

**ÇUKUROVA UNIVERSITY
INSTITUTE OF NATURAL AND APPLIED SCIENCES**

MSc THESIS

Ali Can YILMAZ

DESIGN AND APPLICATIONS OF HYDROXY (HHO) SYSTEM

DEPARTMENT OF MECHANICAL 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 12/07/2010.

.....
Prof. Dr. Kadir AYDIN
SUPERVISOR

.....
Assoc.Prof.Dr.Hüseyin AKILLI
MEMBER

.....
Asst.Prof.Dr.Murat AKSOY
MEMBER

This MSc Thesis is written at the Department of Institute of Natural And Applied Sciences of Çukurova University.

Registration Number:

Prof. Dr. İlhami YEĞİNGİL
Director
Institute of Natural and Applied Sciences

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ABSTRACT

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Supervisor :Prof. Dr. Kadir AYDIN

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Jury :Prof. Dr. Kadir AYDIN

:Assoc. Prof. Dr. Hüseyin AKILLI

:Asst. Prof. Dr. Murat AKSOY

In this study, hydroxy gas (HHO) is produced by the electrolysis process of different catalysts ($\text{KOH}_{(\text{aq})}$, $\text{NaOH}_{(\text{aq})}$, $\text{NaCl}_{(\text{aq})}$) with various electrode designs in a leak proof plexiglass reactor container (hydrogen generator). Catalysts are used to diminish oxygen and hydrogen bonds. Hydroxy gas is used as a supplementary fuel in a four cylinder, four stroke, compression ignition (diesel) and a single cylinder, two stroke spark ignition (gasoline) engine without any modification and without need for storage tanks. Its effects on hydrocarbon (HC), carbon monoxide (CO) emissions, engine performance characteristics and specific fuel consumption (SFC) are investigated.

Key Words: Hydrogen, Hydroxy, Electrolysis, Performance, Emission

ÖZ

YÜKSEK LİSANS TEZİ

HİDROKSİ (HHO) SİSTEM DİZAYNI VE UYGULAMALARI

Ali Can YILMAZ

**ÇUKUROVA ÜNİVERSİTESİ
FEN BİLİMLERİ ENSTİTÜSÜ
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:Doç. Dr. Hüseyin AKILLI

:Yrd. Doç. Dr. Murat AKSOY

Bu çalışmada hidroksi gazı, sızdırmaz bir reaktör kabı (hidrojen jeneratörü) içerisine konulan farklı katalistlerin ($\text{KOH}_{(aq)}$, $\text{NaOH}_{(aq)}$, $\text{NaCl}_{(aq)}$) sulu çözeltilerinin, özel üretilmiş elektrotlar ile elektrolizi sonucu elde edilmiştir. Katalistler sudaki hidrojen ve oksijen bağlarını zayıflatmak amacıyla konulmuştur. Hidroksi gazı, dört silindirli, dört zamanlı bir sıkıştırma ile ateşlemeli (dizel) motor ve tek silindirli, iki zamanlı bir buji ateşlemeli (benzinli) motorda, motorin ve benzine karıştırılarak, motorlar üzerinde hiç bir modifikasyon yapılmadan ve depolama tankına ihtiyaç duyulmadan kullanılmıştır. Hidroksi gazının hidrokarbon (HC) ve karbon monoksit (CO) emisyonları ile motor performansı ve özgül yakıt tüketimi üzerindeki etkileri incelenmiştir.

Anahtar Kelimeler: Hidrojen, Hidroksi, Elektroliz, Performans, Emisyon

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LIST OF ABBREVIATIONS AND NOMENCLATURE

HHO	: Hydroxy
SI	: Spark Ignition
CI	: Compression Ignition
LPG	: Liquefied Petroleum Gas
CO	: Carbon monoxide
HC	: Hydrocarbon
CO ₂	: Carbon dioxide
NO _x	: Nitrogen oxide
Φ	: Equivalence ratio
HEV	: Hybrid Electric Vehicle
H ₂ ICE	: Hydrogen fueled Internal Combustion Engine
RON	: Research Octane Number
MON	: Motor Octane Number
KOH	: Potasium Hydroxide
NaOH	: Sodium Hydroxide
NaCl	: Sodium Chloride
EEW:	: Electrically Expanded Water
W	: Watt
P _e	: Electron Density
q	: Charge of Electrons
SFC	: Specific Fuel Consumption
HECU	: Hydroxy Electronic Control Unit
TDC	: Top Dead Center
BDC	: Bottom Dead Center
ATDC	: After Top Dead Center
BTDC	: Before Top Dead Center
EEPROM	: Electrically Erasable Programmable Read-Only Memory
F	: Force
T	: Torque
rev.	: Revolution

P_b	: Brake Power
bmep	: Brake Mean Effective Pressure
n_r	: Number of Crank Revolutions for One Complete Cycle
BSFC	: Brake Specific Fuel Consumption
Q_{LHV}	: Lower Heating Value of Fuel
\dot{m}_f	: Mass Flow Rate of the Fuel
\dot{m}_a	: Mass Flow Rate of the Air
h_{bth}	: Thermal Efficiency
CR	: Compression Ratio
AFR	: Air-Fuel Ratio
a	: Cut-off Ratio
g	: Ratio of Specific Heats
EGR	: Exhaust Gas Recirculation
SOF	: Soluble Organic Fraction
rpm	: Revolutions Per Minute
CATIA	: Computer Aided Three dimensional Interactive Application
V	: Volt
A	: Amper
CCM	: Cubic Centimeters per Minute
SLM	: Standard Liters per Minute
DC	: Direct Current
AC	: Alternating Current
mF	: Micro Farad
MOSFET	: Metal Oxide Semiconductor Field Effect Transistor
PWM	: Pulse Width Modulation
f	: Frequency
MAP	: Manifold Absolute Pressure

1. INTRODUCTION

Energy is a quantity that can be assigned to every particle, object and system of objects as a consequence of the state of that particle, object or system (Harper, 2007). Different forms of energy include kinetic, potential, thermal, gravitational, sound, elastic, light and electromagnetic energy. The forms of energy are often named after a related force. All forms of energy are equivalent energy in one form can disappear but the same amount of energy will appear in another form (Resnick and Halliday, 1960).

Energy is the primary and most universal measure of all kinds of work by human beings and nature. Everything that happens in the world is the expression of flow of energy in one of its forms. Energy is an important input in all sectors of a country's economy. The standard of living is directly related to per capita energy consumption (Vader and Joshi, 2005). Conventional energy sources based on oil, coal, and natural gas have proven to be highly effective drivers of economic progress, but at the same time damaging to the environment and to human health. Furthermore, they tend to be cyclical in nature, due to the effects of oligopoly in production and distribution. These traditional fossil fuel-based energy sources are facing increasing pressure on a host of environmental fronts, with perhaps the most serious challenge confronting the future use of coal being the Kyoto Protocol greenhouse gas reduction targets (Herzog et al., 2006). Spark ignition (SI) and compression ignition (CI) engines are commonly used for transportation. Fossil fuels used as energy resources for internal combustion engines (ICEs) are known as "automotive fuels" (gasoline, diesel fuel, liquefied petroleum gas (LPG), natural gas). Gasoline is always used in SI engines, with a current maximum compression ratio of eleven. At higher ratios, the mixture in the combustion chamber ignites spontaneously, causing engine knock, which may lead to severe engine damage. Very recently, gasoline has come into consideration as a hydrogen carrier for fuel-cell vehicles. Diesel (diesel oil) burns in CI engines. Pure LPG can function only in SI engines—a gasoline engine retrofitted for dual-fuel use, a dedicated gas engine or a heavy-duty diesel engine retrofitted with a SI system (Koch, 1999).

The sharp increase in energy consumption particularly in the past several decades has raised fears of exhausting the globe's reserves of petroleum and other resources in the near future. The huge consumption of fossil fuels has caused visible damage to the environment in various forms. Approximately 90% of our energy consumption comes from fossil fuels. Due to industrializations and population growth our economy and technologies today largely depend upon natural resources, which are not replaceable (Venkatamaran and Elango, 1998). Diesel and spark plug engines are main sources of pollutions in urban areas. According to studies, petrol and diesel engines are the main causes in almost to studies, half of carbon monoxide (CO), hydrocarbon (HC), carbon dioxide (CO₂) and nitrogen oxides (NO_x) produce during the burning of fossil fuels. Emissions like SO₂, lead and smoke too are health hazards caused by motor vehicles. Especially, diesel engines leading pollutant producers of SO₂ and smoke (Tekin and Çavuşoğlu, 1997).

The increased power demand, depleting fossil fuel resources and growing environmental pollution have led the world to think seriously for other alternative sources of energy. Basic concept of alternative energy relates to issues of sustainability, renewability and pollution reduction. In reality alternative energy means anything other than deriving energy via fossil fuel combustion. Various forms of alternative energy sources are solar, wind, biogas/biomass, tidal, geothermal, fuel cell, hydrogen energy, small hydropower, etc (Alias, 2005). Solar energy is available and clean source of energy with low operating cost. The main limitation lies in availability with time, dilute source of energy, with large initial cost. Collection and storage add to the cost (Mitkari and Kari, 2005). Wind energy is renewable and a clean way of energy production. However, even a 100% efficient wind generator is able to convert maximum up to 60% of the available energy in wind into mechanical energy. In addition to this, losses incurred in the generator or pump decrease the overall efficiency of power generation to 35% (Vader and Joshi, 2005). Biodiesel, an alternative diesel fuel, is made from renewable biological sources such as vegetable oils and animal fats. It is biodegradable and non-toxic, has low emission profiles and so is environmentally beneficial (Krawczyk, 1996). With the increase in global human population, more land may be needed to produce food for human

consumption (indirectly via animal feed). The problem already exists in Asia. Vegetable oil prices are relatively high there. The same trend will eventually happen in the rest of the world. This is the potential challenge to biodiesel. From this point of view, biodiesel can be used most effectively as a supplement to other energy forms, not as a primary source (Fangrui and Milford, 1999).

Hydrogen has long been recognized as a fuel having some unique and highly desirable properties, for application as a fuel in engines (King and Rand, 1955). Its long-term renewability and less-polluting features make hydrogen attractive. In addition, hydrogen is non-toxic, odorless and results in complete combustion (Fulton et al., 1993). When hydrogen burns, it produces only water, as shown in Equation 1.1, except for the formation of NO_x .



Due to these characteristics, researchers are focusing their attention on hydrogen as an alternative fuel in ICEs and in the development of fuel cell powered vehicles and hybrid electric vehicles (HEVs) (Saravanan and Nagarajan, 2008). Hydrogen can be used as a sole fuel in SI engine, either by carburetion or by direct injection (Das, 2002). In a CI engine, however, H_2 cannot be directly used due to its higher self-ignition temperature, but it can be used in the dual fuel mode. H_2 is only one of many possible alternative fuels that can be derived from natural resources, such as coal, oil shale and uranium or from renewable resources based on solar energy. H_2 can be commercially produced from electrolysis of water and by coal gasification; it can also be produced by several other methods such as the thermo-chemical decomposition of water and solar photo-electrolysis, although these are currently still in the laboratory stage (Saravanan and Nagarajan, 2008).

Hydrogen has been used as a fuel in ICEs, fuel cells and as an additive to conventional fuels such as gasoline, diesel fuel, methane, etc. Using hydrogen as an additive offers the possibility of enhancing the mixture by taking advantage of properties from both fuels. The properties of hydrogen along with those of diesel fuel and gasoline are listed for comparison in Table 1.1. (D'Andrea et al., 2004).

Hydrogen has a flame speed more than five times greater than the hydrocarbon fuels listed here. Also, it has a lean limit (mixture at which flame will not propagate due to excess air) of $\Phi=0.1$ (equivalence ratio), where Φ is defined as the ratio of the actual fuel/air mass ratio to the stoichiometric fuel/air mass ratio, much lower than the theoretical limit of gasoline ($\Phi=0.6$). Theoretically, it is possible to extend the lean limit of the mixture, by adding a small amount of hydrogen to a liquid or gaseous HC fuel. Operating with abundant excess air ensures more complete combustion, improves efficiency and results in a decrease in peak temperatures, which aids in lowering NO_x , while eliminating problems commonly associated with operating on lean mixtures (Jingding et al., 1998; Stebar and Parks, 1974). Secondly, the higher flame speed increases the rate of combustion of the mixture and lowers cycle-to-cycle variations (Apostolescu and Chiriac, 1996; Varde, 1981). Hydrogen has a higher diffusivity compared to hydrocarbon fuels, which improves mixing, enhances turbulence and increases homogeneity in the charge. The lower ignition energy requirement for hydrogen ensures prompt ignition and eases cold starts. The quenching gap refers to the largest passage that will extinguish a flame. The smaller quenching gap exhibited by hydrogen in comparison to gasoline means that the flame could travel closer to the cylinder wall and farther into crevices resulting in more complete combustion (Bell and Gupta, 1997). The ability for hydrogen internal combustion engines (H_2 ICEs) to operate with near-zero engine-out emissions is primarily owed to the coupled effect of two characteristics unique to hydrogen: (i) in principle, NO_x are the only undesirable engine out emissions, formed by the thermal dissociation and oxidation of N_2 in atmospheric air during combustion, and (ii) the low lean-flammability limit of hydrogen allows stable combustion at highly dilute conditions. The coupled effect is that during ultra-lean operation, combustion temperatures are low enough such that NO_x formation rates are too slow and engine-out emissions are near zero (Das, 1991). The ability to operate efficiently is in part also owed to (ii) because unthrottled operation is possible at low-loads. The power output of a direct injected hydrogen engine was 20% more than for a gasoline engine and 42% more than a hydrogen engine using a carburetor. While direct injection solves the problem of pre-ignition in the intake manifold, it does not necessarily

prevent pre-ignition within the combustion chamber. In addition, due to the reduced mixing time of the air and fuel in a direct injection engine, the air/fuel mixture can be non-homogenous. Studies have suggested this can lead to higher NO_x emissions than the non-direct injection systems (Masood et al., 2007). Direct injection systems require a higher fuel rail pressure than the other methods.

Table 1.1. Comparison of hydrogen to diesel and unleaded gasoline (D'Andrea et al., 2004)

Properties	Diesel	Unleaded Gasoline	Hydrogen
Auto-ignition Temperature (K)	530	533-733	858
Minimum Ignition Energy (mJ)	-	0.24	0.02
Flammability Limits (vol % in air)	0.7-5	1.4-7.6	4-75
Stoichiometric Air-Fuel Ratio on Mass Basis	14.5	14.6	34.3
Limits of Flammability (equivalence ratio)	-	0.7-3.8	0.1-7.1
Density at 16°C and 1.01 bar (kg/m ³)	833-881	721-785	0.0838
Net Heating Value (MJ/kg)	42.5	43.9	119.9
Flame Velocity (cm/s)	30	37-43	265-325
Diffusivity in air (cm ² /s)	-	0.08	0.63
Octane Number (RON/MON)	30/-	92-98/80-90	130/-
Cetane Number	40-55	13-17	-

The unique combustion characteristics of hydrogen that allow clean and efficient operation at low engine loads present difficulties at high engine loads. Here, the low ignition energies of hydrogen–air mixtures cause frequent unscheduled combustion events, and high combustion temperatures of mixtures closer to the stoichiometric composition lead to increased NO_x production. Both effects, in practical application, limit the power densities of H₂ICEs. The recent research thrust and progress on this front is the development of advanced hydrogen engines with improved power densities and reduced NO_x emissions at high engine loads (White et al., 2006). Some researchers tested diesel engines, using hydrogen as a sole fuel however; it was very difficult to operate a diesel engine with hydrogen just by increasing the compression ratio, due to its high self-ignition temperature. Therefore, glow plug or spark plug is often used (Wong, 1990). For hydrogen fueled SI engines, the problems that are to be overcome are backfire, pre-ignition and NO_x emissions (Lee et al., 2001). The addition of hydrogen increases combustion temperatures therefore creating conditions where it is easier for NO_x to form if proper tuning is not

utilized. Several studies have shown that if mixtures are made lean and spark timing is retarded NO_x can be reduced to a point below normal hydrocarbon combustion (Jingding, 1998). Engines fueled with hydrogen suffer from reduced power output, due mainly to the very low heating value of hydrogen on volume basis and resorting to lean mixture operation. Therefore, the mass of the intake air is reduced for any engine size because of the relatively high stoichiometric hydrogen to air ratio (Block and Veziroğlu, 1994). Hydrogen as a compressed gas at 200 atmospheres and atmospheric temperature has merely around 5% of the energy of gasoline of the same volume. This is a major shortcoming particularly for transport applications (Kondo et al., 1996). A hydrogen engine needs to be some 40–60% larger in size than for gasoline operation for the same power output. This could impose some reduction to engine speed, increased mechanical and motoring losses and reduced tolerance to knocking. Also, some engine design modifications are needed (Peschka, 1998).

Hydrogen is only one of many possible alternative fuels that can be derived from natural resources such as coal, oil shale and uranium or from renewable resources based on solar energy. Hydrogen can be commercially produced from electrolysis of water and by coal gasification; it can also be produced by several other methods such as the thermo-chemical decomposition of water and solar photo-electrolysis, although these are currently still in the laboratory stage. Hydrogen fueled ICE vehicles built with current technology are not competitive with synthetic gasoline or methanol vehicles on the basis of coal consumption or fuel cost (Ravi et al., 1992). However, the development of practical and highly efficient end-use converters of H_2 , such as fuel cells, should lead to a dramatic reduction in cost and improvement in efficiency of H_2 production, in addition to providing safe and convenient onboard storage.

Hydroxy system is basically a hydrogen generator by the electrolysis of water. Hydroxy gas (Brown's Gas) is a mixture of monoatomic and diatomic hydrogen and oxygen and a special form of water called "Electrically Expanded Water (EEW)" or "Santilli Magnecules". Brown's Gas is produced by a similar design of the electrolyzer that will split water into its various components. Browns gas has a plethora of unusual characteristics that seem to defy current chemistry. It

has a cool flame about 130⁰C yet is able to melt steel, brick and many other metals. The system can be used for both SI and CI engines. Within the compact structure of the system, tap water is electrolyzed by special designed electrodes. Brown's gas can fuse brick, steel, sublimate tungsten, flame is cool, glaze quarts, neutralize nuclear waste, fuse two dissimilar substances and many more things. Brown's Gas burns with a clean flame. It uses no atmospheric oxygen, and creates only pure water as its combustion product. One liter of water produces about 1866 liters of gas. When this gas is ignited, the volume is reduced to the original one liter of water (Michrowski, 1993). Every material gets hot when used as a resistor for electricity. When this electric steam hits a piece of metal the extra electrons from the 'electric steam' travel with the speed of the gas plus original speed around the excited water molecule. Normally, the field present in the wire would create a net acceleration in the same direction of the force; however the constant collisions of electrons create a drag effect. The effect on a hole is an average group velocity referred to as the drift velocity (V_d). It is followed by the equation of;

$$V_d = \frac{W}{P_e \times q} \quad (1.2)$$

where,

W : Energy consumed during the related process (Joule)

P_e : The free electron density of desired material (water) which the electron is traveling in (Coulombs/m³)

q : Charge of electron (1.602 x 10⁻¹⁹ Coulombs)

This equation helps to determine the amount of energy and the speed of the electrons. In case of the Brown's gas, it is like a moving wire at a speed of 7.4 m/s plus the original speed of the electrons (if the gas is stationary), so the material that is being hit by this high energy gas has those extra electrons transferred into the new material. Those electrons disperse causing high heat due to the electrical resistance of that material (that is why it changes for every material it hits) (Michrowski, 1993).

These high energy electrons do not travel as fast as the gas. When they hit the surface of something, they lose big amount of their speed. Thus, they release their

kinetic energy as heat, the more dense and resistive material gets hotter, the less dense or more conductive material results the less generated heat (however it still has a high heat because of the kinetic energy of the electrons). The Brown's Gas flame is a unique method for transmitting electrical energy directly into the atomic structure of materials.

The flame, upon application to an element or compound of elements, increases its temperature due to an interactive combustion property which is one of the unique characteristics of Brown's Gas (Chieh, 2007).

Oxygen attracts electrons much more strongly than hydrogen (more electronegative), resulting in a water molecule having a positive charge on the side where the hydrogen atoms are and a negative charge on the other side, where the oxygen atom is. Electrical attraction between water molecules is due to this dipole nature of individual water molecules to pull each other closer together, making it more difficult to separate the molecules (meaning the charge differences will cause water molecules are attracted to each other). This attraction is known as hydrogen bonding. Surface tension is a manifestation of this unique bonding (important later). Also hydrogen bonding is a comparatively weak attraction compared to the covalent bonds within the water molecule itself. Hydrogen bond in water is largely due to electrostatic forces and some amount of covalence bonds. Hydrogen bonding also gives water its unusual behavior when freezing (Mccarthy, 2008).

The aim of this experimental investigation is, to make a spectacular combination of anodes and cathodes in a simply adaptable ambient within the fuel system and to obtain an enhancement in combustion and reduction in exhaust emissions with electrolysis reaction without the need for storage tanks. In this experimental study, instead of pure hydrogen addition to diesel fuel, produced hydrogen gas along with oxygen (hydroxy gas, HHO, Brown's gas) is fed to the intake manifolds of a direct injection CI and a single cylinder SI engine by a hydroxy system and a hydroxy electronic control unit (HECU) under various loads.

In this study, HHO is produced by the electrolysis process of different electrolytes ($\text{KOH}_{(aq)}$, $\text{NaOH}_{(aq)}$, $\text{NaCl}_{(aq)}$) with various electrode designs in a leak proof plexiglass reactor (hydrogen generator). Electrolytes are used to diminish

oxygen and hydrogen bonds. HHO is used as a supplementary fuel in a four cylinder, four stroke, CI and a single cylinder two stroke SI engine without any modification and without need for storage tanks. Its effects on HC, CO emissions, engine performance characteristics and specific fuel consumption (SFC) are investigated.

In the first stage of this study, experiments are achieved to specify the most appropriate catalyst and electrode. Then, experiments are carried out for those chosen catalyst and electrode. Reduction in SFC and exhaust emissions and increment in engine torque output are observed for both CI and SI engines at all engine speeds with the aid of HECU. Detailed explanations about the members of the HHO system and the experimental set-up are done in Section 4.1 and 4.2, respectively.

2. LITERATURE REVIEW

Most early engine experiments have been designed for burning a variety of gases, including natural gas and propane. When hydrogen is used in these engines it may backfire. Since hydrogen burns faster than other fuels, the fuel-air mixture is ignited in the intake manifold before the intake valve closes. Injected water controls the backfiring. Hydrogen gives less power than gasoline with or without the water.

There have been many investigations on hydrogen-enriched combustion in ICEs. Rudolf A. Erren has made practical the hydrogen-fueled engine in the 1920s and converted over 1,000 engines. His projects have included trucks and buses. After World War II the allies have discovered a submarine converted by Erren to hydrogen power. Even the torpedoes have been hydrogen powered (Erren and Campbell, 1933).

Since World War I hydrogen and pure oxygen have been considered for submarine use because the crew can get drinkable water from the exhaust. Hydrogen has been also considered for use in powering airship engines. The gas used for buoyancy has also been used for fuel. Even if helium has been used to provide lift, hydrogen gas can be used to supply additional buoyancy if stored at low pressure in a light container (King and Rand, 1955).

Stebar and Parks investigated the effects of hydrogen-supplemented fuel on emission control with lean operation and the test results demonstrated that with small additions of hydrogen to the fuel, very low NO_x and CO emissions were achieved for hydrogen-isooctane mixtures leaner than equivalence ratio of 0.55. Also, significant thermal efficiency improvements resulted from the extension beyond isooctane lean limit operation (Stebar and Parks, 1974).

Robert Zweig has converted a pickup truck to hydrogen power. It has been running ever since. He has solved the backfiring problem by using an extra intake valve to admit hydrogen separately from air. It is a simple, elegant vehicle that uses compressed hydrogen. The American Hydrogen Association has been displaying the Zweig hydrogen pickup trucks in public exhibits (Zweig, 1992).

Cunningham et al. made researches on method and apparatus for enhancing combustion in an ICE through electrolysis and produced hydrogen along with oxygen yielded enhanced combustion at low engine loads for all types of engines (Cunningham et al., 1992).

The Laboratory of Transport Technology (University of Gent, Belgium) has specialized in alternative fuels for the past 10 years or so. Natural gas, LPG, hythane and hydrogen have been the subject of extended research. In a first stage, a Valmet 420D engine, a natural aspirated diesel engine with direct injection has been converted to a SI engine for the use of hydrogen. This engine has been used mainly for the development of a multipoint timed injection system and the study of different types of electromagnetic gas injectors (Sierens, 1998).

Dülger and Özçelik experimentally studied on fuel economy improvement by on board electrolytic hydrogen production kit which could be installed on different vehicles of various types and sizes of engines. Test results under city traffic conditions showed that the fuel consumption for the Volvo 940 dropped to 6 l/100 km from 10.5 l/100 km, a reduction of 43% in fuel consumption. It was 36% for Mercedes 280 (Dülger and Özçelik, 2000).

The fuel induction systems have been developed and designed to provide two intake paths; one for hydrogen and one for air. The fuel and air are kept separate until entering the cylinder to prevent backfiring (Peavey, 2003).

Masood et al. have studied on experimental verification of computational combustion and emission analysis of hydrogen–diesel blends and the test results showed that the hydrogen–diesel co-fueling solved the drawback of lean operation of hydrocarbon fuels such as diesel, which were hard to ignite and resulted in reduced power output, by reducing misfires, improving emissions, performance and fuel economy (Masood et al., 2006).

Saravanan et al. have experimentally investigated the hydrogen-enriched air induction in a diesel engine system. The test results have showed that an efficiency of 27.9% has been achieved without knocking over the entire load range with 30% hydrogen enrichment. Also, they have observed that SFC decreased with increase in hydrogen percentage over the entire range of operation (Saravanan et al., 2007).

Ji and Wang experimentally studied on combustion and emissions performance of a hybrid hydrogen–gasoline engine and they concluded that wide flammability and fast burning velocity of hydrogen yielded reduced CO and HC emissions at idle and lean conditions (Ji and Wang, 2009).

3. INTERNAL COMBUSTION ENGINE (ICE) FUNDAMENTALS

A typical ICE such as that generally used in automobiles, trucks and other similar vehicles uses hydrocarbon fuels for combustion. It is well known that the burning of such fuels is not one hundred percent efficient and further produces large amount of pollutants as a byproduct of the combustion process.

3.1. Spark Ignition Engine

SI engine is one of the two most common reciprocating ICE types in current use. Basic SI engines have not fundamentally changed since the early 1900s with the possible exception of the introduction of the Wankel rotary SI engine in the 1960s. However, major advances in the areas of materials, manufacturing processes, electronic controls, and computer aided design have led to significant improvements in dependability, longevity, thermal efficiency, and emissions during the past decade. Electronic controls, in particular, have played a major role in efficiency gains in SI automotive engines through improved control of the fuel injection and ignition systems that control the combustion process. Electronic control of diesel fuel injection systems is also becoming more common and is producing improvements in diesel emissions and fuel economy.

IC engines may be classified by a wide variety of characteristics, the primary ones being SI vs. CI, four-stroke vs. two-stroke, and reciprocating vs. rotary. Other possible categories of classification include intake type (naturally aspirated vs. turbocharged or supercharged), number of cylinders, cylinder arrangement (in-line, V-type, opposed), cooling method (air vs. water), fueling system (injected vs. carbureted), valve gear arrangement (overhead cam vs. pushrod), type of scavenging for two-stroke engines (cross, loop or uniflow), and type of injection for diesel engines (direct vs. indirect) (Kahraman, 2005; from Heywood, 1998).

3.1.1. Spark Ignition (SI) Engine Operation

Figure 3.1 is a cross-section schematic of a four-stroke SI engine. The SI engine relies on a spark plug to ignite a volatile air-fuel mixture as the piston approaches top dead center (TDC) on the compression stroke. This mixture may be supplied from a carburetor, a single throttle-body fuel injector, or by individual fuel injectors mounted in the intake port of each cylinder. One combustion cycle involves two revolutions of the crankshaft and thus four strokes of the piston, referred to as the intake, compression, power, and exhaust strokes. Intake and exhaust valves control the flow of mixture and exhaust gases into and out of the cylinder, and an ignition system supplies a spark-inducing high voltage to the spark plug at the proper time in the cycle to initiate combustion. On the intake stroke, the intake valve opens and the descending piston draws a fresh combustible charge into the cylinder. During the compression stroke, the intake valve closes and the fuel-air mixture is compressed by the upward piston movement. The mixture is ignited by the spark plug, typically somewhat before TDC. On the expansion stroke, the combustion products expand, forcing the piston down. Near bottom dead center (BDC) the exhaust valve opens and the cylinder pressure drops rapidly to near atmospheric. The piston then returns to TDC, expelling the exhaust products. At TDC, the exhaust valve closes and the intake valve opens to repeat the cycle again (Kahraman, 2005; from Heywood, 1998).

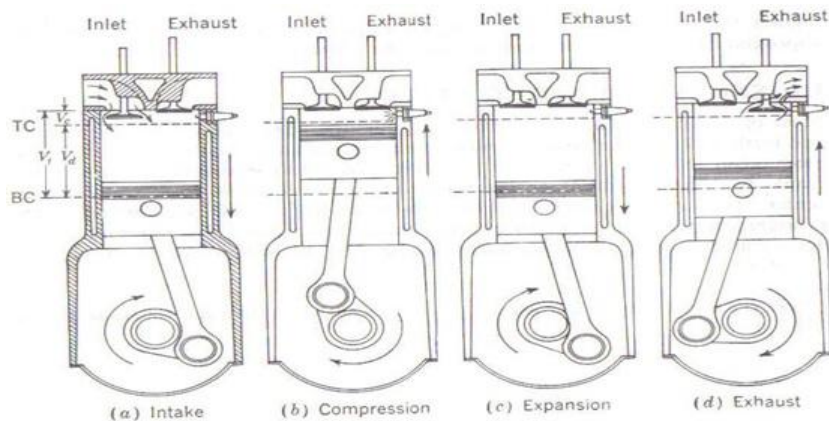


Figure 3.1. Cross-section schematic of a four stroke SI engine (Heywood, 1998)

In SI engines the air and fuel are usually mixed together in the intake manifold using either a carburetor or fuel injection system. In automobile applications, the temperature of the air entering the intake system is controlled by mixing ambient air with air heated by the exhaust manifold. For gasoline the ratio of mass flow of air to mass flow of fuel must be held around 15 to ensure reliable combustion. The carburetor meters an appropriate fuel flow for the engine air flow in the following manner. The air flow through the venturi (a converging-diverging nozzle) sets up a pressure difference between the venturi inlet and throat which is used to meter an appropriate amount of fuel from the float chamber. Just below the venturi is a throttle valve or plate which controls the combined air and fuel flow as shown in Figure 3.2.

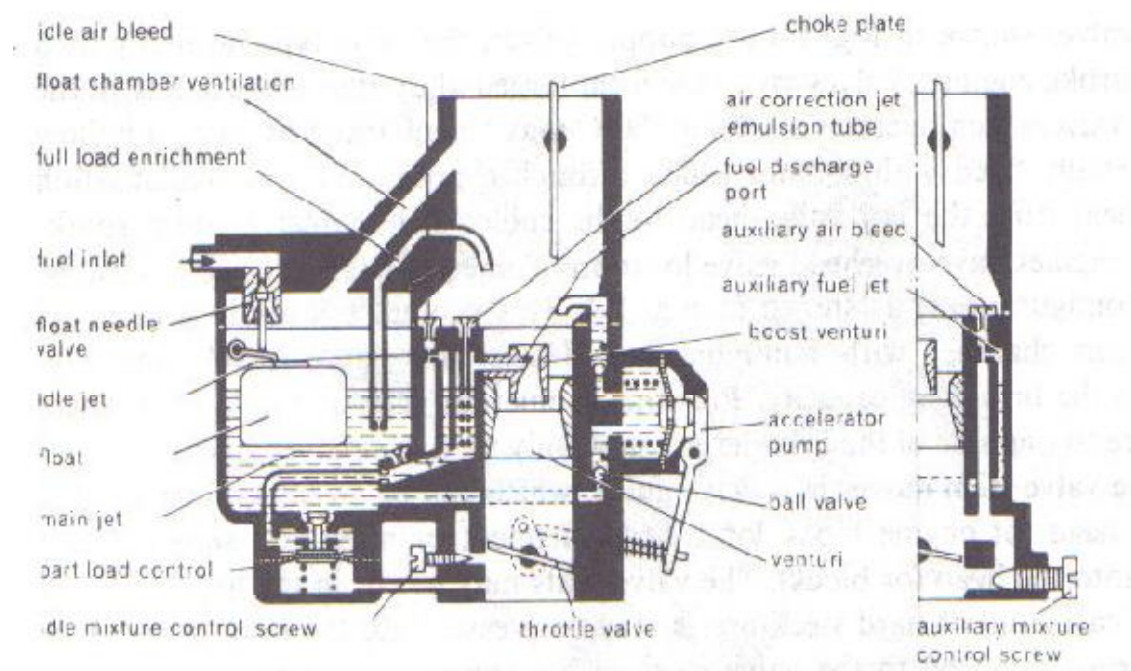


Figure 3.2. Cross section of a single barrel carburetor (Heywood, 1998)

The intake flow is throttled to below atmospheric pressure by reducing the flow area according to the power required. The maximum power is obtained when the throttle is wide open. The intake manifold is usually heated to promote faster evaporation of the liquid fuel thus obtaining a more uniform fuel distribution.

Fuel injection into the intake manifold or inlet port is a widely used alternative to a carburetor. With port injection, fuel is injected through individual injectors from a low-pressure fuel supply into each intake port. There are several different types of injection systems. Mechanical injection in which an injection pump driven by the engine continuously injects fuel to the inlet port and electronically controlled injection where an electronic control unit (ECU) measures the air flow rate and accordingly supplies fuel. Figure 3.3 shows an electronically controlled system where the air flow rate is metered directly; the ECU actuates the injection valves by pulses whose durations are predefined and recorded in the ECU's electrically erasable programmable read-only memory (EEPROM). Another alternative is the use of a single fuel injector located above the throttle plate in the position normally occupied by the carburetor. This approach gives the electronic control at reduced cost but also reduced accuracy (Kahraman, 2005; from Heywood, 1998).

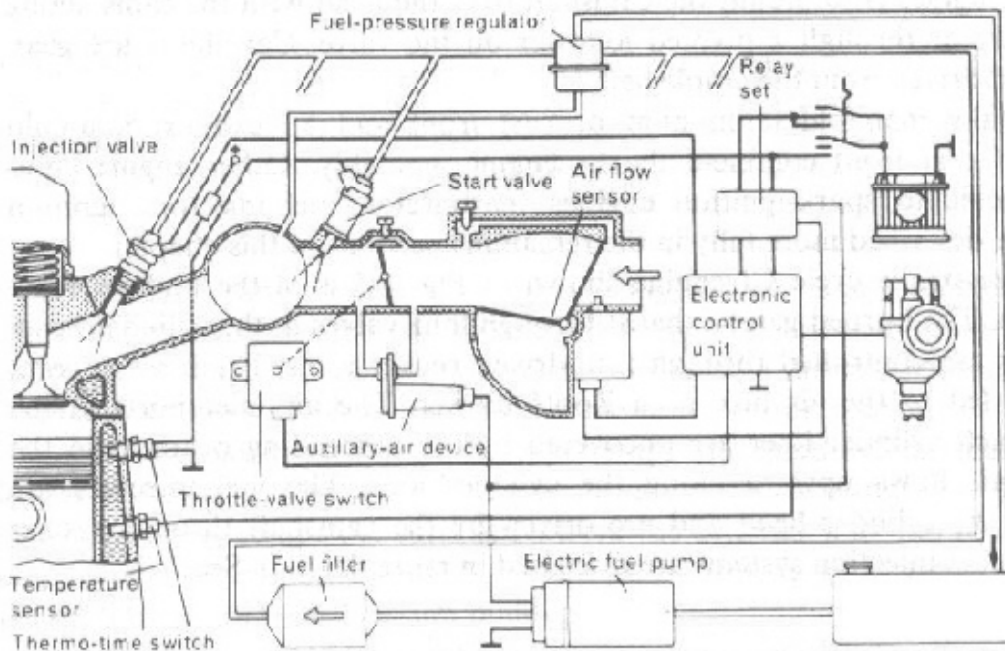


Figure 3.3. Schematic drawing of L-Jetronic port electronic fuel injection system (Heywood, 1998)

3.1.2. Important Engine Characteristics

An engine's primary factors that are important to its user are its performance over the operating range, its fuel consumption within the operating range and the cost of the fuel, the engine's noise and air pollutant emissions, its initial cost and the durability as well as reliability throughout its operating life. Geometrical relationships and other parameters characterize an engine. Engine performance, efficiency and emission characteristics are the most common considerations. Engine performance is more precisely defined by the maximum power at rated speed and maximum torque at rated speed. Rated speed is the speed at which these maximum values are reached. In general the rated speed for maximum power is close to the engine's maximum allowable speed whereas the maximum torque is developed around or slightly above the half of maximum operating speed.

Engine torque is normally measured with a dynamometer. The engine is secured to a test bench where its output shaft is coupled to the dynamometer rotor. Figure 3.4 illustrates the operating principle of a dynamometer. The rotor is braked either by electromagnetic, hydraulic or mechanical friction. The energy supplied by the engine is converted to heat and therefore the dynamometer needs adequate cooling. The opposing torque exerted on the stator is measured by balancing weights, springs, pneumatic or electronic means (Kahraman, 2005; from Heywood, 1998).

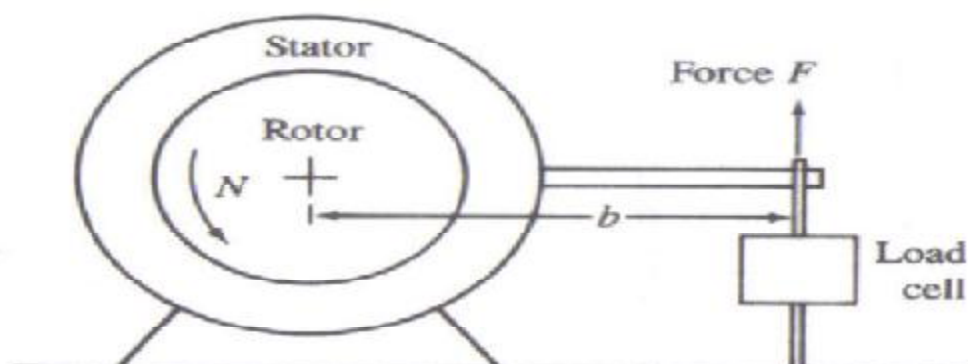


Figure 3.4. Schematic of a dynamometer (Heywood, 1998)

The load cell shown in Figure 3.4 reads the force F applied at a distance b from the center of the rotor. The torque applied by the engine on the dynamometer is T :

$$T (Nm) = F (N) \times b(m) \quad (3.1)$$

Torque is the engine's ability to do work, whereas power is the rate of this work done. The power P delivered by the engine and absorbed by the dynamometer is the product of the torque and angular speed:

$$P = N \times T \quad (3.2a)$$

Using proper units Equation 2.2a becomes:

$$P (kW) = 2\pi w (rev/s) \times T(Nm) \times 10^{-3} \quad (3.2b)$$

The value of engine power measured as described above is called **brake power** (P_b). This is the usable power delivered by the engine to the load.

Another engine performance parameter is the “mean effective pressure”. Since both torque and power depend on engine size, dividing these values by the total volume swept by the cylinders of the engine, gives a more useful relative engine performance measure. The power used in the calculation is the brake power so the term is called **brake mean effective pressure** ($bmep$).

$$bmep (kPa) = \frac{P_b (kW) \times n_r \times 10^3}{V_d (dm^3) \times w (rev/s)} \quad (3.3)$$

where,

n_r : the number of crank revolutions for one complete cycle, 2 for the four-stroke engines and 1 for the two-stroke engines

V_d : the total displaced volume of the cylinders.

In engine tests, the fuel consumption is measured as a flow rate. Again the dependence of the flow rate on engine size makes the use of a parameter called **brake specific fuel consumption** (*bsfc*) necessary, the fuel flow rate per unit power output. It measures how efficiently an engine is using the fuel to do useful work.

$$bsfc \text{ (g/HP}\cdot\text{h)} = \frac{m_f \text{ (g/h)}}{P_b \text{ (HP)}} \quad (3.4)$$

As seen, the *bsfc* has units. A dimensionless parameter that relates the desired engine output (power) to the necessary input (fuel flow) would be of more fundamental value. The ratio of the work produced to the amount of heat energy that can be released in the combustion process is called brake thermal efficiency.

$$\eta_{bth} = \frac{P_b \text{ (kW)}}{m_f \text{ (kg/s)} \times Q_{LHV} \text{ (kJ/kg)}} \quad (3.5)$$

The fuel energy supplied that can be released by combustion is given by the mass of fuel supplied m_f to the engine times the lower heating value Q_{LHV} of the fuel. The heating value of a fuel is determined in a standardized test procedure in which a known mass of fuel is fully burned with air, and the thermal energy released by the combustion process is absorbed by a calorimeter as the products cool down to their original temperature.

Practically, the energy supplied to the engine by the fuel is not fully released as thermal energy in the combustion process because the actual combustion is incomplete. When enough air is present in the cylinder to oxidize the fuel completely, almost all (more than 96%) of this fuel energy supplied is transferred as thermal energy to the working fluid (Kahraman, 2005; from Heywood, 1998).

3.1.3. Combustion in Spark Ignition Engines

In SI engines, combustion of the fuel-air mixture is initiated by a spark generated between the electrodes of a spark plug. The intake and compression strokes are designed to prepare the mixture for combustion by completely vaporizing the fuel and heating the mixture to just below its auto-ignition temperature. This is one reason, in addition to controlling emissions, for the current practice of limiting the maximum compression ratio of SI engines to about 10:1. Near the end of compression, the mixture is well conditioned for combustion and the spark is discharged to initiate the combustion process. For best fuel economy, the combustion process must be completed as close as possible to TDC. This requires that the spark timing be controlled for varying operating speed and load conditions of the engine. Fuel metering and control, according to the engine load requirements, and with minimum variation from cylinder to cylinder and cycle to cycle, is essential for good fuel economy, power output, and emission control of the engine. Both carburetors and fuel injection systems are used for fuel-metering control. Because of the superior control capabilities of fuel injection systems, they are nearly universally used today in production automotive applications. Carburetors are used for applications with less-stringent emission requirements, like small engines for lawn and garden equipment. Optimum fuel economy, coinciding with maximum thermal efficiency, is obtained at part throttle with a lean mixture as a result of the fact that the heat release from lean mixtures suffers minimal losses from dissociation and variation of specific heat effects when compared with stoichiometric and rich fuel-air ratios. Maximum power is obtained at full throttle with a slightly rich mixture, an indication of the full utilization of the air inside the cylinders. Idling, with a nearly closed throttle, requires a rich mixture because of the high percentage of exhaust gas residuals that remains in the cylinders. The fuel-air mixture requirement under transient operation, such as acceleration, requires a rich mixture to compensate for the reduced evaporation caused by the sudden opening of the throttle. Cold starting also requires a rich mixture to ensure the vaporization of sufficient amounts of the highly volatile components in the fuel to achieve proper ignition. The combustion processes in SI

engines can be divided into two categories, normal and abnormal. The normal combustion process occurs in three stages: initiation of combustion, flame propagation, and termination of combustion. Combustion normally starts across the spark plug gap when the spark is discharged. The fuel molecules in and around the spark discharge zone are ignited and a small amount of energy is released. The important criterion for the initial reaction to be self-sustaining is that the rate of heat release from the initial combustion be larger than the rate of heat transfer to the surroundings. The factors that play an important role in making the initial reaction self-sustaining, and thereby establishing a flame kernel, are the ignition energy level, the spark plug gap, the fuel-air ratio, the initial turbulence, and the condition of the spark plug electrodes. After a flame kernel is established, a thin spherical flame front advances from the spark plug region progressively into the unburned mixture zone. Flame propagation is supported and accelerated by two processes. First, the combined effect of the heat transfer from the high-temperature flame region and the bombardment by the active radicals from the flame front into the adjacent unburned zone raises the temperature and accelerates the rate of reactivity of the unburned mixture region directly adjacent to the flame front. This helps to condition and prepare this zone for combustion. Second, the increase in the temperature and pressure of the burned gases behind the flame front will cause it to expand and progressively create thermal compression of the remaining unburned mixture ahead of the flame front. It is expected that the flame speed will be low at the start of combustion, reach a maximum at about half the flame travel, and decrease near the end of combustion. Overall, the flame speed is strongly influenced by the degree of turbulence in the combustion chamber, the shape of the combustion chamber, the mixture strength, the type of fuel, and the engine speed. When the flame front approaches the walls of the combustion chamber, the high rate of heat transfer to the walls slows down the flame propagation and finally the combustion process terminates close to the walls because of surface quenching. This leaves a thin layer of unburned fuel close to the combustion chamber walls which shows up in the exhaust as unburned hydrocarbons.

Abnormal combustion may occur in SI engines associated with two combustion phenomena: knock and surface ignition. Knock occurs near the end of the combustion process if the end portion of the unburned mixture, which is being progressively subjected to thermal compression, autoignites prematurely before the flame front reaches it. As a result of the sudden energy release, a violent pressure wave propagates back and forth across the combustion chamber, causing the walls or other parts of the engine to vibrate, producing a sharp metallic noise called knock. If knock persists for a period of time, the high rate of heat transfer caused by the traveling high pressure and temperature wave may overheat the spark plug electrode or ignite carbon deposits that may be present in the combustion chamber, causing uncontrolled combustion and pre-ignition. As a result, loss of power and serious engine damage may occur. Knock is sensitive to factors that increase the temperature and pressure of the end portion of the unburned mixture, as well as to fuel composition and other time factors. Factors that increase the probability of knock include increasing the temperature of the mixture by increasing the charge intake temperature, increasing the compression ratio, or turbo/supercharging; increasing the density of the mixture by turbo/supercharging or increasing the load; advancing the spark timing; increasing the time of exposure of the end portion of the unburned mixture to auto-ignition conditions by increasing the length of flame travel or decreasing the engine speed and turbulence; and using low-octane fuel and/or maximum power fuel-air ratios. Engine design factors that affect knock in SI engines include the shape of the combustion chamber and the location of the spark plug and inlet and exhaust valves relative to the location of the end portion of the unburned mixture.

Surface ignition is the ignition of the unburned mixture by any source in the combustion chamber other than the normal spark. Such sources could include overheated exhaust valves or spark plug electrodes, glowing carbon deposits, or other hot spots. Surface ignition will create secondary flame fronts which cause high rates of pressure rise resulting in a low-pitched, thudding noise accompanied by engine roughness. Severe surface ignition, especially when it occurs before spark ignition,

may cause serious structural and/or component damage to the engine (Kahraman, 2005; from Heywood, 1998).

3.1.4. Exhaust Emissions

The products of combustion from ICEs contain several constituents that are considered hazardous to human health, including CO, unburned HC, and oxides of nitrogen (NO_x). The concentration of gaseous emissions in the engine exhaust gases are usually measured in parts per million (ppm) or percent (%) by volume. There are principal schemes present for the reduction of these pollutants.

CO is a colorless, odorless, and tasteless gas that is highly toxic to humans. Breathing air with a small volumetric concentration (0.3%) of CO in an enclosed space can cause death in a short period of time. CO results from the incomplete combustion of hydrocarbon fuels. One of the main sources of CO production in SI engines is the incomplete combustion of the rich fuel mixture that is present during idling and maximum power steady state conditions and during such transient conditions as cold starting, warm-up, and acceleration. Uneven fuel distribution, poor condition of the ignition system, very lean combustion, and slow CO reaction kinetics also contribute to increased CO production in SI engines.

When unburned hydrocarbons combine with NO_x in the presence of sunlight, ozone and photochemical oxidants form that can adversely affect human health. Certain HC's are also considered to be carcinogenic. The principal cause of HC in SI engines is incomplete combustion of the fuel-air charge, resulting in part from flame quenching of the combustion process at the combustion chamber walls, and engine misfiring. Additional sources in four-stroke engines may include fuel mixture trapped in crevices of the top ring land of the piston and out gassed fuel during the expansion stroke that was absorbed into the lubricating oil film during intake. In two stroke SI engines, the scavenging process often results in a portion of the fresh mixture exiting the exhaust port before it closes, resulting in large HC emissions. Engine variables that affect HC emissions include the fuel-air ratio, intake air temperature, and cooling water temperature.

Oxides of nitrogen; nitric oxide (NO) is formed from the combination of nitrogen and oxygen present in the intake air under the high-temperature conditions that result from the combustion process. As the gas temperature drops during the expansion stroke, the reaction is frozen, and levels of NO persist in the exhaust products far in excess of the equilibrium level at the exhaust temperature. In the presence of additional oxygen in the air, some NO transforms to nitrogen dioxide (NO₂), a toxic gas. The NO and NO₂ combined are referred to as oxides of nitrogen or NO_x. The production of NO_x is in general aggravated by conditions that increase the peak combustion temperature. In SI engines the most important variables that affect NO_x production are the air/fuel ratio, spark timing, intake air temperature, and amount of residual combustion products remaining in the cylinder after exhaust.

Lean mixture combustion, which promotes good thermal efficiency, also results in low HC and CO production but causes high levels of NO_x emission. Increasing the fuel/air ratio to reduce NO_x results in increased CO and HC emissions. Approaches to reduce total emissions fall under two categories; the first concentrates on engine design and fuel modifications and the second involves treatment of exhaust gases after leaving the engine. In SI engines, the first approach focuses on addressing engine variables and design modifications which improve in-cylinder mixing and combustion in an effort to reduce CO and HC emissions. To reduce NO_x, attention is focused on factors that reduce peak combustion temperature and reduce the oxygen available in the flame front. Design and operating parameters that have been implemented or modified for decreased emissions include compression ratio reduction, increased coolant temperature, modification of the combustion chamber shape to minimize surface-to-volume ratio and increase turbulence, improvement of intake manifold design for better charge distribution, use of fuel injection instead of carburetors for better mixture control, use of exhaust gas recirculation to reduce NO_x by lowering combustion temperatures, positive crankcase ventilation to reduce HC, and increased aromatic content in gasoline.

In the second approach, several devices have been developed for after treatment of exhaust products. A thermal reactor may be used to oxidize HC and CO. These typically consist of a well-insulated volume placed close to the exhaust

manifold, with internal baffles to increase the gas residence time and an air pump to supply fresh oxygen for the oxidation reactions. Thermal reactors are ineffective for NO_x reduction and thus have limited application. Catalytic converters utilize a catalyst, typically a noble metal such as platinum, rhodium, or palladium, deposited on a ceramic substrate to promote reactions at lower temperatures. Two types are in use, oxidation converters and reduction converters. Oxidation catalytic converters use the excess air available in lean mixtures (or supplied from an external air pump) to oxidize CO and HC emissions. Reduction catalytic converters operate with low levels of oxygen to cause reduction of NO_x . Sometimes, dual catalytic converters are employed to treat all three pollutants with a reducing converter, to reduce NO_x , placed upstream of an oxidation converter for treating CO and HC. This arrangement requires that the engine be operated with a rich mixture which decreases fuel economy. Three-way catalytic converters are a recent development that permits treatment of NO_x , CO, and HC in a single device, thus reducing size and weight of the exhaust system. Proper operation of a three-way catalyst requires very nearly stoichiometric combustion. If the combustion is too lean, NO_x is not adequately reduced, and if it is too rich, HC and CO are not adequately oxidized. There is a narrow band for equivalence ratio from about 0.999 to 1.007 within which conversion efficiency is 80% or better for all three pollutants. Maintaining engine operation within this narrow mixture band requires a closed-loop fuel-metering system that utilizes an oxygen sensor placed in the exhaust system to monitor excess oxygen and control the fuel injection to maintain near stoichiometric combustion (Kummer, 1980).

3.1.5. Combustion Stoichiometry

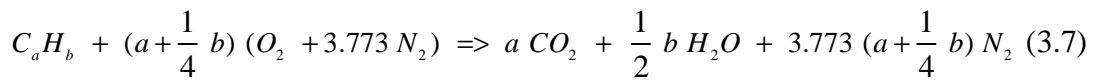
To develop a relation between the composition of the reactants (fuel and air) of a combustible mixture and the composition of the products, it is necessary to meter the air inlet and fuel supply rate.

The ratio of the air mass flow rate \dot{m}_a to the fuel mass flow rate \dot{m}_f is called the air fuel/ratio (A/F).

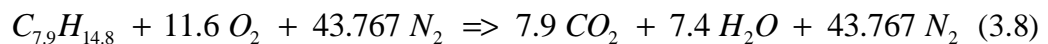
$$A/F = \frac{\dot{m}_a}{\dot{m}_f} \quad (3.6)$$

The normal operating of A/F for a conventional SI engine using gasoline fuel is between [12,18].

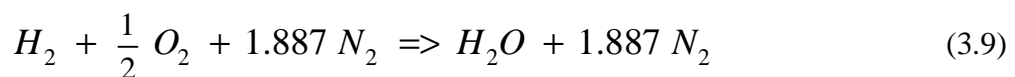
The relation between the composition of the reactants and the composition of the products depends only on the conservation of mass of each chemical element in the reactants, only the relative elemental composition of the fuel and the relative proportions of fuel and air are needed. If sufficient oxygen is available, a hydrocarbon fuel can be completely oxidized. The carbon in the fuel is then converted to carbon dioxide CO_2 and the hydrogen to water H_2O . The general equation for the complete combustion of one mole of a hydrocarbon with air:



This is the equation for the stoichiometric (theoretical) proportions of fuel and air. That is, just enough air is present to oxidize all of the fuel. It is obvious that the stoichiometric air/fuel or fuel/air ratios depend on the chemical fuel composition. For gasoline (a reasonable approximation is $C_{7.9}H_{14.8}$) the equation becomes:



Whereas for hydrogen H_2 it is:



The molecular weights of oxygen, atmospheric nitrogen, atomic carbon, and atomic hydrogen are 32, 28.16, 12.001, and 1.008, respectively. Substituting these values and a simplification $y = b/a$ in Equation 3.6 results in the expression:

$$(A/F)_s = \frac{34.56(4+y)}{12.011+1.008y} \quad (3.10)$$

$$\text{Gasoline} = \text{C}_{7.9}\text{H}_{14.8} \quad (A/F)_s = 14.6$$

$$\text{Hydrogen} = \text{H}_2 \quad (A/F)_s = 34.3$$

Fuel-air mixtures with more than or less than the stoichiometric air requirement can be burned. With excess air or fuel-lean combustion, the extra air appears in the products in unchanged form. With less than the stoichiometric air requirement, with fuel-rich combustion, there is insufficient oxygen to oxidize fully the fuel. The products are a mixture of CO_2 and H_2O with CO and hydrogen as well as N_2 . The product composition cannot be determined from an element balance alone and additional assumption about the chemical composition of the product species must be made. Since the composition of the combustion products is significantly different for fuel-lean and fuel-rich mixtures, and because the stoichiometric fuel/air ratio depends on the fuel composition, the ratio of the actual fuel/air ratio to the stoichiometric ratio (or its inverse) is a more informative parameter for defining mixture composition. The fuel/air equivalence ratio f :

$$f = \frac{(F/A)_{actual}}{(F/A)_s} \quad (3.11)$$

The inverse of f , the relative air/fuel ratio l ,

$$f^{-1} = I = \frac{(F/A)_s}{(F/A)_{actual}} \quad (3.12)$$

For fuel-lean mixtures: $f < 1$ or $I > 1$

For stoichiometric mixtures: $f = I = 1$

For fuel-rich mixtures: $f > 1$ or $I < 1$

In practice, although with excess air condition, the composition of the products of combustion does not occur as in Equation 2.6. At normal combustion temperatures significant dissociation of CO_2 and of H_2O occurs. Whether, at low temperatures, recombination brings the product composition to that indicated by these overall chemical equilibrium equations depends on the rate of cooling of the product gases (Kummer, 1980).

3.2. Compression Ignition (CI) Engine

CI engine is quite different from that in an SI engine. Whereas combustion in an SI engine is essentially a flame front moving through a homogeneous mixture, combustion in a CI engine is an unsteady process occurring simultaneously at many spots in a very non-homogeneous mixture at a rate controlled by fuel injection. Air intake into the engine is unthrottled, with engine torque and power output controlled by the amount of fuel injected per cycle. Because the incoming air is not throttled, pressure in the intake manifold is consistently at a value close to one atmosphere (Pulkrabek, 2004). This makes the pump work loop of the engine cycle shown in Figure 3.5 very small, with a corresponding better thermal efficiency compared to an SI engine.

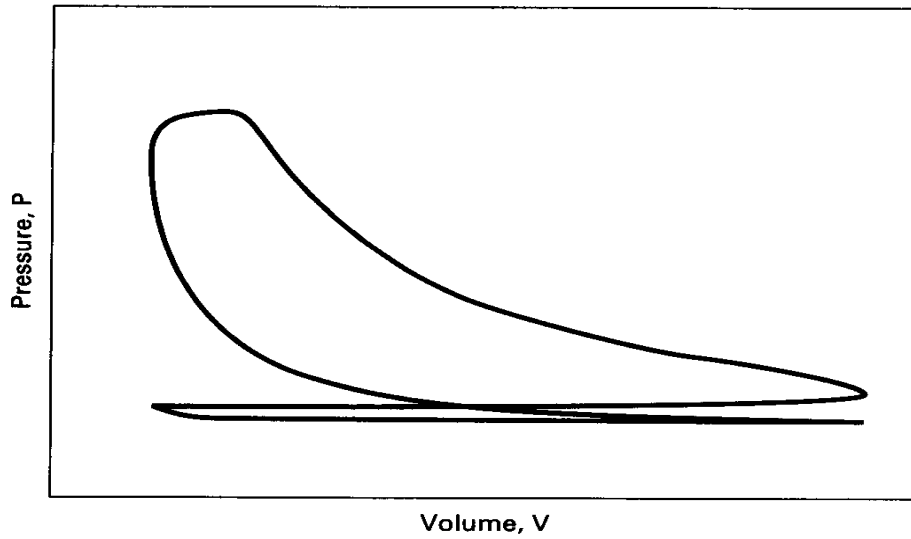


Figure 3.5. Indicator diagram of a modern CI engine operating on a four stroke cycle (Pulkrabek, 2004)

This is especially true at low speeds and low loads when an SI engine would be at part throttle with a large pump work. Thermal efficiency (h_{th}) for ideal diesel cycle is given as:

$$(h_{th})_{diesel} = 1 - \frac{1}{r^{g-1}} \left[\frac{a^g - 1}{g(a-1)} \right] \quad (3.13)$$

where,

r : compression ratio

a : cut-off ratio (ratio between the end and start volume for the combustion phase)

g : ratio of specific heats (C_p / C_v)

An engine ideally would be CI but would operate on the Otto cycle. CI would operate on the more efficient higher compression ratios, while constant-volume combustion of the Otto cycle would give higher efficiency for a given compression ratio.

The modern high-speed CI engine accomplishes this in part by a simple operating change from early diesel engines. Instead of injecting the fuel late in the compression stroke near TDC, as was done in early engines, modern CI engines start

to inject the fuel much earlier in the cycle, somewhere around 20° BTDC. The first fuel then ignites late in the compression stroke, and some of the combustion occurs almost at constant volume at TDC, much like the Otto cycle. Peak pressure still remains high into the expansion stroke due to the finite time required to inject the fuel. The last of the fuel is still being injected at TDC, and combustion of this fuel keeps the pressure high into the expansion stroke. The resulting cycle shown in Figure 3.5 is a cross between an SI engine cycle and the early CI cycles. The air-standard cycle used to analyze this modern CI engine cycle is called a “Dual cycle”, or sometimes a “Limited Pressure cycle” (Figure 3.6) (Pulkrabek, 2004).

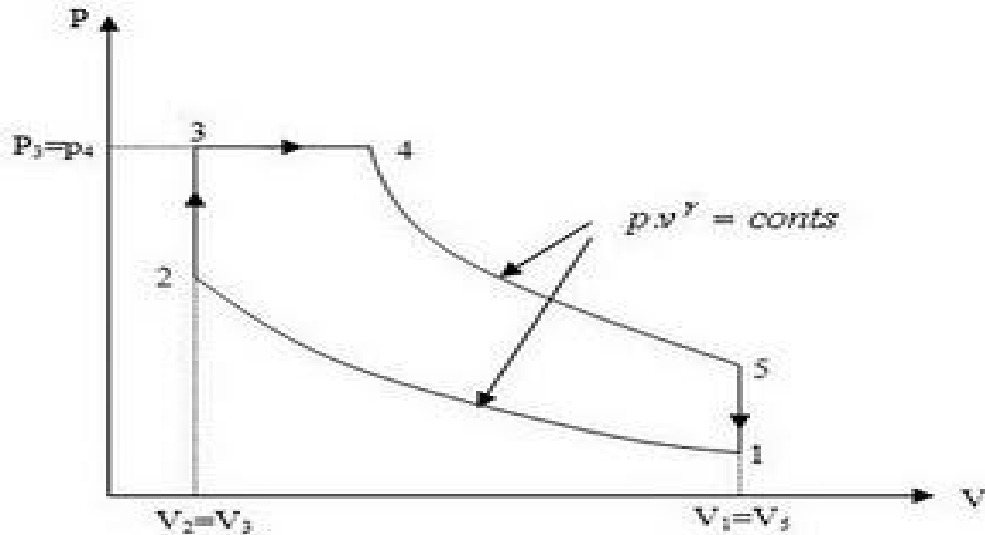


Figure 3.6. Pressure-specific volume coordinates of Dual cycle (Pulkrabek, 2004)

It is a dual cycle because the heat input process of combustion can best be approximated by a dual process of constant volume followed by constant pressure. It can also be considered a modified Otto cycle with a limited upper pressure. Thermal efficiency of a dual cycle is given as:

$$(h_{th})_{dual} = 1 - \frac{1}{r^{g-1}} \left[\frac{r_p r_c^g - 1}{(r_p - 1) + g r_p (r_c - 1)} \right] \tag{3.14}$$

where,

$$r_p = \frac{P_3}{P_2} \quad \text{and} \quad r_c = \frac{v_4}{v_3} \quad (3.15)$$

For CI engines, only air is contained in the cylinder during the compression stroke, and much higher compression ratios are used in CI engines. Compression ratios of modern CI engines range from 12 to 24. Compared to normal SI engines, high thermal efficiencies (fuel conversion efficiencies) are obtained when these compression ratios are used in Eqs. (3.13) and (3.14). However, because the overall air-fuel ratio on which CI engines operate is quite lean (equivalence ratio $\bar{f} = 0.8$), less brake power output is often obtained for a given engine displacement.

3.2.1. Combustion in Compression Ignition Engines

Fuel is injected into the cylinders late in the compression stroke by one or more injectors located in each cylinder combustion chamber. Injection time is usually about 20° of crankshaft rotation, starting at about 15° BTDC and ending about 5° ATDC. Ignition delay is fairly constant in real time, so at higher engine speeds fuel injection must be started slightly earlier in the cycle.

In addition to the swirl and turbulence of the air, a high injection velocity is needed to spread the fuel throughout the cylinder and cause it to mix with the air. After injection the fuel must go through a series of events to assure the proper combustion process:

1. *Atomization:* Fuel drops break into very small droplets. The smaller the original drop size emitted by the injector, the quicker and more efficient will be this atomization process.

2. *Vaporization:* The small droplets of liquid fuel evaporate to vapor. This occurs very quickly due to the hot air temperatures created by the high compression of CI engines. High air temperature needed for this vaporization process requires a minimum compression ratio in CI engines of about 12:1. About 90% of the fuel injected into the cylinder has been vaporized within 0.001 second after injection. As the first fuel evaporates, the immediate surroundings are cooled by evaporative

cooling. This greatly affects subsequent evaporation. Near the core of the fuel jet, the combination of high fuel concentration and evaporative cooling will cause adiabatic saturation of fuel to occur. Evaporation will stop in this region, and only after additional mixing and heating will this fuel be evaporated.

3. *Mixing*: After vaporization, the fuel vapor must mix with air to form a mixture within the AF range which is combustible. This mixing comes about because of the high fuel injection velocity added to the swirl and turbulence in the cylinder air. Figure 3.7 shows the non-homogeneous distribution of air-fuel ratio that develops around the injected fuel jet. Combustion can occur within the equivalence ratio limits of $\bar{f} = 1.8$ (rich) and $\bar{f} = 0.8$ (lean).

4. *Self-Ignition*: At about 8° BTDC, 6-8° after the start of injection, the air-fuel mixture starts to self-ignite. Actual combustion is preceded by secondary reactions, including breakdown of large hydrocarbon molecules into smaller species and some oxidation. These reactions, caused by the high-temperature air, are exothermic and further raise the air temperature in the immediate local vicinity. This finally leads to an actual sustained combustion process.

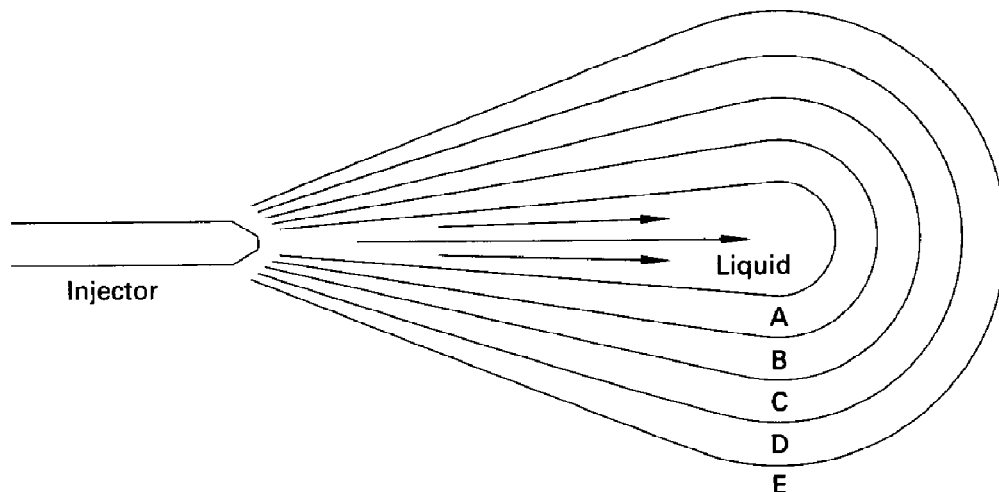


Figure 3.7. Fuel jet of a CI engine showing air-fuel vapor zones around the inner liquid core. The liquid core is surrounded by successive zones of which are (A) too rich to burn, (B) rich combustible, (C) stoichiometric, (D) lean combustible and (E) too lean to burn. Self-ignition starts mainly in zone B. Solid carbon soot is generated mainly in zone A and B (Pulkrabek, 2004)

5. *Combustion:* Combustion starts from self-ignition simultaneously at many locations in the slightly rich zone of the fuel jet, where the equivalence ratio is $f = 1$ to 1.5 (zone B in Fig. 3.7). At this time, somewhere between 70% and 95% of the fuel in the combustion chamber is in the vapor state. When combustion starts, multiple flame fronts spreading from the many self-ignition sites quickly consume all the gas mixture which is in a correct combustible air-fuel ratio, even where self-ignition wouldn't occur. This gives a very quick rise in temperature and pressure within the cylinder, shown in Figure 3.8. The higher temperature and pressure reduce the vaporization time and ignition delay time for additional fuel particles and cause more self-ignition points to further increase the combustion process. Liquid fuel is still being injected into the cylinder after the first fuel is already burning. After the initial start of combustion when all the air-fuel mixture that is in a combustible state is quickly consumed, the rest of the combustion process is controlled by the rate at which fuel can be injected, atomized, vaporized, and mixed into the proper AF. This rate of combustion, now controlled by injection rate, can be seen in Figure 3.8 in the slower pressure rise that occurs after the initial fast rise. Combustion lasts for about 40° to 50° of engine rotation, much longer than the 20° of fuel injection. This is because some fuel particles take a long time to mix into a combustible mixture with the air, and combustion therefore lasts well into the power stroke. This can be seen in Figure 3.8, where the pressure remains high until the piston is 30° - 40° ATDC. About 60% of the fuel is burned in the first third of combustion time. Burning rate increases with engine speed, so the burn angle remains about constant. During the main part of the combustion process, anywhere from 10% to 35% of the fuel vapor in the cylinder will be in a combustible AF.

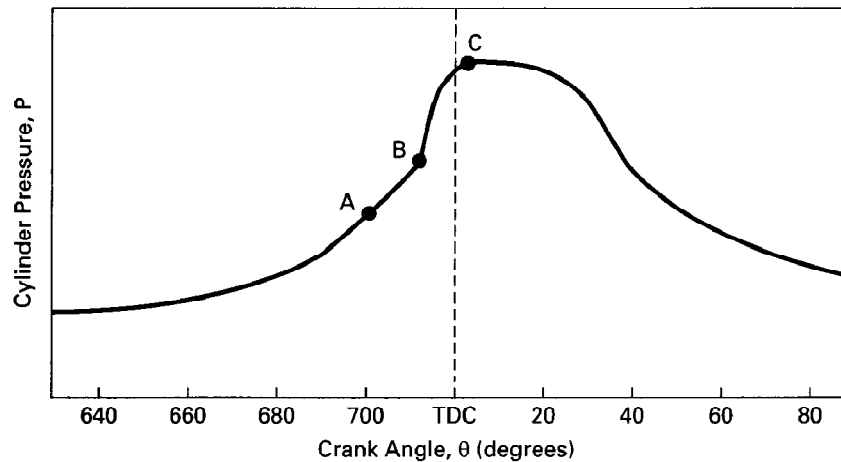


Figure 3.8. Cylinder pressure as a function of crank angle for a CI engine. Point A is where fuel injection starts, A to B is ignition delay, and point C is the end of fuel injection. If the cetane number of the fuel is too low, a greater amount of fuel will be injected during ignition delay time. When combustion then starts, the additional fuel will cause the pressure at point B to increase too fast, resulting in a rough engine cycle (Pulkrabek, 2004).

3.2.2. Exhaust Emissions

Catalytic converters are used with CI engines but are not efficient at reducing NO_x due to their overall lean operation. HC and CO can be adequately reduced, although there is greater difficulty because of the cooler exhaust gases of a CI engine (because of the larger expansion ratio). This is counter-balanced by the fact that less HC and CO are generated in the lean burn of the CI engine. NO_x is reduced in a CI engine by the use of EGR, which keeps the maximum temperature down. EGR and lower combustion temperatures, however, contribute to an increase in solid soot. Platinum and palladium are two main catalyst materials used for converters on CI engines. They promote the removal of 30-80% of the gaseous HC and 40-90% of the CO in the exhaust. The catalysts have little effect on solid carbon soot but do remove 30-60% of the total particulate mass by oxidizing a large percent of the HC absorbed on the carbon particles. Diesel fuel contains sulfur impurities, and this leads to poisoning of the catalyst materials. However, this problem is being reduced as legal levels of sulfur in diesel fuels continue to be lowered.

CI engine systems are equipped with particulate traps in their exhaust flow to reduce the amount of particulates released to the atmosphere. Traps are filter-like systems often made of ceramic in the form of a monolith or mat, or else made of metal wire mesh. Traps typically remove 60-90% of particulates in the exhaust flow. As traps catch the soot particles, they slowly fill up with the particulates. This restricts exhaust gas flow and raises the back pressure of the engine. Higher back pressure causes the engine to run hotter, the exhaust temperature to rise, and fuel consumption to increase. To reduce this flow restriction, particulate traps are regenerated when they begin to become saturated. Regeneration consists of combusting the particulates in the excess oxygen contained in the exhaust of the lean-operating CI engine.

Carbon soot ignites at about 550°-650°C, while CI engine exhaust is 150°-350°C at normal operating conditions. As the particulate trap fills with soot and restricts flow, the exhaust temperature rises but is still not high enough to ignite the soot and regenerate the trap. In some systems, automatic flame igniters are used which start combustion in the carbon when the pressure drop across the trap reaches a predetermined value. These igniters can be electric heaters or flame nozzles that use diesel fuel. If catalyst material is installed in the traps, the temperature needed to ignite the carbon soot is reduced to the 350°-450°C range. Some such traps can automatically regenerate by self-igniting when the exhaust temperature rises from increased back pressure. Other catalyst systems use flame igniters.

Another way of lowering the ignition temperature of the carbon soot and promoting self-regeneration in traps is to use catalyst additives in the diesel fuel. These additives generally consist of copper compounds or iron compounds, with about 7 grams of additive in 1000 liters of fuel being normal. To keep the temperatures high enough to self-regenerate in a catalytic system, traps can be mounted as close to the engine as possible, even before the turbocharger.

On some larger stationary engines and on some construction equipment and large trucks, the particulate trap is replaced when it becomes close to filled. The removed trap is then regenerated externally, with the carbon being burned off in a furnace. The regenerated trap can then be used again.

Various methods are used to determine when soot buildup becomes excessive and regeneration is necessary. The most common method is to measure pressure drop in the exhaust flow as it passes through the trap. Pressure drop is also a function of exhaust flow rate, and this must be programmed into the regeneration controls. Another method used to sense soot buildup is to transmit radio frequency waves through the trap and determine the percent that is absorbed. Carbon soot absorbs radio waves while the ceramic structure does not. The amount of soot buildup can therefore be determined by the percent decrease in radio signal. This method does not readily detect soluble organic fraction (SOF).

Modern particulate traps are not totally satisfactory, especially for automobiles. They are costly and complex when equipped for regeneration, and long-term durability does not exist. An ideal catalytic trap would be simple, economical, and reliable; it would be self-regenerating; and it would impose a minimum increase in fuel consumption (Pulkrabek, 2004).

4. EXPERIMENTAL SET-UP AND PROCEDURE

4.1. Material and Method

In this experimental study, hydroxy gas (HHO) which is produced by the electrolysis process of different electrolytes ($\text{KOH}_{(\text{aq})}$, $\text{NaOH}_{(\text{aq})}$, $\text{NaCl}_{(\text{aq})}$) with different reactor (electrode) designs in a leak-proof plexiglass reactor (hydrogen generator) is used as a supplementary fuel in a single cylinder, two stroke, SI and a four stroke, four cylinder, direct injection CI engine without any modification. HHO was firstly sent to a water safety system to prevent backfire using a 1/3 water-filled pot before being sent to the intake manifold. Sensors are located on the container to observe excess growth of water temperature and gas pressure. Non-turn valve is used to prevent rising of gas pressure over 1 bar in the container. An electronic hydrogen leak detector is used for probable gas leaks.

HHO is used as a sole fuel for gasoline and diesel fuel is in brown color and the form of unseparated hydrogen and oxygen generated by the electrolysis process of water by a unique electrode design. Hydrogen and oxygen do not form into O_2 and H_2 molecules. They are in their mono-atomic state (a single atom per molecule). In this state, which is an unstable state of H_2O vapor, more energy is achieved compared to hydrogen burning with oxygen. Pulverized water clashes the fuel and they unite. Water becomes the core and the fuel tends to be the water shell (due to density difference). During compression stroke, pressure and heat increases, the water explodes to steam and consequently, the fuel gets atomized. After ignition, in-cylinder temperature increases rapidly which results water to be splitted into hydrogen and oxygen and reignition occurs which yields increased combustion efficiency. Due to the simultaneous production and consumption of hydrogen, no storage is necessary, which results in safe operation. HHO is generated and used as a sole fuel in SI and CI engines to benefit from peculiar features and minimize disadvantages of hydrogen.

Technical specifications of the SI and CI engines used in this experimental study are shown in Table 4.1 and Table 4.2 and photographs of the engines are

demonstrated in Figure 4.1 and 4.2, respectively. NETFREN trademarked hydraulic dynamometer is used which has torque range of 0-1700 Nm and speed range of 0-7500 rpm for CI engine tests. COUSINS trademarked hydraulic dynamometer is used which has torque range of 0-11 Nm and speed range of 0-5800 rpm for SI engine tests. Effects of HHO addition to the fuel on exhaust emissions and engine performance are investigated. All of the experiments are conducted in Çukurova University Automotive Engineering Department Laboratories.

Table 4.1. Technical specifications of the CI test engine

Configuration	Four-stroke, four cylinder diesel engine
Type	Direct injection with glow plug
Swept Volume (cm ³)	3567
Bore (mm)	104
Stroke (mm)	105
Maximum Torque (Nm)	255 at 1800 rpm

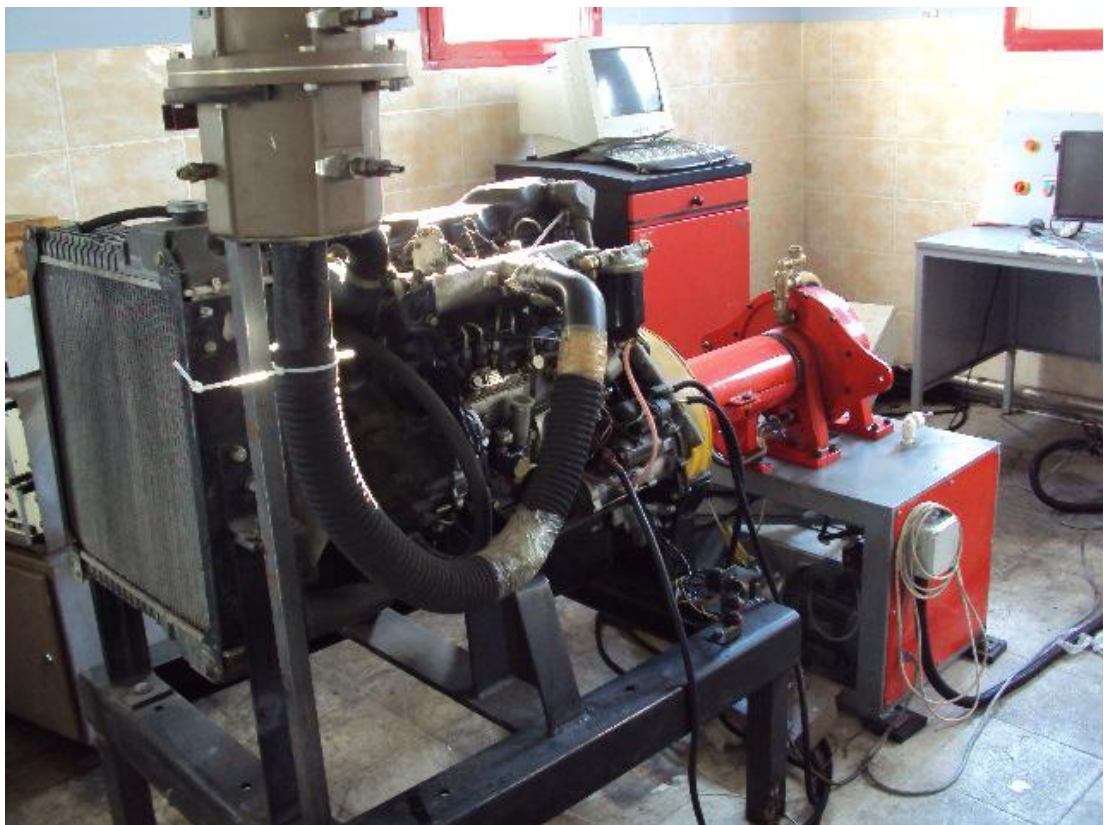


Figure 4.1. CI test engine

Table 4.2. Technical specifications of the SI test engine

Configuration	One-cylinder
Type	SI gasoline engine
Swept Volume (cm ³)	151
Bore (mm)	60
Stroke (mm)	54
Compression Ratio	8:1
Maximum Torque (Nm)	10.6 at 3000 rpm
Maximum Brake Power (kW)	4.4 at 4500 rpm

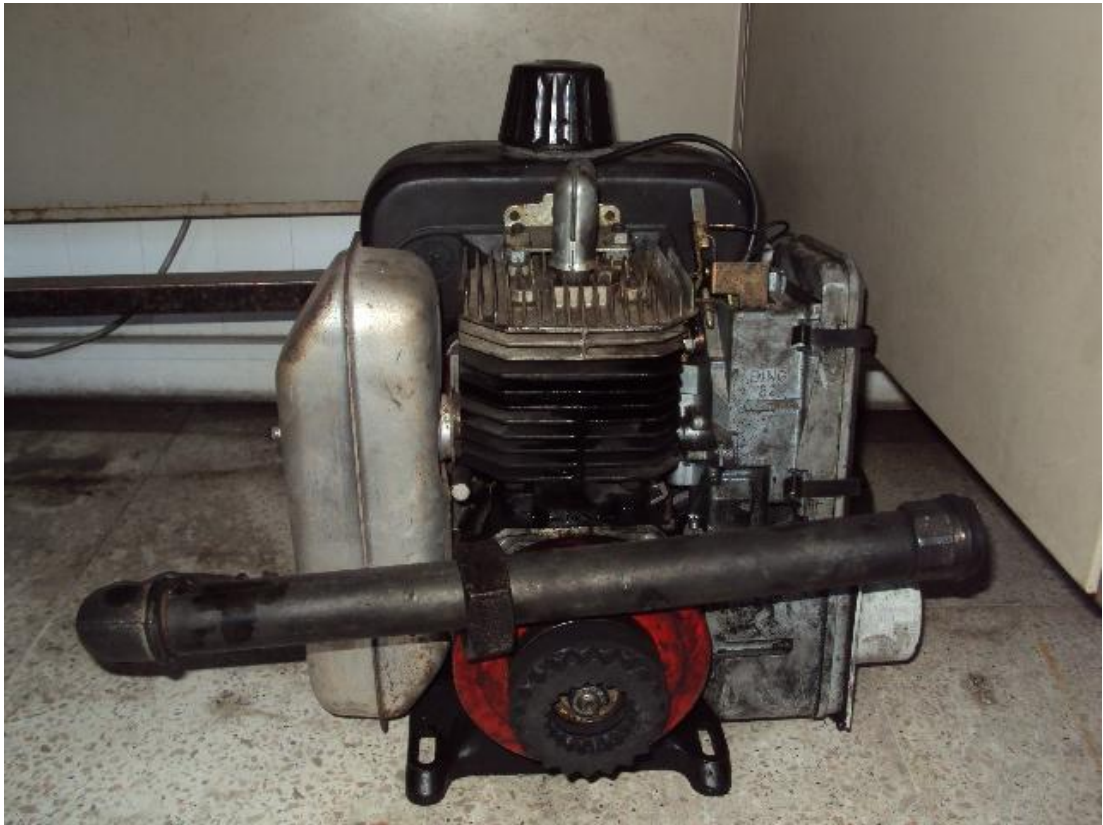


Figure 4.2. SI test engine

This experimental study consists of two stages. In the first stage, experiments are achieved to specify the most suitable catalyst. It is observed that NaOH is the best one. In the second stage, the best reactor design is chosen and observations depict that the plate type electrode-NaOH aqueous solution is the most appropriate and efficient combination for main experiments. Experiments on aqueous solutions of catalysts demonstrate that HHO flow rate increases in relation to mass fraction of catalyst in water. However, if the molality of NaOH in solution exceeds 1% by mass, current supplied from battery increased dramatically due to the too much reduction

of electrical resistance. The members are expressed in Sections 4.1.1 to 4.1.9 and the whole experimental set-up is shown in Section 4.2.

High corrosion resistance is considered as one of the major factors for electrode material due to its high corrosive working ambient. Therefore, electrodes used in this study are made of 316L stainless steel due to its high corrosion resistance. Plexiglass is used as material of reactor container due to its high resistance against a probable explosion and its transparency which allows a clear vision during the reaction. The lid material used for reactor container is polyamide due to its easy machinability in lathe. Basic system materials and members are seen in Table 4.3. Letters “D”, “L”, “T”, “W” and “H” represent “diameter”, “length”, “thickness”, “width” and “height”, respectively.

Table 4.3. Basic system materials and members

Item	Quantity
Cylindrical plexiglass (170mm x 400mm x 1,5mm / D x L x T)	1
316 L stainless steel electrode plates (115mm x 70mm x 2mm / D x L x T)	16
Polyamide rods (8mm x 190mm / D x L)	2
316 L stainless steel pipes (22mm x 146mm x 1.5mm / D x L x T)	6
316 L stainless steel pipes (28mm x 146mm x 1.5mm / D x L x T)	6
316 L stainless steel wire (2mm x 8500mm / D x L)	1
Cylindrical polyamide reactor lids (170mm x 75mm / D x L)	4
Cylindrical polyamide braces (120mm x 30mm / D x L)	2
Plexiglass plates (350mm x 130mm x 20mm / L x H x T)	2
HECU (expressed detailed in section 4.1.9)	1
Water level system (float)	1
Water tank (200mm x 200mm x 150mm / W x L x H)	1
DC device (automobile battery)	1
Multimeter	1
Gas analyzers (for SI and CI engines)	2
Catalysts (NaOH, KOH and NaCl)	~1% by mass

4.1.1. Reactors (Electrodes)

Three types of reactors are designed and manufactured for this experimental study. Three views and 3D models in CATIA and the photographs of the reactors are shown in Figure 4.3, 4.5 and 4.7.

4.1.1.1. Plate reactor

Figure 4.3 shows the photograph and the 3D model of the reactor and Figure 4.4 depicts the HHO flow rate in relation to mass fraction (%) of catalysts in aqueous solutions.

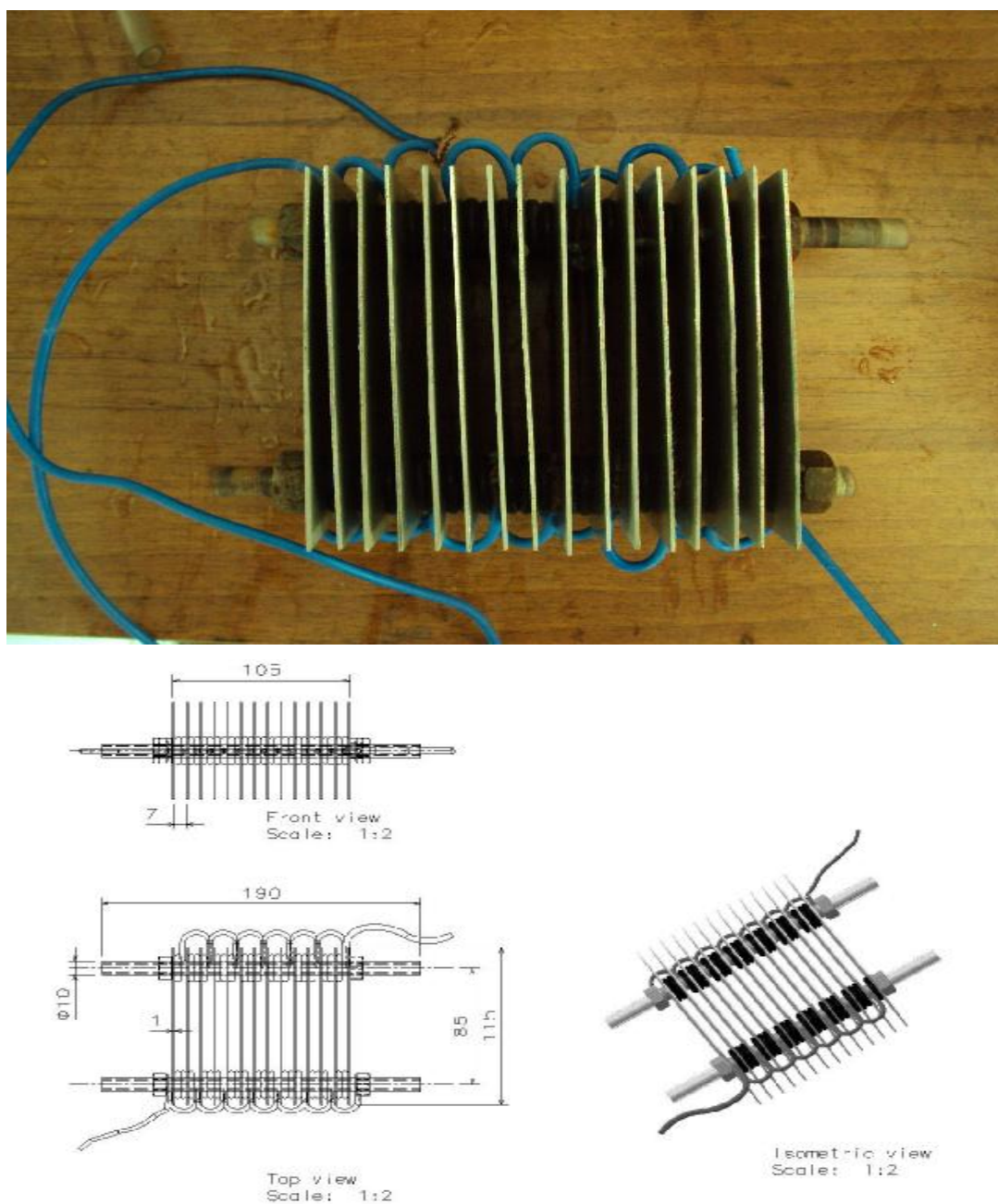


Figure 4.3. Photograph and CATIA drawings of the plate reactor

The electrical power (P) consumed during electrolysis process with plate reactor without HECU at mass fraction of about 1% is:

$$P = V \times I = 12 \text{ V} \times 10.6 \text{ A} = 127.2 \text{ W} \approx 0.13 \text{ kW} \quad (4.1)$$

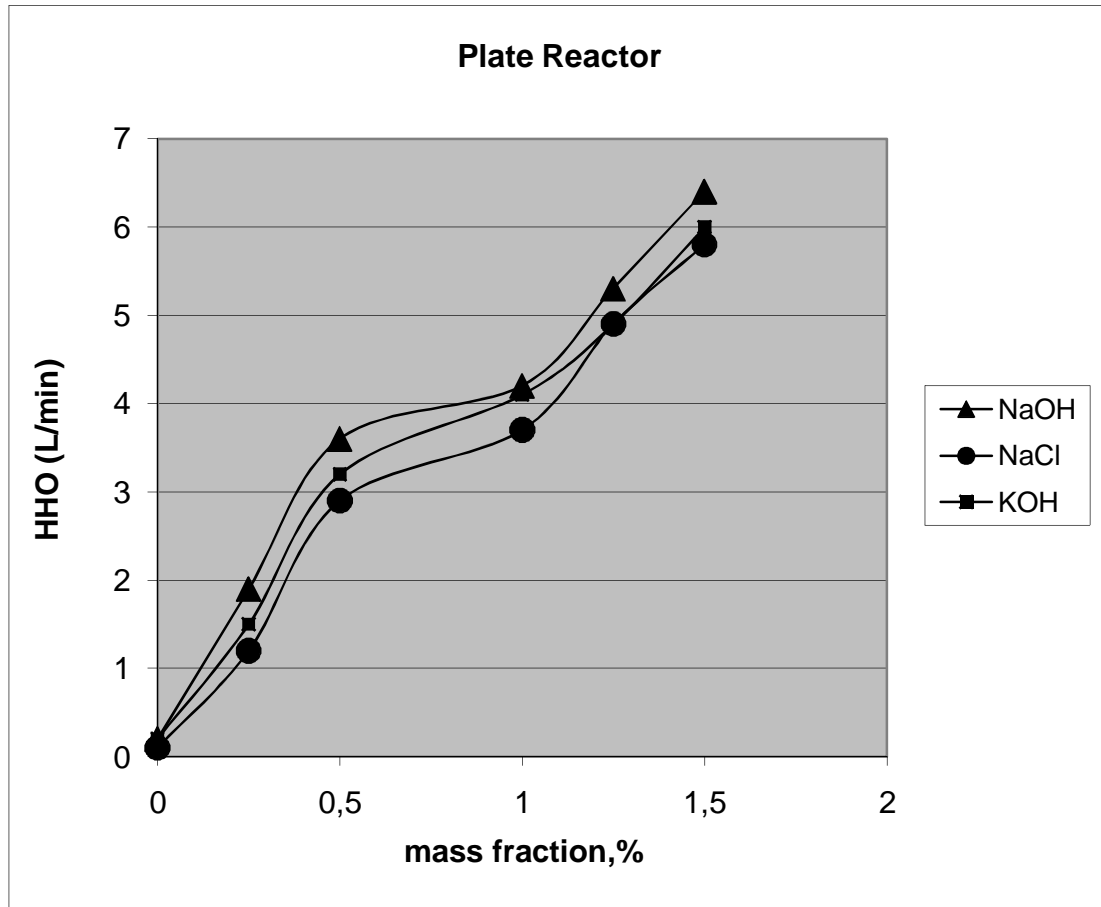


Figure 4.4. Variation of HHO flow rate with catalysts mass fraction

4.1.1.2. Cylindrical Reactor

Figure 4.5 shows the photograph and the 3D model of the electrode and Figure 4.6 depicts the HHO flow rate in relation to mass fraction (%) of catalysts in aqueous solutions.

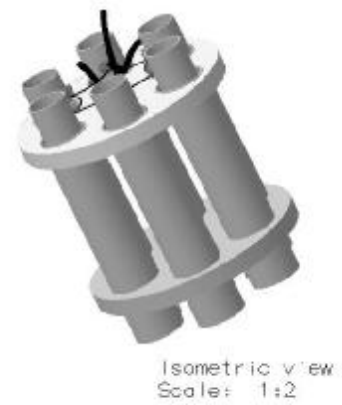
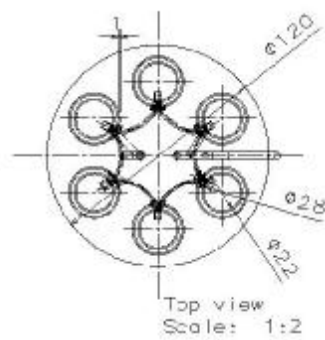
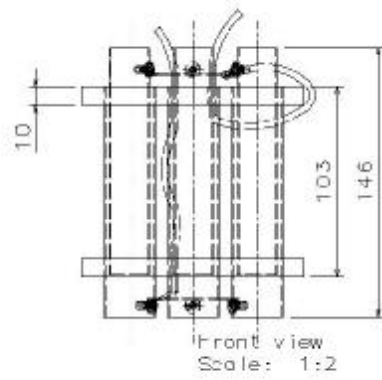


Figure 4.5. Photograph and CATIA drawings of the cylindrical reactor

The electrical power (P) consumed during electrolysis process with cylindrical electrode without HECU at mass fraction of about 1% is:

$$P = V \times I = 12 \text{ V} \times 12.2 \text{ A} = 146.4 \text{ W} \cong 0.15 \text{ kW} \quad (4.2)$$

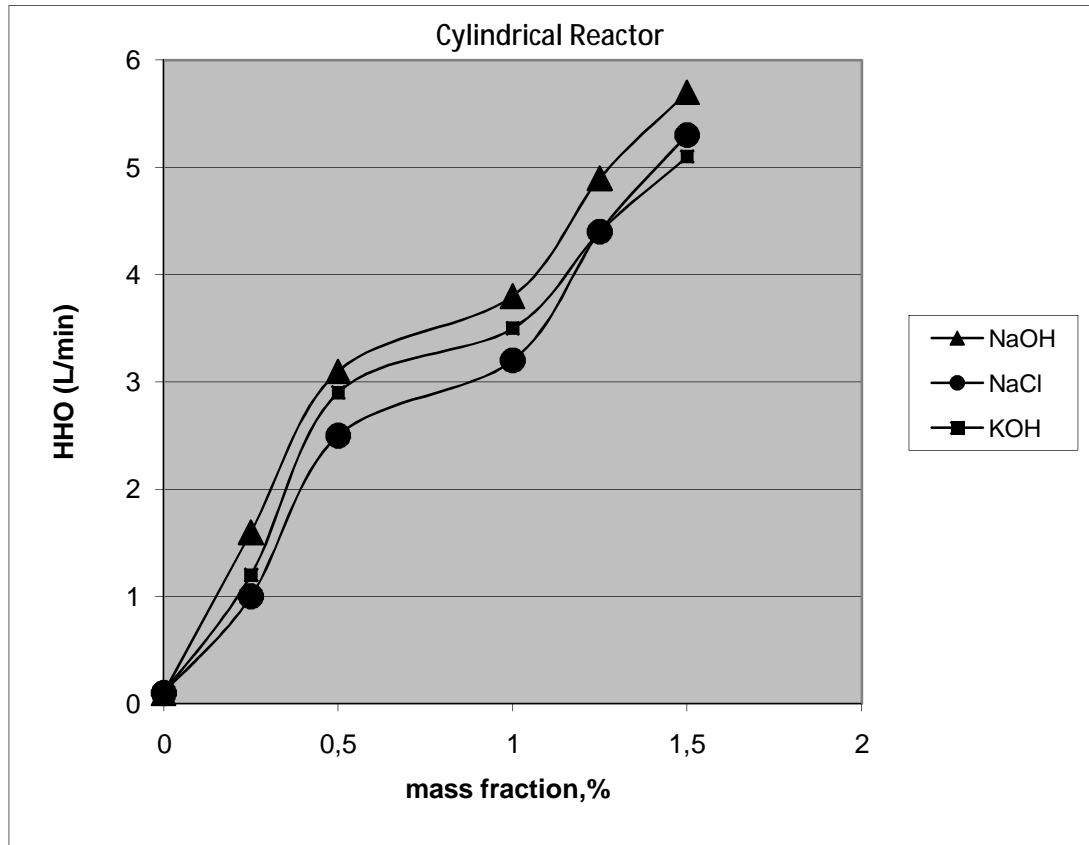


Figure 4.6. Variation of HHO flow rate with catalysts mass fraction

4.1.1.3. Wire Reactor

Figure 4.7 shows the photograph and the 3D model of the electrode and Figure 4.8 illustrates the HHO flow rate in relation to mass fraction (%) of catalysts in aqueous solutions.

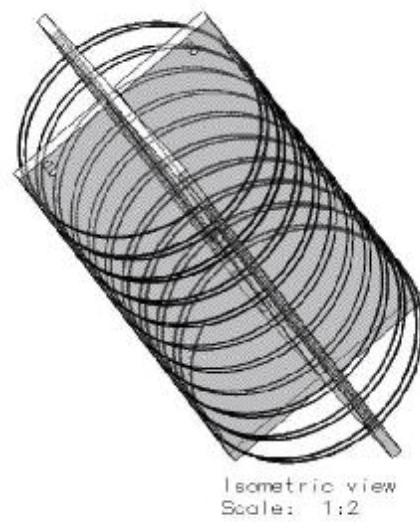
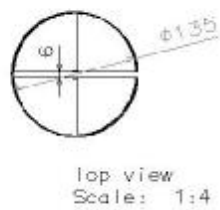
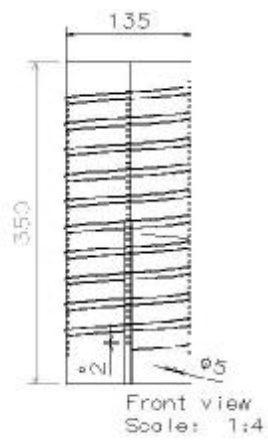


Figure 4.7. Photograph and CATIA drawings of the wire reactor

The electrical power (P) consumed during electrolysis process with wire electrode without HECU at mass fraction of about 1% is:

$$P = V \times I = 12 \text{ V} \times 14,1 \text{ A} = 169.2 \text{ W} \cong 0.17 \text{ kW} \quad (4.3)$$

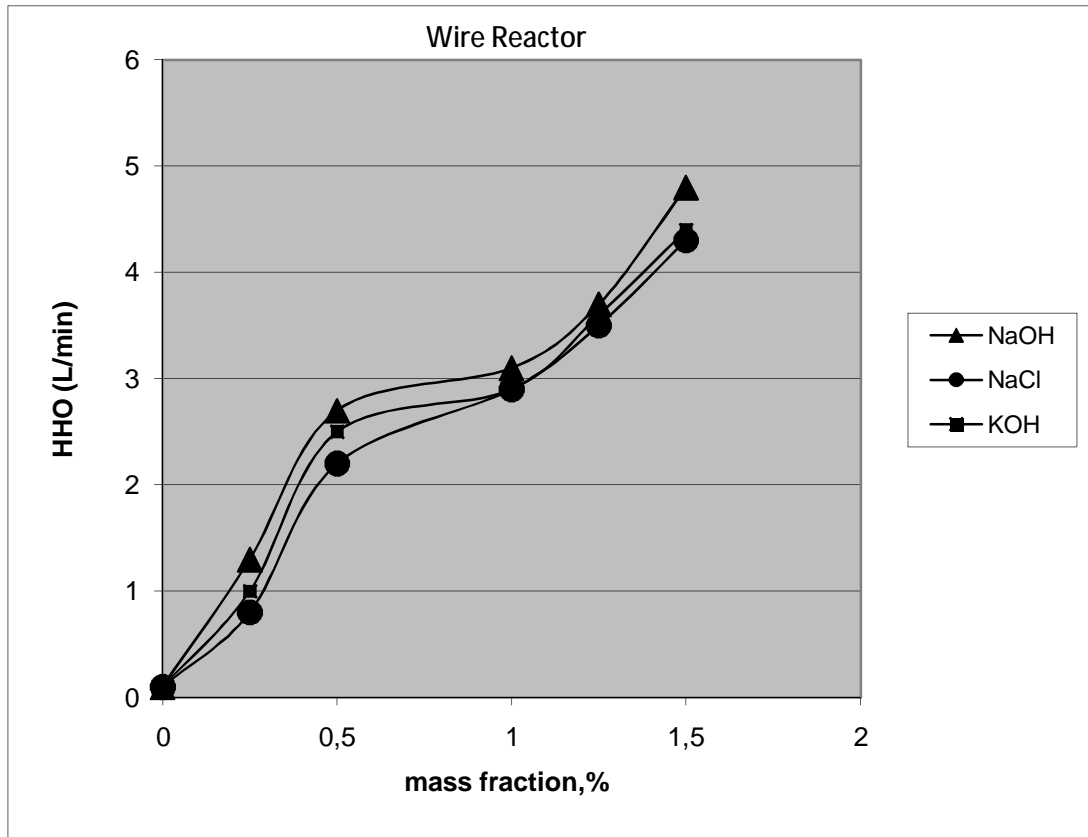


Figure 4.8. Variation of HHO flow rate with catalysts mass fraction

As seen from the graphs, the most efficient catalyst is NaOH and reactor is the plate type. Therefore, the experiments are achieved with plate reactor and about 1% NaOH solution by mass.

4.1.2. Reactor Container

Reactor container is a cylindrical pot in which the reaction takes place. Plexiglass is used as material of reactor container due to its high resistance against a probable explosion and its transparency which allows a clear vision during the

reaction. The lid material used for reactor container is polyamide due to its easy machinability in lathe. The photograph of the reactor container is seen in Figure 4.9.

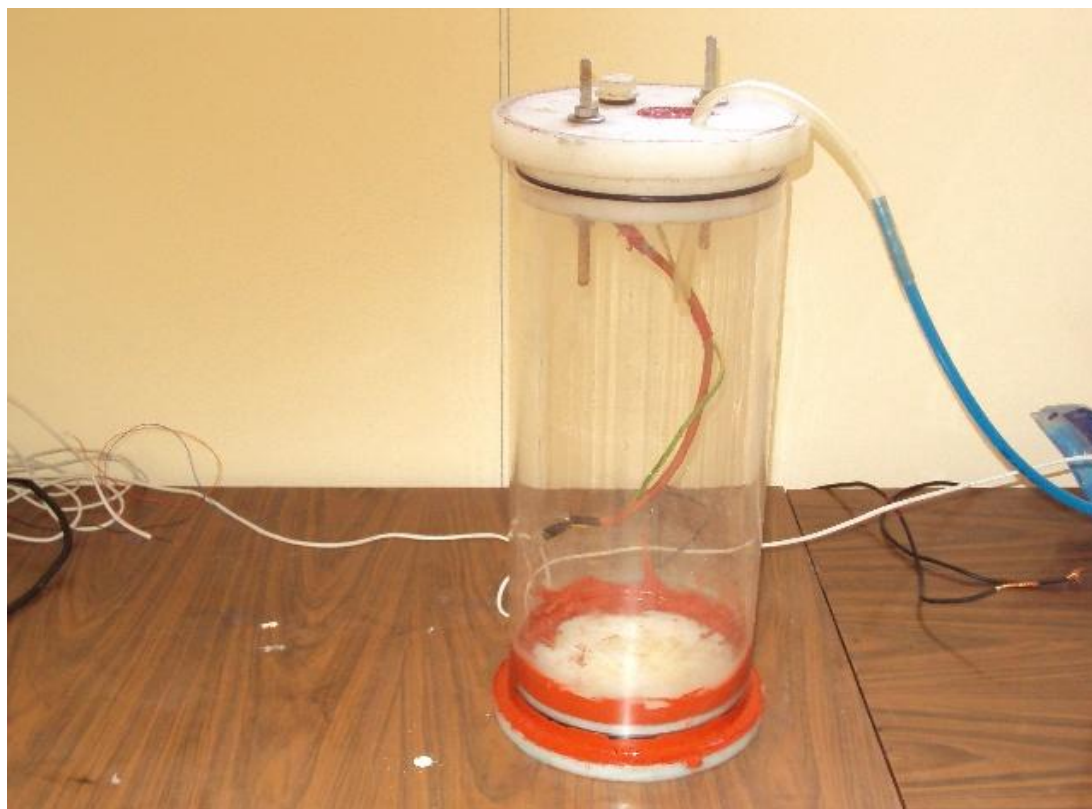


Figure 4.9. Plexiglass reactor container with polyamide lids

4.1.3. Water-level (Float) System

Water-level control (float) system is assembled into the reactor container to prevent the water level drop below the highest point of the reactor to minimize short circuits and a probable explosion through HHO expanded in reaction pot due to its high electrical conductivity.

4.1.4. Water Safety Pot

HHO is firstly sent to a water safety system to prevent backfire using a 1/3 water-filled pot before being sent to the intake manifold.

4.1.5. SFC Measurement Device

SFC for the SI test engine is measured via a device shown in Figure 4.10. SFC is estimated according to the engine's approximately 10 ml fuel consumption time for all speeds, which the measurements are achieved for, during the experiments. Then, the consumption time is converted to g/kWh.



Figure 4.10. SFC measurement device for the SI test engine

4.1.6. Gas Analyzers

Two different devices are used for measuring the exhaust compositions of the set-up. SUN MGA 1200 Exhaust Gas analyzer and its software are used for reading CO and HC emissions of the SI engine. It can also measure engine speed in terms of revolutions per minute (rpm), oil temperature and AFR. TESTO 350 XL gas analyzer and its software are used to measure CO and HC emissions of CI engine. The gas analyzers are seen in Figure 4.11.



Figure 4.11. Gas analyzers of SI and CI engines

4.1.7. Hydrogen Flowmeter

An electronic hydrogen flow meter is used to measure the hydrogen flow rate coming out from the reactor container. Photograph and the technical specifications of the flow meter are seen in Figure 4.12 and Table 4.4, respectively.



Figure 4.12. Electronic hydrogen flowmeter

Table 4.4. Technical specifications of the hydrogen flowmeter

Material	316 L stainless steel
Output Signal	0~5 VDC linear min. load 1000 Ω or 4~20mA linear, loop resistance 500 Ω
Input Power	24VDC standard, optional power supply 15VDC 115VAC, 220VAC@500mA151
Accuracy	$\pm 1\%$ (including linearity)
Repeatability	$\pm 0.15\%$ or better
Range	0~10 SCCM to 0~100 SLM
Response time	1 second

4.1.8. Multimeter

Electrolysis voltage, current and short circuits are checked via a digital multimeter. Photograph and detailed information about the multimeter is denoted in Figure 4.13 and Table 4.5.

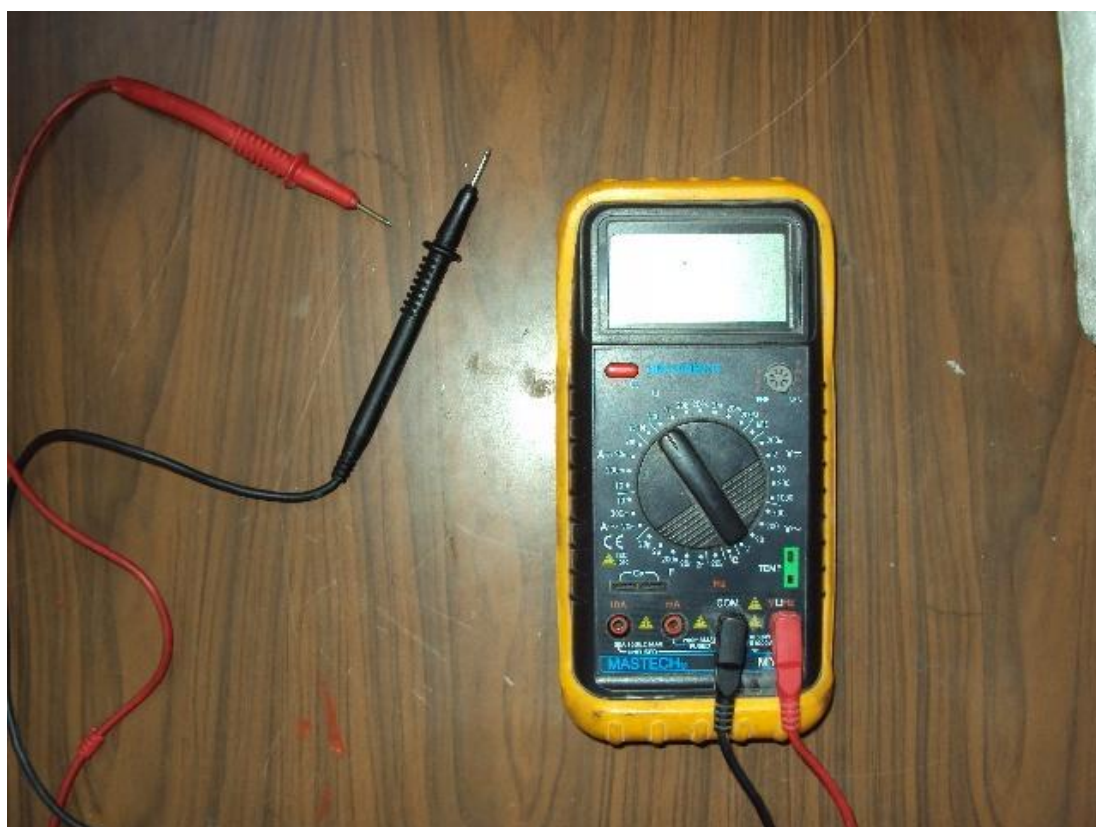


Figure 4.13. Mastech M-9502 Digital multimeter

Table 4.5. Technical specifications of the digital multimeter

DC Voltage	200m/2/20/200V±0.5%, 1000V±0.8%
AC Voltage	200mV±1.2%, 2/20/200V±0.8%, 700V±1.2%
DC Current	2m/20mA±0.8%, 200mA±1.2%, 20A±2.0%
AC Current	20mA±1.0%, 200mA±1.8%, 20A±3.0%
Resistance	200/2k/20k/200k/2MΩ±0.8%, 20MΩ±1.0%, 200MΩ±5.0%
Capacitance	2n/20n/200n/2μ/20μF±4.0%
Power Source	9 V x 1

4.1.9. Hydroxy Electronic Control Unit (HECU)

Experiments show that, when the engine speed decreases under the critical value (2500 rpm for SI and 1750 rpm for CI engines in this experiment), flow rate has to be decreased to prevent HHO occupy too much volume due to the long opening time of manifolds at low speeds. Otherwise reduction in volumetric efficiency is inevitable unless hydroxy electronic control unit (HECU) is serial connected to the HHO system (constant voltage, current and HHO flow rate), due to minimum ignition energy of HHO which is a strongly decreasing function of equivalence ratio, pre-ignition and knock occurrence is probable strongly. Also, low lean flammability limit of HHO results advantages only under dilute (lean) conditions.

Electronic control unit is designed and manufactured to decrease HHO flow rate by decreasing voltage and current. Experiments depict that voltage around 7.1 V and current around 5.4 A are suitable values for both SI and CI engines below the critical speeds. Therefore, an electronic control unit has to be designed and manufactured to set the voltage and current to desirable values when the engine speed decreases under critical readings.

The most common method of speed control is pulse width modulation (PWM). The 555 IC timer chip is widely used as both a monostable and astable multivibrator (Millman and Grabel, 1987). Pulse width modulation is the process of switching the power to a device on and off at a given frequency, with varying on and off times with aid of a Metal Oxide Semiconductor Field Effect Transistor (MOSFET). The proportion of the time period occupied by the pulse is defined as “duty cycle”. Figure 4.14 indicates the waveform of approximately 61% duty cycle

(appropriate value for HHO flow rate below critical speeds) signals which means average $V_{dc}=7.1$ V. Schematic diagram and the photograph of the HECU are shown in Figure 4.15 and Figure 4.16, respectively. The pin descriptions for the 555 Timer are illustrated in Table 4.6.

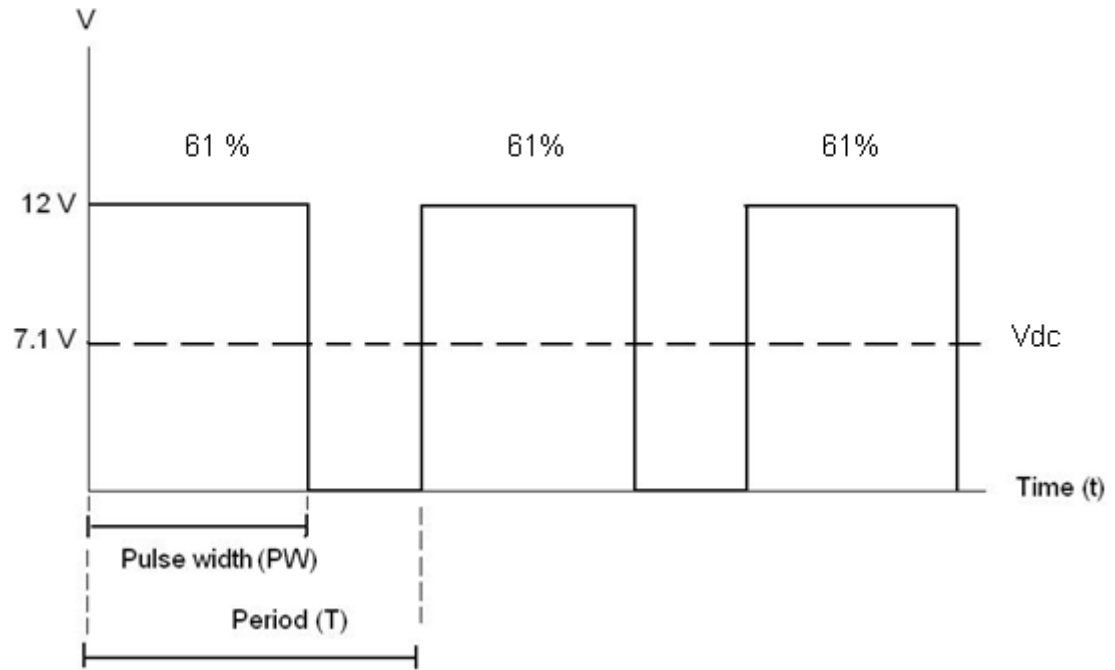


Figure 4.14. Waveform of 61% duty cycle signals

Pulse width can be found by the Equation 4.4 given below:

$$Duty\ cycle = PW \times f \times 100 \quad (4.4)$$

From the Equation 4.4, for 61% duty cycle and $\frac{1}{T} = f = 144$ Hz frequency

(set overall frequency for this circuit) $PW = 4.24$ ms.

The 555 timer in the PWM circuit is configured as an oscillator. This means that once power is applied, the 555 oscillates without any external trigger.

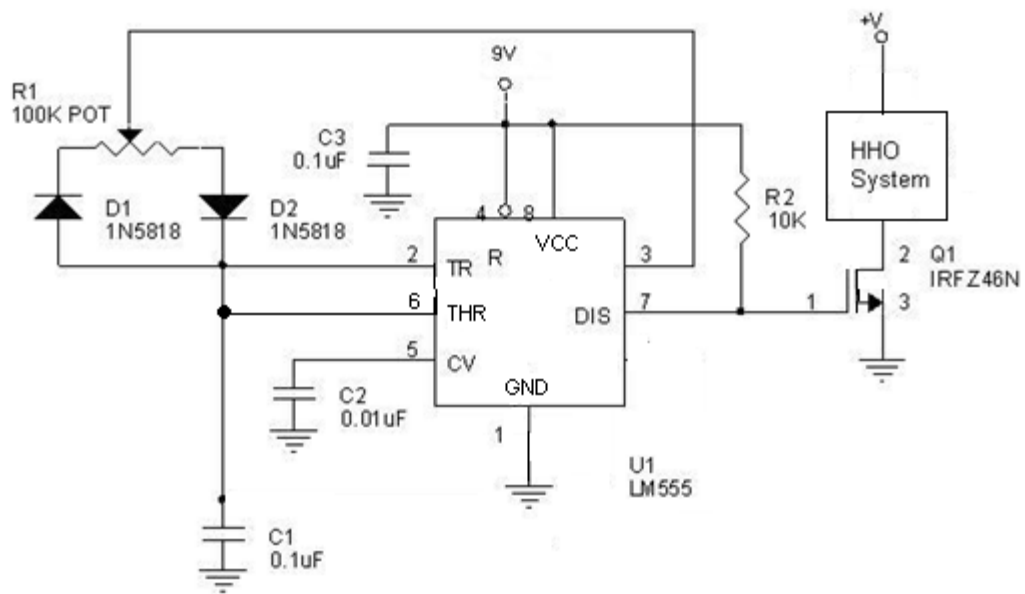


Figure 4.15. The schematic diagram for the 555 PWM circuit (DPRG, 2005)

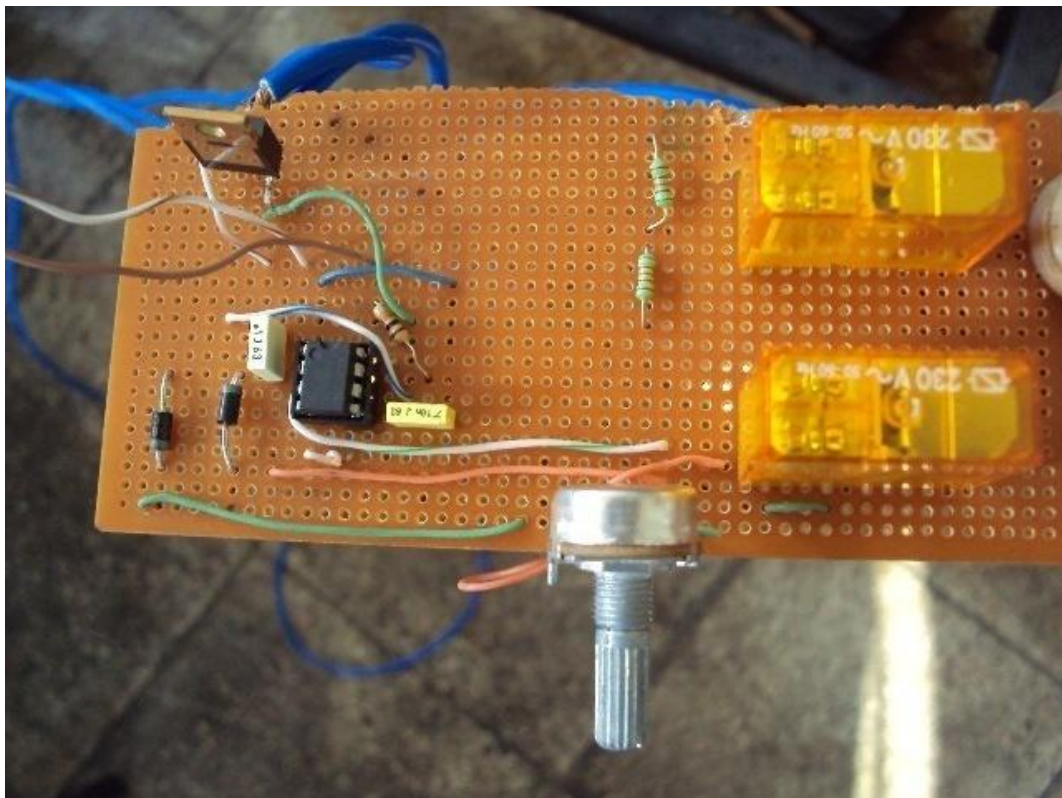


Figure 4.16. The photograph of HECU

Table 4.6. Pin descriptions for the 555 Timer

Pin	Description	Purpose
1	Ground	DC Ground
2	Trigger	The trigger pin triggers the beginning of the timing sequence. When it goes LOW (0), it causes the output pin to go HIGH (1). The trigger is activated when the voltage falls below 1/3 of +V on pin 8.
3	Output	The output pin is used to drive external circuitry. It has a "totem pole" configuration, which means that it can source or sink current. The HIGH output is usually about 1.7 volts lower than +V when sourcing current. The output pin can sink up to 200mA of current. The output pin is driven HIGH when the trigger pin is taken LOW. The output pin is driven LOW when the threshold pin is taken HIGH, or the reset pin is taken LOW.
4	Reset	The reset pin is used to drive the output LOW, regardless of the state of the circuit. When not used, the reset pin should be tied to +V.
5	Control Voltage	The control voltage pin allows the input of external voltages to affect the timing of the 555 chip. When not used, it should be bypassed to ground through an 0.01uF capacitor.
6	Threshold	The threshold pin causes the output to be driven LOW when its voltage rises above 2/3 of +V.
7	Discharge	The discharge pin shorts to ground when the output pin goes HIGH. This is normally used to discharge the timing capacitor during oscillation.
8	+V	DC Power (9V battery for this experiment)

Pin 5 is not used for an external voltage input, so it is bypassed to ground with an 0.01uF capacitor. Capacitor C1 charges through one side of R1 and discharges through the other side. The sum of the charge and discharge resistance is always the same, therefore the period of the output signal is constant. Only the duty cycle varies with R1.

The overall frequency of the PWM signal in this circuit is determined by the values of R1 and C1. In the schematic above, this has been set to 144 Hz. Considering R1 as constant, the frequency (f) can be varied with changing the capacity of C1 by the Equation 4.4 shown below:

$$f = \frac{1.44}{R1 \times C1} \quad (4.4)$$

4.2. Experimental Set-up

The technical specifications of the HHO system and general view of experimental set-up are demonstrated in Table 4.7 and Figure 4.17, respectively.

Table 4.7. Technical specifications of the HHO system

Maximum gas supply	5 l /min
Electrodes (anode-cathode)	316L stainless steel plates
Maximum electrolysis voltage and current (≥ 2500 rpm for SI and ≥ 1750 rpm for CI)	12 V – 14.1 A
Electrolysis voltage and current (≤ 2500 rpm with HECU for SI and ≤ 1750 rpm with HECU for CI)	7.1 V – 5.4 A
Electrolyte (1% by mass)	NaOH aqueous solution
Reactor container volume	8.5 l
Water level control	Float system



Figure 4.17. General view of experimental set-up

5. RESULTS AND DISCUSSION

The aim of this experimental investigation is, to make a spectacular combination of anodes and cathodes in a simply adaptable ambient within the fuel system and to obtain an enhancement in combustion and reduction in exhaust emissions with electrolysis reaction without the need for storage tanks. In this experimental study, instead of pure hydrogen addition to diesel fuel, produced hydrogen gas along with oxygen (hydroxy gas, HHO, Brown's gas) is fed to the intake manifolds of a direct injection CI and a single cylinder SI engine by a HHO system and a HECU under various loads.

In this study, HHO is produced by the electrolysis process of different electrolytes ($\text{KOH}_{(\text{aq})}$, $\text{NaOH}_{(\text{aq})}$, $\text{NaCl}_{(\text{aq})}$) with various electrode designs in a leak proof plexiglass reactor (hydrogen generator). Electrolytes are used to diminish oxygen and hydrogen bonds. HHO is used as a supplementary fuel in a four cylinder, four stroke, CI and a single cylinder two stroke SI engine without any modification and without need for storage tanks. Its effects on HC, CO emissions, engine performance characteristics and SFC are investigated.

5.1. Spark Ignition (SI) Engine

Figure 5.1, 5.2, 5.3 and 5.4 represent the variation of engine torque, SFC, HC and CO emissions with engine speed, respectively for SI test engine.

5.1.1. Engine Torque

An average of 32.4% increment in engine torque is obtained with using HHO system compared to pure gasoline operation. This means an increment of 27% in average engine power during the experiments compared to pure gasoline operation is achieved. The increase in power is due to oxygen concentration of HHO and better mixing of HHO with air and fuel that yield enhanced combustion. Also, high laminar flame velocity of HHO yields shorter combustion period that provides lower heat

losses, much closer to ideal constant-volume combustion which results increased thermal efficiency. The results show that the addition of HHO can significantly enlarge the flammable region and extend the flammability limit to lower equivalence ratios. At high speeds (≥ 2500 rpm) the weakened in-cylinder charge flow and increased residual gas fraction are formed, which block the fuel to be fast and completely burnt at low manifold absolute pressures (MAPs). Since HHO has a low ignition energy and fast flame speed, the HHO-gasoline mixture can be more easily ignited and quickly combusted than the pure gasoline. Thus, improved torques at high speeds can be obtained. Low lean-flammability limit of HHO allows stable combustion at highly dilute (lean) circumstances. It is observed that HHO cannot have a positive effect on power output at around stoichiometric (richer) conditions. Also, due to minimum ignition energy of HHO which is a strongly decreasing function of equivalence ratio, pre-ignition and knock are inevitable. Since the energy density of hydrogen on volume basis is much lower than that of gasoline, the reduced fuel energy flow rate is attained and finally results in the dropped engine torque at low speeds. The impairments of HHO at low speeds can be turned into advantages with the aid of HECU.

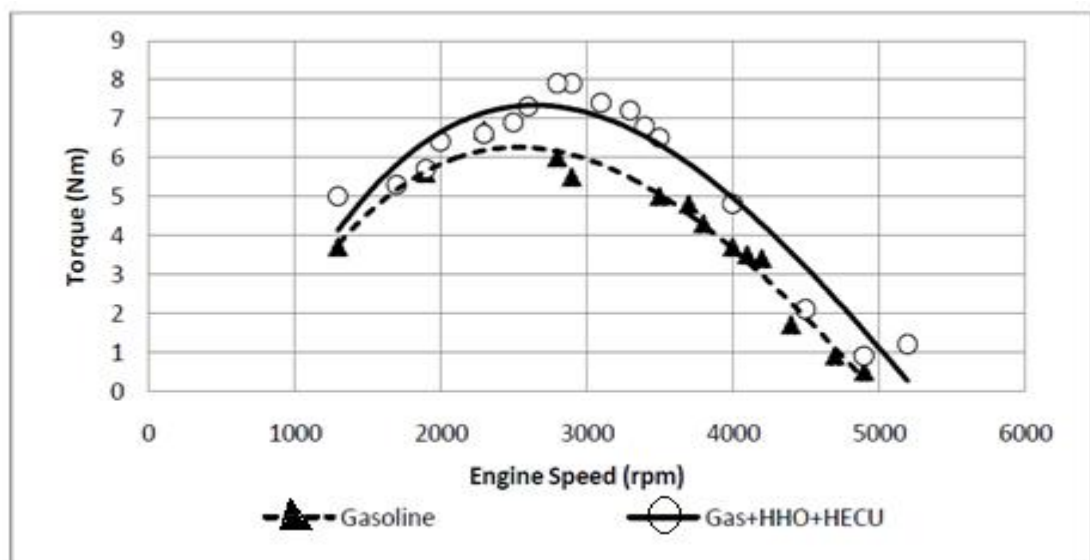


Figure 5.1. Variation of engine torque with engine speed

5.1.2. Specific Fuel Consumption (SFC)

An average gain of 16.3% is achieved in SFC by using HHO system. Brake thermal efficiency is usually used to symbolize the engine economic performance. The improvement in engine brake thermal efficiency for the HHO enriched SI engine is more evidently seen at low manifold absolute pressure (MAP) conditions. The reduction in SFC is due to uniform mixing of HHO with air (high diffusivity of HHO) as well as oxygen index of HHO gas which assists gasoline during combustion process and yields better combustion. This can be attributed to that, at high speeds, the gasoline is hard to be completely burnt at lean conditions due to the increased residual gas fraction and poor mixing. Since HHO gains a high flame speed and wide flammability, the addition of hydrogen would help the gasoline to be burned faster and more complete at high speed conditions. Also, low ignition energy of HHO-air mixture derives gasoline even to be burned safely under leaner conditions. However, at low speeds (≤ 2500 rpm), low lean flammability limit prevents HHO to have positive influence on combustion efficiency due to mixture requirement around stoichiometric conditions and high volume occupation of HHO causes reduced volumetric efficiency unless HECU is included to the system.

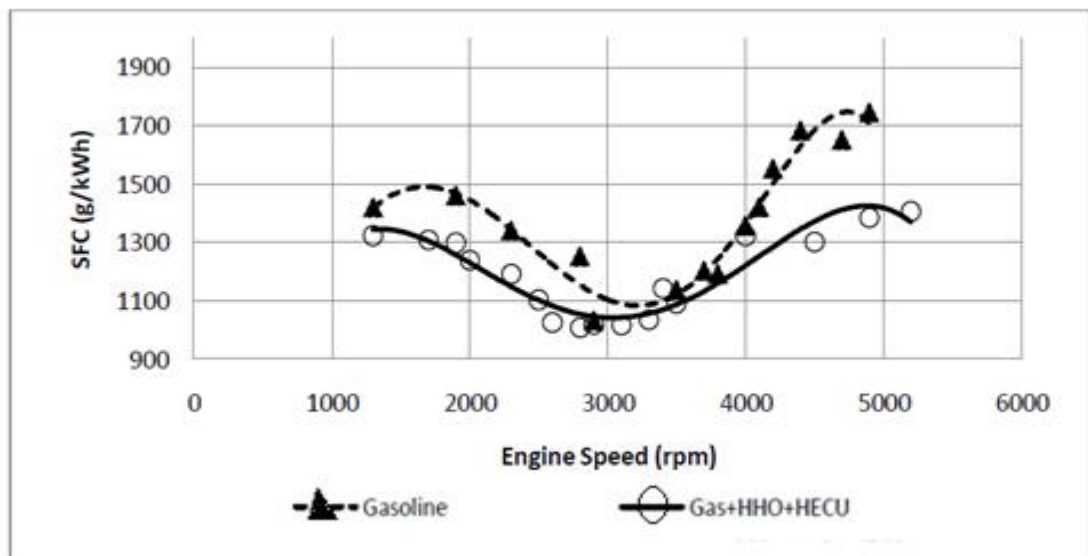


Figure 5.2. Variation of SFC with engine speed

5.1.3. Hydrocarbon (HC) Emissions

An average reduction of 6.7% is obtained in HC emissions compared to pure gasoline operation. At high speed conditions, short opening time of manifolds prevents adequate air to be taken into the cylinder and gasoline cannot be burned sufficiently. Short quenching distance and wide flammability range of hydrogen yield engine to expel less HC emissions especially under high speed conditions and low speed conditions with the aid of HECU. Besides, oxygen index of HHO yields better combustion which diminishes HC emission. At low engine speeds, due to high volume occupation of HHO, correct air cannot be taken into the cylinders which prevents gasoline to be combusted completely if HHO flow rate is not diminished at about 1.6 l/min.

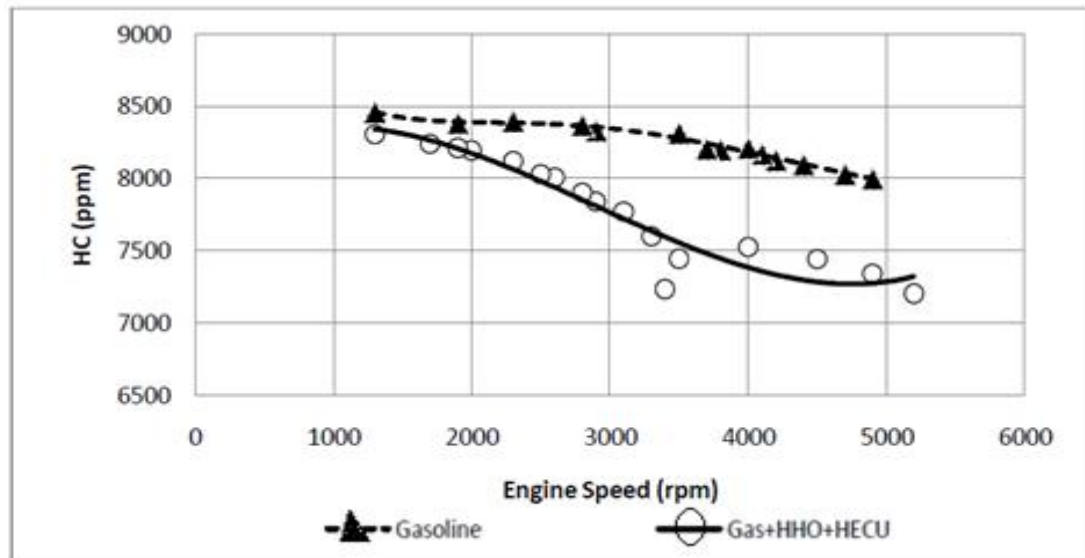


Figure 5.3. Variation of HC emissions with engine speed

5.1.4. Carbon monoxide (CO) Emissions

An average reduction of 14.4% is achieved in CO emissions compared to pure gasoline operation. Absence of carbon in HHO gas is a major reason for CO reduction. Wide flammability range and high flame speed of HHO ensure engine to

be operated at low loads. The HHO-gasoline mixture burns faster and more completely than the pure gasoline. Thus, CO emission at high speed and lean conditions is effectively reduced after hydrogen addition. Since HHO contains oxygen, higher combustion efficiency is obtained and increment for CO emission is slower unless HHO flow rate is diminished to appropriate flow rate values while approaching low speeds.

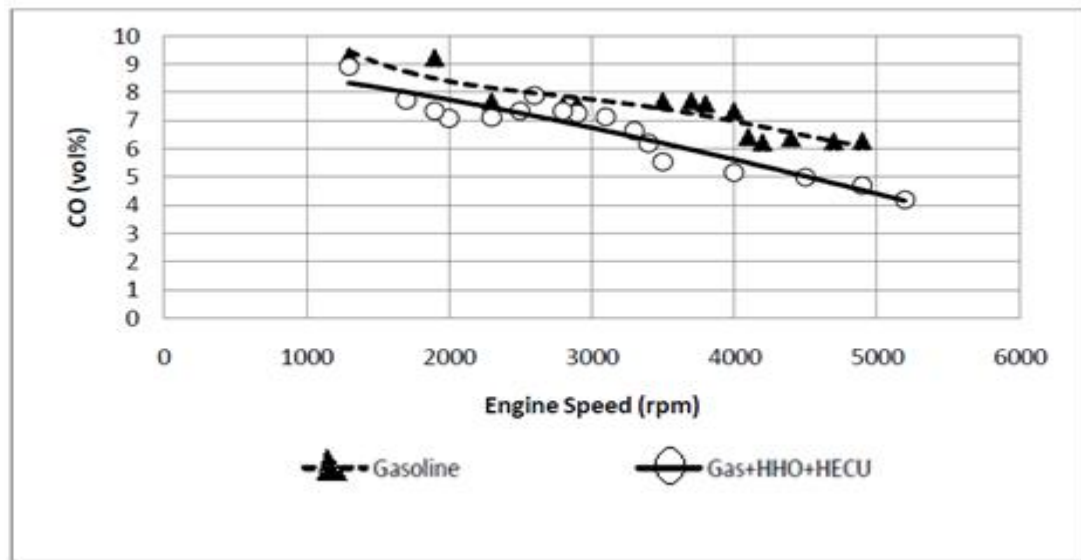


Figure 5.4. Variation of CO emissions with engine speed

5.2. Compression Ignition (CI) Engine

Figure 5.5, 5.6, 5.7 and 5.8 indicate the variation of engine torque, SFC, HC and CO emissions with engine speed, respectively for CI test engine.

5.2.1. Engine Torque

An average of 19.1% increment in engine torque is obtained with using HHO system compared to pure diesel operation. The increase in power is due to oxygen concentration of HHO and better mixing of HHO with air and fuel that yield enhanced combustion. High laminar flame velocity of HHO yields decreased ignition delay and shorter combustion period that provides lower heat losses, much closer to

ideal constant-volume combustion which results increased compression ratio and thermal efficiency. High burning velocity of HHO provides faster increment in pressure and temperature which may minimize the knocking especially at idle conditions (low or no load). Also, ignition delay period reduction yields diminished engine noise. The results show that the addition of HHO can significantly enlarge the flammable region and extend the flammability limit to lower equivalence ratios. At high speeds (≥ 1750 rpm) the weakened in-cylinder charge flow and increased residual gas fraction are formed, which block the fuel to be fast and completely burnt. Since HHO has a low ignition energy and fast flame speed, the HHO-diesel mixture can be more easily ignited and quickly combusted than the pure diesel fuel. Thus, improved torques at high speeds can be obtained. Low lean-flammability limit of HHO gas allows stable combustion at highly dilute (lean) circumstances. However, it is observed that HHO gas cannot have a positive effect on power output at around stoichiometric (richer) conditions. Since the energy density of hydrogen on volume basis is much lower than that of diesel fuel, the reduced fuel energy flow rate is attained and finally results in the dropped engine torque at low speeds. The impairments of HHO at low speeds can be turned into advantages with the aid of HECU.

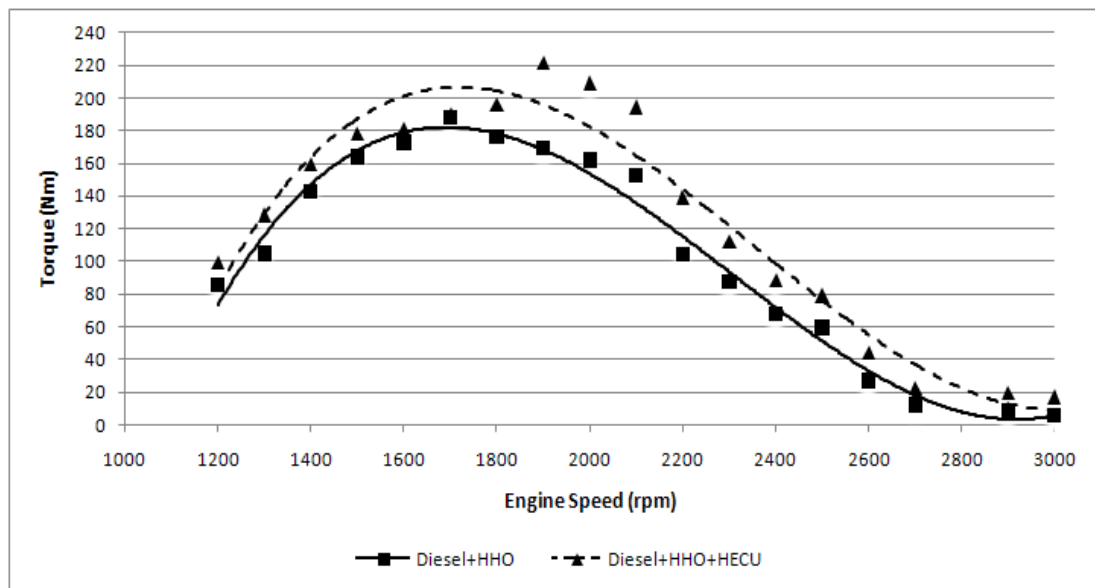


Figure 5.5. Variation of engine torque with engine speed

5.2.2. Specific Fuel Consumption

An average gain of 14% is achieved on SFC by using HHO system. Brake thermal efficiency is usually used to symbolize the engine economic performance. The improvement in engine brake thermal efficiency for the HHO enriched CI engine is more evidently seen at high speed conditions. The reduction in SFC is due to uniform mixing of HHO with air (high diffusivity of HHO) as well as oxygen index of HHO gas which assists gasoline during combustion process and yields better combustion. This can be attributed to that, at high speeds, the diesel fuel is hard to be completely burnt at lean conditions due to the increased residual gas fraction and poor mixing. Since HHO gains a high flame speed and wide flammability, the addition of hydrogen would help the fuel to be burned faster and more complete at high speed conditions. Also, low ignition energy of HHO-air mixture derives diesel fuel even to be burned safely under leaner conditions. However, at low speeds (≤ 1750 rpm), low lean flammability limit prevents HHO to have positive influence on combustion efficiency due to mixture requirement around stoichiometric conditions. Increased CR may cause pre-ignition and high volume occupation of HHO causes reduced volumetric efficiency unless HECU is included to the system.

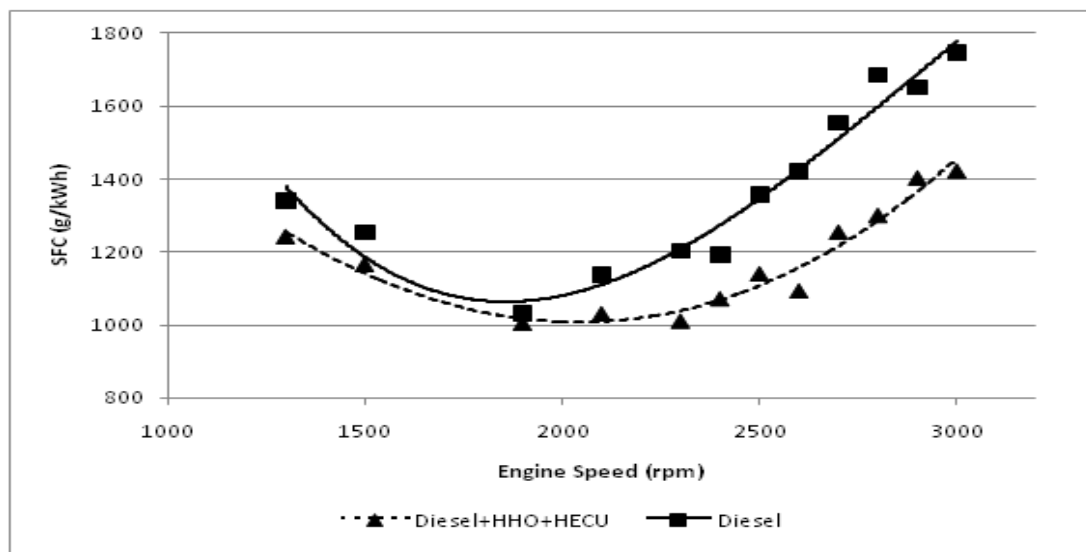


Figure 5.6. Variation of SFC with engine speed

5.2.3. Hydrocarbon Emissions

An average reduction of 5% at HC emission is achieved above the engine speed of 1750 rpm. At high speed conditions, short opening time of manifolds prevents adequate air to be taken into the cylinder and diesel fuel cannot be burned sufficiently. Short quenching distance and wide flammability range of hydrogen yield engine to expel less HC emissions especially under high speed conditions and low speed conditions with the aid of HECU. Besides, oxygen index of HHO yields better combustion which diminishes HC emission. At low engine speeds, due to high volume occupation of HHO, correct air cannot be taken into the cylinders which prevents fuel to be combusted completely. Besides, higher in-cylinder pressure, temperature and high volume occupation of HHO especially at low engine speeds may increase soot formation if HHO flow rate is not diminished at about 1.6 l/min.

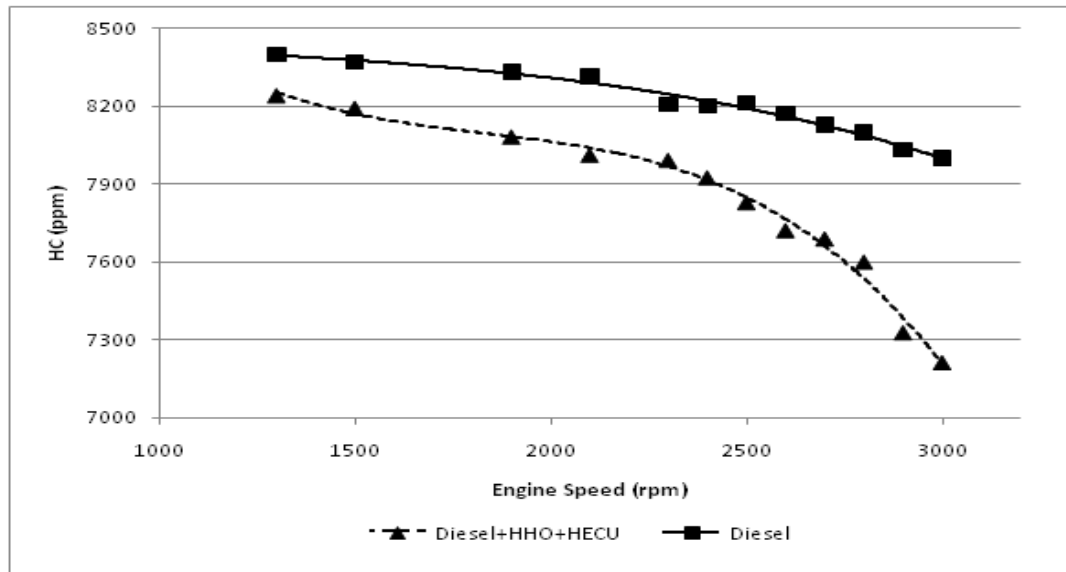


Figure 5.7. Variation of HC emissions with engine speed

5.2.4. Carbon monoxide Emissions

An average reduction of 13.5% is gained at CO emissions at mid and higher engine speeds (≥ 1750 rpm). Absence of carbon in HHO is a major reason for CO reduction. Wide flammability range and high flame speed of HHO ensure engine to be operated at low loads. The HHO-gasoline mixture burns faster and more completely than the pure diesel fuel. Thus, CO emission at high speed and lean conditions is effectively reduced after hydrogen addition. Since HHO gas contains oxygen, higher combustion efficiency is obtained and increment for CO emission is slower unless HHO flow rate is diminished to appropriate flow rate values while approaching low speeds.

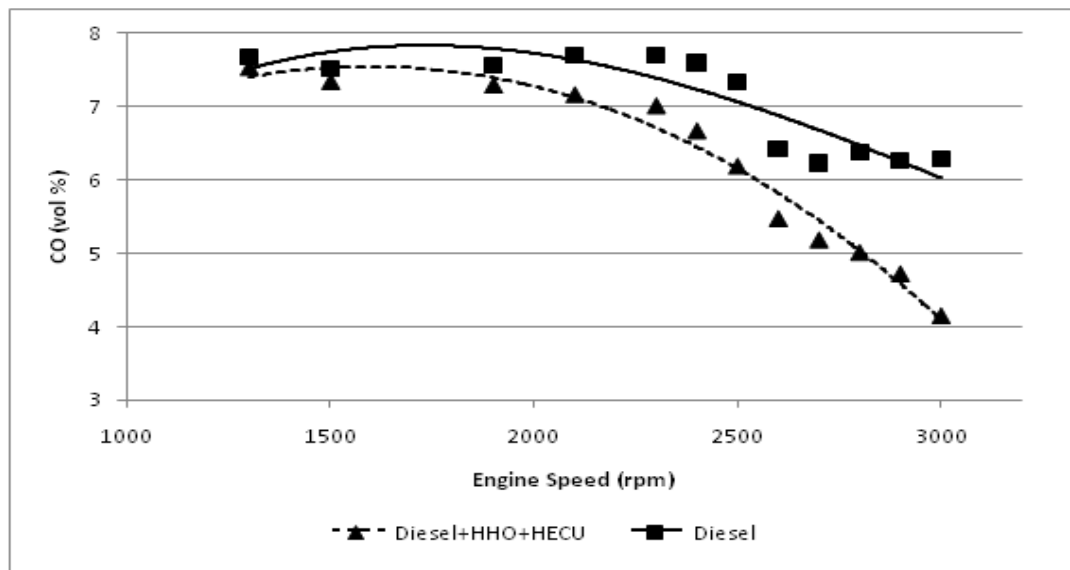


Figure 5.8. Variation of CO emissions with engine speed

5.3. Cost Analysis

The main parts of the experimental set-up and their prices are seen in Table 5.1. Density of 316L stainless steel is taken as 8027 kg/m^3 and $1 \text{ Euro} \approx 2 \text{ TL}$.

Table 5.1. Cost analysis of the main members of the system

Item	Quantity	Unit Price	Cost
Plexiglass pipe (170mm x 900mm x 1,5mm / D x L x T)	1	215 TL/m	~215 TL
316 L stainless steel electrode plates (115mm x 70mm x 2mm / D x L x T)	16	~ €10/kg = ~20 TL/kg	42 TL
316 L stainless steel pipes (22mm x 146mm x 1.5mm / D x L x T)	6	~ €10/kg = ~20 TL/kg	14 TL
316 L stainless steel pipes (28mm x 146mm x 1.5mm / D x L x T)	6	~ €10/kg = ~20 TL/kg	~18 TL
316 L stainless steel wire (2mm x 8500mm / D x L)	1	~ €10/kg = ~20 TL/kg	4 TL
Cylindrical polyamide reactor lids (170mm x 75mm / D x L)	4	300 TL/m	~100 TL
Plexiglass plates (350mm x 130mm x 20mm / L x H x T)	2	220 TL/m ³	44 TL
HECU	1	~20 TL	~20 TL
Water level system (float)	1	5 TL	5 TL
		TOTAL	462 TL

6. CONCLUSION

The results of this experimental study are summarized as follows:

1. At mid and higher engine speeds, the HHO system with diesel fuel and gasoline yields higher engine torque output compared to pure diesel and gasoline fueled engine operation unless HECU is added to the system. High burning velocity and low ignition energy of HHO-air mixture minimize the effect of the weakened in-cylinder charge flow and increased residual gas fraction which block the fuel to be fast and completely burnt at high speeds. Also, high burning velocity of HHO yields higher resistance against knocking which provides higher compression ratio (CR) and increased thermal efficiency. However, increased CR may cause pre-ignition and this can be minimized by direct HHO injection into the cylinder. At low engine speeds, low lean-flammability limits of HHO causes challenges at higher equivalence ratios. Due to the long opening time of intake manifold at low speeds, high volume occupation (reduced volumetric efficiency) of HHO becomes inevitable. Since minimum ignition energy of HHO-air mixture is a decreasing function of equivalence ratio till stoichiometric (richer) conditions, torque is reduced after HHO addition. A control unit has to be used to obtain appropriate electrolysis voltage and current (gas flow rate) to terminate the impairments of HHO gas at low speeds.
2. Uniform and improved mixing of HHO-air and oxygen content of HHO stimulate combustion which has a major effect on SFC by using an adequate capacity HHO system. Wide flammability range, high flame speed and short quenching distance of HHO yield gasoline and diesel fuel to be combusted completely under high speed conditions without HECU and low speed conditions with HECU.
3. High burning velocity, wide flammability range, oxygen content and absence of carbon make HHO gas an appropriate fuel addition to obtain adequate combustion which yield reputable reduction of HC and CO emissions when a sufficient HHO system is used at mid and higher speeds of engine without HECU and low speed conditions with HECU.
4. A control unit, which decreases electrolysis voltage and current automatically when the engine speed decreases under 1750 rpm for CI and 2500 rpm

for SI (critical speeds for this experimental study), has to be designed and manufactured to eliminate the impairments of HHO enriched diesel fuel combustion at low speeds and to provide energy economy.

5. The average power increment in test engines during experiments is bigger than the electrical power consumed and fuel economy obtained with the aid of HHO system as well. This indicates that the system is efficient.

REFERENCES

- ALIAS, A., 2005. Two Days National Seminar on Alternative Energy Resources. August, 27-28, Thane, India.
- APOSTOLESCU, N., and CHIRIAC, R., 1996. A Study of Combustion of Hydrogen-Enriched Gasoline in a Spark Ignition Engine. SAE Paper.
- BELL S. R., and GUPTA, M., 1997. Extension of The Lean Operating Limit For Natural Gas Fuelling of a Spark Ignited Engine Using Hydrogen Blending. *Combustion Science and Technology*, 123:23–48.
- BLOCK, D. L., and VEZIROGLU T. N., 1994. Hydrogen Energy Progress X. Proceedings of The Tenth World Hydrogen Energy Conference. June, 20-24, Florida, USA.
- CHIEH, C. “Bond Lengths and Energies”, University of Waterloo. <http://www.science.uwaterloo.ca/~cchieh/cact/c120/bondel.html> (2007) (Access date: 11 March 2010)
- CUNNINGHAM, J. E., and GOODENOUGH, R. Method and Apparatus for Enhancing Combustion in an Internal Combustion Engine Through Electrolysis. United States Patent, April, 21, 1992.
- DALLAS PERSONAL ROBOTICS GROUP, “Pulse Width Modulation Tutorials”, <http://www.dprg.org/tutorials/2005-11a/index.html> (2005) (Access date: 21 February 2010)
- D’ANDREA, T., HENSHAW, P. F., and TING, D. S., 2004. The Addition of Hydrogen to a Gasoline-Fuelled SI Engine. *International Journal of Hydrogen Energy*, 29:1541-52.
- DAS, L. M., 2002. Near-Term Introduction of Hydrogen Engines For Automotive and Agriculture Application. *International Journal of Hydrogen Energy*, 27:479–87.
- _____, 1991. Exhaust Emission Characterization of Hydrogen Operated Engine System: Nature of Pollutants and Their Control Techniques. *International Journal of Hydrogen Energy*, 16:765–75.

- DULGER, Z., and OZCELIK, K. R., 2000. Fuel Economy Improvement by On-Board Electrolytic Hydrogen Production, *International Journal of Hydrogen Energy*, 25:895-97.
- ERREN, R. A., CAMPBELL, W. H., 1933. Hydrogen: A Commercial Fuel for Internal Combustion Engines and Other Purposes. *J Inst Fuel*. 6:277-90.
- FANGRUI, M., and MILFORD, A., 1999. Biodiesel Production: A Review. *Bioresource Technology*, 70:1-15.
- FULTON, J., LYNCH F., and MARMORA, R., 1993. Hydrogen For Reducing Emissions From Alternative Fuel Vehicles. SAE paper, 471–9.
- HARPER, D. “Online Etymology Dictionary”, <http://www.etymonline.com/index.php?search=energy&searchmode=none> (2007) (Access date: 12 April 2010)
- HERZOG, A. V., LIPMAN T. E., and KAMMEN D. M., 2006. Renewable Energy Sources Report. Encyclopedia of Life Support Systems Publication. University of California, USA, 63 p.
- HEYWOOD, J. B., 1998. Internal Combustion Engine Fundamentals, McGraw-Hill Publication. Massachusetts Institute of Technology, Boston, USA. 15-371.
- JI, C., and WANG, S., 2009. Combustion And Emissions Performance of a Hybrid Hydrogen–Gasoline Engine at Idle and Lean Conditions. *International Journal of Hydrogen Energy*, 35(1): 346-55.
- JINGDING, L., 1998. Formation and Restraint of Toxic Emissions in Hydrogen-Gasoline Mixture Fueled Engines. *International Journal of Hydrogen Energy*, 23:971-5.
- JINGDING, L., LINSONG, G., and TIANSHEN, D., 1998. Formation and Restraint of Toxic Emissions in Hydrogen–gasoline Mixture Fueled Engines. *International Journal of Hydrogen Energy*, 23(10):971-5.
- KAHRAMAN, E., 2005. Analysis of a Hydrogen Fueled Internal Combustion Engine. The Graduate School of Engineering and Sciences of İzmir Institute of Technology. M.Sc. Thesis 71 p.

- KING, R. O., and RAND, M., 1955. The Hydrogen Engine. *Canadian Journal Technology*, 33:445–69.
- KING, R. O., and RAND, M., 1955. The Hydrogen engine. *Canada Journal of Technology*, 33:452–73.
- KOCH, H. J., 1999. *Automotive Fuels For The Future*. Third Edition. International Energy Agency Press. Denmark, 92 p.
- KONDO, T., HIRUMA, M., and FURUHAMA, S., 1996. A Study on The Mechanism of Backfire in External Mixture Formation Hydrogen Engines. *Proceedings of the 11th World Hydrogen Energy Conference*. June, 23-28, Stuttgart, Germany.
- KRAWCZYK, T., 1996. Second Edition. American Oil Chemists Society. *Biodiesel – Alternative Fuel Makes Inroads but Hurdles Remain*. California, USA,
- KUMMER, J. T., 1980. Catalysts for Automobile Emission Control. *Proceedings of Energy Combustion Science*, 6:177-199.
- LEE, J. T., KIM, Y. Y., and LEE C.W., 2001. An Investigation of a Cause of Backfire and Its Control Due to Crevice Volumes in a Hydrogen Fueled Engine. *ASME*, 123:204-10.
- MASOOD, M., ISHRAAT, M. M., and REDDY, A. S., 2007. Computational Combustion and Emission Analysis of Hydrogen–Diesel Blends with Experimental Verification. *International Journal of Hydrogen Energy*, 32:2539-47.
- MASOOD, M., ISHRAT, M. M., and REDDY, A. S., 2006. Computational Combustion and Emission Analysis of Hydrogen–Diesel Blends with Experimental Verification. *International Journal of Hydrogen Energy*, 32:2548-52.
- MCCARTHY, J. *Hydrogen*. Stanford University. <http://www-formal.stanford.edu/jmc/progress/hydrogen.html> (2008) (Access date: 10 March 2010)
- MICHROWSKI, A., 1993. Anomalous Water Explained by Brown's Gas Research. *Planetary Association for Clean Energy Newsletter*. 6 (4):10-11.

- _____, 1993. Yull Brown's Gas. Planet. Assoc. Clean Energy Newsl., 6 (4):10-11.
- MILLMAN, J., and GRABEL, A., 1987. Microelectronics. Second Edition. McGraw-Hill Inc. Publication. New York, USA. 690 p.
- MITKARI, S. G., and KARI, M. G., 2005. Solar PV Power Plant the Need of the Day. Two Days National Seminar on Alternative Energy Resources. August, 27-28, Thane, India.
- PEAVEY, M. A., 2003. Fuel from Water: Energy Independence with Hydrogen. Eleventh Edition. New York, USA. 256 p.
- PESCHKA, W., 1998. Hydrogen: The Future Cryofuel in Internal Combustion Engines. International Journal of Hydrogen Energy, 23:27-43.
- PULKRABEK, W. W., 2004. Engineering Fundamentals of The Internal Combustion Engine, Prantice Hall Press. Second Edition. New Jersey, USA. 425 p.
- RAVI, M., RAO, A. N., RAMASWAMY, M. C., and JAGADEESAN, T. R., 1992. Experimental Investigation on Dual Fuel Operation of Hydrogen in a C.I. Engine. Proceedings of the National Conference on I.C. Engines and Combustion, Indian Institute of Petroleum, September, 15–18, Bangalore, India.
- RESNICK, R., and HALLIDAY, D., 1960. Heat, a Form of Energy. Library of Congress Catalog Card Number 66-1152, Washington DC, USA.
- SARAVANAN, N., and NAGARAJAN, G., 2008. An Experimental Investigation of Hydrogen-Enriched Air Induction in a Diesel Engine System. International Journal of Hydrogen Energy, 33:1769-75.
- SARAVANAN, N., and NAGARAJAN, G., 2008. An Experimental Investigation of Hydrogen Enriched Air Induction in a Diesel Engine System. International Journal of Hydrogen Energy, 33:1771 – 80.
- SARAVANAN, N., NAGARAJAN, G., DHANASEKARAN, C., and KALAISELVAN, K. M., 2007. Experimental Investigation of Hydrogen Port Fuel Injection in DI Diesel Engine. International Journal of Hydrogen Energy, 32:4071-80.

- SIERENS, R., and ROSSEEL, E., 1998. Variable Composition Hydrogen-Natural Gas Mixtures for Increased Engine Efficiency and Decreased Emissions. ASME Spring Engine Technology Conference. April, 26-29, Fort Lauderdale, Florida, USA.
- STEBAR, R. F., and PARKS, F. B., 1974. Emission Control with Lean Operation Using Hydrogen-Supplemented Fuel. SAE Paper (Document No. 740187).
- STEBAR, R. F., and PARKS, F. B., 1974. Emission Control with Lean Operation Using Hydrogen-Supplemented Fuel. SAE Paper (Document No. 740192).
- TEKIN, M., and ÇAVUŞOĞLU, Y., 1997. Pollutant Formation and Its Control in Internal Combustion Engines. 1st Automotive Technology Congress with International Participation, May, 26-30, Adana, Turkey.
- VADER, N. V., and JOSHI, V. A., 2005. Wind Power Generation Technology. Two Days National Seminar on Alternative Energy Resources. August, 27-28, Thane, India.
- VADER, N. V., and JOSHI, V. A., 2005. Wind Power Generation Technology. Two Days National Seminar on Alternative Energy Resources. August, 27-28, Thane, India
- VARDE, K. S., 1981. Combustion Characteristics of Small Spark Ignition Engines Using Hydrogen Supplemented Fuel Mixtures. SAE Paper.
- VENKATARAMAN, B., and ELANGO, D., 1998. Renewable Energy Sources. Hindustan College of Engineering, Padur, India, 235 p.
- WHITE, C. M., STEEPER, R. R., and LUTZ, A. E., 2006. The Hydrogen-Fueled Internal Combustion Engine: A Technical Review. International Journal of Hydrogen Energy, 31:1292-1305.
- WONG, J. K. S., 1990. Ignition of Hydrogen in a Direct Injection Diesel Engine Modified to Operate as a Low Heat Rejection Energy. International Journal of Hydrogen Energy, 15:507.
- ZWEIG, R. M., 1992. Proceedings of the Ninth World Hydrogen Energy Conference. October, 8-12, Paris, France.

BIOGRAPHY

He was born in Mersin, in 02/06/1984. He has completed his elementary education at İsmet İnönü Primary Education School. He went to high school at Gündoğdu College and graduated from there with the 1st degree in 1999. Then he deserved to educate in Seyhan Rotary Anatolian High School. He has completed his B.Sc. degree in the Mechanical Engineering Department of Istanbul University with the 2nd highest degree in 2007. He worked for İlke Construction Supervision Limited Company as a project control engineer for 1.5 years. He has been working as Research Assistant in the Mechanical Engineering Department of Çukurova University since October, 2009.

Internal combustion engines and renewable energy are the major research interests of him.