ÇUKUROVA UNIVERSITY
INSTITUTE OF NATURAL AND APPLIED SCIENCES

MSc THESIS

Oğuz BAŞ

EFFECT OF SPARK PLUG ALTERATION ON PERFORMANCE AND EMISSIONS USING HYDROGEN ENRICHED GASOLINE IN SI ENGINE UNDER VARIOUS LOADS AND COMPRESSION RATIOS

DEPARTMENT OF AUTOMOTIVE ENGINEERING

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We certify that the thesis titled above was reviewed and approved for the award of degree of the Master of Science by the board of jury on 16/07/2018

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ABSTRACT

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In this study, performance and emission parameters of a variable compression spark ignited engine was investigated with different spark plugs and hydrogen enrichment. The tests were carried out with a four stroke, single cylinder, naturally aspirated, variable compression ratio (VCR) engine. Two different compression ratios (CR) of 8.5:1 and 10:1 under %50 part throttle condition were implemented throughout the experiments. Moreover, engine loads of 8 Nm, 13 Nm and 17 Nm were applied to evaluate effects of different spark plugs and hydrogen usage at different engine loads. Copper, iridium and platinum spark plugs were tested for each experiment condition. In addition, hydrogen was added through the intake manifold with flow rates of 0, 2 and 4 lit/min to enhance combustion of VCR engine. Brake power (BP), brake specific fuel consumption (BSFC), carbon monoxide (CO), carbon dioxide (CO_2) nitric oxides (NO_x) and unburned hydrocarbons (UHC) values were determined with this study. According to test results, iridium and platinum spark plug usage, hydrogen addition and higher compression ratio improved BP and BSFC values and the engine emitted lower CO and UHC. However, higher CO_2 emitted because of increased completeness of the combustion and NO_x gases have risen due to increment in-cylinder temperatures. These variances occurred more obvious with platinum spark plug usage comparing to iridium spark plug. In addition, effects of spark plug alteration, hydrogen addition and higher CR on enhancements of BP, BSFC and CO were comparatively lower at higher engine loads.

Key Words: Platinum, Iridium, Spark Plug, Hydrogen, Compression ratio

ÖZ

YÜKSEK LİSANS TEZİ

FARKLI YÜK VE SIKIŞTIRMA ORANLARI ALTINDA HİDROJEN İLE ZENGİNLEŞTİRİLMİŞ BENZİNLİ MOTORDA FARKLI TİP ATEŞLEME BUJİLERİNİN MOTOR PERFORMANS VE EMİSYON DEĞERLERİNE ETKİSİ

Oğuz BAŞ

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Bu çalışmada, kıvılcım ateşlemeli değişken sıkıştırma oranlı bir motorun farklı bujilerle ve hidrojen zenginleştirmesiyle performans ve emisyon parametreleri incelenmiştir. Deneyler, dört zamanlı, tek silindirli, sıkıştırma oranı değiştirilebilir (VCR) bir motorda yapılmıştır. Deneyler süresince, 8.5:1 ve 10:1 olarak iki farklı sıkıştırma oranında (CR), %50 kısmi gaz yükü uygulanılmıştır. Ayrıca farklı buji ve hidrojen kullanımın farklı motor yüklerinde etkilerini değerlendirmek için 8 Nm, 13 Nm ve 17 Nm motor yükleri kullanılmıştır. Bakır, iridyum ve platinyum bujiler her bir deney durumu için test edilmiştir. Bunun yanı sıra, VCR motorunun yanmasını geliştirmek için emme manifoldundan 0, 2 ve 4 l/dk akış oranlarında hidrojen eklenmiştir. Bu çalışmayla, efektif güç (BP), özgül yakıt tüketimi (BSFC), karbon monoksit (CO), karbon dioksit (CO_2), azot oksit (NO_x) ve yanmamış hidrokarbon (UHC) değerleri belirlenmiştir. Test sonuçlarına göre, iridyum ve platinyum buji kullanımı, hidrojen ilavesi ve yüksek sıkıştırma oranı BP ve BSFC değerlerini iyileştirmiş ve motor daha düşük CO ve UHC vermiştir. Bunumla birlikte, geliştirilen yanma ile daha yüksek CO_2 ve artan silindir içi sıcaklığın dolayısıyla da daha fazla NO_x gazları salınmıştır. Bu değişimler, platinyum buji kullanımında iridyum bujiye kıyasla daha belirgin olmuştur. Ayrıca, buji değişimi, hidrojen ilavesi ve yüksek sıkıştırma oranının, BP BSFC ve CO değerlerini iyileştirmedeki etkileri yüksek motor yüklerinde daha az gerçekleşmiştir.

Anahtar Kelimeler: Platinyum, İridyum, Buji, Hidrojen, Sıkıştırma Oranı

EXTENDED ABSTRACT

Internal combustion engines are thermal machineries that convert chemical energy into mechanical energy and used in transportation and industry in a wide extent. This conversion is performed energy out from fuel that is burnt in the combustion chamber. However, rapid depletion of fossil fuels and increasing environmental concerns make development of internal combustion engines very critical. Moreover, automotive giants are increasingly turning to alternative engines for diesel engines as diesel engines break out on the backs of emission cheating software scandals and manufacturers want to avoid tightening controls and paying astronomical penalties. These alternatives are foreseen LPG, CNG, electrical and hybrid application of them. Despite the fact that vehicle manufacturers are expected to adapt electric vehicles in the long run, petrol engines, which are already in widespread use, are at the forefront in the short and medium runs. It is the most important indicator of the situation that the manufacturers recently have to remove the diesel engine option on the flagship and other product ranges or to keep it in a very limited number and to offer new generation gasoline engines instead of these engines. In this respect, improvements in the efficiency of gasoline engines, which are around 25-35%, are among the important targets of the automotive industry.

Petrol (spark ignited) engines are machines that are widely used especially in road vehicles because of their high power to weight ratio, less maintenance requirements than diesel engines, they are cheap and compatible with CNG and LPG fuels. However, the fact that thermal efficiency and torque values are low limits the widespread use of gasoline engines, including areas such as heavy duty machineries. It is possible to solve the above mentioned problems with the improvement of the combustion However, gasoline engines are not able to operate in very poor mixtures without sacrificing stability, silence and emission due to problems caused by ignition, such as slow flame initiation and propagation.

It is also expected that in the future, higher compression ratios and poorer air-fuel mixtures will be adapted for improvements of combustion of and fuel economy in spark-ignition engines. These developments are predicted that they will lead to electrode erosion and deterioration. To avoid these drawbacks, the use of long-lasting, durable and efficient spark plugs is crucial. For the development of the spark plugs, the manufacturers produce spark plugs with different geometries, number of electrodes, gaps and electrode materials. Spark plugs such as yttrium, iridium and platinum are widely found in market however they were seen suspicious by the consumer for their relatively high prices. With the use of this type of noble metals, the main goal of the manufacturers is to reach the optimum operating temperatures of the engine as quickly as possible, improve the cold working characteristics and ignition in the cylinder, achieve smoother and more quiet combustion and reduce electrode wear rates. However, it is a necessity to evaluate the engine performance and emission values with these plugs from the perspective of electrode wear and carbon residue formation as well.

On the other hand, hydrogen has high diffusivity and flame speed, high stoichiometric air fuel ratio, very silent combustion and carbon-free structure make hydrogen one the most important alternative fuel. Hydrogen fuels, which can be used with fuel cell power or directly in combustion chamber, are among the most important fuels of the future.

In this study, the effects of different spark plugs, hydrogen additions and compression ratios on engine performance and emission values were investigated at different engine loads.

Experimental studies have been carried out with two different compression ratios (8.5: 1 and 10: 1) and three different spark plug types (copper (conventional), iridium and platinum) with a variable compression ratio spark ignition engine (Kirloskar Oil Engines - 240). Hydrogen enriched gasoline fuel was tested with hydrogen flow rates of (H0), 2 l/ min (H2) and 4 l / min (H4). During the experiments, the part throttle (50%) was kept constant. Before each experiment, the engine was operated for 5 minutes to reach the stable operating conditions. In the experiments, as performance data; carbon monoxide (CO), carbon dioxide (CO_x), nitrogen oxides (NO_x) and unburned hydrocarbons (UHC) values were obtained and the results were investigated in terms of brake power (BP) and brake specific fuel consumption (BSFC).

When experimental results are evaluated, iridium and platinum spark plugs have enhanced significantly carbon monoxide (CO) and unburned hydrocarbon (UHC) emissions, along with brake power (BP) and brake specific fuel consumption (BSFC) values. As comparison, it was found that the platinum spark plug gave relatively better results for improving these values. However, it turns out that platinum spark plugs increased carbon dioxide (CO_2) emissions. The most important reason for these findings is thought to be the improvement of the burning reactions by increasing the electric charge density in the cylinder with the use of alternative spark plugs. On the other hand, nitrogen oxides (NO_x) emissions have been determined to increase due to the risen in-cylinder temperature.

On the other hand, by addition hydrogen to the fuel, significant improvement has been observed in BP, BSFC and emission values of CO and UHC. An increment in CO_2 and NO_x emissions was also detected by the addition of hydrogen. It can be interpreted that the effect of accelerating the combustion reactions with high diffusivity and flame speed of the hydrogen and the increment of the combustion temperature with high thermal values gave these results. On average, when H2 fuel is used, compared to the case where no hydrogen is used; improvement of 11.87% for BP, 10.83% for BSFC, 8.93% for CO and 16.62% for HC; when H4 fuel is used; improvement of 21.87% for BP, 22.29% for BSFC, 13.68% for CO and 25.92% for HC have been obtained. On the other hand, on the average compared to the non-hydrogen fuel, the CO_2 and NO_x emissions were increased for H2 fuel by 16.83% and 82.16%, respectively. CO_2 and NO_x emissions increased for H4 by 45.13% and 204.53% respectively.

As a result of increasing the compression ratio from 8.5: 1 to 10: 1, similar changes in performance and emission data have been observed in parallel with the results of hydrogen enrichment and the use of different spark plugs. It can be said that these changes are due to significant improvement in combustion delay, combustion completeness and temperature rise in the with high compression ratios. On average, increasing the compression ratio from 8.5: 1 to 10: 1 improved BP by 21.05%, BSFC by 16.47%, CO emissions by 25.15%, and UHC emissions by 16.82%. On the other hand, increasing the compression ratio increased CO_2 and NO_x emissions by 37.39% and 28.13%, respectively.

Moreover, effects of usage of different spark plugs, addition of hydrogen and application of higher compression ratio on performance and emission values were comparatively lower at higher engine loads.

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CONTENTS **PAGE**

LIST OF TABLES **PAGE**

LIST OF FIGURES **PAGE**

LIST OF ABBREVIATIONS AND NOMENCLATURE

η_{th}	:	Thermal Efficiency
η_v	:	Volumetric Efficiency
A	:	Area
ATDC	:	After Top Dead Center
BDC	:	Bottom Dead Center
BMEP	:	Brake Mean Effective Pressure
BSFC	:	Brake Specific Fuel Consumption
BTDC	:	Before Top Dead Center
BTHE	:	Brake Thermal Efficiency
c	:	Specific heat
c_p	:	Specific heat at constant pressure
c_v	:	Specific heat at constant volume
CA	:	Crank Angle
CO	:	Carbon Monoxide
CO_2	:	Carbon Dioxide
COV	:	Coefficient of Variations
COV_{imep}	:	Coefficient of Variations in IMEP
COV_{rpm}	:	Coefficient of Variations in engine speed
CR	:	Compression Ratio
D	:	Diameter
G	:	Gram
HC	:	Hydrocarbons
IMEP	:	Indicated Mean Effective Pressure
kVA	:	Kilo Volt-Amper
L	:	Stroke

LOL	:	Lean operation limit
MAP	:	Manifold Absolute Pressure
mJ	:	Mega Joule
n	:	Engine Speed
nm	:	Nanometer
Nm	:	Newton-meter
NO	:	Nitrogen Oxide
NO_2	:	Nitrogen Dioxide
NO_x	:	Nitrogen Oxides
P	:	Pressure
PM	:	Particulate matter
PPM	:	Particulate Per Million
Q	:	Heat
CR	:	Compression Ratio
RPM	:	Revolution Per Minute
SI	:	Spark Ignition
SFC	:	Specific Fuel Consumption
T	:	Temperature
TDC	:	Top Dead Center
V	:	Volume
V_c	:	Clearance Volume
VCR	:	Variable Compression Ratio
V_s	:	Swept Volume
WOT	:	Wide open throttle
γ	:	Ratio of specific heats (c_p/c_v)

1. INTRODUCTION

The main energy demand in the world is supplied from the fossil based substances such as petroleum, coal and natural gas. With the development of industry and economy, energy shortage and environmental problems have been becoming more and more serious. The internal combustion engine and automotive industry are confronted with severe and realistic challenges. Emission regulations and raised fuel cost are among these challenges.

World's energy sources are exhausting with expedition. Thus, investigations of renewable and sustainable alternative energy sources have attracted great importance. Beside to that, other issues; increasing petroleum products prices, air pollutions and global warming may be solved with renewable energy investigations (Awad et al., 2014). By the depletion of fossil reserve potentials, the requirement to new technologies for sustainable energy of human being is increasing.

As known people and goods transportation is provided by utilization of fossil fuels such as gasoline, diesel, compressed natural gas, liquefied petroleum gas, etc. Despite the fact that diesel market share is still very big among them in European Union nations, in mid-term, it is expected that the diesel engine inevitably will lose its share in the market rapidly ('Cars in Europe: diesel's not dead yet | Euronews', n.d.). Especially, considering recent emission - cheating scandals, development of spark ignition engine is being more and more essential for the future of automotive industry. From this point, it can be said that gasoline engine must be developed more and usage of alternative fuels have to be assessed carefully from different aspects.

Gasoline engine is known for their high power to weight ratio, stable and silent performance, low prices, less maintenance needs and costs and feasibilities for CNG and LPG fuels comparing with similar volume of diesel engine. However, their thermal efficiencies and torque outputs are lower comparing with modern diesel engines (Stone and Ball, 2004). These main problems limit the widespread usage of gasoline engines such as heavy duty machineries.

Lean burn is one of the most effective ways to solve the above-mentioned problems. Nevertheless, current engines cannot be run sufficiently lean due to ignition related problems such as the slow flame initiation and propagation along with potential misfiring. In the future of spark ignited engine it is expected that gasoline engines will operate with much higher compression ratios, faster compression rates, and much leaner fuel-to-air ratios for the sake of combustion enhancement and fuel economy. Furthermore it will aggravate the electrode degradation and erosion of the spark plugs more. In order to accomplish these strong challenges, usage of alternative spark plugs is seemed urgent for spark ignited engines.

Spark plugs invention was a milestone for the production of an effective gasoline engine. In 1860, Etienne Lenoir put an electrical spark plug in his first internal combustion gas engine having a piston, so it is generally attributed to the invention of the spark plug device.

Early patents for spark plugs were got by Nikola Tesla (Tesla, 1898), Frederick Richard Simms and Robert Bosch. However, the invention of the first commercially available high-voltage spark plug as part of a magneto ignited system by Robert Bosch's engineer Gottlob Honold in 1902 made possible the development of the spark ignition engine. Essential production improvements may also be attributed to Albert Champion, the Lodge brothers and Kenelm Lee Guinness, of the Guinness brewing family, who developed the KLG brand. Helen Blair Bartlett also played an important role in manufacturing the insulators for spark plugs in 1930. From the twentieth century to present day, necessary components of the modern spark plug are same (Figure 1.1): two electrode, the anode is connected to the ground by the metal casing of engine block and separated by a ceramic insulator from the central cathode, which connects to the pulse generator that is powered from ignition coils of the engine (Ortiz et al., 2013).

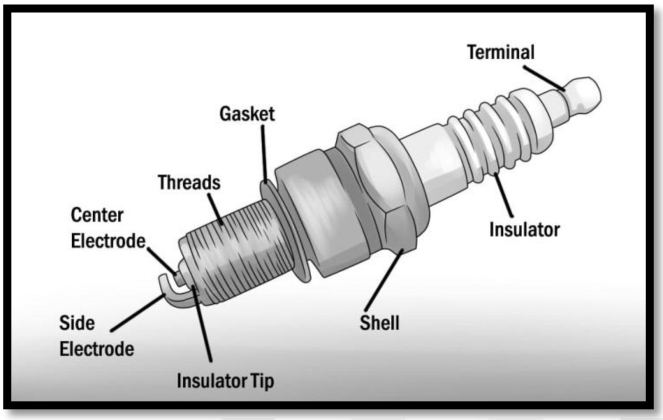

Figure 1.1. Diagram of a typical spark plug

Spark plug is one of the most vital components in a car with spark ignited engine. A spark plug is not only responsible for providing reliable combustion initiation but also play a critical role for obtaining optimum engine performance and stable operation(‘Spark plugs - Technology and developments’, n.d.).

The purpose of a spark plug is to ensure controlled combustion of the fuel in the engine. To do so a high voltage generated by the ignition coil is introduced into the combustion chamber and the compressed fuel/air mixture is then ignited by the electric spark passing between the electrodes. Charge volume and density of the spark plug in combustion chamber is another critical factor that influences engine performance and emission characteristic. With increasing combustion volume, charge density or elongating ignition time, combustion completeness can be enhanced. However, there are some obstacles to improve those features. For instance, in order to increase charge density ignition system requires higher secondary coil voltage to initiate combustion in a spark ignition engine, which is using conventional spark ignition system. However, producing the required voltage with these conditions would cause to spark electrode erosion short service lifetime of the spark plug (‘NGK - Requirements’, n.d.).

Central electrode of the spark plug (Figure 1.1) needs to perform in the temperature range 350-700°C (572-1292°F) in order to attain adequate performance. When electrode is too hot, pre-ignition and knock may come over. On the other hand, if the electrode temperature is too low, carbon deposits can build up on the central insulator that causes electrical breakdown. Moreover, shape of the electrodes determines operation temperature range of a spark plug and the heat flows from the central electrode through the ceramic insulator. A cool-operating gasoline engine needs a hot spark plug having a long heat flow path in the central electrode. As to a hot-operating engine, like a high-performance application or an engine with a high-compression-ratio, needs a cool spark plug. A cool spark plug constructed with much shorter heat flow path for a cool spark plug. The spark plug needs to run with a voltage of 5-15 kV in order to create good spark. The greater electrode gap size of the plug and higher in cylinder pressure means that greater voltage is required (Stone and Ball, 2004).

The voltage and above mentioned spark plug properties that produce the spark highly depends on factors such as combustion chamber pressure during ignition, distance between the electrodes (gap size), cylinder gas temperature, spark location, electrode's geometry, materials that used in construction of spark plug and manufacturing methods (Robert Bosch GmbH., 2004). Beside, by operating multiple spark plugs for each cylinder or employing the ignition point at an optimum place inside the combustion chamber could lead reduction in flame travel path and develop the combustion. Yet, it is quite difficult to apply multiple spark plugs in multi cylinder engines, because already overcrowded cylinder head by exhaust and intake valves (Jung and Iida, 2018).

In order to improve spark plugs, manufacturers have presented to the market different types of spark plugs with various electrode materials such as yttrium, iridium, platinum etc. Main purpose of usage of these noble metal materials, based on desires of rapid attainment of operating temperature, increasing cold starting reliability, enhancing ignition, smooth engine operation and reducing wear rate of electrode. Nevertheless, these types of spark plug that are available in market must be evaluated from different aspects such as performance characteristics, emission values, electrode erosions and carbon buildup etc (‘How spark plug is made - material, making, history, used, parts, dimensions, composition, industry, machine, History’, n.d.)..

On the other hand, fuels such as CNG, LPG, ethanol, methanol, bio fuels, hydrogen and hydroxyl have been investigating for many years regarding different points in order to decrease dependence on gasoline. Amongst, hydrogen is one of the most promising alternative fuel which can be benefited in various ways such as producing electricity for EVs (Ahmadi et al., 2018; Brooks et al., 2018; DeCicco and Li, 2016; Hames et al., 2018), direct usage (Salanki and Wallace, 1996; Wakayama et al., 2006) or hybrid utilisation (Amrouche, Erickson, Varnhagen, et al., 2016; Arat and Surer, 2017; Di Iorio et al., 2016; Du et al., 2016, 2017; Fan et al., 2015; Y. Karagöz et al., 2015; Kim et al., 2017; Su et al., 2017; Wu et al., 2016) for both spark and compression ignition engines. However, hybrid usage seems more promising because some issues. Firstly, hydrogen fuel is well-known for their miss fire and abnormal combustion boosting features because of their easy-combustible characteristic. Moreover, hydrogen storage is rather challenging since its small molecular structure lead to hydrogen leakage which is very unsafe and may cause fatal accidents. In order to eliminate leakage, very thick and strong materials are used for storage (Brooks et al., 2018). However, this brings another problem; over-weight which reduces benefits of hydrogen usage and even creates a space requirement for a large hydrogen tank. In addition to these, hydrogen usage in an engine solely restricts the mileage of the vehicle because of hydrogen’s lower volumetric energy content than gasoline fuel (Wakayama et al., 2006). Consequently, when taking into consideration these, hydrogen usage in a dual-fuel mode is more reasonable.

On the other hand, hydrogen usage brings a lot of advantages due to its some superior properties. Hydrogen is a colourless, inodorous and zero carbon emitted fuel when fired with oxygen. The chemical reaction of two hydrogen and an oxygen atoms generate energy outputs and product of the reaction is water only. Characteristic features of hydrogen like flammability limits, low ignition energy, high burning rate ensure more stable combustion process and engine operation even for ultra-clean air-fuel mixtures. So, hydrogen enables to increase the combustion limits. Hydrogen has wider flammability range in air which allow for engine operation with either rich or lean mixtures. Thus, by leaning the air-fuel mixture better fuel economy can be attainable because of increased combustion completeness of the fuel. In addition to that, lean combustion results in lower NO_x emission because of lower combustion temperature (Çelebi et al., 2017).

The quenching distance of hydrogen is 0.6 mm while the gasoline has 2 mm. Comparing with gasoline, it is more difficult quenching hydrogen flame so that it may cause misfire and abnormal combustions such as backfire, knock etc. (Shinagawa et al., 2004). In order to alleviate this problem, spark timing retarding, injection timing delaying, leaning the air-fuel mixture or hydrogen enrichment with some proportion that is identified according to engine characteristic are main solutions.

When stoichiometric conditions are employed, hydrogen fuelled engine approaches ideal cycle thermodynamically more, due to high flame velocity of hydrogen. The properties such as high flame velocity and adiabatic flame temperature ameliorate the engine parameters in term of combustion stability, thermal efficiency, emissions etc. Thank to fast burning features of hydrogen the engine may operate within high speed range that results in higher power output in case of sacrificing the lean operation (Ji et al., 2016).

Ignition energy is the minimum energy required to initiate combustion of the fuel which is very low for hydrogen comparing to gasoline. Minimum ignition energy of the hydrogen-air mixture is 0.02 MJ while gasoline-air mixture needs 0.24 MJ (Srinivasan and Subramanian, 2014). For this reason, hot spots and gases in combustion chamber could be potential for undesirable ignitions that result in misfire for hydrogen usage in the gasoline engine. Because of their lower minimum ignition energies, sources like even a resistance, hot wire or glow plug can initiate the combustion process. Nevertheless, hydrogen fuel cannot be combusted without a spark plug or in other word with only compression temperature rise of gasoline engine. Because, auto-ignition temperature of hydrogen as high as 585 °C. As a result of that it is difficult to ignite hydrogen fuel with increment of heat only, so an external ignition source is inevitably indispensable. In addition to that, auto ignition temperature is the most important factor to acquire maximum compression ratio of an engine, inasmuch as temperature rise depends only on compression ratio of an engine throughout compression process. On the other hand, high auto-ignition temperature of hydrogen gain advantage of greater compression ratio that can be applied for hydrogen fuelled engine comparing with gasoline engine. As a result, ignition performance of the hydrogen fuelled hydrogen engine must be improved from in terms of energy analysis, abnormal combustion, emissions, storage etc. (Jaber et al., 2012).

Hydrogen is quite diffusive and disperses into air faster than gasoline. In this regard, hydrogen is superior in two points; mixture formation is excellent and if any leakage occurs, hydrogen disperses into the air quickly which is essential for preventing unsafe conditions.

As a result, hydrogen utilisation can be applicable in without major content of modifications in the existing engine system. Employment of hydrogen in the SI engine solely constraints the driving mileage of the vehicle. Moreover, larger space is occupied by hydrogen tank in case of sole usage of hydrogen. On the other hand, hydrogen implementation with dual fuel mode is more feasible with this respect. Moreover, wider flammability limit of the hydrogen also stabilizes engine operation, reduces engine noise and vibrations and cyclic variations. In addition to these, hydrogen existence in the combustion chamber could make a significant enhancement in performance parameters. Because of higher calorific value, flame speed and diffusivity of the hydrogen, air fuel can premix more homogenous and completeness of combustion can be increased with hydrogen blended gasoline engine. More complete combustion ensured by hydrogen enrichment lead to reduction of the emission products. Zero carbon structure of the hydrogen combustion decreases CO, CO_2 and HC emissions. On the contrary, hydrogen existence in combustion rises the cylinder temperature with stoichiometric conditions, thus NO_x emission increases. However, NO_x emission rise can be compensated by the lean operation owing to diminished combustion temperature (Amrouche, Erickson, Park, et al., 2016; Wang et al., 2016).

On the other hand, compression ratio is one of the core elements that influence the engine performance and design. Engine efficiency can be improved further with a higher compression ratio that enables increment in the expansion ratio allow effective utilisation of the fuel. Furthermore, increment of the compression ratio yields decreased exhaust gas dilution of fresh mixture which increases cylinder temperature and pressure. This allows shorter ignition delay and combustion duration as a consequence of increased flame speed. Also, compression ratio increment of an engine is functional way to attain wider backfire free running range of the engine. Besides, increasing compression ratio minimises residual gas amount in the combustion chamber and hereby reduces the possibility of backfire (Salvi and Subramanian, 2016).

From those points of views, although there are studies on spark plug locations, spark plug gap, erosion and failure characteristics and alteration of spark plugs in literature the effect of usage of different types of spark plugs in gasoline engine on performance and emission characteristics has not been investigated in detail. In this study, the effect of different spark plug types on performance and emissions characteristics of a spark ignition engine fuelled with hydrogen enriched gasoline at various engine loads and compression ratios were focused. Brake power (BP), brake specific fuel consumption (BSFC), brake specific emission values of carbon monoxide (CO), carbon dioxide (CO_2), nitric oxides (NO_x) and unburnt hydrocarbon (UH) emissions were determined to evaluate test conditions.

2. PRELIMINARY WORK

2.1. Hydrogen (Alternative Fuels) Literature Review

Within last decades, there have been many developments for alternative fuel technology because of strict emission legislations and increasing energy demand. Rising petroleum prices, harmful effects of exhausted emissions from engines to the environment and global warming issue must be focused intensively international content (Ji et al., 2016).

Scientists have been researched alternatives to petroleum fuels. Hydrogen usage is one of the most important alternatives to decrease fossil fuel dependence.

The total amount of hydrogen existing on the Earth is undeterminable and quite small since its low density suppress the gravitational force of the planet exerting on the hydrogen molecules. Hence, hydrogen must be produced by other existing compounds such as natural gas, oil, coal and water. Hydrogen is an energy source which requires external energy to produce it such as fossil fuels: gasoline, diesel, biodiesel, ethanol, methanol, CNG, LPG, nuclear and renewable energy like solar, wind, geothermal and hydroelectric power. Hydrogen production can be conducted mainly in 3 way: natural gas reforming, gasification, electrolysis (McAllister et al., 2011). Hydrogen utilisation for internal combustion engine can weaken the dependence on fossil fuels. Nevertheless, in recent years, approximately 95% of the total hydrogen generation is carried out from fossil fuels used methods and only a minor ratio is from renewable and relatively clean resources such as geothermal, biomass, wind and solar energy. For this reason, the hydrogen production is still costly and high pollutant process. The main issue is whether hydrogen can be generated economically and to a large content by clean and renewable sources (Dimitriou and Tsujimura, 2017).

Ji et al., (2010) investigated hydrogen-gasoline dual fuel mode under varied loads and excess air ratios for combustion and emission characteristics. They applied %0 and %3 of hydrogen volume fractions, 1.2 and 1.4 excess air ratios and various manifold absolute pressures representing the engine load to their test engine. They reported that the engine BMEP was increased after hydrogen blending at low load conditions. However, hydrogen-gasoline engine operated with smaller BMEP than only gasoline fuelled engine at high loads. The BTHE of the engine was raised significantly with the increment of MAP for both the gasoline and hydrogen-gasoline blended operation. Cyclic variations in IMEP (COV_{imep}) for the HHGE were dropped with the increase of engine load. The hydrogen addition was effective on enhancing gasoline engine instability at low load and lean conditions. HC and CO emissions were reduced whereas NO_x emissions were raised with the increment of engine load. The hydrogen enrichment was more effective on engine combustion and emissions parameters at low MAPs than at high MAPs (Ji et al., 2010).

In their another investigation, Ji et al., (2016) carried an investigation out to evaluate hydrogen addition on combustion and emission performance of a Wankel engine at partial load and stoichiometric conditions. The engine was operated at 3000 rpm engine speed, with MAP of 37.5 kPa, the stoichiometric air-fuel ratio and spark timing 25ºCA BTDC. The results depicted that the combustion pressure, BMEP, cylinder temperature and BTHE were raised with the hydrogen addition. Moreover, the addition of hydrogen was effective on reduction of flame propagation and development periods and increasing combustion completeness. HC emissions were decreased by 44.8% when the hydrogen enrichment was executed. Beside, CO and CO_2 emissions were dimished with the hydrogen addition because of reduced flame propagation and development periods and carbon-free hydrogen fuel (Ji et al., 2016).

Sopena et al., (2010) conducted modifications to convert the an spark ignited gasoline engine of a Volkswagen Polo 1.4 to operate it with hydrogen. They reported that results of hydrogen fueled engine are within the expected values: hydrogen fueled engine was capable of giving a brake torque of 63 Nm at 3800 rpm and maximum brake power of 32 kW at 5000 rpm. The BTHE of the hydrogen engine was higher than gasoline-fueled engine except for excess air ratios greater than 1.8. On the other hand, the BTHE reduced while excess air ratio dropped from 3 to 1.6. Performance of the hydrogen fueled engine was adequate for powering the Volkswagen Polo 1.4. They also stated that hydrogen fueled vehicle was enable to reach a maximum speed of 140 km/h in horizontal and straight line trajectory. Their modified engine also provided safe operation of the hydrogen-fueled engine without missfire such as a knock, backfire and preignition as well as with acceptable low NO_x emissions. (Sopena et al., 2010).

Greenwood et al., (2014) investigated effects of hydrogen addition to ethanol at ultra-lean operation conditions. They utilised a 2-cylinders, 0.745 L SI engine that can operate on both hydrogen and ethanol fuels. They studied at part throttle and fixed RPM operation with 0%, 15%, and 30% of hydrogen volume fractions and in ultra-LOL. Their results has shown that running the engine near the LOL with hydrogen enrichment decreased engine NO_x emissions out by more than 95% as compared with stoichiometric gasoline operation. Power output, brake thermal efficiency and volumetric efficiency were not influenced by the hydrogen enrichment at a certain equivalence ratio. Nevertheless, hydrogen induction provided an increment in the LOL leaded further reduction in NO_x emissions, but also decreased power and BTHE (Greenwood et al., 2014).

Hao et al., (2016) modified a spark ignition gasoline engine to a hydrogen-gasoline dual fuel engine by addition a hydrogen injection system aiming to decrease cold start emissions of HC and CO. The engine started operation with hydrogen fuel and then changed to run on gasoline with hydrogen enrichment or on only gasoline operation to prevent high NO_x emissions. They stated that the cold start emissions of HC and CO were significantly minimised when the engine run on hydrogen-gasoline mode. Nevertheless, the NO_x emissions increased because of the higher combustion speed and risen cylinder temperature of hydrogen (Hao et al., 2016).

Ji and Wang (2010) rettrofitted a 1.6 L port fuel injection SI engine to a hybrid hydrogen–gasoline engine fuelled with the hydrogen–gasoline blend by assembling an electronically controlled hydrogen injection system on intake manifolds while keeping the original injection system of the gasoline fuel unmodified. They adapted an engine speed 1400 RPM and a manifolds absolute pressure of 61.5 kPa to investigate the performance of an hybrid hydrogen gasoline engine at lean burn limits. Hydrogen were induced by 1%, 3% and 4.5% of volume fractions in the total intake gas for investigations. They reduced the gasoline flow rate until the engine operated the lean operation limit that COV_{imep} was 10%. They reported that COV_{imep} at the same excess air ratio was drastically decreased with the increment of hydrogen volume fractions. The excess air ratio of hybrid hydrogen gasoline engine at the LOL was extended by 4.5% comparing with gasoline engine. The engine BTHE, CO, HC and NO_x emissions at lean operation limit were enhanced with hybrid hydrogen gasoline (Ji and Wang, 2010).

Shi et al., (2017) studied effect of spark timing on performance and emissions characteristics of a hybrid hydrogen-gasoline engine. They implemented engine speed of 1500 rpm and excess air ratio at 1.2. They found that when the hydrogen volume fraction was increased from 0% to 10% and the hydrogen injection timing was 110 ºCA BTDC a good stratified effect was attained. They varied spark timing from 4 to 16 ºCA BTDC. This study depicted that the brake thermal efficiency was increased at first and thereafter dropped with the increment of the spark advance angle. Beside this, BTHE enhanced with the increment of hydrogen volume fraction. Delayed spark timing leaded to rise the maximum instantaneous heat release rate. Moreover, when spark advance angle was increased to peak cylinder pressure and the maximum rate of pressure increased, too. Delayed the spark timing also enabled reduction of NO_x, HC and CO emissions. However, NO_x emissions increased and HC and CO emissions decreased due to increment of the hydrogen volume fraction according to their experimental results the (Shi et al., 2017).

Karagöz et al., (2015) used a conventional electrolyser in order to generate hydrogen and oxygen simultaneously and induced to a 1.1 litter Peugeot SI engine. Main purpose of this study was to eliminate the requirement of a gas storage device. They performed all tests under idle condition and hydrogen + oxygen gas mixture were utilised as additional fuels. Hydrogen energy fractions were varied such as 0%, 5%, 8%, 10% and 15%. Their test results have demonstrated that, COV_{rpm}, COV_{imep}, energy flow rate, indicated thermal efficiency, indicated specific fuel consumption, peak in-cylinder temperature, THC and CO were enhanced with the increment of hydrogen energy fraction. Nevertheless, the dramatic increase of NO_x emissions was observed with increment of hydrogen and oxygen amount (Yasin Karagöz et al., 2015).

Amrouche et al., (2016) carried out an experimental study to determine effects of hydrogen addition on Lean Operation Limit (LOL) of a gasoline fuelled rotary (Wankel) engine. They retrofitted a mono rotor rotary engine running at 3000 rpm engine speed and wide open throttle to operate on gasoline-hydrogen dual fuel mode. They varied hydrogen energy fractions as 0%, 3% and 6% of in the intake manifold with excess air ratios from stoichiometric ratio to the engine lean operation limit. According to their test results, hydrogen addition to gasoline extended lean operation limit of the original engine while enhancing stability of the rotary engine. With hydrogen enrichment, increased brake thermal efficiency and shorter burn duration, reduction in brake specific energy consumption, and HC, CO and CO_2 emissions were acquired. Moreover, the extending of the engine lean operation limit with hydrogen enrichment reduced the NO_x emissions levels of the gasoline rotary engine to very low amounts without adapting a 3-way catalytic converter. They also pointed out that a Wankel engine is more resistant to combustion abnormalities such as misfire, detonation, knock etc. when hydrogen is utilised as a combusted fuel due to its three physically separated combustion rooms. This physical separation allows occurrence of each combustion phase in separate rooms. More clearly, it means hot spots like spark plug and exhaust port or valve cannot come into contact with the room that is on intake phase. (Amrouche et al., 2016)

Zang et al., (2015) investigated effects of spark timing on combustion and emissions performances of a hydrogen – methanol fueled test engine. They conducted the experiment on a four-cylinder gasoline engine equipped with an electronic controlled hydrogen port-injection system. They adapted engine speed at 1400 rpm and the MAP at 61.5 kPa with an excess air ratio of 1.20 and hydrogen volume fractions of 0%, 1.5% and 3% in the total intake gas. The spark timings for each hydrogen volume fraction were altered from 18 to 46 ºCA BTDC with the increment of 2 ºCA. The obtained experimental data showed that the indicated thermal efficiency was risen at first and then dropped with the increment of spark advance. By increasing of spark advance, flame development duration was extended while the flame propagation was shortened. The COV_{imep} was first reduced and then a little increment with the advance of spark timing was observed. Reduction in HC and CO emissions were attained by hydrogen enrichmend. They also reported that NO_x emissions out from the hydrogen-gasoline engines can be diminished by the way of retarded the spark timing (Zhang et al., 2015).

Dhyani and Subramanian (2018) conducted an experimental investigation on a multi-cylinder SI engine fueled with hydrogen in order to evaluate its effects of knock on backfire and control of misfire by changing operation parameters such as:, injection timing, spark timing and equivalence ratio. They determined that retarding spark timing and delaying injection timing of the hydrogen fueled engine could decrease potential combustion abnormalites significantly at high equivalence ratio. Backfire and knock are related each other for the hydrogen fueled spark ignited test engine at high equivalence ratio (Dhyani and Subramanian, 2018).

Ravi et al., (2017) studied effects of compression ratio on a LPG fuelled lean burn spark ignited engine in order to to evaluate performance, emission and combustion characteristics at partial throttle position. As a result, their experimental results have shown that hydrogen enrichment improved the combustion and heat release rate, minimised cyclic variations, and extended the lean operation limit of the test engine. They obtained notable enhancement in BTHE, brake power and significant drops in HC, CO and CO_2 emissions. However, because of ignition timing retarding, the NO_x emission rises slightly (Ravi et al., 2017).

2.2. Variable Compression Ratio Literature Review

Following the 70's oil crisis, researchers focused on the issue of alternative fuels for internal combustion engines which are generated from renewable sources, low cost and less harmful to the environment compared with fossil fuels. However, in real world, the engine efficiencies also depend on other parameters. Hence, researchers attracted their interest on engine parameters such as spark timing and compression ratio. Performance and emission characteristics are highly depend on such kind of important parameters of spark ignited engines. Thus, there are many investigations on the effects of these parameters by using variable compression ratio (VCR) engine. Experimental investigations on engines running at different compression ratios become critical.

Raj Saxena and Kumar Maurya (2017) performed a series of experiments to assess the effects of fuel premixing ratio, injection timing and engine compression ratio on the soot particle emissions in nano-size from compression ignition test engine. They stated that the peak of particle number concentration was greater for lower compression ratios implemented and peak of the particle number concentration turned into smaller sized particles at higher compression ratios (Raj Saxena and Kumar Maurya, 2017).

Serin and Yıldızhan (2017) carried out an investigation to evaluate performance and emission characteristics of diesel, biodiesel, and small amount of alcohol blends fuelled variable compression ratio diesel engine. Their study has shown that increment of the compression ratio enhanced performance characteristics: BTHE and BSFC of the test engine for all used fuels. Nevertheless, increment of the engine compression ratio leaded to rise in NO_x and CO_2 emissions

(Serin and Yıldızhan, 2017).

Bhasker and Porpatham (2017) made investigation examining effect of higher compression ratio on engine performance, emission generation and combustion parameters in CNG fuelled spark ignited engine in order to extend lean operation limit with hydrogen enrichment at engine speed of 1500 rpm. Three different compression ratios: 10.5:1, 11.5:1 and 12.5:1 were implemented and the effect of hydrogen enrichment was investigated with 5% and 10% hydrogen energy fractions on the optimized 12.5:1 compression ratio at WOT condition. They pointed out that the brake power and BTHE improved slightly with increment of compression ratio and maximum BTHE of 30.2% was obtained at higher compression ratio of 12.5:1 and equivalence ratio of 0.7. The lean operation limit of CNG combustion was also extended to an equivalence ratio of 0.5 with increasing compression ratio to 12.5:1. Moreover, increasing the hydrogen energy fraction to 10% improved the combustion rate further and lean operation limit to an equivalence ratio of 0.42 (Bhasker and Porpatham, 2017).

Zhen et al. (2013) studied knock in a spark ignited (SI) methanol fuelled engine which had a high compression ratio by using multi-dimensional simulation. Knocked combustion under different engine operation conditions were simulated by them and the effects of spark timing, EGR (Exhaust Gas Recirculation) techniques, mixture concentration and combustion chamber shape for preventing knock in a high compression ratio spark ignited methanol engine were simulated. According to their simulation results, engine knock could not be fully prevented only by postponed spark timing due to the high compression ratio, they stated that in order to suppress knock it was inevitable to implement other methods (Zhen et al., 2013).

Salvi and Subramanian (2016) performed an experimental study on hydrogen fuelled spark ignition generator set by converting the generator from single cylinder, air cooled gasoline fuelled generator set with power of 2.1 kVA at 3000 rpm. They conducted the study at various compression ratios: 4.5:1, 6.5:1 and

7.2:1, spark timings: 2-20 ºCA BTDC and exhaust gas recirculation up to 25% by volume. Their experimental results demonstrated that running the engine at higher compression ratio increased the brake thermal efficiency and reduced the backfire occurrence as well as residual gas fraction minimised with the increment if compression ratio. Nevertheless, NO_x emission levels have risen with higher compression ratio. They run the engine with retarded spark timings and various EGR percentages in order to minimise the NO_x emission at source level. According to the test results, retarded spark time method is not logical option for reducing NO_x emission without sacrificing on power output in hydrogen fuelled spark ignited engine (Salvi and Subramanian, 2016).

Thomas et al., (2016) conducted an experimental study in order to determine effects on varied compression ratio in a single-cylinder carburetted SI engine at different loads with two different fuels on engine parameters such as brake thermal efficiency exhaust gas temperature and exhaust emissions. Their experiments were performed at three different compression ratios of 7:1, 8.5:1 and 10:1. They utilised pure gasoline and 20% n-butanol blend (B20) in gasoline fuels in the investigation. As a result, they found out that brake thermal efficiency improved with CR at all loads.The study has also shown that the usage of alcohol fuels such as n-butanol has increased the engine performance especially at higher compression ratios. (Thomas et al., 2016).

Yıldızhan et al., (2017) investigated fuel properties, performance and emission characteristics in a variable compression ratio diesel engine fuelled with waste cooking oil (WCO) based biodiesel. They represented that compression ratio increment ameliorated brake thermal efficiency and specific fuel consumption values and increased exhaust gas temperature for all blended fuels. Higher compression ratio reduced CO emissions while it caused to rise in CO_2 emissions and NO_x emissions (Yıldızhan et al., 2017).

2.3. Ignition System Studies

Due to lower thermal efficiency of stoichiometric burn which is the main combustion mode for spark ignited engines, these engines suffer from ensuring improved fuel economy and lower emissions. To attain stable, more complete and fast combustion characteristic leading to enhanced lower energy consumption and lower emission out, successful inflammation is vital that is creating a flame kernel that remain its laminar state and transitions into fully developed turbulent ignition. Hence, flame properties highly depend on ignition system and its development. In order to improve combustion quality there were many studies performed by modifying ignition system.

Javan et al., (2012) measured spark plug temperature in SI engine's various operation conditions with two kinds of fuels: CNG and gasoline fuels. Their experimental results indicated that, center electrode temperature is lower than ground electrode and maximum temperature difference between them is about 110 C^o at 2500 rpm engine speed and full load conditions. Spark plug's maximum temperature was seen with compressed natural gas under full load conditions and at 6380 rpm. They stated that temperature of ground electrode reached about 960 C^o which is very prone to pre-ignition. Morever, temperature of center electrode is 195 C^o higher than under same condition of gasoline combustion which leaded greater electrode erosion rate according to their investigation. Due to temperature rise they suggested cold type spark plug selection because of its better heat transfer. They also observed spark plug erosion after performing tests with CNG as a fuel. In these tests, they reported electrodes with non uniform wear patterns and consequently gap growth was not uniform, too. Besides, the growth of average gap for two sets of spark plugs after two similar 200 hour performance tests is 49.6% (Javan et al., 2012).

Jung et al., (2017) examined effects of spark discharge energy and in-cylinder turbulence level on lean operation limits and cyclic variations of combustion for spark ignited engine operation in order improve the thermal efficiency by extending the lean-stability limit. They varied the spark discharge energy by a high-energy inductive ignition system via ten spark coils and enhanced the in-cylinder turbulence level by a custom adapter installed in the intake port. According to their test results, increment of spark discharge energy by ten spark coils was effective on extending the lean stability limit to leaner operation comparing with that of a single spark coil. With the extension of lean-stability limit, they reported a considerable improvement of thermal efficiency comparing to stoichiometric operation. Moreover, a combination of boosting in-cylinder turbulence level and increment of spark discharge energy makes it possible to attain stable operation with more extended lean-stability limit. Despite indicated thermal efficiency is amelioriated by increment of spark discharge energy, it did not effect variations of the combustion, since total spark discharge energy does not affect both durations of the spark timing-to-CA5 and the CA10-to-CA90 and eventually the heat-release efficiency (Jung et al., 2017).

Lin et al., (2005) performed a study in order to determine corrosion and erosion mechanisms of spark plug that used in natural gas engine and examine the primary life limiting processes during field operation. They reported that analysis a strong calcium signal was appeared on spark plugs via optical emission spectroscopic and significant formation of calcium-enriched glassy oxide phases were seen on the electrode surfaces according to scanning electron microscopy. Beside, intergranular cracking was exposed in the subsurface region of both iridium and platinum–tungsten alloy electrode insert tips. The subsequent growth of these cracks could speed up the wear of the spark plug electrodes and shorten the lifetime of the spark plugs. (Lin et al., 2005).

Ceper (2012) carried out an experimental investigation to examine performance and emissions of a commercial hydrogen fueled spark ignition engine operating at partial (50% WOT) and full wide open throttle (100% WOT) positions. She utilized a four-stroke, 6-cylinder, 4.9 L, port fuel injection and hydrogen fueled SI engine with a compression ratio of 13.5:1. She adobted 3 different spark plug gaps of 0.4, 0.6 and 0.8 mm, engine speeds of 1000-3000 rpm and two ignition timing of 10º and 15º CA BTDC. It was stated that spark plug gas is a key factor effecting the performance of the engine depending on the engine parameters such as compression ratio,volume, ignition system etc.. She obtained maximum power values at 0.6 mm spark plug gap for both 50% and 100% WOT at ignition timing of 10º and 15º CA BTDC. At 50% WOT, maximum efficiency was observed with 0.8 mm spark plug gap. the maximum efficiency value was obtained with a 0.6mm spark plug at ignition timing values of 10º and 15º CA BTDC at 100% WOT position. A significant decrease in NO_x emissions was attained by using hydrogen for all WOT and spark plug gaps (Ceper, 2012).

Badawi et al., (2017) carried out optical and thermal tests in order to evaluate effects of spark plug electrode gap on flame kernel growth, engine performance and emissions characrteristics of a constant volume combustion chamber and single cylinder gasoline direct injection engine. They used high speed schlieren visualization method in order to investigate the flame kernel development at different equivalence ratios. They determined that when the spark plug gap was increased the flame kernel development area increased, too. Planar laser-induced imaging for the combustion process inside engine cylinder has shown that a larger flame kernel directly related with the larger spark plug gap. If spark plug gap increased, maximum cylinder pressure, turbulent flame speed, heat release rate and the mass fraction burned improved according to test results. The engine power output increased marginally and more stable combustion process was attained by means of decreased cyclic variations as the gap increased. The engine emitted minimum HC emissions and particulate number concentration with the maximum spark plug gap. However, NO_x emissions were increased as the gap became wider depending on the higher temperature resulted from the increament in flame speed and in-cylinder pressure (Badawy et al., 2017).

Jung and Iida (2018) investigated the effects of multiple spark discharge on lean SI operation for improving thermal efficiency of SI engine by using the multi-coil ignition system featuring ten spark coils for a single spark plug. According to their test results, the indicated thermal efficiency enhanced almost linearly with increasing λ for all multiple spark discharges. Morever, due to combination of the advanced spark timing and the advanced initial combustion phase obtained by multiple spark discharge especially with spark discharge time interval of 2 ms, they achieved more advanced of main combustion phase and shorter the combustion duration. Due to shorter combustion duration, more stable ultra-lean operation at around λ=1.94 could be attained with the highest indicated thermal efficiency of 47.0% by them. (Jung and Iida, 2018).

Rodrigues Filho et al. (2016) designed a prototype with stratified torch ignition engine from a commercial baseline engine. In the engine, they initiated combustion process in a pre-combustion chamber and pressure increment in this chamber pushed the combustion jet flames through a calibrated nozzle into the main combustion chamber. These combustion jet flames were introduced with high kinetic and thermal energy in order to obtain a stable lean combustion process. According to their experimental test results, considerable increment in the torch ignition engine's performance were attained in comparison with the baseline engine. Moreover, torch ignition enabled to enhance engine efficiency and obtain remarkable reduction in exhaust emissions (Rodrigues Filho et al., 2016).

Altın and Bilgin (2009) adapted a spark ignited engine cycle model to investigate the effects of spark plug location on twin-spark plug spark ignited engine performance. They located spark plugs diametrically opposite to each other on cylinder head axisymmetrically for the twin-spark arrangement. They considered five dimensionless distance from the cylinder center to spark plug location on cylinder head (rsd = 0, 0.25, 0.50, 0.75, and 1.0). They stated that spark plug location of rsd = 0 corresponds to the single - spark arrangement which spark plug is located at the center. In order to compare results, they also considered single-spark plug arrangements for other selected spark plug locations. As a result of their study, they found that central single-plug arrangement demonstrated the best engine performance and lowest fuel consumption. However, the results have shown that if central located of spark plug is not possible to use due to some design constraints, twin-spark plug configurations gave better performance than single-non central arrangements. (Altın and Bilgin, 2009).

On the other hand, laser ignition is emerging and very promising strong concept for alternative ignition in spark ignition engines. Laser ignition is more advantageous over conventional spark plug ignition from some aspects. Laser ignition is a electrode-free ignition system that there is no spark energy loss to the electrodes, also this eliminate erosion effects from the ignition system. Furthermore, laser ignition allows to choose spark location and it enables superp performance under high in-cylinder pressures. There have been considerable research for the sake of evaluating of laser ignition's positive and negative sides and improving and adabtion this superior ignition technology in different types of internal combustion engines.

Srivastava and Agarwal (2014) evaluated performances of laser ignition and conventional spark ignition systems regarding cyclic in-cylinder pressure variation, combustion stability, fuel consumption, rate of heat release, power output and exhaust emissions at similar engine operating conditions. They conducted the comparison in a single cylinder engine for various of excess air ratios (λ) and ignition timings. They found that maximum cylinder pressure and rate of heat release were significantly higher for laser ignition comparing with spark ignition for same λ and ignition timing. While λ decreased, difference in maximum cylinder pressure between spark ignition and laser ignition decreased. For laser ignition, higher maximum cylinder pressure and rate of heat release rate were seen due to earlier start of combustion. They also depicted that combustion advanced for laser ignition in the range of 1–4º CA comparing with spark ignition. Lower COV_{IMEP} in laser ignition was obtained compared to spark ignition. Also, they attained higher brake power for laser ignition compared to spark ignition. They attributed these results to more efficient combustion for the laser ignition. Lower brake specific fuel consumption was measured for laser ignition up to $\lambda = 1.10$. After this excess air ratio, brake specific fuel consumption was observed virtually same as that of spark ignition. Brake specific NO_x emissions were measured to be higher for laser ignition comparing with spark ignition up to $\lambda = 1.15$. For further reduction in λ, the study stated that NO_x emission values were almost identical for spark ignition and laser ignition. In addition, brake specific total HC and brake specific CO emissions were slightly lower for laser ignition over spark ignition (Srivastava and Agarwal, 2014).

Endo et al., (2017) investigated ability of laser ignition by ignition of lean-fuel propane–air mixture in a constant volume chamber by a spark inducing with a pair of electrodes and a spark driver, spark inducing with focused high-power laser, or a spark inducing with focused high-power laser between a pair of dummy electrodes, where the deposited energy into the gas was the same in all cases. In their experiments, they altered mole fraction of the fuel in the mixture and obtained test data of pressure on the inner wall of the chamber, ignition probability, and the multi-frame high-speed schlieren images of the ignition process. According to their experimental results, the ignition performance of a laser-induced spark was better than conventional electrical spark in near the lean operation limits, and laser-ignition was the lack of heat loss to electrodes and also a large initial flame kernel whose effective energy was enhanced by rapid heat release from the combustible mixture sucked into the kernel by a non-spherical inward flow created by a laser ignition. Besides, they pointed out that due to the increment of the total absorbed energy, reinjection of the transmitted laser light into plasma empowered the ignition of the laser. (Endo et al., 2017)

Pal and Agarwal (2016) developed a prototype hydrogen fuelled engine with port fuel injection and they implemented laser ignition system in the test engine and tried to reduce the laser pulse energy and optimise it for best engine performance in order to develop a practical laser ignition system. A Q-switched Nd:YAG laser with wavelength of 1064 nm and pulse duration of 6-9 ns was adapted for laser ignition of hydrogen air mixture in the test engine. They used two different laser pulse energies (16.6 mJ/ pulse and 12.1 mJ/pulse) for the study and the effect of variation of laser pulse energy on combustion, performance and emissions characteristics of the laser ignited hydrogen fuelled engine was examined. They found out peak in-cylinder pressure, rate of pressure rise and rate of heat release were higher for greater laser pulse energy of the laser ignited hydrogen fuelled test engine. They obtained considerably advanced start of combustion for greater energy laser pulse although the combustion duration was comparatively shorter. Brake thermal efficiency and brake specific fuel consumption were improved for laser ignition with greater energy pulse. Additionally, exhaust gas temperature for the greater energy laser pulse was relatively higher at all loads that speeded up NO_x formation. Other emissions out such as unburned hydrocarbons, CO and CO_2 were reduced slightly, however this reduction seemed insignificant according to their study (Pal and Agarwal, 2016).

Xu et al., (2014) investigated minimum ignition energy (MIE) and ignition probability of premixed gasoline–air mixture for different equivalence ratios by using laser ignition with a nanosecond pulse at 532 and 1064 nm from a Q-switched Nd:YAG laser in a constant-volume combustion chamber. They compared the test result with the spark ignition. Their experimental findings showed that within the flammable range, the ignition probability increased when the ignition energy increased and the distribution of minimum ignition energy with the equivalence ratios was reported as U-shape for both laser and spark ignition. When the incident energy is higher than 110mJ or the absorbed energy is high than 31 mJ, 100% of ignition could be obtained within equivalence ratios of 0.8–1.6 for laser ignition with 532 nm. For 1064 nm pulse, it was 235mJ and 30 mJ. In order to obtain the same ignition probability of gasoline–air mixture with identical equivalence ratio, the 1064 nm incident energy is twice more than 532 nm incident energy, while the absorbed energy values were almost the same. The measured MIE with laser ignition was greater than that measured by electrical sparks according to their test results. They measured MIE for laser ignition at equivalence ratio of 1.0 at 532 nm and 1064 nm pulses were 13.5 mJ and 9.5 mJ, respectively. However, the MIE was 3.76 mJ with equivalence ratio of 1.6 for spark ignition. They also added that laser ignition could extend the lean operation limit from 0.8 to 0.6 for their test engine (Xu et al., 2014).

3. MATERIAL AND METHOD

The all experimental studies were conducted in Engine Laboratory of the Department of Automotive Engineering at Cukurova University.

3.1. Material

3.1.1. Engine Test Rig

Engine tests were conducted on a four stroke, single cylinder, naturally aspirated, water cooled, variable compression engine that can operate with both gasoline and diesel fuel by replacing the engine head. Technical specifications of the engine were given in Table 3.1 VCR spark ignited engine test rig, compression ratio adjustment tools and schematic representation test rig were shown in Figure 3.1 and Figure 3.2, respectively.

Table 3.1. Technical specifications of the engine

Brand – Model	Kirloskar Oil Engines-240
Configuration	Single Cylinder
Type	Four Stroke,
Cooling	Water Cooling
Displacement	661 cc
Bore	87.5 mm
Stroke	110 mm
Minimum / Maximum Operating Speed	1200/1800rpm
Power	4.5 Kw @ 1800 rpm
CR range	6:1-10:1
Injection Variation	0-25º BTDC
Peak Pressure	77.5 kg/cm^2
Weight	160 kg
Lubricating System	Forced Feed System

Figure 3.1. VCR spark ignited engine test rig

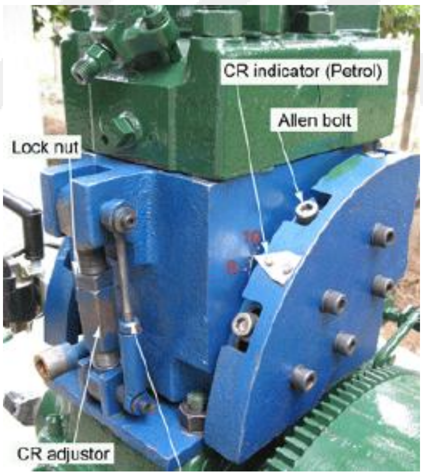

Figure 3.2. Compression ratio adjustment

In these experiments, an eddy current dynamometer and a Sa-Beam load cell was used for determination of torque and power output. The AG Series eddy current dynamometers designed for the testing of engines up to 400kW (536bhp) and may be used with various control systems. The dynamometer is bi-directional. The shaft mounted finger type rotor runs in a dry gap. A closed circuit type cooling system permits for a sump. Dynamometer load measurement is from a strain gauge load cell and speed measurement is from a shaft mounted three hundred sixty PPR rotary encoder. Technical specifications of the dynamometer and load cell are given in Table 3.2 and Table 3.3.

Table 3.2. Technical specifications of the dynamometer

Model	AG10
Make	Saj Test Plant Pvt. Ltd.
End flanges both side	Cardan shaft model 1260 type A
Water inlet	1.6 bar
Torque	11.5 Nm
Hot coil voltage max	60
Continuous current amps	5.0
Speed max.	10000 rpm
Load	3.5 kg
Weight	130 kg

Table 3.3. Technical specifications of the load cell

Model	60001
Type	S Beam Universal
Capacity	0-50 kg
Non-linearity	<+/-0.025%
Non-repeatability	<+/-0.010%
Operating temperature range	-20 ^{0}C to +70^{0}C
Combined Error	<+/-0.025%

MRU Air Delta 1600 V mobile exhaust gas analyzer was used to measure exhaust emissions (Table 3.4). Emission data was collected with the help of analyzer software. Accuracy of the gas analyzer is ±10 ppm for CO, 1% for CO_2 and ±1 ppm for NO_x.

Table 3.4. Measurement ranges, accuracy and resolution of emission device

CO	0-10%
CO_2	0-20%
HC	0-20000 ppm
O_2	0-22%
NO	0-4000 ppm
NO_2	0-1000 ppm
Lambda	0-9.99
Accuracy	According to OIML-class 1
Ambient Temperature	+5^o - +45 ^{o}C
Exhaust Gas Temperature	Max 650 ^{o}C

3.1.2. Hydrogen Enrichment Set-up

During engine experiments, H_2 which has 99.99% purity was added through intake manifold of the engine in order to ameliorate performance and emissions characteristics of the spark ignited test engine. In Figure 3.3, schematic representation hydrogen enriched test engine rig was shown in detail.

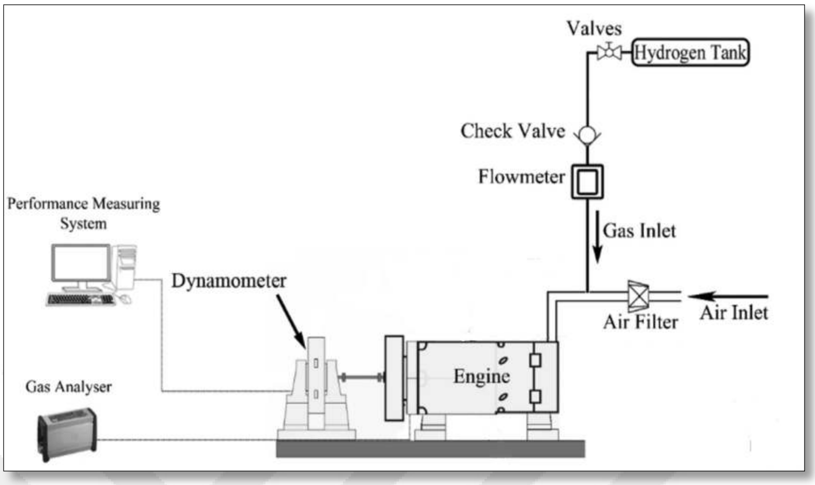

Figure 3.3. Schematic representation hydrogen enriched test engine rig

Equipments which were used in the hydrogen enrichment and their functions were explained below:

Dynamometer: System was connected to AG10 Eddy current dynamometer

Gas Analyzer: MRU Delta 1600 V gas analyzer was used to measure the exhaust gas emission values. This equipment was connected to host computer and using software of MRU.

Performance measuring system: Performance data were obtained with specific software of MRU via host computer.

Air Filter: Air filter was used to enhance air quality of the engine.

Flow meter: Gas flows were measured and set by Alicat volumetric flow meters which are located to sections before mixing chamber and after in-cylinder.

Hydrogen Regulator: was used to set flow motion of gas. Hydrogen fuel supplying to engine was adjusted to test results in order to obtain more effective results.

Hydrogen Tank: (170 bars pressurized) was filled out with 99.99% Hydrogen.

Flashback arrestor: a special gas safety device was used to stop the flame or reverse flow of gas back up into the equipment or supply line and it prevents the user and equipment from damage or explosions.

Data Logger: Outputs were transferred to computer system by data logger.

3.1.3. Spark Plugs

The ignition system on petrol-driven engines – in contrast to diesel engines – is external: during the compression cycle the combustion of the compressed fuel-air mixture is triggered by an electrical spark produced by the spark plug. It is the task of the spark plug to generate this spark. Created by the high voltage produced by the ignition coil, it leaps between the electrodes. A flame front spreads from the spark and fills the combustion chamber until the mixture has been burned. The heat released increases the temperature, there is a rapid build-up of pressure in the cylinder and the piston is forced downwards (Power stroke). The movement is transferred via the connecting rod to the crankshaft; this drives the vehicle via the clutch, the gears and the axles.

In order for the engine to operate smoothly, powerfully and in an environmentally friendly manner, a number of requirements have to be met: the correct amount of perfectly balanced fuel/ air mixture must be present in the cylinder, and the high-energy ignition spark must leap between the electrodes precisely at the predetermined moment. For this purpose spark plugs have to meet the highest performance requirements: they must deliver a powerful ignition spark between around 500 and 3500 times a minute (in 4-stroke operation) - even during hours of driving at high revs or in stop-and-go traffic conditions. Even at -20 °C, they have to ensure a completely reliable ignition. High-tech spark plugs provide low-emission combustion and optimum fuel efficiency – without misfiring, which can cause unburnt fuel to get into the catalytic converter, and destroying it.

In this study, three different spark plugs which are available commercially were used in order to evaluate their effects on performance and emission characteristics of a spark ignited-hydrogen blended engine. Technical specifications of copper (conventional), iridium and platinum spark plugs used in these experiments were given Table 3.5, Table 3.6 and Table 3.7, respectively.

Table 3.5. Technical specifications of copper spark plug used in experiments

Band	*BOSCH*
Parts No.	*UR3DC*
SHELL	
Thread Size	*10 mm*
Thread Pitch	*1.0 mm*
Seat Type	*Gasket*
Resistor	*Yes*
Resistor Value	*5K Ohm*
Reach	*19 mm (3.4")*
Hex Size	*16 mm (5.8")*
Terminal Type	*Threaded Stud*
Overall Height	*ISO*
Spark plug gap	*0.7 mm*
SPECIFICATIONS	
Heat Range	*8*
Longevity	*30k*
CENTER ELECTRODE	
Material	*Copper*
Type	*Fine wire*
Size	*2.5 mm*
Projection	*Projected*
GROUND ELECTRODE	
Material	*Copper*
Type	*Standard*
Shape	*Taper-Cut*
Quantity	*1*

Table 3.6. Technical specifications of iridium spark plug used in experiments

Band	*NGK*
Parts No.	*CR9EIX*
SHELL	
Thread Size	*10 mm*
Thread Pitch	*1.0 mm*
Seat Type	*Gasket*
Resistor	*Yes*
Resistor Value	*5K Ohm*
Reach	*19 mm (3.4”)*
Hex Size	*16 mm (5.8”)*
Terminal Type	*Threaded Stud*
Overall Height	*ISO*
Spark plug gap	*0.8 mm*
SPECIFICATIONS	
Heat Range	*8*
Longevity	*40-50k*
CENTER ELECTRODE	
Material	*Iridium*
Type	*Fine wire*
Size	*0.6mm*
Projection	*Non-Projected*
GROUND ELECTRODE	
Material	*Nickel*
Type	*Standard*
Shape	*Taper-Cut*
Quantity	*1*

Table 3.7. Technical specifications of platinum spark plug used in experiments

Band	*NGK*
Parts No.	*CR9EHVX-9*
SHELL	
Thread Size	*10 mm*
Thread Pitch	*1.0 mm*
Seat Type	*Gasket*
Resistor	*Yes*
Resistor Value	*5K Ohm*
Reach	*19 mm (3.4")*
Hex Size	*16 mm (5.8")*
Terminal Type	*Threaded Stud*
Overall Height	*Bantam*
Spark plug gap	*0.7 mm*
SPECIFICATIONS	
Heat Range	*8*
Longevity	*50k*
CENTER ELECTRODE	
Material	*Platinum*
Type	*Fine wire*
Size	*0.6 mm*
Projection	*Non-Projected*
GROUND ELECTRODE	
Material	*Platinum*
Type	*Standard*
Shape	*Taper-Cut*
Quantity	*1*

3.2. Methods

3.2.1. Hydrogen Enrichment

Hydrogen is characterized by the highest mass energy density and the highest stoichiometric air/fuel ratio of any other fuel. Hence, hydrogen addition may reduce the BSFC of the spark ignited engine. Moreover, hydrogen has a wider flammability range, a low ignition energy and high adiabatic flame temperature and diffusion rates that could help to improve the homogeneity of the fuel-air mixture, enhance the flame stability and reduce the quenching effect in the SI engine (Amrouche et al., 2014). Therefore, hydrogen addition should be helpful to reduce unburned hydrocarbons emissions and increase thermal efficiency of an internal combustion engine. In Table 3.8, physical and chemical properties of hydrogen and gasoline fuels were shown.

Table 3.8. Comparative physical and chemical properties of hydrogen and gasoline [Data derived from (Negurescu et al., 2012)

Property		Gasoline	Hydrogen
Molecular Mass [kg/kmol]		114	2.016
Theoretical air-fuel ratio [kg/kg]		14.5	32.32
Density t 0 ºC and 760 mmHg [kg/m^3]		0.735 - 0.760	0.0899
Flammability limits in air at 20 ºC and 760mmHg	% vol.	1.48 - 2.3	4.1 - 75.6
	λ	1.1 - 0.709	10.12 - 0.136
Flame velocity in air (λ=1), at 20 ºC and 760 mm Hg		0.12	2.37
Octane Number		90 - 98	>130
Min. Ignition energy in air [mJ]		0.2 - 0.3	0.018
Auto-ignition temperature [K]		753 - 823	848 - 853
Lower Heating Value (gas at 0 ºC and 760 mmHg and stoichiometric fuel-air mixture)	[kJ/m^3]	3661	3178
	[kJ/kg]	42690	119600

In these experiments, performance and emission parameters were tried to be enhanced by hydrogen injection with different flow rate into the combustion chamber. In addition, hydrogen usage with different modern spark plug were performed under different dynamometer loads and compression ratios to assess if it is useful in practice Three different hydrogen amounts were applied with a ratio of 0 L/min, 2 L/min and 4 L/min throughout the experiments.

3.2.2. Variable Compression Ratio (VCR) Test Engine

Compression ratio is a critical parameter for internal combustion engines. Since CR directly changes the combustion characteristics of an engine, CR should be chosen as high as possible. A high compression ratio is desirable because it allows an engine to extract more mechanical energy from a given mass of air-fuel mixture due to its higher thermal efficiency. This occurs because internal combustion engines are heat engines, and higher efficiency is created because higher compression ratios permit the same combustion temperature to be reached with less fuel, while giving a longer expansion cycle, creating more mechanical power output and lowering the exhaust temperature. It may be more helpful to think of it as an "expansion ratio", since more expansion reduces the temperature of the exhaust gases, and therefore the energy wasted to the atmosphere.

Higher compression ratios will however make gasoline engines subject to engine knocking (also known as detonation) if lower octane-rated fuel is used. This can reduce efficiency or damage the engine if knock sensors are not present to modify the ignition timing (Glassman et al., 2008).

The thermodynamic cycle that describes the SI engine is the Otto cycle. Thus, thermodynamic efficiency of a SI engine under idealized conditions (standard air assumption) is given by:

$$\eta_{th} = 1 - \frac{1}{CR^{\gamma-1}} \qquad (3.1.)$$

where γ is the ratio of specific heats c_p/c_v, and CR is the compression ratio, V_{max}/V_{min}. It is interesting that the thermal efficiency depends only on compression ratio and γ. Higher compression ratios lead to higher flame temperatures and therefore one and γ anticipates an increase in thermodynamic efficiency (McAllister et al., 2011).

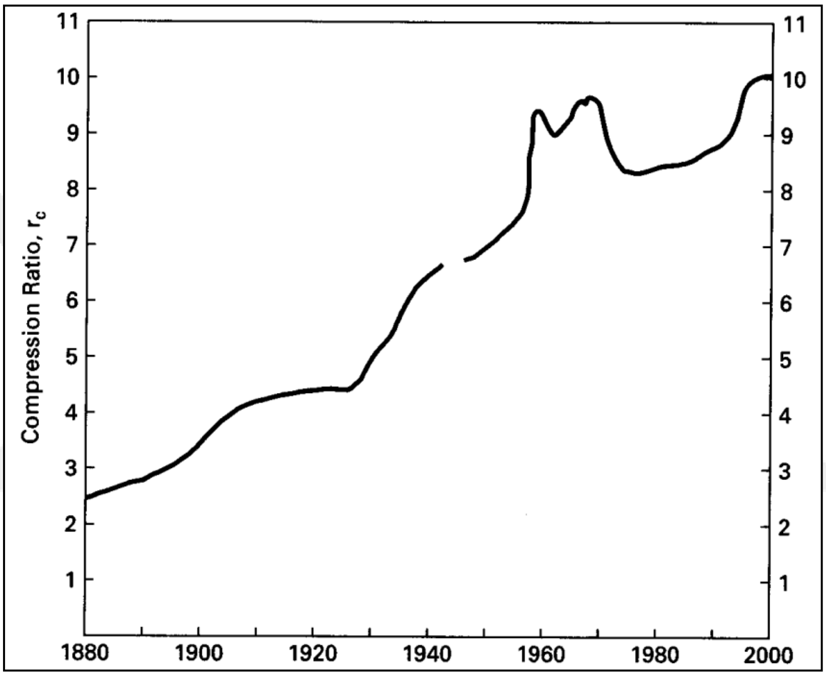

Figure 3.4. Average compression ratio of American spark ignition automobile engines as a function of year. Adapted from (Charles A., 1990)

Various attempts have been made to develop engines with a variable compression ratio. One such system uses a split piston that expands due to changing hydraulic pressure caused by engine speed and load. Some two-stroke cycle engines have been built which have a sleeve-type valve that changes the slot opening on the exhaust port. The position where the exhaust port is fully closed can be adjusted by several degrees of engine rotation. This changes the effective compression ratio of the engine.

Four strokes, single cylinder, multi-fuel variable compression ratio research engine equipped with an eddy current type dynamometer operated during the experiments. The operation mode of the experimental engine can be changed from petrol to diesel or from diesel to petrol with some necessary equipment changes. For the both diesel and petrol modes, the compression ratio of the cylinder can be changed without ceasing the engine and changing the geometry of combustion chamber by tilting cylinder block arrangement.

An engine with fixed compression ratio can be modified to VCR by modifying with additional variable combustion space. There are some different arrangements which can provide this process. Tilting cylinder block method is one of these arrangements where the compression ratio can be varied without changing combustion geometry. By the way of this method, the compression ratio can be changed within designed range without ceasing the engine (Figure 3.5).

In these experiments, 8.5:1 and 10:1 compression ratios were performed for each fuel and spark plug combination.

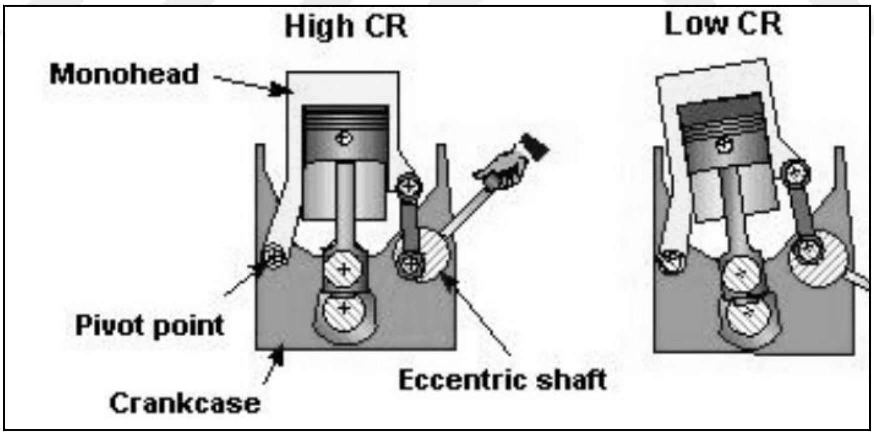

Figure 3.5. Tilting cylinder block arrangement

Experimental test rig is equipped with necessary instruments for measurement of measurement of fuel flow, interface airflow, measurements of temperatures and load which is interfaced with computer. The setup includes a stand-alone panel box consisting of two fuel tanks for duel fuel test, air box, transmitters for air, fuel measuring unit, manometer and measurements of fuel flow, indicators of process and hardware interface. Rotameters are included for water cooling and also calorimeter for measurement of water flow. The setup enables to investigate of VCR engine performance for indicated power, brake power, frictional power, brake mean effective pressure, indicated mean effective pressure, indicated thermal efficiency, brake thermal efficiency, volumetric efficiency, mechanical efficiency, specific fuel consumption, air/fuel ratio and heat balance. Labview based Engine Performance Analysis software package “Enginesoft” is included for on line performance analysis (Kirloskar Oil Engines Instruction Manual, 2010).

3.2.3. Combustion in Gasoline Engines

In order to describe an engine’s combustion process some terms should be defined clearly:

- Engine cylinder diameter (bore) (D): The nominal inner diameter of the working cylinder.
- Piston area (A): The area of a circle of diameter equal to engine cylinder diameter (bore). $A = \pi / 4 \times D^2$
- Engine stroke length (L): The nominal distance through which a working piston moves between two successive reversals of its direction of motion.
- Dead center: The position of the working piston and the moving parts, which are mechanically connected to it at the moment when the direction of the piston motion is reversed (at either end point of the stroke).
- Bottom dead center (BDC): Dead center when the piston’s position is at nearest to the crankshaft.
- Top dead center (TDC): Dead center when the position is farthest from the crankshaft.
- Swept volume ($_{Vs}$): The nominal volume which is generated by the piston

working when travelling from bottom dead center to top dead center, calculated from the product of piston area and stroke. The capacity described by engine manufacturers in cc is the swept volume of the engine. $V_s = A \times L = \pi / 4 \times D^2 \times L$

- Clearance volume (V_c): The nominal volume of the space on the combustion side of the piston at top dead center. Cylinder volume: The sum of swept volume and clearance volume. $V=V_s+V_c$
- Compression ratio (CR): The numerical ratio of the smallest volume to highest volume, calculated by cylinder volume divided by the numerical value of clearance volume. $CR = V / V_c$

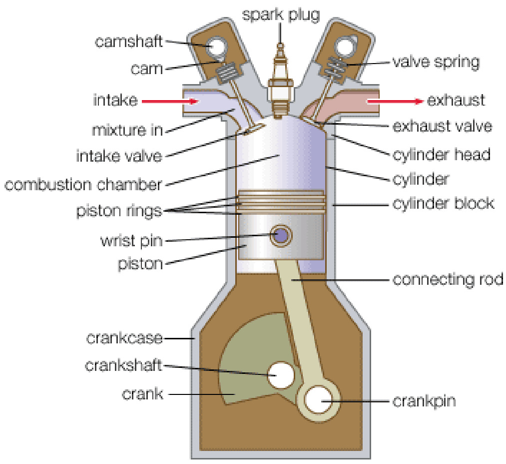

Figure 3.6. Typical piston cylinder arrangement of an SI engine

In four-stroke cycle engine, a full cycle of operation is completed in four strokes of the piston or two revolutions of the crankshaft. Each stroke of crankshaft consists of 180° rotation and thus a full cycle of crankshaft rotation consists of 720° (Kirloskar Oil Engines Manual, 2010). The series of operation of an ideal four-stroke engine are as follows:

- **The induction stroke**: The inlet valve is open, and the piston travels down the cylinder, drawing in a charge of air-fuel mixture.
- **The compression stroke:** Both valves are closed, and the piston travels up the cylinder to compress and heat up the air-fuel mixture.
- **The expansion, power, or working stroke:** Combustion propagates throughout the charge, raising the pressure and temperature, and forcing the piston downward. At the end of the power stroke, as the piston approaches bottom dead center (TDC), the exhaust valve opens, and the irreversible expansion of the exhaust gases is termed "blow-down."
- **The exhaust stroke:** The exhaust valve remains open, and the piston travels up the cylinder and expels most of the remaining gases. At the end of the exhaust stroke, when the exhaust valve closes, some exhaust gas residuals will remain. These will dilute the next charge.

Indicated thermal efficiency (η_t): Indicated thermal efficiency is the ratio of energy in the indicated power to the fuel energy.

$$\eta_t = \frac{\text{Indicated Power (kW)} \times 3600}{\text{Fuel Flow (kg/hr)} \times \text{Calorific Value (KJ/kg)}} \times 100 \quad (3.2)$$

- **Brake thermal efficiency (η_{th}):** A measure of overall efficiency of the engine is given by the brake thermal efficiency. Brake thermal efficiency is the ratio of energy in the brake power to the fuel energy.

$$\eta_t = \frac{\text{Brake Power (kW)} \times 3600}{\text{Fuel Flow (kg/hr)} \times \text{Calorific Value (KJ/kg)}} \times 100 \quad (3.3)$$

- **Mechanical efficiency (η_m):** Mechanical efficiency is the ratio of brake horse power (delivered power) to the indicated horsepower (power provided to the piston).

$$(\eta_m) = Brake\ Power\ /\ Indicated\ Power \tag{3.4}$$

$$Frictional\ Power = Indicated\ Power - Brake\ Power \tag{3.5}$$

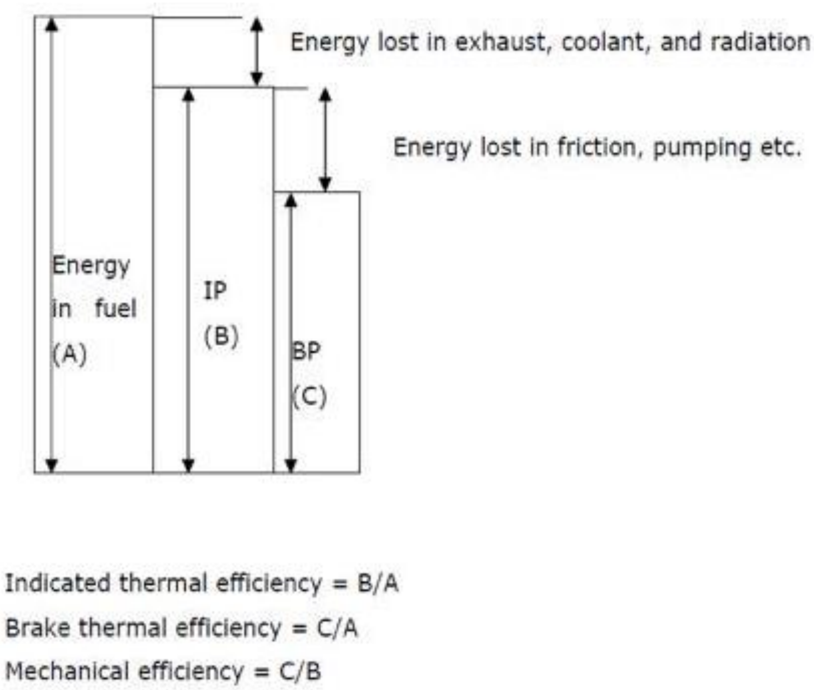

Figure 3.7. The difference between reversible work and actual useful work (*Kirloskar Oil Engines Instruction Manual*, 2010)

- **Volumetric efficiency (η_v):** The engine output is limited by the maximum amount of air that can be taken in during the suction stroke, because only a certain amount of fuel can be burned effectively with a given quantity of air. Volumetric efficiency is an indication of the 'breathing' ability of the engine and is defined as the ratio of the air actually induced at ambient conditions to the swept volume of the engine. In practice the engine does not induce a complete cylinder full of air on each stroke, and it is convenient to define volumetric efficiency as:

$$(\eta v) \; = \; Actual\ air\ inducted \ / \ Theoretical\ air \tag{3.6}$$

$$\eta_v(\%) = \frac{\text{Air Flow (Kg/hr)}}{[\pi/4 \times D^2 L\ (m^3)] \times \left[\frac{N(rpm)}{n}\right] \times \text{N.of Cycle} \times \rho_{air}(kg/m^3) \times 60} \times 100 \tag{3.7}$$

Where n= 1 for 2 stroke engine and n= 2 for 4 stroke engine.

- **Specific fuel consumption (SFC):** Brake specific fuel consumption and indicated specific fuel consumption, abbreviated BSFC and ISFC, are the fuel consumptions on the basis of Brake power and Indicated power respectively.
- **Fuel-air (F/A) or air-fuel (A/F) ratio:** The relative proportions of the fuel and air in the engine are very important from standpoint of combustion and efficiency of the engine. This is expressed either as the ratio of the mass of the fuel to that of the air or vice versa.
- **Calorific value or Heating value or Heat of combustion:** It is the energy released per unit amount of the fuel, when the combustible is burned and the products of combustion are cooled back to the initial temperature of combustible mixture. The heating value so obtained is called the higher or gross calorific value of the fuel. The lower or net calorific value is the heat released when water in the products of combustion is not condensed and remains in the vapour form.
- Power and Mechanical efficiency: Power is defined as rate of doing work and equal to the product of force and linear velocity or the product of torque and angular velocity. Thus, the measurement of power involves the measurement of force (or torque) as well as speed. The power developed by an engine at the output shaft is called brake power and is given by;

$$Power\ (kW) \; = \frac{N\ (rpm) \times T(Nm)}{60000} \tag{3.8}$$

$$T = W \times R \tag{3.9}$$

Where W = 9.81 × Net mass applied (kg) and R= Radius (m).

The combustion process of SI engines can be divided into three broad regions: ignition and flame development, flame propagation, and flame termination. Flame development is generally considered the consumption of the first 5% of the air-fuel mixture (some sources use the first 10%). During the flame development period, ignition occurs and the combustion process starts, but very little pressure rise is noticeable and little or no useful work is produced (Figure 3.8). Just about all useful work produced in an engine cycle is the result of the flame propagation period of the combustion process. This is the period when the bulk of the fuel and air mass is burned (i.e., 80-90%, depending on how defined). During this time, pressure in the cylinder is greatly increased, and this provides the force to produce work in the expansion stroke. The final 5% (some sources use 10%) of the air-fuel mass which burns is classified as flame termination. During this time, pressure quickly decreases and combustion stops.

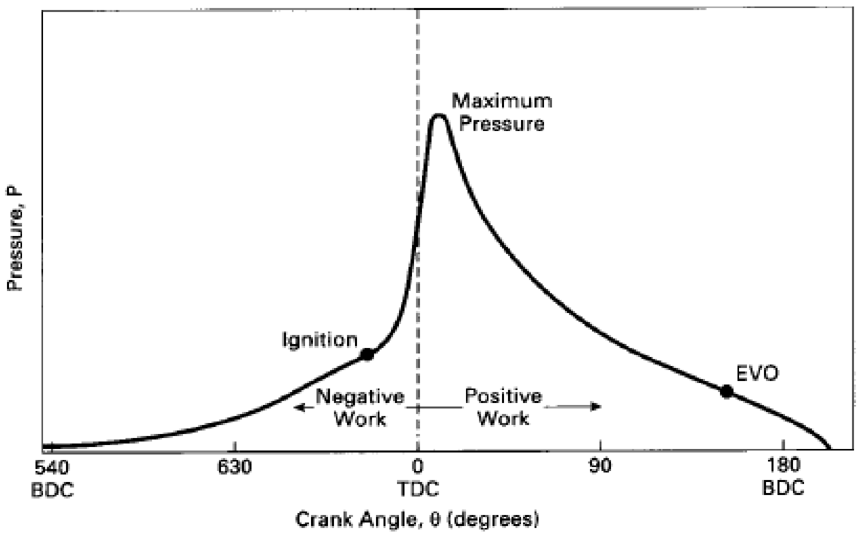

Figure 3.8. Variance of pressure inside the combustion chamber [Adapted from (Pulkrabek, 2013)]

In an SI engine, combustion ideally consists of an exothermic subsonic flame progressing through a premixed homogeneous air-fuel mixture. The spread of the flame front is greatly increased by induced turbulence, swirl, and squish within the cylinder. The right combination of fuel and operating characteristics is such that knock is avoided or almost avoided.

On the other hand, when a four-stroke cycle SI engine is run at less than WOT conditions, air-fuel input is reduced by partially closing the throttle (butterfly valve) in the intake system. This creates a flow restriction and consequent pressure drop in the incoming air. Fuel input is then also reduced to match the reduction of air. Lower pressure in the intake manifold during the intake stroke and the resulting lower pressure in the cylinder at the start of the compression stroke are shown in Figure 3.9. Although the air experiences an expansion cooling because of the pressure drop across the throttle valve, the temperature of the air entering the cylinders is about the same as at WOT because it first flows through the hot intake manifold.

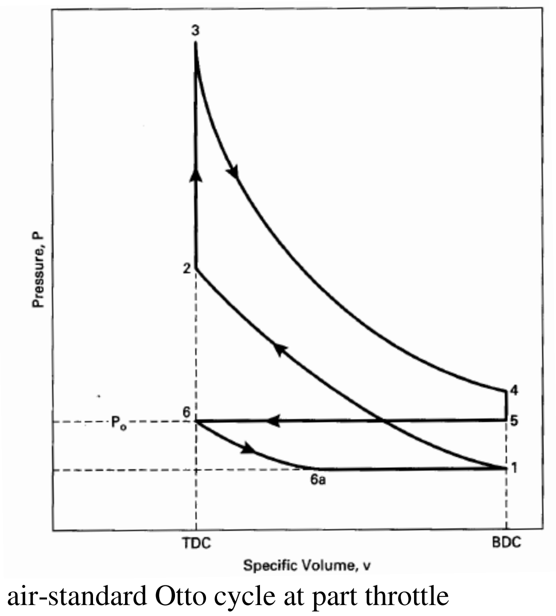

Figure 3.9 Four stroke air-standard Otto cycle at part throttle

3.2.4. Experimental Procedures

Engine performance and emission measurement experiments were executed with two different compression ratios (8.5:1 and 10:1) under %50 part throttle condition. During the experiments engine loads of 8 Nm, 13 Nm and 17 Nm were applied in order to assess effects of spark plugs and hydrogen usage under different engine loads. Copper (conventional), iridium and platinum spark plugs were tested respectively for each experiment condition. In addition, hydrogen was blended through the intake manifold with flow rates of 0 lit/min (H0), 2 lit/min (H2) and 4 lit/min (H4) to enhance combustion of VCR engine.

Before experimental measurements engine was run 5 minutes to attain stable operation conditions. "Enginesoft" software was used for logging experimental data. This software logs for 60 seconds when logging starts and at the end of this period gives the average values of experimental data.

As a result of these experiments, brake power (BP), specific fuel consumption (BSFC), brake specific emissions of; carbon monoxide (CO), carbon dioxide (CO_2), nitric oxides (NO_x), total unburned hydrocarbon (UHC) were obtained to evaluate effects of applied methods on performance and emission characteristics of a four stroke, naturally aspirated, water cooled and spark ignited test engine.

4. RESULTS AND DISCUSSIONS

4.1. Performance Characteristics

Brake power (BP) and brake specific fuel consumption (BSFC) defined as power output from flywheel or crankshaft of the engine is one of the most critical performance criteria for internal combustion engines. Figure 4.1-4.6 shows the brake power output results of the experiments and Figure 4.7-4.12 illustrates the brake specific fuel consumption (BSFC) values with different compression ratios (8.5:1 and 10:1) and engine loads (8 Nm, 13 Nm and 17 Nm).

As can be seen from graphs, usage of iridium and platinum spark plugs improved BP and BSFC values at all engine loads. However, platinum spark plug enhanced performance parameters more than iridium spark plug. The improvement of these spark plugs can be attributed that increased charge density in combustion chamber enhanced flame propagation. For iridium spark plug, between 2.27% - 11.43% and for platinum spark plug between 4.76% - 16.67% increments in BP were obtained. In addition, BSFC values reduced with iridium and platinum spark plugs between 3.45% - 16.67% and 6.45% - 19.05%, respectively. In general, performance enhancement depending on spark plugs and hydrogen enrichment was slightly lower at higher engine loads.

On the other hand, hydrogen induction into the combustion chamber expedited performance improvement, remarkably. Superior chemical and physical properties of hydrogen fuel such as high diffusion and flame speed boosted combustion completeness of the test engine. Nevertheless, as mentioned above, at higher engine loads, performance improvement due to hydrogen injection were less comparing to lower loads. Table 4.1-4.2 shows the average variance in performance parameters with hydrogen induction at different engine loads.

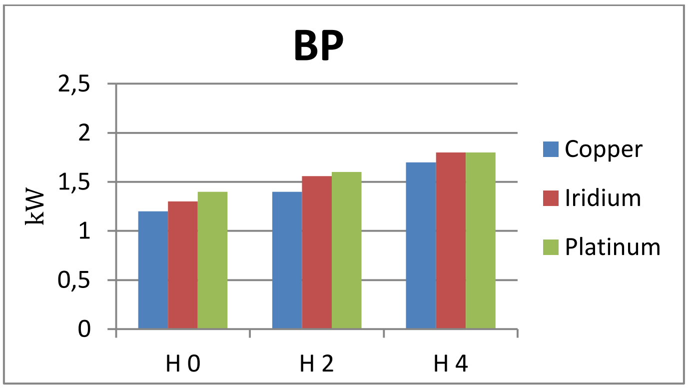

Figure 4.1. BP values for engine load of 8 Nm and CR of 8.5:1

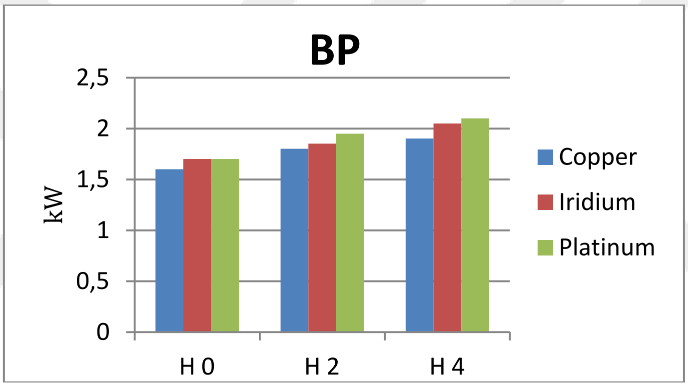

Figure 4.2. BP values for engine load of 13 Nm and CR of 8.5:1

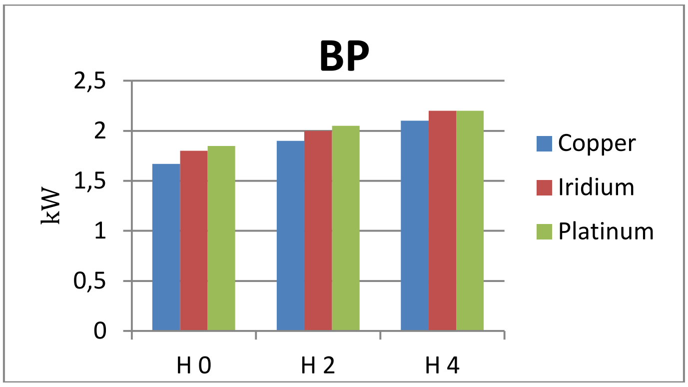

Figure 4.3. BP values for engine load of 17 Nm and CR of 8.5:1

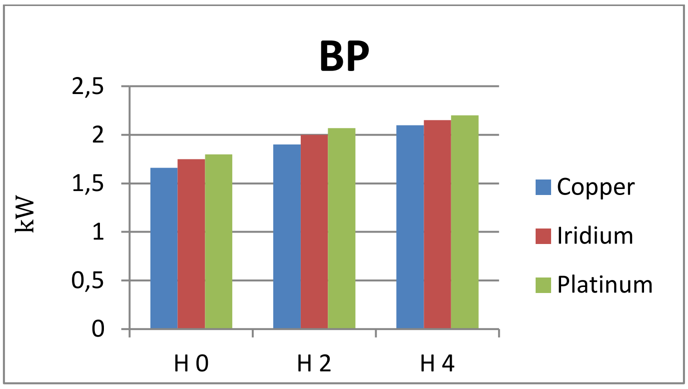

Figure 4.4. BP values for engine load of 8 Nm and CR of 10:1

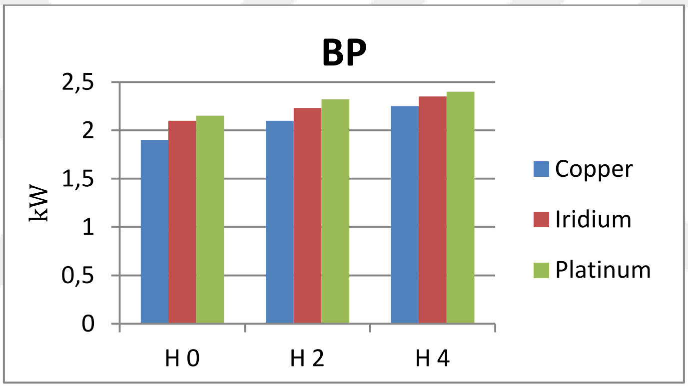

Figure 4.5. BP values for engine load of 13 Nm and CR of 10:1

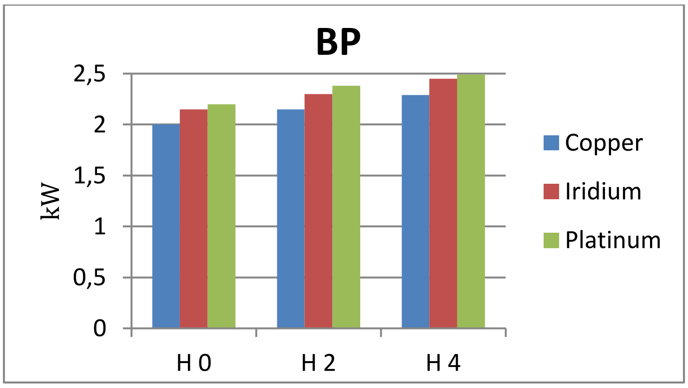

Figure 4.6. BP values for engine load of 17 Nm and CR of 10:1

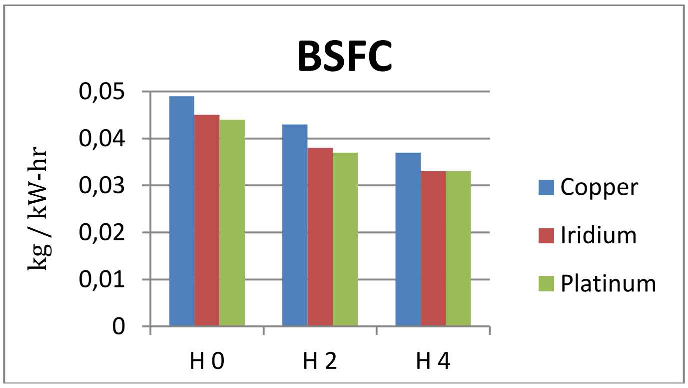

Figure 4.7. BSFC for engine load of 8 Nm and CR of 8.5:1

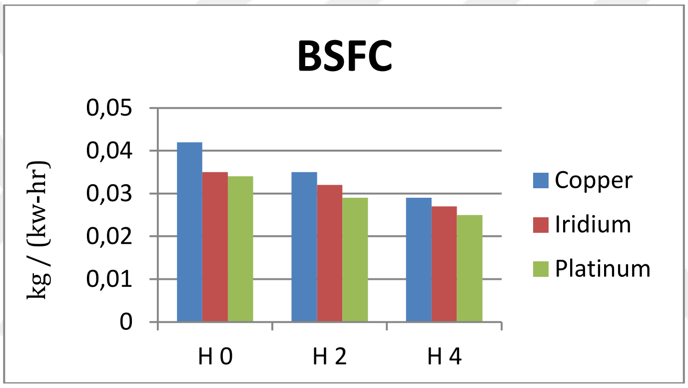

Figure 4.8. BSFC for engine load of 13 Nm and CR of 8.5:1

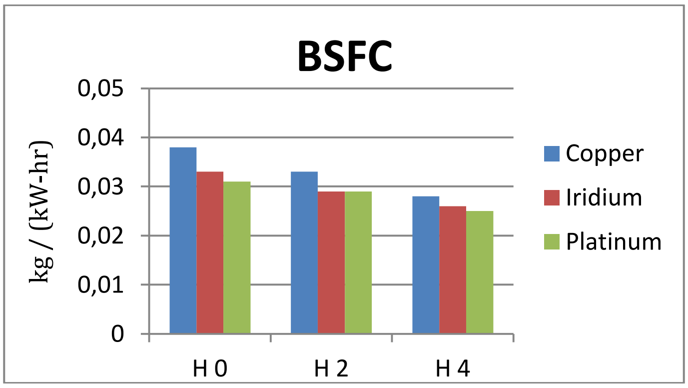

Figure 4.9. BSFC for engine load of 17 Nm and CR of 8.5:1

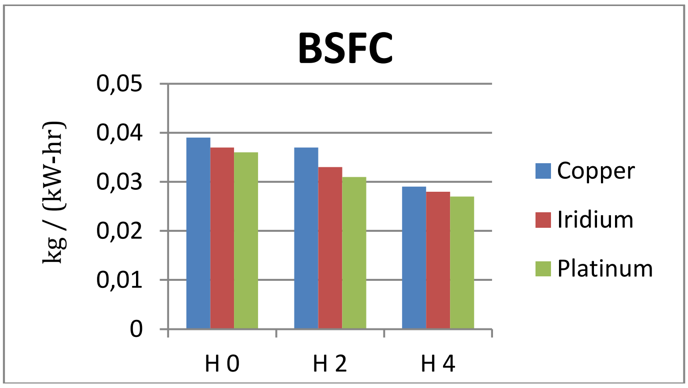

Figure 4.10. BSFC for engine load of 8 Nm and CR of 10:1

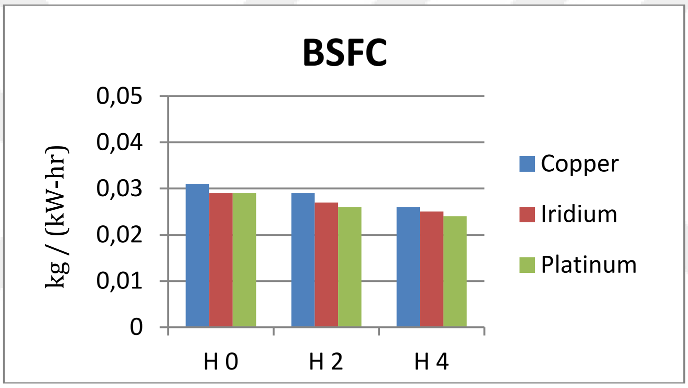

Figure 4.11. BSFC for engine load of 13 Nm and CR of 10:1

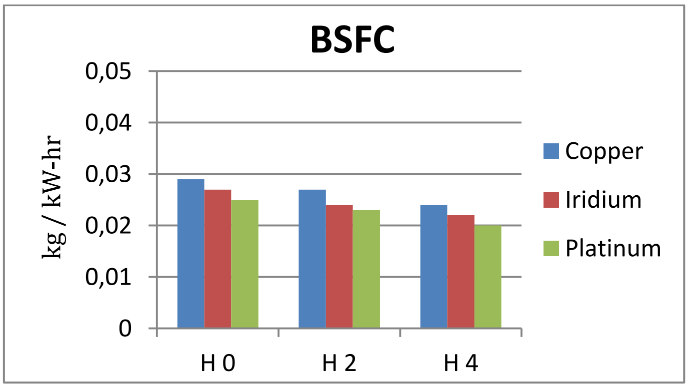

Figure 4.12. BSFC for engine load of 17 Nm and CR of 10:1

Table 4.1. Average variance in BP and BSFC with hydrogen induction at different engine loads and 8.5:1 compression ratio

		Copper		Iridium		Platinum	
		H0 - H2	H0 - H4	H0 - H2	H0 - H4	H0 - H2	H0 - H4
BP	8 Nm	17%	42%	20%	38%	14%	29%
	13 Nm	13%	19%	9%	21%	15%	24%
	17 Nm	14%	26%	11%	22%	11%	19%
SFC	8 Nm	-17%	-31%	-16%	-27%	-16%	-26%
	13 Nm	-13%	-26%	-12%	-23%	-15%	-25%
	17 Nm	-12%	-24%	-9%	-21%	-6%	-19%

Table 4.2. Average variance in BP and BSFC with hydrogen induction at different engine loads and 10:1 compression ratio

		Copper		Iridium		Platinum	
		H0 - H2	H0 - H4	H0 - H2	H0 - H4	H0 - H2	H0 - H4
BP	8 Nm	14%	27%	14%	23%	15%	22%
	13 Nm	11%	18%	6%	12%	8%	12%
	17 Nm	8%	15%	7%	14%	8%	13%
SFC	8 Nm	-7%	-26%	-11%	-24%	-14%	-25%
	13 Nm	-6%	-16%	-7%	-14%	-10%	-17%
	17 Nm	-5%	-17%	-7%	-15%	-8%	-18%

4.2. Emission Characteristics

4.2.1. CO Emissions

Figure 4.13-4.18 illustrates the brake specific CO emission results of the experiments. Formation of CO emission is directly related with combustion completeness of the fuels and insufficient amount of oxygen (Serin and Yıldızhan, 2017).

It is obvious that using iridium and platinum spark plugs dropped the CO emissions significantly. Higher charge density of these spark plugs triggered combustion completeness and helped to reduce incomplete combustion products. However, platinum spark plug reduced CO toxic emissions more than iridium spark plug in general. CO emissions were reduced between 1.01% - 15.77% for iridium spark plugs and 7.66% - 18.32% for platinum spark plug.

Moreover, hydrogen enrichment decreased CO emissions due to its combustion reaction accelerator function and carbonless structure. Induction hydrogen with the flow rate of 2 lit/min (H2) and 4 lit/min (H4) reduced CO emissions by 8.93% and 13.68% respectively in average comparing to unhydrogenated operations. However, at higher engine loads, CO reduction depending on hydrogen enrichment became lower comparing with low engine loads. Table 4.3-4.4 depicts average variance in brake specific CO emissions with hydrogen addition at different engine loads.

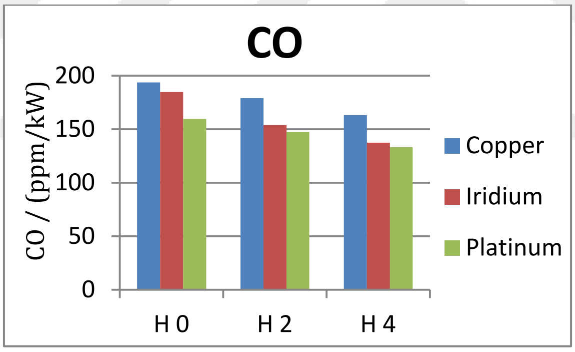

Figure 4.13. Brake specific CO emission values for engine load of 8 Nm and CR of 8.5:1

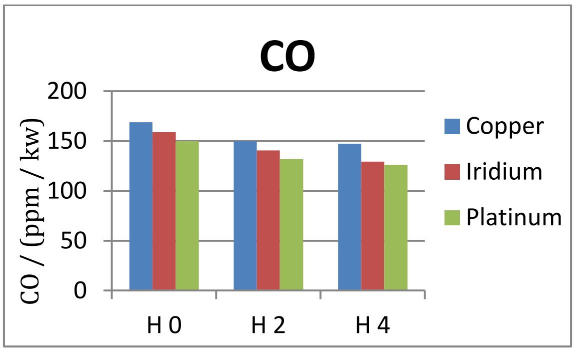

Figure 4.14. Brake specific CO emission values for engine load of 13 Nm and CR of 8.5:1

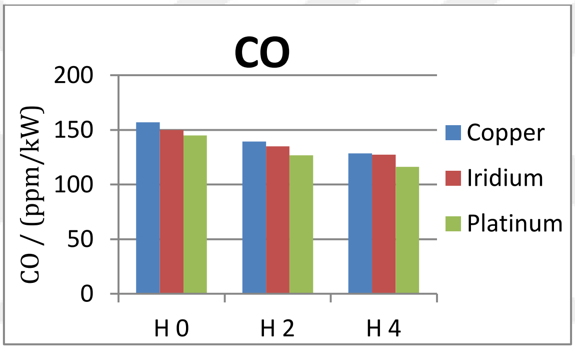

Figure 4.15. Brake specific CO emission values for engine load of 17 Nm and CR of 8.5:1

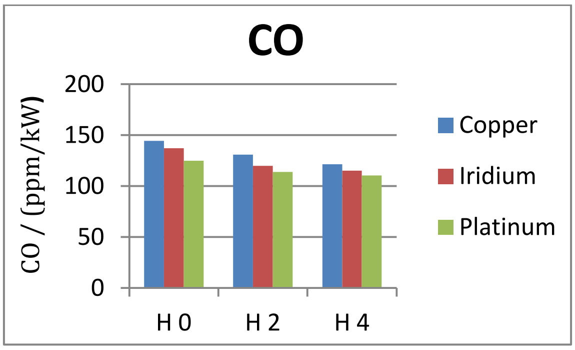

Figure 4.16. Brake specific CO emission values for engine load of 8 Nm and CR of 10:1

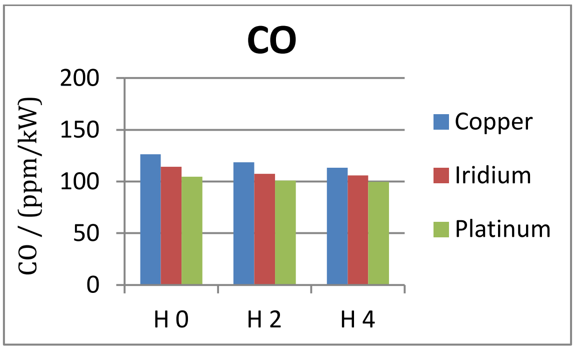

Figure 4.17. Brake specific CO emission values for engine load of 13 Nm and CR of 10:1

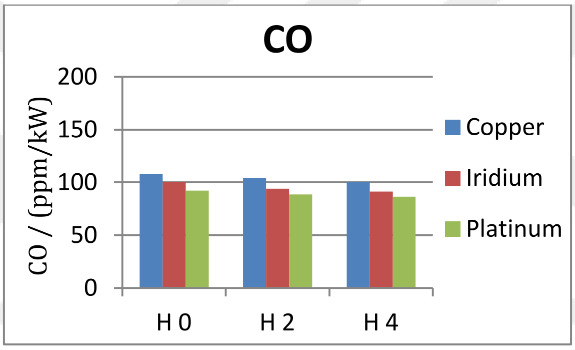

Figure 4.18. Brake specific CO emission values for engine load of 17 Nm and CR of 10:1

Table 4.3. Average variance in CO emissions with hydrogen induction at different engine loads and 8.5:1 compression ratio

		Copper		Iridium		Platinum	
		H0 - H2	**H0 - H4**	**H0 - H2**	**H0 - H4**	**H0 - H2**	**H0 - H4**
	8 Nm	-11%	-19%	-17%	-26%	-12%	-20%
CO	**13 Nm**	-8%	-16%	-12%	-19%	-8%	-16%
	17 Nm	-8%	-13%	-10%	-15%	-7%	-16%

Table 4.4. Average variance in CO emissions with hydrogen induction at different engine loads and 10:1 compression ratio

		Copper		Iridium		Platinum	
		H0 - H2	H0 - H4	H0 - H2	H0 - H4	H0 - H2	H0 - H4
	8 Nm	-9%	-16%	-13%	-16%	-9%	-12%
CO	13 Nm	-6%	-10%	-6%	-9%	-4%	-7%
	17 Nm	-4%	-7%	-7%	-7%	-4%	-6%

4.2.2. CO_2 Emissions

It can be seen from the Figure 4.19-4.24, for iridium and platinum spark plugs, higher CO_2 emissions were measured since these spark plugs improved the combustion and thus CO and UHC compositions are converted into CO_2 emissions.

If they need to be compared, platinum spark plug increased CO_2 emissions more than iridium spark plug. CO_2 emissions were increased between 2.91% - 14.11% for iridium spark plugs and 5% - 25.49% for platinum spark plug. Nevertheless, hydrogen addition increased CO_2 formation, too. CO_2 emissions may have expected to be lower because of combustion of zero-carbon fuel, however due to hydrogen's burning speed increasing function expedited more complete combustion and increased UHC conversion into CO_2.

As a result, hydrogen addition of H2 and H4 increased CO_2 emissions by 16.83% and 45.13% respectively in average comparing with H0 fuel. In addition, CO_2 increment ratio depending on hydrogen induction recorded generally higher for lower engine loads. Table 4.5-6 shows the average variance in brake specific CO_2 emissions with hydrogen addition at different engine loads.

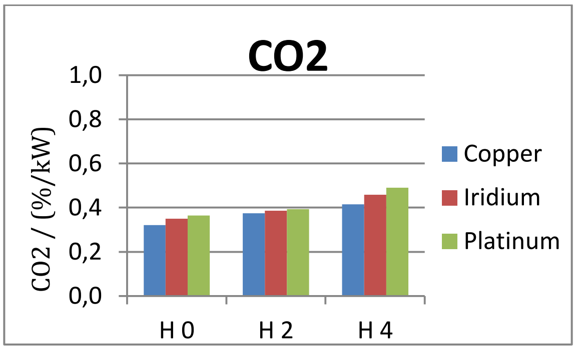

Figure 4.19. Brake specific CO_2 emission values for engine load of 8 Nm and CR of 8.5:1

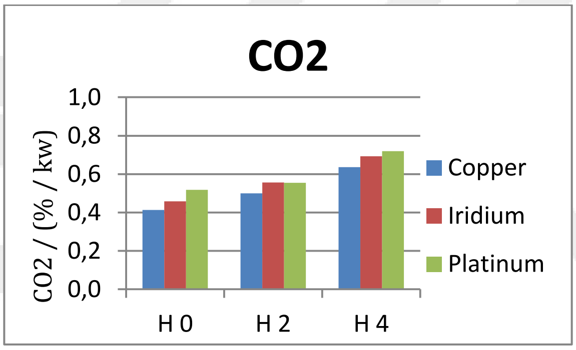

Figure 4.20. Brake specific CO_2 emission values for engine load of 13 Nm and CR of 8.5:1

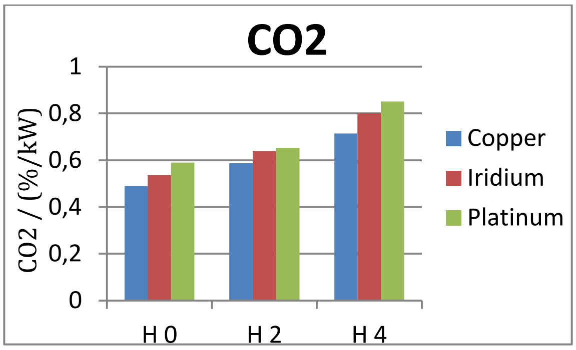

Figure 4.21. Brake specific CO_2 emission values for engine load of 17 Nm and CR of 8.5:1

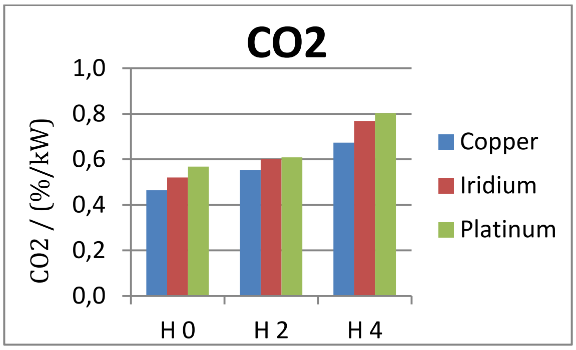

Figure 4.22. Brake specific CO_2 emission values for engine load of 8 Nm and CR of 10:1

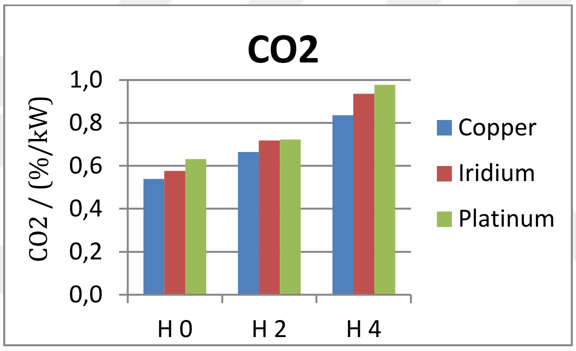

Figure 4.23. Brake specific CO_2 emission values for engine load of 13 Nm and CR of 10:1

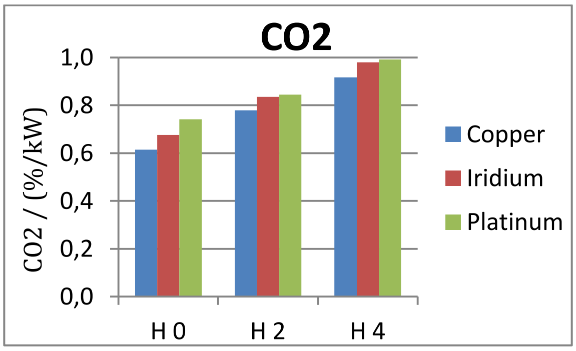

Figure 4.24. Brake specific CO_2 emission values for engine load of 17 Nm and CR of 10:1

Table 4.5. Average variance in CO_2 emissions with hydrogen induction at different engine loads and 8.5:1 compression ratio

		Copper		Iridium		Platinum	
		H0 - H2	H0 - H4	H0 - H2	H0 - H4	H0 - H2	H0 - H4
	8 Nm	21%	40%	21%	51%	11%	44%
CO_2	13 Nm	20%	35%	19%	35%	8%	39%
	17 Nm	17%	27%	10%	26%	7%	34%

Table 4.6. Average variance in CO_2 emissions with hydrogen induction at different engine loads and 10:1 compression ratio

		Copper		Iridium		Platinum	
		H0 - H2	H0 - H4	H0 - H2	H0 - H4	H0 - H2	H0 - H4
	8 Nm	27%	55%	25%	62%	14%	55%
CO_2	13 Nm	22%	45%	16%	48%	7%	41%
	17 Nm	23%	39%	17%	45%	7%	34%

4.2.3. NO_x Emissions

NO_x results of all tests are shown in Figure 4.25-30. NO_x emissions were increased as the platinum and iridium spark plug were used due to the higher temperature accompanied with the increase in flame speed and in-cylinder pressure. For platinum spark plug rise in NO_x became more obvious.

Hydrogen enrichment triggered NO_x formation, too. Higher end-combustion temperature and flame velocity resulted from hydrogen combustion caused dramatic rise in NO_x formation. In average, 82.16% and 204.53% rise in NO_x emissions were measured for H2 and H4 test conditions, respectively. Additionally, NO_x emissions percentage change with hydrogen occurred more apparent at lower engine loads comparing with higher loads. Table 4.7-8 represents the average variance in brake specific NO_x emissions with hydrogen addition at different engine loads.

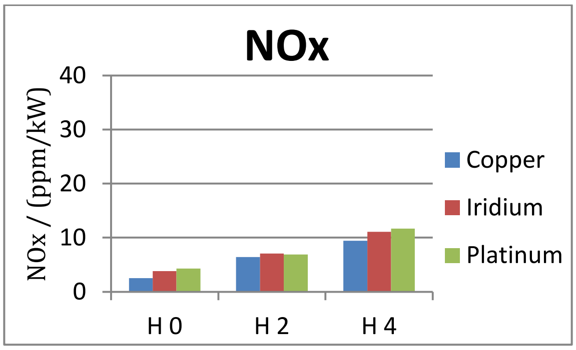

Figure 4.25. Brake specific NO_x emission values for engine load of 8 Nm and CR of 8.5:1

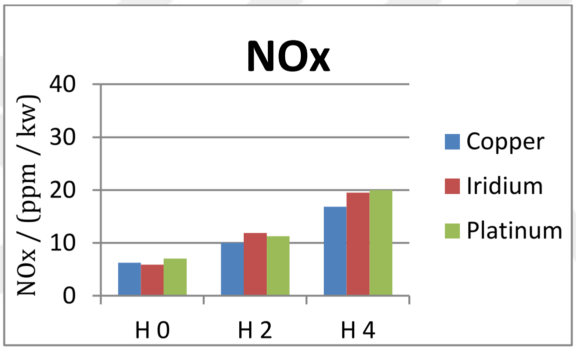

Figure 4.26. Brake specific NO_x emission values for engine load of 13 Nm and CR of 8.5:1

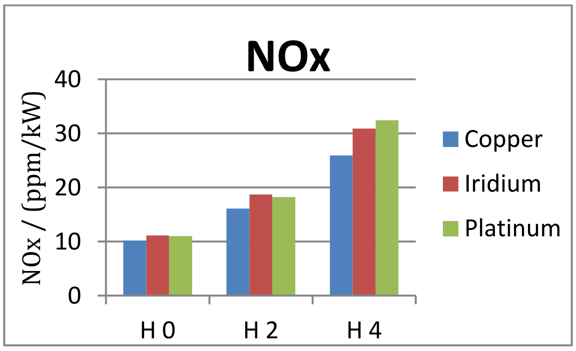

Figure 4.27. Brake specific NO_x emission values for engine load of 17 Nm and CR of 8.5:1

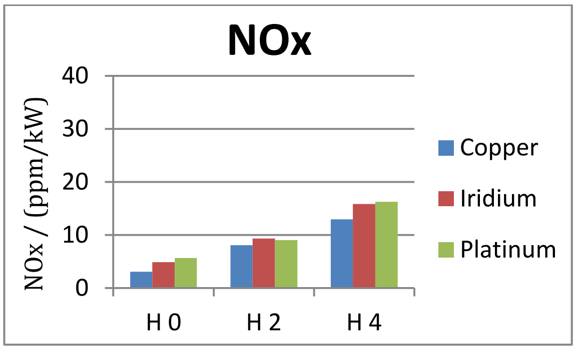

Figure 4.28. Brake specific NO_x emission values for engine load of 8 Nm and CR of 10:1

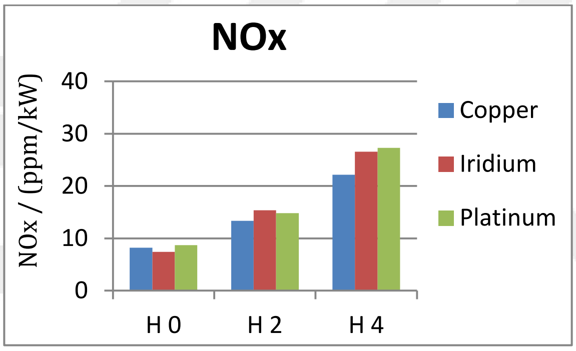

Figure 4.29. Brake specific NO_x emission values for engine load of 13 Nm and CR of 10:1

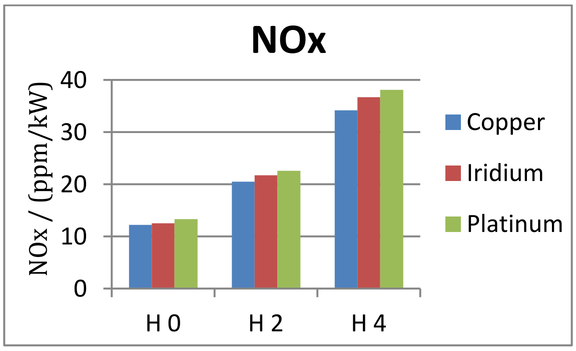

Figure 4.30. Brake specific NO_x emission values for engine load of 17 Nm and CR of 10:1

Table 4.7. Average variance in NO_x emissions with hydrogen induction at different engine loads and 8.5:1 compression ratio

		Copper		Iridium		Platinum	
		H0 - H2	**H0 - H4**	**H0 - H2**	**H0 - H4**	**H0 - H2**	**H0 - H4**
	8 Nm	157%	276%	102%	232%	65%	193%
NO_X	**13 Nm**	60%	169%	83%	189%	60%	183%
	17 Nm	58%	154%	68%	178%	60%	172%

Table 4.8. Average variance in NO_x emissions with hydrogen induction at different engine loads and 10:1 compression ratio

		Copper		Iridium		Platinum	
		H0 - H2	**H0 - H4**	**H0 - H2**	**H0 - H4**	**H0 - H2**	**H0 - H4**
	8 Nm	162%	322%	107%	257%	89%	186%
NO_X	**13 Nm**	63%	170%	93%	226%	70%	214%
	17 Nm	67%	179%	73%	193%	69%	186%

4.2.4. UHC Emissions

It can be seen from Figure 31-36, iridium and platinum spark plug decreased UHC emission because of improved combustion, remarkably. Platinum spark plug reduced UHC emissions more than iridium spark plug. UHC emissions were decreased between 2.51% - 19.59% for iridium spark plugs and 7.62% - 24.12% for platinum spark plug.

Moreover, UHC emissions were effectively reduced with the increment of hydrogen flow rate. According to the test results, the brake specific UHC emissions were dropped by 16.62% and 25.92% with H2 and H4 test fuel, respectively. This result can be attributed to the improvement of the combustion quality through the shorter combustion duration and after hydrogen addition, which lead to complete the combustion. On the other hand, drop of brake specific UHC emissions with hydrogen enrichment were comparatively lower at higher engine loads. Table 4.9-10 show the average variance in brake specific UHC emissions with hydrogen addition at different engine loads.

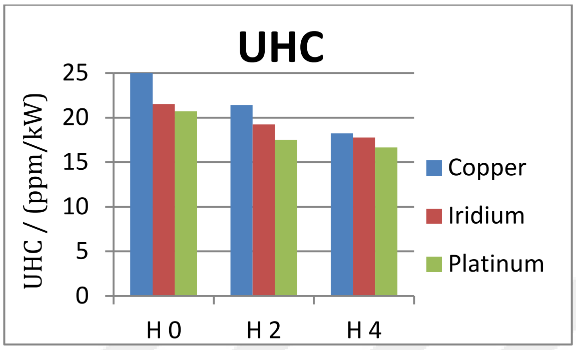

Figure 4.31. Brake specific UHC emission values for engine load of 8 Nm and CR of 8.5:1

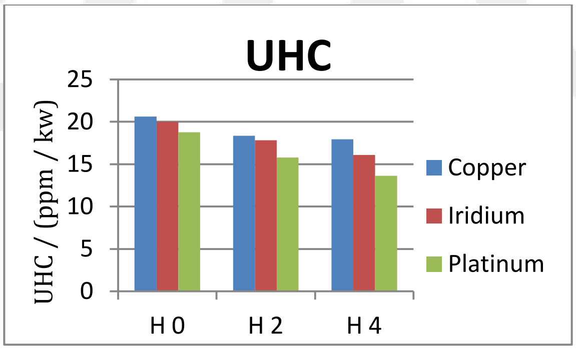

Figure 4.32. Brake specific UHC emission values for engine load of 13 Nm and CR of 8.5:1

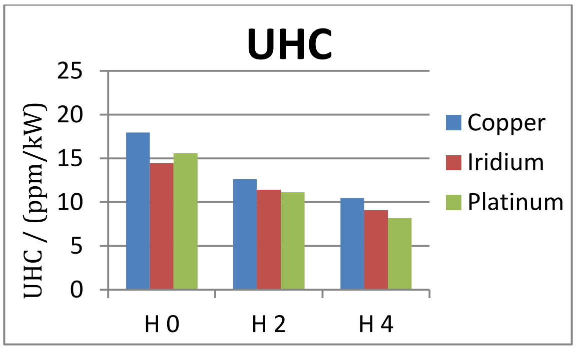

Figure 4.33. Brake specific UHC emission values for engine load of 17 Nm and CR of 8.5:1

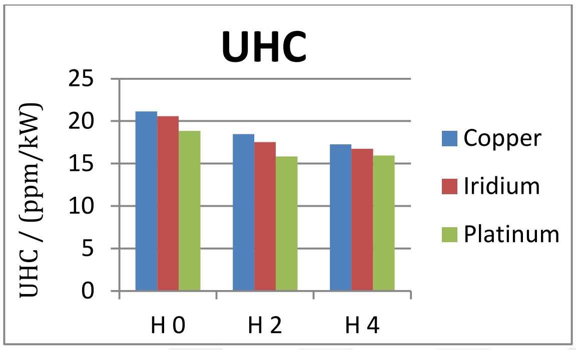

Figure 4.34. Brake specific UHC emission values for engine load of 8 Nm and CR of 10:1

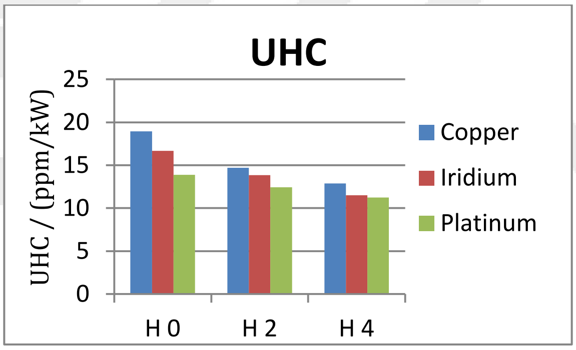

Figure 4.35. Brake specific UHC emission values for engine load of 13 Nm and CR of 10:1

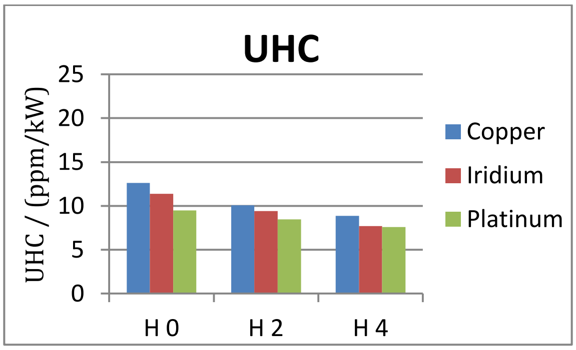

Figure 4.36. Brake specific UHC emission values for engine load of 17 Nm and CR of 10:1

Table 4.9 average variance in UHC emissions with hydrogen addition at different engine loads and 8.5:1 compression ratio

		Copper		Iridium		Platinum	
		H0 - H2	H0 - H4	H0 - H2	H0 - H4	H0 - H2	H0 - H4
UHC	8 Nm	-30%	-42%	-21%	-37%	-29%	-47%
	13 Nm	-14%	-27%	-11%	-20%	-19%	-27%
	17 Nm	-11%	-13%	-11%	-17%	-16%	-25%

Table 4.10 average variance in UHC emissions with hydrogen addition at different engine loads and 10:1 compression ratio

		Copper		Iridium		Platinum	
		H0 - H2	H0 - H4	H0 - H2	H0 - H4	H0 - H2	H0 - H4
UHC	8 Nm	-22%	-32%	-18%	-29%	-16%	-19%
	13 Nm	-20%	-28%	-15%	-20%	-11%	-15%
	17 Nm	-20%	-23%	-15%	-22%	-11%	-16%

4.3. Effects of Higher Compression Ratio

4.3.1. Performance Characteristics at Higher Compression Ratio

Compression ratio is a key factor for internal combustion engines. Because CR directly affects the performance and emission characteristics of an engine, CR should be chosen as high as possible.

In this study, compression ratio of 8.5:1 was increased to 10:1 in order to obtain better performance. It can be seen from Table 4.11-4.13, increment of CR improved performance of the engine regardless hydrogen flow rate and spark plug. However, average enhancement of BP and BSFC values higher at lower engine loads. In average, 21.05% more BP and 16.47% less SFC were attained thanks to higher compression ratio of test engine.

On the other hand, at higher engine loads enhancements in BP and SFC were occurred comparatively lower.

Table 4.11. Variation of performance values of test fuels with CR increment at 8 Nm engine load

CR from 8.5 to 10									
Increment – Decrement Variance in % at 8 Nm Engine Load									
	Copper			Iridium			Platinum		
	H 0	H 2	H 4	H 0	H 2	H 4	H 0	H 2	H 4
BP	38.33	35.71	23.53	34.62	28.21	19.44	28.57	29.38	22.22
SFC	-20.41	-13.95	-21.62	-17.78	-13.16	-15.15	-18.18	-16.22	-18.18

Table 4.12. Variation of performance values of test fuels with CR increment at 13 Nm engine load

CR from 8.5 to 10									
Increment – Decrement Variance in % at 13 Nm Dynamometer Load									
	Copper			Iridium			Platinum		
	H 0	H 2	H 4	H 0	H 2	H 4	H 0	H 2	H 4
BP	18.75	16.67	18.42	23.53	20.54	14.63	26.47	18.97	14.29
SFC	-26.19	-17.14	-10.34	-17.14	-15.63	-7.41	-14.71	-10.34	-4.00

Table 4.13. Variation of performance values with CR increment at 17 Nm engine load

CR from 8.5 to 10									
Increment – Decrement Variance in % at 17 Nm Dynamometer Load									
	Copper			Iridium			Platinum		
	H 0	H 2	H 4	H 0	H 2	H 4	H 0	H 2	H 4
BP	19.76	13.16	9.05	19.44	15.00	11.36	18.92	16.10	13.18
SFC	-23.68	-18.18	-14.29	-18.18	-17.24	-15.38	-19.35	-20.69	-20.00

4.3.2. Emission Characteristics at Higher Compression Ratio

Higher compression ratio is advantageous because it permits an engine to obtain more mechanical energy from a certain air-fuel mixture. This is the result of higher thermal efficiency is created by reaching the same combustion temperature with less fuel, while having longer expansion cycle, giving more mechanical power output and reducing the exhaust temperature. When regarding CR as an expansion ratio, it decreases the temperature of the exhaust gases, and therefore the energy wasted to the atmosphere.

From Table 4.14 to 4.16, percentage change of emission values by increment of CR from 8.5:1 to 10:1 were given. From results, it can be said that increasing CR of the test engine reduced CO and UHC emissions significantly at all engine loads with all spark plugs and hydrogen flow rates. However, CO_2 and NO_x emission outputs increased due to higher CR. Reduction of CO and UHC emissions can be attributed to enhanced combustion. On the other hand, rise in CO_2 emission is a result of that increased combustion completeness of the engine. Moreover, higher temperature and pressure due to increased CR resulted in more NO_x emissions output from the exhaust.

Numerically, due to higher CR, 25.15% and 16.82% reductions in CO and UHC emissions were observed in average, respectively. Conversely, 37.39% and 28.13% increments in CO_2 and NO_x emissions were recorded respectively in average.

Table 4.14. Variation of emission values with CR increment at 8 Nm engine load

CR from 8.5 to 10

Increment – Decrement Variance in % at 8 Nm Dynamometer Load

	Copper			**Iridium**			**Platinum**		
	H 0	**H 2**	**H 4**	**H 0**	**H 2**	**H 4**	**H 0**	**H 2**	**H 4**
CO	-25.38	-26.76	-25.61	-25.71	-22.00	-16.28	-21.70	-22.71	-17.16
CO_2	44.58	47.37	61.90	48.57	56.00	67.44	55.56	54.59	63.64
NO_X	22.89	25.26	37.62	26.29	32.60	42.33	32.22	31.40	39.09
HC	-15.42	-13.79	-5.29	-4.49	-8.74	-5.81	-9.00	-9.57	-4.27

Table 4.15. Variation of emission values with CR increment at 13 Nm engine load

CR from 8.5 to 10

Increment – Decrement Variance in % at 13 Nm Dynamometer Load

	Copper			**Iridium**			**Platinum**		
	H 0	**H 2**	**H 4**	**H 0**	**H 2**	**H 4**	**H 0**	**H 2**	**H 4**
CO	-25.15	-20.95	-23.10	-28.04	-23.42	-18.03	-30.23	-23.47	-21.08
CO_2	30.67	33.00	31.25	25.61	28.98	34.98	22.13	30.42	35.95
NO_X	31.37	33.71	31.73	26.29	29.42	36.09	23.35	31.12	36.50
HC	-8.13	-19.77	-28.19	-16.67	-22.35	-28.63	-25.99	-21.33	-17.40

Table 4.16. Variation of emission values with CR increment at 17 Nm engine load

CR from 8.5 to 10

Increment - Decrement Variance in % at 17 Nm Dynamometer Load

	Copper			**Iridium**			**Platinum**		
	H 0	**H 2**	**H 4**	**H 0**	**H 2**	**H 4**	**H 0**	**H 2**	**H 4**
CO	-31.16	-25.30	-21.88	-33.02	-30.43	-28.16	-36.30	-30.10	-25.80
CO_2	25.38	32.70	28.35	25.72	30.83	22.39	25.70	29.34	16.55
NO_X	20.24	27.26	32.05	32.97	16.25	18.85	21.09	24.03	17.56
HC	-29.86	-20.39	-15.38	-21.11	-17.54	-15.14	-38.97	-23.76	-7.23

5. CONCLUSIONS

In this study the performance and emission characteristics of a spark ignition engine at various compression ratios fuelled with gasoline and hydrogen enriched gasoline fuels were investigated. During the experiments the engine compression ratio was set as 8.5:1 and 10:1 and copper, platinum and iridium spark plugs performed one by one. Besides, each test condition was executed at different dynamometer loads which are 8 Nm, 13 Nm and 17 Nm. According to experimental results, the followings can be summarized;

- Changing conventional (copper) spark plug with iridium and platinum spark plugs expedited combustion development resulting in improved BP, BSFC and lower CO and HC emissions at all compression ratios, engine loads and hydrogen flow rates.
- Platinum and iridium spark plugs caused higher CO_2 and NO_x emissions due to improved combustion completeness and in-cylinder temperature rise depending on greater charge density, respectively.
- Variations in performance and emission parameters mentioned above were more obvious for platinum spark plug than iridium type.
- Similar to the spark plug replacement, hydrogen addition increased BP, CO_2 and NO_x and reduced BSFC, CO and HC values comparing with unhydrogenated fuels.
- Increasing the hydrogen flow rate, triggered the variation in emission and performance parameters depending on hydrogen’s combustion accelerator function.
- Higher CR resulted with lower HC and CO emissions due to better combustion and thus higher CO_2 emission was observed.
- Due to increased cylinder temperature NO_x values were increased when engine was operating with higher CRs.

- Increasing CR improved BSFC and BP values for all test fuels.
- At higher engine loads, changes in values of performance and emissions parameters depending on higher compression ratio and variation of spark plugs were lower comparing to lower engine loads.

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