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Compressed Natural Gas Direct Injection: Comparison Between Homogeneous and Stratified Combustion

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Abstract

Due to abundance of natural gas, the use of natural gas for automotive use, particularly for internal combustion engine (ICE), is more practical and cheaper than their future successors. Even though natural gas is a cleaner fuel than other fossil fuels and has a higher octane number and can lead to higher thermal efficiency, its low carbon number makes it less attractive as compared to gasoline and diesel. Based on its potential, an engine referred to as compressed natural gas direct injection engine (CNGDI) was designed, developed and tested to operate on compressed natural gas (CNG) as monofuel directly and centrally injected into the engine. Computational and experimental works have been performed to investigate the viability of the design. Computational fluid dynamics (CFD) simulations and experimental works with homogenous combustion showed that the results were in good agreement. From experimental works, it is found that combustion characteristics could be improved by using a stratified charge piston configuration with some drawback on performance. In terms of exhaust emissions, stratified configuration causes slight increase in the emission of CO, CO_2 and NO_x, which highlight a need for further study on this issue.

Keywords: CNGDI, internal combustion, performance, emissions, homogenous mixture, stratified mixture

1. Introduction

In development of any engine, it is desirable to optimise the engine parameter and configuration in order to maximise its performance while keeping the emission within stipulated limits.

Due to abundance of natural gas, the use of natural gas for internal combustion engine (ICE) is more practical and cheaper than their future successors, such as electric and fuel cell cars. Several advantages related to natural gas utilisation in ICEs are its higher thermal efficiency and relatively lower exhaust emissions due to the higher octane level and lower ratio of carbon and hydrogen ratio, respectively [1].

It is understood that configuring conventional ICEs to improve efficiency while reducing exhaust emissions is difficult where strategies to improve engine efficiency will eventually increase harmful emissions, such as carbon monoxide (CO) and nitrogen oxides (NO_x) [2]. The significant advantage that compressed natural gas (CNG) has in antiknock quality is related to the higher auto-ignition temperature and higher octane number compared to that of gasoline. As the air-fuel ratio for natural gas is 17.23, which requires less fuel required that is less compared to other fuels such as gasoline and diesel, it is possible to obtain the maximum cylinder pressure and rate of heat release at the shorter combustion duration through a right combination of fuel injection, valve and ignition timings while keeping a low level of HC and CO emissions [3].

The use of alternative fuels, including CNG, has attracted popularity along with the importance of emergent alternative fuel technology because the progressively strict regulatory limits on emission levels [4]. Direct injection (DI) in the spark ignition engine considerably raises the engine volumetric efficiency and declines the requirement to use the throttle valve for regulatory purposes [5].

Hence, this chapter presents aspects of the design and development of a compressed natural gas direct injection engine (CNGDI), which can be optimised to yield maximum performance while keeping the emission low. In order to accomplish this study, two types of mixture will be analysed, namely, homogenous and stratified mixtures.

2. Development of the CNGDI engine

In this work, a monofuel CNGDI engine was designed and developed based on a gasoline port injection (PI) engine as shown in **Figure 1**. This new engine design was designed with a specific purpose to enhance the natural gas engine performance as well as to minimise the exhaust emissions. It was also designed using two types of piston crown designs, which are a homogenous piston crown for optimum performance, and stratified piston crown for reduction of exhaust emission.

In general, the design and development process of the main components in an automotive engine are not straightforward. One of the main engine components is its cylinder head, where careful design is required for optimum performance and emission of a vehicle.

The cylinder head is also the platform that is heavily loaded with mechanisms for internal combustion process such as valve train and fuel rail [6]. Hence, the cylinder head of a direct injection engine is highly influenced by the geometry of the injector location at the combustion chamber and has to withstand a very high combustion pressure and temperature. In fact,
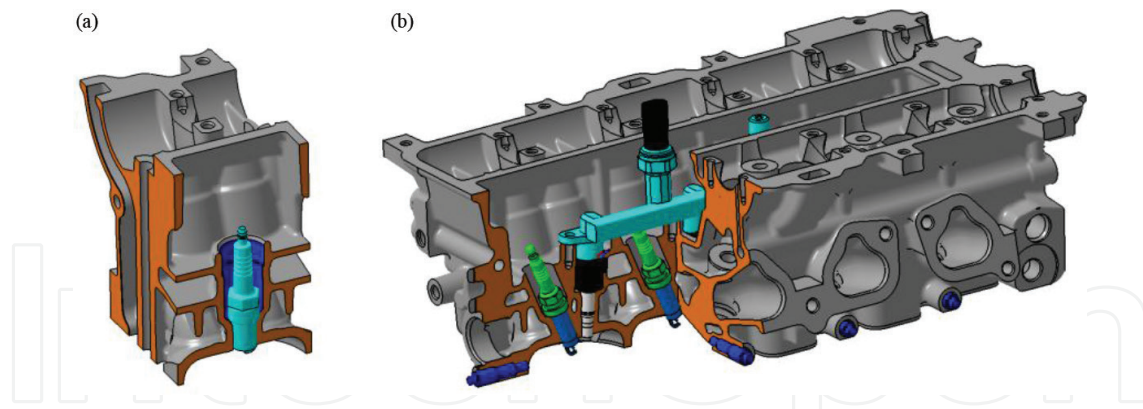
Figure 1. CNGDI engine head showing the location of spark plug and fuel injector: (a) cylinder head for referral gasoline engine and (b) cylinder head for CNGDI engine.

small changes in the cylinder head geometry can lead to considerable changes in the air-fuel mixture distribution and performance of the engine [7].

The cylinder head could also fail during operation due to thermal fatigue cracking especially in the water jacket cooling area because of narrow path between the valves and the exhaust valve seat [8], as shown in **Figure 2**. For its structural strength, the finite element analysis can be used to obtain the stress and strain profiles of the cylinder structure, which could be analysed further to ensure that the maximum stress does not exceed the allowable yield strength limit [9]. Furthermore, the stress-strain and displacement distributions at various loads and pressure can be simulated. Besides, other researchers also had conducted similar analyses under combustion loading and critical assembly parts in the cylinder head [10]. In this work, the effect of gas pressure during combustion process was examined through the stress and displacement distributions, as shown in **Figure 3**.

Designs of the cylinder head intake and exhaust were guided by a computational fluid dynamics (CFD) simulation in order to ensure the smooth flows of air-fuel mixture and combustion products into and away from the combustion chamber, respectively. With optimal
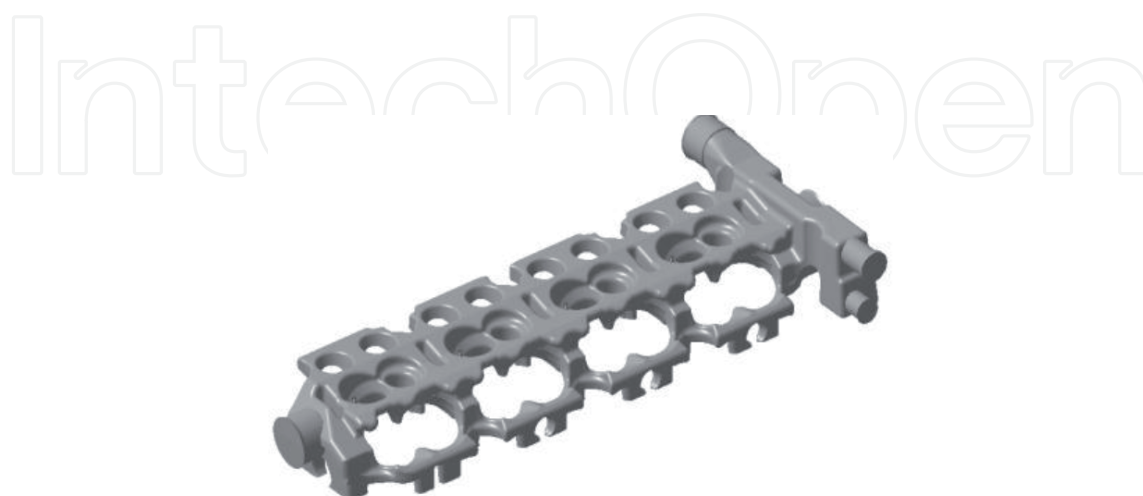

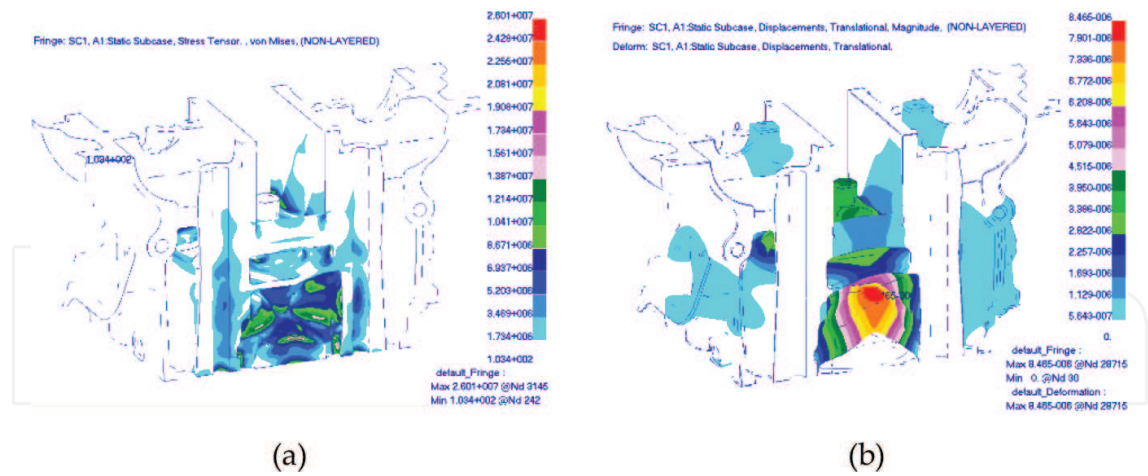**Figure 2.** Water jacket design in CNGDI cylinder head.

Figure 3. Finite element analysis of the cylinder head: (a) stress and (b) displacement.

shape design, the inflow and outflow can increase the efficiency of combustion engines and influence the engine performance and exhaust emissions.

During the design of the cylinder head, the intake and exhaust valves were orientated a few degrees from the central axis of the cylinder bore in order to accommodate the spark plug and the fuel injector vertically close to the central axis. The combined valve train and cam system was analysed for kinematic and dynamic responses in order to optimise the angle of valve orientation and some other characteristic parameter for the cam system. By combining the finite element analysis on the stress-strain profile of the cylinder head, the CFD analysis of the cooling system and the dynamic response analysis of the valve train and cam system, the improved design for cylinder head can be produced. Then, prototypes of the CNGDI single-cylinder engine and the CNGDI multi-cylinder head were fabricated by casting as shown in **Figure 4**. These prototypes were installed and single- and multi-cylinder engine test beds for further internal combustion experiments.
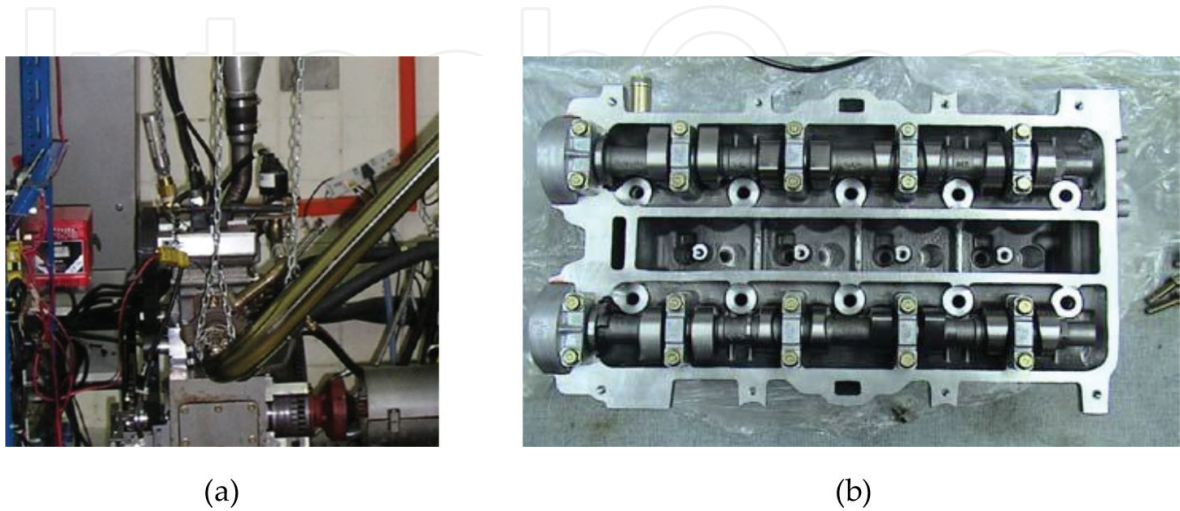

Figure 4. Cylinder head prototypes: (a) single-cylinder engine and (b) multi-cylinder head.

3. Methodology

For the internal combustion study, the analysis started with a CFD simulation of combustion process, which was used to derive the optimum shape of critical engine components that form the combustion chamber as well as the characteristic parameters such as injection and ignition timings. Based on the optimal CFD results, the experiment was performed to further explore the performance and emissions of the newly designed engine.

3.1. Engine specifications and fuel properties

A four-cylinder spark ignition engine direct injection connected to CNG tanks as the source of fuel was used in this work. The engine specifications are given in **Table 1**.

In addition, **Table 2** lists key properties of CNG for internal combustion as compared to gasoline. Based on the table, it can be deduced that there are several properties in which CNG has advantages, such as a better antiknock quality, that is related to higher auto-ignition temperature and higher octane number, as well as higher air-fuel ratio and heating value. In Malaysia, the typical composition of commercially available CNG is 94.42% methane, 2.29% ethane, 0.03% propane, 0.25% butane, 0.57% carbon dioxide, 0.44% nitrogen and 2% other compounds.

Parameter	Value	Unit
Number of cylinders	4	-
Type	Inline	-
Capacity	1596	cm^3
Bore	76	mm
Stroke	88	mm
Connecting rod length	131	mm
Crank radius	44	mm
Compression ratio	14	-
Intake valve opening	12	° before TDC
Intake valve closing	48	° after BDC
Exhaust valve opening	45	° before BDC
Exhaust valve closing	10	° after TDC
Maximum intake valve lift	8.1	mm
Maximum exhaust valve lift	7.5	mm

Table 1. Engine specifications.

Properties	Gasoline	CNG
Motor octane number	80–90	120
Molar mass (g/mol)	110	16.04
Carbon weight fraction (mass %)	87	75
Air-fuel ratio	14.7	17.23
Stoichiometric mixture density (kg/m³)	1.38	1.24
Heating value (MJ/kg)	43.6	47.4
Heating value of stoichiometric mixture (MJ/kg)	2.83	2.72
Flammability limits (vol. % in air)	1.3–7.1	5–15
Spontaneous ignition temperature (°C)	480–550	645

Table 2. Combustion-related properties of gasoline and CNG.

3.2. Model for CFD simulation

In internal combustion engine spark ignition, a complete combustion can produce better engine performance by controlling of engine combustion parameters. Therefore, a numerical study and optimisation of combustion parameters in a CNGDI engine were performed using a computational fluid dynamics (CFD) simulation with single cylinder moving mesh modelling with a source code developed and incorporated as user subroutine to the base CFD code. In developing the moving mesh and boundary algorithms, every event is made as a function of crank angle and represents different configurations of mesh and boundary geometries for an engine cycle [11], as shown in **Figure 5**.
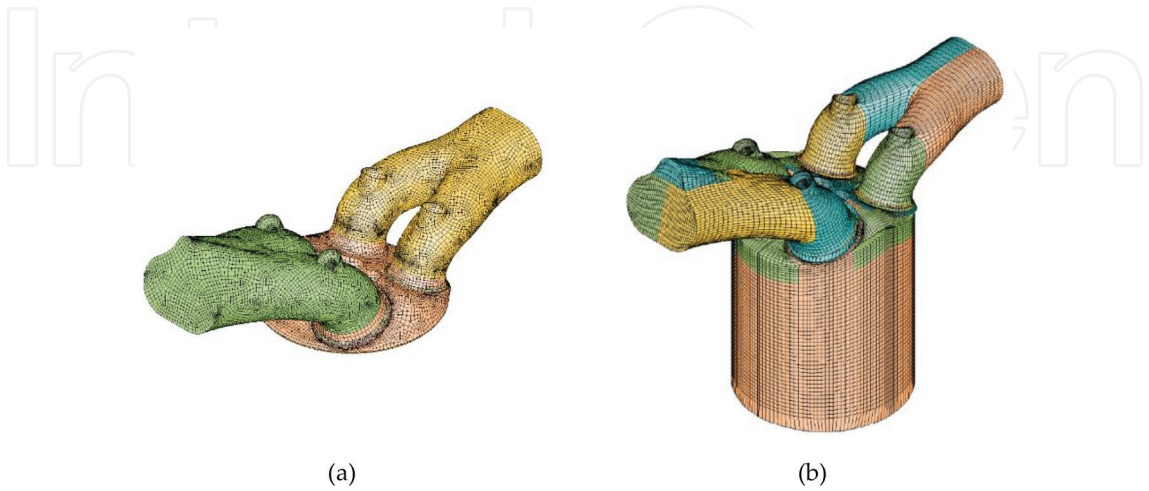

(a)   (b)

Figure 5. Mesh formation using CFD software: (a) surface mesh and (b) volume mesh.

The combustion process was simulated using the Eddy break-up model with the three global reaction schemes. The injection and ignition events were implemented to the engine computational mesh to control the combustion parameters. The CFD analysis used moving mesh simulation, which has been programmed to follow the motion of the intake valves, the exhaust valves and the piston. As the piston moves from bottom dead centre (BDC) to top dead centre (TDC), the height of the cylinder and valve position varies depending on the specific time step used and certain designated events.

During CFD simulation, several engine speeds were simulated in order to investigate the effect of the three engine parameters, namely, timings for the start of injection (SOI), the end of injection (EOI) and the spark ignition (SI), which will affect the engine performance and emission levels of CO, HC and NO. With a correct combined set of timings of SOI, EOI and SI, the indicated power can be optimised while maintaining a sound CO and NO levels. Then, the optimisation work was done by the combined CFD, the Gaussian process and genetic algorithm (GA) methods at every engine speed. The development of single-cylinder model was developed based on the following five stages:

1. The construction and generation of moving mesh for the combustion chamber model to provide an approximation of the actual piston motion

2. Modelling of the combustion process using a CFD code (STAR-CD) at speeds of 1000, 2000, 3000 and 4000 rpm

3. Validation of CFD simulations with the experimental data from the single-cylinder test bed for the above speeds

4. A parametric analysis using a coupled neural network and the Gaussian process at speeds of 1500, 2500 and 3500 rpm

5. Optimisation using multi-objective GA (MOGA) for all the seven speeds, i.e. 1000, 1500, 2000, 2500, 3000, 3500 and 4000 rpm

3.3. Experimental configuration

Using the CNGDI engine installed on a test bed, experimental investigations can be performed in order to study the performance and exhaust emissions of homogenous mixture and stratified mixture [12].

The engine test bed used in this experiment is depicted in **Figure 6**. Furthermore, investigation of optimum injection timings and the in-cylinder combustion pressure measurement of both homogeneous and stratified combustions were carried out, analysed and compared. The test was performed at engine speeds of 1500–4000 rpm. Some of the results were compared with the CFD analysis mentioned earlier. An engine control system and portable exhaust gas analyser were used for controlling engine operations and recording engine performance and emission data. The software used for the test bench shown above is Kronos 4. Fuel system had the pressure regulator to keep fuel pressure around 20 bar, which was maintained at the fuel entry into the combustion chamber. An air mass flow sensor was installed before the throttle
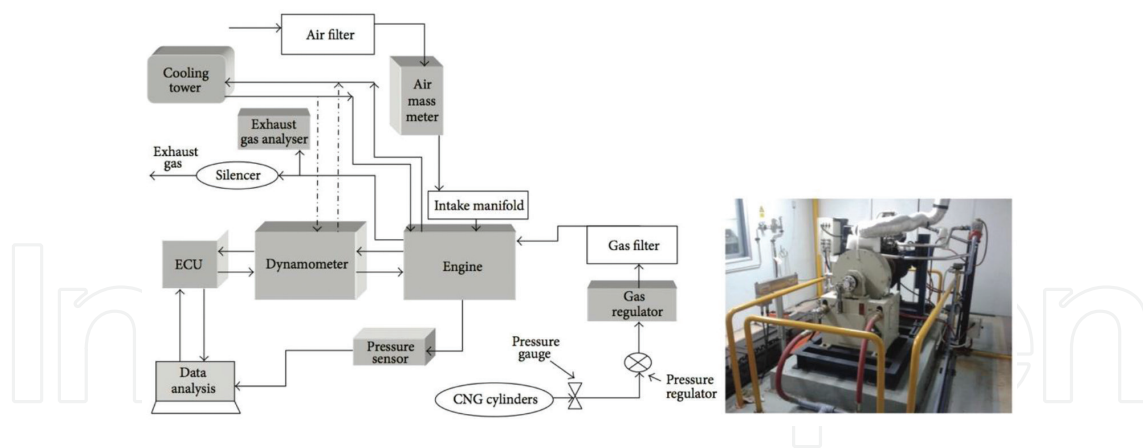
Figure 6. Experimental setup.

valve to record air mass flow, a dynamometer typed FR250 with maximum load of 800 Nm was connected and the torque measured was calibrated using the control levers and weights.

The results were recorded in a steady-state condition and at ambient pressure, temperature and humidity. All readings were recorded in order to estimate the optimal air inlet density. The portable Kane-May exhaust gas analyser was used and calibrated for each test to ensure correct results. The pressure was measured by a pressure sensor of type 6125B-Kistler in the first cylinder to record in-cylinder pressure with specified accuracy. The setting of the electronic control unit (ECU) is modified using the MoTeC software.

The engine was run under the full load wide-open throttle conditions, and the experiments were run using two types of piston configuration, as follows:

1. The homogenous piston configuration is used as shown in **Figure 7** in the first test.

2. The stratified piston configuration is used as shown in **Figure 8** in the second test.
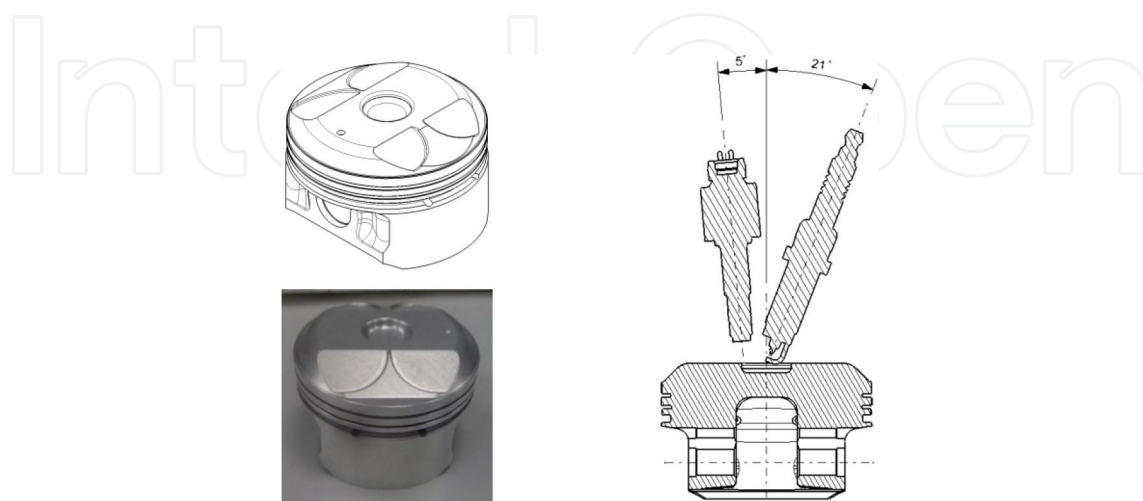

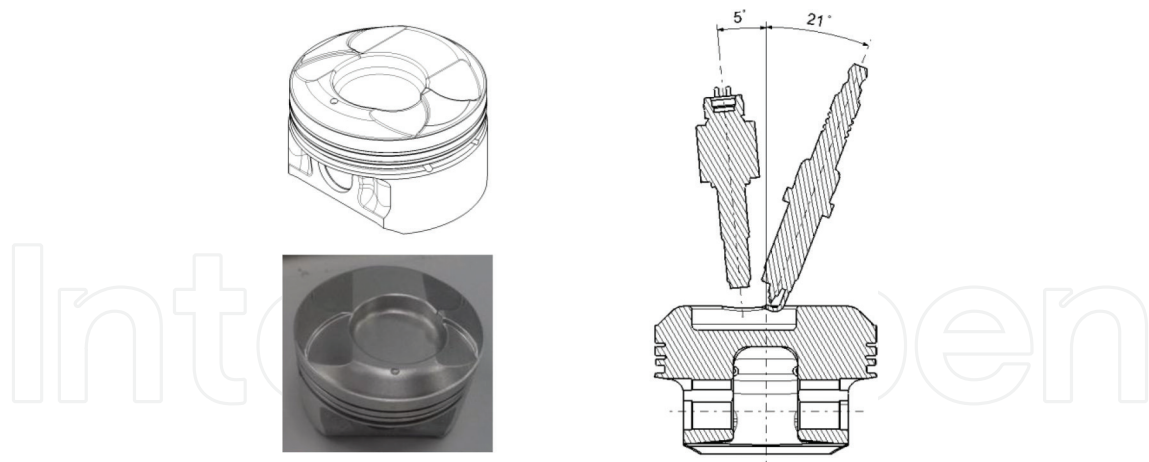**Figure 7.** Homogenous piston configuration.

Figure 8. Stratified piston configuration.

4. Results and discussion

This section displays some results from the CFD simulation, which has been used as the basis for design optimisation of the cylinder head before it was fabricated. Then, the cylinder head together with the valve train, the cam system and the fuel rail was installed on the current based engine on the test bed before it was tested for various engine speeds.

4.1. CFD simulation of combustion process

The CFD simulation results of the same work have been presented and discussed in Refs. [1, 2, 11]. The simulation was performed on a combustion chamber of the engine cylinder specified in **Table 1** for a stoichiometric air-fuel mixture. The simulation results for 3000 rpm were given in **Figure 9** for different crank angles, where the fuel is considered as 100% methane [13].

The simulation depicts the progression of air-methane profile in a shrinking cavity of combustion chamber at pre-ignition stage during the compression stroke. The post-ignition stage
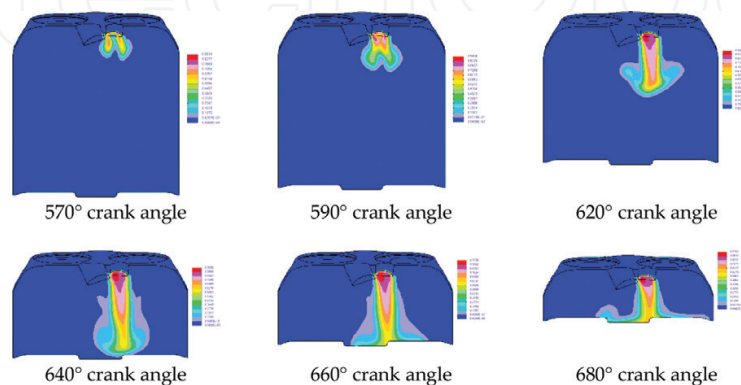

Figure 9. Simulated air-fuel mixture contour during compression stroke.

resulted in rapid increase of in-cylinder pressure, which produced torque and power output as depicted in **Figure 10** in the graph of pressure with the engine displacement. The CFD results were compared with the experimental results as explained in the next section.

4.2. Comparison between homogenous and stratified combustions

The next subsections show the results obtained from experiment using both types of piston configuration, where the results are plotted in the same graphs. The results for power and torque were also compared with the CFD results for homogenous piston configuration. Together with the CFD simulation, the experiment was limited to a speed of 4000 rpm.

4.2.1. Power and torque

Figure 11 shows the brake power and brake torque with the engine speed. The results were recorded from 1500 to 4000 rpm for both homogenous and stratified combustions. From the results, higher power and torque were obtained in stratified combustion with improvement observed at low speeds between 1500 and 2500 rpm. The maximum power recorded with stratified combustion was 54.75 kW at 4000 rpm, which is 3% higher than that of homogenous combustion. This finding is consistent with the study carried out by Sendyka and Cygnar [14] on gasoline direct injection engine.

In both combustion modes, the torque curves exhibit dual-peak profile, which is the typical characteristic of a double overhead cam (DOHC) engine with unmodified cam profile as used in the present investigation. Higher power obtained is the results of higher pressure and higher heat release rate. In addition, a late injection timing with high pressure and suitable combustion duration increases the engine performance, and good propagation flame is obtained [15]. Furthermore, high heat released with the lean mixture produced high indicated power.
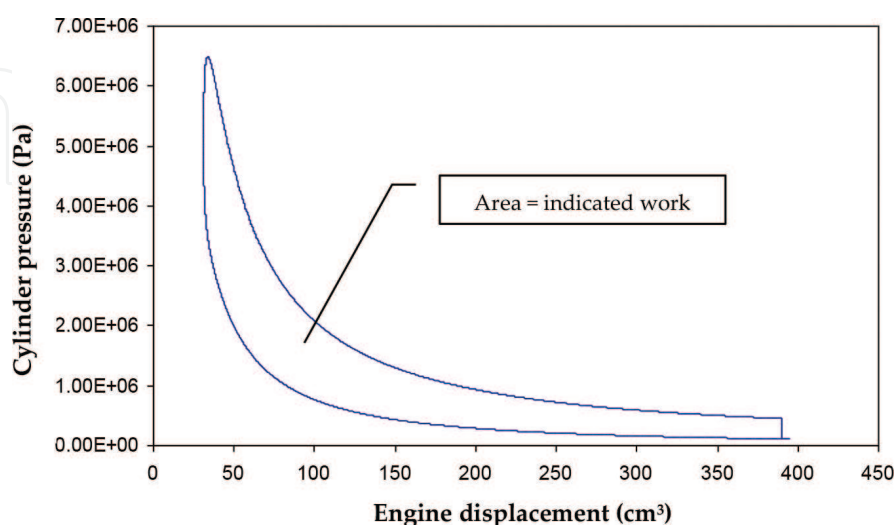

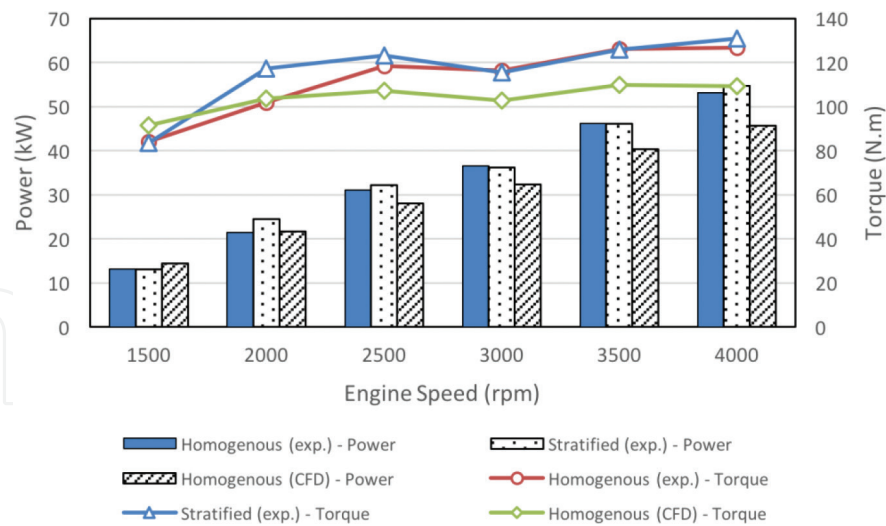**Figure 10.** Simulated pressure and engine displacement in a *p-v* diagram.

Figure 11. Power and torque of homogeneous and stratified combustions at different engine speeds.

As comparison, in another study by Kalam and Masjuki [16], the average brake power over the test cycle obtained was 47.39, 36.90 and 45.37 kW the gasoline port injection, CNG bi-fuel and CNG direct injection engines, respectively, using homogenous piston. The main factor affecting the brake torque is the lack of chemical energy conversion to mechanical energy, which is strongly related to volumetric efficiency, fuel mixing, net heat release rate and cylinder pressure [12].

4.2.2. Brake mean effective pressure

Generally, the brake mean effective pressure (BMEP) is affected by heat release rate, good mixture and sufficient combustion time. **Figure 12** depicts the BMEP with the engine speeds. The results show that BMEP increases with the engine speed and higher BMEP was obtained with the stratified combustion, especially at several low speeds as shown. The maximum value recorded for the stratified combustion is 10.3 bar at 4000 rpm.
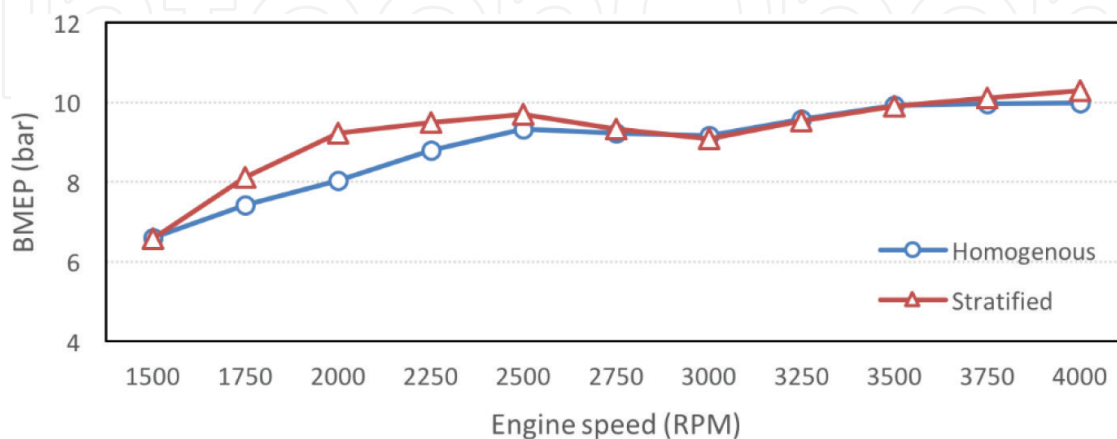

Figure 12. BMEP of homogeneous and stratified combustions at different engine speeds.

4.2.3. Brake-specific fuel consumption

Figure 13 shows brake-specific fuel consumption (BSFC) versus engine speed. The results show a lower BSFC obtained with stratified combustion in comparison with homogenous combustion for all engine speeds.

The main reason is that stratified combustion has lower fuel consumption where the fuel surrounding the spark plug is richer. This leads to ignition and initiation of combustion at this area that produces sufficient energy to propagate the flame and sustain the combustion smoothly through the leaner layers of air-fuel mixture, resulting in a more efficient combustion. This is consistent with the experiment by Baeta et al. on their torch ignition engine [17].

4.2.4. Lambda and volumetric efficiency

Figure 14 depicts the lambda and volumetric efficiency versus engine speed. The recorded lambda was more than one ($\lambda \geq 1$) accounting for a lean combustion which in turn decreases the fuel consumption. The lambda is more at the stratified combustion than the homogenous combustion. Consequently, for the stratified combustion, the volumetric efficiency is higher. From these results, there were two critical points at engine speeds, i.e. at 2000 and 3000 rpm as shown by the volumetric efficiency values in the graph.

4.2.5. Combustion pressure

Based on the same experimental data, the in-cylinder pressure generated from the combustion process is plotted against engine displacement or piston swept volume. **Figure 15** shows the cylinder pressure of the homogenous combustion versus cylinder volume and crank angle. The maximum recorded pressure was 186 bar at 4000 rpm just after the top dead centre.

From the *p-v* diagram, the indicated work and heat release can be calculated. Other factors affecting the *p-v* cycles include the air-fuel ratio and ignition timing. At 2500 rpm, the BMEP
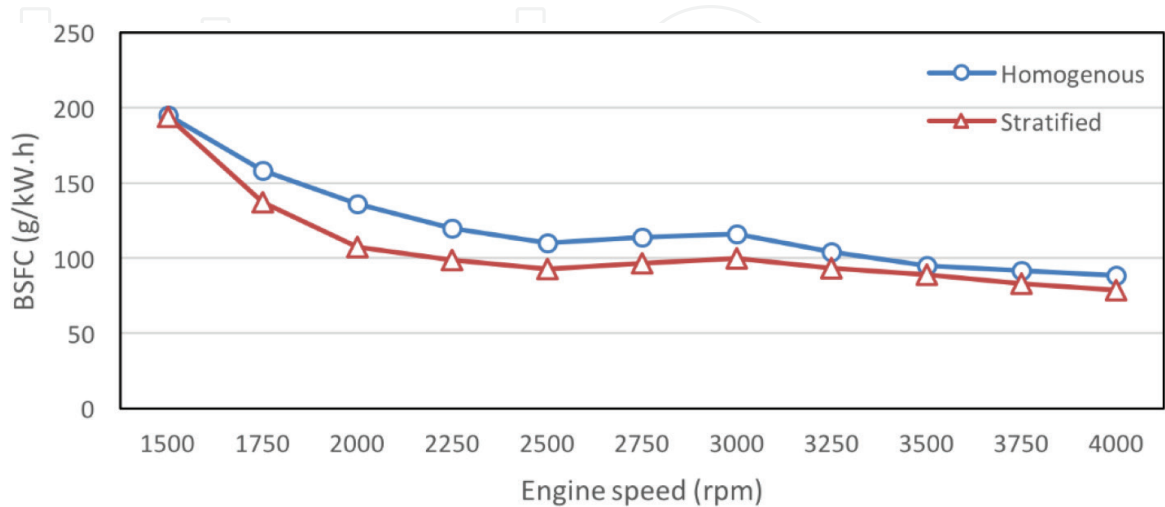

Figure 13. BSFC of homogeneous and stratified combustions at different engine speeds.
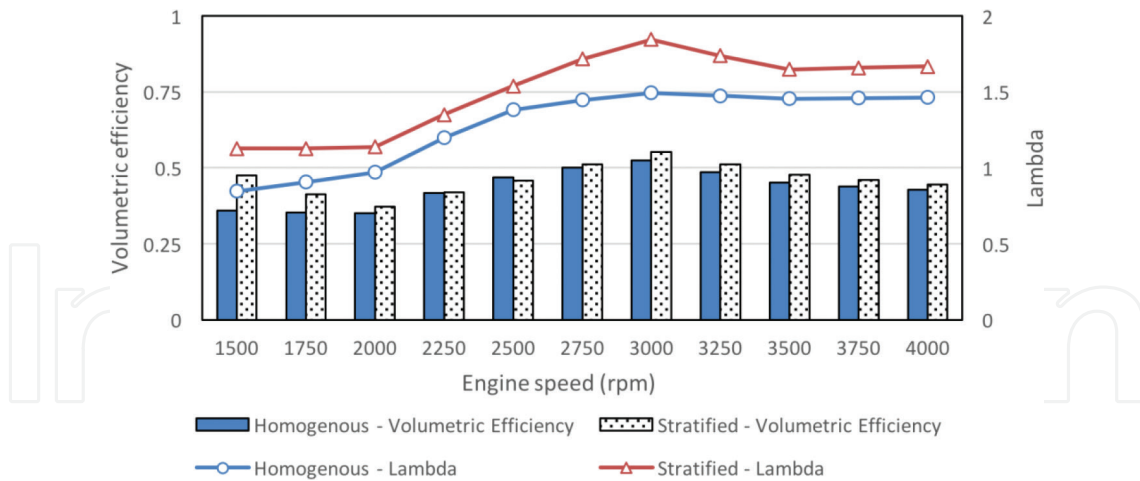
Figure 14. Lambda and volumetric efficiency of homogeneous and stratified combustions at different engine speeds.

and torque produced higher than that of 3000 rpm, which explains the double-peak phenomenon for the DOHC engine with fixed cam profiles. The work then increased at 3500 rpm and eventually at 4000 rpm. The combustion pressure influences the brake mean effective pressure as shown, which in turn affects the power and torque produced in **Figure 11**. All the peak pressure occurred just after TDC.

As comparison, **Figure 16** shows the cylinder pressure of a stratified combustion versus engine displacement and crank angle. A high combustion pressure of 145 bar was observed at 4000 rpm at 7° after TDC. However, all the peak values of pressure are about 20% lower than the ones produced by the homogenous combustion.

Similarly, the indicated work at 3000 rpm is lower than at 2500 rpm and subsequently at 3500 and 4000 rpm. Consequently, the BMEP is more at 2500 rpm than at 3000 rpm with values of 9.69 and 9.08 bar, respectively, as shown in the figure. The most likely reasons that lead to this kind of pressure degree variation are the same with that for the homogenous combustion.
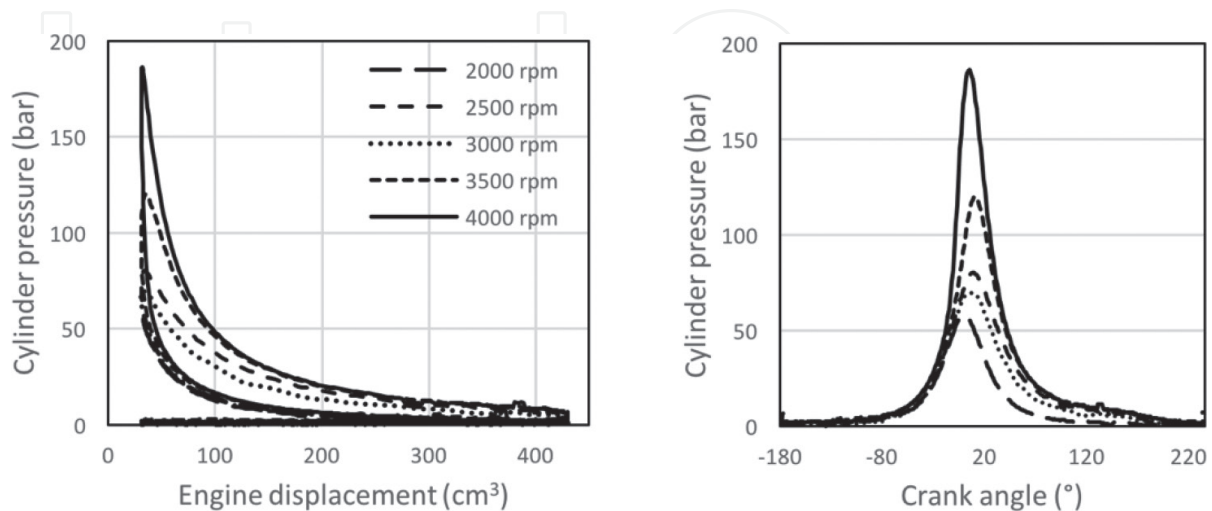

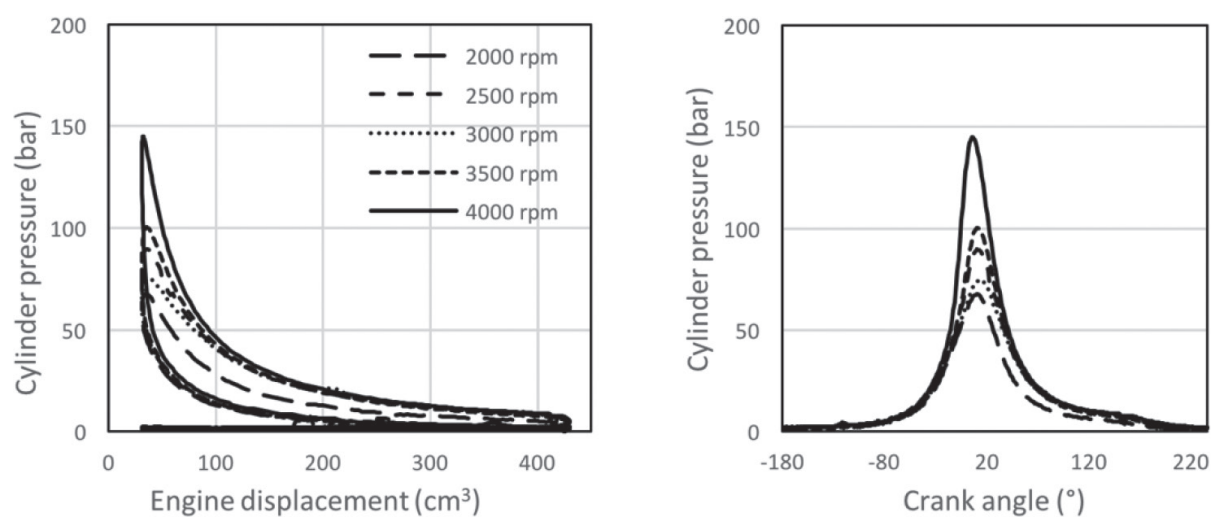**Figure 15.** Cylinder pressure in homogenous combustion.

Figure 16. Cylinder pressure in stratified combustion.

4.2.6. Emissions

For the final sets of result, various emissions are plotted versus the engine speeds as shown in **Figure 17**. The results demonstrate a high CO at stratified combustion at a low engine speed, but CO decreases at high speed compared to homogenous combustion. Meanwhile, CO_2 is also lower for stratified combustion in high speed, whereas the value of CO_2 increases with the engine speed increase for both combustion modes.
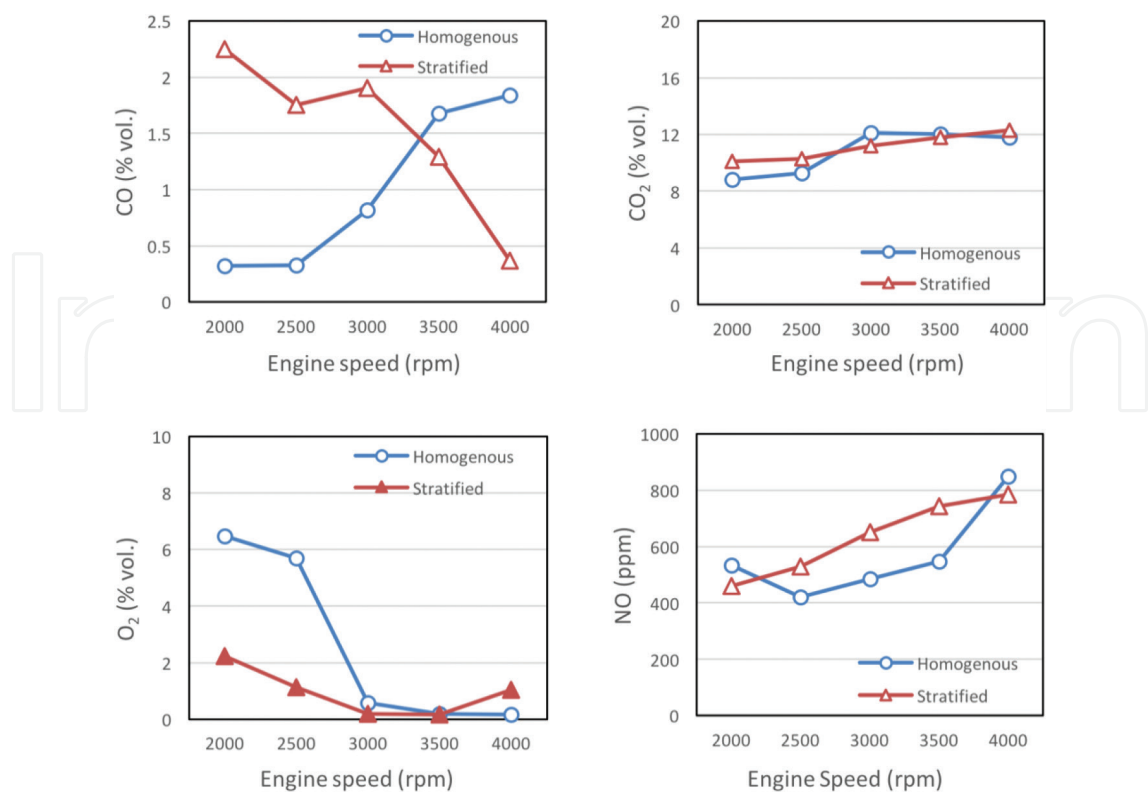

Figure 17. Emissions for homogeneous and stratified combustion at different engine speeds.

Moreover, NO is recorded lower with stratified combustion mode only at 2000 and 4000 rpm, while slightly higher than homogeneous combustion at other speeds. However, the study of Baeta et al. [17], which employed the side injection strategy as opposed to the central injection strategy used in this work, reported reductions of NO_x, CO and CO_2 with stratified combustion in gasoline engine. This highlights the need for further studies, which are underway to look into the emission issue for the central injection strategy, particularly on NO.

5. Conclusions

A CNGDI engine that operates on monofuel, namely, natural gas, has been designed, developed and tested. Computational and experimental works have been performed to investigate the viability of the new design. Baseline computational and experimental works with homogenous combustion showed that the results were comparable. From computational work, detailed CNGDI homogeneous combustion characteristics were obtained via CFD simulation and studied. Combustion characteristics could be improved by using a stratified charge piston configuration. Experimental work on stratified piston configuration has shown that it improves the engine torque and power especially at low speeds as compared to the homogeneous piston. In terms of exhaust emissions, stratified CNGDI design causes slight increase in the emission of CO, CO_2 and NO_x. This highlights the need for further studies, which are underway to look into this emission issue.

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References

[1] W.H. Kurniawan and S. Abdullah. Numerical analysis of the combustion process in a four-stroke compressed natural gas engine with direct injection system. Journal of Mechanical Science and Technology. 2008;**22**(10):1937-1944. DOI: 10.1007/s12206-008-0737-6

[2] S. Abdullah, W.H. Kurniawan, M. Khamas and Y. Ali. Emissions analysis of a compressed natural gas direct injection engine with a homogenous mixture. International Journal of Automotive Technology. 2011;**12**(1):29-38. DOI: 10.1007/s12239-011-0004-1

[3] K. Zeng, Z. Huang, B. Liu, L. Liu, D. Jiang, Y. Ren and J. Wang. Combustion characteristics of a direct-injection natural gas engine under various fuel injection timings. Applied Thermal Engineering. 2006;**26**(8-9):806-813. DOI: 10.1016/j.applthermaleng.2005.10.011

[4] A.R.A. Aziz, Firmansyah and R. Shahzad. Combustion analysis of a CNG direct injection spark ignition engine. International Journal of Automotive and Mechanical Engineering. 2010;**2**(Jul-Dec):157-170. DOI: 10.15282/ijame.2.2010.5.0013

[5] T. Yusaf, P. Baker, I. Hamawand and M.M. Noor. Effect of compressed natural gas mixing on the engine performance and emissions. International Journal of Automotive and Mechanical Engineering. 2013;**8**(Jul-Dec):1416-1429. DOI: 10.15282/ijame.8.2013.29.0117

[6] F. Zhao, M.C. Lai and D.L. Harrington. Automotive spark-ignited direct-injection gasoline engines. Progress in Energy and Combustion Science. 1999;**25**(5):437-562. DOI: 10.1016/S0360-1285(99)00004-0

[7] B. Yadollahi and M. Boroomand. The effect of combustion chamber geometry on injection and mixture preparation in a CNG direct injection SI engine. Fuel. 2013;**107**(May):52-62. DOI: 10.1016/j.fuel.2013.01.004

[8] C.C. Lee, K.N. Chiang, W.K. Chen and R.S. Chen. Design and analysis of gasket sealing of cylinder head under engine operation conditions. Finite Element in Analysis and Design. 2005;**41**(11-12):1160-1174. DOI: 10.1016/j.finel.2004.12.007

[9] S.W. Chyuan. Finite element simulation of a twin-cam 16-valve cylinder structure. Finite Elements in Analysis and Design. 2000;**35**(3):199-212. DOI: 10.1016/S0168-874X(00)00002-0

[10] E. Danielson, D. Turner, J. Elwart and W. Bryzik. Thermomechanical stress analysis of novel low heat rejection cylinder head designs. SAE Technical Paper. 1993;**03**(01):930985. DOI: 10.4271/930985

[11] S. Abdullah, W.H. Kurniawan and A. Shamsudeen. Numerical analysis of the combustion process in a compressed natural gas direct injection engine. Journal of Applied Fluid Mechanics. 2008;**1**(2):65-86.

[12] S. Aljamali, S. Abdullah, W.M.F. Wan Mahmood and Y. Ali. The effect of injection timings on performance and emissions of compressed natural-gas direct injection engine. Journal of Combustion. 2016;**2016**:6501462 (7 pages). DOI: 10.1155/2016/6501462

[13] W.H. Kurniawan. Optimisation of Combustion Processes in Four Stroke Direct Injection Engine Using Compressed Natural Gas [thesis]. UKM Bangi: Universiti Kebangsaan Malaysia; 2007. 226 p.

[14] B. Sendyka and M. Cygnar. Stratified charge combustion in a spark-ignition engine with direct injection system. In: K.N. Hoon, editor. Advances in Internal Combustion Engines and Fuel Technologies. InTech; 2013. DOI: 10.5772/53971

[15] S. Aljamali, S. Abdullah, W.M.F. Wan Mahmood and Y. Ali. Effect of fuel injection timings on performance and emissions of stratified combustion CNGDI engine. Applied Thermal Engineering. 2016;**109**(2016):619-629. DOI: 10.1016/j.applthermaleng.2016.08.127

[16] M.A. Kalam and H.H. Masjuki. An experimental investigation of high performance natural gas engine with direct injection. Energy. 2011;**36**(5):3563-3571. DOI: 10.1016/j. energy.2011.03.066

[17] J.G.C. Baeta, F.A. Rodrigues Filho, M. Pontoppidan, R.M. Valle and T.R.V. da Silva. Exploring the performance limits of a stratified torch ignition engine using numerical simulation and detailed experimental approaches. Energy Conversion and Management. 2016;**126**:1093-1105. DOI: 10.1016/j.enconman.2016.08.073