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SALAH ELDIN MOHAMMED	IR. Dr. MASRI BAHAROM
Juba University	Universiti Teknologi PETRONAS
College of Engineering	Bandar Seri Iskandar, 31750 Tronoh
Khartoum Sudan	Perak Malaysia
Date: ____________________	Date: _________________________

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"ANALYSIS OF ENGINE CHARACTERISTICS AND EMISSIONS FUELED BY IN-SITU MIXING OF SMALL AMOUNT OF HYDROGEN IN COMPRESSED NATURAL GAS"

by

SALAH ELDIN MOHAMMED ELFAKKI HASSAN

The undersigned certify that they have read, and recommend to The Postgraduate Studies Programme for acceptance this thesis for the fulfillment of the requirements for the degree of Master of Science in Electrical and Electronics Engineering.

Signature: ______________________

Main Supervisor: IR. Dr. MASRI BAHAROM

Signature: ______________________

Co-Supervisor: ASSOC. PROF. Dr. A. RASHID A. AZIZ

Signature: ______________________

Head of Department: ASSOC. PROF. Dr. AHMAD MAJDI A.RANI

Date: ______________________

UNIVERSITI TEKNOLOGI PETRONAS

ANALYSIS OF ENGINE CHARACTERISTICS AND EMISSIONS FUELED BY IN-SITU MIXING OF SMALL AMOUNT OF HYDROGEN IN COMPRESSED NATURAL GAS

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SALAH ELDIN MOHAMMED

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DECLARATION OF THESIS

Title of thesis	ANALYSIS OF ENGINE CHARACTERISTICS AND EMISSIONS FUELED BY IN-SITU MIXING OF SMALL AMOUNT OF HYDROGEN IN COMPRESSED NATURAL GAS

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hereby declare at the thesis is based on my original work except for quotations and citations which have been duly acknowledged. I also declare that it has not been previously or concurrently submitted for any other degree at UTP or other institutions.

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________________________	________________________
Signature of Author	Signature of Supervisor
SALAH ELDIN MOHAMMED	IR. Dr. MASRI BAHAROM
Juba University	Universiti Teknologi PETRONAS
College of Engineering	Bandar Seri Iskandar, 31750 Tronoh
Khartoum Sudan	Perak Malaysia
Date: ___________________	Date: ________________________

I would like to dedicate my thesis to my beloved parents who taught me that knowledge is the key to success.

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ABSTRACT

The use of gaseous fuels in internal combustion engines has long been observed as a possible method of reducing emissions while maintaining engine performance and efficiency. Most of the research interests is focused on the use of compressed natural gas as alternative fuel, mainly due to its wide availability, high thermal efficiency and lower exhaust emissions compared to other hydrocarbon fuels. But compressed natural gas has the penalty of slow burning velocity and poor lean burn ability. One effective way to solve this problem is to mix the compressed natural gas with a fuel that possesses the high burning velocity. Hydrogen is the best additive candidate to natural gas due to its unique characteristics in promoting flame propagation speed, which stabilizes the combustion process.

This research investigated the engine characteristics and emissions of a CNG-DI engine fueled by low levels of hydrogen enrichment (lower than 10%) in CNG utilizing an in-situ mixing system. Prior to the main experiment, two pre-experiments were conducted to determine the best and most suitable parameters for optimization of engine performance, combustion as well as emissions. The first experiment was to determine the suitable injector type to be used, and it was found that the wide cone angle injector of 70^o was better for the applications. The second experiment was to determine the suitable injection timing, and it was discovered that the earlier injection timing was the best for this work.

In this research, the engine used was a 4-stroke single cylinder, with a swept volume of 399.25 cc and a compression ratio of 14:1. The injection timing was set to 300^o crank angle before top dead center as determined in the pre-experiment; the engine speed from 2000 to 4000 rpm and the spark timing for all the operating conditions were set to maximum brake torque. All the experiments were conducted at full load and relative air-fuel ratio λ =1.0. The injection pressure was fixed at 14 bar for all the cases.

The findings revealed that the brake torque, brake power and brake mean effective pressure increased with the increase of hydrogen fraction at low and medium engine speeds. The brake specific energy consumption decreased and brake thermal efficiency increased with the increase of hydrogen percentage. In general, significant changes have been observed with the engine characteristics at low engine speed but the rate of increase/decrease of the parameters decreased was less significant with the addition of higher percentages of hydrogen as well as with the increase in engine speeds.

For all the cases, the cylinder pressure and the heat release rate increased while the flame developement and rapid combustion duration decreased with the increase in the amount of hydrogen in the blends. The phenomenon was more obvious at the low engine speed, suggesting that the effect of hydrogen addition in the enhancement of burning velocity plays more important role at relatively low cylinder air motion.

Exhaust THC, CO and CO_2 concentrations decreased with the increase of hydrogen fraction due to the increase in hydrogen to carbon ratio (H/C). However, the variation in the NO_x emissions was found to be negligible with the addition of hydrogen.

ABSTRAK

Penggunaan gas bahan api dalam enjin telah diperhatikan sebagai cara yang berkemungkinan dapat mengurangkan pelepasan dan dalam masa yang sama mengekalkan prestasi dan kecekapan enjin. Kebanyakkan kajian memfokuskan kepada penggunaan gas asli mampat sebagai bahan api alternatif kerana sumber yang banyak, kecekapan terma yang tinggi, dan kurangnya pelepasan berbanding lain-lain bahan api asli. Akan tetapi gas asli mampat mempunyai seperti halaju pembakaran yang perlahan dan kurang kemampuan membakar. Cara untuk menyelesaikan masalah ini ialah dengan mencampur gas asli mampat dengan gas asli yang mengandungi halaju pembakaran yang tinggi. Hidrogen ialah calon bahan tambahan yang terbaik kepada gas asli kerana sifat uniknya yang menggalakkan kelajuan pergerakkan jelaga, yang mengstabilkan proses pembakaran.

Ujikaji ini telah mengkaji sifat-sifat enjin dan pelepasan oleh enjin CNG-DI yang dijanakan oleh pengkayaan hidrogen tahap rendah (kurang dari 10%) dalam CNG menggunakan in-situ sistem campuran. Sebelum ujian dijalankan, dua pra-ujikaji telah dijalankan untuk mengetahui parameter yang terbaik dan paling sesuai untuk mengoptimisasikan prestasi enjin, pembakaran, dan juga pelepasan. Ujikaji yang pertama adalah untuk mengetahui jenis suntikan yang sesuai untuk digunakan, dan suntikan sudut kon luas 70^o adalah yang terbaik untuk tujuan ini. Ujian yang kedua adalah untuk mengetahui masa suntikan yang sesuai, dan didapati suntikan awal masa adalah yang terbaik untuk tujuan ini.

Dalam ujikaji ini, enjin yang digunakan adalah 4-lejang 1 silinder, dengan isipadu sapuan 399.25 cc dan nisbah mampatan 14:1. Masa pancitan ditentukan pada 300^o sudut engkol sebelum top dead center seperti yang ditentukan dalam pra-ujikaji; julat kelajuan enjin pada 2000 ke 4000 putaran per minit dan masa nyalaan untuk semua kondisi operasi ditentukan pada tahap torque brek maksimum. Kesemua ujikaji dijalankan pada beban penuh dan relatif nisbah udara kepada bahan bakar ialah λ =1.0. Tekanan pancitan dikekalkan pada 14 bar untuk semua keadaan.

Keputusan menunjukkan brek torque, kuasa brek, dan min tekanan efektif enjin meningkat dengan peningkatan kandaungan hidrogen pada kelajuan enjin yang rendah dan sederhana. Penggunaan tenaga spesifik brek berkurang, dan kecekapan terma brek meningkat dengan peningkatan hidrogen. Secara amnya, perubahan besar dapat diperhatikan dengan sifat-sifat enjin pada kelajuan enjin yang rendah, tetapi kadar peningkatan dan pengurangan parameter berkenaan menurun sedikit dengan penambahan hidrogen kepada peratusan yang lebih tinggi dan juga dengan peningkatan kelajuan enjin.

Untuk semua kes, tekanan silinder dan kadar pelepasan haba meningkat sementara perkembagan jelaga dan tempoh pembakaran tinggi berkurang dengan peningkatan kandugan hidrogen dalam campuran itu. Keadaan ini lebih ketara pada kelajuan enjin yang rendah, menandakan kesan penambahan hidrogen dalam meningkatkan halaju pembakaran memainkan peranan yang lebih penting pada pergerakan udara silinder yang rendah secara relatifnya.

Pelepasan kepekatan THC, CO dan CO_2 berkurangan dengan peningkatan kandungan nisbah hidrogen kepada karbon (H/C). Walaubagaimanapun, variasi dalam pelepasan NO_x ditemui berada dalam keadaan yang boleh diabaikan dengan penambahan hidrogen.

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LIST OF ABBREVIATIONS

ATDC	After Top Dead Center
A/D	Analog/Digital
AFR_{stoich}	Air Fuel Ratio at Stoichiometric
BMEP	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
BP	Brake Power
BSEC	Brake Specific Energy Consumption
BTE	Brake Thermal Efficiency
BT	Brake Torque
BTDC	Before Top Dead Center
BSHC	Brake Specific Hydrocarbon
$BSNO_x$	Brake Specific Nitric Oxide
CA	Crank Angle
CFR	Cooperative Fuel Research
CH_4	Methane
CO	Carbon Monoxide
CO_2	Carbon Dioxide
COV	Coefficient of Variation
CNG	Compressed Natural Gas
CV_m	Calorific Value of Mixtures
DC	Direct Current
DI	Direct Injection
ECU	Engine Control Unit

EGR	Exhaust Gas Recirculation
ERI	Engine Remote Interface
FSO	Full Scale Output
FTIR	Fourier Transformed Infrared
GHGs	Green House Gases
H_2	Hydrogen
HC	Hydrocarbon
HCCI	Homogeneous Charge Compression Ignition
IMEP	Indicated Mean Effective Pressure
LHV	Lower Heating Value
LPG	Liquid Petroleum Gas
MBT	Maximum Brake Torque
MFB	Mass Fraction Burn
MPI	Multi Port Injection
NAI	Narrow Angle Injector
NG	Natural Gas
NG-H_2	Natural Gas and Hydrogen Fuel Mixtures
NO_x	Oxides of Nitrogen
ppm	parts per million
rpm	revolutions per minute
SCRE	Single Cylinder Research Engine
SI	Spark Ignition
TDC	Top Dead Center
THC	Total Hydrocarbon
TO	Throttle Open

vpm	parts per million by volume
V_s	Engine Swept Volume
WAI	Wide Angle Injector
WOT	Wide Open Throttle

NOMENCLATURE

λ	relative ratio $\left(1/_{\phi}\right)$
f	The hydrogen fraction
M_{CH_4}	The molecular weight of methane (g/mol)
M_{H_2}	The molecular weight of hydrogen (g/mol)
M_{air}	The molecular weight of air (g/mol)
T	Torque output of the engine (Nm)
N	Engine speed (rpm)
$\dot{m}_f$	The fuel consumption (g/s)
ϕ	equivalence ratio

CHAPTER 1

INTRODUCTION

1.1 Research Background and Motivation

With the dramatic increase in number of vehicles, the energy needs in the transportation sector is growing rapidly. Fossil fuels, particularly crude oil, have become the main energy sources and the primary fuel for transportation. Continuing depletion of such fossil fuels and the danger of air pollution linked to vehicular emissions have widely become the focus of concerns of every designer and researcher all over the world. The total fossil fuel reserves are undoubtedly limited and will be inevitably depleted at some point in the future. Therefore, the improvement in the fuel economy of combustion engines is quite important until new affordable alternative energy resources are found and widely used. As a result, petroleum companies continue to find new sources of crude oil and develop innovative technologies to reduce the dependencies on liquid fuels (diesel and gasoline).

Furthermore, automotive sector has largely contributed to the global air pollution leading to the deterioration of the environment. Correspondingly, interest is growing to reduce the exhaust emissions of the regulated pollutants. In particular, the hazardous polluting gases related to the fossil fuel combustion engines are hydrocarbon (HC), carbon monoxide (CO) and nitrogen oxide (NO_x) which can bring hazards to humans. Another class of gaseous pollutants is the group of gases commonly referred to as "greenhouse" [1]. Green House Gases (GHGs) can cause global warming due to the increased carbon dioxide (CO_2) emissions, which in turn, results in chain of tragedies that jeopardize human life and ecosystem at large. It is clear from the previous descriptions that alternative fuels may eventually be required in order to meet future emissions standards. Reducing automobile emissions is part of a larger strategy to tackle the negative health and environmental effects of air pollution created by all sectors. Emission standards are requirements that set specific limits to the amount of pollutants that can be released into atmosphere. Many emissions standards (United States Emissions Standards, European Emissions Standards, India Emissions Standards, etc) focus on regulating pollutants released by automobiles and others from industries.

Due to limited reserves of crude oil and environmental problems, development on alternative engine fuel has attracted more attention in the engine community. Alternative fuels are usually cleaner fuels compared to conventional liquid fuels such as gasoline and diesel fuel in the combustion process of engine. The introduction of these alternative fuels helps reduce fuel shortage and may also be beneficial to reducing engine exhaust emissions [2, 3]. The alternative fuels under studies are mainly methanol, ethanol, liquefied petroleum gas (LPG), compressed natural gas (CNG) and hydrogen (H_2). The most common alternative gaseous fuels are often considered as CNG and hydrogen H_2.Over the years, a few possible approaches for improving engine combustion and reducing exhaust emissions have been proposed including the application of low carbon level fuel such as methane (CH_4) [4, 5] or carbon free fuel such as hydrogen (H_2) [6, 7].

1.1.1 Natural gas and Hydrogen as Alternative Fuels

The use of gaseous fuels in internal combustion engines has long been observed as a possible method for reducing emissions while maintaining engine performance and efficiency. Most of the research interest is focused on the use of natural gas as alternative fuel, mainly due to its wide availability and low cost compared to other gaseous fuels.

Compressed natural gas is the most favorite for fossil fuel substitution. The use of compressed natural gas as engine fuel has been studied for many years and realized in both spark-ignition engines and the compression-ignition engines. Compressed natural gas is a mixture of different gases where methane is a major component and it is widely used gaseous fuel considered as a potential alternative to traditional liquid fuel. There are several benefits of using CNG in internal combustion engines such as high thermal efficiency due to higher octane value, desirable combustion properties including its superior knock resisting capability and lower exhaust emissions due to its simple chemical structure [8, 9]. Recently, hydrogen has received increased attention both popular and scientific as a potential alternative for fossil fuel based power generation and transportation applications. Internal combustion engine using hydrogen is considered as a suitable pathway to hydrogen economy [10].

Most researchers around the world support the use of hydrogen because, it can be produced from abundant sources and it is environmentally secure and clean. In terms of ignitability, hydrogen combustion processes have been found to have better characteristics compared to any other hydrocarbon fuel. Moreover, hydrogen has a wide flammability range which allows higher efficiency with leaner operation for reduced toxic emissions, low ignition delay, and higher flame stability. These inherent advantages are associated with hydrogen [7, 11]. However, there are serious problems which are affecting the development of hydrogen vehicles such as hydrogen storage, cost and methods of the fuel production.

Hydrogen is the best additive candidate to natural gas due to its unique characteristics in promoting flame propagation speed, which stabilizes the combustion process. The improvement of engine efficiency and reduction of emissions had been realized by adding H_2 into NG [12, 13].

Many studies has been conducted using natural gas and hydrogen blends on homogeneous charge port-injection engine. However, there are still some lingering obstacles associated with the port injection [14]. While CNG engine with turbocharger is beneficial at increasing power due to higher intake charge, the power increase is still lower at low engine speed [15].

1.2 Problems Statement

The main disadvantage of compressed natural gas as engine fuel is its low flame propagation speed, narrow combustible range and high ignition energy. Low flame propagation of CNG results in a longer combustion duration which in turn leads to incomplete combustion and high misfire ratio. In addition, at low engine speeds, CNG combustion becomes less complete resulting in lower torque and power as well as higher CO and THC emissions. In view of better and more improved combustion, undoing these problems associated with CNG becomes imperative. This can be done by adding hydrogen gas along with CNG into the DI-CNG engine which would be expected to ease these problems due to its faster burning velocity. However, addition of H_2 reduces the ignition energy of the mixture which may result in uncontrolled auto-ignition problems especially for the high H_2 fraction cases. The present work will investigate enriching of CNG with small amount of H_2 in in-situ mixing process in the DI-CNG engine.

1.3 Scope of Study

This research focuses on the experimental studies of engine performance at different parameters. The research also involved the analysis of engine combustion data and measurement of exhaust gas emissions of a direct-injection CNG spark-ignition engine enriched with a small amount of hydrogen (3%, 5%, and 8%) at different engine speeds: low (2000 rpm), medium (3000 rpm) and high (4000 rpm) with the H_2 injection pressure set at 14 bar. The experiments were carried out at wide open throttle (WOT) conditions with ignition timing set for maximum brake torque, and the air-fuel ratio was kept at stiochiometric, while three injection timings were selected to be at 120^o, 180^o, and 300^o CA BTDC. The engine performance characteristics are brake torque, brake power, brake mean effective pressure and brake specific energy consumption; the combustion characteristics are cylinder pressure, heat release rate, mass fraction burned, flame development duration and rapid combustion duration while the exhaust emissions are CO, CO_2, NO_x, and THC.

Prior to the main experiment, two pre-experiments were conducted to determine the best or most suitable parameters for optimized engine performance, characteristics and emissions. The first experiment was to determine the suitable injector type to be used. Two types of injector were selected, narrow cone angle injector of 30^o and wide cone angle injector of 70^o. The second experiment was to determine the suitable injection timing, three injection timings were selected to be at 120^o, 180^o, and 300^o CA BTDC.

1.4 Objectives of the Study

The main objectives of this work are

1. To investigate the combustion and engine performance of small amount of H_2 addition to CNG in a CNG-DI engine using in-situ mixing concept.
2. To investigate the effects of small amount of H_2 addition to CNG on emissions of a CNG-DI engine using in-situ mixing concept.

1.5 Thesis Outlines

This thesis is divided into five chapters. Chapter 1 discusses the background of depletion of fossil fuels, the air pollution caused by conventional liquid fuel and renewable clean burn alternative fuels. It also introduces the concept of hydrogen and natural gas mixtures, and provided preliminary indications of the potential benefits of such fuel. In addition, the chapter illustrates the research problems, objectives and scope of work.

Chapter 2 highlights the ground works related to the application of CNG, hydrogen and hydrogen-natural gas blend in spark ignition port–injection engine. It also discusses the application of a direct-injection spark-ignition engine fueled with natural gas-hydrogen mixtures; and presents the earlier research investigation of natural gas-hydrogen blends relevant to the present study.

Chapter 3 describes the detailed methodology of the experimental engine which includes experimental setup, data acquisition, gas supply systems, tests and calibration.

Chapter 4 presents the results and discussion of the research. The experimental results are presented in graphical forms which elaborated the results in detail.

Finally, Chapter 5, which is the last chapter shows the major conclusion and recommendations as well as the future work.

CHAPTER 2

LITERATURE SURVEY

2.1 Compressed Natural Gas as an Engine Fuel

The rapid depletion of the world oil reserves, high cost of oil refining, and the danger of air pollution linked to vehicular emission have widely become the focus of concerns of engine researchers all over the world. The situation therefore led the transportation industries to concern themselves with the fuel which is low in cost, readily available for use in abundant quantity and less harmful to the environment. The use of gaseous fuels in internal combustion engines has long been observed as a possible method for reducing emissions while maintaining engine performance and efficiency. Most research has focused on the use of compressed natural gas as alternative fuel, mainly due to its wide availability and low cost compared to other gaseous fuels. CNG is a mixture of different gases and concentration of these gases may differ from one reserve to another. The primary constituent of CNG is methane (CH_4), which typically makes up from 90% to 99% of the total volume [16]. Methane can be produced from gas wells or can be obtained from renewable sources, such as organic matter decomposed in landfills, and through processing of animal waste.

The use of CNG from the point of view of engine efficiency has some advantages and disadvantages. One of the advantages of CNG is that it possesses high octane number which means that it can have a higher compression ratio. The use of higher compression ratio increases the expansion ratio of the burned gases as a result increasing the cycle efficiency. The flammability limits of CNG fuel are wider than gasoline, so it can operate on a very lean limit [2, 17]. As for the results of wider flammability limits, CNG can be operated at lower fuel consumption than that of gasoline. In addition, from the environmental perspective and cost, it is cleaner and cheaper than either gasoline or diesel fuels [4, 18].

Despite the advantages of CNG as one of the most important alternative fuels, CNG has some drawbacks and disadvantages. The primary disadvantage of CNG as fuels is that, firstly, power produced from CNG engine shows approximately 10% lower output as compared to gasoline fuel. This is due to the gaseous state of CNG that decrease the engine volumetric efficiency [19]. Secondly, the lower flame speed of CNG would increase the combustion duration [20]. This could be overcome somewhat by changing the spark timing but some reduction in fuel economy and power will occur. Thirdly, natural gas engine conversion from gasoline fueled engine requires some modification on the fueling systems, valve train and ignition system [21]. However, other systems in CNG fueled engine basically operate on the same fundamental concept as that of gasoline fueled engine.

On the other hand, the performance of CNG fueled internal combustion engine however can be significantly improved by adjusting air-fuel ratio, using maximum knock free compression ratio, adjusting ignition timing and using other methods such as turbo-charging, stratified charge engine and revised combustion chamber shape [22]. For natural gas engines, direct injection improved the performance of the engine due to increased volumetric efficiency.

2.2 Direct Injection of Compressed Natural Gas Engine

The process of injecting natural gas directly into the cylinder near TDC is an entirely different process than injection of diesel fuel under the same conditions. The major difference is that the natural gas is in gaseous state when it is injected while the diesel fuel is injected as liquid. For current passenger car standard applications, a power drop of approximately 10% is noticed by the use of CNG, which occurs from reduced volumetric efficiency. But this drawback can be compensated by direct injection of CNG straight into the combustion chamber. It was noticed that direct injection system has 9-35% higher indicated mean effective pressure (IMEP) compared to multi-port-injection (MPI) system from the effect of improved volumetric efficiency [23, 24].

It was discovered that the emission level of CNG-DI engine is higher compared to MPI system especially for NO_x and HC emission near stoichiometric condition [25]. However, by controlling the mixture preparations and regulating injection parameters such as injection timing, ignition timing, and injection pressure, the drawbacks in emission and the increase in power could be improved [25].

There are two types of mixture preparation methods namely, homogeneous charge and stratified charge. Homogeneous systems on gasoline direct injection offers good exhaust emission [26, 27]. The combustion characteristics can be improved when the stratified charge is controlled [9, 28].

From the previous studies, it can be summarized that the possibility of direct injection system to be applied on CNG engine could increase the performance of the engine. In order to achieve more power output, one has to optimize the parameter that controls the injection events in direct injection systems. Injection parameters such as injection timing [25, 29-32], injection pressure [24, 33, 34] and type of injectors [34-37] had proven to be important parameters that affect the combustion and performance of direct injection engine.

Due to the low flame propagation speed of the CNG and high ignition energy, the improvement is limited especially at lower engine speeds where the combustion becomes less complete. An effective method to increase the flame propagation speed is to mix the CNG with a fuel which has higher flame speed. Hydrogen is the best additive candidate to natural gas due to its unique characteristics in promoting flame propagation speed and it also stabilizes the combustion process.

2.3 Hydrogen as a Supplementary Fuel

Hydrogen has long been realized as a true alternative fuel as it can be produced by several methods, from non-fossil fuel source. Hydrogen is a better fuel for spark ignition engines because it mixes easily with air and having highly desirable combustion properties [7, 38]. Furthermore, hydrogen offers many advantages for the improvement of the combustion process. This is due to some favorable combustion properties such as wide flammability limits, low ignition in air, high heating value, and it has exceptionally higher flame velocity than most common fuels which leads to high thermal efficiency. Therefore, it can be estimated that the addition of hydrogen to the slow burning fuels such as methane and gasoline can accelerate the flame propagation, extend the lean operational limits and improve combustion [39, 40]. Nevertheless, recent studies showed that the desirable combustion properties of H_2 make it the most likely candidate to finally replaced conventional liquid fuels [41]. Despite these advantages, hydrogen has some serious problems relating to the development of hydrogen vehicles such as occasional occurrence of back-fire, pre-ignition, hydrogen storage, cost and methods of the fuel production [42, 43].

Over the years, scientists and engineers around the world have suggested hydrogen as a supplementary fuel not only for gasoline and diesel [44, 45] but also for NG and other fuels [46, 47], in order to improve engine performance and enhance the combustion stability.

Hydrogen-enriched gasoline engine has also attracted many researchers to study its combustion and emission characteristics [48-52]. Results of these studies showed that the hydrogen addition could increase the flame propagation rate, extend the lean equivalence ratio for smooth operation from 0.8 to 0.5 [52]. This enhances the combustion process and improves the effective thermal and mechanical efficiency. Ji and Wang [53] studied the effect of hydrogen addition on improving spark ignition engine idle performance at stoichiometric condition. The experimental results showed that thermal efficiency, combustion performance, and NO_x emission are improved. While the HC and CO emissions first decrease with the increasing hydrogen enrichment level, but when hydrogen energy fraction exceeds 14.44%, it was observed to increase again at idle and stoichiometric conditions. According to Aly and Siemer [54] the CO_2 was found to decrease when hydrogen is added to existing natural gas in internal combustion engines. Thus, these results showed there is an increase in thermal efficiency and exhaust emission reduction of CO_2, CO, NO_x and unburned HC's.

2.4 Use of NG-H_2 Mixture in Internal Combustion Engines

The idea of using hydrogen as additive, to improve the combustion rate in spark ignition engines was first suggested for conventional gasoline fuelling [55, 56]. Several more recent researchers have investigated the effects of blending NG and H_2 for use in homogenous charge of spark ignition engines [47, 57-62]. The data have shown varying positive and negative results.

The most common application of hydrogen addition to CNG is to extend the lean limit of mixtures [57, 58, 60]. This has been attributed to enhance combustion rate due to the shorter ignition delay [12, 57]. For a given air-fuel ratio neither stoichiometic nor lean, NO_x emissions are higher with the hydrogen addition, due to higher temperature, while CO and HC emission are reduced [58, 60].

Because of the hydrogen's ability to extend the lean limit, lower NO_x emissions could be achieved by running the engine at leaner air-fuel ratio with hydrogen addition [59, 60]. Hydrogen addition could broaden the range of EGR while maintaining the engine to operate at low cyclic variations and low level of NO_x emission, and flame stability in the presence of EGR was also improved [47, 61].

The effects of hydrogen addition on engine efficiency appear to depend on operating condition, with some studies indicating improved efficiency [16, 57, 58] and other reporting reduced efficiency [61, 62].

Due to the NG features which have low laminar burning velocity and high ignition energy, the NG is even more prone to cyclic variation. The H_2 addition to NG has increased the burning velocity of mixture and also led to the reduction of the cycle-by-cycle variation of the engines [63-65].

Laminar burning velocities are fundamentally significant in regard to developing and justifying the chemical kinetics mechanism, as well as the performance and emission of combustion systems [66]. The laminar flame velocities of natural gas-hydrogen-air mixtures were studied in order to investigate the combustion fundamental characteristics of natural gas-hydrogen-air mixtures at various hydrogen fractions (volume fraction from 0 to 100%) [67]. They found that the laminar burning velocities increased exponentially with the increase of hydrogen fractions in mixtures, while the Markstein number decreased and flame instability increased with the increase of hydrogen fractions in mixtures. In general, the fundamental concept of hydrogen addition to CNG is to increase the combustion flame speed.

2.5 Port-injection of NG-H_2 Mixtures

Many studies had been carried out on using natural gas and hydrogen blends in port-injection spark ignition engines. One of the studies was conducted experimentally on a one-cylinder research engine in Germany by Nagalingam et al. [68]. Blends of hydrogen in methane of 0, 20, 50, and 100% by volume were studied at one engine speed. The addition of hydrogen was found to extend the lean limit of combustion due to its inherent nature, but it decreased the power due to an overall lower volumetric heating value. Indicated thermal efficiency decreased with H_2 addition, possibly due to a decreasing ratio of brake power to friction power and increasing heating value of the fuel.

Blarigan and Keller [69] investigated the feasibility of a hydrogen fueled internal combustion engine for both stationary and auxiliary power applications. These findings indicated that it is possible to construct a highly efficient power with equivalent zero emissions, fueled by 100/0, 70/30, and 100/0 CH_4/H_2 gas mixtures. Karim et al, [70] studied the engine performance and emission fueled by various hydrogen fractions in natural gas. Also, Bauer and Forest [62] reported a test conducted on a single cylinder cooperative fuel research engine (CFR) operating on mixtures of hydrogen in methane of 0%, 20%, and 60% by volume. Each fuel was tested at the speed of 700 and 900 rpm, full and part loads, and equivalence ratios from stoichiometric to partial burn limit. However, these experimental results showed that hydrogen enrichment reduced the value of spark advance for the best torque and decreased power due to a reduction in volumetric lower heating value. Furthermore, their experiments yielded unusual results which are rarely found in other researchers on engines were thermal efficiency dropped as hydrogen fraction increased.

Considering the experimental studies on NG and H_2 mixtures in combustion engine, some emissions from the engine fueled with the mixture of NG and H_2 were discovered. According to Wallace and Cattelan [71] experiments, the emissions from the engine fueled with the mixture of NG and H_2 is approximately 15% by volume. The experiments were conducted using a Chevrolet Lumina, which has six cylinders, four stroke cycles, water cooled, and a total engine cylinder volume of 3.135 l, bore of 89 mm, stroke of 84 mm and compression ratio 8.8:1. In their study, BSFC of 85/15 CNG/H_2 mixture was found to be less than that of NG. The BSFC values reduce for both NG and 85/15 CNG/H_2 mixture while spark timing (BTDC) values increased. BSHC of CNG is higher than that of the fuel mixture. But $BSNO_x$ emission values of 85/15 CNG/H_2 mixture are higher than that of CNG. If a catalytic converter was used, BSNOx values would decrease drastically. But, Collier et al. [60] examined the untreated exhaust emissions of hydrogen enriched compressed natural gas (H_2-CNG) production engine. The addition of hydrogen has increased NO_x emission for a given equivalence ratio while decreased total hydrocarbon emission. The effects of a spark ignition engine fueled by hydrogen and methane have been experimentally considered, for four cylinder tests with mixture of hydrogen in methane of 0, 10, 20, and 30% by volume. Similarly, Akansu et al. [72] found that by varying the equivalence ratio from 0.6 to 1.20, each fuel had been investigated at 2000rpm under constant load condition. These results showed that NO_x emission increased while, HC, CO_2 and CO emission value decreased and brake thermal efficiency (BTE) value increased with the increasing of hydrogen percentage.

IIbas et al. [73] experimentally studied laminar burning velocities of hydrogen-air and hydrogen-methane-air mixture. They concluded that increasing the hydrogen percentage in the hydrogen-methane mixture brought about an increase in the resultant burning velocity and caused a widening of the flammability limit. But, Shrestha and Karim [74] investigated proportions of 100/0, 90/10, 80/20, 70/30, 20/80 and 10/90 CH_4/H_2 percentages in different compression rates by varying equivalence ratio. They studied that the addition of some H_2 to CH_4 in a spark ignition engine enhanced the performance, particularly when operating on relatively low equivalence ratio mixtures. The optimum concentration of hydrogen in the mixture for producing a power gain and avoiding knock appears to be about 20-25% by volume over the range of conditions considered. Therefore, the idea of adding hydrogen into conventional vehicle fuels to improve thermal efficiency and inhibit cyclic variation could date back to several decades ago.

Many researchers conducted their experiments at MBT spark timing [14, 75, 76]. However, Ma et al. [13] studied the effect of hydrogen addition on thermal efficiency and emissions both at unchanged spark timing and changed spark timing to MBT, by doing this they found that optimizing spark timing according to hydrogen’s special combustion characteristics was critical to the engine’s overall performance and emissions.

Thus, the lower energy density of the gaseous charge can be enhanced through turbocharger. However, this further increases the chance of knock at high hydrogen concentration [59, 60]. The previous study, mainly concentrated on homogeneous mixture fueled from the port and very few studies were reported on direct-injection engine.

2.6 Direct-injection of NG-H_2 Mixtures

The disadvantage of port-injection gas engine is that its volumetric efficiency decreases as gas fuel occupies certain portion of intake charge, leading to the decrease of power output. Direct-injection gas engine can avoid the problem of the decrease in volumetric efficiency and it can maintain high engine power output. Meanwhile, the direct injection system can realize the stratified charge combustion and extend the lean mixture combustion capability, leading to the increase in thermal efficiency and the decrease in exhaust emissions [14, 77].

Preliminary studies have been conducted on direct-injection spark-ignition engine fueled with NG and H_2 blends at low compression ratio 8 and hydrogen volumetric fraction less than 20% [2, 77, 78]. The results showed that the heat release rate increased and combustion duration decreased when hydrogen volumetric fraction was over 10%. Nevertheless, Wang et al. [14] investigated the combustion and emission of a direct-injection engine fueled with NG-H_2 blends at extended hydrogen fraction of over 20%, and increased compression ratio of 12. These results showed that the brake effective thermal efficiency increased with the increase in hydrogen fraction for low and medium engine loads. The rapid combustion duration decreased, the heat release rates and exhaust NO_x increased with the increase in hydrogen fraction of the blends. Therefore, their findings suggested that the optimum hydrogen volumetric fraction in NG-H_2 blends is around 20% in order to obtain good engine performance and emission.

The fraction of hydrogen in the fuel varies from studies to studies. Most of the previous researchers used high percentage of hydrogen which is more than 10% in order to investigate important parameters [14, 68, 71, 74]. All the previous studies showed that the exhaust hydrocarbon emissions decreased when hydrogen was added to natural gas. However, NO_x was found to increase due to the increase in the combustion temperature. The addition of hydrogen with the higher percentages as previously mentioned extend the lean operation limit, improved the engines lean burn ability and decreased the burn duration. Additionally, the cycle-by-cycle variation was found to decrease and lower the brake Specific Fuel Consumption (BSFC) as compared to the pure CNG [7]. However, with the addition of higher percentages of hydrogen it was found to decrease the power output due to decreasing of the lower heating value of the mixtures.

The methods of mixing hydrogen and NG used in the past were mainly pre-mixed port injection [68, 69, 74], premixed direct injection [14, 77], and the combination of both [79]. Shudo et al. [79] argued that the effects on combustion and emissions of methane fueled direct injection of a stratified charge engine, and premixed port injection of hydrogen lean mixture. Their results revealed that the combustion system achieved higher thermal efficiency due to higher flame propagation velocity and lower exhaust emissions. The increase in NO_x emission can be maintained at a lower level with retarded ignition timing without deteriorating the thermal efficiency. Also, the effect of the hydrogen fraction on the engine torque studied and investigated by Chapman and Patil [80] at full load and stoichiometric air to fuel ratio using 0, 10, and 20% H_2 in NG. They found that a reduction in engine torque was measured as hydrogen concentration increased. One reasonable explanation is that the increased presence of hydrogen in natural gas fuels starts to adversely affect the power output due to lower volumetric heating value of the natural gas-hydrogen blends as shown in Figure 2.1. A reduction in the engine torque was also measured by Shrestha and

Karim [74] for natural gas-hydrogen mixtures more than 20%.

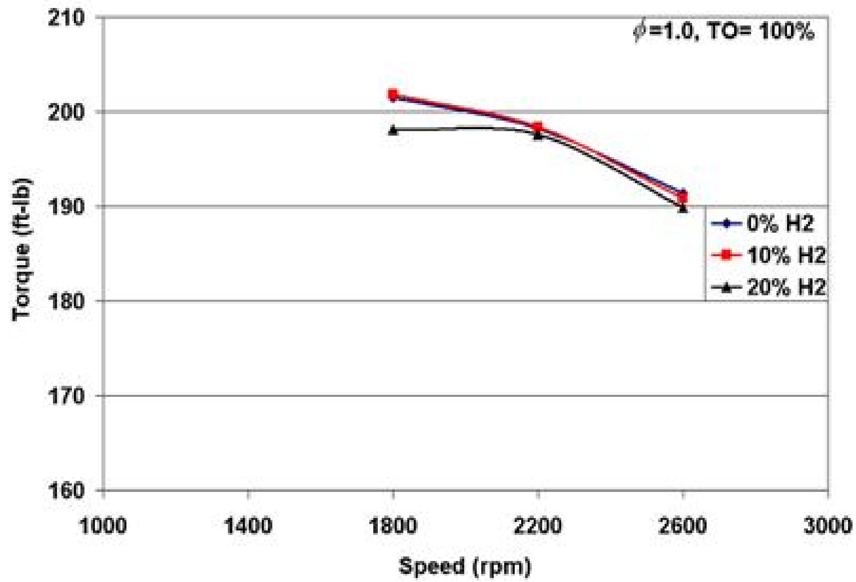


Fig 2.1: Variation of torque against speed for ϕ of 1.0 and TO of 100% [80]

2.7 Summary

The literature review advocates that hydrogen is the best additive candidate supplemented dual-fuel engine operation due to its unique characteristics in promoting flame propagation speed, which stabilizes the combustion process, reduce exhaust emissions from combustion engines and at the same time improve the engine efficiency.

Previous researchers have been conducted to investigate the premixed NG-H_2 blend combustion in spark ignition engine at higher levels of hydrogen percentage. However, only few studies were identified that investigated the engine performance characteristics of a small amount of hydrogen in natural gas blends. They found that higher percentage of hydrogen reduced the exhaust emissions close to zero; while other research conducted at the same percent of fuel blends showed a reduction in CO emissions and increase in NO_x emissions. A significant reduction in power was also observed for a 20% natural gas-hydrogen mixture. The major benefits of hydrogen-supplemented engine operation mentioned in the literature are reduced greenhouse gas emissions, reduced fuel consumption, and improved overall engine efficiencies. However, there are still some lingering obstacles that limit the wide application of hydrogen in SI engines, such as the occasional occurrence of back-fire, pre-ignition, the onset of knock and reduced engine power.

The previous study mainly concentrated on homogeneous mixture fueled from the port and few literatures were reported on direct-injecting engine using CNG-H_2 blends. Also most of the previous researchers used high percentages of hydrogen i.e. more than 10% in order to investigate important parameters. Consequently, the performance of the engine decreases due to the increased in hydrogen.

There are potentials for futher improvement of engine efficiency and reduce exhaust emission by adding small amount of hydrogen in CNG. This research aimed at investigating engine characteristics and emissions of CNG-DI engine with low levels of hydrogen (less than 10%) enrichment with CNG utilizing an in-situ mixing system.

The results of this work suggest that small amounts of hydrogen which exists within the locality of CNG utilizing in-situ mixing system is expected to improve the performance, combustion and reduce engine emissions.

CHAPTER 3

EXPERIMENTAL WORKS

This chapter describes the experimental setup and procedures adopted for collecting and analyzing the needed data for this project. The equipment which was used, the calibration of devices, and the engine parameters and data collection systems are described.

3.1 The Experimental Setup

In general, the experimental setup consists of a four-stroke, single cylinder, water cooled and direct injection CNG engine. The schematic layout of the experimental setup is shown in Figure 3.1. The engine was coupled to a direct current dynamometer that allowed the engine braking and motoring while the performance parameters were measured. The concentration of the exhaust emission (CO, CO_2, THC, and NO_x) and lambda were measured using a GASMET gas analyzer. The type and specification of the equipment used during this experimental work for testing and measurement are described in the subsequent sections.

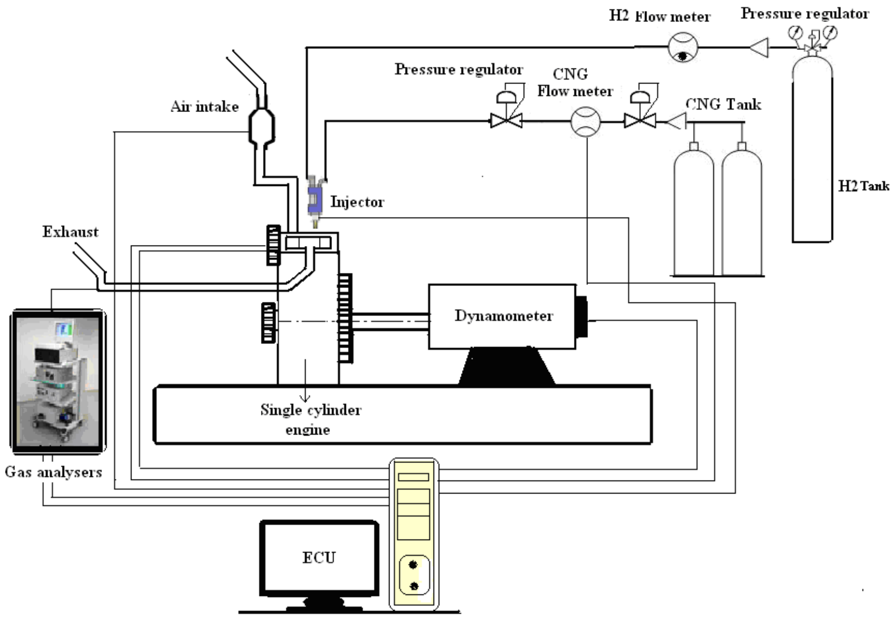


Fig 3.1: The schematic diagram of the experimental setup

3.1.1 The Engine and Accessories

3.1.1.1 Engine

The engine used for the experiment was a HYDRA engine with some modification done to its cylinder head to enable direct injection of gaseous fuel. It was a single-cylinder, four-stroke spark-ignition engine with a swept volume of 399.25 cc and a compression ratio of 14:1. The detailed specifications of the engine are given in Table 3.1.

The engine was originally a gasoline engine but it was modified for natural gas application. The data for gasoline as well as natural gas were made available for comparisons with other fuels. For this experiment no modification was done to the engine when using CNG and H_2 mixtures because the content of H_2 is very small and it was expected that the combustion characteristics would not vary in a large scale. A programmable Electronic Control Unit, or ECU connected to a computer was used to control the engine. Engine parameters such as injection timing, ignition timing, injection duration and the amount of fuel were controlled by ECU that is connected with ECU Remote Interface, or ERI installed in the personal computer. The real time data is available from the engine ECU and can be viewed and recorded accordingly.

Table 3.1: The specifications of the single cylinder engine [24]

Engine Properties	
Displacement volume	399.25 cm^3
Cylinder Bore	76 mm
Cylinder Stroke	88 mm
Compression Ratio	14:1
Exhaust Valve Closed	350^o BTDC
Exhaust Valve Open	225^o ATDC
Inlet Valve Open	372^o BTDC
Inlet Valve Closed	132^o BTDC
Dynamometer	Direct Current with maximum reading is 50 Nm
ECU	No of Inputs: 21 analog and 4 digital inputs No of Outputs: 14 multi-purpose outputs Battery Voltage: 8 to 16 V Power Supply dropout: o.1 ms

3.1.1.2 Injector and Spark Plug Position

Figure 3.2 shows the geometry of the combustion chamber with injector and spark plug location. The fuel injector is placed at the top centre of the combustion chamber with the spark plug next to it with an offset of 6 mm. A spark plug with a longer tip as compared to the standard one is used. In this experiment, original NG direct injector is used for the study without any modification. As hydrogen has low density, Wide Angle Injector, or WAI of 70^{o} (refer to Section 3.4.2.1) is chosen to allow maximum fuel spray distribution.

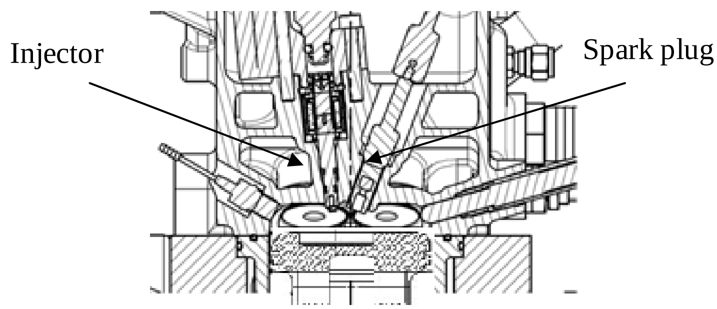


Fig 3.2: Cut-off view of the engine showing the injector and spark plug position [81]

3.1.1.3 Piston

The experimental work was done in a stratified condition using a stratified piston as illustrated in Figure 3.3. It has a bigger cup positioned away from the center as shown in the sectional view. This configuration is specially designed to achieve stratified conditions of the mixture, in which fuel is deflected back from the piston head to the spark plug, so that a rich mixture is created near the spark plug.

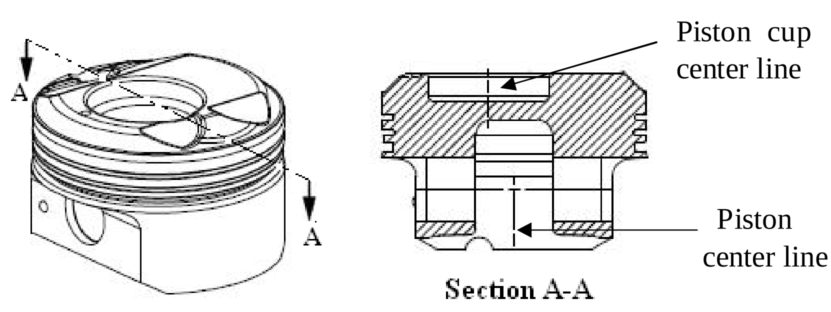


Fig 3.3: Stratified piston head shape [81]

3.1.2 The Engine Dynamometer

For the toque measurement during the experimental work, a dynamometer is coupled to the engine. Direct current or DC dynamometer is used to measure the brake torque. Table 3.2 shows the specifications of the dynamometer.

The DC dynamometer has the capability to motor the engine. Engine oil temperature, coolant temperature, intake air temperature were recorded manually from the engine control panel.

Table 3.2: The dynamometer specifications

Make and Model	David McClure DC30
Type	Direct current
Capacity	30 KW
Maximum Speed	5000 rpm
Maximum Torque	50 Nm

3.1.3 Exhaust Gas Analyzers

The Fourier Transformed Infrared or FTIR gas analyzer used in this study was called GASMET, manufactured by Temet Instruments. It uses the CALCMET software to compute the concentrations of the components present in the sample gas from the absorbance spectrum. The gas sample is introduced into the gas cell through standard gas line connectors. The transmitted infrared radiation is finally detected by a thermoelectrically cooled detector. The gas analyzer was used to measure the concentration of CO, CO_2, NO_x, THC and lambda with the accuracy of ±2% for each gas. The GASMET analyzer is capable of measuring about 50 gas species in the exhaust and providing the reading in parts per million, or ppm.

For oxygen concentration measurement, a GASMET oxygen analyzer was used. It utilized a Zirconia Measurement Cell that gave out voltage in proportion to the oxygen concentration. The analyzer was able to calculate the lambda values based on the oxygen concentration. Appendix A shows the general specification of the GASMET analyzer, its measuring parameters and the general specification of the oxygen analyzer.

3.1.4 Pressure Sensor and Cylinder Pressure Data Acquisition

The in-cylinder pressure has been measured using Kistler Piezoelectric pressure transducer. A piezoelectric pressure transducer generates electric charge proportional to the pressure. The signal for the transducer is then amplified using a charge amplifier that gave output in terms of voltage which was proportional to the charge. The transducer temperature variation can affect the calibration of the sensor due to the expansion of the casing as a result of the decompression of the crystal. Thus, the piezoelectric transducer is designed to have a water cooling passage to maintain its temperature in order to ensure correct pressure reading. Table 3.3 shows the specification of the transducer.

A crank angle encoder is used to establish the top dead center position and the phasing of cylinder pressure to crank angle. Typical set up for the cylinder pressure data capture system is as shown in Figure 3.4.

Combustion characteristics can be generated from the pressure reading obtained from the pressure sensor. A computer-based combustion analysis hardware and software are used to acquire and analyze the pressure data. The schematic of the system is shown Figure 3.4.

The hardware consists of high speed A/D data acquisition system and dedicated digital signal processors. The software performs statistical and thermodynamic analysis of the pressure data in real time. The data from the measurements of cylinder pressure can be used to determine not only the location of peak pressure but also the instantaneous heat release, burn fraction, and gas temperature [82]. Lab-view software of high speed data acquisition is used to record the reading of cylinder pressure. Pressure acquired data of 100 engine cycles are averaged to analyze the cylinder gas pressure, and the resolution of the pressure data is 0.5^{o} CA.

Table 3.3: The specifications of the pressure transducer

Make and model	KistlerThermoCOMP
Range	0-250 bar
Linearity all ranges	< ± 0.5% FSO
Sensitivity shift, cooled 50±35^{0}C	< ± 0.5%

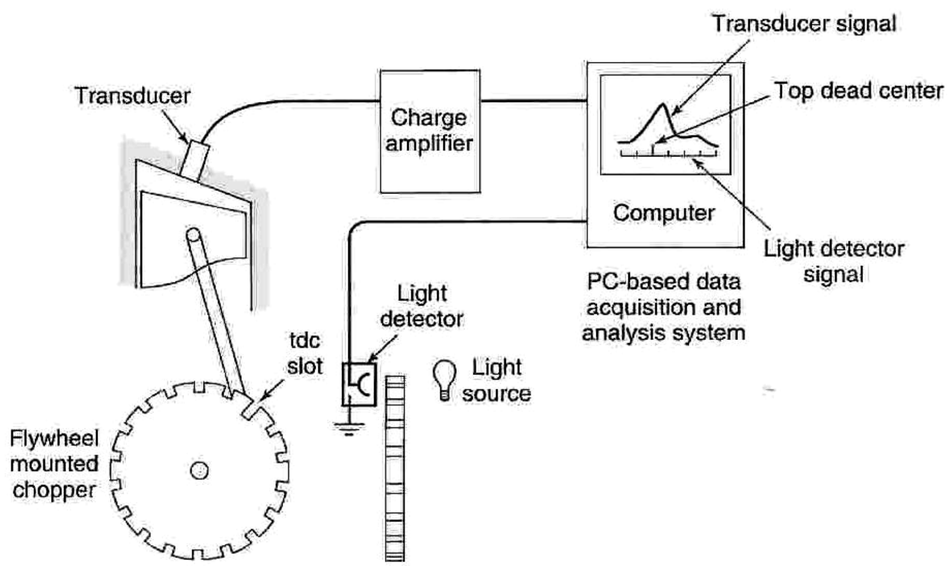


Fig 3.4: Cylinder pressure data capture system [82]

3.2 Fuel Composition and Properties

The fuel tests are carried out with gaseous fuel such as pure CNG and hydrogen supplemented CNG gas mixtures. The hydrogen is supplied by MOX Sdn. Bhd. in gas bottles of 200 bar pressure. The hydrogen has high purity of 99.999% and its specification is shown in Table 3.4.

Compressed natural gas supplied by Gas Malaysia Snd. Bhd. is used in the study of engine performance and combustion comparisons, and its specification is shown in Table 3.5. In this experiment, four fuel blends with the volumetric fraction of hydrogen in CNG of 0, 3, 5, and 8% are studied. The small amount of H_2 is preferred because it can be easily transported through the present natural gas delivery systems. The fuel properties of the natural gas and hydrogen are listed in Table 3.6.

Table 3.4: The specifications of hydrogen [83]

Purity	99.999%
Moisture	<3 vpm
Oxygen	<3 vpm
Hydrocarbon	<1 ppm
CO	<1 ppm
CO_2	<1 ppm

Table 3.5: Typical composition of the CNG in Malaysia [19]

Component	Symbol	Volumetric (%)
Methane	CH_4	94.42
Ethane	C_2H_6	2.29
Propane	C_3H_8	0.03
Butane	C_4H_{10}	0.25
Nitrogen	N_2	0.44
Carbon dioxide	CO_2	0.57
Others	H_2O+	2.00

Table 3.6: Fuel properties of natural gas and hydrogen [78]

Fuel properties	Natural gas	Hydrogen
Density in 1 atm, at 300 K (kg/m^3)	0.754	0.082
Stoichiometric air to fuel ratio (vol%)	9.396	2.387
Stoichiometric air to fuel ratio (wt%)	0.062	0.029
Laminar flame speed (m/s)	0.380	2.900
Quenching distance (mm)	1.900	0.600
Mass lower heating value (MJ/kg)	43.726	119.220
Volumetric heating value (MJ/Nm^3)	32.970	10.220
Octane number	120	
C/H ratio	0.251	0

3.2.1 Fuel Gas Supply System

The fuel gas supply system consisted of CNG and H_2 cylinders, pressure regulators, CNG mass flow meter, H_2 flow meter, and injector. Figure 3.5 shows the schematic diagram of the fuel supply system. The CNG is supplied from the gas bottles and the pressure regulators are used to reduce the pressure of the CNG in the main tank from 200 bar to 30 bar. A micromotion CMF010 ELITE series fuel flow meter is used to measure the fuel flow. The specification of the flow meter is given in Appendix B. The flow meter is placed after the pressure regulator and it has a sensitivity of 0.0001 g/s, and an inlet fuel pressure control system is placed after the flow meter. A gas compressor is coupled to the fuel supply system in order to maintain the fuel pressure along the fuel rail. By using a two stage hydrogen pressure regulator, the downstream pressure in the fuel line is always kept constant at 14 bar.

The photo of the CNG-H_2 mixture injector is shown in Figure 3.6a and its schematic diagram is shown in Figure 3.6b. The fuel injection system was specially designed to be used with high injection pressure in order to get constant pressure and flow to the injector. CNG is supplied through Inlet gas 1 at injection pressure of 14 bar while hydrogen is supplied through Inlet gas 2 at the same injection pressure.

Both CNG from Inlet gas 1 and H_2 from Inlet gas 2 are introduced in a short mixing chamber before entering the injection nozzle. Since the height of the mixing chamber is very short, both CNG and H_2 are expected not to be thoroughly mixed. Therefore, some amount of standalone H_2 is expected to exist within the locality of CNG. Due to the assumption that the injection system does not allow CNG and H_2 to completely mix, the system is referred to as 'in-situ mixing' as shown in Figure 3.7b. This mixing technique was chosen because H_2, by its nature, has relatively fast laminar burning velocity. It is therefore expected that the presence of standalone H_2 can enhance the combustion characteristics of the CNG.

The flow rate of hydrogen is controlled by a flow meter, CONCOA model 560, specifically calibrated for hydrogen. The amount of the injected H_2 was varied by adjusting its flow rate. The specification of the hydrogen flow-meter is given in Appendix C.

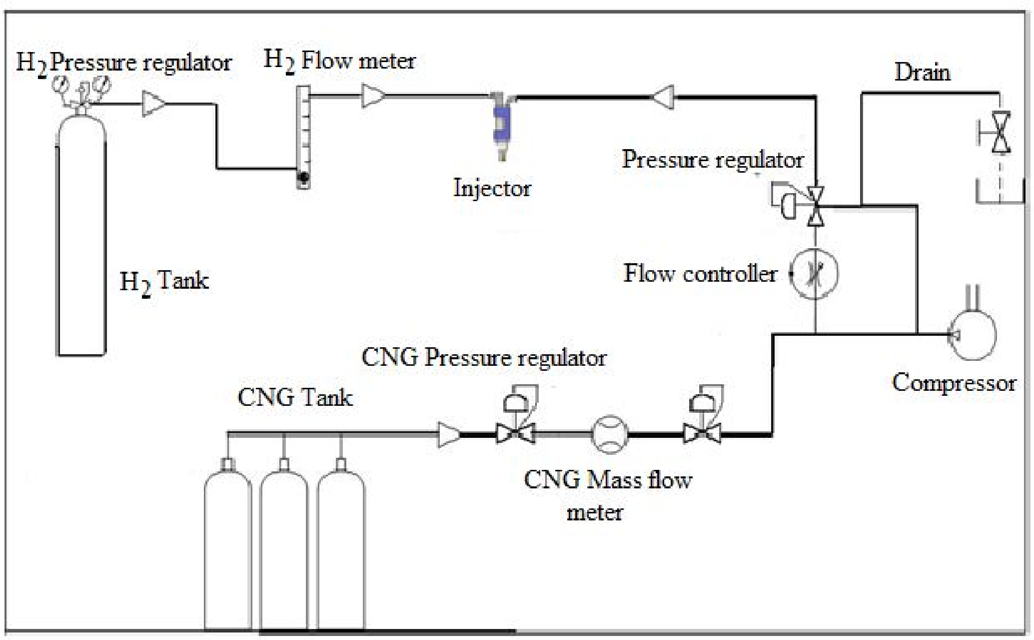
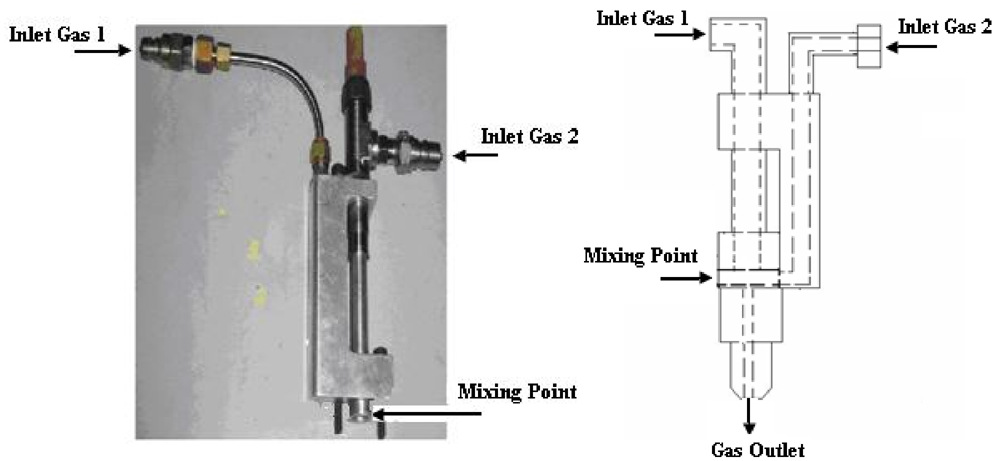


Fig 3.5: Schematic diagram of fuel supply system



a) Photo b) Schematic Diagram

Fig 3.6: Fuel injector used for testing

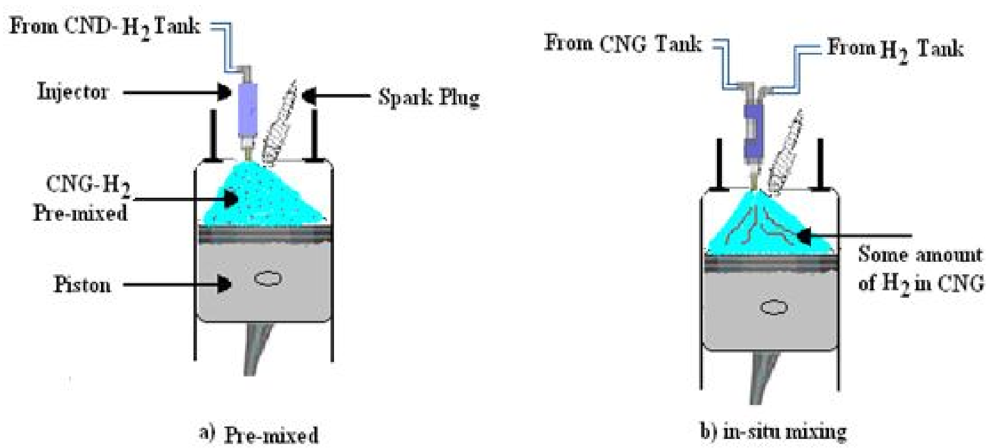


Fig 3.7: Differences between pre-mixed and in-situ mixing

3.2.2 Leakage Test

Due to the involvement of poisonous and highly flammable gases, detailed safety and operating procedures were consistently followed. One of the most important precaution is gas leakage test, therefore a soap solution was used to detect leakage.

The leakage test of gaseous fuels into laboratory will be very serious due to its toxic and explosive hazards. The tests were carried out on daily basis before and at the end of experiments to ensure that the system is leakage free.

3.3 Engine Operating Conditions

In the present work, all tests and measurements are conducted on a CNG-DI, four-stroke, and single-cylinder research engine. The injection timing is set at 300^{o} CA BTDC (refer to Section 3.4.2.2), the air-fuel ratio is kept at stoichiometric, while the ignition timing is adjusted to obtain the maximum brake torque. The experiments are performed at low, medium and high engine speeds of 2000, 3000, and 4000 rpm with each operating at full-load conditions wide open throttle (WOT).

3.4 Test Procedures

This study aims at investigating engine characteristics and emissions when fueled with a small portion of hydrogen with natural gas at compression ratio of 14:1 utilizing in-situ mixing system. In this work, experimental works are carried out to compare the engine performance using pure CNG and the mixtures of CNG-H_2. Before conducting the experiments, the equipment is checked and any fault found had been rectified.

Before start of the experiments, the engine was warmed up until the cooling water and lubricant oil temperatures reached stable values of 60^{o} C and 70^{o} C respectively. As hydrogen is very costly, the CNG is used to warm up the engine. Once the engine is warmed and when all necessary equipment put in place, hydrogen fuel tank with the two-stage regulator pressure and flow meter is supplied to the fuel line system. The engine is then re-started with the CNG-H_2 mixtures. A duration range of 4 to 5 minutes is specified before any data can be recorded, this will allow the fuel line to be flushed while the temperature of the engine is re-stabilized.

3.4.1 Fuel Blends

The fuels used in the tests were pure CNG and the mixtures of 3, 5, and 8% hydrogen. Due to the low percentage of hydrogen in CNG, it was assumed that the calculation of stoichiometric air to fuel ratio is almost the same. Given that the fraction of hydrogen *f*, the balance equation for CH_4 and H_2 mixture with air is [82]:

$$(1-f)CH_4 + fH_2 + (2-1.5f)(O_2 + 3.76N_2) = (1-f)CO_2 + (2-f)H_2O + (2-1.5f)(3.76N_2) \qquad 3\text{-}1$$

The stoichiometric air-to-fuel ratio (AFR_{stoich}) can be calculated as follow:

$$AFR_{stoich} = {(2-1.5f)*3.76M_{air}} \Big/ {(1-f)M_{CH_4} + fH_2} \qquad 3\text{-}2$$

Table 3.7: The stoichiometric air-to-fuel ratio values

H_2 fraction f	AFR_{stoich}
0%	17.20
3%	17.26
5%	17.31
8%	17.36

The summarized results, of the AFR_{stoich} are given in Table 3.7. Based on the above result, it can be concluded that the value of AFR_{stoich} is almost the same for all case with the maximum error of about 0.9%.

The engine was calibrated with pure CNG in order to know how much fuel the engine demands in mg/cycle. The calibration was made at different engine speeds of 2000, 3000, and 4000 rpm. The units were then converted from mg/cycle into g/s. Since the hydrogen flow meter is a volumetric type, therefore the amount of fuel had to be converted from g/s to L/min. Finally, the amount of hydrogen percentage was adjusted using a hydrogen flow meter and the summarized values are shown in Table 3.8.

Table 3.8: Fuel blends preparation

Engine speed	3% H_2 (L/min)	5% H_2 (L/min)	8% H_2 (L/min)
2000 rpm	0.853448	1.422414	2.275862
3000 rpm	1.339854	2.23309	3.572944
4000 rpm	1.866048	3.11008	4.076127

3.4.2 Selection of Injection Parameters

Two pre-experiments to determine the type of spray angle injector and the injection timing were carried out prior to the main experiment. The procedures for the experiments and selection criteria are discussed.

3.4.2.1 Pre-experiment to Determine the Suitable Spray Cone Injector

In this experiment, two types of injector spray angle were tested. One was a Narrow Angle Injector, or NAI that had a spray angle of 30^o, and other was a Wide Angle injector or WAI with a 70^o spray-angle. The main objective of this experiment was to determine the suitable injector type to be used. The capture of images was performed by a high speed video camera (Photron, FASTCAM-APX) operated at a speed of 4,000 frames per second with effective pixel size of 640*128. A Nikon 60mm f/2.8D Micro-Nikkon lens was used to accompany the camera. Finally, the images were recorded by an ECU. In order to get clear images, the knife-edge should be adjusted to the focal point of the light. The images were captured when the injector operated at 14 bar under atmospheric conditions.

3.4.2.2 Pre-experiment to Determine the Suitable Injection Timing

The objective of this experiment is to determine the suitable injection timing to be used in this work. The tests were first performed at an engine speed of 2000 rpm, followed by 3000 rpm and 4000 rpm with a wide open throttle at stoichiometric air fuel ratio. Three injection timings were selected for the study, with the injection pressure fixed at 14 bar for all the cases. For comparisons, two fuels were selected namely pure CNG and CNG-H_2 mixtures.

The selected injection timings were chosen based on the late injection, part injection and earlier injection timings. In this test, 120^o CA BTDC was chosen for late injection, 180^o CA BTDC for part injection and 300^o CA BTDC for earlier injection timings. The duration of the injection was set such that the air fuel ratio is close to stiochiometric.

The ignition timing was adjusted to obtain the maximum brake torque, while the injection pressure was fixed at 14 bar for all cases.

3.5 Device Calibration

3.5.1 Dynamometer Calibration

The dynamometer was calibrated using calibrated weights. The weights were put to the extension arm on the dynamometer and the torque readings were recorded from the control panel. Table 3.9 shows the calibration weights and the corresponding torque readings. After dynamometer achieves its maximum reading, the weight was unloaded and the error in the reading was within 0.3 Nm. For other arm of dynamometer, same calibration procedure was applied.

Table 3.9: The calibration of the dynamometer

NO	Weight (kg)	Torque (Nm)
1	1	4
2	2	8
3	5	20
4	10	40
5	12	48

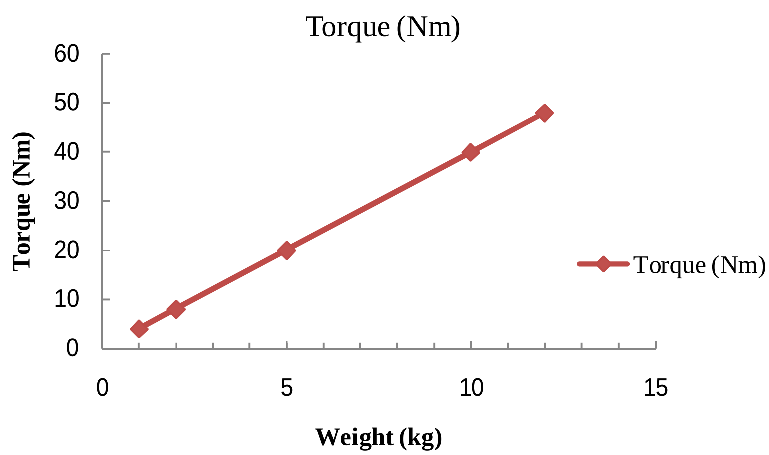


Fig 3.8: The calibration of the dynamometer

Based on Figure 3.8, the calibration values are almost linear. This explains that the dynamometer is reliable for experiments.

3.5.2 Pressure Data Acquisition Systems Calibration Check

The pressure reading was calibrated by using pressure testing device for engine. The procedure for pressure sensor calibration was:

- The sparkplug was removed from the engine
- Compression tester was installed on spark plug position.
- The engine was motored at low speed less than 3000 rpm.
- The reading of the cylinder pressure was recorded by using data acquisition system.
- Both results from manual pressure device and pressure sensor were compared.

The maximum pressure was compared using the above procedure. The manual pressure gauge reading indicated pressure in the cylinder. Based on the calibration results, the pressure sensors showed the same reading with pressure of the gauge.

3.5.3 Exhaust Gas Analyzer Calibration

Zero calibration of the exhaust gas analyzer was carried out daily before starting any experimental works. This allowed the analyzer to measure the background spectrum before exhaust gas was sampled. For zero calibration, the sample must be filled with pure substance such as N_2 to make sure that there is no unwanted sample in the test cell. The spectrum obtained was used by the analyzer as a baseline for the measurement process.

The background spectrum represents the actual absolute intensity of infrared radiation that is transmitted through zero gas filled sample. A typical background spectrum is presented in Figure 3.9.

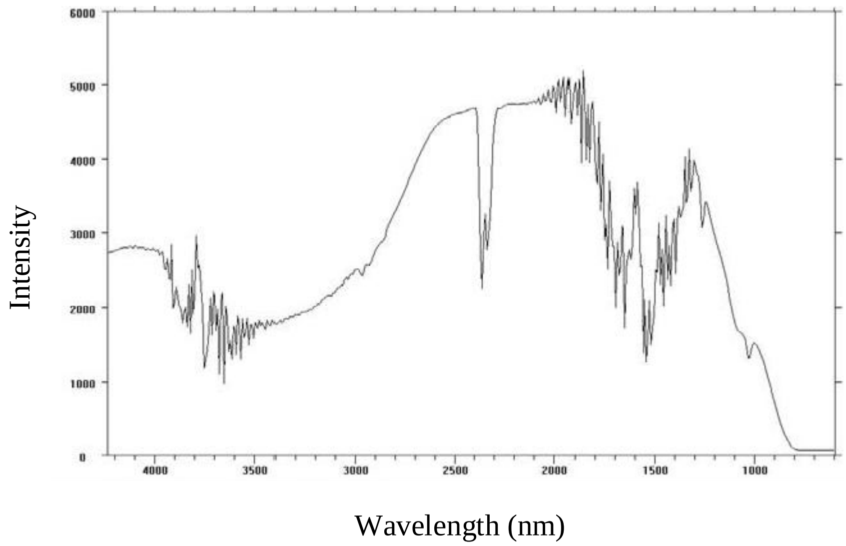


Fig 3.9: Calibration spectrum on FTIR system for emission analysis

3.6 Engine Parameters and Data Collections

In this project, all the data collection for experiment followed the SAE standard for engine performance and testing. The detailed description of the standard is stated in "SAE J1995, Engine Power Test Code-Spark Ignition and Compression ignition-Gross power rating".

3.6.1 Engine Performance Parameters

Data such as torque, engine speed, engine temperatures and exhaust gas temperature were manually recorded from the dynamometer control panel. Automatic data recording was available for the real time data from the ECU as well as for the emissions. For these experiments, the dynamometer was capable of maintaining the speed to an accuracy of about ±0.1 rev/sec. For each operating point, the data were recorded once the engine had stabilized, after 4-5 minutes. The engine performance parameters such as BP, BMEP, BSFC, BSEC, and BTE were calculated based on the following equations.

$$BP = {}^{2\pi NT}/_{60}\,(W) \tag{3-3}$$

$$BMEP = {}^{2\pi T}/_{V_s} \qquad (KPa) \tag{3-4}$$

$$BSFC = {}^{\dot{m}_f}/_{BP} \left({}^{g}/_{kW.hr}\right) \tag{3-5}$$

$$BSEC = BSFC.CV_m \left({}^{kJ}/_{kW.hr}\right) \tag{3-6}$$

$$BTE = {}^{BP}/_{\dot{m}_f.CV_m} \tag{3-7}$$

3.6.2 Analysis of Combustion Characteristics

The cylinder pressure data and the corresponding crank angle position were captured via a high speed data acquisition system. Cylinder pressure data was used to determine the IMEP, COV, heat release rate and the mass burn fraction. The calculations of the combustion parameters were done in an Excel spreadsheet. A special MACRO code was developed in Microsoft Excel to analyze the data. This code selects the data relevant to the calculations, analyses and summarizes it.

3.6.3 Exhaust Gas Concentration

A GASMET exhaust gas analyzer interfaced was used to measure emission concentration of the engine. This enable the emission data and engine operating data can be logged instantaneously during the test. The gas analyzer was used to measure the concentration of CO, CO_2, NO_x, THC and providing the reading in parts per million (ppm).

CHAPTER 4

RESULTS AND DISCUSSIONS

4.1 The Results of the Pre-experiment to Determine a Suitable Spray Injector

In this experiment, two types of injector spray angles were tested. Figure 4.1 shows the injector spray image at atmospheric condition for both injectors, NAI and WAI. The images are captured when the injector operated at 14 bar under atmospheric conditions. It was found that the intensity of the injected gas for narrow cone angle injector was higher than the wide cone angle injector before the time reached 5.0 ms after the start of injection. This phenomenon occurred because the fuel for the wide angle case had already been mixed with surrounding air. At the time of 6.25 ms after the start of injection, the images of a wide cone angle injector shows that the gas has already disappeared while the narrow angle injector showed some residual gaseous.

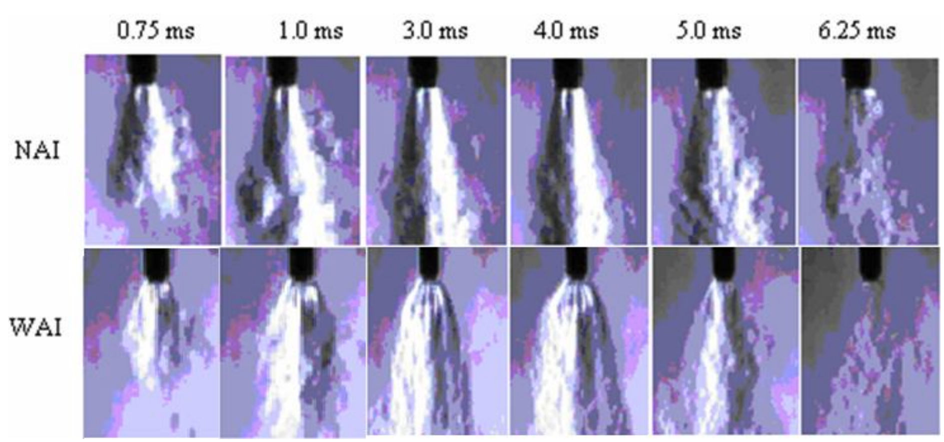


Fig 4.1: Schlieren image of spray pattern of CNG at atmospheric conditions and 14 bar using NAI and WAI

Figure 4.2 shows the schlieren images for the comparisons of the fuel spray penetration and cone angle between hydrogen and NG using WAI at 14 bar injection pressure under atmospheric conditions. The pictures were taken in order to observe the behavior of the spray characteristics of CNG and H_2. It shows that although the penetration of hydrogen is quite similar to CNG, the distribution of hydrogen is wider. It was found that the wide spray angle of H_2 could improve mixing rate of mixture due to the larger distribution area for the injected gas.

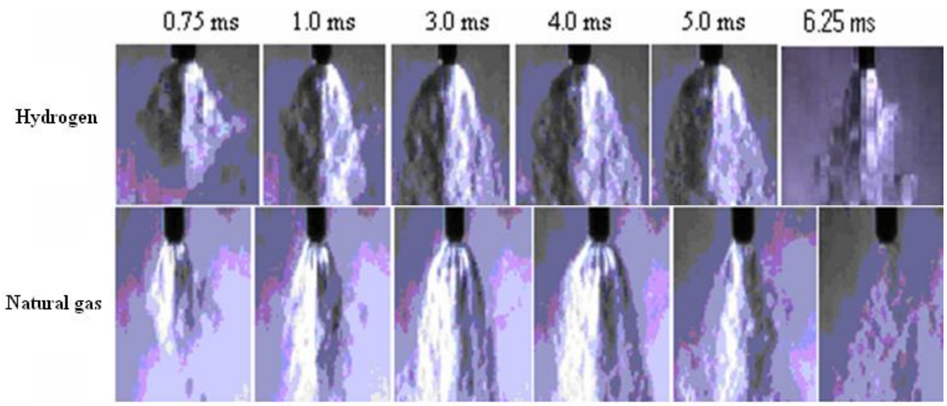


Fig 4.2: Schlieren image of spray pattern of hydrogen in comparison to natural gas at atmospheric conditions and 14 bar using WAI

4.1.1 Conclusion for the Pre-experiment to Determine a Suitable Spray Injector

Based on the above results, it can be concluded that, the WAI had better mixing rate as compared to NAI. Therefore, the WAI injector was used for the entire experiments.

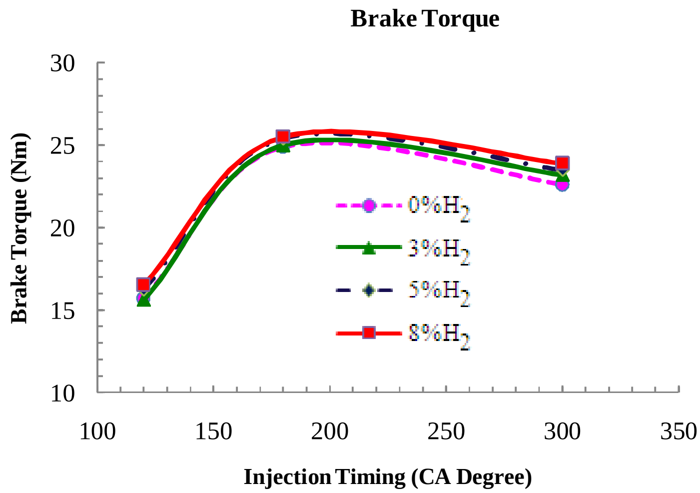
4.2 The Results of the Pre-Experiments to Determine the Effect of the Injection Timings on the Engine Performance of CNG-DI Engine

The effect of injection timing on the engine performance of CNG-DI engine is discussed in this section. Figure 4.3(a-c) gives the BT, BP, and BMEP versus the start of fuel injection timings for different hydrogen fractions at wide open throttle and λ=1.0, respectively. The three data points for each case of hydrogen fractions in CNG were connected using interpolated curves in order to predict the graphical trends. In general, it can be seen clearly that the BT, BP, and BMEP showed an increasing trend when injection timing was in the range of 120^o to 180^o CA BTDC, while slight decreasing trend was observed when the injection timing was in the range of 180^o to 300^o CA BTDC. This can be explained by the fact that the retardation of the fuel injection has reduced the available time between fuel injection and fuel ignition which in turn decreased the time for the fuel to mix with the air resulting in a non-homogenous mixture in the cylinder. In addition, the retardation of the fuel injection also decreased the penetration distance of the fuel jet after intake valve closing, resulting in a higher fuel concentration in the region near the injector nozzle, leading to a long ignition delay due to the lean mixture around the spark plug. It is expected that the combustion would be unstable and incomplete combustion would occur in the case of highly retarded fuel injection timing (e.g. less than 120^o CA BTDC) and this resulted in a decrease in BT, BP, and BMEP of the engine.

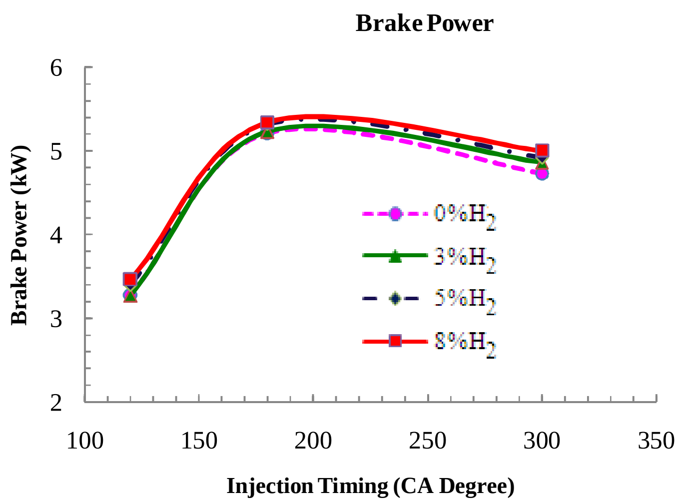
Advancing the fuel injection timing (e.g. 120^o to 180^o CA BTDC) is expected to increase the available time for air-fuel mixing, which improved the quality of the air-fuel mixture and shortened the ignition delay. These phenomenon lead to the increase in engine brake torque, brake power and brake mean effective pressure (BMEP). However, further advancement of injection timing (e.g. 180^o – 200^o CA BTDC) made little difference to the BT, BP, and BMEP. Slight decrease in engine performance was observed when the injection timing is advanced further from 200^o to 300^o CA BTDC. It was thought that the advancement of injection timing beyond 200^o CA BTDC could result in a lean mixture and thus a reduced flame propagation speed and hence an increase in the combustion duration.

For all the cases of injection timings, a small amount of hydrogen enrichment in CNG was found to increase the BT, BP and BMEP as compared to pure CNG, This phenomenon can be explained from the fact that the availability of hydrogen in CNG improves the mixture ignitability, increase the burning velocity of the CNG, and shortens the ignition delay.

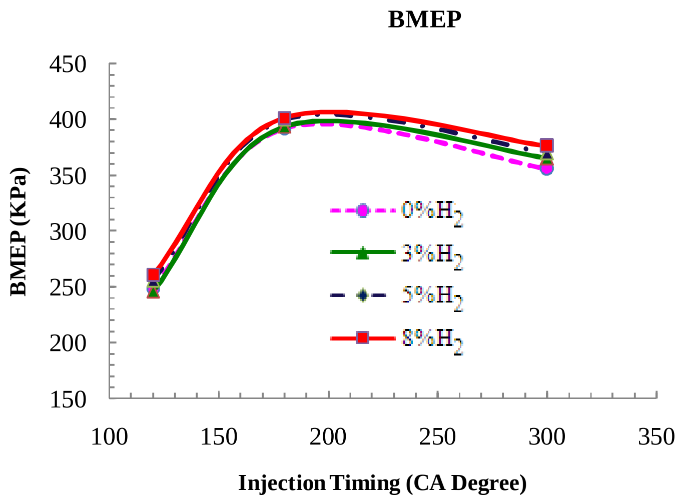
From Figure 4.3(a-c), it was found that the engine performance increased to about 9%, 2.5% and 5.7% at late, part, and earlier injection timing, respectively, when hydrogen is added. The findings showed that the highest rate of increase in engine performance occurred at the injection timing of 120^o CA BTDC; however, this injection timing is not preferable because it resulted in low engine brake torque as shown in Figure 4.3a. Although the 300° injection timing has higher rate as compared to the 180° injection timing, the 180° injection timing is preferable due to the highest engine torque.



(a)



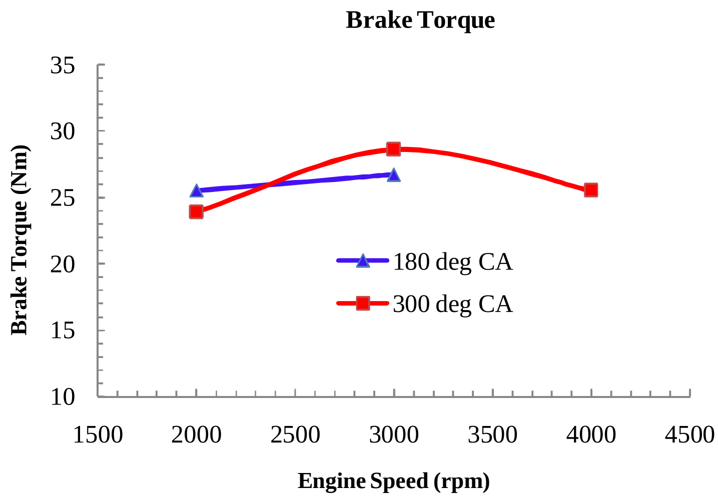
(b)



(c)

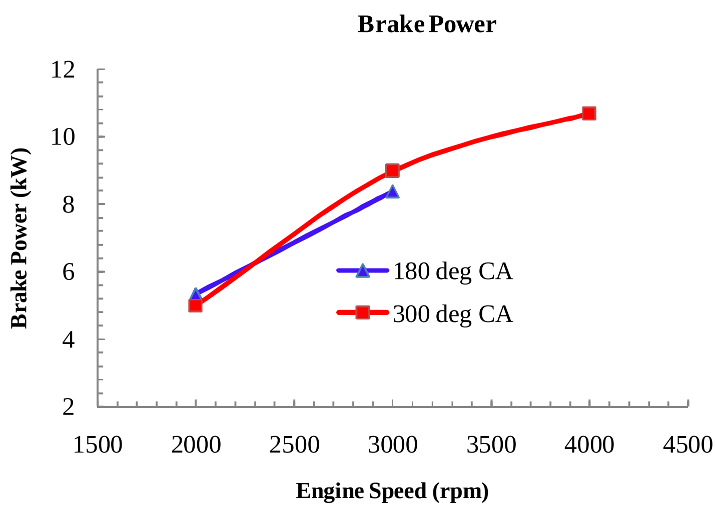
Fig 4.3: Engine performance versus fuel injection timings for different H_2 fractions

Figure 4.4(a-c) shows the engine performance of 180° and 300° injection timings versus engine speeds for CNG-H_2 fuel mixture (8% H_2) at λ =1.0. It can be observed that at low engine speed i.e. 2000 rpm, the 180° injection timing shows better results as compared to 300° injection timing. However, when the engine speed is increased to 3000 rpm, the 300° injection timing shows the maximum BT and BMEP. The explanation to this phenomenon is due to the fact that when the engine speed increases, the injection timing should be advanced in order to complete the combustion process. At 14 bar injection pressure and high engine speed (e.g. 3000 rpm and above), the combustion characteristics become unstable for the case of 180° injection timing, due to less available time for the fuel and air to be thoroughly mixed resulting in lower engine performance.



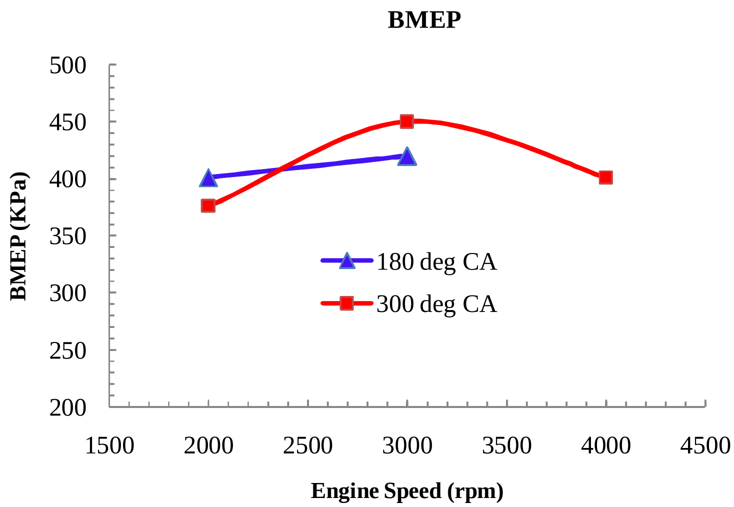
Brake Torque
35
30
25
20
15
10
Brake Torque (Nm)
1500
2000
2500
3000
3500
4000
4500
Engine Speed (rpm)
180 deg CA
300 deg CA

(a)



Brake Power
12
10
8
6
4
2
Brake Power (kW)
1500
2000
2500
3000
3500
4000
4500
Engine Speed (rpm)
180 deg CA
300 deg CA

(b)



(c)

Fig 4.4: Engine performance versus engine speeds two different Injection timings at CNG-H_2 mixture

4.2.1 Conclusion of the Injection Timing Test

By selecting the 300^o CA BTDC at 14 bar more experimental data could be obtained at higher engine speeds because the engine is stable at these conditions. It is clear from the previous descriptions that the 300^o injection timing was the best for this work.

For a specific injection timing, the addition of hydrogen to the natural gas led to an increase in the burn rate of the mixture, resulting in the enhancement of the engine performance as compared to pure CNG.

4.3 Analysis of the Engine Characteristics and Emissions of a Small Amount of Hydrogen

Analysis on the effects of adding a small amount of hydrogen in CNG using in-situ mixing technique to the combustion characteristics and emissions of CNG-DI engine will be discussed in this section. The engine test bed is a single cylinder research engine (SCRE) with a compression ratio of 14:1. The trigger signal was set for injection timing at 300° CA BTDC, the air-fuel ratio was kept at stoichiometric, while the ignition timing was adjusted to obtain the MBT. Furthermore, the tests were performed at low, optimum and high engine speeds of 2000, 3000, and 4000 rpm with each one operating at full-load (WOT) conditions.

4.3.1 Engine Performance

The performance characteristics of the engine are shown in Figures 4.5(a-e). Figure 4.5a shows the engine brake torque versus the engine speeds for difference percentages of hydrogen and λ =1.0. In general, it can be clearly seen that the torque curved for all the cases increased to the maximum value at about 3000 rpm and then started to decrease. At low engine speed, i.e. 2000 rpm, the torque was found to increase with the increase in the percentage of H_2. This can be explained by the fact that at low engine speed, the turbulence in the cylinder was low; and the addition of H_2 with its fast burning velocities increased the combustion rate of the CNG.

In Figure 4.5a, the increase in torque from pure CNG to 3% H_2 was found to be about 3.1%, from 3% H_2 to 5% H_2 was 1.5% and from 5% H_2 to 8% H_2 was 1.2%. This suggested that the rate of torque enhancement diminished with the increase in the percentage of H_2. It was observed from Figure 4.5a that the torque decreased for all the cases of CNG-H_2 mixtures when the engine speeds exceeded 3000 rpm. The value of torque at 4000 rpm for the mixtures of the H_2 was found to drop by about 1% from its value of pure CNG. The drop in engine torque was noted to increase with the increase in the ratio of H_2 enrichment. The explanation to this phenomenon is due to the fact that H_2 has a lower volumetric heating value as compared to CNG.

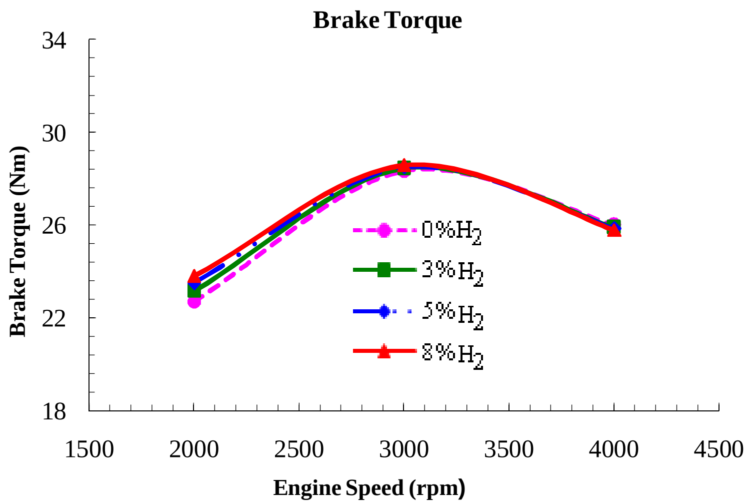
Figure 4.5b illustrates the experimental results of BMEP versus engine speeds for CNG- H_2 combustion. BMEP shows good relation with brake torque and similar trend curves as brake torque. Maximum value is 450 KPa when the engine speed at 3000 rpm, and it reduced to 401 KPa when the speed is at 4000 rpm.

As it is previously stated, hydrogen addition to NG will decrease the power output due to the decrease in LHV of NG-H_2 mixtures, depending on the hydrogen content while maintaining a better efficiency. Hence, suggestion that the small amount of H_2

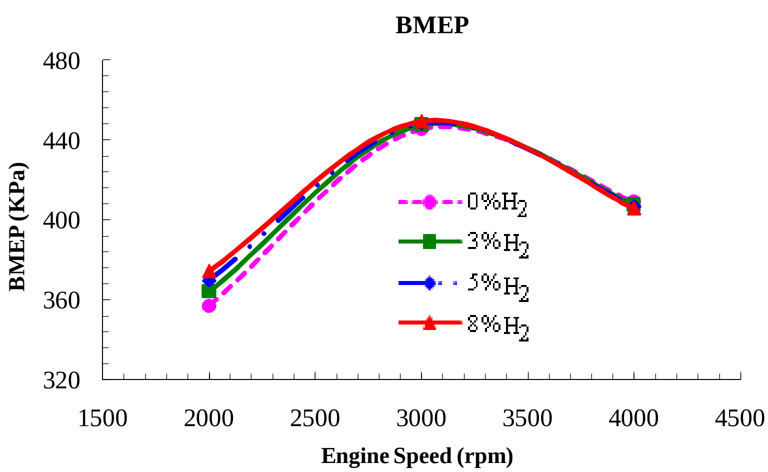
that needs to be added to the CNG may maintained the higher efficiency and power output. Effect of hydrogen addition on power is illustrated in Figure 4.5c, the results show that no decrease is observed in the power when engine speed is at 2000 and 3000 rpm based on hydrogen addition. That means a small amount of hydrogen addition to CNG can improve the power output. But at high speed 4000 rpm the performance slightly drops (1%) below CNG operation.

Since the brake specific fuel consumption is not an effective parameter in comparing two fuels having different calorific values and density, brake specific energy consumption is a better choice for comparisons because the heating value and density of fuel are considered. Figure 4.5d illustrates the BSEC plotted against engine speeds at λ =1.0. In general, the BSEC decreased with the increase in engine speed until about 3000rpm and then started to increase when the engine speeds were increased further. The BSEC was observed to decrease at different rates with the increase in H_2 percentage at low engine speed, i.e. 2000rpm. The BSEC for all the cases was found to drop to the lowest value of almost the same magnitudes when the engine speed was about 3000 rpm; which is the point of the maximum torque. It should be noted from Figure 4.5d, as the engine speed increases, the BSEC for all the cases increases mainly due to greater friction losses. At 4000 rpm, the addition of H_2 resulted in a slight increase in BSEC as compared to the pure CNG. The explanations to the previous findings are mainly due to the volatility and fast burning velocity of H_2 which enhances the combustion rate of CNG and result in a high energy output. Due to this, the BSEC reduced in the case of CNG-H_2 mixtures as compared with pure CNG.

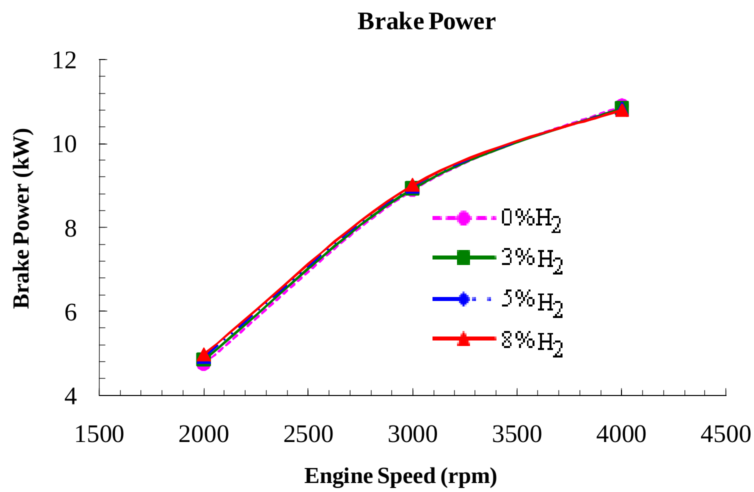
Figure 4.5e shows the BTE for mixtures at λ =1.0. As expected, the characteristics of the BTE plot are the inverse of the plot of BSEC. It can be observed from the figure that the BTE values are found to increase with the addition of hydrogen to CNG. The reason for the increase in efficiency as a result of hydrogen addition is because the burn rate and combustion efficiency increased.


Brake Torque
34
30
26
22
18
1500
2000
2500
3000
3500
4000
4500
Brake Torque (Nm)
Engine Speed (rpm)
0%H2
3%H2
5%H2
8%H2

(a)

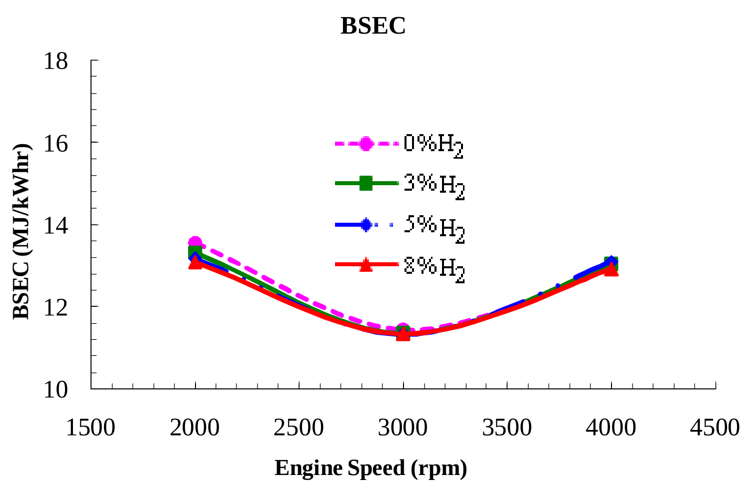

BMEP
480
440
400
360
320
1500
2000
2500
3000
3500
4000
4500
BMEP (KPa)
Engine Speed (rpm)
0%H2
3%H2
5%H2
8%H2

(b)



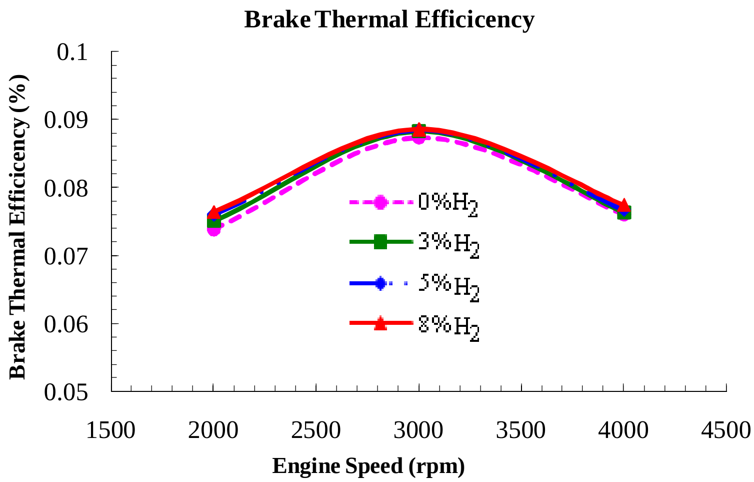
Brake Power
12
10
8
6
4
Brake Power (kW)
0%H2
3%H2
5%H2
8%H2
1500
2000
2500
3000
3500
4000
4500
Engine Speed (rpm)

(c)



BSEC
18
16
14
12
10
BSEC (MJ/kWhr)
0%H2
3%H2
5%H2
8%H2
1500
2000
2500
3000
3500
4000
4500
Engine Speed (rpm)

(d)



(e)

Fig. 4.5: Engine performance characteristics of fuel blends versus engine speeds

4.3.2 Engine Combustion

4.3.2.1 Cylinder Pressure

Figure 4.6(a-c) shows the cylinder pressure at engine speeds of 2000, 3000 and 4000 rpm for different values of H_2 percentages (0, 3%, 5% and 8%) and λ =1.0. For all the cases, the cylinder pressure increased with the increase in the amount of H_2.

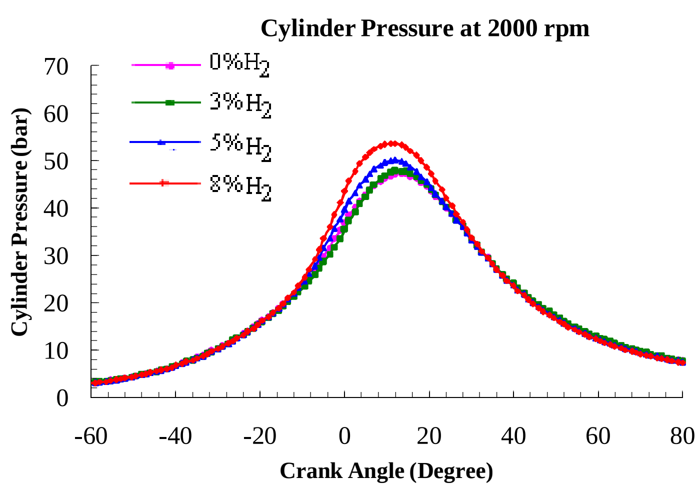
Figure 4.6a shows the engine combustion pressure at low engine speed, i.e. 2000 rpm, as a function of crank angle. The cylinder pressure was observed to increase at higher rate with the increase in H_2 injection. The maximum pressures for the 8% H_2, 5% H_2 , 3% H_2 and pure CNG occurred at 11^o, 12^o, 12.5^o, and 13.5^o CA ATDC respectively.

At an engine speed of 3000 rpm as shown in Figure 4.6b, the maximum cylinder pressures occurred at 13.5^o CA ATDC with their magnitudes being the highest of all values of H_2 percentage.

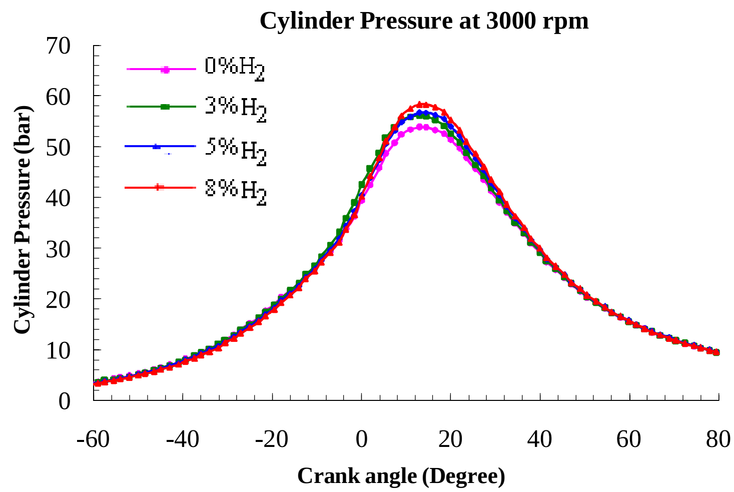
At high engine speed, i.e. 4000 rpm as shown in Figure 4.6c, the cylinder pressure was noted to increase with the increase in the amount of H_2, but the rate of increase for all cases was almost equal. The values for the crank angles ATDC for the 8% H_2, 5% H_2, 3% H_2 and pure CNG were found to be 10^o, 11.5^o, 14^o and 14^o.

The maximum cylinder pressure values recorded for the engine speed of 2000, 3000, and 4000 rpm were 54, 58, and 52 bar, respectively. It can be obviously seen that the maximum peak cylinder pressure occurred at engine speed of 3000 rpm which combustion process is the best as compared to other conditions. The lowest maximum cylinder pressure occurred at 4000 rpm of the engine speed. In this case, the combustion process was so rapid that it led to the decrease in its performance. In other words, at higher engine speed, the heat transfer to the combustion chamber walls increased and this caused the peak temperature, pressure and thermal efficiency of the engine to decrease.

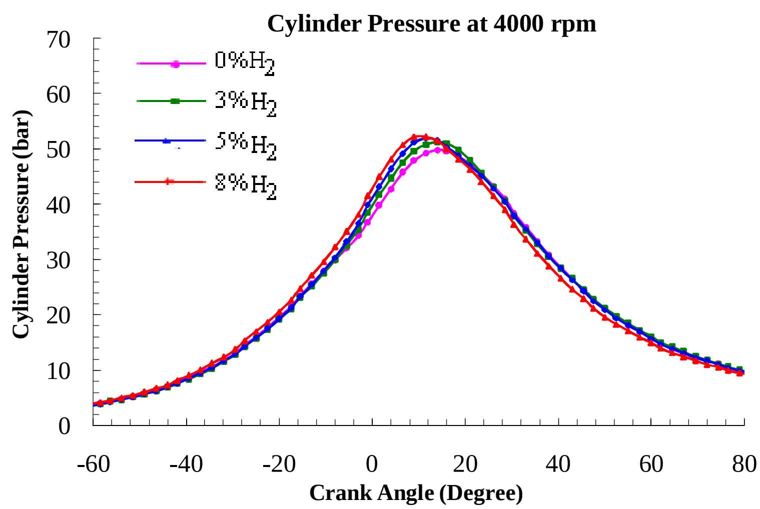
For all the previous cases, the cylinder pressure increased with the increase in the amount of H_2. The explanation to this phenomenon is mainly due to fact that the flame speed of hydrogen is faster than the flame speed of CNG. Therefore, burning CNG in the presence of a small amount of hydrogen will result in faster and more complete combustion. This will result in higher peak pressure closer to TDC and it will produce a higher effective pressure.



(a)



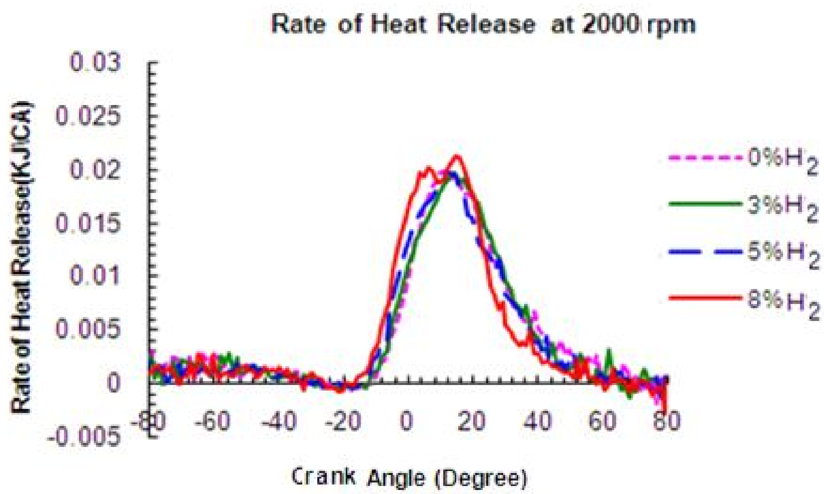
(b)



(c)

Fig 4.6: Cylinder pressure values versus the crank angle for different engine speeds and different H_2 fraction

The heat release characteristics for different engine speeds and various hydrogen fractions in the fuel blends at λ =1.0 are shown in Figure 4.7(a-c). Similar to those of the cylinder pressure, at 2000 rpm engine speed the phenomenon was obvious when increase the hydrogen fraction in the fuel blends the heat release is advanced, as the enhancement of burning velocity. The maximum values of heat release were found at engine 3000 rpm (0.027 kJ/CA). Heat release pattern can explain the combustion process that occurs in the systems. Higher heat release shows that better combustion efficiency and higher NO_x emission takes place in the process. While lower heat release leads to longer combustion duration.



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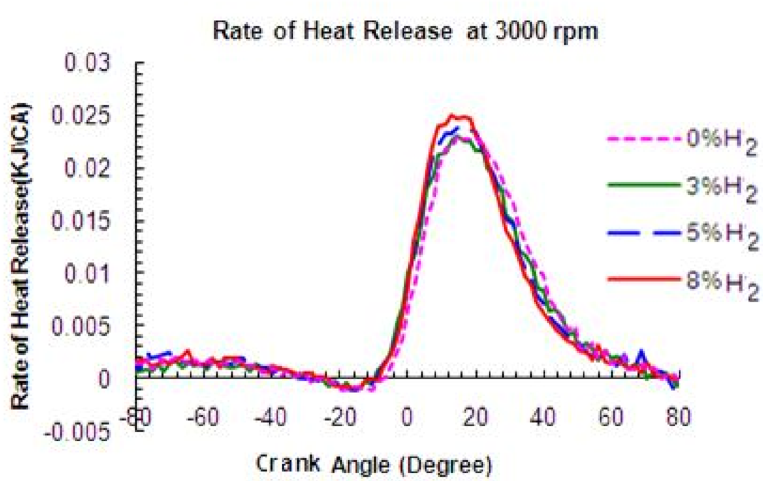
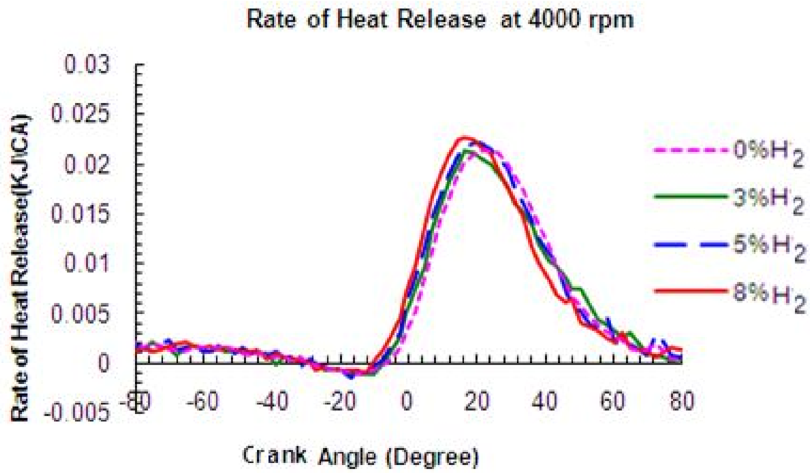
a)



b)



c)

Fig 4.7: Heat release rate versus the crank angle for different engine speeds and different H_2 fraction

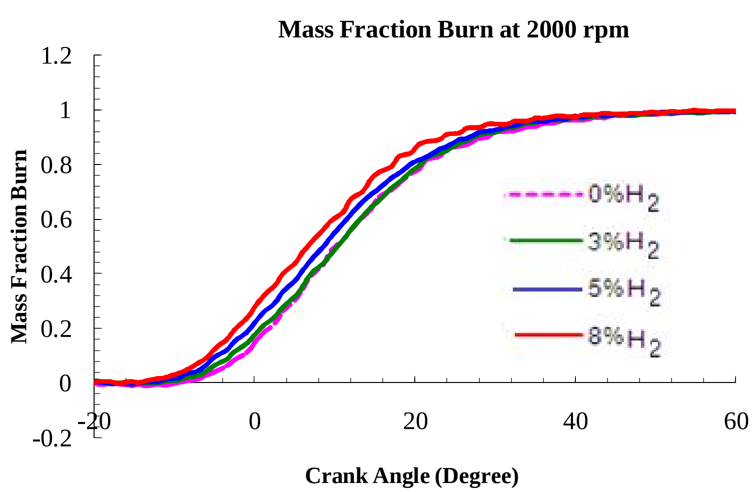
4.3.2.2 Mass Fraction Burned

The effects of adding a small amount of hydrogen in CNG on mass fraction burned, as a function of the crake angle are shown in Figure 4.8(a-b), for two different engine speed 2000 and 4000 rpm at full load and λ =1.0. It is clear that the positive effect of hydrogen addition to CNG, which results in an increase of combustion speed for any investigated condition. The improvement of combustion speed is more evident at higher percentage of hydrogen, as shown in Figure 4.8a. When the hydrogen percentage is 5%, the reduction of combustion duration is about 8.5%, in terms of crake angle, compared to CNG. In order to verify the findings, the plots of flame development and combustion duration expressed in crank angles were obtained and shown in Figure 4.9(a-b).

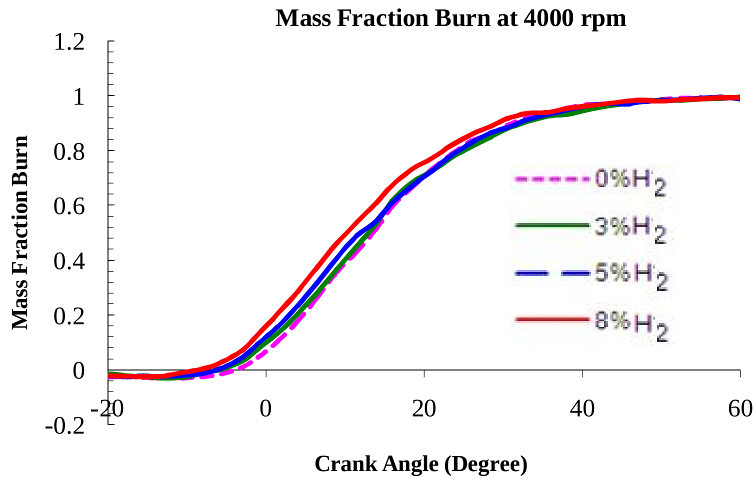
Figure 4.9(a-b) illustrates the flame development duration and rapid combustion duration expressed in crank angles versus hydrogen fractions and different engine speeds at λ =1.0. The flame development duration is defined as the interval of the crank angle from start of ignition to that of 10% mass fraction burnt, or MFB. And the rapid combustion duration is defined as the interval of the crank angle from 10% MFB to that 90% MFB. As shown in Figure 4.9(a-b), the flame development and rapid combustion duration reduced with the increase in the hydrogen fraction. The presence of H_2 in CNG improved the mixture ignitability and caused early flame development. The addition of H_2 also led to the increase in the rate of fuel utilization. At a low engine speed, i.e. 2000 rpm, the flame development and combustion duration in terms of crank angles were also low but the values keep on decreasing with the increase in H_2 percentage.

At 3000 rpm and 4000 rpm, the addition of H_2 was found to have less significant effect on the flame development as well as the combustion duration due to the high turbulence which occurred at higher engine speed. In other words, at high engine speed, the turbulence intensity of the fuel mixture is already high that the addition of small amount of H_2 does not give any effect to the flame development and rapid combustion duration. Since the data for 3000 and 4000 rpm are very close, this implies that at higher engine speed, i.e. more than 4000 rpm, the results will become much less significant. However, higher change in combustion development is expected at engine speed lower than 2000 rpm.

The combustion rate was observed to increase with the increase in H_2 fraction while, the opposite was found for the rate of increase in the engine torque. The main reason was possibly due to the requirement of readjusting the injection parameters in order to achieve the best engine performance.

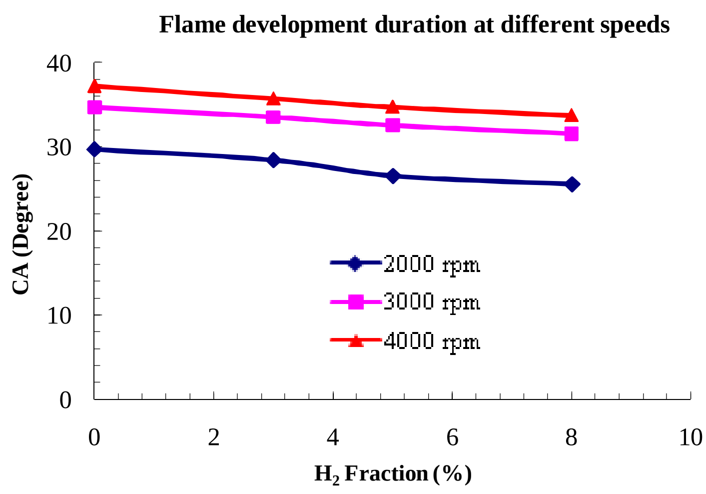


(a)

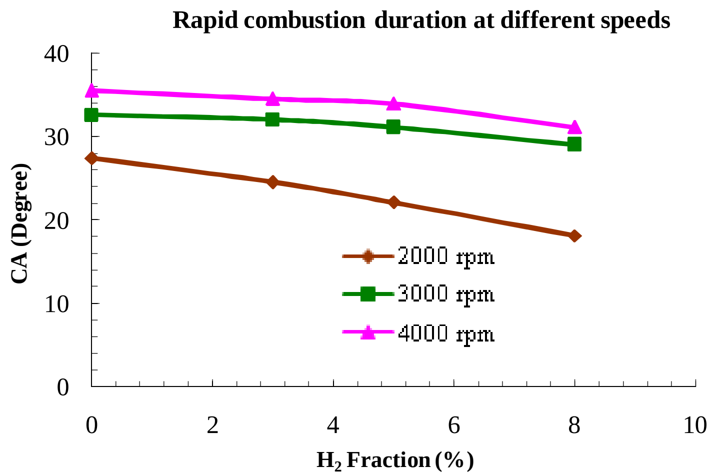


(b)

Fig 4.8: Mass fraction burned versus the crank angle for different H_2 fraction and 2000-4000 rpm



(a)



(b)

Fig 4.9: Flame development duration and rapid combustion duration versus hydrogen fractions.

4.3.3 Engine Exhaust Emission

Figure 4.10(a-d) shows the emissions characteristics at various engine speeds with the variation in H_2 portions at λ =1.0.

Figure 4.10a, shows the plot of the THC values against engine speeds for various H_2 percentages. It is observed from Figure 4.10a, that the THC decreased for all cases when the engine speed increased. At low engine speed, i.e. at 2000 rpm, THC tended to decrease at almost similar rates for all values of H_2. It can be said that the decrease in the carbon fraction in the fuel blends and the increase in combustion temperature is due to the increase in H_2 fractions which is the main reasons contributing to the THC reduction. As previously stated, the addition of H_2 enhanced the combustion. THC reached its lowest value at 4000 rpm for both CNG and CNG-H_2 combustions as observed form the flatness of the curves.

Figure 4.10b, shows the NO_x emission for various H_2 fractions. It can be noted that NO_x emission was the highest at engine speed of 3000 rpm which also coincided with the highest combustion pressure. The result also revealed that the NO_x emission was not significantly affected by the addition of H_2 which may be due to the retarding of spark timing at maximum brake torque. Normally, the NO_x emission should have increased due to the combustion improvement.

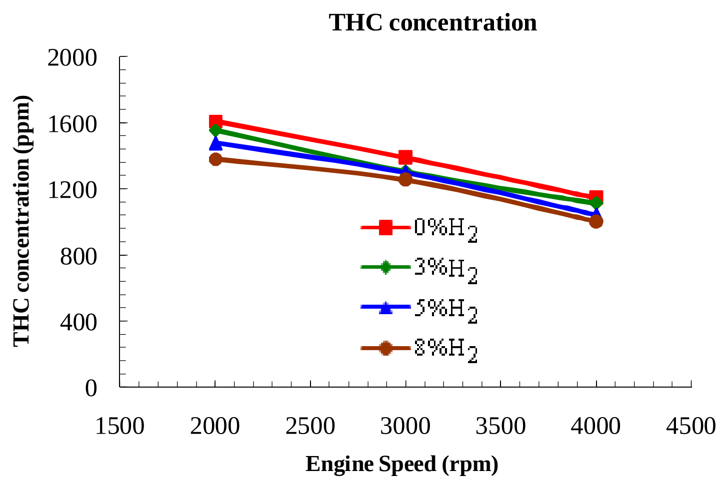
The CO_2 emission is represented in Figure 4.10c. It is observed from Figure 4.10c, that the CO_2 increased for all cases when the engine speed increased. CO_2 formation depends upon the carbon-hydrogen ratio of the fuel, so CO_2 concentrations decrease with the increase in H_2 fractions in the fuel mixtures. The equation of equilibrium in Table 4.1 can better illustrate the findings. In this formulation, the CNG is assumed to mainly consists of 100% of Methane (CH_4) and the remaining gases such as Ethane, Propane, and Butane are assumed negligible.

Table 4.1: The equation of equilibrium at different hydrogen percentage

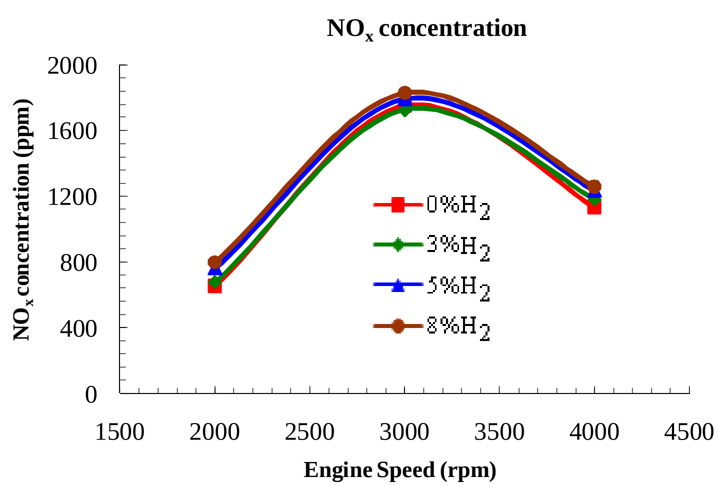
H_2%	Equation of Equilibrium
0	$CH_4 + 2(O_2 + 3.76N_2) = \mathbf{CO_2} + 2H_2O + 2(3.76N_2)$
3	$0.97CH_4+0.03H_2 + 3.91(O_2 + 3.76N_2) = \mathbf{0.97CO_2} + 1.97H_2O + 3.91(3.76N_2)$
5	$0.95CH_4+0.05H_2 + 3.85(O_2 + 3.76N_2) = \mathbf{0.95CO_2} + 1.95H_2O + 3.85(3.76N_2)$
8	$0.92CH_4+0.08H_2 + 3.76(O_2 + 3.76N_2) = \mathbf{0.92CO_2} + 1.92H_2O + 3.76(3.76N_2)$

It can be seen from Table 4.1 that the increase in hydrogen atoms resulted in the decrease of CO_2. Ideally, the graphs were expected to be linear; however, the nonlinearity behavior was expected due to the change in the engine speed which contributed to different turbulence intensity in the cylinder which influenced the completeness of the combustion process.

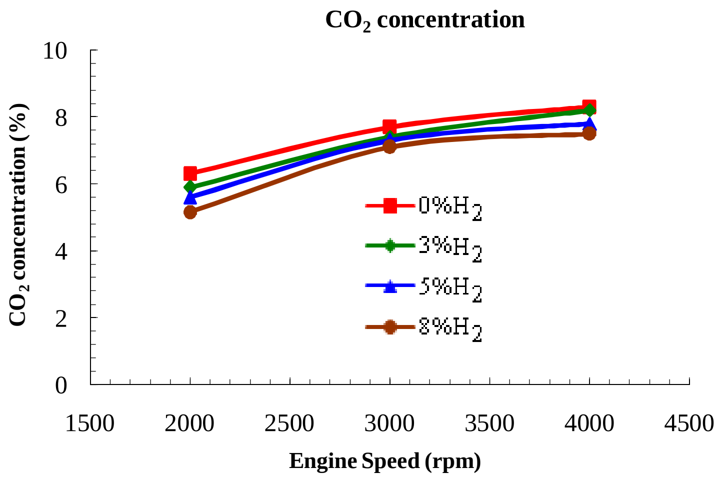
Figure 4.10d illustrates the CO emission. As seen from the figure, the increase in H_2 fraction resulted in a decrease in CO emissions until the engine speed reaches 3000 rpm and then started to increase with the increase in engine speed. The main reason for the decrease was the completeness of combustion process and sufficiency of oxygen, and the reason for the increase was due to the poor combustion as a result of timing retardation.



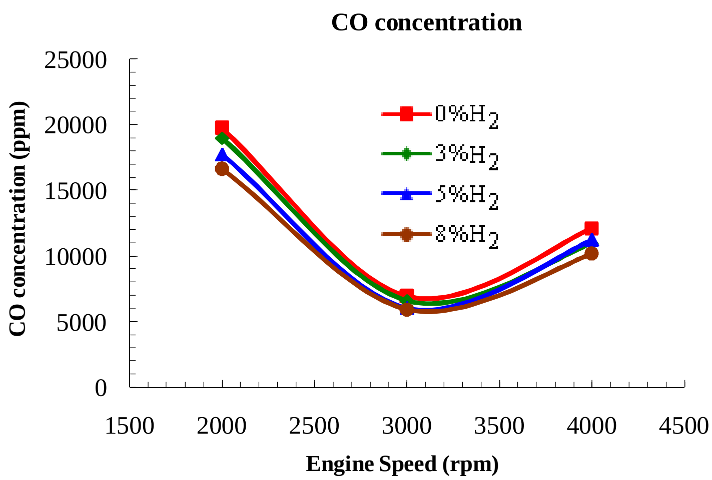
(a)



(b)



(c)



(d)

Fig 4.10: Engine emissions versus engine speeds and different H_2 fractions

4.3.4 Conclusion of the Main Experiment

The effects of adding a small amount of hydrogen to CNG using an in-situ mixing system at WOT and stoichiometric air-fuel ratio (λ=1.0) were investigated. The results showed that the introduction of small amount of hydrogen had improved the engine performance, combustion as well as emissions. In general, significant changes have been observed with the engine characteristics at low engine speed. At high engine speeds, i.e. 4000 rpm, the values of the engine performance were found to drop slightly.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 Conclusion

The experimental works were carried out to investigate the engine characteristics and emissions of a CNG-DI, spark-ignition engine fueled by a small amount of H_2 in CNG using an in-situ mixing system. Pre-experiments were conducted to determine the best and most suitable parameters for optimization of engine performance, combustion as well as emissions. The first experiment was to determine the suitable injector type to be used, and it was found out that the wide cone angle injector of 70^o was better for the applications. The second experiment was to determine the suitable injection timing, and it was discovered that the earlier injection timing i.e. 300^o CA BTDC, was the best for this work. Finally, the following conclusions were obtained based on the investigation.

- The introduction of small amount of hydrogen had improved the engine performance at low engine speeds. The rate of increase in the engine torque was found to decrease with the addition of higher percentages of hydrogen. At high engine speeds, i.e. 4000 rpm, the value of torque was found to drop slightly. Due to the higher hydrogen heating values based on mass, the BSEC was found to decrease while the brake thermal efficiency increases with the addition of hydrogen.

- The cylinder pressure increased with the increase in hydrogen fractions for all engine speed ranges with the prominent values occurring at low engine speeds. The flame development duration and rapid combustion duration in terms of crank angle decreased with the increase in the hydrogen fraction; and the rate of decrease was higher at low engine speeds.

- From the experimental results, it was inferred that there was a significant reduction in THC, CO and CO_2 emissions due to the increase in hydrogen to carbon ratio (H/C) with the increase of hydrogen fraction. However, the variation in the NO_x emissions was found to be negligible with the addition of hydrogen.

- The improvement of engine performance and emissions at low engine speeds suggested that the effect of adding a small amount of hydrogen into pure CNG enhanced the burning velocity of the mixture. At high engine speeds, the effect of adding small amount of hydrogen to the CNG mixture became insignificant due to the high turbulence intensity of the mixtures. Moreover, the addition of small amount of hydrogen at high engine speeds could decrease in engine torque although an increase in cylinder pressure was observed.

5.2 Recommendations

The present research has made contribution towards improving engine characteristics and emission fueled with small amount of hydrogen in CNG using in-situ mixing system. However, there is still a need to continue the experimental approaches to achieve further reductions in emissions and improvements in efficiency.

The exhaust emissions such as THC, CO, and CO_2 were decreased when using CNG-H_2 blends compared with CNG alone. However, NO_x emission increased as the combustion temperature increases. Also at higher engine speed the addition of small amount of hydrogen shows negative impact to the engine performance and this is not desirable. Based on these, the following are thereby offered as recommendations for the improvement in future work:

- The use of EGR may improve engine emissions especially NO_x under CNG-H_2 fuelling. Experiments should be conducted to investigate the effect of low level of hydrogen in CNG emissions. With the varying level of EGR under different loads and speeds.

- Since the ultimate aim of every engine manufacturer is to produce an engine with a good fuel economy and reduced emission without sacrificing the performance of the engine; it is highly recommended to conduct the experiment using natural gas-hydrogen blends operation at lean condition

- The effect of ignition timing, injection pressure, and injector types on the engine performance and emissions of small amount of hydrogen in CNG, under different loads and speeds need to be considered. By optimizing these parameters the improvement of the engine performance and reduce out emission could be obtained.

- Although, the stratified combustion in DI-injection improve the fuel economy and reduce engine out emission. But in view of high efficiency and extremely low emission HCCI combustion is highly recommended.

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Appendix A

General specification of GASMET FTIR analyzer [84]

General parameters	
Measuring principle	Fourier Transform Infrared, FTIR
Performance	Simultaneous analysis of up to 50 gas compounds
Response time	Typically <25s, depending on the gas flow and measurement
Operating temperature	15-25^{o} C non condensing
Storage temperature	-20-60^{o} C non condensing
Power supply	110-115 or 230V / 50-60 Hz
Power consumption	300 W
Measuring parameter	
Zero point calibration	24hours, calibration with nitrogen
Zero point drift	<2% of measuring range per zero point calibration interval
Sensitivity drift	None
Linearity deviation	<2% of measuring range
Temperature drifts	<2% of measuring range per 10 K temperature change

Pressure Influence	1% change of measuring value for 1% sample pressure change. Ambient pressure changes measured and compensated
Oxygen Analyzer	
Measuring principle	ZrO_2 measuring cell
Detection limit	< 1ppm O_2
Response time	< 1second
Sample Gas temp	120^o C to 300^o C, non condensing
Environm. Temp.	20^o C to $+40^o$ C
Reference gas	instruments air, dew point less than -40^o C, no oil, about 30 L/h
Measuring gas	dry or wet, no combustibles
Calibration gas	instrument air as above or test gas from bottles

Appendix B

The specification of the fuel flow meter [85]

Flow accuracy	+/- 0.05% of flow rate
Gas accuracy	+/- 0.35% of flow rate
Density accuracy	+/- 0.0002% of flow rate
Wetted material	403 L, 316 L Stainless Steel or Nickel alloy
Temperature ranging	-240 to 427^{o} C
Pressure ranging	100 to 413 bar

Appendix C

The specification of the fuel flow meter [86]

Accuracy	+/- 3% full scale
Repeatability	+/- 25% full scale
Maximum Pressure	14 bar
Maximum Temperature	95^{o} C
Useful Flow range	10:1

LIST OF PUBLICATIONS

[1] Salah E. Mohammed, D. Iffa, M.B Baharom, A. Rashid A. Aziz, "Spray Characteristic Comparisons of Compressed Natural gas and Hydrogen Fuel using Digital Imaging," Journal of Applied Sciences. (Published, 2011), ISSN 1812-5654 / DOI: 10.3923/jas. 2011.

[2] Salah E. Mohammed, M.B. Baharom, A. Rashid A. Aziz, "Analysis of Engine Characteristics and Emissions Fueled by In-situ mixing of Small amount of Hydrogen in CNG," International Journal of Hydrogen Energy, Elsevier (Published 4 February 2011),Volume 36, Issue 6, Pages 4029-4037.

[3] Salah E. Mohammed, M.B. Baharom, A. Rashid A. Aziz, Firmansyah "The Effects of Fuel Injection Timing at Medium Injection Pressure on the Engine Characteristics and Emissions of a CNG-DI Engine Fueled by a small amount of Hydrogen in CNG," International Journal of Hydrogen Energy (submitted).