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THE IMPACT OF HYDROGEN BLENDING ON EMISSIONS AND PERFORMANCE OF A NATURAL ASPIRATED ENGINE

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THE IMPACT OF HYDROGEN BLENDING ON EMISSIONS AND PERFORMANCE OF

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A THESIS APPROVED FOR THE

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To my wonderful Mom and Dad and my little sister: Thank you so much for always being by my side and for supporting me in everything that I have ever done. Your unconditional love and sacrifices are the reasons I am who I am today. “I love you for giving me your eyes, for staying back and watching me shine.”

- Tu

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Abstract

There has been a significant focus on developing alternative fuels and energy sources in order to mitigate the adverse effects of fossil fuel combustion on the environment. Among promising options, hydrogen stands out owing to its high energy density and clean combustion characteristics. The use of hydrogen in internal combustion engines, particularly in four-stroke engines, has been extensively studied. However, research on hydrogen combustion in two-stroke engines is limited, even though these engines are widely used in various industries, including the oil and gas sector. This study aims to report novel test results highlighting the exhaust emissions and performance of a large-bore internal combustion reciprocating two-stroke engine fueled with hydrogen-natural gas mixtures, that is the AJAX DPC-81. Natural gas-hydrogen mixtures with different hydrogen concentrations are examined with various engine loads and equivalence ratios. The emission concentrations of nitrogen oxides (NOx), carbon monoxide (CO), carbon dioxide (CO_2), unburned methane (CH_4), and volatile organic compounds (VOCs) are measured with corresponding mentioned parameters. Indicated thermal efficiency is computed to expose the combustion performance of the engine. Results found in this study can be adopted for engines that have a similar combustion system to that of the AJAX DPC-81. It is found that emissions are reduced significantly with increasing hydrogen concentration and engine load. The maximum reduction in NOx, CO, CO_2, CH_4, and VOCs emissions are approximately 90%, 33% 13%, 33%, and 55% respectively at 40% volumetric H_2 concentration. Overall, indicated thermal efficiency increases steadily with hydrogen volume fraction in the fuel mixture. At 75% load and 40% volumetric H_2 concentration, the engine achieves the highest efficiency of approximately 33%. The optimal operating condition for the AJAX DPC-81 is also determined to be at 60% load with 40% H_2 concentration and 60% bypass valve position combined.

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Nomenclatures

C_P: Specific heat in constant pressure of the fuel

C_V: Specific heat in constant volume of the fuel

$\frac{dP}{d\theta}$: Rate of change of in-cylinder pressure with respect to crank angle

$\frac{dQ_n}{d\theta}$: Heat release rate

$\frac{dV}{d\theta}$: Rate of change of cylinder volume with respect to crank angle

dV: The difference between each cylinder volume at each crank angle

H_2O%: Water composition in exhaust gas measured in percentage

HV: Heating value of fuel

$\dot{m}_{fuel}$: Mass flow rate of fuel

O_2%: Oxygen composition in exhaust gas measured in percentage

P: In-cylinder pressure at each crank angle

ppm: Parts per million of gas on dry basis

ppmvw: Parts per million of gas on wet basis

V: Cylinder volume at each crank angle

V_{swept}: Swept volume of the cylinder

RPM: Engine speed in revolution per minute

x: Normalized mole fraction of the gas in ppm

$x_{exhaust}$: Mole fraction of the gas measured in ppm

$\eta_{indicated}$: Indicated thermal efficiency

Abbreviations

NG: Natural gas

AFR: Air-to-fuel ratio

BDC: Bottom-dead-center

BTE: Brake thermal efficiency

BSFC: Brake specific fuel consumption

CH_2O: formaldehyde

CNG: Compressed natural gas

CO: Carbon monoxide

CO_2: Carbon dioxide

COV: Coefficient of variation

COx: Carbon oxides

EPA: Environmental Protection Agency

EGR: Exhaust gas recirculation

H_2: Hydrogen

H_2S: Hydrogen sulfide

HC: Hydrocarbon

HCNG: Hydrogen-compressed natural gas

HNG: Hydrogen-natural gas

HRR: Heat release rate

ICE: Internal combustion engine

IMEP: Indicated mean effective pressure

ITE: Indicated thermal efficiency

N_2: Nitrogen

NMHC: Nonmethane-hydrocarbon

NSPS: New Source Performance Standards

NOx: Nitrogen oxides

O_2: Oxygen

PM: Particulate matter

RE: Reciprocating engine

SIE: Spark ignition engine

SOx: Sulfur oxides

SMR: Steam methane reforming

TDC: Top-dead-center

VOCs: Volatile organic compounds

Chapter 1: Introduction

Hydrogen economy is a popular topic that attracts lots of investments and attention as one of the efforts to decarbonize many economic sectors. Many studies have been carried out on multiple aspects of hydrogen energy such as production, transportation, storage, and application for sustainable development. More and more methods to integrate hydrogen into the energy sector and current infrastructure are introduced to promote the growth of hydrogen economy. The transition to hydrogen economy and the utilization of hydrogen energy could lead to a more sustainable energy system and more novel technology developments. Nevertheless, the widespread adoption and development of hydrogen-powered engines remain significant challenges that need to be addressed.

Two-stroke engines have been popular for their light weight and high power-to-weight ratio [1], which make them suitable for a wide range of applications, including motorcycles, boats, and many off-road vehicles. However, two-stroke engines are known for their high pollutant emissions such as hydrocarbons (HCs), carbon monoxide (CO), and nitrogen oxides (NOx), compared to four-stroke engines [2]. Unlike four-stroke engines which burn pure fuel, two-stroke engines burn lubricating oil while combusting and a considerable amount of this oil is released as unburned gas. Due to this lubricating oil consumption and scavenging process, two-stroke engines produce high HCs emission. Moreover, methane slip, i.e., uncombusted methane that escapes from engines, is a prevalent phenomenon in gas-burning engines which operate according to Otto cycle. There are several sources that allow methane to evade combustion and be ejected via the engine ventilation such as scavenging process, quenching effects, or delayed combustion. Otto gas engines, especially those with lean-burn combustion system, generally emits the highest amount of unburned methane among different engines concepts [3]. Methane slip also occurs the most at low engine loads because of poor combustion. The aforementioned emissions contribute to air pollution and have detrimental effects on human health and the environment.

In recent years, there has been growing interest in developing alternative fuels and technologies to mitigate emissions from two-stroke engines. One promising approach is using the blended fuels of methane (CH_4) and hydrogen (H_2), which are known for their clean-burning properties and low emissions [4]. Hydrogen is often produced as a byproduct in various industrial processes, such as oil refining or chemical manufacturing. Blending surplus hydrogen with methane can provide valuable uses for the hydrogen that might otherwise be wasted or released into the atmosphere, thereby reducing waste and maximizing resource utilization. Moreover, hydrogen infrastructure, such as production, storage, and transportation, is still being developed. Blending hydrogen with methane is a way to leverage the existing natural gas infrastructure for hydrogen utilization and thus provide a transitional step towards a more a hydrogen-based energy system [5, 6]. Blending hydrogen can also be treated as a hydrogen transportation method by using natural gas pipelines and as a result, it creates opportunities for remote hydrogen production. Furthermore, blending hydrogen with methane can enable the use of hydrogen in existing natural gas equipment, such as boilers, furnaces, and engines, without requiring significant modifications [6]. This can be a cost-effective way to start incorporating hydrogen into existing infrastructure and utilizing hydrogen as an alternative or complement to natural gas. The present study is one of the real-life approaches to incorporate hydrogen fuel to the existing engine structure by using H_2-CH_4 blends for a two-stroke engine.

Although the addition of hydrogen to methane can improve combustion properties and reduce emissions by promoting complete combustion, increasing flame speed, and reducing ignition delay, there remains several challenges and complications. Hydrogen has a higher flame speed compared to methane, which means it burns faster [7]. This causes challenges in achieving the correct ignition timing in two-stroke engines, which relies on precise timing of the air-fuel mixture ignition to operate efficiently. If the ignition timing is not properly adjusted, it can result in poor engine performance, reduced power output, or even engine damage. In addition, combustion of hydrogen-methane blends can have different characteristics compared to pure methane or conventional hydrocarbon fuels. For example, hydrogen has a high flame speed and wide flammability, which can affect combustion stability, flame propagation, and emissions [7]. Managing these combustion characteristics can be challenging and may require careful optimization of engine design, fuel injection or carburetion, and ignition timing. Moreover, while hydrogen is a clean-burning fuel that produces only water vapor when being burned, the combustion of hydrogen-methane blends can still result in emissions of NOx and other pollutants. Achieving low emissions levels may require additional emissions control technologies, such as exhaust gas recirculation (EGR) or catalytic converters, which can add complexity and cost to the engine system.

Many researchers have investigated the impact of hydrogen blends on engines; however, experimental study of two-stroke engines is not pursued. Considering the current market situation and interest of different industries in using hydrogen in large scale engines including large two-stroke engines, the findings of a large-bore two-stroke natural gas-fired reciprocating engine running at different loads and mixture of hydrogen-natural gas fuel are presented in this thesis. Without altering the ignition timing, the tests are designed to examine how the hydrogen-natural gas fuel affects emissions and engine performance. This thesis experimentally evaluates the impact of blending ratios, engine operating conditions, air-fuel ratios, and the effects of catalysts on emissions reduction and combust characteristics. Engine emission and performance aspects are the main problems examined in this thesis. Additionally, trade-offs between emissions and efficiency are considered to determine an optimal condition of hydrogen volume fraction and engine load. The results can be adopted for engines that have a similar combustion system and working mechanism to those of the AJAX DPC-81.

Figure 1. AJAX-DPC 81 engine at the Sustainable Energy and Carbon Management Center, University of Oklahoma.

Chapter 2: The Problem

Fossil fuel shortage and severe environmental crises are inevitable with the current rate of impact from human activities. These issues raise worldwide awareness about environmental conservation and the future of humanity. Hence, there has been numerous efforts trying to solve the fuel substitution and emission reduction problems. Whether an engine is two- or four-stroke, fuel oxidation occurs in the form of combustion through the energy conversion process. Burning fossil fuels is associated with the emissions of NOx, CO, CO_2, sulfur oxides (SOx), and unburned HCs. These emissions contribute to global warming and many abnormal weather phenomena. In addition, geopolitical conflicts brought on by the uneven distribution of fossil resources worldwide impact the future of fossil fuel energy. Hence, hydrogen-based energy solutions are becoming increasingly popular.

Currently, natural gas is known as one of the cleanest and most efficient fuels. However, there are some penalties that come with such advantages. Engines run with NG potentially have high misfire ratio and incomplete combustion which are caused by the low burning velocity and poor lean-burn ability of NG [9]. Compared to other fuels, the burning of NG produces less particulate matter (PM), but still creates considerable amounts of CO, VOCs, and formaldehyde (CH_2O). Beside NG, hydrogen is another potential candidate. However, to use pure hydrogen as fuel, significant engine modifications are required because of issues like low power density, excessive NOx emissions, or anomalous combustion, i.e., backfires or knocking [6, 10]. In order to directly address the issues with pure hydrogen, it becomes interesting to employ hydrogen that has been combined with other fuels. Therefore, studying how well engines run on hydrogen, natural gas, and their mixtures is an essential task.

Many previous studies on classical internal combustion engines (ICEs) have shown that blending hydrogen with conventional hydrocarbon fuels leads to numerous advantages [5, 7, 10]. Bauer and Forest [10] and Wong and Karim [11] examined engine performance and combustion characteristics of a spark ignition engine (SIE) by blending various hydrogen volume fractions with NG. They discovered that adding a small amount of hydrogen could increase the flame propagation rate. Moreover, ignition delay period and combustion time loss were also reduced [10]. Furthermore, various researchers also carried out experimental studies on the lean-combustion capability of natural gas-hydrogen blend [10, 11]. The results show that there was a reduction in HC, CO, and CO_2 emissions. However, the concentration of NOx might increase due to the increased flame propagation speed if the engine was operated at rich-burn conditions [11]. Consequentially, a lean mixture combustion would decrease the NOx concentration [10]. These studies above even though included various types of hydrogen blends and engine concepts, most of them focus on four-stroke engines whereas many engines operated in the oil and gas industry are two-stroke which actually produce higher emissions and contribute significantly to the overall pollutant emissions.

Furthermore, the U.S. Environmental Protection Agency (EPA) established the New Source Performance Standards (NSPS) which all stationary reciprocating internal combustion engines must comply. This NSPS regulates hazardous air pollutants such as NOx, CO, and VOCs. To remain eligible for operation, each stationary engine must be tested regularly for compliance. Therefore, it is crucial to develop a method to control emissions, and hydrogen fuel blend is one of the promising approaches. Many studies were carried out to investigate the effect of hydrogen on engines; however, most of them focused on transportation four-stroke ones. To complete the understanding of hydrogen's effect on reciprocating engines, this thesis examines two-stroke engines, particularly, the industrial-sized ones used in oil and gas sector. The table below shows the emissions standard that the AJAX DPC-81 must comply with.

Table 1. New Source Performance Standards for stationary engines [12].

Engine type and fuel	**Maximum engine power**	**Manufacture date**	**Emission standards (ppmvd ref. 15% O_2)**		
			NOx	**CO**	**VOCs**
Non-emergency SI NG and non-emergency SI lean-burn LPG (except lean-burn 500 ≥ HP < 1350)	HP < 500	7/1/2008	220	610	80
		7/1/2011	150	610	80

Chapter 3: Background Information

I. Reciprocating engine

Reciprocating engine (RE) is the most common engine type worldwide. There are billions of such engines in service around the world. They power everything, from cars to ships or even power plants. Internal combustion engines make up the majority of the number above. Due to its cost-efficiency and robustness, internal combustion reciprocating engine is the most widely used engine, especially in the oil and gas industry. Similar to most engine types, reciprocating engines use fossil fuel to convert chemical energy to mechanical energy. This process produces waste materials that are hazardous to the environment. Moreover, the rapid depletion of fossil fuel supplies also impacts the future of ICEs.

I.1 Two-stroke engine

Two-stroke engine is one of the most widely used engines around the world along with four-stroke engine. Two-stroke engine is a type of ICEs which completes one power cycle within one up and down movement of the piston, which is called stroke. Figure 2 shows the diagram of a conventional two-stroke engine.

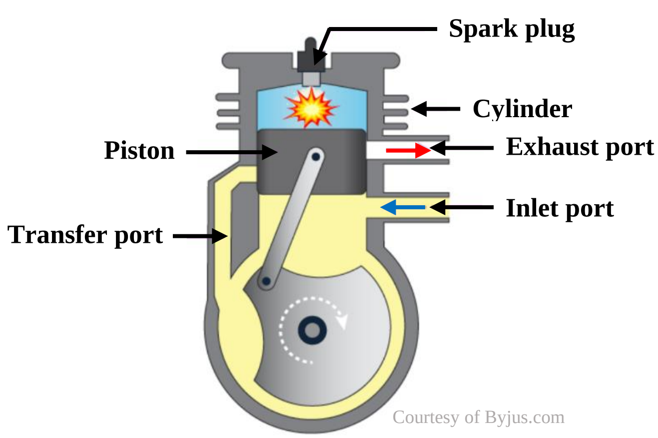


Figure 2. Construction of a two-stroke engine.

In the first stroke or down stroke, as the piston moves from the top-dead-center (TDC) to the bottom-dead-center (BDC), a fresh air-fuel mixture is drawn into the crankcase through the inlet port. At the same time, the mixture flows into the combustion chamber through the transfer port. In the second stroke or up stroke, the piston moves from the BDC to the TDC compressing the air-fuel mixture. When the mixture is compressed, it is ignited by the spark plug, and the second cycle starts. The hot gas expands and pushes the piston down to the BDC reducing the volume of the crankcase and creating a partial vacuum in the combustion chamber. Therefore, the new air-fuel mixture is forced into the combustion chamber. Simultaneously, as the fresh mixture occupying the combustion chamber volume, it expels the exhaust fumes out through the exhaust port. In a two-stroke engine, the end of the combustion stroke and the beginning of the compression stroke happen concurrently, while the intake and exhaust port function at the same time.

This type of engine is characterized by its mechanical simplicity and flexibility as it contains fewer moving parts and can work in any orientation since it does not have any reservoir or depend on gravity. Two-stroke engines have high power-to-weight ratio and significant power boost. However, in exchange for high power and instantaneous power delivery, two-stroke engines produce significantly more emissions as a result of incomplete combustion. They are also less efficient in terms of fuel consumption and thermal efficiency. Therefore, when it comes to environmentally friendly concerns, two-stroke engines have been phased out in most on-road vehicle uses.

I.2 Reciprocating compressor

A reciprocating compressor is a device that uses piston-driven crankshafts to pressurize and deliver gas. A compressor can have one or multiple cylinders. The cylinders can be arranged in many orientations such as inline or oppose. The working mechanism of reciprocating compressor is very much similar to that of a two-stroke engine.

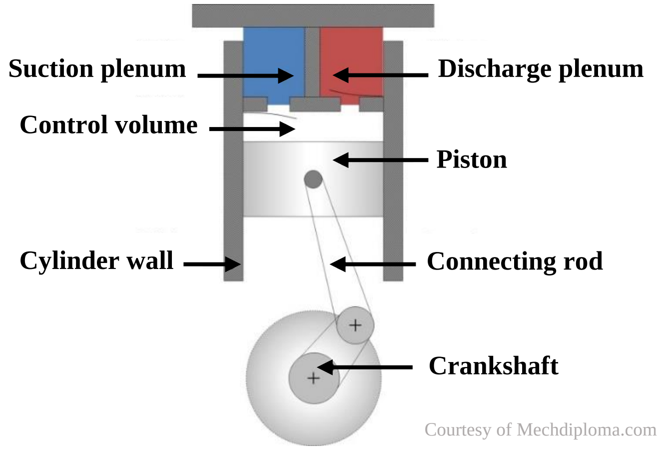


Figure 3. Construction of a reciprocating compressor.

Each cylinder has a suction valve and a discharge valve which operate on pressure difference. These valves are located in the head of the cylinder connecting the control volume to the suction plenum and discharge plenum. A compressing process in one cylinder is completed with one revolution of the crankshaft. As the piston moves to its BDC, since both valves are closed, this movement creates a low pressure volume in the cylinder. When the pressure is lower than the suction pressure, the suction valve opens allowing gas to flow into the cylinder. Then, when the piston moves to the TDC, it compresses the gas in the cylinder. When the gas is pressurized enough, the discharge valve opens expelling the compressed gas out of the cylinder.

Reciprocating compressors are one of the most critical machines in the oil and gas industry. They are also expensive and usually require high initial installment costs. Gas compressors can be found in almost any production facility such as well sites, petrochemical plants, oil refineries, and transmission gas pipelines.

II. Natural gas

Natural gas is one of the most widely consumed types of fossil fuel along with gasoline and diesel. It is a mixture of different gases including light HCs, CO_2, hydrogen sulfide (H_2S), and nitrogen (N_2) [13]. Due to being formed naturally, the composition of NG is not constant, but methane is always the major component. It occupies anywhere from at least 90% to 99% of the total volume [13, 14]. Up to now, NG has been known as the cleanest form of fossil fuel. In fact, it is the most promising alternative to petroleum [15]. NG has been used as an additive in ICEs because gasoline and diesel engines can be converted to adapt to NG easily [16]. Not only NG is widely available, but it is also less expensive than gasoline or diesel. Since methane is composed mostly of hydrogen, its combustion produces less harmful emissions compared to other forms of petroleum. Additionally, possessing high flammability, NG is able to burn easily and mostly achieve complete combustion [13]. Compared to other types of fossil fuels, NG is one of the safest because of its high auto-ignition temperature and narrow flammability range [13]. NG is also lighter than air allowing it to dissipate easily in case of leakages and thus would not cause suffocation [13]. Moreover, since it is an energy source, NG can be consumed in its natural state making NG an efficient fuel [17].

Compared to gasoline and diesel, NG produces better thermal efficiency and higher power output [18]. Such advantages are yielded by high octane number in NG allowing engines to operate smoothly with little knocking even at high compression ratios [18]. NG has a broad combustion range with lean-burn quality which lowers exhaust emissions and fuel consumption [19]. These advantages make NG an excellent fuel for ICEs. However, NG still possess certain penalties. NG has low burning velocity and poor lean-burn capability which cause engines running with NG to potentially have high misfire ratio and incomplete combustion [20]. Engines that run on NG have low thermal efficiency and large cycle-by-cycle variation which generates lower power output while having higher fuel consumption [20]. Despite having high volumetric heating value, NG's low flame speed requires special burners for better combustion [21]. Also, even though the combustion of NG generates lower exhaust emissions compared to the combustion of petroleum, it still releases harmful gases to the environment such as NOx, CO, CO_2, and VOCs. These reasons have enhanced the need for the investigation and development of an alternative approach to improve engine performance and decarbonize the world.

III. Hydrogen

Hydrogen is an appealing alternative because of its distinctive and valuable properties. First, hydrogen is a renewable energy carrier and the most abundant element on Earth [22]. Next, hydrogen possesses anti-knocking ability due to its high self-ignition temperature [22]. The high energy-mass coefficient of hydrogen reduces engines' specific fuel consumption and increases combustion efficiency [23]. Especially, hydrogen has high diffusivity which accommodates its mixing potential with other gases [24]. Only a small amount of hydrogen mixed with air is needed to combust, thus hydrogen can create ultra-lean combustible mixtures which produce low flame temperature, higher engine efficiency, and lower NOx exhaust emissions [25, 26, 27]. From experimental investigations, Carlucci found that automobiles run on NG and hydrogen mixture emitted less NOx, CO, and HCs [28]. Previous research has shown that improvements in combustion properties and reduction in NOx were achieved when injecting hydrogen to NG engines, explicitly for lean-burn cases [25, 26, 27]. Hence, hydrogen has long been considered as a fuel additive to many conventional fuels, especially those used in ICEs.

Hydrogen has a high burning velocity which is about six times that of NG leading to higher thermal efficiency [22]. Additionally, due to the small quenching distance of hydrogen, hydrogen flame can burn closer to engine cylinder wall and deeper into crevices resulting in more complete combustion [29]. Hydrogen also ranks among the fuels with the highest heating values per mass unit, which indicates that combustion of hydrogen is able to release more heat than most other conventional fuels with the same mass. Engines operating on hydrogen consume less fuel while having increased efficiency owing to hydrogen's highest energy-mass coefficient [21]. Moreover, hydrogen is famous as the cleanest known fuel since its combustion produces zero carbon emissions. The combustion products of hydrogen and air only contain water and a reduced amount of NOx [22]. The lean combustion of hydrogen and air produces significantly less NOx than the combustion of gasoline and air [25]. These properties make hydrogen an excellent additive for NG to improve its burning velocity, enhance combustion characteristics, and minimize environmental impacts.

Despite having such potential, utilizing hydrogen energy remains challenging. Even though hydrogen is the most abundant element, it is not an energy source but rather an energy carrier. This means that hydrogen must be produced from original resources or other substances. Hydrogen production alone is already a challenge, since there are not many currently known ways to produce hydrogen and only a few of them are considered clean. Additionally, due to its high flammability, low volumetric energy density, and smallest molecule size, hydrogen storage is a complicated problem that is still under research. The most important issue that makes it challenging to adopt hydrogen to the energy sector is the high cost of hydrogen production. Currently, 95% of hydrogen today is produced by steam methane reforming (SMR) which costs almost three times the price of NG [30] and clean hydrogen is even more expensive [31]. Besides, steam methane reforming without carbon capture also generates a considerable amount of CO_2 per kilogram of produced hydrogen [30]. Thus, replacing NG with hydrogen produced by SMR defeats the purpose of achieving net-zero emissions. One can argue that hydrogen produced from water electrolysis does not produce CO_2, but then the problem is traced down to the source of the electricity used for electrolysis. If the electricity is generated from nuclear power, then the hydrogen produced using it cannot be considered as clean. Therefore, switching to hydrogen fuel is far from economically and practically viable at the moment. However, the U.S. Department of Energy launched a project aiming to reduce the price of clean hydrogen to $1 per kilogram in one decade [31]. This is a good sign promising that hydrogen fuel will become feasible in the near future.

IV. Hydrogen-natural gas blend

One effective approach to the reduction of emissions problem is to introduce hydrogen into the energy sector. Many researchers have proven that hydrogen-natural gas (HNG) is an ecofriendly fuel for ICEs. The future of HNG becoming a global fuel will be promising if low cost hydrogen production and hydrogen supply infrastructures are made available [32]. As discussed above, even NG has the upper hand in efficiency, supply, price, and environmental effects compared to petroleum, it still has certain disadvantages. The most critical one is its slow burning characteristic. To improve the burning velocity, NG can be mixed with a fast burning velocity gas such as hydrogen. HNG mixture shall possess the characteristics in between those of NG and hydrogen. This mixture is expected to have lean-burn characteristics while decreasing emissions.

Moreover, most pollutants are combustion products of carbon-based fuels, adding hydrogen to NG will reduce the pollutants such as CO, CO_2, and VOCs. However, there are some main drawbacks of HNG fuel. Since hydrogen has a higher mass heating value compared to NG, the HNG mixture burns hotter and creates more NOx unless it is lean-burn condition [14]. If the combustion temperature is higher than the temperature threshold of the engine, the cylinder can be damaged. Furthermore, since hydrogen has a very low volumetric energy density, as hydrogen's volume fraction increases, this value drops. If too much hydrogen is added to the mixture, there are chances that the combustion of HNG will not supply enough energy to keep the engine working.

V. Emissions

Emissions from engine operation contribute significantly to the overall air pollution. The combustion of natural gas in reciprocating engines is even though cleaner, still generates considerable amounts of emissions. The primary pollutants from natural gas-fired engines are NOx, COx, CH_4, and VOCs. These hazardous gases are harmful to the environment and human health; therefore, it is crucial to develop emission reduction methods.

V. 1. Nitrogen oxides

NOx emission usually stands for nitric oxide (NO) and nitrogen dioxide (NO_2). NOx formation is not a direct product of combustion, but a product of the nitrogen-oxygen reaction initiated by the heat supplied by the burning of fuel. Thus, the amount of NOx is associated with the temperature in the combustion chamber [33]. This type of NOx usually consists of 95% NO and only 5% NO_2 [14]. Due to the dominant dependency on temperature, fuel-rich combustion facilitates rapid NOx formation [33]. NOx is responsible for many environmental concerns such as smog, acid rain, and global warming. This toxic gas is also hazardous to humans since it causes severe respiratory problems [33].

V. 2. Carbon oxides

COx refers to carbon monoxide and carbon dioxide. While CO_2 is a product of almost every combustion of any organic compound, CO is a product of incomplete combustion, i.e., lack of oxygen [14]. Both of the gases are harmful to the environment as they can cause climate change by warming up the planet. CO is well-known as a fatal gas since it deprives human lungs of oxygen, leading to poisoning or suffocation. ICEs emit the highest amount of CO if operated without catalytic converters [35]. In fact, 22% of CO emissions are produced by off-roads vehicles which include stationary engines like the one studied in this paper [35].

V. 3. Methane

Methane is a strong greenhouse gas. CH_4 emissions are often the remaining fuel from combustion procedure. The unburned fuel can be a result of misfire or abnormal combustion. Furthermore, wall quenching influence, incomplete combustion, oxygen insufficiency, and low combustion temperature are also the main causes of CH_4 emissions [36]. Due to operating on high CH_4 concentrated fuel, natural gas and petroleum systems contribute the most to the overall CH_4 emissions.

V. 4. Volatile organic compounds

VOCs, also called as non-methane non-ethane hydrocarbons, are released by a large range of products. Any products containing organic solvents or fuel made of organic chemicals can emit VOCs even when they are stored. In ICEs, when the higher hydrocarbons in the fuel are burned incompletely, VOCs are generated [37]. VOCs are dangerous gases since some of them cause cancer and many of them can deplete the Ozone layer [38].

VI. Performance

To assess the performance of the engine, the following parameters are usually investigated.

VI. 1. Indicated mean effective pressure

Indicated mean effective pressure (IMEP) expresses the work transferred to the piston by the gas. It can be thought of as the constant average pressure acting on the piston throughout one cycle disregarding what cycle portion the piston is at, i.e., independent of engine displacement. This parameter is fundamental to examine engine efficiency [39].

VI. 2. In-cylinder pressure

In-cylinder pressure is the pressure profile across the combustion chamber of an engine with respect to crank angle during its working strokes. Since the pressure inside the cylinder continuously changes, the pressure profile should be in the shape of a curve. The pressure curve indicates certain combustion characteristics such as peak pressure, knocking, misfire, primary ignition, and IMEP.

VI. 3. Heat release rate

Heat release rate (HRR), also known as power, is the rate at which energy is generated by combustion over a unit of time. In this paper, heat release rate is measured with respect to crank angle in Joule per degree unit, thus it will not be used interchangeably with power. Since HRR characterizes the type of combustion, it is used to define fire phenomena such as fire hazards. Higher HRR means higher chances of fire hazards happening; therefore, combustion product gases also increase in parallel to HRR.

VI. 4. Indicated thermal efficiency

Indicated thermal efficiency (ITE) is the ratio of the theoretical energy developed in the engine cylinder to the total heat of the consumed fuel. This thermodynamic variable indicates the amount of useful heat and measures how well the engine is able to utilize the total fuel energy. It should be noted that ITE assumes no energy loss due to friction between piston and cylinder wall or crankshaft and bearings.

Chapter 4: Literature Review

I. Effects of HNG on engine emissions

Many previous studies on classical ICEs have shown that blending hydrogen with conventional hydrocarbon fuels leads to numerous advantages. Shudo et al. performed experiments on a direct injection stratified charge engine with premixed hydrogen and methane to examine its combustion and emission characteristics [40]. They discovered that the engine achieved higher thermal efficiency and produced lower emissions [40]. The combustion process could be stabilized more if the amount of premixed hydrogen is increased [40]. Consequently, it would produce less HCs and CO in the combustion products yet increase the amount of NOx [40]. Raman et al. experimentally analyzed a four-stroke engine running on hydrogen-natural gas blend from 0% to 30% of hydrogen in volume [41]. The obtained results showed that from 15% to 30% hydrogen, ultra-lean combustion occurred, thus the amount of emitted NOx reduced while HCs increased slightly [41]. Larsen and Wallace tested hydrogen-natural gas blend on heavy-duty engines [42]. In contrast to Raman's findings, they observed a reduction in CO, CO_2, and HCs emissions and some improvement in efficiency [42]. Nevertheless, the amount of NOx recorded by Hoekstra and Larsen and Wallace were surprisingly low [42, 43].

Akansu et al. varied engine speed, engine load, and hydrogen mass fraction [44]. The results of their investigation show that increased hydrogen fractions contributed to lower CO, HCs, and CO_2 emissions. Additionally, an increase in hydrogen fraction led to an increase in NOx emission [44]. CO emissions were found to be 9%, 8%, 6%, 5%, and 5.5% for the respective mass fractions [44]. The largest difference from fossil fuels is found at 33.8% H_2 fraction with a difference of 4% [44]. HC values were found to be 237, 165, 141, 107, and 94 ppm with the respective mass fractions of H_2 [44]. The largest difference was found at 45.1% H_2 mass fraction with a difference of 143 ppm in HC content [44]. CO_2 emissions dropped by 6% at maximum hydrogen fraction at 45.1% [44]. Decreases in these carbon-based emissions could be attributed to the smaller content of carbon within the blended fuels, especially considering the 33.8% and 45.1% H_2 fraction levels [44]. NOx emissions increased with the addition of blended hydrogen, with 1800 ppm NOx at 0% and a maximum of 4500 ppm at 33.8% H_2 fraction [44]. There was also a decrease in NOx presented from 33.8% to 45.1% H_2 fraction of 1420 ppm, suggesting higher hydrogen fraction may be able to decrease the overall NOx [44]. The increased NOx value was due to the increase in flame temperature which causes a chemical reaction between the oxygen and nitrogen components of air [44]. Collier et al. conducted similar experiments to those of Akansu where they tested an engine with hydrogen-natural gas mixture with various hydrogen volume fractions [45]. They concluded that adding hydrogen increased NOx emissions and decreased total HC emissions which aligns well with Akansu's conclusion. They also discovered that the addition of hydrogen extended the combustion lean limit [45].

Wang et al. experimentally studied how hydrogen fraction in the fuel blend impacted the combustion characteristics of a direct injection engine [46]. They found that at low and medium engine loads, addition of hydrogen caused HRR to increase and NOx emission also increased along with that [40]. From the results, they suggested the optimal hydrogen volume fraction in hydrogen-natural gas mixture is approximately 20% to satisfy both engine performance and emissions with minimum trade-offs [46]. Meanwhile, Sierens and Rossel discovered in their experiment that the optimal hydrogen-natural gas blend compositions should vary with engine load [47]. Bauer and Forest studied the combustion characteristics of HNG blend in a SIE [10]. They demonstrated in their paper that NOx increased with increasing hydrogen percentage [10]. Moreover, they observed a general reduction in CO, and they concluded that the production of CO was not highly dependent on equivalent ratio but rather on the engine's nature [10]. Furthermore, experimental studies on lean combustion capability of natural gas-hydrogen blend were also carried out by various researchers [45, 48, 49]. The results show that there was a reduction in HCs, CO, and CO_2 emissions [48, 49]. However, the concentration of NOx might increase due to the decreased flame propagation speed if the engine was operated at rich mixture conditions [49]. Consequentially, a lean mixture combustion would decrease the NOx concentration. On a similar study, Hoekstra et al. noted that NOx reduced as hydrogen percentage increased up to 30%, yet no further reduction was observed beyond 30% hydrogen fraction [43].

Aydin et al. varied fuel type by blending hydrogen with diesel [50]. This study tested emissions with increasing engine speeds and hydrogen fractions from 20% to 30% [50]. It was found in this study that with the addition of hydrogen all emissions other than NOx decreased [50]. Hydrogen-diesel blend reduced the amount of CO emitted [50]. Particularly, CO was reduced by a maximum of 38% [50]. However, there was an increase in NOx emissions by the averages of 16% [50]. The decrease in CO and increase in NOx could be attributed to the hydrogen mass heating value being greater than that of fossil fuels and the lowered amount of burning carbon in the blends [50].

Ceper et al. investigated a four-stroke, four-cylinder engine on various hydrogen-methane compositions with different engine loads and excess air ratios [51]. They observed that as the excess air ratios increased, CO and CO_2 emissions decreased with increasing hydrogen fraction [51]. Similarly, Kahraman et al. examined the performance and emissions of an SI engine running on hydrogen-methane blends with different hydrogen fractions at various engine speeds and excess air ratio [52]. Their results showed that CO emissions decreased when engine speed and excess air ratio increased which comes in good agreement with Ceper's results [52]. Dimopoulos et al. modified a car engine for hydrogen-natural gas blend fuel and high exhaust gas recirculation [53]. They found that the addition of hydrogen promoted combustion which resulted in improvement in efficiency [53]. The engine generated 7% less greenhouse gas emissions compared to when it is fueled by diesel [53].

Das investigated an SIE with compressed natural gas (CNG) and 18% hydrogen-compressed natural gas (HCNG) fuels [54]. They noticed that CH_4 emission was significantly lower and HCs emissions dropped slightly in the case of HCNG. Additionally, when the engine ran with CNG only, it produced a high amount of formaldehyde, but with the addition of hydrogen, this gas was controlled [55]. When comparing CH_4-H_2 mixture to pure CH_4 operations, Yusuf observed that CH_4-H_2 mixture increased NOx while reducing the level of unburned HCs and CO [18]. Moreover, after being enhanced with hydrogen, the lean limit combustion of NG dropped from 0.61 to 0.54 [18]. Konoplev et al. conducted experimental investigations on the effects on emissions of lean HNG on an SIE [56]. They confirmed that lower concentrations of pollutants in exhaust gas were attained by HNG as expected. Since they experimented on a lean-fuel engine, their primary emission concerns were NOx and HCs. From their results, implications between NOx and HCs emissions and HNG admix can be drawn.

According to the discussion above, hydrogen was proven to have positive impacts on engine emission as supposed. However, the increased amount of NOx after injecting hydrogen can be an obstacle preventing the expansion of hydrogen fuel blend. Even though the working mechanism of the engines studied in the research above can be different from the one used in this study, similar results shall be expected. Therefore, the present study aims to observe the impacts of H_2-NG blend on the emission of two-stroke engines.

II. Effects of HNG on engine performance

Besides emissions, engine performance with enhancement of hydrogen is another crucial aspect that must be considered before implementing hydrogen blended fuels. Nagalingam et al. studied the properties of hythane, i.e., hydrogen enriched compressed natural gas [57]. In their paper, they concluded that hythane produced lower power than that of methane since hydrogen has a lower volumetric heating value [57]. Moreover, while studying how hydrogen fraction in the fuel blend can impact the combustion properties, Wang discovered that at low and medium engine loads, addition of hydrogen increased brake thermal efficiency [46]. Wong and Karim examined engine performance by blending various hydrogen volume fractions with NG [11]. It was concluded from their result that the enrichment of hydrogen extended engine's operation regions and reduced cyclic variations [11]. Bauer and Forest analytically studied HNG in an SIE and discovered that the flame propagation rate could be increased by adding a small amount hydrogen [10]. Moreover, ignition delay period and combustion time loss were also reduced [10].

In the study of Akansu et al. that was mentioned in the previous section, interesting findings about combustion characteristics were recorded [44]. The results of their investigation show that thermal efficiency generally increased with hydrogen fraction. The increased hydrogen fractions contributed to lower CO, CO_2, and HCs emissions. Increase in hydrogen fraction also led to increase in NOx emission [44]. Thermal efficiencies were recorded at 0%, 10.7%, 21.3%, 33.8%, and 45.1% H_2 mass fractions blended with gasoline and ethanol [44]. These fractions gave values of 25.6%, 28.7%, 29.5%, and 31.1%, and 30.9% respectively for thermal efficiency [44]. The increase in thermal efficiency can be attributed to the high heating value and flame propagation of hydrogen [44]. The maximum increase is seen at 33.8% H_2 fraction with an increase of 5.5% [44].

Das analytically studied the performance of an SI engine running on an 18% HCNG blend [54]. The results indicate that without adjusting ignition timing, the engine consumed 18% more fuel [54]. When implementing a catalytic converter and exhaust gas recirculation, the fuel consumption decreased slightly [54]. Additionally, Konoplev et al. conducted experiments on an AVL engine with various HCNG concentrated mixtures, from 0% hydrogen to 100% hydrogen [58]. They noticed that engine output and thermal efficiency were lost about 23% and 12% respectively when pure hydrogen was used [58]. For HCNG cases, an increase in flame speed occurred because the optimal spark timing was found to drop by up to 20° before TDC [58]. From the results above, it can be concluded that the performances of HCNG engines shall be in between those of pure hydrogen and NG [58, 59]. Genovese et al. varied the hydrogen volumetric content in HCNG blends from 5% to 25% [60]. They found that average engine efficiency increased with hydrogen concentration and NOx was highest with 20% and 25% hydrogen fractions even when lean air-fuel ratios and delayed ignition timing were accounted for [60]. Furthermore, in Yusuf's comparison of CH_4-H_2 mixture to pure CH_4 operations at the same equivalent ratio, they concluded that the mixture increased brake thermal efficiency (BTE) yet decreased the best efficiency spark advantage [18]. Ceper et al. also confirmed in their study that adding hydrogen caused peak pressure to approach the top dead center, and thus increased BTE [51].

Overall, the literature shows that the further increase of hydrogen improves engine performance in terms of less knocking and more stability. Moreover, thermal efficiency increases with hydrogen concentration in the fuel mixture. While most previous studies experimented upto 30% H_2 volume fraction blends, the approach of this present study shows that it is possible to inject H_2 more than that concentration.

Chapter 5: Methodology

I. Apparatus

I. 1. AJAX DPC-81 integral engine – compressor

The experiments were conducted at the Sustainable Energy and Carbon Management Center located on the campus of the University of Oklahoma. The engine used for this study was an AJAX DPC-81 compressor. AJAX DPC-81 is a large-bore two-stroke natural gas-fired internal combustion reciprocating engine which was produced by Cooper Machinery Services. Its detailed specifications are described in Table 2. This engine is an integral compressor-engine unit which indicates that there is a compressor installed on the same crankshaft as the engine. The engine is able to operate at different loads which is defined by three factors including maximum discharge pressure, average compressor suction pressure, and engine speed. The load of the engine was computed by PowerFlow software provided by Cooper Machinery Service. The abovementioned factors were adjusted manually to set engine load.

Table 2. AJAX DPC-81 engine specifications.

Engine type	SI, two-stroke, natural gas aspirated, water-cooled
Rated power	81 bhp
Rated speed	475 rmp
Number of cylinders	1
Bore × stroke	10.5 in. × 12 in.
Swept volume	1039 in^3
Combustion type	Lean-burn

The AJAX DPC-81 is an integral engine-compressor which consists of a gas compressor and a reciprocating engine mounted in the same frame. There is only one crankshaft that runs the pistons of both the power cylinder and the compressor cylinder. Due to this single frame and crankshaft design, this type of engine is commonly used for field installation where low maintenance, low fuel efficiency, and long lifetime are crucial. Similar to the working mechanism of any two-stroke engine, the engine cycle starts with the intake air being drawn into the combustion chamber. After being compressed by the piston, the air-fuel mixture combusts and pushed the piston backward. The energy is transferred through the crankshaft driving the compressor piston and compressing gas in the compressor cylinder.

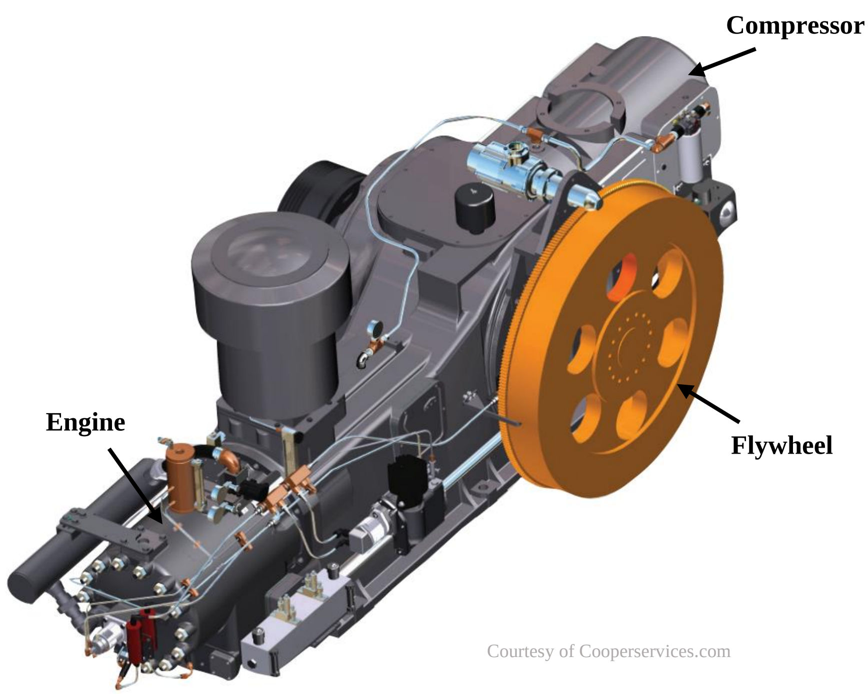


Figure 4. AJAX DPC-81 integral engine-compressor.

Some slight modifications were provided to accommodate the scope of this study and the data acquisition process.

- An oxidizer catalyst was attached to the engine's exhaust pipe to reduce the amount of exhaust CO.
- Hydrogen injecting system were connected to the settling tank of the engine to promote the blending process of hydrogen and natural gas.
- Thermal couples were arranged along the exhaust pipe to record exhaust temperature.
- A magnetic speed sensor was attached to the flywheel to measure the number of revolutions per minute.
- Flowmeters were installed on the fuel line and air intake port to measure volume flow rate.
- Combustion pressure sampling valve was manufactured in the engine head for in-cylinder pressure measurement.
- Exhaust gas sampling probes were placed before and after the catalyst for emission measurement.
- A bypass pipe with a controllable valve was added into the engine to adjust its air-to-fuel ratio (AFR).

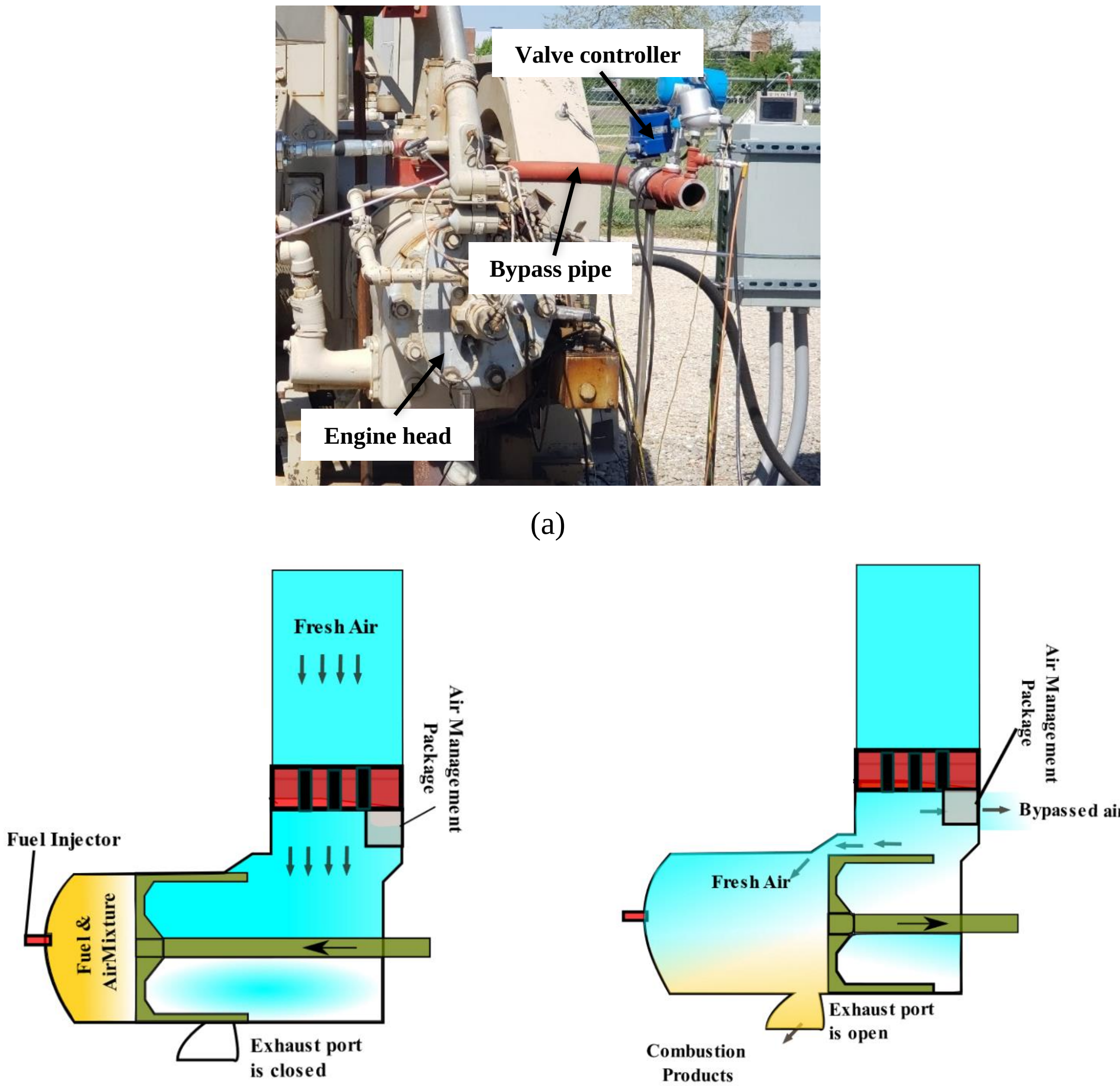


Figure 5. Bypass pipe components and operating mechanism.
a) Bypass pipe location. b) Bypass pipe closed. c) Bypass pipe opened.

Figure 5 shows the construction of the bypass system used in this experiment. This bypass system includes a bypass pipe and a butterfly valve. This valve allows a partial amount of intake air to exit before it enters the combustion chamber. This means that the more the bypass valve opens, the less air enters the combustion chamber or the richer the combustion becomes. Bypass mechanism is the most simplistic way to control the AFR during combustion by reducing the excess air in the combustion chamber.

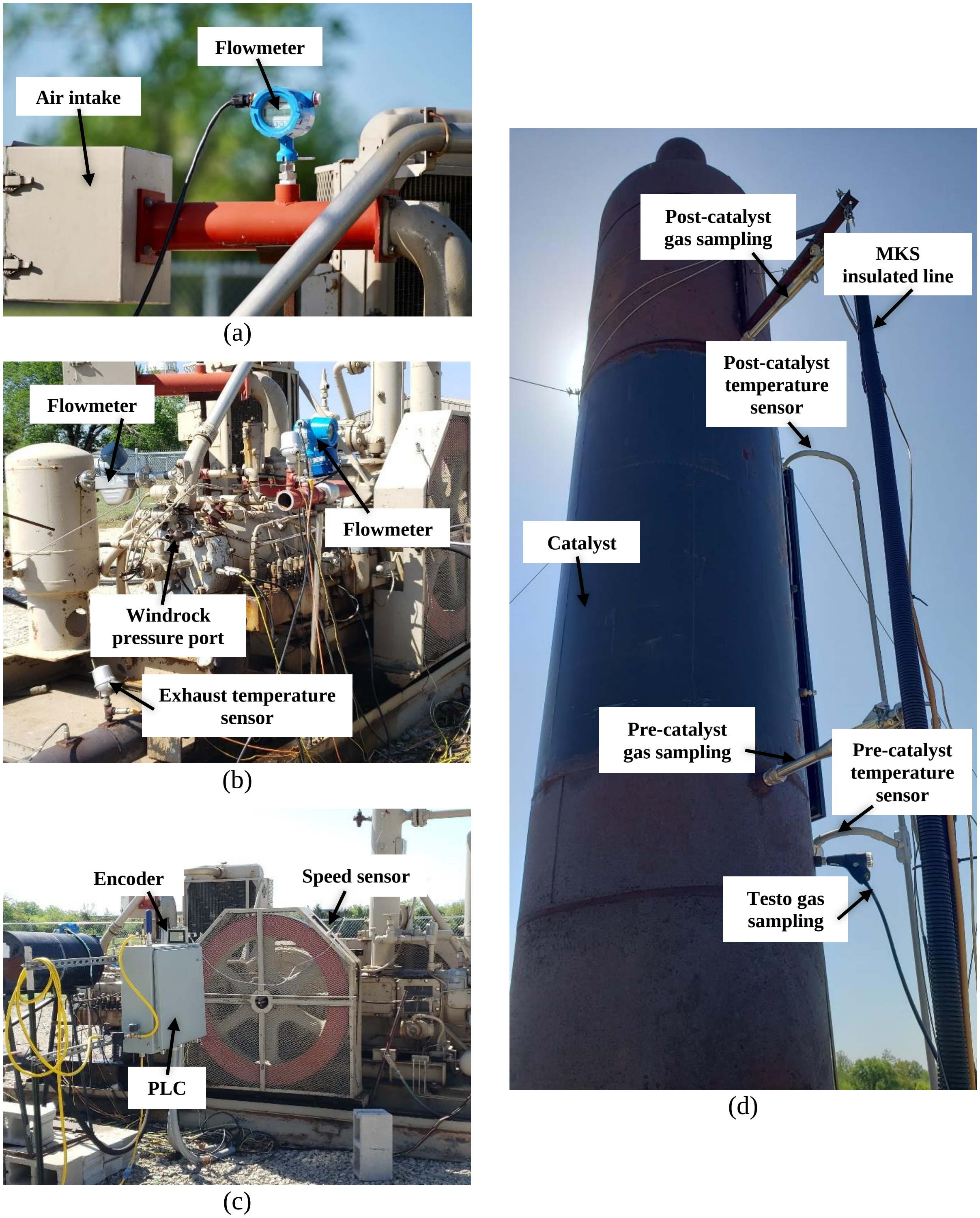


Figure 6. Engine modifications.

The engine is equipped with a catalyst on its exhaust pipe to convert hazardous emissions into less harmful substances. A catalyst is a substance that enables a chemical reaction to proceed faster without itself being consumed during the process. In other words, a catalyst is neither the reactant nor the product of the reaction. In a catalysis process, catalysts lower the activation energy, i.e., the energy required to trigger a reaction, causing the reaction to occur more quickly and efficiently. Therefore, in the presence of a catalyst, atoms break free easily from their molecules to form new substances. There are many types of catalysts, yet some of the most used in engines are three-way catalyst, oxidation catalyst, and diesel oxidation catalyst. Each of these catalysts are specified for certain types of engines to achieve the best result. For instance, three-ways catalysts are often used in rich-burn or stoichiometric engines [61]. In this type of catalyst, NOx, CO, and HCs in the exhaust gas react to oxidize one another to water, nitrogen, and CO_2 [61]. Oxidation catalysts are effective on lean-burn engines [61]. These catalysts mainly control NOx, CO, VOCs, nonmethane-hydrocarbon (NMHC), formaldehyde (CH_2O) [61]. Lastly, as the name suggests, diesel oxidation catalysts are used for diesel engines specifically.

Since AJAX DPC-81 engine is classified as lean-burn, it is equipped with an oxidation catalyst. This catalyst consists of a 304 stainless steel outer frame containing specialty stainless steel foils. The steel foils are corrugated to 300 cells per square inch forming a honeycomb structure to optimize emission performance and pressure drop. To properly fit the heavy-duty applications of AJAX DPC-81 engines, the foils are brazed to be durable in high mechanical stress and thermal shock.

Table 3. AJAX DPC-81 engine oxidation catalyst specification

Catalyst model	DC3W catalytic element (brazed)
Catalyst type	Oxidation
Number of elements	2
Cell density	300 cpsi

Table 4. Conversion efficiency of the DC3W catalyst.

Exhaust gas component	**Converter output (g/bhp·hr)**
NOx	NA
CO	0.588
NMNEHC	0.144
CHOH	0.13

I. 2. Data collection equipment

Since the scope of this study focuses on investigating engine performance and emissions, many thermocouples, pressure transducers, flowmeters, gas analyzers, and engine monitors were used. The three paramount devices are the MKS gas analyzer, Testo gas analyzer, and Windrock engine monitor. Two gas analyzers were used to collect emission data from the exhaust gas. The MKS 2030 Fourier Transform Infrared Spectroscopy (FTIR) analyzer is a device used for gas analysis applications such as detection and measurement. It is capable of collecting a large variety of gases, including NOx, COx, CH_4, and VOCs, by analyzing their infrared spectra. However, it is unable to record high concentrated gases such as oxygen or nitrogen. Since measurement of oxygen is required to optimize combustion, a Testo 300 combustion analyzer was used for this purpose. Additionally, a Windrock 6400 engine analyzer was used to assess engine performance and combustion stability. This device collects in-cylinder pressure with respect to crank angle, records flywheel speed, and monitors primary ignitions of the engine.

Courtesy of Testo.com | Courtesy of MKS.com | Courtesy of ChampionX.com

(a) (b) (c)

Figure 7. Data collection equipment.
a) Testo gas analyzer. b) MKS gas analyzer. c) Windrock engine monitor.

II. Experimental setup

To examine the effects of hydrogen on combustion characteristics and emissions, hydrogen was blended with natural gas to fuel the engine. An engine can be operated with different loads depending on the need of its user. With each load, the engine can perform differently. Thus, it is essential that the experiment should be conducted from the minimum load the engine is normally operated at to the maximum load the engine can handle.

On well sites, compressors pump gas up from the producing well and transfer it to the transportation pipelines. It depends on how much gas is in the well that those compressors are set to work at certain load to achieve the highest efficiency. At compressor stations, compressors mostly work at fixed loads to maintain the pressure and flow of gas in the pipeline to deliver it to the end users. The working scenario of the AJAX compressors was recreated for the purpose of this research. However, it is not possible to set up the research at an actual well site or a compressor station, the compressor in this study used natural gas from city pipeline to operate. To avoid wasting gas, the compressor was installed in a circuit setup as shown in Figure 8.

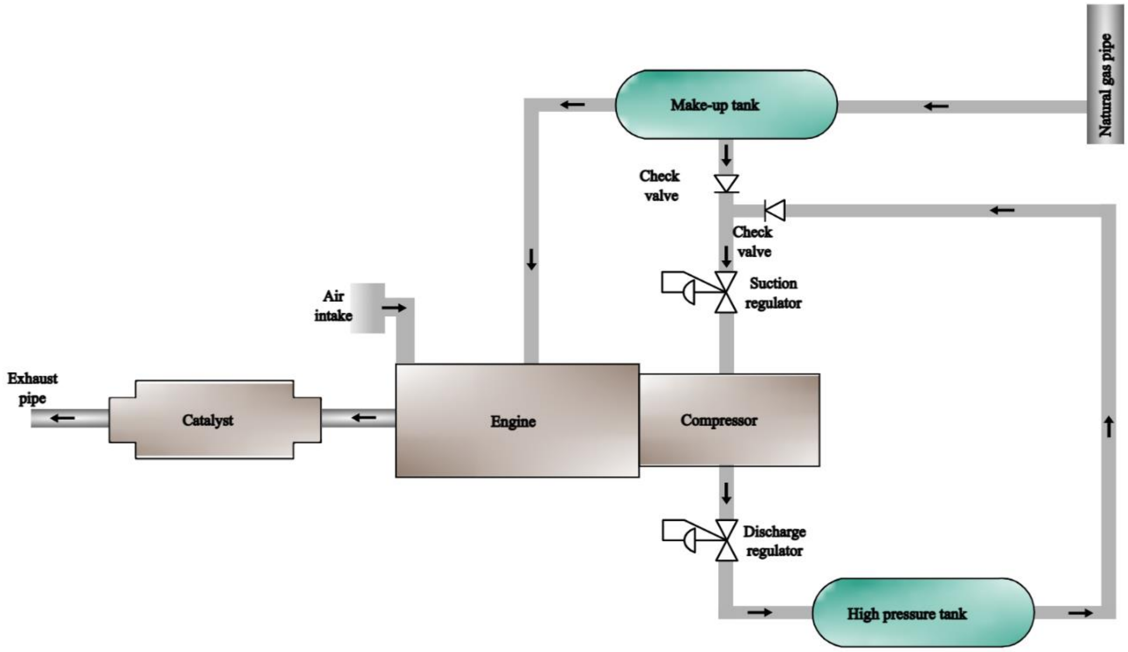


Figure 8. Engine setup.

The engine part of the AJAX draws fuel from a make-up tank containing natural gas from the city pipeline. The make-up tank, the compressor, and a high pressure tank are connected in a circuit. The compressor compresses gas supplied from the make-up tank and builds pressure in the high pressure tank. When the pressure is high enough, the compressor will stop taking gas from the make-up tank, and the suction regulator between the compressor and the make-up tank will keep the natural gas cycling in the circuit. A check valve was installed to stop the gas from following back into the make-up tank while allowing the tank to supply more gas to the circuit if the pressure is too low. The suction regulator and discharge regulator are used to set suction and discharge pressure at desired loads.

Figure 9 shows how the equipment and hydrogen injections system were connected to the AJAX engine.

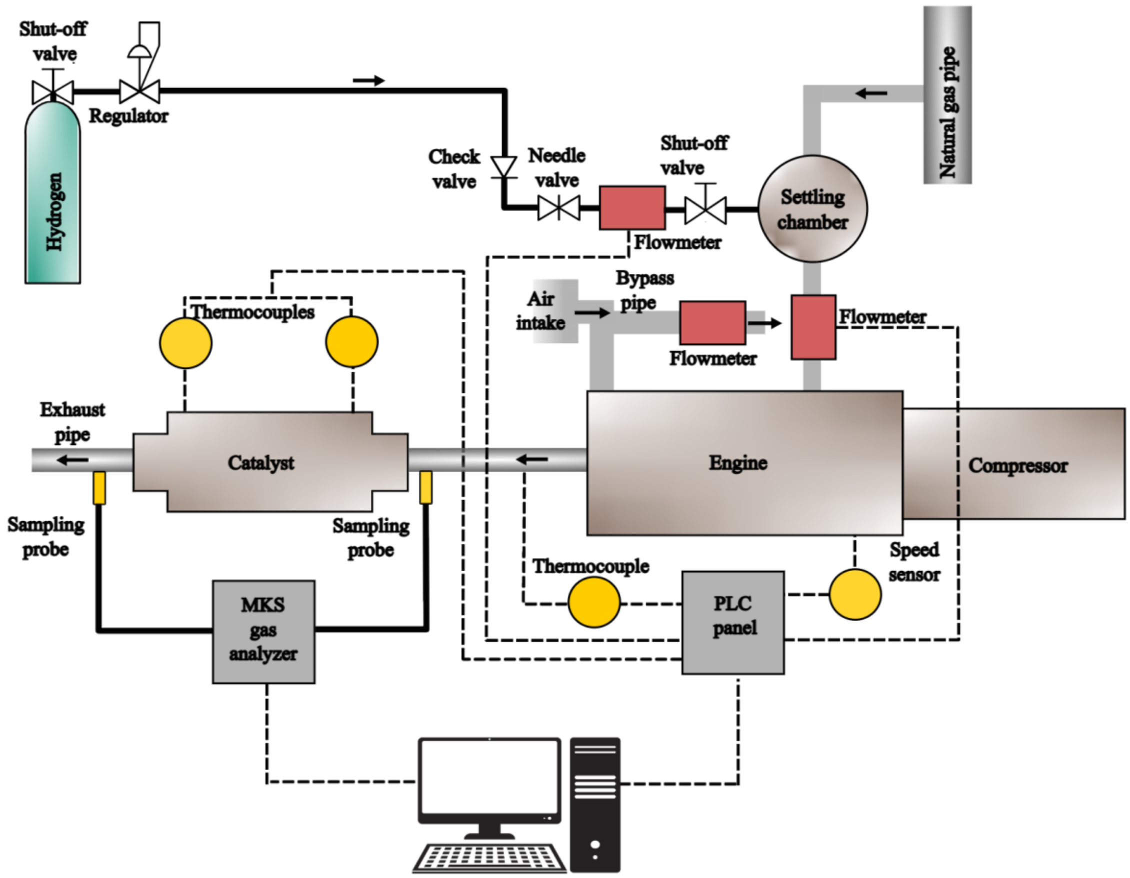


Figure 9. Experimental setup.

Ultra-pure hydrogen from Airgas, at least 99.9% pure, was used to blend with natural gas. In order to accommodate the pre-mixing process and to prevent possible flashback, hydrogen was injected into a settling tank instead of directly into the combustion chamber. The hydrogen cylinders were connected to the settling tank using ¼ in. stainless steel tube. Along the line, a series of shut-off valves and check vales were installed for easier flow control and backflow prevention. The volume flowrate of hydrogen was recorded using a Fox Thermal flow meter. The volume flowrate of the hydrogen-natural gas mixture was recorded by a Coriolis Micro Motion flow meter. The flowrate measured by the flow meters were used to compute the volume fraction of hydrogen in the mixture. A bypass pipe was installed after the air intake manifold to control the air to fuel ratio. The bypass and intake air flowrates were also recorded.

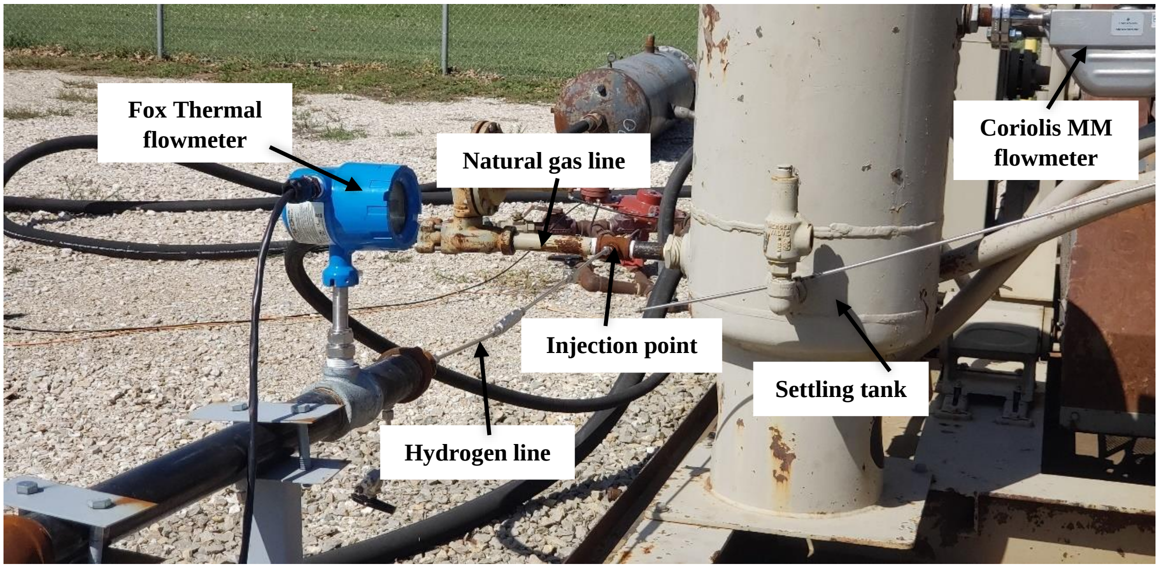


Figure 10. Hydrogen injection point.

III. Experimental procedure

The AJAX DPC-81 engine was tested from 40% load, which is equivalent to 32.4 BHP, to 75% load, which is equivalent to 60.8 BHP. Due to the way the engine was set up, it cannot run at 100% load. 75% load is the highest load the engine could achieve in this study. Table 5 displays the operating conditions of various loads.

Table 5. Testing conditions.

Operating parameters	**Loads**				
	40%	50%	60%	70%	75%
Power (BHP)	32.4	40.5	48.6	56.7	60.8
Max discharge pressure (psi)	30	40	50	55	60
Average suction pressure (psi)	420	430	450	520	550
Engine speed (rpm)	350	380	410	430	450

The experiment consists of two segments. One did not involve the use of the bypass valve, and one did. In the first segment, at each tested load, the engine was run with a hydrogen-natural gas blend up to 40% by volume with a 10% increment. At each condition, emission data was recorded pre- and post-catalyst. In-cylinder pressure and primary ignition were also collected for combustion analysis. In the second segment, the bypass valve was used to examine the effect of hydrogen and equivalent ratio on emission and combustion characteristics. Two bypass valve positions were tested in this segment, which were 30% and 60%. Similar to the first segment, at each tested load, the bypass valve was open at 30% and 60%, and hydrogen was blended up to 40% by volume.

IV. Analytical method

IV. 1. Emission analysis

To eliminate the effect of moisture on gas commodities and collect more reliable data, the compositions were converted to dry volume basis. Moreover, since the engine does not operate at one constant oxygen content but rather at various levels, a benchmark is required to normalize the measurements of the exhaust gas. Therefore, the emission data received from the MKS analyzer was converted to dry basis and normalized to 15% O_2 to compare with emission standards for stationary spark ignition engines issued by the NSPS.

$$\mathrm{x} = \mathrm{x}_{\mathrm{exhaust}}\left(\frac{21-15}{21-\mathrm{O}_2\%}\right) \tag{1}$$

$$\mathrm{ppm} = \mathrm{ppmvw}\left(\frac{100}{100-\mathrm{H}_2\mathrm{O}\%}\right) \tag{2}$$

where,

x: normalized mole fraction of the gas in ppm

$x_{exhaust}$: mole fraction of the gas measured in ppm

ppm: parts per million of the gas on dry basis

ppmvw: parts per million of the gas on wet basis

H_2O%: water composition in exhaust gas measured in percentage

O_2%: oxygen composition in exhaust gas measured in percentage

IV. 2. Heat release rate

Heat release rate (HRR) is a variable that depicts the rate at which energy is generated from the combustion of fuel. The released heat is mostly transferred through the piston and converted into mechanical energy. Based on the first law of thermodynamics, the heat release rate can be calculated from the model below.

$$\frac{dQ_n}{d\theta} = \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dP}{d\theta} \tag{3}$$

$$\gamma = \frac{C_P}{C_V} \tag{4}$$

where,

$\frac{dQ_n}{d\theta}$: heat release rate

$\frac{dV}{d\theta}$: rate of change of cylinder volume with respect to crank angle

$\frac{dP}{d\theta}$: rate of change of in-cylinder pressure with respect to crank angle

P: in-cylinder pressure at each crank angle

V: cylinder volume at each crank angle

C_P: specific heat in constant pressure of the fuel

C_V: specific heat in constant volume of the fuel

It should be noted that gas properties vary with in-cylinder temperatures. In other words, the value of γ varies with each cycle. Since the variation of specific heat capacity within a certain range of temperature is modest, it is acceptable to assume constant in-cylinder temperature to estimate the HRR. The specific heat capacities of hydrogen and NG are taken at normal temperature and pressure condition (68°F, 14.7 psi).

IV. 3. Indicated mean effective pressure

Indicated mean effective pressure (IMEP) is a measure of the indicated work output per unit swept volume. It is a parameter that is independent of engine's size, cylinder count, and speed. In other words, it is the average pressure obtained in the combustion chamber during the entire working cycle. IMEP is also known as the work performed by the gas on the piston, i.e., the area enclosed by the p-v diagram of an engine. The pressure at each crank angle was collected by the Windrock analyzer for 250 cycles. These values were used to calculate IMEP following the equation below.

$$\mathrm{IMEP} = \frac{\oint \mathrm{P\,dV}}{\mathrm{V_{swept}}} \tag{5}$$

where,

V_{swept}: swept volume of the cylinder

dV: the difference between each cylinder volume at each crank angle

The coefficient of variation (COV) is a variable that displays how the data point disperses around the mean. In this study, this coefficient was used to interpret how much the IMEP changes after each cycle. COV was determined as follows.

$$\mathrm{COV} = \frac{\sigma_{\mathrm{IMEP}}}{\overline{\mathrm{IMEP}}} \tag{6}$$

where,

σ_{IMEP}: standard deviation of IMEP for 250 cycles

$\overline{IMEP}$: average IMEP for 250 cycles

IV. 4. Indicated mean thermal efficiency

Indicated thermal efficiency is determined by the ratio of the engine's indicated power to the power produced by the combustion of its fuel. The indicated thermal efficiency could be computed by the equation below.

$$\eta_{indicated} = \frac{IMEP \cdot V_{swept} \cdot RPM}{\dot{m}_{fuel} \cdot HV} \tag{7}$$

where,

RPM: engine speed

$\dot{m}_{fuel}$: mass flow rate of fuel

HV: heating value per mass unit of the fuel

It should be noted that hydrogen was blended with NG in volume fraction, which is estimated based on volume flowrate. Therefore, the flowrate recorded in this experiment is volume flowrate. However, the flowrate used in Equation 7 is mass flowrate. Proper conversions need to be carried out to convert volume flowrate to mass flowrate by using density. Moreover, the heating value of each substance is different from one another. Hence, it must be recalculated for each hydrogen concentrated blend. In order to do that, the mass fraction of hydrogen and NG in the blends must be known. Assuming all the gases used are ideal gases, volume fraction is equal to mole fraction. Converting mole fraction to mass fraction, and the heating value per mass unit of the blends can be determined using their mass fractions. Since heating value is dependent on temperature, to make the computation less intricate, the heating values of hydrogen and NG are assumed to be constant at normal temperature and pressure condition (68°F, 14.7 psi).

Lastly, to determine the range of tolerance, the uncertainty of the results is calculated. The methods outlined by Ganji and Wheeler are used for the uncertainty calculation and propagation process [62] (Table 7, Appendix).

Chapter 6: Results and Discussion

This thesis focuses on analyzing two aspects which are emission and performance. The data for hydrogen fractions from 0% to 40% were recorded for engine loads from 0% to 75% and bypass position of 30% and 60%. The data for hydrogen fraction higher than 40% and for loads higher than 75% and bypass higher than 60% were not considered due to safety issues and the constraints of the engine.

I. Engine emissions

Comparing the collected emission data to the values listed on Table 1, it is concluded that in most cases the AJAX-DPC 81 complies well with the NSPS. CO and VOCs emissions from the engine are significantly lower than the standards. The highest amount of CO and VOCs the engine produced are approximately 350 and 45 ppm respectively, while the standards for those gases are 610 and 80 ppm. The main concern for this engine is its NOx emission. The engine passes the standard for most cases except for 40% load. When operating at 40% load, the engine emitted about 250 ppm which is 30 ppm higher than the standard. Therefore, emission control methods must be introduced to the AJAX-DPC 81 to maintain its compliance.

I. 1. Nitrogen oxides

In every testing condition, the engine produces extremely low NOx emissions, below 250 ppm. This is certainly due to the nature lean-burn operation of the AJAX.

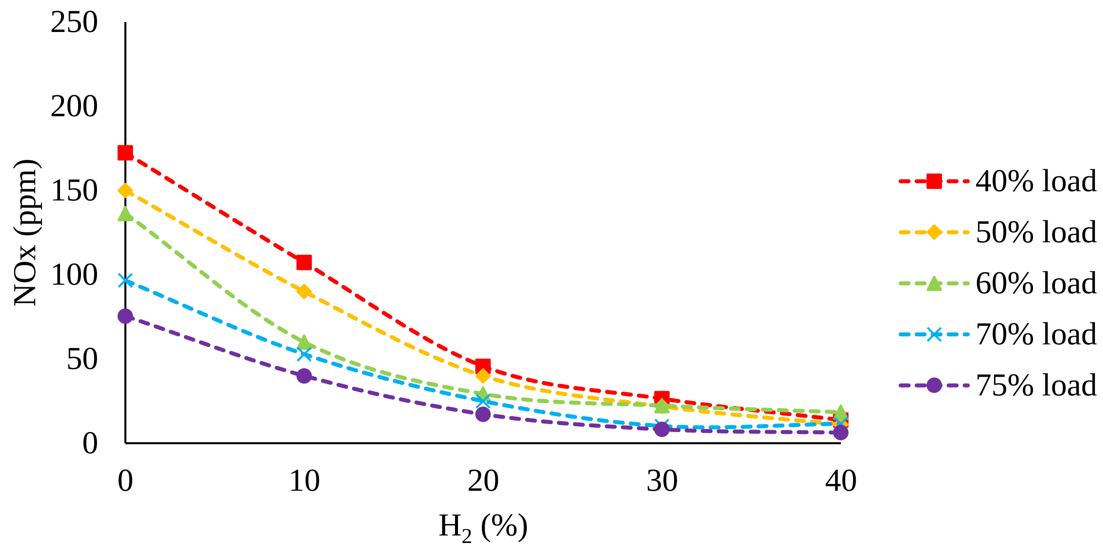


Figure 11. NOx emission before catalyst at rising H_2 fractions.

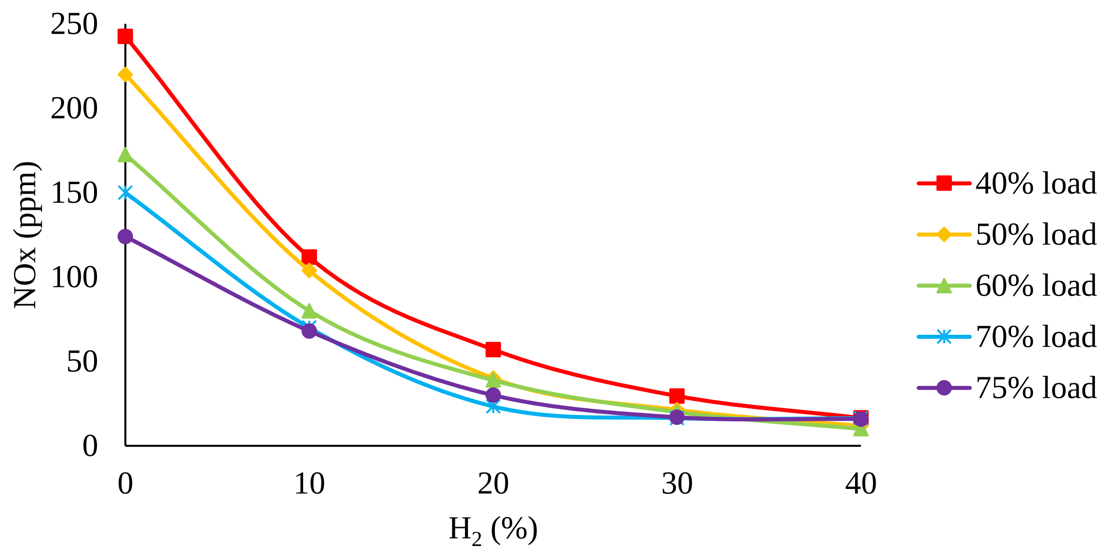


Figure 12. NOx emission after catalyst at rising H_2 fractions.

Figures 11 and 12 show that NO_x decreases as hydrogen fraction increases in the blend fuel, most drastically at 40% load. For all cases, NOx emission reduces to approximately 25 ppm at 40% hydrogen content. The level of NOx also seems to drop at higher engine loads. It can be noticed that the emission of NOx is higher after catalyst compared to before catalyst, approximately 50 ppm at each condition. As shown in Table 4, the oxidation catalyst used in this experiment should not affect NOx emissions. However, it can be inferred from the figures that more NOx is generated during the catalysis process. One possible explanation is that since the engine operates at lean-burn condition, there is an abundant amount of excess air in the products; consequently, under high exhaust temperature, oxygen and nitrogen in the left-over air continue reacting as the exhaust gas passing through the catalyst raising NOx emission level.

It should be noted that the decrease in NOx observed in this experiment is infrequent since hydrogen is supposed to increase the combustion temperature and thus increase the NOx emission [10, 11, 44, 63]. This possibly means that adding hydrogen makes the combustion more fuel-lean.

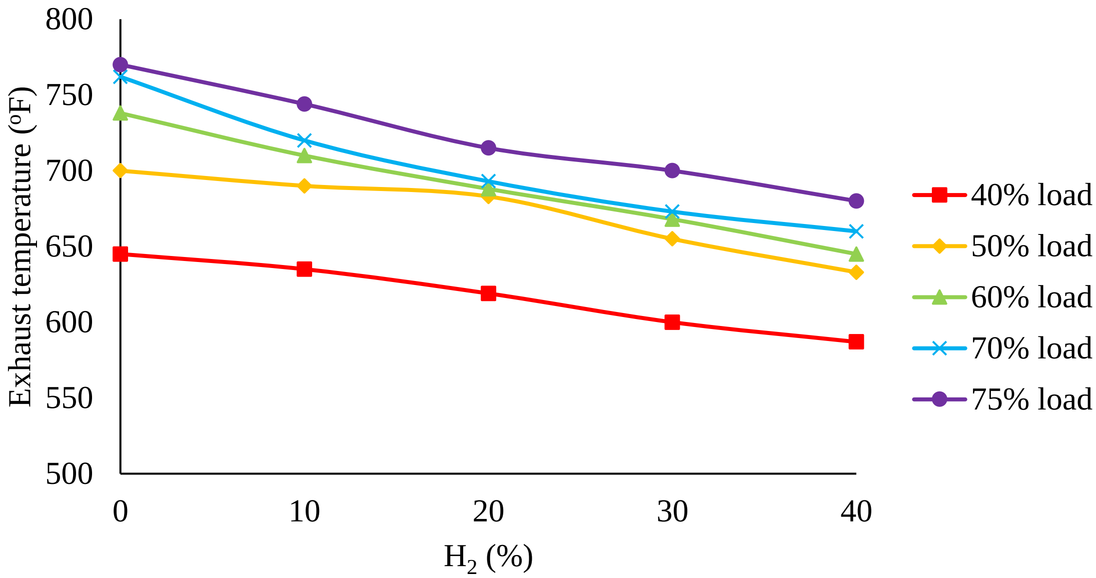


Figure 13. Impact of H_2 increment on the exhaust temperature.

It is noticed that after adding hydrogen, the exhaust temperature steadily drops, Figure 13, which indicates the temperature inside the combustion chamber is lowered as well. The emitted NOx, which is called thermal NOx, is not generated directly from the combustion of hydrogen but rather from air itself. Nitrogen and oxygen in air react to one another under the heat provided by the combustion of air and fuel. Thus, higher temperature promotes this process and creates more NOx. However, hydrogen has high adiabatic flame temperature, which is around 500°F higher than that of NG [64]. This means addition of hydrogen is supposed to increase the combustion temperature and consequentially create more NOx. But the findings of the present study do not align with the above interpretation. There is one possible explanation for the reduction of NOx, that is ultra-lean combustion leads to lower flame temperature and so lower NOx emission [65, 66]. Besides, Zhen reported that the ignition delay increased with the addition of hydrogen [67]. They found that when the engine load was 10%, the extra air ratio fell as the quantity of hydrogen increased and longer ignition delays may result from insufficient air in the mixture, which might have caused lower NO_x emission [67].

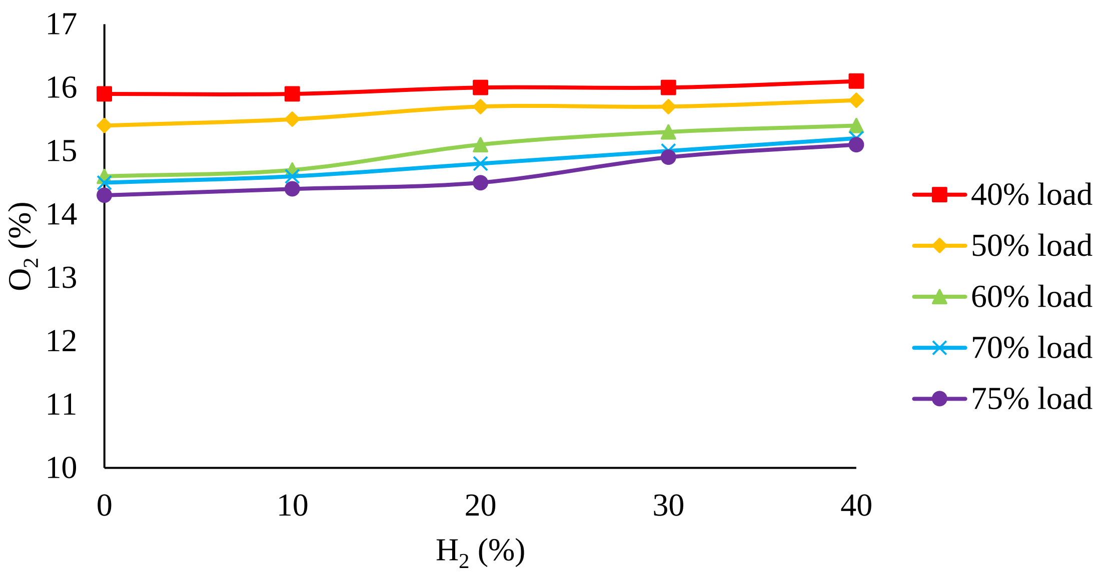


Figure 14. Variation of O_2 content in exhaust gas with increasing H_2 fractions.

From the figure above, it can be seen that as hydrogen content increases, oxygen content in the exhaust increases as well. This indicates that less air is burned inside the combustion chamber. Oxygen content can also be used to determine if a combustion is lean or rich. According to Pai, the oxygen volume fraction in exhaust of a lean-burn engine is usually higher than 6% while that of a rich-burn engine only ranges from 0.5% to 0.6% [68]. As mentioned earlier, hydrogen is highly diffusive so only a small amount of hydrogen is needed to create a combustible mixture [25, 26, 27]. Furthermore, Soltic and Hilfiker highlighted in their study that the addition of hydrogen accelerates combustion and extends lean operation limit [69]. Figure 14 shows that even at 0% hydrogen, the oxygen content is already high, and it increases with increasing hydrogen. Since there is too much air and too little fuel to heat it up with, the combustion temperature drops, Figure 13. Moreover, the equivalent ratios of the reactions are extremely low which further supports the explanation above (Table 6, Appendix). Combining all the stated reasons, it can be concluded that the phenomenon that reduces NOx emission is ultra-lean combustion.

While almost all literature shows that injecting hydrogen causes the emissions to reduce except NOx, this result proves otherwise [10, 11, 44, 63]. However, there is a noticeable number of researchers that obtained increasing NOx results [41, 42, 43]. Although the results are not predominant and contradict many previous studies, the logic behind them is coherent and convincing. Since the NOx concern seems to be solved, especially in the case of ultra-lean burn, this could be a good sign heralding the hydrogen era of two-stroke engines.

I. 2. Carbon oxides

Generally, the emissions of CO and CO_2 decrease with increasing hydrogen content. The post-catalyst level of CO is significantly reduced due to being converted to CO_2.

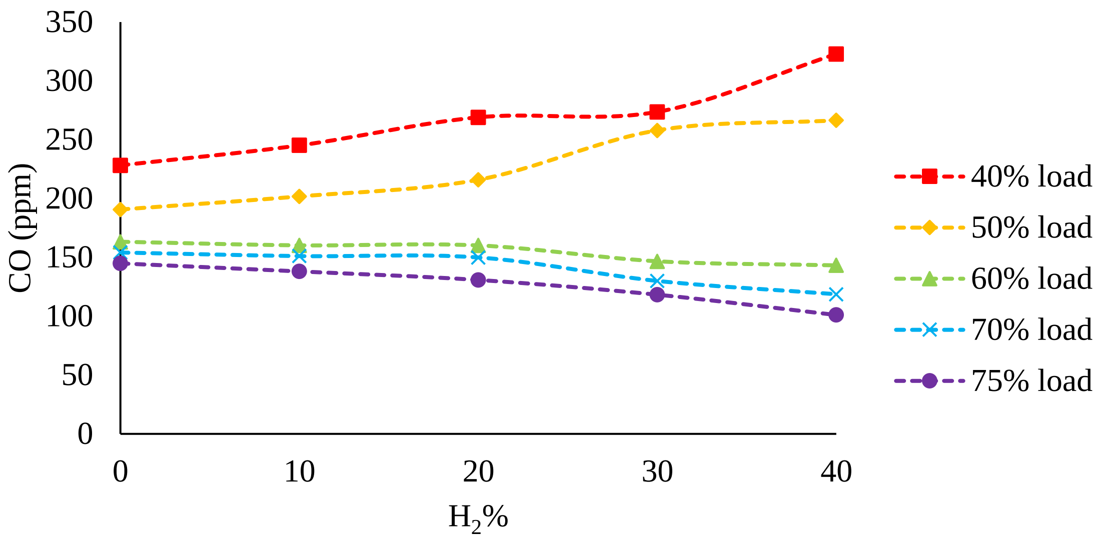


Figure 15. CO emissions before catalyst at rising H_2 fractions

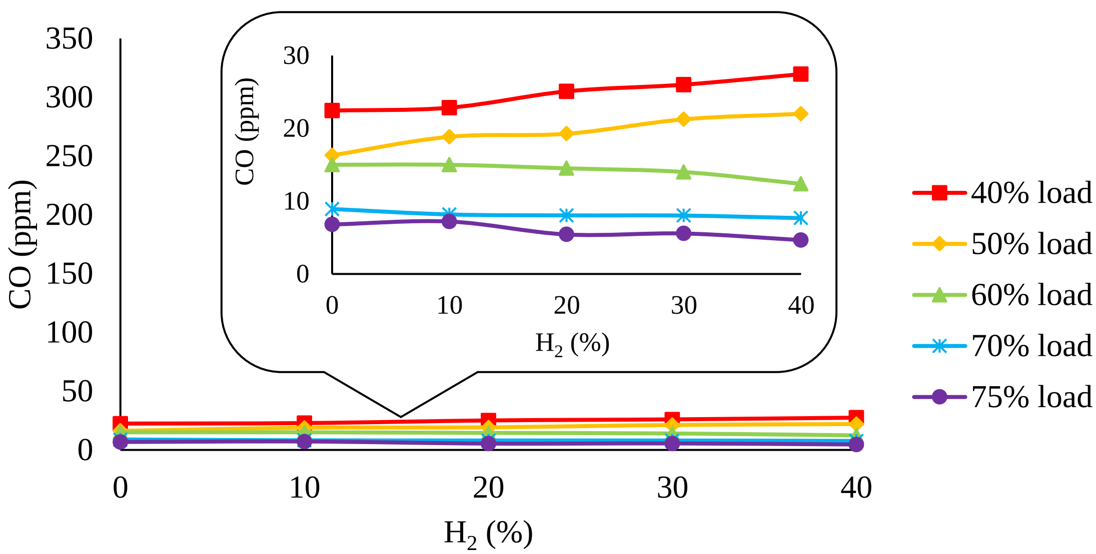


Figure 16. CO emissions after catalyst at rising H_2 fractions.

Insufficient combustion of fuel is the main factor causing CO emission. Overall, CO emission decreases as hydrogen fraction increases. Adding hydrogen into NG means decreasing hydrocarbon concentrations, thus reducing the amounts of carbon-based products. In general, the level of CO emission decreases as engine load increases, i.e., engine speed increases. However, Figure 15 illustrates an infrequent trend of CO emissions, that is, with the increase of H_2%, CO concentration rises at lower engine loads, 40% and 50% loads, and it drops at the higher loads, 60% to 75% loads. This phenomenon depicts a relationship between CO emission and engine load for HNG blend.

It is suggested in the previous section that the engine is operated under lean-burn conditions, which means that there is always more air than fuel in the cylinder. However, the engine is not able to burn all the fuel with that amount of excess air causing incomplete combustion whose evidence is CO formation. This could be caused by poor mixing or short retaining time in the cylinder. When the air-fuel ratio is near to stoichiometric requirement, CO emission increases in parallel to hydrogen addition unless hydrogen is injected under lean operation conditions, i.e., high air-fuel ratios [14]. This reason explains what happens at higher loads, yet it does not seem to work for lower ones. It should be noted that the formation of CO is highly dependent on combustion temperature. Figure 13 shows that exhaust temperature increases with higher loads and adding hydrogen to the blend made the temperature go down. This means that at 40% and 50% loads the combustion temperature of the blend is already barely enough for the oxidation of CO into CO_2, the addition of hydrogen causes the temperature to be even lower to a point that it is insufficient to promote the oxidation process. Furthermore, emission of two-stroke engine is influenced by two other factors which are scavenging pressure and EGR. The quantity of internal EGR is reduced when the scavenging pressure increases and vice versa [70]. In the present study, there are high possibilities of having low scavenging pressure at lower load and consequently internal EGR increases and causes insufficient combustion of the mixed gas, resulting higher CO emission.

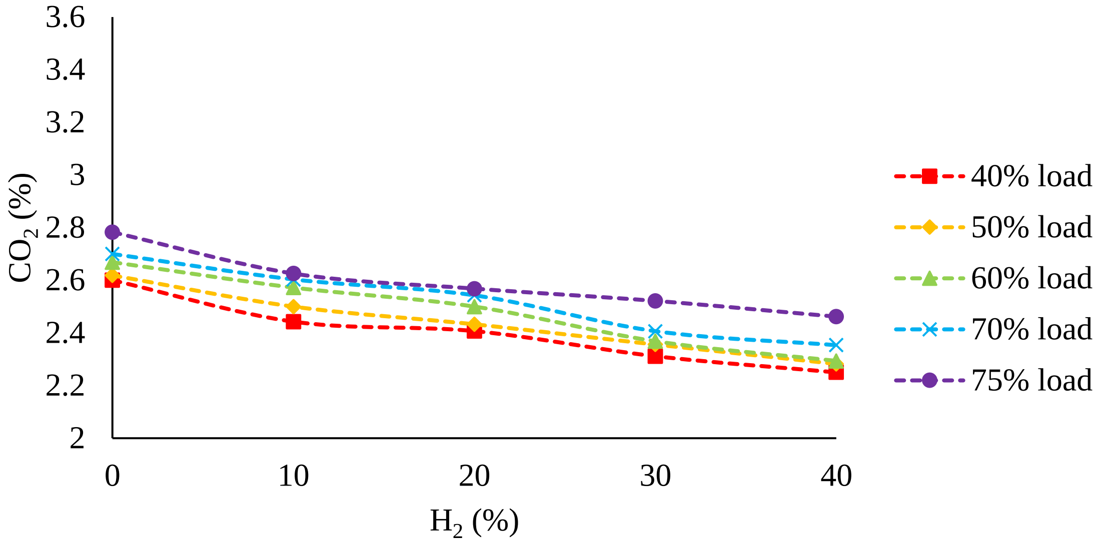


Figure 17. CO_2 emissions before catalyst at rising H_2 fractions.

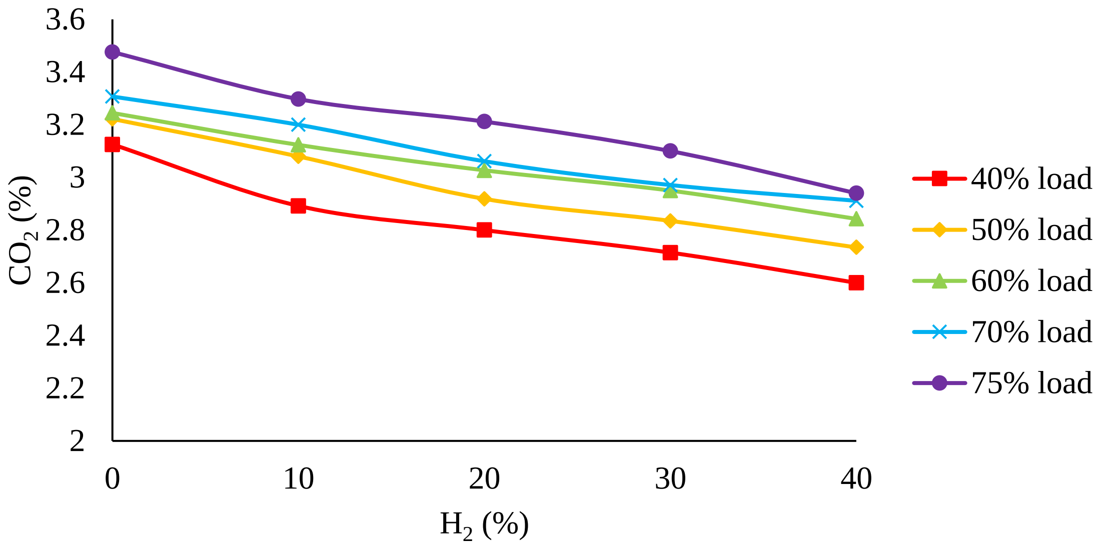


Figure 18. CO_2 emissions after catalyst at rising H_2 fractions.

The CO_2 emission trend shown in Figures 17 and 18 are consistent over different loads, it decreases as H_2 fraction increases. CO_2 emission drops about 0.3% at 40% hydrogen. The cause of this phenomenon is the same as that of the case of CO emission. The hydrocarbon fraction in the fuel decreases as the hydrogen fraction increases causing carbon-based products to decrease as well. The level of CO_2 before the catalyst is lower than that after the catalyst. This result shows that more CO_2 is generated during the catalysis process. Combining this difference with the drop in CO shown in Figures 15, 16, 17, and 18, it can be easily interpreted that most of the amount of CO is converted into CO_2 by the catalyst. Figures 17 and 18 clearly display the relationship between loads and CO_2 emission. As the load increases, CO_2 emission increases as well. On the other hand, CO emission decreases at higher load. This indication further supports the logic that more complete combustions occurs at higher loads generating more CO_2 instead of CO.

It is obvious from the figures that CO_2 emission dropped drastically, and this result is entirely expected since the catalyst is specifically designed for CO-CO_2 conversion. The difference between the level of CO pre- and post-catalyst is significant. CO emission is reduced from approximately 200 ppm to 25 ppm after passing through the catalyst. CO_2 emission rises about 0.6% after the catalyst.

A simple stoichiometry analysis was carried out to verify the experimental results. Theoretically, stoichiometric condition is where all the reactants react to one another completely to generate products. In this condition, the reaction temperature is the highest. Assuming the fuel and air combusts stoichiometrically at every testing condition, the following chemical equations can be obtained.

0/100 HNG: $$\mathrm{CH_4 + 2\ (O_2 + 3.76\ N_2) \longrightarrow CO_2 + 2\ H_2O + 7.5\ N_2}$$

10/90 HNG: $$\mathrm{(9\ CH_4 + H_2) + 18.5\ (O_2 + 3.76\ N_2) \longrightarrow 9\ CO_2 + 19\ H_2O + 69.7\ N_2}$$

20/80 HNG: $$\mathrm{(4\ CH_4 + H_2) + 8.5\ (O_2 + 3.76\ N_2) \longrightarrow 4\ CO_2 + 9\ H_2O + 32.0\ N_2}$$

30/70 HNG: $$\mathrm{(2.3CH_4 + H_2) + 5.2\ (O_2 + 3.76\ N_2) \longrightarrow 2.3\ CO_2 + 5.7\ H_2O + 19.4\ N_2}$$

40/60 HNG: $$\mathrm{(1.5\ CH_4 + H_2) + 3.5\ (O_2 + 3.76\ N_2) \longrightarrow 1.5\ CO_2 + 4\ H_2O + 13.2\ N_2}$$

Assuming all the gases are ideal gases, their volume fractions are equal to their mole fractions. Hence, the volumetric concentrations of CO_2 for each condition from 0% to 40% hydrogen fractions are 9.5%, 9.2%, 8.9%, 8.5%, and 8% respectively. Because of stoichiometric conditions, these values are the highest amount of CO_2 that could be generated. Therefore, they should be higher than the actual values. Comparing these to Figure 17 the actual amounts of CO_2 obtained is only about 33% of the theoretical amounts. Thus, the collected emission data are validated. Furthermore, it can be concluded that a large amount of NG and hydrogen, in reality, does not react and is flushed out of the cylinder at the end of each power cycle.

I. 3. Methane

Methane is a potent greenhouse gas. It is 80 times more potent than carbon dioxide. It is so harmful that it is better to burn methane to CO and CO_2 than releasing it into the air. Methane has been responsible for roughly 30% of global warming since pre-industrial times and is spreading faster than at any other time since records began in the 1980s [71]. However, blends of methane and hydrogen emit the fewest greenhouse gases. This could be explained by the effect of broad flammable limits and quick hydrogen flame propagation, which promote combustion and lower VOCs emissions. Figures 19 and 20 show a significant reduction of CH_4 emission with the increase of H_2 fraction and engine loads, though emission rate is slower with the increase of engine load. This result indicates that more complete combustion of CH_4 occurs with the higher addition of H_2. It might be attributed to the better diffusivity resulting from more H_2 addition and the higher combustion temperature at higher loads. Moreover, CH_4 emission decreases with engine loads. This is also caused by more complete combustion and less knocking at higher loads. Also, when operating at higher loads, the engine is supposed to generate more power, so logically it also consumes more fuel. Besides, literature shows that above 50% reduction of CH_4 emission from two-stroke engine is a substantial attainment. With the increase of hydrogen percentage in methane, combustion can be enhanced to improve thermal efficiency [72, 73]. Similarly, CO_2 and

HCs emissions are reduced when compared to pure methane because the carbon content of the fuel is reduced [44].

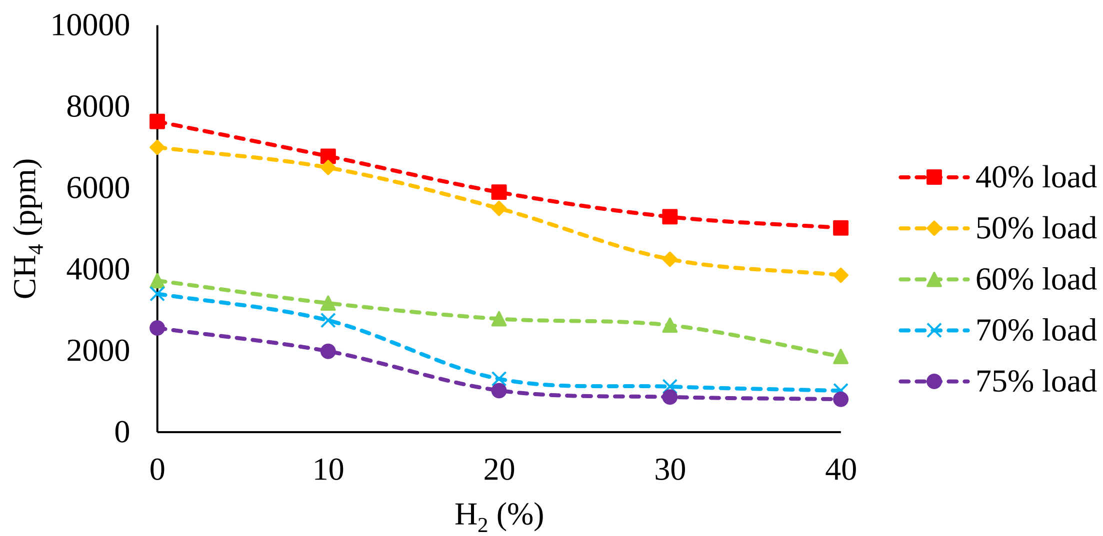


Figure 19. CH_4 emissions before catalyst at rising H_2 fractions.

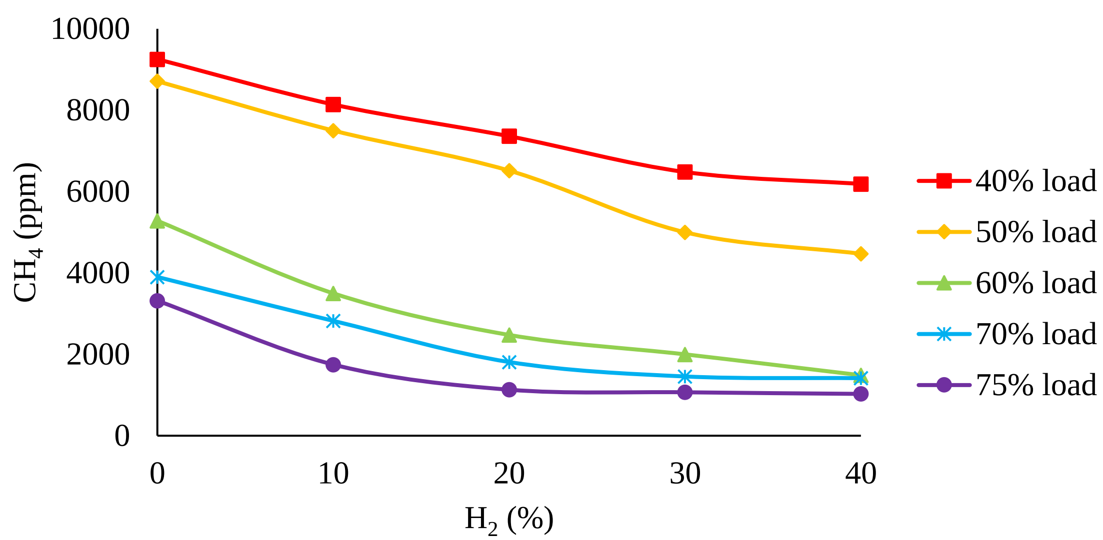


Figure 20. CH_4 emissions after catalyst at rising H_2 fractions.

Comparing Figures 19 and 20, it can be seen that the level of CH_4 increases after the catalyst, approximately 1500 ppm for each condition. The oxidation process of exhaust gas should not produce more methane; therefore, the level of methane is not supposed to change after going through the catalyst. The rise of CH_4 shown in the figures indicates that the high temperature of exhaust gas possibly facilitates the catalytic oxidation products to react with combustion products and produce methane. Additionally, the stoichiometry analysis proves that about 2/3 of the air-fuel mixture does not burn so the excess hydrogen in the fuel could have reacted with CO and CO_2 in the products forming methane. This methanation of COx can be one of the reasons behind this increase. The increase of water concentration in the post-catalyst exhaust gas also supports this reasoning (Figures 43 and 44, Appendix).

$$\mathrm{CO_2} + 4\,\mathrm{H_2} \longrightarrow \mathrm{CH_4} + 2\,\mathrm{H_2O}$$

$$\mathrm{CO} + 3\,\mathrm{H_2} \longrightarrow \mathrm{CH_4} + \mathrm{H_2O}$$

Moreover, it should be noticed that even in the case of 0% H_2 concentration, the post-catalyst concentration of CH_4 also increases. It is suspected that steam methane reforming reaction might have taken place in the combustion chamber to create hydrogen so this product hydrogen can allow the methanation of COx to happen.

$$\mathrm{CH_4} + \mathrm{H_2O} \longrightarrow 3\,\mathrm{H_2} + \mathrm{CO}$$

$$\mathrm{CH_4} + 2\,\mathrm{H_2O} \longrightarrow 4\,\mathrm{H_2} + \mathrm{CO_2}$$

I. ***4. Volatile organic compounds***

There are various routes for the formation of VOCs, such as short circuiting, HCs from wall and bulk quenching, and the combustion or evaporation of lubricating oil. Yet, the major factors are suggested to be quenching and lubricating oil. Furthermore, formaldehyde, one of the VOC gases, is formed as an interstage product of the oxidation of CH_4. The reasons behind and the amount of this toxic gas depend on the design of the engine's combustion system. These are the main contributors to the emission of VOCs.

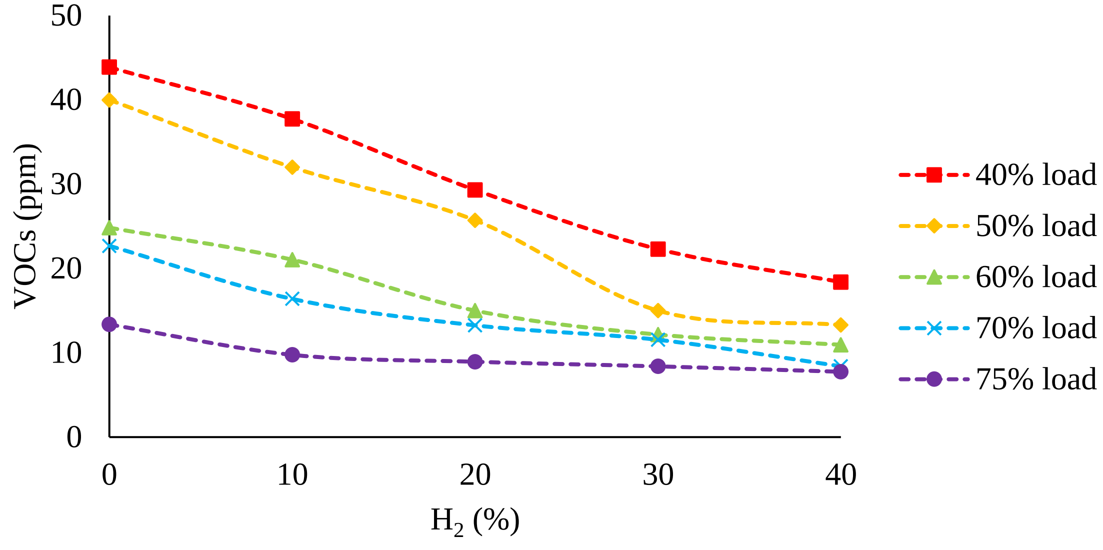


Figure 21. VOC emissions before catalyst at rising H_2 fractions.

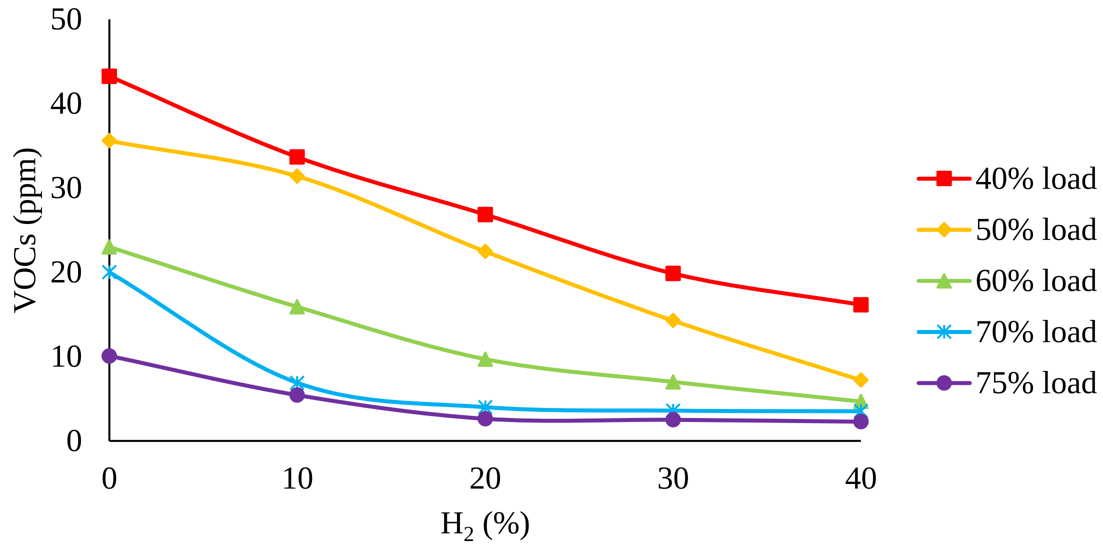


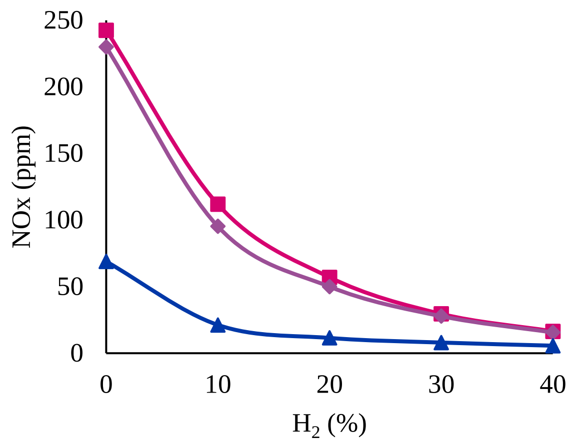
Figure 22. VOCs emissions after catalyst at rising H_2 fractions.

The graphs, Figure 21 and 22, show that the level of VOCs consistently decreases with hydrogen concentration in all cases. At all testing conditions, the engine produces extremely low VOCs, less than 50 ppm. With the addition of hydrogen, the amount of VOCs drops around 50% at 40% H_2. At 40% load, VOCs emission decreases the most, by about 20 ppm. There seems to be a minimal limit of VOCs content since all the curves approach approximately 5 ppm. Moreover, there is a clear difference in emissions between the lower loads, 40% and 50%, and the higher loads, 60% to 75%. Except for NOx, visible gaps in emissions can be observed in every other case. This could indicate that the engine operates better and more efficiently at higher loads.

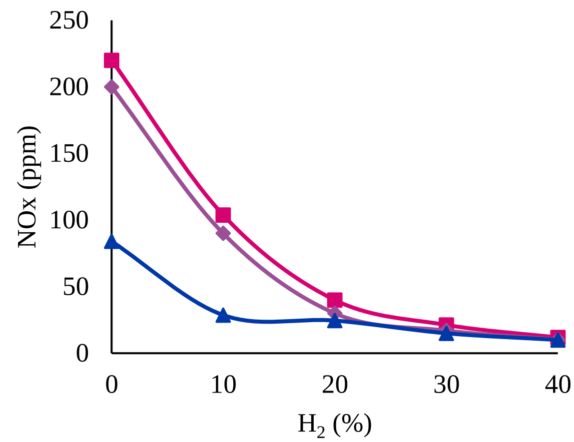
There are noticeable differences between data recorded pre- and post-catalyst. The amount of VOC after catalyst seemed to be lower compared to before catalyst and is reduced by approximately 5 ppm. The catalyst control formaldehyde which is one of the exhaust products included in VOCs. Therefore, the decrease in VOCs after catalyst is certainly the contribution from the catalyst.

I. 5. Integration of air management bypass valve

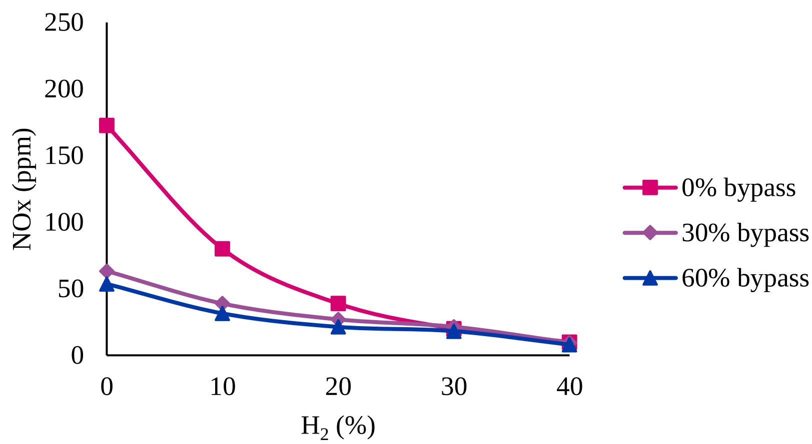
In this section, the effects of hydrogen and air-fuel ratio on emissions are discussed. The bypass valve used in this experiment is an air management system. It alters operating conditions by reducing the amount of air flowing into the combustion chamber. It should be noted that the intake air flowrate stays constant with engine speed and bypass valve does not affect this value. Also, each bypass valve position corresponds to a certain flowrate. Therefore, at each bypass valve position and engine load, the engine has a specific AFR (Table 6, Appendix).



(a)



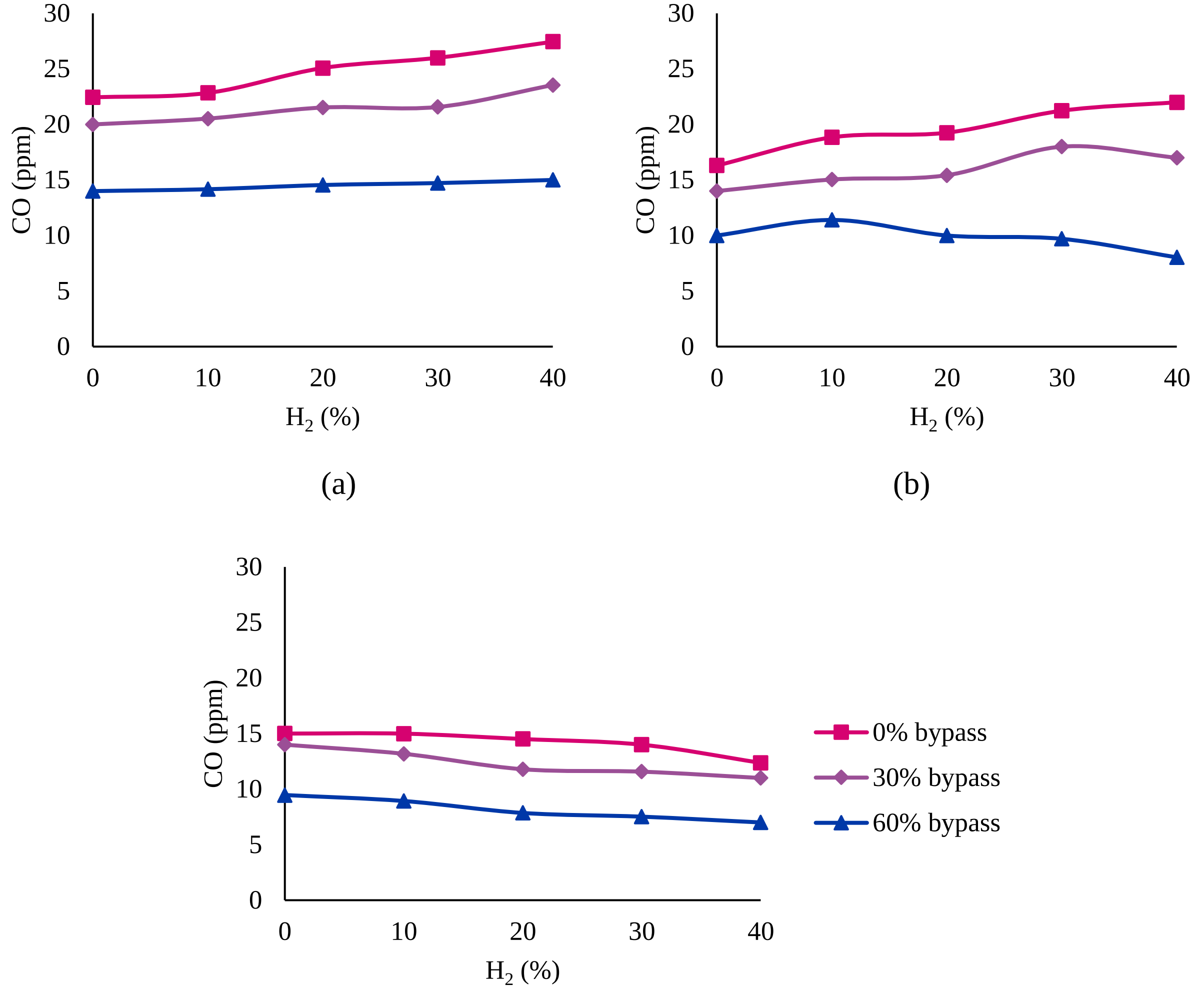
(b)



(c)

Figure 23. NOx emissions after catalyst at rising H_2 volume fractions and various loads. a) 40% load. b) 50% load. c) 60% load.

Figure 23 shows that NOx emission decreases as bypass valve position increases. In most cases, when operating at 60% bypass and 0% H_2, the engine produces the amount of NOx that is just below that of the 0% bypass and 10% H_2 case. Moreover, the wider the bypass opens, the higher the exhaust temperature gets. Despite the increase in temperature, NOx still decreases. This is because of the reduction in air at high bypass position. When the bypass opens, a large amount of air bypasses the cylinder leading to less excess air. Thus, there is less oxidation reaction between O_2 and N_2 occurs which reduces the level of NOx.



(c)

Figure 24. CO emissions after catalyst at rising H_2 fractions and various loads.
a) 40% load. b) 50% load. c) 60% load.

As displayed in Figure 24, CO emission decreases with increasing bypass percentage. At 40% and 50% load conditions, the level of CO increases just as discussed in the previous section. However, at 60% bypass, CO emissions at 50% load condition starts decreasing with increasing H_2% which follows the same trend as other conditions. This can be explained that, as the fuel becomes richer, more complete combustions occur producing CO_2 instead of CO. Moreover, more combustion leads to more heat being generated and raises the exhaust temperature. High temperature also promotes the oxidation process of CO to CO_2.

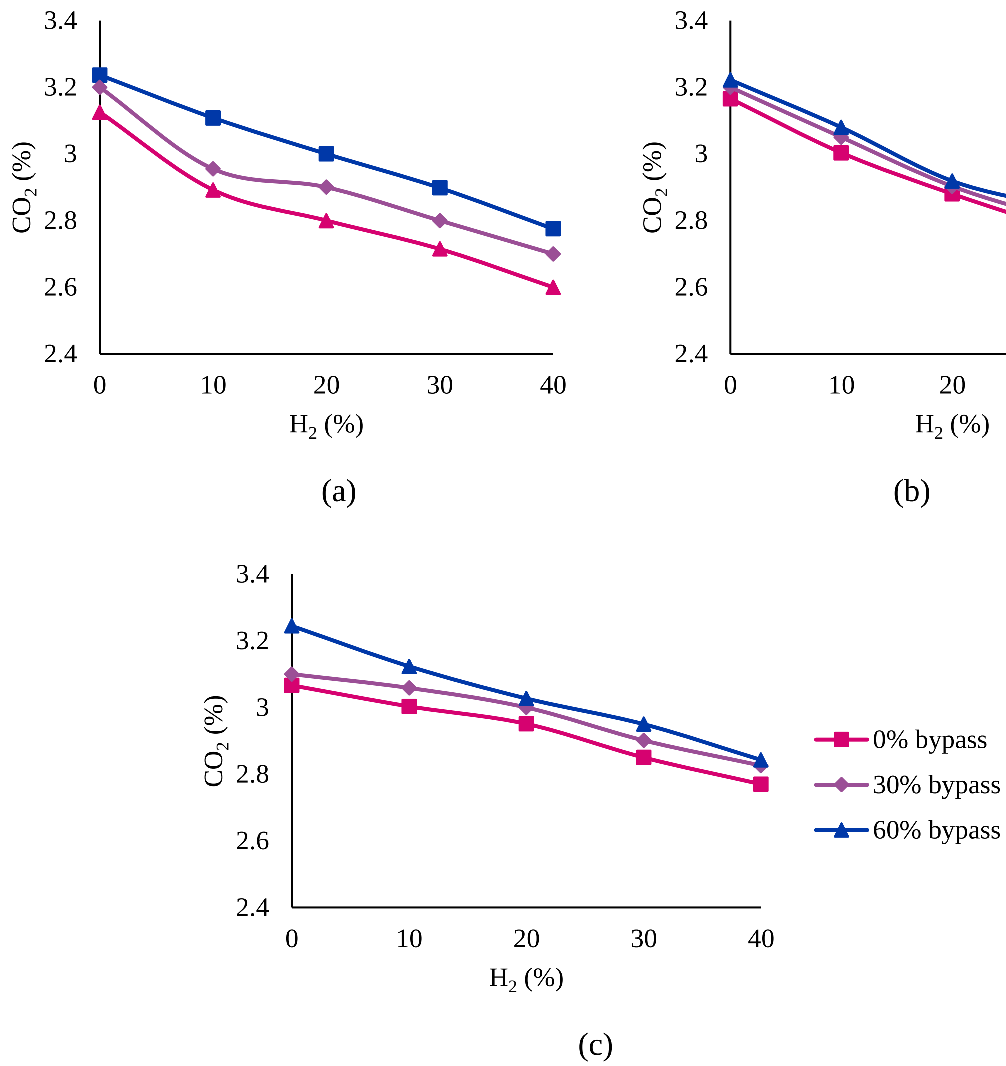


Figure 25. CO_2 emissions after catalyst at rising H_2 fractions and various loads.
a) 40% load. b) 50% load. c) 60% load.

From Figure 25, the concentration of CO_2 in the exhaust gas increases in parallel to bypass valve position. Comparing this trend to the trend of CO, this can be inferred as a sign of complete combustion. The increase of CO_2 also means that there is improvement in combustion performance or fuel utilization. At all cases, CO_2 emission seems to reduce to approximately by 2.8%. 50% load is the condition that has the lowest increase in CO_2.

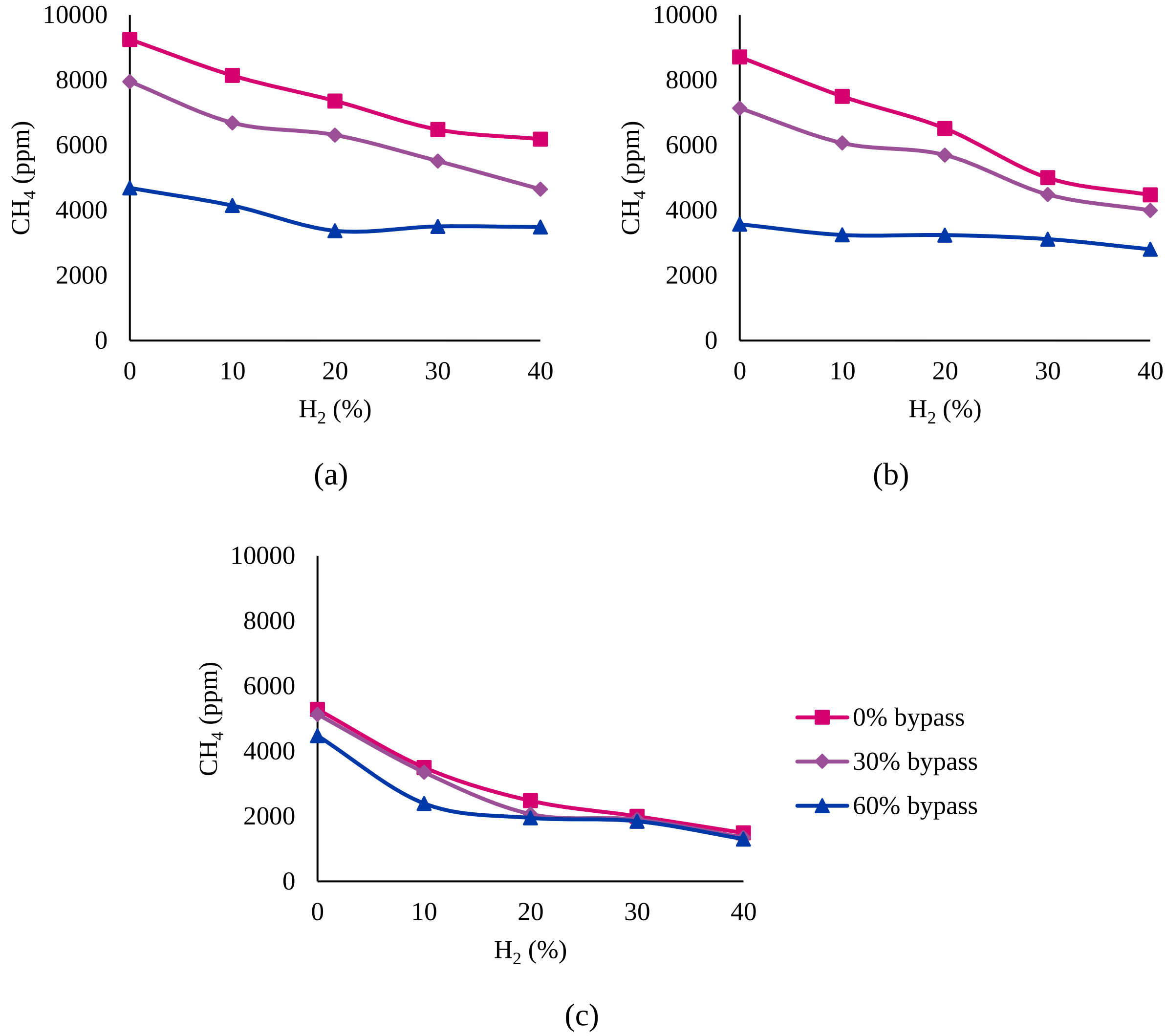


Figure 26. CH_4 emissions after catalyst at rising H_2 fractions and various loads.
a) 40% load. b) 50% load. c) 60% load.

As the bypass percentage gets higher, the air-fuel mixture became richer. Therefore, less heat is lost for heating excess air but rather is utilized to initiate more reactions. The amount of excess methane drops significantly in the cases of 40% and 50% loads which is caused by complete combustion, Figure 26. The trends seen in CO and CO_2 also further support this reasoning. Moreover, the concentration of oxygen in exhaust gas decreases as bypass valve percentage increases, Figure 27. This signifies that more fuel is burned leading to less exhaust air.

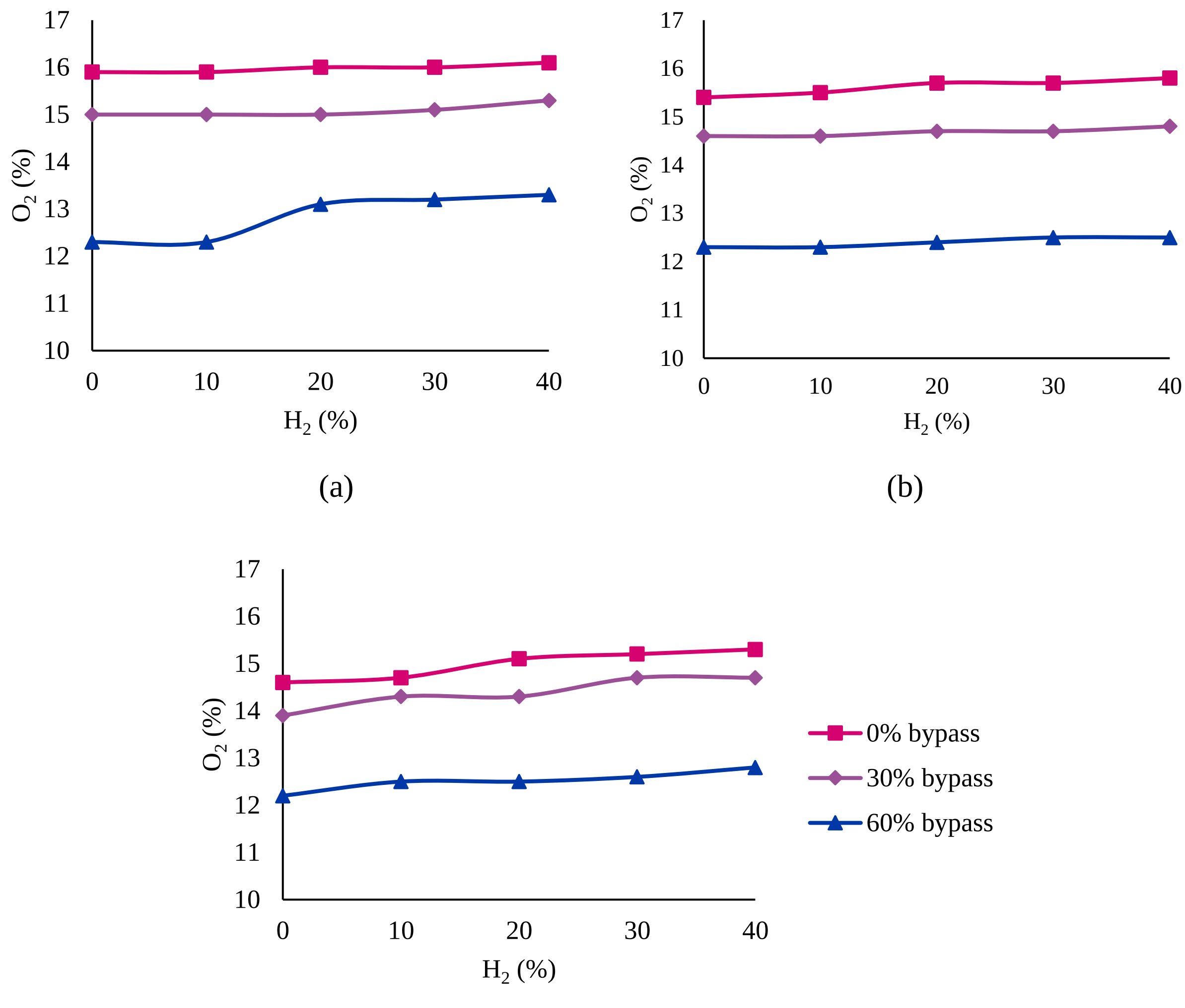


Figure 27. O_2 concentrations in exhaust gas at rising H_2 fractions and various loads.
a) 40% load. b) 50% load. c) 60% load.

Figure 28 proves that the engine produces extremely low VOCs at all conditions. Since the engine works more efficiently and stably when bypass valve is opened, VOCs emission decreases significantly. At 60% bypass, VOCs emission drops the most. As mentioned earlier, 40% and 50% loads seem not to work efficiently in the presence of hydrogen. However, with the integration of bypass valve there are some improvements seen in engine efficiency, especially at 60% bypass. This is proven by the big gaps in CH_4 and VOCs emissions data at 40% and 50% loads. These loads also have the highest reduction in VOCs. Even at 0% H_2, the engine is able to produce as low as 20 ppm VOCs with 60% bypass.

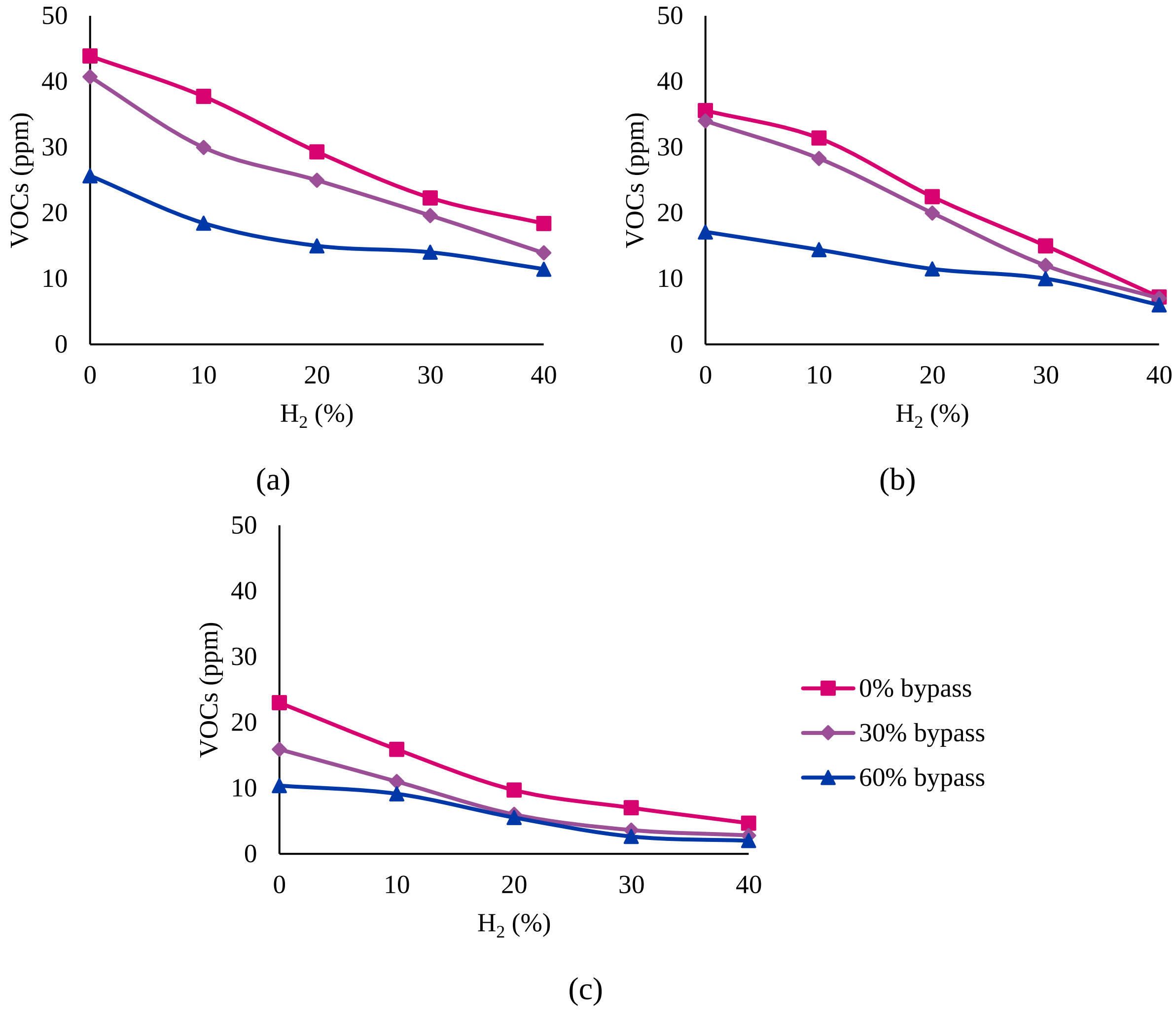


Figure 28. VOCs emissions after catalyst at rising H_2 fractions at various loads.
a) 40% load. b) 50% load. c) 60% load.

II. Engine performance

II. 1. Indicated mean effective pressure

IMEP profile can be used to predict combustion characteristics inside the cylinder. Misfire and knocking cycles are displayed on IMEP dispersion plot as shown below.

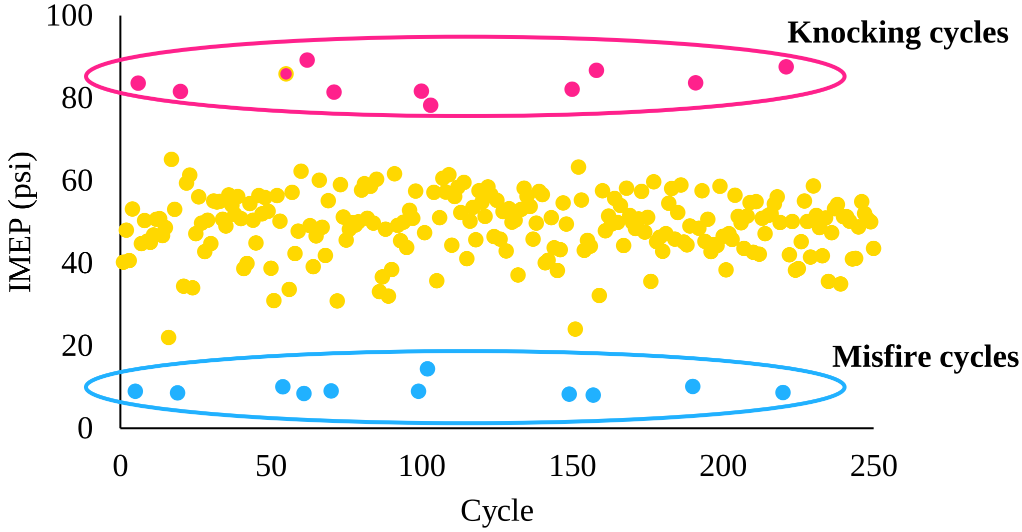


Figure 29. IMEP at 50% load with 0% bypass and 30% H_2.

Misfire is an incomplete combustion phenomenon. When this happens, the cylinder does not produce power. Knocking is when the fuel burns unevenly giving a sudden rise in pressure. Neither of these phenomena is beneficial to the engine. The desirable IMEP profile should have the minimum number of knocking and misfire cycles. The effects of hydrogen addition and bypass valve in stabilizing combustion can be drawn from interpretating IMEP plots. IMEP value changes noticeably with engine load, hydrogen concentration, and bypass valve position as shown in Figure 30 below.

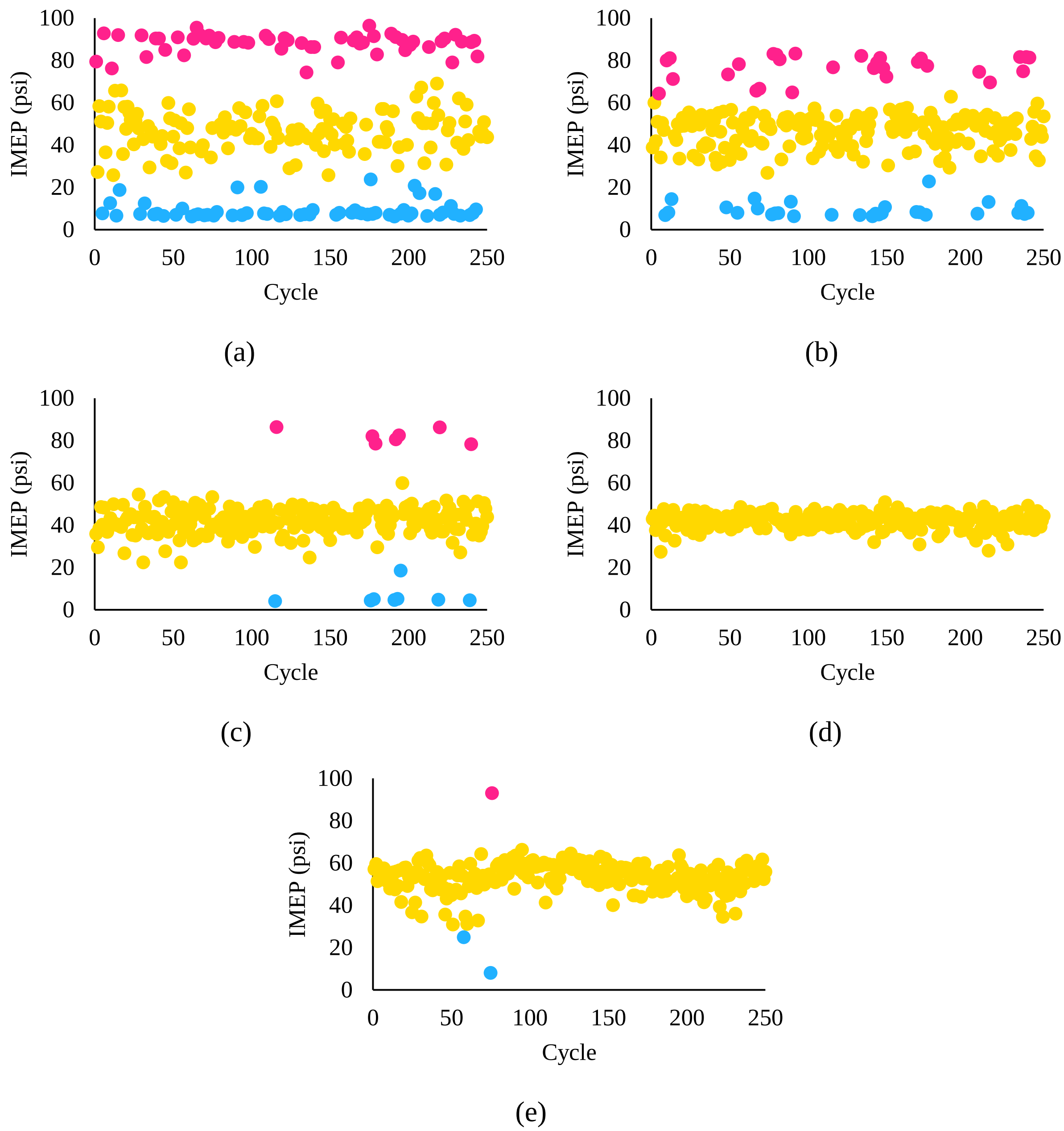


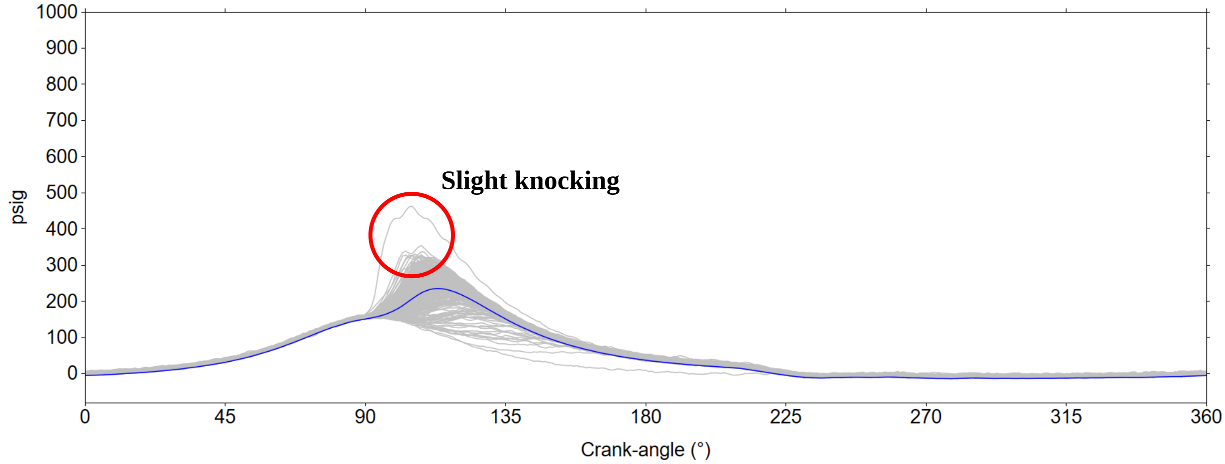
Figure 30. Engine IMEP at various testing conditions.
a) 40% load, 0% bypass, 0% H_2. b) 40% load, 0% bypass, 40% H_2.
c) 40% load, 60% bypass, 0% H_2. d) 40% load, 60% bypass, 40% H_2.
e) 60% load, 0% bypass, 0% H_2.

The COV of IMEP decreases drastically over engine load and bypass valve position. The COV of Figures 30 a), b), c), d), and e) are 59.3%, 38.6%, 25.8%, 8.6%, and 14.3% respectively. Comparing Figures 30 a), b), c), d), and e) to each other, it is clear that engine load has the largest impact on IMEP. COV value drops by 33.5% with the contribution of bypass valve. Hydrogen also reduces COV even though it is not as significant as the other two factors. Furthermore, since IMEP implies the average work exerting on the piston, the lower the COV is, the more stable the engine becomes. The engine operates a lot smoother at higher loads or with the involvement of hydrogen or bypass valve. Therefore, significant improvement on combustion stability can be achieved at higher loads, higher bypass positions, and higher hydrogen concentrations.

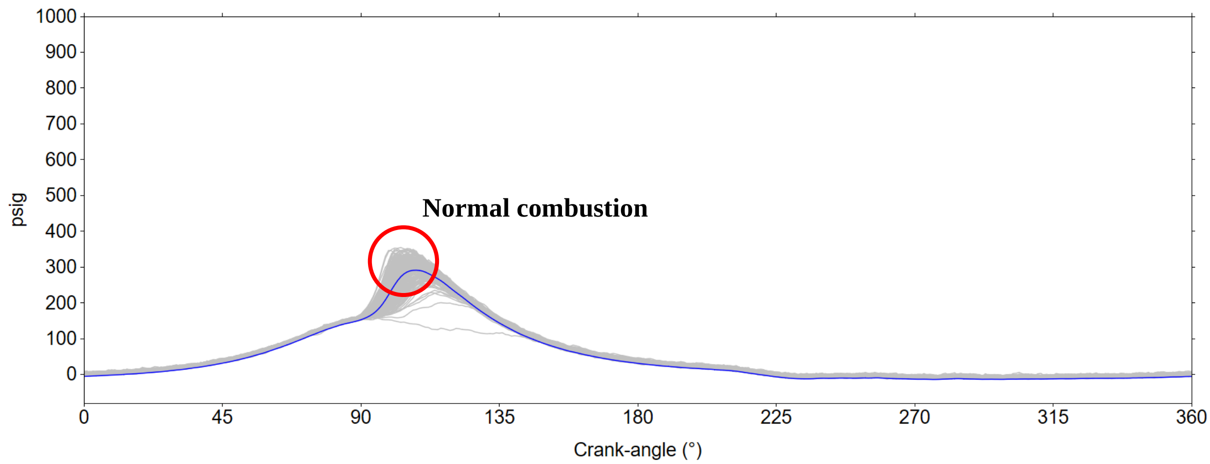
II. 2. In-cylinder pressure

As the hydrogen concentration increases, the in-cylinder pressure becomes more uniform between each cycle, Figure 31. Peak pressure is also reduced with increasing hydrogen fraction. Less knocking occurs in the presence of hydrogen in the fuel. Additionally, the average in-cylinder pressure varies very little with either load, hydrogen fraction, or bypass position.

As displayed in Figure 31, the engine works more stable in the presence of hydrogen. There is significantly less knocking observed, Figure 31 b). In a properly functioning engine, flame propagates from the spark plug creating stable combustion. However, in the case of knocking, the fuel does not burn evenly. When this abnormal combustion happens, pockets of air-fuel mixture ignite outside the envelope of the flame propagation giving a rapid pressure rise. The high pressure difference creates violent vibration leading to loss in engine performance and possible engine damage. In Figure 31 a), multiple peaks can be clearly observed in various cycles. A sudden rise in pressure, almost 200 psi higher than the average, is also spotted.



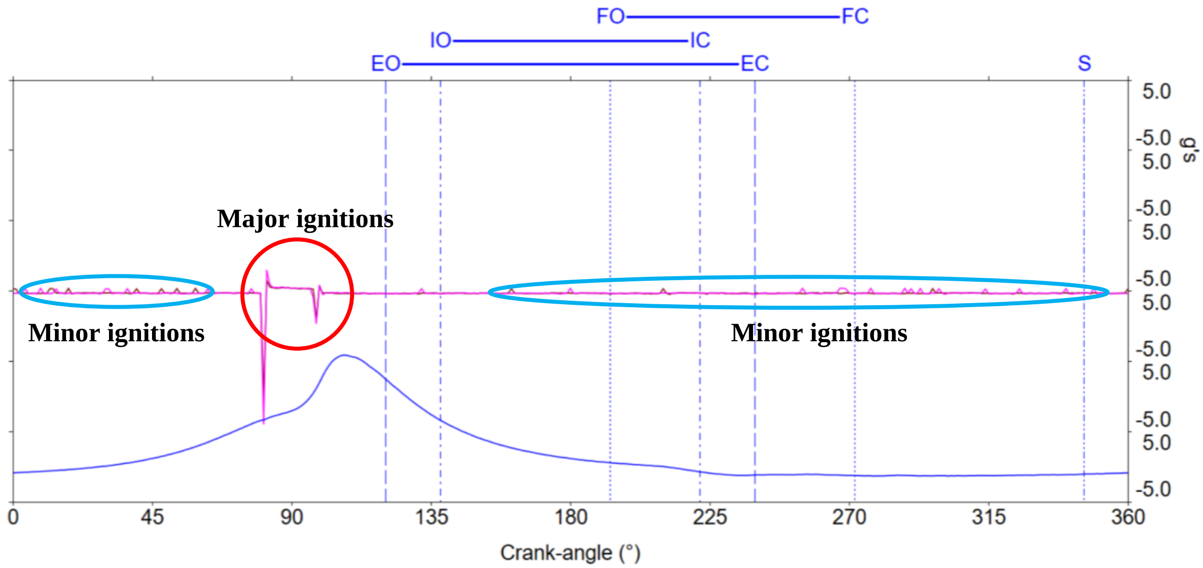
(a)



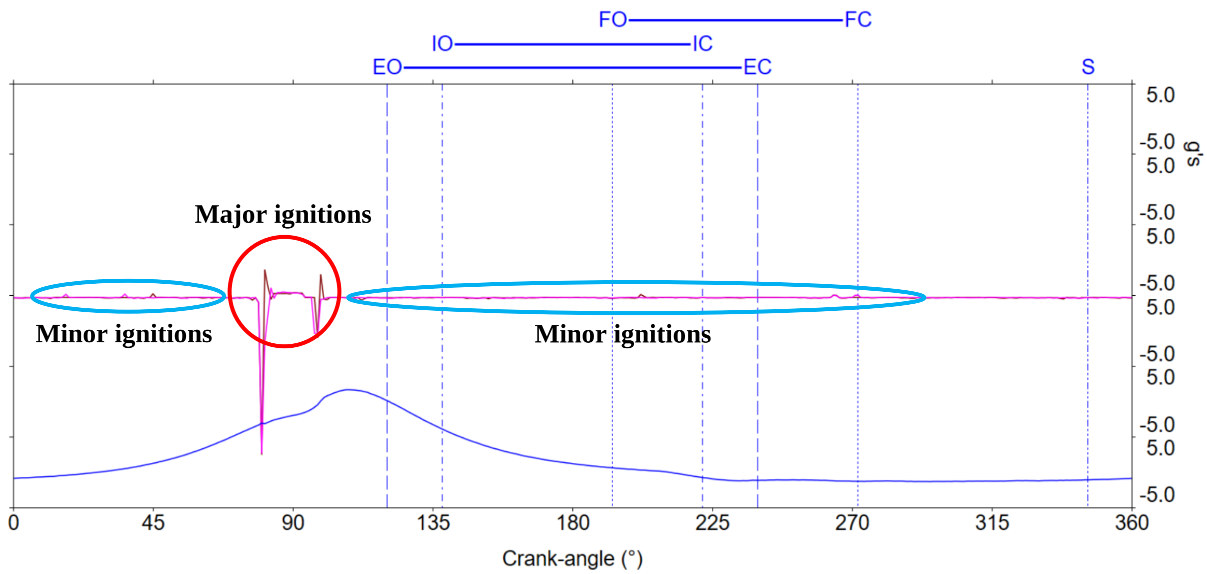
(b)

Figure 31. In-cylinder pressure at various H_2 fractions at 60% load.
a) 0% H_2. b) 40% H_2

There is no change in ignition timing caused by the addition of hydrogen or the integration of bypass valve. The ionization angle stays constant at approximately 81° before TDC as it was pre-set by the manufacturer. Moreover, there seemed to be two major ignitions in each cycle, one primary at 81° before TDC and one secondary at 93° before TDC. Looking closely to Figure 32 a) and b), there are more minor ignitions in the presence of hydrogen. These ignitions can be detonations, i.e., combustion happens after the spark plug fires, and potentially prevent the engine from achieving the highest possible efficiency. This sign indicates that the ignition timing may need to be adjusted to suit the HNG mixture.



(a)



(b)

Figure 32. Major ignitions and in-cylinder pressure at 40% load.
a) 0% H_2. b) 40% H_2

Moreover, ignition timing is subjected to change with many factors such as engine speed, combustion chamber size, or fuel type. Generally, ignition timing needs to be advanced for higher engine speed or larger combustion chamber. If the fuel has high octane number, more ignition timing is also required since the fuel needs more time to burn. Also, if the fuel has high flame speed, ignition timing retarding, i.e., less ignition timing, is needed. Since HNG has higher flame speed and lower octane number compared to NG, the engine's ignition timing may need to be retarded to reduce detonation and achieve higher efficiency.

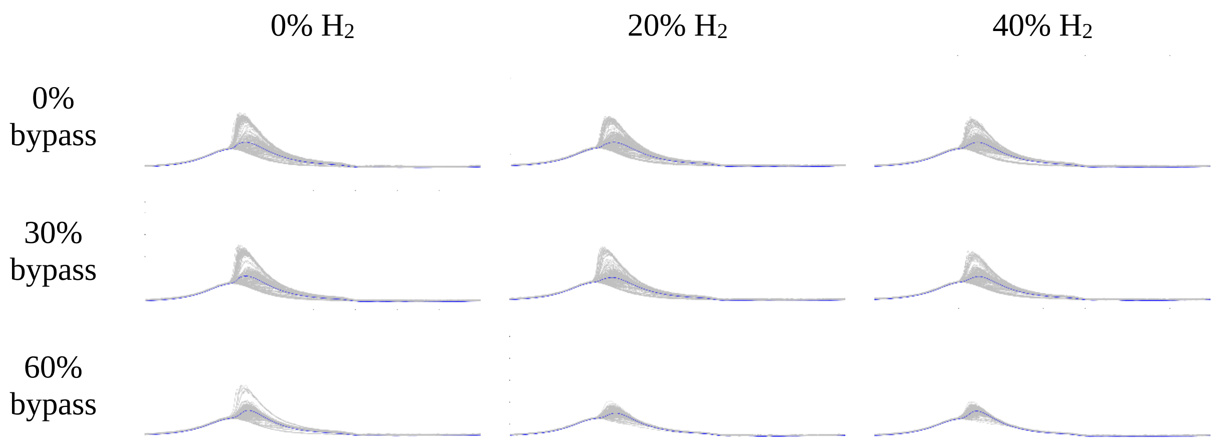


Figure 33. In-cylinder pressure at various H_2 fractions and bypass positions at 40% load.

From Figure 33 it can be confirmed that the average in-cylinder pressure does not seem to vary much with either hydrogen content or AFR. However, peak cyclic in-cylinder pressure dropped drastically with increasing bypass percentage (Figure 47, Appendix). Hydrogen addition also shares similar effects though it was not dominant (Figure 45, Appendix). Standard deviation between each cycle decreases significantly at 60% bypass and 40% hydrogen.

II. 3. Heat release rate

Heating value is the amount of energy obtained when a specific unit quantity of a substance undergoes complete combustion under standard conditions. Therefore, a high heating value is always favored in energy production. The heating value per volume unit of HNG blends decreases linearly with addition of hydrogen, Figure 34. This means that if too much hydrogen is added, the fuel mixture will not be able to supply enough energy to keep the engine running.

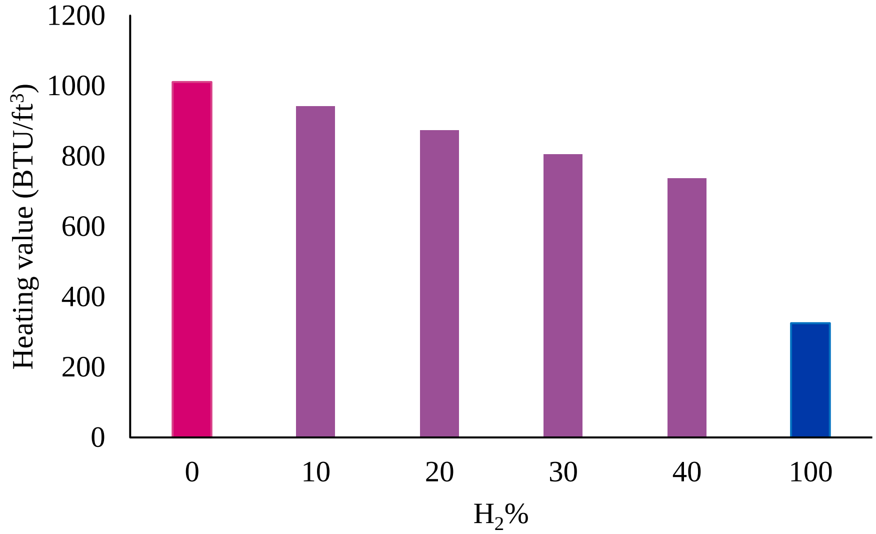


Figure 34. Volumetric heating values of various HNG blends.

It is shown in Figure 35 that HRR reduces slightly with increasing load. Similar results can also be seen on Figure 36, the maximum HRR drops as bypass valve opens wider. It can be inferred from the reasons above that without the presence of hydrogen, HRR decreases with increasing combustion temperature. These logics contribute to the reasons behind the rise in emissions at 40% load with 0% bypass.

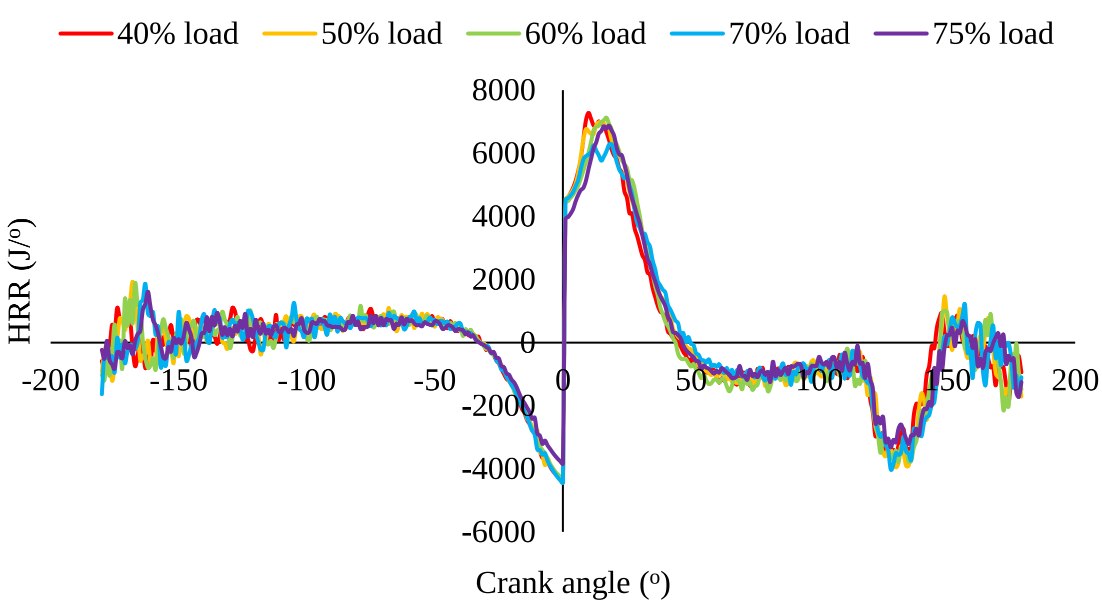


Figure 35. HRR at various engine loads.

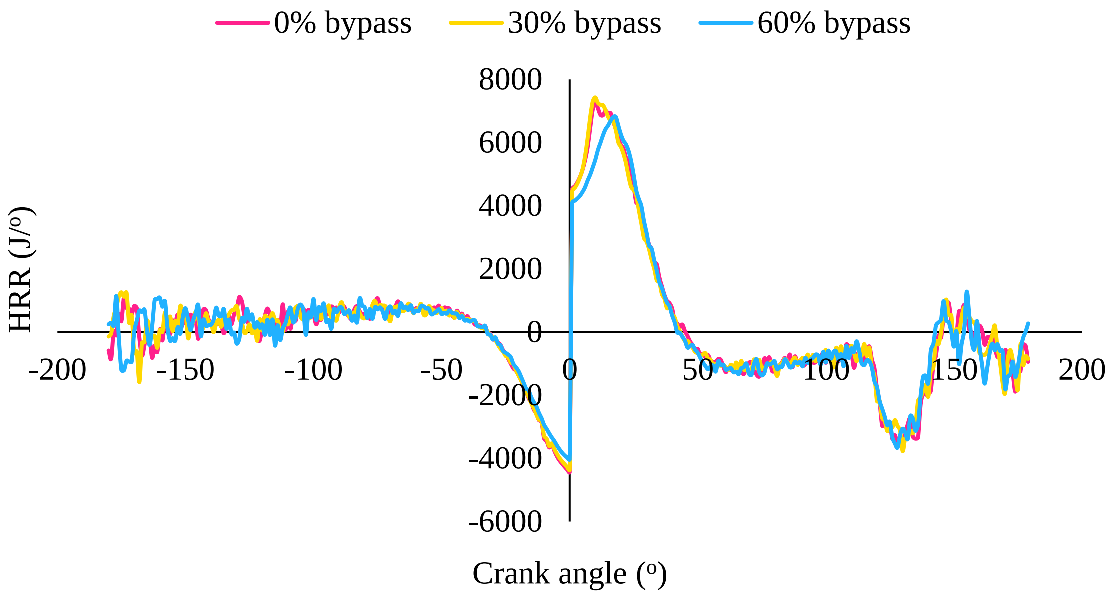


Figure 36. HRR at various bypass valve positions.

On the other side, with the presence of hydrogen, HRR decreases with increasing hydrogen content. Hence, at 100% NG, the engine has the highest HRR. Moreover, HHR provides indication of fire phenomenon. The higher HRR is, the higher amount of exhaust gas and smoke is generated. This partially explains why most emissions, including NOx, are highest at 0% hydrogen. The peak HRR has the most noticeable differences, other regions on the curves vary very little, Figure 37.

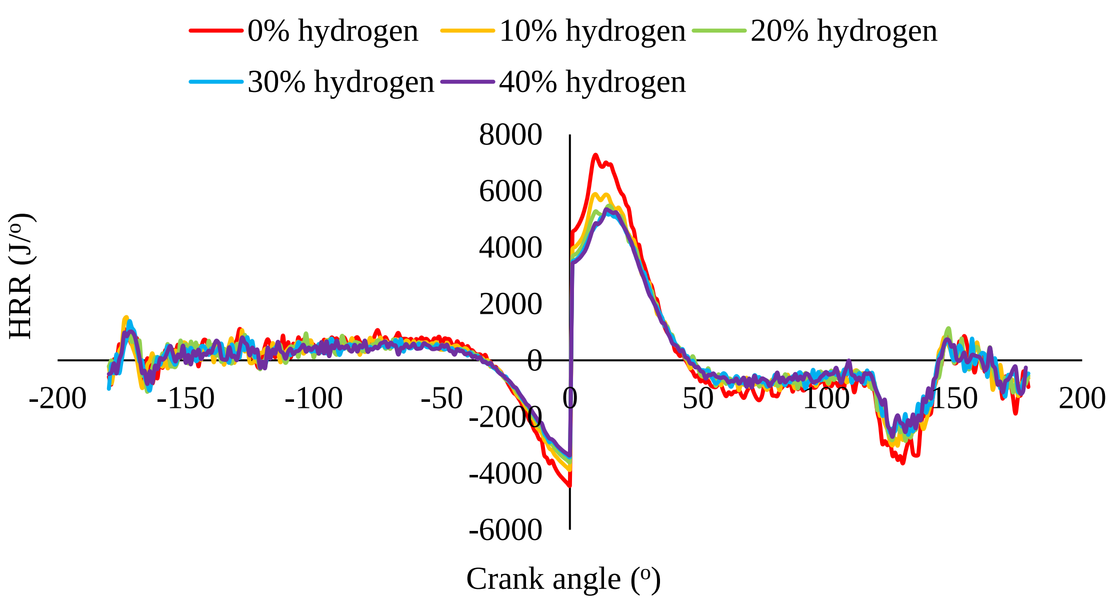


Figure 37. HRR at various H_2 fractions at 40% load.

II. 4. Indicated thermal efficiency

One of the major goals of this work is to find a hydrogen and NG blend that provides a good balance between thermal efficiency and pollutant emissions compared to NG alone. The ITE was calculated, since various energy losses were not considered like heat loss from cylinder walls, friction loss between cylinder and walls, crankshaft bearing, and heat lost in exhaust gas.

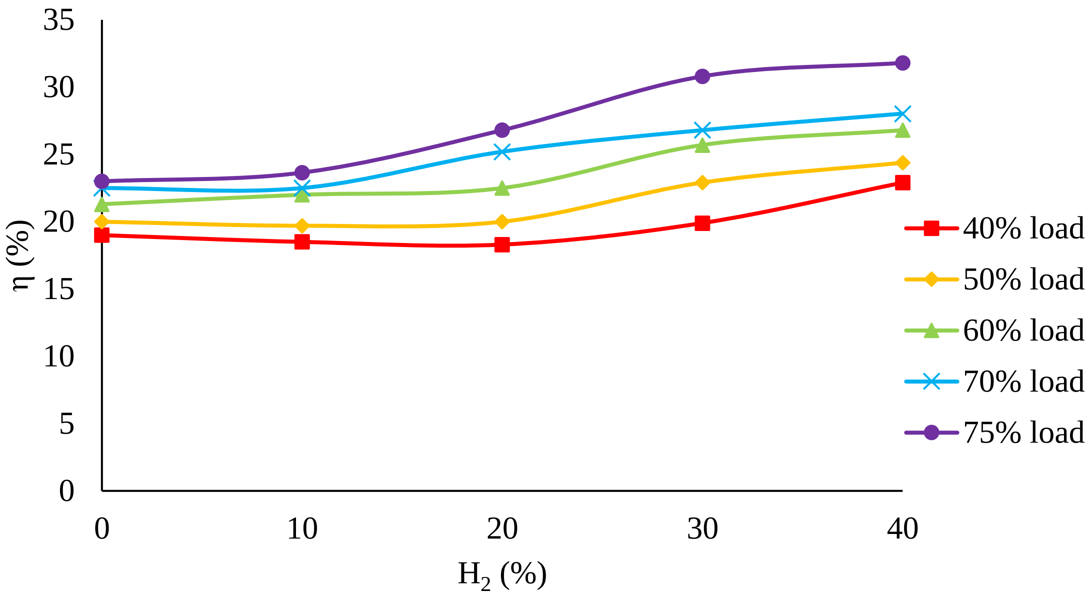


Figure 38. Indicated thermal efficiency at various loads.

The ITE is affected by combustion efficiency. When using leaner equivalence ratios, combustion efficiency is high, as evidenced by the low amount of incomplete combustion products, for example CH_4. On the other hand, combustion efficiency decreases for richer equivalence ratios because there is not enough oxygen to complete combustion [52]. Figure 38 shows that the ITE at various loads improves with hydrogen concentration. The maximum 8.8% increment of efficiency is observed at 75% load with 40% H_2. Looking closely at Figure 38, there are slight decreases happening at 40% and 50% loads up to 20% H_2. This could be the reason behind the increase in CO emissions and the big gaps in data compared to other higher loads. Figure 38 also depicts slight instability in engine performance at 40% and 50% loads. The probable reason is the reduction of equivalence ratio might cause unstable engine performance at lower hydrogen addition [33, 63]. From the efficiency analysis, the engine would work best at 75% load with 40% hydrogen blend. However, it should be noted that higher hydrogen concentration could lead to insufficient energy. More experiments are needed to determine the hydrogen concentration limit for this engine.

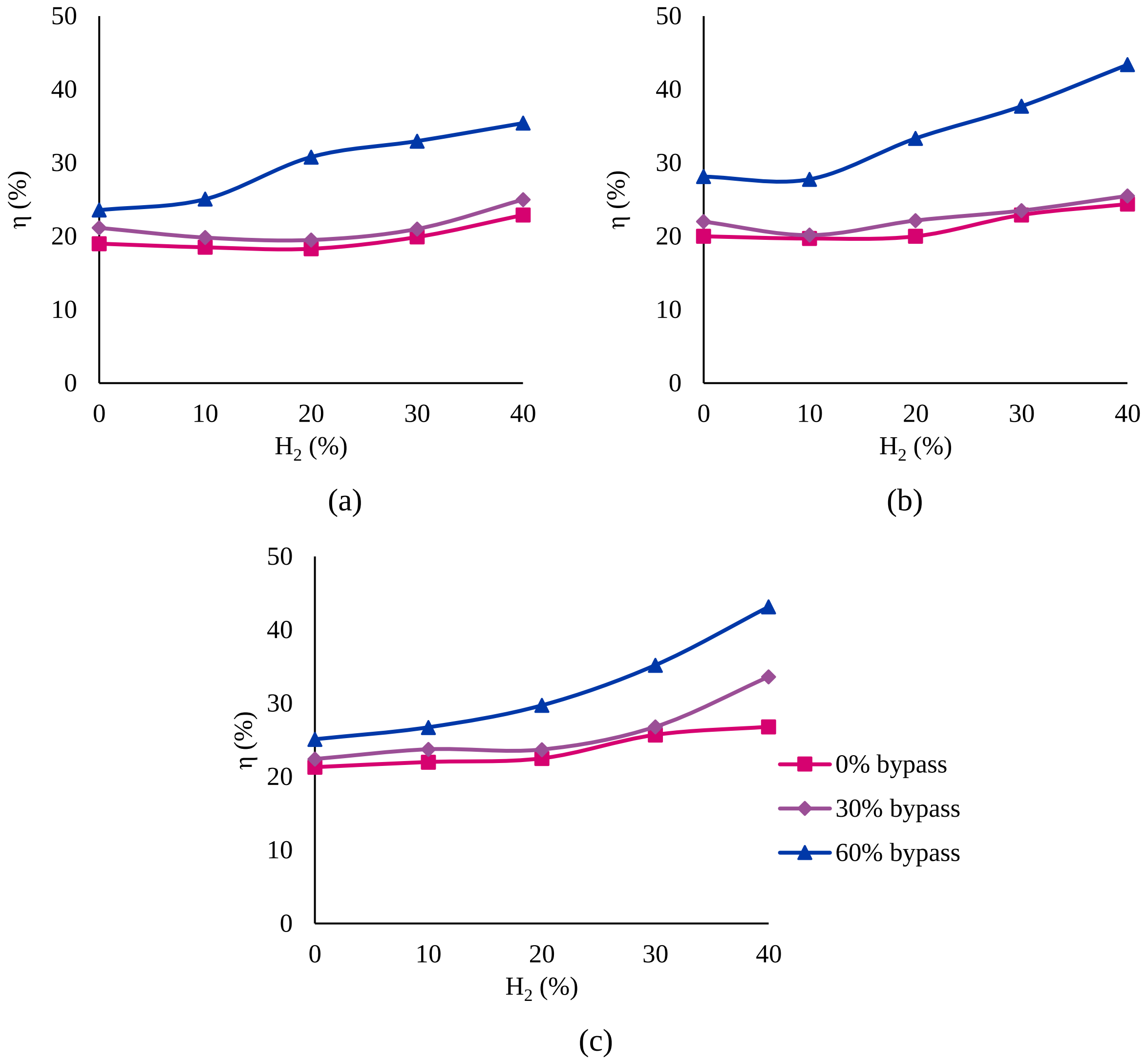


Figure 39. Indicated thermal efficiency at various H_2 fractions and loads.
a) 40% load. b) 50% load. c) 60% load.

Figure 39 shows that the ITE of the engine improves significantly with hydrogen addition and bypass valve implementation. As bypass percentage increases, ITE increases as well. At 40%

and 50% load cases, there are some slight drops in ITE at 10% and 20% H_2. When higher H_2% is injected, the ITE starts rising. The ITEs at 0% and 30% bypass are relatively close to one another for all cases. 60% bypass has the most noticeable improvement, especially at lower loads. Moreover, since the ITE was computed based on the fuel being supplied to engine, a high ITE implies that better fuel utilization happens. Therefore, at lower loads, without the contribution of high hydrogen content and bypass valve, the engine cannot utilize the fuel well. These results coincide with those seen in emissions. The maximum improvement in ITE is seen at 60% loads with 60% bypass and 40% hydrogen. In this case, the ITE increases by 18%. Hence, according to this study, it can be concluded that the optimal operation condition for this engine is 60% loads with 60% bypass and 40% hydrogen combined. However, one should be mindful that having too high bypass percentage would lead to insufficiency of air or high fuel consumption while high hydrogen content results in low volumetric heating value. Moreover, increasing bypass position makes the combustion approach stoichiometric condition which increase combustion temperature. The exhaust temperature threshold of the AJAX DPC-81 is 800°F. An improper combination of high bypass valve position and engine load can easily push the exhaust temperature surpass the temperature constraint and possibly damage the engine. Therefore, considerations must be carefully made to minimize trade-offs and lengthen engine's lifetime.

III. Validation

Hassan also performed similar experiments on the AJAX DPC-81 on emissions and efficiency [74]. Their results can be used to validate the credibility of this study. In their study, Hassan examined the effects of the bypass system on post-catalyst emissions and efficiency of the engine [74]. The emission results of the engine from Hassan's study are shown in Figure 40 below.

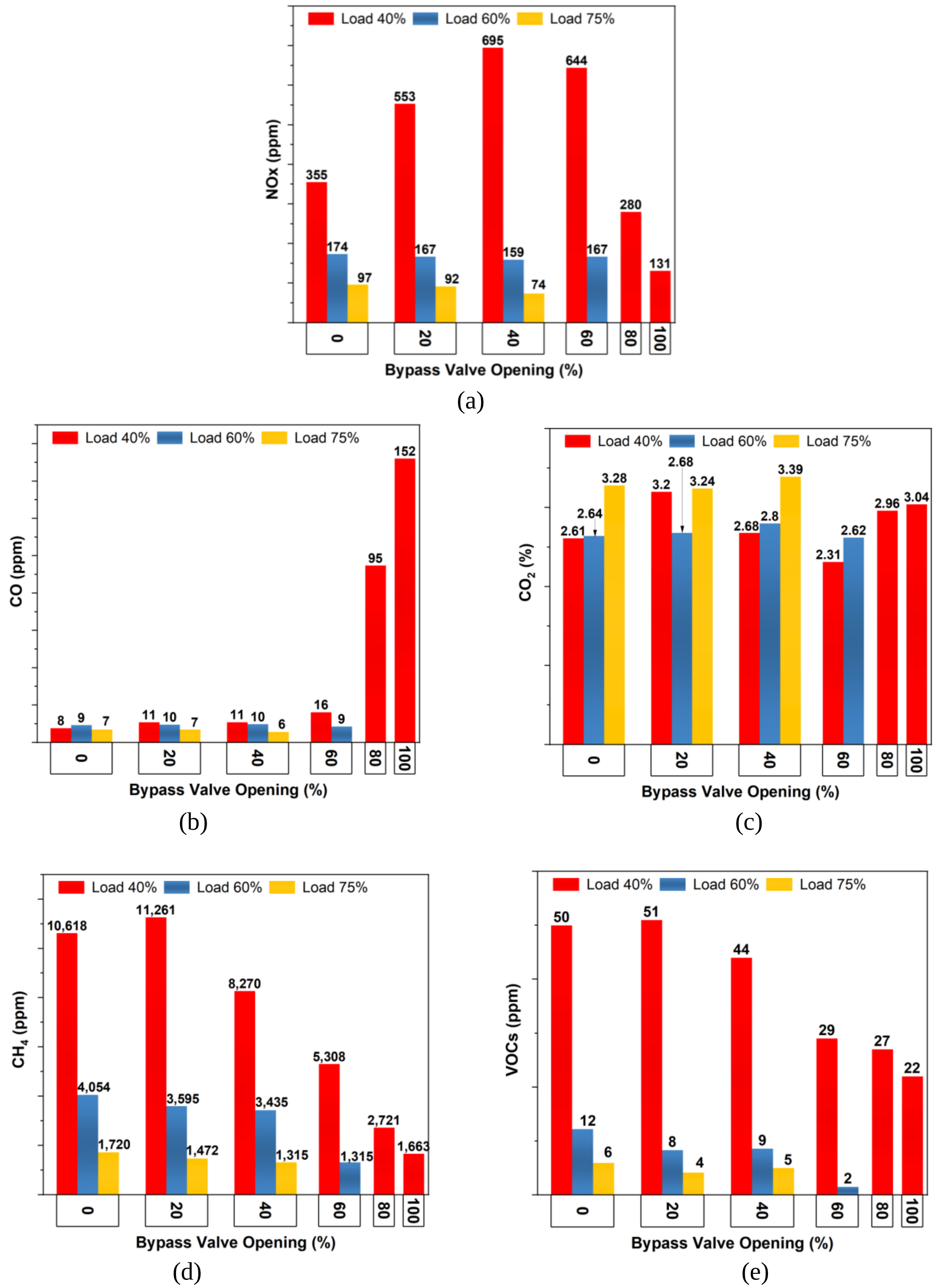


Figure 40. Post-catalyst emissions at various loads and bypass valve positions by Hassan [74].

In Hassan's research, the engine was run at 40%, 60%, and 75% loads with 5 bypass valve position from 0% to 100% with 20% increment [74]. Comparing the results from this study to Hassan's, it is clear that similar trends in emissions can be observed. In most cases, the emissions decrease when the bypass valve opens wider, except for CO [74]. The only three difference trends seen between both studies are NOx at 40% load and CO_2 at 60% and 70% loads. In Hassan's study, CO_2 stayed relatively constant with bypass valve position at higher loads [74] while it should be decreasing according to this present study. Nevertheless, the differences are modest and acceptable.

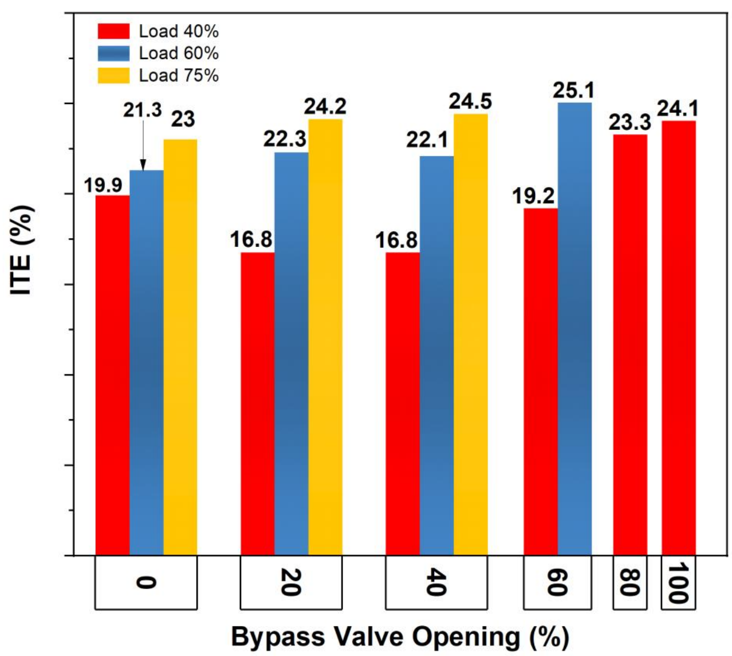


Figure 41. Engine ITE at various loads and bypass valve positions by Hassan [74].

From Figure 41, ITE increases with bypass valve positions which agrees with the trends seen in Figure 38. The highest ITE Hassan was able to achieve is 25.1% which is at 60% load and 60% bypass [74]. The ITE at the same condition in this paper is approximately 26% which is really close to Hassan's result. Due to the temperature constraint of the engine, Hassan could not test with 70% load at 60% bypass, as the temperature raise with bypass valve position (Figure 42, Appendix). According to their experiment, the optimal operating condition is at 60% load and 60% bypass which comes to agreement with the optimal operating condition determined by this study.

Chapter 7: Conclusion

The AJAX-DPC 81 engine was examined with various hydrogen concentrated HNG blends, up to 40% in volume fraction, and three bypass valve positions, from 0% to 60%, at several engine loads, 40%, 50%, 60%, 70%, and 75%. Emission data of NOx, CO, CO_2, CH_4, and VOCs was recorded to evaluate the contribution of HNG blend in reducing emissions. In-cylinder pressure, HRR, IMEP, and ITE were computed to investigate the effects on combustion characteristics of hydrogen. The results of this paper can be summarized as follows.

- Because of its lean-burn characteristics, the AJAX produces fairly low NOx emissions, below 250 ppm.
- NO_x decreases as hydrogen fraction increases in the blend, most drastically at 40% load. For all cases, NOx emission drops to approximately 25 ppm at 40% hydrogen content.
- Ultra-lean burn is the reason causing NOx to decrease parallel to exhaust temperature.
- Since hydrogen is highly diffusive, addition of hydrogen accelerates combustion and extends lean operation limit.
- Generally, due to the reduction of carbon-based component in the fuel, emission of CO decreases with increasing hydrogen fraction, except at 40% and 50% loads where the temperature is too low to facilitate complete burning.
- Emission of CO_2 increases with engine loads while emission of CO decreases. This is caused by complete combustion.
- CO_2 drops with increasing hydrogen fraction due to the same reason as why CO is reduced.

- Excess methane drops as hydrogen content and engine load increase. This also is caused by more complete combustion and less knocking occurring at higher load.
- Lower loads, 40 and 50%, seem to not operate as efficiently as higher loads since the excess methane at higher loads is significantly higher than that of lower loads.
- Complete combustion also reduces VOCs at higher loads and hydrogen contents.
- The catalyst installed with the engine works efficiently in converting CO into CO_2. After the catalyst, CO emission drops by about 200 ppm.
- After the catalyst, the amount of CH_4 increases. This could be caused by excess hydrogen reacting to CO or CO_2 in the exhaust gas under the promotion of high temperature and the catalyst.
- After implementing the bypass valve, the amount of all emissions decreases as the valve opens wider.
- At lower load, with the aid of 60% bypass position, the engine is able to achieve complete combustion reducing CO emission and rising CO_2 emission.
- From the IMEP plots, it can be concluded that the engine combusts more stably and efficiently at higher loads, higher hydrogen contents, and higher bypass valve positions.
- The heating value per volume unit of HNG blends decreases linearly with the addition of hydrogen. This could potentially cause insufficiency of energy if hydrogen content is too high.
- ITE increases in parallel to engine load and hydrogen content in most cases. At lower loads, ITE slightly decreases to 20% H_2 before starting to rise at 30% H_2.
- Higher bypass percentages also causes ITE to increase.

- The optimal condition for the AJAX-DPC engine to obtain both highest efficiency and lowest emission is 60% engine load with 40% hydrogen fraction and 60% bypass.

To further investigate the effect of hydrogen on engine performance and emissions, future works will include, first and foremost, the limit at which blending more hydrogen will cause energy insufficiency. Ignition timing should also be studied. Since the AJAX DPC-81 was designed to operate on NG, its ignition timing could possibly be not compatible with HNG. Determining proper ignition timing not only increases engine efficiency but also lengthens engine's lifetime. Vibrational analysis can also be examined at increasing load, hydrogen content, and bypass valve position to further understand the impact of such factor on combustion characteristics. Overall, it can be concluded that HNG blend is compatible to be used on AJAX DPC-81 engines to reduce emissions and improve engine performance.

Appendix

Table 6. The associated AFR and ER to each engine load and bypass valve position.

Testing conditions	**AFR**	**ER**
75% load 0% bypass	86.1	0.116
70% load 0% bypass	81.0	0.123
60% load 0% bypass	70.7	0.142
50% load 0% bypass	70.0	0.145
40% load 0% bypass	57.8	0.147
60% load 30% bypass	67.9	0.173
50% load 30% bypass	54.4	0.184
40% load 30% bypass	50.0	0.200
60% load 60% bypass	48.9	0.205
50% load 60% bypass	41.9	0.239
40% load 60% bypass	37.5	0.267

Table 7. Table of uncertainties.

Variables	**Uncertainty**
NOx	± 5.35 ppm
CO	± 0.12 ppm
CO_2	± 0.013 %
CH_4	± 59.10 ppm
VOCs	± 0.18 ppm
O_2	± 0.69 %
H_2O	± 0.72 %
T	± 4.23 °F
P	± 10.03 psi
IMEP	± 27.75psi
HRR	± 30.27 J/^{o}
ITE	± 3.47 %
AFR	± 0.86

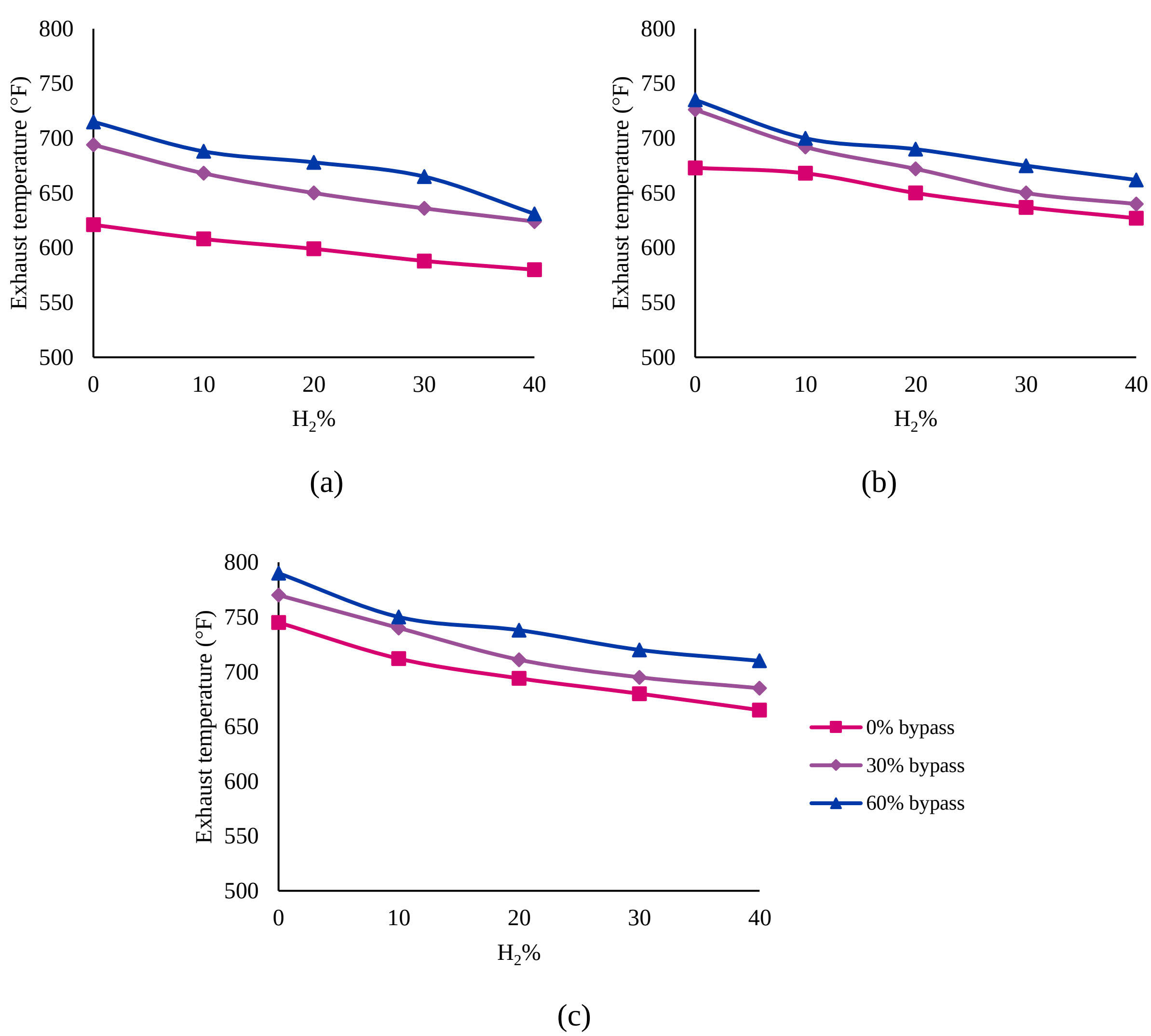
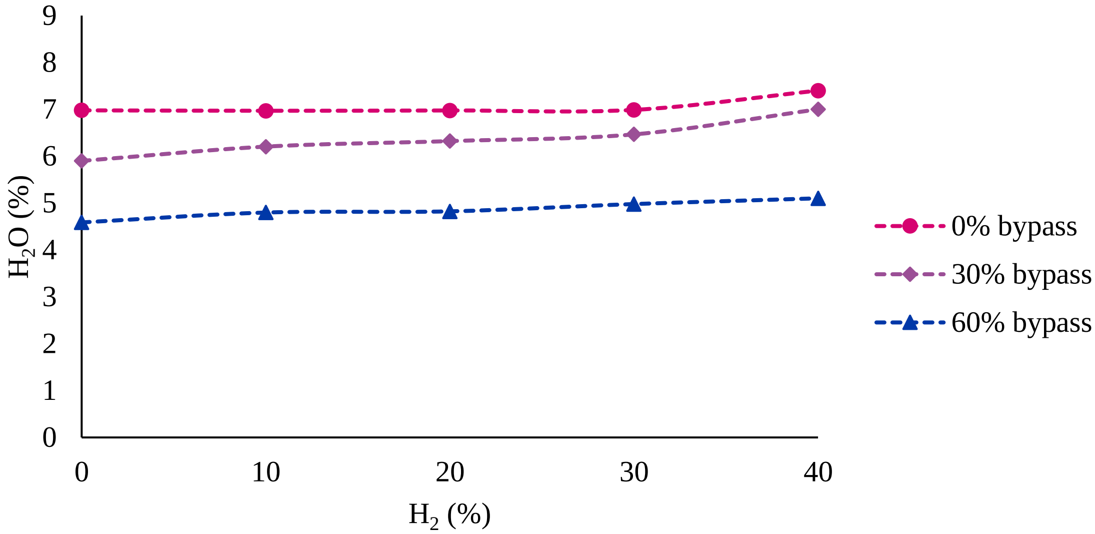
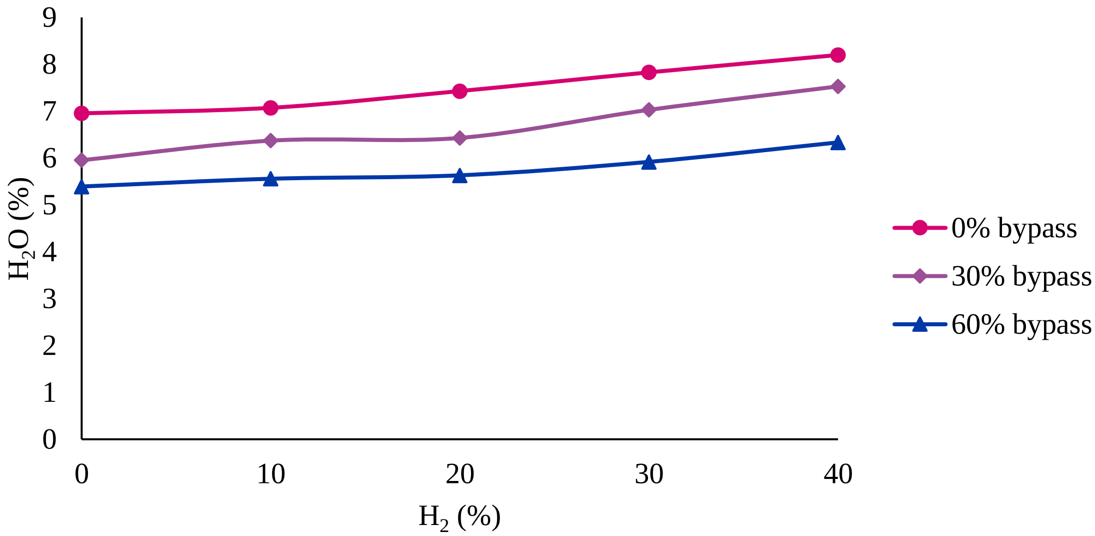


Figure 42. Exhaust temperature at rising H_2 fractions and various loads.
a) 40% load.   b) 50% load.   c) 60% load.

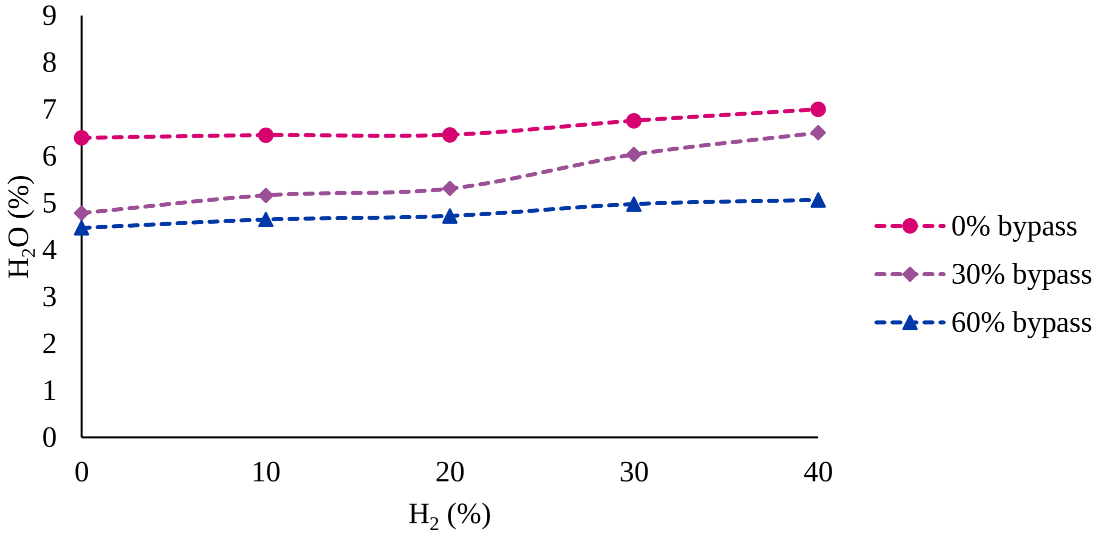


(a)

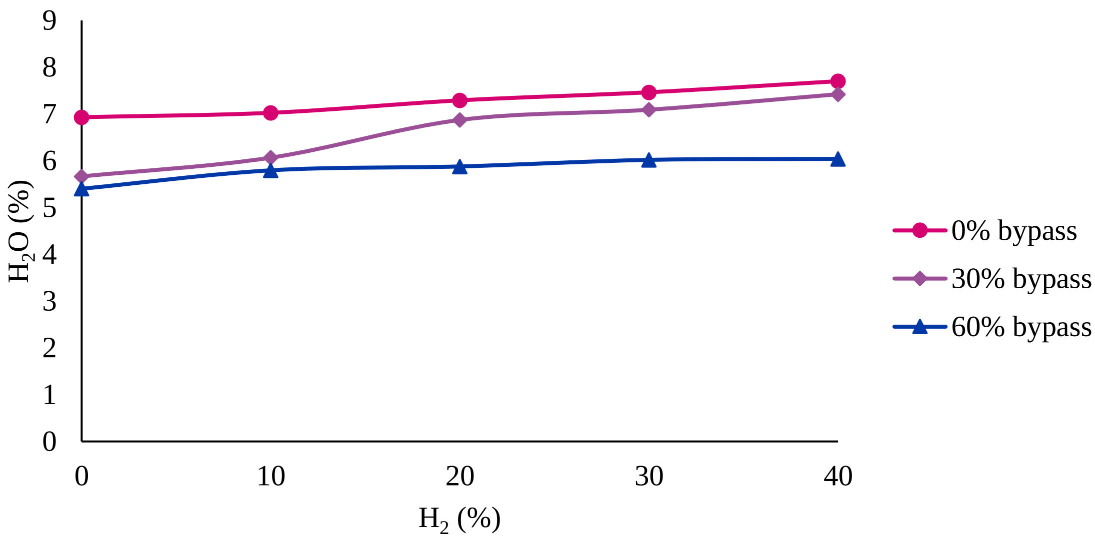


(b)

Figure 43. Water concentration in exhaust gas at 50% load.
a) Pre-catalyst. b) Post-catalyst.



(a)



(b)

Figure 44. Water concentration in exhaust gas at 60% load.
a) Pre-catalyst. b) Post-catalyst.

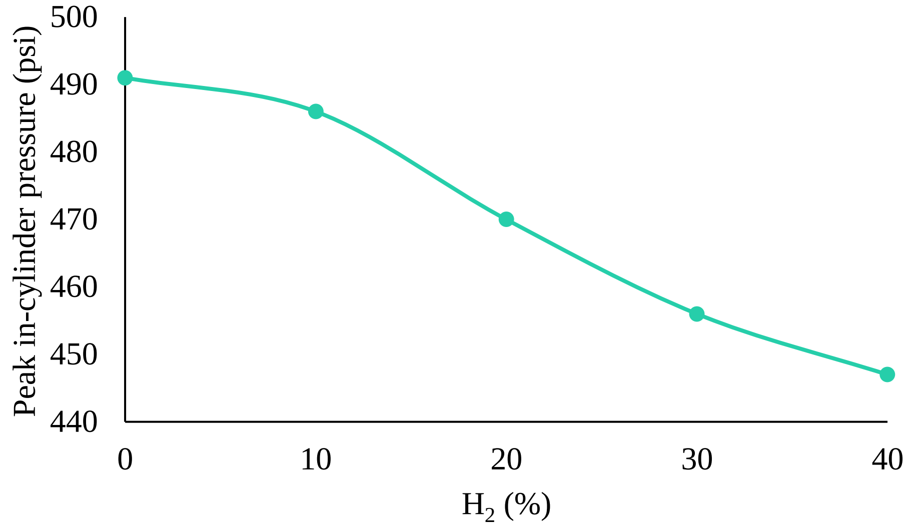


Figure 45. Variations of peak in-cylinder pressure with hydrogen concetration.

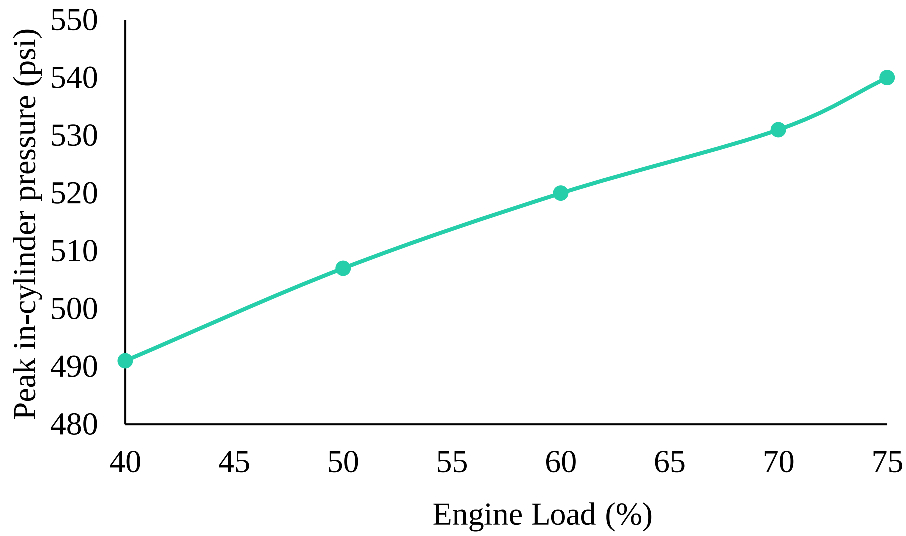


Figure 46. Variations of peak in-cylinder pressure with engine load.

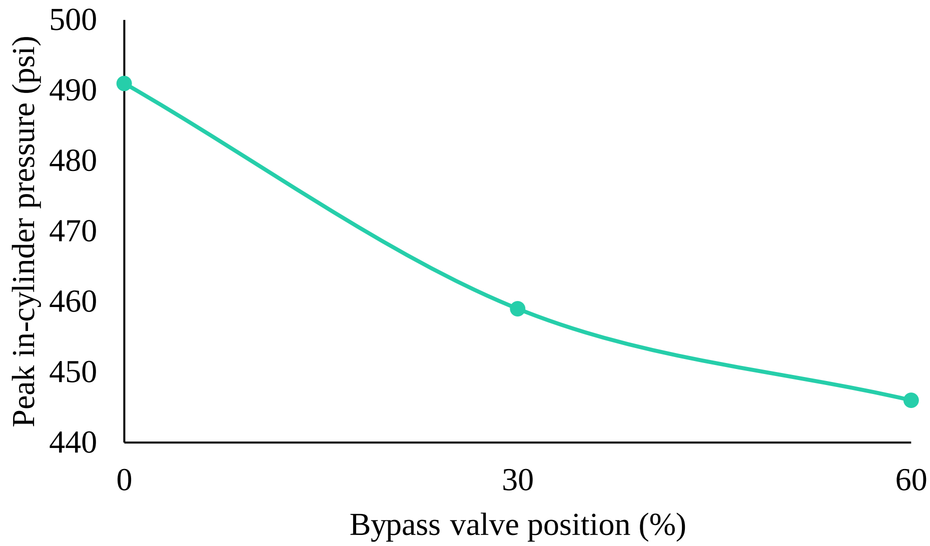


Figure 47. Variations of peak in-cylinder pressure with bypass position.

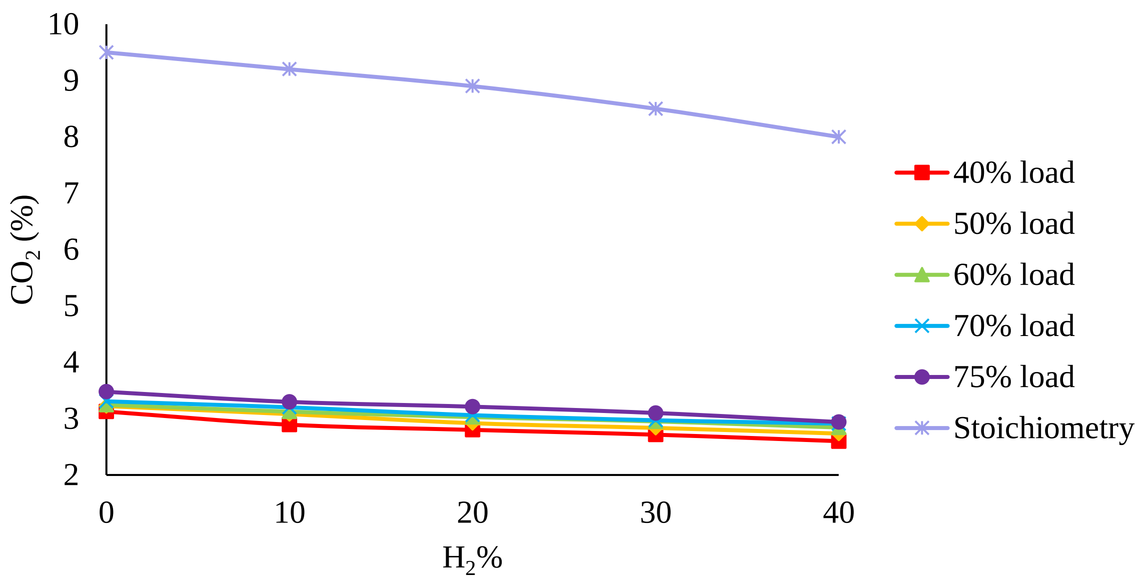


Figure 48. Comparison between experimental and stoichiometric CO_2 emissions.

References

[1] M. Voldsund M et al., “Low carbon power generation for offshore oil and gas production,” *Energy Conversion and Management*: *X*, vol. 17, pp. 100347, Jan. 2023, doi: 10.1016/j.ecmx.2023.100347.

[2] O. A. Odunlami et al., “The Effect of Air-Fuel Ratio on Tailpipe Exhaust Emission of Motorcycles,” *Fuel Communications*, vol. 11, pp. 100040, Jun. 2022.

[3] CIMAC WG 17 ‘Gas Engines’, “CIMAC Position Paper: Methane and Formaldehyde Emissions of Gas Engines,” *CIMAC International Council on Combustion Engines*, Apr. 2014, doi: 10.1016/j.jfueco.2021.100040.

[4] G. Lopez-Ruiz, I. Alava, and J.M. Blanco, “Impact of H_2/CH_4 Blends on the Flexibility of Micromix Burners Applied to Industrial Combustion Systems,” *Energy*, vol. 270, pp. 126882, May 2023, doi: 10.1016/j.energy.2023.126882.

[5] J. Ogden et al., “Natural Gas as a Bridge to Hydrogen Transportation Fuel: Insights from the Literature,” *Energy Policy*, vol. 115, pp. 317 – 329, Apr. 2018, doi: 10.1016/j.enpol.2017.12.049.

[6] D. Hjeij, Y. Biçer, and M. Koc, “Hydrogen Strategy as an Energy Transition and Economic Transformation Avenue for Natural Gas Exporting Countries: Qatar as a Case Study,” *International Journal of Hydrogen Energy*, vol. 47, no. 8, pp. 4977 – 5009, Apr. 2022, doi: 10.1016/j.ijhydene.2021.11.151.

[7] M. Kuznetsov et al., “Flammability Limits and Laminar Flame Speed of Hydrogen-Air Mixtures at Sub-Atmospheric Pressures,” *International Journal of Hydrogen Energy*, vol. 37, no. 22, pp. 17580 – 17588, Nov. 2012, doi: 10.1016/j.ijhydene.2012.05.049.

[8] M. Kamil and M. Rahman, “Performance Prediction of Spark-Ignition Engine Running on Gasoline-Hydrogen and Methane-Hydrogen Blends,” *Applied Energy*, vol. 158, pp. 556 – 567, Nov. 2015, doi: 10.1016/j.apenergy.2015.08.041.

[9] M. R. A. Mansor, Mohamad, and O. Sabah, “Numerical Investigation on Combustion and Emissions in a Direct Injection Compression Ignition Engine Fueled with Various Hydrogen-Methane-Diesel Blends at Different Intake Air Temperatures,” *Energy Reports*, vol. 7, pp. 403 – 421, Nov. 2021, doi: 10.1016/j.egyr.2021.07.110.

[10] C. Bauer and T. Forest, “Effect of Hydrogen Addition on the Performance of Methane-Fueled Vehicles. Part I: Effect on SI Engine Performance,” *International Journal of Hydrogen Energy*, vol. 26, no. 1, pp. 55 – 70, Jan. 2001, doi: 10.1016/S0360-3199(00)00067-7.

[11] Y. Wong and G. Karim, “An Analytical Examination of the Effects of Hydrogen Addition on Cyclic Variations in Homogeneously Charged Compression-Ignition Engines,” *International Journal of Hydrogen Energy*, vol. 25, no. 12, pp. 1217 – 1224, Dec. 2000, doi: 10.1016/S0360-3199(00)00030-6.

[12] United States Environmental Protection Agency. “Compliance Requirements for Stationary Engines.” https:// www.epa.gov/stationary-engines/compliance-requirementsstationary-engines (accessed Mar. 13, 2023).

[13] B. Viswanathan, *Energy Sources: Fundamentals of Chemical Conversion Processes and Applications*. Chennai, The Netherlands: Elsevier, 2017.

[14] S. Sanni and Ba. Oni, “The Roles of Hydrogen and Natural Gas as Biofuel Fuel-Additives Towards Attaining Low Carbon Fuel-Systems and High Performing ICEs,” *Greener and Scalable E-fuels for Decarbonization of Transport*, vol. 1, pp.193 – 219, 2022.

[15] C. Park, C. Kim, and Y. Choi. “Power Output Characteristics of Hydrogen-Natural Gas Blend Fuel Engine at Different Compression Ratios,” *International Journal of Hydrogen Energy*, vol. 37, no. 10, pp. 8681 – 8687, May 2012, doi: 10.1016/j.ijhydene.2012.02.052

[16] J. Kubesh, S. R. King, and W. E. Liss, “Effect of Gas Composition on Octane Number of Natural Gas Fuels," *SAE Technical Paper*, Oct. 1992, doi: 10.4271/922359.

[17] A. L. Simon. *Energy Resources*. Akron (Ohio). UK: Pergamon Press, 1975.

[18] M.R. Swain MR, M. J. Yusuf, Z. Dulger, and M. N. Swain, “The Effects of Hydrogen Addition on Natural Gas Engine Operation,” *SAE Technical Paper*, Oct. 1993, doi: 10.4271/932775.

[19] Y. Lee and G. Kim, “Effect of Gas Compositions on Fuel Economy and Exhaust Emissions of Natural Gas Vehicles,” *SAE Technical Paper*, Jun. 2000, doi: 2000-05-0169.

[20] E. Hu, Z. Huang, J. He, J. Zheng, and H. Miao, “Measurements of laminar burning velocities and onset of cellular instabilities of methane–hydrogen–air flames at elevated pressures and temperatures,” *International Journal of Hydrogen Energy*, vol. 34, pp. 5574 – 84, Jul. 2009, doi: 10.1016/j.ijhydene.2009.04.058.

[21] W. Francis and M. C. Peters, “Data Sheet No. 102 – Natural Gas,” *Fuels and Fuel Technology*, pp. 321 – 324, 1980, doi: 10.1016/B978-0-08-025249-0.50058-6.

[22] B. Çeper, “Use of Hydrogen-Methane Blends in Internal Combustion Engines,” in *Hydrogen Energy: Challenges and Perspectives*, InTech, 2012, pp. 175 – 200, 2012.

[23] T. N. Veziroglu, T. Petkov, and J. W. Sheffield, “An Outlook of Hydrogen as an Automotive Fuel,” *International Journal of Hydrogen Energy*, vol. 14, no. 7, pp. 449 – 74, 1989, doi: 10.1016/0360-3199(89)90031-1.

[24] B. A. Oni, S. E. Sanni, A. J. Ibegbu, and A. A. Aduojo, "Experimental Optimization of Engine Performance of a Dual-Fuel Compression-Ignition Engine Operating on Hydrogen-Compressed Natural Gas and Moringa Biodiesel," *Energy Reports*, vol. 7, pp. 607 – 619, Nov. 2021, doi: 10.1016/j.egyr.2021.01.019.

[25] D. C. North, "An Investigation of Hydrogen as an Internal Combustion Fuel," *International Journal of Hydrogen Energy*, vol. 17, no. 7, pp. 509 – 12, Jul. 1992, doi: 10.1016/0360-3199(92)90150-U.

[26] A-B. M. Asadiq, and A-J. H. Ashahad, "A Prediction Study of the Effect of Hydrogen Blending on the Performance and Pollutants Emission of a Four-Stroke Spark Ignition Engine," *International Journal of Hydrogen Energy*, vol. 24, no. 4, pp. 363 – 75, Apr. 1999, doi: /S0360-3199(98)00040-8.

[27] A. S. Al-Baghdadi Maher, and A. S. Al-Janabi Haroun, "A Prediction Study of a Spark Ignition Supercharged Hydrogen Engine," *Energy Conversion Manage*, vol. 44, no. 20, pp. 3143 – 50, Dec. 2003, doi: 10.1016/S0196-8904(03)00127-4.

[28] A. P. Carlucci AP, A. de Risi, D. Laforgia, and F. Naccarato, "Experimental Investigation and Combustion Analysis of a Direct Injection Dual-Fuel Diesel-Natural Gas Engine," *Energy*, vol. 33, pp. 256 – 263, Feb. 2008, doi: 10.1016/j.energy.2007.06.005.

[29] P. Malogorzata. *Mitigation of Landfill Gas Emissions*. Taylor and Francis. 2014

[30] C. Clifford. "The Clean Hydrogen Energy Economy Was a Dream. The Climate Bill Could Make It a Reality This Decade." *CNBC*. https://www.cnbc.com/2022/09/08/clean-hydrogen-industry-got-huge-boost-from-inflation-reduction-act.html (accessed Apr. 16, 2023).

[31] Hydrogen and Fuel Cell Technologies Office. "Hydrogen Shot." *Office of Energy Efficiency and Renewable Energy*. https://www.energy.gov/eere/fuelcells/hydrogen-shot#:~:text=Currently%2C%20hydrogen%20from%20renewable%20energy%20costs%20about%20%245%20per%20kilogram (accessed Apr. 16, 2023).

[32] O. H. Ghazal, "Performance and Combustion Characteristic of CI Engine Fueled with Hydrogen Enriched Diesel," *International Journal of Hydrogen Energy*, vol. 38, pp. 15469 – 15476, Nov. 2013, doi: 10.1016/j.ijhydene.2013.09.037.

[33] J. B. Heywood, *Internal combustion engine fundamentals*, McGraw-Hill Education, 2018.

[34] S. Verhelst, "Recent Progress in the Use of Hydrogen as a Fuel for Internal Combustion Engines," *International Journal of Hydrogen Energy*, vol. 39, pp. 1071 – 1085, Jan. 2014, doi: 10.1016/j.ijhydene.2013.10.102.

[35] S. Dey and G. C. Dhal, "Materials Progress in the Control of CO and CO_2 Emission at Ambient Conditions: An Overview." *Materials Science for Energy Technologies*, vol. 2, no. 3, pp. 607 – 623, Dec. 2019, doi: 10.1016/j.mset.2019.06.004.

[36] Y. Belyi, and T. R. Teregulov, "Hydrogen Energy: Advantages and Disadvantages," *Journal of Science and Education*, vol. 12, pp. 8 – 10, 2016.

[37] Y. Ren, D. Lou, P. Tan, Y. Zhang and X. Sun, "Emission Reduction Characteristics of After-Treatment System on Natural Gas Engine: Effects of Platinum Group Metal Loadings and Ratios," *Journal of Cleaner Production*, vol. 298, pp. 126833, May 2021, doi: 10.1016/j.jclepro.2021.126833.

[38] C. E. Baukal Jr., *The John Zink Hamworthy Combustion Handbook: Volume 1-Fundamentals*. CRC Press, 2021.

[39] A.J. Martyr and M.A. Plint, “The Combustion Process and Combustion Analysis” in *Engine Testing*, Butterworth – Heinemann, pp. 375 – 406, 2012.

[40] T. Shudo, K. Shimamura, and Y. Nakajima, “Combustion and Emissions in a Methane DI Stratified Charge Engine with Hydrogen Pre-Mixing,” *JSAE Review*, vol. 21, no. 5, pp. 3 – 7, Jan. 2000, doi: 10.1016/S0389-4304(99)00061-2.

[41] V. Raman et al., “Hythane – An Ultra-clean Transportation Fuel,” *10th World Hydrogen Conference*, vol. 3, pp. 1797 – 1806, 1994.

[42] J. F. Larsen and J. S. Wallace, “Comparison of Emissions and Efficiency of a Turbocharged Lean Burn Natural Gas and Hythane-Fueled Engine,” *Journal of Engineering for Gas Turbines and Power*, vol.119, pp. 218 – 226, Jan. 1997, doi: 10.1115/1.2815553.

[43] R. L. Hoekstra at al., “Experimental Study of a Clean Burning Vehicle Fuel,” *International Journal of Hydrogen Energy*, vol. 20, pp. 737 – 745, Sep. 1995, doi: 10.1016/0360-3199(95)00008-2.

[44] S. Akansu et al., “Experimental Study of Gasoline-Ethanol-Hydrogen Blends Combustion in an SI Engine”, *International Journal of Hydrogen Energy*, vol. 42, pp. 25781 - 25790, Oct. 2017, doi: 10.1016/j.ijhydene.2017.07.014.

[45] K. Collieret al., “Untreated Exhaust Emissions of a Hydrogen-Enriched CNG Production Engine Conversion,” SAE Technical Paper, Feb. 1996, doi: 10.4271/960858.

[46] J. Wang et al., “Combustion Behaviors of a Direct-Injection Engine Operating on Various Fractions of Natural Gas-Hydrogen Blends,” *International Journal of Hydrogen Energy*, vol. 32, no. 15, pp. 3555 – 3564, Oct. 2007, doi: 10.1016/j.ijhydene.2007.03.011.

[47] R. Sierens and E. Rosseel E., “Variable Composition Hydrogen-Natural Gas Mixtures for Increased Engine Efficiency and Decreased emissions,” *Journal of Engineering for Gas Turbines and Power*, vol. 122, pp. 135 – 140, Jan. 2000, doi: 10.1115/1.483191.

[48] So. O. Akansu, N. Kahraman, and B. Ceper, “Experimental Study on a Spark Ignition Engine Fuelled Methane-Hydrogen Mixtures,” *International Journal of Hydrogen Energy*, vol. 32, no. 17, pp. 4279 – 4284, Dec. 2007, doi: 10.1016/j.ijhydene.2007.05.034.

[49] G. A. Karim, I. Wierzba, and Y. Al-Alousi, “Methane-Hydrogen Mixtures as Fuels,” *International Journal of Hydrogen Energy*, vol. 21, no. 7, pp. 625 – 631, Jul. 1996, doi: 10.1016/0360-3199(95)00134-4.

[50] K. Aydin and R. Kenanoğlu, “Effects of Hydrogenation of Fossil Fuels with Hydrogen and Hydroxy Gas on Performance and Emissions of Internal Combustion Engines”, *International Journal of Hydrogen Energy*, vol. 43, pp. 14047 – 14058, Jul. 2018, doi: 10.1016/j.ijhydene.2018.04.026.

[51] B. A. Ceper, S. O. Akansu, and N. Kahraman, “Investigation of Cylinder Pressure for H_2/CH_4 Mixtures at Different Loads,” *International Journal of Hydrogen Energy*, vol. 34, pp. 4855 – 4861, Jun. 2009, doi: 10.1016/j.ijhydene.2009.03.039.

[52] N. Kahraman et al., “Investigation of Combustion Characteristics and Emissions in a Spark-Ignition engine Fueled with Natural Gas-Hydrogen Blends,” *International Journal of Hydrogen Energy*, vol. 34, pp. 1026 – 1034, 2009, doi: 10.1016/J.IJHYDENE.2008.10.075.

[53] P. Dimopoulos et al., “Hydrogen-Natural Gas Blends Fueling Passenger Car Cngines: Combustion, Emissions and Well-to-Wheels Assessment,” *International Journal of*

Hydrogen Energy, vol. 33, pp. 7224 – 7236, Dec. 2008, doi: 10.1016/j.ijhydene.2008.07.012.

[54] L. M. Das, “Utilization of Hydrogen-CNG Blends in an Internal Combustion Engine”, *11th World hydrogen energy conference*, pp. 1513–1535, 1996.

[55] A. Boretti, “Advances in Diesel-LNG Internal Combustion Engines,” *Applied Science*, vol. 10, no. 4, pp.1296, Feb. 2020, doi: 10.3390/app10041296.

[56] V. N. Konoplev, Z. G. Melnikov, and V. Sarbaev, “The Main Aspects of Safety of hydrogen Energy in Relation to the Motor Transport Process on Gas Engine Fuels,” *Transportation Research Proccedia*, vol. 36, pp. 295 – 302, 2018, doi: 10.1016/j.trpro.2018.12.086.

[57] B. Nagalingam, F. Duebel, and K. Schmillen, “Performance Study Using Natural Gas, Hydrogen-Supplemented Natural Gas and Hydrogen in AVL Research Engine,” *International Journal of Hydrogen Energy*, vol. 8, no. 9, pp. 715 - 720, 1983, doi: 10.1016/0360-3199(83)90181-7.

[58] V. N. Konoplev et al., “Hydrogen Energy in the Strategy of International Energy Consumption and Its Relationship with the Perspective of the Development of Transport Technologies,” *AIAA SciTech Forum*, pp. 58 – 63, 2018.

[59] E. Aslan, M. Ergeneman, and C. Sorusbay, *Use of Hydrogen in Internal Combustion Engine as Fuel*, Istanbul, Istanbul Technical University, 1991.

[60] A. Genovese et al., “On-Road Experimental Tests of Hydrogen-Natural Gas Blends on Transit Buses,” *International Journal of Hydrogen Energy*, vol. 36, pp. 1775 – 1783, Jan. 2011, doi: 10.1016/j.ijhydene.2010.10.092.

[61] DCL International, “Catalyst Element: Metalcor,” DCL. [Online]. Available: http://www.dcl-europe.com/pdf/replacement-elements/metalcor-brochure.pdf

[62] A. J. Wheeler and A. R. Ganji, *Introduction to Engineering Experimentation,* Pearson, 2010.

[63] Moreno, F., et al., “Efficiency and Emissions in a Vehicle Spark Ignition Engine Fueled with Hydrogen and Methane Blends,” *International Journal of Hydrogen Energy*, vol. 37, no. 15, pp. 11495 – 11503, Aug. 2012, doi: 10.1016/j.ijhydene.2012.04.012.

[64] H. Zhao, *Advanced Direct Injection Combustion Engine Technologies and Development: Diesel Engines*, Elsevier, 2009.

[65] A. A. Al-Harbi et al., “Effect of Operation under Lean Conditions on NOx Emissions and Fuel Consumption Fueling an SI engine with Hydrous Ethanol-Gasoline Blends Enhanced with Synthesis Gas,” *Energy*, vol. 238, pp. 121694, Jan. 2022, doi: 10.1016/j.energy.2021.121694.

[66] B. Masum et al., “Study on the Effect of Adiabatic Flame Temperature on NOx Formation Using Ethanol Gasoline Blend in SI engine,” Advanced Materials Research, vol. 781 – 784, pp. 2471 – 2475, Sep. 2013, doi: doi.org/10.4028/www.scientific.net/amr.781-784.2471.

[67] X. Zhen et al., “Comparative Study on Combustion and Emission Characteristics of Methanol-Hydrogen, Ethanol-Hydrogen, and Methane-Hydrogen Blends in High Compression Ratio SI Engine,” *Fuel*, vol. 267, pp. 117193, May 2020, doi: 10.1016/j.fuel.2020.117193.

[68] A. Pai. “Clean Burn Gas Generator Sets.” Caterpillar Inc. https://www.cat.com/en_US/by-industry/electric-power/Articles/White-papers/lean-burn-gas-generator-sets.html. (accessed Apr 01, 2023)

[69] P. Soltic and T. Hilfiker, “Efficiency and Raw Emission Benefits from Hydrogen Addition to Methane in a Prechamber-Equipped Engine,” International Journal of Hydrogen Energy, vol. 45, no. 43, pp. 23638 – 23652, Sep. 2020, doi: 10.1016/j.ijhydene.2020.06.123.

[70] X. Li et al., “Combustion and Emission Characteristics of a Two-Stroke Diesel Engine Operating on Alcohol,” *Renewable Energy*, vol. 30, no. 13, pp. 2075 – 2084, Oct. 2005, doi: 10.1016/j.renene.2004.05.014.

[71] Z. Zhang et al., “Emerging Role of Wetland Methane Emissions in Driving 21st Century Climate Change,” *Proceedings of the National Academy of Sciences*, vol. 114, no. 36, pp. 9647 – 9652, Aug. 2017, doi: 10.1073/pnas.1618765114.

[72] L. He et al., “A Review on Catalytic Methane Combustion at Low Temperatures: Catalysts, Mechanisms, Reaction Conditions, and Reactor Designs,” *Renewable and Sustainable Energy Reviews*, vol. 119, pp. 109589, Mar. 2020, doi: 10.1016/j.rser.2019.109589.

[73] F. Ma et al., “Combustion and Emission Characteristics of a Port-Injection HCNG Engine under Various Ignition Timings,” *International Journal of Hydrogen Energy*, vol. 33, no. 2, pp. 816 – 822, Jan. 2008, doi: 10.1016/j.ijhydene.2007.09.047.

[74] H. A. Hassan, “Integrated System to Reduce Emissions from Natural Gas-Fired Reciprocating Engine,” M. S. Thesis, School of Aerospace and Mechanical Engineering, University of Oklahoma, Oklahoma, 2022. [Online]. Available: https://shareok.org/handle/11244/335966.