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Effects of using a novel fuel vaporizer on partially premixed charge compression ignition (PPCCI) engine emissions, performance, and combustion characteristics

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Abstract. Environmental concerns of toxic emissions and depleting of fossil fuel supplies due to their excessive usage as the main source of energy have raised interests in the creation of novel combustion modes that result in reduction of combustion temperatures and produce low emissions. In comparison to conventional diesel engines, the partially pre-mixed charged compression ignition (PPCCI) combustion strategy has demonstrated its ability to significantly reduce emissions carbon monoxide (CO), unburned hydrocarbon (HC), oxides of nitrogen (NO_X), carbon dioxide (CO_2) and smoke opacity. In order to compare the results with those of conventional engines, the current experimental work's objective is to investigate the combustion, performance, and exhaust emissions characteristics of PPCCI engines. A single-cylinder, air-cooled, 4-stroke, direct-injection diesel engine that had been modified to run in PPCCI mode was used for the experiments. An external mixture formation technique with a fuel vaporizer is added to create the homogeneous mixture for PPCCI combustion. After being heated to the point of vaporization, liquid diesel fuel vapor was mixed with some fresh air and then the mixture directed to the intake manifold, where it was mixed with the remaining fresh air to create an external homogenous mixture that filled the combustion chamber. The tests were conducted at different premixed ratios of diesel fuel proportions of 15%, 20%, and 30% in the intake port. However, the fuel vaporizer chamber was kept at fixed temperature of 100 ºC, 105 ºC, 110 ºC, 115 ºC, and 120 ºC. The PPCCI engine results were compared with the conventional engine data. Results from the PCCI technique at various premixed ratios indicate a certain decrement for HC, CO, $NO_{X,}$ and smoke emissions, rising in BTE "brake thermal efficiency". At 30% premixed ratio of the fuel vapour inducted at 110 ºC in PCCI mode give the best results as the brake thermal efficiency raised from 28.8% for CDC mode to 34.2% for PCCI mode at full load. Additionally, NO_X emissions decreased from 615 PPM to 550 PPM, HC emission decreased to 30 PPM, CO emission decreased from 0.09% to 0.06% and a decrease in smoke opacity from 38% to 19.3%.

Keywords: Fossil diesel fuel; Low temperature combustion; PCCI engine; External mixture formation; Fuel vaporizer; Engine emissions; Engine performance

1. Introduction

At present, due to their extensive use as the main source of energy, fossil fuel resources are rapidly running out [1, 2]. Internal combustion engines (ICEs) is the main consumer of fossil fuels additionally, ICEs produce a significant amount of toxic emissions that are bad for the environment and people's health [3]. The essential components of the exhaust gas are carbon monoxide (CO), unburned hydrocarbon (HC), oxides of nitrogen (NO_X), carbon dioxide (CO_2), smoke opacity and particulate matter (PM) [1, 4]. Compression ignition (CI) engines are the most appropriate converter device for use in transportation applications and enormous power requirements because they have high combustion efficiency, reliability and durability due to their high compression ratio and require lower maintenance compared to engines with spark ignition (SI) [5]. Hence, CI engine obtained a great concern the formation of NO_X and smoke emissions. The after-treatment devices "diesel particulate filter (DPF), lean NO_X traps (LNTs) and selective catalytic reduction (SCR)" represent an effective way in controlling NO_X and soot emissions but they are costly sub-systems. Additionally, they require repetitive regeneration over a period of time and increase fuel consumption [6, 7].

Owing to the high increase of these pollutants and the depletion of fuel resources, there is a strong requirement to develop an efficient combustion system with low emissions and fuel-flexibility by improving the in-cylinder combustion [8]. Low-temperature combustion (LTC) methods are being developed in order to sustainably lower NO_X and soot emissions to an ultralow level with sustaining higher/the-same thermal efficiency. LTC used various strategies "Premixed Charge Compression Ignition (PCCI), Homogenous Charge Compression Ignition (HCCI), Reactivity Controlled Compression Ignition (RCCI), and Stratified Charge Compression Ignition (SCCI)" to keep the in-cylinder temperature relatively low than in CDC [9]. The HCCI combustion is considered a hybrid combination of spark ignition and compression ignition engine operating cycles since the charge is well premixed like in SI engine and is ignited automatically as a result of compression like in CI engine, combustion of the charge occurs spontaneously with the absence of any diffusion flame [10]. The RCCI engine uses multiple fuel injection techniques that use fuel with low reactivity (LRF) is injected through the port fuel injection system during the intake stroke, creating a homogeneous air–fuel charge [11]. Next, fuel with high reactivity (HRF) is injected into the cylinder via the direct injection system during the compression stroke to create a stratification of reactivity regions inside the cylinder. Combustion phasing for RCCI is controlled by varying the ratio of the two fuels with varies auto-ignition characteristics [12, 13]. HCCI mode of combustion can be achieved by using two strategies, namely improvement the homogeneity of the external mixture formation "A/F ratio and fuel properties" and the formation of the in-cylinder mixture "injection time, injection pressure and temperature" [14]. The HCCI engines' limitations in commercial applications owing to knocking occurring at high loads as a result of combustion pressure are their main drawbacks, misfire during cold start operation and the difficulty of controlling combustion phasing due to controlling the timing of the ignition and the rate of combustion is complicated [15].

To overcome these difficulties, PCCI engine has been introduced. PCCI engine was developed from HCCI to overcome the major disadvantages such as stability and noise. PCCI strategy (which includes PCCI and partial PCCI) is extremely promising and essential for accomplishing both high/the same efficiency and reducing both NO_X and PM emissions [16, 17]. Table 1 compares the characteristics of CI, HCCI, and PCCI modes. PCCI is characterized by reducing the heterogeneity in the fuel-air mixture charge and this is attained by dividing the quantity of fuel into two parts [18]. At first, small part is injected early to prepare the homogenous mixture, while the remaining part injection timing is responsible for controlling the fuel ignition [19, 20]. In the PCCI engine, combustion occurs through two stages. At first spontaneous ignition occurs at unspecified points. Then, the flame is rapidly developing throughout the combustion chamber [21]. According to S. S. Bhurat et al. [22], In comparison to direct injection (DI) mode, PCCI mode emits fewer NO_X and soot emissions with a slight increment in HC and CO emissions. When compared to direct injection (DI) mode, brake thermal efficiency (BTE) was marginally decreased by 2.5%. PPC technique shows high instability in combustion for fuels with high RON at low engine load operation due to low in-cylinder temperature, while poor combustion has a major impact on emissions. For stable combustion, a high intake temperature is required since high RON fuel has low reactivity and strong auto-ignition resistance.

Table 1. Comparison of the fundamental concepts of combustion in the CI, SI, and HCCI modes.

Mode	CI	HCCI	PCCI
Fuel type	High-cetane-like diesel	Flexible fuels	Flexible fuels
air-fuel ratio	$\Phi < 1$	$\Phi \leq 1$	$1 \leq \Phi \leq 2$
Ignition type	Auto-ignition	Auto-ignition	Auto-ignition
injection technique	Injection just before TDC	Early injection	Early injection + TDC injection
Combustion type	diffusion	Premixed	Premixed + diffusion
Mechanism control burning rate	injection timing	chemical kinetics	Time of injection + chemical kinetics
Thermal efficiency	High	Partially high	Partially high
Emissions	High NO_X & smoke opacity	Low NO_X & smoke opacity	Low NO_X & smoke opacity
Combustion temp.	Partially high	Relatively low	Relatively low

Pre-heated air was used in several experiments to help fuel combustion by promoting the vaporization of high RON fuel. Wang Pan et al. [21] examined the effects of increasing the temperature of intake air on emissions with methanol fuel; they discovered that NO_X and PM emissions increased while CO and HC emissions reduced with increasing the intake air temperature. A similar pattern was discovered in another study by K. S. Kumar and R. T. Raj [23]. They examined at how intake-air temperature affected combustion of fuel that was a blend of ethanol and biodiesel on the exhaust emissions characteristics and found that as intake air temperature increased, NO_X emissions increased while CO, HC, and soot emissions decreased. As the intake air temperature is reduced, it was discovered that NO_X emission decreased, HC and CO emissions increased with high RON fuels due to incomplete combustion and the fuel impingement impact to wall since the combustion temperature did not reach auto-ignition [24, 25].

Another essential and significant aspect influencing the PCCI performance and emission characteristics is the proportion of premixed fuel. Therefore, many studies have been implemented to reveal the impact of the amount of premixed injection on the PCCI engine's performance and emissions characteristics. For example, V. Vinodkumar and A. J. E. Karthikeyan Studied [26] the influences of manifold injection at various concentrations of n-decanol like 10%, 20%, and 30% as a secondary injection with a primary fuel of NB20 (80% diesel and 20% biodiesel) on the engine characteristics. From the results, it was observed that BTE increases, and NO_X reduced while HC and CO increased for all manifold injections. Shyam Pandey et al. [27] studied the impact of ethanol at several proportions through manifold in PCCI mode with constant preheat intake temperature at 40 ^{o}C on the engine characteristics. According to the research results, adding more premixed ethanol resulted in a reduction in HC and CO emissions while increasing NO_X and smoke emissions. In another study by Swapnil et al. [28]. In order to evaluate the effects of 15 and 30 percentage diesel vapour, they conducted experiments in a single-cylinder, 4-stroke, direct-injection diesel engine that has been modified to operate in PCCI mode, and then they compared those results with CDC. It was observed that by increasing the proportion of diesel vapour, the BTE, volumetric efficiency, NO_X, and soot emissions reduced while HC and CO emissions continued to increase.

From all the above works, it is evident that use of the PCCI combustion is promising and sustainable. Although there are many works related to the PCCI concept, but the work related to PCCI with different premixed fuel proportions at different intake temperature is very limited. Thereby, the current work focused on converting CI engine to PCCI engine through adding some components to the CI engine without any modification in the main engine components with minimal cost. The objective for this work would be to study the impact of PCCI mode works with different premixed diesel fuel proportions "15%, 20%, and 30%" in intake manifold at various premixed temperature "100 ºC, 105 ºC, 110 ºC, 115 ºC, and 120 ºC" on the performance, emissions, and combustion characteristics of the engine. In this work, a new fuel system attached to an external mixture formation technique has been developed to improve the homogeneity of the various fuel mixtures. Experimental investigation into the engine performance and emission characteristics has been conducted to evaluate the impacts of manifold injection at various concentrations of diesel that are computed under various load circumstances in PCCI and compared combustion and emission characteristics to those of CDC engines.

2. Materials and methods

The major goal of this work is to determine how a single cylinder CI engine's combustion, emission, and efficiency characteristics are impacted by the ratio of diesel vapour passing through the intake manifold. The impact of various diesel vapor ratio and temperature has been mapped, and the results are compared to those from Conventional Diesel Combustion (CDC).

2.1 Experimental setup

The schematic diagram and actual photo of the experimental PCCI engine setup are shown in Figure 1 and Figure 2. A single-cylinder, air-cooled, four-stroke, and DI diesel engine was used for the experiments. The operating conditions within the engine were kept stable with a constant speed of 1500 rpm, lubricating oil temperature maintained at 90 ºC throughout the experiment, fuel pumped at a pressure of 220 bar at a 32 CA before TDC, and a compression ratio of 17. The engine is equipped with a sensor to measure the engine speed, a knock sensor to measure the engine vibrations, switches to control the load, and equipment for measuring temperature, pressure and flow rate were employed in the experiments [29]. A detailed description of the conventional direct injection CI engine used in this study is given in Table 2. An orifice system is used to measure the amount of intake air flowing into the engine additionally; a damping reservoir is used to eliminate the pulsations results from the suction stroke. A generator was coupled to the engine, which enabled to change speed and torque of the engine. According to Figure 3., there are two main component assemblies for the generator type dynamometer. The first is a 5 kW A.C. Synchronous generator attached on the diesel engine's cylinder crankshaft. The load circuit for the dynamometer controller is the second essential part assembly. Lamps and an electric voltage variac device are used to control the generator output power by varying the voltage.

To function in PCCI mode, the engine has already been modified. An external mixture formation technique was added to the engine's intake system that facilitated the homogeneity formation for the charge mixture. The mechanism for forming an external mixture consists of a mixing chamber with two inlet ports and one outlet port. The first inlet port provides a fresh air as it connects to a blower that is the supplier of a small amount of air which passes over an electrical heater to rise its temperature before entering the mixing chamber. The blower is connected to a variac to specify the amount if air. The second inlet port connects to an electrical fuel injector with five holes that is controlled by an electrical circuit as shown in Figure 4. (Consists of an arduino and a tachometer sensor connected to the engine shaft) and in our experiments this injector is kept open for 8 CA degrees for each engine cycle. While, the outlet port is for the air-fuel vapor mixture, the mixing ratio and vapor quality mainly depend on the mixing camber temperature and the temperature of the mixing chamber is controlled by using variac device connected to the heater. Finally, prior to entering the engine, this air-fuel vapour mixture is blended with fresh air inside the intake manifold. The Path between the mixing chamber and the engine is fully insulated to Keep the mixture temperature.

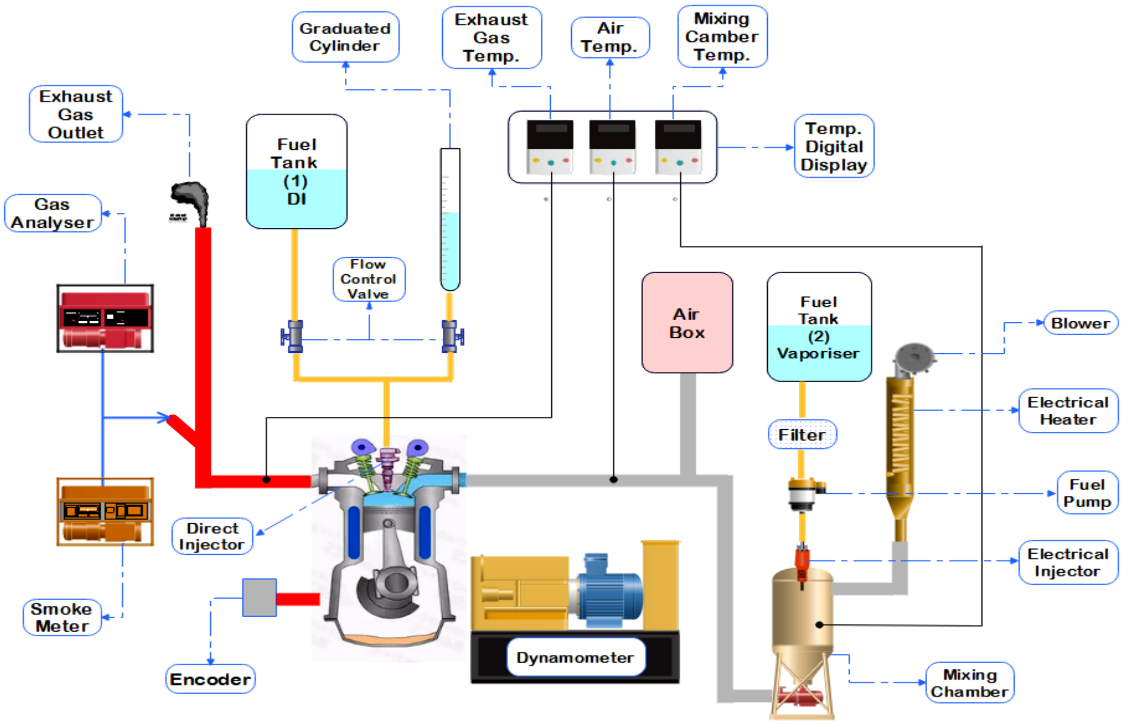


Figure 1. PCCI engine setup schematic diagram with fuel vaporizer.

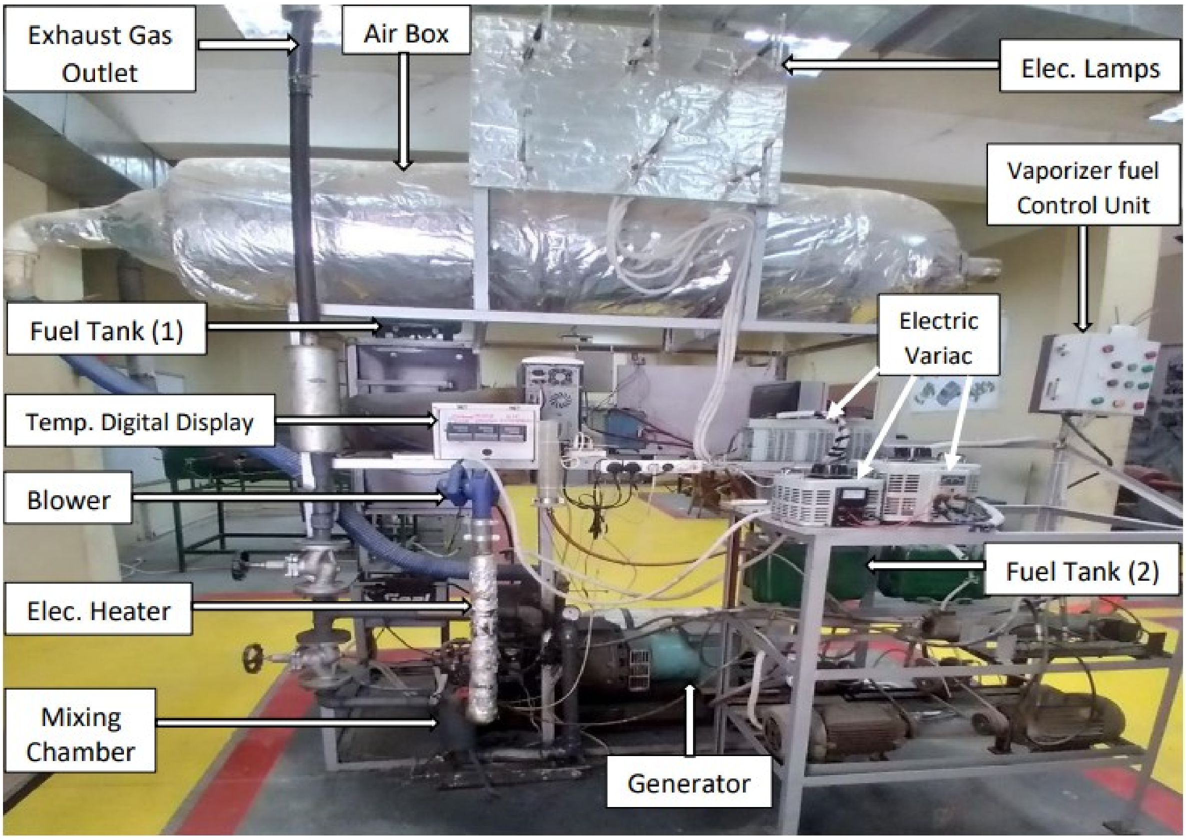


Figure 2. Actual photo of the engine's experimental setup with a fuel vaporizer.

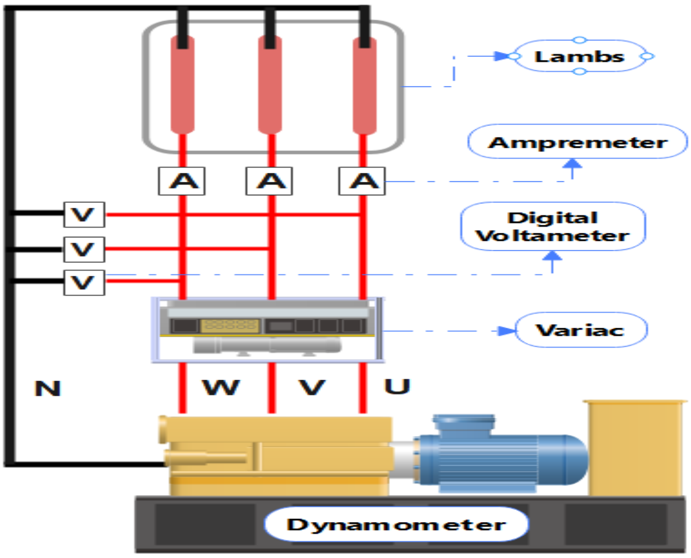


Figure 3. Schematic diagram for a load and dynamometer circuit.

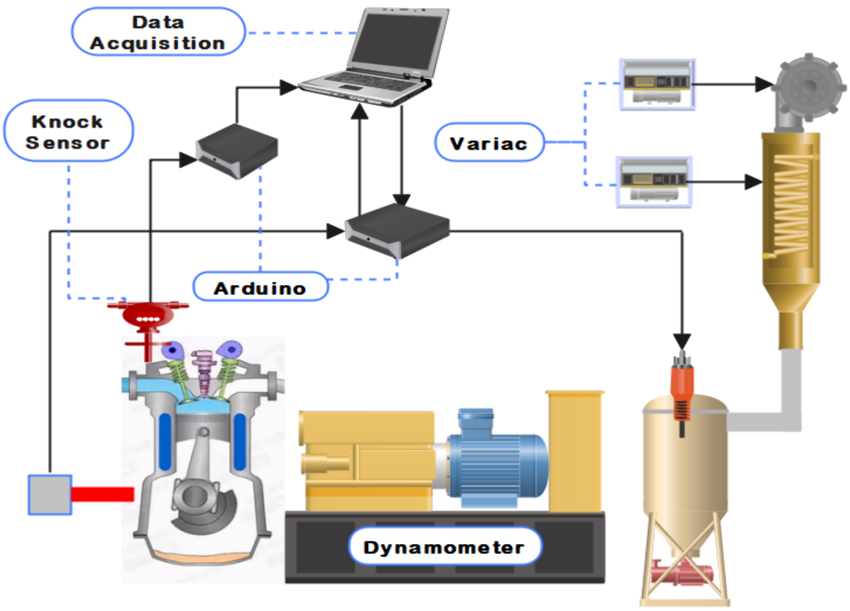


Figure 4. Vibration and vaporizer fuel circuit schematic diagram.

Additionally, the mixing chamber has a vent in the bottom to discharge the condensed fuel on the mixing chamber walls. From the above discussion, it is obvious that there are two different fuel-feeding techniques, one of which is the engine's DI diesel main technique and the other of which is the external feeding technique used in the formation of the external mixture. Similarly, there are two different air-feeding mechanisms: one is the traditional air mechanism that runs via the engine manifold, and the other of which is the external air feeding mechanism used in the formation of the external mixture. Table 3 listed the properties of the test diesel fuel used. The number of samples used, and its composition is mentioned in Table 4. K type thermocouples were used to measure the temperatures of the mixing chamber, exhaust, and inlet air with a 1 ^{o}C accuracy. Additionally, the knock sensor is a small device in the engine that is used to detect the engine block vibrations as depicted in Figure 4. The knock sensor detects the vibrations of engine block, converts it to an electronic signal, and delivers it to an arduino, which connected to a computer where this date was plotted and gave a clear diagram of the engine vibration. Then, we evaluate the data and determine whether the vibrations are abnormal which indicates to the engine knocking.

Table 2. Engine's technical specification.

parameters	specifications
Engine Make	Deutz diesel engine
No. of Cylinders	One
Total cylinder volume	825 cm3
Cylinder bore diameter	10 cm
Piston stroke	10.5 cm
Cycle	4- strokes
Cooling mode	Air Cooled
Injection mode	Direct injection
Maximum power	5 kW @ 1500 rpm
Compression Ratio	17
Inlet valve opens	32 ºCA before TDC
Inlet valve closes	59 ºCA after BDC
Exhaust valve Opens	71 ºCA before BDC
Exhaust valve Closes	32 ºCA before TDC

Table 3. Properties of Diesel fuel.

Parameters	Property value
Kinematic Viscosity (at 30 ºC) (mm^2/s)	2.75
Specific Gravity (@ 15 ºC)	0.83
Pour Point (ºC)	-12
Flash Point (ºC)	50
Fire Point (ºC)	0
Calorific Value (MJ/kg)	43
H/C (molar ratio)	1.8
Cetane Number	54
Range of flammability (vol. %)	0.6–5.6
Sulfur Content (wt %)	0.154
Ash Content (% mass)	0.013
Carbon Reside (wt %)	0.014
Acid Value (mg KOH/g)	0.15

CO_2, CO, O2, HC and NO measurements were measured using the GASBOARD-5020 emission gas analyzer. CO_2, CO, and O2 were estimated in %Vol, while NO_X and HC were calculated in ppm. The soot emissions were performed using the GASBOARD-6010 opacity meter. Uncertainty of the measured and calculated quantities can be calculated through the following equations 1 & 2 [30]. Each observed value's COV (coefficient of variation) is calculated to assess the procedure's accuracy and measurement repeatability and were found in the allowable limit with less than 2 %. Accurate uncertainty analysis has been performed in order to specify the limiting errors related to each measured value. The accuracy of the instruments and the measured values both affect the limiting errors of the measurements. The precision and uncertainty of the several instruments utilized for this experimental investigation are displayed in Table 5 below. The uncertainty related to the measuring instruments in Table 5 demonstrates that the conclusion will be not affected by the uncertainty.

$$\mathrm{COV} = \frac{\sigma}{X_m} \times 100\% \quad (1)$$

$$\sigma = \sqrt{\frac{\sum_1^n (x_i - X_m)^2}{n-1}} \quad (2)$$

Where; σ is the standard deviation, X_m is the average of the variable (x), n is the number of samples cycle

Table 4. Fuel and Blends used for Experimentation.

Samples	Fuel	Premixed Percentage	Premixed Temperature
		CDC Mode	
1	Diesel	______	______
		PCCI Mode	
2	Diesel	15 %	100 ºC
3	Diesel	20 %	105 ºC
4	Diesel	30 %	110 ºC
5	Diesel	30 %	115 ºC
6	Diesel	30 %	120 ºC

Table 5. Specifications for instruments and uncertainty**.**

Instrument	Parameter	Accuracy	Range
	CO_2	± 4%	0-20%
	CO	± 1%	0-20%
Emission gas analyzer	O2	± 3%	0-25%
	HC	± 5%	0-9999 ppm
	NO	± 5%	0-5000 ppm
Opacity meter	Smoke opacity	± 0.01%	0-100%
Thermocouple of type K	Temperature of the exhaust gas	±1%	0 to 800 ºC
Rotary encoder	Engine speed	±0.2%	3000 rpm
Inclined-limb Manometer	Flow rate of air	±2%	0–2.99 m^3/h
Graduated cylinder	Flow meter of diesel	± 1 %	1-30 cm3
Indicator of load	Load	±0.2%	1-1000 watt

2.2 Experimental methodology

All experiments in this work were performed on a single-cylinder DI diesel engine with efficient modification. The experiments were carried out at a constant rotational speed "1500 rpm" under various load conditions, such as no load, 0.9 BMEP, 1.8 BMEP, 2.8 BMEP and 3.8 BMEP. The experiments were conducted in 2-steps. Diesel was tested in the conventional mode in the first step. In the second stage, a part of diesel was injected through the mixing chamber fuel injector to the intake manifold and the remaining diesel fuel injected directly into the combustion chamber in PCCI operation mode. The combustion, emission, vibration, and performance characteristics of the engine were measured in both stages. Finally, the second stage's results have been plotted and contrasted with the first stage's results.

The premixing ratio is calculated using the below equation 3. The direct injected fuel flow rate measured by using a graduated cylinder and a stopwatch. The premixed fuel flow rate that mixed with air in the mixing chamber is determined by the blow equation 4. The fuel flow rate injected by the electrical injector from a separated external tank is fixed and already known. While, the fuel flow rate condensed in the mixing chamber is collected in a tank through the mixing chamber vent, then by using an electrical pump the collected fuel is sucked and discharged to a graduated cylinder, where the flow rate of the returning fuel can be calculated by using a stopwatch before back to the separated external tank.

$$\mathrm{PMR} = \frac{Q_{\mathrm{Premixed}}}{Q_{\mathrm{Premixed}} + Q_{\mathrm{Direct}}} \tag{3}$$

$$Q_{Premixed} = Q_{injected} - Q_{condenced} \tag{4}$$

Where, Q_{Premixed} is the flow rate of fuel premixed with air, Q_{Direct} is the flow rate of the direct injected fuel, Q_{Injected} is the fuel flow rate of the electrical injector, $Q_{\mathrm{Condenced}}$ is the flow rate of fuel condensed in the mixing chamber.

3. Results and discussion

Our results suggested that the premixed mixture percentage and temperature of an engine powered with diesel fuel are important parameters for achieving stable combustion and reducing emissions. Results of engine vibration, performance, combustion, and emissions are markedly affected by the premixed mixture percentage and temperature.

3.1 Engine Vibrations

Experimental measurements of the vibration data were obtained by collecting information from knock sensor. We can determine whether the engine is operating steadily or experiencing abnormal combustion by looking at the vibration profile. The fluctuation in vibrations for the conventional mode and PCCI mode works with various premixed diesel fuel proportions and with various premixed temperatures under all the engine loads is indicated at Figure 5. The engine vibrations demonstrated how to detect the knocking occurrence by the appearance of higher values for the abnormal vibrations. From the figure, it is clear that for 110 ºC the peak vibration had the lowest value. In other words, the engine operation improved and became smoother because of efficient combustion as a result of more homogeneity of premixed mixture with suitable A/F ratio. With the increasing of the premixed mixture temperature (115 & 120 ºC), the peak vibration recorded higher values than conventional mode which indicates to higher tendency to engine knocking, as a result of advancing combustion timing with respect to top dead center (TDC) due to more homogeneity of mixture in addition to higher premixed temperature. Owing to this, we stopped the experiments at 120 ºC to avoid knocking. While (100 & 105 ºC) almost had the same peak vibration value as the conventional mode.

3.2 Engine Performance

3.2.1 Brake thermal efficiency.

Brake Thermal Efficiency of the engine is estimated by the following equation 5. At all engine loads, the variance in BTE has been investigated. Figure 6 clearly illustrates the effects of varying temperature and the fraction of the premixed mixture during combustion. With an increase in engine load, brake thermal efficiency raised, as is evident throughout the whole operation condition of the engine. At full engine load, the higher improvement of BTE is noticed due to increased useful power demand at full engine load that decreased heat loss from cooling air and exhaust gas. It is comprehended from the figure that for PCCI mode, (a) at the 4-cases (15% at 100 ºC - 20% at 105 ºC - 30% at 110 ºC - 30% at 115 ºC) with increasing fuel vapour induction and increasing premixed temperature, the BTE increased due to the premixed homogenous mixture formation compared to the CDC mode. (b) 30% at 110 ºC has produced maximum reward in BTE by 5.4% at full load in contrast to CDC mode due to better quality of premixed fuel and more homogeneity of mixture. (c) 30% at 120 ⁰C has demonstrated reduction in BTE by 1.3% at full load in contrast to CDC mode owing to the creation of a rich mixture, which results in extra fuel being left unmixed and a correspondingly decreased level of BTE.

$$\mathrm{BTE} = \frac{\text{Brake power (Kw)}}{\sum_{i=1}^{i=n} \dot{m}_i * \mathrm{LHV}_i} \quad (5)$$

3.2.2 Brake specific fuel consumption.
Brake specific fuel consumption (BSFC) is the amount of fuel consumed to produce one unit of brake power in one unit of time [31]. BSFC value in g/kWh is calculated by applying the following equations 6 & 7. Figure 7 shows the fluctuation of BSFC and break power. From the figure, it is obvious that we have four cases for PCCI mode. In all the tested cases with increasing fuel vapour induction with increasing premixed temperature, the BSFC decreased in contrast to the CDC mode owing to the premixed homogenous mixture formation leading to better combustion which producing more power. Meanwhile, the $^0m_{\text{direct injected}}$ is reduced to keep the engine speed at 1500 rpm. At the case of 30% fuel vapour induction with 110 ⁰C has shown minimum BSFC at all engine loads compared to CDC mode due to better quality of premixed fuel and more homogeneity of mixture results in minimum $^0m_{\text{direct injected}}$. However, at 30% fuel vapour induction with 120 ⁰C has shown maximum BSFC in all engine loads compared to CDC mode due to increasing the fuel vapour volume while the suction volume of engine is constant. This causes the amount of fresh air flowing into combustion chamber to be reduced in addition to the $^0m_{\text{direct injected}}$ cannot be reduced below a specific value as it has a minimum value. Owing to these aspects, the mixture became rich, and the engine consumes more fuel while keeping increasing premixed temperature.

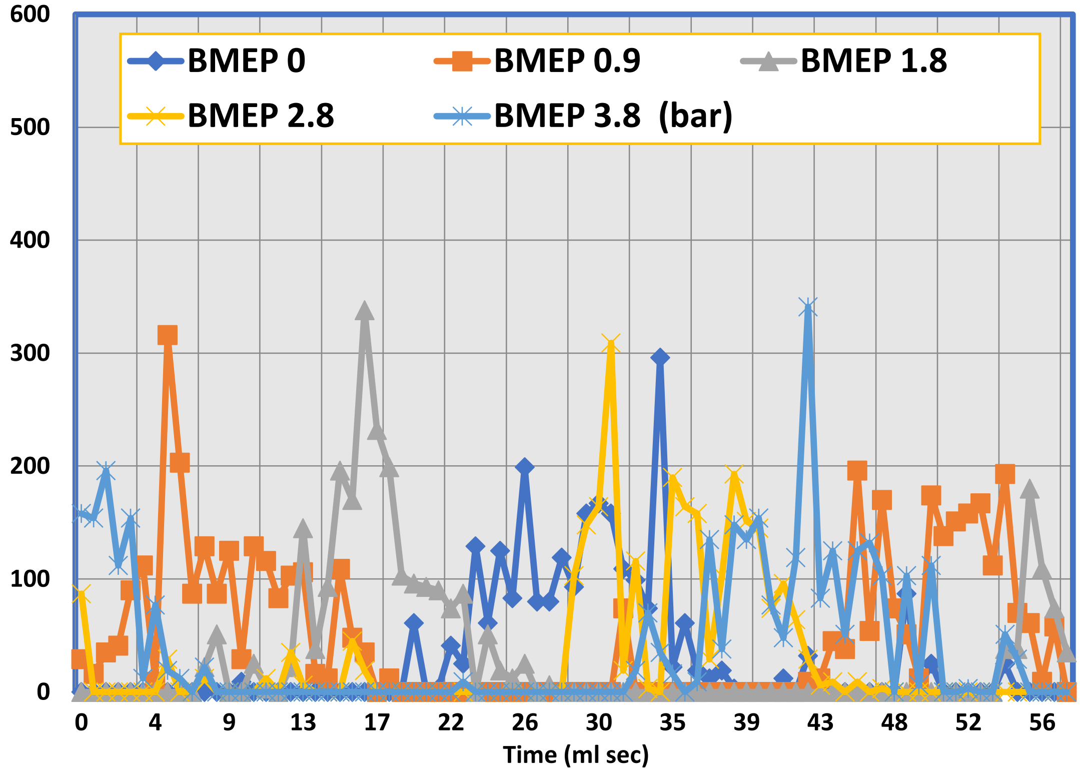
$$\mathrm{BSFC} = \frac{^0\mathrm{m_f}}{\mathrm{P_b} \times 3600} \quad (6)$$

$$^0\mathrm{m_f} = {}^0\mathrm{m}_{\text{direct injected}} + {}^0\mathrm{m}_{\text{premixed}} \quad (7)$$

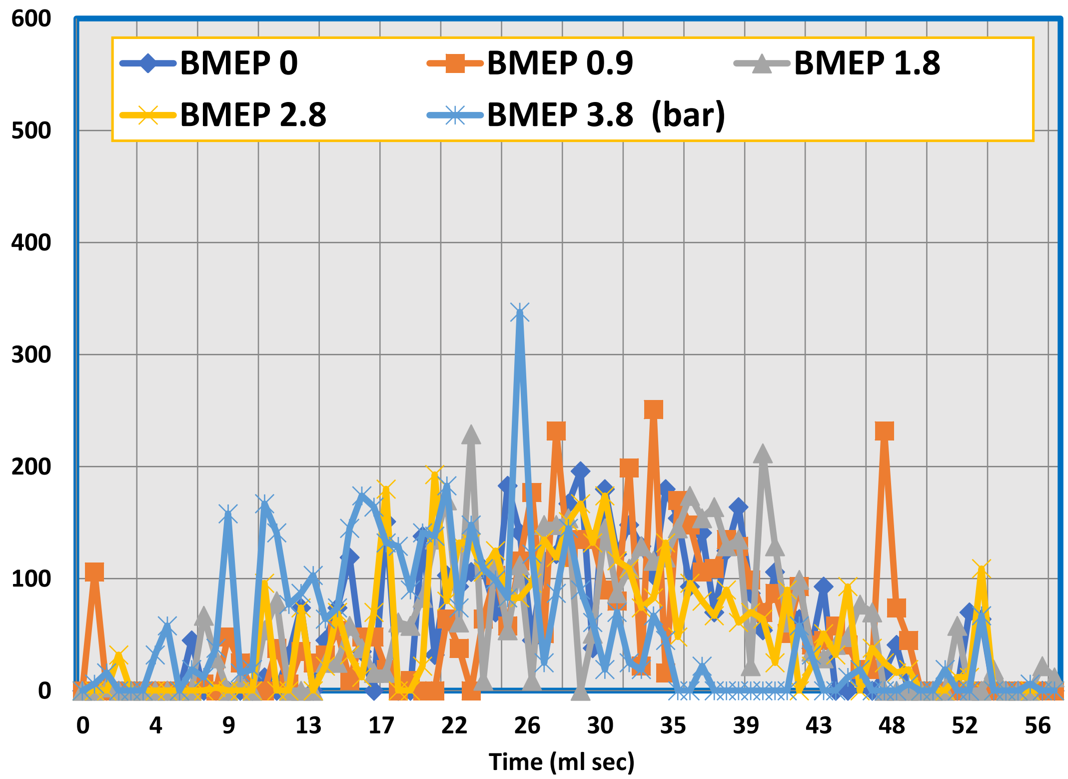
Where, P_b is the brake power (kw), ^0m_f is the total fuel mass flow rate used to produce power (g/sec), $m_{\text{direct injected}}$ is the rate at which fuel is directly injected into combustion chamber, $^0m_{\text{premixed}}$ is the mass flow rate of premixed fuel vapour with air.

3.2.3 Exhaust gas temperature.

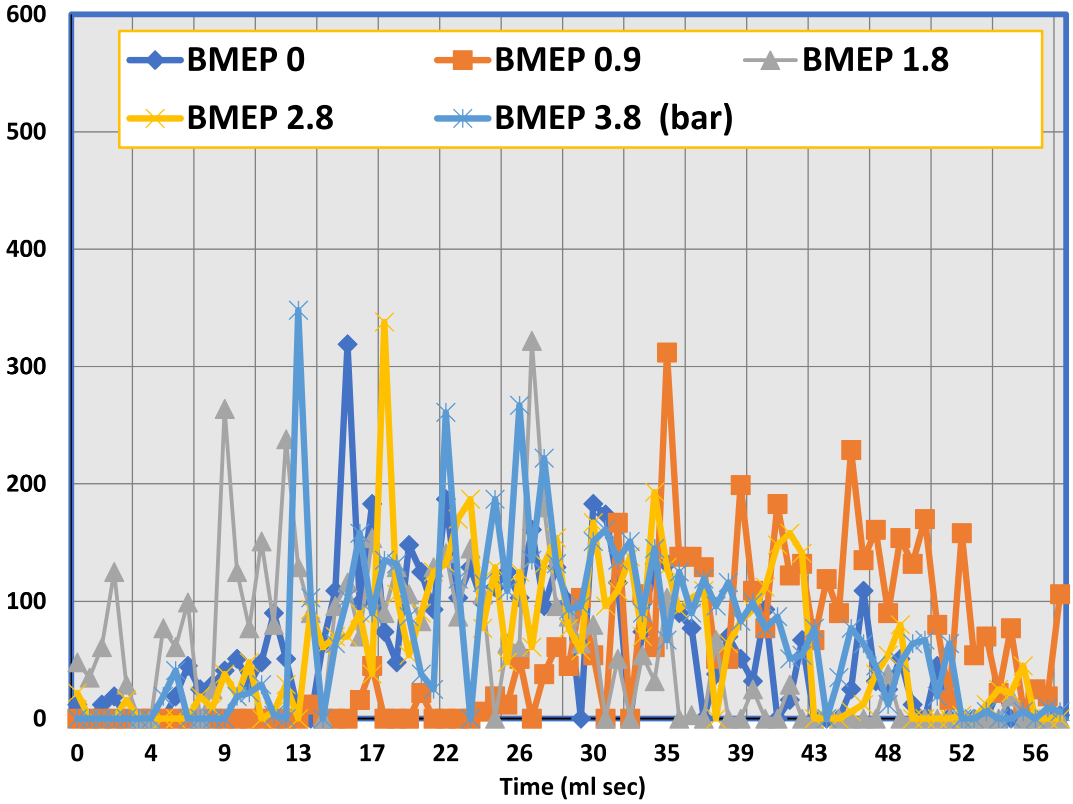
Exhaust gas temperature (EGT) is the temperature of products in the outlet port results after the combustion process of charge in the engine cylinder and it is essential in determining the combustion temperature and the heat loss from the exhaust gases [32]. The variation of EGT and break power is indicated in Figure 8. When the engine is operating at full capacity, the EGT increases as the engine load rises. In the CDC mode and the PCCI mode, the amount of fuel combusted increases as engine load increases, resulting in a rich mixture that promotes the combustion and raises exhaust gas temperature. It is comprehended from the figure that for PCCI mode at all operation condition except (30% fuel vapor induction with 120 ⁰C), there is a decrement in EGT when with rising fuel vapour induction and increasing premixed temperature. This is due to the better homogeneity of the mixture as a result of external mixing of evaporated diesel with the air that prevents fuel accumulation in cylinder, contributing towards controlled combustion and help in the reduction of diffusion phase combustion. 30% fuel vapour induction with 120 ⁰C has indicated maximum EGT due to rich mixture formed results in increase the amount of fuel burned during expansion stroke during the final stages of combustion. Hence, the diffusion combustion is higher.



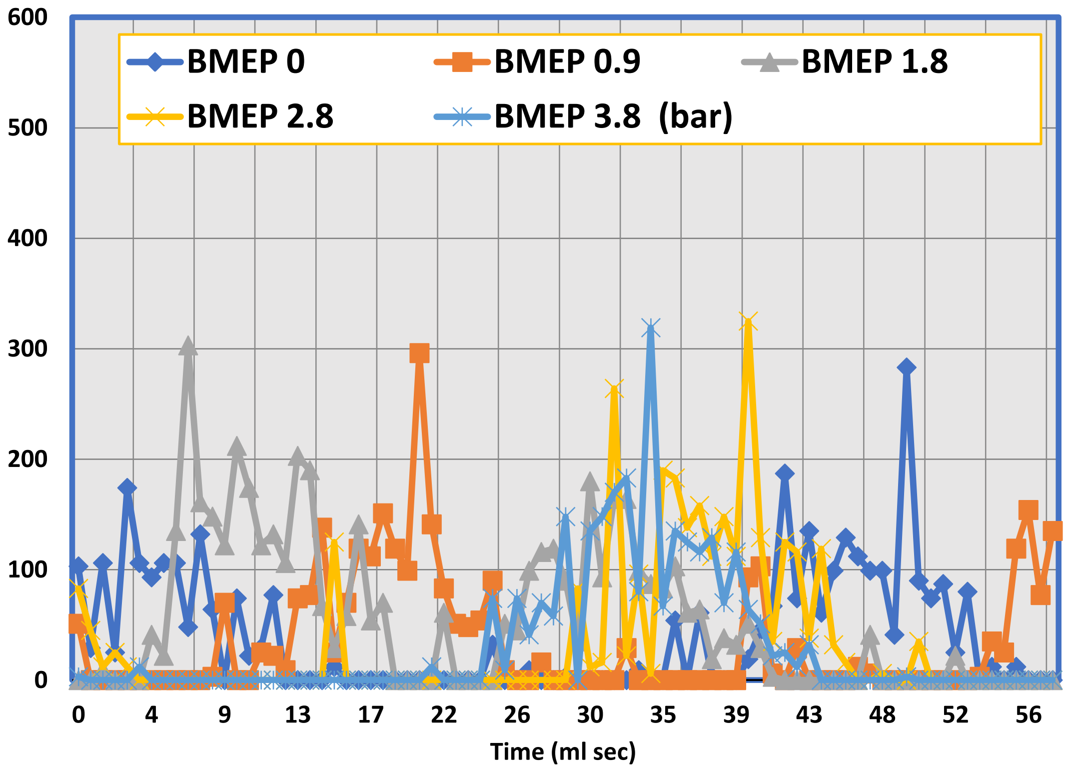
(a) CDC mode



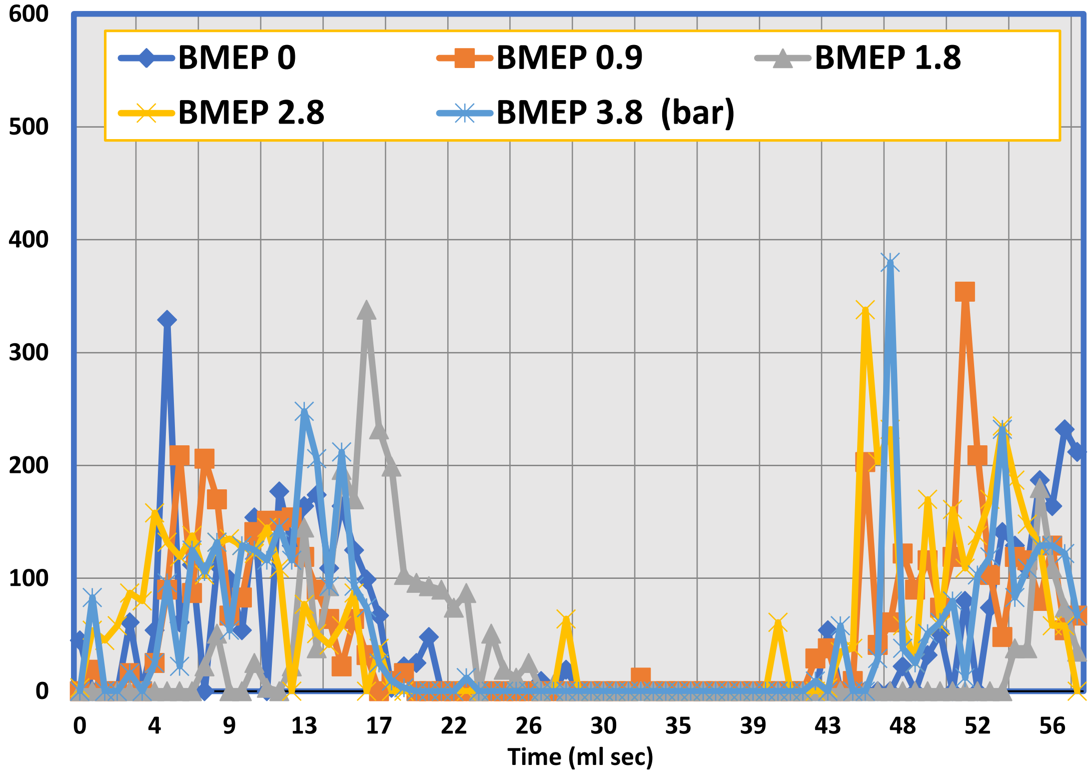
(b) 15% at 100 ºC PCCI



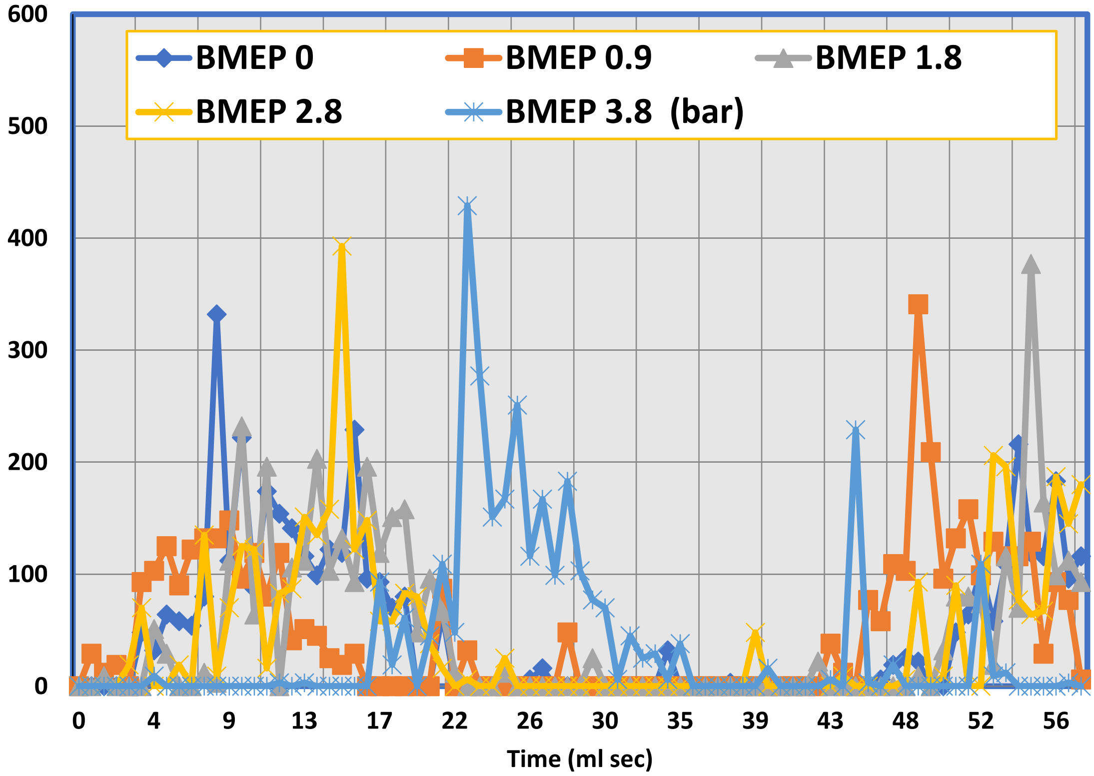
(c) 20% at 105 ºC PCCI



(d) 30% at 110 ºC PCCI



(e) 30% at 115 ºC PCCI



(f) 30% at 120 ºC PCCI

Figure 5. The variation in engine vibrations under various conditions

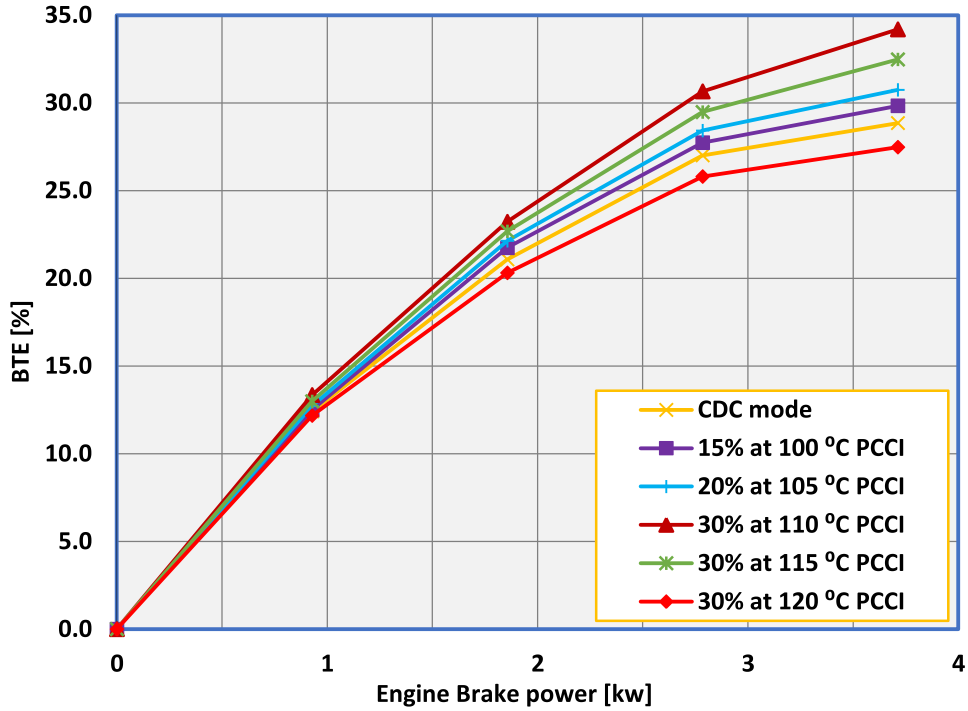


Figure 6. Effects of premixed proportion and temperature on PCCI BTE under various loads

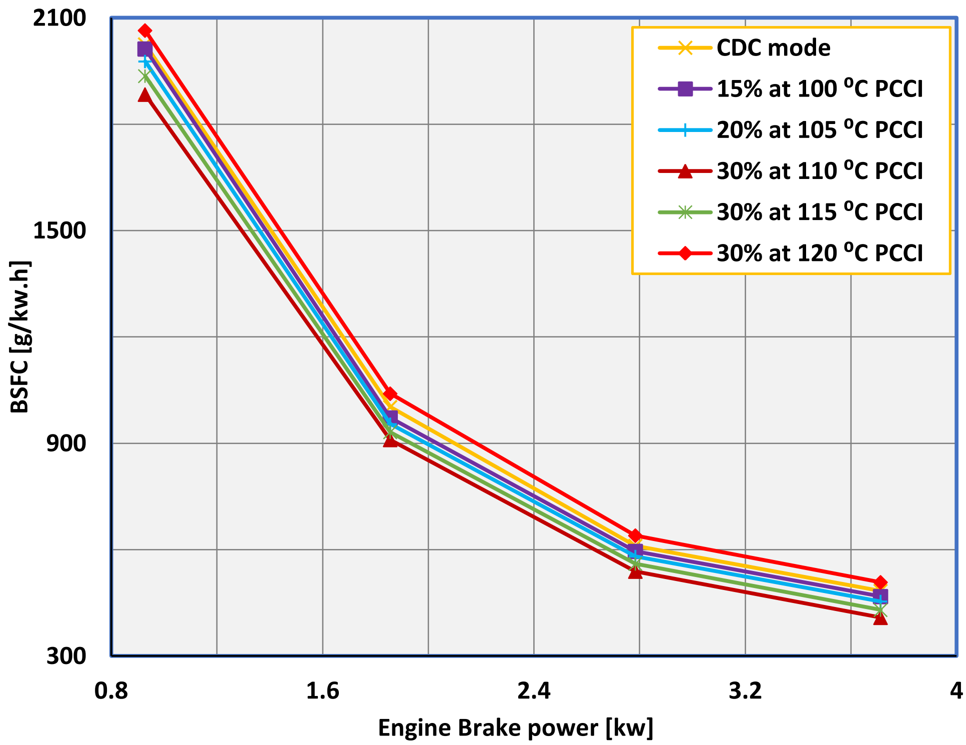


Figure 7. Effects of premixed proportion and temperature on PCCI BSFC under various loads

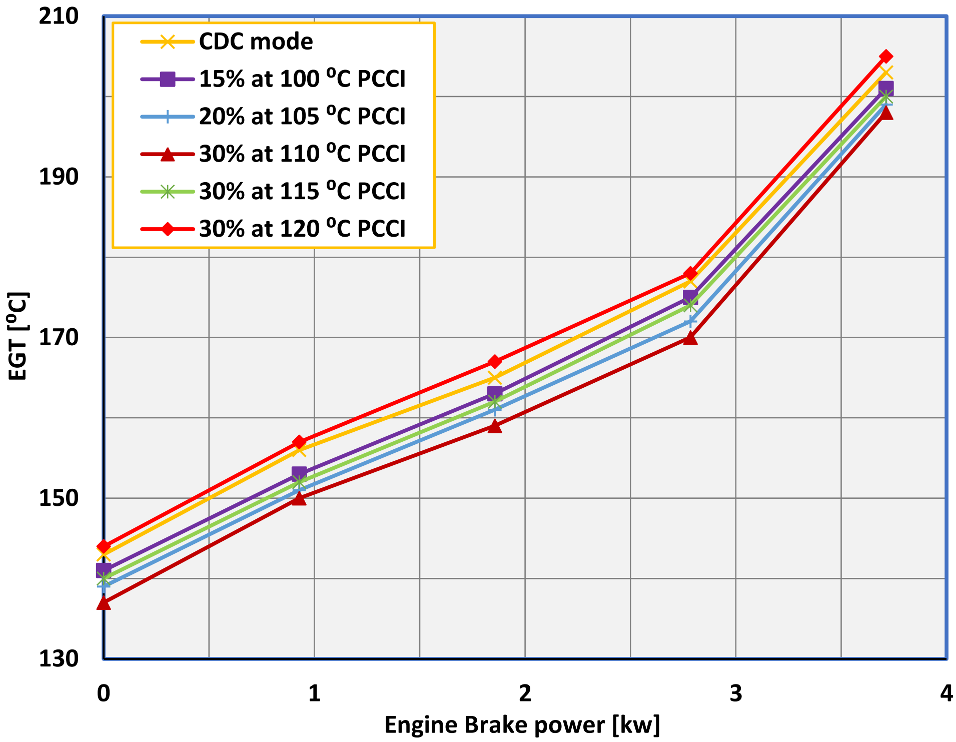


Figure 8. Effects of premixed proportion and temperature on PCCI EGT under various loads

3.3 Engine Emissions

3.3.1 NO_X emission.

The NO_X emission formation rate is greatly dependent on temperature. The variation of NO_X emission and break power is indicated in Figure 9. The figure makes it very clear that the NO_X concentration rises as engine load increases through all engine-operating conditions. In CDC, the diffusion phase of combustion dominates, and insufficient fuel atomization caused a heterogeneous mixture that produced a rich zone. These factors caused the peak cylinder temperature to be high, which increased the formation of NO_X emissions. [33]. In PCCI mode, the homogeneous air-fuel mixture formation results from the induction of vapor fuel that preventing the formation of pockets of rich mixture and ensuring that combustion occurs largely during the pre-mixed phase and these factors lower the combustion temperature hence, NO_X emission is reduced in the PCCI's entire operating capacity. The NO_X emission is reduced at 30% fuel vapor induction with 110 ºC at all engine load because of improved combustion efficiency and decrease in combustion temperature. The minimum NO_X emission is achieved at 30% fuel vapor induction with 120 ºC because of decreasing combustion temperature despite the formation of over-rich air-fuel mixture.

3.3.2 CO emission.

Carbon monoxide (CO) is produced from incomplete combustion. As a result, CO emissions are significantly influenced by the homogeneity of the combustible mixture and the temperature inside the cylinder. In Figure 10, the relationship between CO emission and breaking power is demonstrated. It is comprehended according to that figure the decrease in CO with the rise in engine load is observed in the whole engine operating conditions as the in-cylinder temperature increases. At first, with premixed ratio of 15% at 100 ºC has demonstrated higher CO compared to the CDC mode due to over-rich mixture formation. Then, with premixed ratio of 20% at 105 ºC has demonstrated lower CO compared to the CDC mode due to lean mixture formation. With increasing premixed ratio and temperature 30% at 110 ºC, the minimum CO emission is obtained due to good quality of mixture and suitable combustion temperature. However, for 30% at 115 ºC keeping increasing temperature with the same premixed ratio but the DI fuel quantity kept constant these aspects create a rich mixture and so CO is higher compared to the CDC mode. With premixed ratio of 30% at 120 ºC has demonstrated the maximum increment in CO compared to CDC mode due to the formation of an excessively rich mixture in addition to reducing the volume of fresh air flowing into combustion chamber results in the absence of O_2 in the reaction area. From the explanation above, it can be seen that the suction of intake air was being hindered by the fuel vapour's excess temperature.

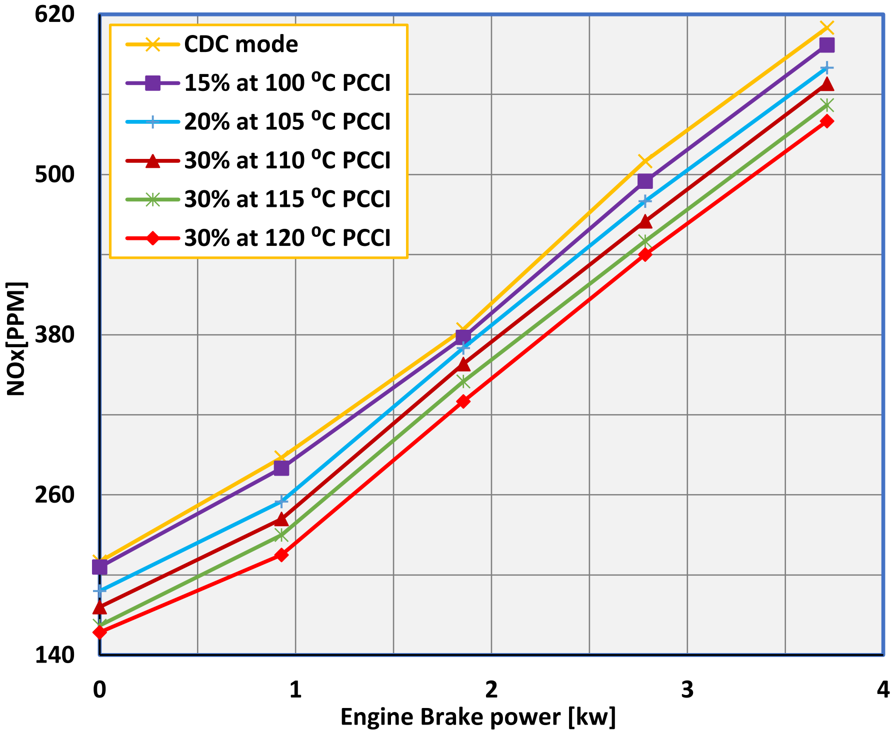


Figure 9. Effects of premixed proportion and temperature on PCCI NO_X emissions under various loads

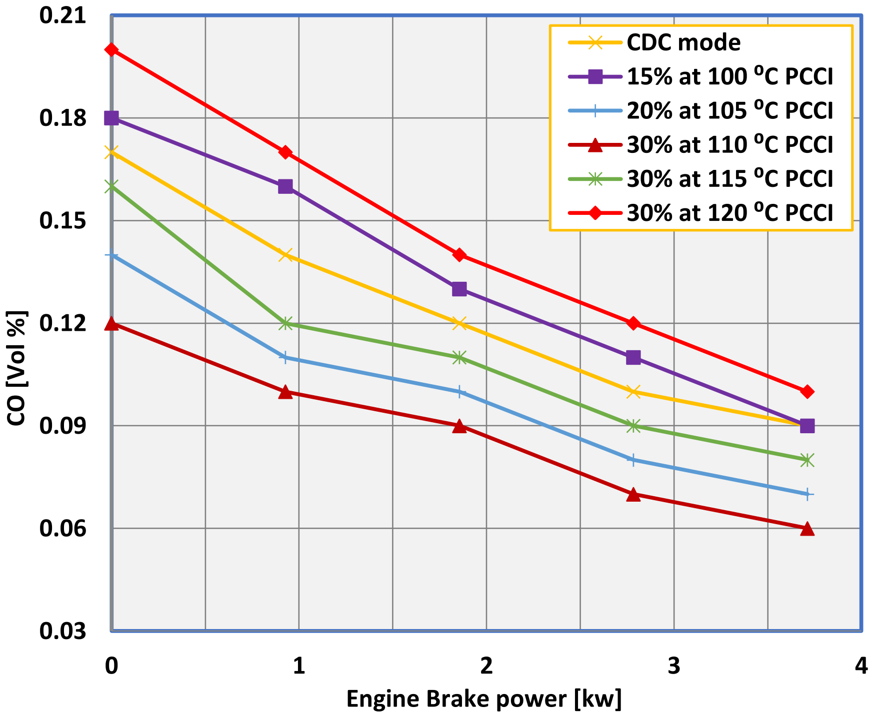


Figure 10. Effects of premixed proportion and temperature on CO emissions of PCCI under various loads

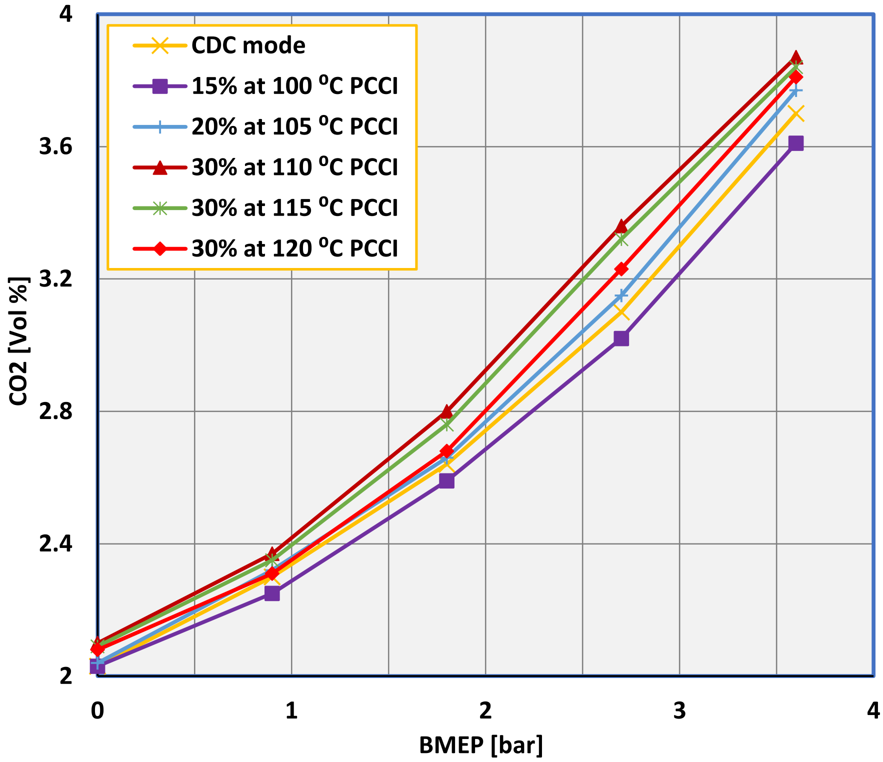


Figure 11. Effects of premixed proportion and temperature on PCCI CO_2 emissions under various loads

3.3.3 CO_2 emission.

Complete combustion usually produces carbon dioxide (CO_2) as a normal product. In ideal case, Only CO_2 and water (H_2O) should be produced during the complete combustion of hydrocarbon fuels. The variation of CO_2 emission and break power is demonstrated in Figure 11. The figure demonstrates that the increment in CO_2 with the increase in engine load is noticed in all operating conditions of engine. As seen in Fig. 11. In PCCI mode, the homogeneous air-fuel mixture formation results from the induction of vapor fuel that prevent the creation of rich mixture pockets and the combustion occurring mostly during the pre-mixed phase these aspects results in occurrence of complete combustion hence, CO_2 emission is increased in the PCCI's entire operating range compared to CDC mode. Except for 15% fuel vapour induction with 100 ºC CO_2 emission decreased because of incomplete combustion as a result of rich air-fuel mixture formation.

3.3.4 HC emission

In Figure 12, the relationship between HC emission and breaking power is demonstrated. It is comprehended from the figure that the decrease in HC with the increment in engine load is observed in all operating conditions this is because of the higher the in-cylinder temperature and fuel-combustion efficiency. CDC showed high HC emission because of incomplete combustion as a result of poor of fuel-air mixing and insufficient atomization. At first, 15% fuel vapour induction with 100 ºC has shown higher HC in contrast to the CDC mode as a result of incomplete combustion of over-rich mixture formation and fuel droplets in crevice volume due to longer time between suction and combustion starting. Then, 20% fuel vapour induction with 105 ºC has shown lower HC compared to the CDC mode due to better mixture formation. With increasing premixed ratio and temperature, 30% at 110 ºC the minimum HC emission is obtained due to good quality of mixture that leads to suitable combustion temperature and higher premixed temperature reduce fuel droplets in crevice volume. However for 30% at 115 ºC keeping increasing temperature with the same premixed ratio is good to prevent fuel droplets in crevice volume but the DI fuel quantity kept constant these aspects creates a rich mixture and so HC is higher in contrast to CDC mode as a result of the In-cylinder wall-wetting. With premixed ratio of 30% at 120 ºC has shown the maximum increment in HC compared to CDC mode due to higher In-cylinder wall-wetting because of the formation of over-rich mixture and decreasing the volume of fresh air entering to the combustion chamber results in the absence of O_2 in the reaction area that leads to poor combustion.

3.3.5 Smoke emission

The main Components of smoke opacity are carbonaceous soot due to combustion and absorbed some organic HC, and sulfates. The variation of soot emission and break power is observed in Figure 13. It is comprehended from the Fig. 13 that the increment in soot with the increase in engine load is noticed in all operating conditions because of the formation of richer mixture that causes the creation of soot and unburned hydrocarbons. With PCCI mode, the smoke opacity is at lowest level, whereas CDC mode has the highest. PCCI mode decreases smoke opacity percentage due to better mixture formation as the premixing of homogeneous air–vapor mixture prevents the creation of rich fuel-air mixture. Due to incomplete combustion brought on by burning more fuel in the combustion chamber with insufficient oxygen, a greater amount of undesirable intermediate products is produced in rich mixture pockets. In PCCI mode, the significant part of the combustion chamber generally has lean air–vapor mixture. Additionally, low levels of soot in the exhaust are noticeable in the incomplete combustion region. Due to the absence of excess fuel and so, these aspects generally result in less opacity in smoke when compared with CDC mode. The ideal case of combustion of smoke opacity is at 30% fuel vapour induction with 110 ºC due to better quality of air-fuel mixture and sufficient combustion temperature, which reduce unburned hydrocarbon. For 30% fuel vapour induction with 120 ºC, at low and medium load it produces lower smoke opacity compared to CDC mode. Despite it has rich air- fuel mixture but the higher temperature of combustion promote the partially oxidation of carbon to CO and so produce lower smoke, while, at high load is observed higher smoke opacity compared to CDC mode due to more over rich air- fuel mixture.

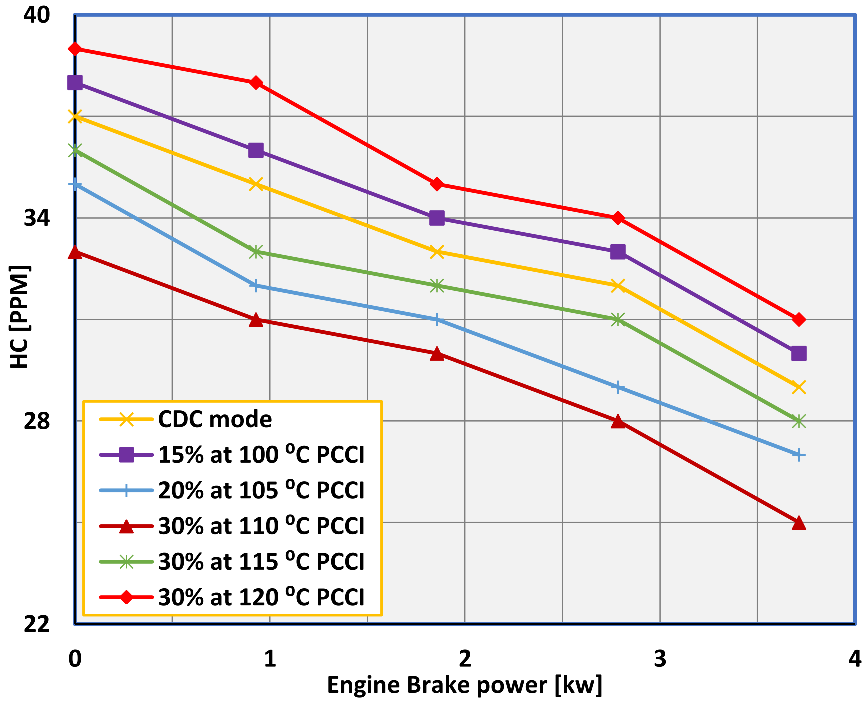


Figure 12. Effects of premixed proportion and temperature on PCCI HC emissions under various loads

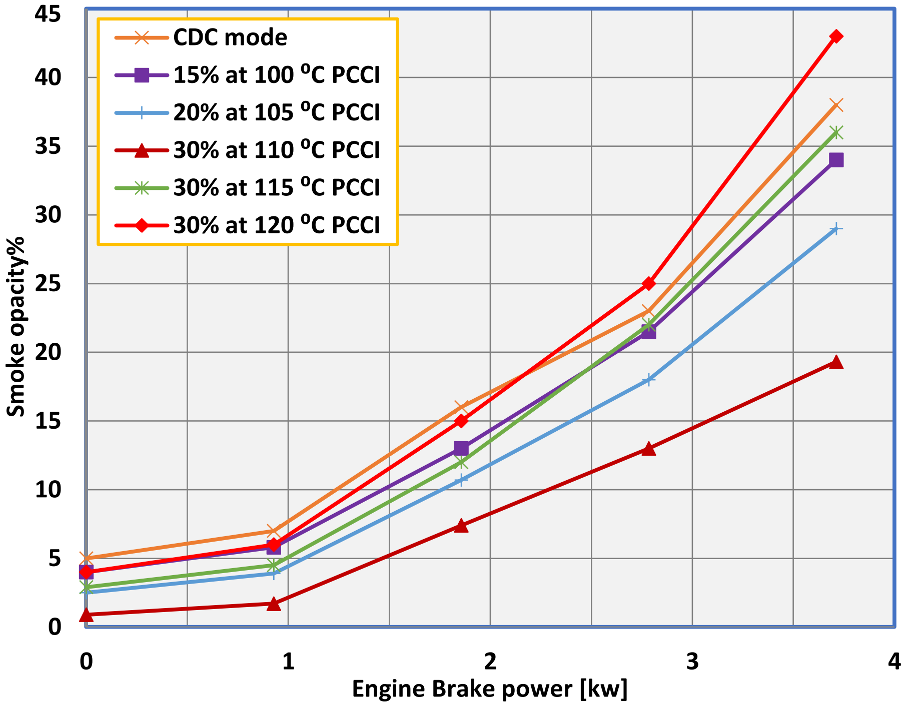


Figure 13. Effects of premixed proportion and temperature on PCCI smoke opacity under various loads

4. Conclusion

In this research, experiments were conducted to compare PCCI engine combustion, performance, and emission characteristics with CDC engine characteristics at the different engine loads. An external air-fuel mixture formation strategy mounted using a fuel vaporizer unit for external mixture preparation. The results showed that the premixed mixture percentage and temperature play an important role in the combustion process. The results demonstrated that temperature and the proportion of the premixed mixture are extremely important factors that influences to the combustion process. The engine characteristics were studied, and the following conclusions can be draw:

- In order to guarantee the system's efficient performance, a comparative study of the design approach used was conducted before using a new fuel vaporizer unit.
- The PCCI engine has demonstrated significant potential for use in CI engines. It offers a solution to the primary diesel engine concerns of NO_X and smoke emissions while maintaining or increasing BTE and reducing CO and HC emissions.
- PCCI mode at 30% fuel vapour induction with 110 °C show the perfect results in contrast to the conventional engine, as NO_X, HC, CO, and smoke opacity emissions and BSFC decreased in addition to an increment in BTE. This could be due to better homogeneity of the mixture with achieving good quality of mixture and suitable combustion temperature.
- PCCI mode with premixed ratio of 30% at 120 °C shows the lowest results compared to the conventional engine, as HC, CO, and smoke opacity emissions and BSFC rose along with a reduction in BTE. This might be caused by the creation of an overly rich mixture in addition to reducing the volume of fresh air flowing into the combustion chamber results in absence of O_2 in the reaction area however, better homogeneity of the premixed air- fuel vapor mixture.

Overall, PCCI mode with increasing the proportion and temperature of premixed air-vapour fuel is very promising and essential for providing both high/the same efficiency and reducing HC, CO, NO_X, and smoke emissions compared to CDC mode as result of better homogeneity of the charge.

5. Future research direction

Future research on the following critically intriguing subjects has been recommended as a result of the discussion of fuel vaporizer technology implications on PCCI engine emissions, performance, and combustion characteristics that was just mentioned.

- For controllability, reliability, and a huge working capacity in combustion, using an electrical injector is suggested as an effective method. The suitable fuel injection pressure variation will be achieved as a comparison with the mechanical injector. In addition, the ability of controlling the amount of fuel those is directly injected into the cylinder according to the premixed fuel vapour in the mixing chamber and avoids knocking occurrence can be reached.
- Using fuel vaporizer technology, a turbocharger is considered a better method to increase the volume of fresh air flowing into the combustion chamber and could avoid the formation of over-rich mixture, which is beneficial because it decreases fuel impingement and trapping into crevice volumes, which helps to minimize HC and CO emissions.
- Renewable fuels could remove some restrictions on the PCCI mode of combustion's stability and reliability, provide additional control flexibility for the PCCI, and help in extending the operational range by simplifying the control strategy.

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7. Nomenclature & Abbreviations:

BMEP	Brake mean effective pressure	LNTs	Lean NO_X traps
BSFC	Brake specific fuel consumption	LRF	Fuel with low reactivity
BTE	Brake thermal efficiency	LTC	Low temperature combustion
CDC	Conventional Diesel Combustion	NO_X	Oxides of nitrogen
CO	Carbon-monoxide	PCCI	Premixed Charge Compression Ignition
CO_2	Carbon-dioxide	PM	Particulate-matter
DI	Direct injection	PPM	Particulate per million
DPF	diesel particulate filter	RCCI	Reactivity Controlled Compression Ignition
EGT	Exhaust gas temperature	rpm	Revolution per minute
HC	Unburned-hydrocarbon	SCCI	Stratified Charge Compression Ignition
HCCI	Homogenous Charge Compression Ignition	SCR	Selective catalytic reduction
HRF	High reactivity fuel	TDC	Top dead center
ICEs	Internal combustion engines	RON	Research octane number

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