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## **Reactivity Controlled Compression Ignition (RCCI) of Gasoline-CNG Mixtures**

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Additional information is available at the end of the chapter

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### **Abstract**

Reactivity controlled compression ignition (RCCI) is a dual fuel combustion method that relies on the significant difference in reactivity of the fuels involved. RCCI had a low performance at high engine speed due to its high tendency on knocking and high pressure rise rate. Therefore, this study investigates the effect of the fuel stratification on the RCCI combustion and its extended to the interaction of two low reactive fuels, gasoline and compressed natural gas (CNG), in the RCCI combustion system. The investigation was experimentally performed on a single cylinder engine and constant volume chamber. The stratification was created by varying injection timing in the engine by injecting CNG at 80° and 120° before top dead center (BTDC) and varying injection gap in the constant volume chamber with the gaps between two fuel injection timing were varied between 0 ms to 20 ms. The results in the engine experiment show that proportions of gasoline and CNG and degree of stratification of CNG were found to be effective means of combustion control within certain limits of engine load and HC and CO emissions could be significantly reduced. While in constant volume chamber it has a significant effect on the combustion phasing. Stratified mixture produces shorter combustion duration while homogeneous mixture produces longer duration.

**Keywords:** reactivity charge compression ignition, gasoline, compressed natural gas, stratification, fuel injection

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## 1. Background

Perhaps the most graceful invention by humankind that ever had a greater impact on society, the economy, and the environment is the reciprocating internal combustion engine, in general, called the IC engine. For decades, this magnificent invention proved to play a vital role in the automobile system, used almost exclusively today. There are two types of internal combustion engines: spark ignition (SI) and compression ignition (CI). For the last decades, rapid improvements in the efficiency have been achieved on both types of IC engine.

Unfortunately, at present, there is a pressing need to develop advanced combustion engines that maximize the engine efficiency and totally mitigate the exhaust pollutants. Profound understanding of both SI and CI combustion principles has been achieved during the last decades to improve the efficiency and reduce the emissions. The conventional SI combustion, which is characterized by flame propagation in near-stoichiometric homogeneous mixtures, produces very low exhaust emissions in combination with a three-way catalytic converter but has a relatively low thermal efficiency, which is its main drawback.

CI combustion, on the other hand, that is characterized by the autoignition of a lean fuel-air mixtures, has a very high thermal efficiency; yet, it has very high soot and  $\text{NO}_x$  (nitrogen oxide) emissions. Diesel engines typically produce lower carbon monoxide (CO) and unburned or partially burned hydrocarbons (HC) compared to the gasoline engines. However,  $\text{NO}_x$ , which comprises nitric oxides and nitrogen dioxides, in addition to particulate matter (PM) or soot, is significant pollutant from diesel engines, which require proper control strategies as they pose adverse health and environmental impacts.

The engine technologies are advancing at a significant rate during the last decades. The engine technology development timeline is depicted in **Figure 1**. The engine performance was the main priority in the first era of engine developments. Technologies such as turbocharger, port fuel injection, high compression ratio, direct fuel injection, and engine lightweight material were the technologies that are focusing on increasing the engine power output to its maximum capabilities. This early era was driven by the abundant amount of fuels relative to its low demand as the automobile was still an exclusive technology.

The second era of the engine technology development was mainly driven by increasing concern about exhaust emissions and efforts in achieving low fuel consumption. In the earlier technologies, efficiency was improved and the engine downsizing was the primary target of the engine development. Homogeneous charge compression ignition (HCCI) engine is one of the promising alternatives in order to achieve these objectives.

The spark ignition engine (SI) and compression ignition engine (CI) are the established engine technologies, and each have their advantages and disadvantages. SI has a faster response and low emissions yet low efficiency, while CI offers high efficiency and low fuel consumption yet higher emissions and slower response. This makes the development of the SI and CI engine followed different approaches. Nevertheless, the main objective of an engine is mainly to achieve high performance, high efficiency with low emissions. In order to achieve this goal,

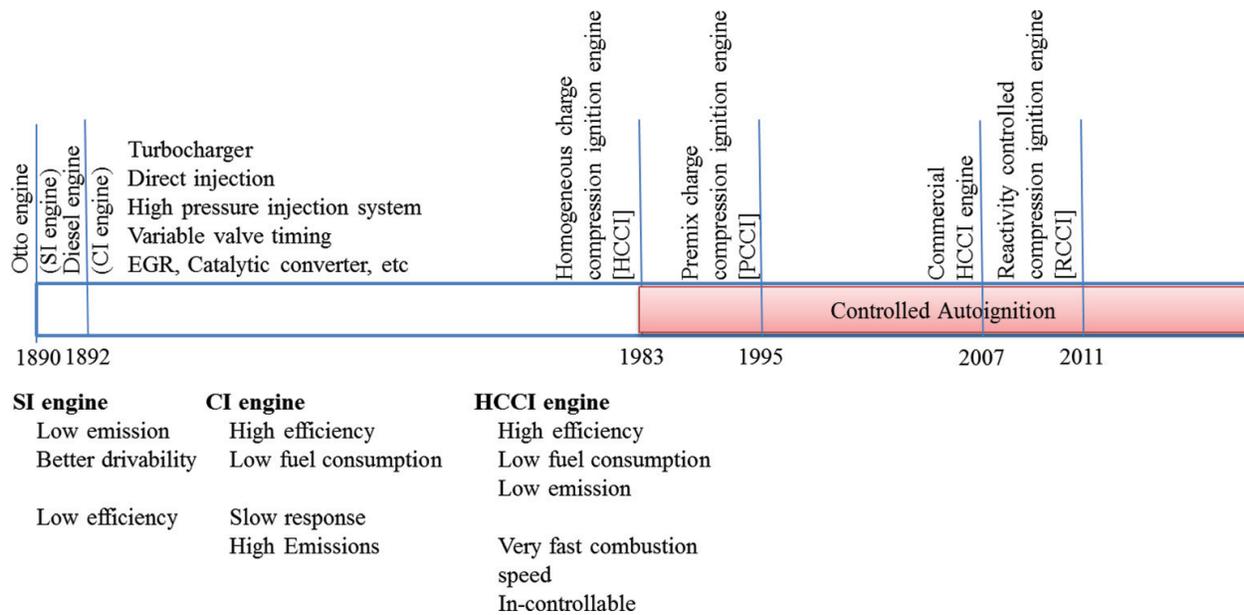


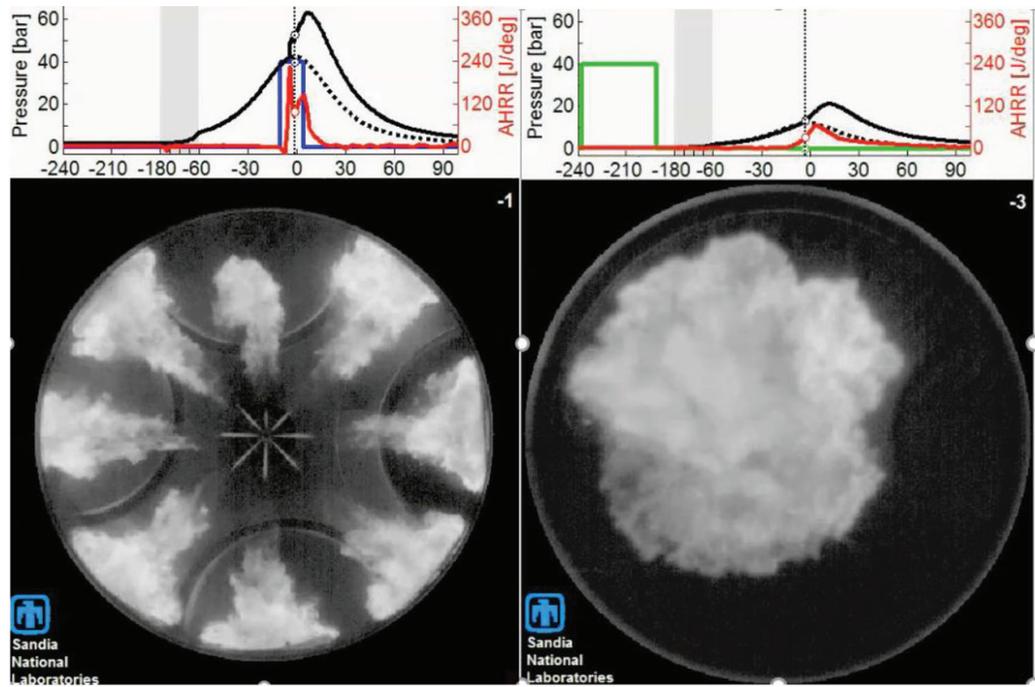
Figure 1. Engine technology development.

HCCI was introduced in 1983. Major obstacle in HCCI is its rapid heat release rate with a very high maximum pressure, which is detrimental to engine structure. In the effort of controlling the combustion in HCCI engines, technologies such as exhaust gas recirculation (EGR) and variable valve timing (VVT) are utilized to improve the controllability of HCCI combustion, which is used in the commercial HCCI engine such as skyActive technology (Mazda) and diesOtto engine (Mercedes).

On the other hand, in an effort to find the best method for controlling HCCI combustion, the HCCI has evolved into various types of controlled autoignition-based engine (CAI) such as premixed charge compression ignition (PCCI) in 1995 [1] and reactivity controlled compression ignition (RCCI) engine in 2011 [2]. These methods are proven to be able to improve the operating range of CAI engine. However, there are no established methods that are proven to be effective in controlling CAI engine.

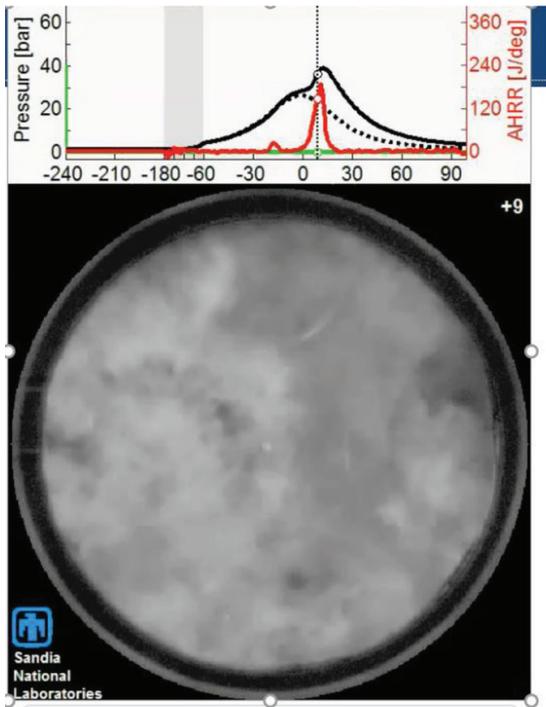
Figure 2 depicts the differences in each combustion system control strategy. The CI system (Figure 2 (a)) creates a very high pressure inside the combustion chamber, and the fuel is directly injected to combust. SI system (Figure 2b)), on the other hand, forms a homogeneous mixture either by direct injection or by port injection before the mixture is ignited by a spark, whereas the HCCI engine (Figure 2 (c)) produces a homogeneous mixture by injecting the fuel during the intake stroke, and the mixture is autoignited due to the compression. HCCI combustion combines the best features of gasoline and diesel engines to produce diesel-like power and efficiency while maintaining gasoline-like soot free emissions within certain operating limits.

The engine performance was limited to the part load conditions and controlling the combustion process was very problematic due to the autoignition being highly dependent on the



(a)

(b)



(c)

Note:

- Injection event
- Heat release rate (HRR)
- Pressure

**Figure 2.** Combustion control strategies (a) compression ignition (CI), (b) spark ignition (SI), and (c) homogeneous charge compression ignition (HCCI). Source: <http://crf.sandia.gov/combustion-research-facility/engine-combustion/fuels/>.

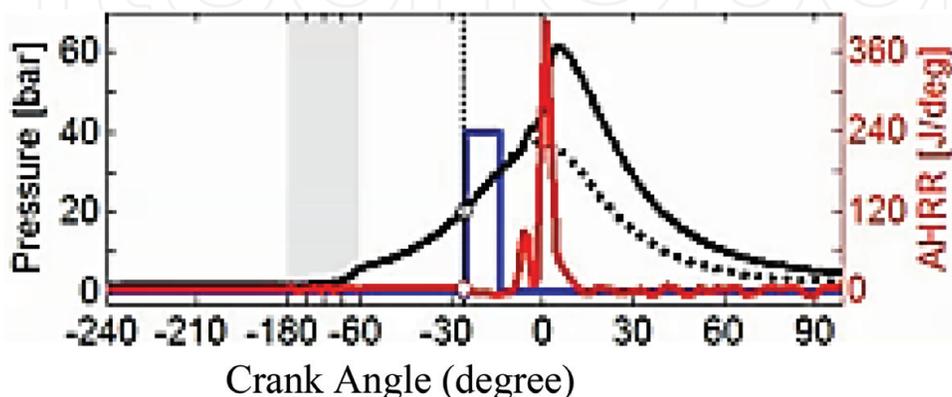
temperature, pressure, and mixture composition inside the combustion chamber. In its development, HCCI was limited by the narrow operating range and the unpredictable combustion delay and behavior. The maximum pressure generated was very high but produced a low mean

effective pressure due to the short combustion duration. Many methods and possibilities were proposed to control HCCI engines. In this process, the paradigm of creating an autoignition process from homogeneous charge is shifting to the method of controlling autoignition. The first development is the premixed charge compression ignition (PCCI). This concept was introduced by Aoyama et al. [1]. Gasoline was subjected into diesel-like environment with high compression ratio 17.4:1. A port injection method was used to create the premixed charge in the combustion chamber where the fuel is injected very close to the intake valve closing time as shown in **Figure 3**.

The combination starts with gasoline and diesel as the low-reactive fuel and high-reactive fuel, respectively. It was found that RCCI combustion was able to operate in a wide range of engine loads with near-zero NO<sub>x</sub> and soot emissions, accepted pressure rise rate and high indicated efficiency. However, RCCI still could not achieve high load operation with power outputs comparable to CI engine. The combustion behavior of RCCI is somewhat still unpredictable. Further investigation on the important parameters in RCCI combustion control will improve the understanding of the combustion process of RCCI that leads to better control of the process and better engine output.

The premixed charge compression ignition (PCCI), reactivity charge compression ignition (RCCI), and spark-assisted HCCI are some of the established methods of controlling the autoignition process in an engine. All of these methods are categorized as CAI engines. Regardless of the limitations of the controlled autoignition (CAI)-based combustion system, CAI offers high efficiency [4], low fuel consumption [5], and low emission [6], which are the main aims of future engine development. The attributes that differentiate SI, CI, HCCI, PCCI, and RCCI are shown in **Table 1**.

These earlier works are the basis of the controlled autoignition engine concept. The primary focus of the CAI combustion concepts is identifying the relevant influencing parameters as well as control parameters of this system to widen the operating range and improve the efficiencies. The need for a thorough understanding of the CAI combustion process initiates further discussion on the method to control its combustion. The next stage is to determine the engine parameters that have a direct effect on the combustion. As these steps are carefully defined, high efficiency, low fuel consumption, and low emissions internal combustion engines are achievable (**Table 2**).



**Figure 3.** PCCI combustion method [3].

Many researches have been done in the effort of controlling CAI combustion-based system. Agarwal et al. [7] summarize the various types of combustion method and some method in controlling the combustion process. There are various areas that require improvement in order to reshape the combustion and improve the efficiency while still having a low emission. Development of control on ignition timing [8], method in slowing down the heat release rate at high load [9, 10] and development of intake and exhaust manifold for multicylinder engine are among the few area of improvement that have been identified in the area of CAI combustion system. Focusing on the RCCI combustion control method, Li et al. [11] categorized the control by two main categories, fuel and engine management. The fuel management includes two fuel strategies [12, 13] and single fuel strategy with additives, while the engine management

Combustion mode	SI	CI	HCCI	PCCI	RCCI
Fuel	Gasoline-like fuels	Diesel-like fuels	Flexible fuels	Diesel-like fuels	Multifuels
Lambda	1	1.2–2.2	>1	>1	>1
Mixture preparation	PFI, GDI	DI	DI, PFI and DI + PFI	DI, PFI	DI, PFI and DI + PFI
Ignition	Spark ignition	Autoignition	Autoignition	Autoignition	Autoignition
Combustion form	Premixed	Diffusion	Premixed but dominated by chemical kinetics	Premixed	Premixed + stratified
Combustion rate limitation	Flame propagation	Mixing rate	Multipoint or spontaneous	Multipoint or spontaneous	Fuel reactivity
Flame front	Y	Y	w/o	w/o	Y
Combustion temperature	High	Partially high	Relatively low	Relatively low	Relatively high

**Table 1.** Traditional and the controlled autoignition (CAI)-based combustion mode.

Fuel type	Gaseous fuel
Fuel supply system	Direct injection (DI) <ul style="list-style-type: none"> <li>a. High-pressure direct injection (12–18 bars)</li> <li>b. Air-assisted low pressure injection (4–6 bars)</li> </ul>
No. of cylinders bore	Single cylinder (399.25 cc) 88 mm
Stroke compression ratio	132 mm 14:1 (Geometric)
No. of valves	4
Valve timing events	12° BTDC
Intake valve open (IVO) intake Valve closure (IVC)	132° BTDC
Exhaust valve open (EVO)	15° BBDC
Exhaust valve closing (EVC)	10° BBDC

**Table 2.** Summary of specifications of the engine.

include fuel ratio [14], injection strategy [15–18], EGR rate [19], compression ratio [16], bowl geometry [20, 21], stability control, and utilization of two injectors.

This chapter introduces two low reactive fuels, gasoline and CNG, in an RCCI combustion system in order to increase the limit of RCCI engine operation. It introduces a method that has different principals compared to RCCI by introducing combination of low reactive fuels rather than combination of high- and low-reactive fuel. Furthermore, this chapter also introduces the method of RCCI combustion control by varying the stratification by varying the injection timing and gap between two fuel injectors. Two approaches were done to investigate the behavior of gasoline-CNG mixtures in the RCCI combustion system. First approach is experimental testing on a single cylinder engine that was converted to dual fuel engine with gasoline injected at intake port, while CNG is directly injected into the combustion chamber. The stratification was done by varying the injection timing of CNG, while gasoline is kept homogeneous.

The second approach is the combustion testing in a constant volume chamber with both fuels that are directly injected into the combustion chamber. In this setup, the stratifications level in chamber is done with varying the injection gap between two fuel injections.

## 2. Test procedure and equipment

The test procedure and equipment used in both approaches are elaborated in this section.

### 2.1. Engine testing and equipment

The engine used for this experimental study houses the fuel system of direct injection of gaseous fuel and has compression ratio of 14. This engine is a single cylinder water-cooled engine coupled to an electric dynamometer that can be used for starting the engine and measuring the brake torque produced by the engine. **Figure 4** shows a schematic drawing of the engine.

An electric heater is provided to heat the lubricant oil to help warm up the engine faster. A separate control unit controls the operation of the pumps and the temperature of the oil and water by temperature controllers. The control unit also controls the operation of the dynamometer, which also serves as the starter motor and the engine can be motored at a wide range of speeds. There are standard features of safety included in the control unit such as emergency switch, automatic shut down upon the excessive rise in the oil and/or coolant temperatures, or any abnormal conditions of electrical power supply (**Figure 5**).

A commercially available gasoline port fuel injector was used and its specifications can be found in **Table 3**. This injector has low flow rates and was selected to match the requirement of injecting very low quantities of gasoline to operate in HCCI mode with ultralean mixtures. The injector comes calibrated in the factory to inject and precisely meter the volume against the specified injection duration.

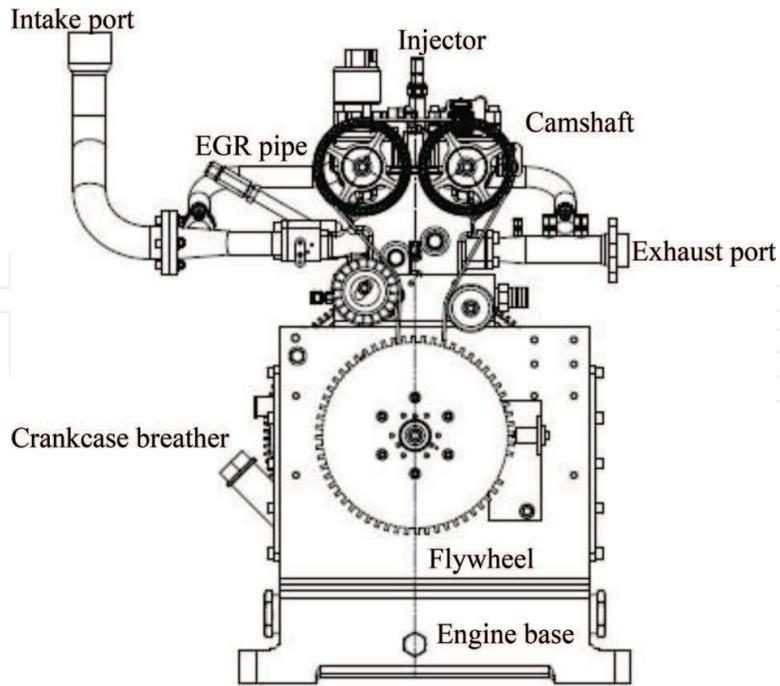


Figure 4. The single cylinder research engine.

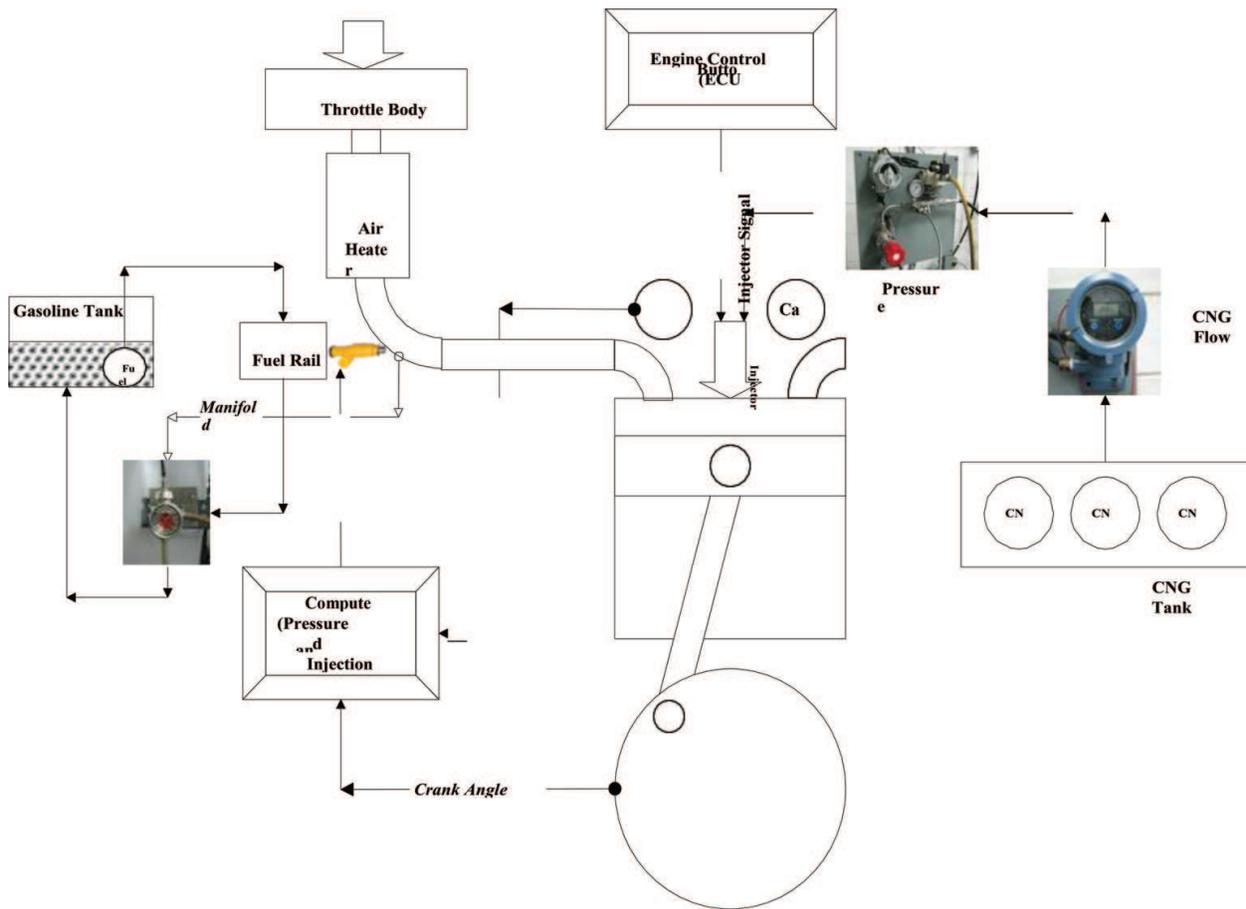


Figure 5. Gasoline and CNG fuel supply systems.

Make/part number	Bosch/0280155710	
Type	High impedance	
Fuel injection pressure	3 bar	
Fuel flow rate	191.8 cc/min	
Power supply	12 V DC	

**Table 3.** Specifications of the gasoline injector.

Engine speed (N)	Equivalence ratio of gasoline ( $\phi_g$ )	Timing of CNG injection (SOI)	CNG quantity (mCNG)
RPM	Ratio	CAD-BTDC	mg/cycle
1200	0.20	300	Up to the knocking limit or unstable engine operation
1500	0.22	240	
1800	0.24	180	
2100	0.26	120	
		80	

**Table 4.** Matrix of experiments.

Fuel compatibility	Standard gasoline and ethanol flex fuels
Fuel pressure	300 kPa
Static flow rate at 300 kPa	7.55 g/s
Offset	0.67 ms
Gain	0.11 = 9.09 mg/ms
Minimum linear PW	1.5 ms
Linear flow range	8.43
Open/closing time	1.3/0.7 ms
Coil resistance	12
Injector inductance	11.6 mH
SCOV/DMOV	4.42/4.92 V
Spray pattern	26° Cone

**Table 5.** Injector technical specification.

The change in air flow rate and volumetric efficiency at different engine speeds were taken into account, and the gasoline injection durations were adjusted so as to operate with constant equivalence ratios of gasoline from 0.20 to 0.26. **Table 4** shows the matrix of experiments and lists the variable parameters of the experiments conducted for this study.

The CNG quantity was varied and limited by the combustion stability of the mixture by limiting the maximum coefficient variation (CoV) of the combustion to 10%. Thus, the readings were obtained at different speeds, different gasoline flow rates, and different injection timings with various quantities of CNG injected (Table 5 and Figure 6).

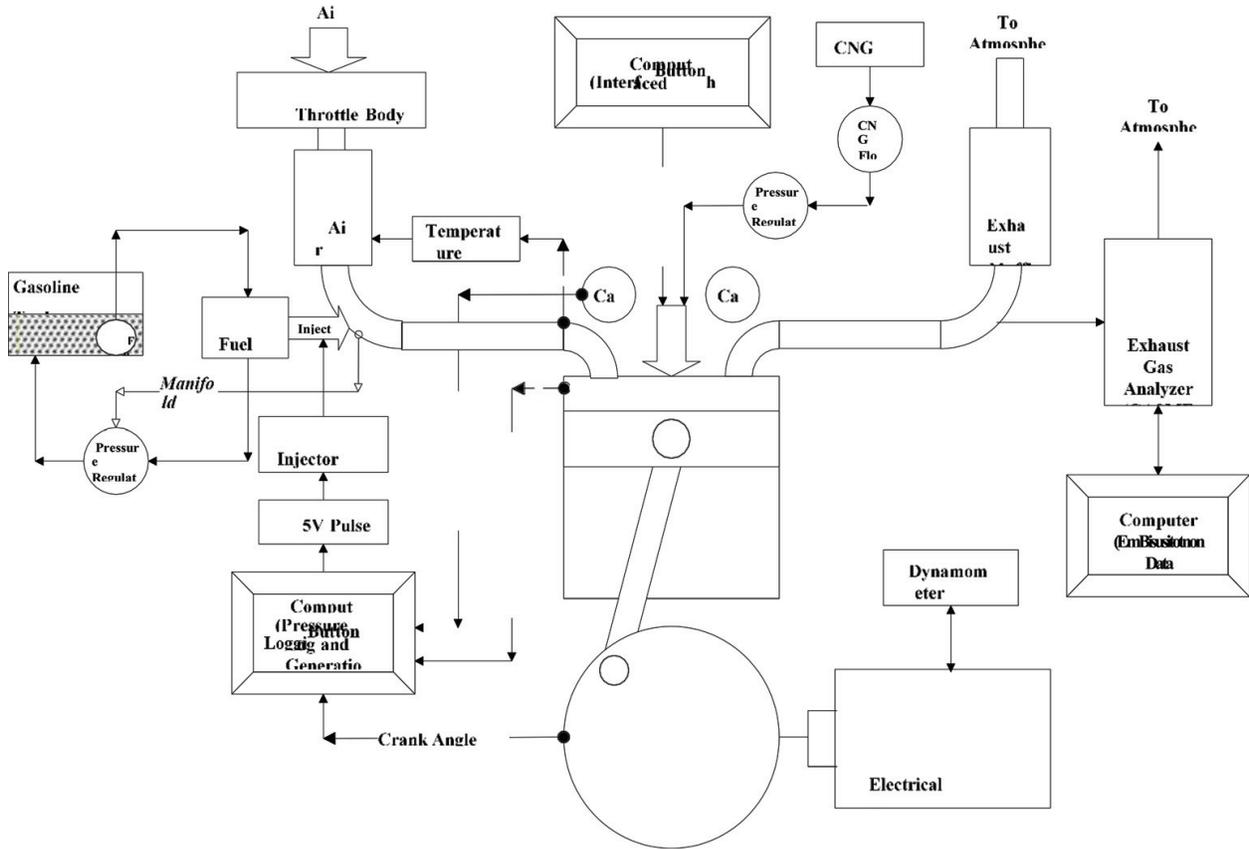


Figure 6. Experimental setup.

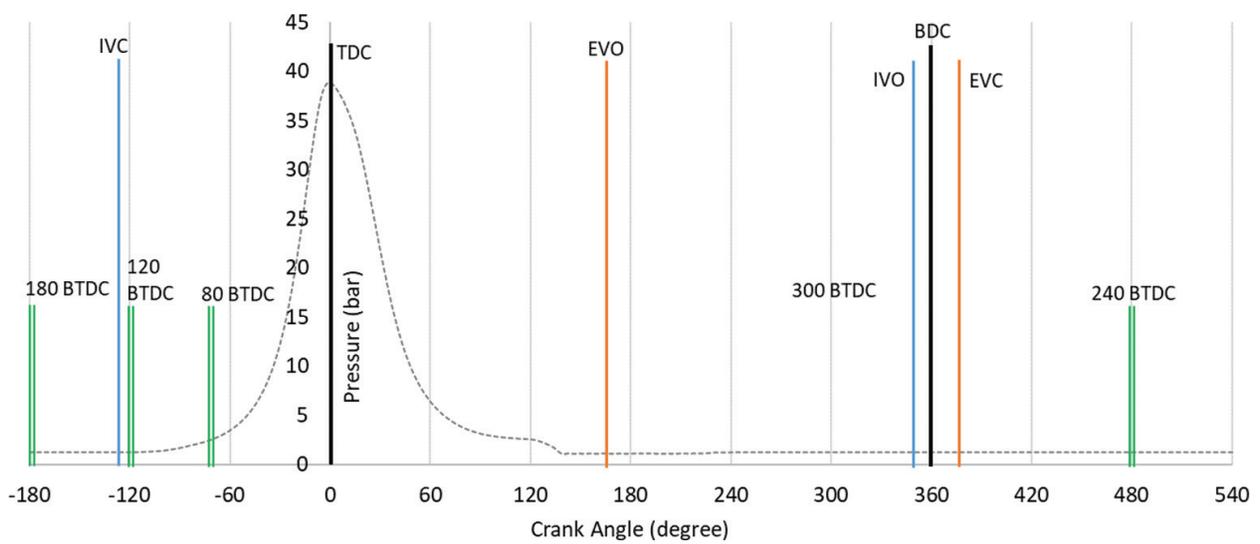


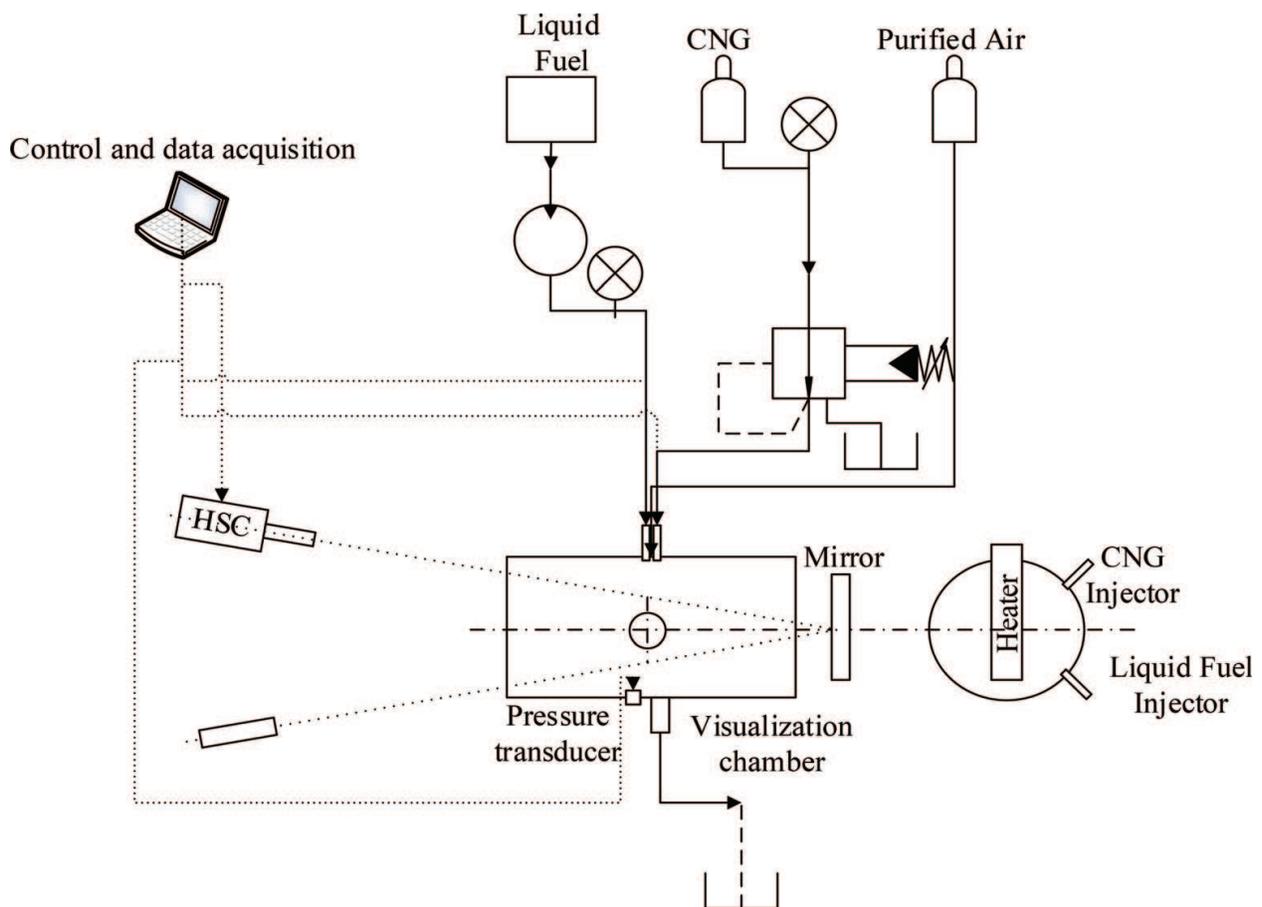
Figure 7. Injection timing of CNG, and intake and exhaust valve timing.

**Figure 7** depicted the injection timing, and intake and exhaust valve timing of the engine. It shows that injection timing 300, 240, and 180° BTDC take place before the intake valve close, while 120 and 80 BTDC take place after intake valve close.

## 2.2. Constant volume combustion chamber testing and equipment

Following section describes the detailed experimental setup and equipment used on the constant volume chamber. Constant volume combustion was used as the primary method for characterizing the interaction of the parameters such as fuel compositions, lambda, and mixing ratio. The equipment and control system were designed, manufactured, and calibrated to accommodate the experimental works. The primary data in this experimental setup are combustion images and pressure trace of which acquired by Schlieren and direct measurement method, respectively.

The experiments were carried out in a constant volume combustion chamber with diameter and length 80 and 100 mm, respectively. A 100 W cartridge heater was placed in the middle of the chamber to increase in-cylinder gas temperature with a maximum temperature of 820 °C. There were two window access planes to the chamber in order to facilitate the Schlieren image visualization method (**Figures 8–10**).



**Figure 8.** Experimental setup.

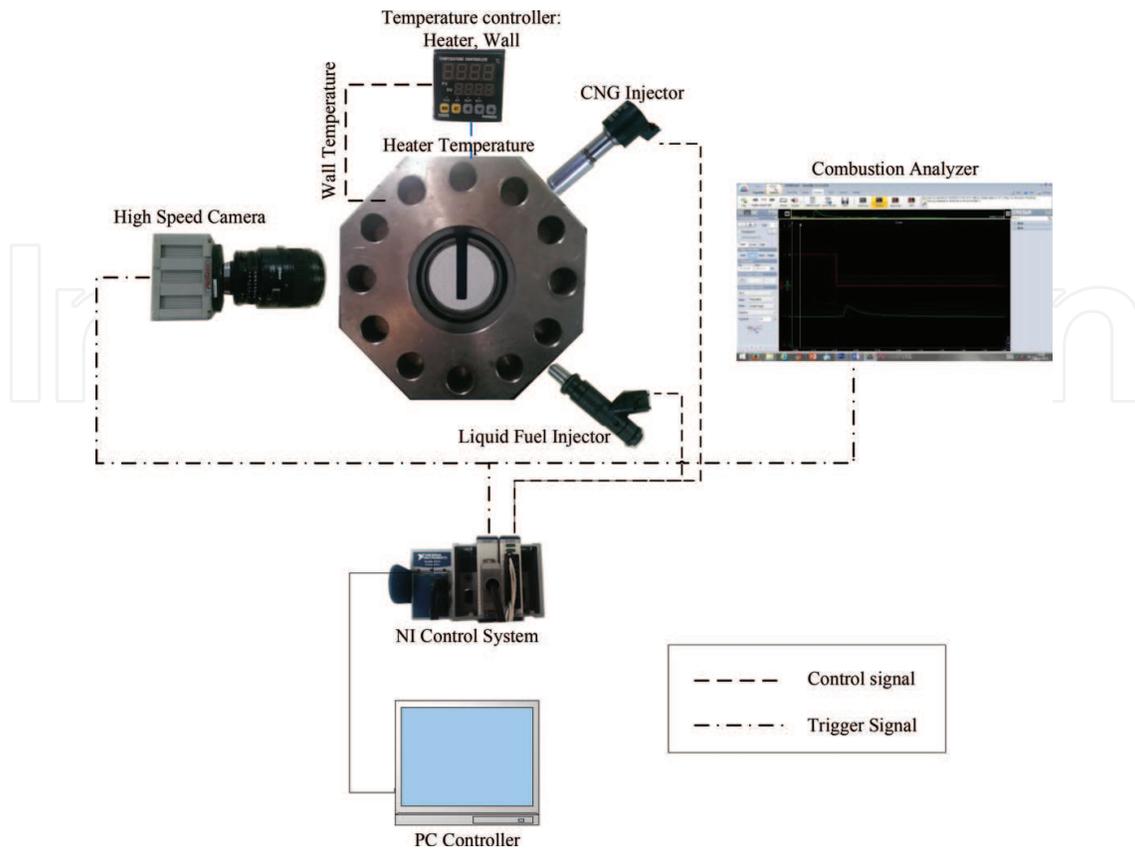


Figure 9. Schematic diagram of the control system and data acquisition.

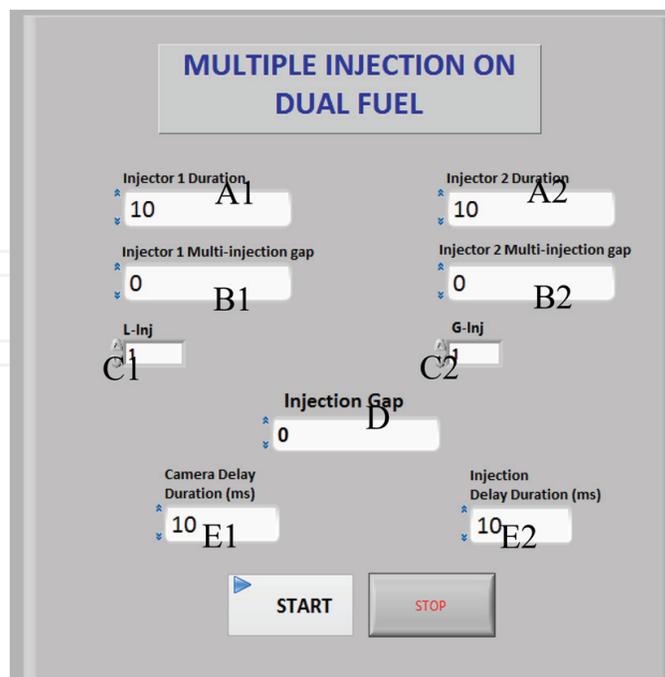


Figure 10. Injector control system interface.

### 3. Gasoline-CNG mixtures in RCCI combustion system

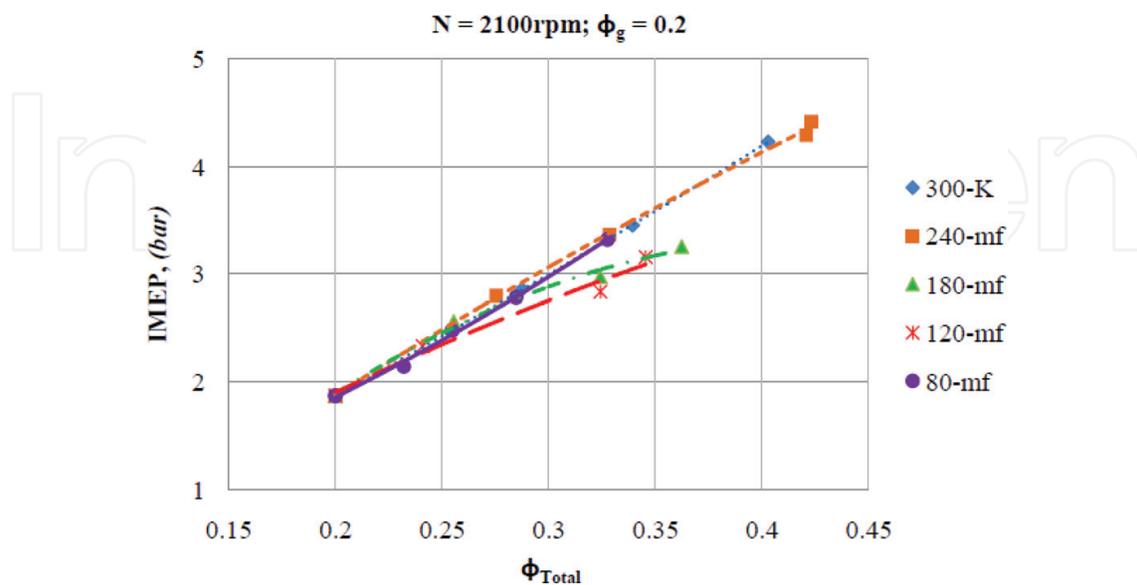
The gasoline-CNG mixtures performance and combustion in RCCI combustion system from both methods are elaborated in this section. It explained the parameters affecting the combustion of low-reactive fuels and method in controlling the combustion process.

#### 3.1. Gasoline-CNG combustion behavior in RCCI combustion-based engine

The degree of stratification of CNG in the total mixture was found to have significant effects on the maximum load in terms of the IMEP attainable and  $\phi_{Total}$ . The degree of stratification is determined by the injection timing with  $300^\circ$  BTDC representing the homogeneous mixture, while  $120^\circ$  BTDC represents the stratified mixture. The  $300^\circ$  BTDC has a very early injection timing, and the fuel is injected during intake valve is open. It is, therefore, the fuel that has sufficient time to be completely mixed with air and create a homogeneous mixture. While on the other hand, for  $120^\circ$  BTDC, the fuel is injected after intake valve closed and mixing time between fuel and air is very short and does not allow a complete mixing to take place.

**Figure 11** shows that at  $300$  and  $240^\circ$  BTDC injection higher total equivalence ratios could be operated. But with  $180^\circ$  and  $120^\circ$  BTDC, the maximum operable  $\phi_{Total}$  was limited with reduced IMEP at a given  $\phi_{Total}$  when compared to the  $300$  and  $240^\circ$  BTDC cases.

The IMEP results show agreement with investigation from Genchi G and Pipitone E [22] where the increased composition of CNG produces higher IMEP. With the highest degree of stratification, although the maximum load was limited, there was no significant drop in the IMEP and the trend was similar to  $300$  and  $240^\circ$  BTDC conditions. The corresponding values of indicated thermal efficiencies are shown in **Figure 12**. The maximum load was observed to be limited by knocking when CNG was injected at  $300^\circ$  BTDC, and for the other cases, increasing CNG injection rate led to unstable operation or misfire.



**Figure 11.** Effect of degree of stratification of CNG on IMEP. (K—limited by knocking; mf—limited by misfire).

From **Figure 13**, it can be seen that the ignition timing could be altered by changing the timing of injection of CNG at a given load. The ignition timing was determined by identifying the start of heat released rate and mass fraction burned derived from the pressure data where the 0% points before the continuous propagation of the mass fraction burned is determined as the start of ignition of the analyzed combustion cycles.

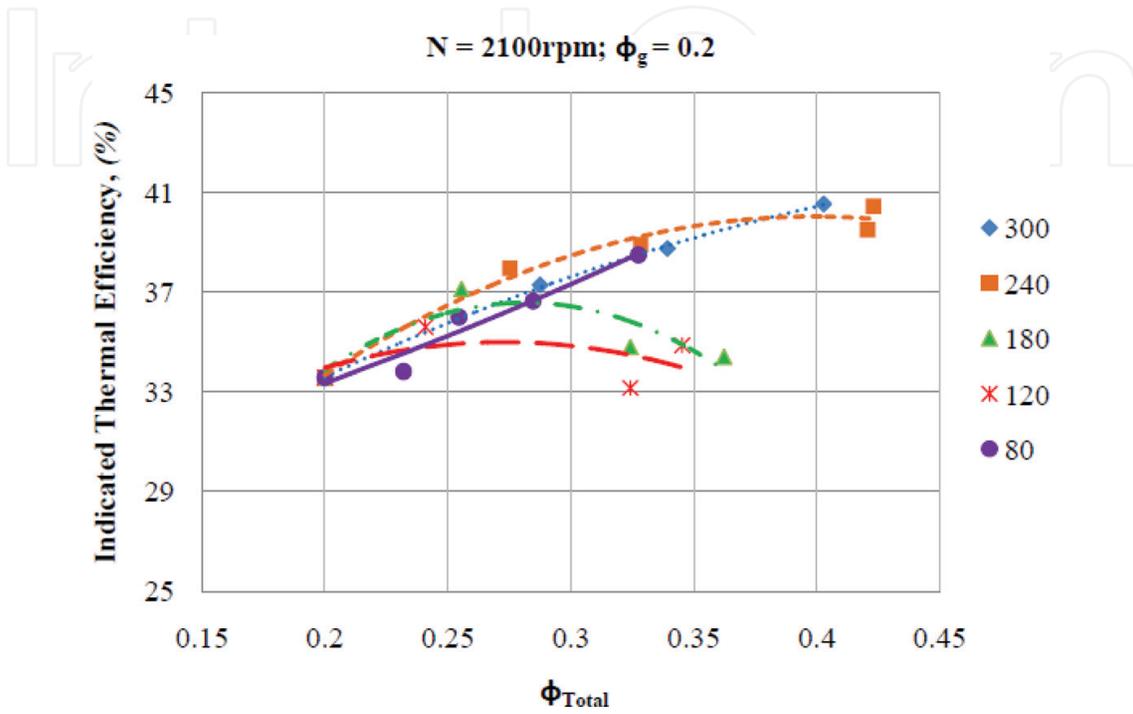


Figure 12. Effect degree of CNG stratification on the indicated thermal efficiency.

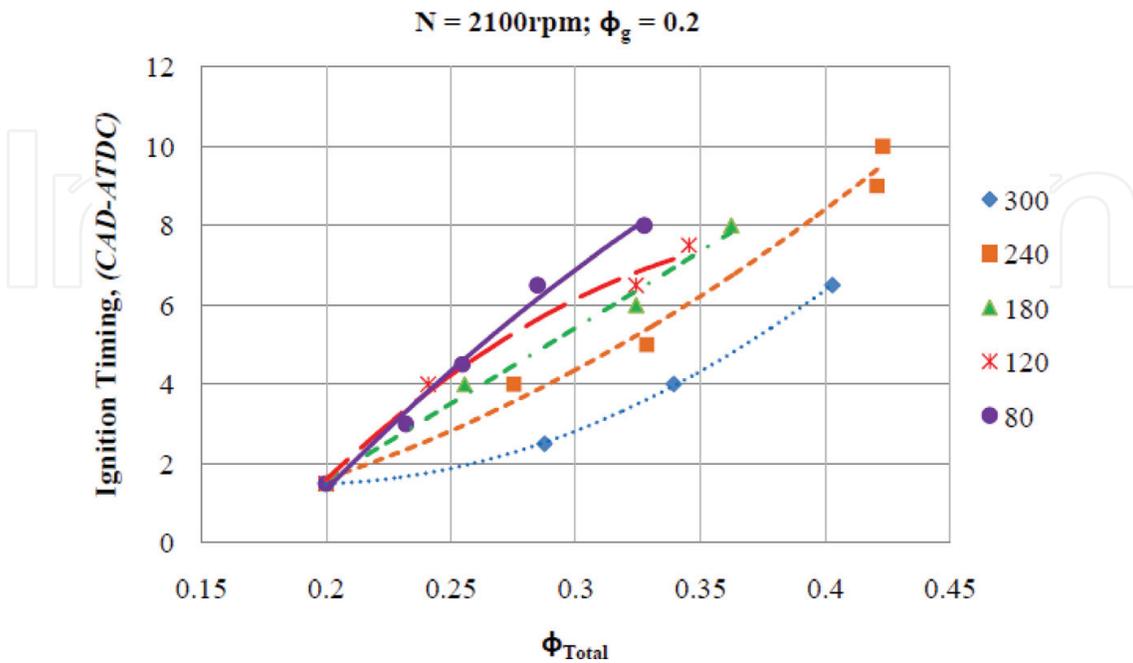


Figure 13. Effect of degree of CNG stratification on ignition timing.

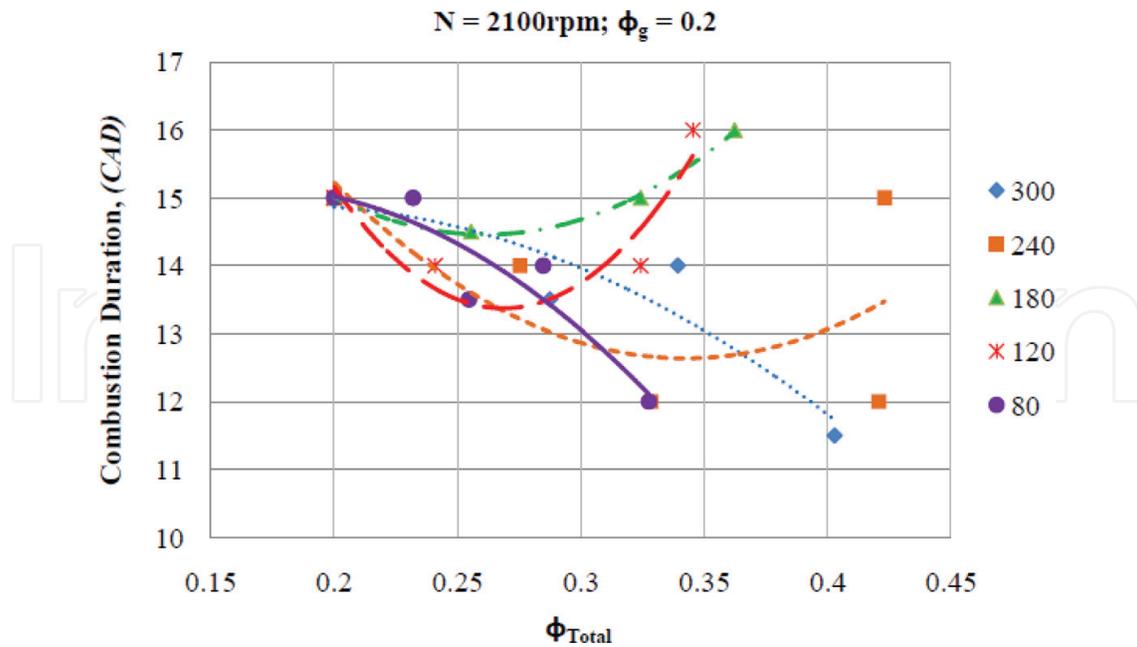


Figure 14. Effect of degree of CNG stratification of CNG on the combustion duration.

When the rate of CNG injection was increased, the ignition timing was delayed due to the higher octane number of CNG. Also, higher degrees of stratification resulted in higher increments in the delay of ignition timing as the CNG injection rate was increased. The slope of the curves was steeper when the injection timing was delayed. For a given increase in the CNG injection rate, the increase in delay in ignition timing was higher when the degree of stratification was increased. That is both the injection rate and the degree of stratification of CNG had significant effects on the ignition timing when operated with  $\phi_g = 0.20$ . However, the maximum total equivalence ratio was less than that obtained with CNG injection at 300 and 240° BTDC.

It was found that the combustion duration was reduced when CNG injection rate was increased at 300, 240, and 80° BTDC. When CNG was injected at 180 and 120° BTDC, the combustion duration was marginally affected and it initially decreased up to certain values of CNG injection rate and then it increased again.

Figures 15–18 show the rate of heat release and pressure rise at various injection timings. Increasing the rate of CNG injection at 300° BTDC was limited by knocking as shown in Figure 14. But with later injection timings, with  $\phi_g = 0.20$ , any increase in CNG injection rate resulted in a delayed autoignition and reduced peak pressure. Therefore, increasing CNG injection rate beyond certain levels led to misfire or no fire, thereby defining the maximum load limit.

As shown in Figure 15, when CNG injection rate was increased, it resulted in delayed ignition timing. Up to  $\phi_{Total} = 0.33$ , the resultant peak pressure increased, and with a further increase in CNG injection rate, it decreased. Also, above  $\phi_{Total} = 0.33$ , the delay in ignition timing was more significant and resulted in decreased peak pressures. As will be discussed later in this section, combustion efficiency of both the fuels increased and CH<sub>4</sub> emissions decreased with an increase in  $\phi_{Total}$  above 0.33 as shown in Figures 23 and 29.

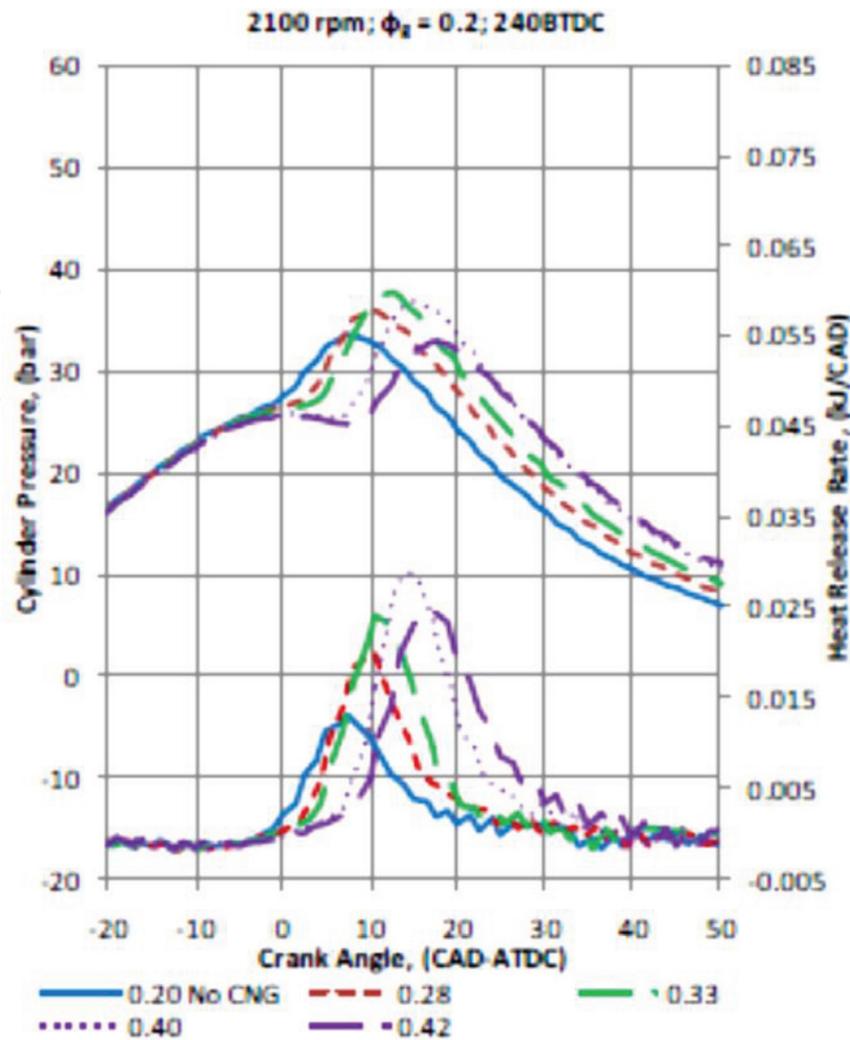
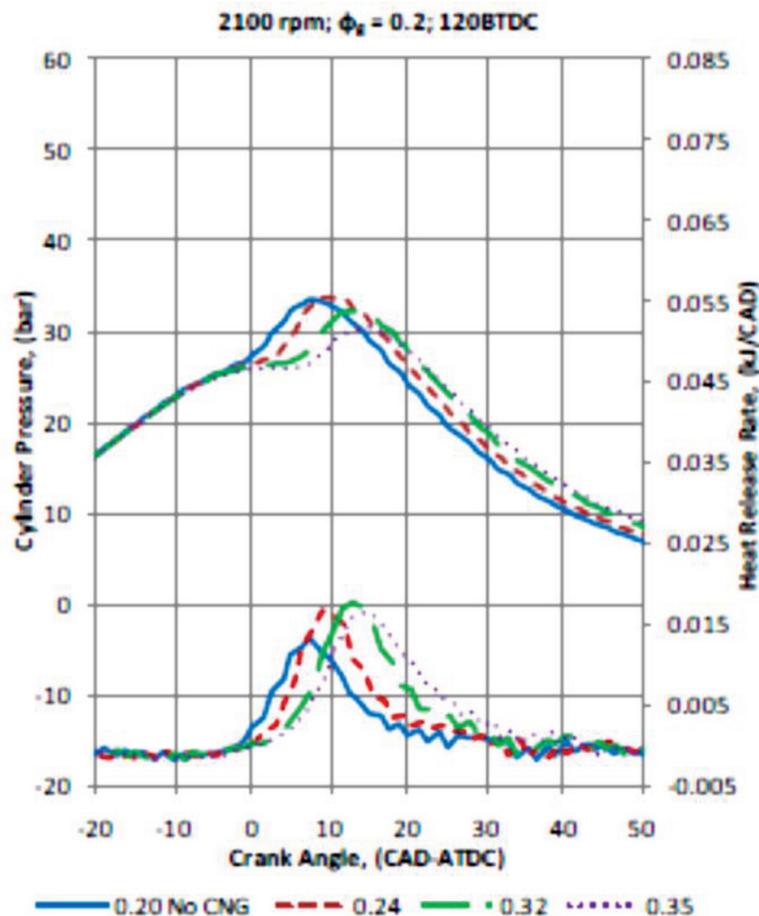


Figure 15. Pressure history and heat release rates with CNG injection at 240° BTDC.

Therefore, it can be concluded that, above  $\phi_{\text{Total}} = 0.33$ , the peak pressure was reduced due to delayed ignition, and the combustion was more complete with increase in injection rate at 240° BTDC. That is, increasing  $\phi_g$  above 0.33 resulted in reduced peak pressures without a decrease in thermal efficiency as shown in Figure 12. Heat release rates increased with an increase in CNG injection rate up to  $\phi_{\text{Total}} = 0.40$  above which it reduced again. Above  $\phi_{\text{Total}} = 0.42$ , increasing CNG injection rate resulted in misfire or no fire, and both gasoline and CNG combustion was quenched.

With CNG injection at 180° BTDC, increase in CNG injection rate resulted in a more significant delay in ignition timing. There was a marginal increase in peak pressure when  $\phi_{\text{Total}}$  was increased to 0.26 above which it reduced again. Thermal efficiency and combustion efficiency increased primarily due to the remarkable increase in the completeness of CNG combustion as suggested by CH<sub>4</sub> emissions as shown in Figure 29. Heat release rate increased with an increase in CNG injection rate as shown in Figure 17. However, increasing CNG injection rate above  $\phi_{\text{Total}} = 0.26$  resulted in decreased overall combustion efficiency and high CH<sub>4</sub>



**Figure 16.** Pressure history and heat release rates with CNG injection at 120° BTDC.

emissions as shown in **Figures 23** and **29**. This suggests that the degree of stratification created at 180° BTDC injection results in deterioration in combustion and leads to decrease in thermal efficiency as shown in **Figure 12**. Similar trends were observed with CNG injection at 120° BTDC when  $\phi_{Total}$  was increased above 0.24 as shown in **Figure 16**.

When CNG injection was retarded to 80° BTDC, increase in injection rate resulted in significant delay in ignition; however, there was less noticeable effect on peak pressures up to  $\phi_{Total} = 0.28$ . Increasing  $\phi_{Total}$  above resulted in a more significant delay in ignition and peak pressure, and heat release rates increased. Thermal efficiency and combustion efficiency increased primarily due to remarkable increase in completeness of CNG combustion as suggested by CH<sub>4</sub> emissions as shown in **Figure 29**.

As shown in **Figure 19**, increasing CNG injection rate at 240° BTDC resulted in delayed ignition. At  $\phi_{Total} = 0.28$  and 0.33, there was a slight increase in burning rate at the last stage of combustion compared to combustion with pure gasoline. At 180 and 120° BTDC, there was no significant effect on the burning rate of the fuels due to increase in CNG injection rate, but it caused a significant delay in ignition as shown in **Figures 20** and **21**. Similar results were obtained with CNG injection at 80° BTDC; however, at  $\phi_{Total} = 0.28$  and 0.33, the combustion was slower at the initial stages and was faster at latter stages as shown in **Figure 22**.

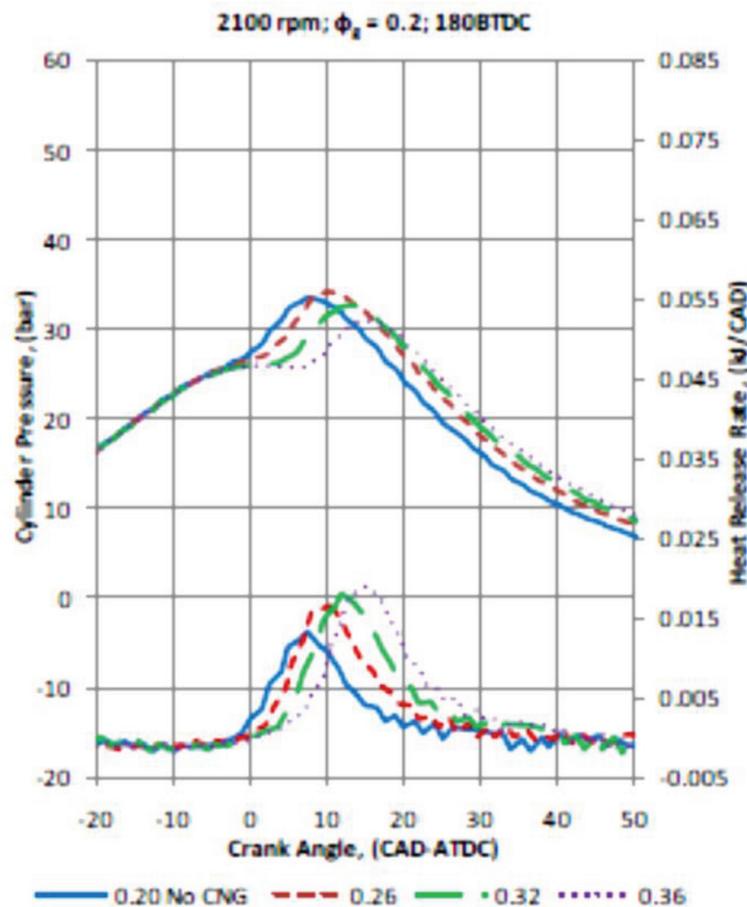


Figure 17. Pressure history and heat release rates with CNG injection at 180° BTDC.

As shown in **Figure 23**, with an increase in  $\phi_{\text{Total}}$  by CNG injection at 300, 240, and 80° BTDC, combustion efficiency increased. The highest increment was obtained with CNG injection at 80° BTDC for a given increase in  $\phi_{\text{Total}}$  due to mixture stratification. However, CNG injection at 180 and 120° BTDC, combustion efficiency increased initially but decreased again and was below 80% for all  $\phi_{\text{Total}}$ .

The exhaust gas temperature was observed to increase as the CNG injection rate was increased as shown in **Figure 24**. The increase in exhaust gas temperature with increasing CNG injection rates at 180 and 120° BTDC was less than that observed with increasing CNG injection rates at 300, 240, and 80° BTDC. When the fuels were homogeneously mixed, it resulted in higher exhaust gas temperatures due to rapid burning. Similarly, when CNG was highly stratified, it also led to higher exhaust gas temperatures.

**Figure 25** shows the indicated specific  $\text{NO}_x$  ( $\text{ISNO}_x$ ) emissions. The  $\text{NO}_x$  emissions were marginally affected and were around the same levels for all test conditions. However, different trends were observed at different injection timings and CNG injection rates.

Increasing the CNG injection rate resulted in a drastic increase in the  $\text{NO}_2/\text{NO}_x$  ratio up to a certain point and then it decreased. As shown in **Figure 26**, the ratio of  $\text{NO}_2/\text{NO}_x$  almost doubled when the CNG injection rate was increased to around  $\phi_{\text{Total}} = 0.33$  before decreasing

again. That is, up to a certain value of CNG injection rate, CNG reduced the combustion temperature and led to formation of higher amounts of  $\text{NO}_2$ .

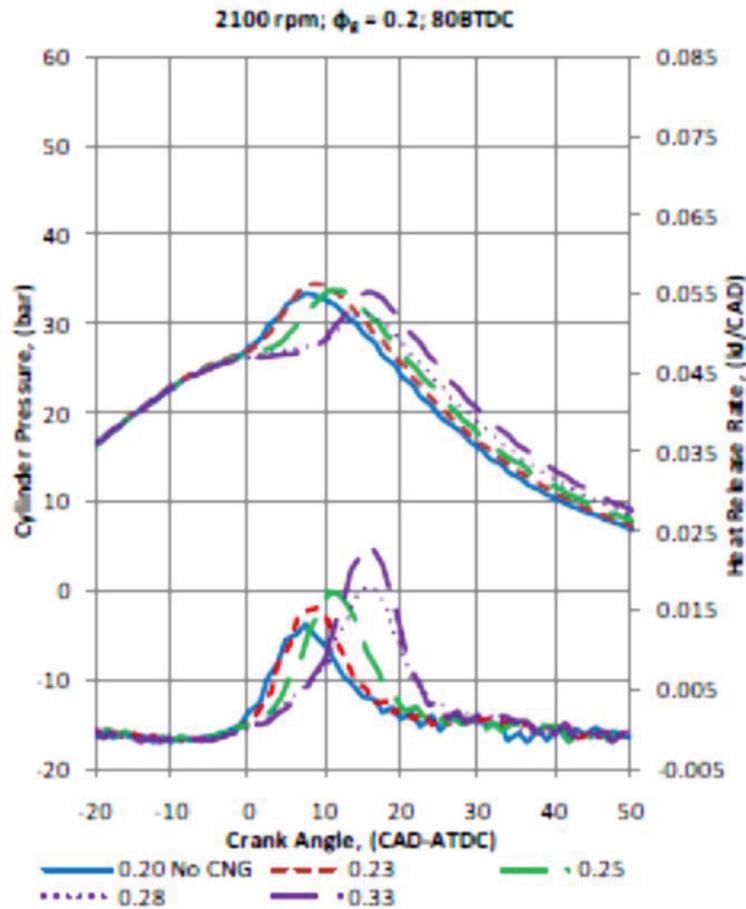


Figure 18. Pressure history and heat release rates with CNG injection at  $80^\circ$  BTDC.

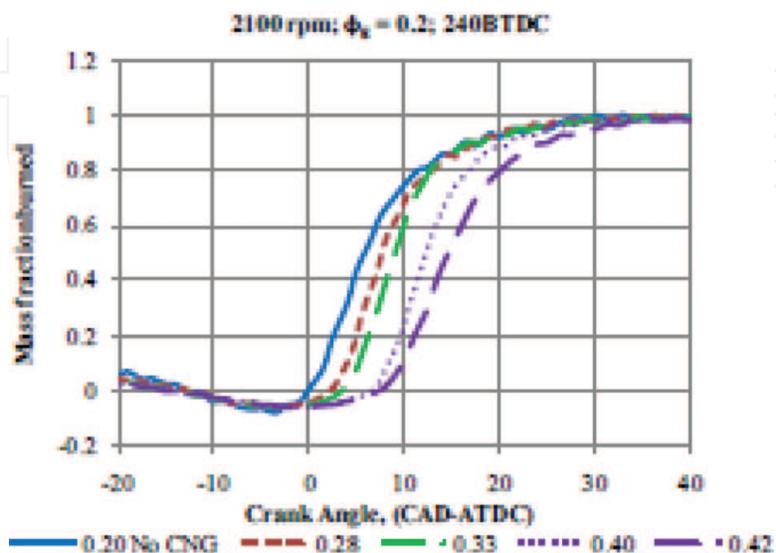


Figure 19. Mass fractions burned with CNG injection at  $240^\circ$  BTDC.

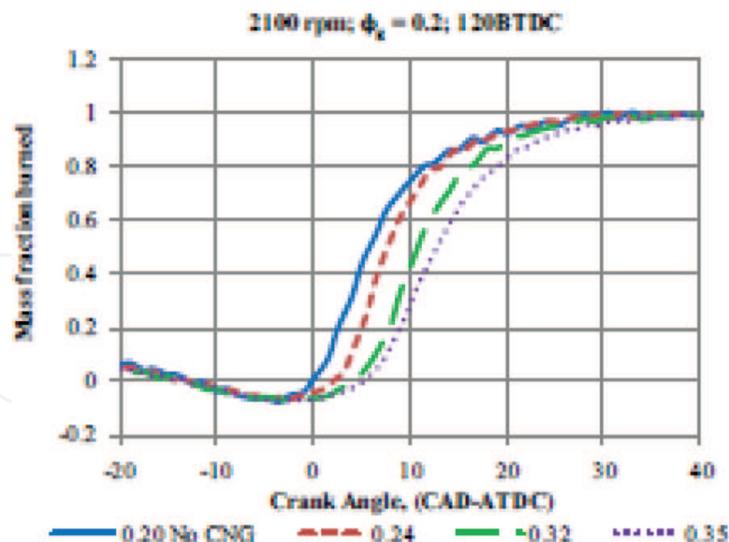


Figure 20. Mass fractions burned with CNG injection at 180° BTDC.

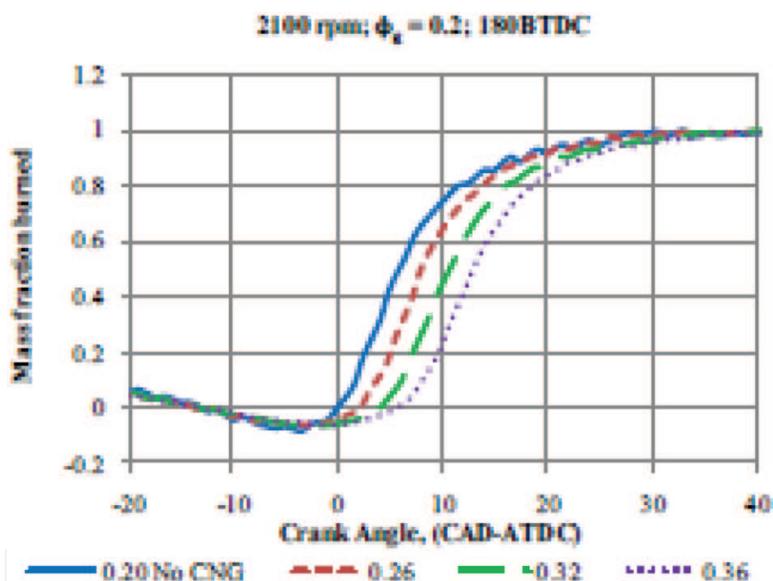


Figure 21. Mass fractions burned with CNG injection at 120° BTDC.

The indicated specific CO (ISCO) emissions were reduced significantly as the mixture was enriched with CNG by direct injection at all injection timings as shown in **Figure 27**. However, the reduction obtained was the highest when CNG was injected at 300 and 240° BTDC. Any increase in CNG injection rate at later injection timings resulted in less reductions in CO emissions. The lowest reduction was obtained at the injection timing of 80° BTDC, as the high degree of stratification of CNG limited the availability and distribution of oxygen and temperature differences in the CNG and air particles.

The HC emissions were found to be significantly affected by the degree of stratification of CNG as shown in **Figure 28**. The highest reduction in HC emissions was obtained with CNG

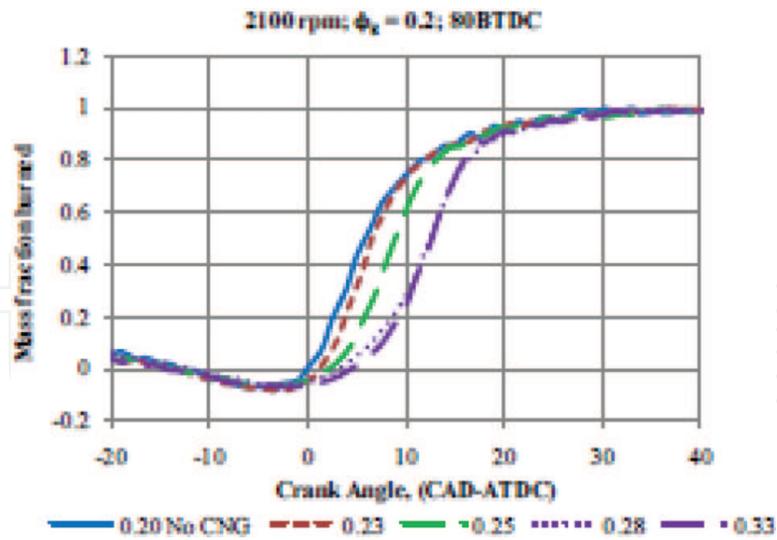


Figure 22. Mass fractions burned with CNG injection at 80° BTDC.

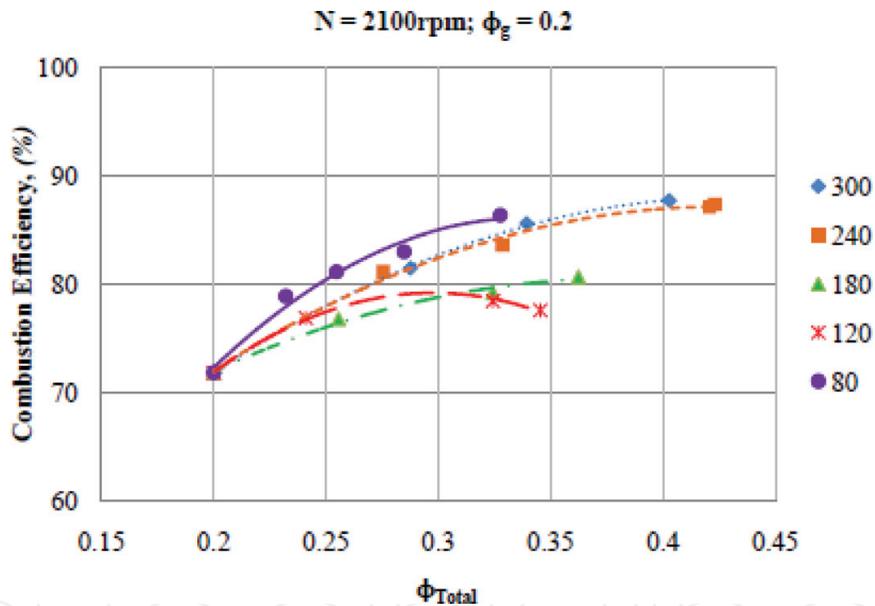


Figure 23. Effect of degree of stratification on combustion efficiency.

injection at 80° BTDC. Higher degrees of stratification of CNG resulted in more complete combustion.

Figure 29 shows the mass ratio of flow rates of CH<sub>4</sub> in the exhaust emissions and mass flow rate of CNG injected into the cylinder. At a given constant gasoline equivalence ratio of  $\phi_g = 0.20$ , CNG direct injection at 80° BTDC resulted in the least emission of CH<sub>4</sub>. Therefore, the combustion of CNG was more complete when it was stratified. CNG injection at 300 and 240° BTDC resulted in moderate levels of CH<sub>4</sub> emissions, and highest values were obtained with CNG injection at 180° BTDC. This was due to the turbulence created and mixing conditions in the cylinder when the piston changed its direction at 180° BTDC.

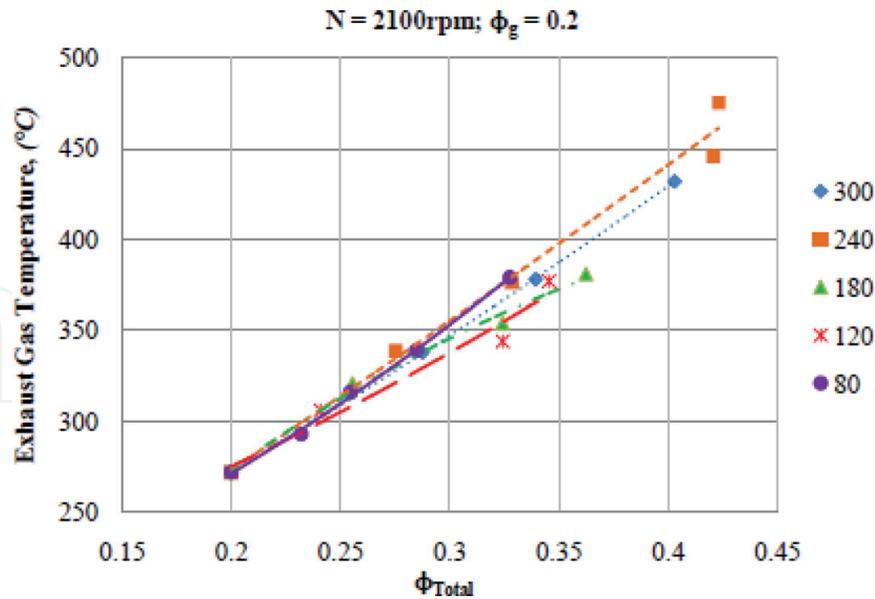


Figure 24. Effect of CNG injection on the exhaust gas temperature.

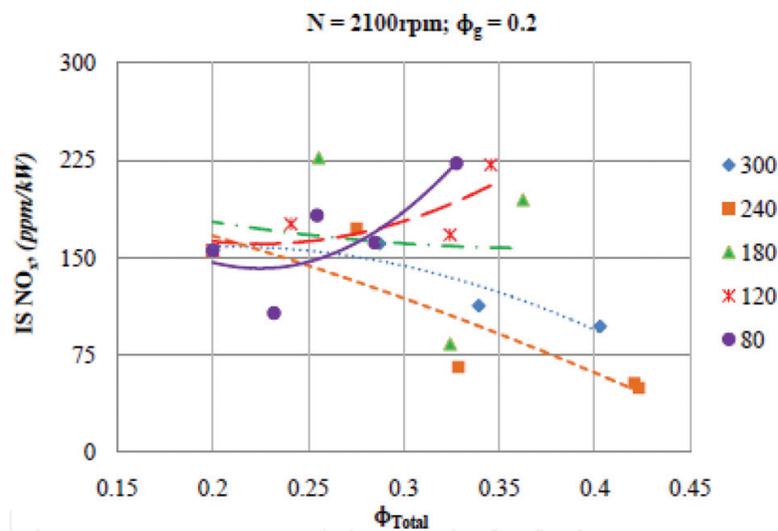


Figure 25. Effect of degree of CNG stratification on  $\text{NO}_x$  emission.

### 3.2. Gasoline-CNG combustion behavior in constant volume combustion chamber

The effect of injection gap on the gasoline-compressed natural gas mixture (GCNG) mixture combustion is discussed below. The injection gap alteration gave direct impact on the mixture distribution inside the combustion chamber. There are five injection gaps tested, 0, 5, 10, 15, and 20 ms. These injection gaps are expected to be able to give direct control to the mixture distribution inside the chamber.

The effect of injection gaps is shown in **Figure 30**. It shows two mixture compositions, 50 and 90% GCNG composition. The injection gap gives different effect between the two compositions. In 50% GCNG, longer injection gap gives higher combustion efficiency, maximum pressure, total heat released (THR) and shorter delay. Furthermore, it also shows longer duration for

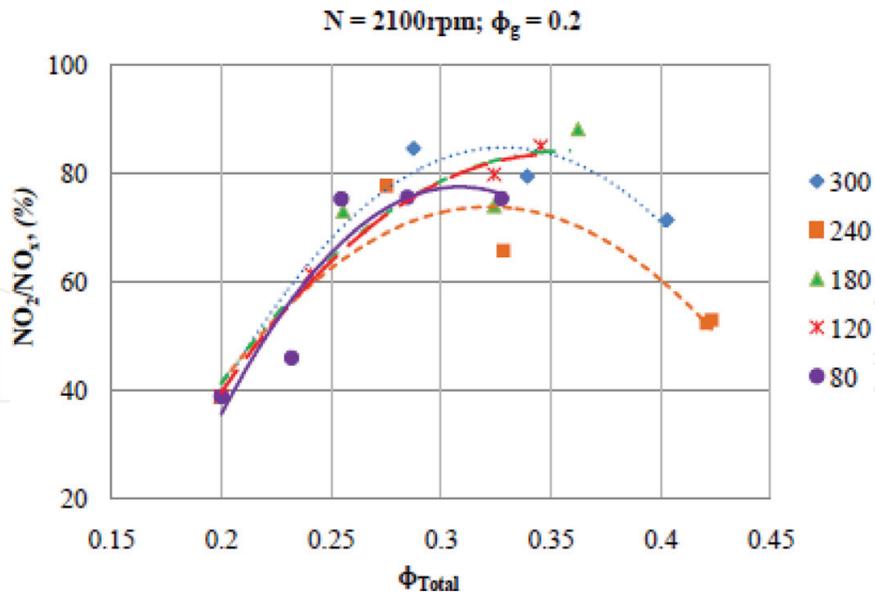


Figure 26. Effect of degree of CNG stratification on NO<sub>2</sub> formation.

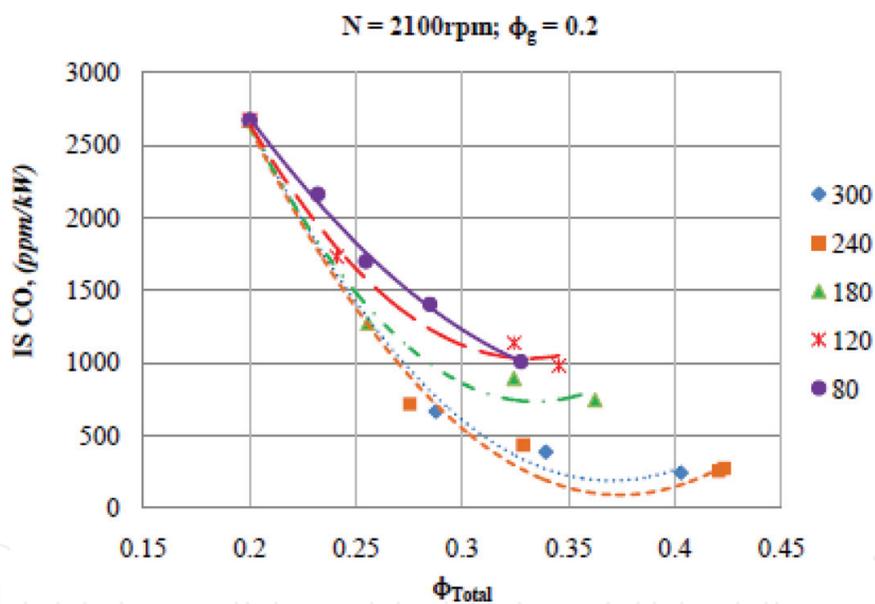


Figure 27. Effect of degree of CNG stratification on CO emissions.

all combustion stages. While on the contrary, longer injection gap reduces the combustion efficiency, maximum pressure, THR and longer combustion delay for 90% GCNG. However, the trends of the combustion duration are similar, longer duration for longer injection gap. **Figure 31** confirmed the variation of injection gap effect to the combustion process of the GCNG mixture. The turning point is shown between 70 and 80% GCNG mixture compositions. For all the mixture above 80% shows decreasing combustion efficiency with the increase in injection gaps which is contrary with the mixtures below 70% that show an increment of combustion efficiency with the increase of injection gaps. These differences may cause by the mixture distribution inside the chamber.

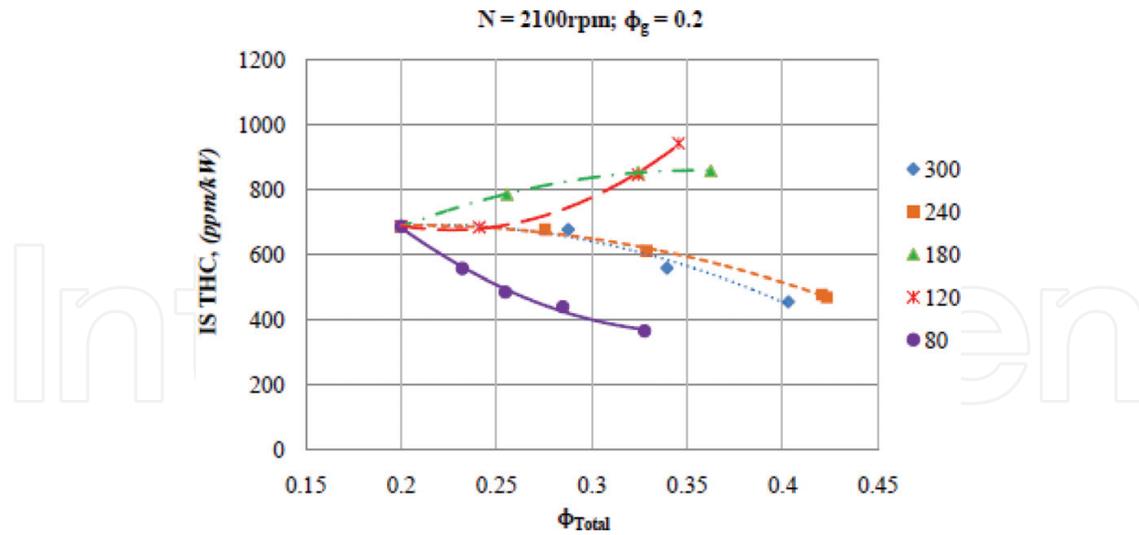


Figure 28. Effect of degree of CNG stratification on HC emissions.

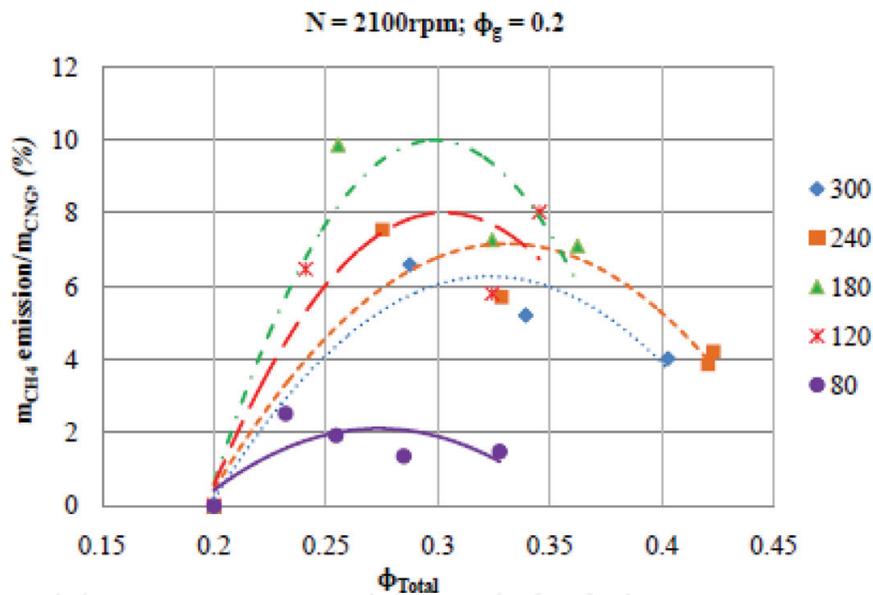


Figure 29. Effect of injection timing on the emission of CH<sub>4</sub> with  $\phi_g = 0.20$ .

The mixture distribution inside the chamber for GCNG mixture 30 and 90% with 0 and 20 ms injection gaps are depicted in **Figure 32**. In the figure, highly stratified mixture for 30% GCNG mixture with 0 ms injection gap. The stratification is marked by darker color on the bottom of the chamber that indicates high density fluid (gasoline). The image shows that most of the gasoline was collected at the bottom of the chamber because of the momentum of CNG injection that prevents the gasoline from reaching the top side of the chamber. 20 ms injection gap, on the other hand, shows better fuel mixing shown by fairly similar image intensity throughout the chamber.

The injection gaps for 30% GCNG mixture improve the mixing rate thus increase the combustion performance. Furthermore, the gasoline fuel is mainly accumulating at the bottom side

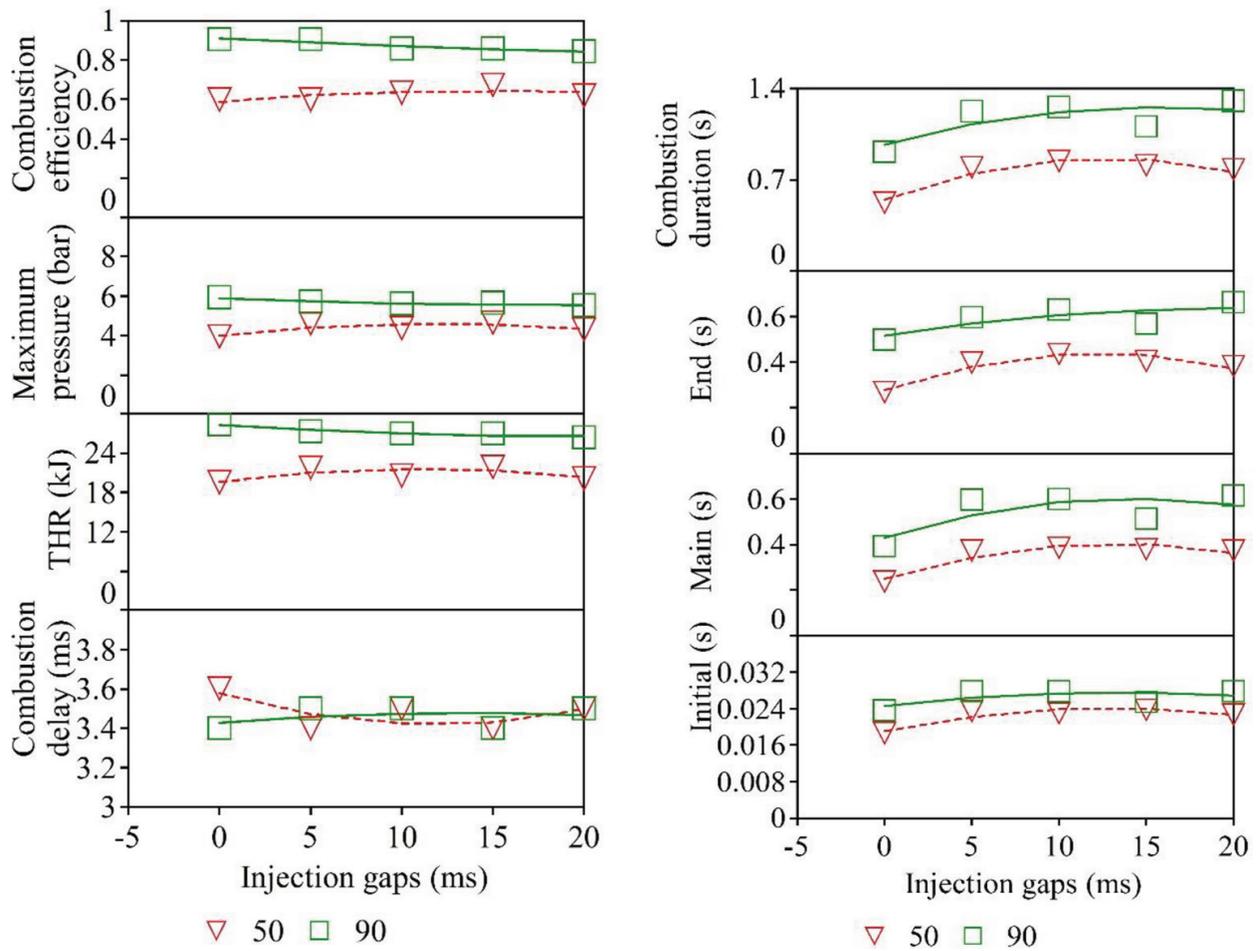


Figure 30. Effect of injection gaps on the combustion characteristics of GCNG mixture for 50 and 90% composition at  $\lambda = 1$ .

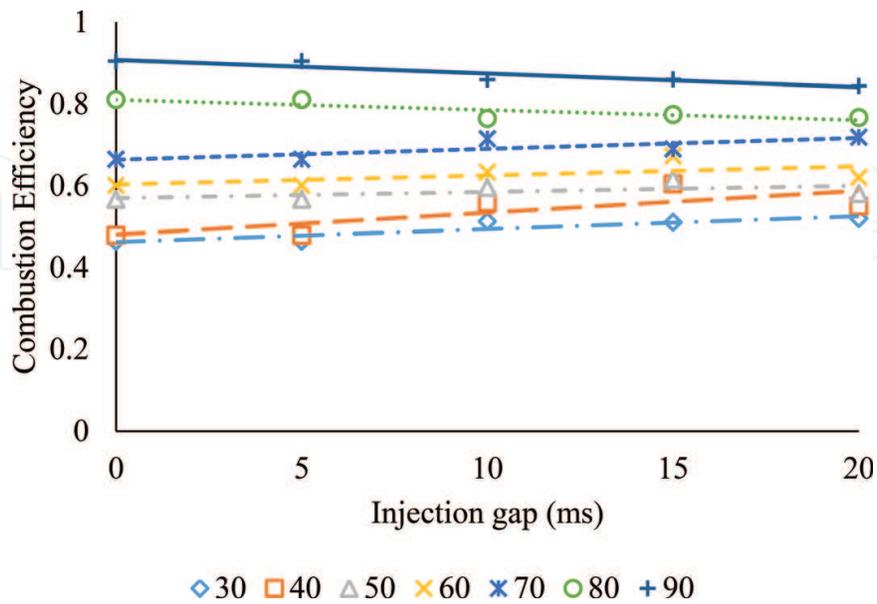
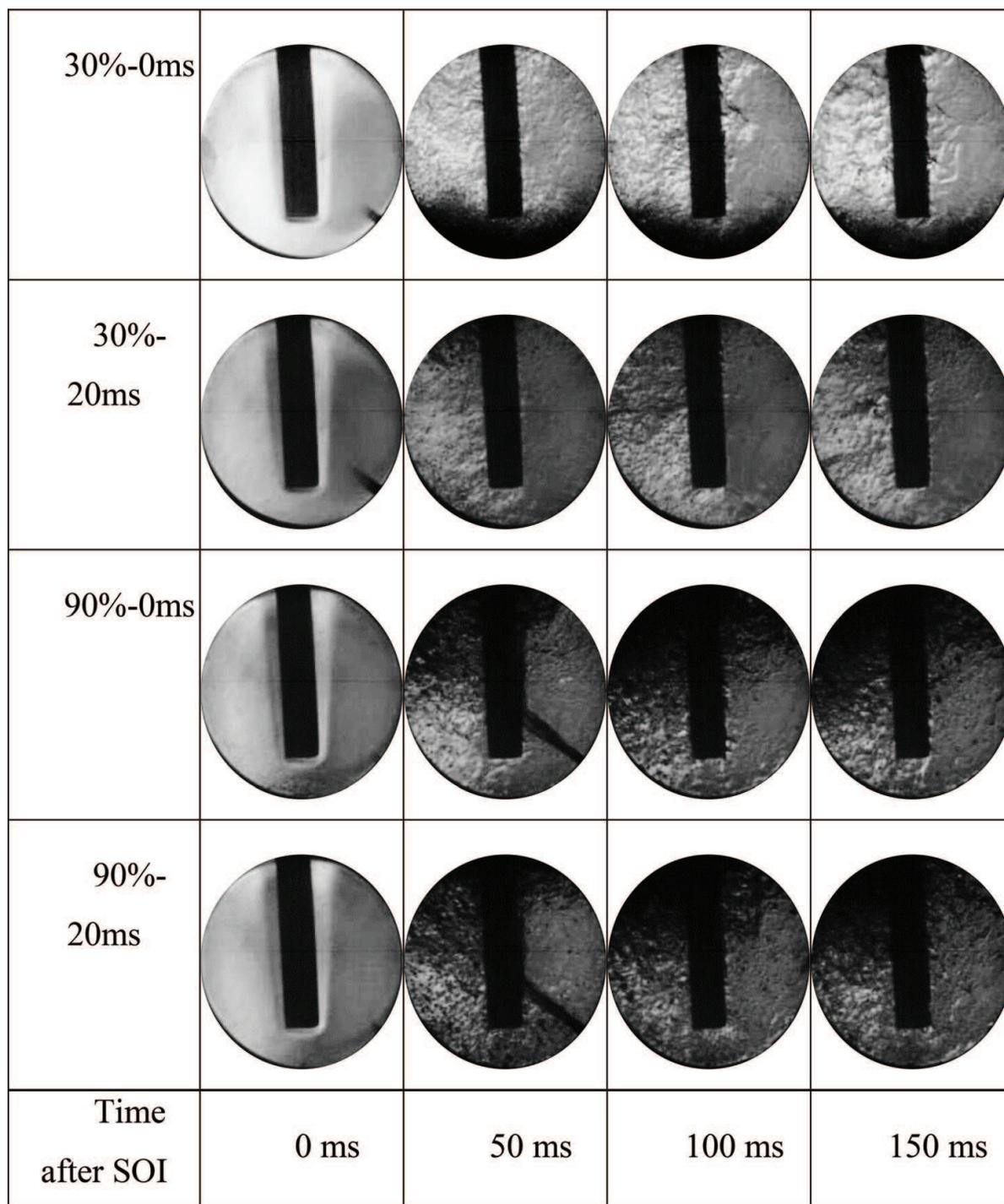


Figure 31. Effect of injection gap to the combustion efficiency for various mixture compositions at  $\lambda = 1$ .



**Figure 32.** Mixture distribution for 30 and 90% mixture composition at 0 and 20 ms injection gaps.

which also has average low temperature compared to the top one. It makes the vaporization rate of the gasoline took longer time which also elongates the combustion delay as shown in **Figure 33**.

The injection gaps at 90% GCNG mixture, on the other hand, have similar liquid fuel distribution as in **Figure 32** where both injection gaps show concentrated fuel distribution at the top of the chamber. In spite of the similarity, the 0 ms injection gap shows the higher intensity of the liquid fuel (darker region) on the top side of the chamber compared to 20 ms injection gap. It

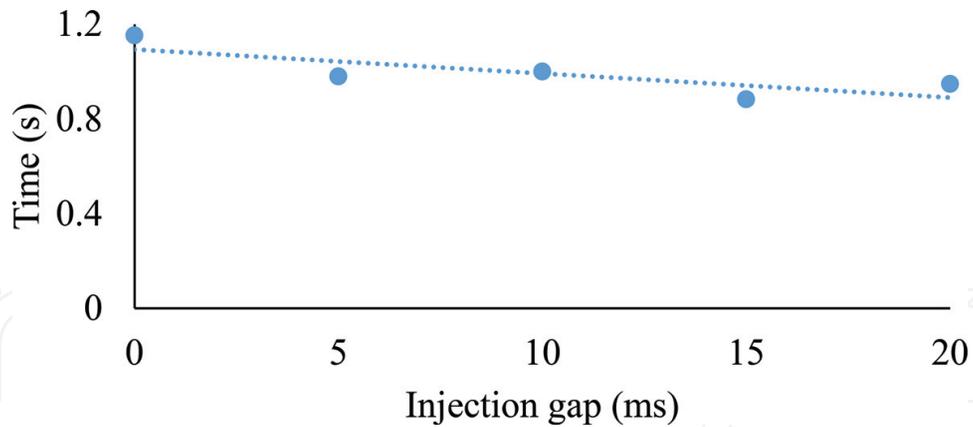


Figure 33. Combustion delay relative to SOI.

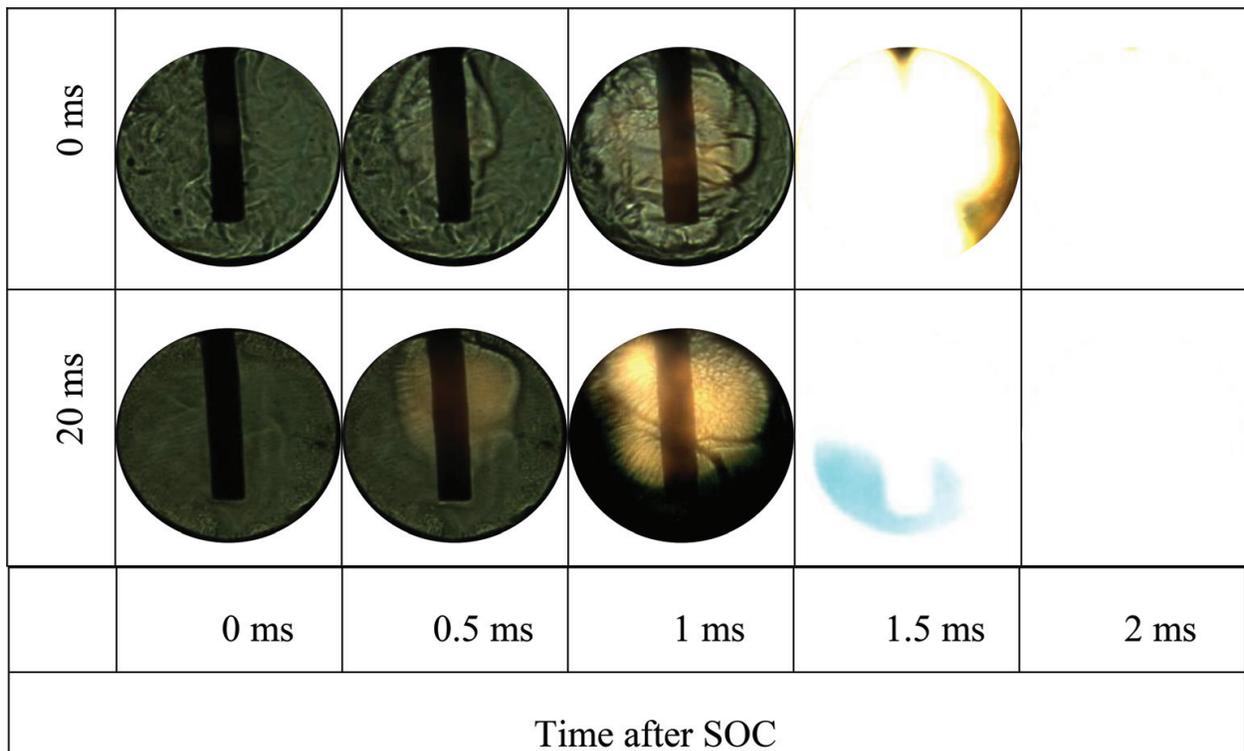
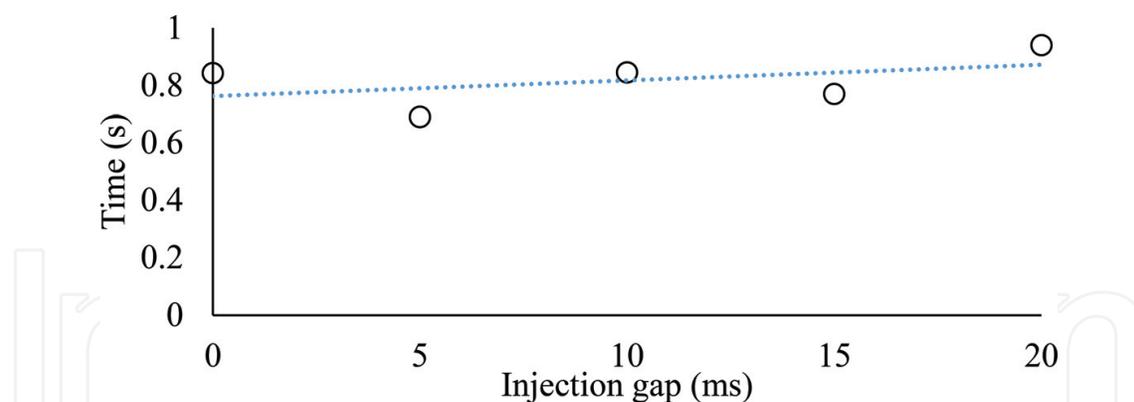


Figure 34. Combustion sequence for GCNG at 60/40 mixture composition and lambda 1.

shows that 0 ms injection gap have higher gasoline population compared to 20 ms injection gap thus the longer time required for vaporization process. This is the main reason for the lower combustion output as well as longer combustion delay for 0 ms compared to 20 ms injection gap.

The combustion sequence for GCNG at 0 and 20 ms injection gaps is depicted in **Figure 34**. The flame speed of 20 ms injection gap is faster than 0 ms injection gap with 37.02 m/s at the first 0.5 ms and 15.9 at the first 1 ms after start of combustion (SOC), while 0 ms injection gap with 30.56 m/s and 16.9 m/s at 0.5 ms and 1 ms, respectively. **Figure 34** also reveals the difference in the flame color for the two injection gaps. A 20 ms injection gap shows light blue color yet with



**Figure 35.** Effect of injection gap for 60/40 gasoline/CNG composition at lambda 1.

lower intensity compared to 100% gasoline, while 0 ms injection gap shows yellow color. It can be assumed that the blue color is the product from the same reaction that generates hydroxyl peroxide and increases the combustion output of the mixture.

In 100% gasoline combustion, the blue flame occurs because of the homogeneous mixture that creates multipoint combustion behind the flame front that significantly increases the combustion output. A similar process occurs in the 20 ms injection gaps, mixture homogeneity is achieved with this mixture as the effect of CNG injection shown by longer combustion delay relative to the start of injection as depicted in **Figure 35**.

Injection gaps proved to have a direct influence on the fuel distribution inside the chamber, thus affecting the combustion characteristics of the mixture. The combustion process in the CVC is mostly affected by the characteristics of the fuel distribution inside the chamber at the time the combustion occurs. The injection gaps, in this case, highly affect the mixture distribution inside the chamber where longer gap promotes the mixing and create a mixture that is more homogeneous.

## 4. Conclusions

1. It was found that more amount of gasoline was needed to ignite stratified CNG. Increasing the degree of CNG stratification resulted in delayed ignition and reduced peak pressures. The combustion efficiency of overall mixture and that of CNG were higher at high degree of CNG stratification. When the degree of CNG stratification was increased, significant reduction in HC emissions could be achieved even at low gasoline equivalence ratios. However,  $\text{NO}_x$  and CO emissions increased especially at low engine loads and high speeds. Maximum load attainable at low engine speeds could be extended by increasing the degree of CNG stratification.
2. Injection gap capable of controlling the stratification level in the CVC shown by longer injection gap produces relatively homogeneous mixture compared to the short injection gaps. The results show that the fuel stratification level has a less significant effect on the combustion stages although it has a significant effect on the combustion phasing. Stratified mixture produces shorter combustion duration especially on the main and final combustion

stages, while homogeneous mixture produces longer duration. This trend is applicable for both GCNG and DCNG mixtures.

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