

## EFFECT OF AIR PREHEATING, EXHAUST GAS RE-CIRCULATION, AND HYDROGEN ENRICHMENT ON BIODIESEL/METHANE DUAL FUEL ENGINE

by

**Kavin Robert MOHANASUNDARAM\* and Nagarajan GOVINDAN**

Internal Combustion Engineering Division, Department of Mechanical Engineering,  
College of Engineering, Anna University, Chennai, India

Original scientific paper  
<https://doi.org/10.2298/TSCI191024146M>

*An experimental study was carried out to investigate the effect of intake air pre-heating, exhaust gas re-circulation, and hydrogen enrichment on performance, combustion, and emission characteristics of CH<sub>4</sub>/waste cooking oil biodiesel fuelled compression ignition engine in dual fuel mode. Methyl ester derived from waste cooking oil was used as a pilot fuel which was directly injected into the combustion chamber at the end of the compression stroke. The CH<sub>4</sub>/hydrogen-enriched methane was injected as the main fuel in the intake port during the suction stroke using a low pressure electronic port fuel injector which is controlled by an electronic control unit. The experiments were conducted at a constant speed and at the maximum load. Experimental results indicated that the increase in energy share of gaseous fuel extends the ignition delay. With air preheating the thermal efficiency increased to 49% and 55% of CH<sub>4</sub> and hydrogen-enriched CH<sub>4</sub> energy share, respectively. The CO and HC emissions were higher in CH<sub>4</sub> combustion with biodiesel when compared to the conventional diesel operation at full load and a reduction in CO and HC was observed with air preheating. Lower NO<sub>x</sub> were observed with gaseous fuel combustion and it further reduced with exhaust gas re-circulation but NO<sub>x</sub> increased by preheating the intake air. Improvement in thermal efficiency with a reduction in HC and CO was observed with hydrogen-enriched CH<sub>4</sub>.*

**Keywords:** hydrogen enriched CH<sub>4</sub>, CH<sub>4</sub>, exhaust gas re-circulation, air pre-heating, premixed combustion, performance, combustion, emissions

### Introduction

Fuel research and development possess equal importance of engine development in the automotive industry. The objective of this research work is to produce economical and environmentally friendly powertrains. Most of the initial fuel research on alternative fuel started after the 1970's due to the oil crisis. Natural gas (NG), hydrogen enriched methane (H<sub>2</sub>Methane), liquefied petroleum gas, bio-diesel, and biogas are the potential alternative fuels that can be used [1]. Compression ignition (CI) engines and spark ignition (SI) engines are very well-established engine combustion modes and each combustion technique has its own

\* Corresponding author, e-mail: robertkavin@gmail.com

advantages and disadvantages. The SI engine produces low emissions and also suffers from low efficiency while in the case of CI engine, it offers high efficiency but suffers from high emissions particularly particulate matter (PM). The main objective of the engine development is to achieve lower emissions and maximum efficiency. To achieve the goals of maximum efficiency and lower emissions, the engine should operate in the hybrid state of SI and CI engines. Dual fuel combustion concept gives the pathway of operating engines with both cetane and octane fuels in the combustion process [2].

Biodiesel is a methyl or ethyl ester extracted from vegetable or animal fat oil by transesterification process which can be used as an alternative source of fuel for diesel [3]. Biodiesel extracted from waste cooking oil (WCO) has five major chemical components, methyl palmitate ( $C_{17}H_{34}O_2$ ), methyl linolenate ( $C_{19}H_{32}O_2$ ), methyl oleate ( $C_{19}H_{36}O_2$ ), methyl stearate ( $C_{19}H_{38}O_2$ ), and methyl linoleate ( $C_{19}H_{34}O_2$ ) [4]. Many researchers conducted the test on oxygen content in WCO biodiesel blend and observed upto 8% by mass of oxygen in 80% WCO biodiesel blend [4-7]. As WCO biodiesel is an oxygenated fuel, the inherent oxygen in the fuel enhance the oxidation process compared to the oxygen content in the air hence oxygenated fuel helps in reduction of HC. At high pressure injection of biodiesel, emissions such as CO, smoke, and HC were reduced compared with diesel fuel and increase in  $NO_x$  emissions was observed. The  $NO_x$  emissions can be decreased by nanofilter and nanoclay methods in biodiesel treatment which separate  $N_2$  from the biodiesel fuel [8, 9]. Porterio *et al.* [10] did an experimental study on engine performance and emission characteristics of CI engine fuelled with WCO at different blending proportions and observed that thermal efficiency decreases with increase in biodiesel blending proportions. Lower CO emissions were recorded with B50 and  $NO_x$  emissions increase with an increase in biodiesel blending proportions. Jeshvaghani *et al.* [11] observed an increase in output power and efficiency with biodiesel (B20) fuel when compared with diesel where B20 gives similar engine performance as that of diesel fuel. Tormos *et al.* [12] used biodiesel fuel (B50) on different urban buses and observed that usage of B50 does not affect the serviceability of the buses.

The  $CH_4$  can be used in engines in three different modes such as pure  $CH_4$  as fuel, gasoline  $CH_4$  bi-fuel ignited by a spark and diesel  $CH_4$  dual fuel ignited by compression. In the bi-fuel engine, the spark-ignition engine can be fuelled with either gasoline or  $CH_4$  without engine modifications. Kalam and Masjuki [13] conducted experiments in a bi-fuel engine fuelled with compression natural gas (CNG)/gasoline and observed 15% reduction in brake power and 15 to 18% reduction in brake specific fuel consumption (BSFC) with CNG as fuel at full load conditions. Jahirul *et al.* [14] made an experimental study on four cylinders, a bi-fuel engine with 9.5 compression ratio (CR) and observed a reduction in brake power and BSFC with an increase in the thermal efficiency of 1.6% and higher  $NO_x$  in CNG operation compared with gasoline combustion. From the literature study, it is observed that thermal efficiency improved due to better combustion of the homogeneous mixture of  $CH_4$ /air when compared to gasoline/air mixture. The CO and HC emissions decrease due to better combustion, increase in  $NO_x$  due to higher thermal efficiency and combustion temperature and lesser  $CO_2$  due to the lower carbon content in  $CH_4$ . Even though  $CH_4$  has a higher heating value of 55.5 MJ/kg compared with gasoline of 44.5 MJ/kg,  $CH_4$  bi-fuel engine suffers from lower power. This is because the volumetric heating value of  $CH_4$  ( $3.36 \text{ MJ/m}^3$ ) is less than gasoline ( $3.82 \text{ MJ/m}^3$ ) which governs the amount of heat that can be converted into useful mechanical work in the engine combustion [15]. To improve the power output of the engine, pure  $CH_4$  engines were designed and manufactured with a higher CR due to its higher octane number which has higher anti-knocking characteristics. From the literature study on dual-fuel (DF)

engines with CH<sub>4</sub>, longer ignition delay (ID) and lower thermal efficiency were observed due to poor utilization ion of CH<sub>4</sub> at low loads. The NO<sub>x</sub> and soot emission was reduced with CH<sub>4</sub> addition with an increase in CO and HC. For improving the thermal efficiency and lower emissions of CH<sub>4</sub> combustion in both SI and CI engines, the homogeneous charge combustion concept was adopted by many researchers. But the application was not viable due to the barriers of higher intake charge temperature and higher CR.

For improving the thermal efficiency of CH<sub>4</sub> combustion, hydrogen was mixed with CH<sub>4</sub> due to its higher laminar burning velocity (290 cm/s) compared with CH<sub>4</sub> (48 cm/s). The concentration of OH radicals in the fuel mixture increases with hydrogen addition which increases the laminar flame speed of CH<sub>4</sub> combustion. He *et al.* [16] conducted an experimental study of hydrogen enriched CNG fuel in SI engines and observed higher indicated thermal efficiency with 55% hydrogen enrichment. Emissions such as HC, CO, and NO<sub>x</sub> were lower when the ignition takes place closer to TDC. From the literature study on hydrogen enrichment in SI engines, higher thermal efficiency and HC were observed with an increase in gaseous fuel energy share and NO<sub>x</sub> reduces with gaseous fuel energy share. Experimental study on the effect of exhaust gas re-circulation (EGR) in CNG reactivity controlled compression ignition (RCCI) engine was also done by Kalsi and Subramanian [17]. Reduction in thermal efficiency was observed with more than 15% EGR and 8% EGR reduced CO, HC, and NO<sub>x</sub> marginally. Liu *et al.* [18] made a 3-D numerical study on the effect of hydrogen addition on CH<sub>4</sub>/dimethyl-ether RCCI engine and concluded that the addition of hydrogen increases peak cylinder pressure and gives higher efficiency in the beginning state of combustion. Addition of hydrogen increases NO<sub>x</sub> with a reduction in CO emissions. Rashid *et al.* [2] made an investigation to study the effect of fuel stratification on the RCCI engine with two low reactive fuels, compressed natural gas (CNG) and gasoline. Increasing energy share of CNG resulted in delayed ignition and lesser peak pressure for both diesel CNG and gasoline CNG. Stratified fuel-air mixture shows shorter combustion duration on the main combustion and final combustion stages, whereas homogeneous charge mixture exhibits longer combustion duration.

In order to overcome the problems associated with both NO<sub>x</sub> and HC, EGR technique can be used which re-circulate the exhaust gas from the previous cycle to re-burn unburned HC in the successive cycle. The CH<sub>4</sub> combustion mostly suffers from higher CO and HC at medium and low loads. This EGR technique may give positive outputs at low and medium loads by re-circulating the unburned methane back into the combustion process with reduction in exhaust HC, CO, NO<sub>x</sub>, and higher thermal efficiency due to the burning of exhaust HC. At higher loads, the trend differs due to the oxygen deficiency for the combustion which suppresses the oxidation of CO and HC. Abdelaal and Hegab [19] conducted a combustion and emission study of CNG DF engine with EGR and observed that ID increases with increase in EGR ratio and both increasing and decreasing trends of thermal efficiency based on the load and EGR ratios. Nabi and Akhter [20] studied the combined effect of air preheating and re-circulation of exhaust gas on Diesel engines and concluded that PH extract 35% of energy from the exhaust unburned gases and NO<sub>x</sub>, HC, and CO reduce significantly with combined EGR and heating.

Engine combustion with CH<sub>4</sub> and H<sub>2</sub>Methane results in higher CO, HC, and poor utilization of gaseous fuels at low loads. To overcome these problems, the intake air is to be preheated. Yilmaz [21] investigated the effect of intake air preheating of 85 °C in biodiesel methanol blend fuelled CI engine and found that preheated air decreases HC and CO emissions with an increase in exhaust temperature and NO<sub>x</sub> emissions. Feroskhan *et al.* [22] conducted an experimental study on intake air preheating (60 °C, 80 °C, and 100 °C) in biogas

fuelled diesel engine and observed that 100 °C reduces volumetric efficiency, CO and HC but increases NO<sub>x</sub>. Intake air preheating of 60 °C reduces NO<sub>x</sub> but CO and HC increases compared to 100 °C preheating. Intake air preheating of 80 °C provides better thermal efficiency with lesser NO<sub>x</sub> compared with intake air preheating of 100 °C and lesser CO and HC compared with intake air preheating of 60 °C.

From the discussions, it is observed that previous literature has focused on the effect of CH<sub>4</sub> and H<sub>2</sub>Methane on performance, combustion and emission characteristics of combustion in dual fuel mode. Only a few works have been identified with re-circulation of exhaust gas using CH<sub>4</sub> and biogas in DF engines. The available literature in CH<sub>4</sub> and H<sub>2</sub>Methane demonstrate higher thermal efficiency up to a certain level of gaseous fuel energy share. However, increased HC and CO emissions still remain challenging because of prevailing lower in-cylinder temperatures due to lower global reactivity gradient and reduced oxidation process. The effect of intake air preheating and re-circulation of exhaust gas on H<sub>2</sub>Methane combustion can effectively reduce CO and HC emissions in dual fuel combustion concept and has not been investigated so far. This study aims to investigate the effect of intake air preheating, EGR, and hydrogen addition on CH<sub>4</sub>/biodiesel fuelled DF engine experimentally. All the experiments were conducted at full load condition on the test engine (*i.e.* 3.7 kW).

### Materials and procedures

The engine used for the present study was Kirloskar make, AV1 type, water-cooled single-cylinder engine. The engine specifications are given in tab 1. The experimental apparatus was constructed as shown in fig. 1. The engine was coupled with a water-cooled eddy current dynamometer. The air intake system was modified for the injection of gaseous fuel. Biodiesel fuel was injected directly into the combustion chamber using the direct mounted conventional injector where the fuel injection rate is controlled by a mechanical governor. The calibrated gases of 20% hydrogen and 80% CH<sub>4</sub> by volume were stored in a high pressure gas cylinder at 130 bar. The gas pressure was reduced to 3-4 bar using a two-stage pressure regulator provided in the cylinder. A flashback arrestor was fitted in the line next to the pressure regulator to arrest any flash that should not reach the gas cylinder. This flashback arrestor also acts as a non-return valve. A gas rotameter with an accuracy of ±1% was used to measure the gaseous fuel flow rate in standard Litres per minute. A wet type of flame arrestor was provided to quench the flame in case. In the flame arrestor, the pressure tank was filled with a liquid and the gas was made to pass through the liquid medium making bubbles and then captured at the other end. The liquid medium used was distilled water. Bosch CH<sub>4</sub> port fuel injector was used to inject the gaseous fuel in the engine. A provision was made in the port for the location of the injector. A digital weighing balance was used to measure the reduction in the weight of the gas cylinder, from which the amount of gas injected was recorded in grams per hour.

**Table 1. Specifications of engine**

Model and make	AV1, Kirloskar make
General	4-stroke/vertical
Type	CI
Number of cylinders	One
Stroke	110 mm
Bore	80 mm
Compression ratio	16.5: 1
Clearance volume	36.87 cc
Swept volume	553 cc
Rated speed	1500 rpm
Rated output power	3.7 kW at 1500 rpm
Type of cooling	Water-cooled
Combustion chamber	Hemispherical open

Nichrome type heating element with fins was used to heat the air in the heating chamber. The power supply was given to the heating element through the contactor which was used as an automatic switch to control the power supply. The K-type thermocouple was used for temperature measurement and the temperature was controlled to a particular value by activating the contactor by the temperature controller.

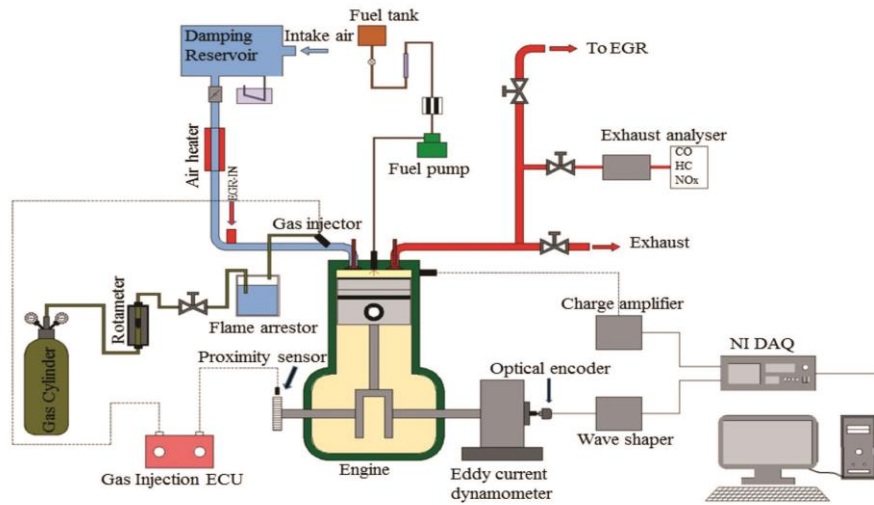


Figure 1. Schematic diagram of the experimental set-up

Initially, experiments were conducted with biodiesel-diesel blended (B50) fuel under conventional mode at a maximum engine power output of 3.7 kW. The intake system was modified to inject the gaseous fuels at the port during suction stroke using low pressure (3 bar) gas injector governed by the specialized gas management system to operate the engine in dual fuel mode. The experiments were repeated with CH<sub>4</sub> and H<sub>2</sub> Methane at different energy shares up to 76% of gaseous energy shares. The experimental set-up was modified with EGR cooler which cools and re-circulates the portion of the exhaust gas back into the combustion chamber for successive combustion cycles. The experiments were conducted with 10% and 20% EGR. The EGR % was calculated [23]:

$$\text{EGR} = \frac{[\text{CO}_2]_{\text{in}} - [\text{CO}_2]_{\text{atm}}}{[\text{CO}_2]_{\text{ex}} - [\text{CO}_2]_{\text{atm}}} \quad (1)$$

The intake system was further modified with an external preheating chamber containing electrical heaters. The experiments were conducted with preheating of intake air to about 80 °C.

Global equivalence ratio,  $\phi$ , was calculated [24]:

$$\phi = \frac{(\text{mass of B50} \cdot \text{AFR of B50}) + (\text{mass of gaseous fuels} \cdot \text{AFR of gaseous fuels})}{\text{Total mass of the air}} \quad (2)$$

Energy share of gaseous fuels was determined by [25]:

$$E_{\text{gaseous fuel}} = \frac{\text{Heat input by gaseous fuel}}{\text{Total heat input}} 100(\%) \quad (3)$$

Thermal efficiency was calculated [26]:

$$TE(\eta_{th}) = \frac{\text{Brake power}}{\text{Total heat input}} 100 (\%) \quad (4)$$

The ID,  $\theta_{ID}$ , was calculated [27]:

$$\theta_{ID} = \theta_{SOC} - \theta_{INJ} \quad (5)$$

where  $\theta_{SOC}$  is the crank angle at the start of combustion (SOC) and  $\theta_{INJ}$  is the crank angle at the start of injection (SOI). The SOC was determined by mass fraction burnt (MFB) curve. The burned mass fraction is the fraction of heat release relative to the total cumulative gross chemical heat release between the SOC and end of combustion. The combustion state occurring in the cylinder can be distinguished by the MFB. From the start of fuel injection, the crank position where the MFB becomes 10% is noted as CA10. This CA10 is the crank angle where the combustion starts. The difference between the fuel injection start point and CA10 is called the flame-development angle or ID.

The combustion characteristics were obtained using National instruments make data acquisition system with Labview software. Pressure crank angle data was obtained from the pressure transducer and encoder for the defined engine load. The net heat release rate (HRR) was calculated based on the First law of thermodynamics. Heat transfer from the gases to the cylinder was computed and deducted from the gross HRR to arrive at net HRR and emissions were measured using AVL digas analyser and bosch smoke meter and analysed by origin software.

The AVL make digas analyser was used to measure CO, HC, and NO<sub>x</sub> emissions and bosch make smoke meter was used to measure smoke emissions.

The uncertainty analysis was done to measure the accuracy of the test results. Such an analysis makes it possible to judge the fitness of the results on the basis of a scientific approach.

Standard deviation methods were used to determine the uncertainty of the instrument and it was calculated [23]:

Standard Deviation,

$$\sigma = \sqrt{\frac{\sum_i^n (x_i - \bar{x})^2}{n}} \quad (6)$$

Mean of sample test data,

$$\bar{x} = \frac{\sum_i^n x_i}{n} \quad (7)$$

Error for 95% confidence level,

$$U_{95} = 2\sigma \quad (8)$$

Uncertainty for test data,

$$a = \frac{2\sigma \cdot 100}{\bar{x}} \quad (9)$$

Total Uncertainty,

$$U = \sqrt{a^2 + b^2} \quad (10)$$

Table 2 shows the specifications and uncertainty of the instruments used.

**Table 2. Specifications and uncertainty of instruments**

Instruments	Measured variables	Range	Accuracy	Uncertainty [%]
Eddy current dynamometer (water cooled)	Torque	0-100 Nm	± 1% FSR	0.8
	Speed	0-10000 rpm	± 1 rpm	0.5
Pressure transducer	In-cylinder pressure	0-100 bar		1
Thermocouple K-type	Temperature	0-1000 °C	± 1 °C	2
Exhaust gas analyser	NO	0-5000 ppm	± 50 ppm	3
	CO	0-10 vol.%	± 0.03 vol.%	3.1
	HC	0-5000 ppm	± 10 ppm	5.6
	CO <sub>2</sub>	0-20 vol.%	±0.1 vol.%	6.2
AVL Smoke meter	Smoke	0-100% opacity	±0.1% opacity	5.2
Digital weighing machine	Gas consumptions	0-100 kg	± 5 grams	2
Gas rotameter	Flow rate	0-25 Lpm	± 2 Lpm	12

### Fuel characterization

The waste vegetable oil was used for the production of biodiesel using transesterification process. Oils extracted from the vegetable seed and animal fats are esters of unsaturated and saturated monocarboxylic acids and triglycerides. This reacts with alcohol to convert into methyl or ethyl esters. This chemical reaction is called as transesterification. Conversion to ester can be made either by a single step or two-step process which is determined by the acid value of the oil. [28]:

$$\text{Acid value} = \frac{\text{Titration value} \cdot \text{normality of KOH} \cdot \text{molecular weight of KOH}}{\text{The weight of the oil sample}} \quad (11)$$

The free fatty acid value of WCO used was 0.8 hence the single-step process was carried out. Initially, before the transesterification process, the WCO was filtered and cleaned by means of the hydrolysis process. An equal amount of water was mixed with the WCO and stirred at a constant speed and a constant temperature of 70 °C for about five hours. This process removed the unwanted waste materials from the WCO. For transesterification process, 1/4<sup>th</sup> of methanol and 10% of KOH was mixed with WCO and allowed to stir continuously for 90 minutes with a constant temperature of 60 °C. It was finally washed and heated. Table 3 shows the combustion properties of the high reactivity fuels used.

Hydrogen was blended with CH<sub>4</sub> in the CH<sub>4</sub> cylinder by reducing its pressure to 150 bar. Hydrogen and CH<sub>4</sub> were blended in the ratio of 1:4 [29]. The hydrogen in CH<sub>4</sub> was about 3% by mass and 7% by energy. The properties of hydrogen and CH<sub>4</sub> are given in the tab. 4 [30].

### Gaseous fuel injection

Bosch gas injector was used for the injection of gaseous fuel at the intake port. The injector works on the principle of solenoid and injection timing and injection duration were controlled by the self-developed electronic control unit (ECU). A constant voltage of 12 V supply was given to the injector. The ECU unit receives input from the proximity sensor. The trigger point was adjusted such that the signal was given to the ECU through the proximity sensor at 45° aTDC during suction stroke.

**Table 3. Combustion properties of high reactivity fuel (B50) in comparison with diesel**

Properties	Diesel	B50
Flash point, [°C]	74	80
Calorific value, [kJkg <sup>-1</sup> ]	47180	43146
Cetane index	60.32	62.82
Density, [kgm <sup>-3</sup> ]	830.4	874.6

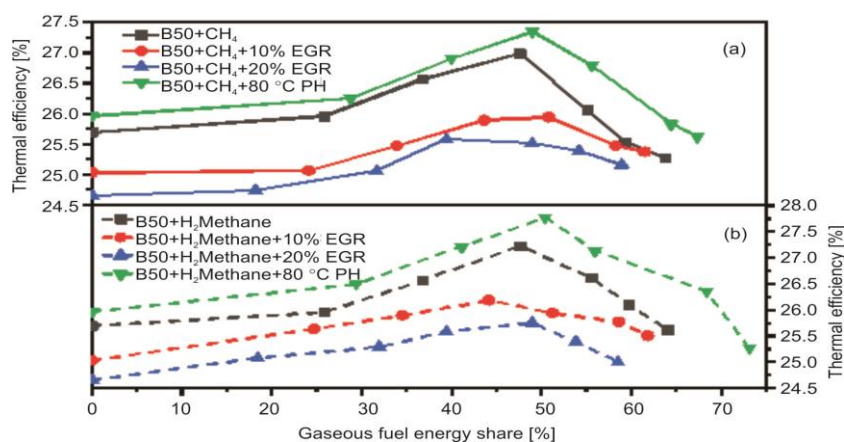
**Table 4. Combustion properties of gaseous low reactivity fuel**

Properties	Methane	Hydrogen
Density [kg/m <sup>3</sup> ]	0.72	0.082
Auto-ignition temperature [°C]	595	585
Laminar flame speed [cm/s]	42	230
Stoichiometric air/fuel ratio	17.2	34.3
Molar mass [kg/kmol]	16.04	2.02
Higher heating value [MJ/kg]	52.23	142.18
Lower heating value [MJ/kg]	47.14	120.21

## Results and discussion

### Performance characteristics

Figure 2 shows the variation of thermal efficiency at different operating conditions with respect to energy shares of gaseous fuels at 3.7 kW. Addition of gaseous fuel enhances the homogeneity of the mixture and higher diffusivity rate of CH<sub>4</sub> increases the thermal efficiency of CH<sub>4</sub> fuelled DF engine [18]. Improvement in thermal efficiency from 25.6% to 27% is observed with B50- CH<sub>4</sub> -fuelled engine up to the gaseous fuel energy share of 48%. Increasing gaseous fuel energy share beyond 48% decreases the thermal efficiency and this may be due to the increase in specific heat capacity (SHC) of the low reactivity CH<sub>4</sub> fuel. Hydrogen enrichment in CH<sub>4</sub> increases the thermal efficiency up to 27.3 at 50% gaseous fuel energy share. Hydrogen enrichment would increase the active OH combustion radicals in the combustion mixtures that result in higher in-cylinder temperature and better combustion. Recirculation of exhaust gas helps in increasing the thermal efficiency mainly at low and medium loads due to the reburn of unburnt fuel in the exhaust [19]. At high loads, lower thermal efficiency is observed with EGR and this is due to the scarcity of oxygen in the combustion mixture which was replaced by the exhaust CO<sub>2</sub>. Increasing EGR% further reduce the thermal efficiency at high loads. Preheating (PH) of the intake charge results in higher thermal efficiency compared to all operations and it extends the gaseous fuel energy share with higher thermal efficiency. In the case of CH<sub>4</sub>, the maximum thermal efficiency obtained was 27.3% at 50% gaseous energy share and in the case of H<sub>2</sub>Methane, 27.9% thermal efficiency was



**Figure 2. Variation in thermal efficiency; (a) CH<sub>4</sub> DF combustion and (b) H<sub>2</sub>Methane DF combustion**



obtained at 54% gaseous energy share. This is due to the increase in charge temperature that provides sufficient temperature to ignite the low reactivity fuel CH<sub>4</sub>/H<sub>2</sub>Methane. In general, the thermal efficiency increases to a certain energy share of gaseous fuel due to the higher degree of homogeneity and decreases with further increase in gaseous fuel energy share due to increase in SHC of the gaseous fuel that takes part in combustion.

### Combustion characteristics

Figure 3 shows the variation of in-cylinder pressure at different energy shares of gaseous fuels at 3.7 kW. Conventional operation is characterized by a non-homogeneous mixture, where the air is only an intake charge while the biodiesel (B50) fuel is directly injected into the combustion chamber closer to the end of the compression stroke. Generally, the non-homogeneous mixture is subjected to a non-premixed combustion process [27]. Initially, air-fuel mixture in ID period attains flammability limit which undergoes prompt combustion in few crank angles.

In gaseous dual fuel operation, non-premixed combustion of pilot biodiesel blend takes place and the gaseous fuel undergoes premixed combustion. The cylinder condition allows the biodiesel blend to ignite spontaneously and the flame propagation makes the air-gaseous fuel mixture to take part in combustion. Better combustion efficiency can be achieved when the air-fuel mixture is around stoichiometric conditions in premixed combustion phase. In dual fuel operation with gaseous fuel, premixed combustion phase suffers from a lean air-fuel ratio which results in the lower in-cylinder pressure. Longer ID in gaseous fuel combustion extends the combustion during expansion stroke which reduces the peak in-cylinder pressure [31]. Increase in in-cylinder pressure is observed with the enrichment of hydrogen gas in CH<sub>4</sub> fuel combustion. This is due to the higher flame speed of hydrogen which burns fuel-air mixture faster resulting incomplete combustion. Hydrogen enrichment increases the mole fraction of OH and O radicals which improves the reaction rate by shortening the ID [1]. The re-circulation of exhaust gas with the intake fuel-air mixture of DF operation lowers the cylinder pressure. The O<sub>2</sub> content in the intake air-fuel mixture was replaced by CO<sub>2</sub>, where a lesser amount of O<sub>2</sub> takes part in combustion which reflects in the lower in-cylinder pressure [32]. Preheating of the intake air reduces the ID compared to gaseous fuel operation, therefore the peak in-cylinder pressure is higher up to a certain energy share of gaseous fuel and the cylinder pressure decreases with an increase in gaseous fuel energy share. Generally, in-cylinder pressure decreases with increase in gaseous fuel energy share at all engine operating conditions due to the low burning rate and higher SHC of low reactivity gaseous fuels.

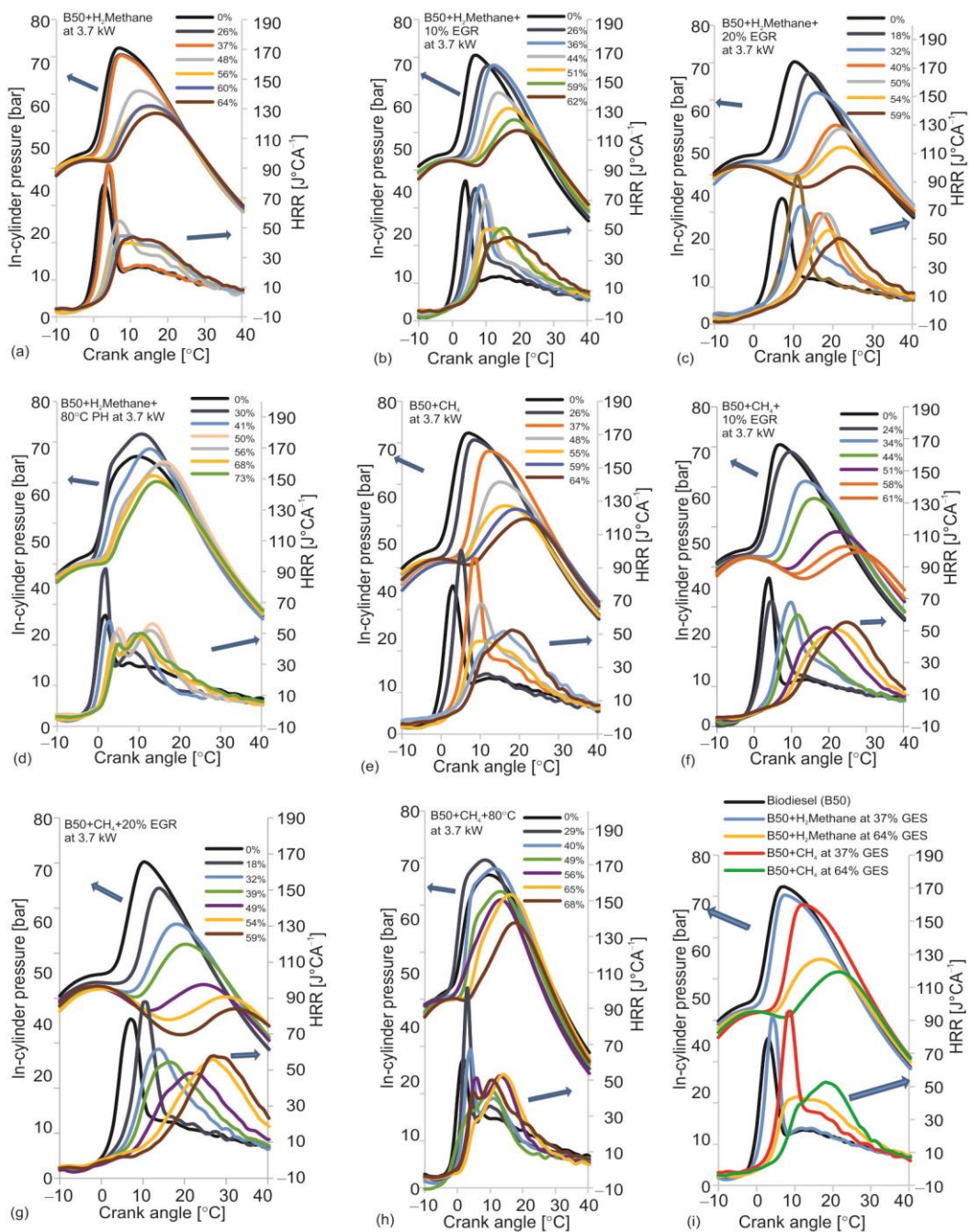
Figure 3 shows the in-cylinder pressure and HRR of (biodiesel and gaseous fuels) dual fuel operation with biodiesel blends at different operating conditions with respect to energy shares of gaseous fuels at 3.7 kW. The HRR was calculated using First law of thermodynamics [33]:

$$dQ_{hr} = dU + dW + dQ_{ht} \quad (12)$$

where  $dQ_{hr}$  is the instantaneous heat release modelled as heat transfer to the working fluid,  $dU$  – the change in internal energy of the working fluid,  $dW$  – the work done by the working fluid, and  $Q_{ht}$  is the heat transmitted away from the working fluid:

$$dU = \frac{C_v}{R}(PdV + VdP) \quad (13)$$

where  $R$  is the gas constant,  $T$ ,  $P$ ,  $V$  are temperature, pressure, and volume, respectively, and  $C_v$  – the specific heat at constant volume.



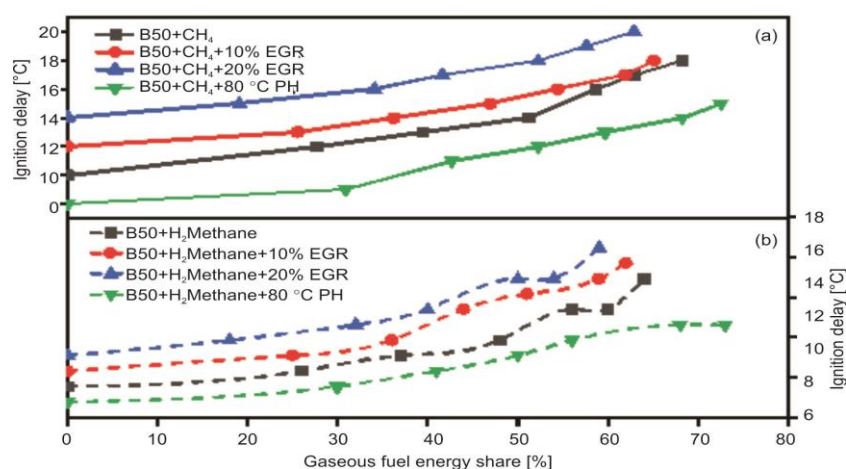
**Figure 3.** Variation in in-cylinder pressure and HRR of biodiesel engine under dual fuel operation; (a) B50+H<sub>2</sub>Methane, (b) B50+H<sub>2</sub>Methane+10% EGR, (c) B50+H<sub>2</sub>Methane+20% EGR, (d) B50+H<sub>2</sub>Methane+ 80 °C PH, (e) B50+CH<sub>4</sub>, (f) B50+CH<sub>4</sub>+10% EGR, (g) B50+CH<sub>4</sub>+20% EGR, (h) B50+CH<sub>4</sub>+ 80 °C PH, and (i) B50+CH<sub>4</sub> and H<sub>2</sub>Methane at different gaseous fuel energy share (GES)

Net HRR:

$$\frac{dQ_n}{d\Theta} = \frac{\gamma}{\gamma-1} p \frac{dV}{d\Theta} + \frac{1}{1-\gamma} V \frac{dp}{d\Theta} \quad (14)$$

where  $(dQ_n)/d\Theta$  is the net HRR and  $\gamma$  – the ratio of specific heats of the fuel and air.

In the case of conventional operation, due to the shorter ID, the SOC is earlier than that of gaseous dual fuel operation [27]. In dual fuel operation, the HRR curve shows two different peaks, the premixed combustion peak curve, and mixing controlled combustion peak curve. Higher premixed combustion phase is observed with CH<sub>4</sub> combustion up to 37% gaseous fuel energy share beyond which the premixed combustion decreases. The lower value of HRR in gaseous fuel operation beyond certain gaseous fuel energy share is due to the lean mixture of gaseous fuel-air, where a large amount of gaseous fuel escapes without taking part in the combustion process. The EGR further lowers the HRR because some O<sub>2</sub> content is substituted by the CO<sub>2</sub> and water vapour which reduce the burning rate and increase the heat capacity of the gaseous fuel mixture [25]. Higher premixed combustion is observed with pre-heating of intake air with 30% gaseous fuel energy share due to the rapid combustion of H<sub>2</sub>Methane fuels during combustion. Retardation in SOC is observed with increase in gaseous fuel energy share due to the charge dilution and SHC of air-gaseous fuel mixed intake charge. The shorter combustion duration and higher premixed combustion are observed with the addition of hydrogen in CH<sub>4</sub>. The active OH and O radicals advance the combustion process.



**Figure 4. Variation in ID under; (a) CH<sub>4</sub> DF combustion and (b) H<sub>2</sub>Methane DF combustion**

Figure 4 shows the variation of ID at different operating conditions with respect to energy shares of gaseous fuels at 3.7 kW. The ID is a measure of the crank angle between the SOI and the SOC. A longer ID is recorded in gaseous dual fuel mode when compared with the conventional operation. The physical delay is due to the absorption of pre-ignition heat release because the homogeneous mixture of gas fuel and air intake charge has a higher SHC when compared to air [27]. The chemical delay is due to the reaction kinetics of pilot biodiesel blend vapour and the gaseous fuel. The addition of hydrogen helps in shortening the ID due to its higher flame speed and burning characteristics. When the exhaust gas is recirculated, it further increases the ID. The EGR will replace a portion of intake air with combustion products. Due to the less availability of oxygen which makes it difficult for combus-

tion initiation, the ID is increased [27]. Preheating of the intake air helps in reducing the ID by providing a sufficient temperature to ignite earlier. Generally, ID increases with an increase in gaseous fuel energy share and EGR percentage.

### Emission characteristics

Figures 5(a) and 6(a) show the variation of CO at different operating conditions with respect to the energy share of gaseous fuels at 3.7 kW. The rate of CO formation is based on

the mixture temperature and oxygen availability which controls the fuel oxidation and decomposition [27]. The CO emissions in CH<sub>4</sub> gaseous fuel engines are higher compared to conventional diesel operation. This is due to the poor utilization of gaseous fuel and suffers from improper fuel decomposition and partial oxidation of HC. Re-circulation of exhaust gas helps in reducing CO at low and medium loads where the unburned active radical reburn converting partially oxidized fuel into CO<sub>2</sub> [34]. At higher loads, the gaseous fuel engine suffers from higher CO emissions due to oxygen deficiency for the combustion process for oxidizing the decomposed fuel particles. Hydrogen enrichment in CH<sub>4</sub> helps in reducing CO emissions compared to neat CH<sub>4</sub> combustion in DF engines. The higher laminar speed of hydrogen provides better combustion and increased cylinder temperature which helps in converting partially oxidized fuel into CO<sub>2</sub>. The CO emissions decrease with preheating of intake air at all operating conditions. Preheating gives sufficient temperature for better combustion of the fuel-air mixture. The CO emissions increase with an increase in gaseous fuel energy share at all the operating conditions.

Figures 5(b) and 6(b) show the variation of CO<sub>2</sub> at different operating conditions with respect to the energy share of gaseous fuels at 3.7 kW. In the case of gaseous dual fuel operation, lesser CO<sub>2</sub> emissions were recorded due to lower carbon to hydrogen ratio (C/H) of the CH<sub>4</sub>/H<sub>2</sub>Methane gaseous. When the engine was operated at full load conditions, the CO<sub>2</sub> emissions are very low due to the incomplete combustion of gaseous fuels. Higher HC and

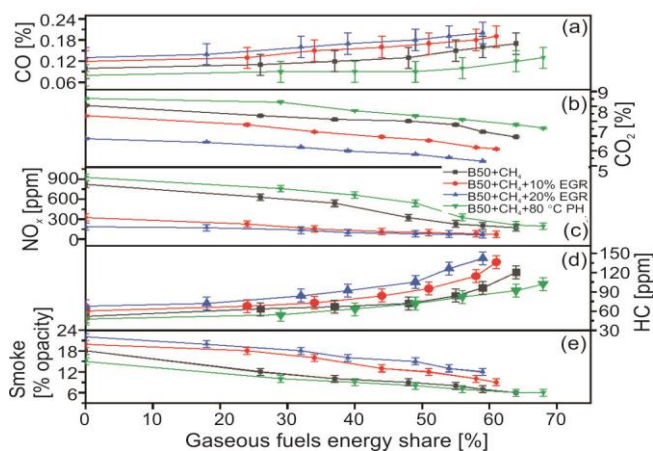


Figure 5. Variation in emissions under CH<sub>4</sub> DF combustion

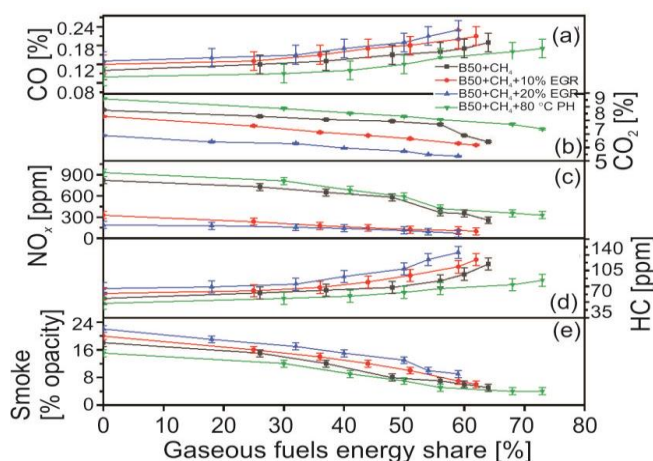


Figure 6. Variation in emissions under H<sub>2</sub>Methane DF combustion

CO emissions at full load conditions, making it difficult for complete combustion of HC fuel which results in lesser CO<sub>2</sub>. Hydrogen enrichment in methane increases CO<sub>2</sub> in all operating conditions when compared with methane DF combustion. Enrichment of hydrogen helps in the formation of higher OH radicals for complete combustion of HC fuels. Re-circulation of exhaust gases in DF operation increases CO<sub>2</sub> emissions only at part loads but at higher loads, the trend is different, which emits lesser CO<sub>2</sub> emissions due to the deficiency in O<sub>2</sub>. Preheating the intake air increases CO<sub>2</sub> at all operating conditions. Heated intake air provides sufficient combustion atmosphere for complete burning of HC.

Figures 5(c) and 6(c) show the variation of NO<sub>x</sub> at different operating conditions with respect to energy shares of gaseous fuels at 3.7 kW. The Zeldovich (thermal) and Fenimore (prompt) mechanisms are responsible for the formation of nitric oxide in the combustion process [35]. The Zeldovich NO<sub>x</sub> formation is due to the high combustion temperature which is above 1400 K [19]. The Zeldovich reaction is influenced by high in-cylinder pressure and oxygen concentration. Fenimore NO<sub>x</sub> is formed in rich, low temperature combustion zone where the active radical is available to react with N<sub>2</sub> [36]. The Zeldovich mechanism of NO formation is considered to be the highest contributor of total NO. In the case of reactivity controlled combustion, a little amount of pilot biodiesel blend takes part in diffusion combustion followed by the premixed combustion of the gaseous fuel. This premixed combustion of gaseous fuel produces very low NO<sub>x</sub> emission only. This results in the reduction of NO<sub>x</sub> emission in the case of gaseous fuel operation with methane and H<sub>2</sub>Methane at part load. The supplement of gaseous fuel in the intake will reduce the air quantity and the deficiency of oxygen concentration in intake charge reduces the combustion temperature and hence thermal NO<sub>x</sub> decreases. With EGR in gaseous fuel mode of operation, a reduction in NO<sub>x</sub> was observed. The formation of NO<sub>x</sub> is mainly dependent on the oxygen concentration in the intake charge and the intake charge temperature. When the exhaust gas is recirculated, the amount of O<sub>2</sub> is replaced by CO<sub>2</sub> and water vapour which dilute and increase the heat capacity of the mixture resulting in lower combustion temperature and reduction in NO<sub>x</sub>. Preheating of intake air helps in enhancing combustion with high in-cylinder temperature, hence NO<sub>x</sub> increases in preheating. The NO<sub>x</sub> emission decreases with an increase in gaseous fuel energy share.

Figures 5(d) and 6(d) show the variation of HC at different operating conditions with respect to energy shares of gaseous fuels at 3.7 kW. Higher HC emissions are observed with CH<sub>4</sub> combustion and increase with an increase in CH<sub>4</sub> fuel energy share. This is due to the poor utilization of CH<sub>4</sub> and high self-ignition temperature of CH<sub>4</sub>. The CH<sub>4</sub> injected may escape in the exhaust without taking part in combustion. The EGR gives a better result in reducing the HC at low and medium loads [18]. A portion of unburned CH<sub>4</sub> in the exhaust was recirculated to re-burn HC in the successive combustion cycle with sufficient oxygen. At high loads, the trend differs with an increase in HC due to oxygen deficiency for the combustion process. Hydrogen enrichment with CH<sub>4</sub> reduces HC emissions at all engine operating conditions compared with CH<sub>4</sub>. This is due to the lower carbon and formation of higher OH radical for better combustion than CH<sub>4</sub>. Preheating of intake air gave a positive approach in reducing the HC emissions at all the operating conditions [37]. This is because preheating provides sufficient temperature for burning high self-ignition temperature fuels such as CH<sub>4</sub> and H<sub>2</sub>methane. The HC increases with an increase in gaseous fuel energy share at all the operating conditions.

Figures 5(e) and 6(e) show the variation of smoke at different operating conditions with respect to the energy share of gaseous fuels at 3.7 kW. In the case of gaseous fuel operation, lesser smoke emissions were recorded due to the lower C/H of the CH<sub>4</sub>/H<sub>2</sub>Methane gas-

eous fuels and higher diffusion coefficient of gaseous fuel [38]. The addition of gaseous fuel increases the ID and gives sufficient time for proper mixing of gaseous fuel in the intake manifold to form a homogeneous mixture and increases premixed charge combustion phasing. In the case of hydrogen enrichment, smoke emission is further reduced when compared with CH<sub>4</sub> combustion due to high hydrogen to carbon ratio and high burning velocity which provides better combustion of fuels. Smoke emission is less with 10% EGR compared to the conventional operation but at higher EGR ratios smoke increase significantly at high loads [18]. This is due to the lack of oxygen available for the combustion process. Preheating of the exhaust gas reduces smoke emissions due to better combustion efficiency. Generally, smoke emission decreases with an increase in gaseous fuel energy share at all the operating conditions.

### Conclusions

The major findings from the investigation of the effect of air preheating and re-circulation of exhaust gas on CH<sub>4</sub>/H<sub>2</sub>Methane and biodiesel fuelled stationary DF engine are summarized as follows.

- The thermal efficiency of the engine improved marginally with gaseous dual fuel mode at a high power output of 3.75 kW. Thermal efficiency increases up to 27% at 47% of energy share of methane and 27.3% thermal efficiency at 50% energy share of H<sub>2</sub>Methane. Increase in gaseous fuel energy share beyond 50% reduces thermal efficiency. Preheating the intake air increases the thermal efficiency to about 27.3% at 50 % energy share of methane and 27.9% thermal efficiency at 54% energy share of H<sub>2</sub>Methane. Lower thermal efficiency was observed with re-circulation of exhaust gas.
- The NO<sub>x</sub> emissions reduce with increase in energy share of CH<sub>4</sub> and it further reduces with re-circulation of exhaust gas. Hydrogen enrichment in methane increases NO<sub>x</sub> compared to CH<sub>4</sub> combustion. Preheating of intake air increase NO<sub>x</sub> in both CH<sub>4</sub> and H<sub>2</sub>Methane fueled engine operation.
- The CO<sub>2</sub> emissions reduce with increase in energy share of CH<sub>4</sub> due to the incomplete combustion of CH<sub>4</sub> fuel and it further reduces with EGR. Hydrogen enrichment in CH<sub>4</sub> provides higher OH radical for combustion and therefore increase in CO<sub>2</sub> emissions was recorded when compared to CH<sub>4</sub> combustion. Preheating the intake air increases CO<sub>2</sub> emissions in both dual fuel operations.
- Smoke emission reduces with increase in energy share of gaseous fuels and it is further reduced by preheating the intake air. Re-circulation of exhaust gas increases smoke at high loads.
- The CO and HC emissions increase with increase in gaseous fuel energy share and EGR further increases CO and HC. Preheating of intake air decreases CO and HC emissions.

Overall, it is concluded that preheating the intake air will reduce CO and HC emissions with a marginal increase in NO<sub>x</sub> emissions. The combined effect of EGR and preheating may help in reducing CO, HC, and NO<sub>x</sub> and increase the thermal efficiency.

### Acknowledgment

The authors acknowledge the Centre for Research, Anna University, for providing Anna Centenary Research Fellowship (ACRF) (Grant Number: CFR/ACRF/2016/15, Dated 21.01.2016) towards conducting this study.

### Acronyms

BSFC – brake specific fuel consumption

CI – compression ignition

CNG	– compressed natural gas	MFB	– mass fraction burnt
CR	– compression ratio	PH	– preheating
C/H	– carbon to hydrogen ratio	PM	– particulate matter
DF	– dual-fuel	RCCI	– reactivity controlled compression ignition
ECU	– engine control unit	SHC	– specific heating capacity
EGR	– exhaust gas re-circulation	SI	– spark ignition
GES	– gaseous energy share	SOC	– start of combustion
HRR	– heat release rate	SOI	– start of injection
H <sub>2</sub> Methane	– hydrogen enriched methane	WCO	– waste cooking oil
ID	– ignition delay		
NG	– natural gas		

## References

- [1] Mehra, R. K., *et al.*, Progress in Hydrogen Enriched Compressed Natural Gas (HCNG) Internal Combustion Engines – A Comprehensive Review, *Renew. Sustain. Energy Rev.*, 80 (2017), Jan., pp. 1458-1498
- [2] Rashid, A., *et al.*, Reactivity Controlled Compression Ignition (RCCI) of Gasoline- CNG Mixtures, *Improvement Trends Intern. Combust. Engines, Intechopen, Chapter-4* (2018), pp. 51-80
- [3] Mohammadi, G. M. F., Utilization of Waste Coral for Biodiesel Production Via Transesterification of Soybean Oil, *International Journal of Environmental Science and Technology*, 11 (2014), 3, pp. 805-812
- [4] Manojkumar, C. V., *et al.*, Reduced Chemical Kinetic Mechanism for a Waste Cooking Oil Biodiesel/N - Pentanol Mixture for Internal Combustion Engine Simulation, *Energy & Fuels*, 32 (2018), 12, pp. 12884-12895
- [5] Cheung, C. S., *et al.*, Effect of Waste Cooking Oil Biodiesel on the Emissions of a Diesel Engine, *Phys. Procedia*, 66 (2015), pp. 93-96
- [6] Garcia-Martin, J. F., *et al.*, Biodiesel Production from Waste Cooking Oil in an Oscillatory Flow Reactor. Performance as a Fuel on a TDI Diesel Engine, *Renew. Energy*, 125 (2018), Sept., pp. 546-556
- [7] Abed, K. A., *et al.*, Effect of Waste Cooking-Oil Biodiesel on Performance and Exhaust Emissions of a Diesel Engine, *Egypt. J. Pet.*, 27 (2018), 4, pp. 985-989
- [8] Afshari, F., *et al.*, The Effects of Nanofilter and Nanoclay on Reducing Pollutant Emissions from Rapeseed Biodiesel in a Diesel Engine, *Waste and Biomass Valorization*, 9 (2018), Mar., pp. 1655-1667
- [9] Nantha Gopal, K., *et al.*, Investigation of Emissions and Combustion Characteristics of a CI Engine Fueled with Waste Cooking Oil Methyl Ester and Diesel Blends, *Alexandria Eng. J.*, 53 (2014), 2, pp. 281-287
- [10] Porteiro, J., *et al.*, Performance and Exhaust Emissions in the Use of Biodiesel in Outboard Diesel Engines, *Fuel*, 86 (2007), 12-13, pp. 1765-1771
- [11] Jeshvaghani, H. S., *et al.*, Performance Analysis of Diesel Engines Fueled by Biodiesel Blends Via Thermodynamic Simulation of an Air-Standard Diesel Cycle, *International Journal of Environmental Science and Technology*, 11 (2014), Apr., pp. 139-148
- [12] Tormos, B., *et al.*, Results of an Operating Experience for Urban Buses Fuelled with Biodiesel Blends ( B50), SAE Technical paper 2009-01-1827, 2009
- [13] Kalam, M. A., Masjuki, H. H., An Experimental Investigation of High Performance Natural Gas Engine with Direct Injection, *Energy*, 36 (2011), 5, pp. 3563-3571
- [14] Jahirul, M. I., *et al.*, Comparative Engine Performance and Emission Analysis of CNG and Gasoline in a Retro Fited Car Engine, *Appl. Therm. Eng.*, 30 (2010), 14-15, pp. 2219-2226
- [15] Yousefi, A., *et al.*, An Experimental and Numerical Study of the Effect of Diesel Injection Timing on Natural Gas/Diesel Dual-Fuel Combustion At Low Load, *Fuel*, 203 (2017), Sept., pp. 642-657
- [16] He, Y., *et al.*, Reducing the Idle Speed of an SI CNG Engine Fueled by HCNG with High Hydrogen Ratio, *Int. J. Hydrogen Energy*, 37 (2012), 10, pp. 8698-8703
- [17] Kalsi, S. S., Subramanian, K. A., Experimental Investigations of Effects of EGR on Performance and Emissions Characteristics of CNG Fueled Reactivity Controlled Compression Ignition (RCCI) Engine, *Energy Convers. Manag.*, 130 (2016), Dec., pp. 91-105
- [18] Liu, Y., *et al.*, An Experimental Investigation of the Engine Operating Limit and Combustion Characteristics of the RI-CNG Engine, *J. of Mec. Science and Technology*, 26 (2012), 11, pp. 3673-3679
- [19] Abdelaal, M. M., Hegab, A. H., Combustion and Emission Characteristics of a Natural Gas-Fueled Diesel Engine with EGR, *Energy Convers. Manag.*, 64 (2012), Dec., pp. 301-312

- [20] Nabi, M. N., Akhter, M. S., Combined Effect of EGR and Inlet Air Preheating on Engine Performance in Diesel Engine, *International Energy Journal*, 9 (2008), pp. 109-116
- [21] Yilmaz, N., Effects of Intake Air Preheat and Fuel Blend Ratio on a Diesel Engine Operating on Biodiesel – Methanol Blends, *Fuel*, 94 (2012), Apr., pp. 444-447
- [22] Feroskhan, M., et al., Effects of Charge Preheating on the Performance of a Biogas-Diesel Dual Fuel CI Engine, *Eng. Sci. Technol. an Int. J.*, 21 (2018), 3, pp. 330-337
- [23] Duraisamy, G., Rangasamy, M., Effect of EGR and Premixed Mass Percentage on Cycle to Cycle Variation of Methanol/Diesel Dual Fuel RCCI Combustion, SAE Technical paper 2019-26-0090, 2019
- [24] Sarkar, A., Saha, U. K., Role of Global Fuel-Air Equivalence Ratio and Preheating on the Behaviour of a Biogas Driven Dual Fuel Diesel Engine, *Fuel*, 232 (2018), June, pp. 743-754
- [25] Kalsi, S. S., Subramanian, K. A., Experimental Investigations of Effects of Hydrogen Blended CNG on Performance, Combustion and Emissions Characteristics of a Biodiesel Fueled Reactivity Controlled Compression Ignition Engine (RCCI), *Int. J. Hydrogen Energy*, 42 (2017), 7, pp. 4548-4560
- [26] Heywood, J. B., *Internal Combustion Engine*, Mc-Graw Hills, New York, USA, 1998
- [27] Liu, Z., Karim, G. A., An Examination of the Ignition Delay Period in Gas-Fueled Diesel Engines, *J. Eng. Gas Turbines Power*, 120 (1998), 1, 225
- [28] Sharma, V., Duraisamy, G., Production and Characterization of Bio-Mix Fuel Produced from a Ternary and Quaternary Mixture of Raw Oil Feedstock Bureau of Indian Standards, *J. Clean. Prod.*, 221 (2019), June, pp. 271-285
- [29] Baltacioglu, M. K., et al., Experimental Comparison of Pure Hydrogen and HHO (Hydroxy) Enriched Biodiesel (B10) Fuel in a Commercial Diesel Engine, *International Journal of Hydrogen Energy* 41 (2016), 19, pp. 3-9
- [30] Baltacioglu, M. K., et al., Effect of Using Hydroxy – CNG Fuel Mixtures in a Non-Modified Diesel Engine by Substitution of Diesel Fuel, *International Journal of Hydrogen Energy* 41 (2016), 19
- [31] Guerry, E. S., et al., Injection Timing Effects on Partially Premixed Diesel-Methane Dual Fuel Low Temperature Combustion, *Appl. Energy*, 162 (2016), Jan., pp. 99-113
- [32] Pirouzpanah, V., et al., Comparison of Thermal and Radical Effects of EGR Gases on Combustion Process in Dual Fuel Engines at Part Loads, *Energy Convers. Manag.*, 48 (2007), 7, pp. 1909-1918
- [33] Stone, R., *Introduction To Internal Combustion Engines*, 4<sup>th</sup> ed., Palgrave Macmillan, London, UK, 2012, pp. 421-423
- [34] Abd-Alla, G. H., Using Exhaust Gas Re-circulation in Internal Combustion Engines: A Review, *Energy Convers. Manag.*, 43 (2002), 8, pp. 1027-1042
- [35] Papagiannakis, R. G., et al., A Combined Experimental and Theoretical Study of Diesel Fuel Injection Timing and Gaseous Fuel/Diesel Mass Ratio Effects on the Performance and Emissions of Natural Gas-Diesel HDDI Engine Operating at Various Loads, *Fuel*, 202 (2017), C, pp. 675-687
- [36] Konigsson, F., On Combustion in the CNG - Diesel Dual Fuel Engine, Ph. D. thesis, KTH Royal Institute of Technology, Stockholm, 2014
- [37] Papagiannakis, R. G., Study of Air Inlet Preheating and EGR Impacts for Improving the Operation of Compression Ignition Engine Running Under Dual Fuel Mode, *Energy Convers. Manag.*, 68 (2013), Apr., pp. 40-53
- [38] Sandalci, T., et al., Effect of Hythane Enrichment on Performance, Emission and Combustion Characteristics of an CI Engine, *Int. J. Hydrogen Energy*, 44 (2018), 5, pp. 3208-3220