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Experimental Investigations on Enhancement of DME Energy Shares in Compression-Ignition Engine Under Dual Fuel Mode Using Reduced Compression Ratio

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Abstract

The effects of the compression ratio (CR), which was reduced from the base (19:1) to (17:1), on the enhancement of Dimethyl Ether Energy Share (DME ES) and emissions reduction in a 5.5 kW DICI (direct injection compression ignition) engine are studied. The experimental tests were conducted on the engine with DME as main fuel and Diesel as pilot fuel at maximum torque and speed of 2200 rpm condition under dual fuel mode (DME-diesel). The maximum DME energy share (DME ES) enhanced drastically from 44% with base CR to 53% with the reduced CR. Smoke emission decreased to nearly zero levels with both CRs (19:1 and 17:1). CO and HC emissions significantly reduced with 53% of DME ES with reduced CR. NOx emission decreased drastically with the reduced CR (17:1). However, brake thermal efficiency is marginally dropped with the re-duced compression ratio. The combustion characteristics are discussed in detail.

Keywords: Compression ratio; Emissions; Diesel engine; DME energy share; Dual fuel mode.

Research Article

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1. Introduction

Diesel engines play a major role in passenger and mass transportation applications. However, these engines emit a high level of smoke / Particulate Matter (PM) and Oxides of Nitrogen (NO_x). Many technologies such as common rail direct injection (CRDI), selective catalyst reduction (SCR), diesel particulate filter (DPF) are adopted to improve performances and reduce emissions. Besides the adoption of present technologies, the issues of emissions, i.e. PM/smoke, oxides of nitrogen, remain unresolved to meet emissions standards as simultaneous reduction of PM and NO_x emissions remains a big challenge worldwide.

The aim for adaptable and affordable alternative fuels to replace diesel fuel part or full in CI engines are Fischer-Tropsch Diesel, biodiesel, and DME. These fuels are a higher cetane number (CN) (Fischer-Tropsch-75, Biodiesel-58, and DME-58) compared with base diesel having CN-51. Comparing these options for CI engines, DME is more preferred due to attaining a gaseous state at atmospheric condition (boiling point-24° C), leading to better mixing with air and a wider range of flammability limit (3.5 to 27), and combustion resulting in lower emissions.

DME is a clean, renewable, and oxygenated alternative fuel suitable for CI engines [1,2]. The production technology for DME fuel is mature and can be produced from natural gas, coal, and biomass. With the higher cetane number (ignition quantity) of DME (58) than the base diesel (51), DME fuelled engine could operate with good cold stability, lower probability of combustion with a knock, lower transient emissions, lower noise, and higher engine's life than the base diesel engine. DME fuel can be utilized in diesel engines without major engine design modifications. The fuel supply system has to be modified completely to replace Diesel with DME fuel or a slight change in the intake manifold design for injecting DME in gaseous form.

The liquid state of DME fuel can be utilized in CI engines by injecting it into the combustion chamber directly during the compression stroke, similar to conventional diesel engines [3-6]. Alternatively, DME can be utilized as auxiliary fuel in diesel-fuelled CI engines known as dual-fuel engines [7,8]. This means that the DME fuel is injected into the intake manifold along with the air during the intake stroke, and diesel fuel is injected directly into the combustion chamber at the end of the compression stroke to start the ignition of DME and achieve controlled auto-ignition. The dual-fuel mode offers fuel flexibility as an engine can operate with two fuels (DME-Diesel). In case, if DME fuel is not available, the engine can run with Diesel alone. The dual-fuel mode approach is preferable until the infrastructure for the DME supply chain system is established. Therefore, this paper represents the use of DME fuel in the compression ignition engine under dual fuel mode, as this technology could be implemented in in-use diesel-fuelled engines/vehicles without major engine hardware modifications along with advantages of fuel flexibility. DME fuel ignites faster than diesel fuel as it exhibits a higher cetane number, thus it is a big challenge to achieve controlled auto-ignition (CAI). However, the CI engines can be operated with certain energy shares of DME fuel with ignition assisted with diesel fuel without the problem of uncontrolled auto-ignition under dual-fuel mode. Thereafter, further enhancement of DME ES can be achieved by reducing the compression ratio of engines taking advantage of a higher cetane number of DME, which affects uncontrolled auto-ignition compared with standard compression ratio. The technology for enhancing DME ES is not reported by the researchers. Kajitani conducted the experiments with a direct-injection compression ignition engine, with the rated power output of 8.45 kW at 2600 rpm and compression ratio (CR) of 17.7. The test was conducted with diesel fuel and 100% DME by modifying the fuel injection system at 960 rpm of mean effective pressure 0.6 MPa without reducing CR. He reported that DME has the potential to use at CR 14. He reported negligible soot emission; however, NO_x emission is higher at lower CR with DME fuel [9].

Chintala et al. scrutinized the experimental tests on enhancement in hydrogen energy share (HES) with emissions characteristics by varying compression ratios (19.5, 16.5, and 15.4) and using water injection technique on single cylinder DICI engine under dual fuel mode. The author reported that the HES increased from 18.8 (conventional mode) to 79% (combined strategy of CR 16.5 and water injection). The BTE decreased with reduced CR and water addition compared to the base mode. The emissions of NO_x, CO, HC, and smoke reduced drastically with reduced CR and water addition compared to conventional mode [10]. Hariram et al. investigated the effect of varying compression ratios (18, 17, and 16) on the performance and combustion of single-cylinder DICI engines at loading conditions. The results showed that reducing CR leads to a reduction in BTE and increases the BSFC, and exhaust gas temperature. The peak in-cylinder pressure was reduced, and the ignition delay period increased on reducing the CR [11]. The experimental work on the impact of EGR on the performance, emissions, and combustion characteristics of six-cylinder CRDI CI engine with DME-biodiesel blends at a constant speed of 1262 rpm with varied EGR rates (0% to 20%) were conducted by Sun et al. The percentage of biodiesel in DME fuel was varied from 0% to 15% by weight. It was found that the combustion duration and ignition delay increased with an increase in EGR rates. NO_x emission decreased by EGR with negative effects on CO and HC emissions. The results indicate that 5% of Biodiesel blend in DME fuel exhibit the highest brake thermal efficiency and the shortest ignition delay [12].

Balasubramanian et al. carried out the experimental tests and studied the effect of varying compression ratios (19, 17, and 21) on performance, combustion, and emissions of single-cylinder automotive DICI fuelled with neat biodiesel. The author reported that the peak combustion pressure increased from 69.74 bar at 19 CR to 72.65 bar at 21 CR, and ignition delay period, combustion duration, and peak HRR reduced from 8.9 to 8.5 CA, 58 to 55 CA, and 78.7 to 72.1 J/CA at 21 CR compared to 19 CR. The BTE increased from 25.43 (19 CR) to 27.84% (21 CR). The emissions of NO_x and HC increased significantly from 8.71 to 9.25 g/kWh and 0.0059 to 0.0087 g/kWh at 21 CR compared to 19 CR. The smoke and CO emissions decreased from 20 to 14.5% opacity and 1.57 to 1.02 g/kWh at CR of 21 compared to 19 CR [13]. Chintala et al. performed and examined the influence of varying compression ratios (19.5, 16.5, and 15.4) on HES enhancement, performance, and emission characteristics of single-cylinder DICI engine fuelled with hydrogen under dual fuel mode. The results showed that the maximum HES enhanced from 19-59-63% at 19.5-16.5-15.4 CR. The emissions of HC and CO were reduced to zero levels when CR was reduced from 19.5 to 15.4. The NO_x emission was reduced significantly by 43% and 48% at 16.5 and 15.4 CR compared to 19.5 CR [14]. Wang et al. examined the influence of LPG addition on the combustion and emission characteristics of a diesel engine under dual fuel mode (DME-diesel) at an engine speed of 1700 rpm and two BMEPs 0.24 and 0.48 MPa. LPG in DME was used as an ignition inhibitor to minimize the detonation problem. The oxides of nitrogen decreased marginally, whereas the concentration of particles increased with an increase in LPG mass fraction in the DME-LPG mixture as compared to DME-diesel dual fuel mode [15].

Kumar et al. investigated the effect of varying compression ratios (19, 17.3, and 21.2) on greenhouse gas (GHG) emissions of a DICI single-cylinder engine fuelled with neat biodiesel (B100). The results indicate that at 19 CR (base), CO_2 emission increased, whereas CH_4 and N_2O emissions decreased compared with base diesel operation. The emissions of CO_2, CH_4, and N_2O were reduced with an increase in CR of the engine [16]. Park et al. studied the effect of biogas on DME-fuelled CRDI engine with the compression ratio of 17.8. Biogas (main fuel) was inducted during intake stroke and DME (pilot fuel) was directly injected into the combustion chamber at the end of the compression stroke. The experiments were conducted on the engine at a constant speed of 1200 rpm with varied energy share of biogas from 20% to 80%. The results showed reductions in peak in-cylinder pressure and peak heat release rate with the increase in biogas mixing ratio. NO_x emission decreased with a negative effect on an increase in HC and CO emissions, whereas the smoke emission reached almost zero levels [17]. Zhao et al. examined the effect of DME with cooled EGR on performance, emission, and combustion characteristics of a four-stroke two-cylinder DICI engine. The results showed that the peak in-cylinder pressure and HRR raised under dual fuel mode (DME-diesel). With EGR, the peak in-cylinder pressure and HRR decreased, and the NO_x emission drastically reduced with a lower level of HC and CO emissions, whereas the smoke emission increased beyond a certain percentage of EGR [18].

Based on the above literature review, it is found that limited studies were reported on different compression ratios for improvement in performance and reduction in emissions in diesel engines with different alternative fuels, but no information is available on the enhancement of DME ES and emission reduction with DME fuel using low compression ratio. Keeping this research gap in mind, the present study aims to enhance the DME energy share (DME ES) in a DICI engine with a 5.5 kW rated power output at 3000 rpm under DME-Diesel mode with a reduction of the compression ratio from 19:1 (standard) to 17:1. Reduction in emissions and performance parameters such as brake thermal efficiency (BTE), brake specific energy consumption (BSEC), was analyzed and combustion parameters such as in-cylinder pressure, rate of pressure rise, net heat release, and combustion duration were studied at 2200 rpm and maximum load.

2. Materials and Methods

2.1 DME Fuelled Engine Setup details

A single-cylinder, four-stroke DICI engine coupled with an eddy current dynamometer was used to conduct the experimental tests under DME-diesel mode. The layout of the experimental test rig is shown in Fig. 1, and the valve timing diagram of the engine is shown in Fig. 2. The DME fuelled engine setup was developed by altering a DICI engine to a dual-fueled engine with minor modification by inducting the DME fuel as the main fuel in the intake manifold. The Diesel (pilot fuel) was injected directly into the combustion chamber at the end of the compression stroke. The detailed specification of the test engine is given in Table 1, and the properties of the fuels are summarized in Table 2. The Coriolis principle-based digital mass flow meter was used to measure liquid (diesel fuel) and gaseous (DME fuel) consumption rates during the experimental tests. Bronkhorst- M15 and Endress + Hauser mass flow meters with the uncertainties of ± 0.40% and ± 0.50% were used to record the consumption rate of diesel and DME fuels. The air filter, airflow meter, and surge tank were connected to the intake air supply line of the engine set up. The thermal-based digital airflow meter (Endress + Hauser) with an uncertainty of ± 0.95% was used to measure the air consumption rate. An Electronic Control Unit (ECU) system was used to control DME injection timings, such as the start and end of DME injection. The injection timings of DME fuel, such as the start and end of injection, were controlled using an Electronic Control Unit (ECU) system. The pressure reducer valve was used to reduce the pressure in the DME fuel line from 6 bar (pressure inside the DME cylinder) to 2 bar. The solenoid injector was used to induct DME fuel (in gaseous form) ranging from 0 to 2 bar in the inlet manifold.

The AVL Di-gas analyzer working on the non-dispersive infrared (NDIR) principle was used to measure the engine exhaust emissions such as NO_x, unburned HC, and CO with an uncertainty of ± 1.4%, ± 1.4%, and ± 1.36% respectively. Smoke emission was recorded by using AVL smoke opacity meter with an uncertainty of ± 1.85%, respectively. The combustion pressure transducer with 45 pC/bar sensitivity was attached to the cylinder head to measure the in-cylinder pressure. The crank angle position was determined by an optical crank angle encoder (0.1 °CA resolution). The AVL-Indicom data acquisition system was used to record the averaged combustion pressure signals of 200 consecutive cycles to study the combustion parameters of the engine.

The standard deviation method was used to obtain the uncertainty of measured parameters such as consumption rate of Diesel and DME etc. Similarly, the error analysis of calculated parameters such as DME ES and BTE were determined using Eq. (1),

$$\Delta f = \left[\left(\frac{\partial f}{\partial x_1}\Delta x_1\right)^2 + \left(\frac{\partial f}{\partial x_2}\Delta x_2\right)^2 + \ldots\ldots\ldots\ldots\ldots\ldots + \left(\frac{\partial f}{\partial x_n}\Delta x_n\right)^2\right]^{0.5} \tag{1}$$

2.2 Methodology adopted for compression ratio (CR) reduction

The compression ratio (CR) was reduced by using the appropriate thickness of the copper gasket. The Photographic view of engine components used to reduce CR is shown in Fig. 3. The details of the engine regarding the calculation of compression ratio are given below:

$$CR = \left(\frac{V}{V_c}\right) = \left(\frac{V_s + V_c}{V_c}\right) \tag{2}$$

$$V_s = \frac{\pi}{4} * B^2 * L \tag{3}$$

where CR is compression ratio, V is the total volume, V_c is the clearance volume, V_s is the swept volume, B is Cylinder Bore (mm), and L is stroke length (mm)

$$V_c = (V_R) + (V_G) + (V_B) \tag{4}$$

$$V_R = \frac{\pi}{4} * B^2 * t \tag{5}$$

$$V_G = \frac{\pi}{4} * B^2 * (h) \tag{6}$$

where V_R is copper gasket volume, V_B is bowl volume, V_G is clearance gap volume, t is the copper gasket clearance thickness, and h is the clearance gap height

$$(CR) = 1 + \frac{(V_s)}{(V_c)} \tag{7}$$

$$(CR_1) = 1 + \frac{(V_s)}{(V_{c1})} \tag{8}$$

$$(CR_2) = 1 + \frac{(V_s)}{(V_{c2})} \tag{9}$$

$$(V_{c1}) = (V_{R1}) + V_G + V_B \tag{10}$$

$$(V_{c2}) = (V_{R2}) + (V_{c1}) \tag{11}$$

$$(V_{R1}) = \frac{\pi}{4} * B^2 * (t_1) \tag{12}$$

$$(V_{R2}) = \frac{\pi}{4} * B^2 * (t_2) \tag{13}$$

where CR_1 and CR_2 corresponds to 19:1 and 17:1 compression ratios, V_{c1} and V_{c2} are the clearance volume corresponds to 19:1 and 17:1 CRs, V_{R1} and V_{R2} are the copper gasket volume corresponds to 19:1 and 17:1 CRs, t_1 and t_2 denotes the copper gasket clearance thickness for 19:1 and 17:1 compression ratios.

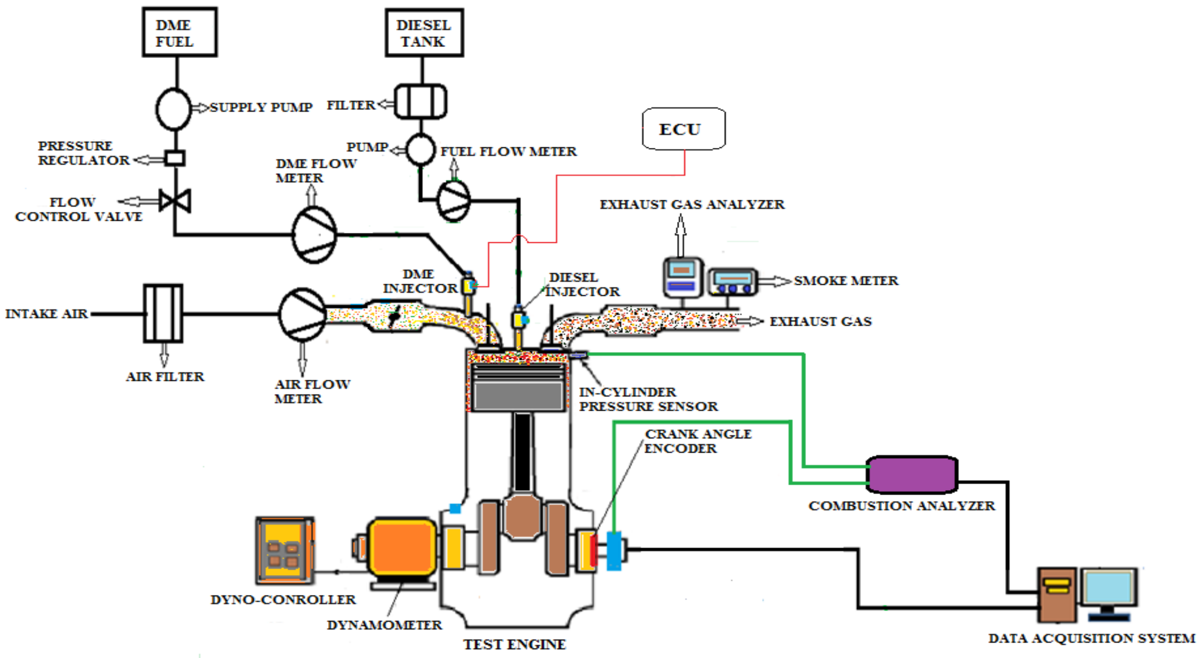

Fig. 1. Schematic diagram of the experimental test setup

Cylinder Bore (B) = 86 mm = 8.6 cm
Stroke Length (L) = 75 mm = 7.5 cm
Swept Volume (V_s) = 435 cm^3
clearance gap height (h) = 0.05 cm
Bowl Volume (V_B) = 18 cm^3
Clearance gap Volume (V_G) = 2.90 cm^3
Copper gasket Volume1 (V_{R1}) = 3.194 cm^3
Copper gasket Volume2 (V_{R2}) = 3.04 cm^3
Clearance volume1 (V_{c1}) = 24 cm^3
Clearance volume (V_{c2}) = 27 cm^3
Compression ratio1 (CR_1) = 19:1
Compression ratio2 (CR_2) = 17:1

2.3 Methodology adopted for experimental tests

All the experiments were performed on the CI engine under DME-diesel mode with two different compression ratios of 19:1 (CR_1) and 17:1 (CR_2) at maximum load (100%) and constant speed of 2200 rpm. DME fuel was inducted by an electronic injector with the help of a programmable ECU through an inlet manifold along with air and diesel fuel injected during the end of the compression stroke. The results of the DME-diesel dual-fuel mode were compared with the neat diesel mode. The DME ES with Diesel can be calculated using Eq. (14).

$$\text{DME ES }(\%) = \frac{\dot{m}_{DME} \times CV_{DME}}{(\dot{m}_{DME} \times CV_{DME} + \dot{m}_{diesel} \times CV_{diesel})} \times 100 \quad (14)$$

Where $\dot{m}_{DME}$ is the flow rate of DME; CV_{DME} is the lower calorific value of DME; $\dot{m}_{Diesel}$ is the flow rate of Diesel and CV_{Diesel} is the lower Calorific Value of Diesel.

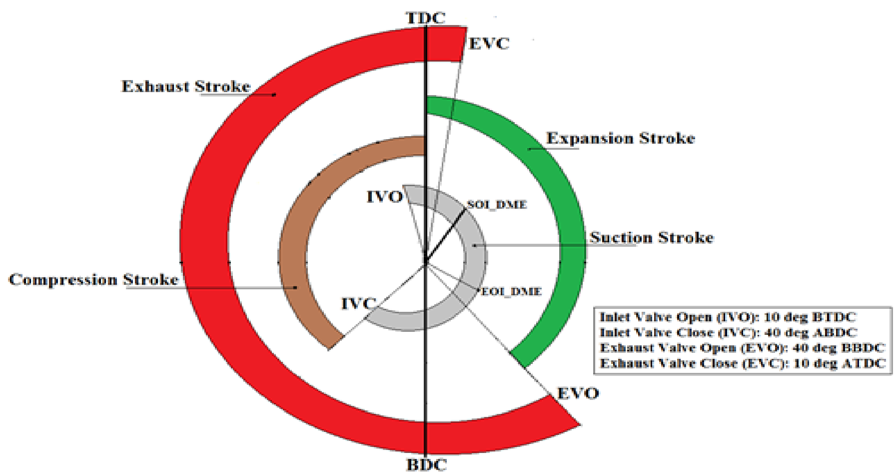

Fig. 2. Valve timing diagram of DME fuelled DICI engine

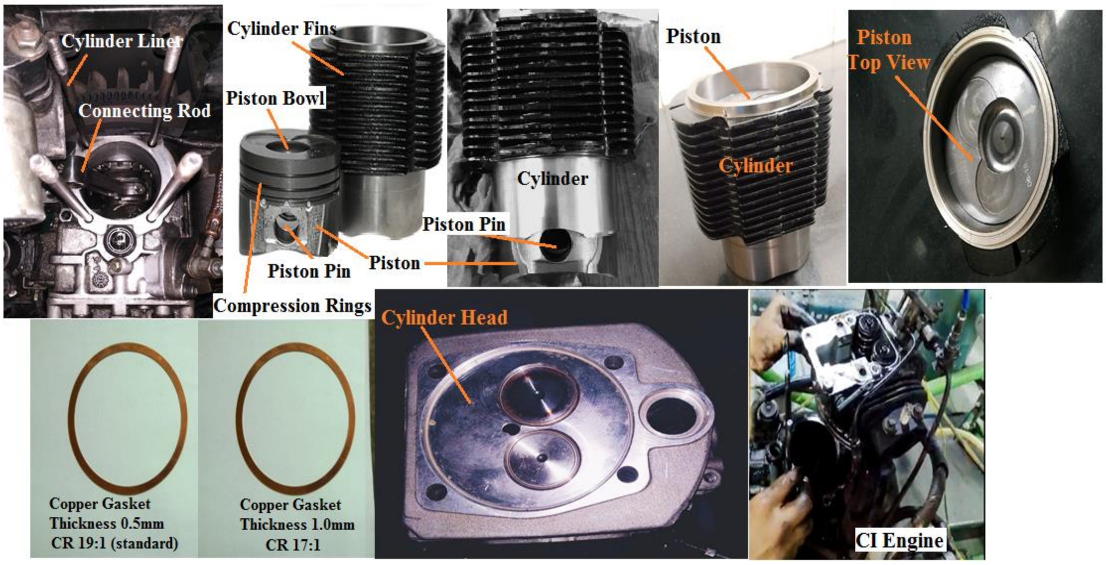

Fig. 3. Photographic view of engine components used for reduction of CR

BTE (η_{th}) of the engine (DME-diesel) can be determined by using Eq. (15).

$$\eta_{th} = \frac{B.P.\ \ (kW)}{(m_{DME}\ .\ \ CV_{DME})+(m_{diesel}\ .\ \ CV_{diesel})}\ X\ 100 \qquad (15)$$

The level of water in the engine's dynamometer and lubricating oil level in the engine was maintained before starting the engine. The engine was run for about a few minutes at no-load conditions. The engine load was slowly increased and set to full load condition, and continued to run for a few minutes to confirm the stability of the engine. Then, the DME fuel was inducted into the intake system of the engine to achieve the maximum DME ES. The performance and emissions parameters were recorded for diesel and DME-diesel mode along with the combustion parameters such as in-cylinder pressure with respect to crank angle. The heat release rate (HRR), rate of pressure rise (ROPR), and cumulative heat release (CHR) were calculated by using the first law of thermodynamics using Eq. (16-18).

$$\frac{dQ_c}{d\theta} - \frac{dQ_{ht}}{d\theta} = \left[\left(\frac{\gamma}{\gamma-1} p\frac{dV}{d\theta} + \frac{1}{\gamma-1} V\frac{dp}{d\theta}\right)\right] \qquad (16)$$

$$\frac{dQ_{ht}}{d\theta} = \left[\ \left(h_i A_i \frac{d(T-T_m)}{d\theta}\right)\right] \qquad (17)$$

The net heat release can be written as,

$$\frac{dQ}{d\theta} = \left[\left(\frac{\gamma}{\gamma-1} p\frac{dV}{d\theta} + \frac{1}{\gamma-1} V\frac{dp}{d\theta}\right)\right] \qquad (18)$$

The temperature of the combustion chamber with respect to crank angle degree is quantified by using the characteristics gas equation (Eq. (19)).

$$\frac{dT}{d\theta} = \frac{\left({}^{dp}/_{d\theta}\right)*\left({}^{dV}/_{d\theta}\right)}{m_{charge}*R} \qquad (19)$$

Where Q_c is heat released by combustion, Q_{ht} is heat transfer through combustion chamber walls, h_i is convective heat transfer coefficient quantified by Woschini's correlation, γ is specific heat ratio, Q is net heat release rate (Joules/degree crank angle), θ is a crank angle (degree), p is combustion pressure (pascal), V is instantaneous cylinder volume (m^3), T is the average temperature (K), A_i is chamber surface area (m^2), T_m is the mean temperature (K), dT is the in-cylinder temperature (K), m_{charge} is mass of air-fuel charge (g/s) and R is characteristic gas constant

Table 1. Engine specifications and features

Description	Values
Engine	Single-cylinder naturally aspirated direct-injection compression ignition engine
Bore	86 mm
Stroke	75 mm
Swept Volume	435 cm^3
Rated power output	5.5 kW at 3000 rpm
Connecting rod length	118 mm
Maximum torque	18 Nm at 2200 rpm
Compression ratio (CR)	19:1

Table 2. Characteristics of fuels used in the experimental study

Properties	Diesel	Dimethyl Ether (DME)
Cetane Number (CN)	50-55	55-60
Density (kg/m^3)	856	1.97
High Heating Value (kJ/kg)	46800	31681
Lower Heating Value (kJ/kg)	43200	28430
Boiling point (°C)	125-400	-24.9
wt% oxygen	0	34.8
wt% hydrogen	14	13
wt% carbon	86	52.2

3. Results and Discussion

3.1 Influence of different Compression Ratio (CR:19 and CR:17) on Brake Thermal Efficiency (BTE)

The effects of compression ratios (CR: 19:1 and CR: 17:1) on brake thermal efficiency (BTE) for both diesel and DME-diesel modes at maximum torque condition at 2200 rpm are shown in Fig. 4. BTE of the DME fuelled dual-fuel engine was determined using Eq. (15). It can be observed from the figure that the BTE of the engine under DME-diesel mode with increasing DME ES at reduced CR (17:1) improved from 14.65% of base diesel to 23.09% with 53% of maximum DME ES and at base CR (19:1), no change in BTE at 26% DME ES, thereafter BTE decreased to 25.87% at 44% of maximum DME ES. The observation of increased BTE at reduced CR (17:1) may be due to reduction in brake specific energy consumption (BSEC), better mixing of the air-fuel mixture, increase in the degree of homogeneity in air-fuel charge, rapid combustion, and favorable physicochemical properties of the DME fuel (the higher latent heat of vaporization, and wider flammability limits). The injection of DME during intake stroke along with air results in a reduction in ignition delay due to the high cetane number of DME than Diesel, which improved and advanced the premixed combustion, thus, results in higher BTE than Diesel at lower CR (17:1) with increasing DME ES. The engine's efficiency with base diesel at 2200 rpm decreased from 28.7% at CR of 19:1 to 14.6% at CR of 17:1, respectively. Similarly, the efficiency with DME-diesel mode decreased from 25.87% with 44% of DME ES at CR of 19:1 to 23.09% with 53% of DME ES at CR of 17:1 respectively. The BTE decreases with a reduction in compression ratio for base diesel fuel due to longer ignition delay, resulting in a decrease in combustion peak pressure and peak temperature and a decrease in combustion rate, which results in lower cumulative heat release. Therefore, pre-mixed and controlled combustion of Diesel is happening in expansion and exhaust stroke, and overall poor combustion results in lower BTE.

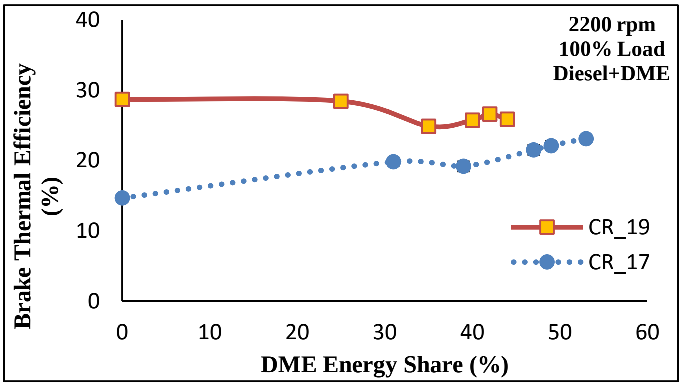

Fig. 4. Variation in brake thermal efficiency (BTE) with respect to DME ES under DME-diesel mode for compression ratios (CR)_19 and 17

Fig. 5 indicates that the total input energy supplied by diesel and DME fuels under dual fuel mode at reduced CR (17:1) decreased sharply with increasing DME ES for the same output energy (brake power) at maximum torque condition 2200 rpm. The output energy (3 kW) was kept constant for both the compression ratios (19:1 and 17:1) operating conditions. For the same output energy (3 kW) at reduced CR (17:1) under dual fuel mode, the total input energy decreased from 20.45 kW with 0% DME ES (base diesel) to 12.97 kW with 53% DME ES respectively. This decreased total energy input resulted in a significant increase in the BTE at reduced CR (17:1).

The effect of compression ratios (CR: 19:1 and CR: 17:1) on brake specific energy consumption (BSEC) for both Diesel and DME-diesel modes at the power of 3 kW condition at 2200 rpm are shown in Fig. 6. The result reveals that the BSEC of the engine under dual-fuel mode with increasing DME ES at reduced CR (17:1) reduced sharply from 24.5 MJ/kWh with base diesel to 15.5 MJ/kWh with 53% of maximum DME ES. However, at base CR (19:1), BSEC increased from 12.5 MJ/kWh with base diesel to 13.9 MJ/kWh with 44% DME ES respectively. The higher BSEC with reduced CR with base diesel due to higher pressure drop across the nozzle at the time of diesel injection could result in over-penetration of base diesel due to lower in-cylinder pressure. The higher penetration resulted in the impingement of liquid fuel on cool surfaces, especially with little or no air swirl. This lowers the mixing rates and increases emissions of unburnt, including partially burnt hydrocarbon. The observation of a reduction in BSEC at reduced CR (17:1) with an increase of DME ES compared with base diesel due to better mixing of the air-fuel mixture, which increases the level of homogeneity leads to better combustion closer to top dead centre and higher conversion fuel energy into mechanical power which resulted in increased BTE of the engine with increasing DME ES.

3.2 Influence of different Compression Ratio (CR:19 and CR:17) on Carbon Monoxide (CO) emission

The effects of compression ratios (CR: 19:1 and CR: 17:1) on Carbon Monoxide (CO) emission for both Diesel and DME-diesel at 2200 rpm are shown in Fig. 7. At base CR (19:1), the CO emission increases with an increase in DME ES under dual fuel mode with base diesel up to a certain percentage of DME ES, and thereafter emission of CO decreases at a higher energy share of DME. The CO emission increases from 2.4 g/kWh with base diesel to 22.4 g/kWh at 25% DME ES and further, decreases to 8.8 g/kWh at 44% of maximum DME ES. The formation of CO due to many sources such as thermal pyrolysis of Diesel and DME into lower carbon chain hydrocarbon and lower chain hydrocarbon subsequent oxidized to CO and CO_2 with air depends on oxygen available, local in-cylinder temperature, etc. CO emission increases with an increase of DME with base CR (19:1) due to trapping of hydrocarbon during compression and combustion, in-cylinder pressure forces some of the mixture in the engine's cylinder into crevices, or narrow volumes, volumes between the piston, rings and cylinder wall.

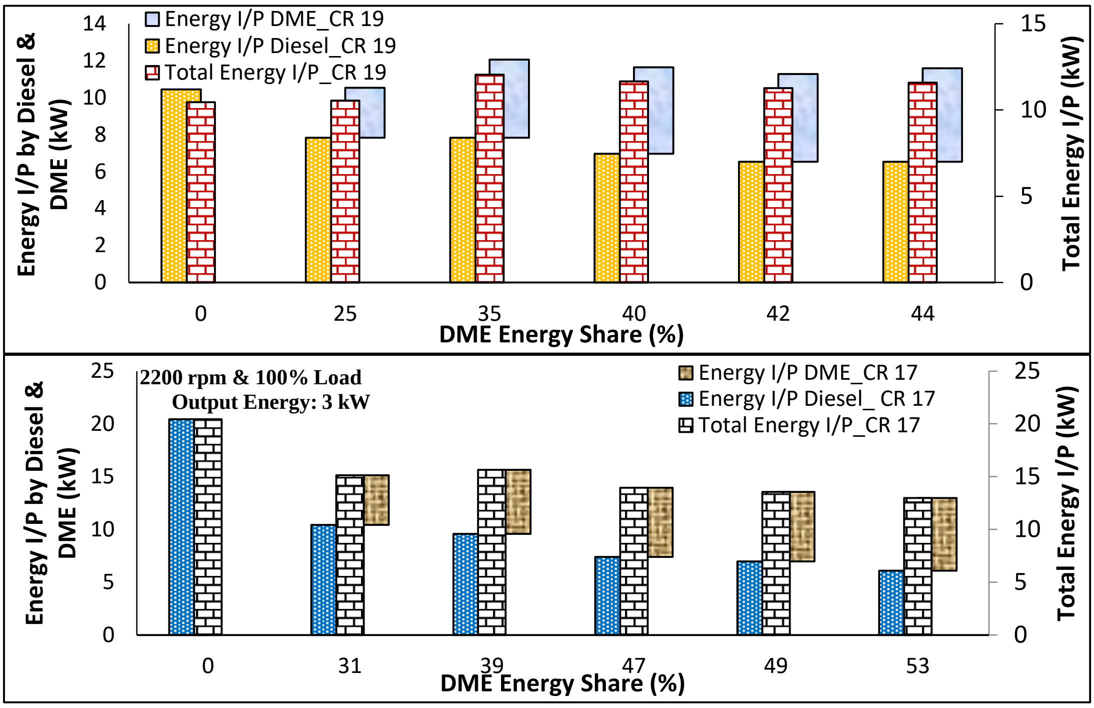

Fig. 5. Influence of input energy supplied by diesel and DME fuels under DME-diesel mode for compression ratios (CR)_19 and 17

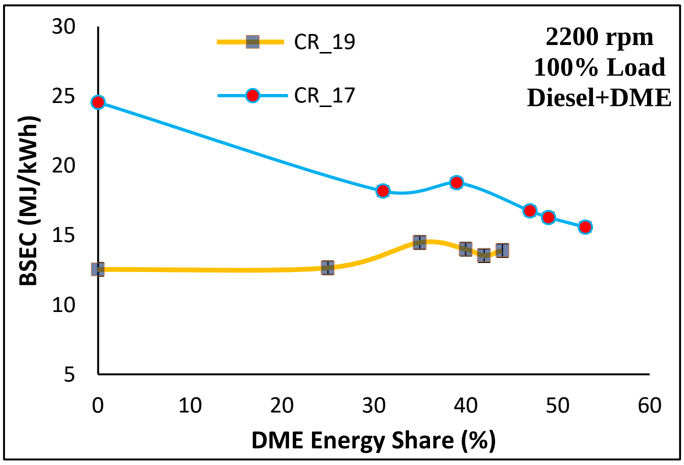

Fig. 6. Variation in brake specific energy consumption (BSEC) with respect to DME ES for different compression ratios (CR) of 19 and 17 under dual fuel mode

Most of this trapped hydrocarbon is unable to combust the entrance of these crevices, which are too narrow for the flame to enter. This gas later leaves crevices that are unable to oxidize during the expansion stroke, resulting in higher CO emission. In addition, the mixture near the cylinder wall is unable to completely oxidized due to the quenching of flame near the wall and results in higher hydrocarbon, and this hydrocarbon is unable to combust due to less oxygen and results in higher CO. The higher formation of CO in cool flame and premixed combustion with DME fuel is due to the rapid formation of CO in the flame zone and oxidized up to the end of the compression stroke. The oxidation of CO becomes slower during expansion and exhaust stroke due to lowering oxygen concentration and in-cylinder temperature. The figure indicates that the CO emission gets reduced with higher DME ES. The emission of CO reduced from 22.4 g/kWh with 25% DME ES to 8.8 g/kWh with 44% DME ES. This could be due to the higher combustion temperature and better oxidation of DME in expansion and exhaust stroke and the higher mixture of DME and air within the flammability range.

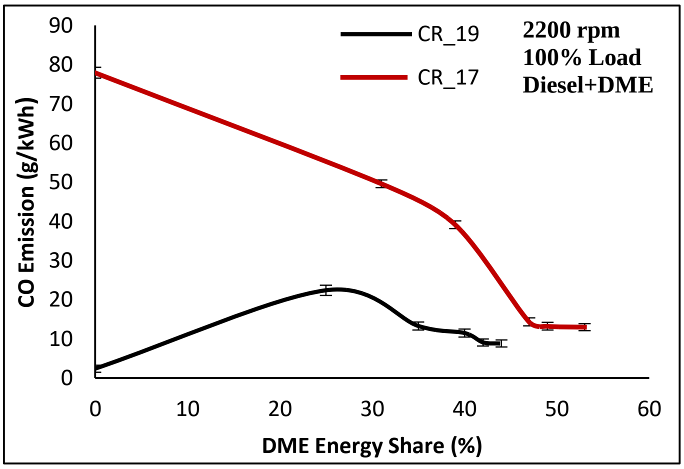

Fig. 7. Variation of Carbon Monoxide (CO) emission with respect to DME ES for different compression ratios (CR) of 19 and 17 under dual fuel mode

The CO emission was reduced with an increase of DME ES with reduced CR (17:1). It can be observed from the figure that the CO emission with increasing DME ES under dual fuel mode at reduced CR (17:1) decreased from 77.9 g/kWh with base diesel to 12.9 g/kWh with 53% of DME ES. At reduced CR (17:1), CO emission was higher compared with base diesel due to higher equivalence ratio and lower in-cylinder temperature, which results in incomplete combustion. Besides, lower chain hydrocarbon is trapped in crevices area and unable to oxidize, resulting in the higher formation of CO. The CO decreased with an increase of DME ES due to higher in-cylinder temperature caused by cool flame and pre-mixed combustion. Also, the equivalence ratio (ø) of DME and air is above 0.22, and the combustion efficiency is higher at equivalence ratio (ø) (above 0.22) and thus decreasing the trend for CO emission [19].

One of the principal reactions in CO formation is in the hydrocarbon combustion mechanism given in Eq. (20),

$$RH \rightarrow R \rightarrow RO_2 \rightarrow RCHO \rightarrow RCO \rightarrow CO \quad (20)$$

Where R stands for hydrocarbon radical, the CO formed in the combustion process via this path is then oxidized to CO_2 at a slower rate, the principal CO oxidation reaction in hydrocarbon-air flame is as under Eq. (21);

$$CO + OH \Rightarrow CO_2 + H \quad (21)$$

3.3 Influence of different Compression Ratio (CR:19 and CR:17) on Hydrocarbon (HC) emission

The effects of compression ratios (CR: 19:1 and CR: 17:1) on Hydrocarbon (HC) emission for both base diesel and DME-diesel modes at 2200 rpm are shown in Fig. 8. At base CR (19:1), HC emission increases slightly with an increase in DME ES under dual fuel mode compared with base diesel. The HC emission increases from 0.02 g/kWh with base diesel to 0.46 g/kWh with 44% of DME ES. The higher HC is due to higher equivalence ratio, fuel pyrolysis, locally lean and over-rich mixture, quenching of flame on the wall and trapped mixture in crevices area. The trapped DME fuel inside the crevice volume with heterogeneous air-fuel (DME + Diesel) mixture which leads to improper combustion in multiple zones like quenching area, cylinder walls, compression clearance, and piston ring, and ultra-lean mixture (beyond flammability limit of DME fuel, i.e. 3.4 to 27% volume) results in the formation of HC emission with the injection of DME during intake stroke at base CR (19:1).

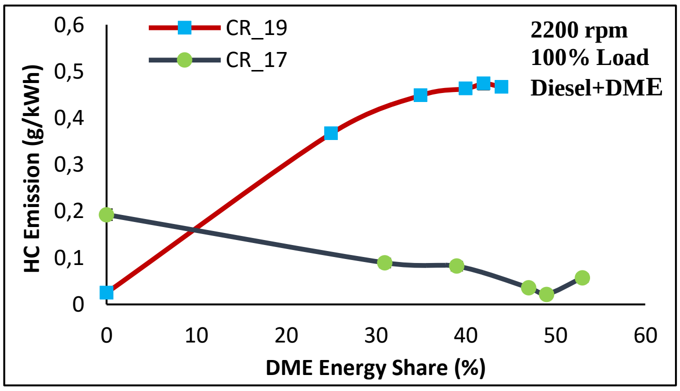

Fig. 8. Variation of Hydrocarbon (HC) emission with respect to DME ES for compression ratio (CR)_19 and 17

The HC emission was reduced with an increase in DME ES at reduced CR (17:1). It can be observed from the figure that the HC emission decreases with the increase of DME ES under dual fuel mode from 0.19 g/kWh of base diesel to 0.05 g/kWh with 53% of DME ES. The high HC with base diesel with reduced CR is due to the mechanical diesel injector. The decreasing trend of HC emission with reduced CR (17:1) could be attributed due to better combustion of the mixture as DME injected during the intake stroke, about 60 to 99% of the energy of DME get burnt from 31 to 53% of DME ES (higher equivalence ratio 0.22 and above of DME and air) which resulted in a lower trend of HC. A similar trend was reported by Sato et al. [19]. Sato et al. reported that HC emission is almost zero when the equivalence ratio for DME fuel with air is more than 0.25.

3.4 Influence of different Compression Ratio (CR:19 and CR:17) on Oxides of Nitrogen (NOx) emission

The effects of compression ratios (CR: 19:1 and CR: 17:1) on oxides of nitrogen (NO_x) for both Diesel alone and DME-diesel under dual fuel mode at 2200 rpm are shown in Fig. 9. It can be observed from the figure that the NO_x formation under dual-fuel mode with increasing DME ES with reduced CR (17:1) increased marginally from 0.8 g/kWh with base diesel to 1.9 g/kWh with 53% of DME ES due to higher in-cylinder temperature. However, at base CR (19:1), NO_x emission marginally increased from 4.0 g/kWh with base diesel to 4.1 g/kWh with 35% DME ES and then increased to 4.7 g/kWh with 44% of DME ES. The marginally increasing trend of NO_x formation with the increase in DME ES is due to the dominance of dilution effect by the gases produced during cool-flame and premixed combustion of DME, which acts as an internal exhaust gas recirculation (EGR), which suppresses the formation of NO_x. However, at higher DME ES, the net heat release rate (HRR) is higher during cool flame and pre-mixed combustion as compared with base diesel and resulting in higher combustion temperature than temperature with diesel fuel alone, indicating that the NO_x formation is higher due to dominant of temperature effect than dilution effect. The oxides of nitrogen (NO_x) emission compared with base CR and reduced CR with base diesel at 2200 rpm decreased from 4.0 g/kWh at base CR (19:1) to 0.8 g/kWh at reduced CR (17:1) due to lower in-cylinder temperature with reduced CR. Similarly, the NO_x formation with DME-diesel mode decreased from 4.8 g/kWh with 44% of DME ES at base CR (19:1) to 1.9 g/kWh with 53% of DME ES at reduced CR (17:1) respectively primarily due to both dilution and thermal effects. The kinetics of the formation of NO_x (NO and NO_x) during the combustion process through the Zeldovich mechanism is given in Eqs. (22 - 24) [20];

Reaction	Temperature Range-K	
$O + N_2 \rightarrow NO + N$	2000-5000	(22)
$N + O_2 \rightarrow NO + O$	300-3000	(23)
$N + OH \rightarrow NO + H$	300-2500	(24)

The NO formation rate can be determined by Eq. (25) [20],

$$[NO] = k \left(e^{-K/T}\right) [N_2][O_2]^{1/2t} \quad (25)$$

Where [NO] is the concentration of NO (moles/cm^3) at time t (sec); k and K are reaction constants; T is the temperature (K); $[O_2]$ and $[N_2]$ are equilibrium concentration at temperature T (K) of oxygen and nitrogen (moles/cm^3); and t, is time (sec).

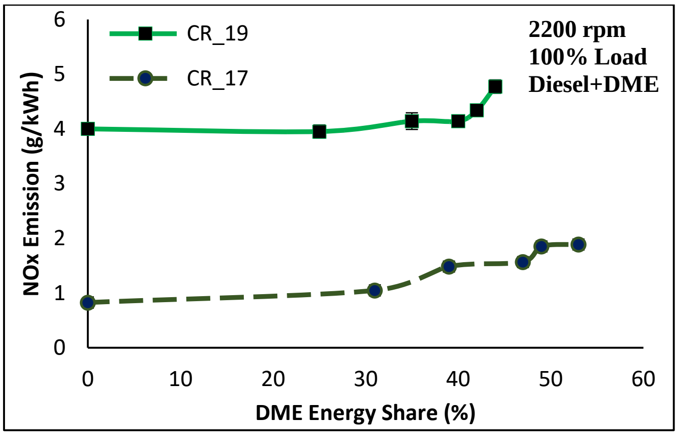

Fig. 9. Variation of Oxides of Nitrogen (NOx) emission with respect to DME ES for different compression ratios (CR) of 19 and 17 under dual fuel mode

3.5 Influence of different Compression Ratio (CR:19 and CR:17) on smoke emission

The effects of compression ratios (CR: 19:1 and CR: 17:1) on smoke emission with diesel alone and DME-diesel dual fuel mode at 2200 rpm are shown in Fig. 10. Smoke is generally defined as liquid and solid particles suspended in the exhaust of Diesel. Smoke emission in CI engines is mainly caused by the heterogeneous air-fuel (diesel) charge during the combustion process as fuel molecules undergo oxidation/pyrolysis, resulting in many unsaturated hydrocarbons (acetone its higher analogs $C_{2n}H_2$, and polycyclic aromatic hydrocarbon). These products are the main precursor for soot formation and smoke emission. The molecular structure of DME is free from the C-C bond and low/no formation of soot precursor during combustion. Smoke decreases with addition/increase of DME due to higher mixing in the level of homogeneity in the charge and replacement of part of diesel fuel with DME. Fig. 10, indicates that the smoke emission under dual-fuel mode with the increase of DME ES with reduced CR (17:1) decreases drastically from 67% opacity with base diesel to 1.1% opacity with 53% of maximum DME ES. Similarly, at base CR (19:1), a similar trend was found as smoke emission reduced substantially from 10.3% opacity with base diesel to 0.9% opacity with 44% DME ES respectively. The reaction processes for DME fuel is commenced with the breaking of H atom from C-H bond of molecular structure and thereafter addition of O_2 with C bond during cool flame and pre-mixed combustion under dual fuel mode of engine operation which resulted in low/no formation of molecular precursors of soot, polycyclic aromatic hydrocarbons (PAH) and acetylene compared with base diesel. Besides, there is no C-C bond in the DME molecular structure with higher diffusivity, and 34.8% oxygen content results in the formation of a better homogeneous mixture between fuel and air, which can also be responsible for low/no smoke emission.

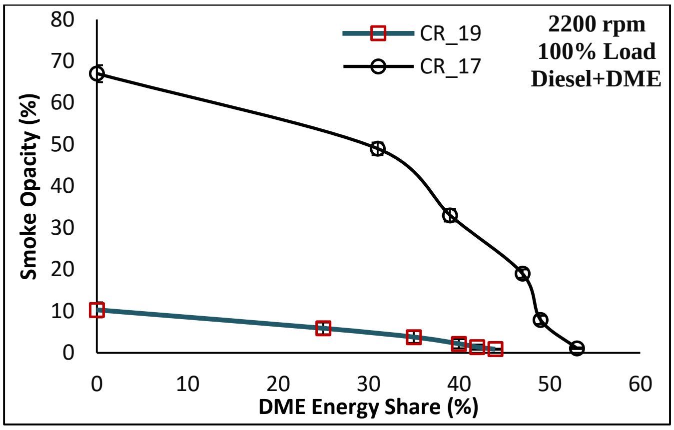

Fig. 10. Variation of smoke emission with respect to DME ES for compression ratios (CR)_19 and 17 under DME-diesel mode

3.6 Variation in DME Energy Share (DME ES) for different Compression Ratios (CR:19 and CR:17)

Fig. 11 demonstrates the variation of the DME Energy Share (DME ES) with different compression ratios (CR:19 and CR:17) for DME-diesel mode at 2200 rpm. The DME ES increased from 44% with base CR (19:1) to 53% with reduced CR (17:1). The enhancement of DME ES at reduced CR (17:1) is primarily due to the advantage of lower ignition temperature, gaseous DME, and higher cetane number of DME fuel, resulting in shorter ignition delay and advancement of premixed combustion, which is near to top dead centre. At base CR (19:1), there is a limitation of the use of DME as the energy released during premixed combustion is not converted into useful mechanical energy, as pre-mixed combustion is advancing much before TDC instead of near TDC.

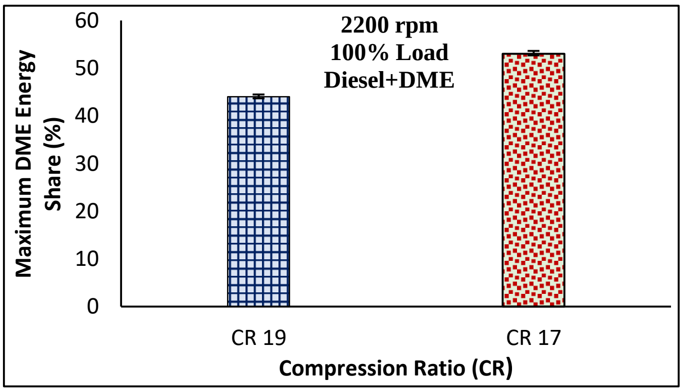

Fig. 11. Variation in maximum DME energy share (DME ES) for compression ratios (CR)_19 and 17

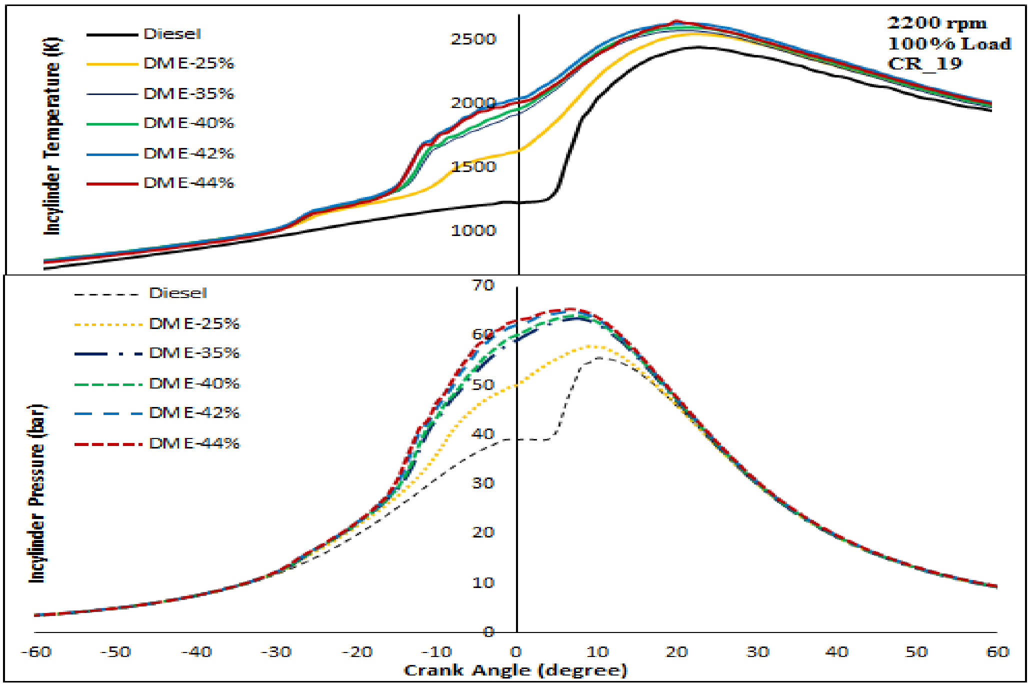

Fig. 12. Variation of In-cylinder pressure and temperature with respect to crank angle (degree) for compression ratio (CR) 19

3.7 Influence of different Compression Ratio (CR:19 and CR:17) on in-cylinder pressure and temperature under dual fuel mode

The influence of compression ratios (CR: 19:1) on combustion pressure and temperature for both Diesel and DME-diesel modes at maximum torque conditions at 2200 rpm are shown in Fig. 12. The combustion pressure and temperature of the engine under dual-fuel mode with increasing DME ES at reduced CR (17:1) increased significantly from 41.67 bar and 1843 K with base diesel to 55.37 bar and 2520 K with 53% of maximum DME ES. Similarly, at base CR (19:1), combustion pressure and temperature increased from 55.63 bar and 2445.4 K with base diesel to 65.41 bar and 2652 K with 44% DME ES respectively. The peak in-cylinder pressure and temperature increased and advanced with the increase in DME ES due to the ignition time of DME fuel which is prior to Diesel which initiates the process similar to HCCI combustion, homogeneous mixture of DME and air as more time for mixing from intake stroke to end of compression stroke before injection of Diesel and thus, results in better combustion. Further, the result also reveals that the combustion pressure of the engine decreased with reduced CR (17:1) compared to base CR (19:1). The reduction in combustion pressure may be attributed due to the increase in the clearance volume after the reduction in CR from 19:1 to 17:1. Similarly, the combustion temperature decreased considerably after reducing CR from 19:1 to 17:1, as the in-cylinder temperature is directly proportional to combustion pressure. The peak combustion pressure and temperature with base diesel at maximum torque condition and 2200 rpm decreased from 55.63 bar and 2445.4 K at base CR (19:1) to 41.67 bar and 1843 K at reduced CR (17:1) due to longer ignition delay, which results in a decrease in combustion rate and hence lower cumulative heat release. Fig. 13 shows that the combustion peak pressure with DME-diesel mode decreased from 65.41 bar with 44% of maximum DME ES at base CR (19:1) to 55.37 bar with 53% of maximum DME ES at reduced CR (17:1) respectively. Further, the peak combustion pressure increased and advanced with the increase in DME ES at reduced CR (17:1).

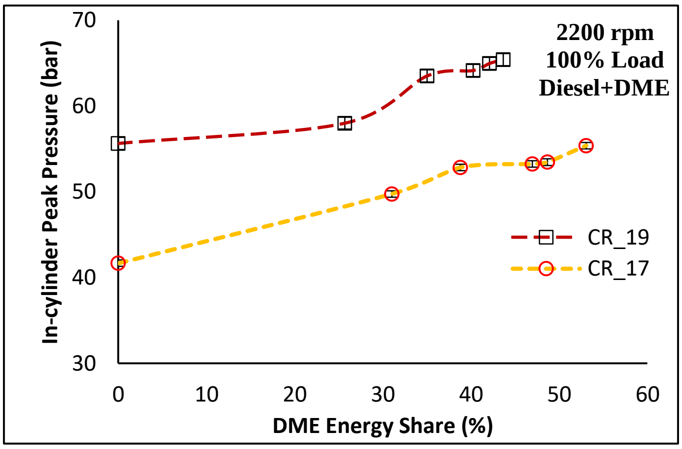

Fig. 13. Variation in combustion peak pressure with respect to DME ES for compression ratio (CR) of 19 and 17

For both compression ratios (19:1 and 17:1), the comparison between peak combustion pressure and temperature with Diesel and DME-diesel modes are summarized in Table 3. It can be observed from the table that the combustion peak pressure and temperature decreased subsequently with reduced CR (17:1) as compared to base CR (19:1) for both modes. At 44% of maximum DME ES, the combustion peak pressure and temperature with the compression ratios (19:1 and 17:1) are enlisted in the table. From the above discussion, it is clearly understood that the combustion pressure and temperatures at Diesel and DME-

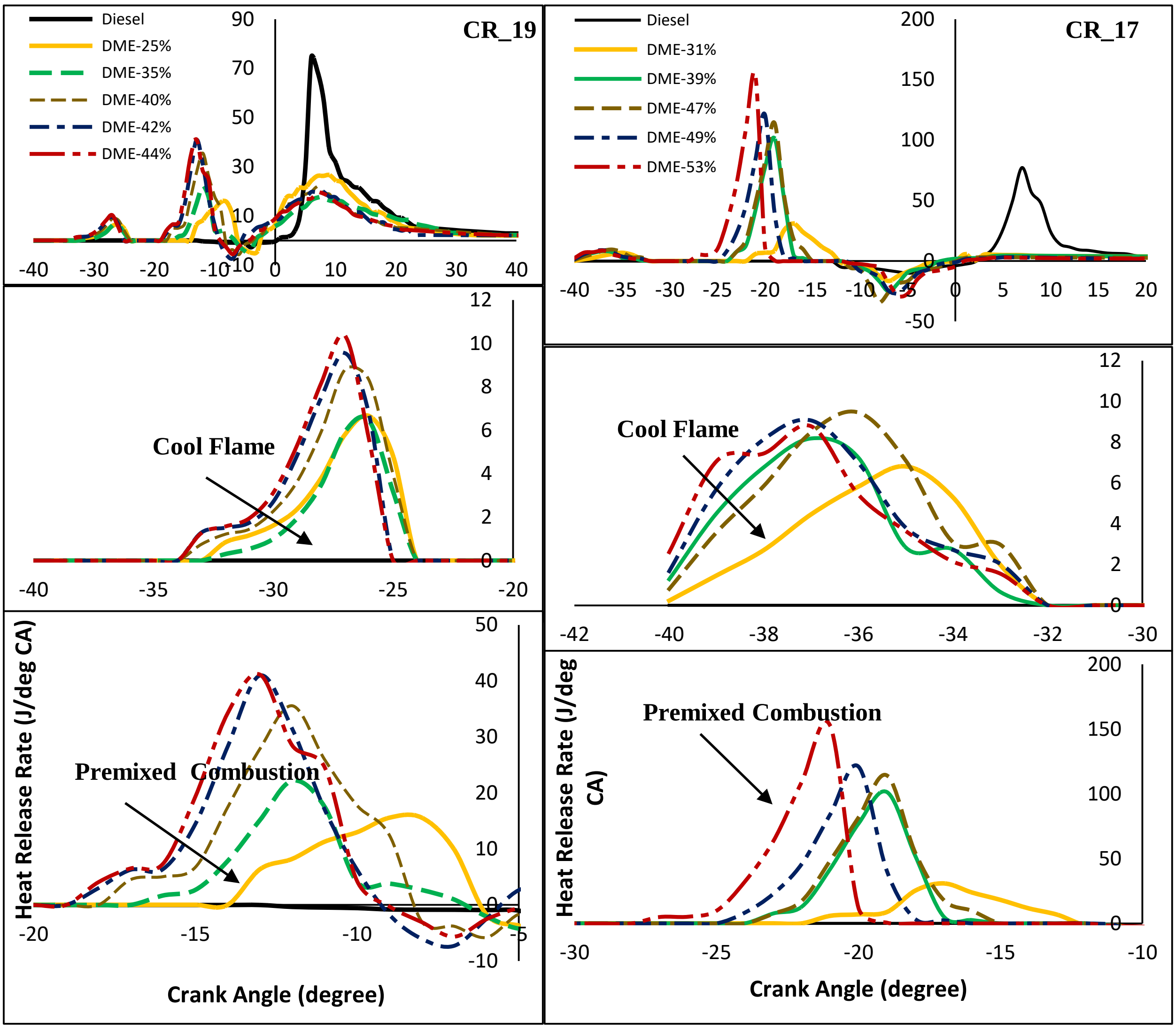

Fig. 14. Variation of the rate of heat release (HRR) with respect to crank angle (degree) for compression ratios (CR) 19 and 17

Diesel modes decreased with a reduction in the compression ratio from base CR (19:1) to reduced CR (17:1).

Table 3. Combustion peak pressure and temperature at compression ratios (19:1 and 17:1)

Compression Ratio (CR)	Base Diesel mode		DME-diesel mode at 44% of maximum DME ES	
	Peak combustion pressure (bar)	Peak combustion temperature (K)	Peak combustion pressure (bar)	Peak combustion temperature (K)
19:1	55.6	2445	65.4	2652
17:1	41.6	1843	53.0	2408

The decrease in in-cylinder peak pressure with reduced CR (17:1) leads to a substantial reduction in knocking occurrence tendency, which endorses further DME fuel substitution under DME-diesel mode in the CI engine. The knock-limited DME ES increased from 44% to 53%, reducing compression ratio (CR) from 19:1 to 17:1 in the CI engine.

3.8 Influence of different Compression Ratio (CR:19 and CR:17) on Heat Release Rate (HRR) under dual fuel mode at 2200 rpm

The influence of compression ratios (CR: 19:1 and CR: 17:1) on heat release rate (HRR) for Diesel alone and DME-diesel modes at 2200 rpm is shown in Fig. 14. The quantitative information on the progress of combustion is obtained from in-cylinder pressure versus crank angle using the first law of thermodynamics. It can be observed from the figures that the DME fuel, due to its higher cetane number, get ignition first, and then further it initiates the ignition of base diesel fuel. As DME fuel gets ignition at lower combustion temperature than diesel leads in advanced combustion mode due to its lower self-ignition temperature (508 K) than Diesel (588 K), leads to increase the in-cylinder pressure from 55.63 bar and 41.67 bar with base diesel to 65.4 bar and 55.37 bar with 44% and 53% of DME ES under DME-diesel mode with CR 19:1 and 17:1 respectively and in-cylinder temperature from 2445 K and 1843 K with base diesel to 2652 K and 2520 K with 44% and 53% of DME ES with CR 19:1 and 17:1 respectively. The heat release rate (HRR) after injection of DME fuel (gas) through inlet line in dual fuel mode is observed three-stage combustion, i.e. cool flame, premixed combustion and diffusion combustion of DME with base diesel. The cool flame occurs at low temperature at 800-900 K at the almost constant crank angle between 40 to 35 degree before top dead center (BTDC) and premixed combustion at a higher temperature at the temperature of 1000 K or more at a crank angle of 10 degrees before top dead center (BTDC) or earlier depending on the increase of DME. The premixed combustion gets advanced with the increasing of DME ES.

The cool flame or low-temperature reaction (LTR) occurs due to breaking the C-H bond of DME, and thereafter, O_2 is attached with C atom in place of H atom, forming $C_2O_3H_5$ and subsequent isomerization reaction results in the addition of the second O_2. In equilibrium conditions, the chain propagation reaction of $C_2O_3H_5$ without the addition of second O_2 results in apparently stopping the oxidation. The reaction rate is forward movement at low temperature leads to chain reactions and results in the formation of cool flame, which increases the reactants temperature during the compression stroke, and this cool flame could work as an ignition source to ignite the charge (DME-air mixture) and hence no external ignition source required as in case of base diesel. However, the reaction rate reversed at high temperature due to these reactions are temperature dependent and the formation of species such as H, H_2, HO_2, and H_2O_2, diminishes or possibly get to be distinct. This could limit the prolongation of a cool flame. The cumulative heat release during cool flame increases with the increase of DME. The cumulative heat released of DME energy during cool flame increases with the increase of DME but in the range of 10-12% of DME energy with reduced CR, whereas DME energy released in the cool flame is in the range of 22-15% with base CR (19:1).

The premixed combustion or high-temperature reaction above 1300 K, occurs due to the breaking of the C-H bond results in the removal of the H atom, and thereafter breaking of the C-O bond continues. After breaking the C-O bond, the broken DME fuel subsequently oxidized before diesel fuel injection. With increasing DME quantity, the HRR increases, which results in higher cumulative HRR (49 to 88%), and commencement pre-mixed combustion gets advanced. Theoretically, with assumptions, i.e. a premixed mixture is homogeneous, the total mass remains the same before and after reactions, no heat transfer (adiabatic change), Ideal gas, the timing or crank angle at which the premixed combustion is dominant that should retard, but as per experimental data's, it is advancing apparently due to presence of residual gases of previous cycles.

The diffusion combustion of DME with base diesel or mixing control combustion is dominant for diesel fuel, and DME energy released in this phase decreased (40 to 2%) with the increase of DME. Under dual fuel mode (DME-diesel), the cool flame and pre-mixed combustion become dominant due to gaseous DME fuel injection, higher time for mixing DME with air (intake stroke to end of compression stroke/injection of base diesel fuel), and better ignition quality of DME.

It is observed from Fig. 14 that the cool flame is in the range from 40°CA to 33°CA before top dead center (BTDC) and release of DME energy is about 10-12% of total DME energy, premixed combustion in range of 21°CA to 14°CA before top dead center (BTDC) at 31% of DME ES with burning of 49% of DME energy and 24°CA to 14°CA BTDC with burning of 89% of DME energy at 53% DME ES with reduced CR (17:1). The premixed combustion with DME ES is dominant and advancing in HRR during combustion compared with base diesel due to higher cetane number of DME than Diesel and better combustion due to more time for mixing of DME with air during intake and compression stroke before injection of Diesel about the end of the compression stroke. The strategy to control the un-controlled auto-ignition of the mixture by reduced CR resulted in the dominance of pre-mixed combustion of mixture close to TDC and advancing of peak in-cylinder pressure at 3°CA BTDC at 53% DME ES at reduced CR (17:1) than peak in-cylinder pressure at 7°CA BTDC at 44% DME ES at base CR (19:1). This resulted in higher utilization of chemical energy into mechanical energy and the enhancement of DME ES from 44% to 53% with reduced CR. Therefore, the conclusion that emerged from the above discussion is that the overall combustion process of DME-diesel mode in dual-fuelled CI engine presents a three-stage (multi-staged) combustion, i.e. cool flame, premixed combustion, and diffusion combustion DME with Diesel as compared to the combustion of base diesel-fuelled of CI engine. A similar kind of results was reported by Zhang et al. [21], Kim et al. [22], and Wang et al. [23] on three-stage heat-releases characteristics in DME-diesel mode in CI engine under dual fuel mode.

3.9 Influence of different Compression Ratio (CR:19 and CR:17) on the maximum rate of pressure rise under dual fuel mode at 2200 rpm

The effects of compression ratios (CR: 19:1 and CR: 17:1) on the rate of pressure rise (ROPR) for both diesel and DME-diesel modes at maximum torque condition at 2200 rpm are shown in Fig. 15. The maximum rate of pressure decreased with an increase of DME ES to some extent of DME at both CR and thereafter, increasing, but the highest value is within the limit of generally design value of 10 bar/°CA. It is observed that the maximum rate of pressure rise (dp/dɵ) at reduced CR 17:1 decreased from 4.62 bar/°CA with Diesel to 3.41 bar/°CA with 31% of DME ES and then further, increased to 9.45 bar/°CA with 53% of DME ES. A similar trend is noticed at base CR 19:1 and maximum dp/dɵ decreased from 5.01 bar/°CA with Diesel to 2.42 bar/°CA with 25% of DME ES and then further, increased to 4.1 bar/°CA with 42% of DME ES respectively. However, the engine design parameter for dp/dɵ is 10 bar/°CA. The increasing trend at above 30-35% DME ES is due to higher DME get burnt at premixed combustion and thus, rise in maximum pressure rate per degree of crank angle.

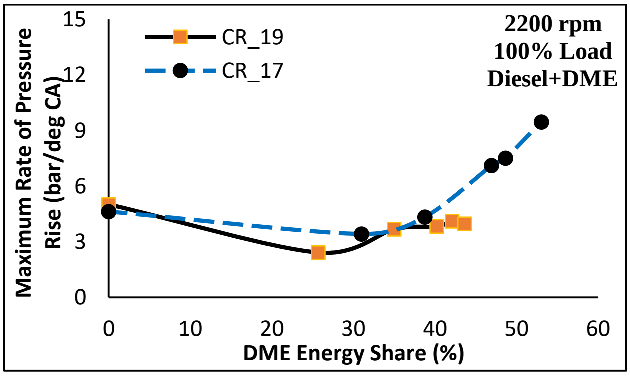

Fig. 15. Variation in the maximum rate of pressure rise with respect to DME ES for compression ratio (CR) of 19 and 17

3.10 Influence of different Compression Ratio (CR:19 and CR:17) on the start of combustion (SoC) under dual fuel mode at 2200 rpm

The effects of compression ratios (CR: 19:1 and CR: 17:1) on the start of combustion (SoC) for both diesel and DME-diesel modes at maximum torque condition at 2200 rpm are shown in Fig. 16. It is observed that SoC gets advanced after injection of DME than Diesel due to better ignition quality of DME than Diesel in terms of cetane number at both CRs. The SoC commenced after ignition delay and represents the time interval between fuel injection and combustion. The injection of DME in a gaseous state also contributes to the reduction in ignition delay. The ignition delay is higher at reduced CR 17:1 compared with base CR 19:1 due to lower in-cylinder pressure at the time of injection of liquid Diesel.

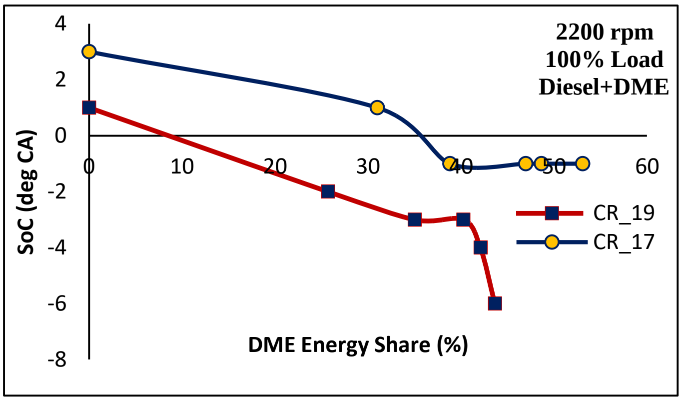

Fig. 16. Variation in the start of combustion (SoC) with respect to DME ES for compression ratio (CR) of 19 and 17

3.11 Influence of different Compression Ratio (CR:19 and CR:17) on combustion duration (CD) under dual fuel mode at 2200 rpm

The influence of compression ratios (CR: 19:1 and CR: 17:1) on combustion duration (CD) for both diesel and DME-diesel modes at maximum torque conditions at 2200 rpm is shown in Fig. 17. There are no significant changes observed in combustion duration variation at reduced CR (17:1). The variation in combustion duration (CD) from 59°CA with Diesel to 55°CA with 53% DME ES under DME-diesel mode at reduced CR (17:1) is observed. A similar pattern is noticed at base CR (19:1), and combustion duration is marginally varied from 46°CA with Diesel to 40°CA with 44% DME ES respectively. The combustion duration at higher DME ES is higher at reduced CR (17:1) than base CR (19:1) due to the lower rate of heat release in the expansion stroke. These are due to a small fraction of fuel may not have burnt, oxygen concentration reduces, the kinetics of the final burnout processes are slower as the temperature of in-cylinder gases decreases and the fraction of fuels energy is present soot, and fuel-rich combustion products can still be released.

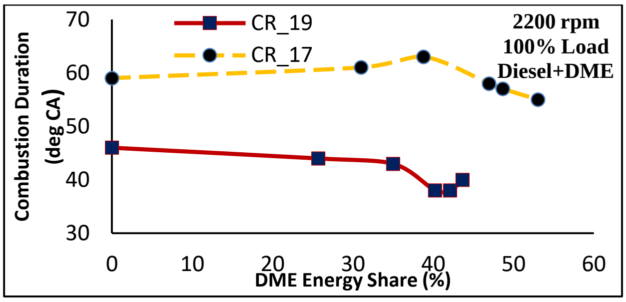

Fig. 17. Variation of combustion duration (CD) with respect to DME ES for compression ratios (CR) of 19 and 17

The summary of performance and emission parameters of the CI engine under DME-diesel mode with compression ratios (19:1 and 17:1) is given in Table 4. It can be observed from the table that the DME ES enhanced significantly from 44% at base CR to 53% at reduced CR after reducing CR from base 19:1 to reduced 17:1. The results show that BTE is 25.87% with base CR (at 44% DME ES), nearly comparable to 23.09% with reduced CR (at 53% DME ES). The smoke, CO, and HC emissions were drastically reduced with 53% of maximum DME ES at reduced CR (17:1) under DME-diesel mode compared to base CR (19:1), respectively. However, the NO_x emission increased marginally with both CR (19:1 and 17:1) but was found as lower in the case of reduced CR (17:1). All the evidence that emerged from the study leads to a conclusion that the reduced CR (17:1) could be an optimum CR for enhancement of DME ES, improved BTE, and reduced emissions such as smoke, CO, and HC.

Table 4. Outline on performance and emission parameters of CI engine under DME-diesel mode with compression ratios (19:1 and 17:1) under same operating condition (2200 rpm, and maximum torque)

Parameters under DME-diesel mode	Remarks on CR reduction from base 19:1 to modified 17:1	
	Base CR (19:1)	Reduced CR (17:1)
DME Energy Share (DME ES)	Achieved 44% of maximum DME ES	Enhanced to 53% of maximum DME ES
Thermal Efficiency(BTE)	Decreased from 28.7% with Diesel to 25.87% with 44% of maximum DME ES	Increased significantly from 14.6% with Diesel to 23.09% with 53% of maximum DME ES
Smoke emission	Reduced substantially from 10.3% opacity with Diesel to 0.9% opacity at 44% of DME ES	Decreased drastically from 67% opacity with Diesel to 1.1% opacity at 53% of DME ES
CO emission	Increased from 2.4 g/kWh with Diesel to 8.8 g/kWh at 44% of DME ES	Decreased from 77.9 g/kWh with Diesel to 12.9 g/kWh at 53% of DME ES
HC emission	Increased slightly from 0.02 g/kWh with Diesel to 0.46 g/kWh at 44% of DME ES	Decreased significantly from 0.19 g/kWh with Diesel to 0.05 g/kWh at 53% of DME ES
NO_x emission	Increased from 4.0 g/kWh with Diesel to 4.7 g/kWh at 44% of DME ES	Increased marginally from 0.8 g/kWh with Diesel to 1.9 g/kWh at 53% of DME ES

4. Conclusions

The influence of different compression ratios base (19:1) and reduced (17:1)) on the enhancement of DME ES and emissions reduction in CI engine at maximum torque and at constant speed of 2200 rpm condition under DME-diesel mode was studied. The following conclusions are drawn from the results obtained from this study:

i. The maximum DME ES with base CR (19:1) in conventional CI engine under dual fuel mode (DME-diesel) was found as 44% due to uncontrolled auto-ignition.

ii. The maximum DME ES enhanced significantly from 44% with base CR (19:1) to 53% with reduced CR (17:1) under dual fuel mode (DME-diesel) in CI engine due to higher premixed combustion close to TDC.

iii. The BTE of the engine under dual fuel (DME-diesel) mode with the increase of DME ES with reduced CR (17:1) improved to 23.09% with 53% of maximum DME ES and comparable to 25.87% with base CR (at 44% DME ES).

iv. Smoke emission reduced drastically and achieved nearly zero with both CRs (19:1 and 17:1) under DME-diesel mode in CI engine at maximum DME energy shares.

v. CO and HC emissions were substantially reduced with 53% of maximum DME ES with reduced CR (17:1) under DME-diesel mode compared to base CR (19:1) due to better combustion in the premixed phase.

vi. NO_x emission increased marginally with both CRs (19:1 and 17:1) but was found much lower with reduced CR (17:1) primarily due to the dilution effect of gases from cool flame and premixed combustion than thermal effect.

vii. It is observed from this study that the reduced CR of 17:1 could be a viable option for utilization of higher ES of DME along with the benefit of reduced emissions, particularly smoke and NO_x.

viii. The CI engine could be used in flexible fuel mode (DME-Diesel) without major modification of the engine as part replacement of diesel fuel (cetane number-51) with DME fuel (cetane number-58) on an energy share basis as DME has a higher cetane number.

Nomenclature

CI	Compression-Ignition	*HRR*	Heat Release Rate (J/°CA)
DICI	Direct Injection Compression Ignition	*ROPR*	Rate of pressure rise (bar/°CA)
CRDI	Common Rail Direct Injection	*CD*	Combustion Duration(°CA)
CAI	Controlled Auto-Ignition	*CHR*	Cumulative Heat Release (J)
HCCI	Homogeneous Charge Compres sion Ignition	*BTDC*	Before Top Dead Centre
CR	Compression Ratio	*TDC*	Top Dead Centre
DME	Dimethyl Ether	*SoC*	Start of Combustion
DME ES	Dimethyl Ether Energy Share	*PM*	Particulate Matter
SCR	Selective Catalyst Reduction	NO_x	Oxides of Nitrogen
DPF	Diesel Particulate Filter	*CO*	Carbon Monoxide
BTE	Brake Thermal Efficiency (%)	*HC*	Hydrocarbon
BSFC	Brake Specific Fuel Consum-ption (g/kWh)	*BSEC*	Brake Specific Energy Consum-ption (MJ/kWh)
CO_2	Carbon dioxide	CH_4	Methane
ECU	Electronic Control Unit	N_2O	Nitrous Oxide
NDIR	Non-dispersive infrared	*PAH*	Polyaromatic hydrocarbon
CV	Calorific Value	*BDC*	Bottom Dead Centre
BP	Brake Power (kW)	*Q*	Net heat release rate (J/°CA)
EGR	Exhaust Gas Recirculation	θ	Crank angle (degree)
p	Combustion pressure (bar)	*T*	Average temperature (K)
T_m	Mean temperature (K)	A_i	Chamber surface area (m^2)
V	Instantaneous cylinder volume (m^3)	*R*	Characteristic gas constant (J/mol-K)
m_{charge}	Mass of air-fuel charge (g/s)	V_c	Clearance volume (cm^3)
V_s	Swept voulme (cm^3)	*B*	Cylinder bore (mm)
L	Stroke length (mm)	V_R	Copper gasket volume (cm^3)
V_B	Bowl voulme (cm^3)	V_G	Clearance gap volume (cm^3)
h	Clearance gap height (cm)	*t*	Copper gasket clearance thickness (cm)

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

Shreemohan Kumar Sinha: Conducted experiment, Manuscript preparation, Data interpretation,
Anilkumar Shere: Conducted experiment, Manuscript preparation, Data interpretation, Graphical presentation,
K A Subramanian: Conceptualization, Investigation, Methodology, Supervision, Writing-original draft

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