# INVESTIGATION OF ETHANOL DISI ENGINE PERFORMANCE AND EMISSIONS



ATHIWAT BUTMARASRI



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| **Thesis** | Investigation of Ethanol DISI Engine Performance and Emissions |
| **Student** | Mr. Athiwat Butmarasri |
| **Student ID** | 54600908 |
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| **Thesis Advisor** | Asst.Prof.Dr. Chinda Charoenphonphanich<br>Dr. Nuwong Chollacoop<br>Prof.Dr. Hidenori Kosaka |

# Abstract

The reduction of particle emissions is nowadays an important issue with respect to emissions from gasoline powered motor vehicle. Using new development technologies and renewable oxygenated fuels are considered the most suitable solution for sustainable future. This research focused on the comparison of particle emissions and performance from gasoline and ethanol DISI (Direct Injection Spark Ignition) engine. Amount of particle emissions would be reduced by using ethanol DISI engine. In addition, physical structure of particle emissions emitted from gasoline and diesel diffusion flames were investigated by using a Scanning Electron Microscopy (SEM) and a Transmission Electron Microscopy (TEM). The DISI engine was tested on engine dynamometer with different loads and injection behaviors, with homogenous and stratified charge. Then, the particle emissions were sampled by smoke meter in order to investigate physical structure and measure the amount of particle emissions. The results in this part showed that particle emissions emitted from ethanol are lower than gasoline. The average primary size of gasoline and diesel fuels particle emissions are approximately 25-60 nm and 50-60 nm, respectively. The accumulated sizes of gasoline and diesel fuels particle emissions are approximately 100-360 nm and 100-500 nm, respectively. Performance of ethanol DISI engine were invested in the point of brake specific fuel consumption (bsfc), brake specific energy consumption (bsec), Maximum torque advance and lean burn limit in stratified charge mode. The results in this part showed gasoline and ethanol distillation curves showed the same trend as spray images from CVCC, that ethanol can diffuse easily and vaporization of ethanol is more complete than gasoline. Spray images also showed burn rate of ethanol is faster than gasoline these lead to the increasing of brake mean effective pressure of ethanol DISI engine compare with gasoline DISI engine.

The findings of this study can serve as guidance for the reduction particle emissions and improving performance from gasoline DISI engines by using ethanol.

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# Chapter 1
# Introduction

## 1.1 Background

Nowadays, the fuel consumption in transportation field is one of the main reasons to realize amount of emissions, in particularly, the limitations of new emission standards. In particular, the total energy consumption in the world depends on the remaining fossil fuels [1].

It is imperative, then, to find out alternative fuels instead of the using fossil fuels. In Thailand, Alternative fuel, Ethanol, has been broadly promoted by the government due to environmental friendly properties of ethanol and it can reduce dependency of imported crude oil [2]. Ethanol can be produced from many source of biomass and were the renewable energy. In addition, the raw material for produced ethanol, cassava or sugarcane was the main economic vegetation in Thailand. Ethanol is the most suitable alternative fuel for spark ignition engines because of the advantages of ethanol, e.g. better knock limit range due to higher octane number of ethanol, higher volumetric efficiency due to cooling effect of higher heat of vaporization and also reduced particle emissions due to more complete combustion from oxygen atom inside ethanol molecule [3-13].

The development of new clean spark ignition engines, such as direct injection spark ignition (DISI) engines is important because of the advantages of DISI engines, e.g. higher thermal efficiency due to direct fuel injection, higher power output than conventional homogeneous charge port injection spark ignition (PSI) engines and lower fuel consumption due to an ultra-lean combustion in stratified charge operating mode [14-16].

In particular, diffusion flame in stratified charge operation of DISI engine is similar to compression ignition (CI) engine that causes particle emissions. Particle emissions consist of a solid fraction (SOL) and a soluble organic fraction (SOF). Primary particles, composed of carbon and metallic ash, are coated with SOF and sulphate. A primary soot particle has two distinct parts: an inner core is located at the central region of the primary particle and another is outer shell. The composition of particle emissions may vary widely depending on the operating conditions and fuel composition [17-20].

The use of direct injection spark ignition engines with fuelled ethanol is to reduce regulated pollutant and particle emissions produced by internal combustion engines, as well as to reduce the greenhouse effect impact of transportation. In addition PM, NOx and HC emissions are prejudicial effects on the environment and human health.

## 1.2 Objectives

To study the use of ethanol direct injection spark ignition engines to reduce regulated pollutant emissions produced by internal combustion engines.

To investigate performance (fuel consumption, bsfc, bsec and maximum torque advance) and emissions (amount and nanostructure) of ethanol DISI engines.

## 1.3 Scope of Work

### 1.3.1 The Properties of Gasoline, Gasohol and Ethanol

The first study would be aimed to characterize fuels properties, e.g. heating value, octane number, heat of vaporization and fuel composition, to know the fuel characteristics.

### 1.3.2 The Properties of Gasoline, Gasohol and Ethanol that Effect on Particle Emissions (Soot Formation and Oxidation Process) and Gaseous Emissions

This study focused on the effects of characterized parameters of each fuels on soot formation and oxidation process and also NOx emissions by using SEM and TEM technique, smoke meter and gas analyzer. The results would be reported to figure out the pollutant occur phenomena.

### 1.3.3 Optimization of Ethanol DISI Engine Best Operating Condition

This study focused on improving performance and emissions of ethanol DISI engine by vary operating conditions (ignition timing, injection timing and duration) of the engine then the results would be analyzed in the view point of engine performance (power, torque, bsfc and bsec) and emissions (emissions concentration, particle emissions concentration)

# Chapter 2
# Research Background

## 2.1 Introduction

### 2.1.1 DISI Engine

With the increasing emphasis on achieving substantial improvements in automotive fuel economy, automotive engineers are striving to develop engines having enhanced brake-specific fuel consumption (BSFC), and which can also comply with future stringent emission requirements. The BSFC [15], and hence the fuel economy, of the compression-ignition, direct-injection (CIDI), diesel engine is superior to that of the port-fuel-injected (PFI) spark-ignition engine, mainly due to the use of a significantly higher compression ratio, coupled with unthrottled operation. The diesel engine, however, generally exhibits a higher noise level, a more limited speed range, diminished start ability, and higher particulate and NOx emissions than the spark ignition (SI) engine. Over the past two decades, attempts have been made to develop an internal combustion engine for automotive applications that combines the best features of the SI and the diesel engines. The objective has been to combine the specific power of the gasoline engine with the efficiency of the diesel engine at part load. Such an engine would exhibit a BSFC approaching that of the diesel engine, while maintaining the operating characteristics and specific power output of the SI engine.

Research has indicated that a promising candidate for achieving this goal is a direct-injection, four-stroke, spark ignition engine that does not throttle the inlet mixture to control the load [21-22]. In this engine, a fuel spray plume is injected directly into the cylinder, generating a fuel–air mixture with an ignitable composition at the spark gap at the time of ignition. This class of engine is designated as a direct-injection, stratified-charge (DISC) engine. This engine type generally exhibits an improved tolerance for fuels of lower octane number and drivability index, and a significant segment of the early work on prototype DISC engines focused on the inherent multi-fuel capability. In a manner similar to that of the diesel, the power output of this engine is controlled by varying the amount of fuel that is injected into the cylinder. The induction air is not significantly throttled, thus minimizing the negative work of the pumping loop of the cycle. By using a spark plug to ignite the fuel as it mixes with air, the engine is provided with direct ignition, thus avoiding many of the requirements of auto ignition quality that are inherent in fuels for the diesel engine. Furthermore, by means of the relative alignment of the spark plug and the fuel injector, overall ultra-lean-operation may be achieved, thus yielding an enhanced BSFC.

### 2.1.1.1 Key Potential Benefits: GDI Engine Versus PFI Engine

The major difference between the PFI engine and the GDI engine is in the mixture preparation strategies [16]. In the PFI engine, fuel is injected into the intake port of each cylinder, and there is an associated time lag between the injection event and the induction of the fuel and air into the cylinder. The vast majority of current automotive PFI engines utilize timed fuel injection onto the back of the intake valve when the intake valve is closed. During cranking and cold starting, a transient film, or puddle, of liquid fuel forms in the intake valve area of the port. This causes a fuel delivery delay and an associated inherent metering error due to partial vaporization, making it is necessary to supply amounts of fuel that significantly exceed that required for the ideal stoichiometric ratio. This puddling and time lag may cause the engine to either misfire or experience a partial burn on the first 4–10 cycles, with an associated significant increase in the UBHC emissions. Alternatively, injecting fuel directly into the engine cylinder totally avoids the problems associated with fuel wall wetting in the port, while providing enhanced control of the metered fuel for each combustion event, as well as a reduction in the fuel transport time. The actual mass of fuel entering the cylinder on a given cycle can thus be more accurately controlled with direct injection than with PFI. The GDI engine offers the potential for leaner combustion, less cylinder-to-cylinder variation in the air–fuel ratio and lower operating BSFC values. The UBHC emissions during a cold start are also potentially lower with direct injection, and the engine transient response can be enhanced. As a result of the higher operating fuel pressure of the GDI system, the fuel entering the cylinder is much better atomized than that of the PFI system, particularly under cold operating conditions, thus yielding much higher rates of fuel vaporization. The mean drop size is typically 16 microns SMD as compared to 120 microns SMD with the PFI system. It is important to note, however, that injection of fuel directly into the cylinder is not a guarantee that fuel film problems are not present. The wetting of piston crowns or other combustion chamber surfaces, whether intentional or unintentional, does introduce the important variable of transient wall film formation and evaporation.

The GDI concept does indeed offer many opportunities for circumventing the basic limitations of the PFI engine, particularly those associated with port wall wetting. The fuel film in the intake port of a PFI engine acts as an integrating capacitor, and the engine actually operates on fuel inaccurately metered from the pool in the film, not from the current fuel being accurately metered by the injector. During a cold start, the fuel from more than 10 cycles must be injected to achieve a steady, oscillatory film of liquid fuel in the intake port. This means that the cold PFI engine does not fire or start on the first few cycles, even though fuel is being repetitively injected into the film pool. Control algorithms must be used to provide significant over-fueling if acceptable PFI start times are to be achieved, even though the catalyst temperature is below the light-off threshold at this condition and UBHC emissions will be increased. Thus it is not unusual for PFI systems to generate 90% of the total UBHC emissions in the US FTP emission test within the first 90 s.

The direct injection of gasoline into the cylinder of a four stroke, gasoline, spark-ignition engine eliminates the integrating fuel film in the intake port. It is well established that the direct injection of gasoline with little or no cold enrichment can provide starts on the second cranking cycle, and can exhibit significant reductions in UBHC spikes during load transients. An excellent example of the comparison of the fuel quantity required to start GDI and PFI engines. It is quite evident that the GDI engine requires much less fuel to start the engine, and that this difference in the minimum fuel requirement becomes larger as the ambient temperature decreases.

Another limitation of the PFI engine is the requirement of throttling for basic load control. Even though throttling is a well-established and reliable mechanism of load control in the PFI engine, the thermodynamic loss associated with throttling is substantial. Any system that utilizes throttling to adjust load levels will experience the thermodynamic loss that is associated with this pumping loop, and will exhibit thermal efficiency degradation at low levels of engine load.

Current advanced PFI engines still utilize, and will continue to require, throttling for basic load control. They also have, and will continue to have, an operating film of liquid fuel in the intake port. These two basic PFI operating requirements represent major impediments to achieving significant breakthroughs in PFI fuel economy or emissions. Continuous incremental improvements in the older PFI technology will be made, but it is unlikely that the long-range fuel economy and emission objectives can be simultaneously achieved. The GDI engine, in theory, has neither of these two significant limitations, nor the performance boundaries that are associated with them. The theoretical advantages of the GDI engine over the contemporary PFI engine are summarized as follows, along with the enabling mechanism:

Improved fuel economy (up to 25% potential improvement, depending on test cycle resulting from:

- less pumping loss (unthrottled, stratified mode);
- less heat losses (unthrottled, stratified mode);
- higher compression ratio (charge cooling with injection during induction)
- lower octane requirement (charge cooling with injection during induction)
- increased volumetric efficiency (charge cooling with injection during induction)
- fuel cutoff during vehicle deceleration (no manifold film).

Improved transient response.

- less acceleration-enrichment required (no manifold film).

More precise air–fuel ratio control.

- more rapid starting;
- less cold-start over-fueling required.

Extended EGR tolerance limit (to minimize the use of throttling).

Selective emissions advantages.

- reduced cold-start UBHC emissions;
- reduced $CO_2$ emissions.

#### 2.1.1.2 Theory of Operation

There are two main variants of DISI technology currently of interest – the ''homogeneous'' DISI operated with stoichiometric mixtures, and the mixed mode DISI, where the mixture is stratified with an overall lean equivalence ratio composition during low load and speed operation, while at the higher load and speed the engine is operated in the ''homogeneous'' mode. The mixed mode DISI, has the highest fuel-economy (FE) benefit potential, and therefore we limit our discussion to the mixed mode type of DISI engines. With lean combustion, pumping losses and heat transfer losses are reduced and the cycle efficiency is improved because of the improved thermodynamic conditions. Moreover, with DISI engines, the engine compression ratio could be raised by a couple of ratios, because of the extra charge-cooling effect offered by directly injecting the fuel into the cylinder. This increase in compression ratio obviously will raise the cycle efficiency further [23].

An example of the operational modes of a DISI engine is shown in Fig.2.1. As mentioned previously, the mixed-mode DISI engine runs with a stratified charge and with lean mixtures during low load/speed operation, and runs with a ''homogeneous'' charge at higher load/speed points. The homogeneous-charge operation could be split into medium load regions where the charge is overall lean, or highly diluted stoichiometric, and higher load regions where the charge is stoichiometric or even rich. In some cases the medium load region is eliminated, and the engine is run either in the lean stratified mode, or in the stoichiometric/rich homogeneous mode during the whole load/speed range. With the exception at the highest loads, exhaust gas recirculation (EGR) is used extensively to control NOx emissions. The level of excess-air and EGR dilution used is determined primarily by the need to maximize fuel economy, while maintaining emissions and combustion stability at acceptable levels.

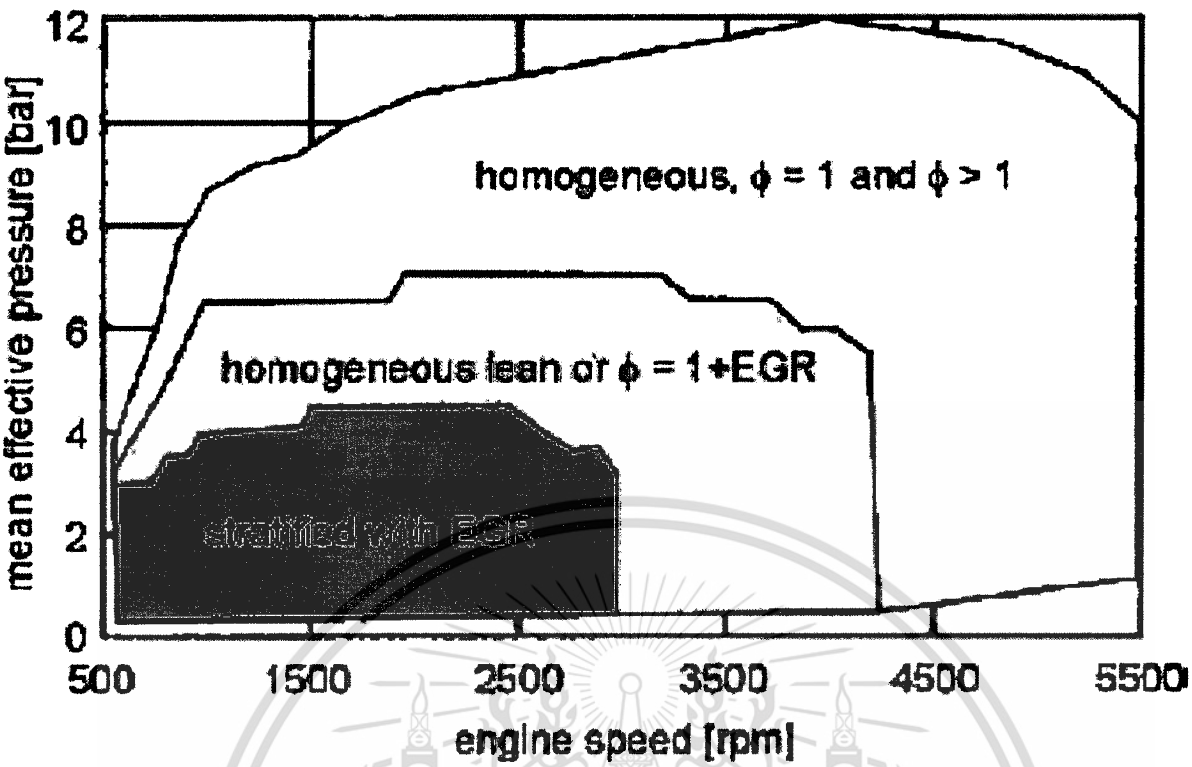


**Fig. 2.1** Operation mode of a DISI engine

The combustion system of the direct-injection spark ignition engines may be classified into three broad categories, shown in Fig. 2.2, according to the mechanism which dominates the mixture formation: air-guided, wall-guided and spray-guided. In the air-guided and wall-guided combustion systems the injector is placed a long distance from the spark plug, and the fuel spray is directed towards the spark plug by a well-defined, in-cylinder air motion or by the interaction of the spray with the piston combustion cavity. In the spray-guided combustion system, the close arrangement of the injector and the spark plug provides a strong coupling between fuel preparation and ignition.

The development of the first generation of DISI engines for production was mainly focus on the wall-guided combustion system, however, most likely, the second generation of DISI may employ a spray-guided system because of potentially additional FE improvements, wider stratified combustion operation, and improved engine-out emissions.

For the wall/charge-motion guided SIDI engines the main contributors to their fuel-economy advantage over port-fuel injection (PFI) engines are: lower pumping losses, favorable mixture properties due to lean/dilute operation, lower heat losses due to the lower charge temperatures, and higher compression ratio enabled by the charge cooling effects of direct injection. The fuel-economy advantage of the SIDI engine would have been even greater but for the lower combustion efficiency, combustion-phasing losses, and the higher friction losses. In the case of the spray-guided SIDI engines, both combustion-efficiency losses and combustion phasing losses are lower, with the result of a significant improvement of the fuel economy of the spray-guided system over that of the wall/charge guided system.

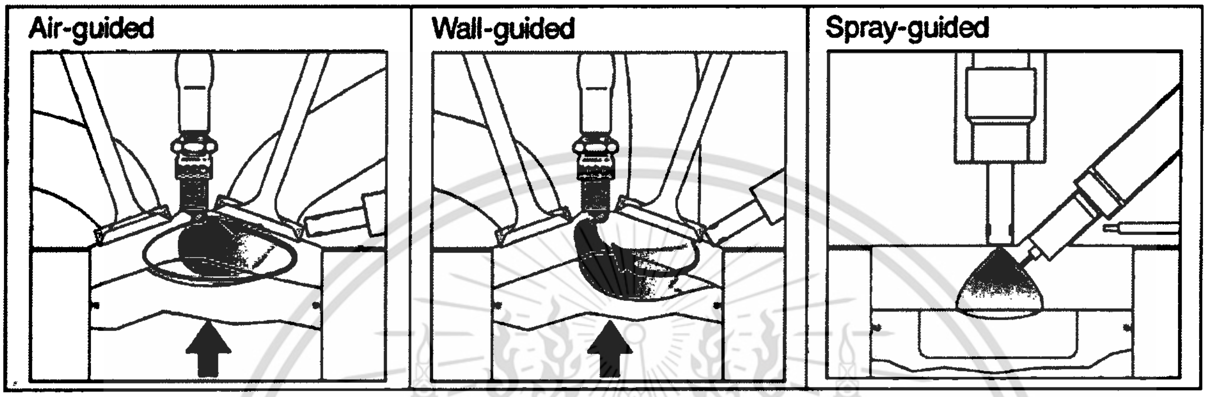


**Fig. 2.2** Classifications of DISI combustion systems

The sub-optimal phasing of stratified DISI combustion, observed by many investigators, is illustrated very clearly, which shows comparisons of heat release histories of various stratified and homogeneous combustion systems. In comparison to the homogeneous SI combustion (optimum phasing), combustion for both, stratified-combustion systems, is advanced, which will result in an increase in negative work, and consequently a reduction in the efficiency of the engine. Fig. 2.2 also shows the phasing advantage of the spray-guided system relative to the wall-guided system, discussed earlier.

## 2.3 DISI Emissions

The emission levels of a spark-ignition engine are particularly sensitive to air–fuel ratio [24]. This can be seen in Fig. 2.3, taken from Heywood (1988) [15], which shows schematically the level of emissions from a spark-ignition or Otto cycle engine as a function of relative air–fuel ratio. At rich air–fuel ratios, with $\phi$ greater than 1.0, unburned HC levels are high since there is not enough air to completely burn all the fuel. Similarly, CO levels are high, because there is not enough oxygen present to oxidize the CO to $CO_2$.

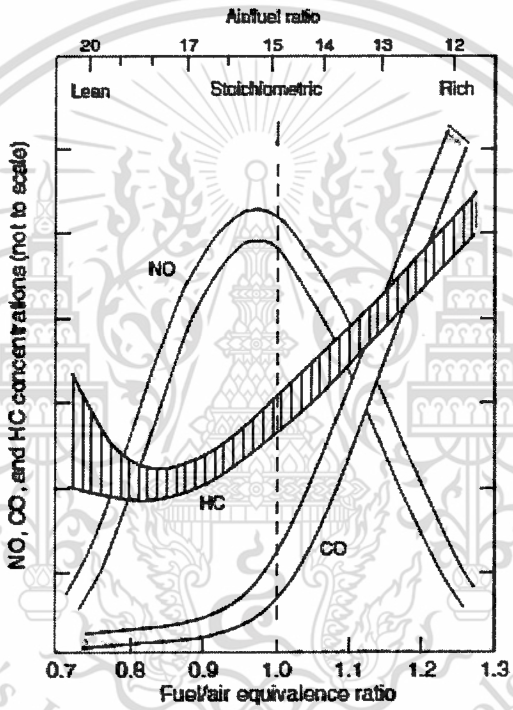


**Fig. 2.3** Emissions of SI engine as a function of fuel/air equivalence ratio

Fig. 2.3 Emissions as a function of fuel–air equivalence ratio $\phi$. For lean mixtures, with $\phi$ less than 1.0, there is always excess air available, so that CO almost completely disappears, while HC emissions reach a minimum near $\phi$=0.9. For $\phi$ less than about 0.9, some increased misfiring occurs because of proximity to the lean misfire limit, and HC emissions begin to rise again. The main factor in production of NO is combustion temperature: the higher the temperature, the greater the tendency to oxidize nitrogen compounds into NO. Since the combustion temperature is at a maximum near stoichiometric conditions where $\phi$=1.0, and falls off for both rich and lean mixtures, the NO curve takes the bell shape shown in Fig. 2.3.

CO emission is very low in GDI engine. CO varies depending on air /fuel ratio. CO is high at rich mixtures. Since GDI engines operate with lean mixture at part loads and stoichiometric mixture at full load, CO is not a problem for these engines. In GDI engine, due to the wetting of the piston and the cylinder walls with liquid fuel, HC emission can increase. Hydrocarbon (HC) emissions are a function of engine temperature and, therefore it can rise during cold start. The cold starts characteristics vary depending on the fuel distribution characteristics, the in-cylinder air motion, fuel vaporization, and fuel-air mixing.

During cold-start of a GDI engine, homogeneous operation can be employed due to a higher exhaust gas temperature resulting in a shorter time for catalyst light-off, and lower engine out HC emissions. Gasoline engines do not emit soot emission normally. Soot emission can occur at very rich mixtures. However, the GDI engines emit soot at stratified-charge operation, as in–cylinder can be areas with very rich mixtures. In addition, in GDI engine, if mixture formation do not realize at full loads due to rich mixture, the soot emission can increase. NOx emission is maximum at high cylinder temperatures and at $\lambda = 1.1$. As torque output rises, temperatures rise and, in turn, the engine-out NOx emissions display an increase. NOx emissions increase especially at full load.

## 2.4 Particle Emissions

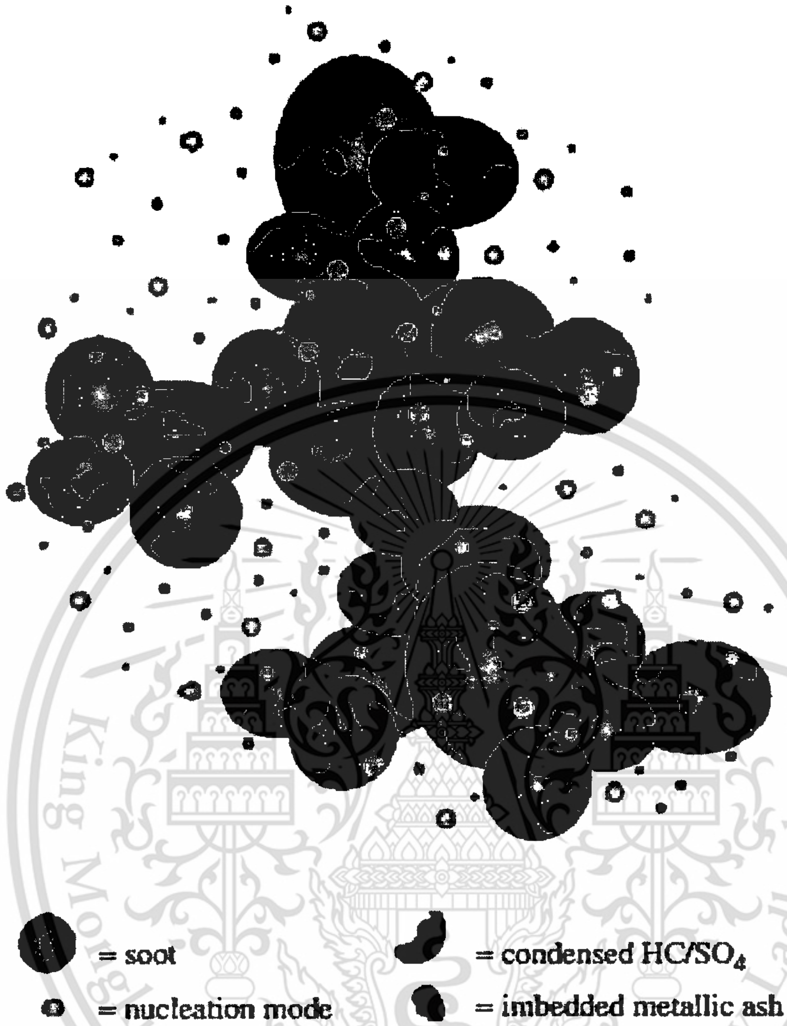


**Fig. 2.4** Conception of particle emissions [

$$CxHy + zO = aCO2 + bH2O + cCO2 + dC(s) + mHC(l) + nNOx$$

Particulate Matter (PM) is one of the six criteria pollutants, and the most important in terms of adverse effects on human health [26-28]. The marked increase in mortality that occurred during air pollution episodes in the small town of Donora, Pennsylvania, in 1948 (20 deaths) and the London fog of 1952 (4000 deaths), provides evidence of the impact that this pollutant has on human health. Since this date, many epidemiological studies of PM health effects have been completed. These studies showed that particulate matter, especially particles that are smaller than 10 microns ($PM_{10}$), are likely to cause adverse health effects including increasing morbidity and mortality in susceptible individuals. These results influenced governments to establish strategies to control air pollution. Air quality guidelines and standards were developed in an attempt to reduce adverse impacts on human health and the environment. The US National Ambient Air Quality Standards (NAAQS) established in 1971 included the first standard for Total Suspended Particulate Matter (TSP). In 1987, the TSP standard was replaced with $PM_{10}$ standard (150 mg/m$^3$ 24-h average and 50 mg/m$^3$ averaged annually), and in 1997, a standard for particles less than 2.5 micrometers in diameter ($PM_{2.5}$) was established (65 mg/m$^3$ 24-h and 16 mg/m$^3$ annual). Despite the significant improvements made in air quality over the last three decades, PM continues to exert a public health impact. Particulate matter (PM) is the term used for a mixture of solid particles and liquid droplets suspended in the air. These particles originate from a variety of sources, such as power plants, industrial processes, and diesel trucks, and they are formed in the atmosphere by transformation of gaseous emissions. Their chemical and physical compositions depending on location, time of year, and weather. Particulate matter is composed of both coarse and fine particles.

Coarse particles ($PM_{10}$) have an aerodynamic diameter between 2.5μ m and 10m m. They are formed by mechanical disruption (e.g. crushing, grinding, abrasion of surfaces); evaporation of sprays, and suspension of dust. $PM_{10}$ is composed of aluminosilicate and other oxides of crustal elements, and major sources including fugitive dust from roads, industry, agriculture, construction and demolition, and fly ash from fossil fuel combustion. The lifetime of $PM_{10}$ is from minutes to hours, and its travel distance varies from < 1 km to 10 km.

Fine particles have an aerodynamic diameter less than 2.5μ m ($PM_{2.5}$). They differ from $PM_{10}$ in origin and chemistry. These particles are formed from gas and condensation of high-temperature vapors during combustion, and they are composed of various combinations of sulfate compounds, nitrate compounds, carbon compounds, ammonium, hydrogen ion, organic compounds, metals (Pb, Cd, V, Ni, Cu, Zn, Mn and Fe), and particle bound water. The major sources of $PM_{2.5}$ are fossil fuel combustion, vegetation burning, and the smelting and processing of metals. Their lifetime is from days to weeks and travel distance ranges from 100s to >1000s km. In addition, fine particles are associated with decreased visibility (haze) impairment in many cities of the U.S

## 2.5 Alternative Fuel : Ethanol

The chemical formula for ethanol is $C_2H_5OH$, sometimes written EtOH or $C_2H_6O$. It is also known under the names ethyl alcohol or hydroxyethane and is the type of alcohol found in alcoholic beverages. Ethanol is a rather simple organic molecule consisting of a group of carbon and hydrogen atoms, with a hydroxyl group (an oxygen and a hydrogen atom) attached. Compared to most gasoline components, the ethanol molecule is small and light, having a molecular weight of just 46 g/mol.

Ethanol is somewhat special in its electrochemistry, the molecule being polar at one end and nonpolar at the other. The polarity of a molecule refers to the distribution of electric load in the molecule and is a significant factor in the physical and chemical behavior of substances. The presence of a hydroxyl group in the ethanol molecule allows it to participate in hydrogen bonding with other ethanol molecules or other polar substances. The bond is relatively weak but strong enough to make ethanol more viscous and less volatile than other similar but less polar substances. The fact that the ethanol molecule has both a polar and a nonpolar end makes ethanol soluble in both polar and nonpolar substances. The polar end makes ethanol miscible with water (and other polar substances), and the nonpolar end makes it miscible with many nonpolar organic substances, such as gasoline and, to a lesser extent, diesel fuel.

The hydrogen bonding in ethanol also causes the substance to have a rather low volatility for a molecule of such relatively small molecular weight. Under atmospheric conditions ethanol is a liquid, although it will gradually evaporate if exposed to the atmosphere. It is colorless, has a distinct taste and smell, and is categorized as a mildly toxic substance.

### 2.5.1 Use of Ethanol as Fuel

As a motor fuel ethanol is found in various forms around the world [29], in blends together with gasoline and diesel containing different amounts of water. Fuel producers design fuel blend specifications to suit local legislation, vehicles, weather, consumer habits, and other conditions of the market in which they operate.

Somewhat more than half of the fuel ethanol used worldwide is used as an additive to gasoline, meaning that ethanol constitutes 5–10 percent of the overall fuel mass in the blend.

There are two major reasons for using ethanol as an additive to gasoline, apart from any reduction in $CO_2$ emissions. First, adding ethanol to gasoline raises the octane number of the fuel blend, thus guarding against engine knock (premature ignition), which can damage the engine. Ethanol is thus able to replace more costly octane-boosting components such as alkylate. Second, because ethanol contains oxygen, ethanol containing gasoline burns more cleanly and reduces the amount of harmful emissions of carbon monoxide (CO), particulates and unburned gasoline components (see section on Emissions). Other oxygen-containing compounds can be added with the same effect.

The ethanol used as an additive is normally anhydrous, in order to prevent phase separation (de-mixing) of the water and gasoline in the blend (see section on Water and Blending Issues) Two other major types of ethanol blends, which are widely used in Brazil, are gasohol, containing roughly 20 percent anhydrous ethanol in gasoline, and E100, hydrous ethanol without gasoline and with a water content of roughly 7 percent by volume. E100 has the advantage of a lower cost of production energy and consequently monetary cost compared to that for anhydrous ethanol, whereas gasohol has a better cold starting capability and a much higher energy content per liter. Additionally, a new type of ethanol blend has recently become more widespread, E85, containing between 71 and 85 percent anhydrous ethanol, with gasoline constituting the rest of the blend. This is primarily used in flex fuel vehicles (FFVs) in the United States and Sweden. At the low temperatures experienced in these countries, the ethanol used in blends with gasoline is required to be almost anhydrous in order to avoid phase separation.

### 2.5.2 Application in SI Engines

This section describes the possibilities and difficulties associated with the application of ethanol in gasoline passenger cars. Some important fuel properties, as well as them compatibility and potential of ethanol fuels in spark-ignited (SI) engines, are discussed.

#### 2.5.2.1 Energy Density

One way of reducing $CO_2$ emissions from a vehicle is to make it more fuel efficient, that is, make it use less energy. Ethanol has a significantly lower energy density (Joule per liter), about two-thirds of that of gasoline, so about 50 percent more fuel (by volume) is needed per kilometer, if a given engine is equally efficient on either fuel. If an engine is equipped to utilize ethanol properly, ethanol usage can increase the energy efficiency of the engine and thereby of-set the otherwise higher fuel consumption. The increased efficiency would lead to lower $CO_2$ emissions, even though more fuel is used. The lower energy density of ethanol in many cases necessitates a higher fuel tank capacity and fuel flow rate, if vehicle range and performance are to be maintained. The degree to which these measures are needed depend on the percentage of ethanol in the fuel. For low ethanol blends (E5 and E10), which are used in unmodified cars, a slight decrease in performance is not unusual. Vehicles that can run on E85, or blends of E85 and regular gasoline (or low-ethanol blends) typically do not experience a performance decrease, since the vehicle is prepared for the properties of ethanol. Fig. 2.5 shows the energy content of typical ethanol fuels currently on the market.

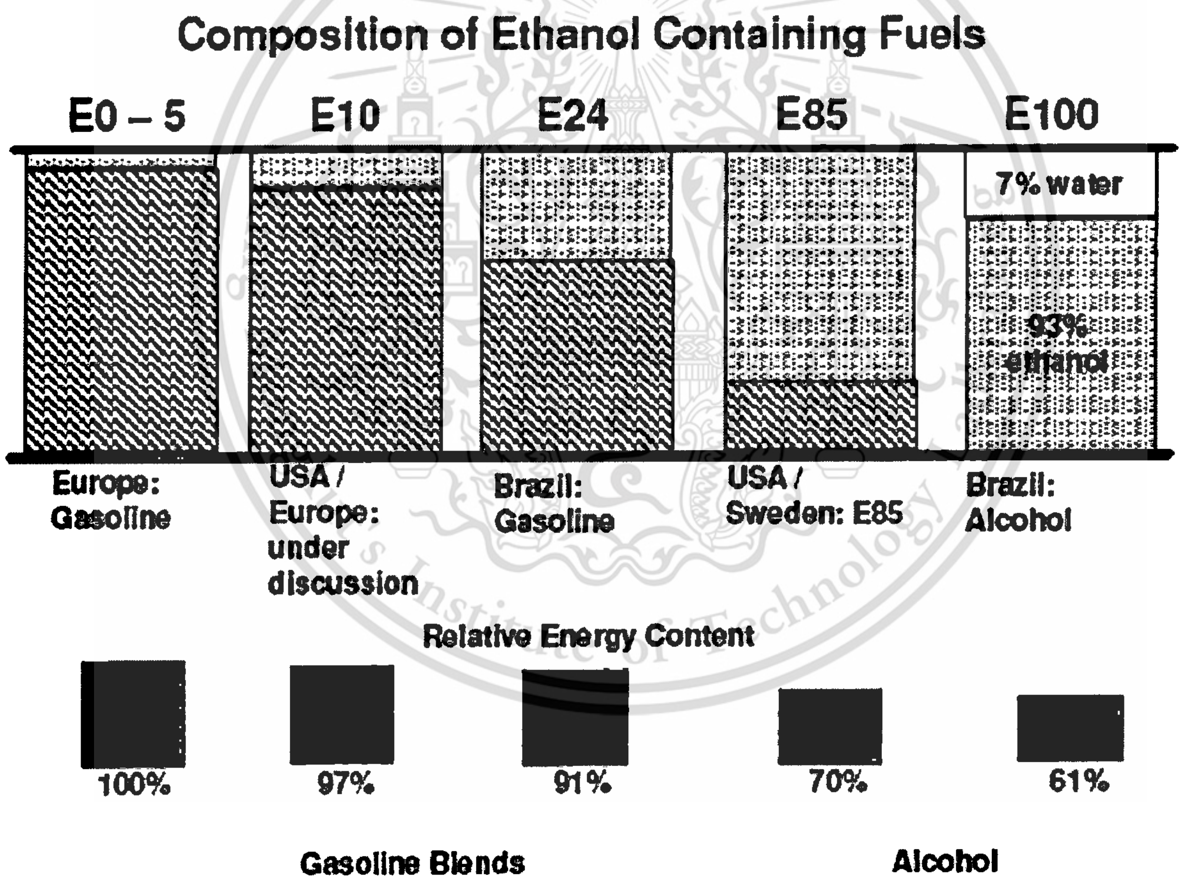


**Fig. 2.5** The relative energy content of ethanol fuels compared to gasoline []

### 2.5.2.2 Oxygen Content

When used as blending components in gasoline, oxygenates are beneficial for both combustion efficiency and exhaust emissions, especially CO emissions. Ethanol contains about 35 percent oxygen by weight and is therefore categorized as an oxygenated. Compared to other oxygenates such as MTBE, ETBE and FAME, ethanol is less toxic and therefore a good alternative.

Because of its oxygen content, ethanol has a lower stoichiometric air-fuel ratio (AFR) than gasoline, that is, 9:1 and 14.7:1, for neat ethanol and gasoline, respectively. Thus more fuel must be injected per engine cycle. In terms of energy content, a given volume (that of the engine cylinders) of stoichiometric air-fuel mixture contains about the same amount of energy with gasoline and ethanol. This is one of the main reasons current gasoline engines do not need a fundamental redesign to run on ethanol and perform similarly with either fuel.

Since the introduction of the three-way catalyst, passenger cars have been equipped with a closed-loop system to measure and ensure a stoichiometric AFR, using the lambda probe (an oxygen sensor). Newer cars are therefore able to automatically adjust the AFR, at least when using low-ethanol blends (E5 and E10). Older cars without a closed-loop system or cars with a carburetor cannot adjust the AFR and will not run with a correct AFR. An incorrect AFR can cause such problems as too lean combustion, possibly resulting in worse exhaust emissions, start problems, lack of power, or engine failures. Ethanol fuel usage is usually not recommended for these types of cars.

### 2.5.2.3 Octane Number

Perhaps the greatest advantage of ethanol as a fuel in SI engines is its high octane number. The efficiency of an SI engine, that is, the ability to convert fuel energy to mechanical energy, mainly depends on the compression ratio. It is therefore advantageous to increase this as much as possible. The major restraint is the fuel octane number – high - octane fuels can be used with higher compression ratios, thus yielding higher energy efficiency.

A drawback is that NOx formation inside the engine increases with increasing compression ratio due to increased peak combustion temperatures. Conversely, higher compression ratios with ethanol use seem to enable high EGR ratios, which can reduce NOx significantly. The net outcome of these two mechanisms depends on the configuration of the engine.

When a gas is compressed, its temperature increases. In an SI engine, if the temperature gets too high during the compression stroke, there is the possibility of premature auto ignition of the fuel and shockwaves forming inside the cylinder. This phenomenon is called knocking and is a design and operating parameter in gasoline and ethanol fuel engines. In SI engines, the fuel–air mixture is ignited at the start of the expansion stroke, and it is not desirable to have a premature ignition before that point because the efficiency of the engine decreases. Furthermore, heavy knocking is very harmful to the engine. The two main parameters (in a well-adjusted engine) determining whether an engine will knock or not is the compression ratio of the engine and the ability of the fuel to withstand auto-ignition. This fuel characteristic is called the anti-knock index, or the octane number. A fuel with a high octane number can thus be used in an SI engine with a high compression ratio, offering a higher overall efficiency, that is, a better fuel economy, and relatively lower $CO_2$ emissions.

Currently, much work is being done by car manufacturers to develop engines that can make optimal use of many different fuels. Operating a gasoline car on low-ethanol blends will likely take advantage of the higher octane number of ethanol to some degree. Raising the compression ratio and utilizing the higher octane rating can be problematic, if the vehicle has to be compatible with both neat gasoline and ethanol blended fuels. The result is therefore that most engines are optimized for regular gasoline. Currently there are no vehicles on the market that can automatically change the compression ratio according to the fuel, but some experimental concepts have been demonstrated, and many of the major car manufacturers are active in this area. At present, unfortunately, variable compression ratio for optimal fuel efficiency in FFVs is not economically feasible.

Alternatively, ethanol in low-level blends can be used while maintaining a regular octane number. Adding ethanol can, instead of boosting octane number, remove the need for other more toxic or expensive octane-boosting gasoline components, such as alkylate or aromatic compounds.

Another important fuel property of ethanol is its rather high latent heat of vaporization, a measure of the amount of energy required to evaporate the fuel. In an SI engine, vaporization of the fuel absorbs energy from the engine surroundings, thus lowering the temperature in the intake manifold and combustion chamber of the materials and air, depending on the injection method. Since ethanol has a much higher heat of vaporization than gasoline, engine temperatures tend to be lower when ethanol fuels are used. This property complements the high octane number, because auto-ignition or knocking is less likely to occur with a cooler running engine. A benefit of the high latent heat, especially for direct fuel injection (DI) engines but also for port fuel injection (PFI) engines, is the charge cooling. A cooling of the intake air--fuel mixture, due to a relatively large amount of ethanol (due to the lower stoichiometric AFR) and the high latent heat, increases the air density, thus allowing more air to enter the fixed volume of the engine cylinders. When more air is forced into the engine, more fuel can be injected and more power is created by the same engine size, resulting in increased efficiency. Furthermore, the lower operating temperatures tend to increase engine efficiency because of lower internal heat losses and also lower exhaust gas heat losses, which is observed as lower exhaust gas temperatures. The work needed during the compression stroke has also been shown to decrease due to the high latent heat, thus contributing to improved engine efficiency. The high heat of vaporization has the major disadvantage of further worsening the engine cold start properties of ethanol fuels.

Octane Number of gasoline

$$(ROM + MON)/2$$

Octane Number of blended fuel

$$O_{MIX} = O_A \times m_{fA} + O_B \times m_{fB} +, \ldots$$

Ethanol combustion equation

$$C_2H_5OH + 3(O_2 + 3.76N_2) \rightarrow 2CO_2 + 3H_2O + 11.28N_2$$

Gasoline combustion equation

$$C_8H_{15} + 11.75(O_2 + 3.76N_2) \rightarrow 8CO_2 + 7.5H_2O + 44.18N_2$$

Air-fuel ration equation

$$AF = m_a / m_f$$

$m_a$ is Air mass.

$m_f$ is Fuel mass.

## 2.6 Literature Reviews

### 2.6.1 Performance

Matthew Brusstar [1], Ethanol-Gasoline Blends: Fuel Economy and Emissions Benefits. Presented at the SAE Government and Industry Meeting in Washington, D.C., 2003

The use of ethanol and gasoline in 1.9 L engine, compression ratio of 19.1:1, shows bmep of ethanol increase 20 bars when compare with gasoline. Moreover laminar flame rate of ethanol is higher than gasoline 10 centimeters/second.

Christpher P.Cooney, Yeliana [30], Combustion Characterization in an Internal Combustion Engine with Ethanol – Gasoline Blended Fuel Varying Compression ratios and Ignition Timing. Energy & Fuels, March 2009.

The investigation of gasoline/ethanol blended (E0, E20, E60 and E84) combustion under a stoichiometric condition shows burning rate of E84 is fastest the E60, E40, E20 and E0, respectively.

Guder OL. (1984) [31], Correlations of laminar combustion data for alternative SI engine fuels. Wagner TO. (1979), Practicality of alcohols as motor fuel. SAE

Faster burning can reduce engine knock and can lead to operation with higher compression ratios and with leaner mixtures; thus, higher efficiency, higher power output and more stable engine operation could be obtained. Furthermore, leaner mixtures may also reduce hydrocarbon emissions from the engine.

Bayraktar, H. (2007) [32]. "Theoretical investigation of flame propagation process in an SI engine running on gasoline–ethanol blends."

Blending ethanol with gasoline up to 25% by volume positively affects the geometric properties of flame and the mass burning rate, leading to faster burning. It also produces higher cylinder pressures and temperatures compared with gasoline. As a result, the mean indicated work, and therefore engine output power and thermal efficiency, may also increase.

H. Serdar Yucesu a, Adnan Sozen, Tolga Topgul, Erol Arcakliog. 2006 [12]. 'Comparative study of mathematical and experimental analysis of spark ignition engine performance used ethanol–gasoline blend fuel.' Applied Thermal Engineering. 27(2007)358-368.

The study of gasoline/ethanol blended (E10, E20, E40 and E60) consumption shows the increasing of ethanol percentage lead to more fuel consumption due to lower heating value of ethanol compare with gasoline. BSFC of E60 is 330g/kW.h and gasoline is 260 g/kW.h. Moreover the comparison of pure ethanol and gasoline shows fuel consumption of ethanol is higher than gasoline 38%.

### 2.6.2 Emissions

Huseyin Serdar Yucesu, Tolga Topgul, Can Cinar, Melih Okur. 2006 [10]. "Effect of ethanol–gasoline Blends on engine performance and exhaust emission in different compression ratio." Applied Thermal Engineering. 26 (2006) 2272–2278.

The increasing of ethanol percentage in blended fuel reduces CO concentration up to 11% and 10.8% in E40 and E60 respectively. In addition, the lowest HC concentration is 16.45% under E60 compare with gasoline.

Catapano, F., Iorio, S.D., "Use of Renewable Oxygenated Fuels in Order to Reduce Particle Emissions from a GDI High Performance Engine," SAE Technical Paper 2011-01-0628, 2011, doi: 10.4271/2011-01-0628 [14].

A strong reduction particles smaller than 10 nm only in idle engine operating condition and ethanol fuelling, suggestion their volatile nature.

The use of ethanol with stratified charge strategies allows the improvement in the performance stability and limits the exhaust particulate matter and HC

Catapano, F., Sementa, P., Vaglieco, B.M., "Thermodynamic and optical characterizations of a high performance GDI engine operating in homogeneous and stratified charge mixture conditions fueled with gasoline and bio-ethanol" Fuel (2012) [19].

For both fuels, the stratified flame front, in the first phase of combustion, spread faster (40%) than the homogeneous case, due to the A/F ratio distribution; moreover the burning of fuel impinged on the piston bowl produces more soot and UHC. Particularly, in stratified mixture combustion, ethanol showed faster burning rates and higher peak pressures (about 25% at 1000 rpm) compared to gasoline. The ethanol flames show always a smaller number of bright spots than gasoline.

Maricq, M.M., Podisiadlik, D.H., "Particulate Emissions from a Direct-Injection Spark-Ignition (DISI) Engine," SAE Technical Paper 1999-01-1530, 1999 [20].

The particulate emissions exhibit a strong sensitivity to injection timing; generally particle number and volume concentrations increase steeply as the injection timing is retarded, except over a narrow portion of the range where the trend reverses. Under both homogeneous and stratified charge operation, advancing the spark timing leads to the emission of a higher number of particles and to an increase in particle size. Increasing speed and load both cause higher particulate emissions; however, for stratified charge operation these general trends are strongly influenced by injection timing.

Maricq, M.M., "Soot formation in ethanol/gasoline fuel blend diffusion flames" Combustion and Flame 159 170-180 (2012) [33].

The relatively small influence of low levels of ethanol/hydrocarbon flame studies. This suggests that in addressing the question of how low level ethanol blends affect motor vehicle PM emissions, other factors such as fuel volatilization and mixing may be equally or more important than combustion chemistry

# Chapter 3
# Experimental Procedures

## 3.1 Apparatuses and Experimental Set Up

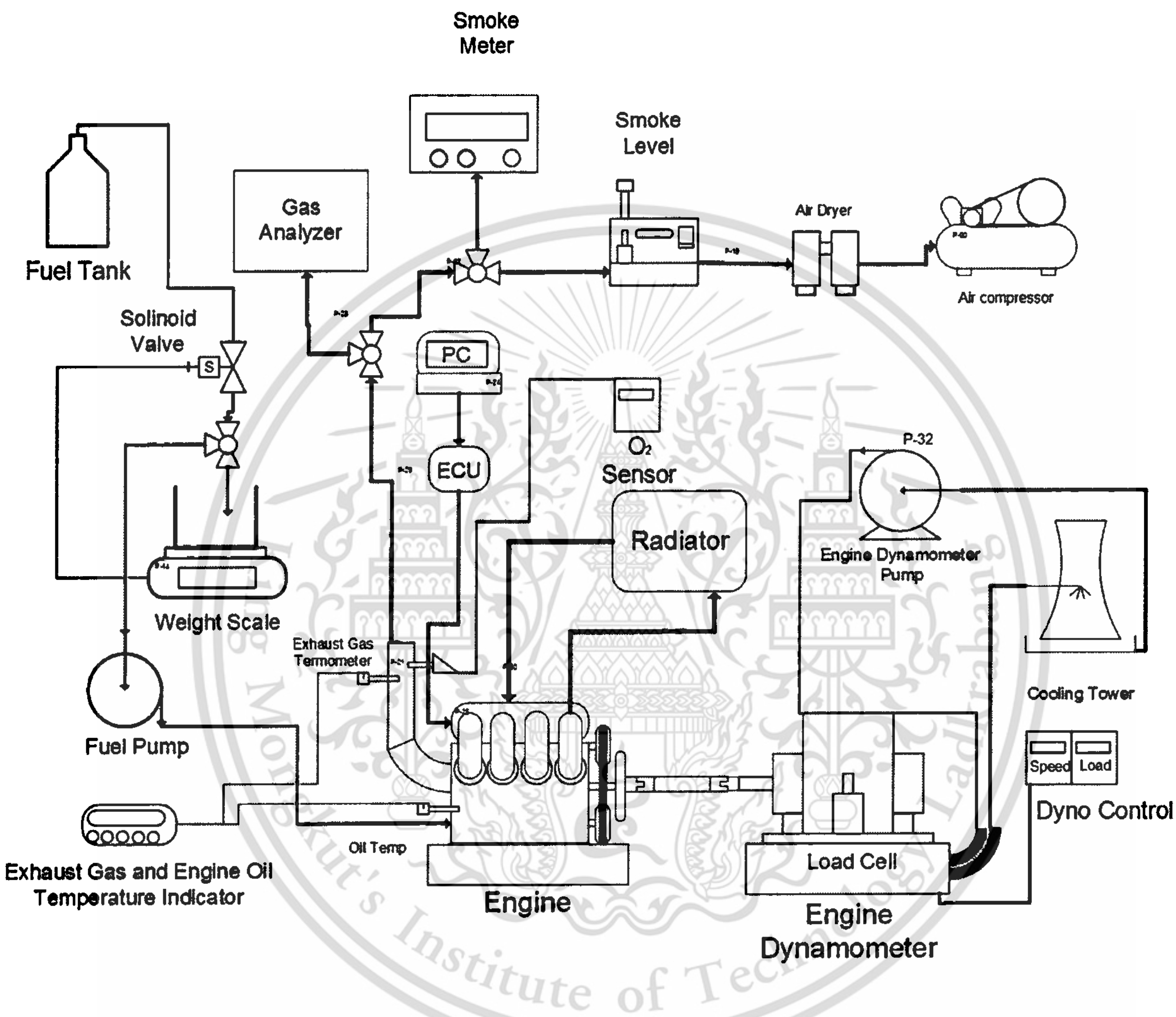


**Fig. 3.1** Schematic diagram of experimental setup

### 3.1.1 Engine

The direct injection spark ignition engine, Mitsubishi model 4G93 GDI, is selected as a test bed for this experiment. The engine was operated on the Eddy current engine dynamometer for both performance and emissions studying at 1000 rpm idle, 1500 rpm and 2000 rpm under 10, 20% loads. The specification of the engine, as shown in Table 3.1 and Fig. 3.2 is direct injection spark ignition, inline 4 cylinders, 4 strokes, 1834 $cm^3$ displacement.

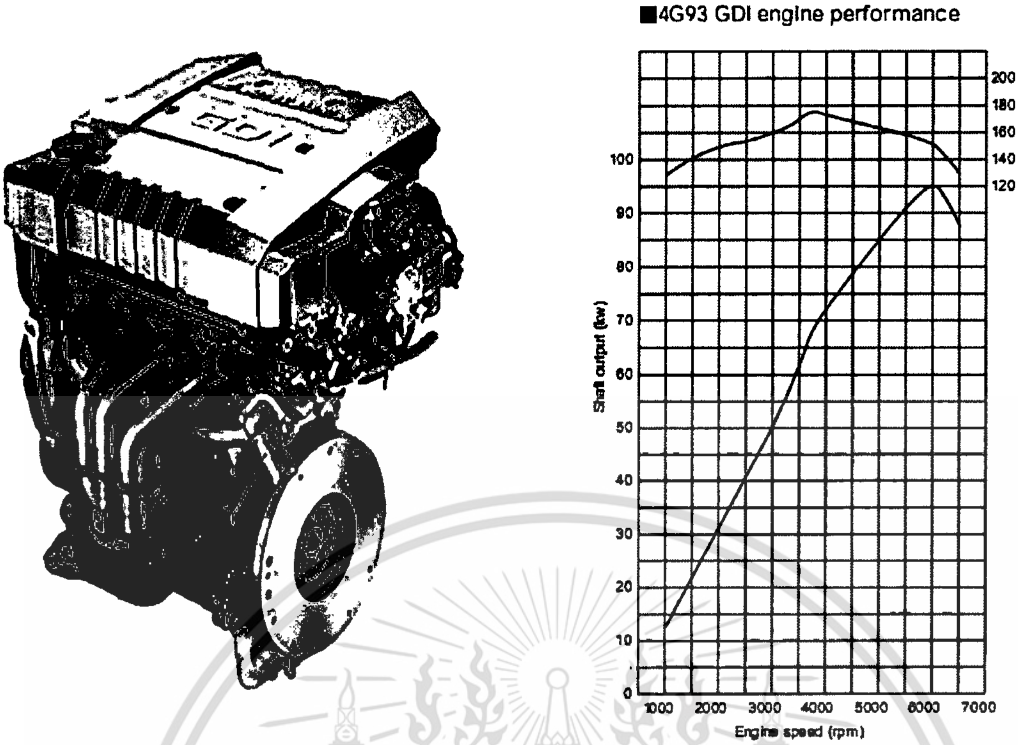


**Fig. 3.2** DISI engine and engine performance

**Table 3.1** Engine specifications

| Description | Specification |
|---|---|
| Brand | Mitsubishi |
| Model | 4G93 GDI |
| Type | In-line OHV,DOHC |
| Number of cylinders | 4 |
| Combustion chamber | Pentroof type |
| Total displacement | 1.834 Liter |
| Cylinder bore (mm) | 81.0 mm |
| Piston stroke (mm) | 89.0 mm |
| Compression ratio | 12 : 1 |
| Max power | 96kW@ 6000rpm |
| Max torque | 177Nm@ 3750rpm |

**Table 3.1** (Continued)

| | |
|---|---|
| Fuel injection system | Direct injection |
| Start system | Electric motor |
| Lubrication system | Wet sump |
| Idle speed | 1000 rpm |
| Ignition type | Spark ignition |

### 3.1.2 Eddy Current Engine Dynamometer

The engine dynamometer, Tokyo Plant model ED-150 Horizontal, was used in the experiment for applying a load on the tested engine and also measuring force, moment of force (torque) and power that the tested engine can produce against the load. The type of the engine dynamometer is Eddy current with external water cooling systems. Eddy current dynamometer can provide a quick load change rate for rapid load setting. Eddy current dynamometer consists of an electrically conductive core moving across a magnetic field to produce resistance to movement. The magnetic field is generated by using variable electromagnets that can change the magnetic field strength to control the amount of braking. The electromagnet voltage is control by a desktop computer, using changes in the magnetic field to match the power output.

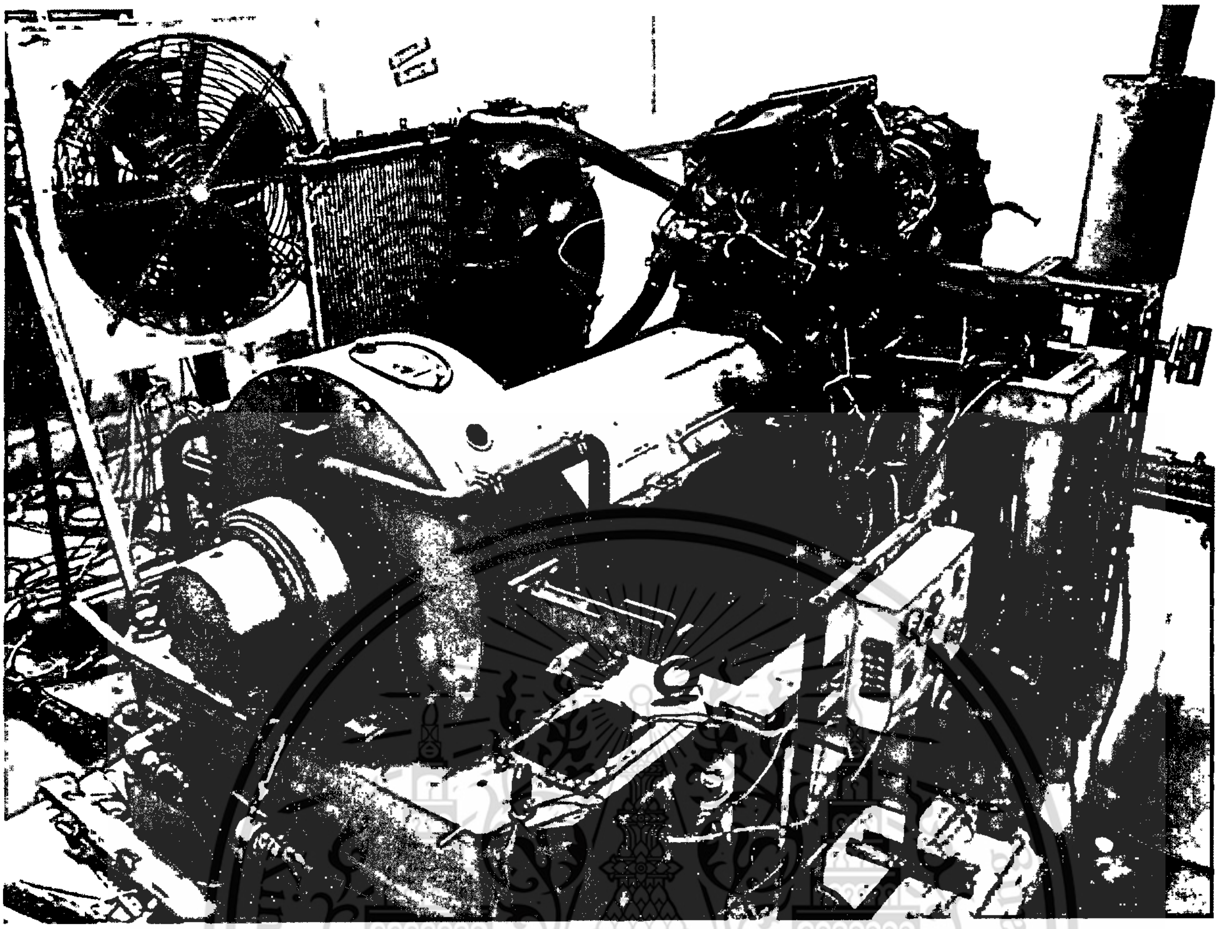

**Fig. 3.3** Engine Dynamometer

### 3.1.3 Fuel Supply System

The fuel supply system was designed to feed the tested fuel from a fuel tank through a filter by a low pressure feed pump, from automotive feed pump. The feed pressure was approximately 0.3 MPa with 2 ltr/min flow rate. After that, excess fuel will be regulated and drain back to the fuel tank by a regulator. The regulated fuel was feed further through another filter before a high pressure pump. (Cam-driven type). The low pressure fuel was pressurized by the plunger within the pump from 0.3 MPa to 4 – 7 MPa. Finally, the pressurized was deliver to the last equipment before discharged to the combustion chamber, which was a GDI injector. The fuel supply system was control by the engine dynamometer program. Using 2 solenoid valves, the 1$^{st}$ valve was used to control the fuel from the tank and the other valve was used to control beaker filling as shown in Fig. The fuel supply system can be set to operate on auto filling mode. The fuel weight was measure by a digital weight scale connected to engine dynamometer program. The weight data was used for BSFC and BSEC calculation.

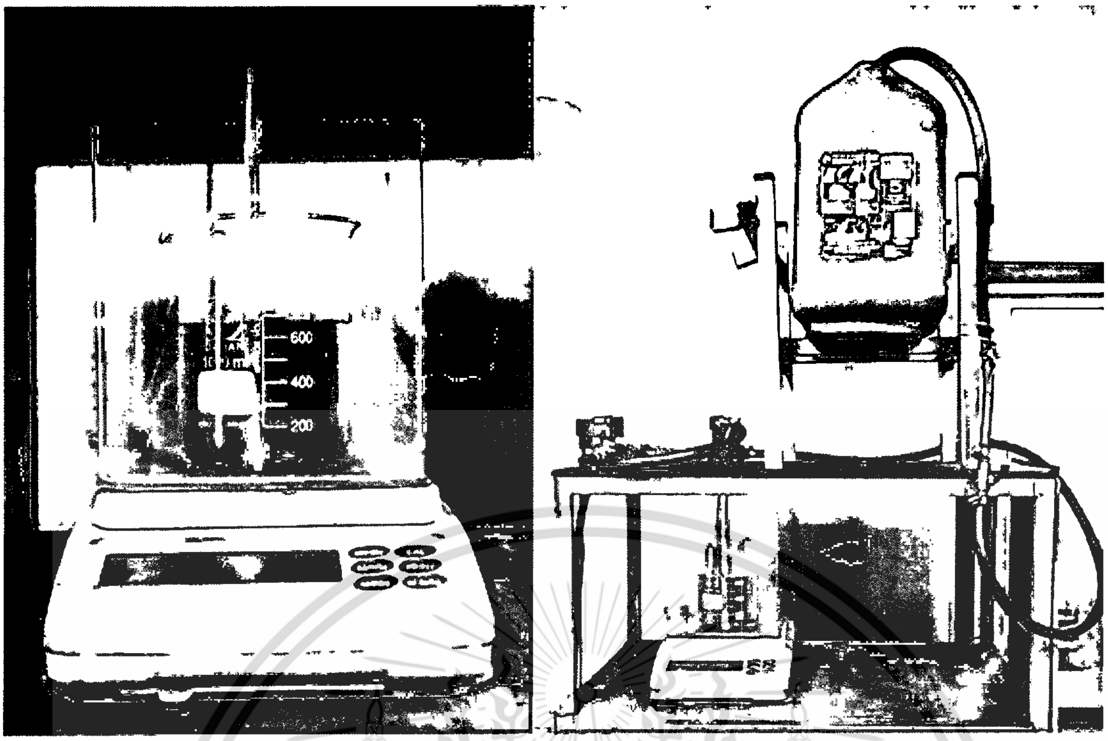


**Fig. 3.4** Fuel supply system

**Fig. 3.5** Test fuels

**Table 3.2** Fuels properties [34]

| Fuels properties | Gasoline [8] | E10 | E20 | E85 | Ethanol [8] |
|---|---|---|---|---|---|
| Formula | $C_4$ to $C_{12}$ | $CH_{2.043}O_{0.015}$ | $CH_{1.63}O_{0.065}$ | $CH_{2.822}O_{0.425}$ | $C_2H_5OH$ |
| Molecular weight [g/mol] | 100-105 | 99.47 | 88.12 | 50.60 | 46.70 |
| Carbon [mass%] | 85-88 | 86.70 | 79.85 | 55.36 | 52.20 |
| Hydrogen [mass%] | 12-15 | 13.20 | 12.88 | 12.89 | 13.10 |
| Oxygen [mass%] | 0 | 1.94 | 7.54 | 31.75 | 34.70 |
| Density, kg/l, at 15°C | 0.72-0.77 | 0.7608 | 0.7645 | 0.78365 | 0.79 |
| Vapor pres., kPa at 38°C | 48-103 | 59.6 | 58.3 | 35-70 | 15.90 |
| Specific heat, kJ/kgK | 2 | | | | 2.40 |
| Heat of vaporization, kJ/kg | 305.00 | | | 610-762.5 | 840.00 |
| Lower, heating val.,, 103 MkJ/lkg | 44.00 | 40.97 | 40.60 | 29.50 | 26.90 |
| Research octane number | 92.4 | 98.1 | 98.3 | 101.6 | 108.60 |
| Motor octane number | 81.2 | 82.3 | 84.6 | 91.1 | 92 |
| (R+M)/2 | 86.8 | 90.2 | 91.45 | 96.35 | 100 |
| Water Tolerance | Negligible | | | | Compl.miscible |
| Stoichiometric air/fuel ratio | 14.70 | 14.05 | 13.51 | 9.87 | 9.03 |
| Initial boiling point, IBP | 35 | 36.5 | 37.8 | 41.3 | 77.6 |
| 10 vol% | 51.5 | 51.6 | 53.5 | 66.6 | 77.8 |
| 20 vol% | 56.5 | 55.7 | 57.8 | 74.4 | 77.9 |
| 30 vol% | 61.8 | 59.7 | 62.5 | 76.8 | 77.9 |
| 40 vol% | 68.6 | 63.8 | 66.8 | 77.4 | 77.9 |
| 50 vol% | 78.2 | 70.2 | 70.8 | 77.5 | 77.9 |
| 60 vol% | 91.5 | 99.4 | 73.7 | 77.6 | 77.9 |
| 70 vol% | 108.6 | 117.9 | 99.9 | 77.7 | 78 |
| 80 vol% | 125.2 | 136.1 | 130 | 77.7 | 78 |
| 90 vol% | 154 | 160.2 | 155 | 77.8 | 78 |
| End boiling point | 197.3 | 187.2 | 184.6 | 80.5 | 80 |

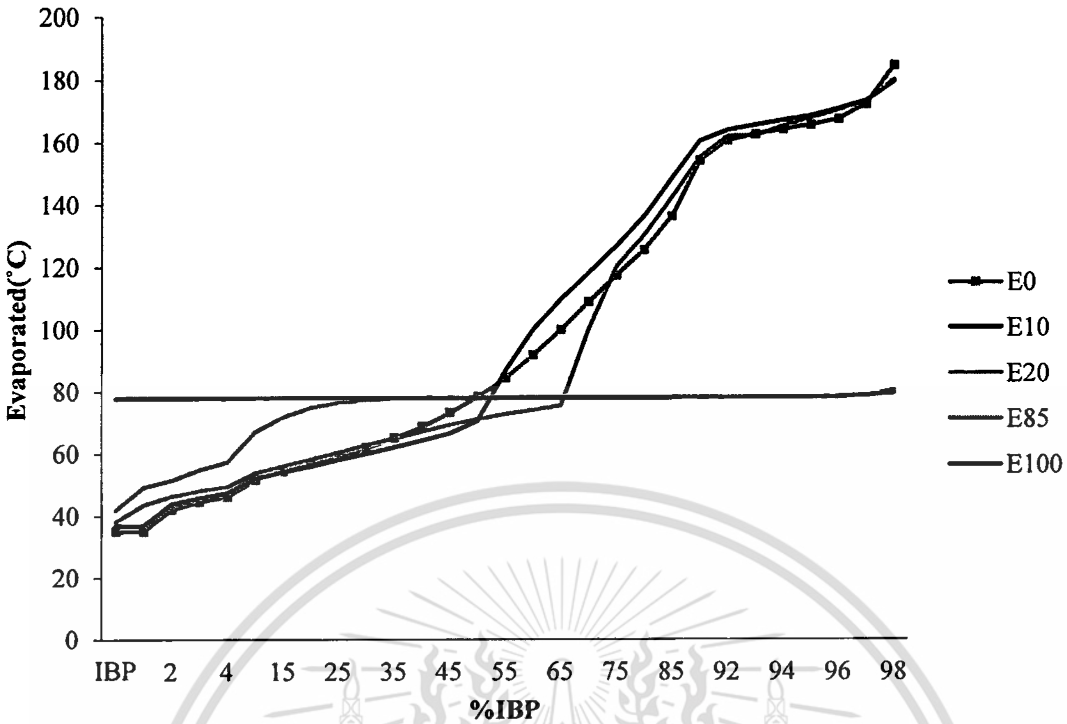


**Fig. 3.6** Gasoline-ethanol distillation curve

### 3.1.4 Fuel Distillation

Distillation curves for ethanol blends are shown in Fig. 3.6 and are provided in tabular form in the Supporting Information. The trends for ethanol-gasoline blends are similar to those observed for methanol-gasoline blends [35]. A similar type of effect has been observed with vapor pressures of blends where adding small percentages of alcohol to gasoline increases the vapor pressure more than adding larger percentages. The observations above are consistent with the well-known phenomenon of azeotropic behavior of mixtures of methanol and the hydrocarbons of the base gasoline. For a true azeotrope, the liquid molar composition is identical to the composition of the vapors formed. During distillation of an azeotrope, both compositions remain constant until the fluid is completely evaporated. Thus, the distillation curve for a true azeotrope should become flat and remain flat until the distillation is complete. With the methanol-gasoline blends, the distillation curve never becomes truly flat and exhibits a steady rise in the late region. Given these considerations, it may be more appropriate to describe this as a "near-azeotropic mixture" of the alcohol and the gasoline hydrocarbons. While the addition of 10 and 20% ethanol leads to little, or no, discernible change in IBP (initial boiling point), there is a substantial decrease in distillation temperature (i.e., increase in volatility) over the middle portion of the distillation curve. As observed with methanol blends, the addition of small amounts of ethanol (E10) gives the largest increase in volatility for the first approximately 30% of the distillation curve. As the fraction of ethanol increases (from E10 to E20), each distillation curve moves progressively closer to that of gasoline over the first approximately 30% of the volume distilled. The IBPs for E85 blends are substantially greater than those of the base gasoline and trend toward that for pure ethanol (78 °C). As similarly observed for methanol blends, the extent of the deviation in the initial portion of the distillation is substantially larger than would be expected simply from the amount of ethanol in the blend. For example, the distillation temperature for E5 increases more slowly than the base gasoline from T10 to T30, but then increases sharply at T50 and thereafter, approaching that of gasoline. These observations indicate that ethanol forms a near-azeotropic mixture with the gasoline hydrocarbons and consequently the impact of ethanol on fuel volatility is substantially greater than expected for ideal mixtures.

### 3.1.5 Temperature Indicator and Thermocouple

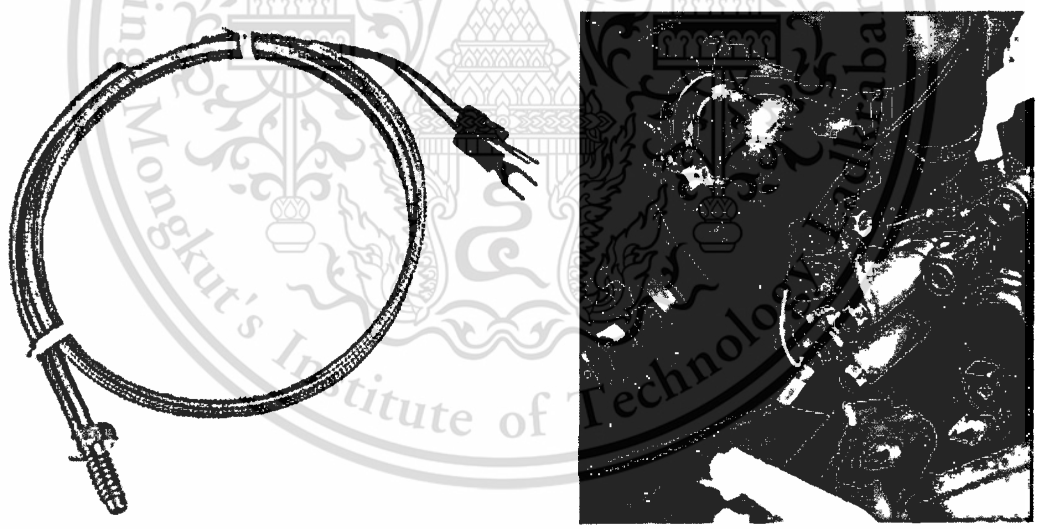

**Fig. 3.7** Thermocouple

In the experiment, thermocouple is very important for monitoring and measuring temperature. A voltage was produced when the temperature of the thermocouple differs from the reference temperature. To monitoring the temperature of each point, a digital temperature indicator was required to convert the voltage from thermocouple into temperature. The indicator provided 4 channels of the thermocouple, cooling system temperature, oil system temperature, exhaust temperature and ambient temperature.

- Cooling system temperature

A cooling temperature sensor was located at thermostat in order to measure and control the temperature of the coolant that circulate inside the engine. Fig. 3.7 shows a location of cooling temperature sensor.

- Oil temperature

An oil temperature sensor was immerged in oil pan replacing dipstick in order to measure the temperature of lubrication system which can be used to ensure the operation of the engine.

- Exhaust temperature

An exhaust temperature sensor was located at an exhaust manifold in order to measure the temperature of the exhaust gas. The exhaust gas temperature can be an indicator of rich or lean air-fuel ratio and dangerous operating conditions that can lead to engine failure.

- Ambient temperature

An ambient temperature sensor was located inside the engine test room in order to observe the temperature of the intake air. The intake air temperature must be controlled to be same temperature in every operation of the engine in order to have the same power.

### 3.1.6 Data Acquisition System

A data acquisition system was connected to the engine dynamometer by an external cable. A desktop computer was used as a commander, which operated by National Instrument LabVIEW system. The program was used to command, measure and store the data from engine dynamometer. The engine dynamometer program includes mainly 2 modes of the operation, constant force and constant speed modes, as shown in Fig. 3.8.

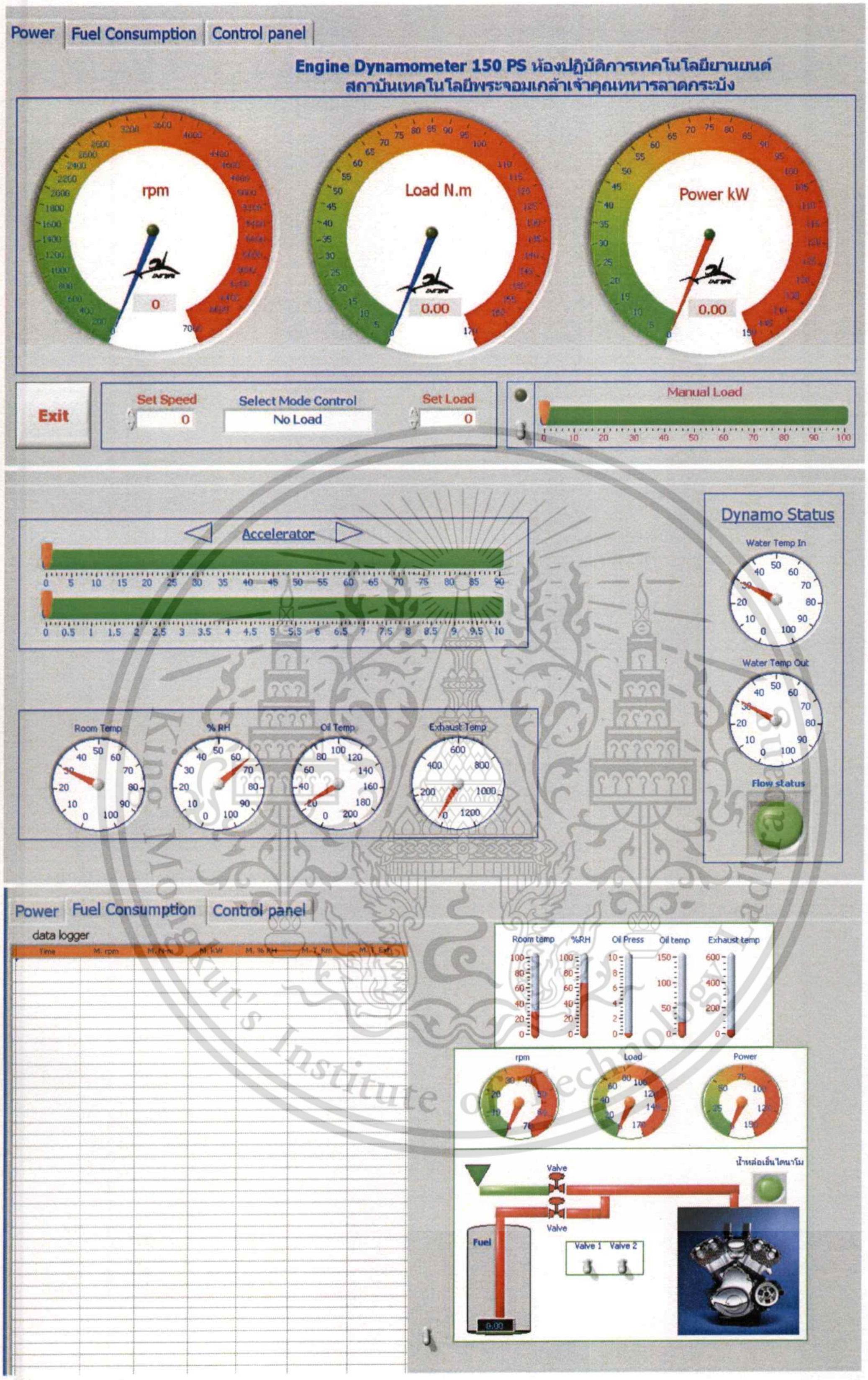


**Fig. 3.8** Engine Dynamometer program interface

### 3.1.7 Black Smoke Meter

Particulate matter emitted from engine combustion was measured by a smoke meter in the unit of black smoke percentage. The smoke meter was applied to measure the concentration of the particulate matter in exhaust gas. The clean filter was filled by particulate matter from exhaust gas then measured by light emitting method. The tested filters of gasoline-ethanol blend are shown in Fig. 3.9. The zero percentage black smoke means no particulate on filter, on the other hand 100 percentage mean the filter is covered by particulate of all area. This smoke meter percentage can be used to summarize the combustion characteristics of DISI engine. The smoke meter which was used in this experiment, Okuda DSM - 240, is shown in Fig 3.9.



**Fig. 3.9** Smoke filter and smoke meter

### 3.1.8 Gas Analyzer

A gas analyzer was applied to measure exhaust gas for DISI engine. With the principle of Non Dispersive Infra-Red, a beam of infra-red light is emitted from a light source which does not disperse or become scattered by substances between the light source and a detector. The exhaust gas was introduced into a sample chamber and absorbed the light from the light source at specific wavelengths. The light detector which was installed in the opposite side of the light source measured the amount of infra-red absorbed by the sample gas in a function of the volumetric concentration. The Technotest Multigas 488 gas analyzer which was used in the experiment in order to measure CO, $CO_2$ and HC is shown in Fig. 3.10. The resolutions of each measured gas (CO, $CO_2$ and HC) are 0.01, 0.1 and 0.1, respectively.


**Fig. 3.10** Gas analyzer

### 3.1.9 Engine Control Module

An Engine Control Unit (ECU), model DTA S60Pro as shown in Fig. 3.11, is an electronics control unit that controls output to actuators and monitor input from sensor. DTA S60Pro is a type of aftermarket modification ECU that can be programmed with a computer connected using a serial or USB cable. In the experiment, the engine control unit was used to control mainly on the amount of fuel to be injected into each cylinder, injection timing of the engine and ignition timing that the spark plug should fire in each cylinder. The variation of the amount of fuel injection depends on the proportion of ethanol in blended fuel, in order to compensate the same lower heating value of each fuel.

The engine control unit was connected to control the engine via two connectors, power and sensor connectors, as shown in Fig. 3.11. The power connector was rewired to supply a signal to 4 coils and 4 injectors. On the other hand, the signals from sensors such as throttle position, crank and cam were delivered to ECU at the sensor connector.


**Fig. 3.11** ECU

**Fig. 3.12** ECU connectors

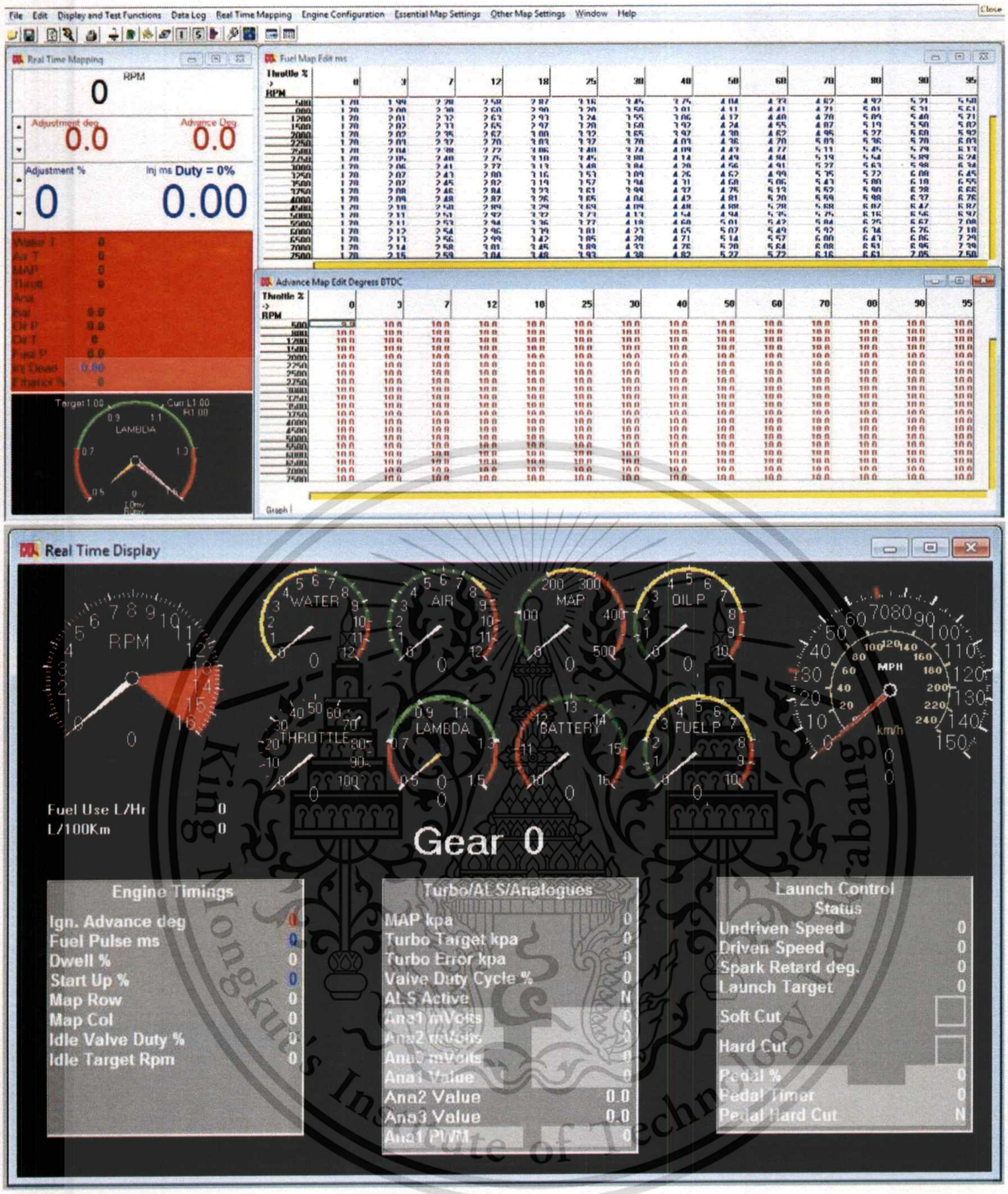

**Fig. 3.13** The DTASwin program interface

### 3.1.10 Electrical Supply System

In all experiments, the voltage of battery was maintained to be constant at 12V. The electrical supply system charged the electricity into the battery by convert the voltage from 220V to 12V to match the battery voltage.

### 3.1.11 Oscilloscope Meter

To measure electrical signal, an oscilloscope meter was required. The oscilloscope was used in the experiment in order to observe signals and frequencies that driven by the ECU and measured from the engine in the process of engine setting, as shown in Fig. 3.15. Fig. 3.13 shows the Carman Scan VG+ oscilloscope meter.


**Fig. 3.14** Oscilloscope meter

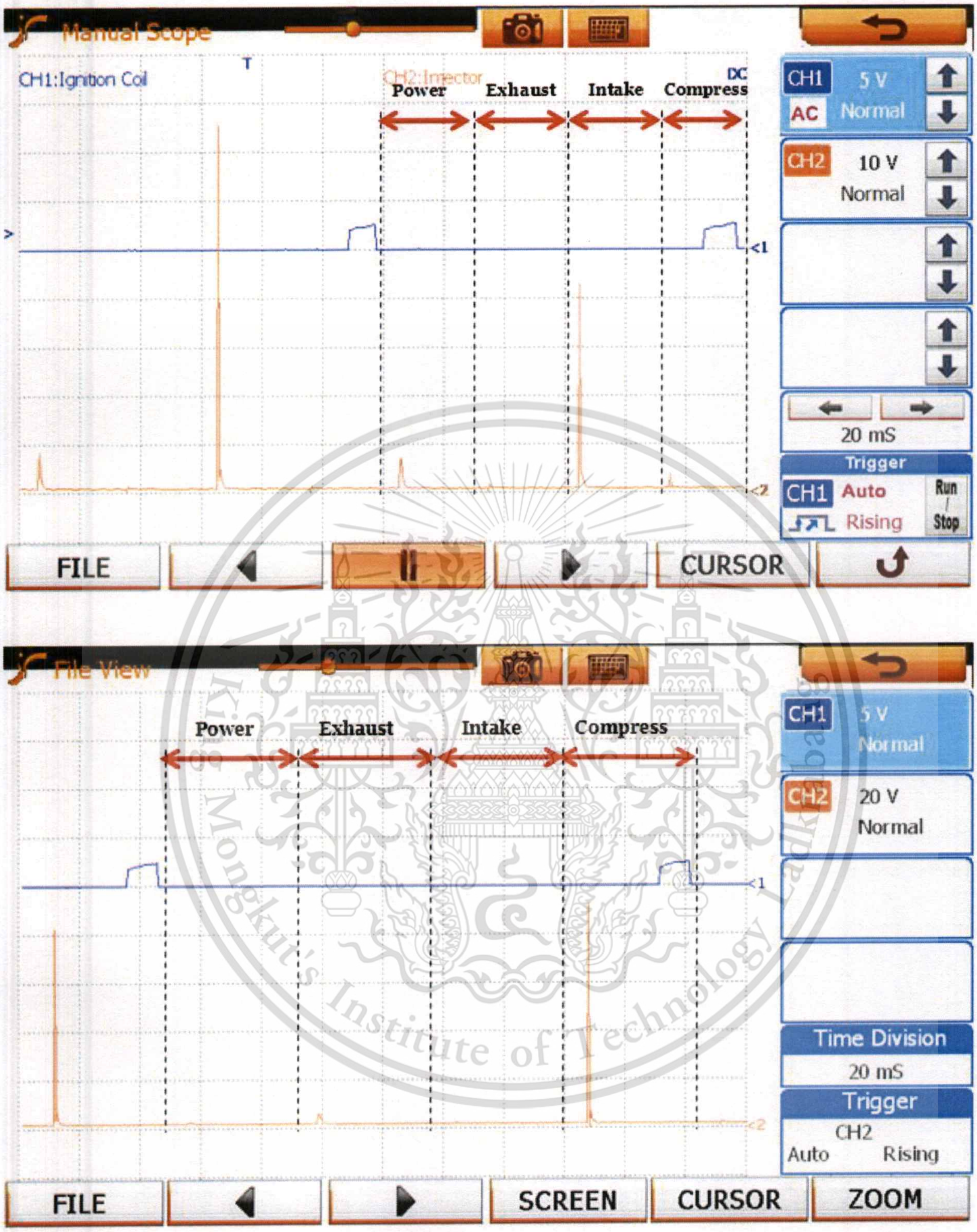


**Fig. 3.15** Signal measurement

### 3.1.12 Oxygen Sensor

An oxygen sensor was used in the experiment for measuring the proportion of oxygen (concentration) in the exhaust gas to determine the air-fuel ratio of combustion, rich or lean, in real time to ensure that how the engine burns the fuel. The signal from the oxygen sensor was convert from voltage signal to air-fuel ratio number by an air-fuel ratio meter, Innovate LM2 model. The oxygen sensor was installed at exhaust manifold which can measure the high temperature of the exhaust gas which can be used to predict combustion phenomena inside cylinder.


**Fig. 3.16** Oxygen sensor

## 3.2 Methodology

### 3.2.1 Experimental Conditions

A direct injection spark ignition (DISI) engine, inline 4 cylinders, 4 strokes, 1834 $cm^3$ displacement was measured emissions at 1000 rpm idle, 1500 rpm and 2000 rpm under 10, 20% loads, using gasoline, ethanol and ethanol-gasoline blended, respectively. The 1000 rpm idle condition was selected to study a critical condition, in terms of stability, for DISI engine, the 1500 rpm and 2000 rpm under 0, 10 and 20% loads were chosen as representative points for urban driving conditions. The injection behaviors were controlled for both homogenous and stratified charges. All the conditions investigated were carried out at $\lambda$=1. The test conditions are shown in Table 3.3 and 3.4. Particle emissions were sampled directly from the exhaust pipe, and then measured for concentration of particle emissions with smoke meter. On the other hand, exhaust emissions such as CO, $CO_2$ and HC were sampled from an exhaust pipe directly in order to be measured by a gas analyzer. In addition, performance: brake specific fuel consumption (BSFC), brake specific energy consumption (BSEC) and maximum torque advance were measured at 2000 under 20, 30 and 50% throttle to understand the variations of performance of the test engine.

**Table 3.3** Test conditions

| | |
|---|---|
| Fuel type | E0, E10, E20, E85 and E100 |
| Fuel injection behavior | Homogeneous and Stratified |
| Engine speed (rpm) | 1000, 1500 and 2000 |
| Engine load (%) | 0, 10, 20, 30 and 50 |
| Engine load (MPa) | 0.65 and 1.3 MPa |
| Lambda ($\lambda$) | 1 |

**Table 3.4** Load and speed

| Wide open throttle (WOT) | |
|---|---|
| Engine Speed (RPM) | Torque (Nm) |
| 1,000 | 130 |
| 1,500 | 140 |
| 2,000 | 150 |
| 2,500 | 155 |
| 3,000 | 160 |
| 3,500 | 170 |
| 3,750 | 177 |
| 4,000 | 174 |
| 4,500 | 169 |
| 5,000 | 163 |
| 5,500 | 158 |
| 6,000 | 153 |
| 6,500 | 130 |

### 3.2.2 Experimental Procedures

In this study, the experimental procedures were divided into 3 parts: Particle emissions, Gas emissions and Performance, to show the dominant advantages of the oxygenated fuel. The procedure of each study is explained by using the flow chart which consists of apparatuses, parameters, conditions and results.

#### 3.2.2.1 Particle Emissions

The diffusion flames of both gasoline and diesel were generated by the lamp which was controlled at the same burning rate of the energy. In order to compare a combustion phenomenon between gasoline and diesel diffusion flame, the length of the diffusion flames were captured by an optical image. On the other hand, core of gasoline and diesel diffusion flames were observed by Schlieren method image. The Schlieren method images showed the comparison of gasoline and diesel particle emissions formed region in the diffusion flame.

Particle emissions of gasoline and diesel diffusion flames were trapped by a paper filter directly. The paper filter was taken an image in Micro-scale by a Scanning Electron Microscope (SEM) to observe particle emissions accumulated size. On the other hand, a power of particle emissions was taken an image in Nano-scale by a Transmission Electron Microscope (TEM) to observe particle emissions primary size. Then, particle emissions images from both techniques were measured the size by average of two diagonal lengths. The results of 100 particles showed a comparison of accumulated and primary size distributions of gasoline and diesel particle emissions.

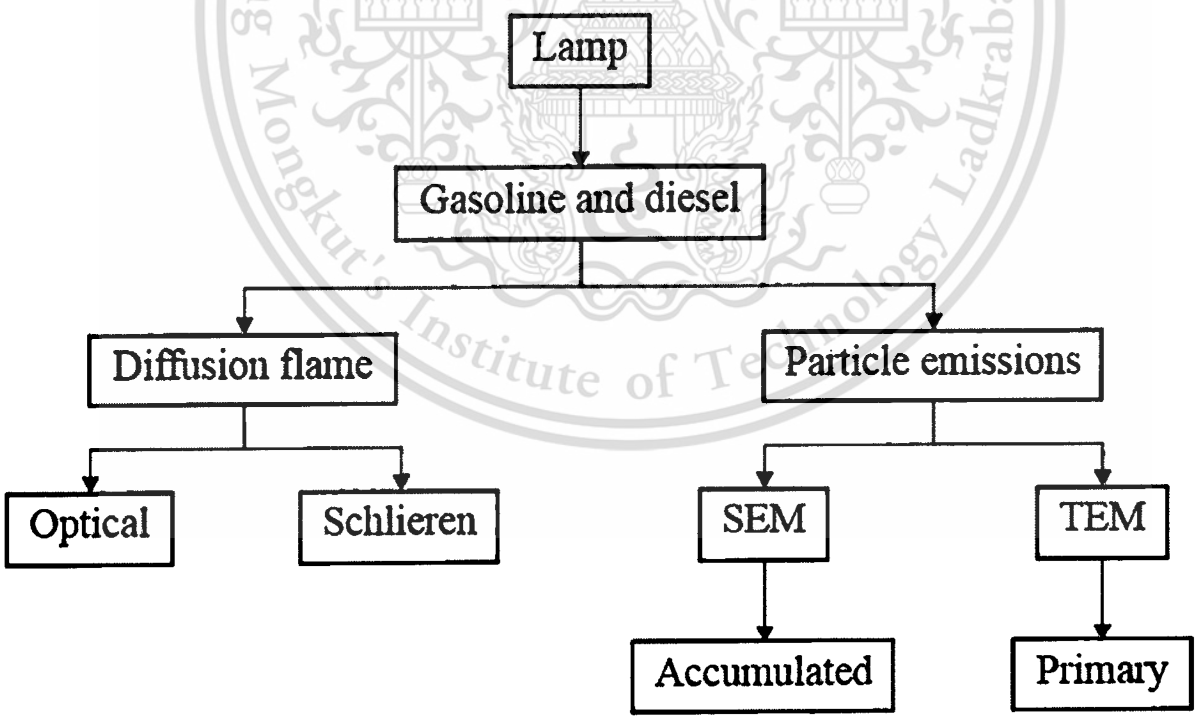


**Fig. 3.17** Particle emissions experimental procedure

#### 3.2.2.2 Gas Emissions

The DISI engine was coupled with Eddy current engine dynamometer which can control the engine load and speed. The engine was fuelled with gasoline, ethanol and gasoline blended fuels. The aftermarket ECU was used in the study to switch the engine operating strategy by adjust both injection timing and duration. A stoichiometric air/fuel ratio was control to be 1 in all test conditions either stratified or homogeneous modes. Exhaust gases from the engine were sampled directly from the exhaust pipe. Smoke meter and Gas analyzer were introduced to measure particle emissions and gas emissions: CO, $CO_2$ and HC concentration, respectively. The results showed the comparison of both particle and gas emissions by using gasoline-ethanol blended fuels under the varying engine loads and speeds condition.

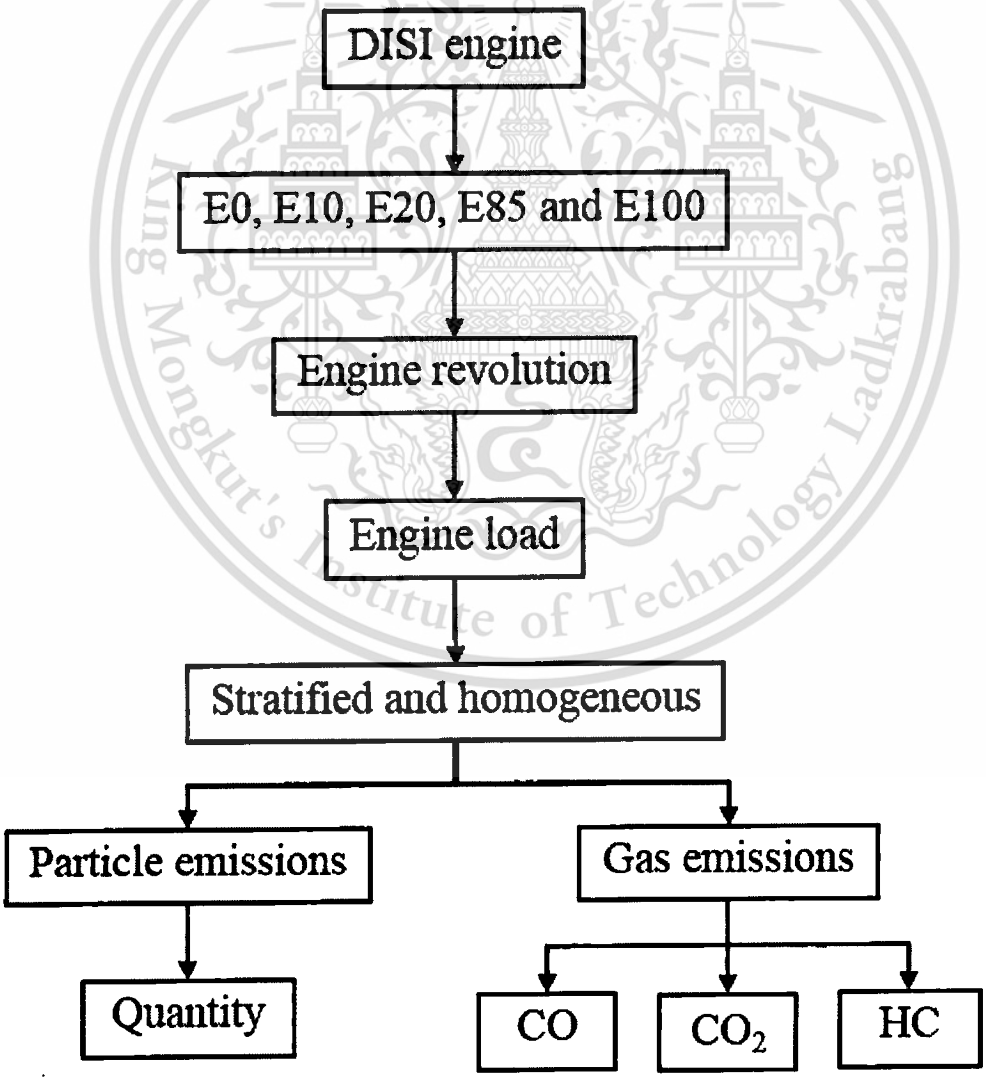


**Fig. 3.18** Gas emissions experimental procedure

### 3.2.2.3 Performance

To investigate ethanol DISI engine performance, the engine was operated under the same procedure as the gas emissions procedure which mentioned previously. In addition, the maximum brake torque was considered by the ignition timing modulation. The ignition timing was modulated in the advanced direction: increment of bTDC degree. Then the maximum brake torque of each operating condition were used in order to calculate BSFC and BSEC.

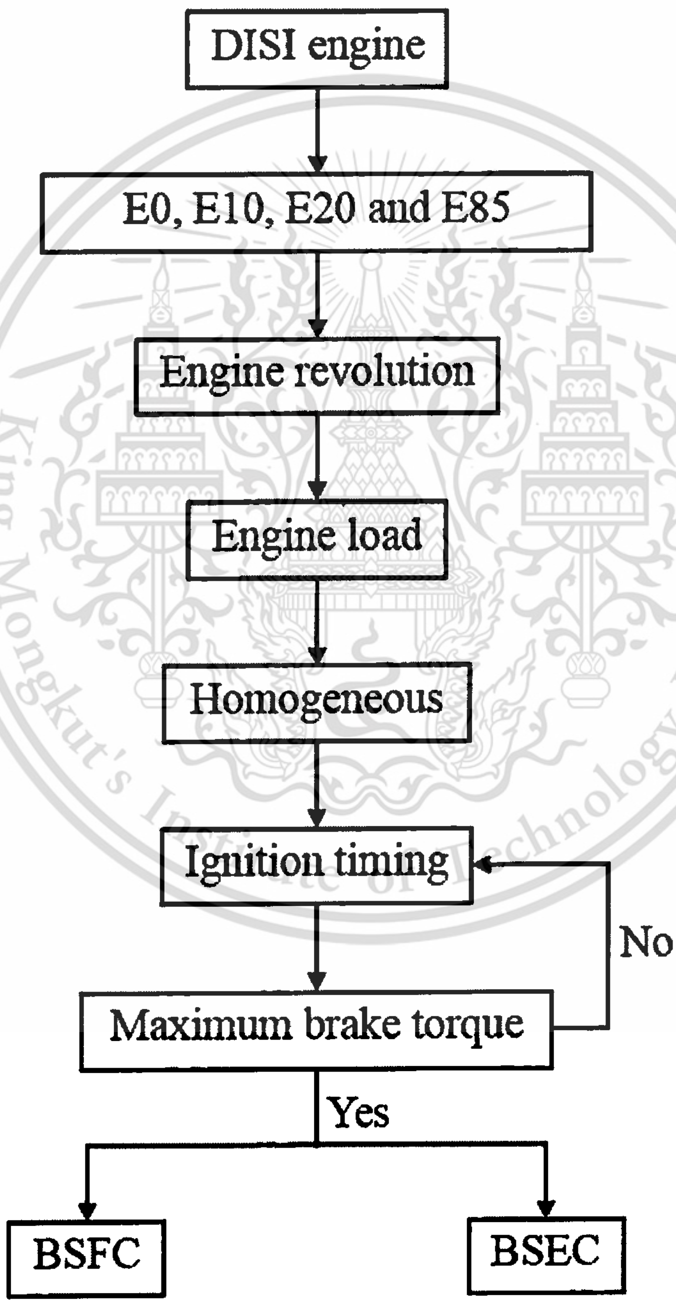


**Fig. 3.19** Performance experimental procedure

# Chapter 4
# Results and Discussions

## 4.1 Effect of Oxygenated Fuel

### 4.1.1 Structure of Particle Emissions

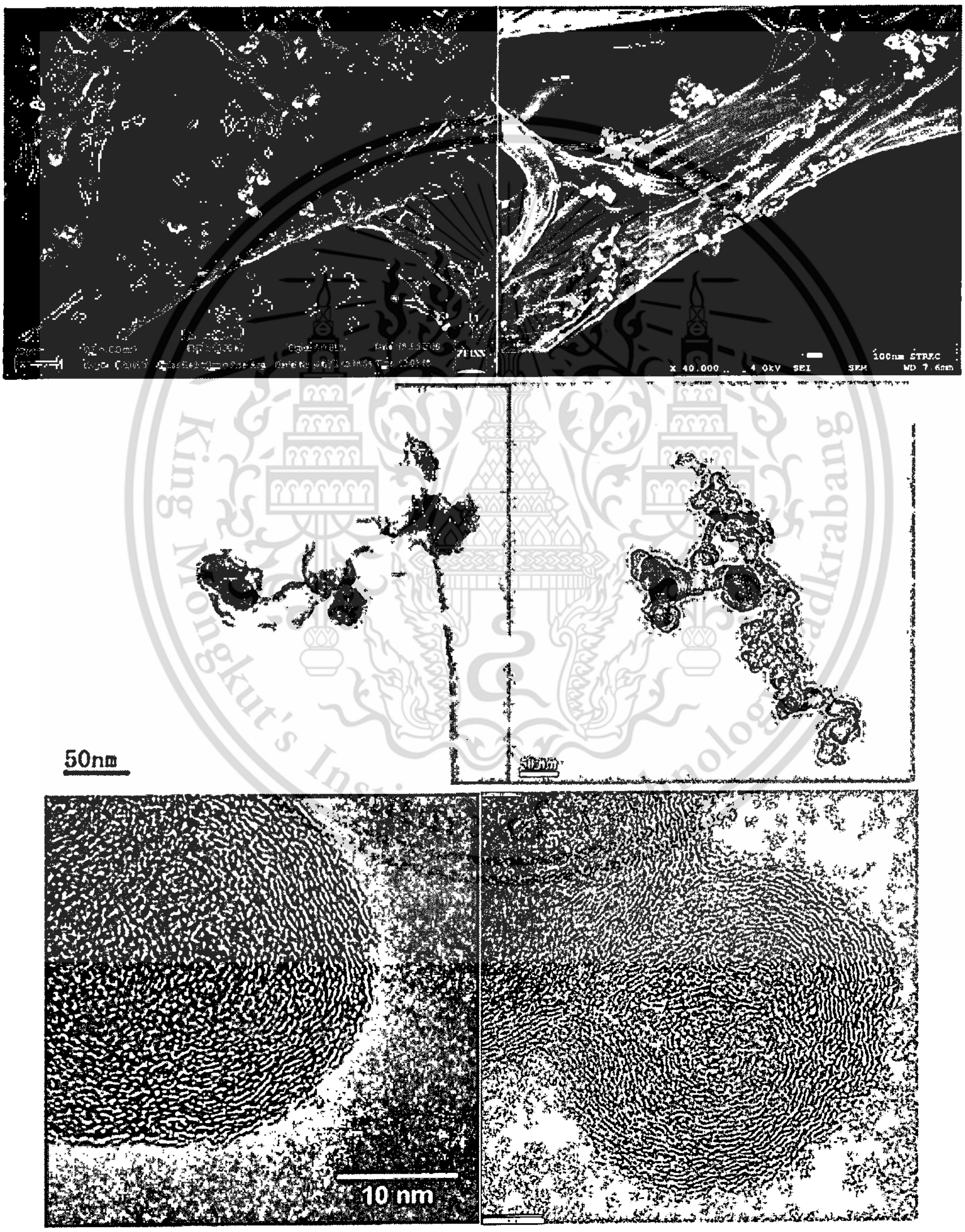


**Fig. 4.1** SEM and TEM images of gasoline (left) and diesel (right) engine particle emissions [36-38]

Using DISI engine fuelled with gasoline has been reported to produce not only more particle emissions than conventional port fuel injection gasoline spark ignition engine but also same morphology asdiesel engine, as shown in Fig. 4.1. It would be better to study the reduction of DISI engine particle emissions as well as diesel engine particle emissions. Gasoline and diesel particle emissions images were taken in SEM and TEM methods in order to verify primary and accumulated size and formation of particle emissions. Gasoline and diesel particle emissions were generated by fuel lamp. Fig. 4.2 and 4.3 show SEM image of gasoline and diesel accumulated particle emissions, respectively. Accumulated sizes of gasoline were slightly smaller than that of diesel, 100-360 nm vs. 100-500 nm, respectively.

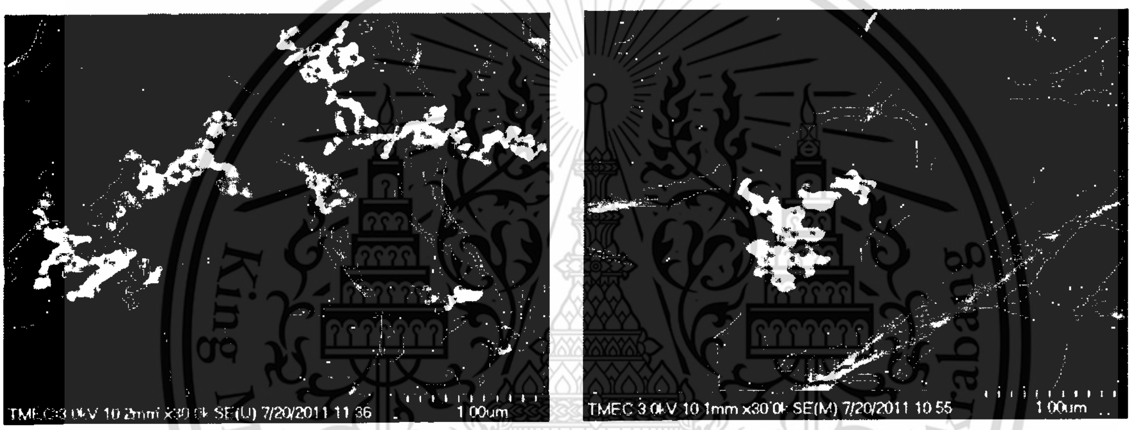

**Fig. 4.2** SEM image of gasoline (left) and diesel (right) particle emissions

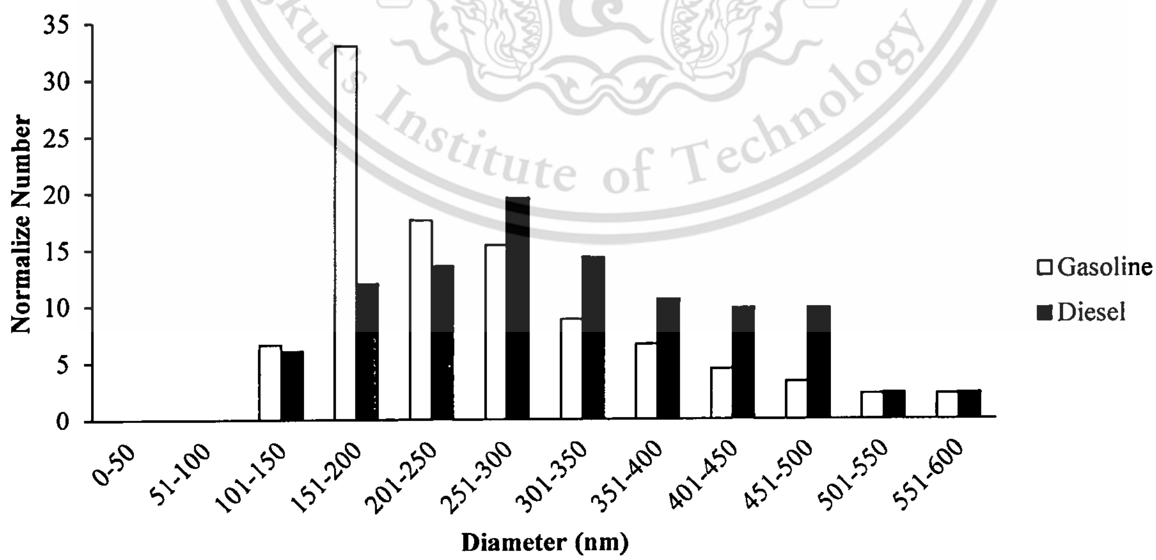


**Fig. 4.3** Accumulated size distribution of gasoline and diesel lamp by SEM images

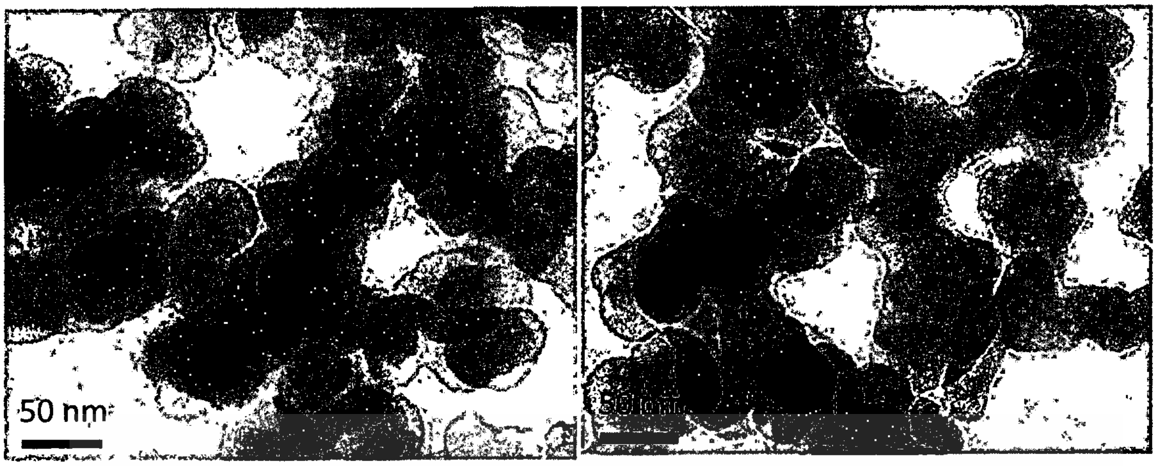


**Fig. 4.4** TEM image of gasoline (left) and (diesel) particle emissions

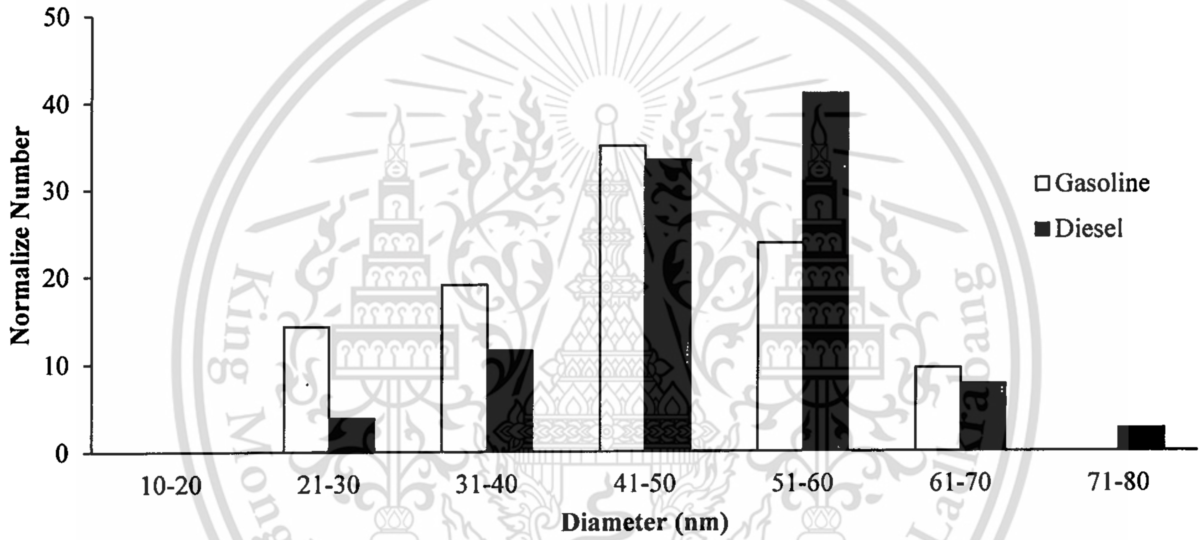


**Fig. 4.5** Primary size distribution of gasoline and diesel lamp by TEM images

However, the primary size of particle emissions is very difficult to measure by SEM image because surface of particle emissions were cover by unburned hydrocarbon. Then, primary size of particle emissions was measure later by TEM image. Primary size of gasoline and diesel particle emissions was shown in Fig. 4.4 and Fig. 4.5, respectively. TEM image was used to verify primary size, because they weren't covered by unburned hydrocarbon. Primary size of gasoline is slightly smaller than that of diesel, primary size of both particle emissions were approximately 25-60 nm and 50-60 nm, respectively. The particle size distribution in primary mode approximately 100 particles was shown in Fig.8. Most of gasoline lamp particle size is around 41-50 nm while the diesel lamp particle size is around 51-60 nm, slightly bigger than that of gasoline lamp particle.

Fig. 4.6 shows gasoline and ethanol diffusion flames by Schlieren method image. The core of ethanol diffusion flame was lighter than that of gasoline, where particle emissions are formed at the center of diffusion flame. Similarly, Fig. 4.7 shows gasoline and ethanol diffusion flames by optical image. The length of ethanol diffusion flame was shorter than that of gasoline due to ethanol contains less carbon content than gasoline and ethanol also contains oxygen atoms that promote more complete combustion.

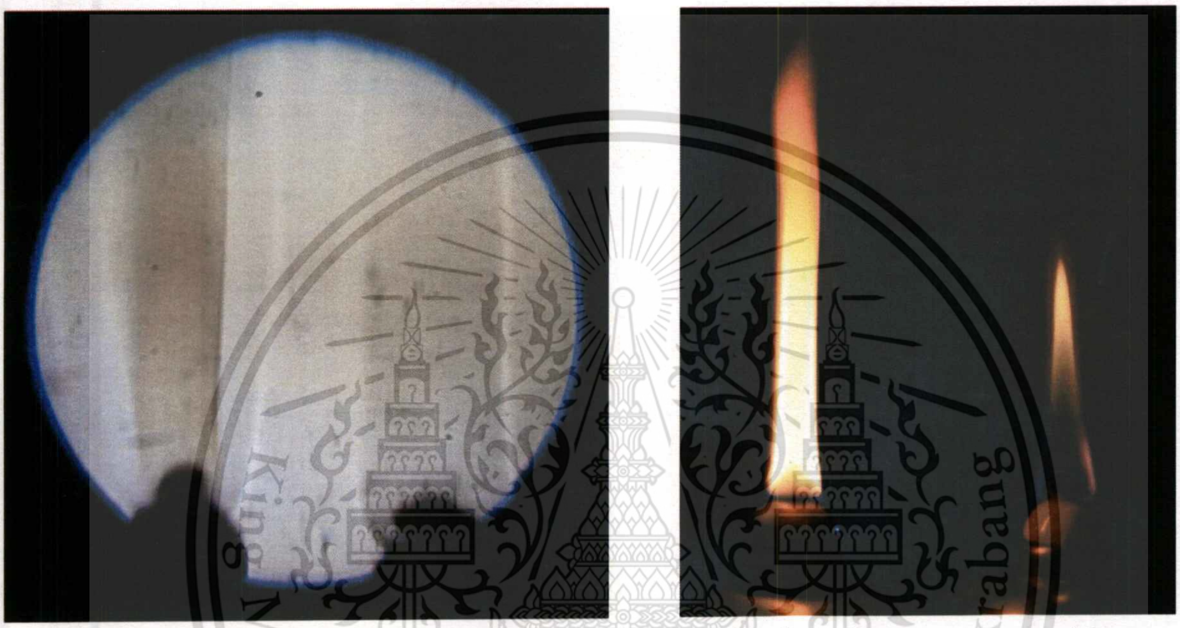

**Fig. 4.6** Gasoline (left) and ethanol (right) diffusion flames captured by Schlieren method and conventional digital camera

**Fig. 4.7** Gasoline (left) and ethanol (right) diffusion flames captured by conventional digital camera

## 4.2 Effect of Conventional Engine

### 4.2.1 Particle Emissions Concentration

The particle emissions were formed in the rich fuel region in the combustion chamber. The exhaust gas was emitted in the exhaust pipe. The remaining particle emissions of gasoline and ethanol were trapped by using a paper filter, as showed in Fig. 4.8, directly from the exhaust pipe. Subsequently, smoke meter was applied to measure the concentration of trapped particle emissions on the paper filter by light opacity method. The zero and 100 percentages of black smoke means no and full of particle emissions on the filter paper, respectively.

**Fig. 4.8** Tested paper of gasoline (left) and ethanol (right) particle emissions

Fig. 4.9 shows the black smoke percentage of gasoline and ethanol under no load condition when varying engine speeds and injection behaviors. The results showed that the percentage of black smoke of gasoline was higher than that of ethanol for all cases. The maximum percentage of black smoke was 1000 rpm, stratified and gasoline condition. Fig. 4.10 shows the black smoke percentage of gasoline and ethanol at 1500 rpm varying engine speeds, injection behaviors and loads. The results showed that the percentage of black smoke of gasoline was higher than that of ethanol for all cases. The maximum percentage of black smoke was 1500 rpm, stratified charge mode, 20% load and gasoline.

Hence, gasoline particle emissions formation was higher than ethanol. This could be explained that more particle emissions were remained in gasoline combustion than those of ethanol. Since ethanol contains oxygen molecules, ethanol is readily oxidized with the available oxygen in the flame zone.

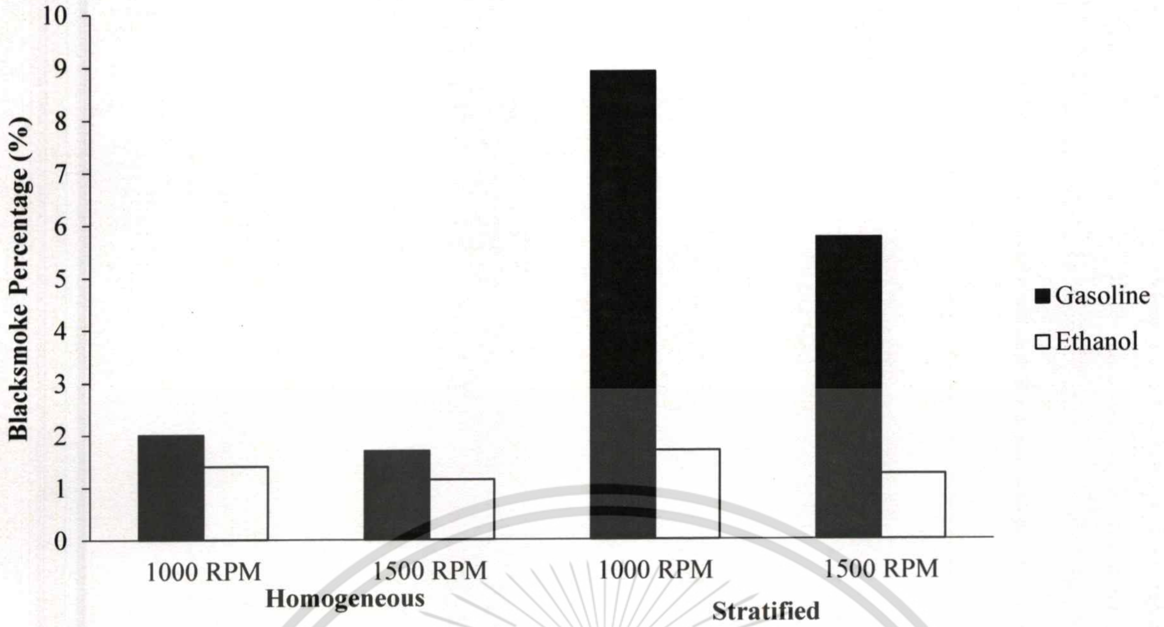


**Fig. 4.9** Quantities of gasoline and ethanol DISI particle emissions under no load condition

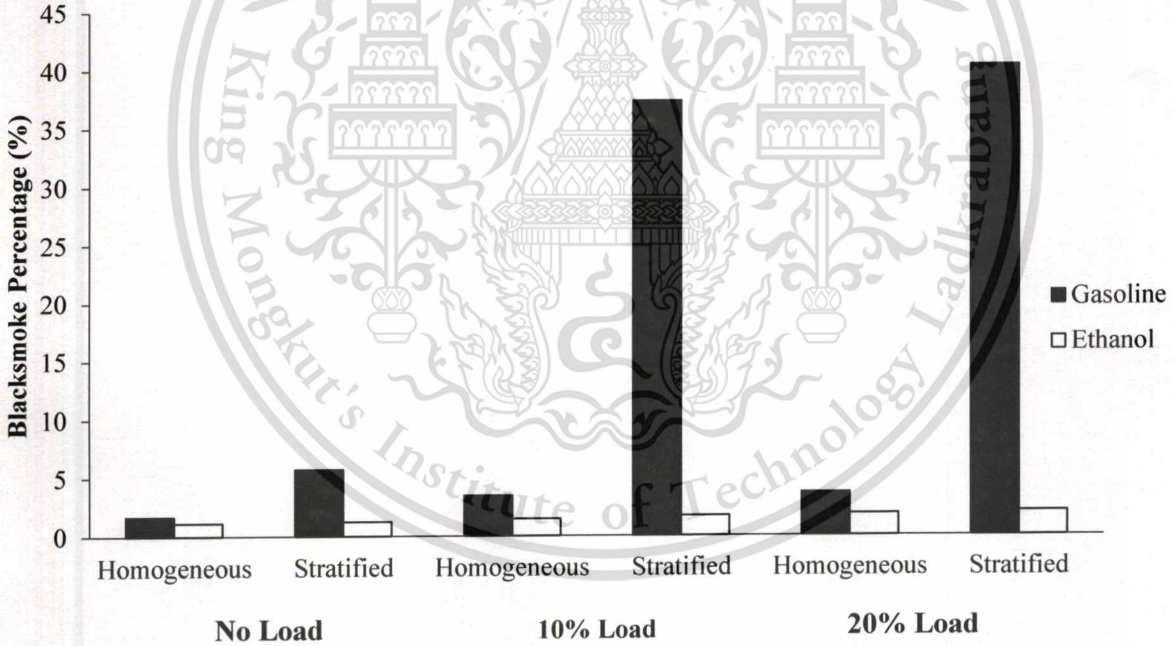


**Fig. 4.10** Quantities of gasoline and ethanol DISI particle emissions under 1500 rpm varying loads condition

Fig. 4.11 shows the black smoke percentage of gasoline, ethanol and ethanol-gasoline blended fuels under 1000 rpm no load condition when varying injection behaviors. The results showed that the percentage of black smoke of gasoline was higher than that of ethanol and ethanol-gasoline blended for all cases. The maximum percentage of black smoke was 1000 rpm, stratified and gasoline condition. Fig. 4.12 shows the black smoke percentage of ethanol-gasoline blended fuels under 1500 rpm varying engine speeds, injection behaviors and loads. The results showed that the percentage of black smoke of gasoline was higher than that of ethanol and ethanol-gasoline blended for all cases. The maximum percentage of black smoke was 1500 rpm, stratified charge mode, 20% load and gasoline.

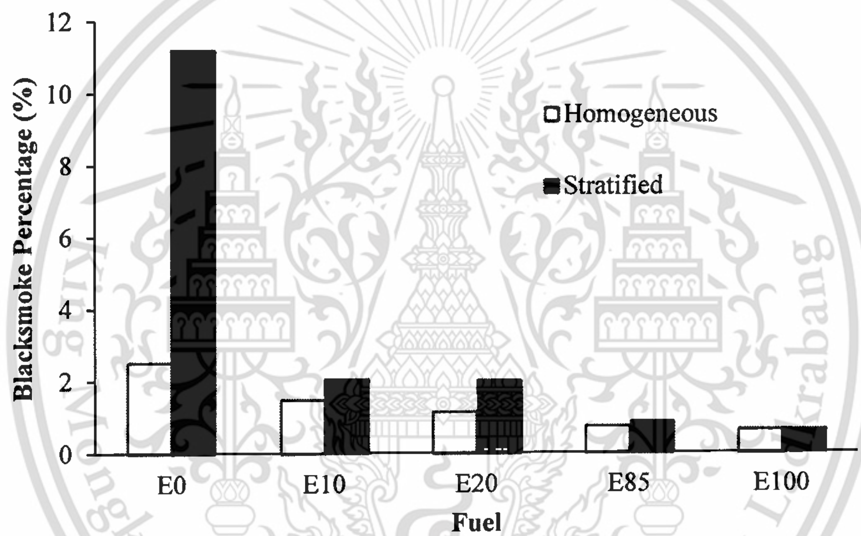


**Fig. 4.11** Quantities of gasoline and ethanol DISI particle emissions under 1000 rpm no load condition

Fig. 4.13 shows the black smoke percentage of ethanol-gasoline blended fuels under 2000 rpm varying engine speeds, injection behaviors and loads. The results showed that the black smoke percentage of gasoline was higher than that of ethanol and ethanol-gasoline blended for all cases. The maximum black smoke percentage was 2000 rpm, stratified charge mode, 20% load and gasoline.

Hence, gasoline particle emissions formation was higher than ethanol and ethanol-gasoline blended. This could be explained that more particle emissions were remained in gasoline combustion than those of ethanol and ethanol-gasoline blended. Since ethanol is an oxygenated fuel that contains oxygen molecules, ethanol is readily oxidized with the available oxygen in the flame zone.

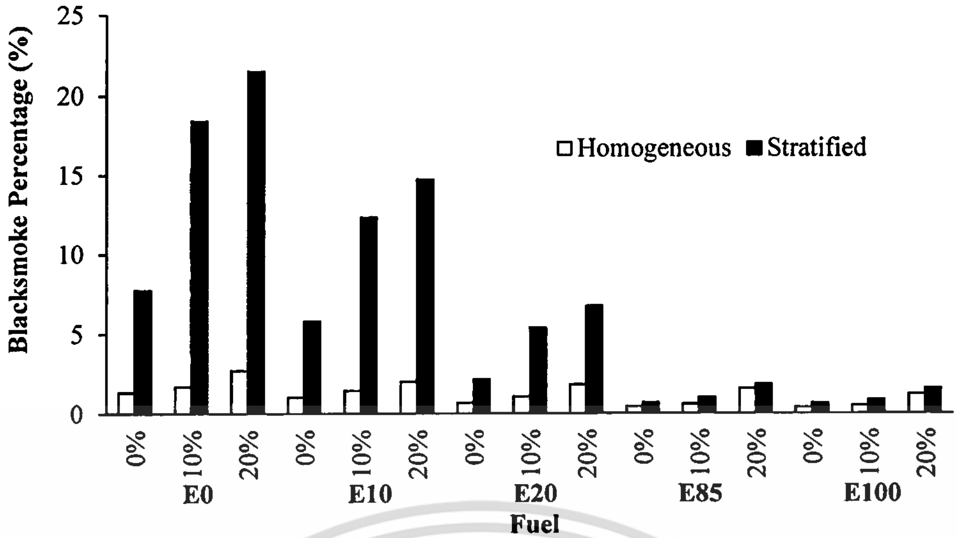


**Fig. 4.12** Quantities of gasoline and ethanol DISI particle emissions under 1500 rpm varying loads condition

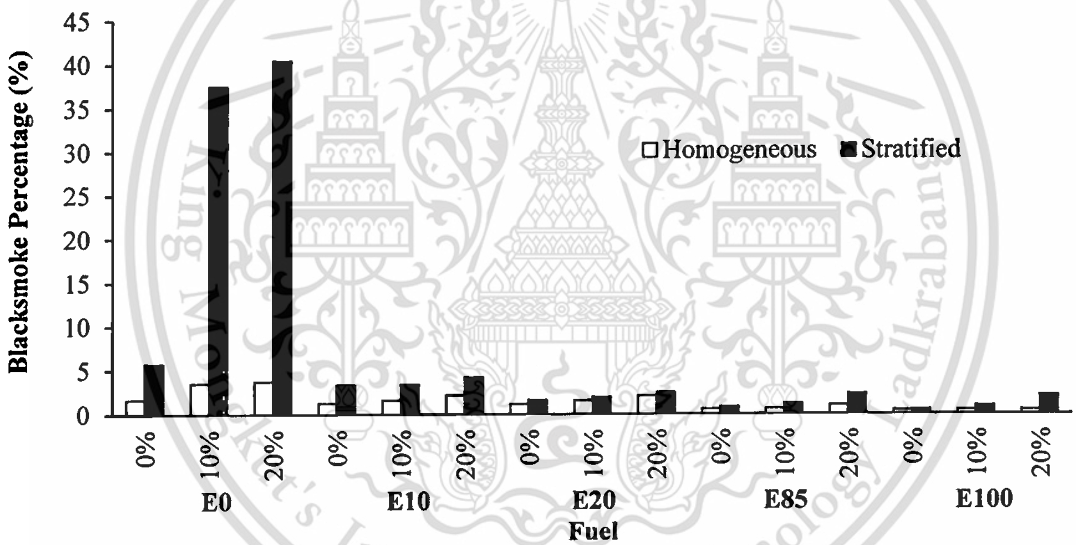


**Fig. 4.13** Quantities of gasoline and ethanol DISI particle emissions under 2000 rpm varying loads condition

## 4.3 Exhaust Emissions

### 4.3.1 CO Emission

CO or Carbon Monoxide is caused by insufficient oxygen distribution and the concentrations of CO are highly dependent on the air- fuel ratio. The comparison of CO emissions at 1000 rpm under different fuels is shown in Fig. 4.14. Compared to gasoline at stratified, the decreases in CO emissions for E10, E20, E85 and E100 are 28.35 %, 80.31 %, 81.10 % and 84.25 %, respectively. At homogeneous, the decreases in CO emissions for E10, E20, E85 and E100 are 25 %, 75 %, 78.41 % and 86.36 %, respectively. The comparison of CO emissions at 1500 rpm under different fuels, loads and speeds are shown in Fig. 4.15. Compared to gasoline at stratified, on average load, the decreases in CO emissions for E10, E20, E85 and E100 are 41.78 %, 76.89 %, 80.22 % and 87.33 %, respectively. At homogeneous, on average load, the decreases in CO emissions for E10, E20, E85 and E100 are 40.61 %, 76.79 %, 81.21 % and 88.95 %, respectively. The comparison of CO emissions at 2000 rpm under different fuels, loads and speeds are shown in Fig. 4.16. Compared to gasoline at stratified, on average load, the decreases in CO emissions for E10, E20, E85 and E100 are 38.89 %, 61.78 %, 69.56 % and 86.89 %, respectively. At homogeneous, on average load, the decreases in CO emissions for E10, E20, E85 and E100 are 45.03 %, 70.44 %, 80.94 % and 84.25 %, respectively. Fig. 4.17 shows the reduction of CO specific emission by increasing ethanol percentage in blended fuel. Which can be explained by the fact that ethanol is oxygenated fuel that contain oxygen atom in ethanol molecule. The oxygenated fuel can provide more oxygen for the combustion process in the rich mixture. That is the reason of CO emissions reduction.

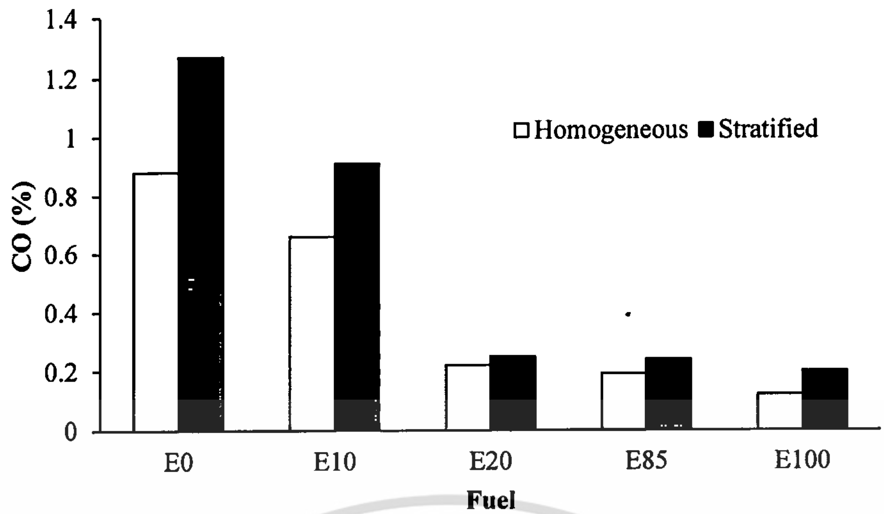


**Fig. 4.14** CO emission at 1000 rpm

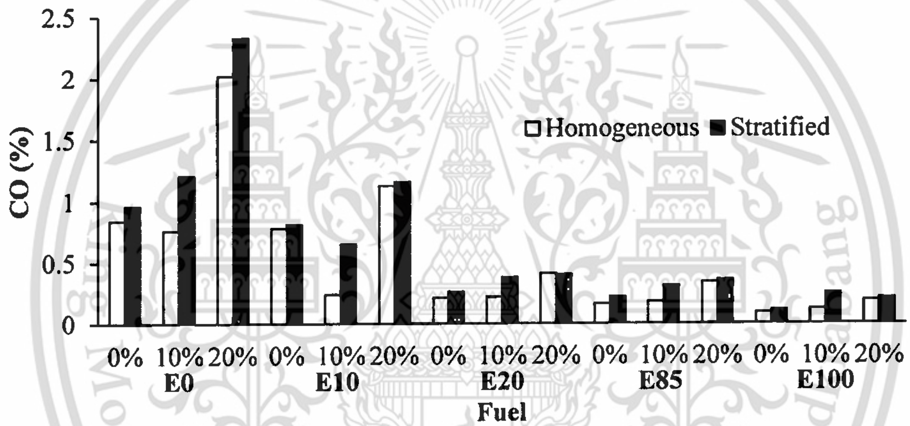


**Fig. 4.15** CO emission at 1500 rpm

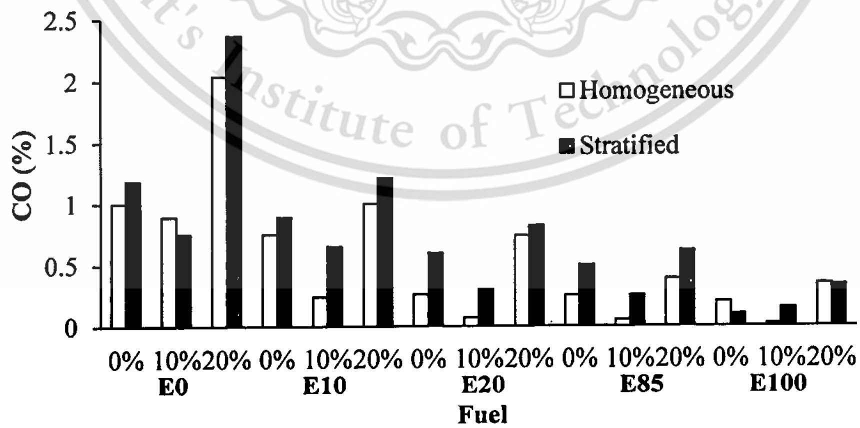


**Fig. 4.16** CO emission at 2000 rpm

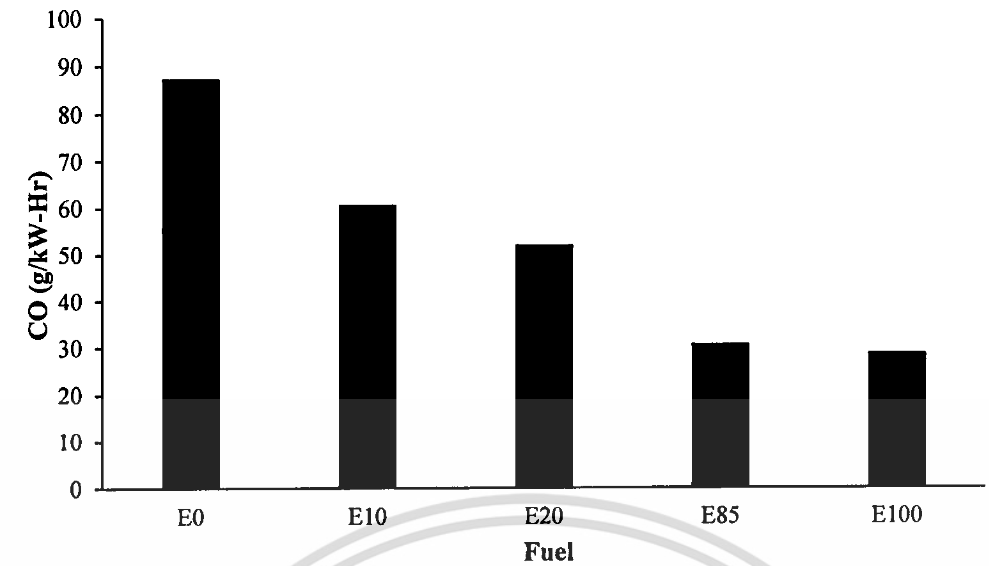


**Fig. 4.17** CO specific emission at 2000 rpm under homogeneous injection

### 4.3.2 $CO_2$ Emission

$CO_2$ or Carbon Dioxide is caused by the complete combustion of hydrocarbon, fuel. The combustion under stoichiometric air-fuel ratio should produce only $CO_2$ and $H_20$ (water). In this study, the effect of ethanol on the exhaust emissions could not be seen clearly. Especially, the effect of different loads and speeds showed unstable trend on $CO_2$ emissions. The comparison of $CO_2$ emissions at 1000 rpm under different fuels is shown in Fig. 4.18. Compared to gasoline at stratified mode, the decreases in $CO_2$ emissions for E10, E20 and E85 are 20.8 %, 16.8 % and 3.2 %, respectively. At homogeneous, the decreases in $CO_2$ emissions for E10, E20, E85 and E100 are 2.46 %, 18.03, 13.93 % and 27.05 %, respectively. The comparison of $CO_2$ emissions at 1500 rpm under different fuels, loads and speeds are shown in Fig. 4.19. Compared to gasoline at stratified, on average load, the decreases in $CO_2$ emissions for E85 and E100 are 2.77 % and 9.32 %, respectively, while the increase in $CO_2$ emissions for E10 is 3.02 %. At homogeneous, on average load, the decreases in $CO_2$ emissions for E10, E20, E85 and E100 are 3.67 %, 3.42 %, 13.69 % and 17.11 %, respectively. The comparison of $CO_2$ emissions at 2000 rpm under different fuels, loads and speeds are shown in Fig. 4.20. Compared to gasoline at stratified, on average load, the decreases in $CO_2$ emissions for E85 and E100 are 6.03 % and 7.53 %, respectively, while the increase in $CO_2$ emissions for E10 and E20 are 1.25 % and 1.51 %, respectively. At homogeneous, on average load, the decreases in $CO_2$ emissions for E10, E20, E85 and E100 are 3.92 %, 0.73 %, 11.27 % and 11.52 %, respectively. $CO_2$ emissions decrease with the increase of ethanol blended percentage. This could be explained that lower carbon content of ethanol and ethanol-blended, compared to gasoline. On the other hand, Fig. 4.21 shows the increment of $CO_2$ specific emission by increasing ethanol percentage in blended fuel. As mention in chapter 2, more complete combustion promotes more $CO_2$ emission in the exhaust gas. Ethanol fuel emitted higher amount of $CO_2$ specific emission than gasoline. These results can be explained as the dominant properties of ethanol as oxygenated fuel promotes complete combustion.

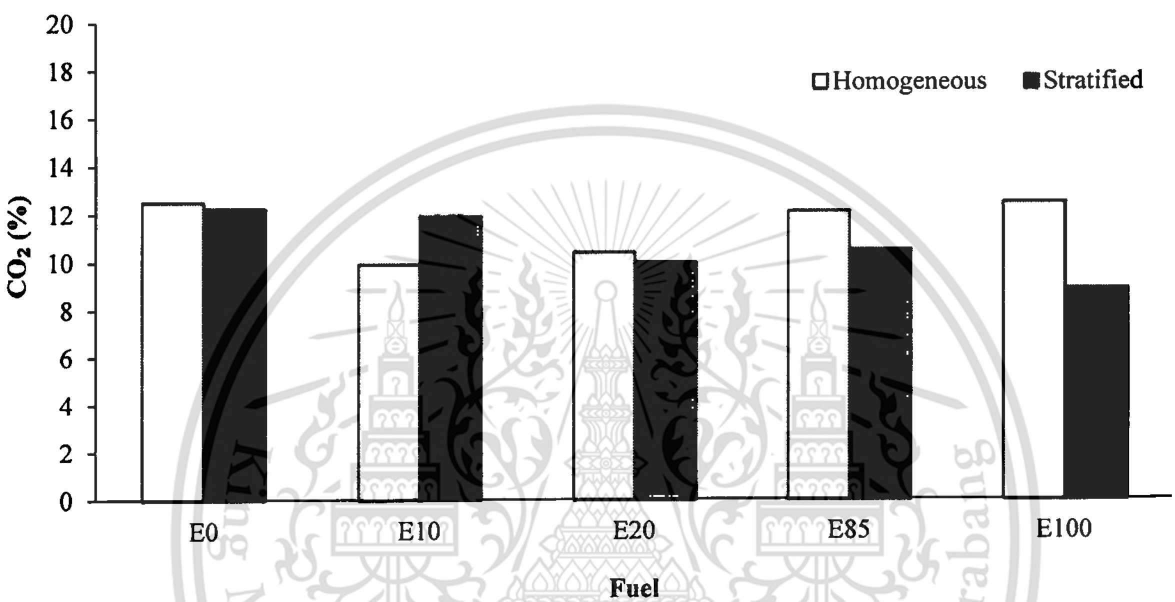


**Fig. 4.18** $CO_2$ emission at 1000 rpm

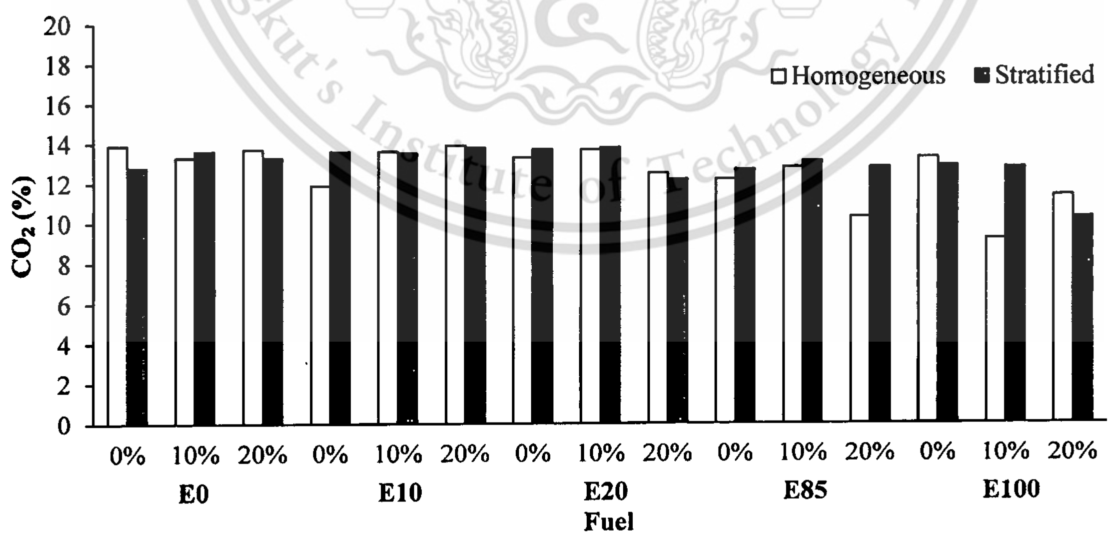


**Fig. 4.19** $CO_2$ emission at 1500 rpm

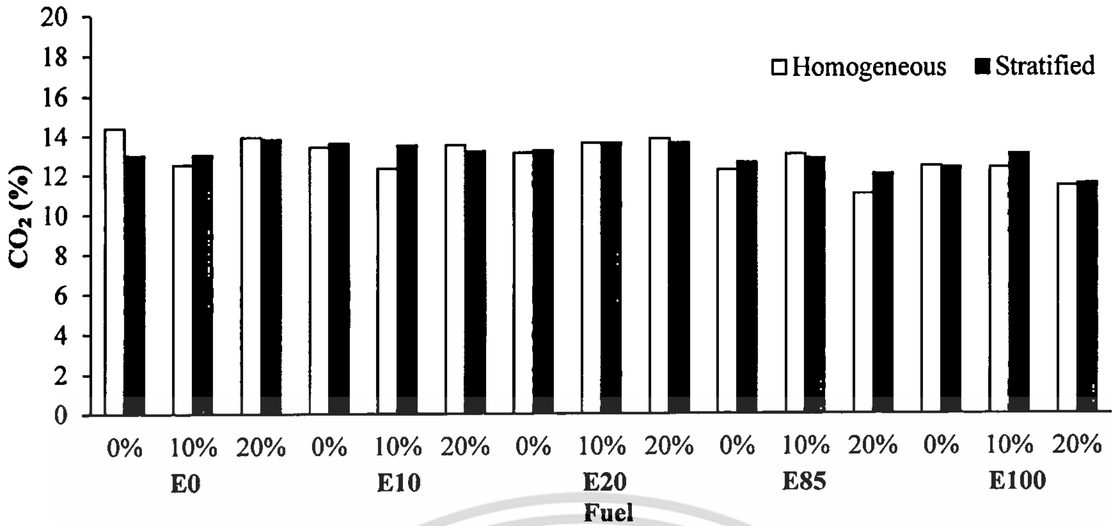


**Fig. 4.20** $CO_2$ emission at 2000 rpm

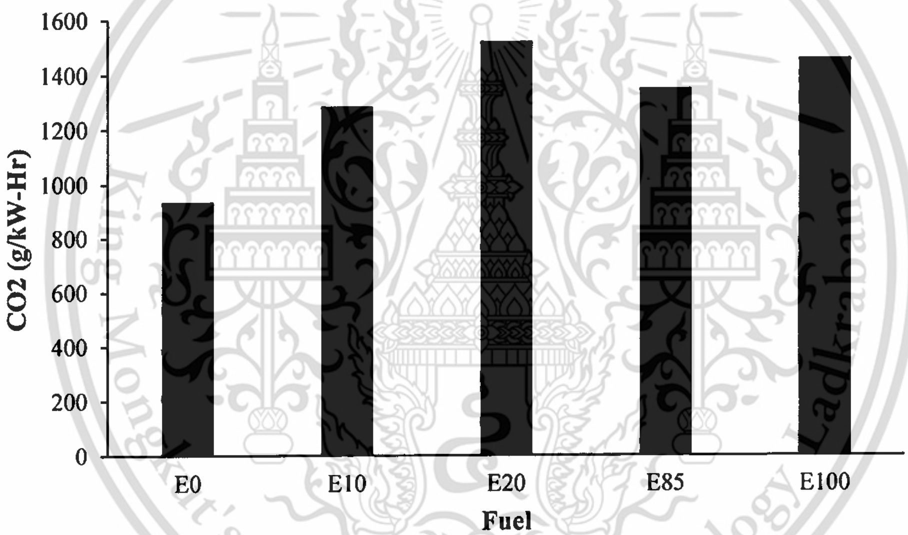


**Fig. 4.21** $CO_2$ specific emission at 2000 rpm under homogeneous injection

### 4.3.3 HC Emission

HC or Hydrocarbon is caused by incomplete combustion of the fuel and the concentrations of HC are highly dependent on the air- fuel ratio. The comparison of HC emissions at 1000 rpm under different fuels is shown in Fig. 4.22. Compared to gasoline at stratified, the decreases in HC emissions for E10, E20, E85 and E100 are 29.67 %, 55.98 %, 61.24 % and 67.94 %, respectively. At homogeneous, the decreases in HC emissions for E10, E20, E85 and E100 are 6.85 %, 41.10%, 49.32 % and 62.33 %, respectively. The comparison of HC emissions at 1500 rpm under different fuels, loads and speeds are shown in Fig. 4.23. Compared to gasoline at stratified, on average load, the decreases in HC emissions for E10, E20, E85 and E100 are 46.94 %, 57.82 %, 58.50 % and 75.51 %, respectively. At homogeneous, on average load, the decreases in HC emissions for E10, E20, E85 and E100 are 5.71 %, 18.57 %, 45.71 % and 50.00 %, respectively. The comparison of HC emissions at 2000 rpm under different fuels, loads and speeds are shown in Fig. 4.24. Compared to gasoline at stratified, on average load, the decreases in HC emissions for E10, E20, E85 and E100 are -25.00 %, 25.00 %, 21.43 % and 41.07 %, respectively. At homogeneous, on average load, the decreases in HC emissions for E10, E20, E85 and E100 are 4.76 %, 7.14 %, 35.71 % and 38.10 %, respectively. The comparison of HC specific emission is shown in Fig. 4.25. Increasing ethanol concentration in blended fuel reduced HC specific emission. Which can be attributed to the higher oxygen content of ethanol and ethanol-blended fuel. Ethanol contains higher oxygen and lower carbon and hydrogen than gasoline which promotes an improved and complete combustion process. Thus HC emission is reduced in the case of using an ethanol.

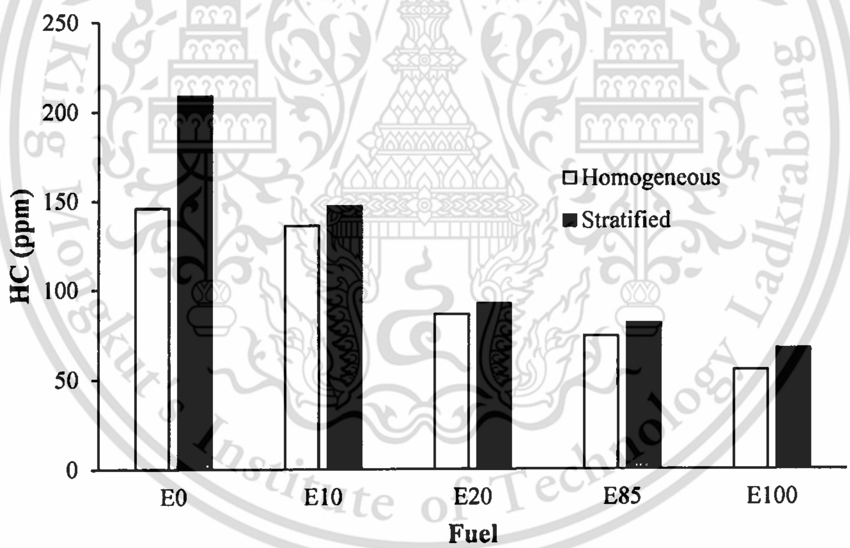


**Fig. 4.22** HC emission at 1000 rpm

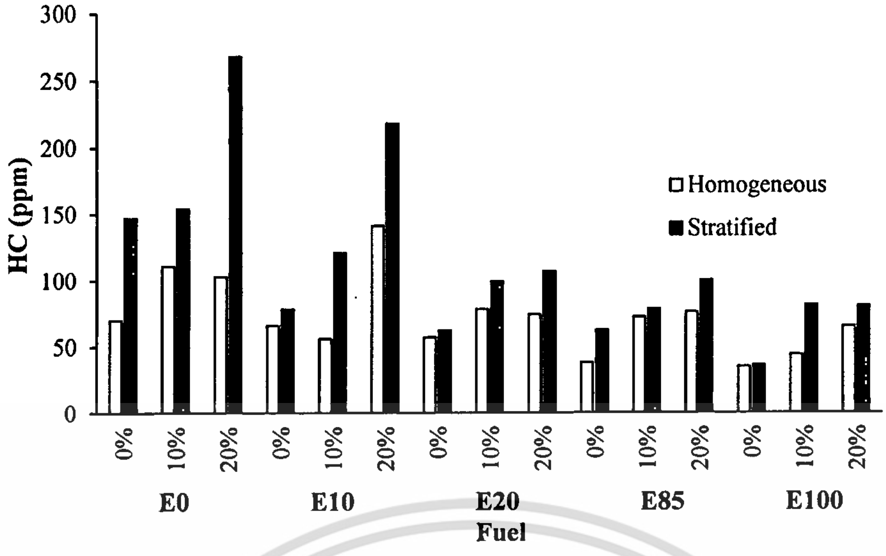


**Fig. 4.23** HC emission at 1500 rpm

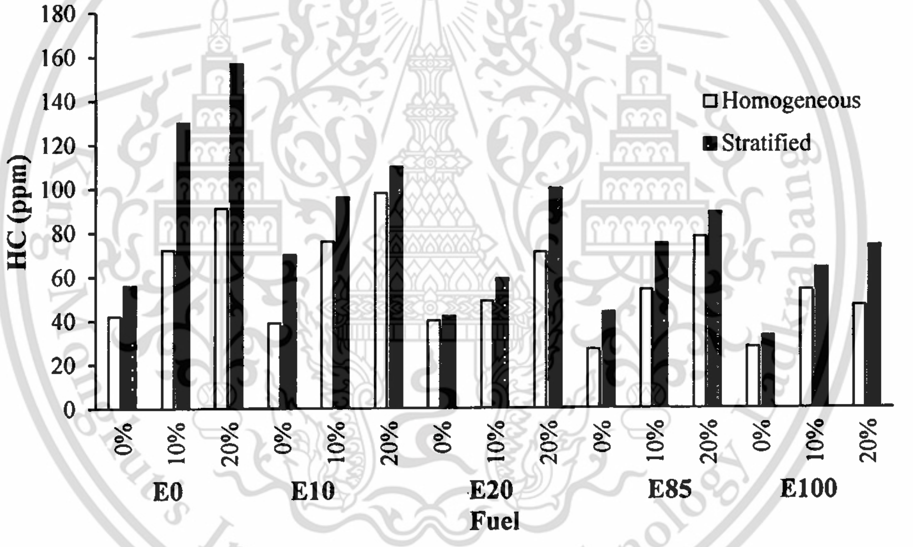


**Fig. 4.24** HC emission at 2000 rpm

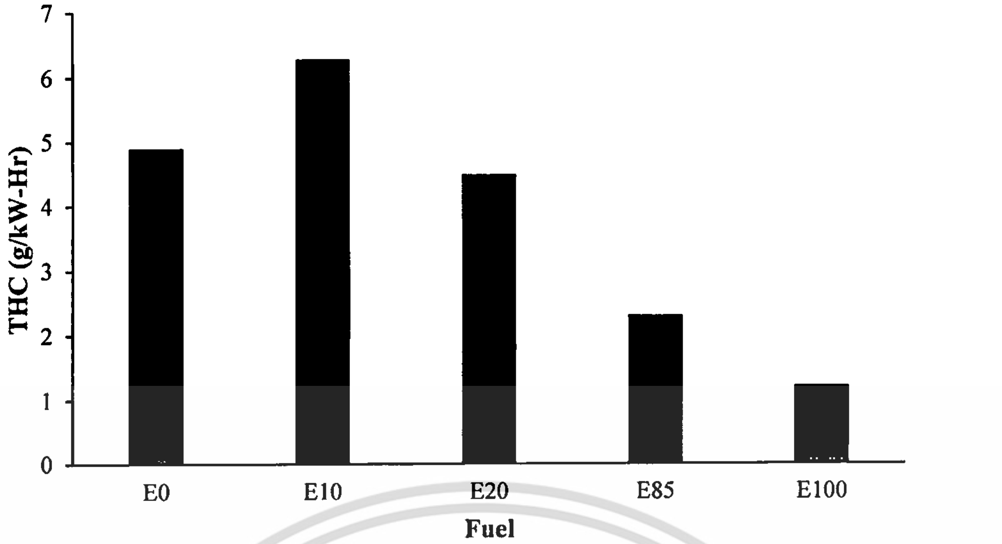


**Fig. 4.25** HC specific emission at 2000 rpm under homogeneous injection

## 4.4 Effect of Ignition Advance on Maximum Brake Torque

For internal combustion engine, maximum brake torque (MBT) is the important value to be realized in order to obtain an optimum ignition timing. An earlier or later ignition timing from this optimum will result in a lower output torque. It is desirable to operate an engine at maximum brake torque to keep the fuel consumption down and to maximize the efficiency of the engine. In this experiment, maximum brake torque were found at the different crank angle with the varying of fuels as 30°, 30°, 31° and 33°bTDC for E0, E10, E20 and E85 respectively. Fig. 4.26 shows the results of ignition advance on maximum brake torque at 20% load. Maximum brake torque of E0, E10, E20 and E85 are 25.50, 27.50, 27.87 and 28.27 Nm, respectively. The using of ethanol blended increases maximum brake torque as 7.84%, 9.29% and 10.86%, compared to those of gasoline. At 30% load as shown in Fig. 4.27, Maximum brake torque of E0, E10, E20 and E85 are 37.80, 38.27, 38.47 and 41.03 Nm, respectively. The using of ethanol blended increases maximum brake torque as 1.24%, 1.77% and 8.55%, compared to gasoline. At 40% load as shown in Fig. 4.28, Maximum brake torque of E0, E10, E20 and E85 are 49.40, 57.10, 57.50 and 60.37 Nm, respectively. The using of ethanol blended increases maximum brake torque as 15.59%, 16.40% and 22.21%, compared to gasoline. The increment of maximum brake torque corresponding with blended ethanol in gasoline can be explain by the ethanol properties. The higher octane number and higher latent heat of vaporization of ethanol bring together with the longer ignition delay. The advancement of ignition timing allows ethanol to combust at the optimum ignition timing for itself there leads to the high peak pressure and high combustion temperature. Hence, thermal efficiency is increased. Fig. 4.29 shows the maximum brake torque trends at optimum ignition timing of each load conditions.

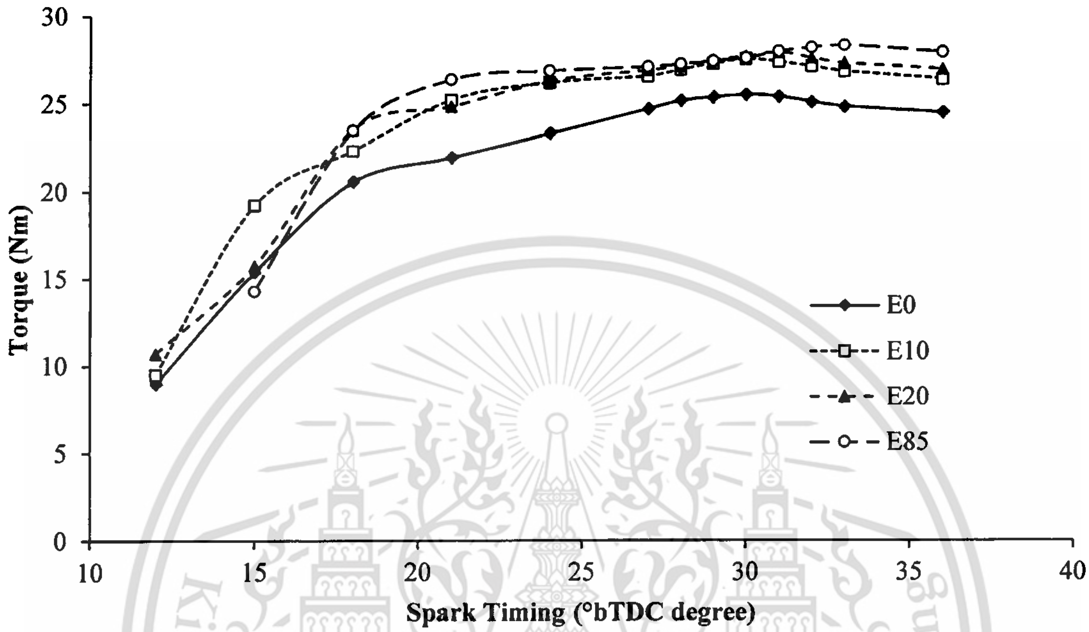


**Fig. 4.26** Torque from DISI engine at 20% load

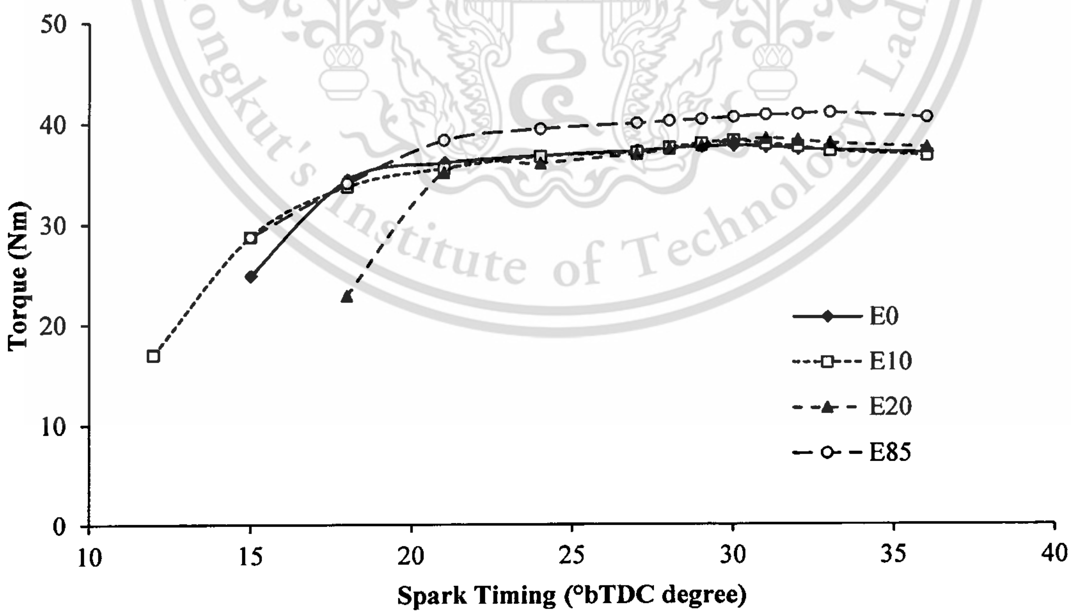


**Fig. 4.27** Torque from DISI engine at 30% load

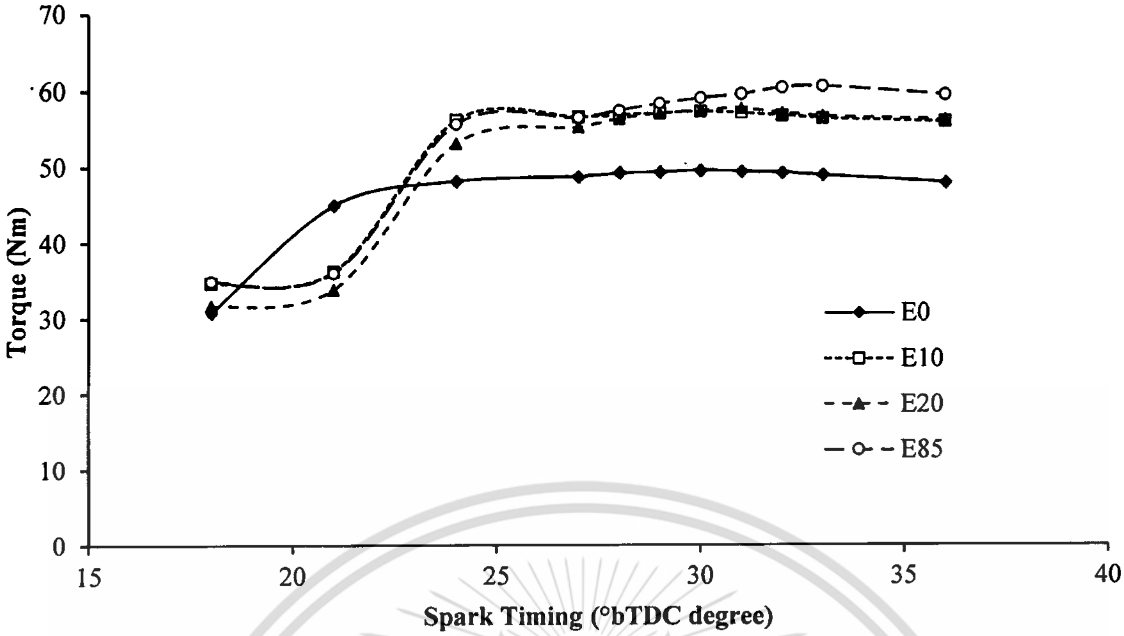


**Fig. 4.28** Torque from DISI engine at 40% load

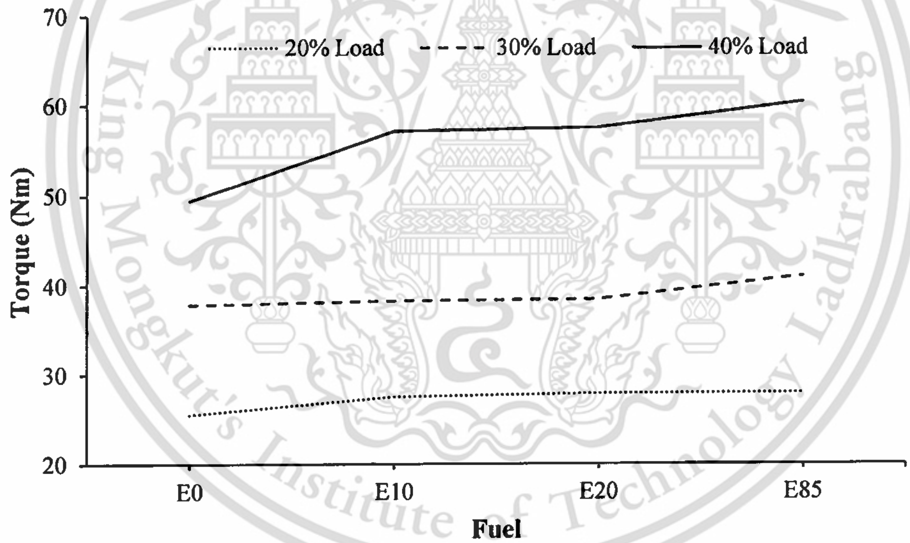


**Fig. 4.29** Maximum brake torque trends at optimum ignition timing

## 4.5 Fuel Consumption

### 4.5.1 Brake Specific Fuel Consumption

From the experimental results, the comparison of Brake Specific Fuel Consumption (BSFC) for test fuels is shown in Fig. 4.30. At the engine speed of 2000 rpm 20% loads, the use of the standalone ECU with the optimum ignition timing decreased BSFC for E10, E20 and E85 by 8.52%, 15.98%, 30.23% and 31.86%, respectively, compared to those of OEM ECU. Considering at the optimum ignition timing, as shown in Fig. 4.31, gasoline shows the lowest BSFC in all loads compared to those of ethanol blended. At 20% load, BSFC of E10, E20 and E85 increased by 2.05%, 2.55% and 20.56%, respectively. At 30% load, BSFC of E10, E20 and E85 increased by 5.38%, 5.43% and 25.67%, respectively. At 40% load, BSFC of E10, E20 and E85 increased by 3.01%, 4.17% and 23.36%, respectively. This caused by the lower energy content of the ethanol, the heating value of ethanol is lower than that of gasoline: 30%. This means that the engine requires a higher amount of fuel to produce the same output power in a gasoline fueled engine. Thus the use of ethanol and ethanol-gasoline blended resulted a slightly increasing in the fuel consumption compared to the use of gasoline.

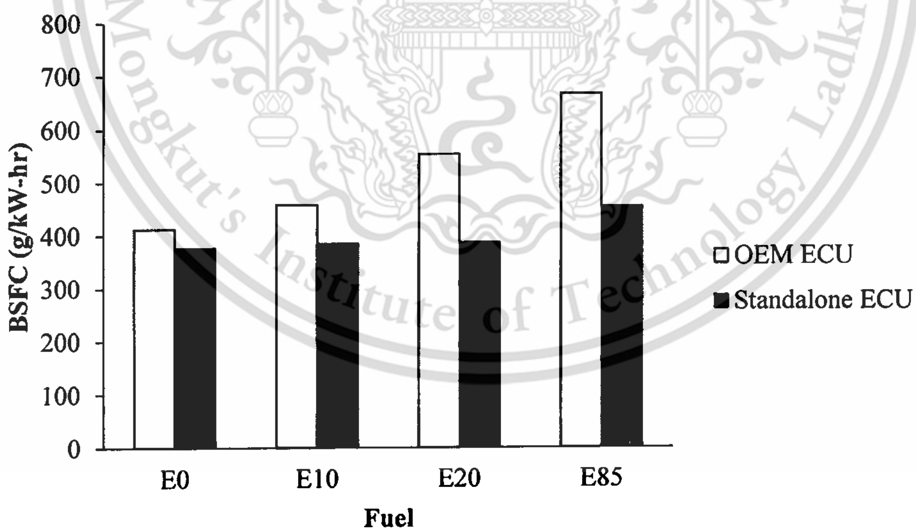


**Fig. 4.30** Comparison of gasoline and ethanol DISI BSFC at 2000 rpm 20% load condition

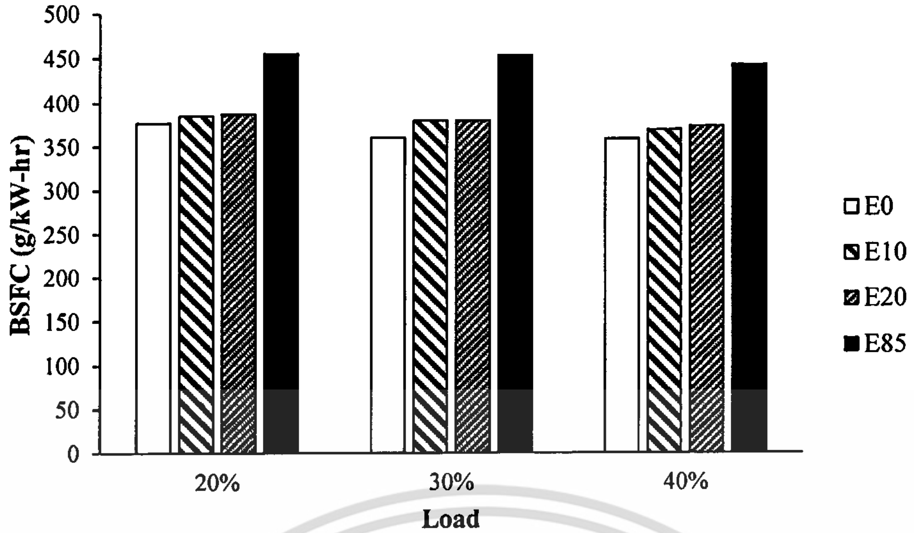


**Fig. 4.31** Comparison of gasoline and ethanol DISI BSFC at 2000 rpm

### 4.5.2 Brake Specific Energy Consumption

The comparison of Brake Specific Energy Consumption (BSEC) for test fuels is shown in Fig. 4.32. At the engine speed of 2000 rpm 20% loads, BSEC of standalone ECU compared with OEM ECU for showed the same results with those of BSFC results due to the calculation of BSEC is a conversion of the quantity of fuel consumed per a unit power in a unit of time with Lower Heating Value (LHV) of each fuel. Fig. 4.33 shows the comparison of BSEC with varying loads. E85 fuel with the oxygen inside its molecule shows the lowest BSEC in all loads compared to those of gasoline. At 20% load, BSEC of E10, E20 and E85 decreased by 4.98%, 5.38% and 19.17%, respectively. At 30% load, BSEC of E10, E20 and E85 decreased by 1.88%, 2.72% and 15.74%, respectively. At 40% load, BSEC of E10, E20 and E85 decreased by 4.08%, 3.88% and 17.29%, respectively. Although the more quantity of ethanol fuel is required, the oxygenated fuel plays an important role to increase combustion efficiency of an internal combustion engines. Blending in higher percentage of ethanol in fuel can be enhanced combustion characteristics by accelerate flame development process and increasing the value of heat release rate. The combustion process occur more advance than that of gasoline. These may cause by the oxygen content within fuel itself can enhance oxidation rate and accelerate the initial stage of combustion regardless the octane rating [24,39-40].

The advantages of ethanol: such as oxygen containing, cooling effect from higher heat of vaporization and high octane, lead to the decrease of BSEC as well.

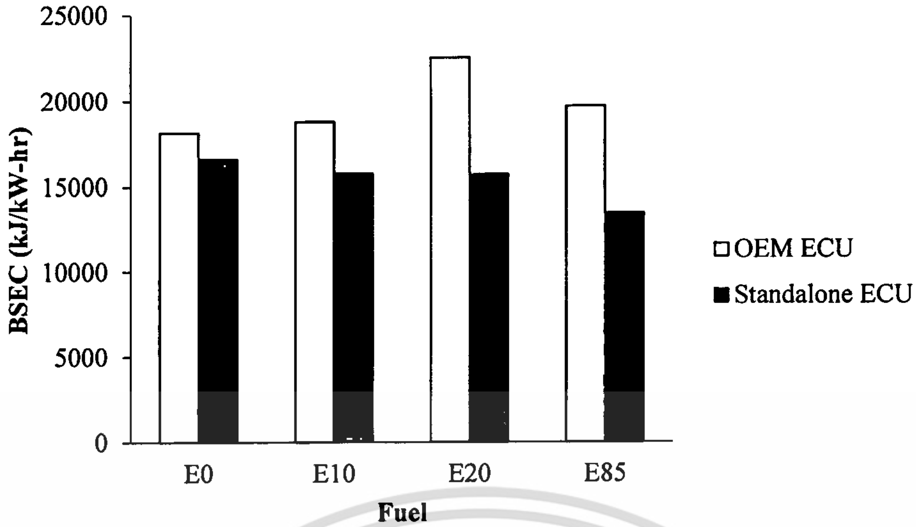


**Fig. 4.32** Comparison of gasoline and ethanol DISI BSEC at 2000 rpm 20% load condition

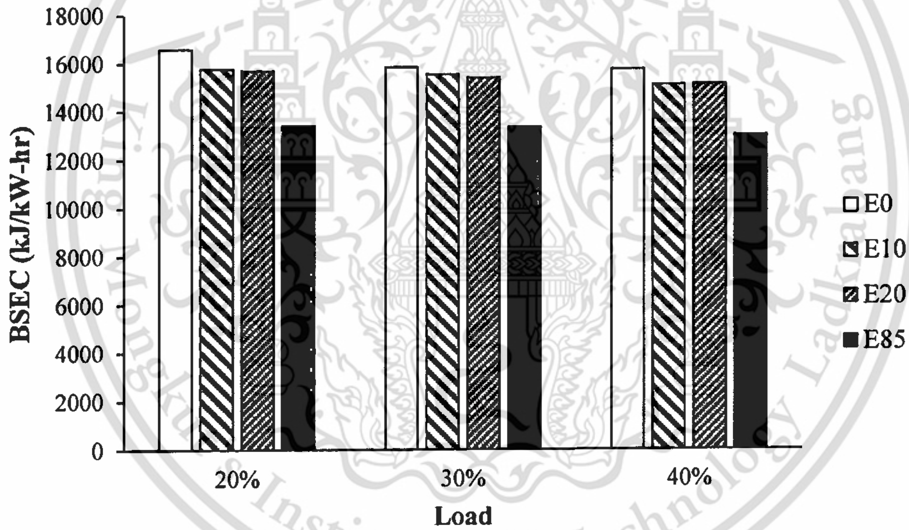


**Fig. 4.33** Comparison of gasoline and ethanol DISI BSEC at 2000 rpm

# Chapter 5
# Conclusions

## 5.1 Conclusions

The investigation of DISI engine fuelled gasoline, ethanol and ethanol-gasoline blended in this study shows the particle emissions concentration of gasoline and ethanol under varying engine speeds, loads and injection behaviors. Particle emissions were analyzed by smoke meter. On the other hand, exhaust emissions such as CO $CO_2$ and HC were measured by a gas analyzer. In addition, performance such as Brake Specific Fuel Consumption (BSFC) and Brake Specific Energy Consumption (BSEC) were investigated. The main conclusions can be summarized as follows:

1 DISI engine can be the cause of particle emissions due to that operating principle as diesel engine, and also cause of global gas emitting and environmental concern. Hence, the investigation of particle emissions of gasoline DISI engine is as important as diesel engine.

2 The remaining particle emissions of gasoline is higher than that of ethanol due to ethanol is the oxygenated fuel so oxygen molecules in ethanol can improve more complete combustion of the engine.

3 The remaining particle emissions of 20% loads operating is higher than that of either 10% or no load due to much amount of fuel injected, rich fuel, causing particle emissions.

4 The remaining particle emissions under stratified charge operating mode is higher than homogeneous charge operating mode since the late injection made less fuel propagation than homogeneous charge operating mode. The rich fuel region around spark plug is the cause of particle emissions under stratified charge operating mode.

5 The CO and HC emissions of ethanol are lower than gasoline due to the oxygen atom in ethanol molecule promotes complete combustion and the $CO_2$ emission of ethanol is lower than gasoline due to lower carbon content of ethanol, compared to gasoline. In addition, lower heating value of ethanol leads to higher fuel consumption, compared to gasoline.

6 Since oxygenated fuel has a strong effect on particle emissions of combustion, using ethanol instead of gasoline may reduce emissions and pollutions from internal combustion engine.

7 Although the use of smoke meter in order to measure particle emissions concentration is an indirect measurement and that can cause some error in results, the method shows significant in the results which lead to good conclusions as well as mass measurement.

8 The use of DISI engine fuelled with ethanol fuel promotes not only the higher thermal efficiency but also volumetric efficiency from its dominant properties, high octane number, high latent heat of vaporization and oxygenated fuel.

9 The advancement of ignition timing for ethanol fuel is required because of longer ignition delay property. Maximum brake torque is increased at the optimum ignition timing.

10 Brake specific fuel consumption of ethanol fuel is lowest among gasoline and gasoline-ethanol blended fuels due to its lower LHV. On the other hand, brake specific energy consumption of ethanol is lowest among gasoline and gasoline-ethanol blended fuels which means ethanol plays an important role to increase combustion efficiency of an internal combustion engines.

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# APPENDICES

# Appendix A: Fuel Distillation

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QUALITY ANALYSIS DEPARTMENT, SUPPLY AND TERMINAL OPERATIONS, OIL BUSINESS
555 ARDNARONG RD. KLONGTOEY, BANGKOK 10260, THAILAND
TEL +66(0)2239-7148 FAX +66(0)2239-7149 WWW.PTTPLC.COM



## Certificate of Analysis

## Product : Gasoline E 0

**Certificate No.** : T-12/29294
**Sample Lab No.** : OP-GSH-1230331
**Customer/Supplier** : International College, King Mongkut's Institute of Technology
International College, King Mongkut's Institute of Technology
Ladkrabang,
Chalongkrung Rd. Ladkrabang BKK 10520
**Sample Location** : LKB
**Batch No.** : -
**Product Source** : -

**Received Date** : 18 Dec 2012
**Date of Test** : 18 Dec 2012
**Date of Sampling** : 18 Dec 2012

**Sample Condition** : Normal

| Test Item | Test Method | Limit | Result |
|---|---|---|---|
| 1. Distillation :Initial Boiling Point,°C | ASTM D 86-11b | - | 35.0 |
| 2. Distillation : 10% vol. Evaporated,°C | ASTM D 86-11b | - | 51.5 |
| 3. Distillation : 50% vol. Evaporated,°C | ASTM D 86-11b | - | 78.2 |
| 4. Distillation : 90% vol. Evaporated,°C | ASTM D 86-11b | - | 154.0 |
| 5. Distillation End Point,°C | ASTM D 86-11b | - | 197.3 |
| 6. Distillation Recovery,% vol. | ASTM D 86-11b | - | 97.9 |
| 7. Distillation Residue,% vol. | ASTM D 86-11b | - | 1.1 |

Approved by : Phurita
( Phurita Pothisuk )
Position Title : Vice President in Quality Analysis Department
Date of Issue : 25 Dec 2012

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# Certificate of Analysis

## Product : E 10

**Certificate No.** : T-12/29295
**Sample Lab No.** : OP-GSH-1230332
**Customer/Supplier** : International College, King Mongkut's Institute of Technology
International College, King Mongkut's Institute of Technology
Ladkrabang,
Chalongkrung Rd. Ladkrabang BKK 10520
**Sample Location** : LKB
**Batch No.** : -
**Product Source** : -

**Received Date** : 18 Dec 2012
**Date of Test** : 18 Dec 2012
**Date of Sampling** : 18 Dec 2012

**Sample Condition** : Normal

| Test Item | Test Method | Limit | Result |
|---|---|---|---|
| 1. Distillation :Initial Boiling Point,°C | ASTM D 86-11b | - | 36.5 |
| 2. Distillation : 10% vol. Evaporated,°C | ASTM D 86-11b | - | 51.6 |
| 3. Distillation : 50% vol. Evaporated,°C | ASTM D 86-11b | - | 70.2 |
| 4. Distillation : 90% vol. Evaporated,°C | ASTM D 86-11b | - | 160.2 |
| 5. Distillation End Point,°C | ASTM D 86-11b | - | 187.2 |
| 6. Distillation Recovery,% vol. | ASTM D 86-11b | - | 97.9 |
| 7. Distillation Residue,% vol. | ASTM D 86-11b | - | 1.0 |

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## Certificate of Analysis

### Product : E 20

**Certificate No.** : T-12/29296
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**Customer/Supplier** : International College, King Mongkut's Institute of Technology
International College, King Mongkut's Institute of Technology
Ladkrabang,
Chalongkrung Rd. Ladkrabang BKK 10520
**Sample Location** : LKB
**Batch No.** : -
**Product Source** : -

**Received Date** : 18 Dec 2012
**Date of Test** : 18 Dec 2012
**Date of Sampling** : 18 Dec 2012

**Sample Condition** : Normal

| Test Item | Test Method | Limit | Result |
|---|---|---|---|
| 1. Distillation :Initial Boiling Point,°C | ASTM D 86-11b | - | 37.8 |
| 2. Distillation : 10% vol. Evaporated,°C | ASTM D 86-11b | - | 53.5 |
| 3. Distillation : 50% vol. Evaporated,°C | ASTM D 86-11b | - | 70.8 |
| 4. Distillation : 90% vol. Evaporated,°C | ASTM D 86-11b | - | 155.0 |
| 5. Distillation End Point,°C | ASTM D 86-11b | - | 184.6 |
| 6. Distillation Recovery,% vol. | ASTM D 86-11b | - | 98.1 |
| 7. Distillation Residue,% vol. | ASTM D 86-11b | - | 1.1 |

Approved by : Phurita
( Phurita Pothisuk )
Position Title : Vice President in Quality Analysis Department
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## Certificate of Analysis

### Product : E 85

**Certificate No.** : T-12/29298
**Sample Lab No.** : OP-GSH-1230334
**Customer/Supplier** : International College, King Mongkut's Institute of Technology
International College, King Mongkut's Institute of Technology
Ladkrabang,
Chalongkrung Rd. Ladkrabang BKK 10520
**Sample Location** : LKB
**Batch No.** : -
**Product Source** : -

**Received Date** : 18 Dec 2012
**Date of Test** : 18 Dec 2012
**Date of Sampling** : 18 Dec 2012

**Sample Condition** : Normal

| Test Item | Test Method | Limit | Result |
|---|---|---|---|
| 1. Distillation :Initial Boiling Point,°C | ASTM D 86-11b | - | 41.3 |
| 2. Distillation : 10% vol. Evaporated,°C | ASTM D 86-11b | - | 66.6 |
| 3. Distillation : 50% vol. Evaporated,°C | ASTM D 86-11b | - | 77.5 |
| 4. Distillation : 90% vol. Evaporated,°C | ASTM D 86-11b | - | 77.8 |
| 5. Distillation End Point,°C | ASTM D 86-11b | - | 80.5 |
| 6. Distillation Recovery,% vol. | ASTM D 86-11b | - | 98.7 |
| 7. Distillation Residue,% vol. | ASTM D 86-11b | - | 0.9 |

Approved by : Phurita
( Phurita Pothisuk )
Position Title : Vice President in Quality Analysis Department
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555 ARDNARONG RD., KLONGTOEY, BANGKOK 10260, THAILAND
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## Certificate of Analysis

**Product : E 100**

**Certificate No.** : T-12/29299
**Sample Lab No.** : OP-GSH-1230335
**Customer/Supplier** : International College, King Mongkut's Institute of Technology
International College, King Mongkut's Institute of Technology
Ladkrabang,
Chalongkrung Rd. Ladkrabang BKK 10520 ,
**Sample Location** : LKB
**Batch No.** : -
**Product Source** : -

**Received Date** : 18 Dec 2012
**Date of Test** : 18 Dec 2012
**Date of Sampling** : 18 Dec 2012

**Sample Condition** : Normal

| Test Item | Test Method | Limit | Result |
|---|---|---|---|
| 1. Distillation :Initial Boiling Point,°C | ASTM D 86-11b | - | 77.6 |
| 2. Distillation : 10% vol. Evaporated,°C | ASTM D 86-11b | - | 77.8 |
| 3. Distillation : 50% vol. Evaporated,°C | ASTM D 86-11b | - | 77.9 |
| 4. Distillation : 90% vol. Evaporated,°C | ASTM D 86-11b | - | 78.0 |
| 5. Distillation End Point,°C | ASTM D 86-11b | - | 80.0 |
| 6. Distillation Recovery,% vol. | ASTM D 86-11b | - | 99.3 |
| 7. Distillation Residue,% vol. | ASTM D 86-11b | - | 0.7 |

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# Appendix B: Publications

# Investigation of DISI Engine Particle Emissions

**Athiwat Butmarasri[1,*], Chanont Jiwattayakul[2], Nattapong Suttikittiyaporn[2]**
**Preechar Karin[1], Chinda Charoenphonphanich[2]**
**Nuwong Chollacoop[3] and Hidenori Kosaka[4]**
[1] *International College, King Mongkut's Institute of Technology Ladkrabang, Thailand 10520*
[2] *Faculty of Engineering, King Mongkut's Institute of Technology Ladkrabang, Thailand 10520*
[3] *National Metal and Materials Technology Center (MTEC), Thailand 12120*
[4] *Department of Mechanical and Control Engineering, Tokyo Institute of Technology, Japan*
* *Corresponding Author: Tel: 081 450 6599*
E-mail: athiwatb@yahoo.com

## Abstract

*This is a part of an ongoing research in using new development technologies and renewable oxygenated fuels are considered the most suitable solution for sustainable future that focused on the comparison of particle emissions that caused by using gasoline and ethanol DISI (Direct Injection Spark Ignition) engines. Amount of particle emission would be reduced by using ethanol DISI engine. In addition, physical structure of particle emissions emitted from gasoline and diesel diffusion flames were investigated by using a Scanning Electron Microscopy (SEM) and a Transmission Electron Microscopy (TEM) and ethanol's properties which affect engine and particle emissions would also be discussed in the present study as well. The DISI engine, inline 4 cylinders, 4 strokes, 1834 $cm^3$ displacement, was tested on engine dynamometer with different loads and injection behaviors with, homogenous and stratified charge then the particle emissions were sampled by smoke meter to measure the amount of particle emissions. The results showed particle emissions emitted from ethanol are lower than gasoline. The average primary size of gasoline and diesel fuels particle emissions are approximately 25-60 nm and 50-60 nm, respectively. The accumulate sizes of gasoline and diesel fuels particle emissions are approximately 100-360 nm and 100-500 nm, respectively. The findings of this study can serve as guidance for the reduction particle emissions from gasoline DISI engines by using ethanol.*

**Keywords:** *particle emissions, ethanol, DISI engine*

## 1. Introduction

Nowadays, the fuel consumption in transportation field is one of the main reasons to realize amount of emissions. In particularly the limitations of new emission standards, emission levels must be furthermore reduced both for spark ignition and diesel engines. In particular, the total energy consumption in the world depends on the remaining fossil fuels.

It is imperative, then, to find out alternative fuels instead of the using fossil fuels. Ethanol is the most suitable alternative fuel for spark ignition engines because of the advantages of ethanol, e.g. better knock limit range due to higher octane number of ethanol, higher volumetric efficiency due to cooling effect of higher heat of vaporization and also reduced particle emissions due to more complete combustion from oxygen atom inside ethanol molecule.

The development of new clean spark ignition engines, such as direct injection spark ignition (DISI) engines is important because of the advantages of DISI engines, e.g. higher thermal efficiency due to direct fuel injection, higher power output than conventional homogeneous charge port injection spark ignition (PSI) engines and lower fuel consumption due to an ultra-lean combustion in stratified charge operating mode [1].

The use of direct injection spark ignition engines with fuelled ethanol is to reduce regulated pollutant emissions produced by internal combustion engines, as well as to reduce the greenhouse effect impact of transportation. In addition PM, NOx and HC emissions [2] are prejudicial effects on the environment and human health.

In particular, diffusion flame in stratified charge operation of DISI engine is similar to compression ignition (CI) engine that causes particle emissions. Particle emissions consist of a solid fraction (SOL) and a soluble organic fraction (SOF). Primary particles, composed of carbon and metallic ash, are coated with SOF and sulphate. A primary soot particle has two distinct parts: an inner core is located at the central region of the primary particle and another is outer shell. The composition of particle emissions may vary widely depending on the operating conditions and fuel composition [3-7].

The objectives of this research are to characterize and measure amount of gasoline and ethanol DISI particle emissions. The advantage of ethanol would be reported in the view point of particle emissions.

## 2. Experimental Apparatus

### 2.1 Fuel

The use of ethanol in the experiment was considered by the reduction of particle emissions and fossil fuel consumption in automotive. The domestic production can reduce amount of transportation activities of gasoline fuels. Higher octane number than gasoline can perform a better anti-knock for increase compression ratio and performance subsequently. Ethanol has a higher heat of vaporization so it provides higher densities in the intake, thus increasing the volumetric efficiency. Moreover, ethanol is an oxygenated fuel, oxygen molecules in the fuel molecule, therefore excess air can react with CO in residual emissions. However, a lower heating value of ethanol compared with gasoline, it govern ethanol to inject more than gasoline in the same amount of total energy.

### 2.2 Engine

A direct injection spark ignition (DISI) engine, inline 4 cylinders, 4 strokes, 1834 $cm^3$ displacement, was used in the experiment to be the source of emissions. An increasing of fuel efficiency and high power output, by increase compression ratio, are the main advantages of DISI engines. In addition, the cooling effect of the injected fuel and the more evenly dispersed mixtures allow for more aggressive ignition timing curves [8]. Emissions levels can also be more accurately controlled with the DISI system. The cited gains are achieved by the precise control over amount of fuel and injection timings which are varied according to the load conditions.

In addition, there are no throttling losses in some DISI engines, when compared to a conventional fuel injected or carburetor engines, which greatly improve efficiency in engines without a throttle plate [9].

## 3. Research Methodologies

A direct injection spark ignition (DISI) engine, inline 4 cylinders, 4 strokes, 1834 $cm^3$ displacement was measured emissions at 1000 rpm idle and 1500 rpm under 10, 20% loads, using gasoline and ethanol, respectively. The 1000 rpm idle condition was selected to study a critical condition, in terms of stability, for DISI engine, and the 1500 rpm under 10, 20% loads were chosen as representative points for urban driving conditions. The injection behaviors were controlled for both homogenous and stratified charges. All the conditions investigated were carried out at $\lambda$=1. Particle emissions were sampled directly from the exhaust pipe, then measured for concentration of particle emissions with smoke meter. On the other hand, gasoline and diesel diffusion flame particle emissions were generated by gasoline and diesel lamps in order to investigate the primary and accumulate particle emissions size by SEM and TEM method.

## 4. Results and Discussions

### 4.1 Structure of Particle Emissions

Fig.1 shows gasoline and ethanol diffusion flames by Schlieren method image. The core of ethanol diffusion flame was lighter than that of gasoline, where particle emissions are formed at the center of diffusion flame. Similarity, Fig.2 shows gasoline and ethanol diffusion flames by optical image. The length of ethanol diffusion flame was shorter than that of gasoline due to ethanol contains less carbon content than gasoline and ethanol also contains oxygen atoms inside an oxygenated fuel molecule that promote more complete combustion.

Gasoline and diesel particle emissions images were taken in SEM and TEM method in order to verify primary and accumulate size and formation of particle emissions. Gasoline and diesel particle emissions were generated by fuel lamp. Fig.3 and Fig.4 shows SEM image of gasoline and diesel accumulate particle emissions, respectively. Accumulate sizes of gasoline were slightly smaller than that of diesel, the size of both particle emissions were approximately 100-360 nm and 100-500 nm, respectively. The particle size distribution in accumulate mode approximately 100 particles was shown in Fig.5. Most of gasoline lamp particle size is around 151-200 nm while the diesel lamp particle size is around 251-300 nm, slightly bigger than that of gasoline lamp particle.

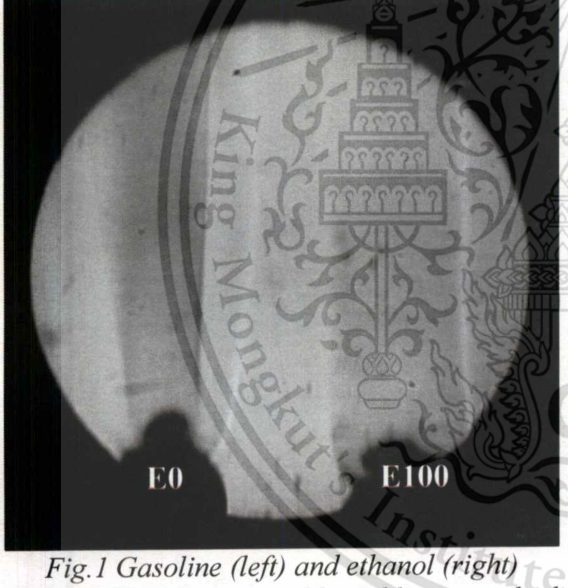

*Fig.1 Gasoline (left) and ethanol (right) diffusion flames captured by Schlieren method*

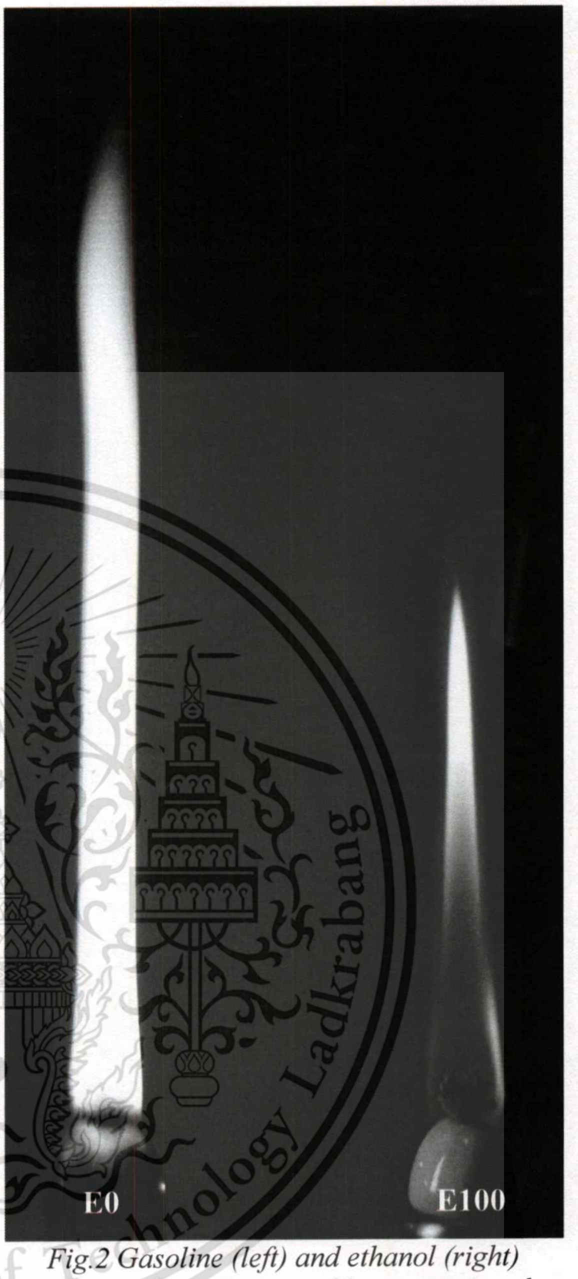

*Fig.2 Gasoline (left) and ethanol (right) diffusion flames captured by conventional digital camera*

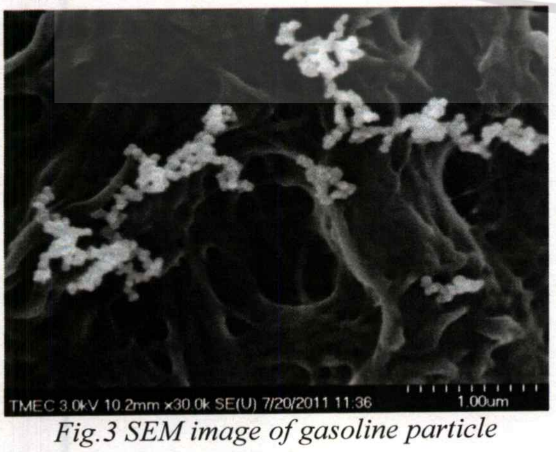

*Fig.3 SEM image of gasoline particle emissions*

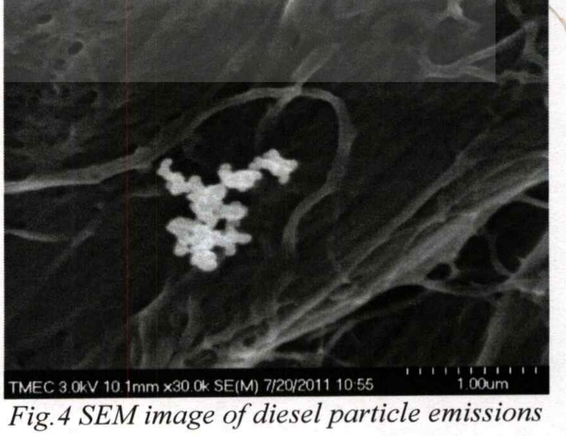

*Fig.4 SEM image of diesel particle emissions*

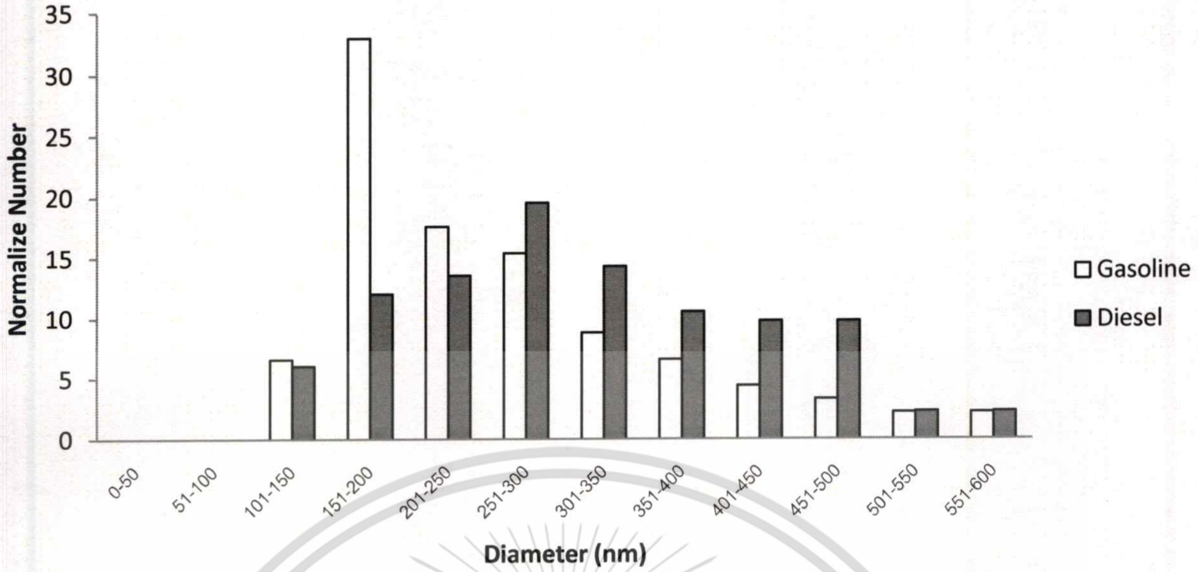


*Fig.5 Accumulated size distribution of gasoline and diesel lamp by SEM images*

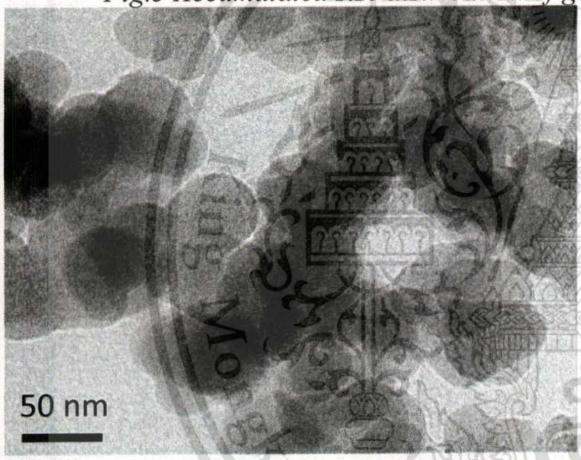


*Fig.6 TEM image of gasoline particle emissions*

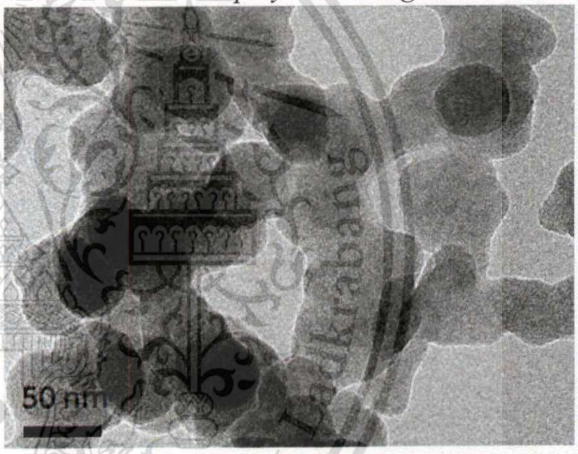


*Fig.7 TEM image of diesel particle emissions [10]*

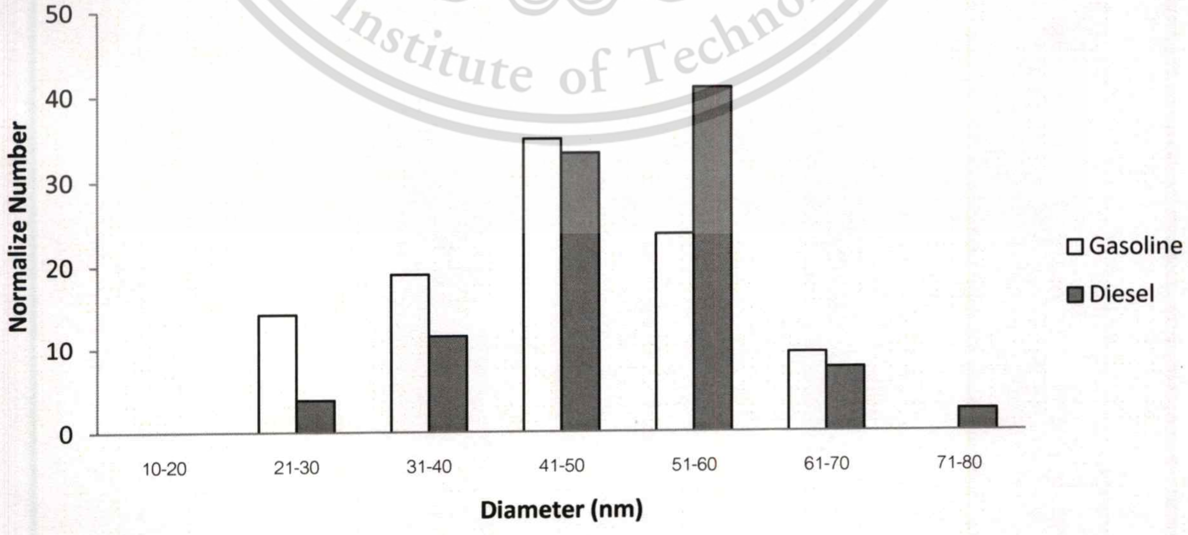


*Fig.8 Primary size distribution of gasoline and diesel lamp by TEM images*

However, the primary size of particle emissions is very difficult to measure by SEM image because surface of particle emissions were cover by unburned hydrocarbon. Then, primary size of particle emissions was measure later by TEM image. Primary size of gasoline and diesel particle emissions was shown in Fig.6 and Fig.7, respectively. TEM image was used to verify primary size, because they weren't covered by unburned hydrocarbon. Primary size of gasoline is slightly smaller than that of diesel, primary size of both particle emissions were approximately 25-60 nm and 50-60 nm, respectively. The particle size distribution in primary mode approximately 100 particles was shown in Fig.8. Most of gasoline lamp particle size is around 41-50 nm while the diesel lamp particle size is around 51-60 nm, slightly bigger than that of gasoline lamp particle.

### 4.2 Particle Emissions Concentration

The particle emissions were formed in the rich fuel region in the combustion chamber. The exhaust gas was emitted in the exhaust pipe. The remaining particle emissions of gasoline and ethanol were trapped by using a paper filter directly from the exhaust pipe. Subsequently, smoke meter was applied to measure the concentration of trapped particle emissions on the paper filter by light opacity method. The zero and 100 percentages of black smoke means no and full of particle emissions on the filter paper, respectively.

Fig.9 shows the percentage of gasoline and ethanol at 1500 rpm when varying engine speeds, injection behaviors and loads. The results showed that the percentage of black smoke of gasoline was higher than that of ethanol for all cases. The maximum percentage of black smoke was under 1500 rpm, stratified charge mode, 20% load and gasoline.

Hence, gasoline particle emissions formation was higher than ethanol. This could be explained that more particle emissions were remained in gasoline combustion than those of ethanol. Because ethanol contains oxygen molecules, ethanol is readily oxidized with the available oxygen in the flame zone.

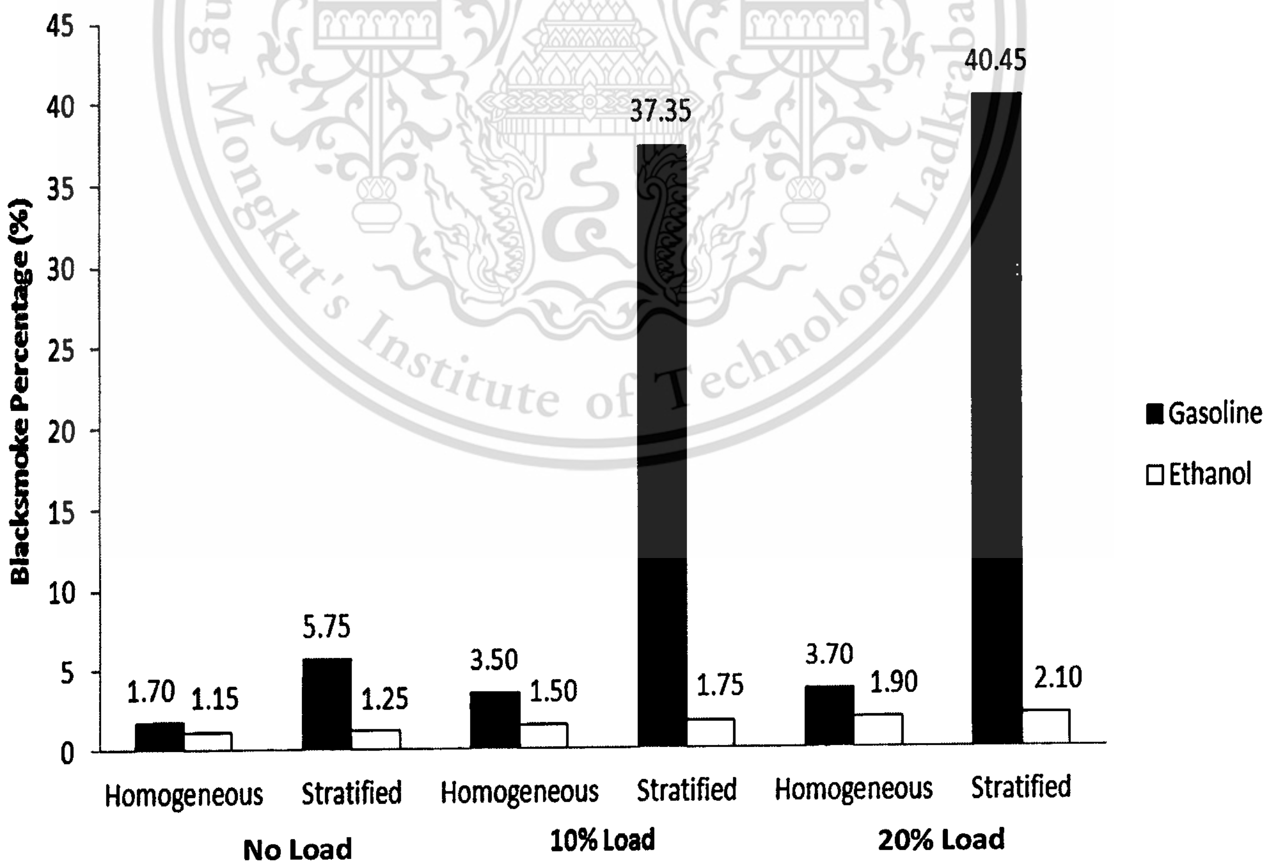


*Fig.9 Quantities of gasoline and ethanol DISI particle emissions under 1500 rpm varying loads condition*

## 5. Conclusions

The investigation of DISI engine fuelled gasoline and ethanol in this study shows the particle emissions concentration of gasoline and ethanol under varying engine speeds, loads and injection behaviors. Particle emissions were analyzed by smoke meter. A part from that, SEM and TEM images of gasoline and ethanol particle emissions from diffusion flame were used to verified the primary and accumulate particle emissions size. The main conclusions can be summarized as follows:

5.1 The primary and accumulate sizes of gasoline are homologous with that of diesel. The similarities of characteristic can be explained that particle emissions of any hydrocarbon by diffusion flame such as spray combustion are similar. That means gasoline can be the cause of particle emissions as diesel, and also causes of global gas emitting and environmental concern. So, the investigation of particle emissions of gasoline is so important as well as diesel.

5.2 The remaining particle emissions of gasoline is higher than that of ethanol due to ethanol is the oxygenated fuel so oxygen molecules in ethanol can improve more complete combustion of the engine.

5.3 The remaining particle emissions at engine speed 1000 rpm is lower than that at 1500 rpm due to higher engine speed relates to higher tumble intensity that causes better mixing formation in combustion chamber.

5.4 The remaining particle emissions of 20% loads operating is higher than that of either 10% or no load due to much amount of fuel injected so the rich fuel causes particle emissions.

5.5 The remaining particle emissions under stratified charge operating mode is higher than homogeneous charge operating mode due to the late injection made less fuel propagation than homogeneous charge operating mode so the rich fuel region around spark plug is the cause of particle emissions under stratified charge operating mode.

The oxygenated fuel is a strongly effect on particle emissions of combustion, then, using ethanol instead of gasoline may reduce emitting emissions, pollutions and also reduce the fossil fuel consumption.

## 6. Acknowledgment

The author is grateful to AE Lab at KMITL for information and good advices, MTEC for supporting SEM and TEM operation and also thanks to TAIST-Tokyo Tech and TGIST for providing scholarship.

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# Investigation of DISI Engine Particle Emissions

Athiwat Butmarasri[1,*], Chanont Jiwattayakul[2], Nattapong Suttikittiyaporn[2]
Preechar Karin[1], Chinda Charoenphonphanich[2]
Nuwong Chollacoop[3] and Hidenori Kosaka[4]

[1] International College, King Mongkut's Institute of Technology Ladkrabang, Bangkok, Thailand 10520
[2] Faculty of Engineering, King Mongkut's Institute of Technology Ladkrabang, Bangkok, Thailand 10520
[3] National Metal and Materials Technology Center (MTEC),Pathumthani, Thailand 12120
[4] Department of Mechanical and Control Engineering, Tokyo Institute of Technology, Japan
* Corresponding Author: Tel: 081 450 6599
E-mail: athiwatb@yahoo.com

***Abstract***

The reduction of particle emissions is nowadays an important issue with respect to emissions from gasoline powered motor vehicle. Using new development technologies and renewable oxygenated fuels are considered the most suitable solution for sustainable future. This is a part of an ongoing research focused on the comparison of particle emissions from gasoline and ethanol DISI (Direct Injection Spark Ignition) engine. Amount of particle emissions would be reduced by using ethanol DISI engine. In addition, physical structure of particle emissions emitted from gasoline and diesel diffusion flames were investigated by using a Scanning Electron Microscopy (SEM) and a Transmission Electron Microscopy (TEM). The DISI engine was tested on engine dynamometer with different loads and injection behaviors, with homogenous and stratified charge. Then, the particle emissions were sampled by smoke meter in order to investigate physical structure and measure the amount of particle emissions. The results showed that particle emissions emitted from ethanol are lower than gasoline. The average primary size of gasoline and diesel fuels particle emissions are approximately 25-60 nm and 50-60 nm, respectively. The accumulated sizes of gasoline and diesel fuels particle emissions are approximately 100-360 nm and 100-500 nm, respectively. The findings of this study can serve as guidance for the reduction particle emissions from gasoline DISI engines by using ethanol.

***Keywords***: Particle Emissions, Ethanol, DISI Engine

## 1. Introduction

Nowadays, the fuel consumption in transportation field is one of the main reasons to both realize the depleting fossil fuel and increasing amount of emissions. In particular, the limitations of new emission standards have set the emission levels further to reduce both spark ignition and diesel engines such as euro 6.

It is imperative, then, to find out alternative fuels instead of using fossil fuels. Ethanol is one of the most suitable alternative fuels for spark ignition engines because of the advantages of ethanol, e.g. better knock limit range due to higher octane number of ethanol, higher volumetric efficiency due to cooling effect of higher heat of vaporization and also reduced particle emissions due to more complete combustion from oxygen atom inside ethanol molecule.

The development of new clean spark ignition engines, such as direct injection spark ignition (DISI) engines is important because of the advantages of DISI engines, e.g. higher thermal efficiency due to direct fuel injection, higher power output than conventional homogeneous charge port injection spark ignition (PSI) engines and lower fuel consumption due to an ultra-lean combustion in stratified charge operating mode [1].

The use of ethanol in direct injection spark ignition engines is to reduce regulated pollutant emissions produced by internal combustion engines, as well as to reduce the greenhouse effect impact of transportation. Needless to say, NOx and HC emissions [2] are prejudicial effects on the environment and human health.

In addition to engine combustion, diffusion flame in stratified charge operation of DISI engine is similar to compression ignition (CI) engine that causes particle emissions consisting of a solid fraction (SOL) and a soluble organic fraction (SOF). Primary particles, composed of carbon and metallic ash, are coated with SOF and sulphate. A primary soot particle has two distinct parts: an inner core located at the central region of the primary particle and outer shell. The composition of particle emissions may vary widely depending on the operating conditions and fuel composition [3-7].

The objectives of this research are to characterize and measure amount of gasoline and ethanol DISI particle emissions. The advantage of ethanol is discussed in the view point of particle emissions.

## 2. Experimental Apparatuses

### 2.1 Fuels

The use of ethanol in the experiment was considered due to the reduction of particle emissions and fossil fuel consumption in automotive. The domestic production can reduce amount of transportation activities of gasoline fuels. Higher octane number than gasoline can perform a better anti-knock for increased compression ratio and performance subsequently. Since ethanol has a higher heat of vaporization, higher densities in the intake can increase the volumetric efficiency. Moreover, ethanol is an oxygenated fuel, a fuel that contains oxygen molecule, therefore excess air can react with CO in residual emissions. However, a lower heating value of ethanol, compared with gasoline, governs ethanol to be injected more than gasoline to achieve the same amount of total energy. Fuels properties are shown in Table 1.

Table 1. Fuels properties [8]

| Fuel properties | Ethanol | Gasoline |
|---|---|---|
| Formula | $C_2H_5OH$ | $C_4$ to $C_{12}$ |
| Molecular weight [g/mol] | 46.70 | 100-105 |
| Carbon [mass%] | 52.20 | 85-88 |
| Hydrogen [mass%] | 13.10 | 12-15 |
| Oxygen [mass%] | 34.70 | 2.70 |
| Density, kg/l, 15/15°C | 0.79 | 0.72-0.77 |
| Boling point, °C | 78 | 27-225 |
| Vapor pres., kPa at 38°C | 15.90 | 48-103 |
| Specific heat, kJkg-1K-1 | 2.40 | 2 |
| Viscosity, mPa s at 20 | 1.19 | 0.37-0.44 |
| Low, heating val., 103 kJ/l | 21.10 | 30-33 |
| Auto ignition temp., | 423 | 257 |
| Research octane number | 108.60 | 98 |
| Motor octane number | 92 | 87 |
| (R+M)/2 | 100 | 92.50 |
| Cetane number | - | 5-20 |
| Flammability lim., Vol% | 4.30/19 | 1.40/7.60 |
| Water Tolerance, Vol% | Compl.miscible | Negligible |
| Stoichiometric air/fuel ratio | 9 | 14.70 |
| Carbonyl [ppm] as C-O | 567 | - |
| Carbonyl [ppm] as acetone | 1117 | - |
| Carbonyl [ppm] as acetaldehyde | 893 | - |
| Sulphur [mg/kg] | <0.80 | 10 |
| Copper [mg/kg] | <0.10 | - |

### 2.2 Engine

A direct injection spark ignition (DISI) engine, inline 4 cylinders, 4 strokes, 1834 $cm^3$ displacement, was used in the experiment as the source of emissions, with the engine specification shown in Table 2. An increased fuel efficiency and high power output [4], by increased compression ratio, are the main advantages of DISI engines. In addition, the cooling effect of the injected fuel and the more evenly dispersed mixtures allow for more aggressive ignition timing curves [9]. Emissions levels can also be more accurately controlled with the DISI system. The cited gains are achieved by the precise control over amount of fuel and injection timings, which are varied according to the load conditions.

In addition, there are no throttling losses in some DISI engines, when compared to a conventional fuel injected or carburetor engines, which would greatly improve efficiency in engines without a throttle plate [10].

Table 2. Engine specification

| Model | 4G93 GDI |
|---|---|
| Type | In-line OHV, DOHC |
| Number of cylinders | 4 |
| Combustion chamber | Pentroof type |
| Displacement | 1.834 Liter |
| Bore | 81.0 mm |
| Stroke | 89.0 mm |
| Compression ratio | 12 : 1 |
| Maximum output | 96 kW @ 6000 rpm |
| Maximum torque | 177 Nm @ 3750 rpm |

### 2.3 Engine Dynamometer

An engine dynamometer was used in the experiment (Model: Tokyo Plant 150 PS, as shown in Fig.1) to measure force, moment of force (torque), or power, as well as control engine speeds and loads. The power produced by an engine can be calculated by simultaneously torque measuring and rotational speed (RPM) of engine.

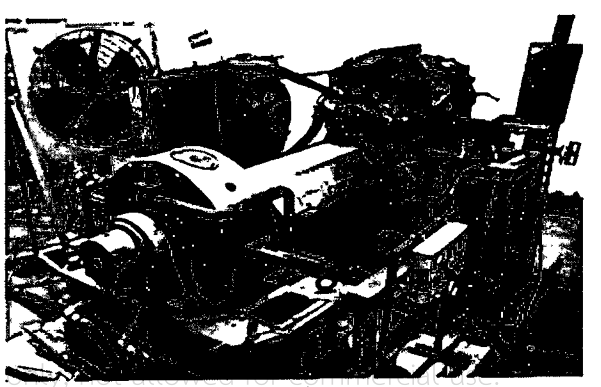

Fig.1. Engine dynamometer

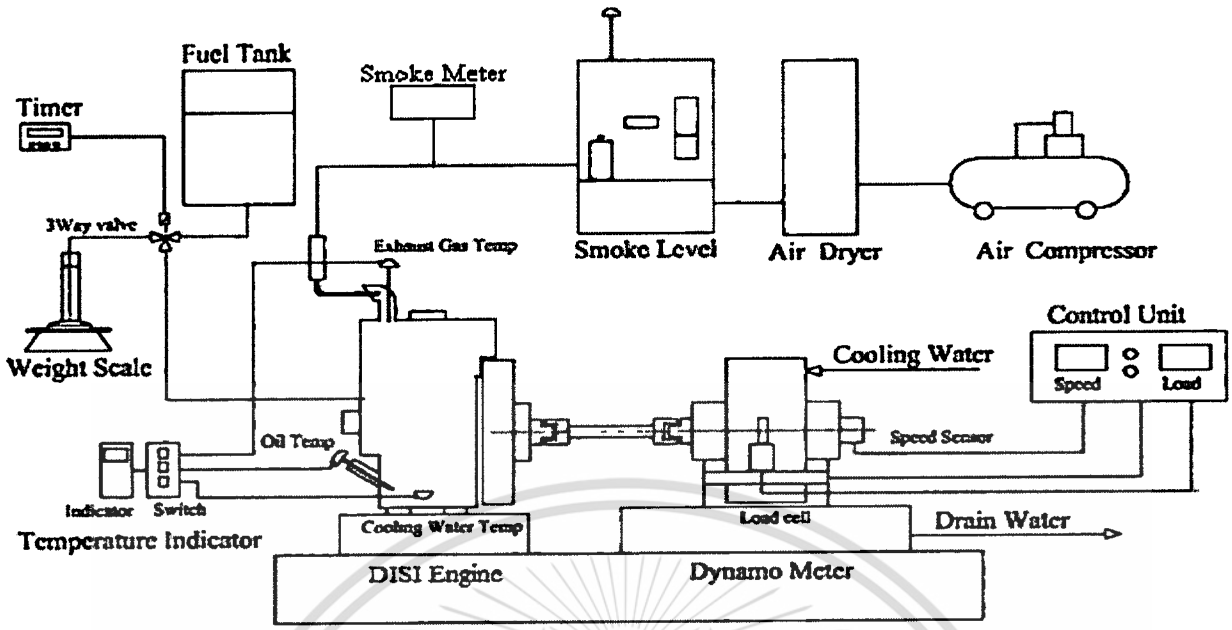


Fig.2 Schematic diagram of experimental setup

**3. Research Methodology and Procedures**

A direct injection spark ignition (DISI) engine, inline 4 cylinders, 4 strokes, 1834 $cm^3$ displacement was measured emissions at 1000 rpm idle and 1500 rpm under 10, 20% loads, using gasoline and ethanol, respectively. The 1000 rpm idle condition was selected to study a critical condition, in terms of stability, for DISI engine, and the 1500 rpm under 10, 20% loads were chosen as representative points for urban driving conditions. The injection behaviors were controlled for both homogenous and stratified charges. All the conditions investigated were carried out at $\lambda$=1. Particle emissions were sampled directly from the exhaust pipe, and then measured for concentration of particle emissions with smoke meter, as shown in Fig.2. On the other hand, gasoline and diesel diffusion flame particle emissions were generated by gasoline and diesel lamps in order to investigate the primary and accumulated particle emissions size by SEM, TEM and image analysis method, summation of longest and shortest lengths divided by two. In addition TEM and SEM images of ethanol particle emissions generated by lamp are in the progress.

**4. Results and Discussions**

**4.1 Structure of Particle Emissions**

Fig.3 shows gasoline and ethanol diffusion flames by Schlieren method image. The core of ethanol diffusion flame was lighter than that of gasoline, where particle emissions are formed at the center of diffusion flame. Similarly, Fig.4 shows gasoline and ethanol diffusion flames by optical image. The length of ethanol diffusion flame was shorter than that of gasoline due to ethanol contains less carbon content than gasoline and ethanol also contains oxygen atoms that promote more complete combustion.

Gasoline and diesel particle emissions images were taken in SEM and TEM method in order to verify primary and accumulated size and formation of particle emissions. Gasoline and diesel particle emissions were generated by fuel lamp. Figs.5(a) and 5(b) shows SEM image of gasoline and diesel accumulated particle emissions, respectively. Accumulated sizes of gasoline were slightly smaller than that of diesel, 100-360 nm vs. 100-500 nm.

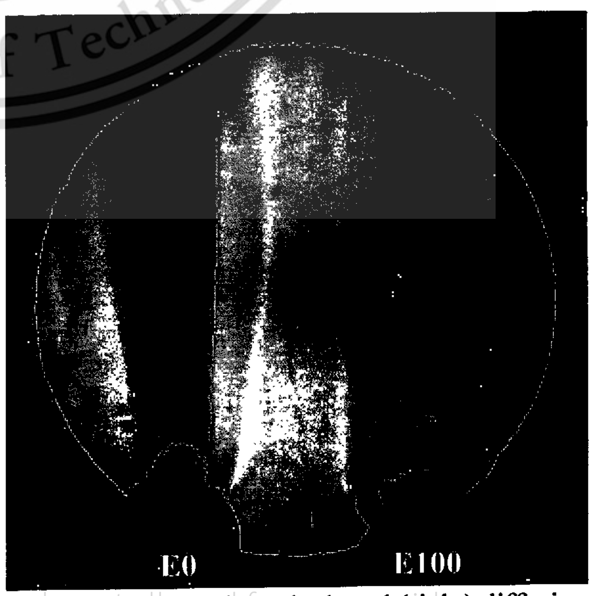


Fig.3 Gasoline (left) and ethanol (right) diffusion flames captured by Schlieren method

flames captured by Schlieren method

## 5. Conclusions

The investigation of DISI engine fuelled gasoline and ethanol in this study shows the particle emissions concentration of gasoline and ethanol under varying engine speeds, loads and injection behavior. Particle emissions were analyzed by smoke meter. In addition, SEM and TEM images of gasoline and ethanol particle emissions from diffusion flame were used to verify the primary and accumulated particle emissions size. The main conclusions can be summarized as follows:

1 The primary and accumulated sizes of gasoline are homologous with that of diesel. The similarities of characteristic can be explained that particle emissions of any hydrocarbon by diffusion flame such as spray combustion are similar. That means gasoline can be the cause of particle emissions as diesel, and also cause of global gas emitting and environmental concern. Hence, the investigation of particle emissions of gasoline is as important as diesel

2 The remaining particle emissions of gasoline is higher than that of ethanol due to ethanol is the oxygenated fuel so oxygen molecules in ethanol can improve more complete combustion of the engine

3 The remaining particle emissions at engine speed 1000 rpm is lower than that at 1500 rpm since higher engine speed relates to higher tumble intensity that causes better mixing formation in combustion chamber.

4 The remaining particle emissions of 20% loads operating is higher than that of either 10% or no load due to much amount of fuel injected, rich fuel, causing particle emissions.

5 The remaining particle emissions under stratified charge operating mode is higher than homogeneous charge operating mode since the late injection made less fuel propagation than homogeneous charge operating mode. The rich fuel region around spark plug is the cause of particle emissions under stratified charge operating mode.

6 Since oxygenated fuel has a strong effect on particle emissions of combustion, using ethanol instead of gasoline may reduce emissions and pollutions from internal combustion engine.

7 Although the use of light opacity method in order to measure particle emissions concentration is an indirect measurement and that can cause some error in results, the method shows significant in the results which lead to good conclusions as well as mass measurement.

## 6. Acknowledgment

The author is grateful to MTEC for supporting SEM and TEM operation and also thanks to TAIST-Tokyo Tech and TGIST for providing scholarship.

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# 14-20135412 A Study of DISI Engine Emissions Using Ethanol-Gasoline Blended Fuels

**Athiwat Butmarasri [1], Preechar Karin [1], Chinda Charoenphonphanich [2]**

**Nuwong Chollacoop [3] and Hidenori Kosaka [4]**

*1) International College, King Mongkut's Institute of Technology Ladkrabang, Bangkok, Thailand 10520*
*(E-mail: athiwatb@yahoo.com)*
*2) Faculty of Engineering, King Mongkut's Institute of Technology Ladkrabang, Bangkok, Thailand 10520*
*3) National Metal and Materials Technology Center (MTEC), Pathumthani, Thailand 12120*
*4) Department of Mechanical and Control Engineering, Tokyo Institute of Technology, Japan*

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**ABSTRACT**: This is a part of an ongoing research focused on performance and particle emissions from gasoline and ethanol DISI engine. The engine was test on engine dynamometer with different loads and injection conditions. The gasoline and ethanol distillation curves showed the same trend as spray images from CVCC, that ethanol can diffuse easily and vaporization of ethanol is more complete than gasoline. Spray images also showed burn rate of ethanol is faster than gasoline. Particle emissions emitted from ethanol are lower than gasoline. In addition, the primary and accumulated size of ethanol particle emissions is smaller than gasoline.

**KEY WORDS:heat engine, spark ignition engine, harmful emissions** Ethanol,Particle emissions (A1)

## 1. INTRODUCTION

The increasing of global energy demand and stringent pollution regulations have promoted research on alternative fuels. In Thailand, ethanol can be produced from many sources of national agriculture products as renewable fuel. The development of new clean spark ignition engines, such as direct injection spark ignition (DISI) engines, is important because of the use of ethanol in DISI engine can reduce regulated pollutant emissions produced by internal combustion engines In addition to engine combustion, diffusion flame in stratified charge operation of DISI engine is similar to compression ignition (CI) engine that causes particle emissions consisting of a solid fraction (SOL) and a soluble organic fraction (SOF). The composition of particle emissions may vary widely depending on the operating conditions and fuel composition [1-5].

P. Ornman et al.[6] found that gasoline has started to vaporize earlier than ethanol but when the tine left, ethanol was fully evaporated in to vapor phase faster than that of gasoline. It can imply that gasoline has more light and heavy fraction that ethanol. Thus, the lighter components tend to be evaporated early in the beginning stage while ethanol wasn't start. After the time past, heavy fraction in gasoline which are comprised of various higher carbon atoms regardless to RVP properties still remained while ethanol which is comprised of only one component, lower carbon atom than the gasoline, already completely changed into the vapor phase.

P. Ornman et al. [6] tested gasoline, ethanol and ethanol-gasoline blended flame propagation using Schlieren technique and taken with high speed video camera at stratified condition. Flame development of ethanol shows the fastest rate compare with lower ethanol concentration. They found that the fuel properties, especially the oxygen content, can affect to the flame speed in term of increasing performance of reaction rate. With increasing the ethanol concentration in the fuel, oxygen content will be also increased due to the ethanol- gasoline blended fuels have more oxygenated properties than the pure gasoline. Thus flame speeds that were depended on oxidation rate were increased.

The objectives of this research are to characterize and measure amount of gasoline, ethanol and ethanol-gasoline blended DISI engine emissions. The advantage of ethanol is discussed in the view point of particle emissions.

## 2. EXPERIMENTAL APPARATUSES

### 2.1 Fuels

The use of ethanol in the experiment was considered due to the reduction of particle emissions and fossil fuel consumption in automotive. The domestic production can reduce amount of transportation activities of gasoline fuels. Higher octane number than gasoline can perform a better anti-knock for increased compression ratio and performance subsequently. Since ethanol has a higher heat of vaporization, higher densities in the intake can increase the volumetric efficiency. Moreover, ethanol is an oxygenated fuel, a fuel that contains oxygen molecule, therefore excess air can react with CO in residual emissions. Fuels properties are shown in Table 1.

Table.1 Fuels properties

| Fuels properties | Gasoline [8] | E10 | E20 | E85 | Ethanol [8] |
|---|---|---|---|---|---|
| Formula | $C_4$ to $C_{12}$ | $CH_{2.043}O_{0.015}$ | $CH_{1.63}O_{0.065}$ | $CH_{2.822}O_{0.425}$ | $C_2H_5OH$ |
| Molecular weight [g/mol] | 100-105 | | 88.12 | 50.60 | 46.70 |
| Carbon [mass%] | 85-88 | 86.70 | 79.85 | 55.36 | 52.20 |
| Hydrogen [mass%] | 12-15 | 13.20 | 12.88 | 12.89 | 13.10 |
| Oxygen [mass%] | 0 | 1.94 | 7.54 | 31.75 | 34.70 |
| Density, kg/l, at 15°C | 0.72-0.77 | 0.7608 | 0.7645 | | 0.79 |
| Vapor pres., kPa at 38°C | 48-103 | 59.6 | 58.3 | 35-70 | 15.90 |
| Specific heat, kJkg-1K-1 | 2 | | | | 2.40 |
| Lower, hheating val.,, 103 MkJ/lkg | 44.00 | 40.97 | 40.60 | 29.50 | 26.90 |
| Research octane number | 92.4 | 98.1 | 98.3 | 101.6 | 108.60 |
| Motor octane number | 81.2 | 82.3 | 84.6 | 91.1 | 92 |
| (R+M)/2 | 86.8 | 90.2 | 91.45 | 96.35 | 100 |
| Water Tolerance, Vol% | Negligible | | | | Compl.miscible |
| Stoichiometric air/fuel ratio | 14.70 | 14.05 | 13.51 | 9.87 | 9.03 |
| Distillation temperature, °C | | | | | |
| Initial boiling point, IBP | 35 | 36.5 | 37.8 | 41.3 | 77.6 |
| 10 vol% | 51.5 | 51.6 | 53.5 | 66.6 | 77.8 |
| 20 vol% | 56.5 | 55.7 | 57.8 | 74.4 | 77.9 |
| 30 vol% | 61.8 | 59.7 | 62.5 | 76.8 | 77.9 |
| 40 vol% | 68.6 | 63.8 | 66.8 | 77.4 | 77.9 |
| 50 vol% | 78.2 | 70.2 | 70.8 | 77.5 | 77.9 |
| 60 vol% | 91.5 | 99.4 | 73.7 | 77.6 | 77.9 |
| 70 vol% | 108.6 | 117.9 | 99.9 | 77.7 | 78 |
| 80 vol% | 125.2 | 136.1 | 130 | 77.7 | 78 |
| 90 vol% | 154 | 160.2 | 155 | 77.8 | 78 |
| End boiling point | 197.3 | 187.2 | 184.6 | 80.5 | 80 |

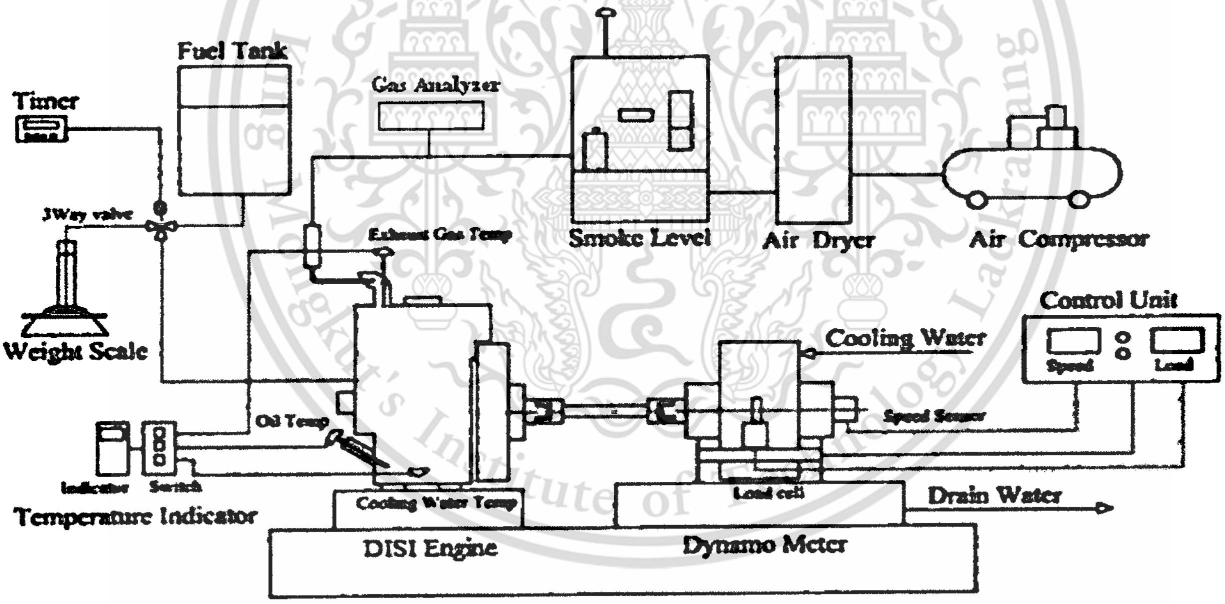


Fig.1 Schematic diagram of experimental setup

**2.2 Engine**

A direct injection spark ignition (DISI) engine, inline 4 cylinders, 4 strokes, 1834 $cm^3$ displacement, was used in the experiment as the source of emissions, with the engine specification shown in Table 2. An increased fuel efficiency and high power output [7], by increased compression ratio, are the main advantages of DISI engines. In addition, the cooling effect of the injected fuel and the more evenly dispersed mixtures allow for more aggressive ignition timing curves [8]. Emissions levels can also be more accurately controlled with the DISI system. The cited gains are achieved by the precise control over amount of fuel and injection timings, which are varied according to the load conditions[9].

Table 2. Engine specification

| Model | 4G93 GDI |
|---|---|
| Type | In-line OHV, DOHC |
| Number of cylinders | 4 |
| Displacement | 1.834 Liter |
| Compression ratio | 12 : 189.0 mm |
| Maximum output | 96 kW @ 6000 rpm12 : 1 |
| Maximum torque | 177 Nm @ 3750 rpm |

### 2.3 Engine Dynamometer

An engine dynamometer was used in the experiment (Model: 150 PS) to measure force, moment of force (torque), or power, as well as control engine speeds and loads. The power produced by an engine can be calculated by simultaneously torque measuring and rotational speed (RPM) of engine.

## 3. Research Methodology and Procedures

A direct injection spark ignition (DISI) engine, inline 4 cylinders, 4 strokes, 1834 $cm^3$ displacement was measured emissions at 1000 rpm idle, 1500 rpm and 2000 rpm under 10, 20% loads, using gasoline, ethanol and ethanol-gasoline blended, respectively. The 1000 rpm idle condition was selected to study a critical condition, in terms of stability, for DISI engine, the 1500 rpm and 2000 rpm under 0, 10, 20% loads were chosen as representative points for urban driving conditions. The injection behaviors were controlled for both homogenous and stratified charges. All the conditions investigated were carried out at $\lambda=1$. Particle emissions were sampled directly from the exhaust pipe, and then measured for concentration of particle emissions with smoke meter, as shown in Fig.1. On the other hand, exhaust emissions such as CO and $CO_2$ were sampled from an exhaust pipe directly in order to be measured by a gas analyzer. In addition, performances such as brake specific fuel consumption (bsfc) and brake specific energy consumption (bsec) were measured at 2000 rpm under 20% load to understand the variations of fuel consumption of the test engine.

## 4. Results and Discussions

### 4.1 Particle Emissions Concentration

The particle emissions were formed in the rich fuel region in the combustion chamber. The exhaust gas was emitted in the exhaust pipe. The remaining particle emissions of gasoline and ethanol were trapped by using a paper filter directly from the exhaust pipe. Subsequently, smoke meter was applied to measure the concentration of trapped particle emissions on the paper filter. The zero and 100 percentages of black smoke means no and full of particle emissions on the filter paper, respectively.

Fig.2 shows the black smoke percentage of gasoline, ethanol and ethanol-gasoline blended fuels under 1000 rpm no load condition when varying injection behaviors. The results showed that the percentage of black smoke of gasoline was higher than that of ethanol and ethanol-gasoline blended for all cases. The maximum percentage of black smoke was 1000 rpm, stratified and gasoline condition. Fig.3 shows the black smoke percentage of ethanol-gasoline blended fuels under 1500 rpm varying engine speeds, injection behaviors and loads. The results showed that the percentage of black smoke of gasoline was higher than that of ethanol and ethanol-gasoline blended for all cases. The maximum percentage of black smoke was 1500 rpm, stratified charge mode, 20% load and gasoline.

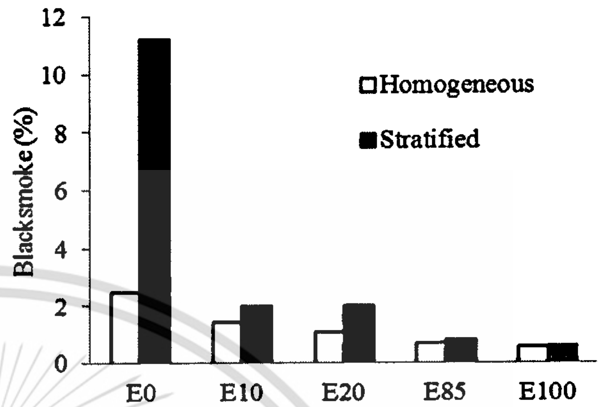


Fig.2 Quantities of gasoline and ethanol DISI particle emissions under 1000 rpm no load condition

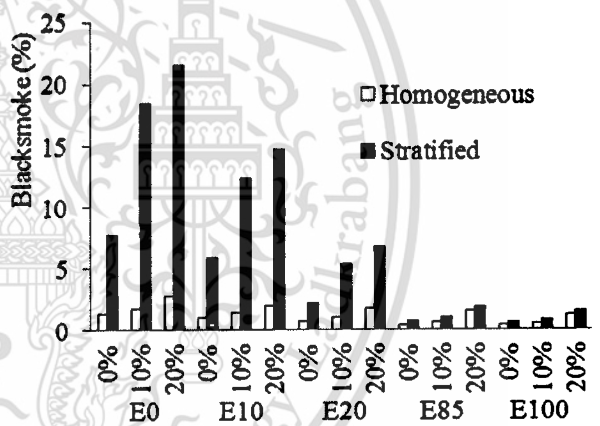


Fig.3 Quantities of gasoline and ethanol DISI particle emissions under 1500 rpm varying loads condition

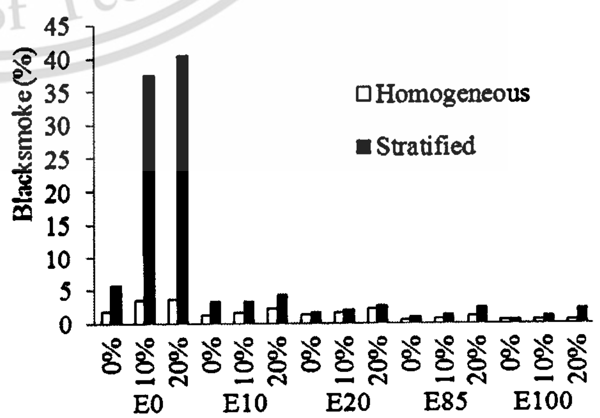


Fig.4 Quantities of gasoline and ethanol DISI particle emissions under 2000 rpm varying loads condition

Fig.4 shows the black smoke percentage of ethanol-gasoline blended fuels under 2000 rpm varying engine speeds, injection behaviors and loads. The results showed that the black smoke percentage of gasoline was higher than that of ethanol and ethanol-gasoline blended for all cases. The maximum black smoke percentage was 2000 rpm, stratified charge mode, 20% load and gasoline.

Hence, gasoline particle emissions formation was higher than ethanol and ethaol-gasoline bleded. This could be explained that more particle emissions were remained in gasoline combustion than those of ethanol and ethanol-gasoline blended. Since ethanol is an oxygenated fuel that contains oxygen molecules, ethanol is readily oxidized with the available oxygen in the flame zone.

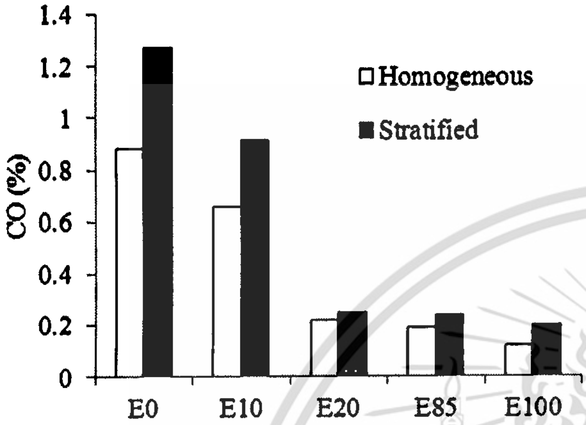


Fig.5 CO emissions at 1000 rpm

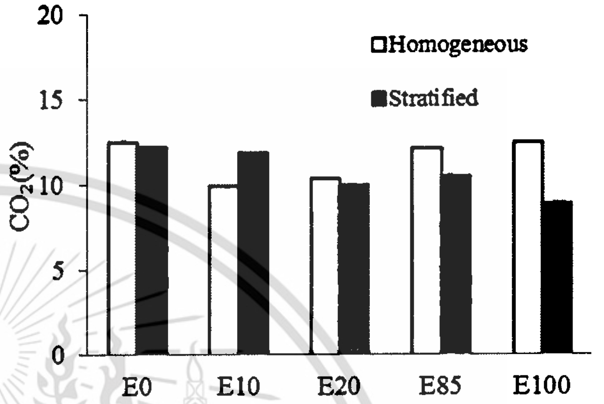


Fig.8 $CO_2$ emissions at 1000 rpm

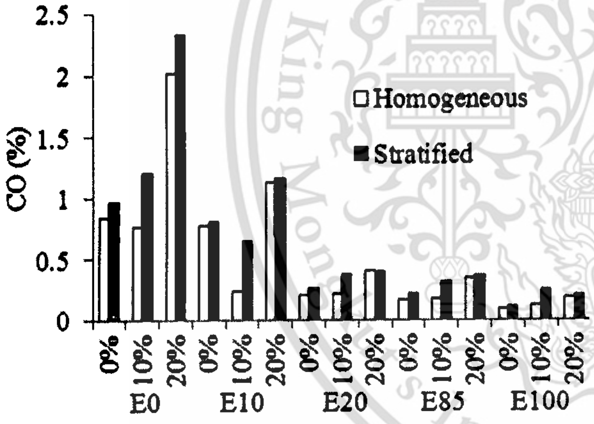


Fig.6 CO emissions at 1500 rpm

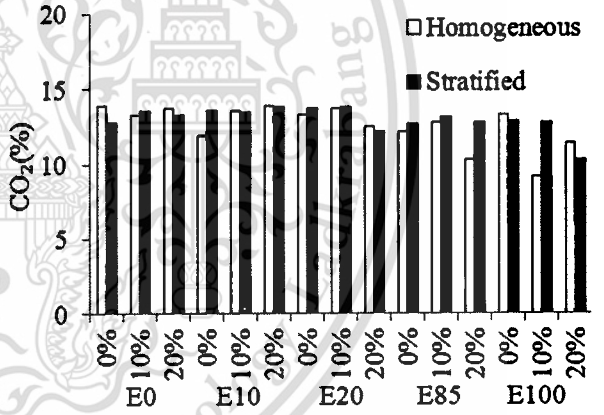


Fig.9 $CO_2$ emissions at 1500 rpm

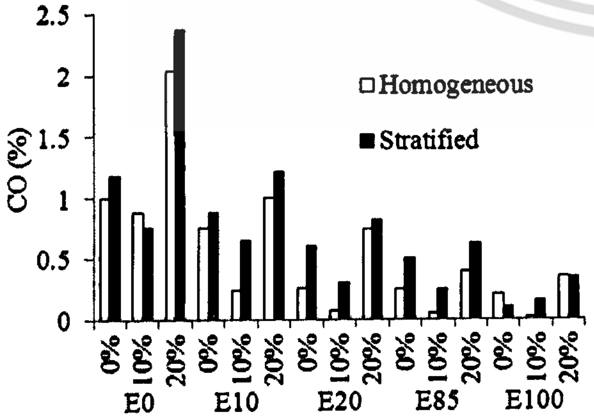


Fig.7 CO emissions at 2000 rpm

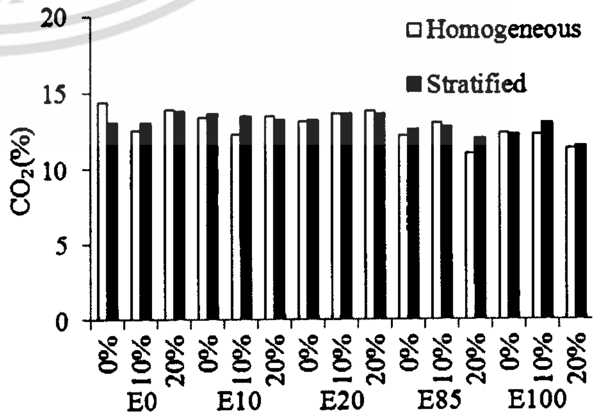


Fig.10 $CO_2$ emissions at 2000 rpm

**4.2 Exhaust Emissions**

CO or carbon monoxide is caused by insufficient oxygen distribution and the concentrations of CO are highly dependent on the air- fuel ratio. The comparison of CO emissions at 1000 rpm under different fuels is shown in Fig.5. Compared to gasoline at stratified, the decreases in CO emissions for E10, E20, E85 and E100 are 28.35 %, 80.31 %, 81.10 % and 84.25 %, respectively. At homogeneous, the decreases in CO emissions for E10, E20, E85 and E100 are 25 %, 75 %, 78.41 % and 86.36 %, respectively. The comparison of CO emissions at 1500 rpm under different fuels, loads and speeds are shown in Fig.6. Compared to gasoline at stratified, on average load, the decreases in CO emissions for E10, E20, E85 and E100 are 41.78 %, 76.89 %, 80.22 % and 87.33 %, respectively. At homogeneous, on average load, the decreases in CO emissions for E10, E20, E85 and E100 are 40.61 %, 76.79 %, 81.21 % and 88.95 %, respectively. The comparison of CO emissions at 2000 rpm under different fuels, loads and speeds are shown in Fig.7. Compared to gasoline at stratified, on average load, the decreases in CO emissions for E10, E20, E85 and E100 are 38.89 %, 61.78 %, 69.56 % and 86.89 %, respectively. At homogeneous, on average load, the decreases in CO emissions for E10, E20, E85 and E100 are 45.03 %, 70.44 %, 80.94 % and 84.25 %, respectively. Which can be explained by the fact that ethanol is oxygenated fuel that contain oxygen atom in ethanol molecule. The oxygenated fuel can provide more oxygen for the combustion process in the rich mixture. That is the reason of CO emissions reduction.

$CO_2$ or carbon dioxide is caused by the complete combustion of hydrocarbon, fuel. The combustion under stoichiometric air-fuel ratio should produce only $CO_2$ and $H_20$ (water). In this study, the effect of ethanol on the exhaust emissions could not be seen clearly. Especially, the effect of different loads and speeds showed unstable trend on $CO_2$ emissions. The comparison of $CO_2$ emissions at 1000 rpm under different fuels is shown in Fig.8. Compared to gasoline at stratified mode, the decreases in $CO_2$ emissions for E10, E20 and E85 are 20.8 %, 16.8 % and 3.2 %, respectively. At homogeneous, the decreases in $CO_2$ emissions for E10, E20, E85 and E100 are 2.46 %, 18.03, 13.93 % and 27.05 %, respectively. The comparison of $CO_2$ emissions at 1500 rpm under different fuels, loads and speeds are shown in Fig.9. Compared to gasoline at stratified, on average load, the decreases in $CO_2$ emissions for E85 and E100 are 2.77 % and 9.32 %, respectively, while the increase in $CO_2$ emissions for E10 is 3.02 %. At homogeneous, on average load, the decreases in $CO_2$ emissions for E10, E20, E85 and E100 are 3.67 %, 3.42 %, 13.69 % and 17.11 %, respectively. The comparison of $CO_2$ emissions at 2000 rpm under different fuels, loads and speeds are shown in Fig.10. Compared to gasoline at stratified, on average load, the decreases in $CO_2$ emissions for E85 and E100 are 6.03 % and 7.53 %, respectively, while the increase in $CO_2$ emissions for E10 and E20 are 1.25 % and 1.51 %, respectively. At homogeneous, on average load, the decreases in $CO_2$ emissions for E10, E20, E85 and E100 are 3.92 %, 0.73 %, 11.27 % and 11.52 %, respectively. CO2 emissions decrease with the increase of ethanol blended percentage. This could be explained that lower carbon content of ethanol and ethanol-blended, compared to gasoline.

**4.3 Fuel consumption**

From the experimental results, the comparison of brake specific fuel consumption (bsfc) for test fuels is shown in Fig.11. At the engine speed of 2000 rpm 20% loads, bsfc for E10, E20, E85 and E100 increased by 25.32 %, 38.29 %, 48.74 % and 56.51 %, respectively, compared to those of gasoline, due to the lower energy content of the ethanol as seen in Table 1, the heating value of ethanol is lower than that of gasoline. This means that the engine needs a higher amount of fuel to produce the same output power in a gasoline fueled engine. Thus the use of ethanol and ethanol-gasoline blended resulted a slightly increasing in the fuel consumption compared to the use of gasoline.

The comparison of brake specific energy consumption (bsec) for test fuels is shown in Fig.12. At the engine speed of 2000 rpm 20% loads, bsec for E10, E20, E85 and E100 increased by 22.20 %, 34.15 %, 25.51 % and 30.99 %, respectively, compared to those of gasoline.

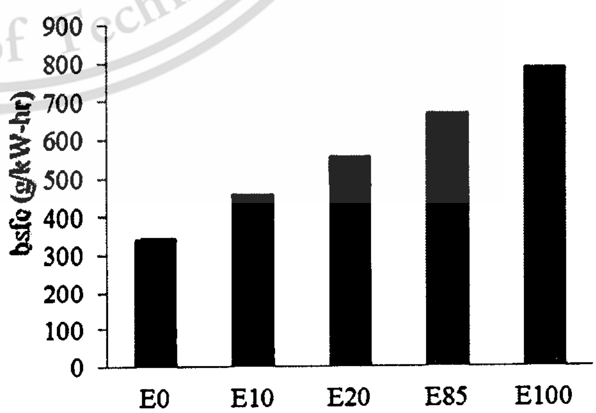


Fig.11 Comparison of gasoline and ethanol DISI bsfc under 2000 rpm 20% loads condition

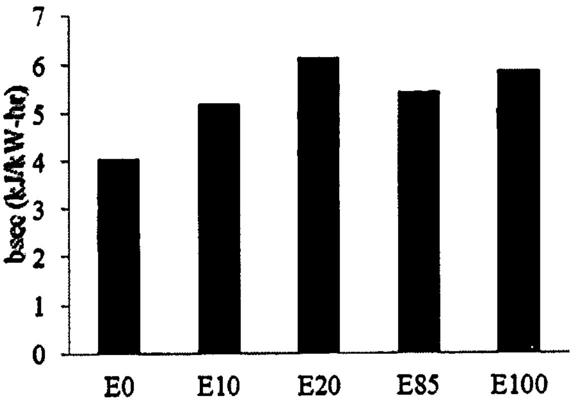

Fig.12 Comparison of gasoline and ethanol DISI bsec under 2000 rpm 20% loads condition

## 5. Conclusions

The investigation of DISI engine fuelled gasoline, ethanol and ethanol-gasoline blended in this study shows the particle emissions concentration of gasoline and ethanol under varying engine speeds, loads and injection behaviors. Particle emissions were ȧnalyzed by smoke meter. On the other hand, exhaust emissions such as CO and $CO_2$ were measured by a gas analyzer. In addition, performances such as brake specific fuel consumption (bsfc) and brake specific energy consumption (bsec) were investigated. The main conclusions can be summarized as follows:

1 DISI engine can be the cause of particle emissions due to that operating principle as diesel engine, and also cause of global gas emitting and environmental concern. Hence, the investigation of particle emissions of gasoline DISI engine is as important as diesel engine.

2 The remaining particle emissions of gasoline is higher than that of ethanol due to ethanol is the oxygenated fuel so oxygen molecules in ethanol can improve more complete combustion of the engine.

3 The remaining particle emissions of 20% loads operating is higher than that of either 10% or no load due to much amount of fuel injected, rich fuel, causing particle emissions.

4 The remaining particle emissions under stratified charge operating mode is higher than homogeneous charge operating mode since the late injection made less fuel propagation than homogeneous charge operating mode. The rich fuel region around spark plug is the cause of particle emissions under stratified charge operating mode.

5 The CO emissions of ethanol is lower than gasoline due to the oxygen atom in ethanol molecule promotes complete combustion and the $CO_2$ emissions of ethanol is lower than gasoline due to lower carbon content of ethanol, compared to gasoline. In addition, lower heating value of ethanol leads to higher fuel consumption, compared to gasoline.

6 Since oxygenated fuel has a strong effect on particle emissions of combustion, using ethanol instead of gasoline may reduce emissions and pollutions from internal combustion engine.

7 Although the use of smoke meter in order to measure particle emissions concentration is an indirect measurement and that can cause some error in results, the method shows significant in the results which lead to good conclusions as well as mass measurement.

## 6. Acknowledgment

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# AUTHOR BIOGRAPHY

Author: Mr. Athiwat Butmarasri
Date of Birth: May 9$^{th}$, 1989
Address: 13 Moo 17 Khampia, Phochai, Roi-Et 45230

Education:

2007-2010 Bachelor Degree in Mechanical Engineering,
King Mongkut's Institute of Technology Ladkrabang

2011-2015 Master Degree in Automotive Engineering
(International Program), International college
King Mongkut's Institute of Technology Ladkrabang

Publications:

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