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Development of a single cylinder CNG direct injection engine and its performance, emissions and combustion characteristics

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Abstract: This paper presents experimental results of a newly developed compressed natural gas (CNG) fuelled direct injection (DI) engine, which was developed by modifying a single cylinder CI engine. Major modifications included: a) modifications in cylinder head and piston; b) development and installation of electronic CNG DI injection system; c) installation of a capacitive discharge ignition system. Experiments were conducted at a constant fuel injection pressure and engine speed and performance, emission and combustion characteristics of this CNG DI engine with varying fuel injection timings [i.e., start of injection (SOI)] and varying engine loads [i.e., brake mean effective pressure (BMEP)] were investigated. Moderate engine loads lead to faster and more complete combustion of the fuel, thus improving engine performance and reducing emissions for specific injection timings. Advanced fuel injection improved engine performance (lower BSFC, higher BTE); reduced emissions and produced faster ROHR whereas retarded injections delivered opposite trends for each engine load. [Received: July 25, 2013; Accepted: February 16, 2014]

Keywords: compressed natural gas; CNG; direct injection; manifold injection; engine performance; emission characteristics; combustion characteristics.

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1 Introduction

With emerging stringent vehicular emission legislations and dwindling resources of liquid fossil fuels, meeting the growing demand for improving fuel efficiency and reduction in emissions has become the prime motivation for engine researchers. SI engines have advantage in terms of higher power-to-weight ratio compared to CI engines, but they suffer from the issue of relatively lower thermal efficiency due to throttling losses and knock limitations. Diesel engines have higher thermal efficiency; however, NO_x and particulate emission still remains a major concern from CI engines (Brehob et al., 1998; Kano et al., 1998).

With abundant availability of natural gas and its low emission potential due to favourable (H: C) ratio, it has emerged as one of the most promising and clean alternative fuels for engine application in stationary as well as transportation sector globally. There are several compressed natural gas (CNG) engine technologies used worldwide, which differ in the way, the fuel is introduced into the engine cylinder, e.g., carburetor technology, port fuel injection, dual fuel technology, etc. To utilise full potential of CNG in engine applications, concept of direct injection (DI) of natural gas has been investigated under various engine operating conditions by varying fuel injection timings, equivalence ratio, cyclic variations, spark timings, etc., by several researchers.

Richards (1992) reported that CNG DI engines have higher power output and higher thermal efficiency compared to a conventional SI CNG engines due to their higher compression ratios, and lower pumping losses at the part load conditions. Ikeda et al. (1995) reported that an eight cylinder CNG DI engine had twice the brake mean effective pressure (BMEP) as compared to conventional port fuel injected SI CNG engine.

Caley and Cathcart (2006) performed a comparative study on manifold CNG injection, manifold gasoline injection and CNG DI and reported that performance of DI CNG was within 3% of stoichiometric gasoline engine performance at low speeds and within 7% at higher speeds. DI of CNG with late start of injection (SOI) improves air flow up to 10% over the manifold CNG operation, which in turn increases engine performance up to 10% at lower speeds but at higher speeds, only 4% improvement in engine performance was reported due to reduction in mixing duration.

Goto (1999) investigated the influence of injection timings and spark timings on the combustion and exhaust emission characteristics of a single cylinder diesel engine and he demonstrated that when λ was close to 1.0 (stoichiometric mixture), combustion became stable by more advanced injection and when λ was more than 2.0, retarded injection provided more stable combustion. Zeng et al. (2006) showed that for advanced SOI timings, HC emissions reduced but NO_x emission increased due to adequate fuel-air mixing, and more complete and faster homogeneous combustion, whereas retarded injection led to reverse trends. Liu et al. (2010) reported that advanced SOI led to superior air-fuel mixtures, and reduced combustion duration, however, the premixed combustion was slightly longer. Huang et al. (2000a, 2000b) investigated CNG DI combustion in a SI rapid compression machine (RCM) in different injection modes and showed same level of unburned HC as that of homogeneous combustion and steeper increase in CO with increase in equivalence ratio. Early injection led to a longer premixed combustion duration, whereas late injection led to a longer mixing controlled combustion duration.

Most research work on CNG DI was performed using RCM which provided some useful insight, however, in order to understand the in-cylinder gas flows and combustion completely, it is essential to understand the effect of fuel injection timing on CNG DI combustion in an actual IC engine. Therefore, the objective of this study is to experimentally investigate the combustion at varying loads and injection timings in a CNG DI engine in order to develop a prototype.

2 Experimental setup and procedure

The schematic of the experimental setup is shown in Figure 1. A single cylinder, four-stroke, naturally aspirated, water-cooled diesel engine was converted into CNG DI engine. It was decided to modify a diesel engine by converting it into a SI engine because such modified SI engine developed from diesel engine was more robust from durability and structural integrity point of view. The specifications of the modified CNG DI engine are given in Table 1.

Figure 1 Schematic of the experimental setup (see online version for colours)

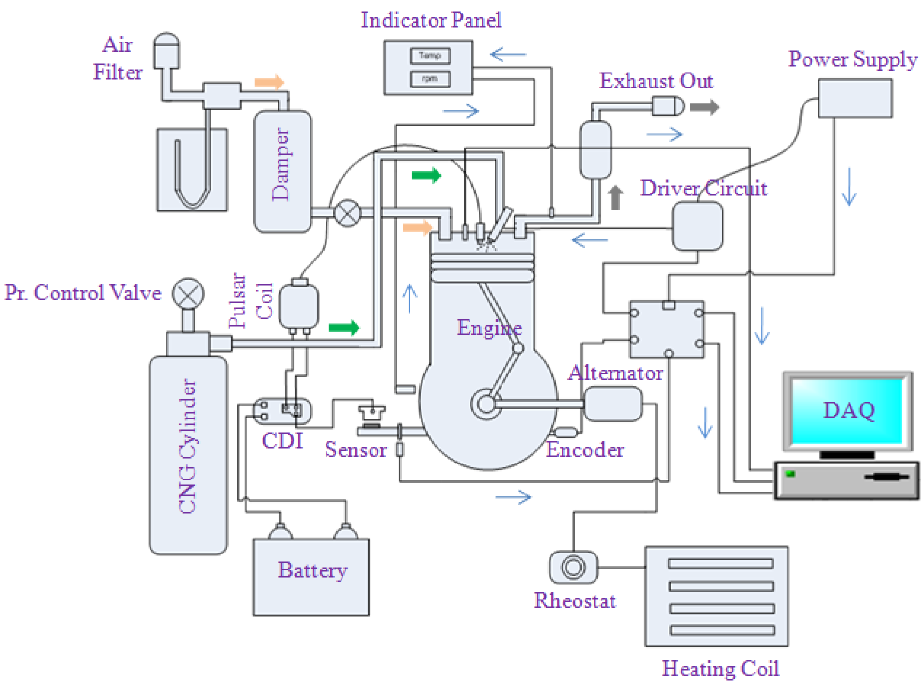


Table 1 Engine specifications after modifications

Make	Kirloskar
Model	DM-10
Bore	102 mm
Stroke	115 mm
Displacement	1,000 cc
Compression ratio	9.5
Combustion chamber	ω Bowl
CNG Injection pressure	50 bar
Ignition source	Spark plug

Major modifications done on the test engine include:

a cylinder head was machined to accommodate a spark plug and a CNG injector

b piston was machined to reduce the compression ratio

c a capacitive discharge ignition (CDI) system was installed for spark ignition

d an electronic fuel injection system developed and installed.

A gasoline direct injection (GDI) injector (Mitsubishi: DIM1000G E7T05071) was used to supply CNG directly into the combustion chamber at 50 bar pressure and the opening, and closing and thus the injection duration was controlled by the electronic circuit, which used TDC signals from a proximity sensor (TAP: GLP18APS). An ignition system with a pickup coil, CDI coil and a long tip spark plug was installed into the engine cylinder head (Goto, 1999). An alternator was coupled to the engine in order to load the engine. The intake air flow rate was measured using an orifice plate and U-tube manometer installed downstream of an air box used for dampening air pulsations. A Coriolis force-based CNG mass flow meter (Emerson: CMF010M) was used to measure the fuel flow rate into the engine. A piezoelectric pressure transducer (Kistler: 6613CQ09-01), a precision optical shaft encoder (Encoders India: ENC58/6-720AB) and a high speed combustion data acquisition system (Hi-Techniques: Synergy) were used for the experiments. Raw exhaust emissions and smoke opacity were measured using exhaust gas emission analyser (AVL: Digas 444) and smoke opacimeter (AVL: 437) respectively.

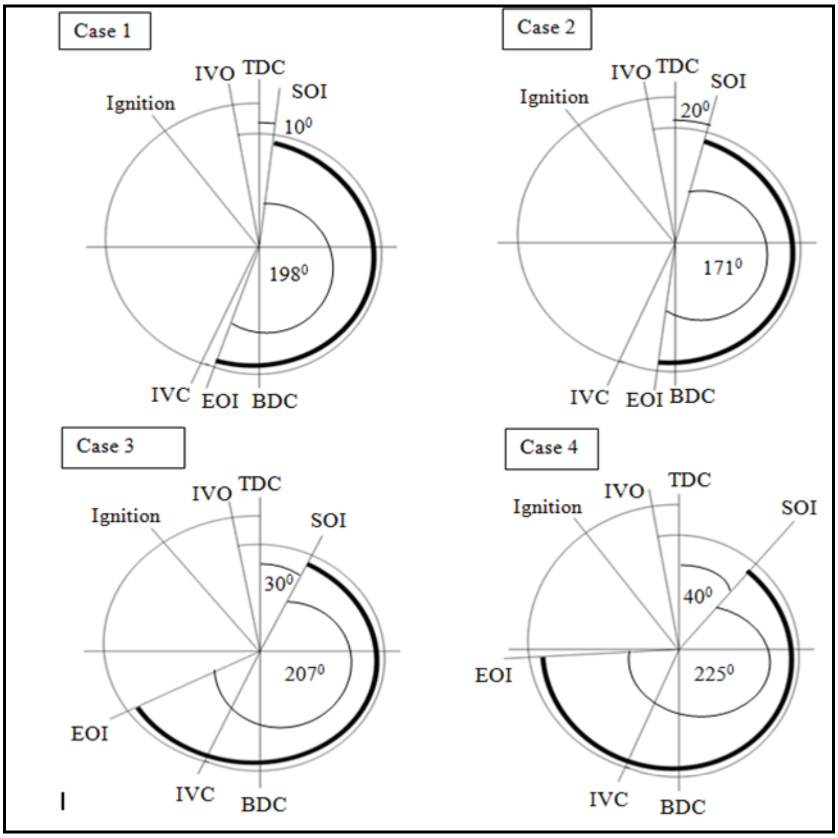
The engine was operated in steady-state at wide open throttle (WOT) condition at a constant engine speed (1,500 rpm) and the spark timing was fixed at 31° BTDC. Four different SOI timings (140°, 150°, 160° and 170° BBDC) were chosen for the experiments and engine load was varied from 1 to 3 kW in steps of 0.5 kW. Fuel injection pressure was maintained at 50 bar.

3 Results and discussion

3.1 CNG injection strategies

The SOI timings and injection duration for four different cases are shown in Figure 2. Due to low energy density of the natural gas, injection duration is relatively longer and in order to complete the injection before the spark ignition, the end of injection (EOI) is done before the intake valve closing (IVC).

Figure 2 CNG injection strategies shown on valve timing diagram



3.2 *Performance characteristics*

Engine performance in terms of its volumetric efficiency, brake thermal efficiency (BTE), brake specific fuel consumption (BSFC) and exhaust gas temperature (EGT) are investigated in this study with varying engine operating conditions. Figure 3 shows the effect of SOI timings on volumetric efficiency and equivalence ratio of the CNG DI engine. It is seen that the volumetric efficiency of CNG DI engine decreases with advanced SOI timings (Goto, 1999). This is due to displacement of higher volume of air by the natural gas during the intake. In case of SOI at 170° BBDC, injection duration (198 CAD) is relative longer compared to 160° BBDC (171 CAD). Therefore the corresponding equivalence ratio is also relatively higher for 170° BBDC SOI.

Figure 3 Volumetric efficiency and equivalence ratio vs. SOI timings (see online version for colours)

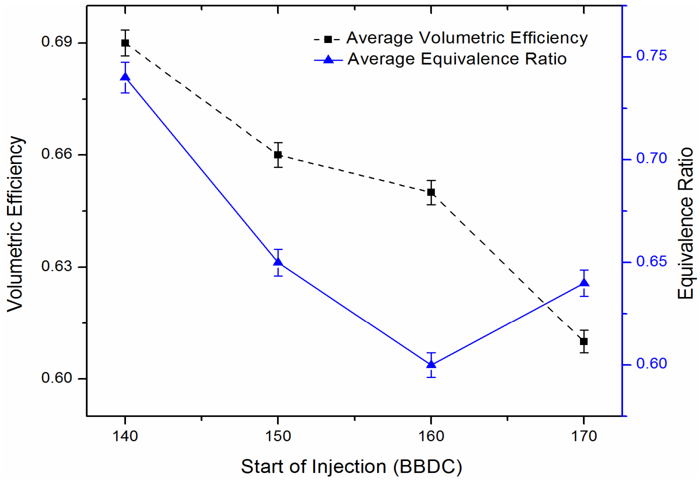


Figure 4 presents various engine performance characteristics curves for various engine loads using different fuel injection strategies at constant engine speed and fuel injection pressures.

Figure 4 Engine performance characteristics curves, (a) BTE vs. BMEP (b) BSFC vs. BMEP (c) EGT vs. BMEP (see online version for colours)

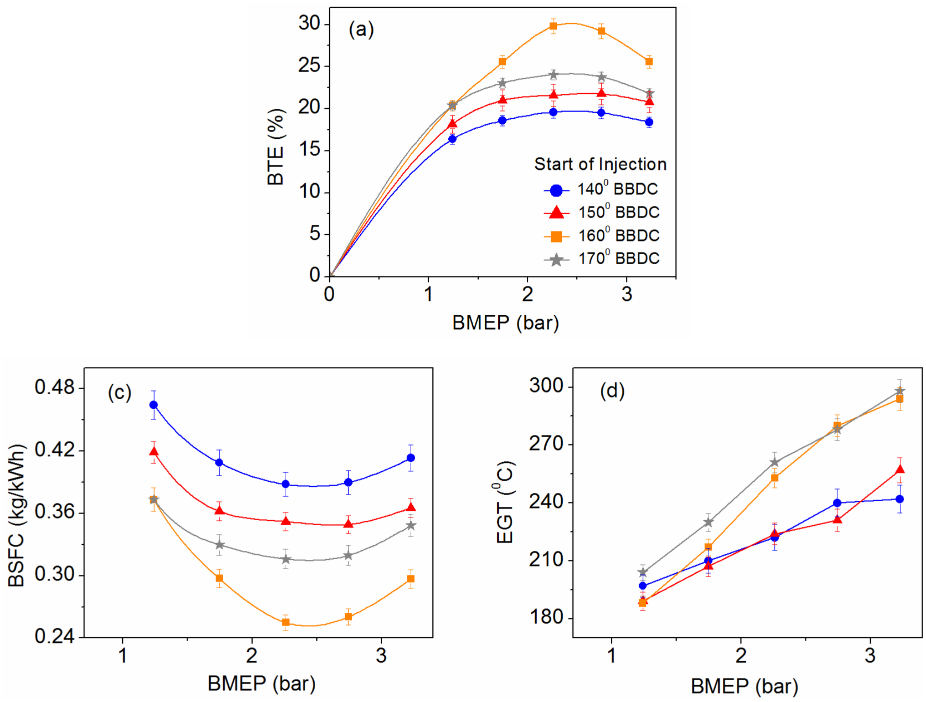


Figure 4(a) shows that BTE increases with increasing engine load (BMEP), and reaches a maxima [which corresponds to the lowest BSFC in Figure 4(b)] and then starts decreasing with further increased BMEP. Among these four SOI timings, 160° BBDC gives the highest BTE at all engine loads. Figure 4(b) shows the BSFC variation with BMEP with different fuel injection strategies. It may be noted that BSFC was relatively higher, both at high and low engine loads (Hassan et al., 2009; Aslam et al., 2006). At high engine loads, requirement of enriched fuel-air mixture in order to increase torque output increases fuel consumption disproportionally whereas large pumping losses at low engine loads increase BSFC. Lowest BSFC was obtained at 2.5 bar BMEP (for 160° BBDC SOI) was 0.26 kg/kWh. Figure 4(c) shows the EGT variation with BMEP for different fuel injection timings. EGT increased with increasing engine load and shows highest temperatures for 170° BBDC SOI. Advanced SOI timings gave relatively lower BSFC, higher BTE and EGT and vice versa. Advanced SOI timings gave more time for fuel-air mixing therefore made the combustible mixture more homogeneous, which results in improved combustion. This also reduced pumping losses during the intake. These factors played an important role in improving the engine performance. On the other hand, retarded fuel injection timings reduced the time available for fuel-air mixing and decreased the fuel jet penetration distance towards the EOI in the later part of injection, after the IVC in the compression stroke. This happens due to increasing cylinder pressure, once the piston starts moving towards TDC during the compression stroke, which offers resistance to the fuel, which is still being injected by the CNG injector, operating at 50 bar. This finally results in relatively inhomogeneous fuel-air mixing, poor combustion and relatively inferior engine performance.

3.3 Combustion characteristics

The cylinder pressure-crank angle history (P-θ diagram) was obtained at different engine loads for CNG DI engine at various SOI conditions. Figure 5 shows P-θ diagram for various fuel injections strategies at different BMEP.

In-cylinder pressure increased with increasing BMEP for all injection strategies. Increasing fuel quantity was injected into the cylinder at higher engine loads and improved volumetric efficiency in these conditions resulted in faster combustion, leading to higher peak in-cylinder pressures. Peak in-cylinder pressure and crank angle, at which this peak pressure occurs, are shown in Figure 6.

Advanced fuel injection timings (160° and 170° BBDC) produced higher in-cylinder pressures at all engine loads. Better homogeneous mixture formations due to advanced injection helped in improving combustion hence relatively higher peak pressures are seen. Figure 6(b) shows that crank angle corresponding to the peak in-cylinder pressure shifts towards TDC for advanced SOI compared to later one. Slower flame development during the late injection cases may be a possible reason for this, however it needs to be verified in an optical engine experimentally.

Figure 7 shows the variation of heat release rate (HRR) for different BMEP at different SOI timings.

Figure 5 P-θ diagram for different BMEP (see online version for colours)

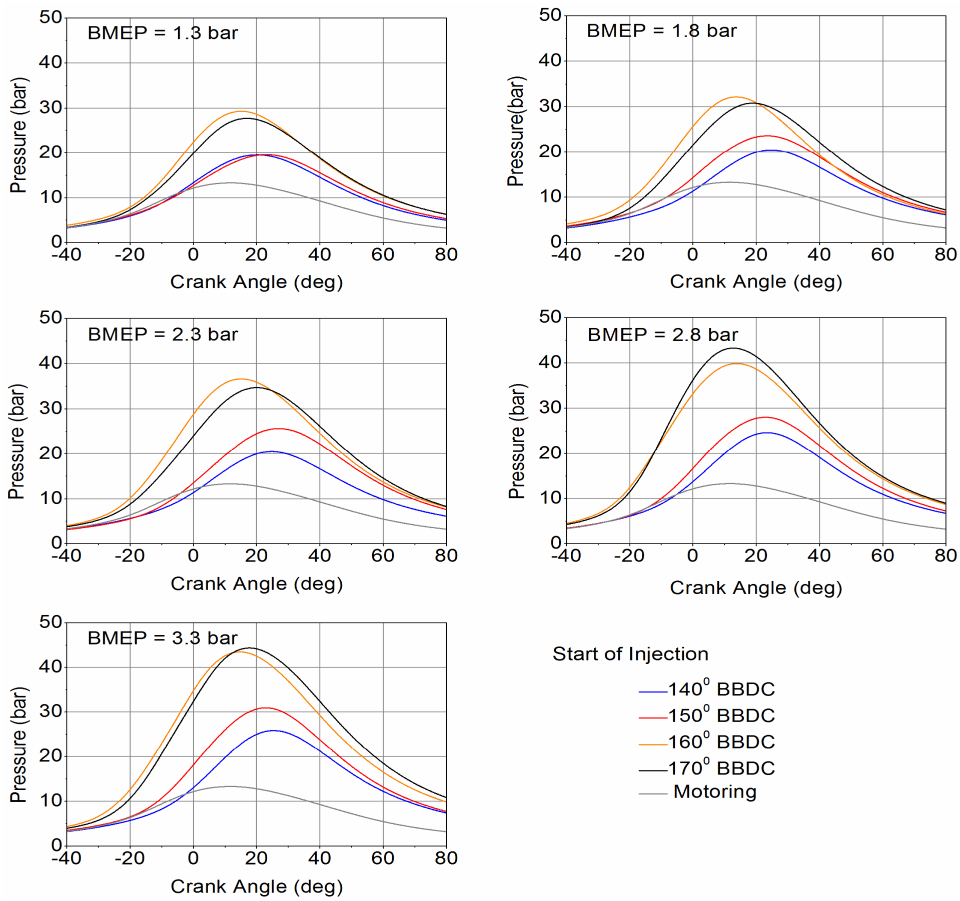


Figure 6 (a) Maximum in-cylinder pressure vs. SOI and (b) crank angle at maximum in-cylinder pressure vs. SOI (see online version for colours)

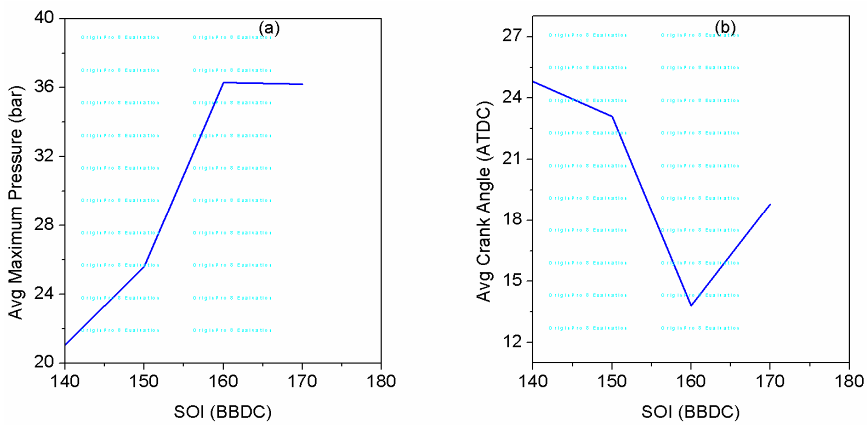
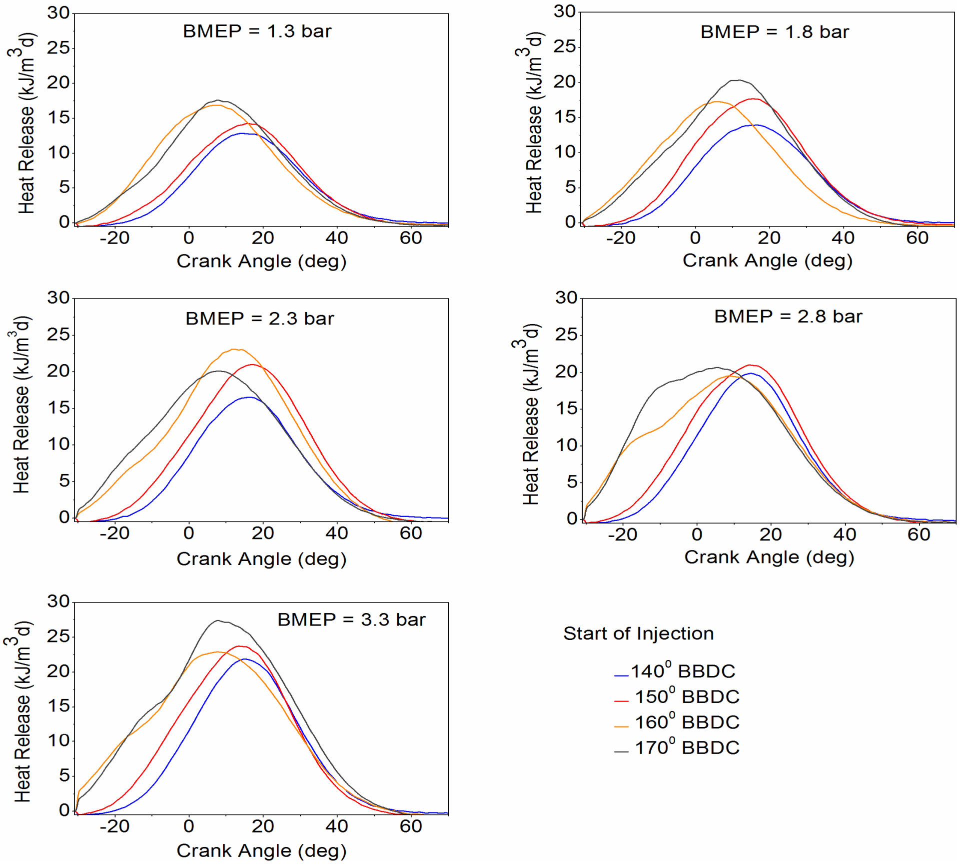


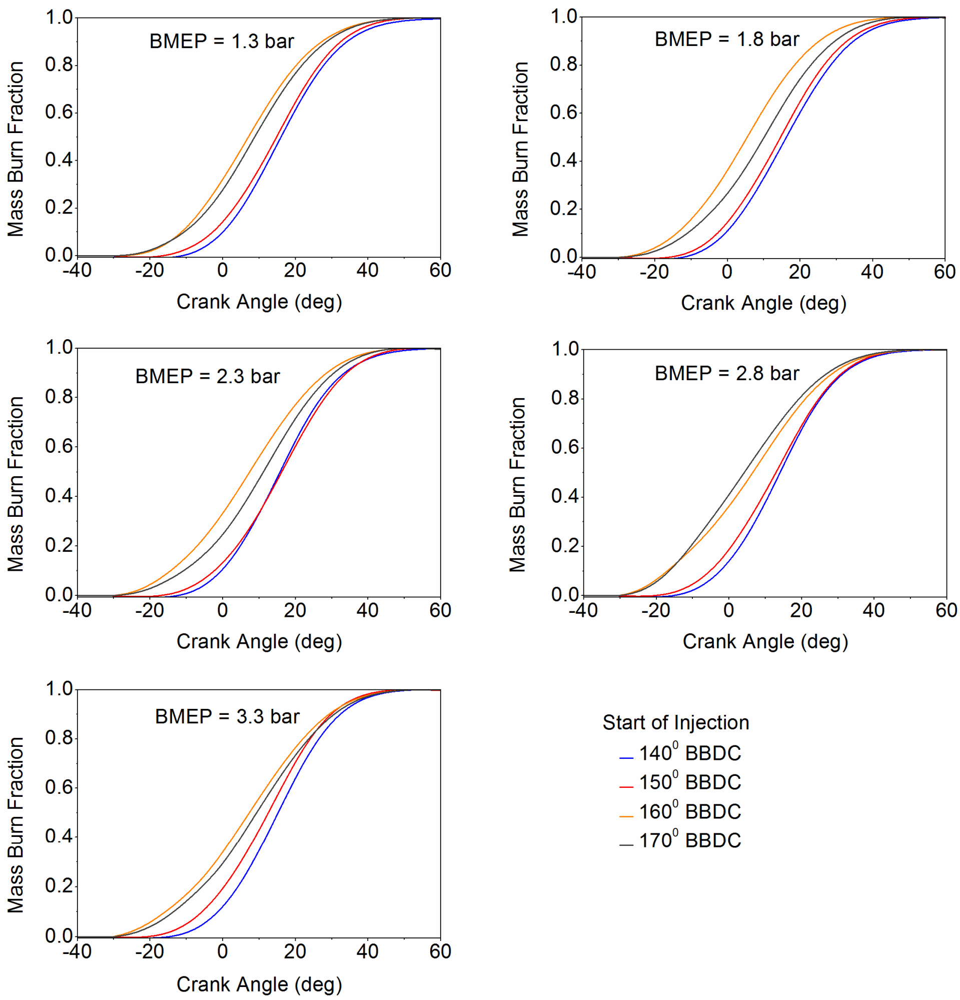
Figure 7 Heat release rate curves for different BMEP (see online version for colours)



HRR increased with increasing BMEP for all fuel injection strategies. This was due to higher fuel quantity injected and faster burn rates of combustible mixtures at higher loads. The results also showed that for most engine loads, advanced injection (160°, 170° BBDC) gave higher HRR compared to their retarded counterparts however at a BMEP of 2.8 bar, both advanced and retarded injections gave almost equal maximum HRR. At all engine loads, the crank angle, at which, maximum HRR occurred was closer to TDC for advanced injection strategies as compared to the retarded ones, resulting in highest pressure rise at the start of expansion stroke. This pointed towards possibly higher flame speeds and faster combustion for advanced fuel injections, because of adequate time availability for fuel-air mixing.

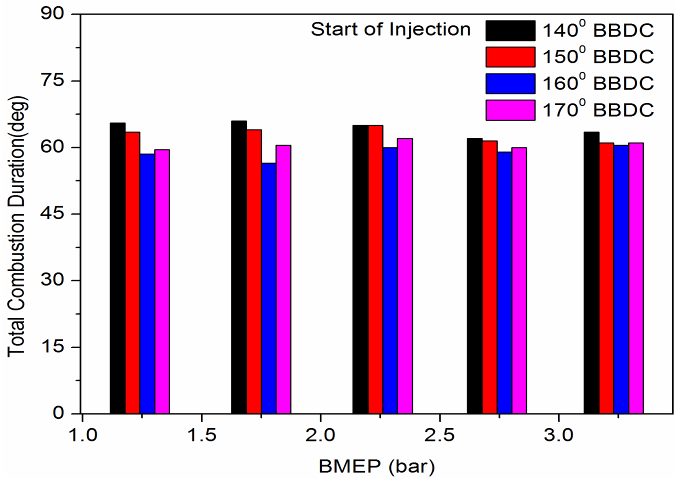
Figure 8 shows the percentage of mass burn fraction (MBF) at different engine loads for various fuel injection strategies.

Figure 8 MBF vs. CAD (see online version for colours)



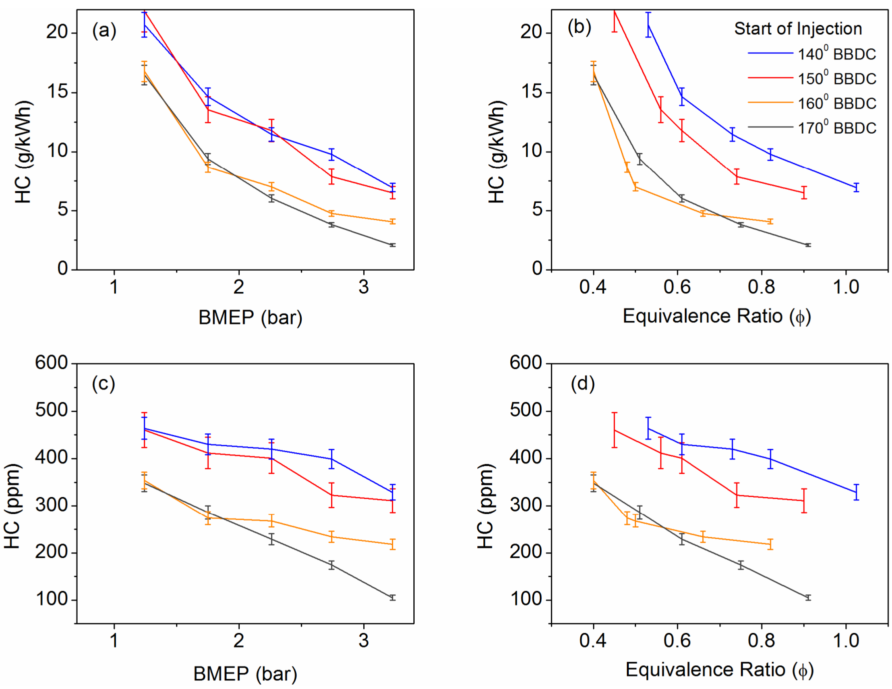
Total combustion duration is defined as the interval (in CAD) from 10% to 90% MBF due to combustion. Figure 9 shows the variation of combustion duration for different engine loads for different SOI timings. Figures 8 and 9 indicate that for all engine loads, late fuel injection leads to longer combustion duration, whereas advanced injection leads to shorter combustion duration. SOI at 160° BBDC was later compared to 170° BBDC (Figure 2), resulting in higher volumetric efficiency and EOI was relatively earlier in the compression stroke therefore it had diminishing effect of lower jet penetration. Both these factors are responsible for improved fuel-air mixing, resulting in superior combustion therefore lower combustion duration.

Figure 9 Combustion duration vs. BMEP (see online version for colours)



There is a slight increase in combustion duration for 170° BBDC SOI timing compared to 160° BBDC SOI inspite of advanced injection of fuel. It is possibly due to slightly higher BSFC at 170^0 BBDC SOI timing (Figure 4).

Figure 10 Unburnt hydrocarbon emissions (a, b) mass emissions of HC vs. BMEP and equivalence ratio (c, d) raw emissions of HC vs. BMEP and equivalence ratio (see online version for colours)



3.4 Emission characteristics

Figure 10 shows raw HC emissions and HC mass emissions for varying engine loads and equivalence ratios respectively with different fuel injection strategies. The results show that at lower BMEP (lower equivalence ratio), HC emission were higher, and it decreased with increasing BMEP and then started increasing with further increasing BMEP because the equivalent ratio exceeded beyond 1.0. At lower engine loads, combustible mixture was too lean to burn completely due to relatively lower flame propagation speeds, which leads to higher HC emissions. On the other hand, lack of oxygen in richer combustible mixtures at higher engine loads increased HC levels in the exhaust. The range of HC emissions emitted from the CNG DI engine over the entire load range varied from 2.0 to 20.7 g/kWh. It was also observed that late injection timing increased fraction of unburnt fuel (HC) in the exhaust compared to advanced injection timings because of inadequate time availability for fuel-air mixing leading to relatively inferior combustion (Goto, 1999).

Figure 11 NOx emissions (a, b) mass emissions of NO vs. BMEP and equivalence ratio (c, d) raw emissions of NO vs. BMEP and equivalence ratio (see online version for colours)

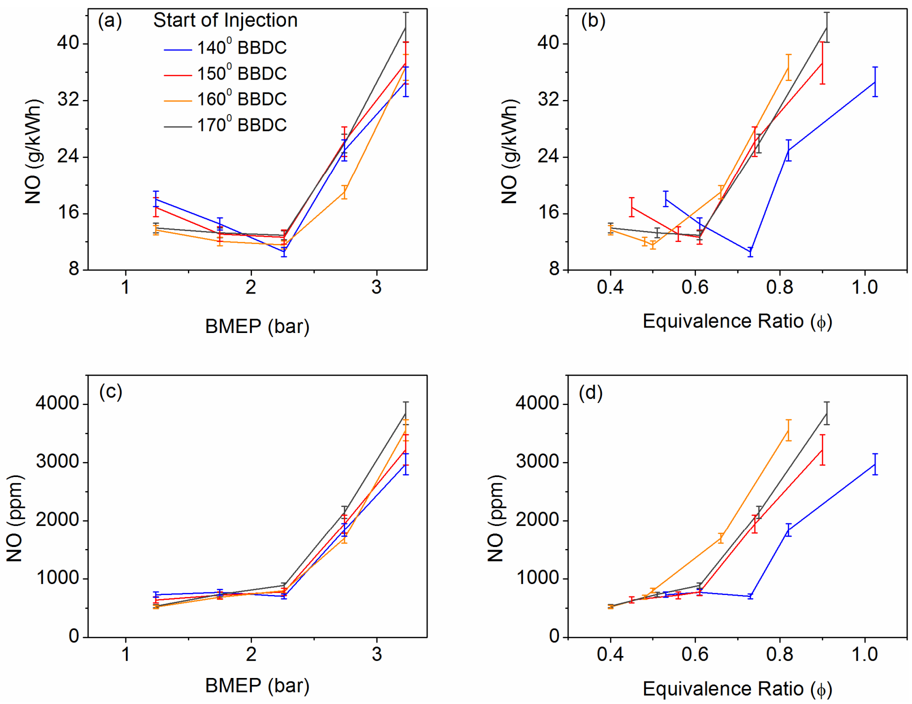


Figure 11 presents NO emissions vs. BMEP and vs. equivalence ratio for various SOI timings at constant speed and fuel injection pressure. It was found that at lower BMEP i.e., low equivalence ratio, NO emissions were lower, which increased rapidly with increasing equivalence ratio (up to 0.9) and then trend may reversed with further increase in equivalent ratio beyond 1.0. At lower engine loads (BMEP), relatively lower combustion temperatures dominate over the high oxygen concentration, resulting in lower NO levels. However, at higher BMEP with equivalence ratios in the range of 0.8 to 0.9, high combustion temperature and sufficient excess oxygen availability produced high NO emissions. Test results showed that NO emissions obtained from CNG DI engine were in the range of 12-42 g/kWh. It was also seen that NO was higher for advanced fuel injections due to relatively faster and more complete combustion and vice versa (Goto, 1999).

Figure 12 describes the variation of CO in the exhaust vs. engine load (BMEP) and equivalence ratio at constant engine speed and different SOI timings. Emission of CO is strongly related to fuel-air mixture strength. The results showed that for lower BMEP (i.e., lower equivalence ratio) due to leaner mixture, CO emissions were very low due to availability of excess oxygen in the engine combustion chamber. However CO emissions increased with increasing equivalence ratio because of formation of richer fuel zones, where sufficient oxygen was not available, leading to relatively incomplete combustion. Different fuel injection strategies showed minor variations in CO emissions, which were in the range of 0.05%–0.1% (v/v). This range was quite low compared to typical gasoline engine CO emissions. Due to inferior combustion observed under relatively retarded fuel injections conditions, slightly higher CO levels were observed.

Figure 12 CO emissions (a, b) mass emission of CO vs. BMEP and equivalence ratio (c, d) raw emissions of CO vs. BMEP and equivalence ratio (see online version for colours)

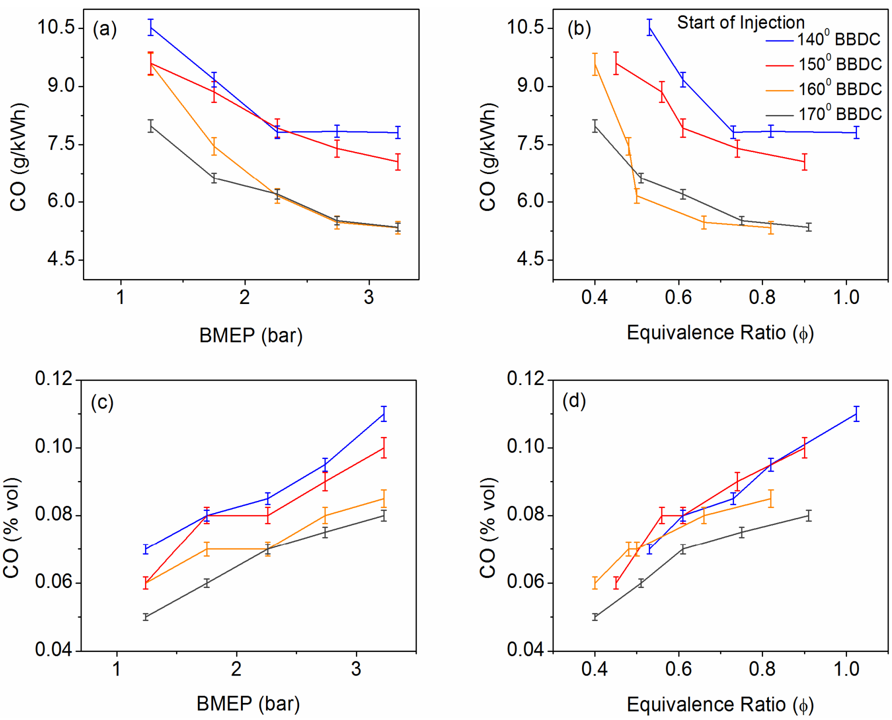


Figure 13 shows variation of CO_2 emissions with the engine load (BMEP) and equivalence ratio under different SOI timings. It can be noticed that CO_2 emissions increased with increasing BMEP and varied from 4-8% (v/v). On comparing these values with typical gasoline SI engine emissions, it can be observed that CNG DI engines produce roughly 20% lesser CO_2 emissions (Aslam et al., 2006) and this was due to lower carbon to hydrogen ratio of the natural gas compared to gasoline. The results also showed that CO_2 emissions increased with retarded fuel injection inspite of its inferior combustion characteristics. The possible reason may be that the mass of fuel required for catering to a particular load was higher in later injection strategy compared to an advanced injection strategy.

Figure 13 CO_2 emissions (a, b) mass emissions of CO2 vs. BMEP and equivalence ratio (c, d) raw emissions of CO_2 vs. BMEP and equivalence ratio (see online version for colours)

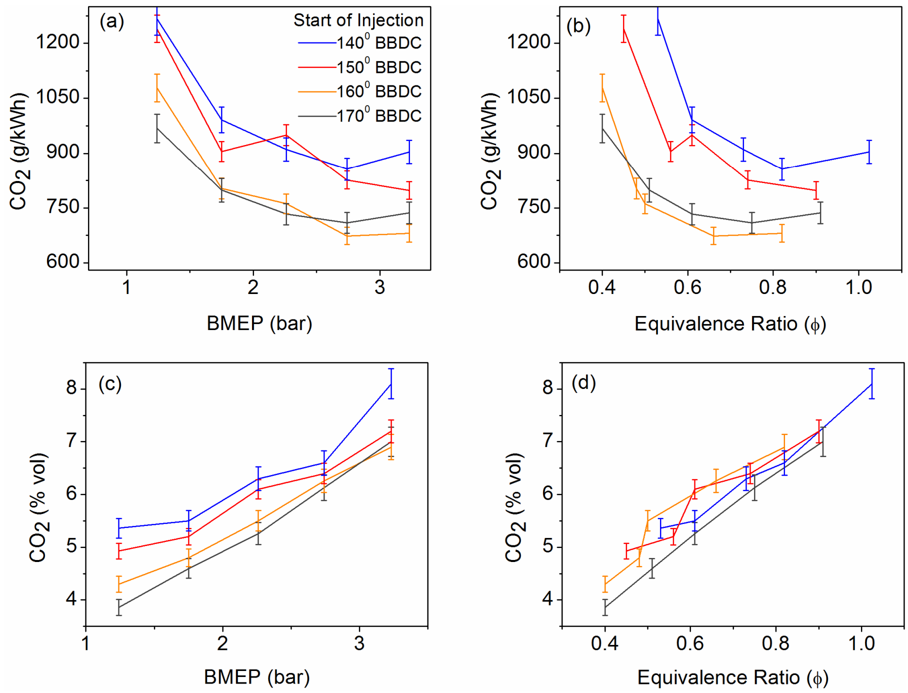
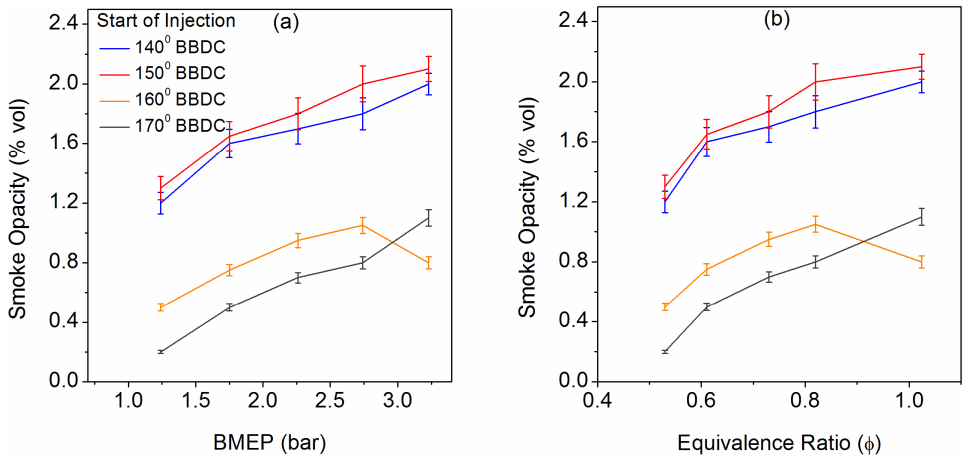


Figure 14 shows the variation of smoke with BMEP and equivalence ratio at constant engine speed under different fuel injection timings. Smoke opacity increased with increasing BMEP as higher BMEP corresponds to richer mixtures in the combustion chamber. The test results showed that the level of smoke emitted by CNG DI engine was in the range of 0.2%–2.1%, which is insignificantly small. With retarded fuel injection timings, smoke opacity of exhaust from CNG DI engine was comparatively higher than the advanced injection timings. Presence of partially burnt/unburnt mixture in the exhaust due to incomplete combustion during late injection might be the reason for this observation.

Figure 14 (a) Smoke opacity vs. BMEP (b) Smoke opacity vs. equivalence ratio (see online version for colours)



4 Conclusions

Experimental investigations of performance, emissions and combustion characteristics of a CNG DI engine operated under different start of fuel injection (SOI) timings and varying loads were carried out and the following conclusions are drawn:

1. Volumetric efficiency decreased with relatively advanced SOI timings. Advanced SOI timings (170°, 160° BBDC) produced lower BSFC, higher BTE and higher EGT compared to retarded SOI timings (150°, 140° BBDC) for all engine loads. This may be due to relatively superior mixture homogeneity for advanced injection strategy. Advanced injections resulted in relatively lower HC and higher NO emissions and vice-versa. CO_2 emissions were higher for retarded injection timings, possibly due to the requirement of higher fuel quantities.

2. At very low and very high BMEP, performance characteristic curves displayed relatively higher BSFC, and lower BTE, whereas intermediate BMEP exhibited lowest BSFC and highest BTE for all SOI timings. HC emissions decreased with increasing engine load up to an equivalence ratio of 0.9 whereas NO and CO_2 emissions increased in this zone for all fuel injection strategies. Both CO emissions and smoke opacity displayed marginal variations in the lean mixtures however the variations increased for richer mixtures at higher BMEP.

3. In-cylinder pressure and heat release rate trends showed lower values for retarded fuel injection strategies compared to the advanced ones. Slower, inferior combustion due to lower jet penetration and relatively lower flame propagation speeds might be possible reasons for this in case of late injection strategies. Moreover, MBF curves and total combustion duration curves exhibited relatively longer combustion duration for retarded fuel injection strategies and shorter combustion durations for advanced injection strategies.

Abbreviations

CNG	Compressed natural gas
BMEP	brake mean effective pressure
BTE	Brake thermal efficiency
BSFC	Brake specific fuel consumption
BSEC	Brake specific energy consumption
EGT	Exhaust gas temperature.

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Appendix

Volumetric efficiency

In intake system, orifice plate along with U-tube manometer is used to measure the actual mass flow rate entering into the engine combustion chamber.

- actual air flow rate

$$A_f = C_d \times \left(\frac{\pi d^2}{4}\right) \times \left[\sqrt{2gH\left(\frac{\rho_w}{\rho_a}\right)}\right] \times \rho_a \times 3,600$$

- theoretical air flow rate

$$A_T = \frac{\pi}{4}\left(D^2 L\right) \times \frac{N}{2} \times n \times \rho_a \times 60.$$

Spark plug and the gas injectors optimised position has been calculated using following equivalence ratio contour in CNG jet (Goto, 1999). In present experimental study, a GDI injector having nozzle hole diameter 1 mm is used. So, for $\Phi = 1$ and injection angle (θ) equal to 20°, the distance between spark plug and injector hole is chosen as 35 mm.