THE COMPARISON BETWEEN BLENDED AND FUMIGATED METHOD TO APPLY ETHANOL FUEL IN A DIESEL ENGINE

SURITH DULANJALA DE SILVA

A THESIS REPORT SUBMITTED IN PARTIAL FULFILLMENT
OF THE REQUIREMENTS FOR THE DEGREE OF
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ABSTRACT

Fossil fuel have been using for couple of decades to achieve day to day needs of human beings such as transportation, electricity generation, household needs military conditions etc. The usage of fossil fuel has been increasing rapidly with industrial revolution and with the achievement of new innovations by late scientists. Formation of internal combustion engine was one of the results of industrial revolution and now it has come to a peak point of its usage and there is a lack of fossil fuels to power them up. Not only the present world faces a lack of fossil fuel to run the engines but also the emission of IC engines causes a huge environmental pollution affecting both natural resources and humans. Toxic emissions of diesel engines directly affect to greenhouse effect, depletion of ozone layer and health problems of people. Lack of fossil fuel has made a huge competition of buying crude oil and sometimes it has become the cause to start war among countries.

Alternative fuels such as bio diesel and ethanol are becoming popular in some countries to reduce the importing cost of crude oil and to low down the emission level of engines. Bio fuels can be found from biomass and it is cheaper compared to crude oil as well. In this experiment also tried to use bio fuel such as bio diesel (B20) and ethanol in a diesel engine using two different operation condition to figure out which one is the good enough in some factors such as power, torque, THC, CO and NOx emissions.

ACKNOWLEDGEMENT

I would like to give my warm regards to my parents who have been supporting me my ups and downs since my childhood and brought me up and helped financially with hearts full of kindness to encourage for my success. I know without both of you I would not have come this far in my life.

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BTE	Brake Thermal Efficiency
BSEC	Brake Specific Energy Consumption
HRR	Heat Release Rate

CHAPTER 1

INTRODUCTION

1.1 Research Background

Due to the existing competition for fossil fuel, most countries try to do experiments on bio fuels with various blending proportions of bio diesel with pure diesel like B7, B10, and B20. Diesel fuel is a nonrenewable fuel made from petroleum. Using biodiesel fuel produces less pollution than using petroleum diesel fuel. Any vehicle that operates on diesel fuel can use biodiesel. Biodiesel fuel has chemical characteristics similar to petroleum-based diesel, so it can be used as a direct substitute for diesel fuel. Biodiesel fuel can also be blended with petroleum diesel in any percentage without reducing vehicle fuel economy. Any diesel engine can use biodiesel at blend levels of 5% by volume (B5) or lower. A blend of 20% biodiesel with 80% petroleum diesel is known as B20. Some federal and state government fleets, such as school and transit buses, snowplows, garbage trucks, mail trucks, and military vehicles, use biodiesel. Public fueling stations that sell biodiesel blends to the public are available in nearly every state. Low-level biodiesel blends like B2 and B5 are popular fuels in the trucking industry because biodiesel has excellent lubricating properties, so the blends can benefit engine performance. Pure biodiesel (often called B100) and biodiesel blends are sensitive to cold weather and may require a special type of anti-freeze, just like petroleum-based diesel fuel. Biodiesel acts like a detergent additive, loosening and dissolving sediments in storage tanks. Because biodiesel is a solvent, B100 may cause rubber and other components to fail in older vehicles. This problem does not occur with biodiesel blends.

Ethanol is another alternative fuel which can be used for both diesel and gasoline engines. Ethanol (CH_3CH_2OH) is a clear, colorless liquid. It is also known as ethyl alcohol, grain alcohol, and EtOH .Ethanol has the same chemical formula regardless of whether it is produced from starch- and sugar-based feedstocks, such as corn grain (as it primarily is in the United States), sugar cane (as it primarily is in Brazil), or from cellulosic feedstock (such as wood chips or crop residues).

Ethanol has a higher octane number than gasoline, providing premium blending properties. Minimum octane number requirements for gasoline prevent engine knocking and ensure drivability. Low-octane gasoline is blended with 10% ethanol to attain the standard 87 octane.

Ethanol contains less energy per gallon than gasoline, to varying degrees, depending on the volume percentage of ethanol in the blend. Denatured ethanol (98% ethanol) contains about 30% less energy than gasoline per gallon. Ethanol's impact on fuel economy is dependent on the ethanol content in the fuel and whether an engine is optimized to run on gasoline or ethanol.

Ethanol can either be blended with bio diesel or fumigate to intake manifold in diesel engines. Due to miscible problems ethanol can be blended only up to a certain point with bio diesel. By using alternative fuels like bio diesel and ethanol in CI engines, the price of importing crude oil can be reduced and by using natural biomass bio fuel can be formed in Thailand. In this experiment try to investigate how bio fuel and ethanol will affect a small size diesel engine.

1.2 Objectives

1. To investigate performances combustion characteristics and emissions of various portion of blend bio diesel with ethanol

2. To investigate performances, combustion characteristics and emissions of ethanol fumigation with bio diesel direct injection

3. To compare the results of blended fuel and fumigated method to conclude which method of injection, which type of fuel obtained are in the acceptable range

1.3 Scope

1. Try to figure out miscibility of bio diesel and ethanol and choose the fuel which can miscible homogenously in large portion of ethanol

2. Check properties of blended fuel

3. Do injection characteristics of the common rail injector for blended fuel

4. Do injection characteristics of the ethanol injector for ethanol fumigation

5. Calculate the rate fumigation level according to energy content of desired blended fuel

6. Test engine using fuels that have been blended to archive results of performances, emission and combustion characteristics.

7. Run the engine1400-2000rpm in full load increment of 200 of engine speed.

CHAPTER 2

LITERATURE REVIEW

Excellent drivability and fuel efficiency of diesel engines have become more popular in various sectors such as agriculture, transportation and power plant industry. (1) Diesel engines emit less carbon monoxide (CO), hydrocarbon (HC) and carbon dioxide (CO2) compared to gasoline engines. However, diesel engines emit high level of nitrogen oxide (NOx) and particulate matter. (2) Greenhouse effect is increased severely by diesel engines due to increased number of diesel engine usage and long distance travelling. Lot of health diseases are caused by air pollution especially respiratory and cardiovascular problems. Increasing global anxiety about combustion related pollutants, such as THC (total hydrocarbon), NOx, SOx (Sulphur oxides), CO, CO2, acid rain and photochemical smog, as well as the depletion of ozone layer, has led many nations to control emissions and implement initiatives to control pollution. In addition, current petroleum reserves are subjected to deplete in the near future (3).

Due to these growing concerns studies on the applicability of clean energy technologies continue to receive attention. Hence, stricter diesel fuel standards have been announced to decrease NOx, particulate and CO2 emissions, and fuel consumption [2]. In addition, the search for sustainable alternative fuels such as alcohols, biodiesel and ethers for diesel engines has recently become important. In order to meet the increasing energy demand and control the environmental pollution, it is important to adopt a sustainable policy that allows the substitution of diesel fuel by neat biofuels such as biodiesel, bioethanol, bio butanol or blends of such biofuels with diesel fuel. Furthermore, studies dedicated towards the reduction of emissions by diesel vehicles have shown that modifications of the specifications of commercial diesel fuel are required to decrease and optimize emissions from diesel engines.

The mixing of oxygenated additives with diesel oil supplies the oxygen required to form CO2 instead of carbon-rich particles. This in turn may considerably reduce PM emissions (4, 5). The use of oxygenates as fuel additives to facilitate cleaner burning of petroleum fuels is about a half century old. Since its introduction, many investigators have studied the impact of blending various oxygenates with diesel fuel. The most commonly investigated oxygenated additives are alcohols and methyl or ethyl esters (biodiesel).

Moreover, the blending of alcohols such as methanol, ethanol and butanol was also found to have significant effects on the fuel properties, combustion and emissions of diesel fuels. The effects of these blends in terms of emissions are almost comparable with ethers, esters and other oxygenates. These blends also significantly decrease the exhaust emissions (6) and enhance octane number when mixed into diesel fuel. Such enhancement is particularly important in unleaded fuels. In addition, due to their complete burning power, alcohols have the potential to increase combustion efficiency. Therefore, the blends of alcohols (methanol, ethanol and butanol)–diesel oil with and without cetane number (CN) improver are potential alternative fuels. Moreover, these alcohols can be produced from biomass resources. As a renewable resource, biomass represents a possibly endless source of feedstock for the production of biofuels. Nowadays butanol is normally synthesized from petroleum sources, but early butanol production was reported to use biological sources. In the first half of the 20th century, the commercial production of butanol from biomass was prominent, and this is what today's butanol users are anticipating to ensue again (1). n-Butanol is a type of primary alcohol that has advantages over methanol and ethanol for use as an alternative fuel in IC engines, since most of the fuel properties of n-butanol are more similar to diesel fuel.

During the 1970s, despite their limited solubility and stability with diesel fuel, low Cetane Number, methanol and ethanol were considered the best diesel fuel additives because of their accessibility, low cost, high oxygen contents, and suitability for use in diesel engines without modifications (7). However, when these blends were later found to exhibit phase separation at certain compositions or as the temperature was reduced. Thereby forming methanol/ethanol-rich bottom phases that could stall engines and prevent normal operation; many investigations were initiated to address these problems. One possible solution is using additives that are capable of increasing the solubility, stability and CN of methanol/ethanol–diesel fuel blends. Oxygenated compounds (fuel additives) have been distinguished to increase the miscibility of methanol/ethanol in diesel fuel and can reduce the particulate emissions from internal combustion engines, and further decline of particulate emissions using these additives depends on the molecular configuration and amount of oxygen in the fuel. Because of their comparable fuel properties with diesel fuel and potential to be used in CI engines in pure form or blended with conventional diesel fuel. It is anticipated that alcohols will be able to slow down the worldwide petroleum consumption and decrease the level of dependence on petroleum fuels.

2.1 Usage of ethanol for IC engines

The use of alcohols as fuel in IC engines is not a new innovation. It has been used since the invention of IC engines. Alcohol may be blended with diesel or gasoline to produce diesohol and gasohol respectively or alcohol may be added to the intake of the engine.

Ethanol is a flammable polar solvent and is miscible with water. It has a vapor density of 1.59, which indicates that it is heavier than air. Consequently, ethanol vapors do not rise, similar to gasoline vapors. The specific gravity of ethanol is 0.79, which indicates that it is lighter than water; however, it is water-soluble. The auto-ignition temperature, boiling point, flash point, and melting point of ethanol are 423℃, 78℃, 13℃ and -114℃, respectively (8). Ethanol is less toxic than gasoline and methanol, and is not carcinogenic. Like gasoline, the major risk when using methanol as a motor fuel is its flammability. In pure form, ethanol burns without any smoke and it is difficult-to-see the blue flame. In denatured form there is little to no smoke, but a slight orange flame may be visible. Due to high octane number, ethanol is a good fuel for compression ignition (CI) engines.

Currently ethanol is one of the most popular alcohol-based fuel, as it is produced by the simple fermentation of natural sugars (sugar cane, sugar beet, molasses), starches (corn, wheat), or cellulosic biomass (corn stover, straw, grass, wood). The most common feedstock is sugar cane or sugar beet, and the second most common feedstock is corn starch. Figure 2.1shows the distribution of ethanol with time.

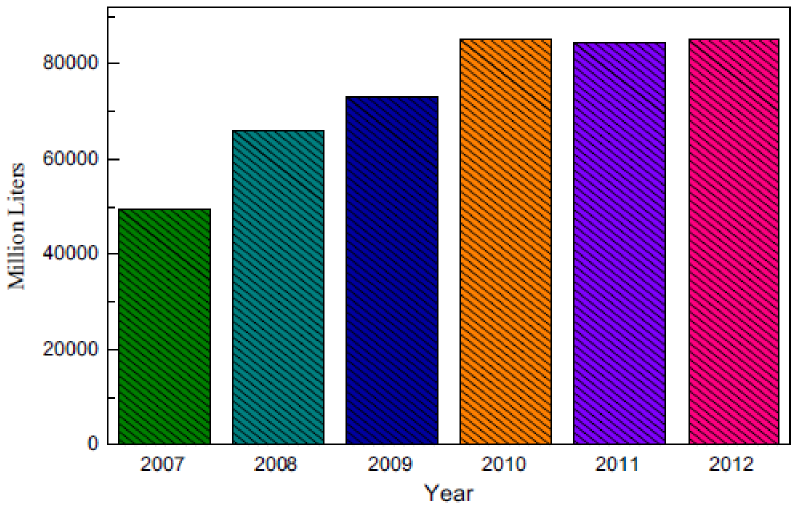

Figure 2.1 Distribution of ethanol with time

The use of lignocellulose biomass is very limited as it requires expensive pretreatment for converting the crystalline structure of cellulose to parent sugars. Although ethanol is produced from sugar beets and molasses, the use of gasohol as an alternative motor fuel has been steadily increasing worldwide for a number of reasons (9). Bioethanol can be derived from a wide range of carbohydrates of general formula (CH_2O) n. Sucrose is fermented using commercial yeast (Saccharomyces Cerevisiae). The chemical reaction comprises the enzymatic hydrolysis of sucrose followed by the fermentation of simple sugars [59–61].

Bioethanol currently accounts for more than 94% of global biofuel production. About 60% of global bioethanol production comes from sugarcane and 40% from other crops. The United States (54,580 million liters as of 2012) and South America (21,335 million liters as of 2012) are the world leaders, together accounting for about 89% of the world bioethanol production using sugarcane and corn, respectively. The USA production as of 2012 amounts to the world production in 2007 (49,625 million liters). The increasing production and use of ethanol as a fuel can decrease dependency on foreign oil, decrease trade deficits, create employment opportunities in rural regions, minimize air pollution, and address the global warming problem.

2.2 Effects of diesel alcohol blends on performances and emissions

The diesel engine is the most proficient power system among all known types of internal combustion engines. Heavy automobiles, urban motor vehicles, and engineering equipment are driven by diesel engines all over the world. Diesel automobiles are becoming more popular, and diesel engines are likely to become dominant power systems in future vehicles. Before that occurs, however, further advancement of diesel emissions controls is required as stricter global environmental protocols call for cutting-edge emission controls and near-zero diesel emission levels in the years to come (2).

In order to achieve diesel engine emission reduction many techniques have been employed. The blending of oxygenates such as alcohols with diesel fuel to provide more oxygen during combustion has been reported as one of the promising methods for controlling diesel engine emissions. There are numerous methods of alcohol diesel dual fuel operation. The ignition of alcohol in dual fuel operation is ensured by the high self-ignition of diesel fuel. The most common methods for achieving dual fuel operation are (I) alcohol–diesel fuel blends, (II) alcohol–diesel fuel emulsions, (III)dual injection, and (IV) alcohol fumigation. The simplest way to use alcohol in diesel engine is to be used in the form of blends. Numerous studies have been executed on CI engines to observe the engine performance and exhaust emissions using alcohol–diesel blends. Collectively these studies indicate that alcohol blends can improve some exhaust emissions without having any adverse effect on performance of diesel engine.

Due to the low miscibility of ethanol with diesel fuel, ethanol does not blend efficiently with diesel to form consistent solutions, which results in phase separation. The miscibility of ethanol in diesel relies on hydrocarbon amount and wax content of the base diesel, ethanol content (low ethanol concentration has reduced immiscibility), and the temperature of the diesel fuel (10). The reduction of diesel aromatic content decreases the miscibility of ethanol. The main advantage of ethanol–diesel blends is its consistency at low temperatures (up to 10–11°C). Anhydrous ethanol has good blending power to mix with diesel fuel and forms consistent solutions of up to 5% ethanol at warm ambient temperatures. However, ethanol has been found to be immiscible in diesel fuel at temperatures below 10°C, at which point the blend splits into two separate phases. This affects the fluidity and filterability of E-diesel in cold conditions.

2.3 Miscibility of ethanol with diesel

Solubility also relies on the amount of water present in ethanol. Being hygroscopic, ethanol easily absorbs water from ambient air and supply systems. Anhydrous ethanol is readily miscible with diesel fuel at ratios of 0–30%. Within these zones, mixture miscibility and cloudiness followed by the phase separation are observed when the water content of the ethanol surpassed 1% [14]. Therefore, low temperatures and water contamination decreases the E-diesel stability mainly due to phase separation. For stabilizing ethanol–diesel fuel blends especially in the context of large amounts of water, and to ensure fuel homogeneity under all temperature conditions, mixing a suitable amount of appropriate additives into E-diesel is mandatory (10). However, such additives may increase the cost of the fuel. In order to maintain the stability of E-diesel blends, two types of additive methodologies are normally employed: the addition of surfactants (emulsifiers) that yield stable emulsions or micro emulsions, and the addition of co-solvents that generate consistent blends. Co-solvents behave as connecting agents through molecular compatibility and bonding to form stable blends. The formulation of E-diesel blends with surfactants is more complex, and generally necessitates heating and rousing. A micro-emulsion consists of droplets or micelles of ethanol dispersed in the diesel fuel phase, and a minor quantity of emulsifier and water are necessary for its formulation. E-diesel preparations are usually micro-emulsions. A study from the1980s concentrated on emulsifiers, and the first E-diesel micro-emulsion was designated by (10), in which about 2% of a commercial surfactant was found to be desirable for each 5% of aqueous ethanol (5% water) added to diesel fuel. This combination formed an impulsive, crystal clear, and thermodynamically stable blend.

Combustion characteristics were measured at low and high load of 1500 and 1800rpms. Start of combustion of BDM5 and BDM10 were delayed than BD50 at low loads, but is almost identical at high engine loads. At low engine loads of 1500rpm, BDM5 and BDM10 show the similar peak cylinder pressure and peak of pressure rise rate to BD50, and higher peak of pressure rise rate to BD50, and higher peak of heat release rate than that of BD50. Residual gas temperature and wall temperature decrease with the increment of engine load because of lower charge temperature and at injection timing and length of ignition delay. Lower cetane number of methanol blends and higher heat of evaporation increase the ignition delay further. Lower viscosity of blended fuel which is positive to form more air fuel mixture and result in a larger percentage of fuel burned in the premixed phase causes higher peak cylinder pressure for BDM5 and BDM10 than BD50 which is shown by figure 2.2.

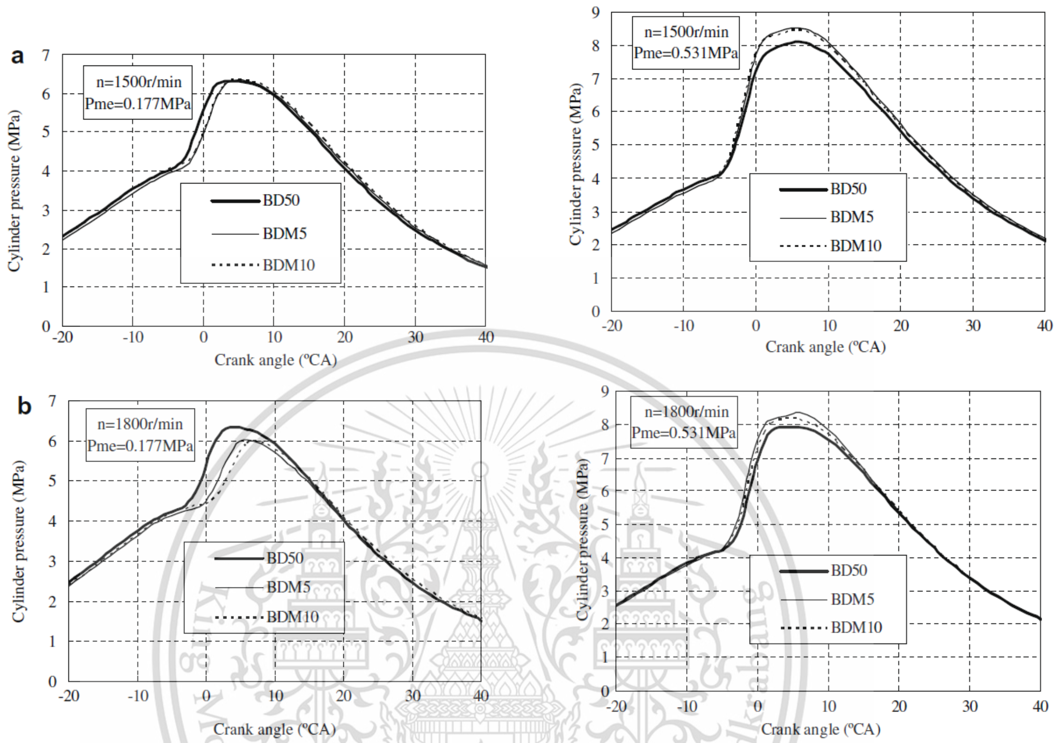

Figure 2.2 Cylinder Pressure with different engine speeds and loads

Reason for negative heat release rate at the beginning is evaporation of fuel accumulated during the ignition delay period. After combustion is initiated, this becomes positive.at low engine loads, although the combustion started later for BDM5 and BDM10, longer the ignition delay, better volatility and lower viscosity of methanol lead to larger amount of fuel accumulation in combustion chamber at the premixed burning phase, causing a higher heat release rate. The graphs of heat release rates of the experiment is shown by figure 2.3.

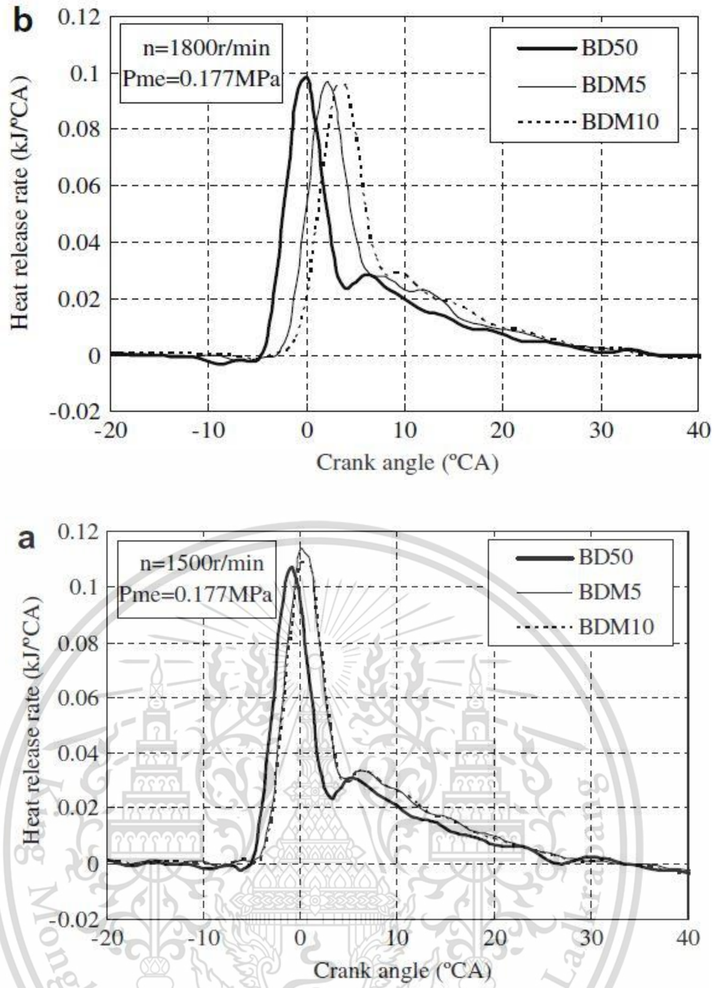

Figure 2.3 Heat Release Rate with engine speed and load

Power and torque output with increasing the rpm show that BDM5 nd BDM10 have lower ouput of power and torque comapre to BD5 by figure 2.4. denisty and lowewr clarifc values are lower by 2.25%4.34% than that BD50. So power and torque should be decreased because of the redueced energy content. BDM5 and BDM10 reduce CO emissions, comapred with BDM50 , especially at low speeds. High oxgen content of blended fuel may lead to complete combustion than BD50. HC emissions of BDM5 and BDM10 are almost similar to that of BD50.

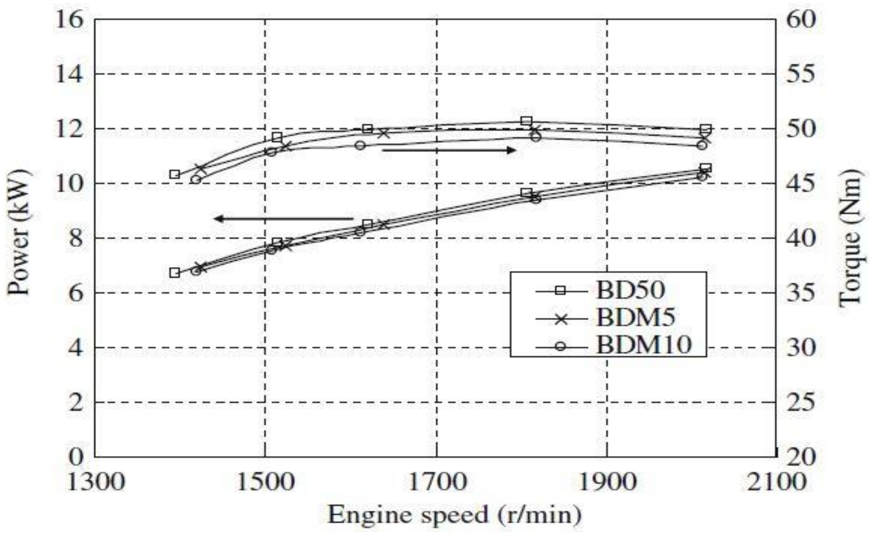

Figure 2.4 Power output for different percentages of ethanol blends

Higher oxygen content of alcohol blends helped to convert CO to CO_2 than diesel fuel. This effect would be expected to be most pronounced for methanol because of larger molecular weight of oxygen than in ethanol. Increasing of engine speed tended towards the volumetric efficiency so that CO emission decreased with increasing of engine speed is shown by figure 2.5.

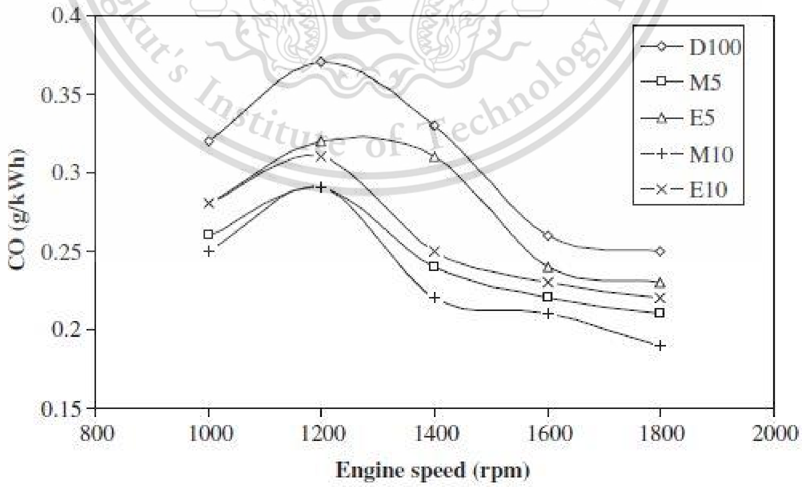

Figure 2.5 CO emission with different rate of ethanol blends

Addition of ethanol and methanol for diesel helped to oxidize THC due to high temperature in the cylinder to make the fuel easier to react with oxygen whih shown by figure 2.6. Alcohols had higher laminar flame speed compared to diesel fuel, hence higher laminar flame speed would reduce combustion duration and increase combustion temperature. Higher combustion temperature promoted more complete combustion and there was less THC emission for ethanol blends than diesel. Methanol showed lowest THC emission due to highest flame speed among fuels. When there was short combustion process at high speeds, the amount of THC would also decreased.

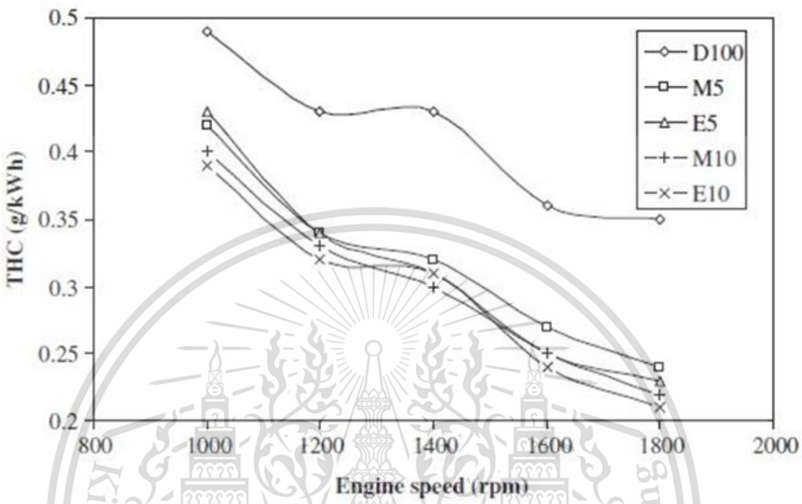

Figure 2.6 THC emissions with different ethanol blends

Cetane number and oxygen content are more effective with regard to increase peak in cylinder temperature. Therefore, NO_X formation increases as the alcohol content was increased in the fuel blend. NO_X concentration increases with increasing engine speed, but after the speed of the maximum torque of the engine its started to decrease.

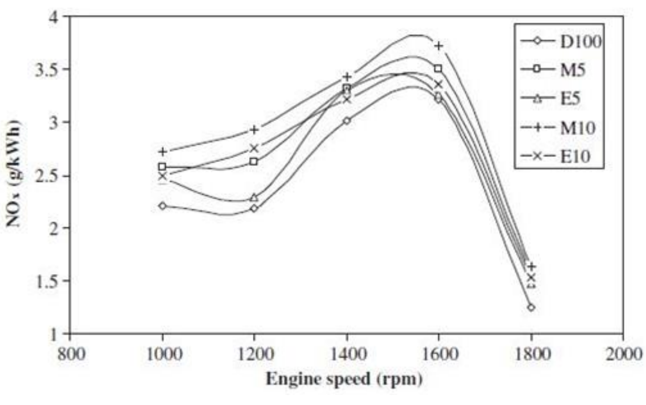

Figure 2.6 NOx emissions with different ethanol blends

Due to lower heating values of blended fuels they showed higher BSFC than diesel. Starting from the least BSFC point, increasing or decreasing speed increased BSFC due to reduction in the engine volumetric efficiency.

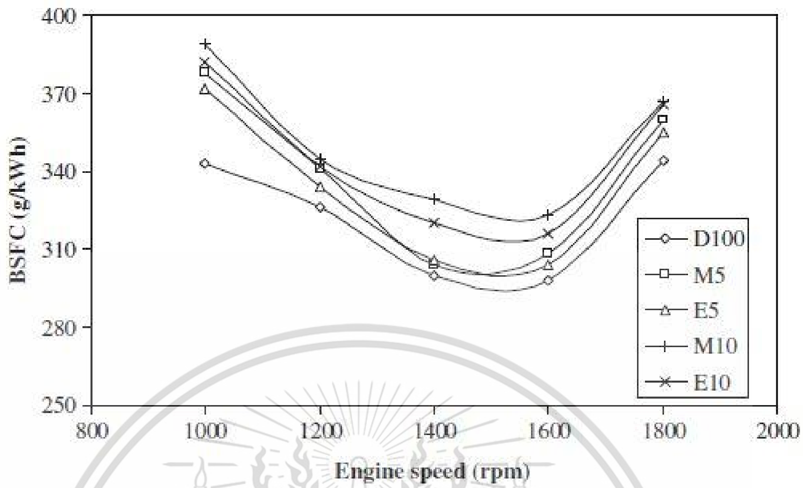

Figure 2.7 Result of BSFC with different ethanol blends

(13) In order to find out the usefulness of coconut and palm oil biodiesels as an alternative fuel, the fuel properties, the combustion characteristics and emissions were investigated. Therefore, the methyl esters of coconut, palm and rapeseed oil (CME, PME and RME) and ethyl ester of palm and rapeseed oils (PEE and REE) were processed and tested using diesel engine.

The properties of the fuels such as the density, kinematic viscosity, pour point, carbon hydrogen and oxygen contents of the test fuels were observed. Engine speed was fixed at 2000rpm and loads were applied from 0%, 25%, 50%, 75% and 100%. Ignition delay of CME was longer than RME and gas oil, but longer than that of PME. It was matching with the order of the cetane number of each of the fuel. Due to lowering of net calorific values of Easter fuels, more fuel was required for combustion process and thus longer fuel injection duration. Higher the initial heat release rate, rapidly emitted after ignition, was probably because of the greater amount of combustible fuel air mixture formed due to longer ignition delay.

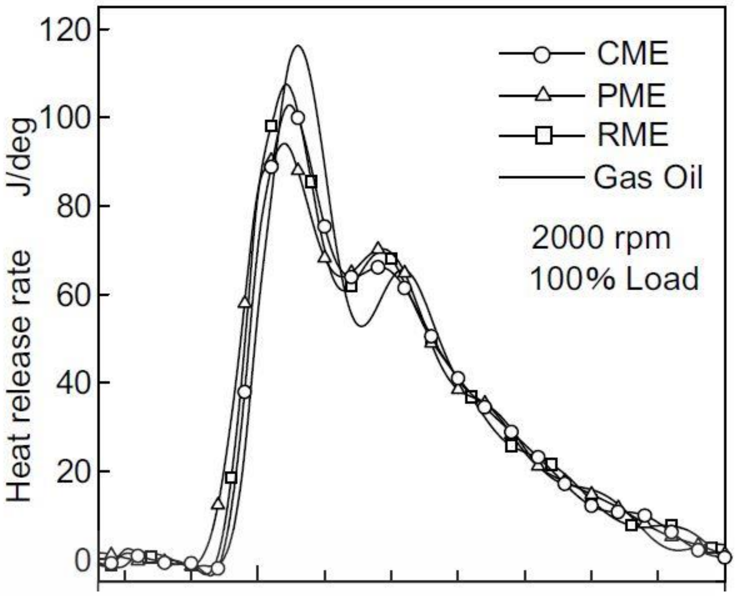

Figure 2.8 Heat Release Rate of Ester fuels

Oxgen in ester fuel supported for combsution process hence HC emission of CME, PME, and RME were lower than that of gas oil. Shorter igntion delay of ester fuel was another factor for less HC emission. The CO emission of CME is the lowest of all the test fuels, and PME was slightly lower and RME is slightly higher than gas oil. RME had less air fuel ratio which can be burn with less air reuqirment for combustion.hence it can observe that RME has lower air fuel ratio in the fuel comaprd to the gas oil, resulting in higher combustion temeprature and NOx emissions. The higher oxgen content in CME increased the present in fuel rich regions of the fuel spray, which caused for the reduction of soot.

(14) simulation was done using the commercial software named Diesel-RK used for this work was capable of predicting performance and combustion characteristics and emission by keeping constant speed and injection timing at 1500rpm and 23^{o} bTDC. The oxygen content of methanol blends improved the combustion efficiency and reduced heat losses in the cylinder due to lowering of flame temperature. Fuel vaporization still went on during compression stroke. So the fuel absorbs heat from the cylinder during the vaporization, the work required for compressing the air fuel mixture decreased and this enhanced the thermal efficiency of the engine.

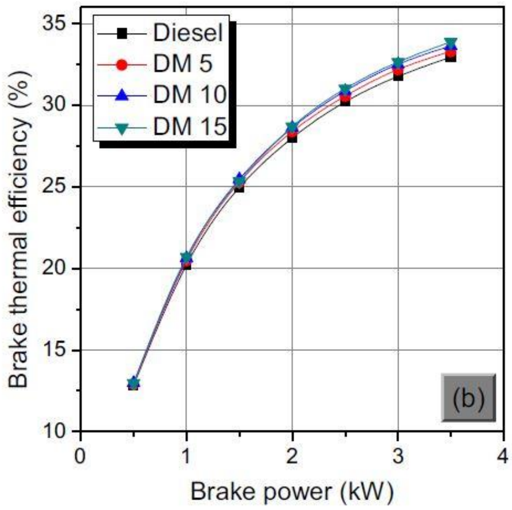

Figure 2.9 Brake Thermal Efficiency of ethanol and methanol blends

The addtion of ethanol and methanol ,caused the position where maximum pressure is reached to move further away from TDC. Starting of combustion was delayed due to low cetane number of blended fuels. lower flame temperature and lower in cylinder pressure rised with ethanol-diesel blended fuels, the in cylinder temperature rise was also low for ethanol – diesel blended fuel comapred to diesel.

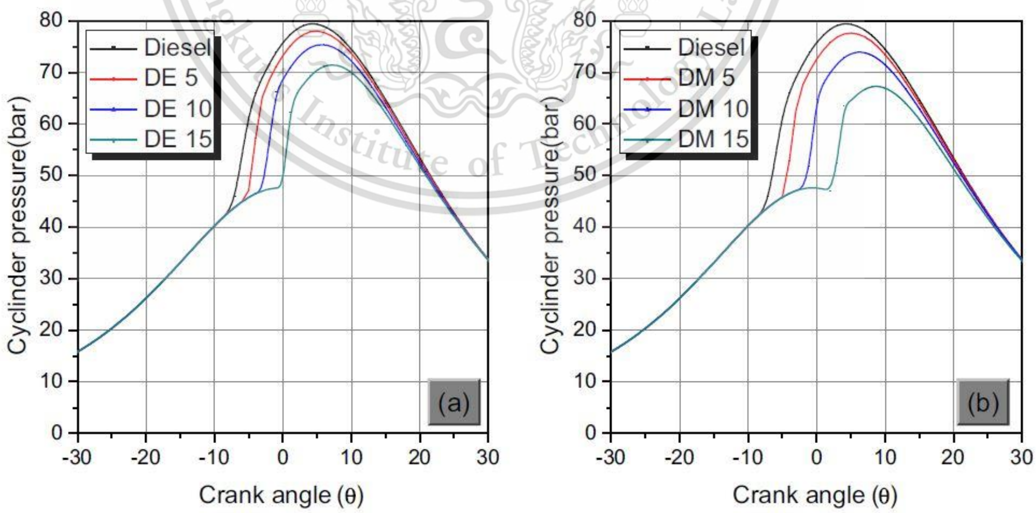

Figure 2.10 Cylinder Pressure of ethanol and methanol blends

Longer the ignition delay lead to longer mixing time of intake of charge, thus more fuel got combusted in the premixed zone resulting in higher heat release rate for ethanol-diesel and methanol-diesel blends. Due to higher oxygen content of blended fuel, it showed that heat release rate of diffusion phase was higher when compared with neat diesel. Increase in the ethanol mass fraction decreased cetane number resulting longer ignition delay and tended to increase of premixed combustion.

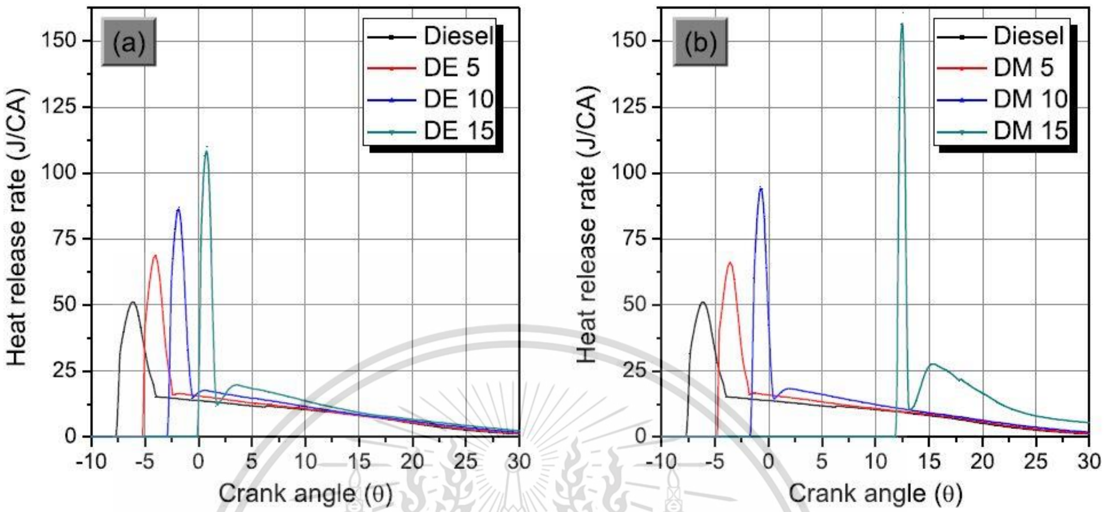

Figure 2.11 Heat Release Rate of ethanol and methanol blends

Higher viscosity and lower compressibility of fuels lead to a faster pressure rise in the fuel line and earlier start of ignition. Since fuel line pressure decreased with increasing engine load, start of ignition is delayed at higher load for all fuels. The injection of B10 started earlier by an average of 0.2^{o} CA compared to the other fuels for all engine loads due to the high viscosity of the higher alcohol butanol. Although higher alcohol butanol had higher cetane number than lower alcohols such as ethanol and methanol, B10 had higher ignition delay than E10 and M10. This was because butanol has higher kinematic viscosity and lower oxygen content than ethanol and methanol.

The addition of biodiesel to diesel fuel reduced CO emissions due to the oxygen content of biodiesel. As the oxygen content of the fuel increased even more with the addition of alcohol to the biodiesel blend, the alcohol blends led to further reductions in CO emissions than did for B20.

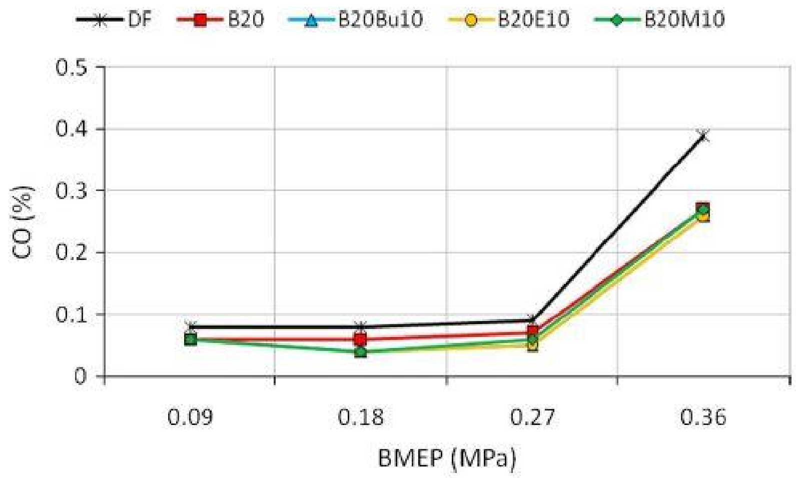

Figure 2.12 CO emission of B20 blends with ethanol

Higher the oxygen content of biodiesel, the HC emission of B20 was slightly higher than that of DF because of the high density and viscosity of the biodiesel deteriorated fuel atomization. In addition, a possible reason for biodiesel to have a higher HC emission than that did for diesel fuel might be the lower heating value and lower combustion temperature of biodiesel. When the heating value of the alcohols was lower. The combustion temperature decreased and HC emission increased. The addition of alcohol showed the cooling effect due to the high evaporation temperatures of the alcohols, which led to lower temperature inside the cylinder, resulting in incomplete combustion and high HC emissions.

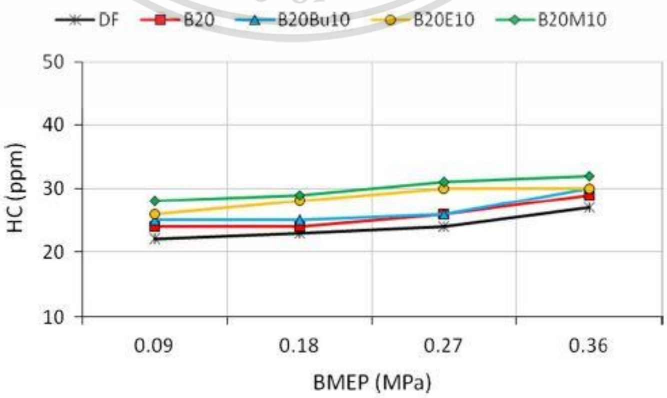

Figure 2.13 HC emission of B20 with ethanol

Addition of biodiesel and alcohol to diesel fuel caused a slight increment in NOx emissions because of their low cetane number and high oxygen content. A low cetane number resulted in a longer ignition delay which led to a higher cylinder pressure, a higher combustion temperature, and hence, higher NOx emissions due to the rapid combustion of the fuel accumulated in the premixed combustion phase. Results showed that the NOx emissions of all the test fuels increased in parallel with the increase in the fuel injection amount as the engine load increased up to 0.27MPa, but it decreased again at 0.36MPa. The reason for this reduction was low premixed combustion phase and the higher diffusion combustion phase at 0.36MPa. In this case, since peak pressures and temperatures did not increase as much as 0.27MPa, HRR max, and consequently, NOx emissions reduced.

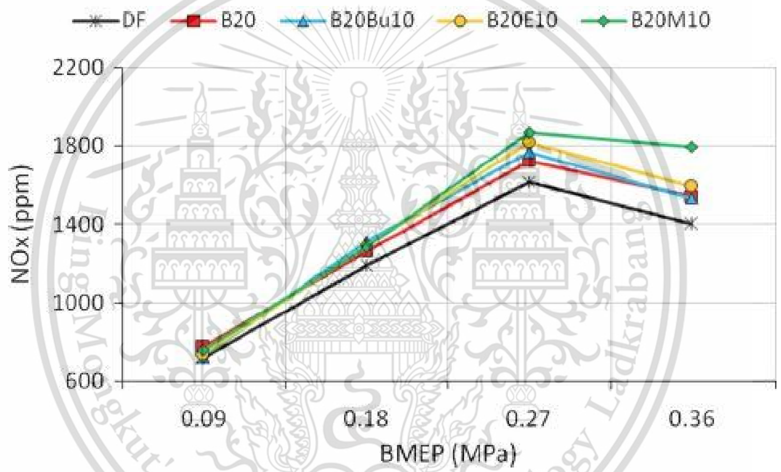

Figure 2.14 NOx emission of B20 with ethanol blends

(16) the alcohol blends obtained by mixing 10% of butanol, ethanol and methanol with diesel fuel (B10, E10, M10) were used to investigate combustion characteristics, performance and exhaust emission. BSFC increaseed with the addition of alcohols due to the decrease energy content. Heating value played important role in fuels. When using fuels with lower heating value such as alcohol blends, more fuel must be injected to achieve a similar power output. The cetane number of the alcohols was lower, the ignition delay period of the alcohol blends increased and BSFC increased. BTE is function of BSFC and heating value of a fuel. D100 had highest BTE because it had lowest BSFC

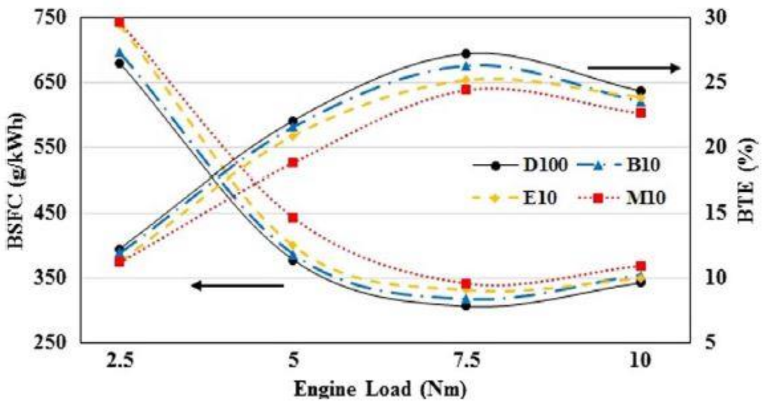

Figure 2.15 BSFC of B10 with ethanol and methanol blends

(17) Used virtual simulation to measure power of the engine by varying the rpm 1000-1500rpm by using different types of blended ethanol E0, E2.5, E5, E7.5 and E10 with diesel. It seemed that the engine power break was quite insensitive beyond 1400 rpm. On the contrary, below 1400 rpm the engine power break is sensitive. However, the engine power break of pure diesel (E0) is slightly lower than those of E2.5-E10, especially for speed above 1400 rpm. This phenomenon is also clearly presented in Fig 4b which discusses the rate of heat release (ROHR) at the 1480 rpm. This figure shows that rate of heat release E2.5-E10 higher than E0. Maximum engine power was measured at 1400 rpm as 11.15 kW for diesel fuel compare to 11.62 kW for E25. The result shows that maximum engine power for diesel engine is slightly lower the ethanol blended. The maximum power of E2.5 increased with four percentages. The combination of changes in the combustion process was contributed from the physical and chemical differences of fuel structure of ethanol and diesel fuel.

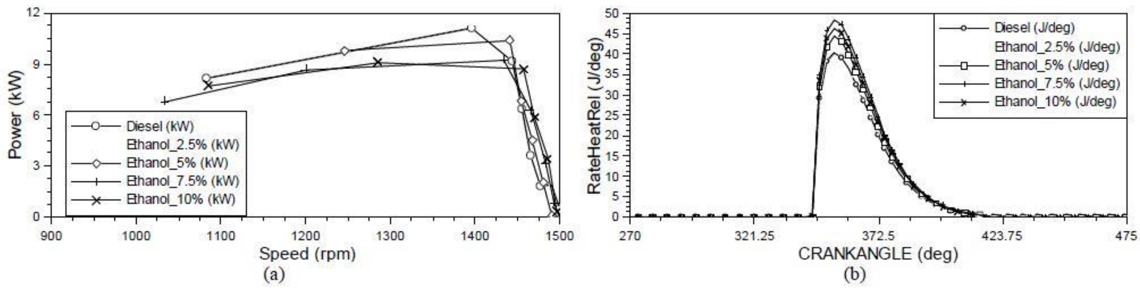

Figure 2.16 Simulated results of power and Rate of Heat Release

2.4 Ethanol Fumigation in CI engines

Comparison between ethanol fumigation and ethanol diesel fuel blends was done by (18) here performance have been measured and emissions of a single cylinder diesel engine have been investigated experimentally. Ethanol had the same effect for all the engine speeds and for Brake Thermal Efficiency results. There was slight gain in BTE with the increment of ethanol percentage which might be caused an increase in ignition delay. Rapid energy was released which reduced the heat loss from the engine as there was no enough time for this heat to leave the cylinder and transfer heat to the coolant. Density of air fuel mixture increased due to evaporation of ethanol which led to the cooling down of the intake temperature and more air was formed in the cylinder, a greater amount of power could be generated if the right proportion of fuel was added.

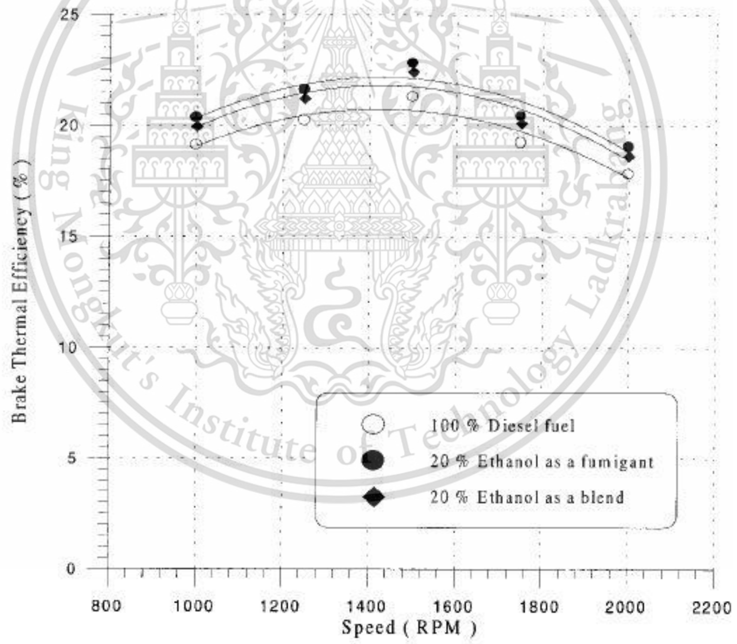

Figure 2.17 BTE of fumigation and blended ethanol

CO formation of fumigation was less than that for blends. Combustion temperature for fumigation might be higher than that for blends, better air utilization due to the availability of homogenous ethanol charge may have lowered the CO emissions. More turbulence in the cylinder and relatively high combustion temperatures compared with the lower speeds tended to cause higher CO emissions at high speeds.

The quench layer of unburned fumigated ethanol might have been caused for increment HC emissions. There was no quench layer with the diesel fuel injection because the combustion is droplet – diffusion- controlled and was completely surrounded by air.

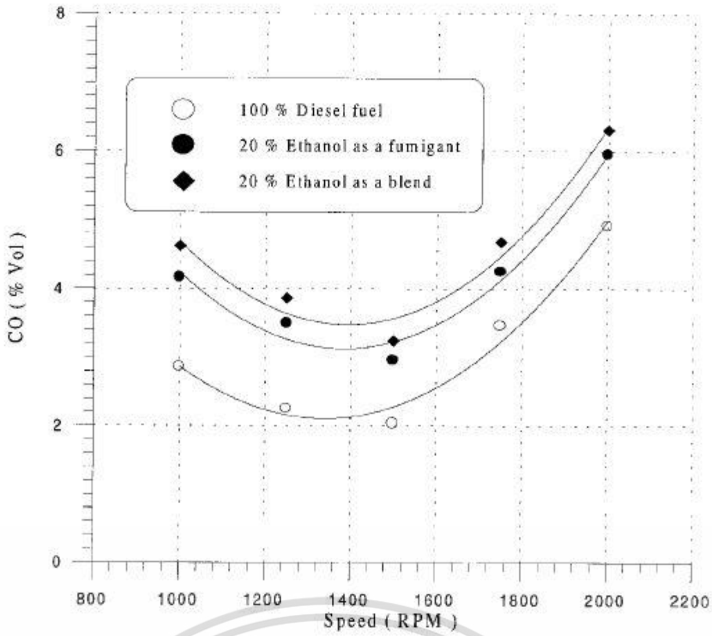

Figure 2.18 CO emission of fumigated and blended ethanol

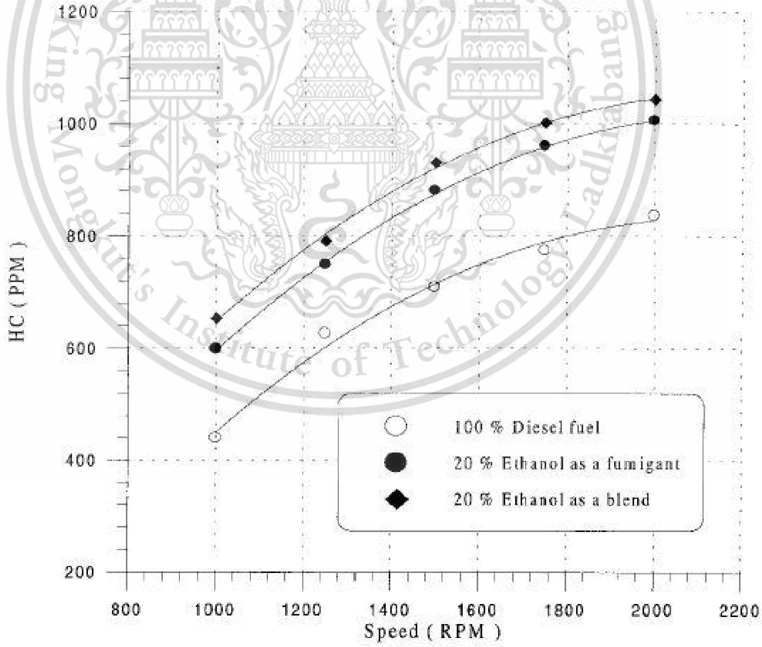

Figure 2.19 HC emission of fumigated and blended ethanol

The objective of (19) was to investigate the thermal performance, exhaust emissions and combustion behaviour of small capacity diesel engine using fumiagted ethanol. Fumigated ethanol at different flow rates were supplied to the cylinder during suction with the help of a simplifed low cost ethanol fuelling system.

Alcohol had high laminar flame propagation speed which cause to increase premixed combustion. The increase of premixed combsution due to enhanced combustion of homogeouns air fuel mixture at high engine loads increased BTE. Higher igntion delay resulted a greater heat release rate and reduced heat loses from the cylinder wall to coolant, which increased BTE.

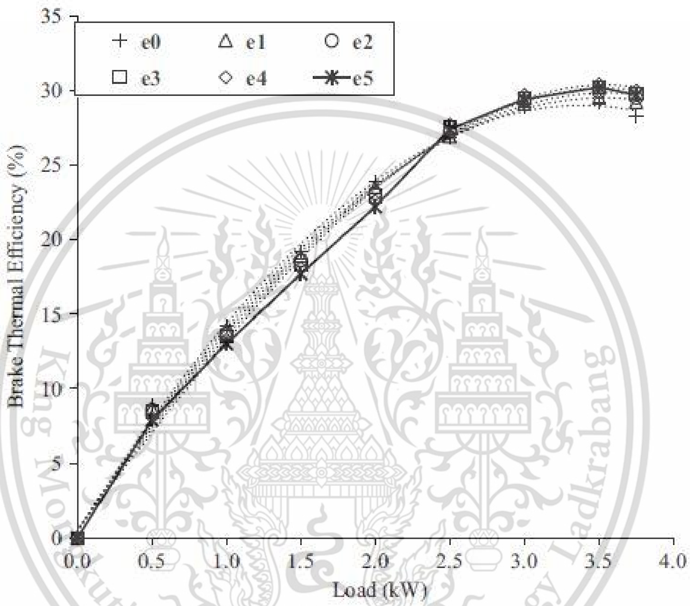

Figure 2.20 BTE for different ethanol fumigation rates

At high engine load, the increase in ignition delay with respect to the increment of the ethanol fraction and richer mixture tended to reduce the cooling effect. However, at high engine load, in the fumigation mode, there was a reduction in relative air/fuel ratio and the diesel fuel was now combusted in a mixture of air and alcohol that adversely affected the availability of oxygen for NOx formation and ultimately resulted in the reduction in NOx emission at high loads when compared to pure diesel.

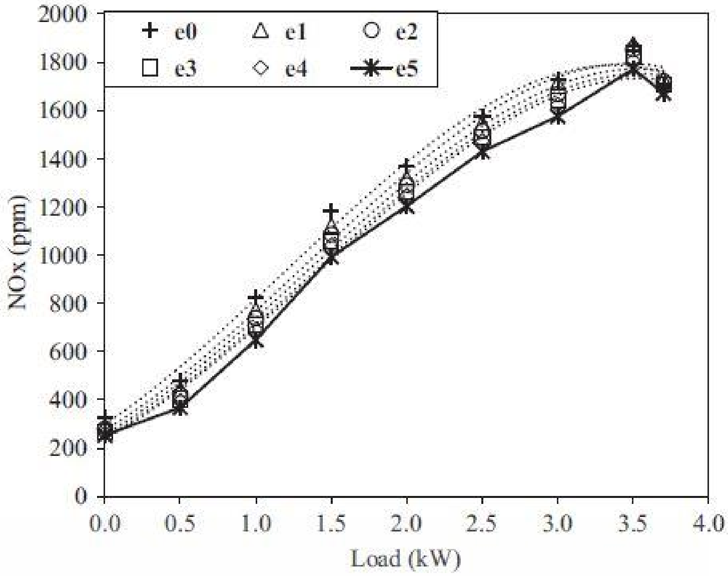

Figure 2.21 NOx emission for different ethanol fumigation rates

The value of peak cylinder pressure increased with the increment of ethanol fraction. The compression process in internal combustion engines has been considered as a polytrophic process, the value of pressure obtained by the compression of working substance of charge was higher in the case of ethanol fumigation as compared to that in diesel only operation.

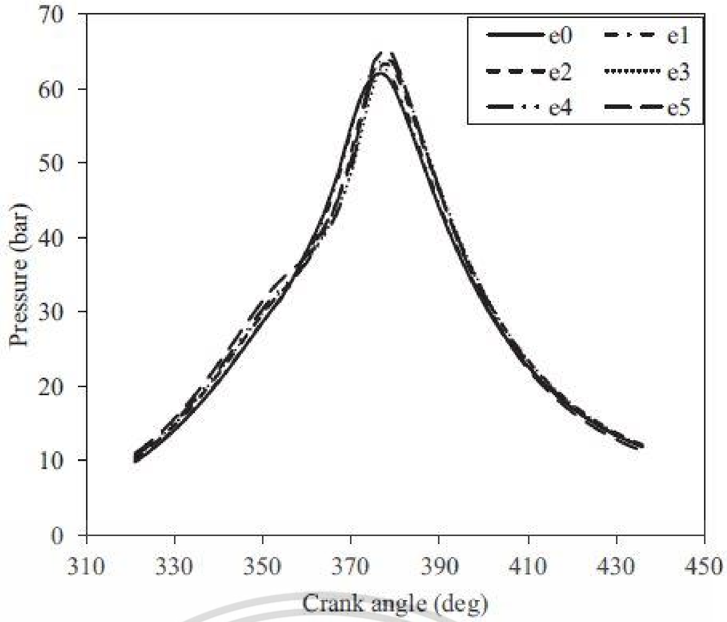

Figure 2.22 In Cylinder Pressure for different ethanol fumigation rates

Bioethanol was fumigated at four different flow rates to figure out performance and emission of a diesel engine. (20) Measured fuel consumption with different Global Equivalence Ratios(Φ). The fuel consumption was found to be lower with global equivalence ratio for bioethanol fumigation. This was due to the low calorific value of the fuel, and less amount of oxygen supplied. At Φ=0.88, Φ=0.9 and Φ=0.74, the ethanol fumigation of 0.48, 0.96, and 1.22kg/h had fuel consumption which was lowered by about 2.2, 4.5 and 20% respectively, then that of diesel. It was obvious that the ethanol fumigation replaced a certain amount of diesel.

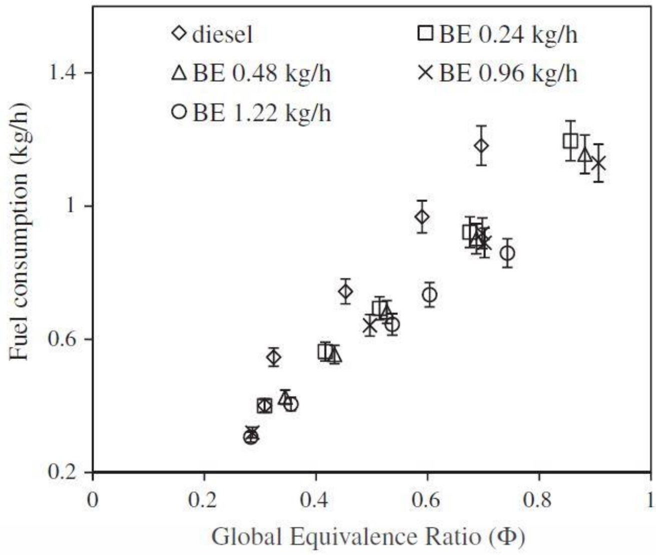

Figure 2.23 Fuel Consumption with Global Equivalence Ratio

CHAPTER 3

RESEARCH METHODOLOGY

3.1 Test Apparatus

3.1.1 Engine Apparatus

The research was done by using Kubota RT140 single cylinder direct injection engine. Movement of the piston is designed in horizontal direction to reduce the vibration of the engine and low down the center of gravity. Specifications of the engine is shown by table 3.1. The Kubota engine was powered by external common rail system for injecting high pressure fuel. The common rail pump was driven by commercial AC three phase motor and Toyota 1KD-FTV EDU (Engine Driver Unit) was used to drive the common rail injector. The Injector was used from Isuzu 4JJ-TC D-Max pickup truck for run the Kubota engine and engine head also had to modify to place the high pressure injector. 500bar of common rail pressure and 346 CA^{o} OF timing were used to do the experiment for all the fuels. Fumigation of ethanol was done by Denso 2350 injector with injection pressure of 3bar.

Table 3.1 Specification of Kubota RT140

Number of Cylinders	1
Bore and Stroke (mm)	97*96
Displacement (cc)	709cc
Max Output (kW)	10.3kW at 2400rpm
Max Torque (kg/m)	5kg/m at 1600rpm
Cooling Water Capacity (L)	2.1L
Fuel Tank Capacity (L)	11L
Combustion System	Direct Injection
Lubricating System	Forced Lubrication with Trochoid Pump
Air Cleaner Type	Wet Type
Direction of Revolution	Counter-clockwise Facing Fly Wheel
Dry Weight (kg)	116kg

Model HFM mass flowmeter and HFC mass flow controller of Teledyne Hasting Instruments (THI) were used to measure intake flow speed. Mass flowmeter was mounted in between the path of the buffer tank and the air filter. Actual mass flow rate was shown by THCD-100 power supply and the display which was connected with the air flowmeter. Specifications of the mass flowmeter is showed by 3.2 table

Table 3.2 Specification of mass flow rate

Accuracy	± 1.0% of F.S.
Repeatability	±0.05% of F.S.
Standard Operating Pressure High Pressure Option	500 psi 1000 psi
Operating Temperature	-10°C to 65°C
Standard Output Optional Output	0-5 VDC 4-20 mA
Power Requirements (±15 VDC) Power Requirements (±24 VDC)	± (14-16) VDC @ ±30mA (< 1 Watt) (14-32) VDC (<1.9 Watt)
Weight	0.66 kg

Figure 3.1 HFC mass flow controller of Teledyne Hasting Instruments (THI)

The front panel of THC consists of a 112×16 dot matrix VFD display with 6 membrane key switches below it. Each switch has legends on and above or below it to indicate its function under different conditions, although there are certain common functions to each screen.

Figure 3.2 THCD-100 Display

Flow meter was broken after using it for one year. Venturi meter was used to measure the airflow rate instead of THC. Venturi which was designed in accordance with the diameter of pipe lines of buffer tanks and rubber gasket was used to prevent air leakage. Pressure difference of venturi was measured by Kistler 4264A small, compact differential pressure transmitter.

Table 3.3 Specifications of Kistler 4264A pressure transmitter

Specifications	Unit	Type 4264AB … Bi-directional Differential						
Pressure range	bard	± 0,1	± 0,17	±0,2	±0,35	±0,5	± 0,7	± 1
	Options	Alternative pressure units available for mA and Voltage outputs: see table 3 for detail						
Proof pressure	bard	>3xFS pressure						
Burst pressure	bard	>5xFS pressure (Positive) / >4xFS pressure (Negative)						
Common mode (line pressure)	barg	14 max.						
Output		mV, V or mA						
Operating temperature range	°C	–55 … 125 (mV or V) / –55 … 100 (mA)						
Compensated temperature range	°C	–40 … 125 (mV or V) / –40 … 80 (mA)						
Accuracy (non-linearity, hysteresis, repeatability), BFSL	±%Span	0,2 (Positive and negative slope considered independently)						
Thermal effects (reference 25 °C)								
–10 … 50 °C	%Span	3,5	3,5	3,5	3,5	2	1	1
–40 … 125 °C (80 °C for mA version)	%Span	10	10	10	10	5	1,5	1,5
Long term stability (12 months)	%Span	±0,1						

Note: For special calibration, please call Kistler

Bernoulli theorem was applied to measure the air flow rate of the engine using venturi.

$$\Delta P = \frac{1}{2}\rho {v_a}^2[\left(\frac{A_a}{A_b}\right)^2 - 1]$$

Where ΔP is the difference pressure (Pa)

ρ is the fluid density, air = 1.2 kg/m^3

v_a is the fluid velocity (m/s)

A_a is the cross section area of the venturi (m^3)

A_b is the decreased cross section area of the venture (m^3)

Buffer Tank was installed to intake air flow path to reduce the back pressure of the engine which forms due to single cylinder geometry. 300L buffer tank was coupled right after air flow sensor and the air filter. A Steel frame was designed to mount the buffer tank to bear the weight of the tank and keep the tank and airflow sensor in linear with the air filter. The connection of the buffer tank and the intake manifold was formed by a flexible rubber tube, both edges were fixed by steel rings to have a better sealing to prevent the leakage of air.

Figure 3.3 Buffer Tank

The cooling system of engine was modified by adding external radiator with water pump to circulate the water effectively. When the engine with the original radiator was run in high speed conditions with load, it caused the engine water to be heated up quickly. Hence a large size external radiator and a water pump was used for a better water circulation of the engine. Original radiator was removed and put a steel plate with two tips for attaching rubber horses as water inlet and outlet lines.

Figure 3.4 Modified water intake and out lines

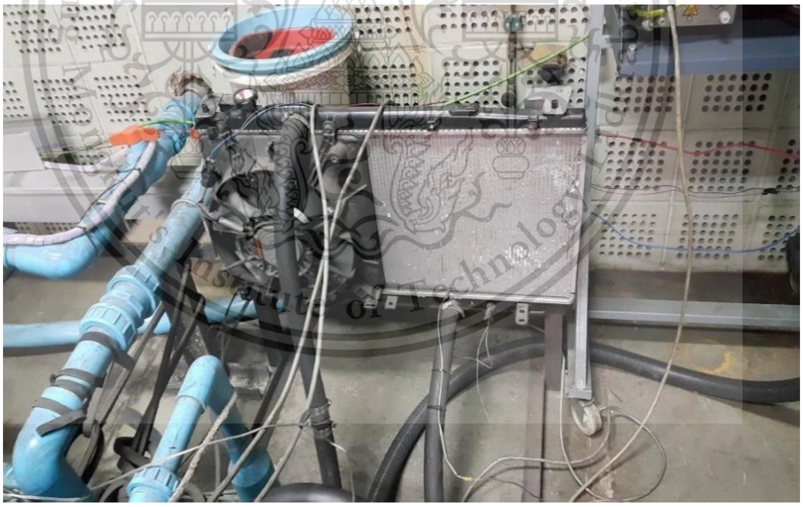

Figure 3.5 External Radiator System

K type thermocouples were used to measure intake air temperature, exhaust air temperature, water in- out temperature and engine oil temperature. Engine oil temperature should be reached around 55-60°C to start the getting of data of the engine by applying loads and the oil temperature should not be exceeded 75°C. Engine should be let to cool down when the oil temperature reaches around 70°C.

Prony brake dyno meter was used to measure the torque of the engine. The dyno meter was designed using Toyota Vios car front brake system. Engine load was measured by Minebe U3B1-220k-B load cell which was fixed on the base with maximum capacity of 1. 96kN.specifications of the load cell is shown by table no 3.5.

Table 3.4 Specifications of the load cell

Specification	
Rated capacity	1.961 kN
Safe overload	150 %R.C.
Ultimate overload	200 %R.C.
Rated output	3 mV/V±0.015 mV/V
Non-linearity	0.05 %R.O.
Hysteresis	0.05 %R.O.
Repeatability	0.03 %R.O.
Excitation, recommended	10 V or less
Excitation, maximum	15 V
Zero balance	±0.03 mV/V
Input resistance	350 Ω±3.5 Ω
Output resistance	350 Ω±5 Ω
Insulation resistance	2 000 MΩ or more (DC50 V) (between bridge and main body)
Temp. range, compensated	-10 ℃ ~ 70 ℃
Temp. range, safe	-20 ℃ ~ 80 ℃
Temp. effect on zero	0.05 %R.O./10 ℃
Temp. effect on output	0.1 %LOAD/10 ℃
Cable	φ6.3, 4-cores shielded 3 m cable 3 m directly attached, Y-crimp type terminal lugs are attached at cable end.
Class of protection	IP64
Material of element	U3B1-20K~200K-B : Aluminum alloy
	U3B1-500K~1T-B : Alloy steel
Durability	1 000 000 times with rated load applied.

Linkage between the brake disc and flywheel was coupled by a propeller shaft. Alignment of the shaft should be done before running of the engine to reduce the movement of tolerance of the shaft. Alignment is done by 3 points of the shafts namely the surface of contact with flywheel, some point at the middle of the shaft and a point on the surface of the shaft which is in contact with the disc. Tolerance was measured by a dial gauge and range of the tolerance should not be exceeded ±0.25.

Figure 3.6 Driving shaft

Brake master pump was used to apply the brake to the disc for applying loads to the engine. Master pump was pressurized by a specially designed manual lever. When the lever was being moved, master pump pulled pressurized brake fluid to grab the brake disc using brake pads. As an example if the needed rpm was 1500 at full throttle opened, lever was moved until the engine got the required rpm by applying brakes.

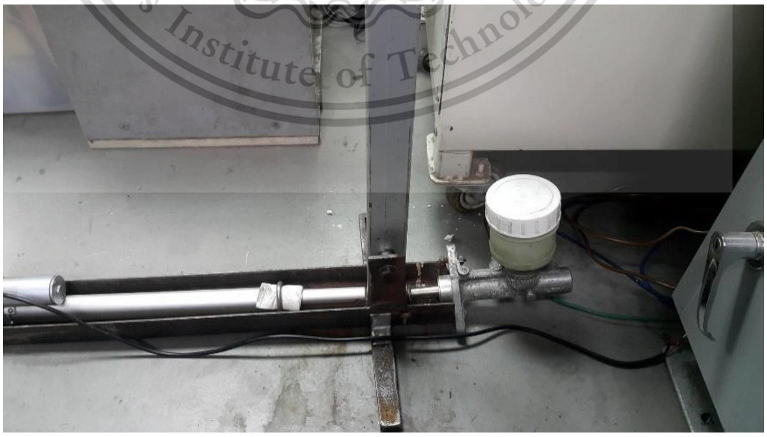

Figure 3.7 Break lever

3.1.2 Interface of controlling injection

All the data of thermocouples and the load cell were recorded by LabVIEW, data acquisition program. The data was obtained in 100Hz frequency and rate of sampling was 100 samples per seconds. National Instrument DAQ USB-6363 was used to transfer the data. Channels and interface of program is shown in figure 3.8 and 3.9.

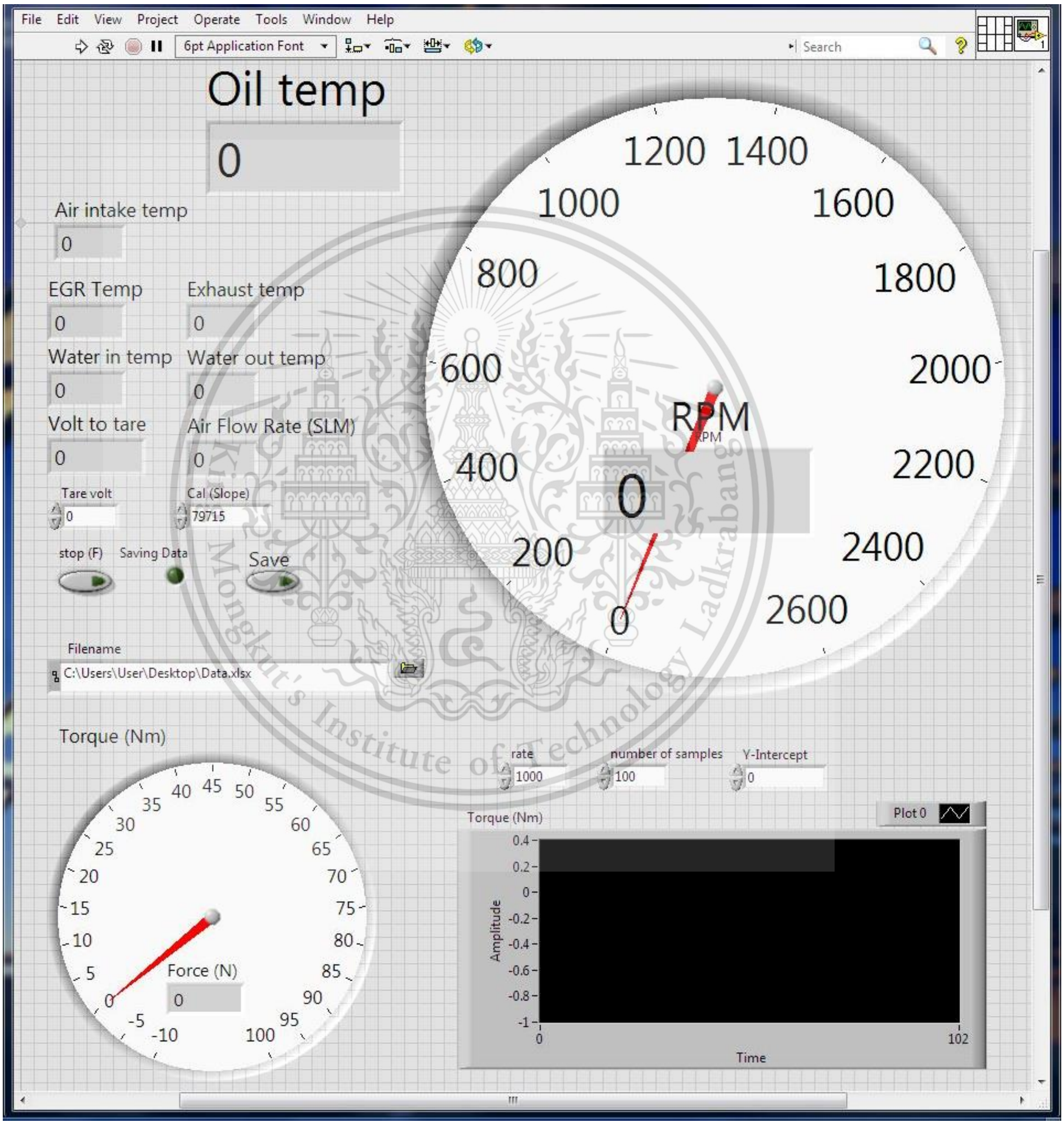

Figure 3.8 Labview Interface

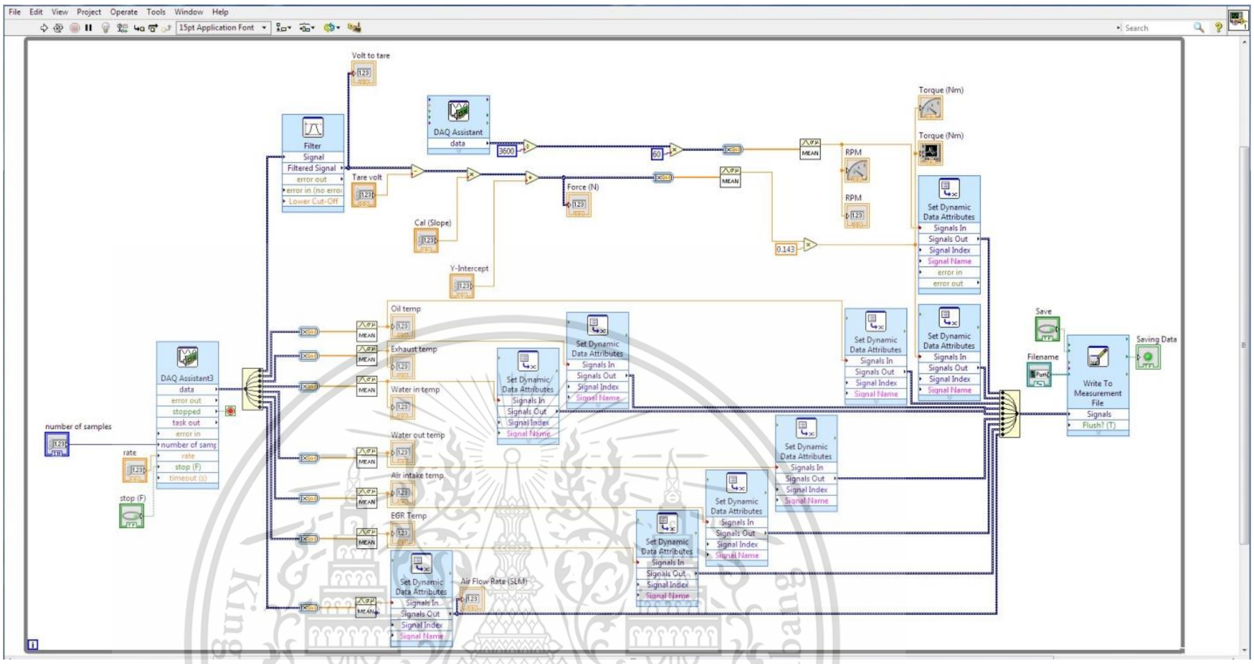

Figure 3.9 Labview Code Interface

3.1.3 Emission Apparatus

Horiba Mexa 1600D emission machine was used to measure emission of the engine. Laboratory-grade analytical emissions system was designed to measure raw exhaust gases from internal combustion engines. Single cabinet included PC-based flow controller and sampling system as well as analyzers for measuring CO, CO_2, THC, NOx, O_2 and (optionally) EGR-CO_2. System was tailored to support R&D work on gasoline and diesel-fueled engines were used in a variety of applications. Main important features of Horiba Mexa 1600D are

Self-contained system in minimal floor-space.

Easy-to-use PC-based flow controller with standard interfaces.

Bench is ISO-8178 complain.

Wide-range heated THC analyzer.

NOx analyzer is hot-dry/atmospheric type.

Minimum CO range is 100 ppm.

High performance: cost ratio due to standard global specification.

Table 3.5 Specifications of HORIBA

Model	**Component**	**Method**	**Range**	**Noise**
AIA-260	NDIR (250-mm cell)	100-3K ppm	3.5 s	
	CO-H	NDIR (10-mm cell)	1-10 vol%	±1.0 FS%
	CO_2	NDIR (10-mm cell)	1-16 vol%	±1.0 FS%
FCA-266	THC	Hot-FID	100-20K ppm	±1.0 FS%
	NO_x	CLD (atmospheric)	100-5K ppm	±1.0 FS%
IMA-262	O_2	MPD	10-25 vol%	±1.0 FS%
	EGR-CO_2	NDIR (10-mm cell)	1-10 vol%	±1.0 FS%

Figure 3.10 Horiba Mexa

The AVL Smoke Meter uses the filter paper method and measures the soot concentration of the exhaust gas of the engine. The variable sampling volume and thermal exhaust conditioning assures a wide applications range, e.g. measurements during engine development or DPF calibration. The AVL Smoke Meter is operated via the AVL Instrument Controller or an automation system. Furthermore, the AVL Smoke Meter is optionally equipped with an AVL in Port, a PC control software, which connects to an Ethernet interface and can be executed in Internet Explorer. Additionally, it can also be controlled with the optionally available AVL in Screen (desktop PC with touchscreen).

The AVL Smoke Meter uses the filter paper method to determine the soot concentration in the exhaust of diesel and GDI engines. A variable, but exactly defined sampling volume is sampled from the engine exhaust pipe and passed through clean filter paper inside the device. The filtered soot causes blackening of the filter paper, which is measured by a photoelectric measuring head and the result is analyzed by a microprocessor. The determined value is the Filter Smoke Number (FSN). The measurement of the AVL Smoke Meter has an excellent repeatability and reproducibility due to the possible manual or automatic adjustment of the sample volume. The proper preselection of the sample volume enables the measuring of low soot concentrations, e.g. on modern GDI engines, as well as of high soot concentrations with high exhaust back pressures and temperatures, e.g. before a diesel particulate filter.

Figure 3.11 AVL smoke Meter

Kistler 6052C piezoelectric pressure transducer coupled with a Kistler 5081 charge amplifier is used to measure in cylinder pressure. The sensor Type 6052C... is an excellent all-rounder. Its rugged construction makes it suitable for measurements at the knock limit as well as for thermodynamic investigations. This sensor is used mainly for complex cylinder head geometry. As well as for motor cycles and other small engines and for combustion analysis in vehicles. Autonics E40HB encoder was used to find the crack position of the cylinder. It could provide the pulse for 3600 pulses per one revolution.

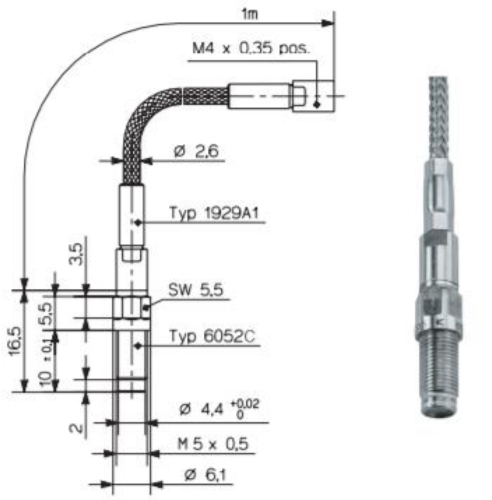

Figure 3.12 Kistler 6052C piezoelectric pressure transducer

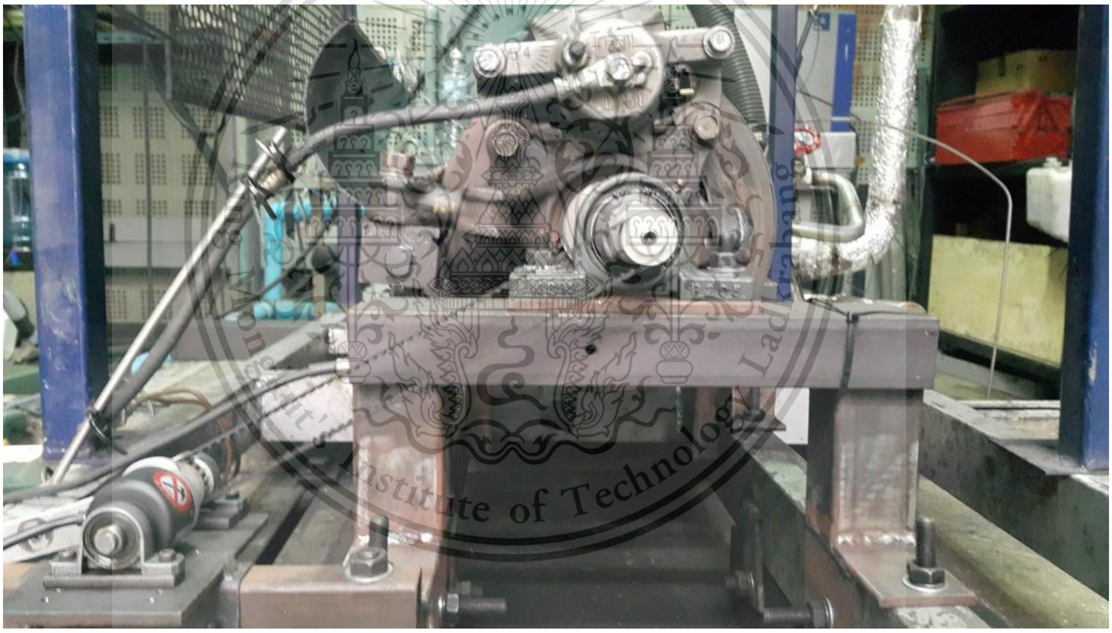

Figure 3.13 Autonics E40HB Encoder

Table 3.6 Specifications of Kistler Type 6052 C

Kistler Type 6052 C		
Measuring range	bar	0 … 250
Calibrated partial ranges	bar	0 … 50, 0 … 100, 0 … 150, 0 … 250
Sensitivity	pC/bar	≈ - 20
Natural Frequency (measuring element)	kHz	≈ 160
Linearity, all ranges (at 23 ˚C)	%/FSO	< ± 0.3
Acceleration sensitivity		
axial radial	bar/g bar/g	< 0.0002 < 0.0005
Operating temperature range Temperature min./max.	˚C	-20 … 350 -50 … 400
Sensitivity change		
200 ˚C ± 50 ˚C 230 … 350 ˚C	% %	< ± 0.5 < ± 2
Thermal shock error (at 1500 1/min, pmi = 9 bar) Δp (short time drift) Δpmi Δpmax	 bar % %	 < ± 0.5 < ± 2 < ± 1.0
Tightening torque	N.m	1.5
Capacitance, without cable	pF	5

3.1.4 Fuel properties apparatus

For this experiment various type of blended fuels were used. Properties of those fuel were figured out using different types of modern measuring equipment. Density of each fuel was measured by using Anton Paar equipment which is shown in figure 3.14

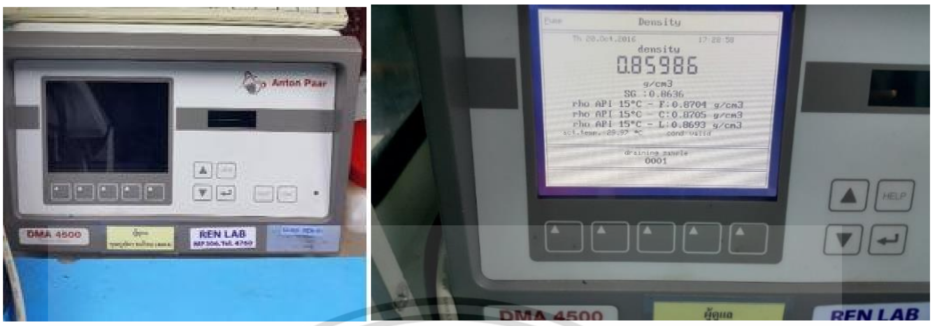

Figure 3.14 Anton Paar density measuring equipment

Cloud point is the temperature at which a cloud of crystals first appears with the body of the fuel due to the solidification of the heavier components of the fuel. Cloud point was also another important property used to compare blended fuels with diesel. Cloud point was measured using CPP 5Gs which took between 20-30 minutes to show the result. The test was done in triplicate for each sample and average value was taken for high accuracy.

Figure 3.15 CPP 5Gs Cloud Point and Pour Point measurer

Viscosity of fuel diesel bio diesel ethanol and blended fuel were measured using mini AV kinematic viscometer

Figure 3.16 miniAV Kinematic Viscometer

Heating value of fuels were measured using LECO AC-350 bomb calorimeter.

Figure 3.17 LECO AC-350 Bomb Calorimeter

3.2 Phase Separation of Fuels

Bio diesel was blended with pure diesel to form B7, B10 and B20. After that each fuel was blended with ethanol by volume of 5%, 10% and 15% to see the phase separation. All the fuel were kept 24 hours to check the separation and found out that B20 had the ability to be well miscible with all the portions of ethanol ratios. The ethanol blending ratio was increased up to 40% and even it also showed a better homogenous mixture.

Figure 3.18 Samples of phase separated blended fuels

Ethanol portion of 10%, 20%, 25%, 30% and 40% with B20 had been selected as blended fuels which were used for testing of the engine. These blended fuels were named as B20E10, B20E20, B20E25, B20E30 and B20E340. Properties of all the samples such as density, viscosity, cloud point and calorific value were measured using the above mentioned measuring equipment.

Measured values of density and calorific value of B20 and ethanol are shown by the following table.

Table 3.7 Properties of B20 and Ethanol

Fuel	Density (g/cm^3)	Calorific Value (MJ/kg)
B20	0.828	44.1
ethanol	0.781	26.9

3.3 Theoretical calculation of heating value

Using above results caloric value of blended fuels were calculated theoretically.

For B20E20

Assuming that volume the B20E20 mixture is 100cm³, and it composed of 80cm³ of B20 and 20cm³ of ethanol.

$$mass\ of\ B20 = density\ of\ B20 \times volume\ of\ B20$$

$$mass\ of\ B20\ in\ B20E20\ mixtue\ = 0.828 \times 80$$

$$mass\ of\ B20\ in\ B20E20\ mixture = 66.24g$$

$$mass\ of\ ethanol = density\ of\ ethanol \times volume\ of\ ethanol$$

$$mass\ of\ ethanol\ in\ B20E20\ mixture = 0.781 \times 26.9$$

$$mass\ of\ ethanol\ in\ B20E20\ mixture\ = 15.62g$$

$$total\ mass\ of\ the\ B20E20\ mixture = mass\ of\ B20 + mass\ of\ ethanol$$

$$total\ mass\ \ of\ the\ B20E20\ mixture = 66.24 + 15.62$$

$$total\ mass\ of\ the\ B20E20\ mixture = 81.86g$$

$$energy\ of\ B20 = (mass\ of\ B20 \times calorifc\ value\ of\ B20) \div 1000$$

$$energy\ of\ b20 = (66.24 \times 44.1) \div 1000$$

$$energy\ of\ b20 = 2.9211MJ$$

$$energy\ of\ ethanol = (mas\ of\ ethanol \times calorifc\ value\ of\ ethanol) \div 1000$$

$$energy\ of\ ethanol = (15.62 \times 26.9) \div 1000$$

$$energy\ of\ B20 = 0.420MJ$$

$$total\ energy\ of\ the\ B20E20\ mixture\ = (energy\ of\ B20 + energy\ of\ ethanol)$$

$$total\ energy = 3.341MJ$$

$$calorific\ value\ of\ B20E20 = (energy\ of\ B20E20 \times 1000) \div mass\ of\ B20E20$$

$$calorific\ value\ of\ B20E20 = 40.818MJ/kg$$

3.4 Injection Characteristics

This experiment was based on the comparison between blended fuel and fumigation of ethanol with bio diesel. Fuel consumption of the engine could be evaluated using the output of torque of the engine.

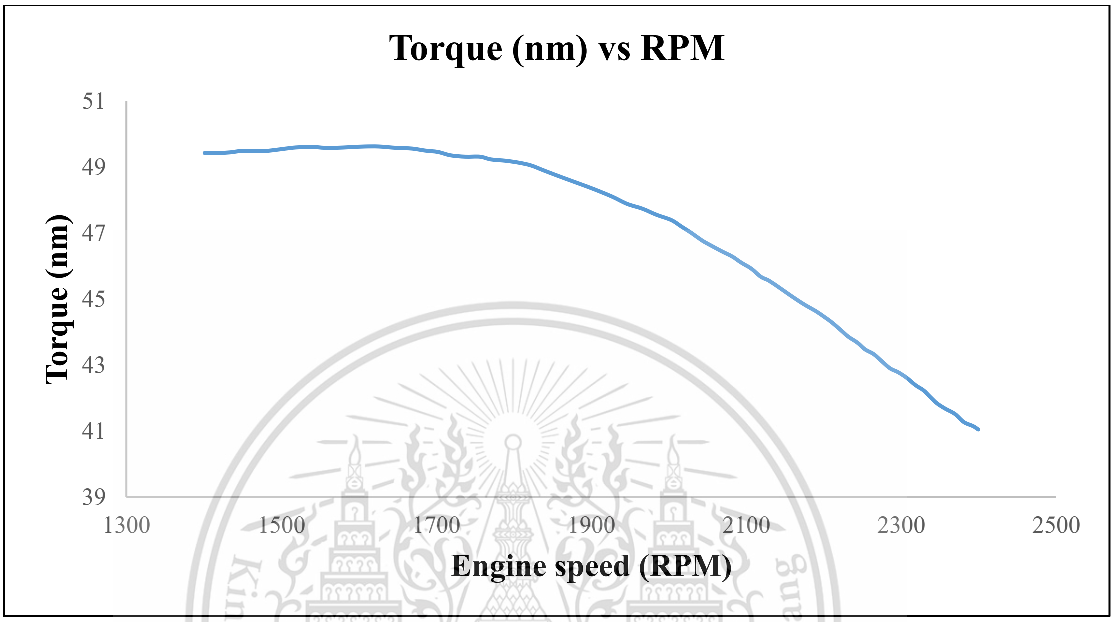

Figure 3.19 Samples of phase separated blended fuels

Power output of the engine varies according to the desired engine speed. Hence different engine speed gave different fuel consumption. The fuel consumption required for each engine speed could be converted to energy input for desired rpm. Duration of injection for each engine speed could be calculated according to these energy inputs. Same energy input for each engine speed should be provided to injectors even if the fuel type was different. Injection characteristics of each fuel were different due to different properties of blended fuels such as density, viscosity and calorific value.

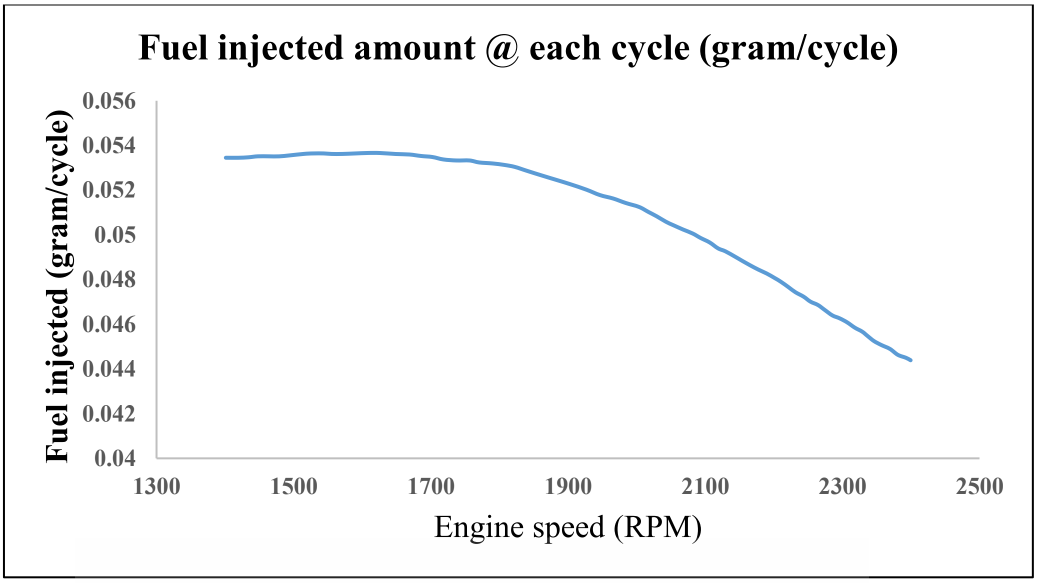

Figure 3.20 Consumption of fuel with engine speed

3.5 Injection durations for each fuel

Injection characteristics of B20, B20E20 and B20E40 were measured and both results had a linear relationship and then it was assumed that injection characteristics of other fuels too are linear.

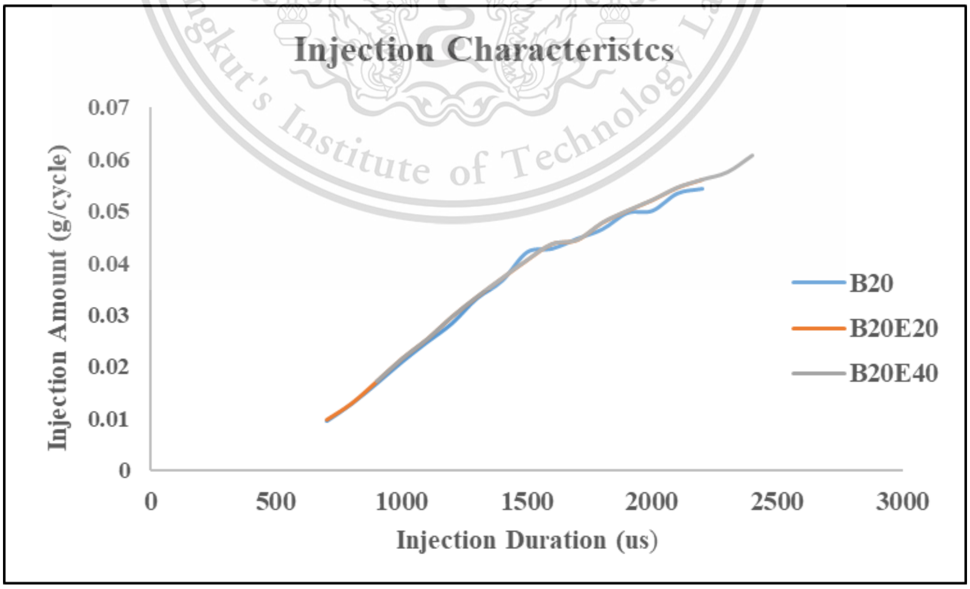

Figure 3.21 Injection Characteristics for various fuel blends

3.5.1 Injection duration calculation

Injection for desired engine speed were calculated using the injection characteristics results. Using fuel consumption of B7(g/cycle) for desired engine speed , it can be convert in to required energy input for desired engine speed using heating value of B7.

Example1: at 1400 rpm fuel consumption of B7 is 0.43355 g/cycle. Heating value of B7 is 44800kJ/kg. So required energy consumption at 1400rpm is

$$Energy\ input = \frac{Heating\ value\ of\ B7}{Fuel\ consumption\ at\ desired\ rpm}$$

$$Energy\ input\ at\ 1400RPM = \frac{44800kJ/kg}{0.43355g/cycle}$$

$$Energy\ input\ at\ 1400RPM = 1942.33088J/cycle$$

When need to find the duration of other type of fuel using same energy input of B7 for reasonable, it can be converted required fuel consumption or energy consumption of desired fuel type using its heating value.

Example 2; finding fuel consumption and energy requirement of B20 at 1400rpm using same energy input of B7. Heating value of B20 is 44100Kj/kg.

$$Fuel\ consumption\ of\ B20\ at\ 1400rpm = \frac{Energy\ input\ of\ B7\ at\ 1400rpm}{Heating\ value\ of\ B20}$$

$$Fuel\ consumption\ of\ B20\ at\ 1400rpm = \frac{1942.33088J/cycle}{44100kJ/kg}$$

$$Fuel\ consumption\ of\ B20\ at\ 1400rpm = 0.044043\ \text{g/cycle}$$

Using injection characteristics results of B20 and fuel consumption values , linear interpolation was used to calculate required duration for desired engine speed.

Table 3.8 Injection Duration of B20

RPM	B7 g/cycle	B 7J/cycle	B20 g/cycle	Duration
1400	0.0433556	1942.33088	0.044043784	1639.0314
1600	0.03853483	1726.360384	0.039146494	1457.3781
1800	0.03253376	1457.512448	0.033050169	1286.4951
2000	0.02787166	1248.650368	0.028314067	1167.8817

Table 3.9 Injection Duration of B20E10

RPM	B20 g/cycle	B20 J/cycle	B20E10 g/cycle	Duration
1400	0.0440438	1942.3309	0.04573634	1737.199
1600	0.0391465	1726.3604	0.040650852	1500.54
1800	0.0330502	1457.5124	0.034320252	1320.948
2000	0.0283141	1248.6504	0.029402147	1192.953

Table 3.10 Injection Duration of B20E20

RPM	B20 g/cycle	B20 J/cycle	B20E20 g/cycle	Duration
1400	0.044044	1942.331	0.047585	1792.735
1600	0.039146	1726.36	0.042294	1553.225
1800	0.03305	1457.512	0.035708	1359.787
2000	0.028314	1248.65	0.030591	1222.844

Table 3.11 Injection Duration of B20E25

RPM	B20 g/cycle	B20 J/cycle	B20E25 g/cycle	Duration
1400	0.044044	1942.331	0.048576	1832.445
1600	0.039146	1726.36	0.043175	1581.475
1800	0.03305	1457.512	0.036451	1380.613
2000	0.028314	1248.65	0.031228	1239.337

Table 3.12 Injection Duration of B20E30

RPM	B20 g/cycle	B20 J/cycle	B20E30 g/cycle	Duration
1400	0.044044	1942.331	0.049615	1877.406
1600	0.039146	1726.36	0.044098	1646.349
1800	0.03305	1457.512	0.037231	1402.488
2000	0.028314	1248.65	0.031896	1256.616

Table 3.13 Injection Duration of B20E40

RPM	B20 g/cycle	B20 J/cycle	B20E40 g/cycle	duration
1400	0.044043784	1942.331	0.051852	1983.383
1600	0.039146494	1726.36	0.046087	1747.722
1800	0.033050169	1457.512	0.03891	1450.589
2000	0.028314067	1248.65	0.033334	1293.835

Ethanol fumigation were done 12%, 16%, 20% and 28% by energy to compare with blended fuels B20E20, B20E25, B20E30 and B20E40. Same energy input was supplied by calculating for fumigation ethanol and B20 direct injection with respective to desired energy input of desired blended fuel. Injection characteristics of ethanol injector also was tested to obtain appropriate energy input for each engine speed. To obtain required energy input of fumigation method, energy input is divided according the fumigation rate of ethanol. If 12% ethanol fumigation is required, 12% of energy input of B7 is used to calculate desired ethanol consumption using heating value of ethanol and injection characteristics data is used to calculate duration of ethanol for desired engine speed. 88% of energy input of B7 is sued to calculate consumption and duration of B20 using its heating value and injection characteristics.

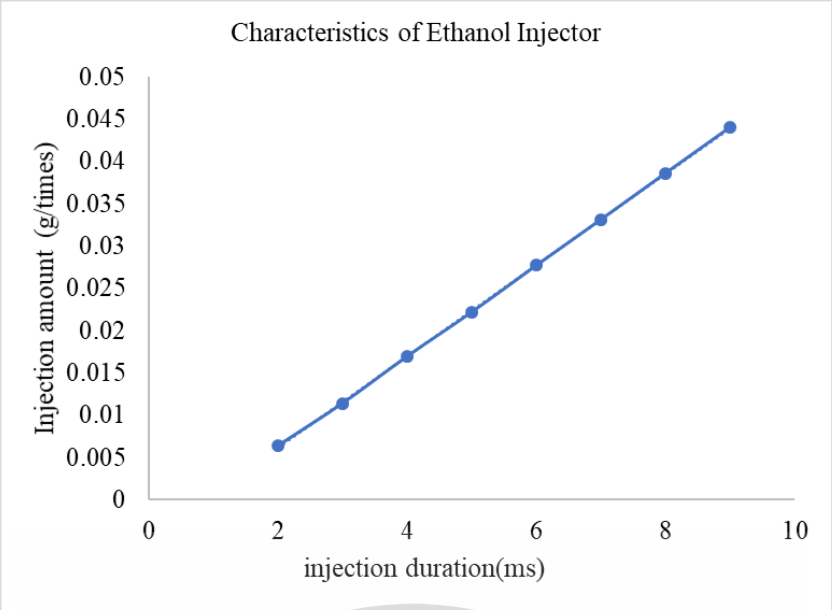

Figure 3.22 Injection Characteristics of Ethanol Injector

Table 3.14 Injection Duration of 12% ethanol fumigation

12% ethanol fumigation			
rpm	B7 J/cycle	B20 g/cycle	Duration
1400	1709.251	0.038759	1446.262
1600	1519.197	0.034449	1322.694
1800	1282.611	0.029084	1185.626
2000	1098.812	0.024916	1089.594
rpm	B7 J/cycle	Ethanol g/cycle	Duration
1400	233.0797	0.008665	1149.82
1600	207.1632	0.007701	1098.1
1800	174.9015	0.006502	1047.195
2000	149.838	0.00557	1000

Table 3.15 Injection duration of 16% ethanol fumigation

16% ethanol fumigation			
rpm	B7 J/cycle	B20 g/cycle	duration
1400	1631.558	0.036997	1395.782
1600	1450.143	0.032883	1282.17
1800	1224.31	0.027762	1155.165
2000	1048.866	0.023784	1057.642
rpm	B7 J/cycle	Ethanol g/cycle	duration
1400	310.7729	0.011553	1433.984
1600	276.2177	0.010268	1247.19
1800	233.202	0.008669	1150.066
2000	199.7841	0.007427	1082.696

Table 3.16 Injection duration of 20% ethanol fumigation

20% ethanol fumigation			
rpm	B7 J/cycle	B20 g/cycle	duration
1400	1553.865	0.035235	1346.557
1600	1381.088	0.031317	1241.646
1800	1166.01	0.02644	1124.704
2000	998.9203	0.022651	1027.36
rpm	B7 J/cycle	Ethanol g/cycle	duration
1400	388.4662	0.014441	1919.706
1600	345.2721	0.012835	1645.602
1800	291.5025	0.010837	1327.253
2000	249.7301	0.009284	1190.186

Table 3.17 Injection duration of 28% ethanol fumigation

28% ethanol fumigation			
rpm	B7 J/cycle	B20 g/cycle	duration
1400	1398.478	0.031712	1251.851
1600	1242.979	0.028185	1164.919
1800	1049.409	0.023796	1057.971
2000	899.0283	0.020386	971.3986
rpm	B7 J/cycle	Ethanol g/cycle	duration
1400	543.8526	0.020218	3011.478
1600	483.3809	0.01797	2575.058
1800	408.1035	0.015171	2031.676
2000	349.6221	0.012997	1673.652

CHAPTER 4

RESULTS AND DISCUSSIONS

The experiment was done in the range of 1400-200rpm with an increment of 100 in full load condition. Injection Pressure of common rail injector was kept at 500bar and injection timing was at 346ºbtdc for all the experimental conditions of performances and emissions. Combustion Characteristics results have been measured at 1600rpm with high load of 30Nm for the blended fuel types and fumigation rates.

Explanations of results of the experiment are described by the following paragraphs. The behavior of the results when using blended fuel has been explained first. Performances, emissions and combustion characteristics of different diesohol fuels have been compared.

The behavior for the results of different fumigation rates have been discussed secondly. Like for diesohol, variation of performances, emissions and combustion characteristics have been explained in that part. The comparison between diesohol and fumigation have been done finally. The comparison was done choosing the blended fuels and fumigation rates which had the same energy input for a much reasonable comparison.

Diesohol fuels were labeled according the energy input of ethanol volume in the mixture for further explanations. Due to that B20E10, B20E20, B20E25, B20E30 and B20E40 which had been named by volume, was labeled as B20E6, B20E12, B20E16, B20E20 and B20E28 according to energy constriction of ethanol of the mixture.

4.1 Diesohol method

4.1.1 Performances

The effects of the ethanol concentrations in the blended fuels are shown in Figure 4.1. At all engine speed, increasing the ethanol fraction in diesohol fuel decreased the engine torque due to the lower calorific values. B20, having the highest calorific value, yielded the highest torque while B20E28 resulted in the lowest torque due to its lowest calorific. The results of torque has a clear trend of reducing torque with engine speed. Higher engine speeds, due to the frictional loss and shortening the time of intake stroke, the engine cylinder cannot be fully charged, which causes a reduction of the engine volumetric efficiency and the engine torque. The reduction of torque with compare to B20 at 1600rpm was 8.05%, 10.56%, 14.9%, 17.875 and 19.84%.

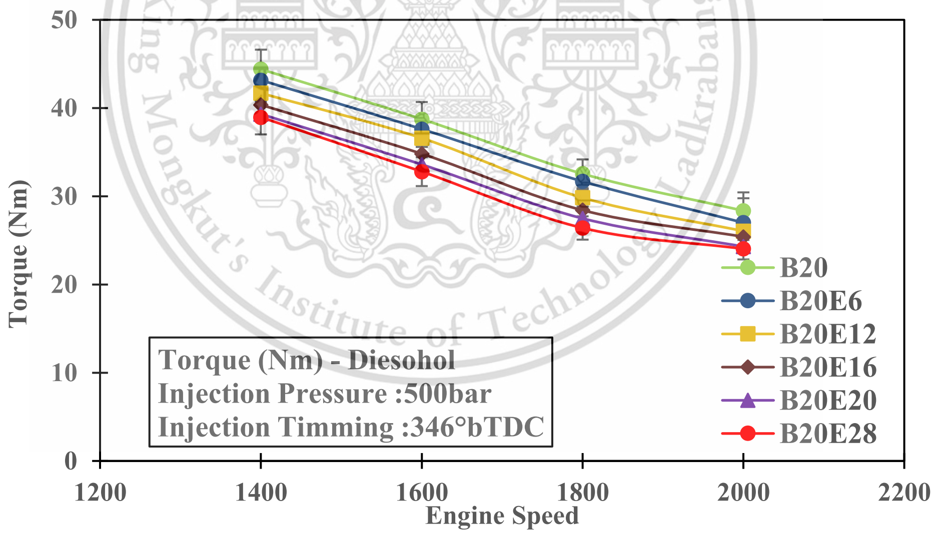

Figure 4.1 Variation of torque with engine speed for diesohol

Figure 4.2 shows the engine power when using diesohol as the fuel at any speed. Similar the results of torque when increasing the ethanol concentration. The power was decreased for all the engine speeds. Increasing the ethanol fraction of the blended fuel causes the lowering of the calorific value which resulted in had caused a reduction in the power. The reduction of heating values of diesohol fuels compare to B20 was by 3.70%, 7.40%, 9.30% , 12.20%, and 15.00%.Reduction of power at 1800rpm when compared to B20 was by 3.28%, 5.84%, 10.3%, 13.23% and 16.66%.

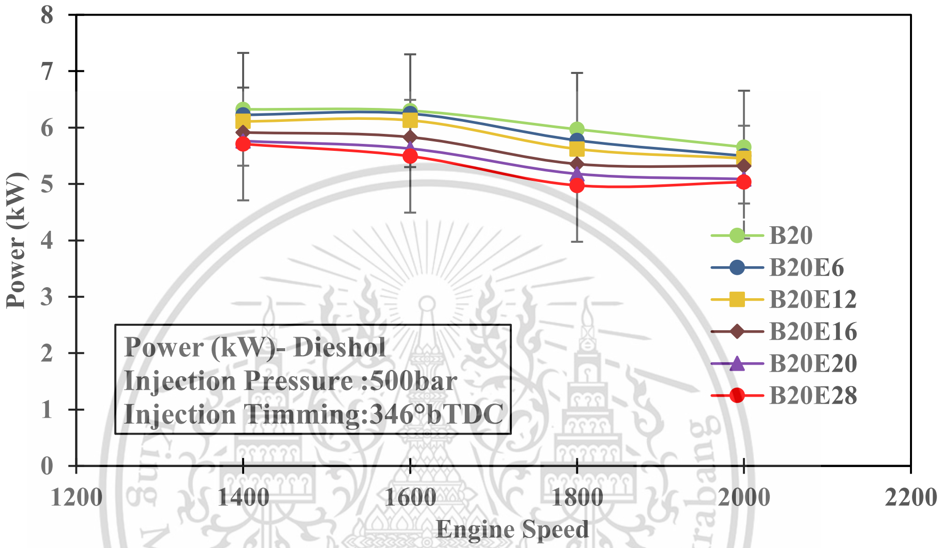

Figure 4.2 Variation of Power with engine speed for diesohol

Brake thermal efficiency (BTE) is defined as the power output divided by the energy. Break thermal efficiency is expressed by the following equation

$$BTE = \frac{P_b}{(q_{m,d} \times LHD_d) + (q_{m,e} \times LHD_e)}$$

P_b = Brake Power

$q_{m,d}$= mass flow rate of diesel or bio diesel

$q_{m,e}$ = mass flow rate of ethanol

LHD_d= latent heat of vaporization of diesel or biodiesel

LHD_e= latent heat of vaporization of ethanol

BTE when using diesohol is shown in figure 4.3. B20 shows the highest brake thermal efficiency when compared to blended fuels. Increasing the ethanol fraction of 10%, 20%, 25%, 30% and 40% by energy had caused to decrease the BTE in 13.31%, 20.00%, 23.8%, 26.3%and 41.26% at 1800rpm with respective to B20. BTE when using B20 shows almost the constant value for all the engine speeds. But BTE when using diesohol was reduced until 1800 rpm and slightly increased afterward.

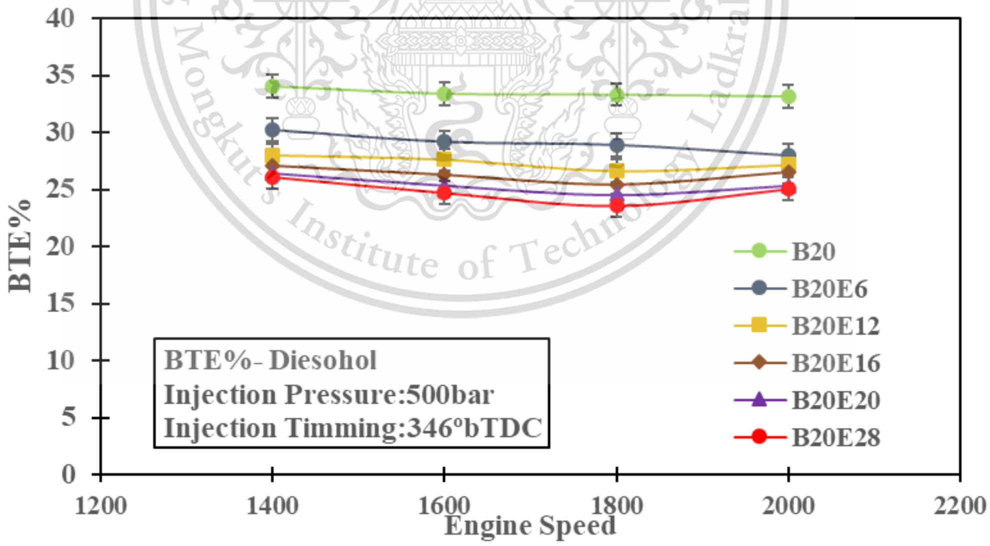

Figure 4.3 Variation of BTE with engine speed for diesohol

Brake specific energy consumption was increased when increasing the portion of ethanol in diesohol fuels which are shown in figure 4.4. Increasing the ethanol amount portion causes to reduce lower down the calorific value of each fuel. B20E40 had the highest portion of ethanol thus it had the lowest calorific value compared to other blended fuel and B20E28 gave highest BSEC. Highest BSEC gave at 1800rpm for all the blended fuels. B20E28 yielded the highest BSEC.

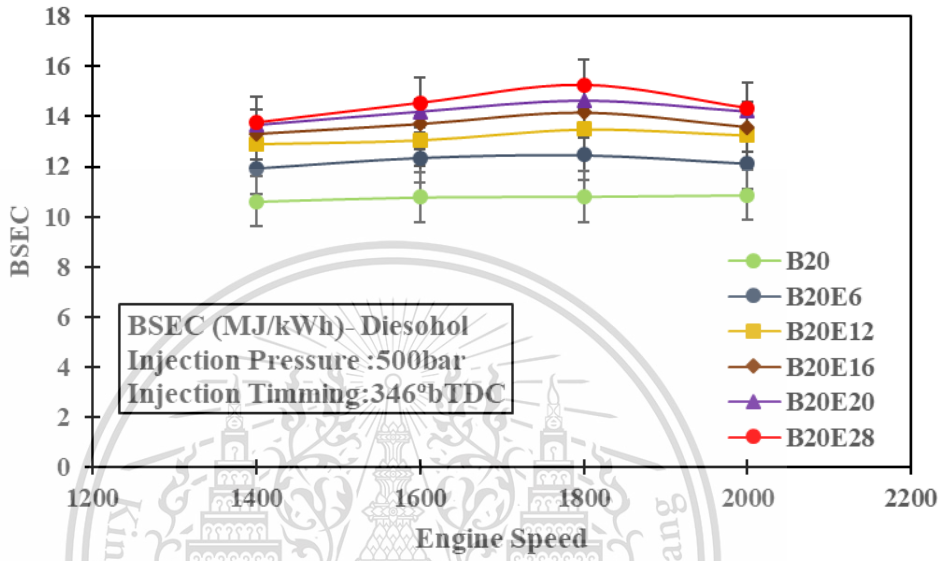

Figure 4.4 Variation of BSEC with engine speed for diesohol

4.1.2 Emissions

The CO emission for diesel fuels had reduced with increase of engine speed and increase in the fraction of ethanol of blended fuel has led to low CO emission which was shown by figure 4.5. Increasing the ethanol fraction means that there was sufficient oxygen for a complete combustion. The use of alcohol allowed a higher relative concentration of oxygen to exist in combustion gases and this achieved a greater conversation of CO to CO2 than B20. B20E28 which had a higher oxygen content gave least CO among all the blended fuels which had higher molecular weight of oxygen. Improve the combustion process at high speed caused for low CO emission. Air fuel ratio is increased with engine speed thus CO emission getting reduce with increment of engine speed.

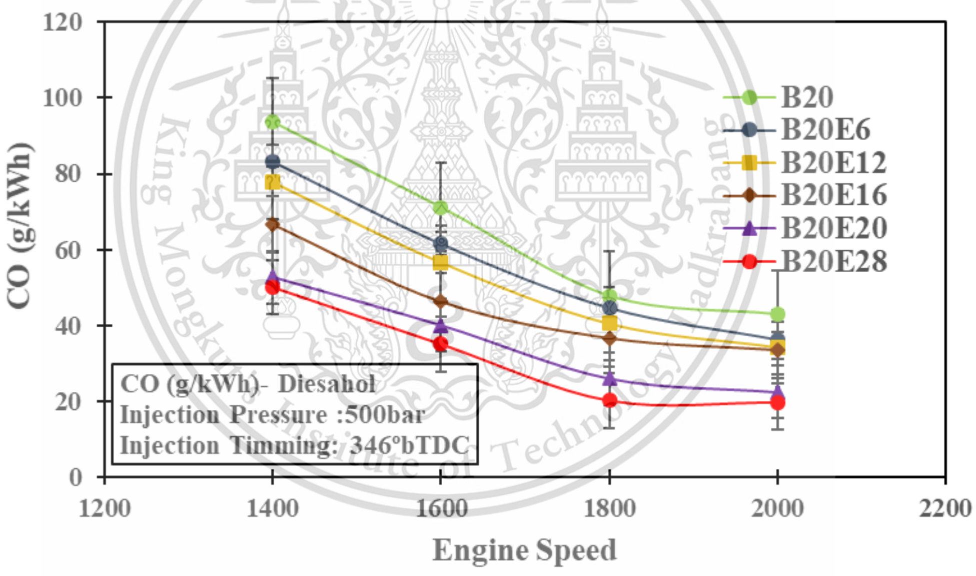

Figure 4.5 Variation of CO emission with engine speed for diesohol

The NOx results of diesahol was different from fumigation results of NOx. NOx was increased when increasing the engine speed and ethanol fraction which was clearly shown in figure 4. B20E28 showed highest NOx formation and B20 gave lowest NOx emissions. Cetane number and oxygen content were more effective than latent heat of vaporization with regard to increasing peak temperature in the cylinder. Ethanol blends have lower cetane number which caused for increasing of ignition delay. Increased the ignition delay, which increased the maximum cylinder pressure and resulted in higher maximum temperatures and consequently increased NOx emissions. Therefore, the concentration of NOx increased as the alcohol content was increased in the fuel blend.

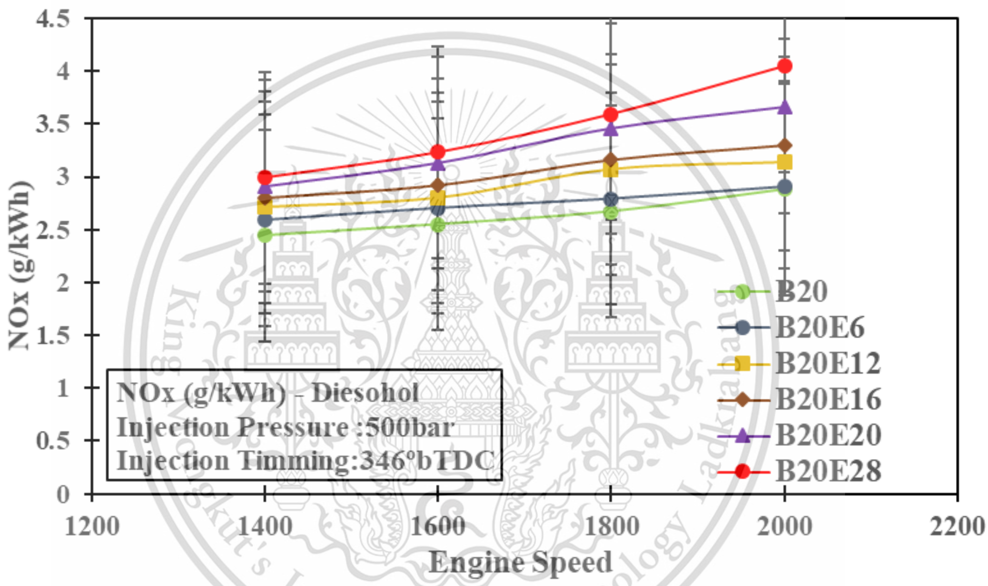

Figure 4.6 Variation of NOx emission with engine speed for diesohol

THC emission of diesol had been shown by figure 4.7.THC emission was primarily caused as a result of engine configuration, combustion temperature, and oxygen availability and residence time. These results showed that increase of ethanol fraction of blended fuel caused for reduction of THC. So that B20 showed highest THC emission meanwhile B20E28 gave the least THC emission. High oxygen content of ethanol blend gave chance to fuel to easily get reacted which cause for oxidation of THC. Laminar flame speed of alcohol was seen to increased compare to diesel fuel. The increase in the flame speed would reduce combustion duration but increase combustion temperature. The higher combustion temperature promoted more complete combustion and hence there were less THC emissions.

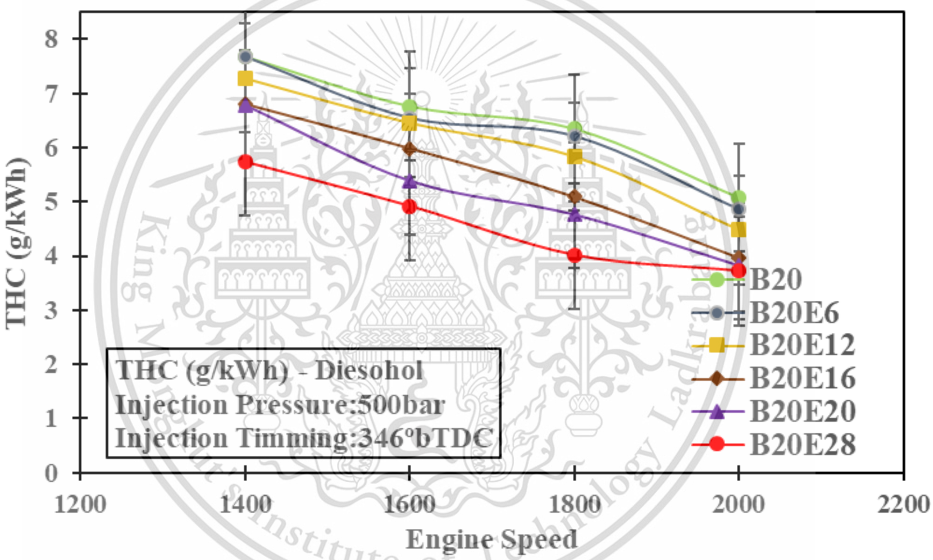

Figure 4.7 Variation of THC emission with engine speed for diesohol

Soot emission fuel was decreased when increasing the engine speed and the portion of amount of ethanol in blend. Soot formation mainly took place in the fuel rich zone, especially within the core region of the each fuel spray. It is commonly assumed that oxygenated blends with bio diesel effectively delivered oxygen to the pyrolysis zone for the burning of the fuel spray resulting in the reduced smoke emission. Thus the highest oxygen contented fuel of B20E28 resulted the lowest soot emission when compared to other blended fuels.

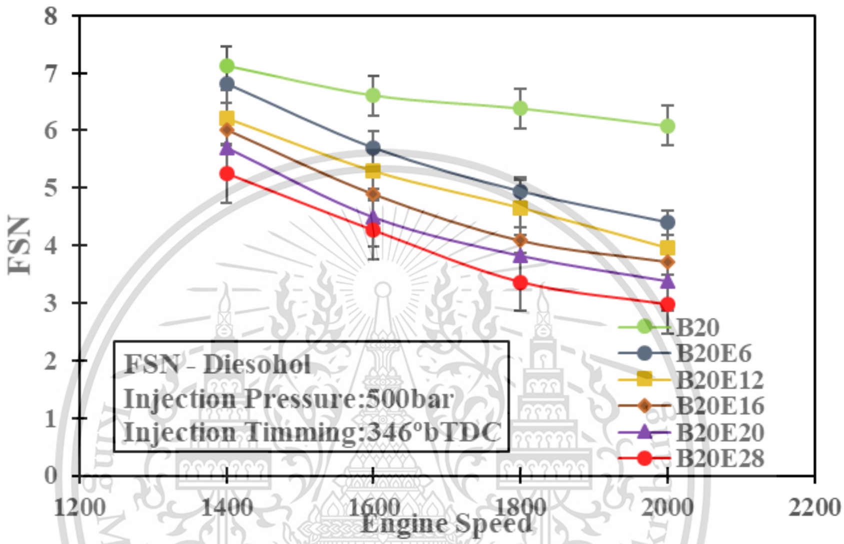

Figure 4.8 Variation of Soot emission with engine speed for diesohol

4.1.3 Combustion Characteristics

The combustion characteristics of Diesohol fuels were measured at 1600rpm with 30Nm load was shown in figure 4.9. Cylinder pressure in a CI engine depended on the burned fuel fraction during the premixed burning phase. Cylinder pressure characterized the ability of the fuel to mix well with air and burn. Highest peak pressure values of diesohol fuels are 98.2101, 99.354, 98.633, 99.49, 98.275 and 104.458 bar respectively. High peak pressure and maximum rate of pressure raised corresponding to the large amount of fuel burned in premixed combustion stage. In cylinder pressure of blended fuels B20, B20E6, B20E12, B20E16, B20E20 and B20E28 shown almost same behavior and no any significant difference. Highest peak pressure was retarded with increasing of ethanol fraction. Due to low cetane number with increasing of ethanol

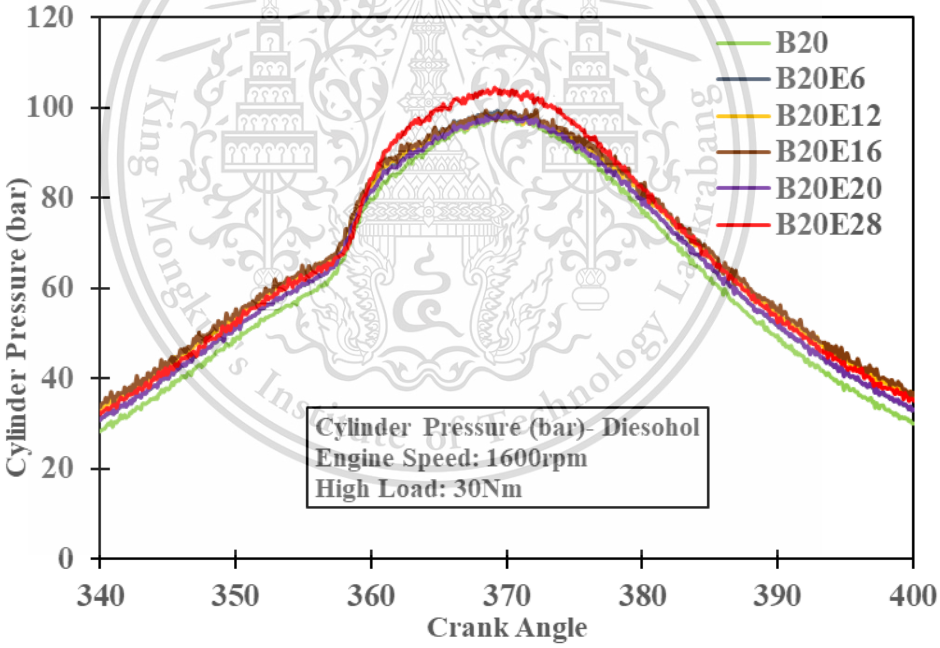

Figure 4.9 Variation of Cylinder Pressure with crank angle for diesohol

Heat release rate was increased with increasing the fraction of the ethanol. Longer ignition delay of increasing ethanol blend ratio lead to longer mixing time of intake charge , thus more fuel get combusted in the premixed zone resulting higher heat release rate, due to that B20E40 gave highest peak of heat release rate. Peak heat release rate of blended fuels are 82.75, 87.08, 95.67, 116.43, 119.57 and 133.15kJ/°CA. All the blended fuels and B20 had diffusion phase combustion due to excess oxygen content might have helped for diffusion combustion. Increasing ethanol fraction caused for lowering down the cetane number of the fuel. Due to that ignition delay was increased with increasing of ethanol fraction. Longer ignition delay helped to fuel to mix with air, atomization and vaporization for better combustion at premixed phase. Thus higher heat release peaks was given by diesohol fuels which blended with higher portion of ethanol. B20E28 had given highest peak of heat release and B20 given lowest peak of heat release according to figure 4.10

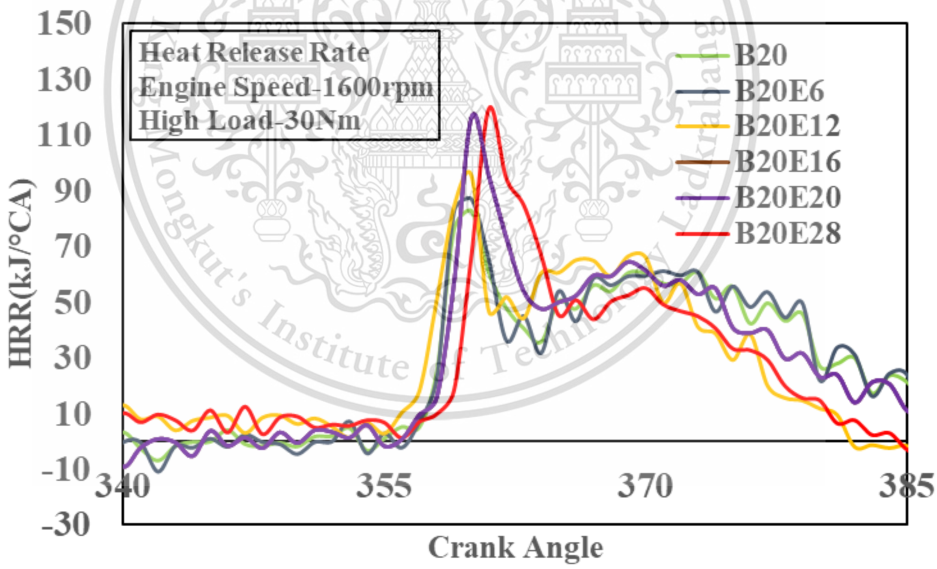

Figure 4.10 Variation of Heat Release Rate with crank angle for diesohol

4.2 Fumigation Method

4.2.1 Engine Performance

Under full load condition, the engine torque variation with respect to engine speed for Ethanol fumigation with B7 and B20 is described in figure 4.10Ethanol fraction of fumigation was varied as 12%, 16% 20% and 28% with B20 and 12% and 20% with B7. Ethanol fumigation percentages with B20 were calculated with blended fuels of B20 with ethanol according to energy input for fair comparison of fumigated method with respective blended fuel. Two percentages of ethanol with B7 were used to see how it behave with different type fuels.The torque was measured the range of 1400rpm and 2000rpm with increment of 100. Torque of all the fuels started to reduce when increasing the engine speed and maximum torque was delivered at 1400 rpm for all the fuels. Increment of the ethanol fumigation fraction lead to a reduction of torque. Ethanol fumigation with B7 showed highest torque for all the engine speeds and 28% ethanol fumigation with B20 gave lowest torque for all the engine speed. Maximum torque of all the fuel were 45.971, 44.399, 43.160, 41.508, 40.530, 39.483 and 37.595Nm at 1400rpm. Fumigation rates of B7 showed highest torque compared to B20. Highest calorific value of B7 than B20 was main reason for highest torque. Increasing the rate of fumigation lead to lowering of caloric value and decrease in cetane number which lead to lower output of torque. Torque was getting reduced with the increment of the engine speed.

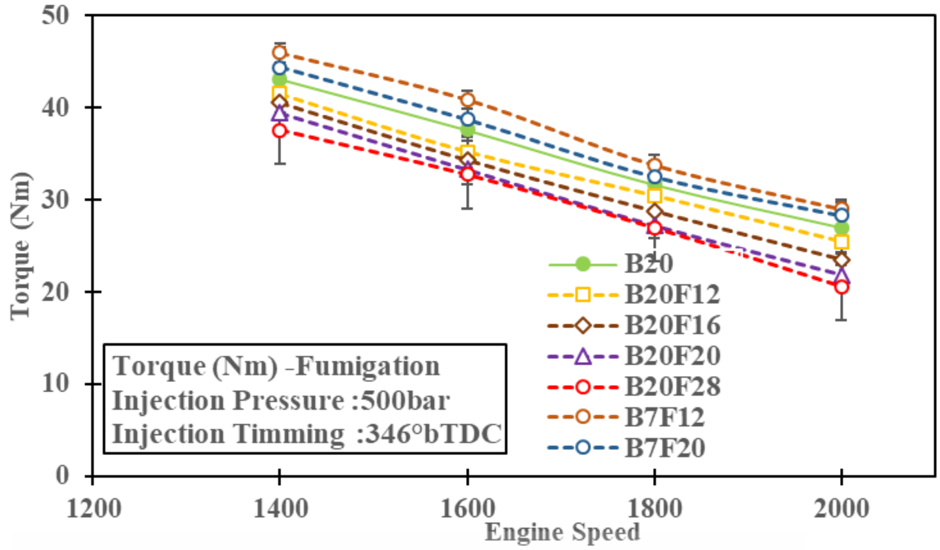

Figure 4.11 Variation of torque with engine speed for fumigation

The results of Power in fumigation method also has shown the same behavior as the results of Torque. Increasing the rate of fumigation has reduced the power. Maximum power of B7F12, B7F20, B20 and B20F28 were given at 1600 rpm. Power reduction from 1400rpm to 2000rpm was 9.7%, 9.86%, 10.59%, 12.3%, 14.9%, 20.96% and 21.64%.it has shown that when increasing the ethanol fraction, the reduction of power with engine speed was also higher.

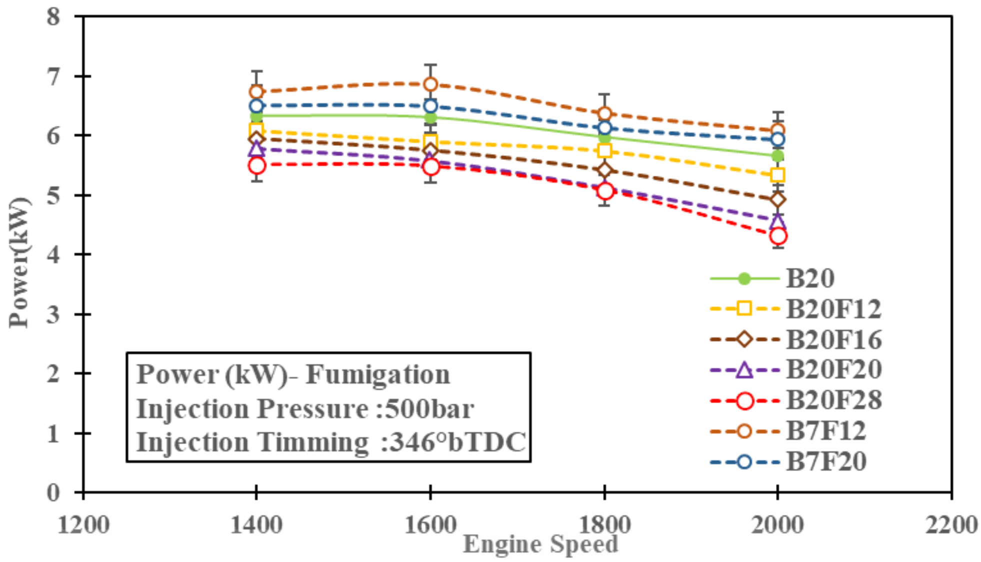

Figure 4.12 Variation of power with engine speed for fumigation

B20 had shown the highest BTE compared to other fumigation rates of ethanol with B7 and B20. Reason was that B20 had higher oxygen content for complete combustion but slightly reduced BTE when increasing the engine speed. Viscosity of bio diesel might have been the reason for improper combustion at high speed which ultimately resulted in a reduction of BTE. When the ethanol fumigation rate was increased by keeping B20 as the base fuel BTE tended to reduce by 5.30%, 6.13%, 7.61%, 10.58%, 14.29% and 18.5% at 1600rpm. It seemed that the cooling effect, together with leaner air/ethanol mixture, resulted in poor combustion thus caused a reduction in BTE. BTE of 12% and 16% fumigation of ethanol which kept B20 as base fuel was higher than 12% fumigation of ethanol with B7. B20 had higher oxygen content than B7 so it has helped for a proper combustion which had ultimately lead to higher BTE. BTE had increased after 1800rpm for all the fumigation conditions because the mixture might have been rich enough to support combustion and caused slight increment of BTE.

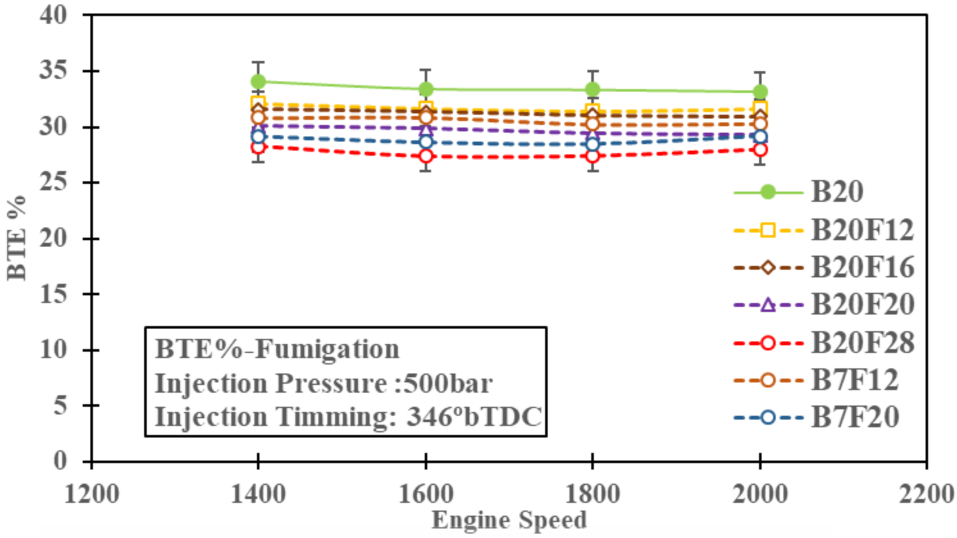

Figure 4.13 Variation of BTE with engine speed for fumigation

The main factor for Brake Specific Energy Consumption is calorific value. If the calorific value was higher the consumption of corresponding fuel was lower because it provided sufficient energy. B20 had lowest BSEC because it had the highest caloric value. When the fumigation rate was increased reducing the portion of B20 compared to base fuel operation it lead to lower energy supply to the engine. Higher portion of ethanol and B20 had to be consumed to achieve the appropriate engine speed

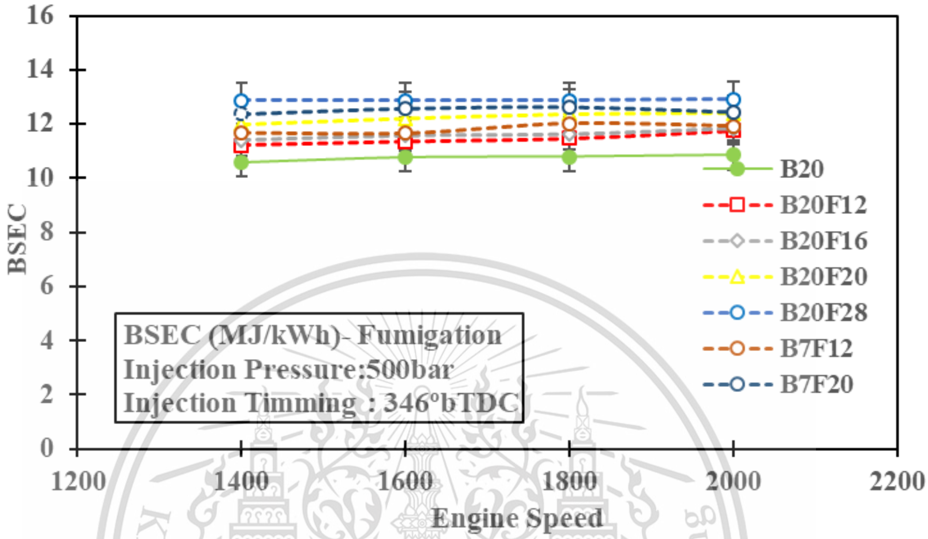

Figure 4.14 Variation of BSEC with engine speed for fumigation

4.2.2 Emissions

Toxic CO gas emission from CI engines has occurred either due to deficiency of oxygen or lower in cylinder temperature during the combustion process. Figure 4.15 and 4.16 showed that CO emission has decreased with the increase of engine speed and CO emission has increased with the increase of ethanol fumigation rate. Figure 4.15 showed how CO emission of fumigation vary with reference fuel of B20.It showed that B20 had huge CO emission than fumigation. Figure 4.16 gives clear explanation of CO emission at different fumigation rates.

Increase in fumigation rate had resulted a lower in cylinder gas temperature due to latent heat of vaporization and it had caused an incomplete combustion at low engine speeds, cooling effect might lead to incomplete oxidation CO to CO2, leading to an increase in CO emission. Another reason for CO emission at low speed might be the trapping of alcohol mixture in crevices which was unable to ignite during the expansion stroke. B7F12 had higher CO emission than B20F12. Higher oxygen content of B20 could have helped for a better combustion than B7 so that B20F12 gave lowest CO emission of the testing.

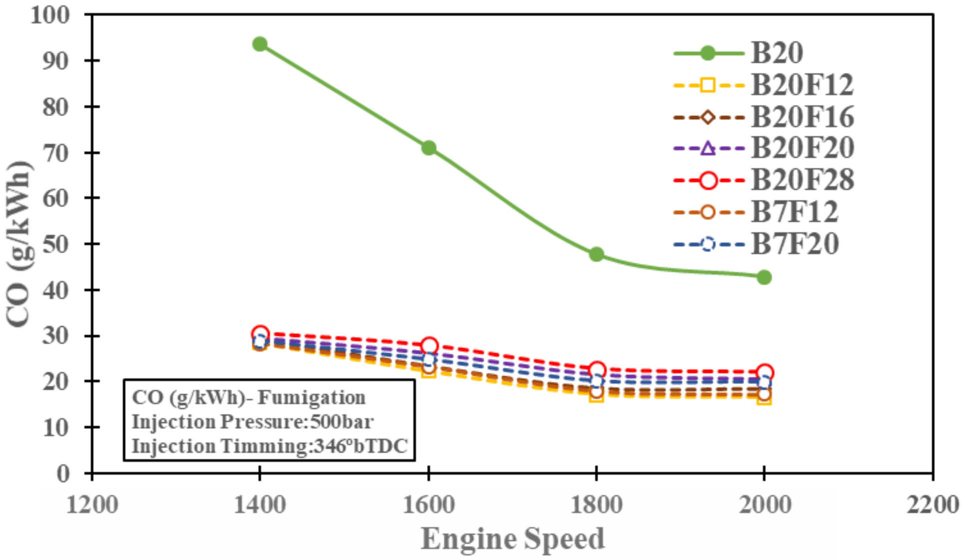

Figure 4.15 Variation of CO including B20 emission with engine speed for fumigation

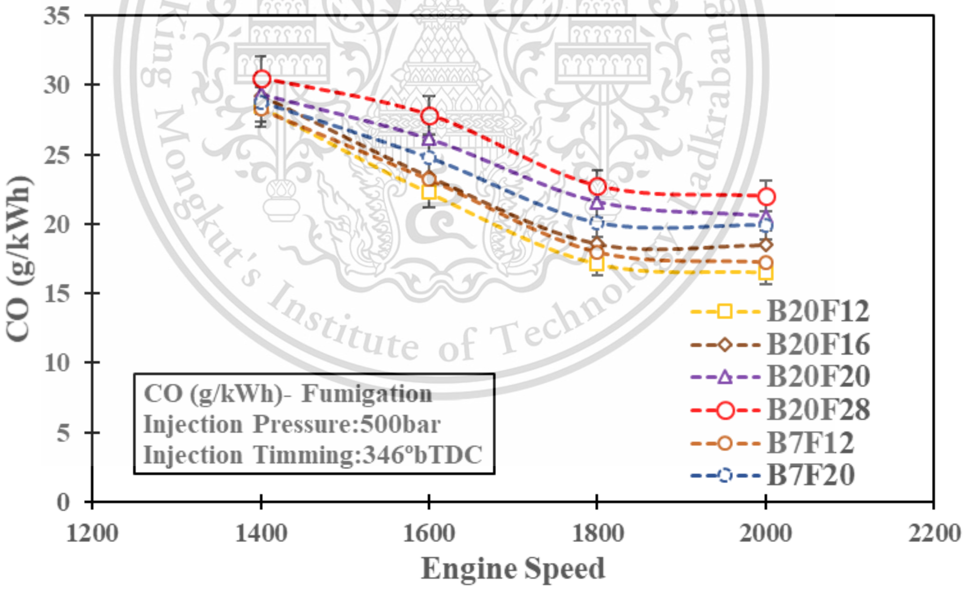

Figure 4.16 Variation of CO emission with engine speed for fumigation

The formation of NOx in CI engine was strongly dependent on the combustion temperature, along with the concentration of oxygen present in the combustion process. These results showed that NOx decreased with the increase of engine speed and fumigation percentage. Higher heat of vaporization, fumigated ethanol could reduce the combustion temperature hence resulted a reduction of NOx formation. Lower combustion temperature and higher air/fuel ratio associated with low engine speed might have resulted in a reduction of NOx. However at high engine speed, in fumigation mode there was a reduction in air fuel ratio and the B20 fuel is now combusted in a mixture of air and alcohol which might had a negative effect on the oxygen availability for NOx formation.

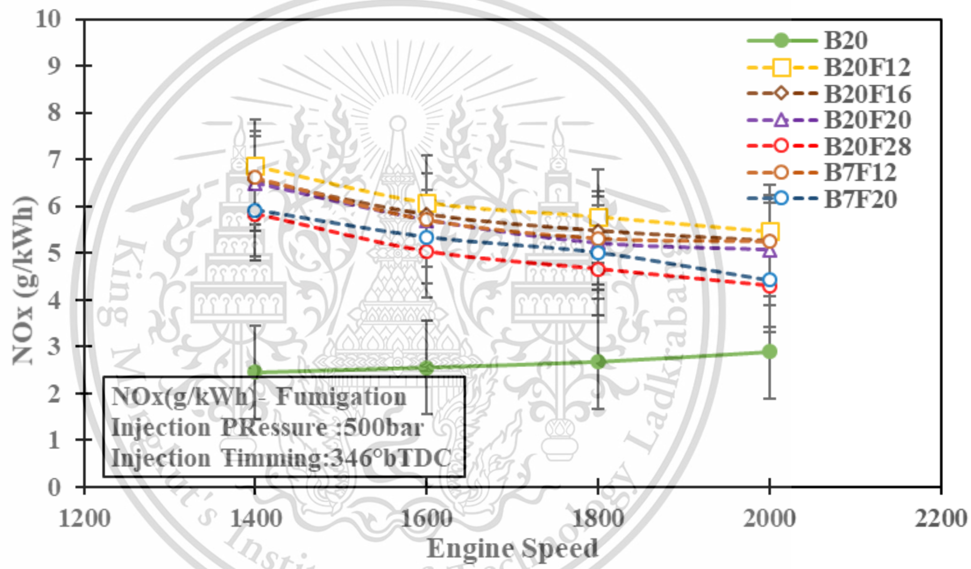

Figure 4.17 Variation of NOx emission with engine speed for fumigation

Figure 4.19 exhibits the increase of THC emission with different rates of ethanol fumigation. THC emission decreased with engine speed and increased with percentage of ethanol fumigation. The high latent heat of vaporization of ethanol slowed down the vaporization and also the mixing of air and fuel, due to that lower combustion temperature lead to poor combustion. The poor combustion enhanced formation of THC that increased with increasing with ethanol percentage. At low engine speeds, due to large amount of air, poor fuel distribution and low exhaust temperature might have been the reasons for higher THC emissions. Reduction of THC in the range of1400rpm and 200rpm were 58.22%, 58.15%, 56.26%, 64.80%, 74.69% and 81.50% respectively.

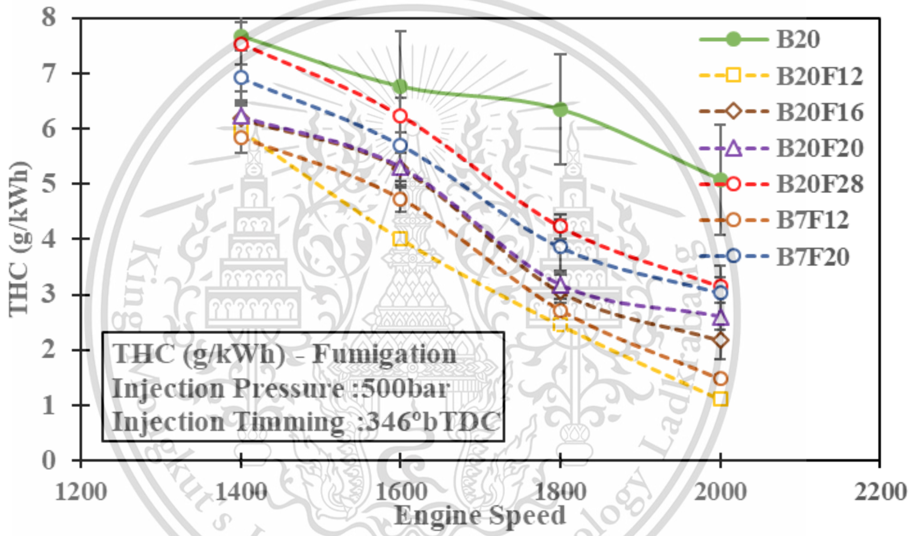

Figure 4.18 Variation of THC emission with engine speed for fumigation

The soot emissions has reduced when increasing of engine speed and ethanol fumigation rate. the charge cooling increases ignition delay and, thus, enhances the mixing of diesel fuel with the ethanol air mixture which, in turn, makes for better air utilization and less smoke. Also, diesel fuel has a high tendency to soot formation due to its low H/C ratio and the nature of its combustion process. B7 has low H/C ratio than B20 thus, B7F12 showed higher Soot emission than B20F12. The reduction of soot emission, with ethanol fumigation modes depends on increasing of ignition delay increases the amount of bio diesel fuel burned in the premixed mode, which reduces the amount of biodiesel fuel burned in the diffusion mode.

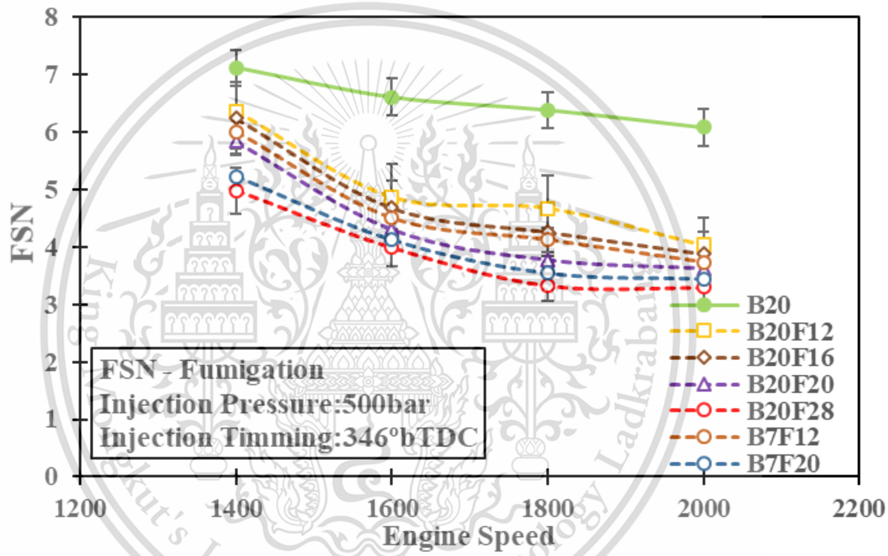

Figure 4.19 Variation of Soot emission with engine speed for fumigation

4.2.3 Combustion Characteristics

Cylinder Pressure of fumigation was also measured same condition as of diesohol, 1600rpm constant engine speed with 30Nm load. Increasing the ethanol fumigation lead to an increase of the cylinder pressure. Max Pressure of each fuel with fumigation were 98.49, 99.0085, 100.95, 103.52, 106.75, 100.06 and 100.075Bar respectively. As more oxygen accumulated during the delay period, the diesel got more oxygen to burn. This might have resulted a rapid pressure rise and peak cylinder pressure. Ethanol had higher flame temperature thus, increasing the ethanol percentage increased the flame temperature which caused for a better combustion and higher cylinder pressure. Cylinder pressure of B7F20 was lower compared to B20F20. B20 had higher oxygen content than B7 which might have been the reason for increasing the pressure due to better combustion and flame temperature.

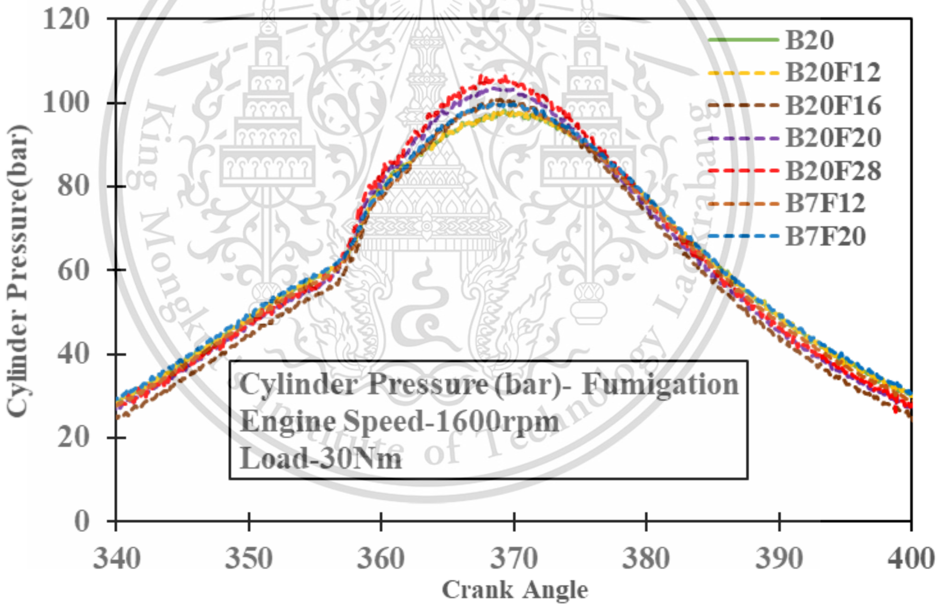

Figure 4.20 Variation of cylinder pressure with crank angle for fumigation

Heat Release rate was increased with increment of ethanol fraction. The increase in premixed combustion rate was mainly due to the combustion of small part of the ethanol entrained the Bio diesel spray. Ethanol and air got mix in the path of intake manifold after the ethanol injection and that ethanol air mixture was homogenous. Homogenous mixture easily got atomized and vaporized due to in cylinder temperature. That homogenous mixture of ethanol and air got mix with B20 and started get combusted when it reached to combustible limits. Due to air ethanol homogenous mixture combust with B20 in premixed phase, Heat Release Rate was increased rapidly. Increasing the ethanol fraction helped for better combustion due to higher oxygen content thus higher heat release peak has raised up suddenly. B20 has higher oxygen amount than B7thus B20 with ethanol fumigation has given higher heat release rate peaks.

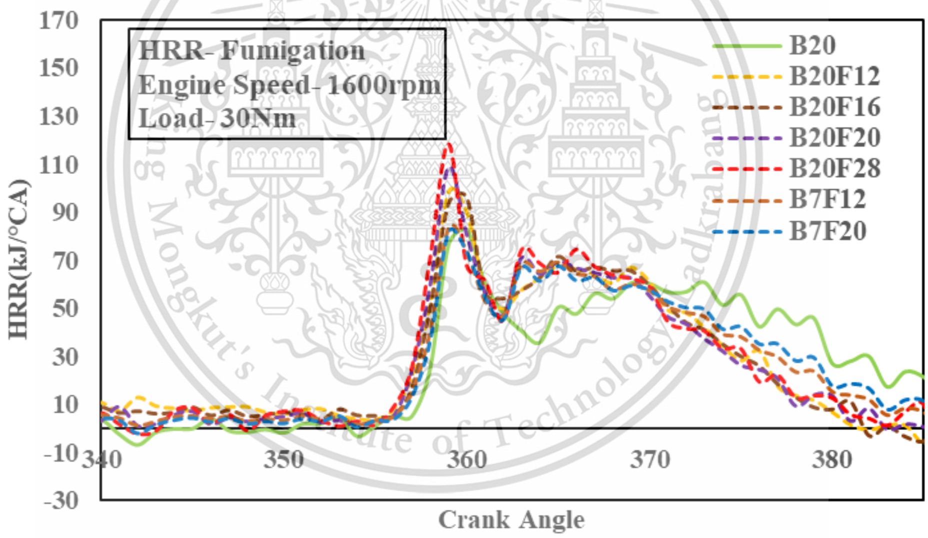

Figure 4.21 Variation of heat release rate with crank angle for fumigation

4.3 Comparison of diesohol method and fumigation method

The comparison of diesohol and fumigation was done using four types of blended fuels and four different rate of ethanol fumigation. The comparison had been done between the diesohol fuel and ethanol fumigated fuel which had the same energy input. B20E20, B20E25, B20E30 and B20E40 diesohol fuels were compared with their same energy input fumigated rate of 12%, 16%, 20% and 28% respectively. The first comparison investigated performance of both methods including torque, power, BTE and BSEC.

4.3.1 Engine Performance

The results of comparison of torque for fumigation of ethanol with B20 and B20 Blended fuels are shown in figure. The comparison of B20F12 with B20E12 has shown almost the same pattern but blended fuel of B20E16, B20E20 and B20E28 showed slight increment of torque after 1800rpm compared to corresponding fumigation rates. The increment was 7.43%, 10.10% and 14.23% which meant increasing blending ethanol ratio tends for enhance torque at high speed compare to fumigation. Less time to mix ethanol with air at high speed may have been a reason for poor combustion and giving low torque at high speeds.

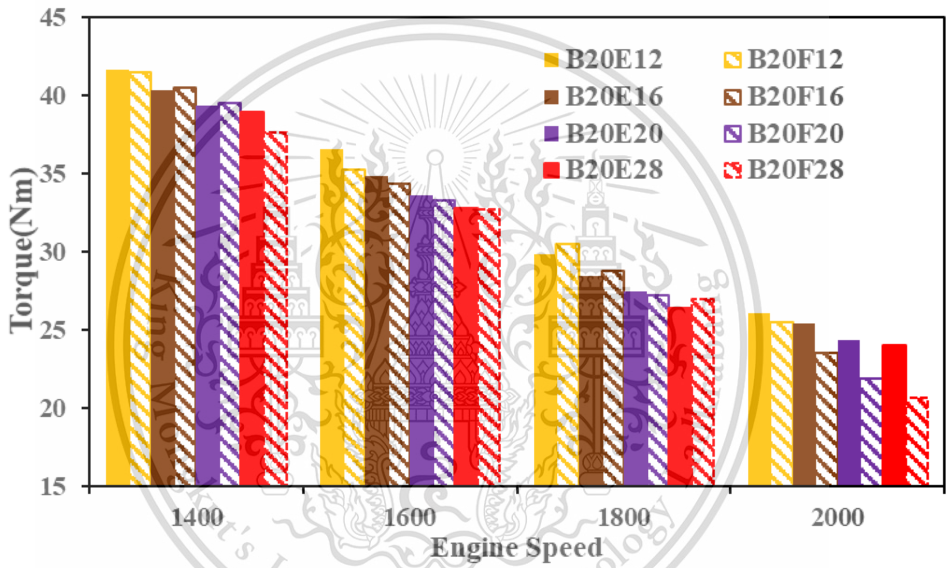

Figure 4.22 Comparison of torque variation with engine speed

Comparison of the power between blended fuel and fumigation corresponding to equal energy input had shown the same behavior as the torque comparison. Increment of power after 1800rpm was shown for blended fuels which was 2.23%, 7.44%, 10.10% and 14.24%. It had shown that increase in the ethanol fraction for both fumigation and blended fuel, the difference of power was getting increased due to lack of time for ethanol and air to mix well and start propagation.

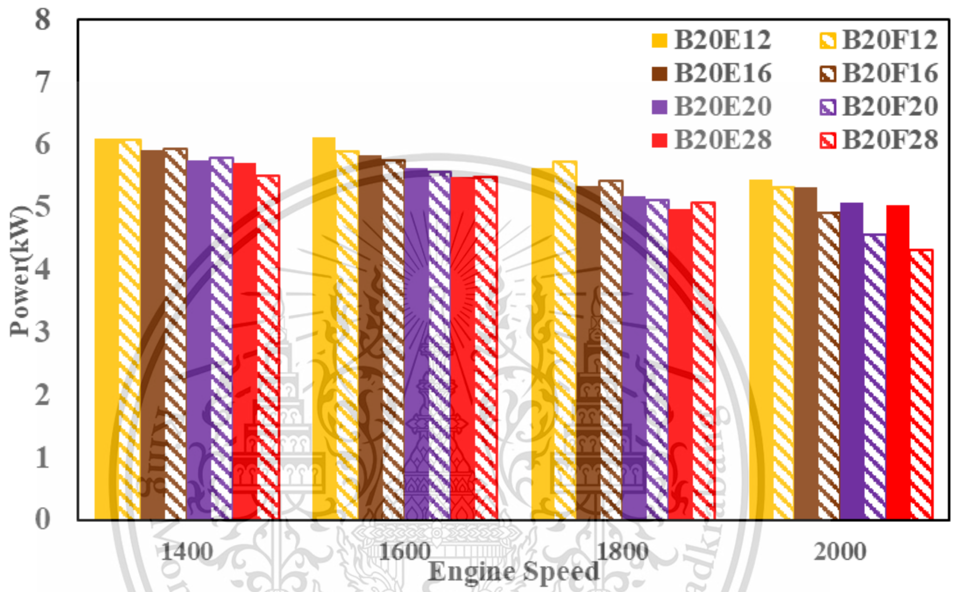

Figure 4.23 Comparison of power variation with engine speed

Comparison BTE between fumigated ethanol and diesohol fuels have shown figure fumigation method had the highest BTE for all four conditions. Homogenous air/ alcohol mixture which burned with more rapid heat release which might have increased increase BTE than for diesohols. Lower cetane number of diesohol had caused for a longer ignition delay which might have occurred sufficiently late in the expansion process for the burning process to be quenched, resulting in incomplete combustion reduce BTE.

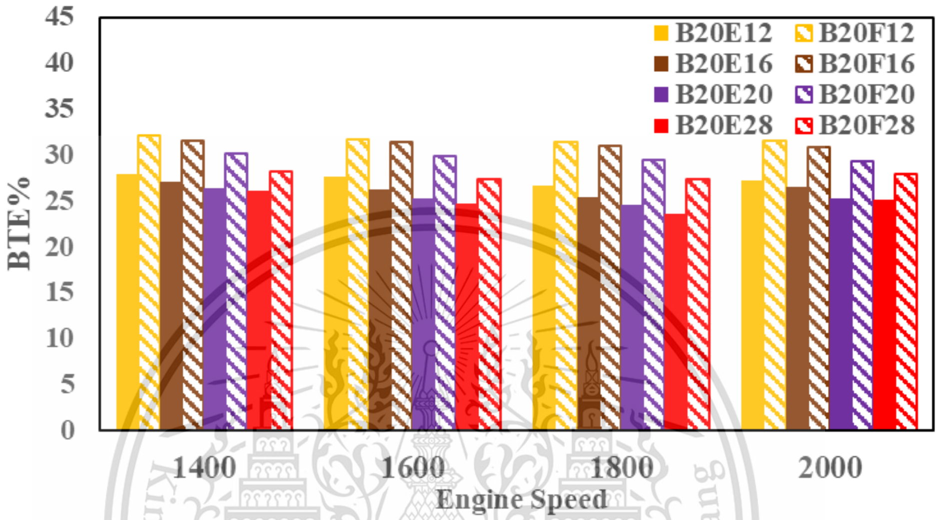

Figure 4.24 Comparison of BTE variation with engine speed

Brake Specific Energy Consumption was increased when increasing the portion of ethanol of diesohol fuels. Increase in the ethanol portion causes to lower down the calorific value of each fuel. B20E28 had the highest portion of ethanol thus it had the lowest calorific value compared to other blended fuel and B20E28 gave highest BSEC. Highest BSEC gave at 1800rpm for all the blended fuel which is 13.37%, 20.00%, 23.8%, 26.00% and 29.19% increased compare to B20.

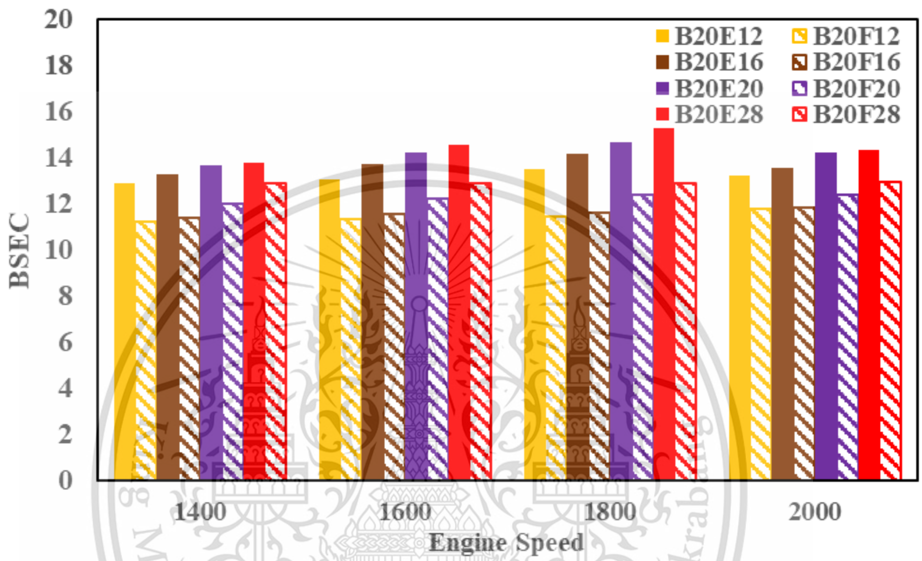

Figure 4.25 Comparison of BSEC variation with engine speed

4.3.2 Emissions

The comparison of CO emission between diesohol fuels and fumigated ethanol are shown in figure 4.21The gap between corresponding fumigation and diesel fuels are getting closer when ethanol fraction was increased. Emission of CO of diesel fuels B20E12, B20E16 and B20E20 were higher than their corresponding fumigated ethanol results. But the gap was closed enough at high speed of B20E20 condition with B20F20. CO emission of B20E28 was higher than B20F28 until 1600rpm and showed lower CO emission than B20F28 after 1800rpm. B20F28 showed highest CO emission in the comparison of fumigation results and B20E28 showed lowest CO emission of Diesohol results. Increasing the percentage of ethanol fumigation might have resulted trapping of ethanol air mixture in crevices and was unable to ignite during expansion stroke. B20E28 was blended fuel with highest oxygen content which might have helped for a complete combustion and good boosting turbulence with sufficient oxygen helped for complete combustion which converted CO to CO2 as well

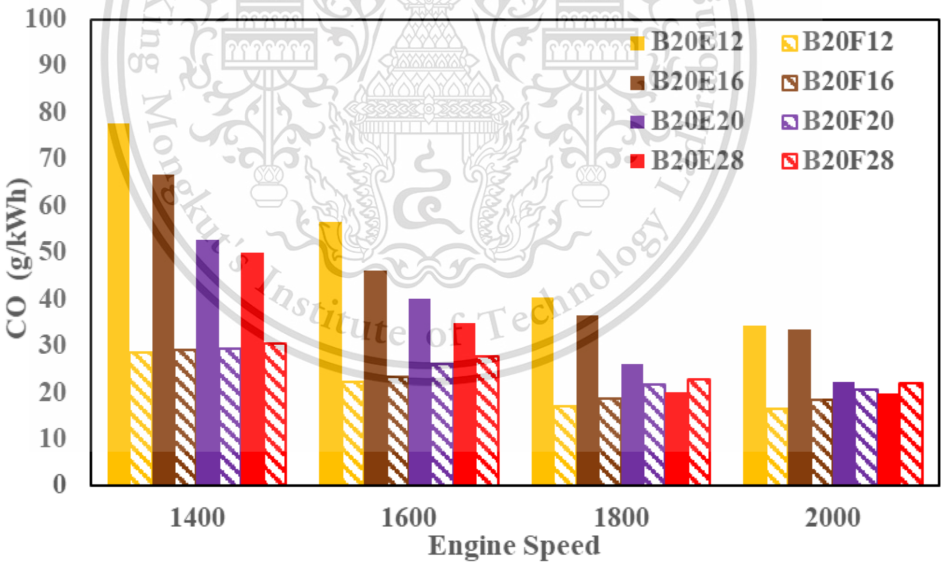

Figure 4.26 Comparison of CO variation with engine speed

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The NOx emission of diesohol and fumigation had an opposite behavior. NOx formation was reduced with engine speed for fumigation method, though NOx formation of diesohol was increased with engine speed. NOx formation was effected in a different way for fumigation where the latent heat of vaporization cooled down the in cylinder temperature with the increase of ethanol percentage. So it caused for a reduction of NOx. Lower cetane number and higher oxygen content when increasing the fraction of ethanol of blends thus caused an increase in NOx formation was increased.

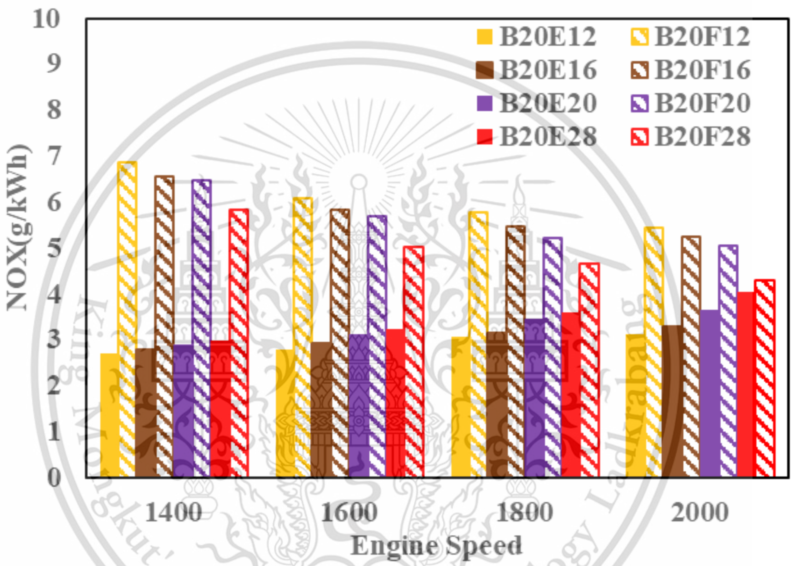

Figure 4.27 Comparison of NOx variation with engine speed

Both fumigation and diesahol fuels showed reduction of THC emission with increasing of engine speed. At low ethanol fumigation rates and low ethanol blend ratios the THC emission gap was higher for THC emission of diesahol for all the engine speeds. But the gap was getting closer with increasing of ethanol. B20F28 THC emission results showed that it had higher THC emission than B20F40. Higher heat of vaporization of ethanol at 28.00% of fumigation was large compared to other fumigation rates. Thus it might have cooled down the in cylinder temperature more than other rate of fumigation of ethanol which may cool down in cylinder temperature. Lower combustion temperature lead to poor combustion. B20E28 had highest oxygen content compared to other diesohol fuels. Highest oxygen content of B20E28 helped for proper combustion and caused less THC emission than B20F28.

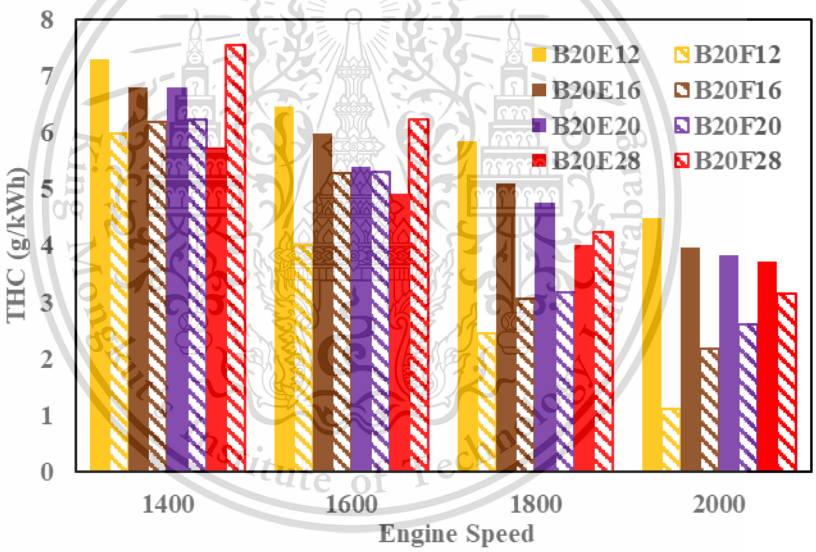

Figure 4.28 Comparison of THC variation with engine speed

The comparison of soot between diesohol and fumigation did have any significant different. There was slight increase of diesohol soot emission than fumigation at high ethanol portion rates.

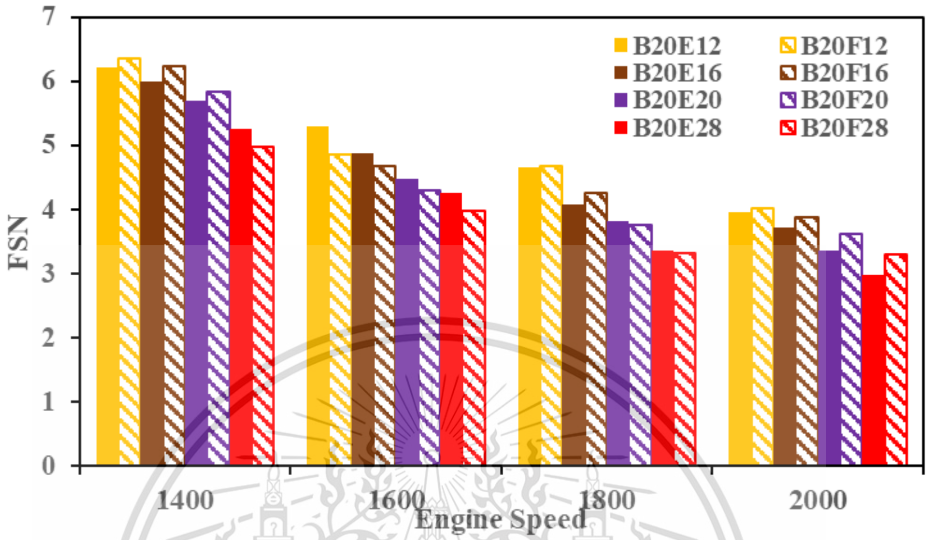

Figure 4.29 Comparison of soot variation with engine speed

4.3.3 Combustion Characteristics

Comparison results of cylinder pressure of diesohol and fumigation has shown higher peak cylinder pressure on diesahol fuel B20E12 and B20E16with respect to the same energy input fumigation rates of 12% and 16%. But after that peak cylinder pressure of fumigation modes showed higher values than diesohol. The combustion quality degraded with percentage increase in ethanol beyond 30 concentration. This retardation in combustion is due to excessive decrease in fuel viscosity, which hampered injection and atomization.

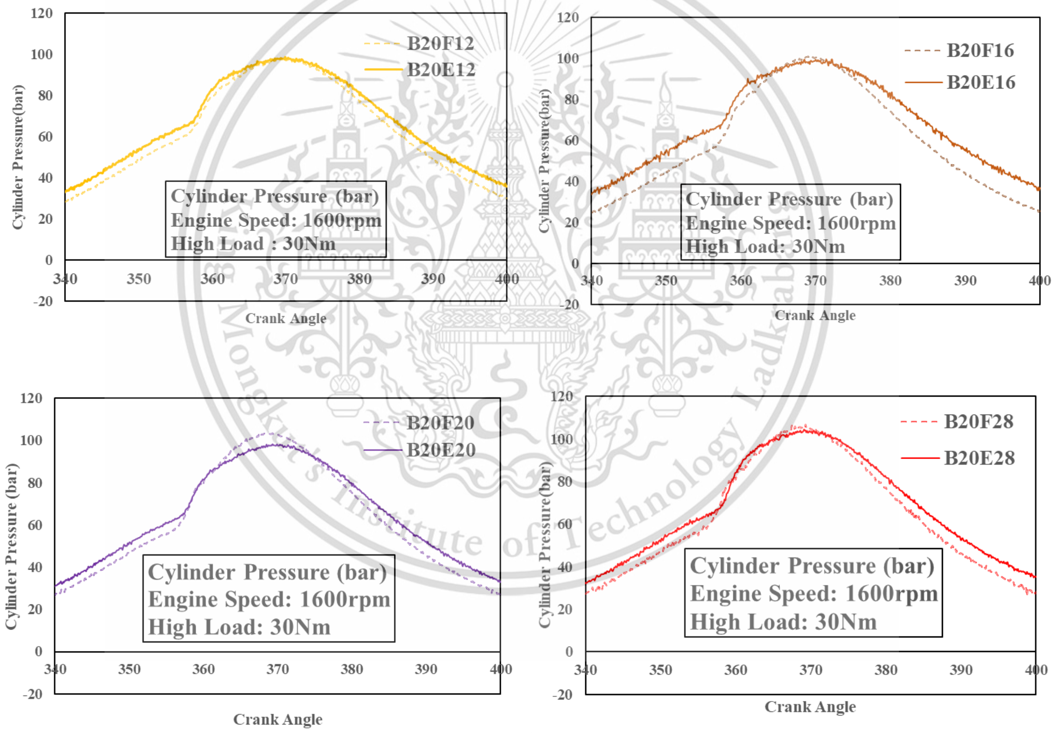

Figure 4.30 Comparison of Cylinder Pressure variation with crank angle

Comparison of Heat release rate showed that diesohol fuel had higher peak of heat release and longer ignition delay than fumigation rates. Increase in ethanol fraction of diesohol caused to low down the cetane thus longer ignition delay had occurred for all the conditions. Both of diesohol and fumigation had diffusion phase combustion due to excess of oxygen which helped for late combustion after premix combustion occurred.

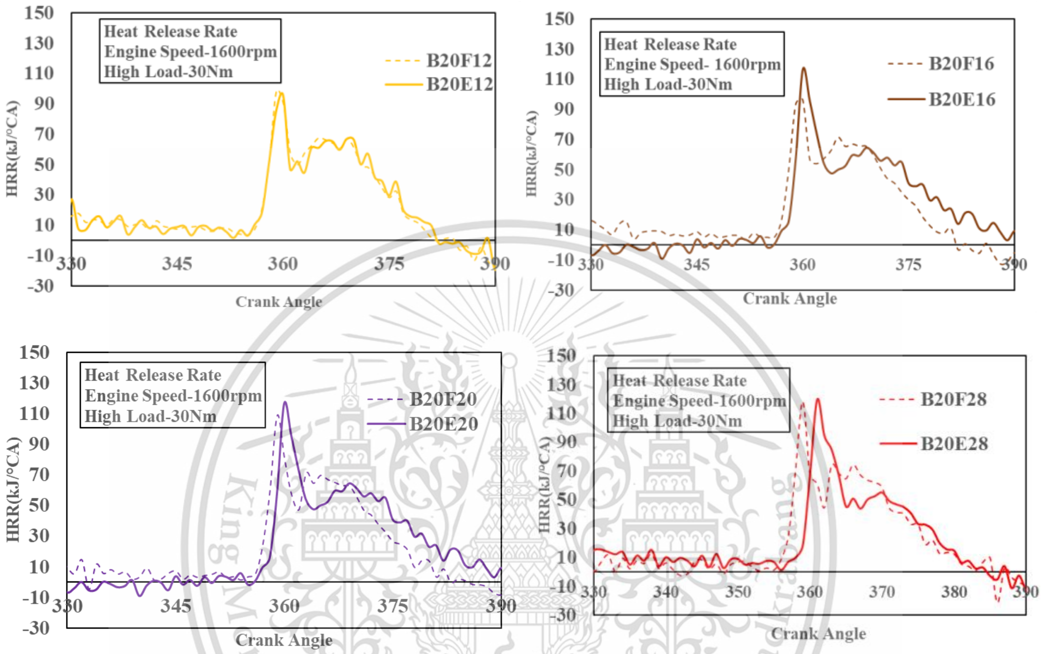

Figure 4.31 Comparison of heat release rate variation with crank angle

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

Torque and power was decreased with ethanol fraction for both diesohol and fumigation method.B20 showed highest power and torque results due to its heating value which was higher than all the blended fuels and fumigation ratios. B20 showed highest BTE and lowest BSEC due to higher heating value and high cetane number. All the time fumigation rates showed highest BTE and lowest BSEC with compare to diesohol. Increasing both fumigation diesohol fraction reduced the gap of BTE and BSEC between diesohol and fumigation results.

Dieshol fuels showed significantly higher CO emission with compare to fumigation rates. The gap between fumigation and dieshol was got closer at high speed and B20F28 CO emission was higher than B20E28 CO after 1800rpm. NOX emission of diesohol and fumigation showed opposite behavior which meant NOX formation was increased with engine speed vice versa NOx formation of fumigation was reduced with engine speed. Both fumigation and diesohol method showed reduction of THC with engine speed. Highest oxygen content of B20E28 helped for proper combustion and caused less THC emission than B20F28. Smoke emission also reduced with engine speed there was not significant different between fumigation and blended smoke emission.Cylinder pressure of blended fuel were higher up ethanol blended ratio of 25% than fumigation. But after 30% ethanol blended fraction, fumigation rates of 20% and 28% showed higher cylinder pressure than B20E30 and B20E40. Heat Release rate of all the diesohol fuels were higher than fumigation rates and had higher ignition delay as well.

Long running of the engine using dieshol fuels and try to see the effect of ethanol blended fuels on the engine such as corrosion of parts, contaminations of oil, wear out of some parts is the future plan of this experiment to make sure how higher portion of ethanol blend fuels will affect to the real world application.

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APPENDIX A

The 28th International Symposium on Transport Phenomena
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IMPACT OF ETHANOL ADDITION TO BIO DIESEL(B20) ON PERFORMANCES COMBUSTION CHARACTERISTICS AND EMISSIONS OF A DIESEL ENGINE

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ABSTRACT

Today the world is affected by a huge energy crisis of satisfying the day to day activities of human beings, running of factories and automobiles due to lack of crude oil. Not only that but also the excessive usage of crude oil has harmed the environment mainly due to the emission of automobiles. bio diesel is mainly used in various countries as an alternative fuel which has almost the same properties like diesel. 20% of bio diesel and 80% of neat diesel are blended together to form B20 which has the ability of being miscible well with ethanol up to 15% by volume of ethanol. Higher heat of vaporization of ethanol may tend to cool down the in cylinder temperature that means it causes low emissions. The aim of the project is to investigate the combustion characteristics and emissions of ethanol blended with bio diesel blend(b20) as an alternative fuel. A single cylinder diesel Kubota RT140 agriculture engine with direct injection system was used for this experiment. B20 is selected to blend with 5 ,10% and 15% ethanol percentages because the blends were miscible well which was figured after checking the properties. In addition, their properties were in an acceptable range when compared with the neat diesel. Try to figure out combustion characteristics such as ignition delay, cylinder pressure, heat release rate and brake thermal efficiency, fuel consumption and emission of HC, CO, NOx and PM varying the engine rpm 1500 to 2500. Combustion characteristics of each fuel are tested 2000 rpm.

torque was reduced when increasing the ethanol portion because property of heating value is getting reduced when increasing the ethanol portion. BSFC was getting reduced as well when increasing the portion of ethanol. NOx emissions is slightly increased when increasing the rpm because of high in cylinder temperature caused the NOx formation. THC emission was less when increasing the ethanol percentage as the high oxygen content helps for a better combustion.

INTRODUCTION

Diesel engine has major role in automobile industry, marine industry diesel power plants and some engineering advanced systems. none of power source replaces the diesel engine 100% so far to fulfill the same work load done by diesel engines. Subsidence of crude oil has been occurred due to huge usage for automobile industry. Huge air pollution and greenhouse effect have been occurred likewise due to the emissions of engines. Advancement of diesel engine emissions controls is required as stricter global environmental protocols call for cutting edge emission controls and zero diesel emission levels in the years to come (1). Alternative fuel such as bio diesel blend with ethanol is another solution for reducing emission of engines.(2) have done simulation of diesel ethanol and diesel methanol operation. Emissions levels are reduced with increasing of alcohol b% and 32.63% by 15% ethanol with diesel blend. BSFC increased up to certain amount with ethanol diesel blends. and methanol diesel blends. Lower Cetane number of alcohol blends have increased the ignition delay period.

(3) ethanol has ability to miscible in water, aluminum, brass and copper may have corrosion effect due to ethanol contamination. Clogging inside fuel pumps may be occurred due to ethanol reacts with rubber. Ethanol is safer to transport and store due to its higher auto ignition temperature than diesel. advantages of ethanol due to high compression ratio, ethanol allows more engine power than gasoline engines. (4) ethanol can easily absorb water from ambient air and other supply systems. Anhydrous ethanol can miscible with diesel 0-30%.

(5) has done experiment of diesel fuel with either 5% and !0% ethanol or 8% and 16% n-butanol on 6 cylinder, Mercedes Benz engine under two engine speeds and three loads. Usage of ethanol and n- butanol decreased in cylinder pressure and increased ignition delay reduced in cylinder temperature as well. Due to that smoke and NOX emissions were decreased when increasing the portion of ethanol and n-butanol.

(6) The research based on study of combustion characterizes, exhaust emissions and spray of four-cylinder diesel engine powered by pure diesel and pure diesel with ethanol blends.in cylinder pressure was increased when advancing the injection timing. NOx emission was decreased when increasing the portion of ethanol with diesel. But increasing the load of the engine and injection timing tended to increase NOx emissions. CO and HC emissions increased with increase in ethanol blends ratio. Low NOx and high CO and HC emissions were observed low engine loads.

(7) Figured out that BSFC increased when increasing the portion of ethanol at constant equivalent ratio(CER). even though combustion duration increased portion of 4-6% of ethanol, it was reduced at high ethanol concentrations. Formation of NO increased at low ethanol concentration levels and started decreased at high ethanol ratios.

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APPENDIX A

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2.EXPERIMENTAL SETUP

2.1 Engine Setup

The experiment is conducted on a single cylinder direct injection Kubota RT140 4stroke engine. Engine is water cooled natural aspirated and using mechanical injector pump to deliver the fuel into the cylinder. Prony brake system is used as dynamometer to control the engine speed and load. Engine load is measured by Minebe U3B1-220k-B load cell which is fixed on the base with maximum capacity of 1.96kN .

Table 1: specifications of the engine

Displaced volume	709 cm^3
Stroke	96 mm
Bore	97 mm
Compression ratio	18:1
Maximum power	14hp @ 2400rpm
Maximum Torque	5kg/m @1600rpm

K type thermocouples were used to measure engine oil temperature, exhaust temperature, intake air temperature and water in and out temperature. Teledyne Hastings HFM 200 S/W air flow meter was used to measure the flow rate of intake air with maximum scale of 750 SLM.

Kistler 6052C piezoelectric pressure transducer coupled with a Kistler 5081 charge amplifier is used to measure in cylinder pressure. The in-cylinder pressures were recorded at every 0.1-degree crank angle, which triggered by using a 3600 ppr incremental encoder.

The gaseous emissions are measured by HORIBA MEXA 1600D which measures CO, NOx, THC. Non-dispersive infrared (NDIR) method of Horiba model AIA-260, was used to measure CO and CO_2 flame ionization detector (FID) analyzer, Horiba model FCA-266 is used to detect THC. Horiba FCA-266 Chemiluminescent analyzer was used to measure the oxides of nitrogen. The smoke number is measured by AVL filter paper type smoke meter.

2.2 Test fuels

Performances, combustion characteristics of the engine were mainly dependent on the type of the fuel which have been used and its thermos physical properties. Ethanol can be produced by sugar cane and cassava. Oxygen content in ethanol can help to reduce soot and particulate matter during the combustion. Therefore, using ethanol in diesel engine as the blends is of interesting. However, because of the polarity of ethanol, it cannot mix well with diesel. The phase separation will be occurred.

Acting as an emulsifier, biodiesel was firstly mixed with diesel such as B5, B7 B10 and B20. The numeric number is the percentage of biodiesel. Then, the blends of biodiesel were blended with ethanol call as disohol to check the proper miscibility of ethanol with bio diesel by varying the percentage of ethanol 5%, 10% and 15 %. The miscibility of diesohol is shown in Fig. 2…Blends of B20 was chosen for testing after the miscibility test of all the blends because B20 was miscible well up to 15% of ethanol.
In this study, 5, 10 and 15% ethanol by volume mixed with B20 were used to test. In addition, neat diesel and B20 were tested as the reference. Properties of test fuels including cloud point, pour point, viscosity, density and calorific value were measured and presented in Table 2.

Table 2 : Properties of Fuels

Properties	Neat Diesel	B20	B20+E5	B20+E10	B20+E15
Calorific Value (MJ/kg)	45.6	44.1	43.7	42.1	39.8
Cloud Point (°c)	5.5	6.1	6.1	6.3	6.8
Viscosity (mm^2/s)	3.2955	3.522	3.052	2.823	2.595
Density (g/cm^3)	0.8199	0.8286	0.8254	0.8231	0.8189

3. RESULTS

Before adopting an alternative fuel for replacing fossil fuel such as neat diesel, it is better to judge the performances and fuel consumptions. Due to addition of alcohol in B2O which may give different performances, emissions and combustion characteristics which have been investigated and discussed in this section.

3.1 Engine Performance

Fig.1 presents the torque of the engine when using diesohol with different ethanol percent. For comparison, the torque of engine when using neat diesel and B20 is also presented. The torque graphs of neat diesel and B20 are almost the same. However, the addition of ethanol caused the torque getting reduced significantly. . Low heating value of alcohol addiction blends may be the reason to show low torque 1500-2100rpm.torque is getting increased 10% and 20% corresponding B20+E10 and B20+E15 at 2100rpm to 2300rpm.high oxygenate content of B20+E10 and B20+E15 may help for a better combustion at high speeds with sufficient air to mix with fuel.

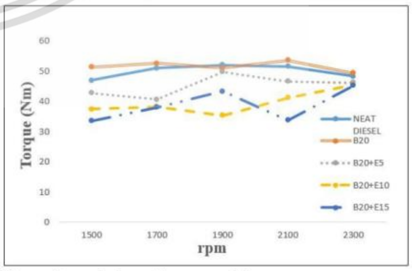


Figure 1: variation of torque with rpm

APPENDIX A

Increasing the alcohol fraction is significantly reducing BSFC.B20+E15, B20+10 and B20+5 show low BSFC ascendingly with compare to neat diesel and B20 fuels. But BSFC of alcohol blended fuels are getting increased gently with high rpms such as after 1900rpm. High oxygen content of alcohol may help to combust fuel with air more efficiency and latent heat of vaporization of alcohol may cool down the in cylinder temperature which may help for thermal efficiency of the engine. This two points may be the reasons of giving low BSFC when using alcohol blends.

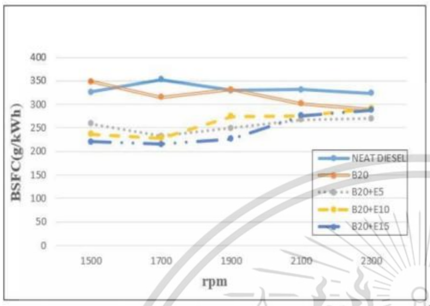


Figure 2: variation of BSFC with rpm

Break Thermal Efficiency(BTE) is the ability of fuel which delivers energy how much that energy can be transformed to mechanical output. BTE of the engine is increased when increasing the fraction alcohol. Diesel has the least BTE. Higher oxygen content of alcohol blends tends to better combustion of the engine cause to higher BTE.

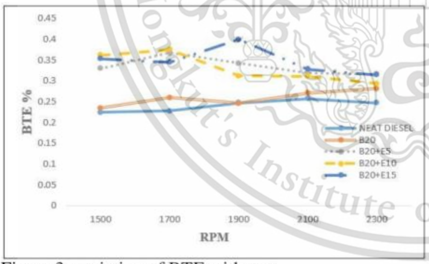


Figure 3: variation of BTE with rpm

3.2 Combustion Characteristics

The variation of in cylinder pressure in 2000rpm with crank angle for different ethanol blended fuels are shown in figure.in peak cylinder pressure curve area of neat diesel and B20 are exactly same. The Max pressure of B20 is 1.5degree advanced compare to neat diesel. When the pressure of neat diesel 60bar at 6degree bTDC pressure of B20 is 52bar.lower Cetane number of B20 may be caused for this manner when compare with neat diesel.in cylinder pressure of B20+E5, B20+E10 and B20+E15 are corresponding 79bar,79bar and 74bar those values are higher than max pressure values of neat diesel and B20.In the speed of 2000rpm, left residual gas may increase the in cylinder temperature which increase the pressure simultaneously. higher content of oxygen in ethanol blend fuel may have chance to react with residual gas and get combusted. These reasons may cause the higher max pressure values of ethanol blend fuels.

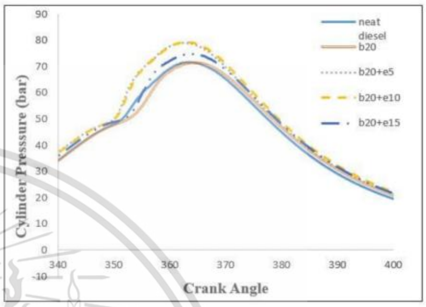


Figure 4: variation of cylinder pressure with crank angle

The net heat release rate for all the tested fuels are shown in figure 4.it is observed that HRR increases with increase of Ethanol fraction. More fuel gets combusted in premixed zone causing to higher heat release rate for ethanol blend fuels. Maximum heat release rate found in B20+E15 and minimum heat release rate found in neat diesel. Due to higher oxygen content, it can be observed that during diffusion phase, the heat release rate is higher for ethanol blends than neat diesel and B20.incresing the ethanol percentage in blended fuel increase premix combustion mode because high ignition lag of ethanol blend fuels. This factor eventually increases the heat release rate and fraction of fuel burned in premixed combustion phase.

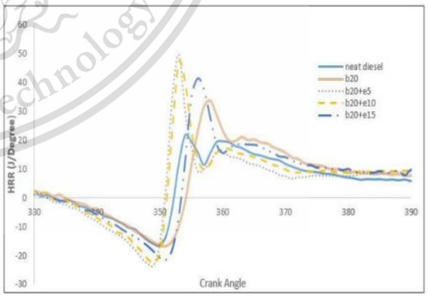


Figure 5: variation of HRR with crank angle

3.3 Emissions

THC emission is significantly less when increasing the fraction of ethanol with B20. The trend of the graphs of all

APPENDIX A

three blends are almost same. THC emission of neat diesel and B20 are getting reduced when rpm is increased. Neat diesel and B20 air mixture trapped in crevices, deposits and quench layer may produce HC emission due to incomplete combustion. Blended fuel with alcohol have high oxygen content which may help to better combustion all the time which leads to less THC emissions compare to neat diesel and B20

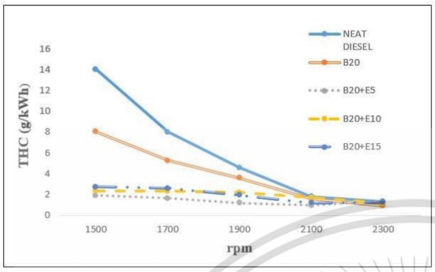


Figure 6: variation of THC with rpm

CO emissions also show results like the trend of THC. Even though neat diesel and B20 show high CO emissions at low rpm and drastically get reduced when increasing the rpm. At the low rpm quench layer might produce CO because of carbon molecules of unburn hydrocarbon may react with oxygen to generate CO. CO emission of B20+E5 little bit higher until 1900rpm when compare to B20+E10 and B20+15.higher oxygen content of B20+E10 and B20+E15 may have great possibility for good combustion than less oxygen concentration of B20+E5.better combustion of high concentration of alcohol fuels may reduce the emission of CO.

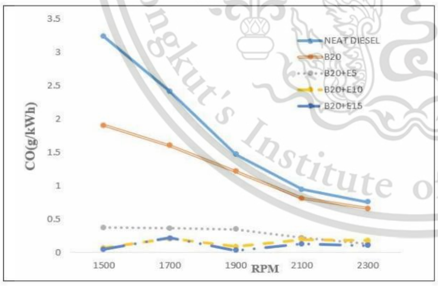


Figure 7: variation of CO emissions with rpm

NOx emission slightly increases when increasing the rpm. neat diesel shows highest NOx emission and trend of graphs of other fuels are almost same. In cylinder temperature is the main factor of forming NOx. In cylinder pressure and temperature are getting high when increasing the rpm which may cause to enhancement of NOx. High oxygen content of blended alcohol fuel may react with N2 and may form NO and NO2.

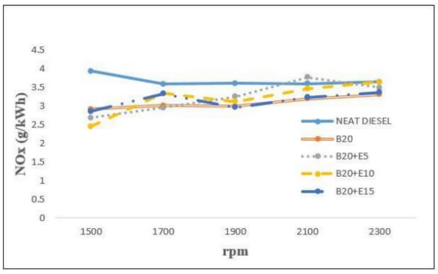


Figure 8: variation of NOx emission with rpm

Soot emission of neat diesel and B20 reduce when increasing the rpm of the engine. Soot emission of alcohol blends slightly increase until 1900 rpm and getting reduced drastically after 1900rpm. Ethanol blended fuels contain more oxygen, which may enhance combustion process and reduce soot emission. Improper combustion at low rpm and burning of heavy lubricant oil may occur increasing the soot level of ethanol blended fuels.

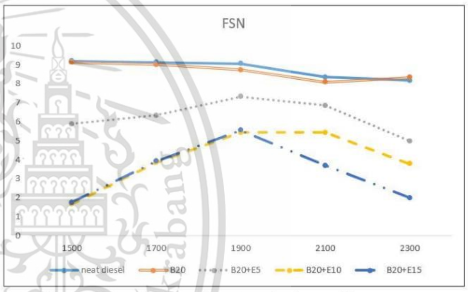


Figure 9: variation of soot emission with rpm

4.CONCLUTIONS

This experiment was based on determine the performances, combustion characteristics and emissions of the Kubota RT140 engine using different kind of fuel blends with alcohol. After examining the results experiment can be concluded that,

- Torque of the engine is reduced when increasing the ethanol percentage and highest torque was given by diesel
- BSFC is reduced when increasing the ethanol percentage, BTE is increased when increasing the fraction of ethanol
- Neat diesel has least in cylinder pressure and HRR value compare to ethanol blends and B20+E5 has highest HRR value.
- THC and CO emissions are reduced significantly with rpm. B20+ ethanol blends have least

APPENDIX A

emissions of THC and CO compare to neat diesel and B20. NOx emission is increased with rpm and neat diesel has highest NOx emission. FSN values of blends tends to increasing until 1900 rpm and reduced drastically after 1900 rpm

5. REFERENCES

(1)Engen: disel engine emissions<http://www.engenoil.com/home/apps/content/environmnet_soceity/cleaner_fuels/did_you_know/diesel_engine_emissions.aspx>.

(2) Datta, Ambarish, and Bijan Kumar Mandal. "Impact of alcohol addition to diesel on the performance combustion and emissions of a compression ignition engine." *Applied Thermal Engineering* 98 (2016): 670-682.

(3) Imran, A., et al. "Review on alcohol fumigation on diesel engine: a viable alternative dual fuel technology for satisfactory engine performance and reduction of environment concerning emission." *Renewable and Sustainable Energy Reviews* 26 (2013): 739-751.

(4) Liotta, Frank J., and Daniel M. Montalvo. *The effect of oxygenated fuels on emissions from a modern heavy-duty diesel engine*. No. 932734. SAE Technical Paper, 1993.

(5) Rakopoulos, D. C., et al. "Combustion heat release analysis of ethanol or n-butanol diesel fuel blends in heavy-duty DI diesel engine." *Fuel* 90.5 (2011): 1855-1867.

(6) Park, Su Han, In Mo Youn, and Chang Sik Lee. "Influence of ethanol blends on the combustion performance and exhaust emission characteristics of a four-cylinder diesel engine at various engine loads and injection timings." *Fuel* 90.2 (2011): 748-755.

(7) Bilgin, Atilla, Orhan Durgun, and Zehra Sahin. "The effects of diesel-ethanol blends on diesel engine performance." *Energy sources* 24.5 (2002): 431-440.

APPENDIX B

COMMENTS AND QUESTIONS

Topic:
The comparison between blended and fumigated method to apply ethanol fuel in the diesel engine

Date:
15 August 2018

Place:
NSTDA Rangsit 2 (Thailand)
and
Tokyo Institute of Technology (Japan)

Examiners:
Asst Prof .Dr .Preecha Karin (KMITL)
Dr. Manida Tongroon (NSTDA)
Prof. Dr. Hidenori Kosaka
Asst Prof. Dr Chinda Charoenphonphanich (KMITL)
Dr Nuwong Chollacoop (NSTDA)

Student:
Surith Dulanjala De Silva

APPENDIX B

Question: how ethanol air mixture behavior while the main injection and combustion of Biodiesel?

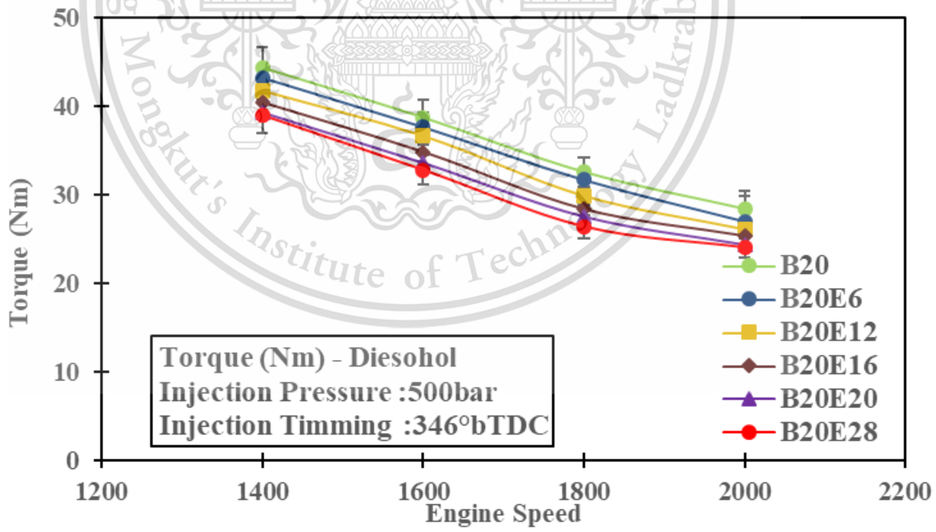
In case of in-cylinder direct injection concept, injection of liquid pilot fuel (B20) takes place first, followed soon after by the injection of ethanol. During the atomization phase of pilot fuel, injected pilot fuel mixes with in-cylinder air, which is called as premixing process. This pre-mixture leads to rapid increase of pressure and temperature once ignition starts. However, ethanol which is injected right after pilot fuel injection has high heat of evaporation and it prevents from prompt increase of pressure and temperature due to heat absorption. The heat from premixed combustion process expedites evaporation of primary fuel. In results, peak heat release value in premix combustion decreases and combustion duration of primary fuel can be reduced.

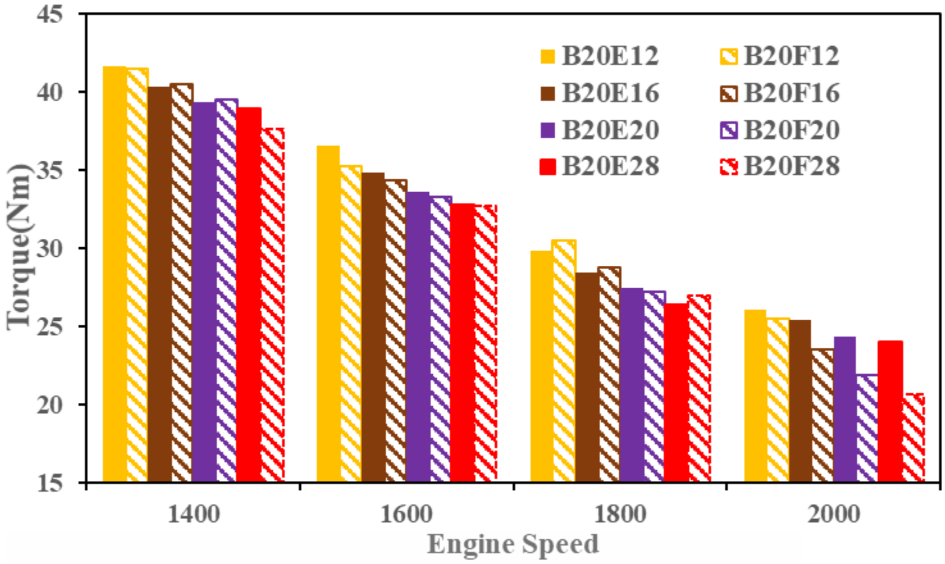
With help from velocity of intake air and swirls in the cylinder, ethanol is well mixed with air. To ignite low reactivity homogenous mixture, the biodiesel is injected close to TDC to induce in-cylinder combustion. Major characteristics of dual fuel combustion can be defined as the combination between mixing-controlled diffusion combustion of CI engine and turbulent flame propagation of SI engines. In dual fuel engine, fuel ratio between pilot fuel and primary fuel (ethanol fuel) highly affects combustion process as it leads to change of ignition delay. Increase of ethanol fuel injection prolongs ignition delay due to its high heat of evaporation value. The larger portion of ethanol fuel enhances the effect of heat absorption and the lower temperature reduces chemical reaction speed. In this respect, the larger portion of ethanol fuel leads to retarded start of combustion. In addition, with increase of ethanol fuel, amount of heat release in premixed combustion phase increases as more pilot fuel is premixed due to longer ignition delay.

APPENDIX B

Comments: Modifying graphs for better understanding

B20 was blended with ethanol 10%, 20%, 25%, 30% and 40% by volume. All of them had been labeled as B20E10, B20E20, B20E25, B20E30 and B20E40. Fumigation of ethanol was done 12%, 16%, 20% and 28% by energy using same energy concentration of blended fuels starting from 20% to 40% by volume of ethanol correspondingly. Ethanol fumigation was labeled as B20F12, B20F16, B20F20 and B20F28. For the comparison as example, B20E10 and B20F12 as same ethanol energy input. Nut numerical value of the names were hard to understand. Names of blended fuels were changed to B20E6, B20E12, B20E16, B20E20 and B20E28 which was the actual ethanol energy input of blended fuels. Following graphs shows the way of modification for better understanding with error bar





45
40
35
30
25
20
15
Torque(Nm)
1400
1600
1800
2000
Engine Speed
B20E12
B20F12
B20E16
B20F16
B20E20
B20F20
B20E28
B20F28

The comparison between blended and fumigated method to apply ethanol fuel in a diesel engine

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Outline

- Introduction
- Objectives
- Literature Review
- Methodology
- Results
- Conclusions and Recommendations

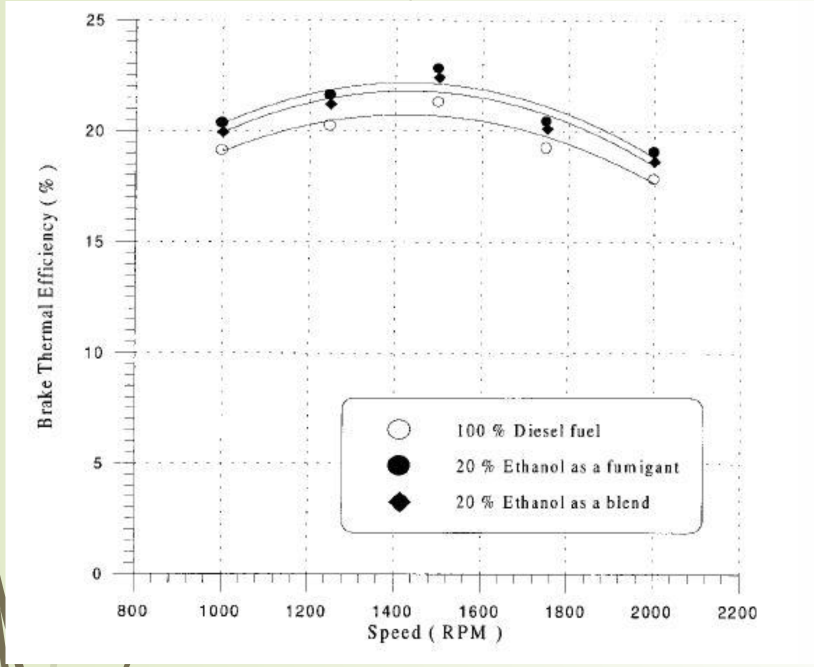
Introduction

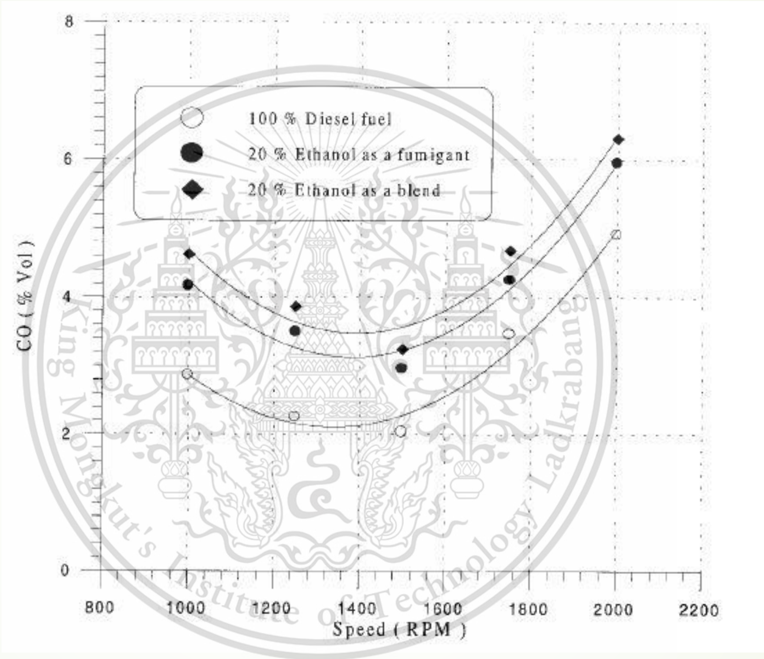
- Today world is affected by a huge energy crisis of satisfying day to day activities of human beings, such as operation of factories and automobiles due to lack of crude oil.
- Excessive usage of crude oil has harmed the environmental conditions in many ways. Among them, emission from automobiles play a major role.
- Alcohols have become an alternative fuel because of it's liquid nature ,high oxygen content ,high octane number, and ability to being produced by renewable biomass
- Bio diesel blends with ethanol may help to reduce the importing cost of crude oil to Thailand , if it is used as fuel for automobile industry

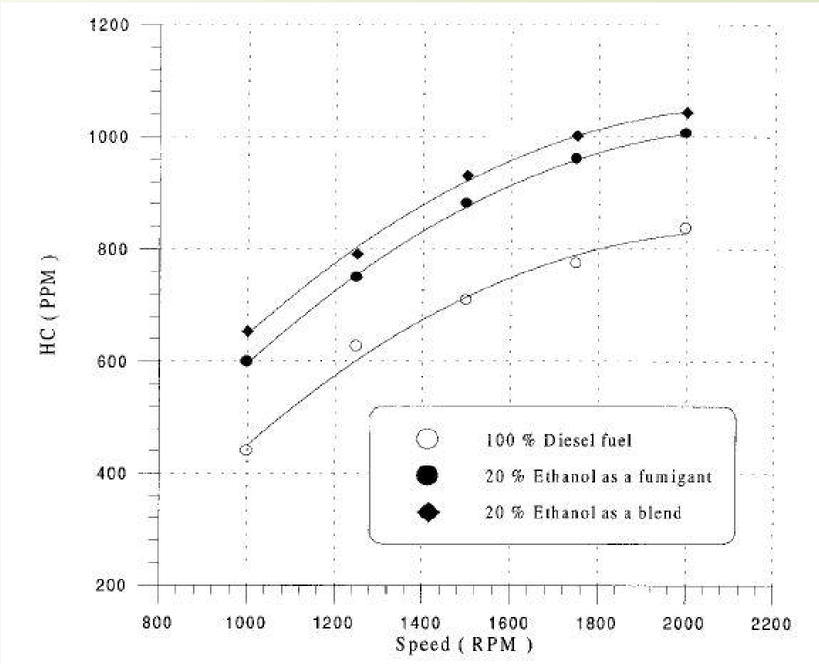
Objectives

- To investigate performances combustion characteristics and emissions of various portion of blend bio diesel with ethanol
- To investigate performances, combustion characteristics and emissions of ethanol fumigation with bio diesel direct injection
- To compare the results of blended fuel and fumigated method to conclude which method of injection, which type of fuel obtained are in the acceptable range

Literature Review

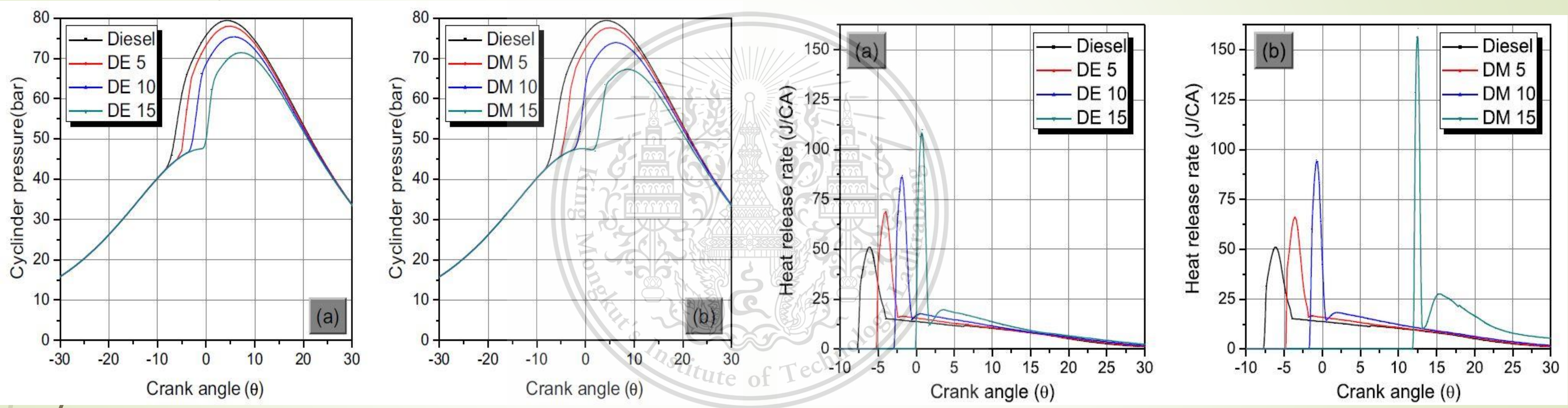






Source: Abu-Qudais, M., Haddad, O., & Qudaisat, M. (2000). The effect of alcohol fumigation on diesel engine performance and emissions. *Energy conversion and management*, *41*(4), 389-399.

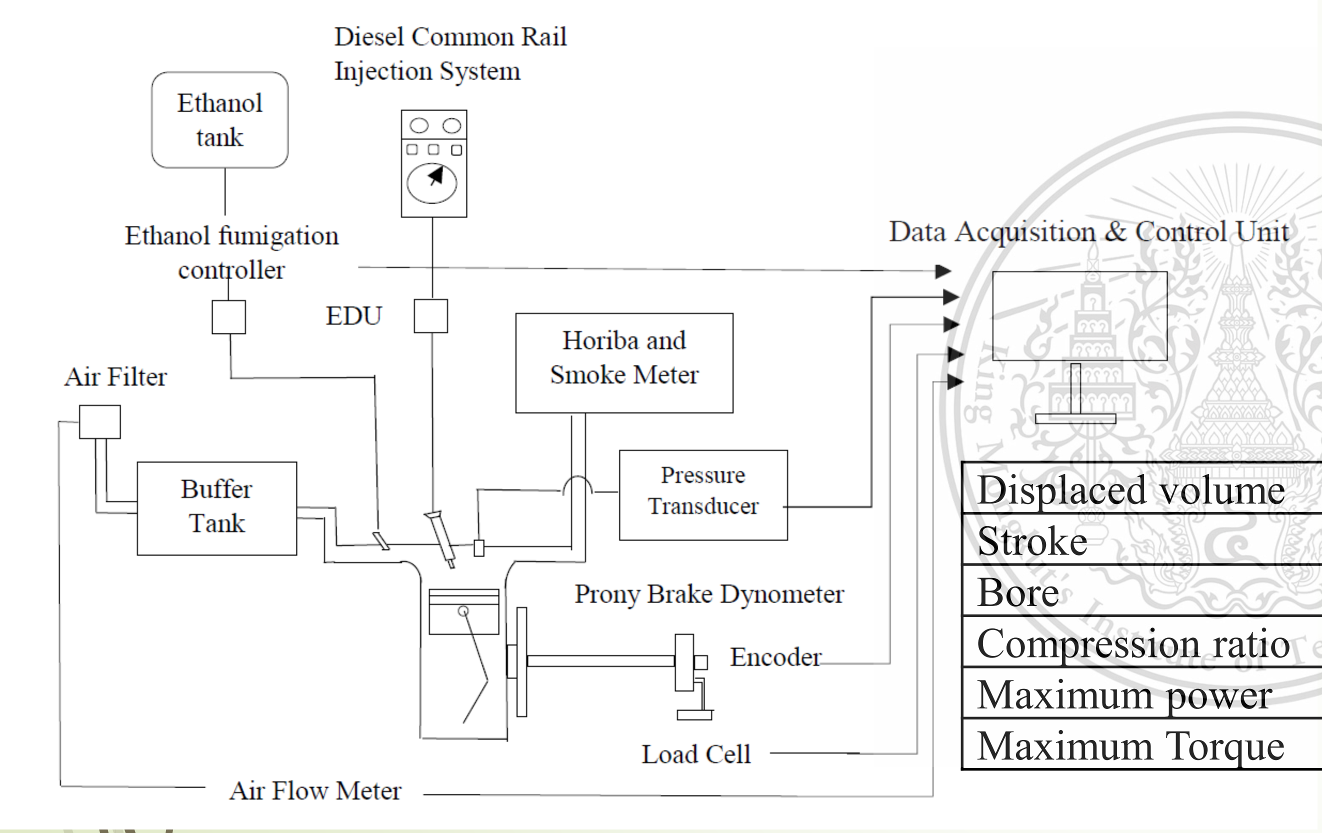
Literature Review



Source: Datta, A., & Mandal, B. K. (2016). Impact of alcohol addition to diesel on the performance combustion and emissions of a compression ignition engine. *Applied thermal engineering*, *98*, 670-682.

Methodology

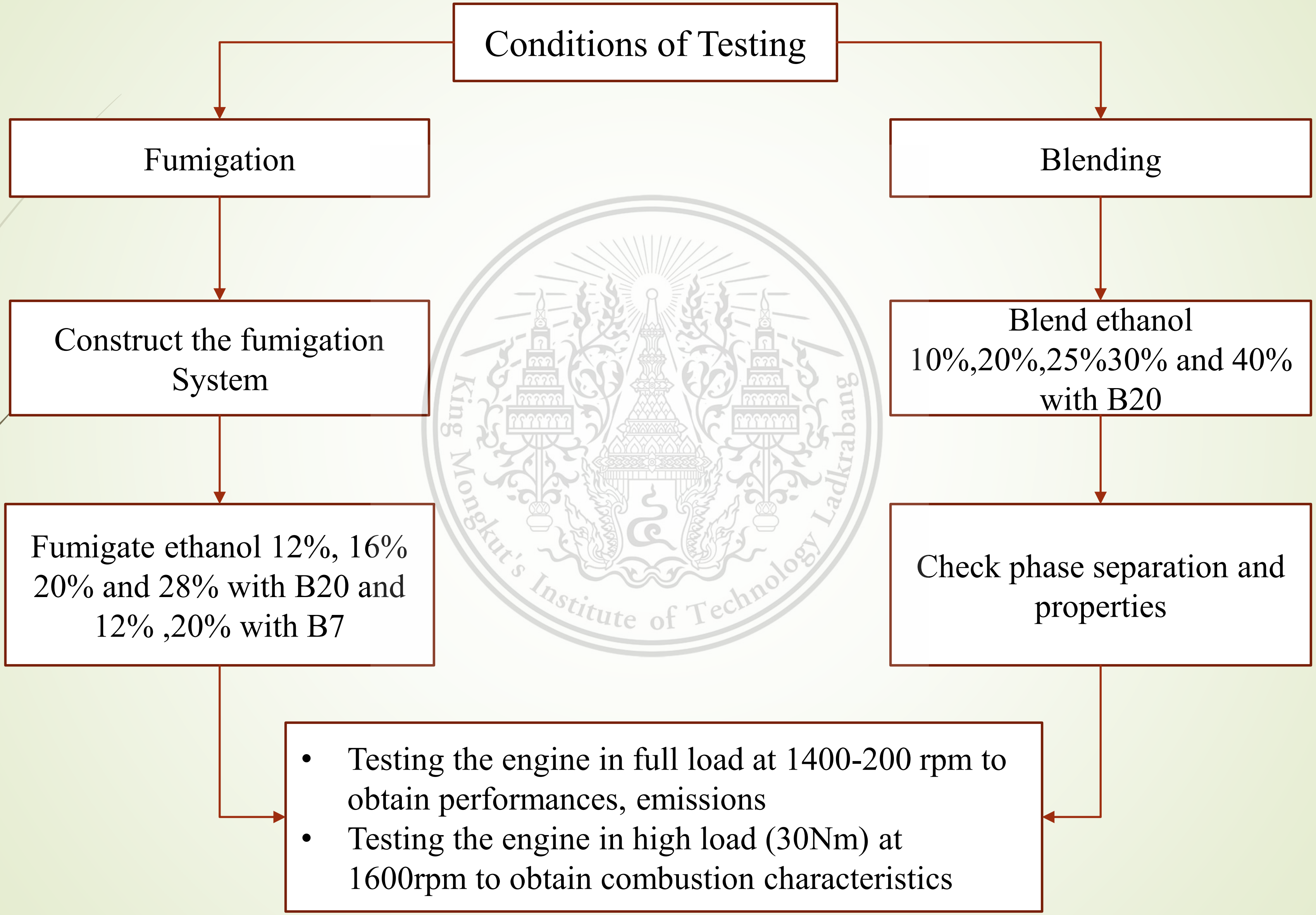
Engine Apparatus



Displaced volume	709 cm^3
Stroke	96 mm
Bore	97 mm
Compression ratio	18:1
Maximum power	10.4kW @ 2400rpm
Maximum Torque	50Nm @1600rpm

- The gaseous emissions were measured by HORIBA MEXA 1600D which measures CO, NOx and THC
- K type thermocouples were used to measure engine oil temperature, exhaust temperature, intake air temperature and water in and out temperature
- Teledyne Hastings HFM 200 S/W air flow meter was used to measure the flow rate of intake air
- Engine load was measured by Minebe U3B1-220k-B load cell which is fixed on the base with maximum capacity of 1.96kN

Methodology



Methodology

Injection duration for different blended fuels and fumigation rates

$$Energy\ input = \frac{Heating\ value\ of\ B7}{Fuel\ consumption\ at\ desired\ rpm}$$

$$Energy\ input\ at\ 1400RPM = \frac{44800kJ/kg}{0.43355g/cycle}$$

$$Energy\ input\ at\ 1400RPM = 1942.33088J/cycle$$

$$Fuel\ consumption\ of\ B20\ at\ 1400rpm = \frac{Energy\ input\ of\ B7\ at\ 1400rpm}{Heating\ value\ of\ B20}$$

$$Fuel\ consumption\ of\ B20\ at\ 1400rpm = \frac{1942.33088J/cycle}{44100kJ/kg}$$

$$Fuel\ consumption\ of\ B20\ at\ 1400rpm = 0.044043\ \text{g/cycle}$$

Methodology

Injection duration for different blended fuels and fumigation rates

Injection Duration of B20

RPM	B7 g/cycle	B 7J/cycle	B20 g/cycle	Duration
1400	0.0433556	1942.33088	0.044043784	1639.0314
1600	0.03853483	1726.360384	0.039146494	1457.3781
1800	0.03253376	1457.512448	0.033050169	1286.4951
2000	0.02787166	1248.650368	0.028314067	1167.8817

Injection Duration of B20E10

RPM	B20 g/cycle	B20 J/cycle	B20E10 g/cycle	Duration
1400	0.0440438	1942.3309	0.04573634	1737.199
1600	0.0391465	1726.3604	0.040650852	1500.54
1800	0.0330502	1457.5124	0.034320252	1320.948
2000	0.0283141	1248.6504	0.029402147	1192.953

Injection duration of 16% ethanol fumigation

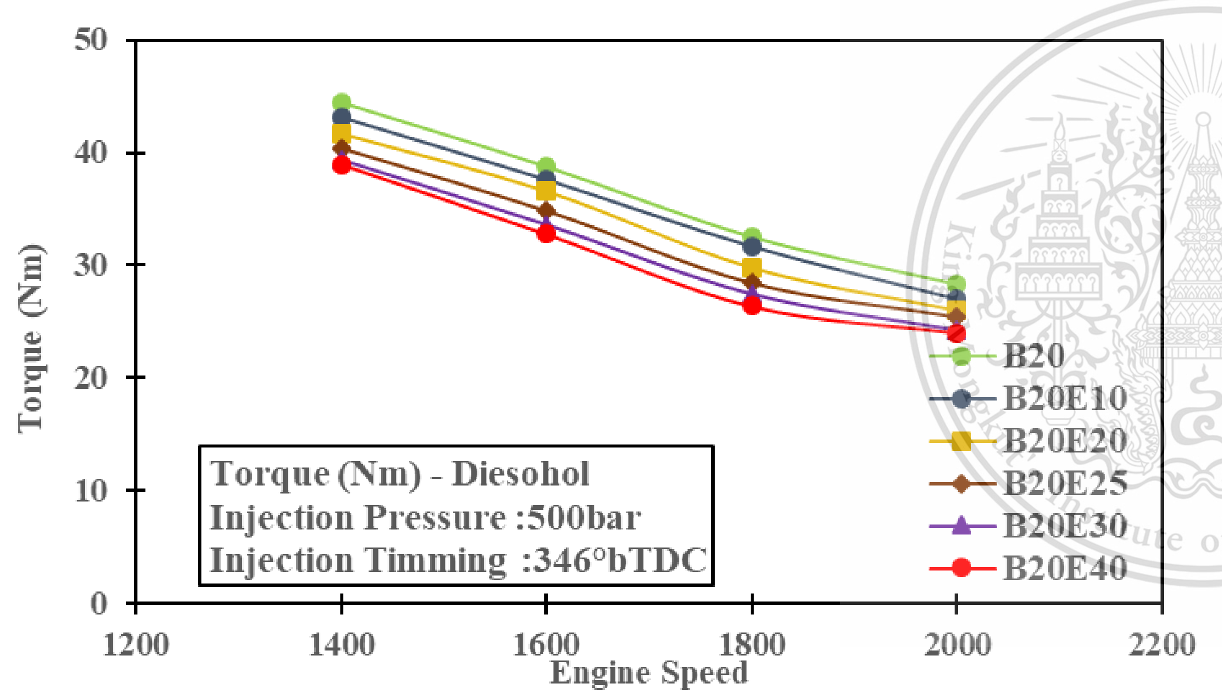
16% ethanol fumigation			
rpm	B7 J/cycle	B20 g/cycle	duration
1400	1631.558	0.036997	1395.782
1600	1450.143	0.032883	1282.17
1800	1224.31	0.027762	1155.165
2000	1048.866	0.023784	1057.642
rpm	B7 J/cycle	Ethanol g/cycle	duration
1400	310.7729	0.011553	1433.984
1600	276.2177	0.010268	1247.19
1800	233.202	0.008669	1150.066
2000	199.7841	0.007427	1082.696

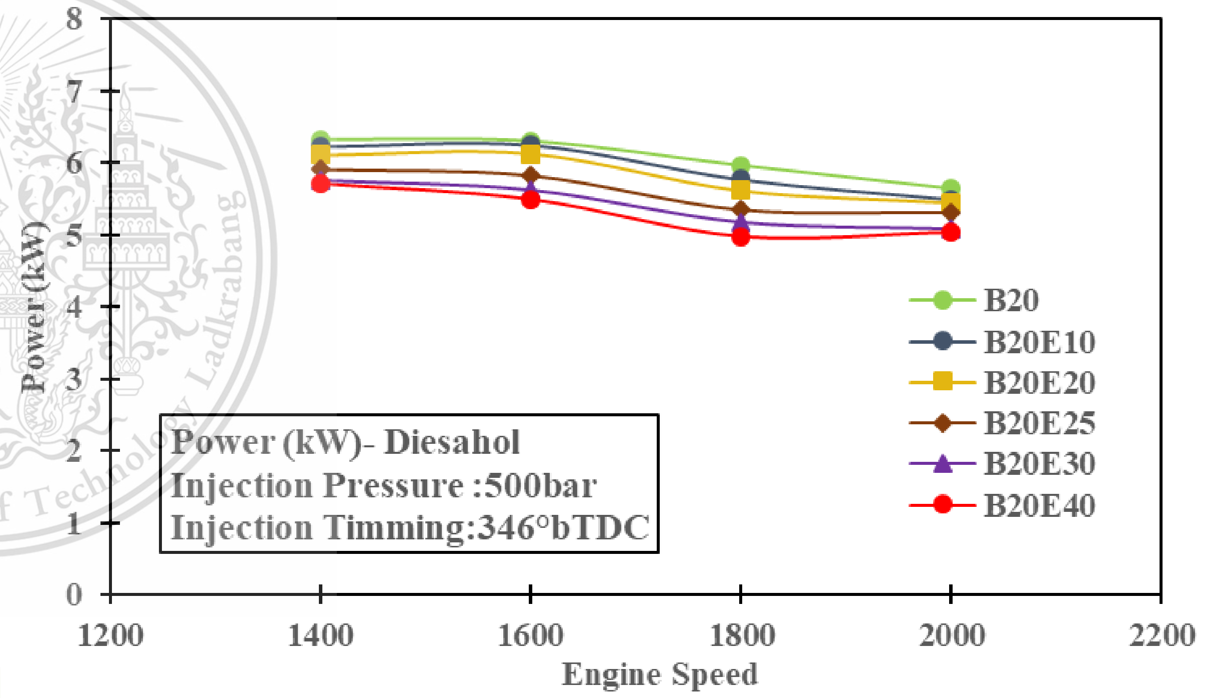
Injection duration of 20% ethanol fumigation

20% ethanol fumigation			
rpm	B7 J/cycle	B20 g/cycle	duration
1400	1553.865	0.035235	1346.557
1600	1381.088	0.031317	1241.646
1800	1166.01	0.02644	1124.704
2000	998.9203	0.022651	1027.36
rpm	B7 J/cycle	Ethanol g/cycle	duration
1400	388.4662	0.014441	1919.706
1600	345.2721	0.012835	1645.602
1800	291.5025	0.010837	1327.253
2000	249.7301	0.009284	1190.186

Results- Performances of Diesohol method

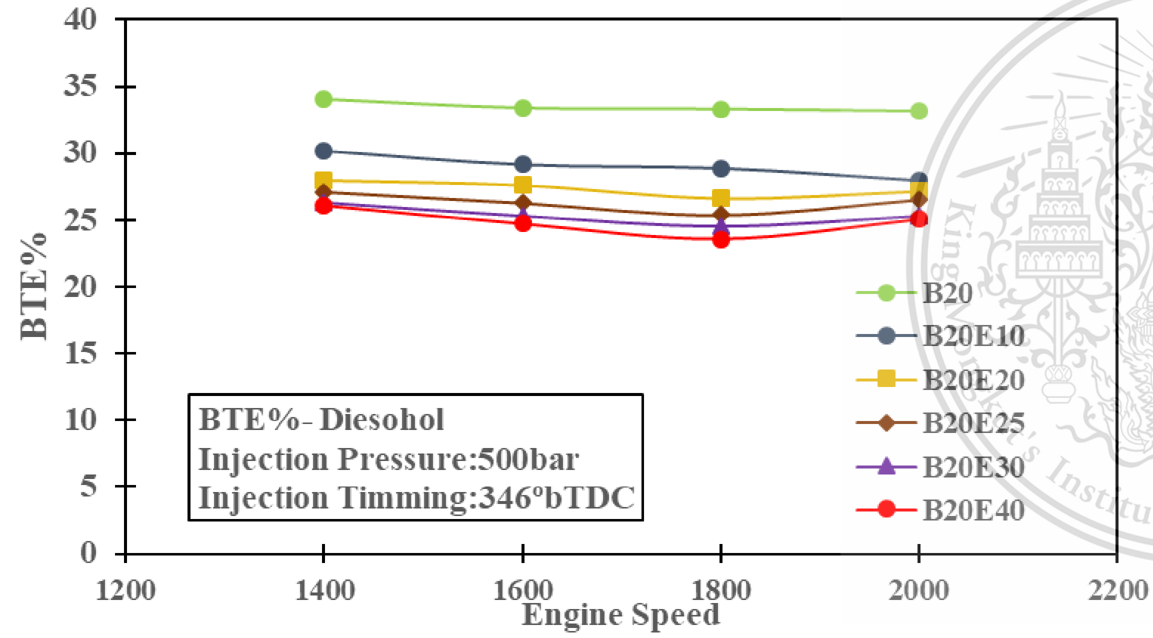
Variation of torque and power with engine speed

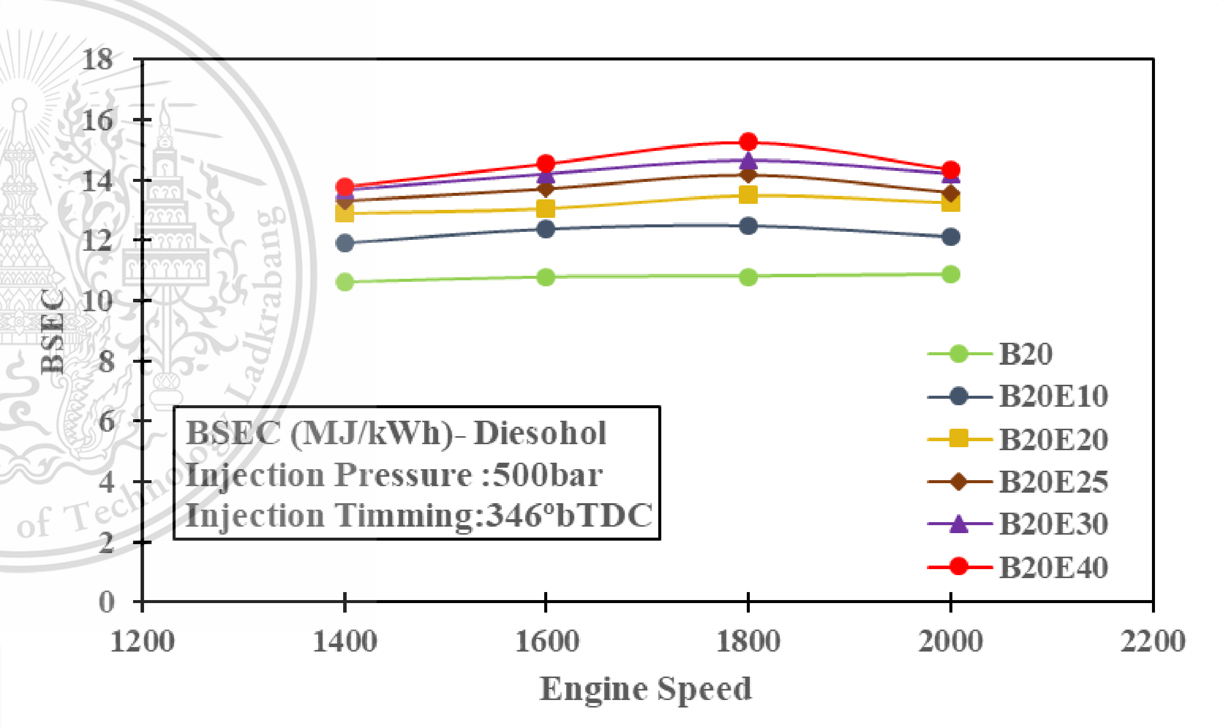




Results- Performances of Diesohol method

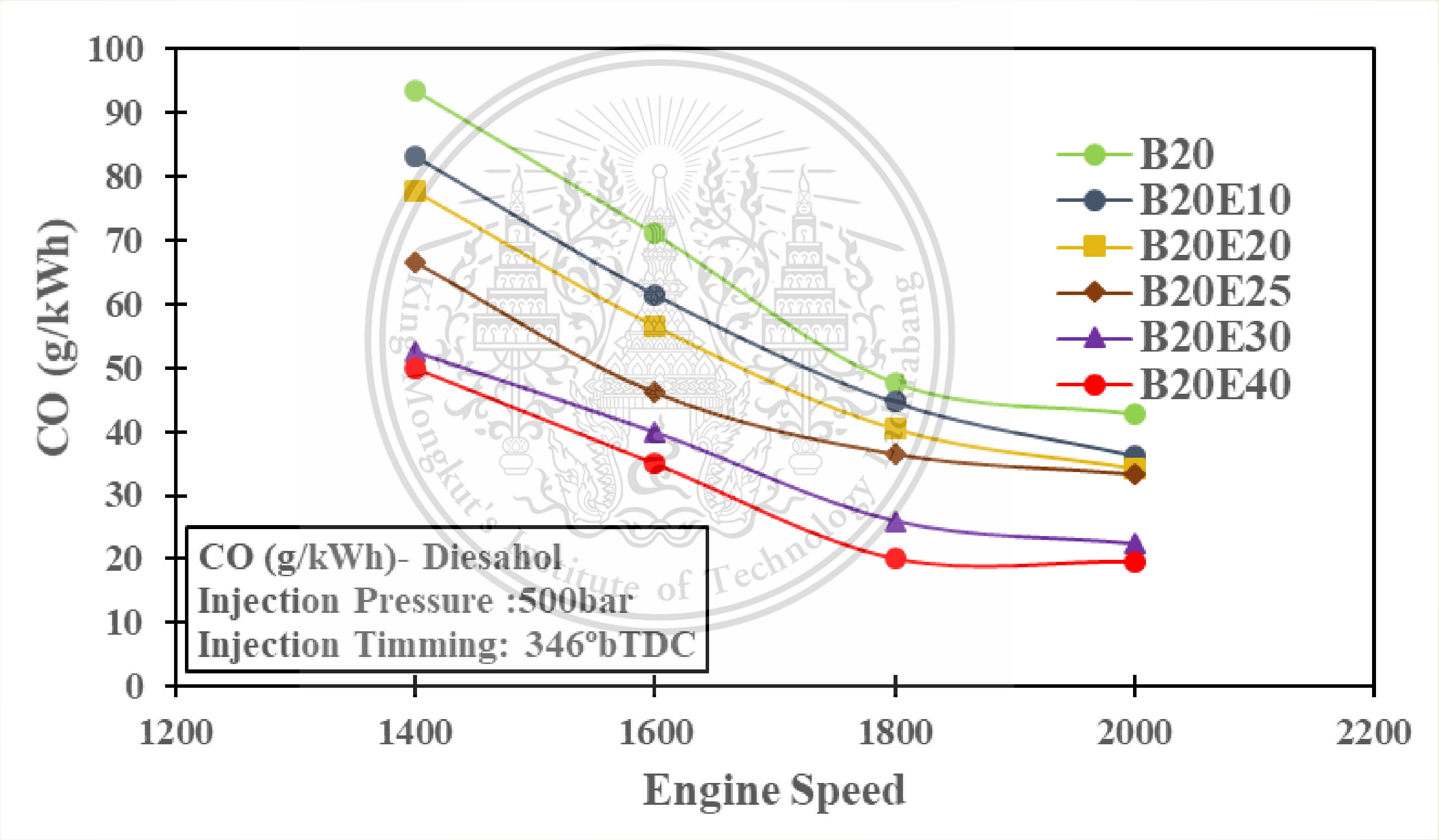
Variation of Brake Thermal Efficiency(BTE) and Brake Specific Energy Consumption(BSEC) with engine speed





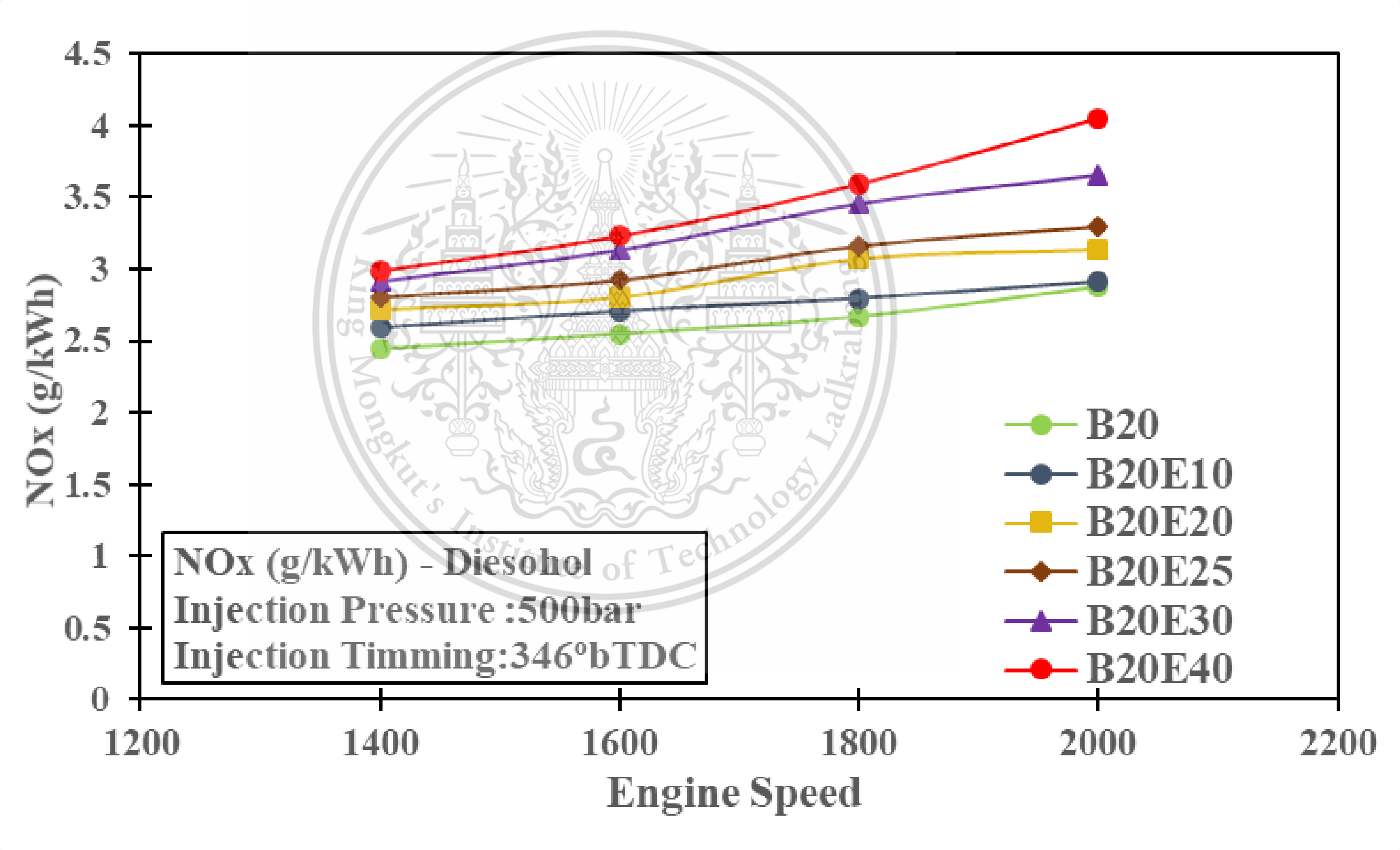
Results- Emissions of Diesohol method

Variation of CO emission with engine speed



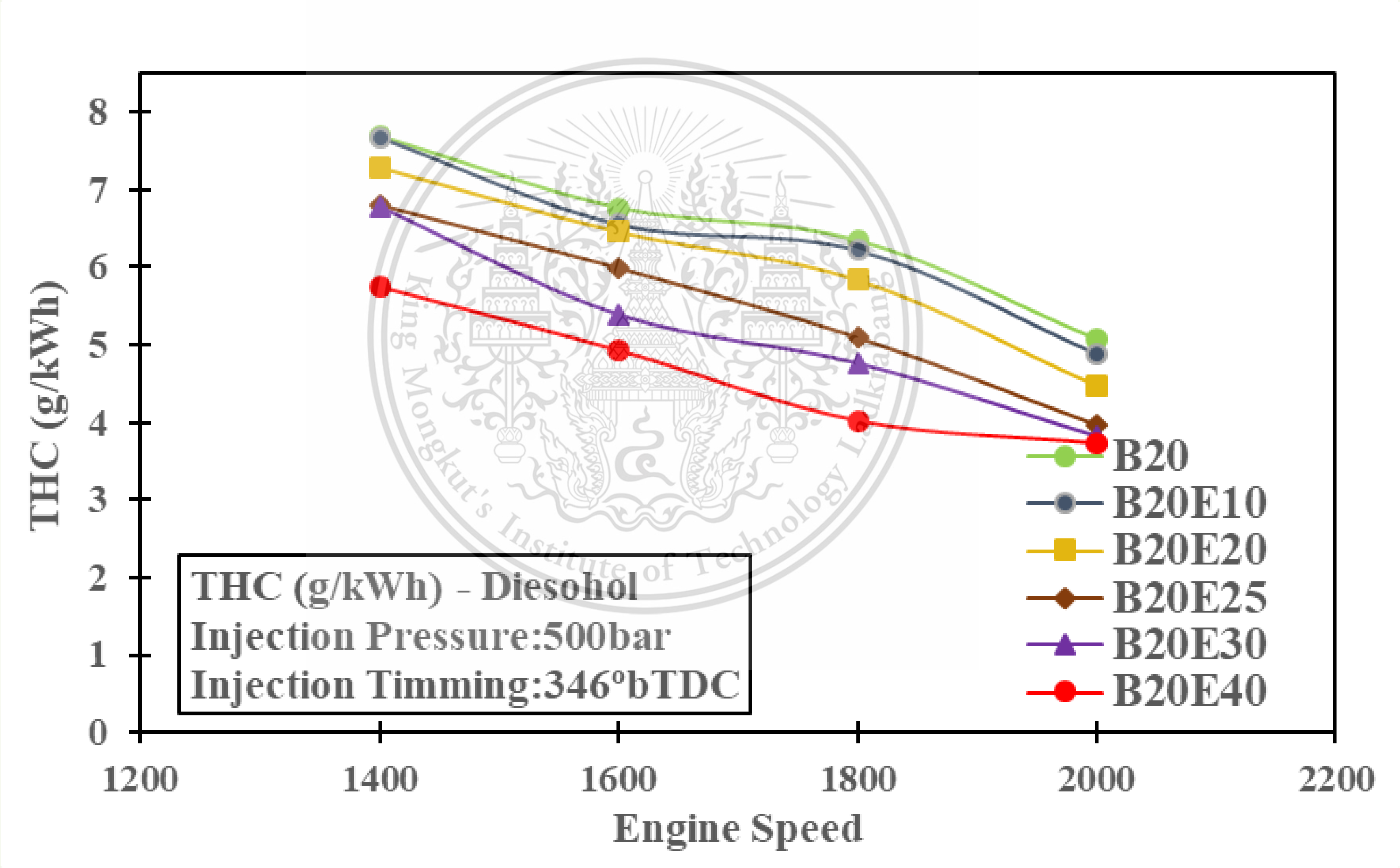
Results- Emissions of Diesohol method

Variation of NO_x emission with engine speed



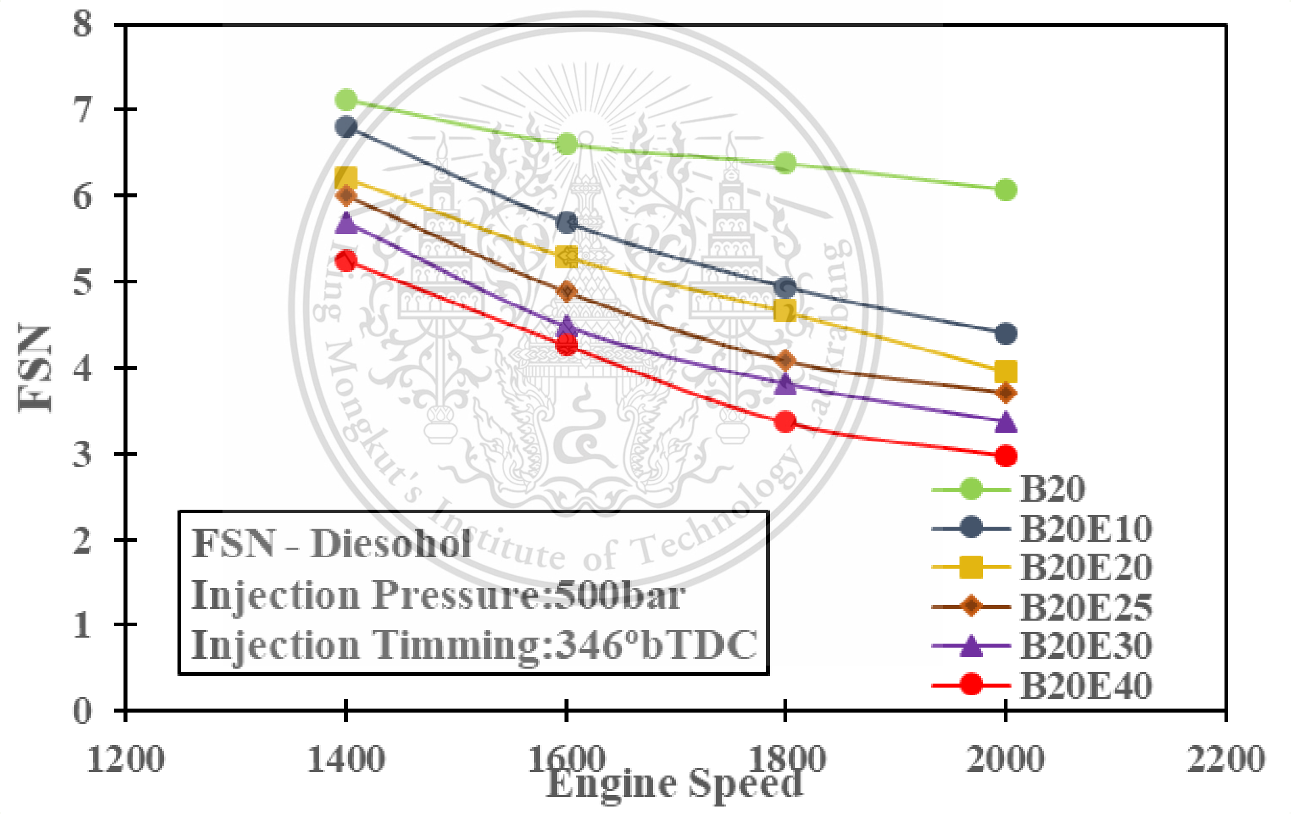
Results- Emissions of Diesohol method

Variation of THC emission with engine speed

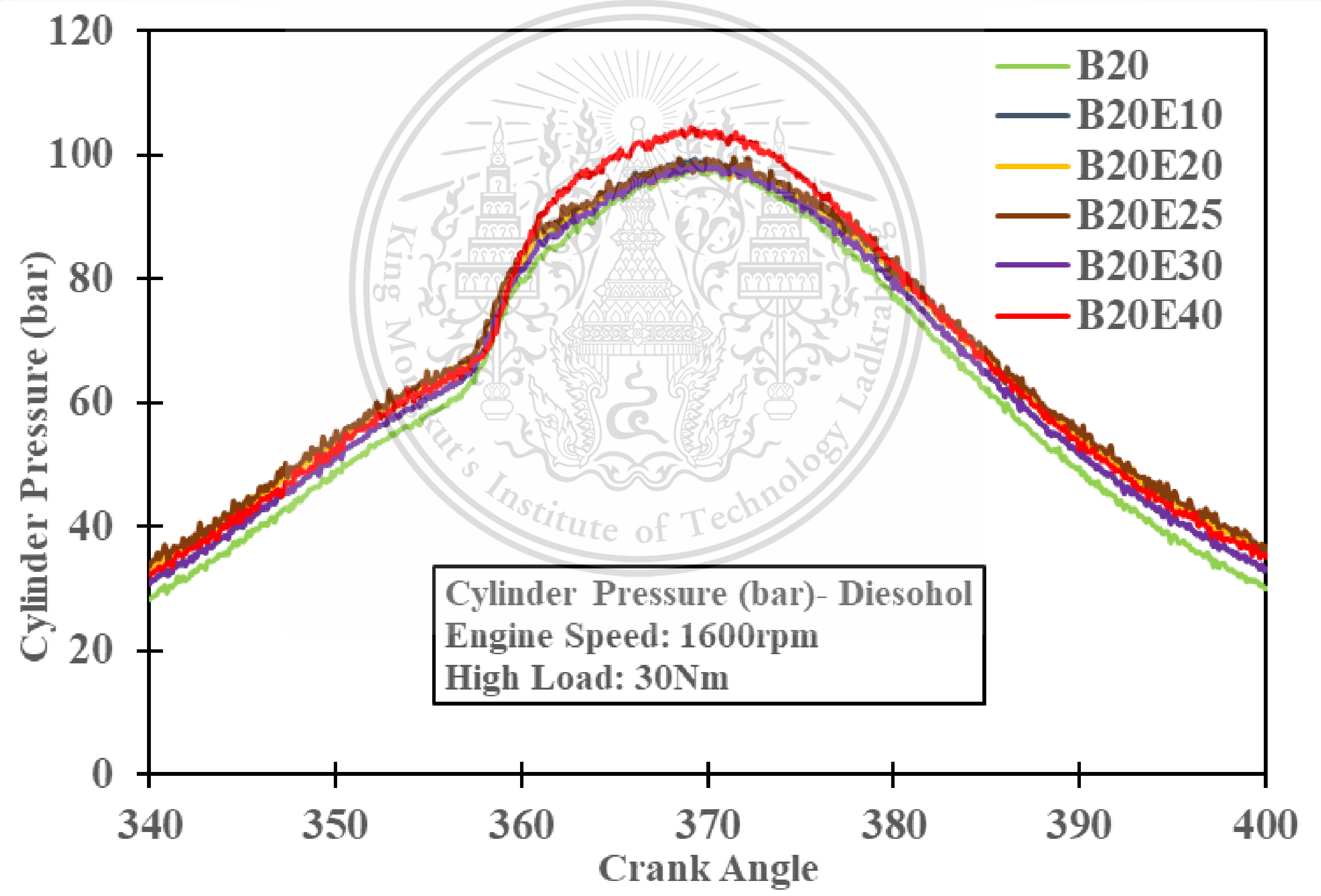


Results- Emissions of Diesohol method

Variation of Soot emission with engine speed

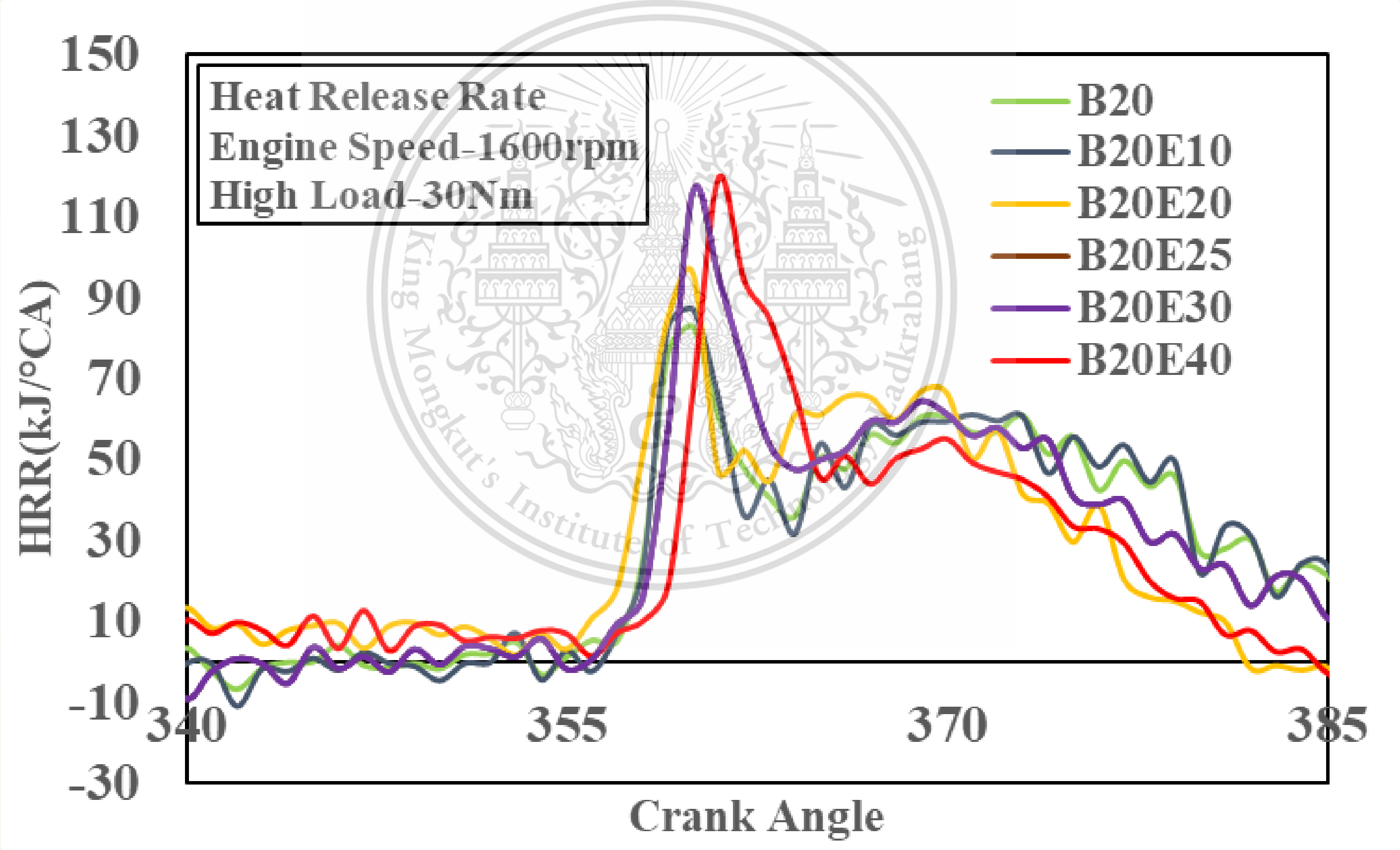


Variation of Cylinder Pressure with crank angle



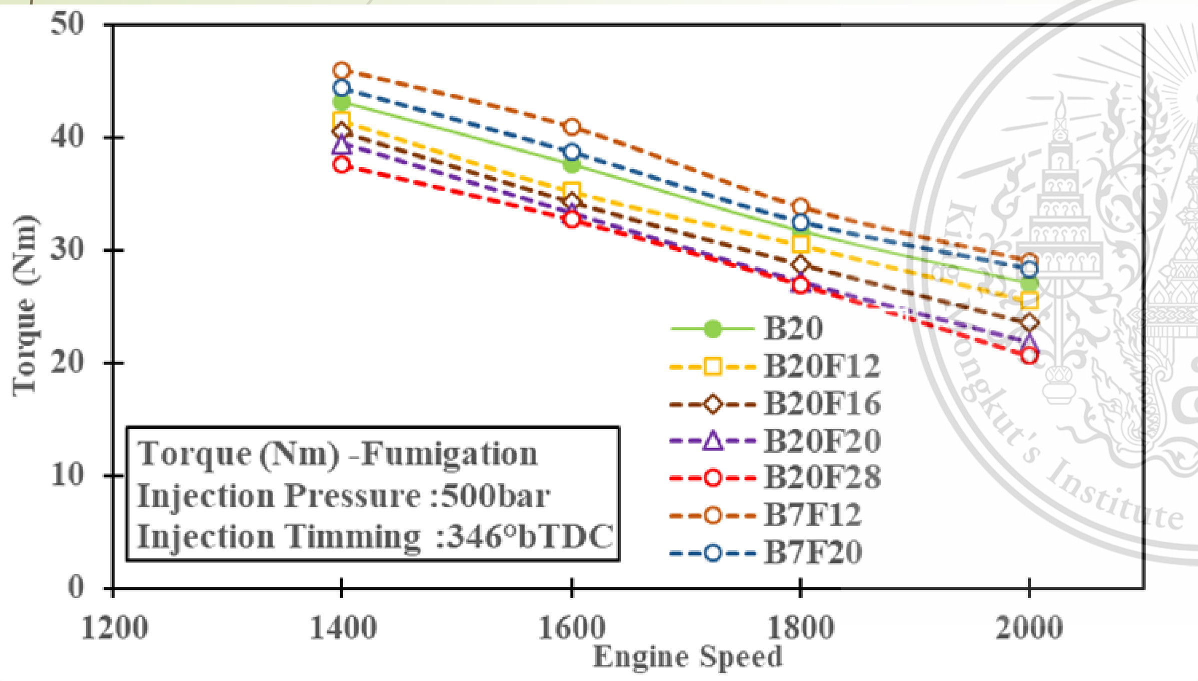
Results- Combustion Characteristics of Diesohol method

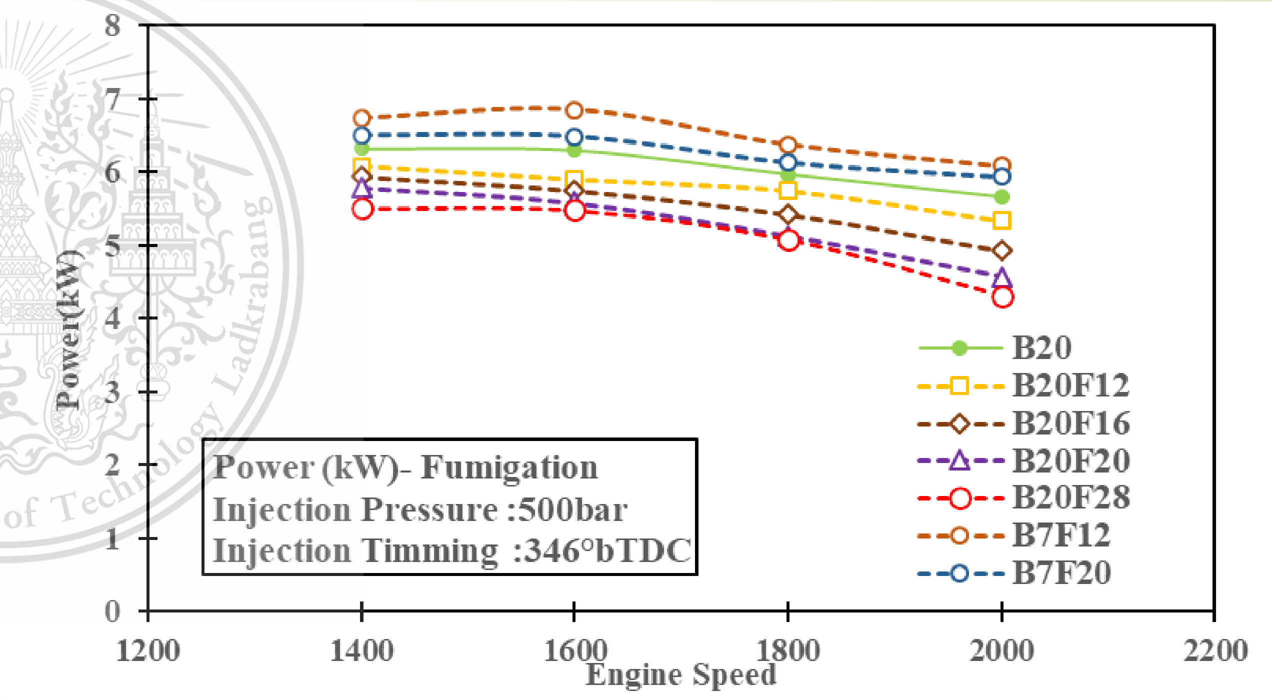
Variation of heat release rate with crank angle



Results- Performances of Fumigation method

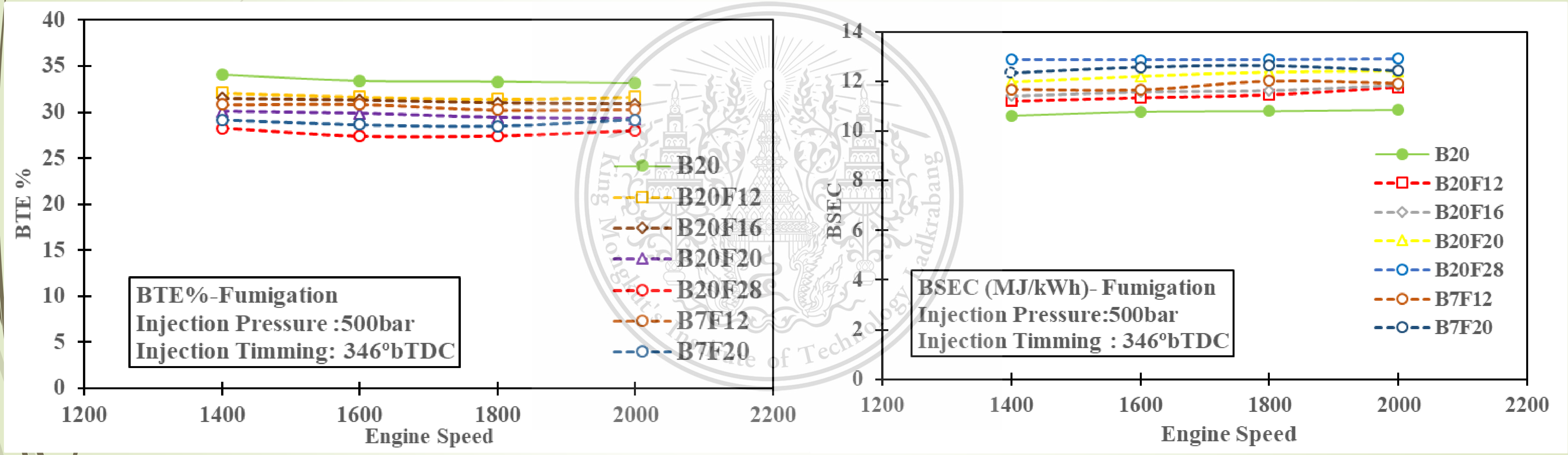
Variation of torque and power with engine speed





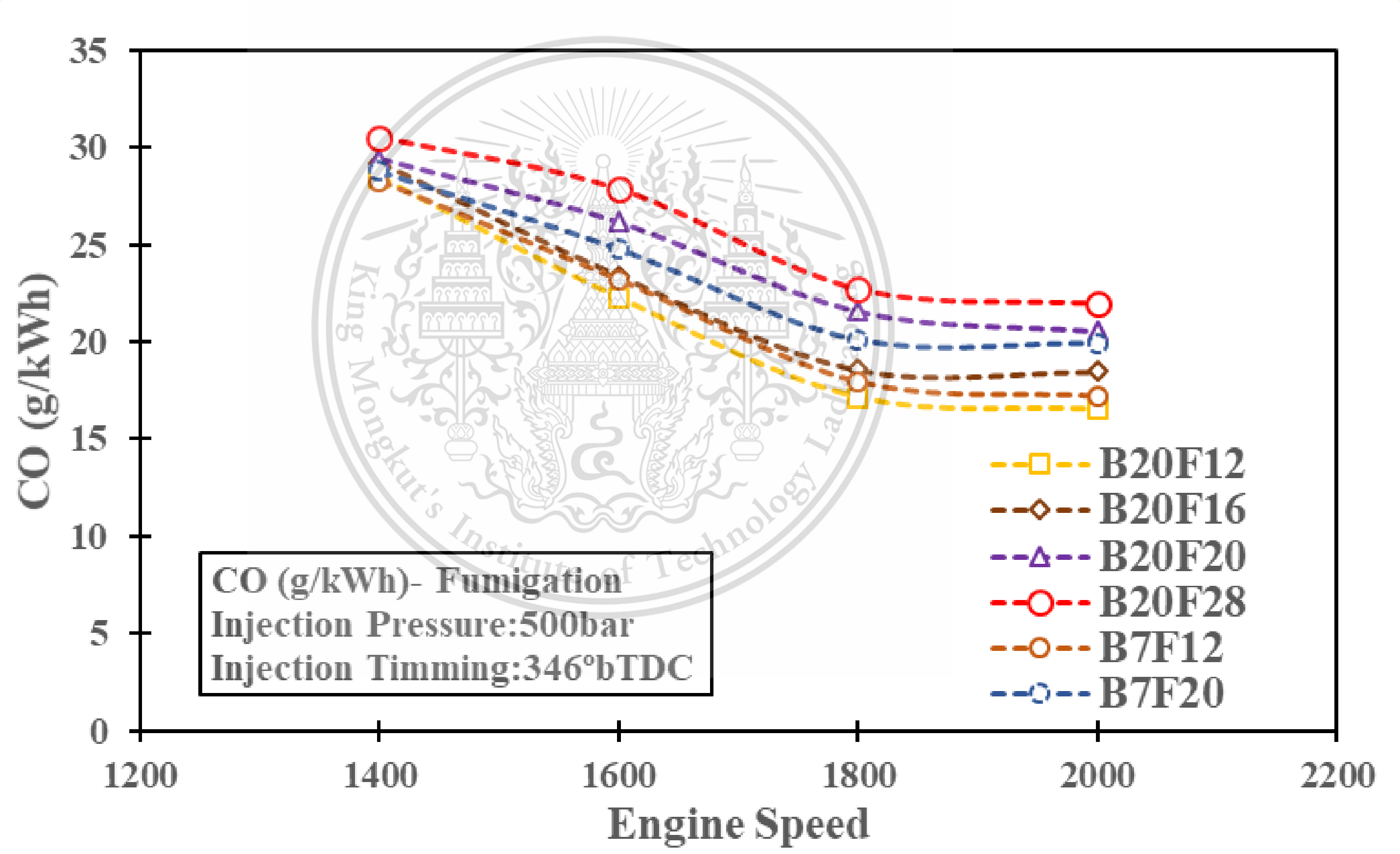
Results- Performances of Fumigation method

Variation of Brake Thermal Efficiency(BTE) and Brake Specific Energy Consumption(BSEC) with engine speed



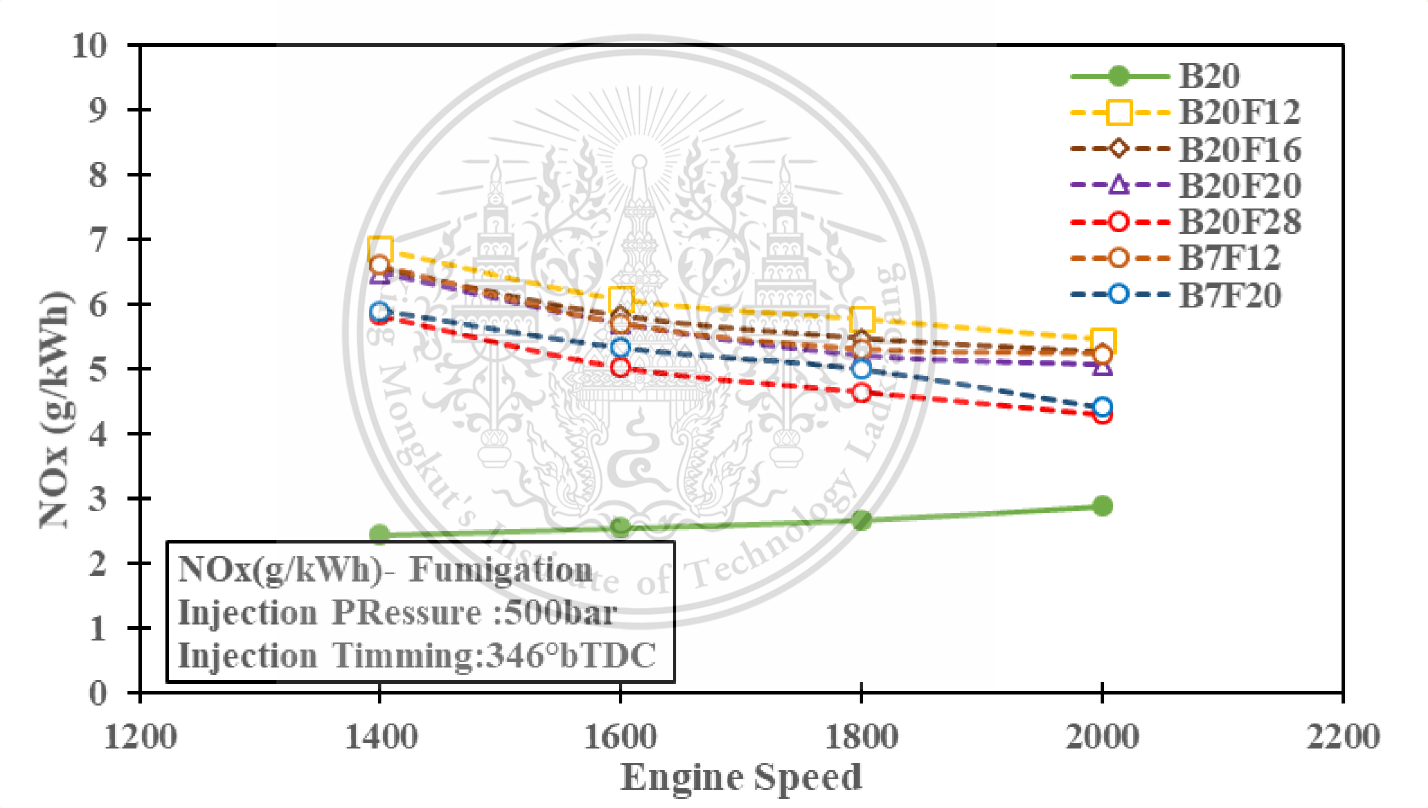
Results- Emissions of Fumigation method

Variation of CO emission with engine speed



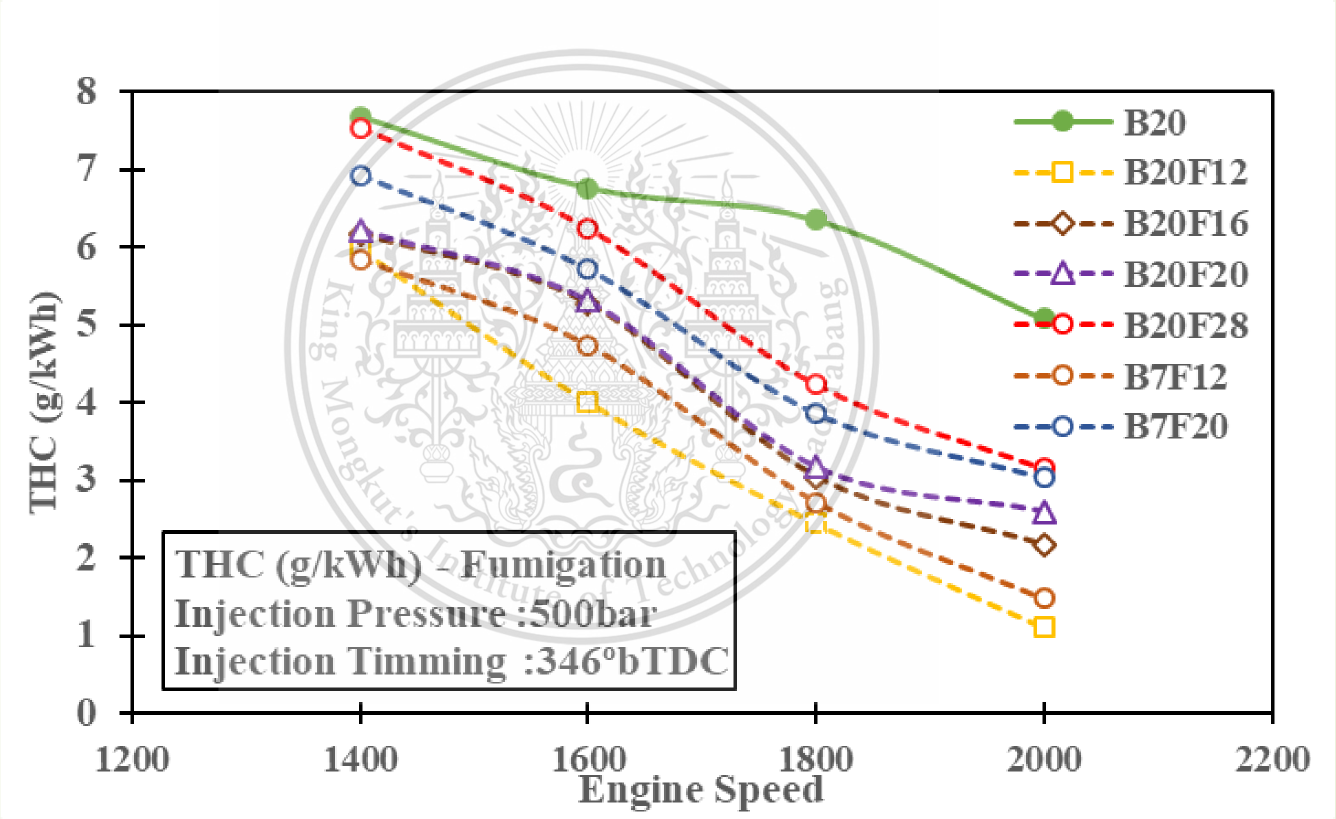
Results- Emissions of Fumigation method

Variation of NO_x emission with engine speed



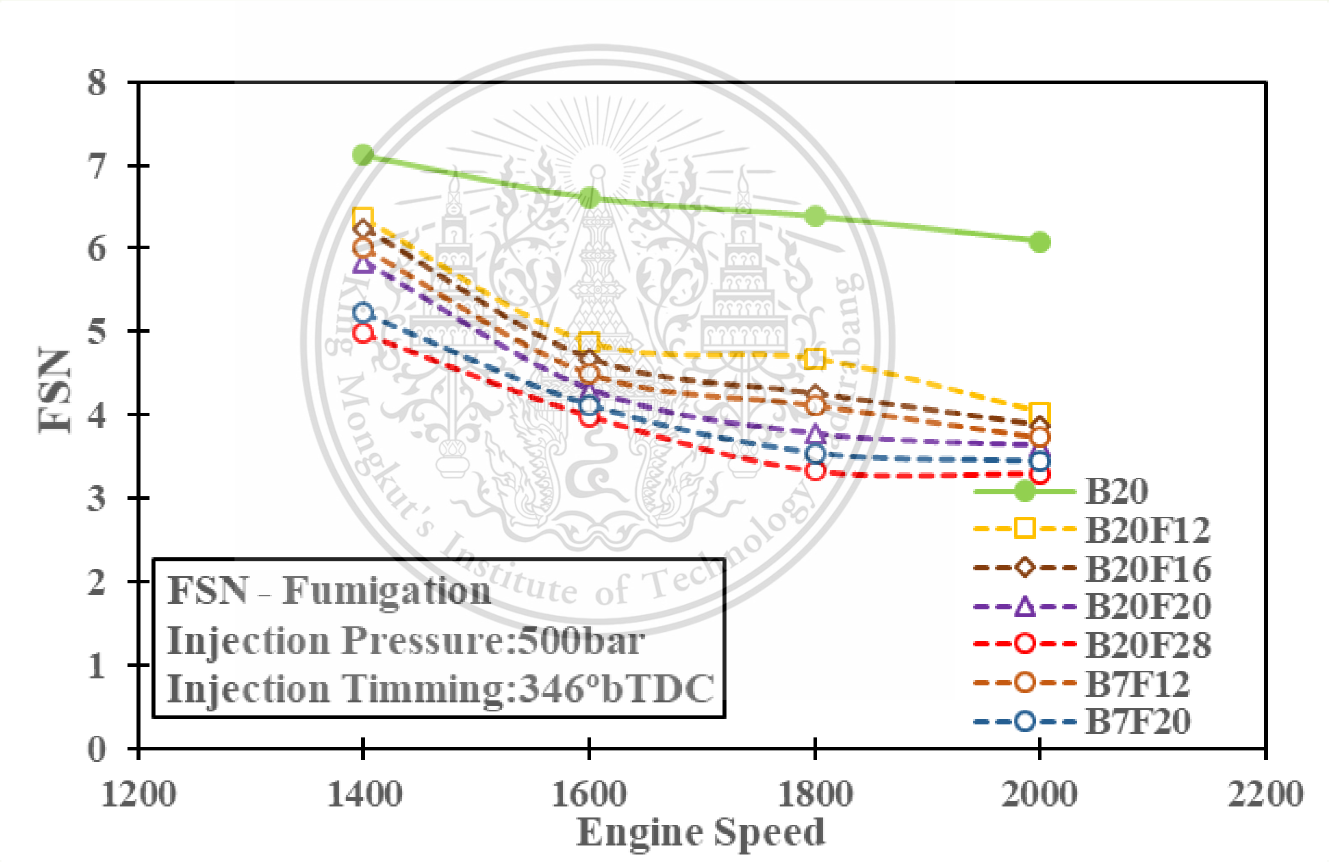
Results- Emissions of Fumigation method

Variation of THC emission with engine speed



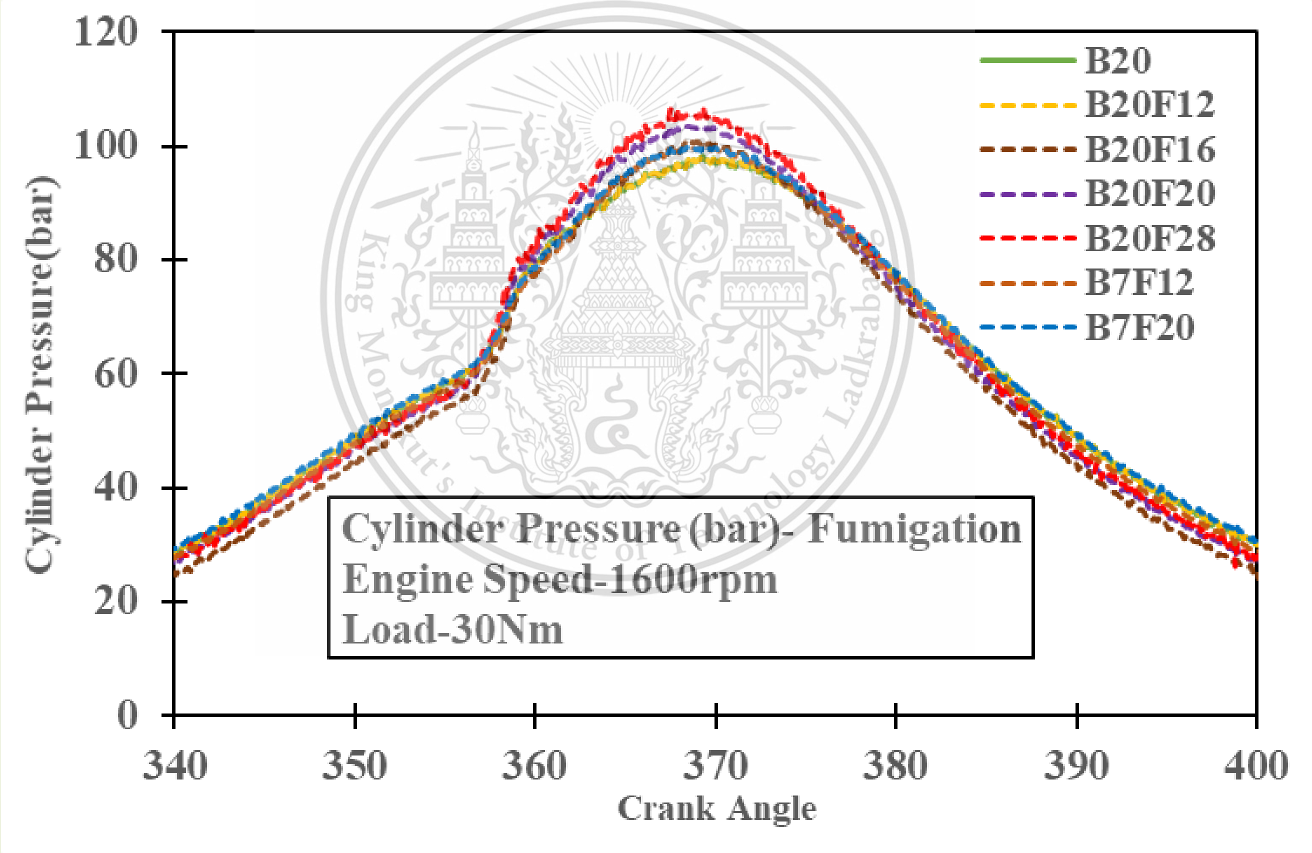
Results- Emissions of Fumigation method

Variation of Soot emission with engine speed



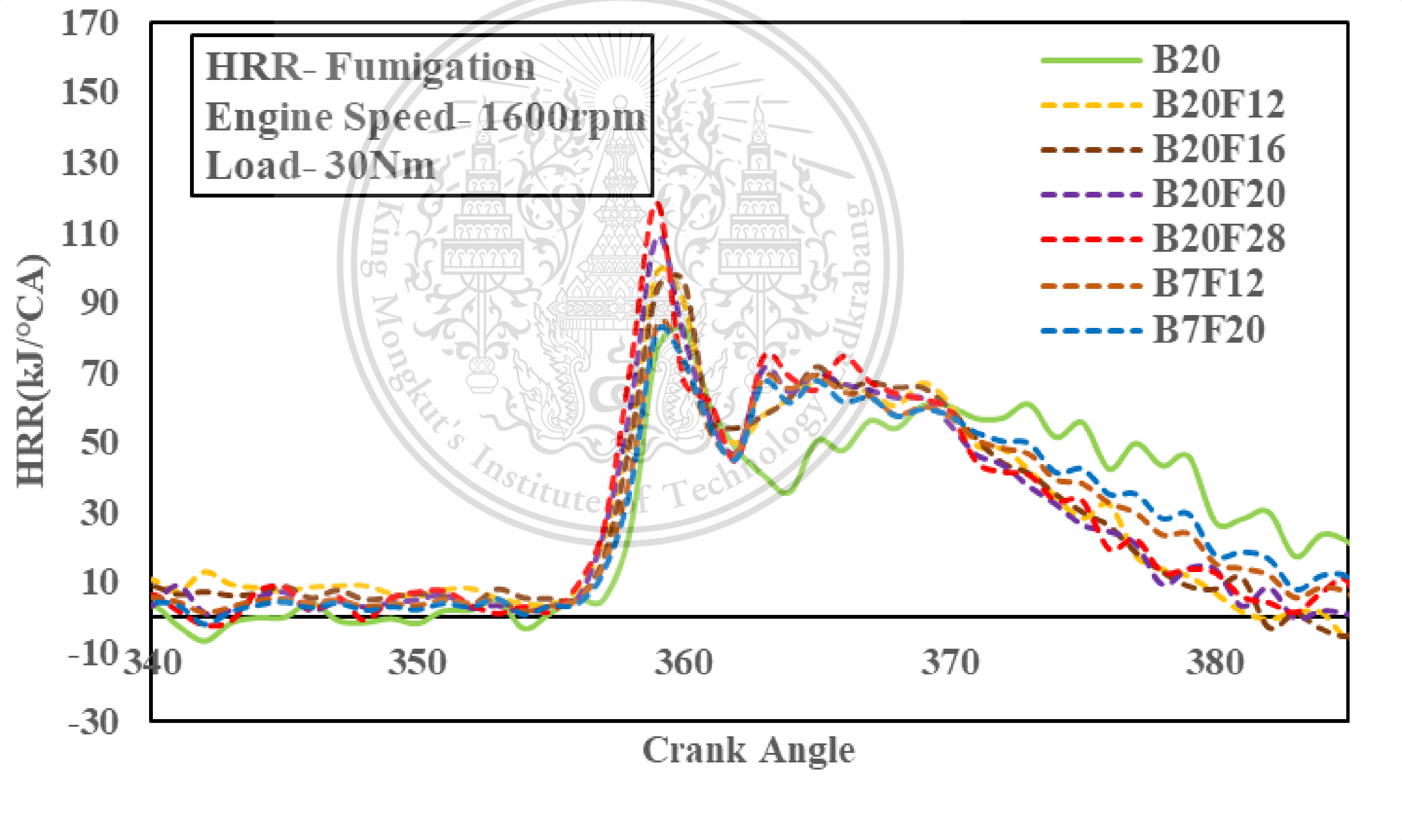
Results- Combustion Characteristics of Fumigation method

Variation of Cylinder Pressure with crank angle



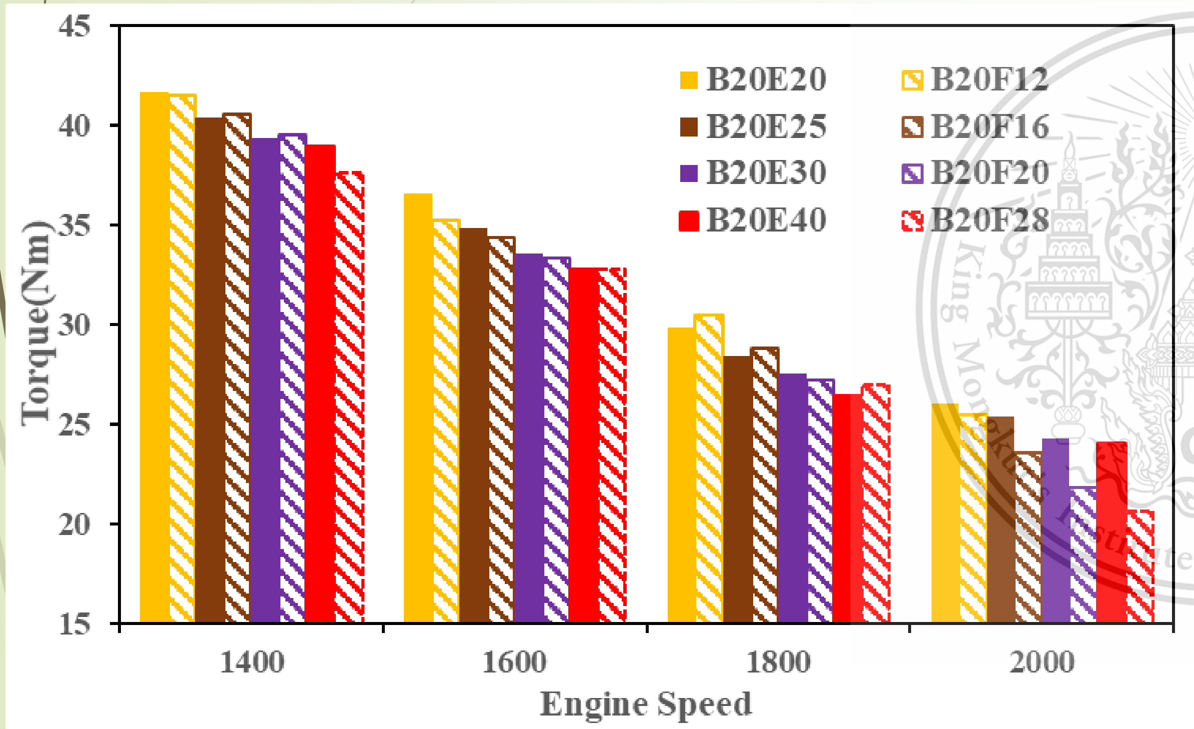
Results- Combustion Characteristics of Fumigation method

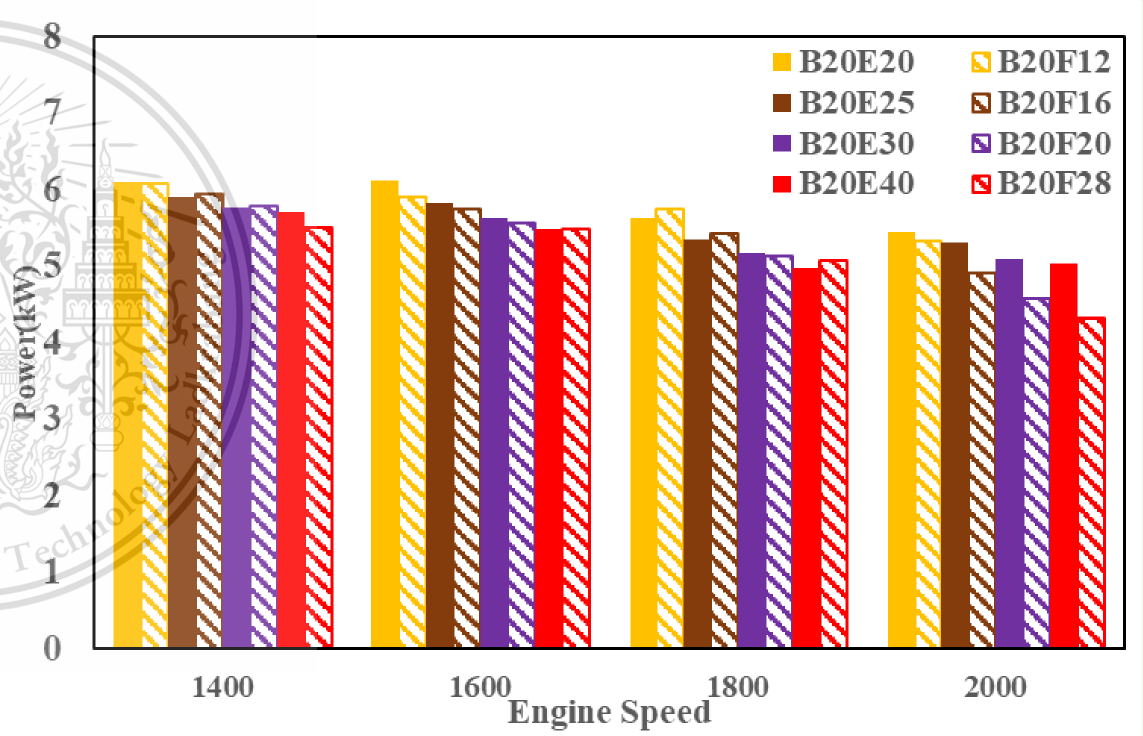
Variation of heat release rate with crank angle



Results- Comparison of performances

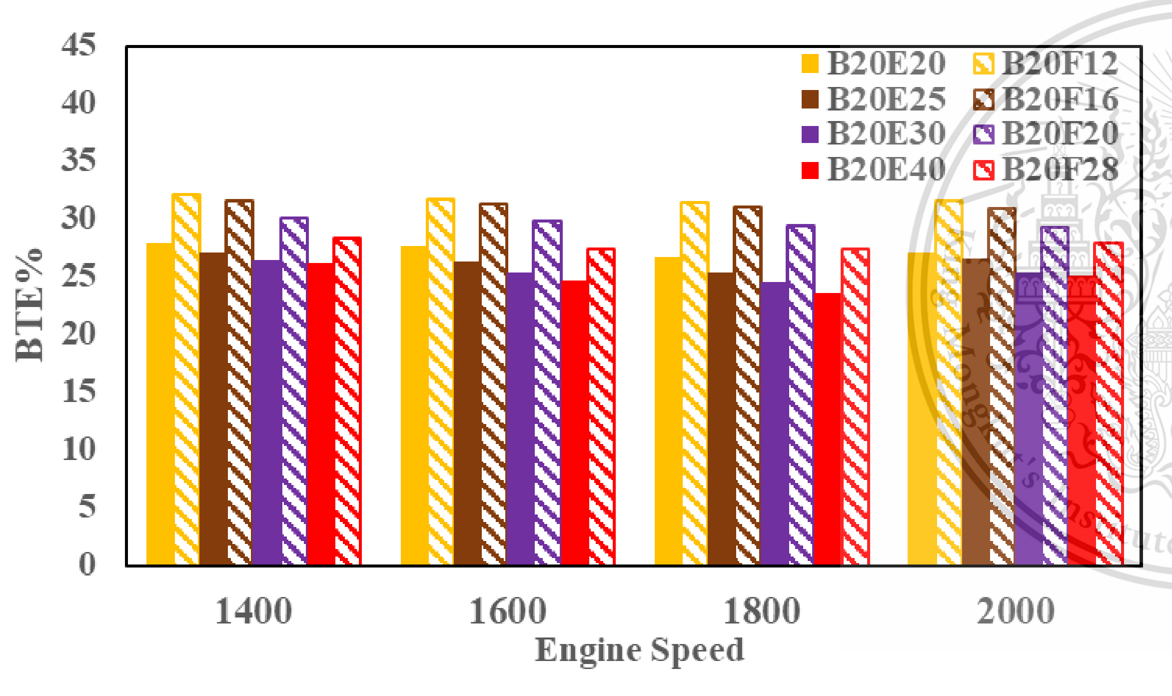
Variation of torque and power with engine speed

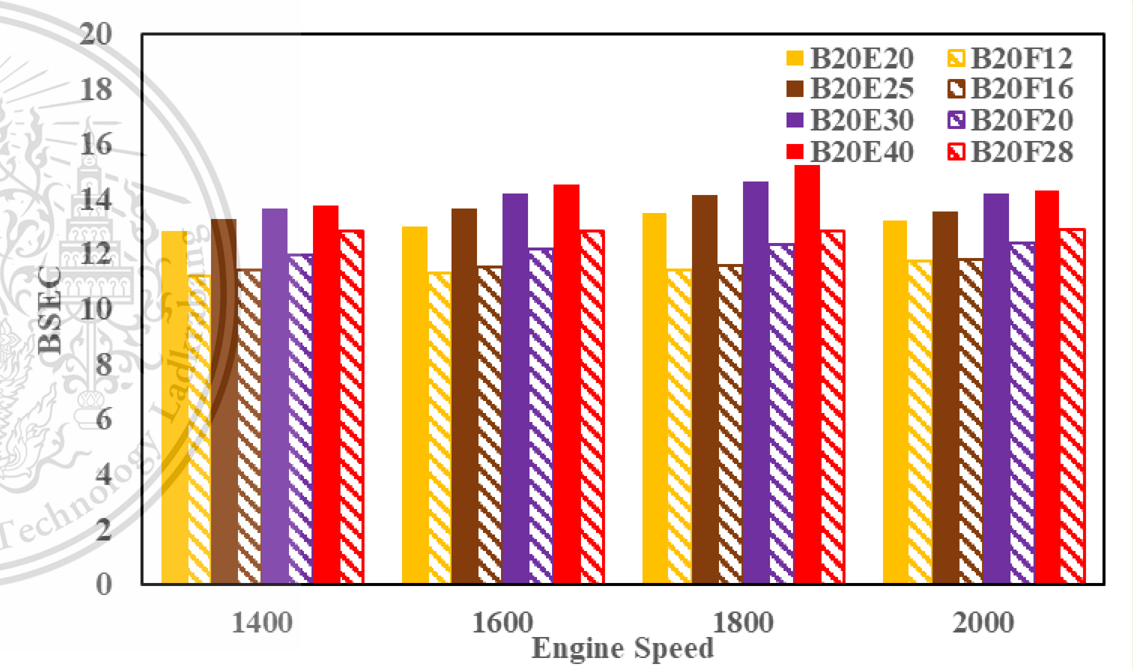




Results- Comparison of performances

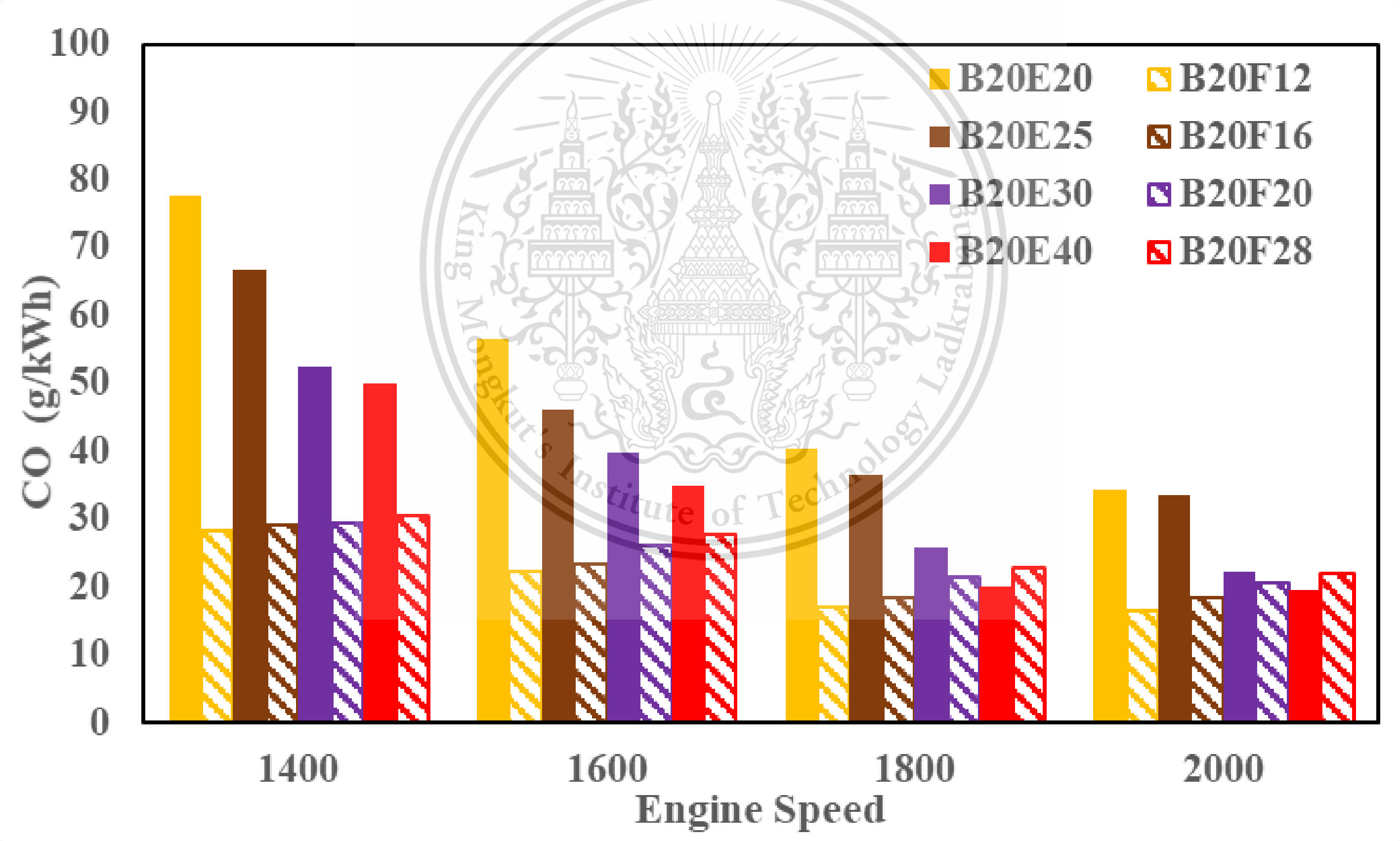
Variation of Brake Thermal Efficiency(BTE) and Brake Specific Energy Consumption(BSEC) with engine speed





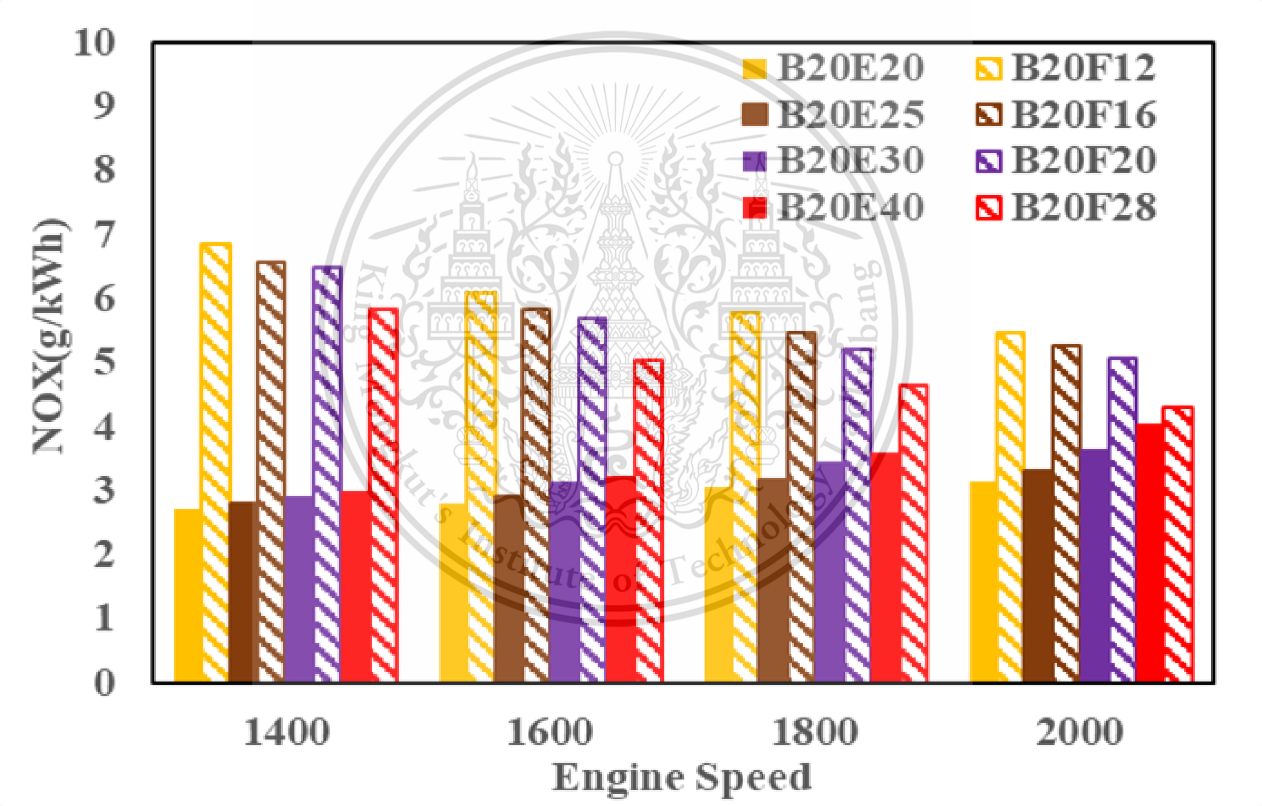
Results- Comparison of emissions

Variation of CO emission with engine speed



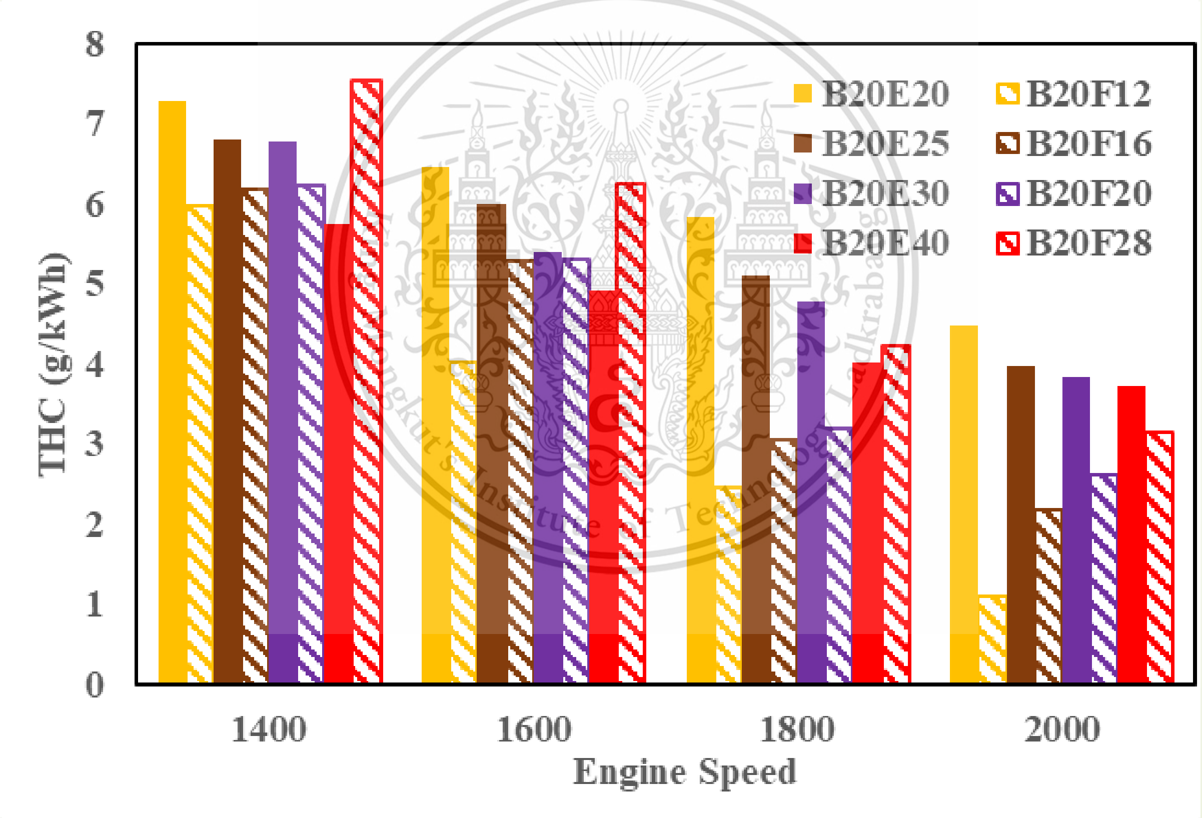
Results- Comparison of emissions

Variation of NO_x emission with engine speed



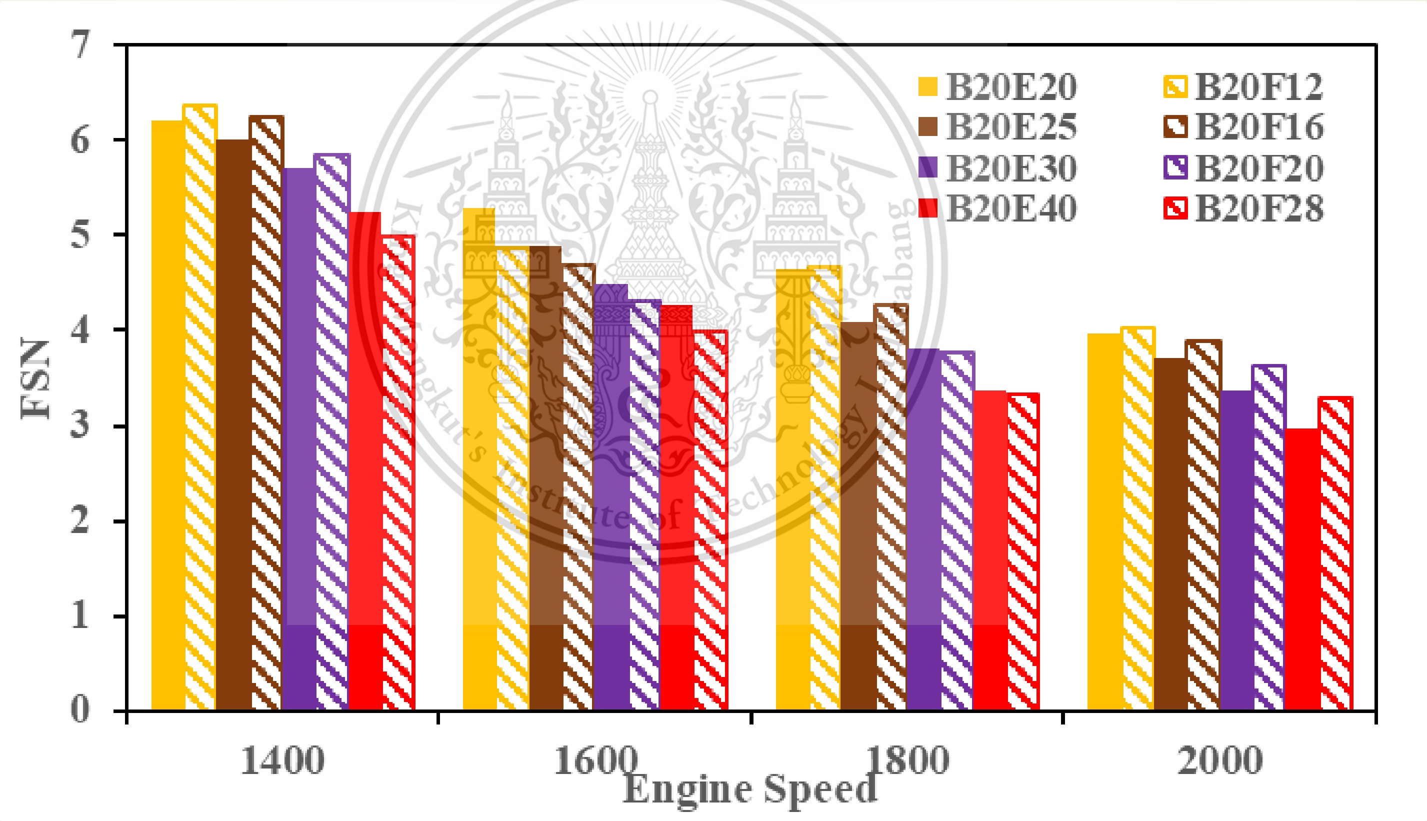
Results- Comparison of emissions

Variation of THC emission with engine speed



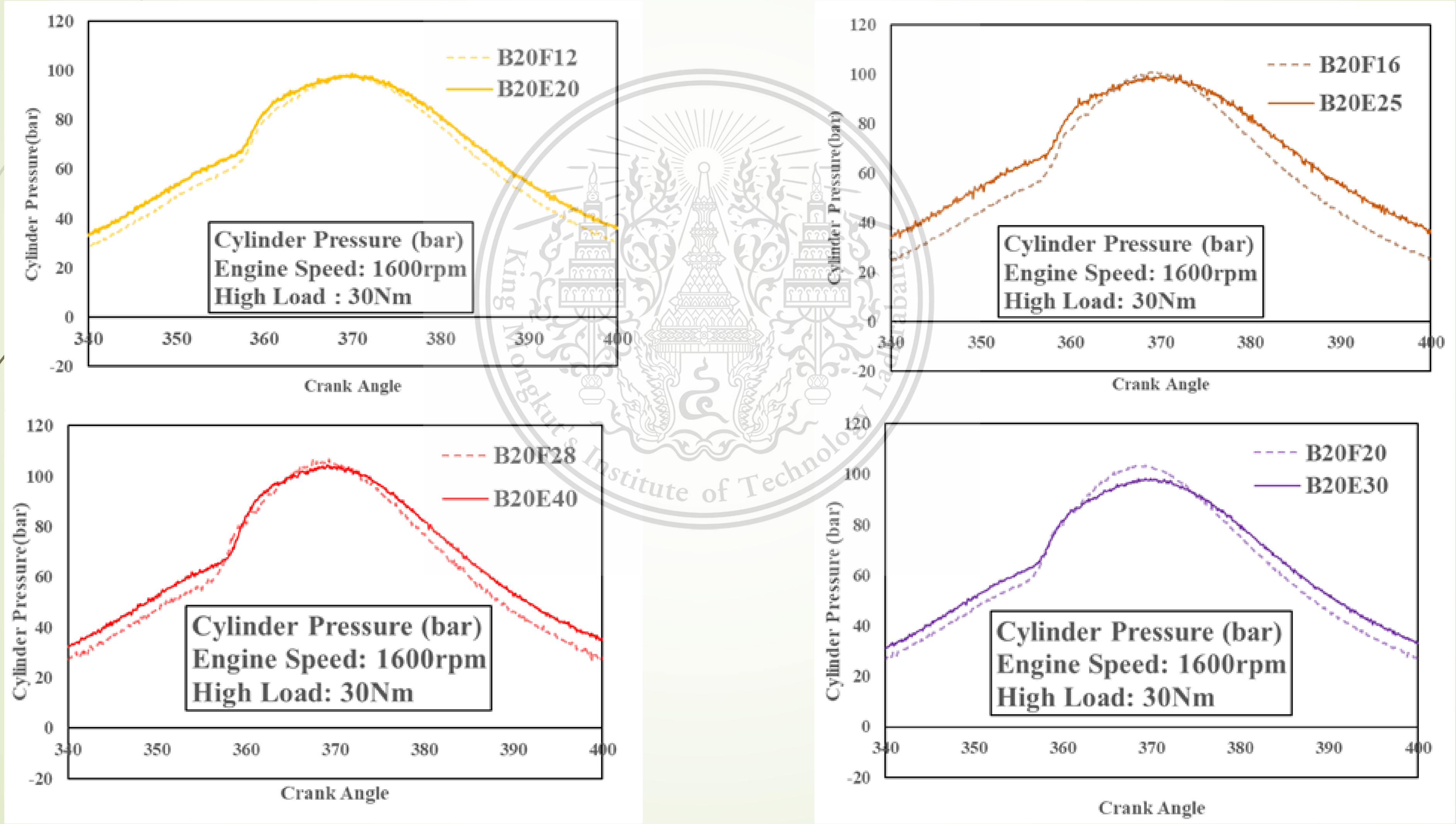
Results- Comparison of emissions

Variation of Soot emission with engine speed



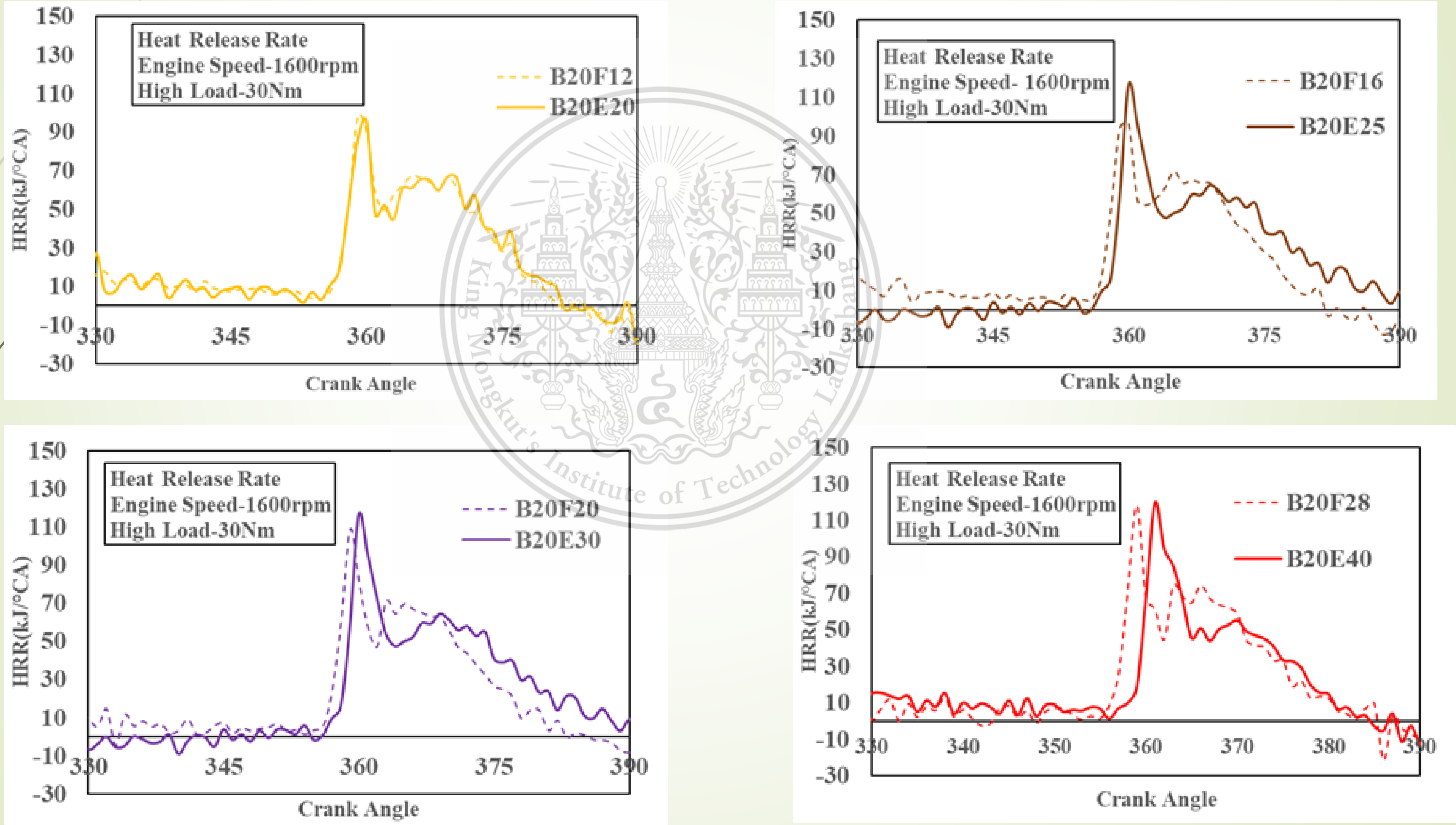
Results- Comparison of combustion characteristics

Variation of Cylinder Pressure with crank angle



Results- Comparison of combustion characteristics

Variation of heat release rate with crank angle



Conclusions and Recommendations

- Torque and power was decreased with ethanol fraction for both diesohol and fumigation method.
- All the time fumigation rates showed highest BTE and lowest BSEC with compared to diesohol. Increasing both fumigation diesohol fraction reduced the gap of BTE and BSEC between diesohol and fumigation results.
- Dieshol fuels showed significantly higher CO emission with compare to fumigation rates.
- Nox emission of diesohol method and fumigation method showed opposite behavior which means NOx formation was increased with engine speed and vice versa
- Both fumigation and diesohol method showed reduction of THC with engine speed.
- Heat Release rate of all the diesohol fuels were higher than fumigation rates and had higher ignition delay as well.
- Long running of the engine using dieshol fuels and try to see the effect of ethanol blended fuels on the engine such as corrosion of parts, contaminations of oil, wear out of some parts is the future plan of this experiment to make sure how higher portion of ethanol blend fuels will affect to the real world application.

THANK YOU

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[1] Surith Dulanjala De Silva, Precha Karin. and Manida Tongroon, and Hidenori Kosaka "Impact of ethanol addition to bio diesel (B20) on performances combustion characteristics and emissions of a diesel engine", The 28 th International Symposium on Transport Phenomena 22-24 September 2017, Peradeniya, Sri Lanka