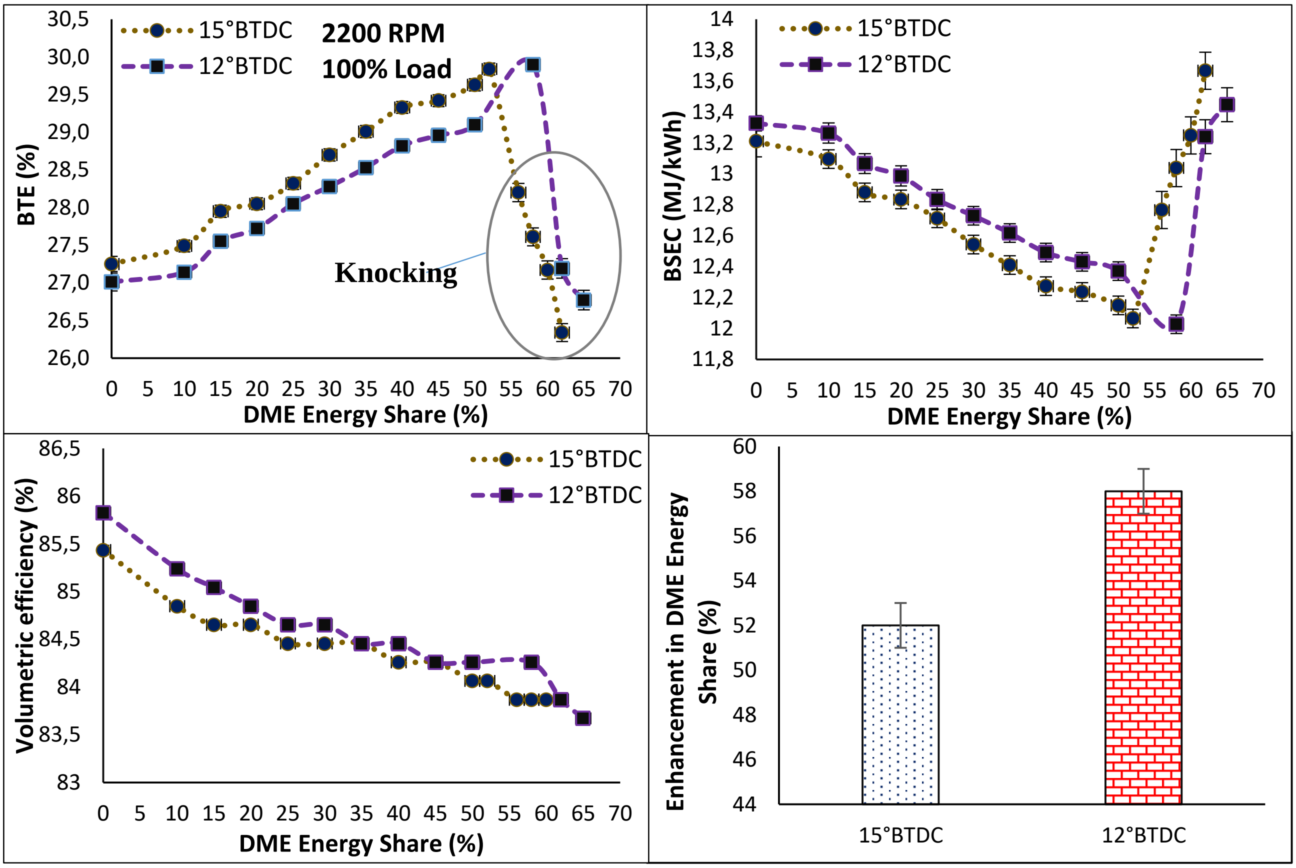

Fig. 8. Performance characteristics at different injection timings under DME-diesel mode

Fig. 9 illustrates the variation in the plots of combustion pressure (bar) – theta (θ) with DME-diesel mode at different injection timings at 2200 rpm and 100% load conditions. It can be clearly observed that peak combustion pressure drops due to the shorter ignition delay (17 deg CA with diesel to 6 deg CA with 52% DME ES at 15°BTDC, and from 14 deg CA with diesel to 5 deg CA with 58% DME ES at 12°BTDC), which supports shifting the combustion phase away from TDC, when injection timing is retarded from 15°BTDC to 12°BTDC. The stability of engine operation was analyzed by using the important parameters, i.e. coefficient of variation (COV) of peak combustion pressure and indicated mean effective pressure (IMEP), and it can be observed from the figure that both the parameters decreased when the fuel injection timing retarded from 15°BTDC to 12°BTDC. This specifies that the late fuel injection timing is suitable for stable DME fueled engine operation. The COV was considered as one of the important parameters in this study to decide the limited or optimum DME ES under DME-diesel mode.

Fig. 10 illustrates the emissions with different fuel injection timings (15°BTDC and 12°BTDC) at 2200 rpm and maximum load conditions under DME-diesel mode. The CO emission just slightly increased, and HC emission increased significantly and linearly, while retarding injection timing from 15°BTDC to 12°BTDC. The increase in CO and HC emissions is attributed to the accumulation of some charge (DME-air) in crevices, which slows down the oxidation reaction and leads to inferior combustion with the formation of CO and HC pollutants. However, the important outcome that emerged from the study is that higher CO and HC emissions tend to increase the possibility of HCCI combustion (as DME HCCI combustion exhibits higher CO and HC emissions) with the increase in the percentage of DME ES. When the fuel injection timing was retarded from 15°BTDC to 12°BTDC, the NO_x emission reduced significantly due to the delayed start of injection, which shifts the combustion phase away from TDC and results in the reduction of peak combustion pressure and temperature, and thus, suppresses the NO_x formation. Smoke emission tends to increase in the case of diesel mode, when the injection timing was retarded from 15°BTDC to 12°BTDC. The reduction in combustion temperature due to delayed start of injection (SoI), leads to improper combustion of the hydrocarbon in fuel and partial oxidation of carbon content in fuel which results in smoke formation. Further, with an increase in the percentage of DME quantity, smoke emission reduced to zero levels. The higher oxygen content (34.8% by mass), absence of C-C bond and better homogeneity of DME-air mixture leads to zero levels of smoke emission. The smoke emission derived from the lubricant was found negligible, and hence, the DME fueled CRDI engine can be concluded as the smoke-free engine under all operating conditions.

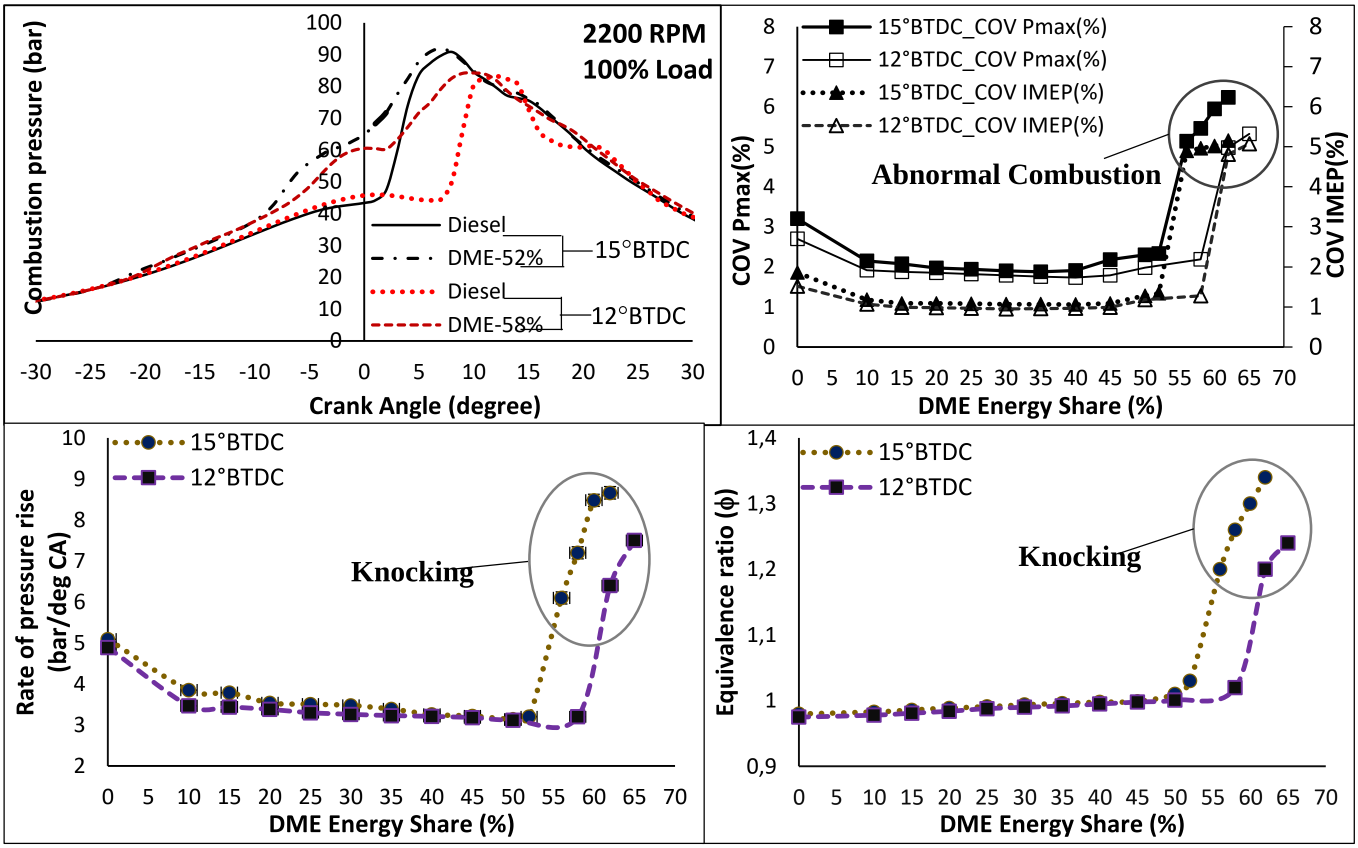

Fig. 9. Combustion characteristics at different injection timing under DME-diesel mode

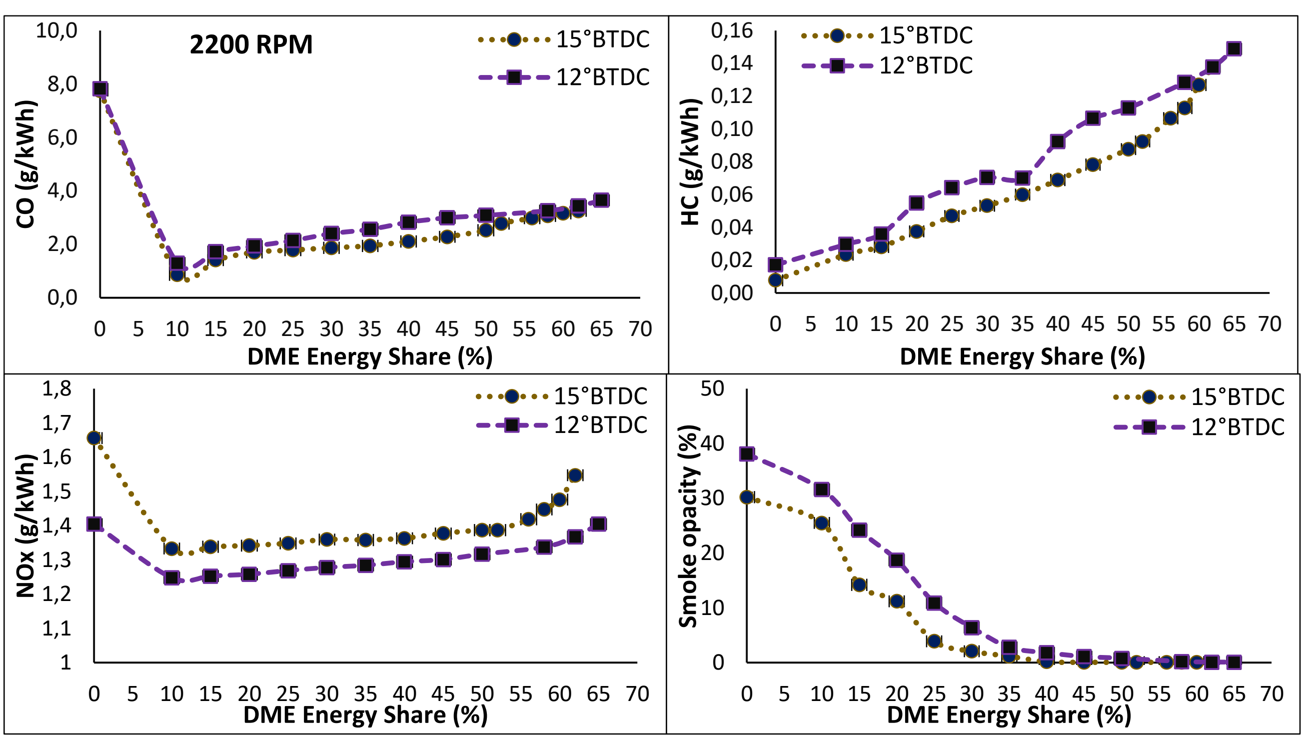

Fig. 10. Emissions characteristics at different injection timing under DME-diesel mode at 100% load

Figs 11, 12 and 13 demonstrate the cycle by cycle variation of the maximum pressure (p_{max}), the maximum rate of pressure rise ($ROPR_{max}$), and the indicated mean effective pressure (IMEP) for 100 consecutive cycles at 2200 rpm and 100% load. For research engines, 5% of the coefficient of variation (COV) value is used as the limit value for COV of each parameter, whereas 6 bar/°CA is used as the standard limiting value for maximum rate of pressure rise ($ROPR_{max}$) for the research engine [18]. The results show that both COV_{pmax} and COV_{IMEP} are found less than 5%, with an increase in the percentage of DME energy share (DME ES). The maximum DME ES under DME-diesel mode with base injection timing (15°BTDC) and retarded injection timing (12°BTDC) under controlled auto-ignition (CAI) is limited to 52% and 58% respectively. Beyond a certain percentage of DME ES (52% with 15°BTDC and 58% with 12°BTDC), a wide range of variations was observed in the maximum rate of pressure rise ($ROPR_{max}$), the maximum pressure (p_{max}), and indicated mean effective pressure (IMEP) with higher values of COV_{pmax} and COV_{IMEP} leads to uncontrolled auto-ignition (UAI) tends to knock occurrence with combustion instability and results in limiting the operating range of the CRDI engine under DME-diesel mode.

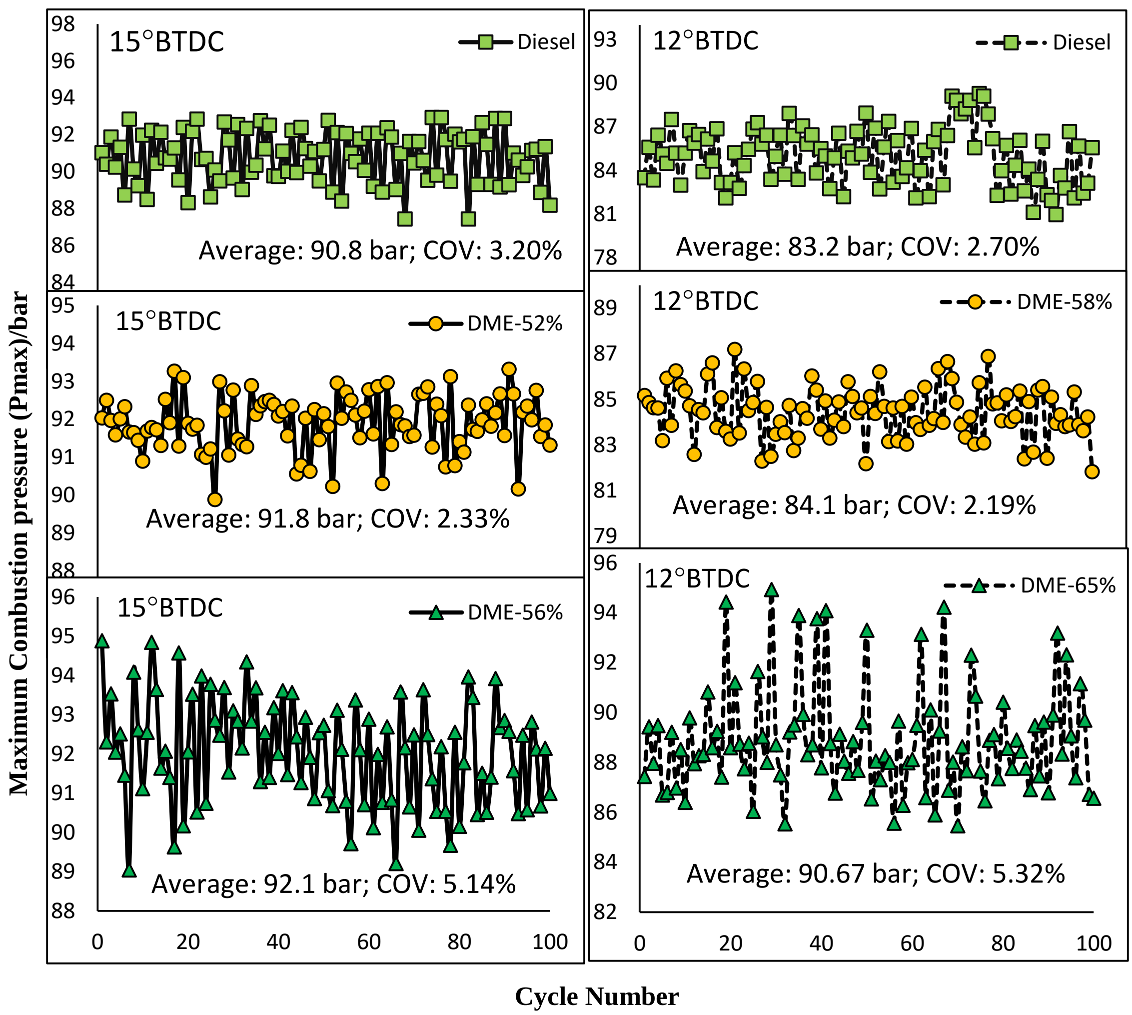

Fig. 11. Cycle-by-cycle variation of maximum combustion pressure at 15°BTDC and 12°BTDC under DME-diesel mode

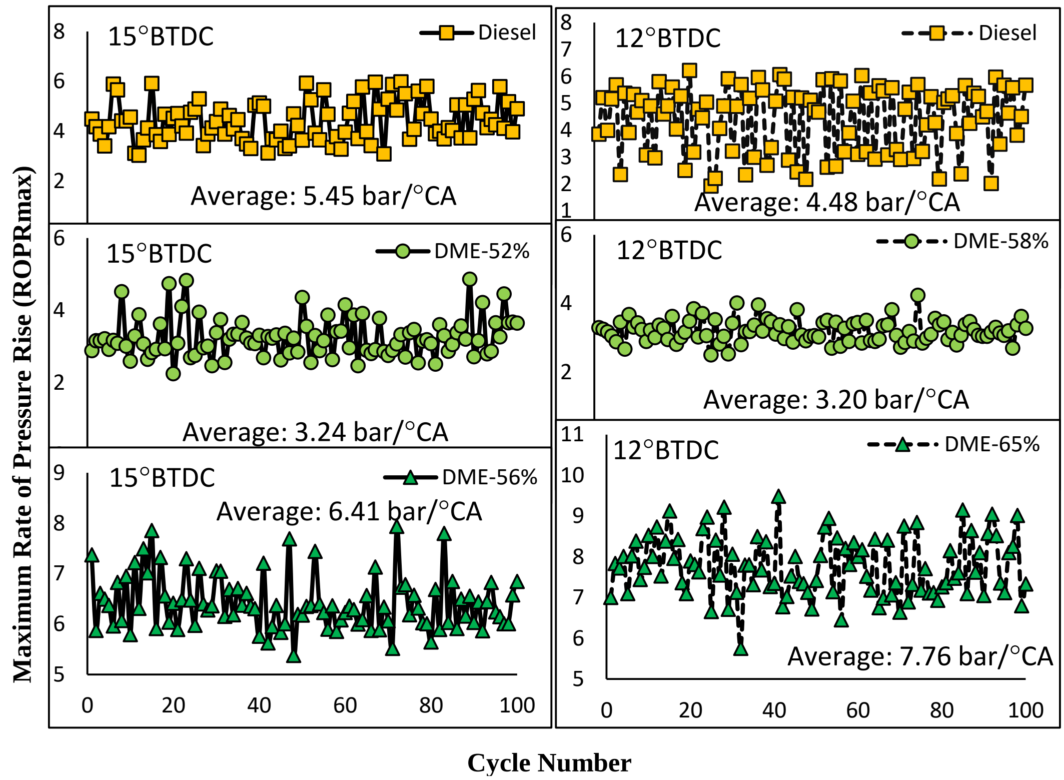

Fig. 12. Cycle-by-cycle variation of the maximum rate of pressure rise (ROPR) at 15°BTDC and 12°BTDC under DME-diesel mode

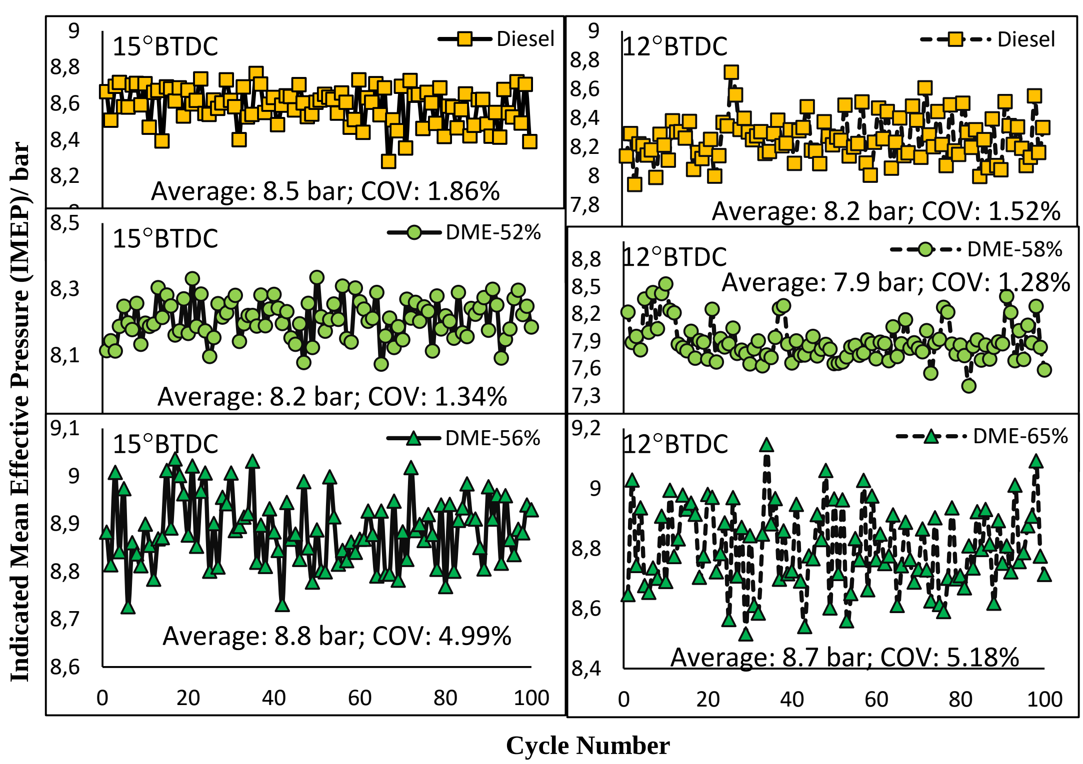

Fig. 13. Cycle-by-cycle variation of indicated mean effective pressure (IMEP) at 15°BTDC and 12°BTDC under DME-diesel mode

4. Conclusions

The impact of different fuel injection timings on an automotive CRDI engine under DME-diesel mode was examined at the speed of 2200 rpm and maximum load conditions. The main outcome of the study are as follows:

i. Increasing engine load results in an increase in brake thermal efficiency with zero levels of smoke emission however, the CO and HC emissions increased significantly with an increase in DME ES. NO_x emission was marginally higher with an increase in DME ES.
ii. Retarding injection timing decreased BTE of the engine however it further increased significantly with an increase in the percentage of DME ES. NO_x emission significantly decreased however, the other emissions like CO and HC increased marginally with retarding fuel injection timing and further increased moderately with an increase in the percentage of DME fuel.
iii. Smoke emission reduced to zero levels with both fuel injection timings.
iv. The combustion duration and ignition delay get shorter with the decrease in the maximum combustion pressure with retarding injection timing.
v. The ROPR, $(COV)_{pmax}$, $(COV)_{IMEP}$, and equivalence ratio (ϕ) was one of the important parameters to decide the optimum DME ES under dual fuel mode.
vi. Retarded injection timing could mitigate knock to a certain extent due to the lower in-cylinder pressure rise and temperature. The DME ES increased from 52% with 15°BTDC to 58% with retarded 12°BTDC injection timing.

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Nomenclature

CI	Compression-Ignition	*CRDI*	Common Rail Direct Injection
DME	Dimethyl Ether	*ULSD*	Ultra-Low Sulphur Diesel
DME ES	Dimethyl Ether Energy Share	*CN*	Cetane Number
CAI	Controlled Auto-ignition	*UAI*	Uncontrolled Auto-ignition
HCCI	Homogeneous Charge Compression Ignition	*BTDC*	Before Top Dead Centre
HRR	Heat Release Rate	*CHR*	Cumulative Heat Release
COV	Coefficient of Variation	*IMEP*	Indicated Mean Effective pressure
LTR	Low Temperature Region	*HTR*	High Temperature Region
BTE	Brake Thermal Efficiency	*BSEC*	Brake Specific Energy Consumption
ECU	Electronic Control Unit	*LHV*	Lower Heating Value
NDIR	Non-dispersive infrared	*ROPR*	Rate of pressure rise
PM	Particulate Matter	NO_x	Oxides of Nitrogen
CO	Carbon Monoxide	*HC*	Hydrocarbon
CO_2	Carbon dioxide	N_2O	Nitrous Oxide

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have influenced the work reported in this paper.

CRedit Author Statement

Anilkumar Shere: Conducted experiment, Manuscript preparation, Data interpretation, Graphical presentation,
K A Subramanian: Conceptualization, Methodology, Supervision, Data interpretation

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