DEVELOPMENT OF A FUEL INJECTION STRATEGY FOR A DIESEL ENGINE FUMIGATED WITH ETHANOL

TRIPOOM PAINRUNGROT

A THESIS REPORT SUBMITTED IN PARTIAL FULFILLMENT
OF THE REQUIREMENTS FOR THE DEGREE OF
MASTER OF ENGINEERING IN AUTOMOTIVE ENGINEERING
INTERNATIONAL COLLEGE
KING MONGKUT'S INSTITUTE OF TECHNOLOGY LADKRABANG
ACADEMIC YEAR 2018
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THESIS TITLE	Development of a Fuel Injection Strategy for a Diesel Engine Fumigated with Ethanol
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ABSTRACT

Diesel engine are widely use all over the world. Because they can generate high torque even in low speed operation, and they give more efficiency from the fuel input compare to gasoline engine. But diesel engine still releases lots of emission which are harmful to environment. So, many researchers are going to improve the efficiency and reduce the emission from the combustion process. Ethanol is a good choice for alternative fuel which is prefer to dual fuel diesel engine. Because ethanol made from plant such as sugar cane, cassava, sorghum, and so on, which can reduce CO_2 in photosynthesis process. In this study, ethanol will be injected in to the intake manifold to cool down the intake temperature and reduce the amount of diesel fuel consumption.

However, there are many parameters that effect to the combustion. Injection timing, injection pressure, and injection amount are some of essential factor that gives different result. The objective of this research is to study the effect of injection timing of dual fuel (diesel and ethanol) on the engine performance and exhaust emissions of a single cylinder 4-stroke direct injection compression ignition engine including ethanol fumigation. By varying the diesel injection timing by 7 different degrees, and also varying the injection pressure of 300, 500, 700 Bar. Then using ethanol fuel as a secondary fuel to replace the energy input from diesel fuel by 10% increment, until the engine cannot run properly. The engine load will be fix at full load. The injection timing of ethanol is 160^{o}ATDC and the injection pressure is 3 Bar.

The expectation of this experiment is to reduce the usage of fossil fuel. Increase the usage of alternative fuel which is cleaner. And another purpose is to reduce

the harmful gases from the emission of diesel engine. Although NO_x, Soot, CO, and THC are reducing by many researchers, there are still remain some significant amount of emission above. Because diesel engine has been using in a huge number for transportation, agricultural, marine engine, and industrial.

ACKNOWLEDGEMENT

Without the contribution of many people, this thesis would not have been existed. It owes the existence to the supports and inspirations from a lot of people. To my thesis advisor Dr. Manida Tongroon (NSTDA) and co-advisor Dr. Chinda Charoenphonphanich of International College at King Mongkut's Institute of Technology Ladkrabang, I would like to express my deepest gratitude for the encouragement and supervision through all obstacles and challenges since the beginning until the end of my study. And National Science and Technology Development Agency (NSTDA) for providing full scholarship, Renewable Energy Laboratory, National Metal and Material Technology Center (MTEC)

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I also want to express my gratitude to all lecturers for your support and guidance to me for the whole two years. Also, I would like to thank all my friends who always be there to support and motivate me as always. Moreover, I also would love to express my gratitude to all respondents who contribute their information and time on this study. And I do believe the study could not been done without their input.

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Tripoom Painrungrot

TABLE OF CONTENTS

Chapter **Page**

LIST OF TABLES

Table **Page**

LIST OF FIGURES

LIST OF DEFINITIONS

ATDC	After Top Dead Center
AHRR	Apparent Heat Release Rate
BSCO	Brake Specific Carbon monoxide
$BSCO_2$	Brake Specific Carbon dioxide
BSEC	Brake Specific Energy Consumption
BSFC	Brake Specific Fuel Consumption
$BSNO_X$	Brake Specific Oxide of Nitrogen
BSPM	Brake Specific Particular Mass
BTDC	Before Top Dead Center
BTE	Brake Thermal Efficiency
CA	Crank Angle
CC.	Cubic Centimeter
CI	Compression Ignition
CR	Compression Ratio
EGR	Exhaust Gas Recirculation
FIP	Fuel Injection Pressure
HCCI	Homogeneous Charge Compression Ignition
HP	Horse Power
HRR	Heat Release Rate

IMEP	Indicated Mean Effective Pressure
KW	Kilo Watt
L.	Liter
LHV	Low Heating Value
LTC	Low Temperature Combustion
MPA	Mega Pascal
NM	Newton Meter
PCCI	Premixed Charge Compression Ignition
P_{MAX}	Maximum Pressure
PM	Particulate Matter
RCCI	Reactivity Controlled Compression Ignition
ROHR	Rate of Heat Release
RPM	Round Per Minuit
SOI	Start of Injection
TDC	Top Dead Center
VVA	Variable Valve Actuation
VVT	Variable Valve Timing

CHAPTER 1

INTRODUCTION

1.1. Research Background

From the concerning of limitation of fossil fuel amount remain in the world from the widely use of diesel engine in many different purposes around the world. Because some advantage of this engine type. Diesel engine can generate high torque even in the low engine speed operation. And because the very high pressure in combustion, the engine is very durable. Diesel engine also require less accessory, because spark plug is not required to ignite for diesel fuel. Relative to gasoline engines, diesel engines have higher compression ratios, more rapid combustion, and lower throttling losses and operate leaner. As a result, diesel engines have a greater thermodynamic efficiency and hence a higher fuel economy (lower fuel consumption) than gasoline engines. Diesel vehicles have higher fuel economy and lower CO_2 emissions than their gasoline counterparts. (SULLIVAN, et al., 2004) By the previous reason. The temperature and pressure in ignition is also lower than gasoline engine. All these reasons show the diesel engine may minimize their maintenance cost. But many papers show the reserves amount of oil which it will last for a future 40 year, if we still consume at the same rate of 2006. Then coal and gas will be depleted in 200, and 70 years, respectively. (Shafiee & Topal, When will fossil fuel reserves be diminished?, 2009)

According to the International Energy Agency (IEA) (2006b), from 2005 to 2030 the demand for oil will grow by 1.3% per annum, broadly in line with global GDP (Gross Domestic Product) averaging 1.7% from 2005 to 2015 and 1.1% from 2015 to 2030. Oil demand should reach 99 million barrels per day by 2015 and 116 million barrels per day by 2030, up from 84 million barrels per day in 2005 and furthermore, the Energy Information Administration (EIA) (2007) has projected that energy consumption will increase at an average rate of 1.1% per annum, from 500 quadrillion Btu in 2006 to 701.6 quadrillion Btu in 2030. The World Energy Outlook (WEO) 2006 claims that energy generated from fossil fuels will remain the major source and is expected to meet about 83% of energy demand in 2030 (International Energy Agency, 2006b). In comparison with other types of fossil fuels, the demand for oil will fall over the next 25 years, even though in absolute terms it will continue to be the main fuel supplying the universal demand for energy until 2030. The rate of consumption of energy will also continue to increase through to 2030. The greatest demand for fossil fuels will be for coal, which will remain the second largest primary fuel source until 2030. (Shafiee & Topal, An econometrics view of worldwide fossil fuel consumption and the role of US, 2008)

There are many papers mention about the prediction of the oil price. Even though the trend of oil price is very complicate and fluctuate in the same time. Most of the prediction shows that oil price will be increase in the future. (Yu, Wang, & Lai, 2008) (Shafiee & Topal, A long-term view of worldwide fossil fuel prices, 2010) (Jammazi & Aloui, 2012) Even though, there are many developers trying to improve the engine efficiency and reduce the emission by adding some accessories like turbocharger, high precise engine control unit, variable valve timing, are not only to increase the efficiency of the engine and by this process, but it also able to reduce the fuel consumption. And by using EGR, many types of catalytic, and improve some properties of the fuel to reduce the emission. Although, there are many new advance techniques were developed to the engine. New type of fuel which can replace the fossil fuel were found and develop in various study. Like biodiesel, this type of fuel was recycled from the cooking oil or oily plant, which is able to produce in Thailand by Transesterification process. For gasoline, ethanol is able to use as an alternative fuel. Most of ethanol were produce by plant such as sugarcane or cassava. These are 2 types of vegetation that are very easy to grow in Thailand. (Energy, 2013)

The most appropriate plant for making ethanol in Thailand is sugarcane (molasses) and cassava. Ethanol has very high-octane number, when blend with gasoline we call Gasohol. When ethanol blended with diesel we call Diesohol. Ethanol also able to fuel as 100% pure without mixing with other fuel. Ethanol are a clean energy, which is release a little emission. And very good choice for new fuel, because it was made from plant that mean we can reduce carbon dioxide by growing of plant. It is also good for economic state. (Energy Policy and Planning Office, 2013)

Ethanol is a kind of alternative fuel, which is appropriate to produce in Thailand. Because of the ability in agricultural. And some property is suitable for using as fuel for internal combustion engine. Ethanol is better to mix with gasoline. But Thailand use diesel engine more than gasoline. So, to increase the amount of ethanol usage with diesel engine is concerned in this project. There is some property that ethanol cannot mix with diesel properly, so there are many technics try to increase the amount of ethanol for using in the combustion. Emulsion is using some additive substance to help this problem. (Water-pacific.com, 2010)

And another main concerning problem is about the emission of diesel engine. Most of the combustion reaction in the engine are various non-ideal process, such as incomplete combustion of the fuel, reactions between mixture components under high temperature and pressure, combustion of engine lubricating oil as well as combustion of non-hydrocarbon components of diesel fuel, such as sulfur compounds and fuel additives. In reality, combustion in diesel engines is never complete. This shortcoming occurs when the chain reactions of combustion are arrested at some intermediate step. In the region surrounding the flame, reaction interruptions occur due to an insufficient provision of oxygen (rich mixture) or due to excessive heat loss (quenching). The first product of incomplete combustion to appear is usually carbon monoxide (CO) as its reaction to CO_2 is slower than rates of the other reactions in the chain. A worsening of conditions may result in the appearance of various hydrocarbons from the broken reaction chains. In the hottest areas of the combustion chamber where there is an excessive fuel presence (rich zone), solid carbon particles are produced which then enter the exhaust in the form of soot. Besides the CO and the unburned hydrocarbons of incomplete combustion, oxides of nitrogen (NO and NO_2 - generally referred to as NO_x) are a considerable product. They are formed by the reaction between atmospheric N_2 and O_2 at high temperatures and will remain in the products as the exhaust gases cool coming out of the exhaust manifold. (Torres, Bello, Sarmiento, Rostkowski, & Brady, 2003) As shown in the Figure 1.1

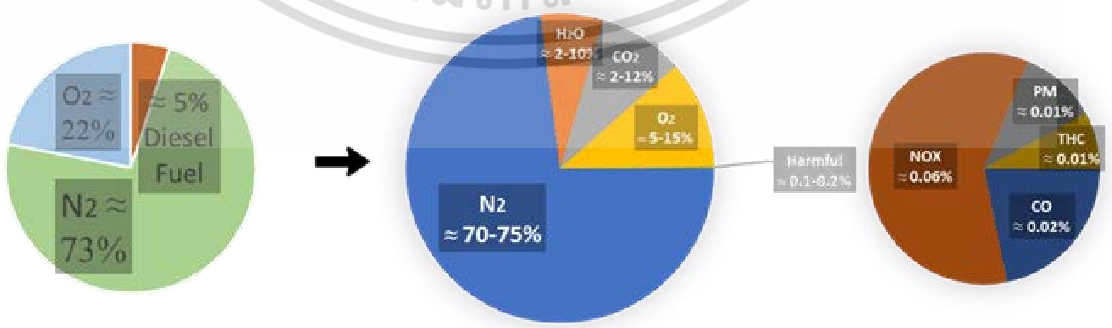


Figure 1.1 Typical diesel engine combustion mass balance

However, actual combustion under no circumstances will be theoretical due to various design and operating conditions. Fuel is not pure hydrocarbon, contain traces of nitrogen, Sulphur and other chemicals and air is comprising hundreds of constituents other than O_2 and N_2, may be amounting to 1% of total constituents. When combustion occurs between fuel and air, in real, a number of compounds appear in the engine exhaust, both as gaseous and solid. High combustion pressure and temperatures during combustion with heterogeneous nature of air fuel mixture further complicates combustion process. Regions of rich and poor mixture ratios along with local stoichiometric composition pockets exist within the combustion chamber. Fuel rich composition produce CO, soot, dark smoke and unburnt HC, on the other hand fuel oxygen rich regions generate NO_x at high temperature conditions. Of all emission constituents, major concerns are NO_x, CO, HC, Smoke (Particles and soot) and in the recent past CO_2 as its accumulation add to greenhouse gases in the atmosphere as shown in Figure 1.2 (Hebbar, 2014)

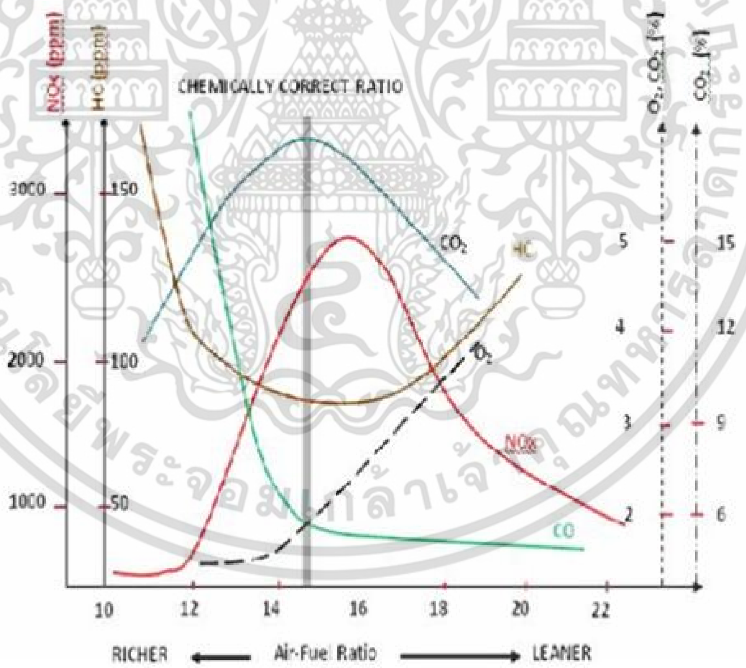


Figure 1.2 Emission from diesel combustion

Also, the Emission control technologies have reduced the emissions of CO, VOCs, PM, HFC-134a, CH_4, and N_2O from modern vehicles to very low levels. Ford Motor Company and elsewhere focused on understanding the nature and quantity of vehicle emissions and developing control technologies to reduce these emissions. Light duty vehicles are generally used as passenger vehicles (e.g., passenger cars, pickup trucks, vans, sports utility vehicles).

Given their commercial importance, and the importance of their emissions into the atmosphere. (WALLINGTON, SULLIVAN, & HURLEY, 2007) And for heavy duty diesel engines. It need to reduce their inventory of old-generation products in preparation for the demand for next-generation products that satisfy new emission regulations. Some researchers have found that "the excessive emissions of greenhouse gas cause the frequent occurrence of extreme weather on Earth, which threatens the safety of human society and the natural system" These toxic emission materials are regulated by law in most countries. In Europe, EURO 5, an exhaust gas regulation for HDDEs, was instituted in 2009 and imposed in 2011. Subsequently, EURO 6 was instated in 2014 and imposed in 2015. The stricter emission regulations induce technological competition and innovation. (Kim, Han , & Sohn, 2017)

NO_x represents a family of seven compounds. Actually, EPA regulates only nitrogen dioxide (NO_2) as a surrogate for this family of compounds because it is the most prevalent form of NO_x in the atmosphere that is generated by anthropogenic (human) activities. NO_2 is not only an important air pollutant by itself, but also reacts in the atmosphere to form ozone (O_3) and acid rain. It is important to note that the ozone that we want to minimize is tropospheric ozone; that is, ozone in the ambient air that we breathe. (Oxides, 1999) And the emission of NO_x and Soot are also depended on both combustion temperature and fuel mixture concentration. (Dempsey, Curran, & Wagner, 2016) As shown in Figure 1.3

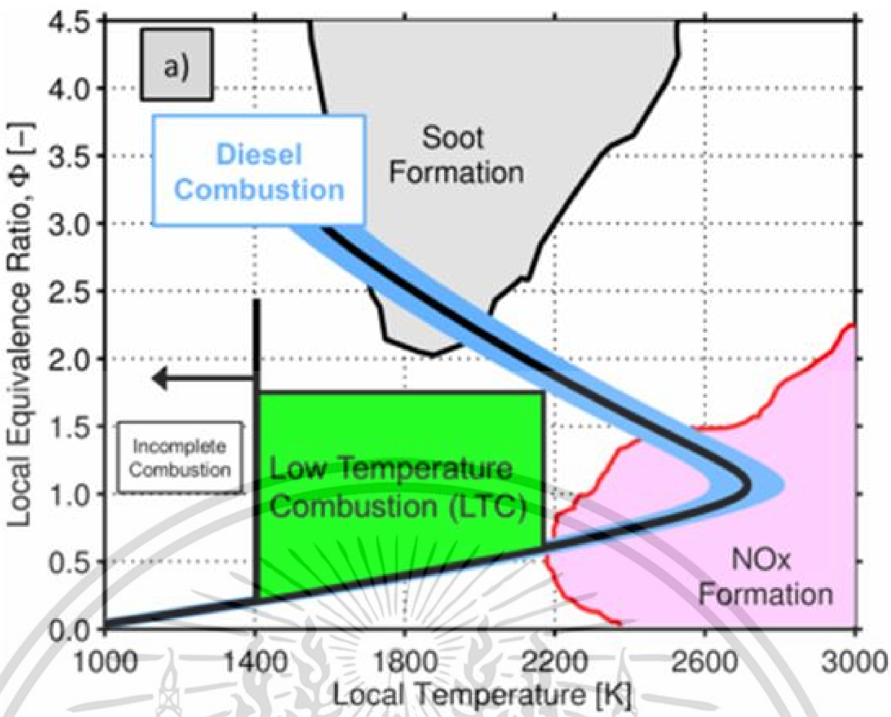


Figure 1.3 NO_x and soot formation in local equivalence ratio vs local temperature

So, this project is trying to reduce the emission, the use of fossil fuel (diesel) and improve the engine performance. To reduce the use of diesel, there are many techniques to achieve this problem. One technique is to use alternative fuel. They filled the engine with ethanol for gasoline or blended it together. But for diesel engine, ethanol cannot mix in the large amount, by some properties of them. So, they try another technique called "ethanol fumigation". This technique can increase the amount of ethanol usage.

1.2. Research Objective

- Study the engine performance from ethanol fumigation when vary the injection strategy of both ethanol and diesel
- Study the emission of the engine from ethanol fumigation when vary the injection strategy of both ethanol and diesel
- To find the optimum strategy for ethanol and diesel injection
- To be a base document for alternative fuel usage in Thailand

1.3. Research Scope

- Vary 7 injection timing
- Vary 3 injection pressure
- Vary both injection timing and injection pressure.
- Vary ethanol fumigation amount by 10% increment of energy

1.4. Experiment Process

1. Modified and apply the fuel injection system from mechanical pump to common-rail system.
2. Install ethanol injection to the intake manifold system and injector controller.
3. Set up the engine to the fuel consumption measuring system, emission analyzer, smoke meter, and pressure and temperature sensors in various position.
4. Run and test the engine with neat diesel 3 times, to get base data.
5. Vary the injection timing of neat diesel for 3-degree increment until the engine do not run properly.
6. Vary the injection pressure of neat diesel for 300, 500, and 700bar.
7. Run and test the engine with diesel and ethanol fumigation by fix the ethanol injection timing first, while varying the diesel injection timing.
8. Run and test the engine with diesel and ethanol fumigation by fix the ethanol injection timing first, while varying the diesel injection pressure.
9. Conclude the result for performance and emission after changing the injection timing and injection pressure of diesel, and ethanol fumigation amount variation to find the optimum point from the result.
10. Observe the combustion characteristic of the engine. Analyze and conclude to result from the test, then report.

CHAPTER 2

LITERATURE REVIEW

2.1. The study of injection timing

Cenk Sayin (Sayin & Canakci, 2009) study, influence of injection timing on the engine performance and exhaust emissions of a naturally aspirated, single cylinder 770 cc diesel compression ratios is 17. This engine can generate 7.4 kW at 1900 rpm and 38.9 Nm at 1650 rpm. By varying ethanol blended in diesel fuel from 0% to 15% with an increment of 5%. The engine load was selected as 15 and 30 Nm. And the engine speed was selected as 1000, 1200, 1400, 1600, and 1800 rpm. The tests were tested at five different injection timings (21, 24, 27, 30 and 33^{o}CA BTDC) by changing the thickness of advance shim. The test was held 3 times in each testing condition, to compare and average. To prepare ethanol blended fuel mixture, two fuels (euro-diesel and ethanol) were used. The fuel properties are shown in Table 2.1

Table 2.1 Properties of fuel tested

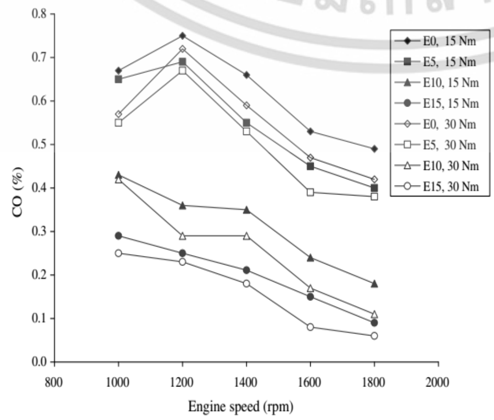
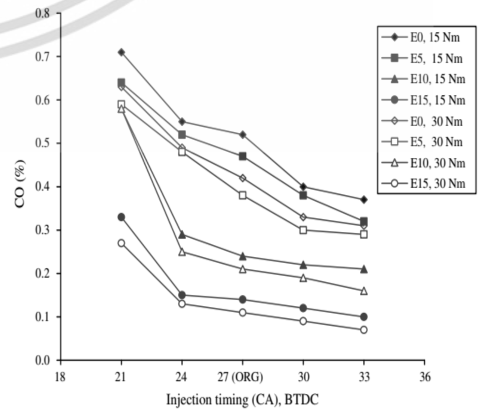
Properties	**Ethanol**	**Euro diesel**
Formula	C_2H_5OH	$C_{12}H_{26}$–$C_{14}H_{30}$
Molecular weight	46.07	170–198
Boiling temperature (˚C)	78.3	190–280
Density (kg/m^3, at 20 ˚C)	811.5s	820–845
Flash point (˚C)	13	52
Autoignition temperature (˚C)	425	300–340
Lower heating value (MJ/kg)	27	43
Cetane number	>15	50>
Vapor pressure (kPa, at 38 ˚C)	17	0.34
Stoichiometric air–fuel ratio	8.96	14.7
Latent heat of vaporization (kJ/kg)	921.1	620

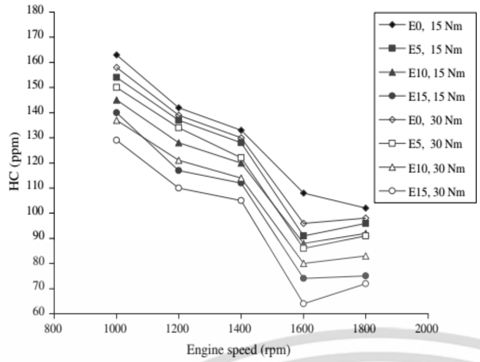
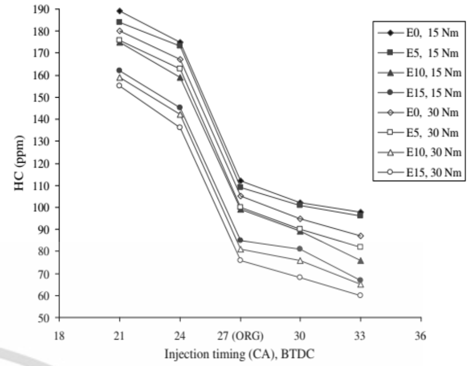
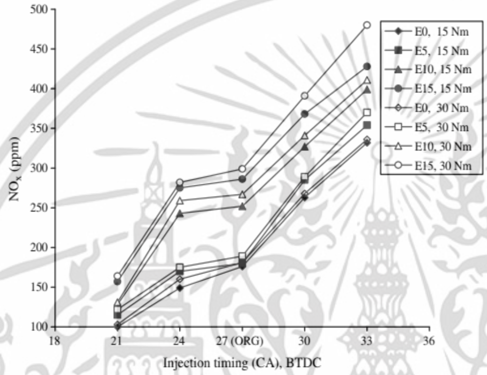
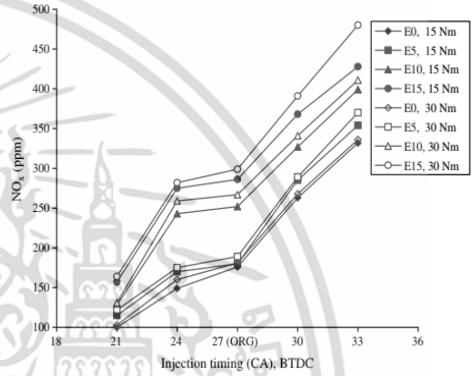
The euro-diesel was blended with ethanol to have four different fuel blends from 0% to 15% with an increment of 5%. The fuel blends were prepared just before starting the experiment to ensure that the fuel mixture is homogenous. A mixer was also mounted inside the fuel tank in order to prevent phase-separation.

The results indicated that NO_x emissions slightly increased; CO and unburned HC emissions decreased dramatically by ethanol addition; and CO_2 emissions increased because of the improved combustion ethanol blended diesel fuels, CO and unburned HC emissions reduced 10–70% and 10–45%, while CO_2 and NO_x emissions increased 10–50% and 5–15%. Increasing the amount of ethanol in the fuel mixture produced higher peak temperature in the cylinder. This effect increased NO_x emissions. Increasing the ratio of ethanol in the mixture leads to increase the BSFC and decrease BTE by about 18% and 17% LHV of the ethanol, which is distinctly lower than that of the diesel fuel. Advancing the injection timing, CO and unburned HC emissions decreased while NO_x and CO_2 emissions increased. Retarding the injection timing 6^o CA BTDC (21^{o}CA BTDC) at 15 Nm load and 1800 rpm presented the minimum results of NO_x and CO_2 emissions. The ORG injection timing gave the best results of BSFC and BTE by about 34% and 32% average value, respectively, compared to the other injection timings. When advanced injection timing, the ignition delay will be longer and speed of the flame will be shorter. These leads to reduction in the maximum pressure and engine output power. Thus, fuel consumption per output power will augment. On the other hand, retarded injection timing means later combustion, and therefore, cylinder pressure increased only when the cylinder volume was getting higher rapidly and the results was a reduced effective pressure to do work.

To make clearer about the result. The writer plots all experimental result on the diagram. All these diagram shows the result of emission and performance from various condition explain above. Table 2.2

Table 2.2 Experimental result

The experimental result on emission and performance		
Title	**Original injection timing**	**Different injection timing (1800 rpm)**
CO		

<table>
<tr><th colspan="3">The experimental result on emission and performance</th></tr>
<tr><th>Title</th><th>Original injection timing</th><th>Different injection timing (1800 rpm)</th></tr>
<tr><td>HC</td><td>
</td><td>
</td></tr>
<tr><td>NO_x</td><td>
</td><td>
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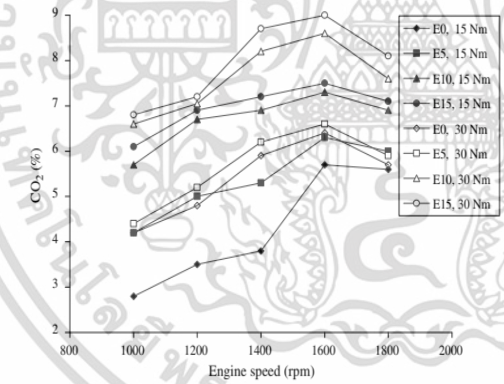
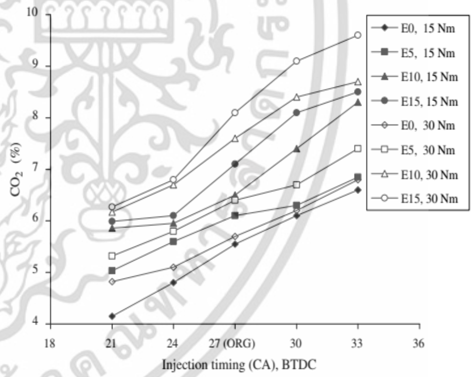
<tr><td>CO_2</td><td>
</td><td>
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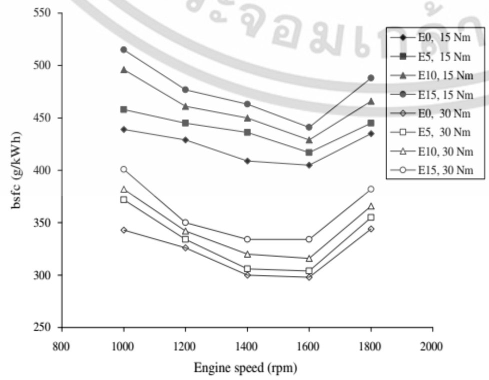
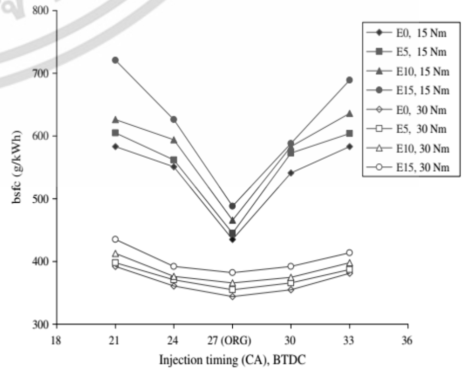
<tr><td>BSFC</td><td>
</td><td>
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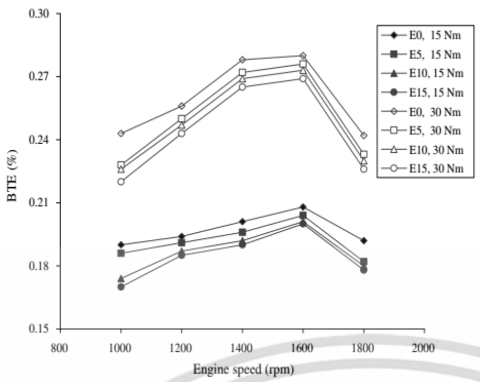
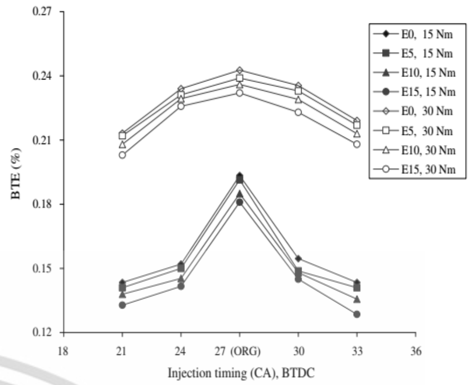
<table>
<tr><th colspan="3">The experimental result on emission and performance</th></tr>
<tr><td>Title</td><td>Original injection timing</td><td>Different injection timing (1800 rpm)</td></tr>
<tr><td>BTE</td><td>
</td><td>
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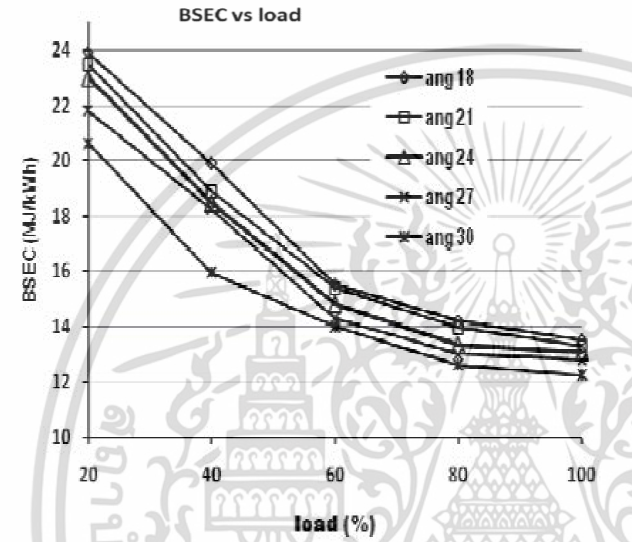
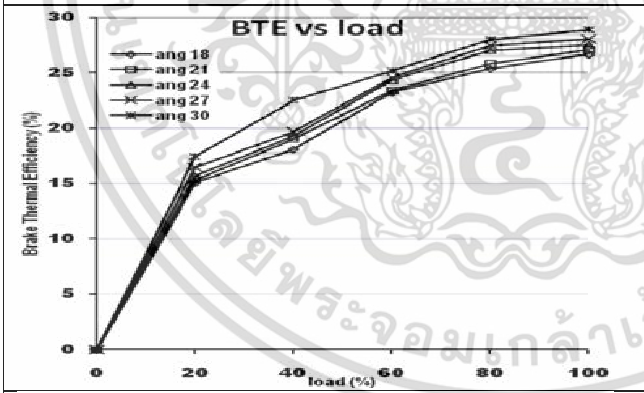
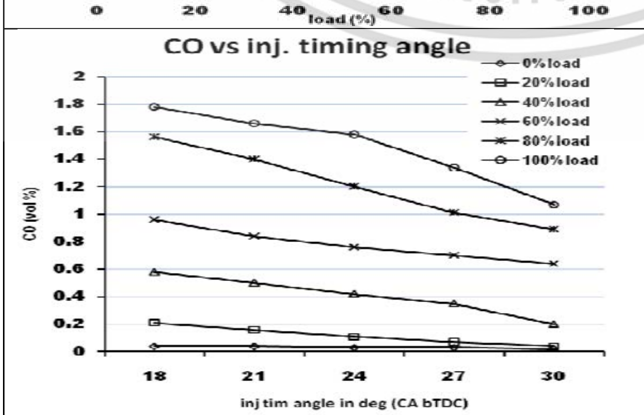
M. Pandian (Pandian, Sivapirakasam, & Udayakumar, 2009) study about Influence of Injection Timing on Performance and Emission Characteristics of Naturally Aspirated Twin Cylinder Compression Ignition Direct Injection Engine Using Bio-diesel Blend B40 as Fuel. To reduce the NO_x emission from the diesel engines employing biodiesel blend as fuel, the injection timing of fuel is altered by either addition or removal of shims in the pump. The effect of changing the injection timing on BSEC, Brake Thermal Efficiency, CO, HC and NO emissions are studied at different injection timings such as 18°, 21°, 24°, 27° and 30° CA BTDC. The properties of blend fuel B40 were shown in Table 2.3 . With the properties of diesel for comparison.

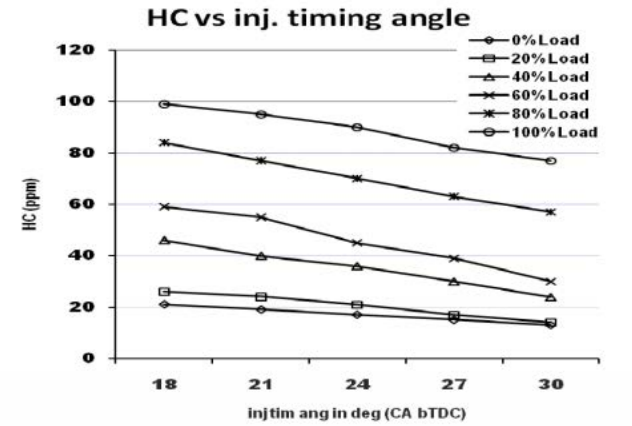
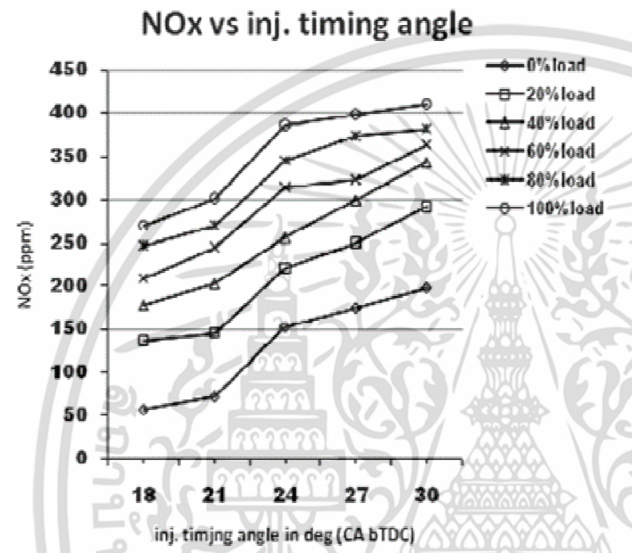
Table 2.3 Properties of blended fuel and diesel

Parameters	Diesel	B40
Kinematic viscosity @ 40°C	2.6	3.85
Cetane No	50	51
Iodine Value	NA	41
Calorific Value (MJ/kg)	42.5	40.1
Specific Gravity @ 15° C	0.835	0.859
Flash point °C	68	81

The engine was always operated at its rated speed of 1500 rpm. And the nozzle opening pressure is 200 bar. Its power is 7.5 kW, 17.5 for compression ratio. Bore and stroke are equal to 80 mm. and 110 mm. In each test, the values of time for 40 cc of fuel consumption, air flow rate, voltmeter, ammeter, the ambient temperature and the exhaust gas temperature were noted from the digital display. Also, the values of pollutants such as CO, HC, CO_2, and NO_x were recorded during the tests. For notifying the exhaust emissions, AVL 5 gas analyzer and AVL Smoke meter were used. All the data were recorded after the engine attained steady state. The engine had an original injection timing of 24°CA BTDC. The tests use five different injection timing (18°, 21°, 24°, 27°, and 30°CA BTDC) values with decreasing or increasing the advance shim. To see the result more precisely and easy to understand. All the tested result will show in Table 2.4

Table 2.4 Result from the test

	BSEC increased by 3. 11% on advancing the injection timing to 30°CA BTDC while reduced by 5% on retarding to 18°CA BTDC from the original injection timing of 24°CA BTDC. Because of advancing the injection of fuel to 30°CA BTDC, complete combustion would have been taken place that results in lesser BSEC. While retarding the injection to 18°CA BTDC, combustion is incomplete that results in higher BSEC.
	It was found that there was 5.07% increase in brake thermal efficiency when injection timing was advanced to 30°CA BTDC but about 3. 08% decrease while retarded to 18°CA BTDC. This was also due to improved combustion when advancing the injection timing and poor combustion while retarding.
	12. 65 % increase in CO emissions was observed on retardation while 32% reduction was noticed on advancement of fuel injection. On advancement About, the fuel blend had sufficient time to undergo the combustion process whereas it had lesser time on retardation.

	HC emission reduced by about 14.44% on advancing the injection timing to 30°CA BTDC but increased about 10% on retarding the fuel injection to18°CA BTDC. Advancing the injection timing caused earlier start of combustion relative to TDC. Hence the cylinder charge had relatively higher temperatures and thus lowered the HC emissions.
	NO_x emission increased as the injection timing was advanced to 27° CA BTDC and 30° CA BTDC from the original injection timing of 24° CA BTDC whereas on retarded to 21° CA and 18° CA BTDC the NO_x emission was reduced. NO_x emission increased at different injection timings due to load increased the fuel consumption with higher oxygen content also increased which lead to the higher NO_x emission.

Jun Cong Ge (Ge, Kim, Yoon, & Choi, 2015) study effects of different pilot injection timings from before top dead center (BTDC) and exhaust gas recirculation (EGR) on combustion, engine performance, and exhaust emission characteristics in a common rail diesel engine fueled with canola oil biodiesel-diesel (BD) blend. Pilot injection timing is 5°, 10°, 15°, 20° BTDC. And main injection timing is 0° TDC.

The experimental engine is four-cylinder 1979 cc. direct injection electronic common rail diesel engine equipped with a turbocharger and compression ratio is 17.7. This engine can generate 82 kW at 4000 rpm and 260 Nm at 2000 rpm. Maximum fuel pressure is 145 MPa with 5 holes' nozzle and 0.17 mm diameter. The EGR rate (%) is defined as the difference between the quantity of fresh air induced without EGR (Q_0) and that of air with EGR (Q_{EGR}) divided by the quantity of fresh air induced without EGR (Q_0).

$$EGR\ (\%) = \frac{Q_0 - Q_{EGR}}{Q_0 \times 100} \qquad (2.1)$$

The pilot injection timing and EGR rate were changed at an engine speed of 2000 rpm and 30 Nm fueled with BD20 (20 vol % canola oil and 80 vol % diesel fuel blend). The fuel pressure in this test is constant at 45 MPa. As the injection timing, advanced, the combustion pressure, brake specific fuel consumption (BSFC), and peak combustion pressure (P_{max}) changed slightly. Carbon monoxide (CO) and particulate matter (PM) emissions clearly decreased at 20° BTDC compared with 5° BTDC, but nitrogen oxide (NO_x) emissions increased slightly. With an increasing EGR rate, the combustion pressure and indicated mean effective pressure (IMEP) decreased slightly at 20° BTDC compared to other injection timings. However, the P_{max} showed a remarkable decrease. The BSFC and PM emissions increased slightly, but the NO_x emission decreased considerably.

Effects of various pilot injection timings without EGR rate. The combustion was started faster as the pilot injection timing retarded. In addition, the ignition delay and the duration of combustion during was shorter when the pilot injection timing was closer to the main injection timing, because the pre-combustion of pilot injection can increase in cylinder temperature to promote combustion. Figure 2.1

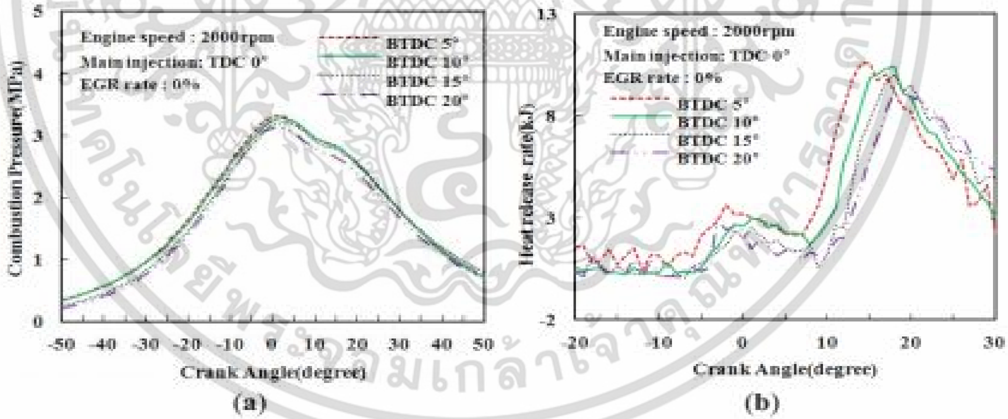


Figure 2.1 Effects of various pilot injection timings without EGR rate on (a) combustion pressure, and (b) HRR.

Comparison of combustion pressure at various pilot injection timings and EGR. The combustion pressure decreased slightly as the EGR rates increased. The oxygen concentration of the intake air is reduced because of the EGR, which causes a significant negative effect on combustion. Figure 2.2

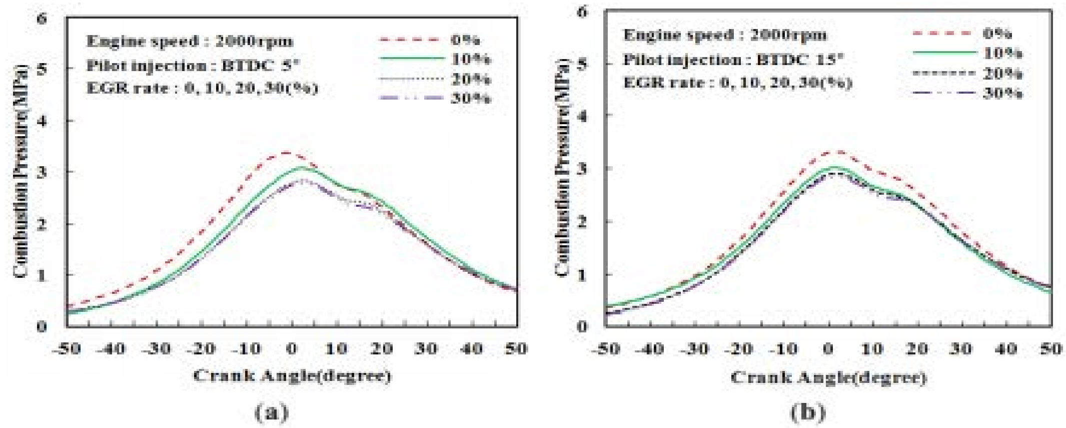


Figure 2.2 Comparison of combustion pressure at various pilot injection timings and EGR rates. (a) BTDC 5° (b) BTDC 15°.

Figure 2.3 shows the HRR for BD20 at various injection timings and EGR rates. The whole combustion process is composed of pilot heat release and main heat release, and the HRR changed slightly with the different pilot injection timings and EGR rates. As shown in Figure 5a, the HRR decreased gradually and the ignition delay was longer as the EGR rates increased at before top dead center (BTDC) 5°, because the oxygen concentration decreased as the EGR rates increased, which is a limiting factor for the pilot combustion. At other pilot injection timings, however, the HRR changed slightly as the EGR rate increased. Increasing the EGR rate slightly does not affect the pilot combustion because the biodiesel contains enough oxygen. These results demonstrate that if the rotational degree of the crankshaft between the pilot injection timing and the main injection timing was long, combustion activation in the main injection deteriorated because the effect of the pilot injection was lost.

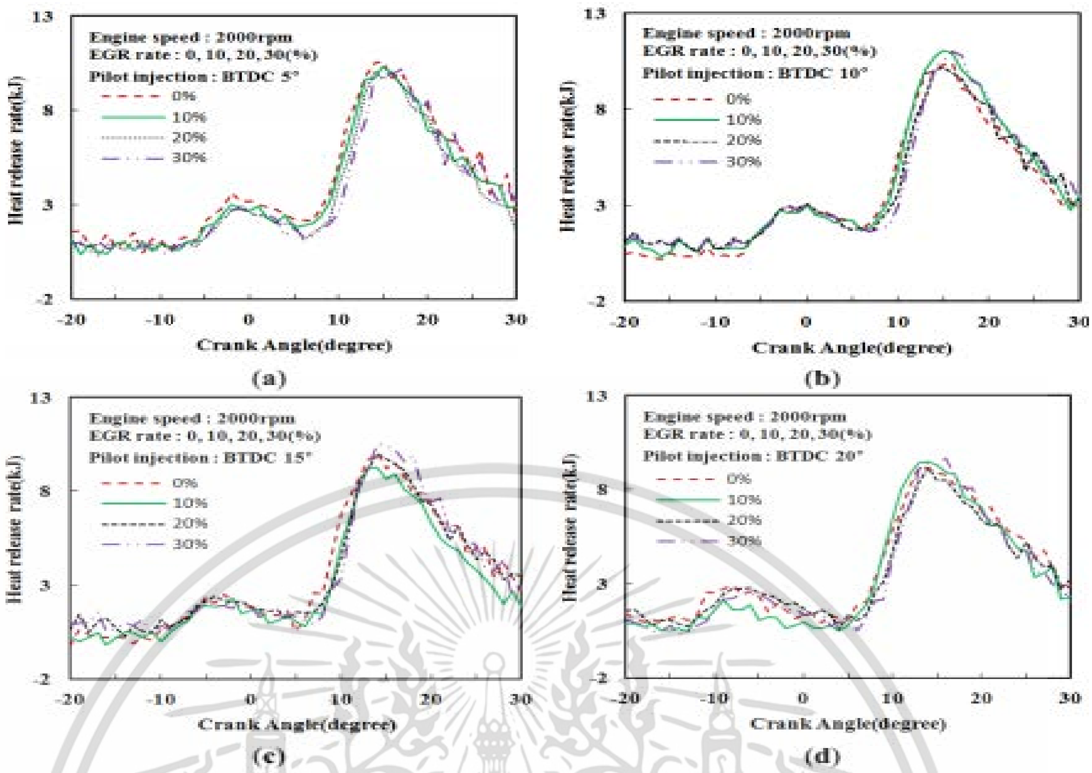
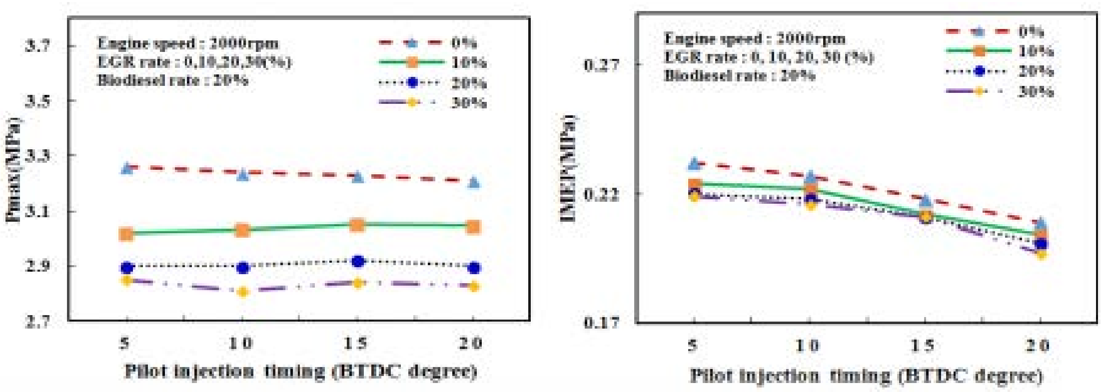


Figure 2.3 Comparison of HRR at various pilot injection timings and EGR rates. (a) BTDC 5°; (b) BTDC 10°; (C) BTDC 15°; (d) BTDC 20°

For the performance of the engine when we increased the EGR rate, the P_{max} showed a remarkable decrease: by 8.7% at 10% EGR, 12.4% at 20% EGR, and 13.6% at 30% EGR, compared to the 0% EGR rate at 15° BTDC. When maintaining the main injection timing at 0° TDC and advancing the pilot injection timing, the decrease ratio was the largest at 15° BTDC as shown in Figure 2.4 , the IMEP increased before BTDC 10° with the advance of pilot injection timing, and as the timing was retarded it decreased. On the other hand, with the increasing of EGR rate, the IMEP decreased slightly. Its highest value at each EGR rate occurred at 10° BTDC.



(a) (b)

Figure 2.4 Effects of various pilot injection timings and EGR rates on the (a) P_{max} and (b) IMEP.

The BSFC of a diesel engine depends on the relationship between the fuel injection system and fuel properties such as specific gravity, viscosity, and heating value. The BSFC decreased as the EGR rate was increased by 1.9%, 2.7%, 3.6% and 2.2% at BTDC 10° compared with that at BTDC 5°, 0.5%, 1.6%, 3.6% and 2.2% at BTDC 15°, and 3.2%, 3.7%, 4.2% and 3.5% at BTDC 20°. As the pilot injection timing, advanced, the BSFC decreased. The lower BSFC at BTDC 20° means that a smaller amount of fuel was required to produce the same amount of power. This is expected because canola fuel has a higher density than pure diesel, so it can be fully burned through early injection and heating. Figure 2.5

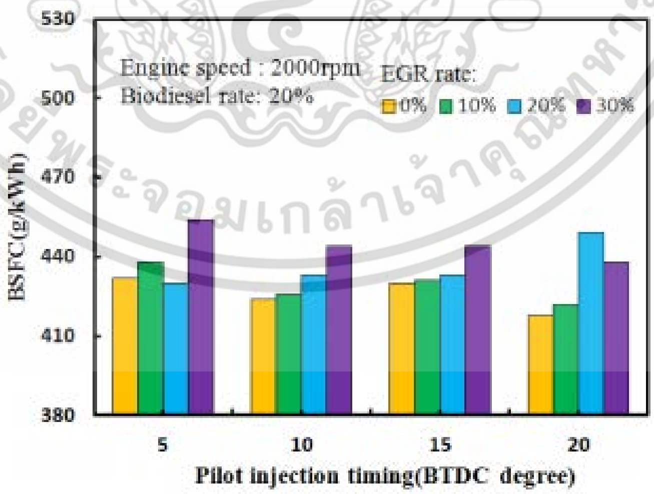


Figure 2.5 Effects of EGR ratio and pilot injection timing on the BSFC.

The BSEC is defined as the product of the BSFC and the heating calorific value of the fuel. It measures the amount of energy consumed to develop a unit of output power. Generally, the BSEC decreases as energy consumption increases. Figure 2.6

represents the variation of the BSEC under different EGR rates and pilot injection timings. It can be seen that the BSEC is slightly increased with EGR rate increasing. The exhaust gas increased in the combustion chamber that will lead to decrease the oxygen concentration when EGR rate increased, so more fuel is needed to produce the same power output. However, the BSEC changed slightly as pilot injection timing increased. It indicates that there is slight effect of pilot injection timing on the BSEC. This is because the combustion is mainly promoted by the oxygen in biodiesel.

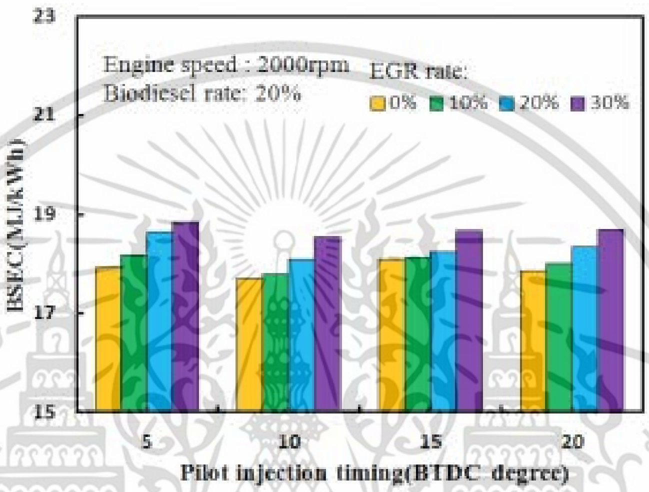


Figure 2.6 Effects of EGR ratio and pilot injection timing on the BSEC.

NO_x emissions tended to decrease as the EGR rate increased and showed a slight increase as the pilot injection timing was advanced while the EGR rate was held steady. The NO_x emission at an EGR rate of 30% decreased by 19.2% at BTDC 5°, 20.07% at BTDC 10°, 20.03% at BTDC 15°, and 19.29% at BTDC 20° on average compared to that at an EGR rate of 0%. When advancing the pilot injection timing, the PM emissions decreased by 16.35% at BTDC 10°, 11.03% at BTDC 15°, and 30.13% at BTDC 20° compared to PM emissions at each condition at BTDC 5° as shown in Figure 2.7

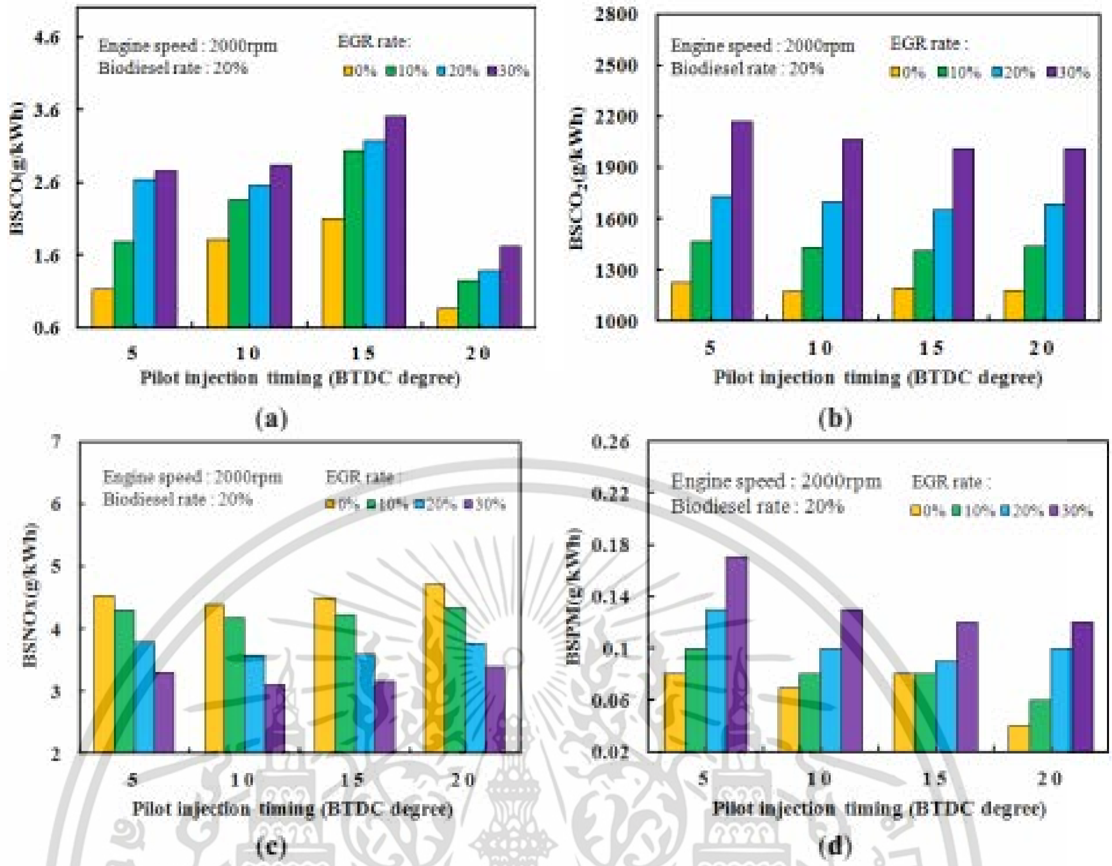


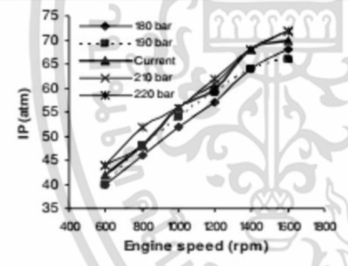
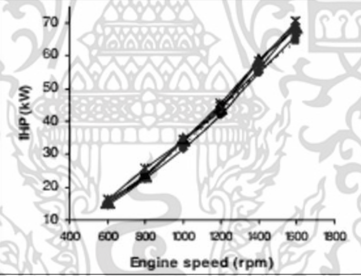
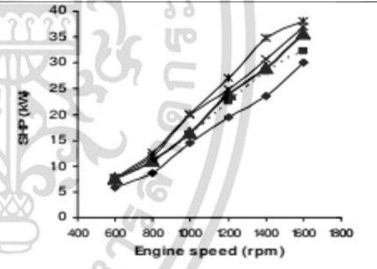
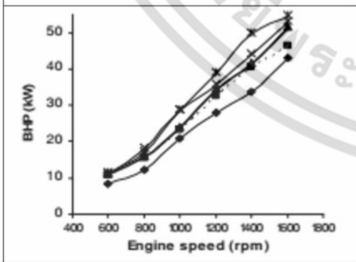
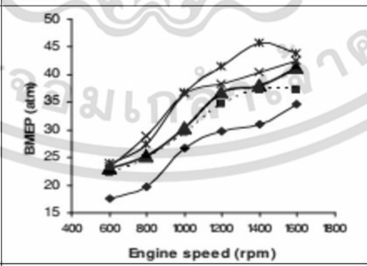
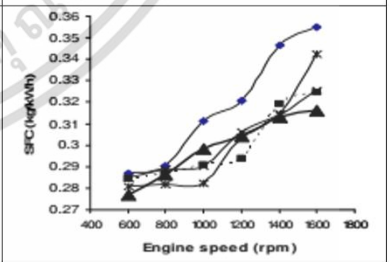
Figure 2.7 Effects of various pilot injection timings and EGR rates on the (a) BSCO; (b) $BSCO_2$; (c) $BSNO_x$; and (d) BSPM emissions.

On the effects on combustion stability: the combustion was delayed as the pilot injection timing retarded. In addition, the ignition delay was shorter when the pilot injection timing was closer to the main injection timing. On the effects on engine combustion: as the pilot injection timing was advanced, the combustion pressure and HRR changed slightly. As the EGR rate was increased, the combustion pressure and HRR decreased slightly. On the effects on engine performance: as the pilot injection timing was advanced, the P_{max}, BSEC, IMEP and BSFC changed slightly. As the EGR rate was increased, the P_{max} and IMEP decreased slightly, and the BSFC and BSEC increased slightly. On the effects on exhaust emissions: as the pilot injection timing was advanced, CO and PM emissions decreased considerably, with a minimum value at BTDC 20°; NO_x emissions increased slightly; and CO_2 emissions decreased slightly. As the EGR rate was increased, NO_x emissions decreased considerably, and CO, CO_2, and PM emissions increased.

2.2. The study of injection pressure

Rosli Abu Bakar (Bakar, Ismail, & Ismail, 2018) do an experiment in both of variation engine speeds - fixed load and fixed engine speed – variation loads by changing the fuel injection pressure from 180 to 220 bar. According to the results, the best performance of the pressure injection has been obtained at 220 bar, specific fuel consumption has been obtained at 200bar for fixed load – variation speeds and at 180bar for variation loads – fixed speed. Engine was tested in range of 600 - 1600 rpm with the interval of 200 rpm and the fuel injection pressure setting from 180 – 220 bar with the interval of 10 bar. In the second experiment the diesel engine loads were tested in 55% - 80% in interval of 5%, engine speed is fixed on 1600 rpm and the fuel injection pressure setting from 180 – 220 bar with the interval of 10 bar. The experiment results on fixed load of 67 kW, in interval engine speeds 600 until 1600 rpm and the fuel injection pressure setting from 180 – 220 bar. The variations for engine performance have given in Table 2.5

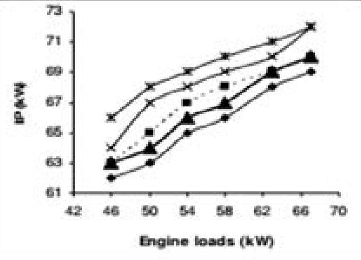
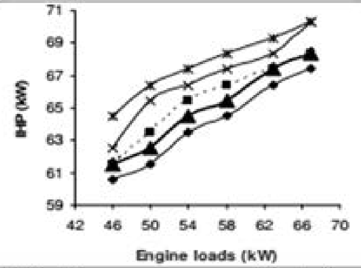
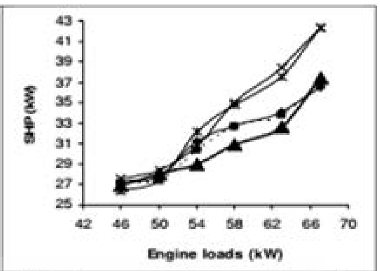
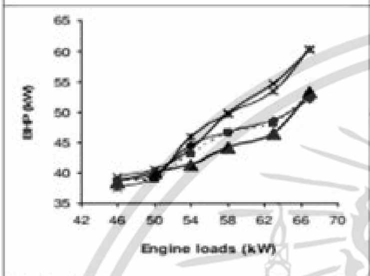
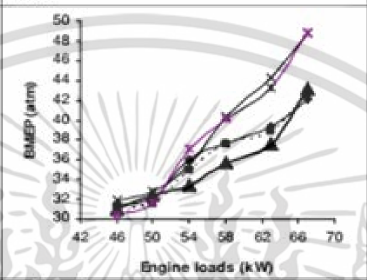
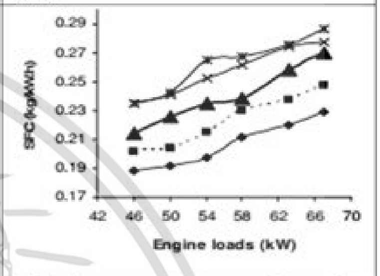
Table 2.5 Diagram form the tested result

		
Injection pressures effect for IP	Injection pressures effect for IHP	Injection pressures effect for SHP
		
Injection pressures effect for BHP	Injection pressures effect for BMEP	Injection pressures effect for SFC

The experiment results on fixed engine speed on 1600 rpm, in interval engine loads were tested in 55% - 80% in interval of 5% and the fuel injection pressure setting from 180 – 220 bar. The variations for engine performance have given in

Table **2.6**

Table 2.6 Diagram form the tested result

		
Injection pressures effect for IP	Injection pressures effect for IHP	Injection pressures effect for SHP
		
Injection pressures effect for BHP	Injection pressures effect for BMEP	Injection pressures effect for SFC

The first experiment on fixed engine load and variations engine speeds the result shows that in the averages performance. The best engine performance for indicated pressure (IP), indicated horse power (IHP), shaft horse power (SHP), break horse power (BHP) and break mean effective pressure (BMEP) obtained at 220 bar and the best engine SFC obtained at 200 bar or in current fuel injection pressure.

In the second experiment on fixed engine speed and variation engine loads, the result shown that increasing injection pressure have given that in engine performance was increased for indicated pressure (IP), indicated horse power (IHP), shaft horse power (SHP), break horse power (BHP) and break mean effective pressure (BMEP). Higher injection pressure given increased higher engine performance power and the highest engine performance was obtained at injection pressure 220 bar.

When the fuel injection pressure is low, fuel particle diameters will enlarge and ignition delay period during the combustion will increase. When injection pressure increased the fuel particle diameters will become small. Since formation of mixing of fuel to air becomes better during ignition period, engine performance will be increase. But, if the fuel injection pressure is too higher, ignition delay period becomes shorter. So, possibilities of homogeneous mixing decrease and combustion efficiency falls down. The fixed load variation speeds and fixed speed-variation loads have been given that the higher engine speed (rpm) given higher engine power. The increasing injection pressure is in line with increasing power. The fuel consumptions experiment results for fixed load-variation speeds and fixed speed-variation loads have been given that increasing injection pressure given increased of fuel consumption for the diesel engine.

2.3. The study of injection timing and injection pressure

Effect of fuel injection timing and pressure on combustion, emissions and performance characteristics of a single cylinder diesel engine was also study. (Agarwal, et al., 2013) The experiments were performed at constant speed (2500 rpm) with two FIPs (500, and 1000 bar) and different SOI timings. Pressure variations and ROHR showed superior combustion characteristics at lower FIP (500 bar), while at higher FIP (1000 bar), knocking was observed under certain engine operating conditions. Advanced injection timings led to rapid combustion hence higher ROHR was observed in early stages of combustion. Engine performance was superior at low FIPs leading to lower BSFC and higher BTE at all engine loads. These parameters can be further improved by advancing the SOI. Lower mass emission of CO_2 CO, HC and NO_x was observed at lower FIP. Emission characteristics improved by advancing the SOI. Particulate number concentration in a CI engine increased with increasing engine load. Increasing the FIP reduced the number concentration of particulates of all sizes at all loads. At higher FIP, advancing the injection timings reduced the particulate number concentration because advanced SOI timings provided longer time for fuel-air mixing before the start of combustion. At lower FIP, particulate number concentration first increased and then decreased with retarding SOI timings because mixing at lower FIP was more sensitive to cylinder pressure and temperature along with time available for mixing before the SOC.

2.4. Study on ethanol fumigation and Dual fuel

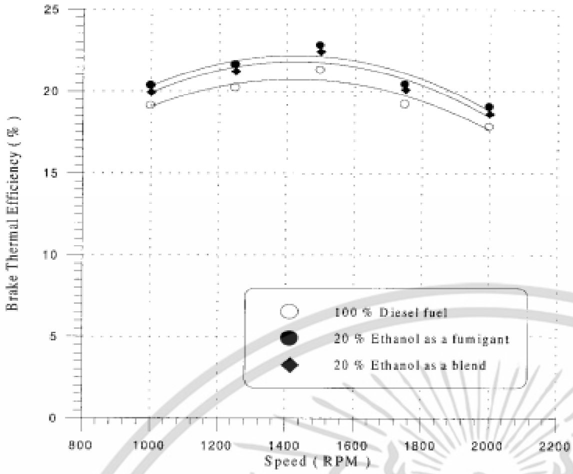
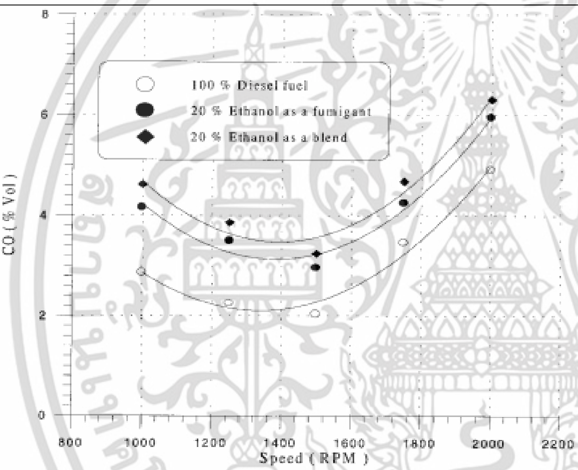
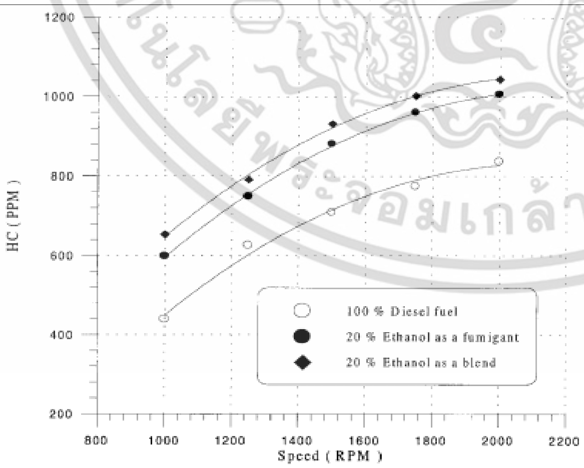
M. Abu-Qudais (Abu-Qudais, Haddad, & Qudaisat, 1999) study the effects of ethanol fumigation (i.e. the addition of ethanol to the intake air manifold) and ethanol diesel fuel blends on the performance and emissions of a single cylinder direct injection, variable compression ratio, diesel engine with a swept volume of 582 cm^3. The engine is naturally aspirated and water cooled 20^o injection timing and 18 compression ratio diesel engines. To determine the optimum percentage of ethanol that gives lower emissions and better performance at the same time. Ethanol delivery through a single hole 0.25 mm. of diameter. The nozzle was positioned approximately 50 cm. ahead of the inlet manifold. This allowed the ethanol to be mixed with the intake air for a sufficient period. The rack settings used were always between 3/4 rack and full rack settings.

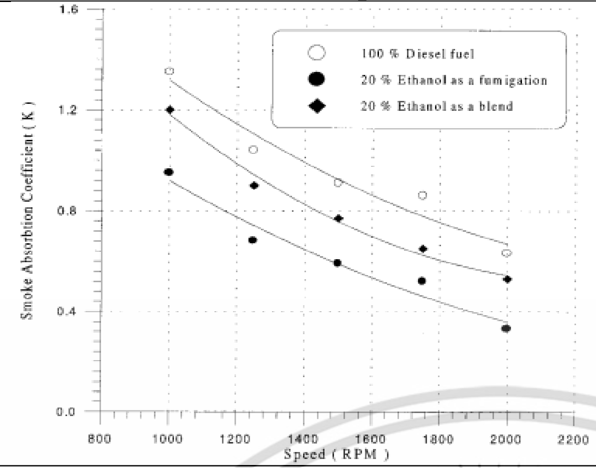
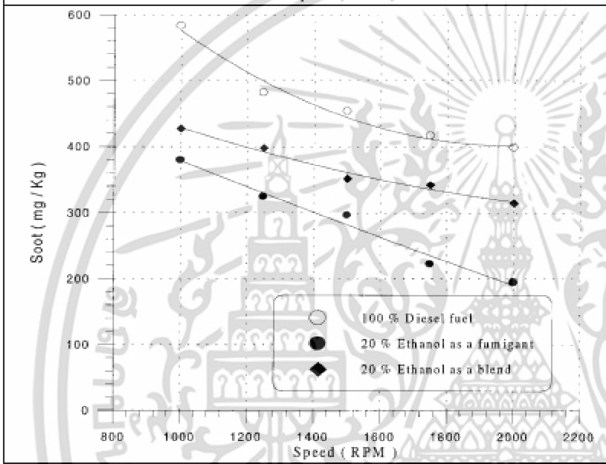
There are some difficulties as large percentages of alcohol do not mix with diesel fuel, hence use of diesel alcohol blends is not feasible. Also, the blends were not stable and separate in the presence of trace amounts of water, Alcohols have extremely low cetane numbers, whereas the diesel engine is known to prefer high cetane number fuels (45-55) which auto-ignite easily and give small ignition delay, Diesel fuels serve as lubricants for diesel engine. Alcohol fuels do not have the same lubricating qualities, and the poor auto-ignition capability of alcohols is responsible for severe knock due to rapid burning of vaporized alcohol and combustion quenching caused by high latent heat of vaporization and subsequent charge cooling.

To solve all difficulties above. There are several techniques involving alcohol-diesel dual fuel operation such as alcohol fumigation (the addition of alcohols to the intake air charge, displacing up to 50% of diesel fuel demand), dual injection (separate injection systems for each fuel, displacing up to 90% of diesel fuel demand), alcohol-diesel fuel blend (mixture of the fuels just prior to injection, displacing up to 25% of diesel fuel demand), and alcohol-diesel fuel emulsion (using an emulsifier to mix the fuels to prevent separation, displacing up to 25% diesel fuel demand).

The results show that both the fumigation and blends methods have the same behavior in affecting performance and emissions, but the improvement in using the fumigation method was better than when using blends. The optimum percentage for ethanol fumigation is 20%. This percentage produces an increase of 7.5% in brake thermal efficiency, 55% in CO emissions, 36% in HC emissions and reduction of 51% in soot mass concentration. This fumigation percentage produces a decrease of 48% in engine smoke and 51% in soot mass concentration. The optimum percentage for ethanol diesel fuel blends is 15%. This produces an increase of 3.6% in brake thermal efficiency, 43.3% in CO emissions, 34% in HC, it can also produce a reduction of 33.3% in engine smoke and 32.5% in the soot mass concentration. All the tested result with many different conditions will be shown as the Table 2.7

Table 2.7 Diagram form the tested result

<table>
<tr><th colspan="2">Effects of ethanol fumigation and ethanol diesel fuel blends on the performance and emissions</th></tr>
<tr><td>
</td><td>Brake thermal efficiency versus speed for ethanol fumigation and for fuel blends diagram. This improvement was approximately 7.5 and 5.4% over the entire speed range for ethanol fumigation and ethanol-diesel fuel blends.</td></tr>
<tr><td>
</td><td>CO emissions versus speed for ethanol fumigation and for fuel blends diagram. For 20% ethanol fumigation, the increase in CO emissions was in the range of 21-55% at the speed range used, and for 20% ethanol as a blend with diesel fuel, the increase in CO emissions was in the range of 28-71.5% at the same speed range used.</td></tr>
<tr><td>
</td><td>HC emissions versus speed for ethanol fumigation and for fuels blends diagram. For 20% ethanol fumigation, the increases in HC emissions were between 20 and 36%, and for 20% ethanol as a blend with diesel fuel, the increases were between 25 and 49% over the entire speed range.</td></tr>
</table>

Effects of ethanol fumigation and ethanol diesel fuel blends on the performance and emissions	
	Engine smoke versus speed for ethanol fumigation and for fuels blends diagram. There is a decrease in smoke coefficient of 30-48% for 20% ethanol as a fumigant, decreases of 11-25% for 20% ethanol as a blend with diesel fuel and a decrease of 18.5-33.3% for 15% ethanol as blend. The optimum percentage of ethanol for smoke reduction are 20 and 15% for the fumigation and blends methods.
	Soot emissions versus speed for ethanol fumigation and for fuels blends diagram. The maximum decrease (over the entire speed range) in soot concentrations was 33-51% for 20% ethanol fumigation, 10-22% for 20% ethanol blends and 15-32.5% for 15% ethanol-diesel fuel blends. Soot formation

Experimental studies on fumigation of ethanol in a small capacity Diesel engine (Chauhan, Kumar , Pal, & Jun, 2011), ethanol is widely investigated for applying in combination with Diesel fuel to reduce pollutants, including smoke and NO_x. Present work aims at developing a fumigation system for introduction of ethanol in a small capacity Diesel engine and to determine its effects on emission. Fumigation was achieved by using a constant volume carburetor. Different percentages of ethanol fumes with air were then introduced in the Diesel engine, under various load conditions. Ethanol is an oxygenated fuel and lead to smooth and efficient combustion. Atomization of ethanol also results in lower combustion temperature.

During the present study, gaseous emission has been found to be decreasing with ethanol fumigation. With fumigation of ethanol carburetion, exhaust temperature has been found to show at decreasing trend under all loading conditions. At higher loads the effect was more dominant. At full load exhaust temperature starts from 300°C and decreases to 265°C as ethanol substitution increases. At 70% load exhaust temperature starts from 250^{o}C and decreases to 220^{o}C as ethanol fumigation increases. Emission of carbon monoxide (CO) has been found that at no load CO% increases up to 30% of ethanol fumigation. However, at 20% and 45% of load, CO% decreases with ethanol fumigation and become constant with increase in ethanol percentage.

At 70% and full load, CO percentage decreases till 15% ethanol fumigation and then increases with increase in the ethanol percentage. At part load, CO_2% decreases as ethanol substitution increases. At full load CO_2% found to be less at 15% ethanol substitution. It was found that at full load NO_x emission decreases up to 16% ethanol fumigation then starts increasing ethanol fumigation. Ethanol fumigation results in the increase of unburned hydrocarbon (HC) emission in the entire load range compared to Diesel operation. However, the increment was low at higher loads and higher at part loads and no-load conditions. It was found that smoke opacity decreases as ethanol fumigation increases. It has been seen that slopes of this decreasing in smoke opacity increases on moving towards higher load conditions.

At high load smoke opacity decreases sharply up to 14% fumigation of ethanol and after that it decreases lightly. By using ethanol fumigation in a Diesel engine all emissions were found to be decreasing except HC emissions. This study shows that ethanol fumigation can be effectively employed in existing CI engine to achieve substantial saving of the scarce Diesel oil and at the same time to obtain improved engine performance with lesser exhaust NO_x and smoke pollution. Based on the above results, the optimum percentage of ethanol appears to be 15% for ethanol fumigation. Results from the experiment suggest that ethanol fumigation can be effectively employed in existing compression ignition engine to achieve substantial saving of the limited Diesel oil. Results show that fumigated

Diesel engine exhibit better engine performance with lower NO_x, CO, CO_2 and exhaust temperature. Ethanol fumigation has resulted in increase of unburned hydrocarbon (HC) emission in the entire load range. Considering the parameters, the optimum percentage was found as 15% for ethanol fumigation. Which show the same trend as many results from reviews, such as Review on alcohol fumigation on diesel engine. The result shows when fumigation alcohol is applied to the diesel engine, BSFC increase with the percentage of fumigation alcohol at all engine loads. Around 7–12% increase of BSFC in mass basis has been found in most of the reviewed studies, which is a consequence of the lower calorific value of alcohol.

Alcohol fumigation decreases BTE at low engine loads but there is a little increase in BTE at medium and high engine loads. The decrease in BTE has been found in the range of 5–13% and increase in BTE has been found in the range of 2–9%. Regarding gaseous emission, alcohol fumigation decreases NO_x emission compared to diesel fuel. NO_x emission is significantly affected by engine loads. The maximum reduction has been found to be 20% compared to pure diesel fuel at lower engine load for 30% fumigation in most of the experiments. Alcohol fumigation increases the CO and HC emission compared to diesel fuel.

The increase in CO emission has been found in the range of 1.00–29.4 g/kWh. On the other hand, increase in HC emission has been found in the range of 0.5–39.05 g/kWh. Alcohol fumigation significantly decreases the CO_2 emission which is corollary of CO emission reduction. Alcohol fumigation can substantially reduce smoke opacity and PM emission compared to diesel fuel. The reductions are mainly associated with the reduction of diesel fuel burned in the diffusion mode. The reductions have been found between a wider range of 14–57% at over all engine load conditions. (Imran , Varman, Masjuki, & Kalam, 2013)

Dual Fuel Engine Fueled with Ethanol and diesel fuel are also study to investigation of the possibility of dual fueling of compression ignition (CI) engine with diesel fuel and additionally with ethanol, which is injected into inlet manifold, was carried out. (Kowalewicz & Pajączek, 2003) Due to physio-chemicals properties of ethanol. Because some property of ethanol such as poor lubrication properties, poor miscibility with diesel fuel in presence of water, corrosion and chemical degradation of engine materials neat ethanol and diesel fuel.

Ethanol mixture cannot be used in CI engines. Investigation was carried out on one-cylinder, direct injection CI engine with modified inlet duct for injection of ethanol. Investigations showed that, energy ratio of ethanol to diesel fuel up to over 50% may be applied, engine thermal efficiency increased, with the increasing load, soot emission was decreased dramatically already for small amount of injected ethanol, emission of greenhouse gas (CO_2) was decreased with increasing amount of ethanol, NO_x emission was decreased for small load but for high load is kept at the same level as when the engine is fueled only on diesel fuel.

The combustion characteristics are also that the combustion time of dual fuel is shorter than combustion time of neat diesel fuel. Maximum combustion pressure depends on the energy ratio of ethanol to both fuels and on the load. They conclude that

Ethanol may be a good additional fuel to CI engines when being injected into inlet port in a proper proportion to diesel fuel, brake fuel conversion efficiency of the dual fuel engine is better than pure diesel engine, dual fueling results also in decrease of some emissions as NO_x, smoke level and greenhouse gas, ratio of ethanol to diesel fuel may be optimized from any point of view (efficiency or any toxic component of exhaust gases), injection timing of diesel fuel may also be optimized. The most promising seems to be 35°CA BTDC for best efficiency and 25°CA BTDC for NO_x emission.

High efficiency enabled by hydrous ethanol use in dual-fuel engines are studied. (Northrop, 2014) (Benajes, García , Serrano, & Boronat, 2016) The purpose of this project is to use hydrous ethanol to demonstrate high efficiency with reduced emissions in a modified diesel engine where ethanol provides up to 80% of the fuel energy input. Our approach improves on traditional ethanol fumigation in diesel engines and opens up new applications for ethanol as a diesel fuel replacement.

A key motivation for this work is previous research that suggests hydrous ethanol is less expensive to produce than anhydrous ethanol and more renewable due to lower fossil energy consumption. Eliminating dehydration and limiting distillation can save considerable energy in production and increases the sustainability of the overall process. Greater use of hydrous ethanol will improve both the economics and life cycle analysis of ethanol. Eliminating the requirement for after treatment devices reduces capital equipment and maintenance costs.

In this project, a new dual fuel combustion mode, reactivity-controlled compression ignition (RCCI) is used to reduce soot and oxides of nitrogen (NO_x) emissions while maintaining very high engine efficiency. In RCCI, a less reactive fuel like ethanol is injected into the intake port and diesel fuel is injected directly into the cylinder. RCCI operation has been successfully demonstrated over a wide range of engine load. NO_x and soot emissions well below the US EPA Tier 4 standards have been achieved, at the expense of slightly lower thermal efficiency and higher HC and CO emissions than in conventional diesel engines. With such remarkable improvement in engine out NO_x and soot emissions, the demand for catalytic after treatment can be significantly reduced.

2.5. Low Temperature Combustion

S. Imtenan (Imtenan , et al., 2014) is review about Impact of low temperature combustion (LTC) attaining strategies on diesel engine emissions for diesel and biodiesels. Future diesel engines will be operated on pure biodiesel and/or blends of biodiesel and crude oil-based diesel. LTC strategies decrease NO_x and PM simultaneously but increase HC and CO emissions. Recent attempts to attain LTC by biodiesel have created a hope for reduced HC and CO emissions. Decreased performance issue during LTC is also being taken care of by latest ideas. Three general methods can be applied to the engines to meet lower regulated emission limits, alternation of fuels, and alternation of combustion processes and after-treatment of the exhaust.

LTC is a general term for Homogeneous Charge Compression Ignition (HCCI) combustion, and Premixed Charge Compression Ignition (PCCI) combustion. LTC takes place at temperatures below the formation regime of NO_x and at local equivalence ratios below the formation regime of diesel soot. NO_x forms in the lean mixture zone where flame temperature is above 2200^o K, whereas soot forms in the rich mixture zone above 1800^o K. LTC techniques like HCCI and PCCI avoid these zones and reduce NO_x and soot simultaneously. Recently, a new approach of LTC, Reactivity Controlled Compression Ignition (RCCI)

HCCI combustion shows two-stage heat release. The first stage is low temperature kinetic reactions and the second stage is main heat release regime. The main advantage of the HCCI combustion over conventional combustion mode is the reduction of NO_x and soot in the exhaust. To achieve HCCI there are many different combustion control strategies e.g. port fuel injection, early direct injection, multiple fuel injection, compound combustion technology, narrow angle injection, late direct injection, variable inlet temperature, variable valve timing, internal or external EGR, etc.

Premixed charge compression ignition or the partially premixed charge compression ignition (PCCI) evolved from the HCCI combustion mode. It is not fully homogeneous like HCCI. It achieves desired ignition delay through enhanced charge motion reduced compression ratio, higher injection pressure and extensive use of EGR. They are, advanced direct injection, port fuel injection and late direct injection. They suffer from fuel spray impingement on the cylinder walls and incomplete fuel evaporation. Consequently, HC and CO emissions increased. However, narrow spray angle injectors and EGR reduce the wall impingement. Late direct injection avoids the fuel-wall impingement and gives a way to switch the combustion style to the conventional at higher loads.

Researchers have tried to increase the high load limits and reduce the emissions of PCCI by applying additives and tuning fuel properties, variable valve timing, multiple injections, and fuel–air mixing enhancement. A newer approach in PCCI introduces air–fuel premixing by early injection followed by a late injection of fuel pulse in the compression stroke, which governs the onset of ignition. Early injected fuel stratifies in the cylinder with the air and as the compression stroke reaches near the TDC (top dead center) it creates HCCI like condition. When the late direct injection occurs, the fuel-rich area of the late injection burns before the fuel lean homogeneous mixture. This variable fuel–air mixture prevents the entire charge from igniting instantaneously which gives a better control over the combustion phase and rate. Moreover, adoption of higher EGR permits longer ignition delay. It permits better premixing of air–fuel, results in less fuel-rich pockets followed by a low temperature combustion, which simultaneously reduces NO_x and soot level

RCCI Reactivity controlled compression ignition is the newest approach where multiple fuels of different reactivity are injected at scheduled intervals which governs the reactivity of the charge in the cylinder for the desired combustion duration and magnitude. In this mode, relatively low reactive fuel is injected (port injection) very early in the engine cycle which mixes with the air homogeneously. Later on, a higher reactive fuel is injected directly into the cylinder.

LTC strategies generally employ enhanced pre-combustion mixing, which helps to avoid locally rich regions. LTC modes take the combustion temperature below the formation temperature of PM. Less locally fuel rich region means low PM formation. Similarly, using EGR results in dilution by pre-combustion mixing which reduces the peak combustion temperature. The peak combustion temperature at LTC can be reduced to about 2000^{o}K or even lower, which drastically reduces the thermal NO_x formation. Low NO_x emission basically depends on higher ignition delay and lower combustion rate, which result in lower in-cylinder temperature. This means higher use of EGR and optimized SOI. For biodiesels, condensation of unburned fuel sometimes increases PM emissions UHC and CO emissions increase during LTC of diesel for the reason of reduction in in-cylinder combustion temperature and oxygen concentration.

Biodiesels reduce UHC and CO emissions than diesel during LTC modes because of their short-premixed combustion duration and higher oxygen content. Still, the emission level remains higher than the conventional combustion system. LTC strategies show higher fuel consumption. Unburned fuel (in the form of HC and CO in the exhaust) due to higher EGR rate or late injection timing is responsible for higher fuel consumption. Using oxygenated fuels or additives, high injection pressure with in-cylinder swirl formation, optimized use of fuel vaporizer and finally, use of different fuels having a wide range of reactivity for the RCCI combustion system can be potential fields for future studies

The internal EGR is used at the gasoline HCCI engines to control the start of the combustion. The internal EGR can be obtained in two modes: by closing the exhaust valve with an advance before the top dead center (TDC) or by reopening the exhaust valve during the intake process. To obtain the homogeneous combustion the internal EGR strategy has to be used to raise the temperature of the fresh charge. To obtain different EGR rates variable valve timing (VVT) mechanisms have to be used. The disadvantage of this type of mechanisms is the fact that the lift curve cannot be modified, it can only be displaced. (COSGAREA, ALEONTE, & COFARU, 2011) Hot exhaust gases are re-admitted into the cylinder, increasing the mixture temperature, and then allowing activation of the oxidation catalyst in the posttreatment system (which permits to decrease HC and CO emissions). It can be achieved using a Variable Valve Actuation (VVA) device. Exhaust valve reopened, consisting of burned gases and unburned air from the previous combustion. Fresh mixture (air + burned gas coming from external EGR) is aspirated into the cylinder through the intake valve. Meanwhile, exhaust gases are re-aspirated into the cylinder, through the exhaust valve.

(Leroy, Bitauld, & Chauvin, 2009) Additionally, or alternatively, it is likewise possible to close the exhaust valve with a delay, i.e. later than usual, so that exhaust gas already expelled from the cylinder can be drawn back out of the exhaust system or out of the region of the internal combustion engine which is associated with the exhaust system again during the next intake stroke. A method for operating an internal combustion engine with internal exhaust gas recirculation is known from DE 10 2009034 763. In this case, an intake valve is opened a second time in order to force out part of the exhaust gas into the fresh gas duct during the exhaust cycle and to draw it back into the combustion chamber during an intake cycle and/or to draw exhaust gas forced out into the exhaust duct back into the combustion chamber by a second opening and closing of the exhaust valve during an intake cycle. This is an example of an internal EGR with back-suction from the fresh air side or from the exhaust side with the aid of an additional opening operation of the respective intake valve or of the respective exhaust valve. The advantages of internal exhaust gas recirculation are considerably lesser space requirements compared with external exhaust gas recirculation. However, the cost in terms of apparatus for adjusting the EGR rate for internal EGR is considerably greater than for external EGR. (Schilling, Otto, & Roessler, 2015) Illustrates in-cylinder mean effective pressure (MEP) as a function of early exhaust valve opening. It further illustrates how increased IEGR reduces achievable engine loads under stoichiometric combustion and constant intake manifold conditions.

"Mean effective pressure" is used in its common meaning as the mean theoretical piston-top pressure of an engine. Both the Brake MEP (BMEP) and the net indicated (nIMEP) MEP decrease with early exhaust valve opening. The pumping MEP (PMEP) is not affected. (Koci, Mehta, & Roberts Jr, 2013) Cost savings may be realized as external passages, an EGR valve and an EGR cooler are not required. Associated external EGR fouling issues Would thus be avoided. Additionally, volumetric efficiency Would be improved with the two-stage expansion and compression, such that a turbocharger may not be required. Emissions Would be improved by enabling a higher EGR percentage While maintaining a high air-to-fuel ratio. (Durrett & Gopalakrishna, 2014)

CHAPTER 3

MATERIALS AND METHOD

3.1. Test apparatus

3.1.1. Engine Layout

The engine was based on commercial agriculture engine. Many parts of the engine need to modified to support the experiment. The cylinder head was replaced with new head which will able to fitted with high injection pressure injector and also the pressure sensor. The new injector will be able to controlled the injection strategy such as injection pressure and injection timing. Many necessary data such as cooling temperature, fuel consumption, emission of the exhaust gas and performance of the engine will be monitor and collected as a data to further analyze. All equipment mentioned above will be connect to the engine in various part as show in Figure 3.1

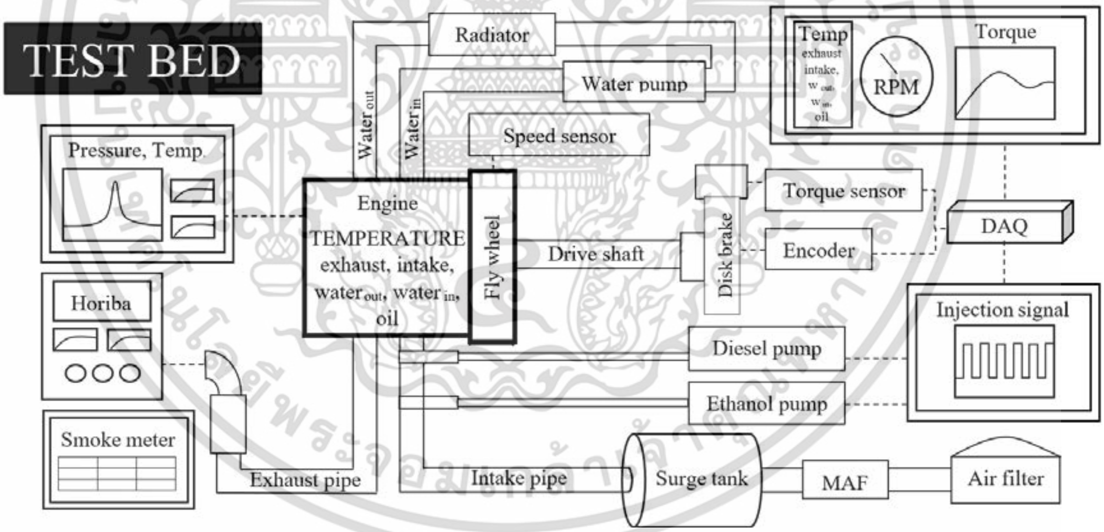


Figure 3.1 Engine layout

3.1.2. Dynamometer System

In this study, base engine is Kubota RT140 Figure 3.2 which is single cylinder water cooled 4-stroke engine. And the engine specification shows in appendix B Table 0.1. The engine will be equipped with common-rail system for the test to control the injection pressure and injection timing properly. By using common-rail system together with direct injection system, then one more injector will be fixed at the intake manifold pipe to inject ethanol in the intake manifold. All injectors will be controlled by FPGA system.

Figure 3.2 Kubota RT140

The cooling system of the engine will be replaced to be as a conventional cooling system in the commercial car to increase the cooling capacity of the engine and to control the coolant temperature. The water cooling system will compose of water pump and the radiator with fan to blow out heat from cooling water. And will be show in Figure 3.3.

Figure 3.3 Water cooling system

The engine will be equipped with performance evaluation system by disc brake system (to measure torque from the tested engine). The performance evaluation system composes of disc braking system as shown in Figure 3.4 . To convert mechanical energy from the engine to be force then generate to the Load cell shown in Figure 3.5. The specification of the load cell is shown in appendix B Table 0.2.

Figure 3.4 Disc brake system

Figure 3.5 Load cell

The pressure in the combustion chamber will be evaluate in many different crank shaft angles. The crank shaft angle will be measure by Rotary Encoder incremental type Autonics E40HB. This shaft encoder resolution is 3600 pulses per revolution (PPR). which will show in Figure 3.6.

Figure 3.6 Rotary encoder

3.1.3. Diesel Injection System

The injection system of diesel fuel will be replaced by a common-rail system from the conventional mechanical pump of the original Kubota engine. This system composes of high pressure, suction control valve, high pressure rail and high-pressure injector. All parts will be shown in Figure 3.7.

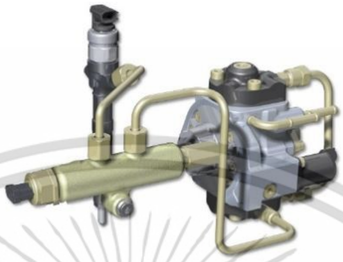


Figure 3.7 Diesel fuel injection system

3.1.4. Ethanol Injection System

The injection system of ethanol will compose of ethanol pump, pressure relive valve which will control the injection pressure for 3 bar, pressure gauge, and the ethanol injector. All above equipment will be show in Figure 3.8.

Figure 3.8 Ethanol injection system

3.1.5. Data Acquisition System

The engine speed will be measure by using proximity sensor. Figure 3.9. This sensor was use to check that the data acquire from the encoder was correct. And during the test. Engine speed in very important parameter. Which need to be controlled by the tester.

Figure 3.9 Proximity sensor

The characteristic of the engine performance will be analyzed by pressure rise in the combustion chamber by using Pressure Transducer piezoelectric type Kitsler Type 6052 C as shown in Figure 3.10 and its specification will show in appendix B Table 0.5.

Figure 3.10 Kistler Type 6052 C

The pressure transducer will be install to engine cylinder head. The signal from pressure transducer will be amplified by Charge Amplifier Kistler Type 5018 as shown in Figure 3.11. to convert the electric signal to pressure unit. Then this data will use to analyze the combustion characteristic.

Figure 3.11 Charge Amplifier Kistler Type 5018

During the test, many parameters will be collected from various sensors. Which is consist of crank angle position sensor, ethanol and diesel injection controller. Temperature will be measured by K Type Thermocouple. Which will show in Figure 3.12. This equipment measurement range is -20 to 1,350 C^o.

Figure 3.12 Thermocouple K type

Engine speed, pressure data, and temperature data at various part of the engine will be collect and recorded in to a computer called Data Acquisition as shown in Figure 3.13 and crank shaft angle signal would be the input data to control the injection timing.

Figure 3.13 Data Acquisition

In this experiment, fuel consumption will be evaluated by using weight change during the operation of the engine. OHAUS PA4102 as shown in Figure 3.14 is the weight scales and have 0.01 gram for resolution

Figure 3.14 Weight Scales

3.1.6. Exhaust gas analyzer System

To measure the emission from of the experiment, Horiba Mexa 1600D emission analyzer was use as shown in Figure 3.15. This analyzer is able to measure carbon dioxide, carbon monoxide, oxide of nitrogen, hydrocarbon compound, and oxygen. While smoke, amount will be measured by AVL Smoke Meter 415SE as shown in Figure 3.16. The specification of emission analyzer Horiba Mexa 1600D and smoke meter specification will show in appendix B
Table **0.3**, and Table 0.4 respectively.

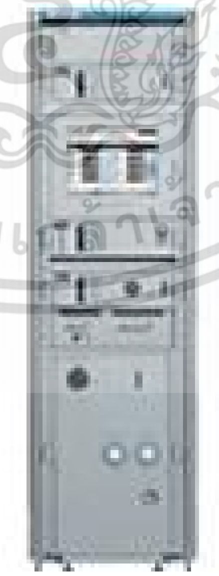


Figure 3.15 Horiba Mexa 1600D

Figure 3.16 AVL Smoke Meter

To measure the mass of air flow into the engine. Pressure sensor was used to measure the pressure different from the venturi. And also compare with the mas air flow sensor. The indicator shows that there is not significant error from these two systems.

3.2. Computation and Analysis method

To do this experiment. The original injection system will be replaced by common-rail system to give more controllable and precision. The injection amount is very necessary parameter in this experiment. Before the injection test every part of the system will be tested. The original injection amount of the original Kubota engine will be determined from the engine specification to do further experiment. To find the injection amount, engine torque was use then convert to diesel fuel amount requirement.

a. Fuel property and fuel amount balancing

Fuel properties	Diesel fuel	Ethanol
Specific gravity [kg/m^3] at 15 ^{o}C	0.84	0.785
Viscosity [cP] at 20 ^{o}C and 1atm	3.35	1.2
Molecular weight	170	46.07
Higher heating value [kJ/kg]	46100	29700
Lower heating value [kJ/kg]	43200	26900
Heat of vaporization [kJ/kg]	270	840
Cetane number	50	8

To find the original parameters like injection amount, the injection timing and the injection duration. First, the energy consumed by the engine must be derive. This mean that the fuel consumption of the engine must be evaluated with the original engine specification Figure 3.17. The fuel consumption can be calculated by using torque output from the engine.

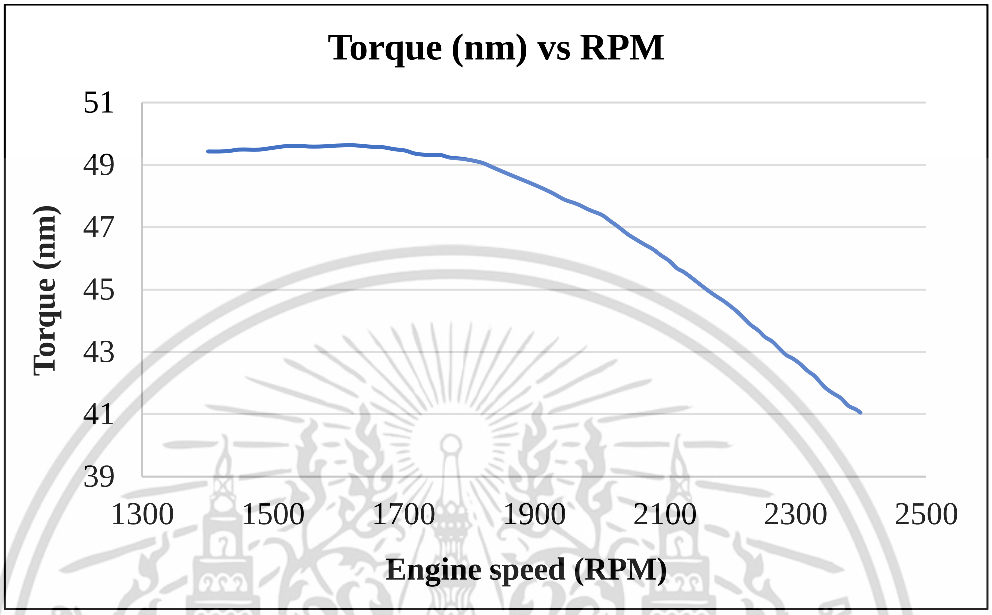


Figure 3.17 Kubota Engine Performance Curve

And converted to fuel consumption. The specific fuel consumption at continuous rated output is 231 g/hp-hr. from the Kubota engine.

$$\text{Fuel consumption}\left(\frac{\text{gram}}{\text{cycle}}\right) = \text{Power(HP)} \times \frac{231}{60 \times \text{rpm}} \qquad (3.1)$$

As the engine speed changed, the power output from the engine will also change. By the previous reason, the engine will consume vast different fuel consumption at different engine speed Figure 3.18.

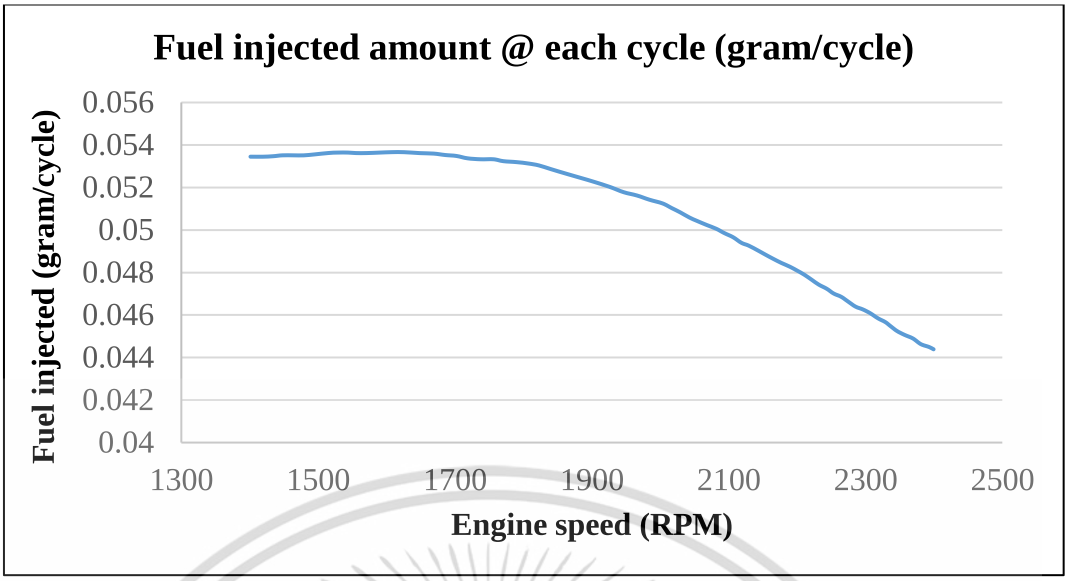


Figure 3.18 Injection amount require

After the fuel consumption amount was determine. This parameter can be convert into energy required at any engine speed and every ethanol fumigation amount as

Figure **3.19**. And this energy requirement at each engine speed will be compare with the characteristic of the injector. Which will convert to injection amount as show in Figure 3.20. The last step is convert to the injection duration of each fuel.

Total energy input amount of diesel and ethanol in each condition								
RPM	1400		1600		1800		2000	
kJ/cycle	Diesel	Ethanol	Diesel	Ethanol	Diesel	Ethanol	Diesel	Ethanol
E0	1.8730	0	1.6647	0	1.4055	0	1.2041	0
E10	1.6857	0.1873	1.4982	0.1665	1.2649	0.1405	1.0837	0.1204
E20	1.4984	0.3746	1.3318	0.3329	1.1244	0.2811	0.9632	0.2408
E30	1.3111	0.5619	1.1653	0.4994	0.9838	0.4216	0.8428	0.3612
E40	1.1238	0.7492	0.9988	0.6659	0.8433	0.5622	0.7224	0.4816
E50	0.9365	0.9365	0.8324	0.8324	0.7027	0.7027	0.6020	0.6020
E60	0.7492	1.1238	0.6659	0.9988	0.5622	0.8433	0.4816	0.7224
E70	0.5619	1.3111	0.4994	1.1653	0.4216	0.9838	0.3612	0.8428
E80	0.3746	1.4984	0.3329	1.3318	0.2811	1.1244	0.2408	0.9632
E90	0.1873	1.6857	0.1665	1.4982	0.1405	1.2649	0.1204	1.0837
E100	0	1.8730	0	1.6647	0	1.4055	0	1.2041

Figure 3.19 Energy input from diesel and ethanol at various condition

Injection amount of diesel and ethanol in each condition								
RPM	1400		1600		1800		2000	
g/cycle	Diesel	Ethanol	Diesel	Ethanol	Diesel	Ethanol	Diesel	Ethanol
E0	0.0434	0	0.0385	0	0.0325	0	0.0279	0
E10	0.0390	0.0070	0.0347	0.0062	0.0293	0.0052	0.0251	0.0045
E20	0.0347	0.0139	0.0308	0.0124	0.0260	0.0104	0.0223	0.0090
E30	0.0303	0.0209	0.0270	0.0186	0.0228	0.0157	0.0195	0.0134
E40	0.0260	0.0279	0.0231	0.0248	0.0195	0.0209	0.0167	0.0179
E50	0.0217	0.0348	0.0193	0.0309	0.0163	0.0261	0.0139	0.0224
E60	0.0173	0.0418	0.0154	0.0371	0.0130	0.0313	0.0111	0.0269
E70	0.0130	0.0487	0.0116	0.0433	0.0098	0.0366	0.0084	0.0313
E80	0.0087	0.0557	0.0077	0.0495	0.0065	0.0418	0.0056	0.0358
E90	0.0043	0.0627	0.0039	0.0557	0.0033	0.0470	0.0028	0.0403
E100	0	0.0696	0	0.0619	0	0.0522	0	0.0448

Figure 3.20 Injection amount of diesel and ethanol at various condition

b. Injection timing and duration

The injection timing is depending on many factors. Such as engine speed, type of the combustion chamber, and etc. Similar to the injection duration. The injection duration in depend on the energy content of each fuel, injection pressure, and the characteristic of the injector. Before the determination of the injection duration. Characteristic of the injector need to be clarify. The diagram below Figure 3.21 will show the characteristic of the injector that was use in this experiment. And in this experiment, there were two injectors. Another injector was use to inject ethanol.

And it also need to find the injection characteristic as well. But for the ethanol injector, the injection pressure will be fixed at 3bar which will show in Figure 3.22. After the amount consuming in each engine speed. Then, the injection duration can be defined by comparing the injection amount required by the engine to the injection amount that the injector can inject in a period of time at each injection pressure.

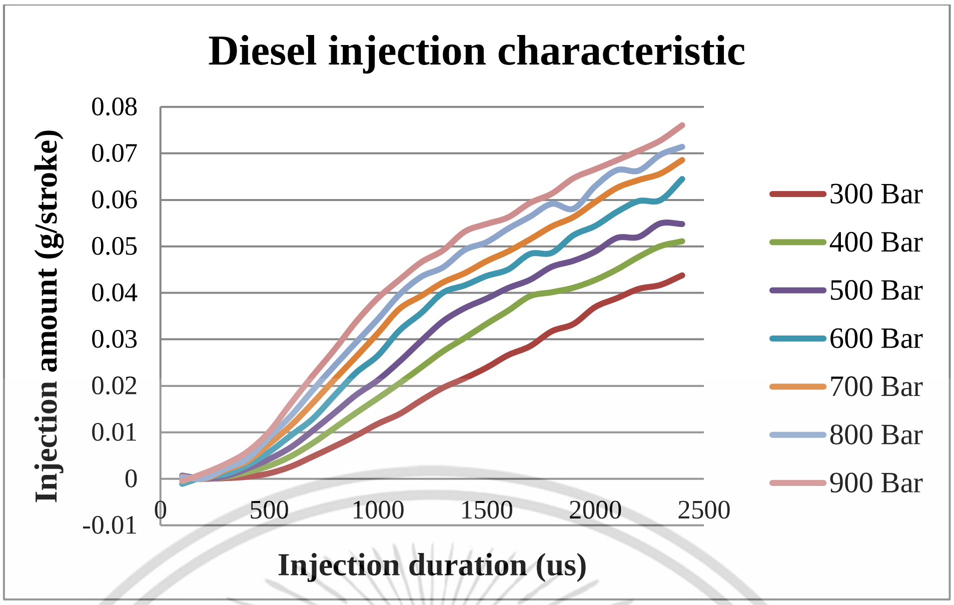


Figure 3.21 Diesel Injector Characteristic

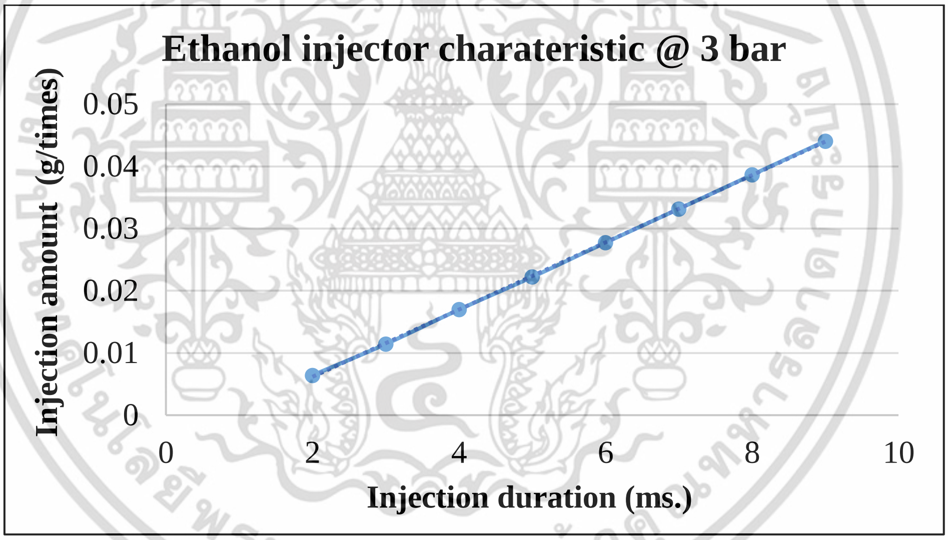


Figure 3.22 Ethanol Injector Characteristic

c. Apparent Heat Release Rate

Heat release rate is the parameter that tells about how much heat were generated by the combustion of the engine. It is typically measured in Joules per second or Watts. But in this experiment. Apparent Heat Release Rate was use to explain the heat produced by the engine compare with the crank angle of the engine by using the pressure that was produce in the combustion stroke.

$$dQ_{net} = \frac{1}{K-1}V\frac{\partial P}{\partial \theta} + \frac{K}{K-1}P\frac{\partial V}{\partial \theta} - \frac{PV}{(K-1)^2}\frac{\partial K}{\partial \theta} \quad (3.2)$$

$$K = 1.386 + 1.778\times10^{-4}T - 5.293\times10^{-7}T^2 + 4.004\times10^{-10}T^3 - 9.932\times10^{-14}T^4 \quad (3.3)$$

Where K = Specific heat ratio, depends on the gas temperature (in this experiment K = 1.3)

T = Average gas temperature (K^o)

The Specific heat of air can be constants, with slice loss of accuracy, Because of the high temperature, and large temperature range was experienced during an engine cycle. So, the specific heat value was fix at 1.3. At low temperature, end of the cycle during intake and start of compression, K = 1.4 (temperature around 25^{o}C). Air flow before enter to the engine. But at the end of combustion, temperature has risen so K=1.3 is more accurate. (Pulkrabek, 1997)

d. BSEC, Brake Specific Energy Consumption (kJ/kW-hr)

BSEC is Brake specific energy consumption is the ratio of energy obtained by burning fuel for an hour to the actual energy or Brake power obtained at the wheels. It is dimensionless. It is indicative how effectively the energy obtained from the fuel is reaching the wheels.

$$BSEC = \frac{\dot{m}_f \times \vartheta}{P} \quad (3.4)$$

Where P = Power (kW)

$\dot{m}_f$ = Mass flow rate (kg/h)

ϑ = Heating Value (kJ/kg)

e. Specific emission (g/kW-hr)

Specific emissions are defined as emissions of pollutants per transport unit (passenger-km or tone-km), specified by mode (road, rail, inland, maritime, air). The pollutants considered include NO_x, HC, PM and CO. For passenger transport, specific emissions are expressed in grams of pollutant (NO_x, HC, PM, CO) per passenger-kilometer. For freight transport, specific emissions are expressed in grams of pollutant (NO_x, HC, PM, CO) per ton-kilometer. In this experiment, the specific emission will express as grams per power output from the engine (kW)

$$(SE)_{NO_x} = \frac{\dot{m}_{NO_x}}{\dot{W}_b} \tag{3.5}$$

$$(SE)_{CO} = \frac{\dot{m}_{CO}}{\dot{W}_b} \tag{3.6}$$

$$(SE)_{HC} = \frac{\dot{m}_{HC}}{\dot{W}_b} \tag{3.7}$$

$$(SE)_{PM} = \frac{\dot{m}_{PM}}{\dot{W}_b} \tag{3.8}$$

Where $\dot{m}_{NO_x}$ = NO_x emission

$\dot{m}_{CO}$ = CO emission

$\dot{m}_{HC}$ = HC emission

$\dot{m}_{PM}$ = PM emission

$\dot{W}_b$ = Power (kW)

f. Air fuel ratio, equivalent ratio, Lambda

AFR, Air–fuel rate is the mass ratio of air to a solid, liquid, or gaseous fuel present in a combustion process. The combustion may take place in a controlled manner such as in an internal combustion engine

$$AFR = \frac{m_{air}}{m_{fuel}} \tag{3.9}$$

Where m_{air} = Mass of air

m_{fuel} = Mass of fuel

Air fuel equivalence ratio (λ) Air–fuel equivalence ratio, λ (lambda), is the ratio of actual AFR to stoichiometry for a given mixture. $\lambda = 1.0$ is at stoichiometry, rich mixtures $\lambda < 1.0$, and lean mixtures $\lambda > 1.0$.

$$\lambda = \frac{\text{AFR}}{\text{AFR}_{\text{stoich}}} \quad (3.10)$$

Where AFR_{stoich} = Stoichiometric AFR

g. Fuel chemical balance

$$(1\text{-}W)(0.93C_{12}H_{22}+0.07C_{19}H_{34}O_2)+W(C_2H_5OH)+\frac{Z}{2}(O_2+3.76N_2)$$

$$\rightarrow X(CO_2)+Y(H_2O)+Z(3.76N_2) \quad (3.11)$$

Where
$C_{12}H_{22}$ = Pure diesel
$C_{19}H_{34}O_2$ = Biodiesel
C_2H_5OH = Ethanol
W = Percentage of ethanol compound
X = $\big((1\text{-}W)(0.93\times12)\big)+\big((1\text{-}W)(0.07\times19)\big)+(W\times2)$
Y = $\big((1\text{-}W)(0.93\times22)\big)+\big((1\text{-}W)(0.07\times34)\big)+(W\times6)$
Z = $(2\times X)+Y\text{-}W$

$$\frac{A}{F}=\frac{m_a}{m_f}=\frac{\dot{m}_a}{\dot{m}_f}=\frac{(0.5Z\times16\times2)+(0.5Z\times14\times3.76\times2)}{\begin{gathered}\left[12\times\big(\big((1\text{-}W)(0.93\times12)\big)+\big((1\text{-}W)(0.07\times19)\big)+(W\times2)\big)\right]+\\ \left[1\times\big(\big((1\text{-}W)(0.93\times22)\big)+\big((1\text{-}W)(0.07\times34)\big)+(W\times6)\big)\right]+\\ \left[16\times\big((1\text{-}W)(0.07\times2)\big)\right]\end{gathered}} \quad (3.12)$$

CHAPTER 4

RESULTS AND DISCUSSIONS

This part of the paper will discuss the influence of diesel direct injection engine with ethanol fumigation operation on performance and exhaust gas emission. These two parameters will represent a pros and cons from the variation of the experiment condition such as increasing the ethanol fumigation amount, changing the injection pressure and injection timing. In each section of the experiment condition, performance and exhaust gas emission will be discuss separately to clearly explain the effect from the variation. But before the experiment start, the boundary condition must be investigated to find the limitation of the engine operation to prevent any factors that will damage to the engine and others equipment. The boundary condition will be investigated in the next chapter.

4.1. Experiment Boundary Limitation

In this experiment, the operating conditions when varying the amount of fumigated ethanol, injection timing and injection pressure were investigated. Three injection pressures (300, 500 and 700 bar) and seven injection timings including advanced and retarded timing were studied while the amounts of ethanol were increased until the engine cannot run properly. Figure 4.1 presents the operating condition in which the test engine can run smoothly without knocking and misfiring. After the knocking occurred, performance of the engine became worst and the engine could not run stably. The engine speed fluctuated. Finally, the engine was shut down.

High ethanol fumigation amount lead to cause knocking because ethanol has extremely low cetane numbers, whereas the diesel engine prefers high cetane number fuels which auto-ignite easily and give small ignition delay, Poor auto-ignition capability of ethanol is responsible for severe knock due to rapid burning of vaporized ethanol and combustion quenching caused by high latent heat of vaporization and subsequent charge cooling. (Abu-Qudais, Haddad, & Qudaisat, 1999)

Higher injection pressure means finer fuel droplet (T. Minami, 1990), results in significantly faster atomization improve fuel–air mixing. followed by faster combustion. The combustion process may start before the piston move to TDC. Some part of the mixture inside the combustion chamber will be ready to combust, those part may ignite before the main combustion process occur before the TDC, which may take place in any part of the chamber. Knocking will be found earlier when the injection pressure is high with the same injection timing.

For advanced injection timing at the same injection pressure, the engine easily got knocking when the amounts of ethanol fumigated were increased. At the most advanced injection timing of 334 degree BTDC, only 30%, 20 % and 10% by energy of ethanol can fumigated into the intake port at injection pressure of 300, 500 and 700 bar while 80%, 40% and 30 % of ethanol can be injected at the retarded timing of 355 degree BTDC. The reason is due to longer delays between injection and ignition means longer time available for prepare the air and fuel mixture before the combustion which lead to unacceptable rates of pressure rise (diesel knock) because too much fuel is ready to burn when combustion eventually occurs. (Nwafor, 2000)

	334°CA			337°CA			340°CA			343°CA			346°CA			350°CA			355°CA		
E0	300	500	700	300	500	700	300	500	700	300	500	700	300	500	700	300	500	700	300	500	700
E10	300	500	700	300	500	700	300	500	700	300	500	700	300	500	700	300	500	700	300	500	700
E20	300	500		300	500		300	500	700	300	500	700	300	500	700	300	500	700	300	500	700
E30	300			300			300	500		300	500		300	500		300	500		300	500	700
E40										300			300			300			300	500	
E50													300			300			300		
E60																300			300		
E60																300			300		
E70																300			300		
E80																			300		

Figure 4.1 Experiment boundary condition

And before discussing the result on performance and emission from the experiment. Another parameter that is very important in this experiment is air and fuel ratio. Because this experiment was directly related to the using of alternative fuel as a duel fuel injection strategy which will affect to the air and fuel ratio compare with conventional diesel engine. In this experiment, the fuel mixture is composed of ethanol, and diesel will mix with air which will affect to the performance and the exhaust gas emission. This parameter will be easily show that there is too high amount of fuel (rich) or too low amount of fuel (lean) of the fuel mixture in the combustion process. This parameter calls lambda (λ).

$\lambda < 1.00$ mean there is not enough air to burn completely the amount of fuel, after combustion there is unburnt fuel in the exhaust gases or rich. $\lambda = 1.00$ mean the mass of air is exact for a complete combustion of the fuel, after combustion there is no excess oxygen in the exhaust and no unburnt fuel or stoichiometric (ideal). $\lambda > 1.00$

mean there is more oxygen than required to burn completely the amount of fuel, after combustion there is excess oxygen in the exhaust gases. (Patreon, n.d.)

There are two parts for lambda in this experiment. The first part is when the ethanol was injected into the intake manifold, the intake charge mixture will change. And Figure 4.2 shows that more ethanol fumigation means richer mixture which gives same character in the second part as lambda at the combustion chamber as show in Figure 4.3 . This character will affect to the engine output. The reason is when ethanol fumigation was applied, the proportion of ethanol injected in the intake manifold will be replace the intake air. This means the air intake to the combustion chamber will be decrease.

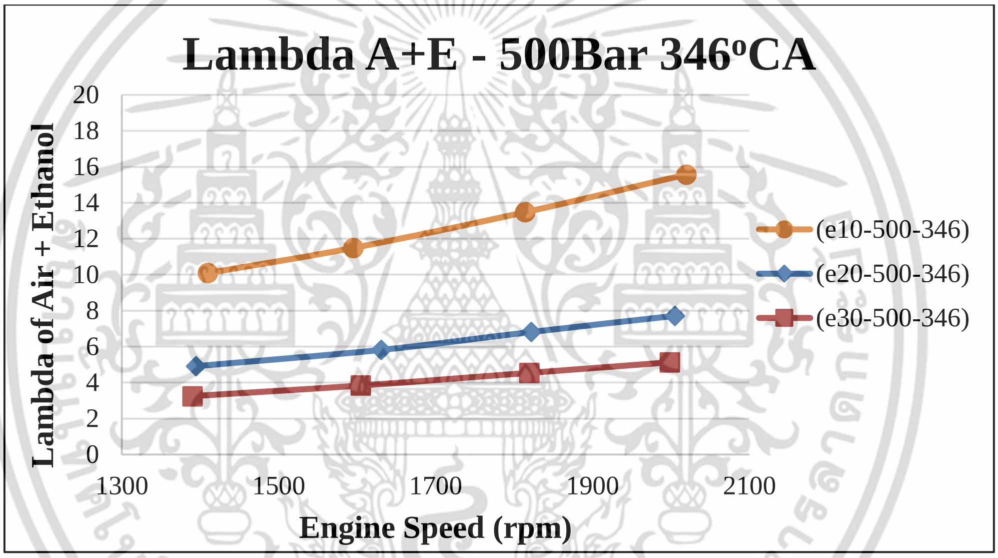


Figure 4.2 Lambda at the intake manifold

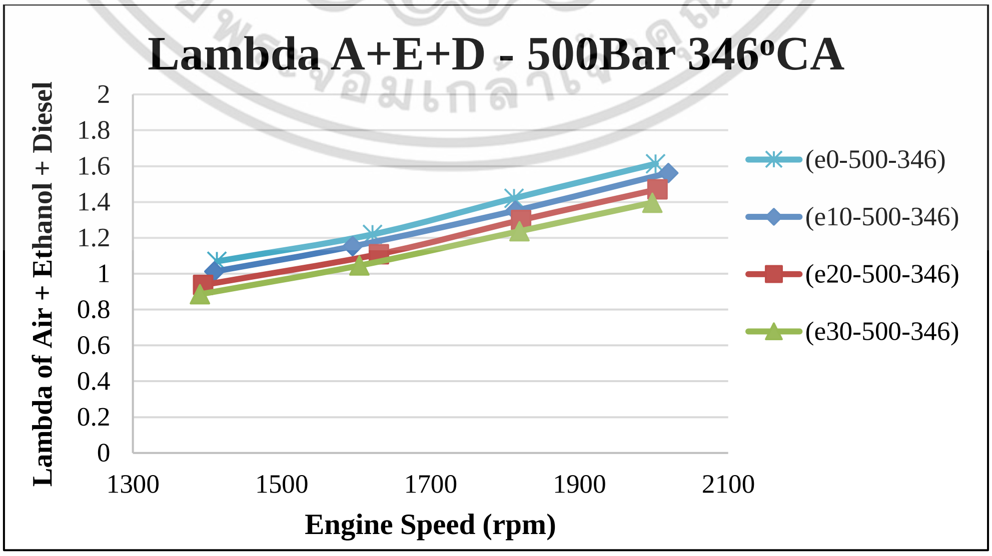


Figure 4.3 Lambda at the combustion chamber

4.2. Effect from Ethanol Fumigation

To explain the effect of ethanol fumigation 500 bar injection pressure at 346 degree crank angle injection timing was chosen with the variation of ethanol fumigation from E0 to E30. Because at this condition, the engine can run from E0 up to E30 smoothly without engine speed fluctuation which is caused by too advance injection timing (knocking), and the engine still able to give high power compare with too retard injection timing. 500 bar injection pressure will be prevented the engine from overheating and knocking. The reason will be discussed later in the follow explanation.

4.2.1. Performance

It is not fair to find the fuel consumption of the dual fuel injection strategy. Because the different of energy content in each fuel. Ethanol has low energy content, so higher amount of ethanol required by mass to achieve the same energy amount. So, in this paper, engine performance will be defined the amount of energy consumed per unit of brake power of the engine in a unit time. This indicates the ability to change the energy from the combustion of fuel to the brake power. BSEC or brake specific energy consumption is a good parameter to show the energy consumption of the engine while increasing the engine speed.

BSEC increase when ethanol fumigation was increased as show in Figure 4.4 . This is because of ethanol will longer ignition delay due to the lower cetane number and gives rapid and high heat release during the combustion process cause the firction loss, which will lower the power output and the BSEC will be increase. This directly implies a poorer combustion efficiency of the dual-fuel operation which results in an increased fuel consumption. (Leermakers, 2012) Due to the increasing of ethanol fumigation amount. The air intake charge will be decrese by the substitution of ethanol. As the engine speed increase the BSEC value increase, this is due to the lower power of engine, from higher engine temperature (M. M. Rahman, 2008). High velocity of air flow in the intake manifold will cause higher friction at the surface of intake manifold which will reduce the intake air amount.

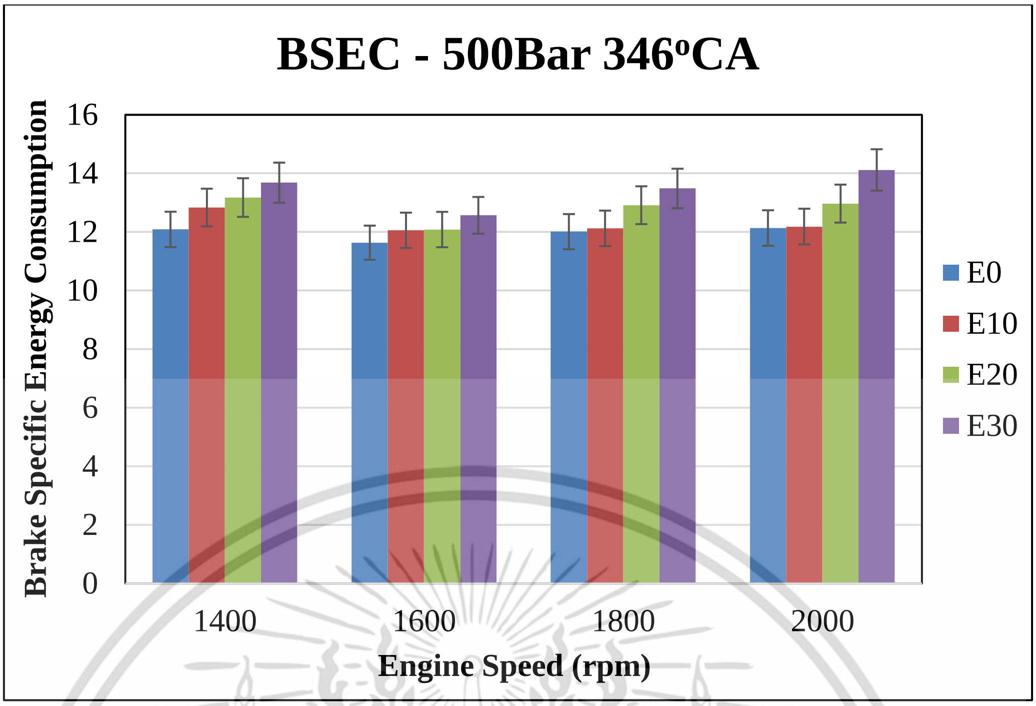


Figure 4.4 BSEC from Ethanol fumigation amount variation

4.2.2. Emission

Compression ignition engine as diesel engines are considered as one of the largest contributors to environmental pollution caused by exhaust emissions. In this paper, the emissions from diesel engines monitored. The four main pollutant emissions from diesel engines (Soot concentration, carbon monoxide-CO, nitrogen oxides-NO_x and Total hydrocarbon-THC) which will be describe below respectively.

- **Soot emission**

For soot emission, when higher amount of ethanol was fumigated, soot concentration was decreased as show in Figure 4.5. The main reason is because the amount of diesel injected into the combustion chamber were lessen when increase the ethanol proportion which. Soot are generated in the combustion chamber in the fuel-rich zones where there is not enough oxygen to convert all carbon to CO_2. Adding ethanol will increase also the ignition delay, which will give better air and fuel utilization before the combustion take place, this reason can also contribute the reason why soot concentration was reduce when the engine speed increase. As turbulence and mass motion continue to mix the components in the combustion chamber at high engine speed, most of these carbon particles find sufficient oxygen to further react and are consumed to CO_2. So, soot concentration will be decrease when the engine operates in high speed. (Abu-Nameh, 2007)

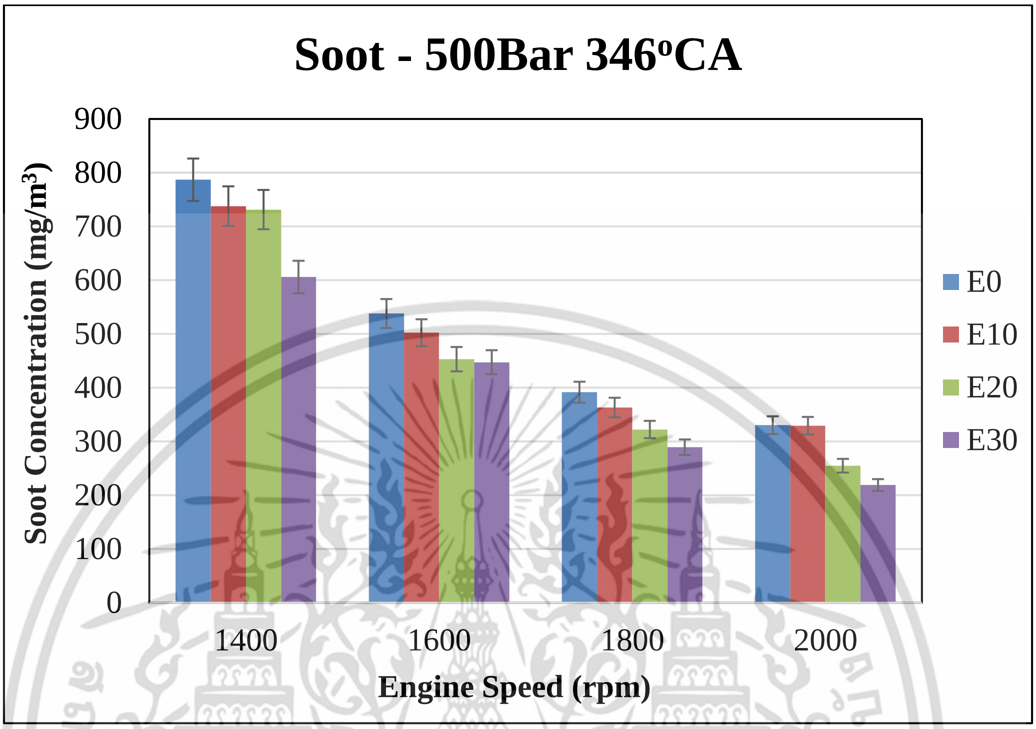


Figure 4.5 Soot emission from Ethanol fumigation amount variation

- **CO emission**

CO increase when increase the ethanol fumigation amount as show in Figure 4.6. Two major causes are a rich fuel mixture (more fuel than is needed), or restricted air supply (dirty or plugged air filter) due to ethanol fumigation which CO production increase. Adding ethanol in the intake manifold may reduce the amount of air intake to the combustion process which means less oxygen to react with carbon atom. Incomplete conversion of fuel carbon to CO_2 results in part from insufficient O_2 in the combustion mixture of fuel-rich conditions and insufficient time to oxidize fuel carbon fully to CO_2 such as in under full load condition. Poor mixing, local rich regions, and incomplete combustion will create some CO. (Gadhia, 2013). Higher ethanol fumigation will cause the richer mixture and it also reduce the combustion duration process which may not sufficient time to convert carbon monoxide to carbon dioxide. Unburn fumigated ethanol may also cause higher CO emission. Higher engine speed will increase the ability of fuel mixing as mentioned before. So, increasing engine speed can reduce the CO emission.

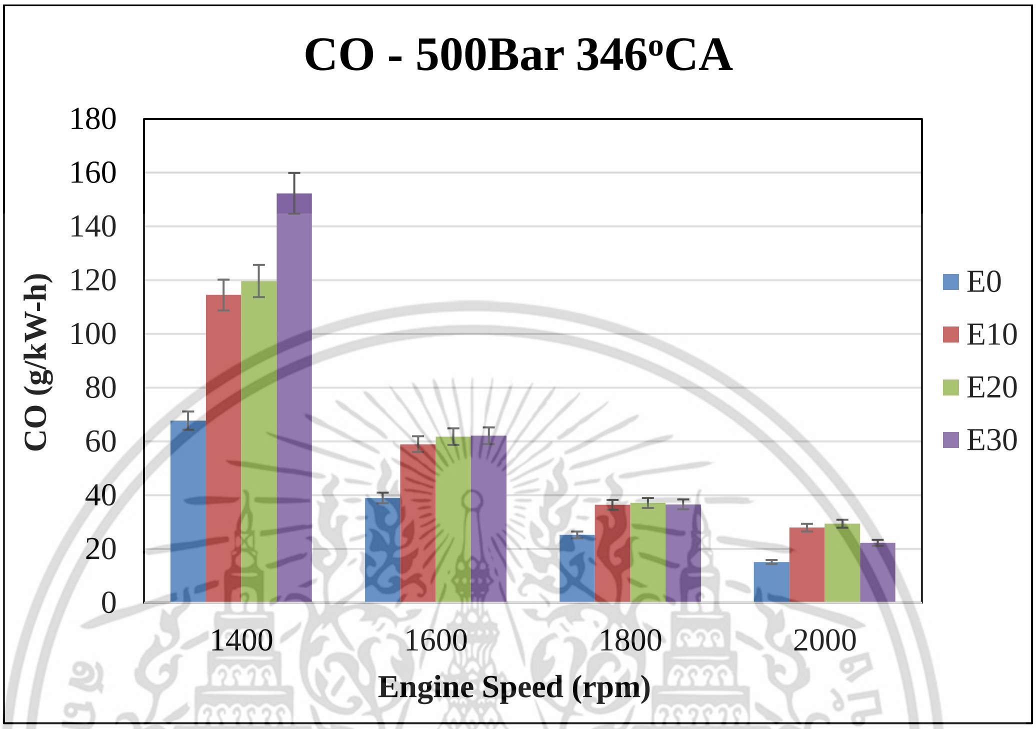


Figure 4.6 CO emission from Ethanol fumigation amount variation

- **NO_x emission**

The generation of nitrogen oxide emissions is a function of the combustion temperature, being greatest near stoichiometric conditions when temperatures are the highest. Peak NO_x emissions occur However, at the very high temperatures that occur in the combustion chamber of an engine, some diatomic nitrogen (N_2) breaks down to monatomic nitrogen (N) which is reactive. The chemical equilibrium constant is highly dependent on temperature. (Hebbar, 2014)

Adding ethanol will give longer ignition delay. Before the combustion start, there are more amount of fuel that ready to combust and also have enough time to get better fuel utilized during the ignition delay period, when the combustion starts, the temperature during the combustion process will be very high compare to no ethanol fumigation case, this reason can explain the increasing of NO_x emission as show in Figure 4.7. NO_x formation is highly dependent on temperature, So, when the engine speed increased, the combustion temperature also increases, more NO_x was generated.

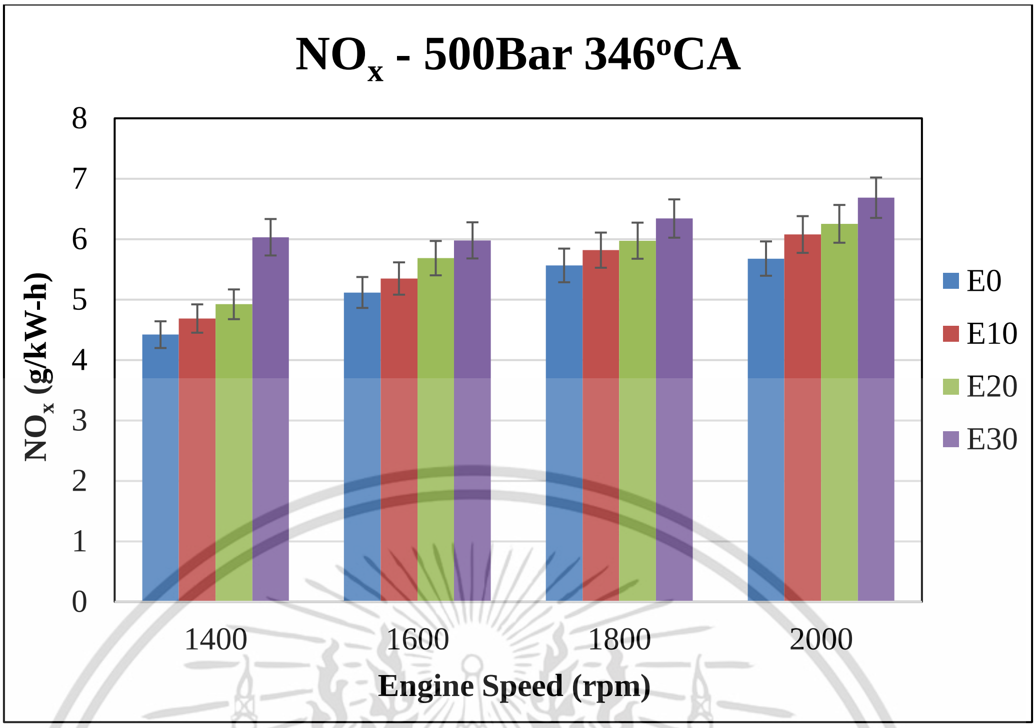


Figure 4.7 NO_x emission from Ethanol fumigation amount variation

- **THC emission**

THC will be produced when the combustion of rich mixture. THC increase when increase the ethanol fumigation amount as show in Figure 4.8. Because there is too less oxygen to react with all the carbon in the fuel. Due to its low energy content, higher proportion of ethanol need to inject which mean richer mixture. And ethanol will longer the ignition delay because of lower cetane number, which results in the significant rise of THC emissions. And another reason is because ethanol has high latent heat of vaporization, which reduces the intake charge temperature and promotes the chilling of cylinder wall, which will give very high temperature during the combustion process. Unburn fumigated ethanol at the quench layer may another source of THC. (Lílian Lefol Nani Guarieiro a, 2014) Adding more ethanol in the combustion process will also increase the combustion pressure Which will lead to increase the unburn leakage. This phenomina calls crevice effect. Higher engine speed will increase the THC when high amount of ethanol fumigation because the higher temperature.

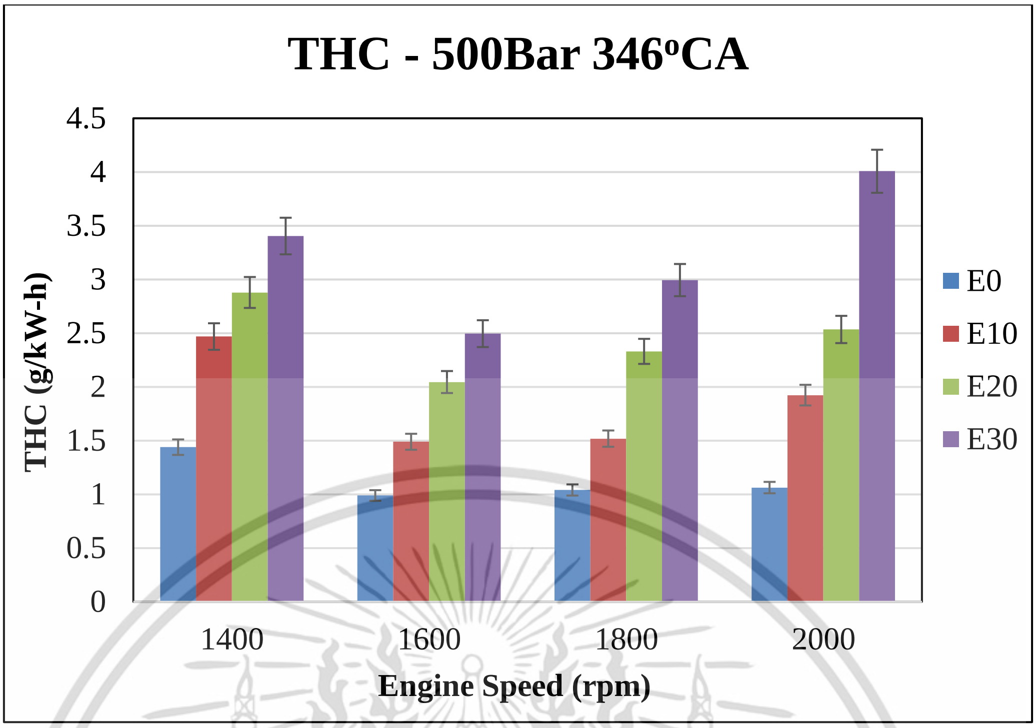


Figure 4.8 THC emission from Ethanol fumigation amount variation

4.3. Effect from Injection Pressure

To explain the effect of injection pressure, E20 with 346 degree crank angle injection timing was chosen with the variation of injection pressure (300, 500, and 700bar). For higher amount of ethanol proportion, higher temperature of the engine was found. Knocking and power lose will be found when the ethanol fumigation amount is too high as mention before. And as mention before. 346 degree crank angle injection timing was fix which will be discuss later in the follow explanation.

4.3.1. Performance

Increase the injection pressure will increase the spray tip penetration, spray area and reduce the fuel droplet size. 500 bar injection pressure give best result for every engine speed for E20 as show in Figure 4.9 Low injection pressure at 300 bar causes larger diameter of the diesel fuel droplet, this reduce the fuel distribution which will lower the ability of mixing with the intake charge. High injection pressure will shorten the ignition delay. Hence, possibility of homogeneous mixing decreases. (Avinash Kumar Agarwala) High injection pressure also causes a rapid pressure rise during the combustion process as a result, combustion efficiency reduces because too high combustion pressure cause higher friction loss. 500 bar injection pressure may suit for the tested combustion chamber to atomize the liquid fuel into small droplets in order to enable rapid vaporization as well for high jet penetration in the combustion chamber, small droplets and high penetration depth of fuel jet enhances the fuel–air mixture quality, which provides shorter ignition delays and more complete combustion. For low engine speed the BSEC value slightly decrease, because of the longer time in each cycle will allows more heat release. And at high engine speed. The BSEC will increase because of the friction lose which mean higher energy consumption.

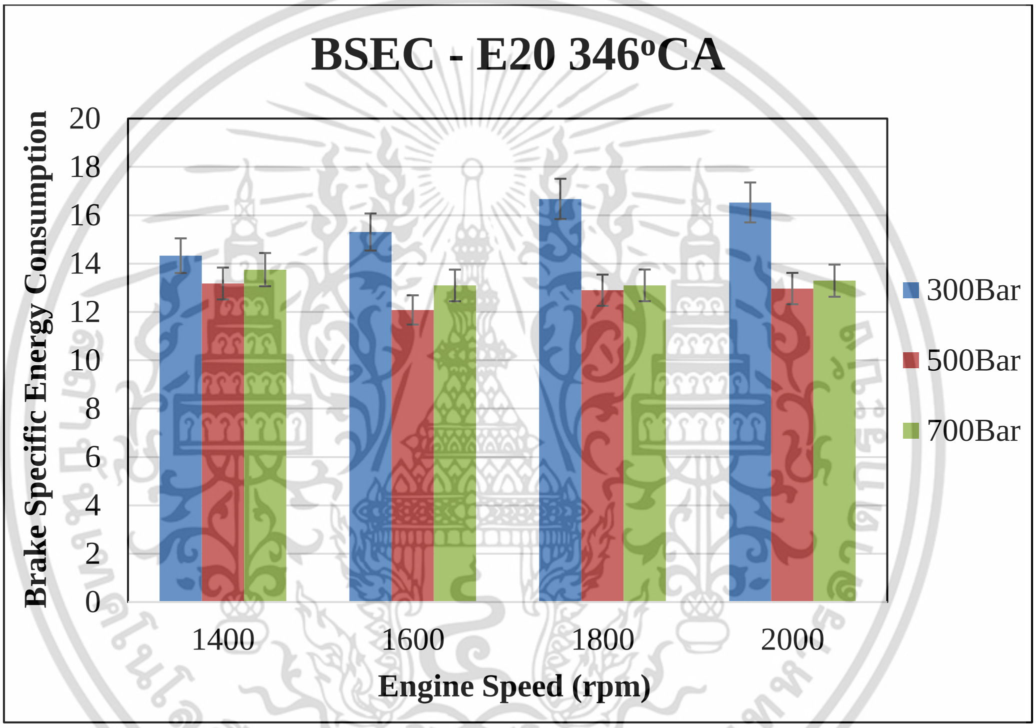


Figure 4.9 BSEC from Injection Pressure variation

4.3.2. Emission

- **Soot emission**

The exhaust of compression ignition engines contains solid carbon soot particles that are generated in the fuel-rich zones within the cylinder during combustion. These are seen as exhaust smoke and are an undesirable odorous pollution. Maximum density of particulate emissions occurs when the engine is under load at wide open throttle. At this condition maximum fuel is injected to supply maximum power, resulting in a rich mixture. The injection amount of diesel will be highest. Wide open throttle or full load condition will generate high smoke because in this reason. (Pulkrabek, 1997)

300 bar injection pressure gives highest soot emission. The reason is about the distribution of fuel. This could be due to relatively big injected diesel droplets. Fuel and air utilization at low injection pressure will less than the high injection pressure condition. Higher injection pressure gives lower soot emission as show in Figure 4.10. The reason is high injection pressure higher injection pressure gives a finer droplet size which lead to increase the contact surface area of air and fuel, spray droplet diameter distribution reduces. This lead to improved fuel–air mixture formation because of superior mixing during ignition delay, therefore soot emission reduce. High engine speed can reduce the smoke emission. Because when engine speed increases, the temperature will rise up, and with high speed, turbulence in the combustion chamber will increase which cause better combustion. When fuel injection pressure increased, the combustion temperature will also increase which accelerate the oxidation.

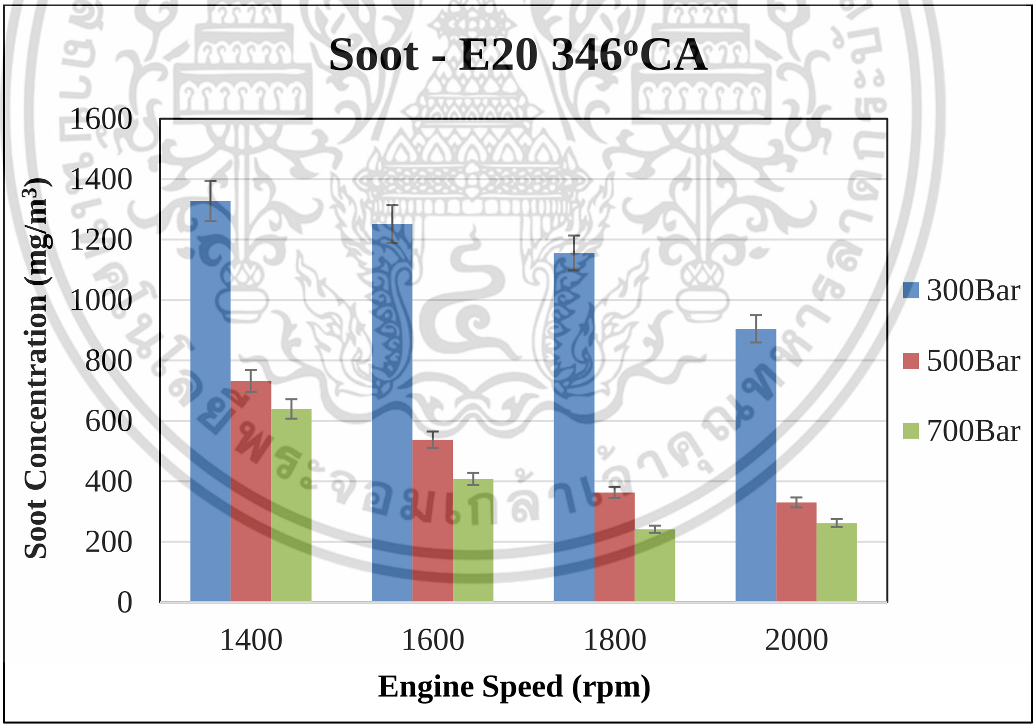


Figure 4.10 Soot emission from Injection Pressure variation

- **CO emission**

Emission of CO is greatly dependent on the air–fuel ratio relative to the stoichiometric proportions. Rich combustion invariably CO. Low injection pressure will give higher CO. This could be due to relatively big injected diesel droplets which may cause the rich mixture zone. When fuel injection pressure increased, smaller spray droplet diameter will increase the distribution. This lead to improved fuel–air mixture formation because of better air fuel mixing before the combustion take place as mention before, CO emission reduce as shown in Figure 4.11. As the engine speed increased, the values of CO decreased. Increase in speed could probably augment volumetric efficiency, boosting turbulence in combustion chamber hence ensure better combustion. (H. J. Parekh, 2018)

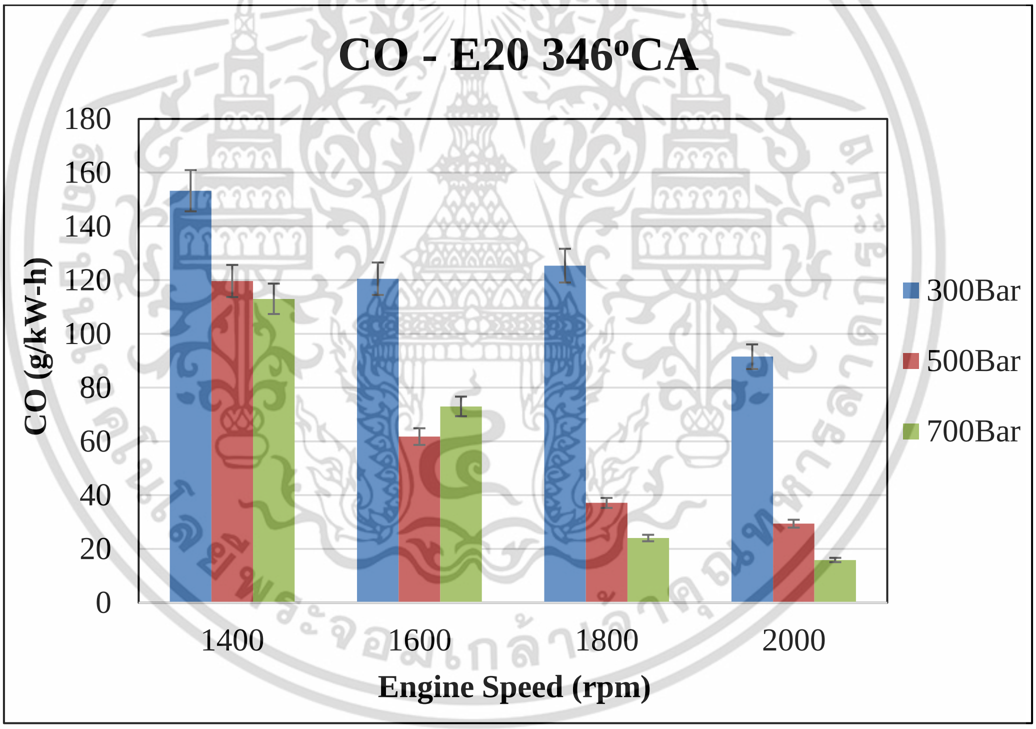


Figure 4.11 CO emission from Injection Pressure variation

- **NO_x emission**

NO_x emissions increased significantly with increasing fuel injection pressure as show in Figure 4.12 Because high injection pressure mean higher heat was produce during combustion process in the combustion chamber due to improved combustion which will be explain later in the combustion analysist section. High injection pressure will promote the fuel distribution first with small fuel droplet and also more turbulence. Most of the fuel will be combust during the combustion stroke. This strategy of combustion will give very high peak combustion temperature. NO_x emissions were higher for 700 bar fuel injection pressure compared to 300 bar fuel injection pressure. The fuel vapor penetration length before the onset of ignition has been significantly altered at a higher injection pressure, which had an important impact on the high temperature zone and thus on the increased formation rate of NO_x emissions. For high engine speed condition, temperature of the combustion will also increase, which cause higher NO_x. (Mateusz Pucilowski, 2017)

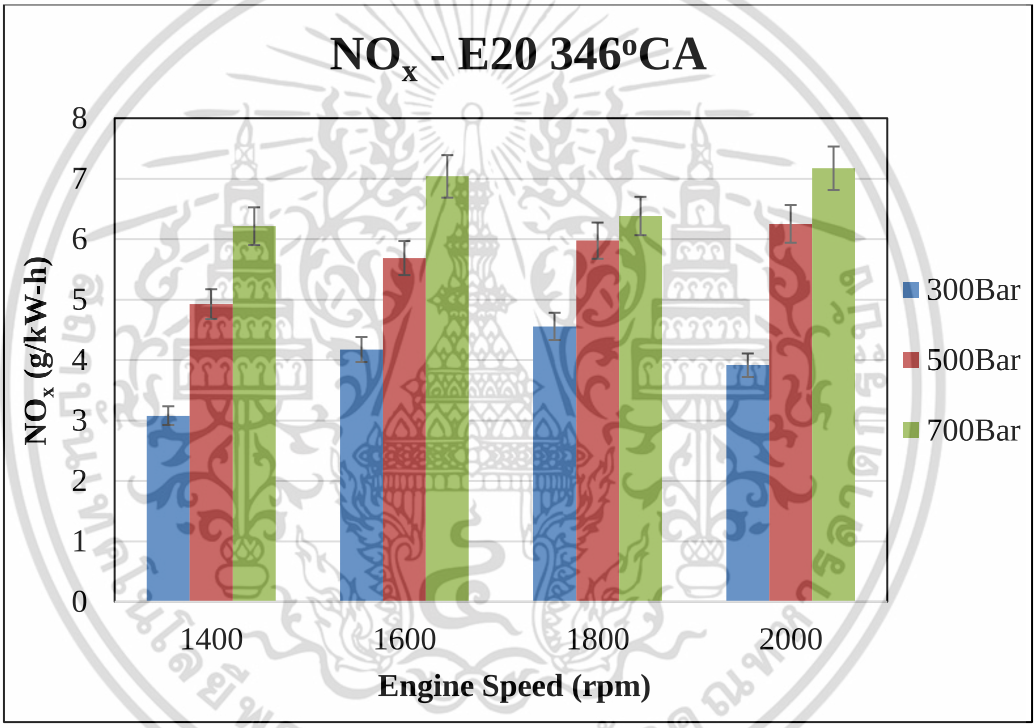


Figure 4.12 NO_x emission from Injection Pressure variation

- **THC emission**

THC emissions are a result of incomplete combustion of the hydrocarbon fuel. And will increase because of poor fuel distribution when the injection pressure was low, large amounts of excess air in some combustion zone, and low combustion temperature which depend on the injection pressure, lean fuel–air mixture regions may survive to escape into the exhaust at low injection pressure condition. During the combustion delay period the fuel-air mixture is too lean, and thereby, outside the flammability boundaries of the fuel. The second reason is at late of the combustion process, fuel leaves the injector with a low velocity. This results in under mixing of the fuel, if there is a droplet of the fuel dribble from the nozzle after the combustion process, this amount of fuel will not combust, but will evaporate during the end of the combustion. This effect called sec volume. Which will cause the increase of hydrocarbon emission. For the diesel direct injection engine, high injection pressure, high turbulence and swirl and precise injection timing are needed, consequently resulting in unburned hydrocarbons. (Casarrubios, June 2015) as show in Figure 4.13. High injection pressure will increase the combustion temperature which can reduce the THC emission.

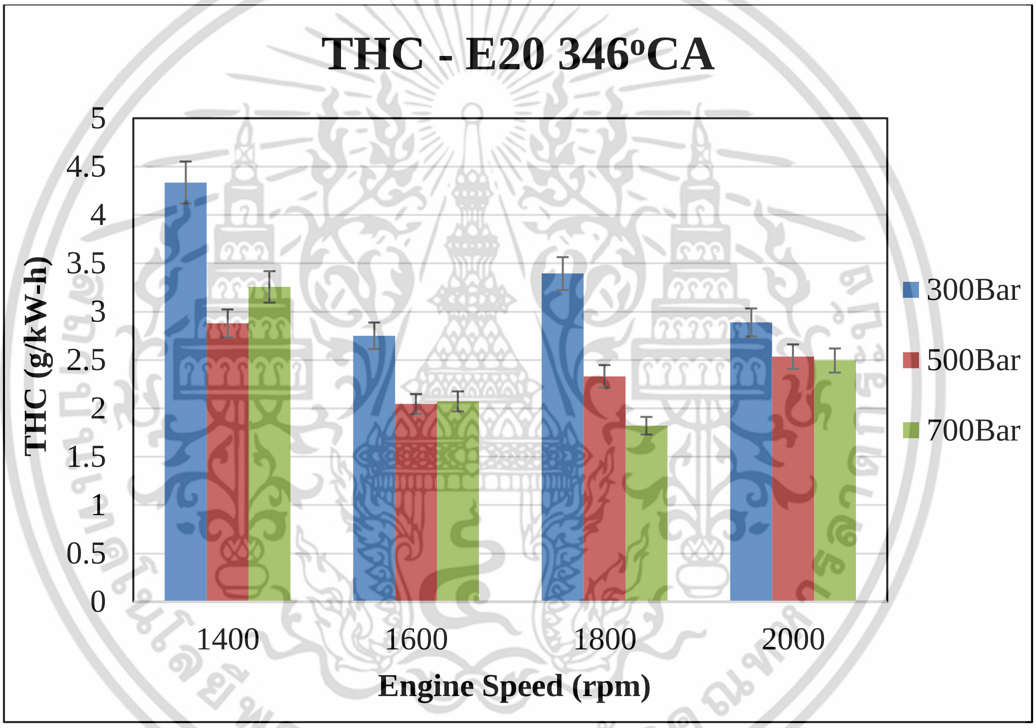


Figure 4.13 THC emission from Injection Pressure variation

4.4. Effect from Injection Timing

To explain the effect of injection timing, E20 with 500 bar injection pressure was chosen with the variation of injection timing. (334, 337, 340, 346, 350, 355 degree crank angle). Because 500 bar injection pressure give best BSEC in every engine speed without seriously knocking and E20 was the appropriate amount of the fumigation as most of the reason mentioned previous.

4.4.1. Performance

BSEC increased. For advanced injection timings, ignition delay period will be longer. The fuel will burn quickly in the combustion process, which leads to causes sudden pressure increase in the cylinder. In some case with too advance injection timing, the combustion will start before the piston reach the TDC. This will cause knocking, very high pressure will happen which lead to damage the engine and cause friction loss. As most of the fuel burns in premixed combustion phase, it causes very high peak heat release and hence power output decrease. These cause reduction of engine output power. Therefore, energy consumption per output power will increase as show in Figure 4.14. (Mohammed EL-Kasaby, 2013) Small retard injection timing decreases the BSEC due to increasing in the engine volumetric efficiency may be due to a lesser disturbance to the intake flow. (A. Rashid A. Aziz, 2010) Small retard injection timing gives the best result. The reason is the combustion process can be take place at the top dead center which will generate the best performance.

On the other hand, high retarding injection timing or late injection means later combustion, and therefore pressure rose only when the cylinder volume was expanding rapidly after the top dead center. Retard injection timing increases the BSEC because all of the fuel cannot burn in the proper time. Some of them may burn after the power stroke which will result in much higher BSEC. (Jindal, 2011)

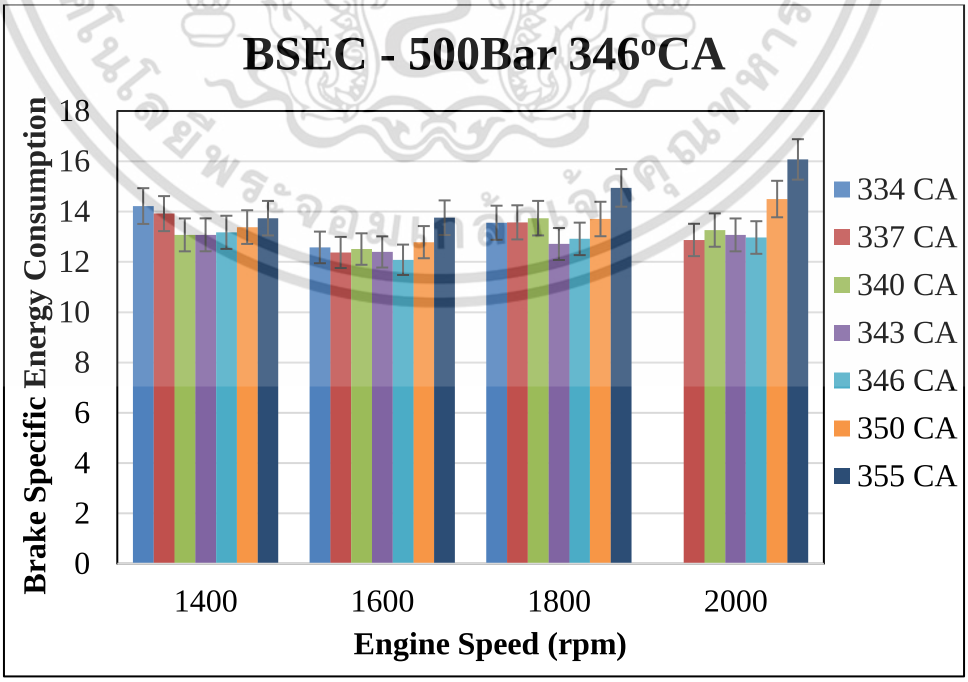


Figure 4.14 BSEC from Injection Timing variation

4.4.2. Emission

- **Soot emission**

Advance injection timing tends to decrease soot emission as show in Figure 4.15. The time available for mixing of fuel droplets with air increased with advanced fuel injection, fuel burns easily due to high cylinder temperatures and pressures prevailing at the time of advanced fuel injection timing more homogeneous mixture was found (Michael Fiebig*, 2014). For retard injection timing, there is only a short period of time before the combustion start. This will be difficult for fuel and air mixing. This result in higher soot emission. Increasing engine speed will promote the air fuel mixture which will lower soot emission.

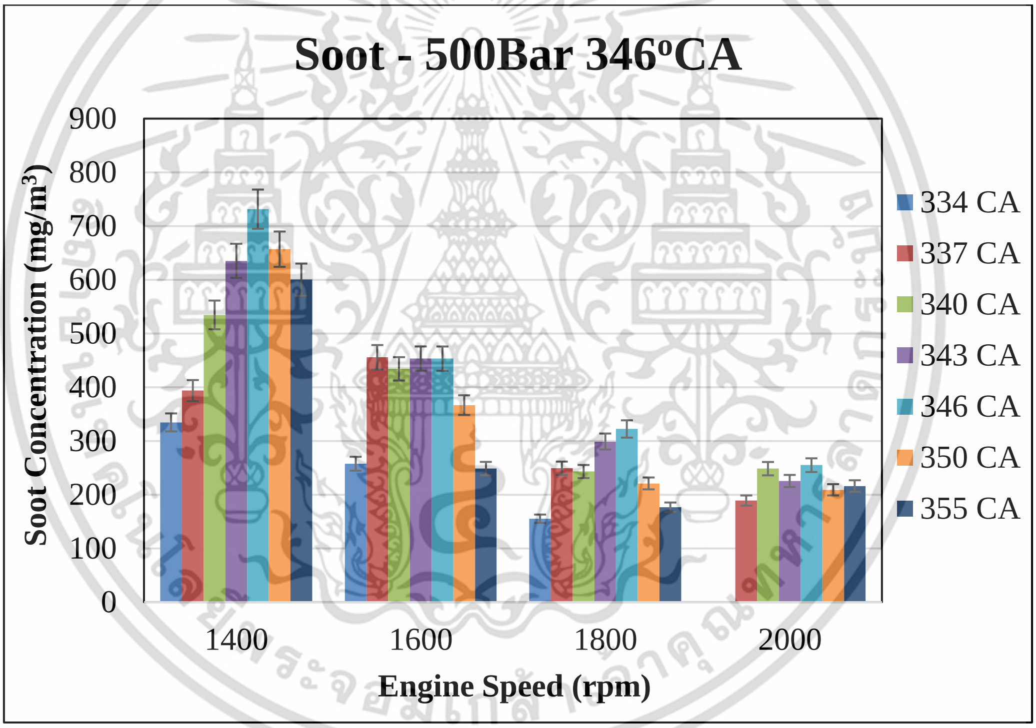


Figure 4.15 Soot emission from Injection Timing variation

- **CO emission**

Advancing the injection timing will longer the period before the combustion start. This reason can give the ability of mixing between fuel and air. High cylinder temperature will be happened due to advancing the injection timing. More fuel was burn for advance injection timing, because of the injection will be end before the combustion happen. Even when the intake air-fuel mixture is stoichiometric or lean, some CO will generate in the engine. Poor mixing, local rich regions, and incomplete combustion will create some CO. High engine speed will boot up the turbulence in the combustion chamber which leads to decrease the CO emission as show in Figure 4.16. Advance injection timing will lower the combustion duration which the formation of CO_2 cannot take place. So that amount of emission will show as CO. Combustion of rich mixture may another cause of CO emission, for advance injection timing most of the fuel get burn, which that fuel mixture is rich. And for this advance injection timing, the combustion duration is much shorter compare to retard injection timing, So the chemical reaction time will be short, which may not enough to form the CO_2. (Sayin & Canakci, 2009)

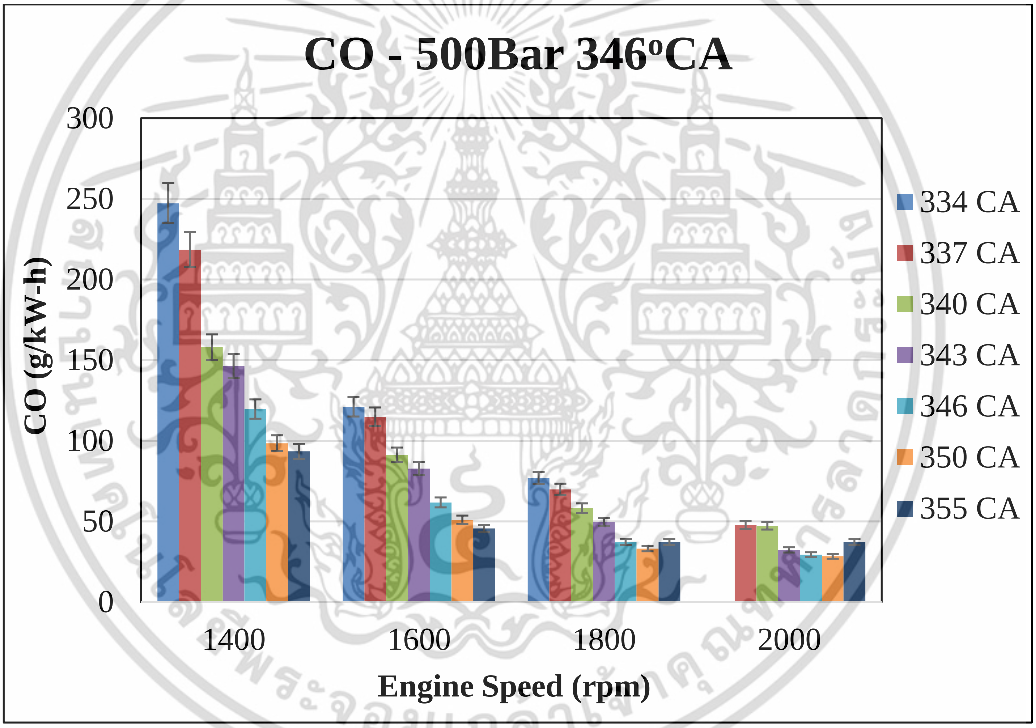


Figure 4.16 CO emission from Injection Timing variation

- **NO_x emission**

Retard injection timing gives lowest NO_x emission. The formation of NO_x is dependent on in-cylinder temperatures, the oxygen concentration and residence time for the reaction to take place is short. Retarding the injection timing decreases the peak cylinder pressure because more of fuel burns after TDC. Lower peak cylinder pressures result in lower peak temperatures. As a consequence, the NO_x concentration starts to diminish. Advance injection timing tends to gives higher NO_x emission due to higher ignition delay and more time available for NO_x to take place as show in Figure 4.17. Because of the high combustion temperature, more fuel burn in this case. Advanced injection timing, the cylinder temperature will be increased and more NO_x will be created. (Sayin & Canakci, 2009)

In addition to temperature, the formation of NO_x depends on pressure which depend in the combustion pressure , air-fuel ratio, and combustion time within the cylinder, chemical reactions not being instantaneous. The amount of NO_x generated also depends on the location within the combustion chamber. The highest concentration is formed around the highest temperatures occurred. (Agency Clean Air Technology Center, 1999)

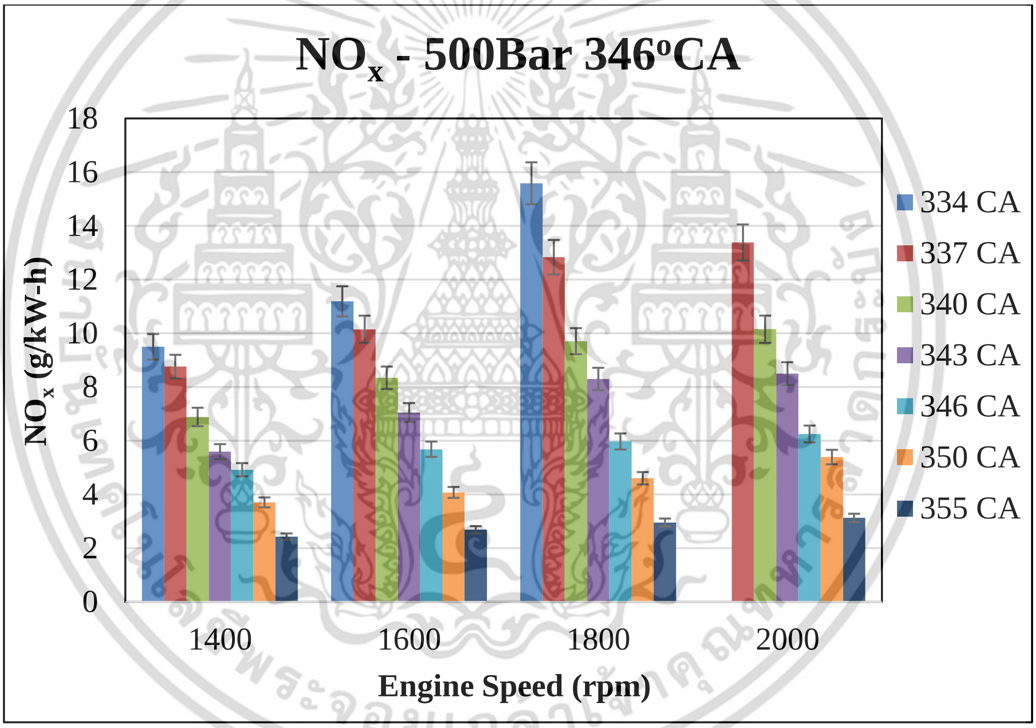


Figure 4.17 NO_x emission from Injection Timing variation

- **THC emission**

There are several causes of THC emission. With undermixing like retard injection timing, some fuel particles in fuel-rich zones never find oxygen to react with. Flame quenching at the walls leaves a small volume of unreacted air-and-fuel mixture. Some of this mixture, near the wall that does not originally get burned, will burn later in the combustion process. In fuel-lean zones, combustion is limited and some fuel does not get burned. Another cause of flame quenching is the expansion which occurs during combustion and power stroke. As the piston moves away from TDC, expansion of the gases lower both temperature and pressure within the cylinder. This slows combustion and finally quenches the flame somewhere late in the power stroke. This leaves some fuel particles unreacted. With overmixing for advance injection timing, some fuel particles will be mixed with already burned gas and will not combust totally so the THC increase as show in Figure 4.18. (Pulkrabek, 1997)

When the pressure rises up very high during the combustion stroke. Few amounts of fuel in the combustion chamber will be forced under very high pressure due to advance injection timing and will be leaked among the small space. Even between the piston ring and the cylinder wall, and etc. Later in the cycle during the expansion stroke, pressure in the combustion chamber will reduced below the crevice volume pressure, and reverse blowby occurred. The fuel and air will flow back into the combustion chamber. During this period, flame reaction has been quenched, and unreacted fuel particles and remain in the exhaust. This reason and cause the increase of emission. Leak past the exhaust valve also one of the examples. As pressure increases during compression and combustion, some air-fuel is forced into the crevice volume around the edges of the exhaust valve and between the valve and valve seat. A small amount even leaks past the valve into the exhaust manifold. When the exhaust valve opens, the air-fuel which is still in this crevice volume gets carried into the exhaust manifold, and there is a momentary peak in HC concentration at the start of blowdown. (Ganesan, 2007)

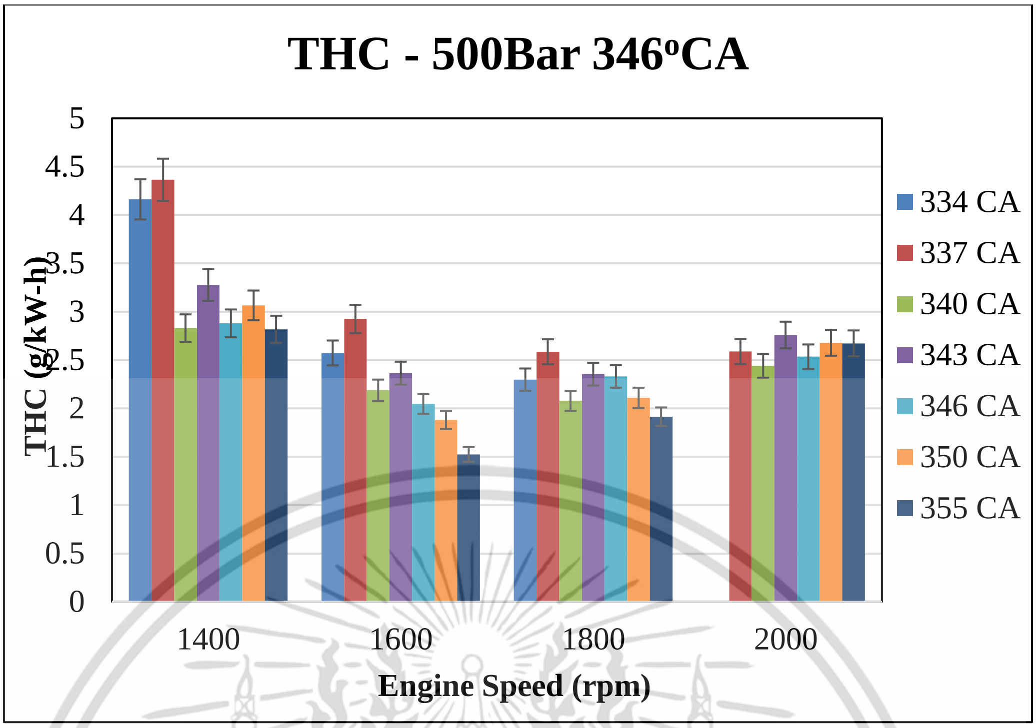


Figure 4.18 THC emission from Injection Timing variation

4.5. Combustion Characteristic

Cylinder pressure data analysis is the most effective tool to analyze engine combustion behavior because cylinder pressure history directly influences power output, combustion characteristics and engine-out emissions. The combustion process can be classified by the following steps.

Ignition delay, fuel is injected directly into the combustion chamber towards the end of the compression stroke. The period between the start of fuel injection into the combustion chamber and the start of combustion. The liquid fuel atomizes into small droplets and penetrates into the combustion chamber. The fuel vaporizes and mixes with the high-temperature and high-pressure air. Premixed combustion phase, combustion of the fuel which has mixed with the air to within the flammability limits (air at high-temperature and high-pressure) during the ignition delay period occurs rapidly in a few crank angles (high heat release characteristics in this phase). If the amount of fuel collected in the combustion chamber during the ignition delay is much high heat release rate results in a rapid pressure rise which causes the diesel knock. For fuels with low cetane number, with long ignition delay, ignition occurs late in the expansion stroke - incomplete combustion, reduced power output, poor fuel conversion efficiency.

Mixing controlled combustion phase, Once the fuel and air which is pre-mixed during the ignition delay is consumed, the burning rate (heat release rate) is controlled by the rate at which mixture becomes available for burning. The rate of burning in this phase is mainly controlled by the mixing process of fuel vapor and air. Liquid fuel atomization, vaporization, pre-flame chemical reactions also effect the rate of heat release. Heat release rate sometimes reaches a second peak (which is lower in magnitude) and then decreases as the phase progresses. Late combustion phase, heat release may proceed at a low rate into the expansion stroke (no additional fuel injected during this phase). A small fraction of the fuel may not yet burn, a fraction of the energy is present in soot and fuel-rich combustion products and can be released. The cylinder charge is nonuniform and mixing during this phase promotes more complete combustion and less dissociated product gases. Kinetics is slower as show in Figure 4.19

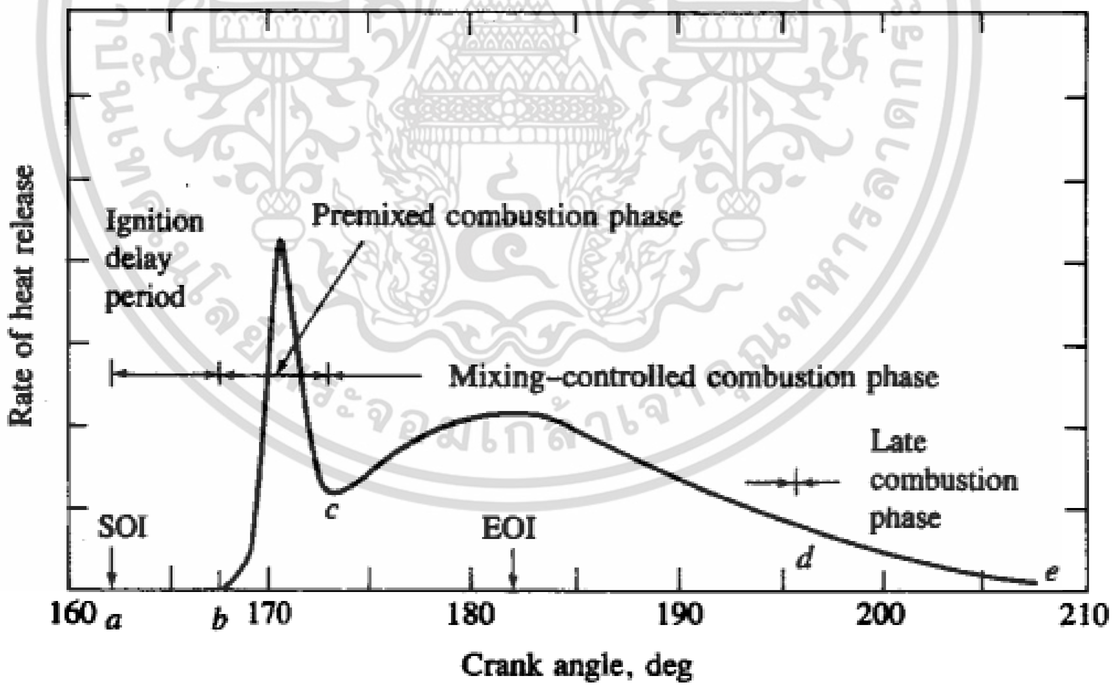


Figure 4.19 Compression-ignition diesel combustion process

4.5.1. Effect from Ethanol

Ethanol is an oxygenated fuel with lower surface tension and boiling point, so the fast vaporization of ethanol can promote the spray performance and the formation of mixture gas. And due to the cetane number is lower than diesel fuel. Longer ignition delay period as show in Figure 4.20. The figure will show the ignition delay for many conditions which can divide into 3 group. The first group is 4 columns on the left side which will show the ignition delay of ethanol fumigation amount variation from E0 to E30 with 500 bar injection pressure 1400 rpm and the injection timing is 346 degree crank angle. The second group is the 3-middle column. These 3 columns will show the ignition delay of injection pressure variation of 300, 500, and 700 bar with E20, 1400rpm and injection timing is still 346 degree crank angle same as the first group. The last 7 columns on the right side will demonstrate the ignition delay of injection timing variation (334, 337, 340, 343, 346, 350 and 355 degree crank angle) with 500 bar injection pressure and fumigation amount is E20.

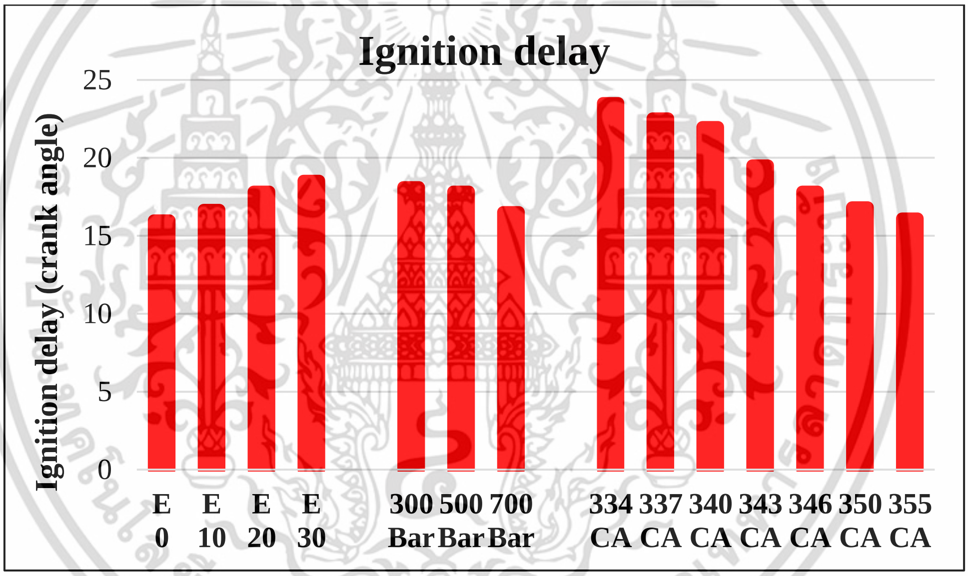


Figure 4.20 Ignition delay at various condition

It is believed that higher fumigated mixtures of alcohol will enhance the pre-combustion reactions, enabling the combustion process to complete more thoroughly, thus achieving 'higher efficiencies. From the ignition delay chart. The result show that E0 will give lowest ignition delay. And when increase the ethanol fumigation amount up to E30, the ignition delay will increase respectively. This longer ignition delay can explain by the lower cetane number of ethanol as mention before. During the ignition delay period. More fuel was store inside the combustion chamber and this reason will promote the fuel utilization before the combustion start due to longer ignition delay. The combustion process will give higher pressure and temperature. Peak cylinder pressure is respectively increase from E0 to E30 as show in Figure 4.21, The peak of the heat release rate during the premix phase seems to be depended on the amount of fuel that can be burned during this phase which most of them is diesel. Liquid diesel is necessary for ignition in this phase. The advantage of this concept is that it makes use of the difference in flammability of the used fuels. The second phase is characterized by the premixed combustion of the major part of the ethanol and the remaining amounts of the diesel. From the heat release rate graph. When increase the ethanol fumigation amount. The injection duration of ethanol will be longer. So, the heat release rate in this phase will still high or will be increase again due to the high remain amount of ethanol and the whole charged appeared in the combustion chamber. When fumigation level is higher. This contributes to a lower temperature at the end of the compression stroke and the subsequent longer ignition delay causes noisy combustion and knocking. The combustion tends to ends faster when ethanol fumigation amount was increase because most of the fuel has burn in premix phase due to the longer ignition delay. So, the peak heat release rate was increased as show in Figure 4.22 (FANG, JUNE, 2016)

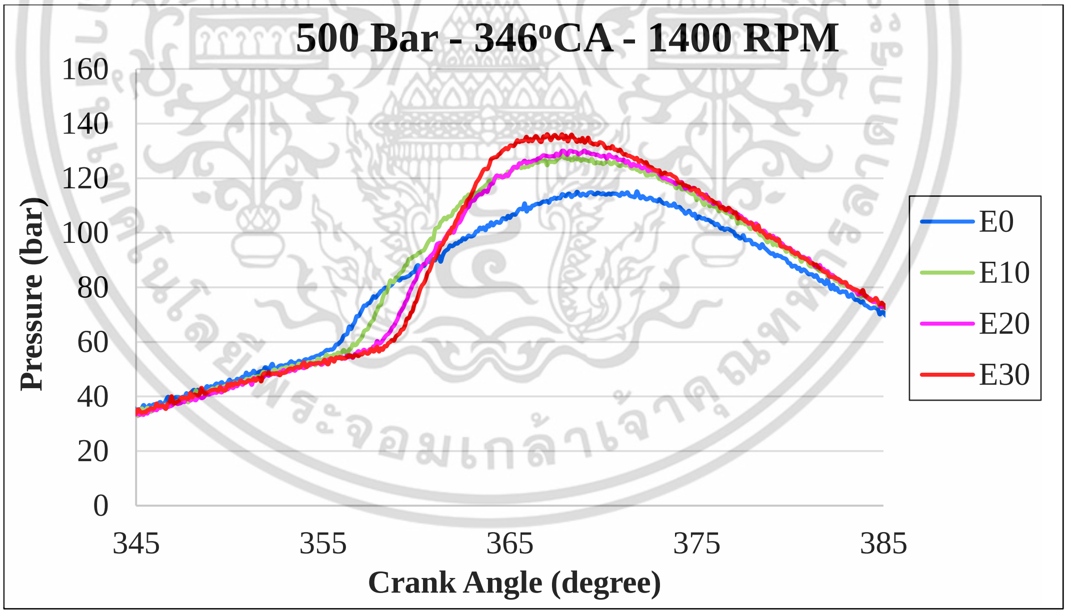


Figure 4.21 Pressure from Ethanol fumigation amount variation

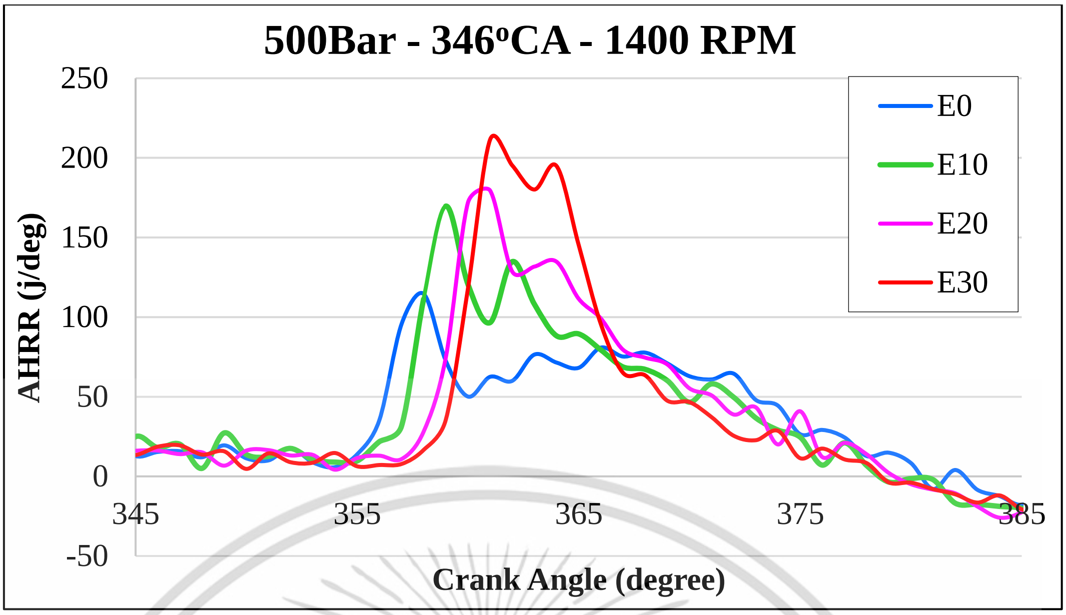


Figure 4.22 Apparent heat release rate from Ethanol fumigation amount variation

4.5.2. Effect from Injection Pressure

High injection pressure will decrease the ignition delay period as show in previous figure. Finer droplet size of fuel due to higher injection pressure and more atomization of the fuel promote the formation of mixing of fuel to air and makes itself become readily for ignite. So, it need less time to present self-ignition. When the combustion starts. Higher fuel injection pressure (700 bars) gave extremely high cylinder pressure in premix combustion phase as show in Figure 4.23. This happens due most of the fuel was injected at the beginning. The high amount of fuel mixture formed inside the combustion chamber at the start of injection, so when the combustion starts. The fuel burns more rapidly in early stages of combustion (premixed combustion) which will give faster and very high apparent heat release rate compare to lower injection pressure as show in Figure 4.24

For lower injection pressure (500 and 300 bar). The ignition delay period was increased. The combustion will start slower respectively. And the also give lower peak combustion pressure and apparent heat release rate. For 500 bar injection pressure, the apparent heat releasee rate curve shows higher and longer diffusion phase than 700 bar. This due to the combustion of remaining amount ethanol and diesel fuel. 500 bar seems to give more homogeneous charge. The appropriate injection duration and pressure may cause diesel and ethanol mixture become more utilized compare to 300 and 700 bar injection pressure. The apparent heat release rate curve show that 300 bar injection pressure will cause the second peak of apparent heat release. This due to the poor mixing ability of low injection pressure cause non-homogeneous charge. And due to longer injection duration. The combustion still occurs to burn the remain fuel. At the late combustion phase, 700 bar injection pressure will end first. Then followed by 500 and 300 bar respectively. It is mainly depending on the remaining fuel amount from the previous combustion phase. High fuel injection pressures will smaller droplet size distribution and larger jet penetration of the fuel spray, which enhanced mixing and reduced ignition delay period but got into knocking regime. (Agarwal, et al., 2013)

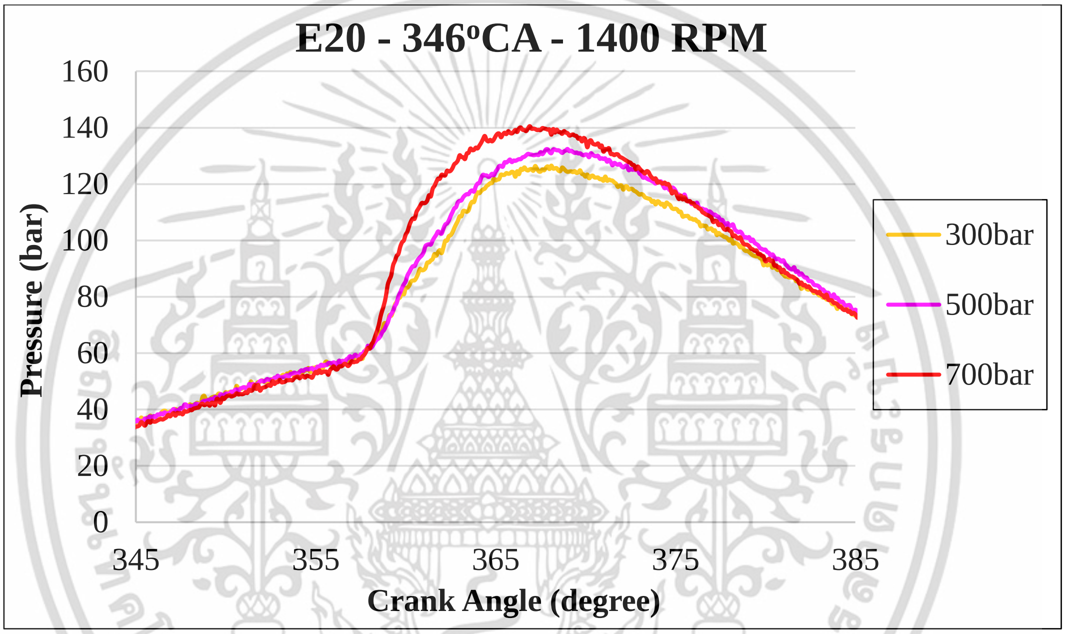


Figure 4.23 Pressure from Injection Pressure variation

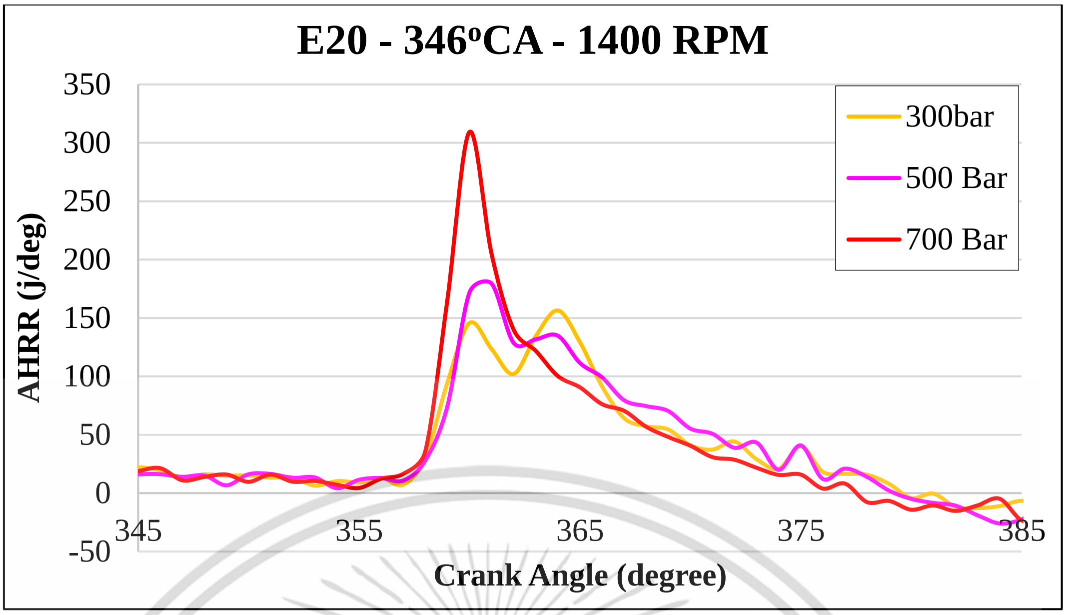


Figure 4.24 Apparent heat release rate from Injection Pressure variation

4.5.3. Effect from Injection timing

For advanced injection timing, injection delay was high as show previously, which promotes premixed combustion and higher maximum pressure when compared to later injection timing conditions. These cause high peak pressure as show in Figure 4.25. With advancing injection timing. The pressure was rapidly increase before the piston move to top dead center. And due to the combustion start before the TDC, engine power will be decrease. Hence combustion duration was shorter. This tendency decreased with retarding injection timing conditions. Therefore, fuel consumption per output power will increase. Knocking tendency increased with advanced injections due to availability of more fuel quantity in early stages of combustion, which promoted erratic combustion.

Advanced injection timings cause a large amount of evaporated fuel accumulates in the combustion chamber during the ignition delay period, and burns quickly, which leads to rapid apparent heat release rate and causes sudden increase in temperature and pressure in the cylinder. As most of the fuel burns in premixed combustion phase, it causes very high peak apparent heat release rate. For diffusion phase, most of the fuel has burn in the premixed phase, the apparent heat release rate in this process will be smaller as the advance injection timing decrease. as show in Figure 4.26

Advancing the injection timing forms a richer injected mixture this improves the pre-combustion reaction and achieves a higher efficiency. The reason is that, the higher diesel quantity supplies a larger ignition source and greater ignition power which increases the apparent heat release rate and hence results in the higher peak in-cylinder pressure. But on the other hand, abrupt ignition of a richer mixture causes a higher peak pressure and a rapidly increased rate of pressure rise which puts more stress on engine components and has such a noisy combustion that it must be avoided. Retarding the injection timing will improve the noise of operation significantly. This is acceptable in exchange for smoother and quieter operating conditions.

Retarding injection timing means later combustion, and therefore pressure rose only when the cylinder volume was expanding rapidly and the results was a reduced effective pressure to do work. End of the combustion will give the similar character to the diffusion phase. The combustion will end faster compared with retard injection timing. With retarded injection, it is clearly visible in and apparent heat release rate curves where the peak of the curve shifted away from TDC in expansion stroke (Rolf D.Reitz, 2015)

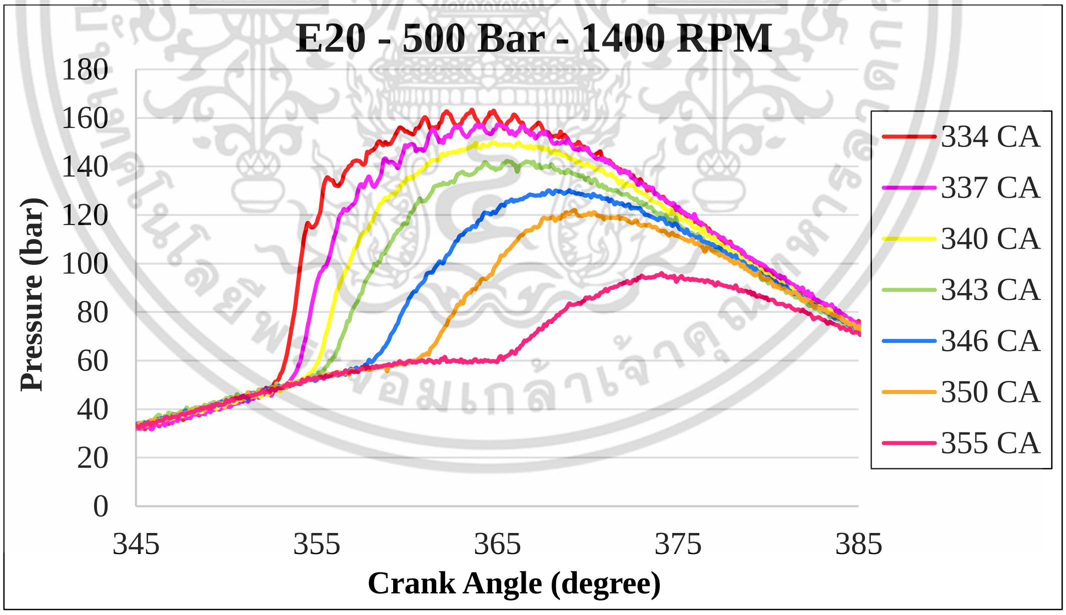


Figure 4.25 Pressure from Injection timing variation

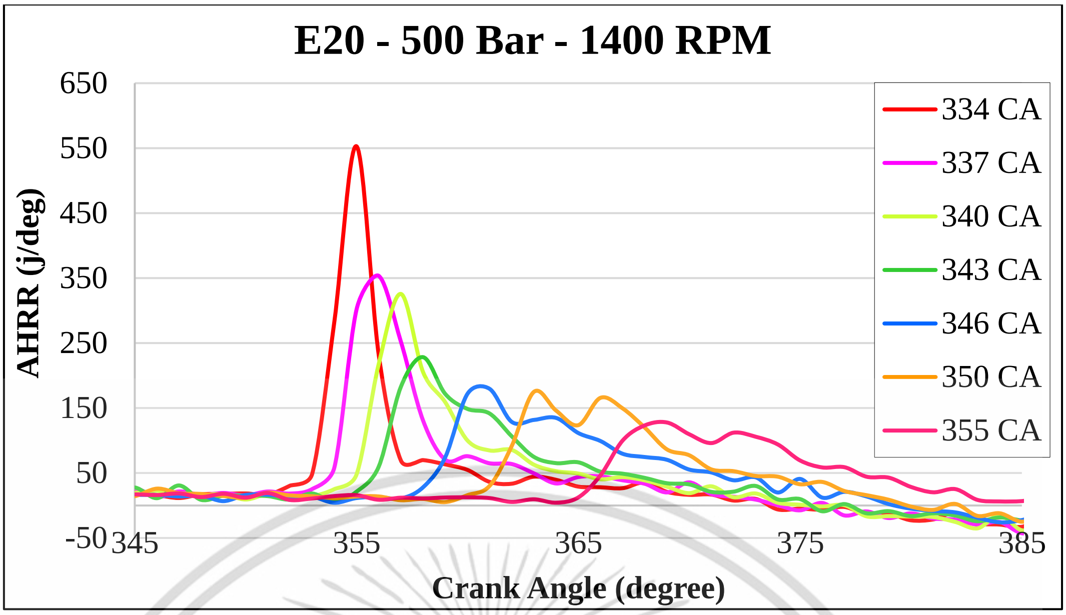


Figure 4.26 Apparent heat release rate from Injection timing variation

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

From the experiment. Engine need different injection pressure, injection timing and injection amount to get the best result for performance and emission and it also depend on engine speed. The result shows that highest torque will generated at 346^{o}CAinjection timing. Because when the piston rotates up near to the top dead center, the pressure and temperature will be very high compare to lower position before the TDC. The combustion will occur rapidly and will gives very high torque. In conclusion, retarding injection timing can increase the ethanol fumigation amount up to 80% by energy before knocking. Before the knocking start, unstable engine torque and noisy operation was found.

BSEC increase when add more ethanol fumigation because of ethanol has lower energy content itself and also the increase of ignition delay will give rapid apparent heat release during the combustion process. Too high injection pressure may cause fuel stuck to the cylinder wall and too low injection pressure causes larger diameter of the diesel fuel. These two reasons can increase the BSEC. For advanced injection timings, BSEC will become high because the ignition delay period will be longer, which will burn quickly in the combustion process, which leads to causes sudden pressure increase in the cylinder.

Soot concentration was decrease when ethanol was fumigated, because the amount of diesel injected into the combustion chamber were lessen and better air and fuel utilization. High injection pressure lead to improved fuel–air mixture formation and finer droplet size will which gives lower soot emission. Advance injection timing decreases soot emission because time available for mixing of fuel droplets with air increased.

CO increase when increase the ethanol fumigation amount because the rich mixture of the fuel in the combustion process. When fuel injection pressure increased, spray droplet diameter distribution reduces so the CO decreased. Poor mixing, local rich regions, and incomplete combustion will create some CO.

NO_x emission will be increase when increase the ethanol fumigation amount because high combustion temperature and also very high ethanol amount was injected compered by volume to achieve the total energy input that was given by diesel fuel.

High injection pressure generates high NO_x because higher combustion temperature. Advance injection timing tends to gives higher NO_x emission due to higher ignition delay cause the high combustion temperature.

THC will be produced when the combustion of rich mixture. Because there is too less of air to react with the fuel. THC increase when increase the ethanol fumigation. Unburn fumigated ethanol at the quench layer can also cause the THC. Poor fuel distribution when the injection pressure was low can cause the THC. With overmixing for advance injection timing, some fuel particles will be mixed with already burned gas and will not combust totally so the THC increase.

The peak in-cylinder pressure and the apparent heat release rate for the ethanol fumigation strategy mode was higher than that for normal diesel mode. The results showed that the peak in-cylinder pressure and maximum pressure rise rate increased with the increase of ethanol quantity. There were two peaks in the apparent heat release rate curve. The first peak which corresponded to combustion of the diesel, while the second peak which corresponded to combustion of the whole charge appeared. In general, the dual fuel mode exhibited higher pressure rise rate compared to the normal diesel engine. At high engine loads, the peak in-cylinder pressure with dual fuel mode became higher than that of normal diesel mode. It was the consequence of an improvement in the gaseous fuel combustion and hence a higher apparent heat release rate under dual fuel mode. In general, fumigation does have a positive role to play in improving engine performance. But careful control is required to avoid the extreme conditions of either misfire (power loss) at low level or knocking at high level and may give better energy consumption.

Parameters	**More fumigation**	**Higher pressure**	**Advance timing**	**Retard timing**
BSEC	Increase	Decrease	Increase	Increase
SOOT	Decrease	Decrease	Decrease	Decrease
CO	Increase	Decrease	Increase	Decrease
THC	Increase	Decrease	Increase	Decrease
NOx	Increase	Increase	Increase	Decrease

Figure 5.1 Conclusion from the test

Too high amount of fumigation can lower engine efficiency and cause high emission. Too high injection pressure will increase the energy consumption and NO_x increase. Too advance injection timing will cause knocking and high emission. Too retard injection will lower the power output from the engine. For this research engine. The appropriate engone operation condition is 20% fumigation by energy 500 bar diesel injection pressure and 346°CA injection timing

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APPENDIX A

PUBLICATIONS

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Development of a fuel injection strategy for a Diesel Engine Fumigated with Ethanol

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ABSTRACT

Ethanol is a good choice for alternative fuel which is prefer to dual fuel diesel engine. In this study, ethanol will be injected in to the intake manifold to cool down the intake temperature and reduce the amount of diesel fuel consumption. And also, use a technique called internal exhaust gas recirculation. The exhaust valve will be reopened during the intake stroke for 4mm. to vaporized the injected ethanol in the combustion chamber. The objective of this research is to study the effect of injection timing of dual fuel (diesel) on the engine performance and exhaust emissions of a supercharged, single cylinder 4-stroke direct injection compression ignition engine including ethanol fumigation and internal EGR, and also varying the injection pressure of diesel. Then using ethanol fuel as a secondary fuel to replace the energy input from diesel fuel by 10, 20, and 30%.

INTRODUCTION

Diesel engine can generate high torque even in the low engine speed operation, higher fuel economy and lower CO_2 emissions than the gasoline [1]. Reserves amount of oil which it will last for a future 40 year [2], [3]. Most of the prediction shows that oil price will be increase in the future. [4], [5] , [6]. New type of fuel which can replace the fossil fuel were found and develop in various study. Most of ethanol were produce by plant such as sugarcane or cassava which are very easy to grow in Thailand. [7] Ethanol is a very good choice for new fuel, because it was made from plant that means the reduction of carbon dioxide amount. It is also good for economic state. [8] But there is some property that ethanol cannot mix with diesel properly, so there are many technics [9]. And another main concerning problem is about the emission of diesel engine. Most of the combustion reaction in the engine are various non-ideal process, such as incomplete combustion. [10] Of all emission constituents, major concerns are NO_x, CO, HC, Smoke (Particles and soot) and in the recent past CO_2 as its accumulation add to greenhouse gases in the atmosphere. [11] Many cars company focused on understanding the nature and quantity of vehicle emissions and developing control technologies to reduce these emissions. [12] In Europe, EURO 6 was instituted in 2014 and imposed in 2015. The stricter emission regulations induce technological competition and innovation. [13]. NO_2 is not only an important air pollutant by itself, but also reacts in the atmosphere to form ozone (O_3) and acid rain. [14] And the emission of NO_x and Soot are depended on both combustion temperature and fuel mixture concentration. [15] .

Cenk Sayin [16] study, influence of injection timing, focusing on the engine performance and exhaust emissions. Increasing the amount of ethanol in the fuel mixture produced higher peak temperature in the cylinder. This effect increased NO_x emissions and leads to increase the BSFC and decrease BTE. Advancing the injection timing, CO and unburned HC emissions decreased while NO_x and CO_2 emissions increased. Retarding the injection timing presented the minimum results of NO_x and CO_2 emissions. The original injection timing gave the best results of BSFC and BTE. The result is quite similar to M. Pandian [17] research which is about influence of injection timing of compression ignition direct injection engine using bio-diesel blend B40. BSEC and BTE increased on advancing the injection timing while reduced on retarding.

Jun Cong Ge [18] study effects of different pilot injection timings from before top dead center (bTDC) and exhaust gas recirculation (EGR) fueled with canola oil biodiesel-diesel (BD) blend. CO and PM emissions clearly decreased, but NO_x emissions increased slightly. With an increasing EGR rate, the combustion pressure and IMEP decreased slightly. However, the P_{max} showed a remarkable decrease. The BSFC and PM emissions increased slightly, but the NO_x emission decreased considerably, the combustion pressure and RoHR decreased slightly. On the effects on engine performance, as the pilot injection timing was advanced, the P_{max}, BSEC, IMEP and BSFC changed slightly. On the effects on exhaust emissions: as the pilot injection timing was advanced, CO and PM emissions decreased considerably. As the EGR rate was increased, NO_x emissions decreased considerably, and CO, CO_2, and PM emissions increased. Effect of fuel injection timing and pressure on combustion, emissions and performance characteristics of a single cylinder diesel engine was also study. [19] Knocking was observed. Advanced injection timings led to rapid combustion hence, higher RoHR was observed in early stages of combustion. Engine performance was superior at low FIP leading to lower BSFC and higher BTE at all engine loads. Lower mass emission of CO_2, CO, HC and NO_x was observed at lower FIP. Particulate number concentration in a CI engine increased with increasing engine load. Rosli Abu Bakar [20] do an experiment in both of variation engine speeds - fixed load and fixed engine speed – variation loads by changing the fuel injection pressure from 180 to 220 bar. According to the results, the best performance of the pressure injection has been obtained at 220 bar, specific fuel consumption has been obtained at 200 bar for fixed load – variation speeds and at 180 bar for variation loads – fixed speed. S. Imtenan [21] is review about Impact of low temperature combustion (LTC) attaining strategies on diesel engine emissions for diesel and biodiesels. LTC strategies decrease NO_x and PM simultaneously but increase HC and CO emissions. Recent attempts to attain LTC by biodiesel have created a hope for reduced HC and CO emissions. Decreased performance issue during LTC is also being taken care of by latest ideas.

M. Abu-Qudais [22] study the effects of ethanol fumigation and ethanol diesel fuel blends on the performance and emissions of a single cylinder direct injection, variable compression ratio, diesel engine with a swept volume of 582 cm^3. The results show that both the fumigation and blends methods have the same behavior in affecting performance and emissions, but the improvement in using the fumigation method was better than when using blends. The optimum percentage for ethanol fumigation is 20%. This percentage produces an increase of 7.5% in brake thermal efficiency, 55% in CO emissions, 36% in HC emissions and reduction of 51% in soot mass concentration. This fumigation percentage produces a decrease of 48% in engine smoke and 51% in soot mass concentration. The optimum percentage for ethanol diesel fuel blends is 15%. This produces an increase of 3.6% in brake thermal efficiency, 43.3% in CO emissions, 34% in HC, it can also produce a reduction of 33.3% in engine smoke and 32.5% in the soot mass concentration. Which gives resemble result to Experimental studies on fumigation of ethanol in a small capacity Diesel engine [23] The reductions are mainly associated with the reduction of diesel fuel burned in the diffusion mode. The reductions have been found between a wider range of 14–57% at over all engine load conditions. [24], [25] conclude that Ethanol may be a good additional fuel to CI engines when being injected into inlet port in a proper proportion to diesel fuel.

High efficiency enabled by hydrous ethanol use in dual-fuel engines are studied. [26] [27] The purpose of this project is to use hydrous ethanol to demonstrate high efficiency with reduced emissions in a modified diesel engine where ethanol provides up to 80% of the fuel energy input. The internal EGR is used at the gasoline HCCI engines to control the start of the combustion. [28] Hot exhaust gases are re-admitted into the cylinder, increasing the mixture temperature, and then allowing activation of the oxidation catalyst in the posttreatment system. Exhaust valve reopened, consisting of burned gases and unburned air from the previous combustion. Fresh mixture is aspirated into the cylinder through the intake valve. [29]

PROCEDURE AND METHODS

Experimental Setup

The engine use in this study is four strokes, water cooled single cylinder, common-rail diesel direct injection engine. The engine has two intake valves and two exhaust valves. One of the exhaust valve is replaced with optical access window. Dynamometer is use to motoring and controlling the engine load and speed. Load cell was equipped to measure the torque. Figure 1 shows engine overview and operation system. Figure 2 shows the picture of tested engine. Table 1 shows the engine specifications. Hydraulic variable valve actuator (VVA) system was used to control the valve lifting condition. The internal EGR system was conducted by reopening the exhaust valve during the intake stroke using VVA. Supercharger which is driven by electrical motor is to control the intake pressure. Port fuel injector (PFI) was installed to inject ethanol to the intake manifold. Ethanol was introduced to the injector by pumping up to 2 MPa. The characteristic of valve lift will be shown in Fig.3.

To measure the pressure, Kistler 6123 type, the piezoelectric pressure transducer was use. Strain gage pressure transducer was used to measure the intake pressure (Kyowa). Water cooled strain gage pressure sensor (Kyowa) was used to measure the exhaust pressure. To monitor the temperature of the intake air, exhaust, lubricant oil and cooling water. The K-type thermocouples were installed. Heaters and thermal controller were used with thermocouples to control the intake air, cooling water and lubricant oil temperature. The measurement of diesel fuel flow rate was carried out using gravity flow meter of (Ono Sokki) type. The air mass flow rate was measured by the laminar air flow meter (Sokken).

Electronically controlled pressure regulating valve is mounted on the intake manifold to control the desirable boost pressure. Similarly, exhaust pressure regulating valve is installed to allow boosting of exhaust pressure and controlling the internal EGR ratio as well. A rotary encoder (Nikon) is connected to the crankcase camshaft and produces 5000 pulses per revolution. Also, a photo sensor is attached to the flange of the crankcase camshaft to produce one pulse per revolution. Pulses from rotary encoder and the photo sensor are necessary for a sampling of cylinder pressure and for controlling injection timing and valve timing. Some software was used to control the system and data acquisition depending on the signal from rotary encoder and photo sensor. The exhaust gases were sampled just after the exhaust manifold for the measurement of NO_x and soot concentrations. A smoke meter was used for soot concentration measurement. The soot sampling line is heated up to 200°C to prevent soot condensation. The NO_x sensor (Horiba, MEXA-720) was used for NO_x measurement from diesel.

Table 1. Engine Specification

Engine Type	4-Stroke single cylinder water-cooled direct injection diesel
Displaced volume	622 cm^3
Stroke	100 mm
Bore	89 mm
Compression ratio	14.3:1
Combustion Chamber	Reentrant type
Injection System	Common rail injection system
Injection Nozzle	8 holes, ϕ = 0.158 mm
Intake System	Supercharged
Valve Train	Hydraulic variable valves (Two intake and one exhaust)

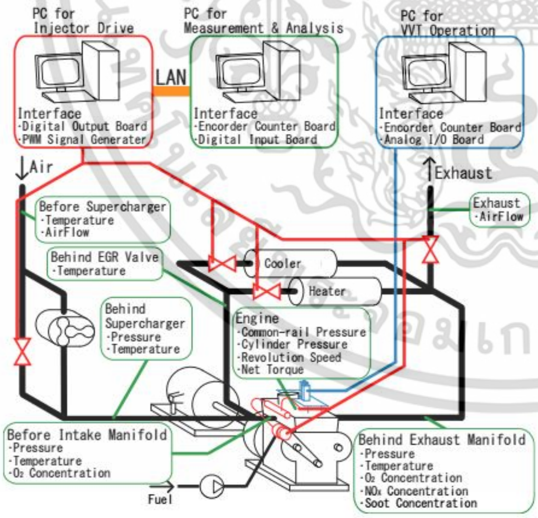

Figure 1. Engine overview and operation system

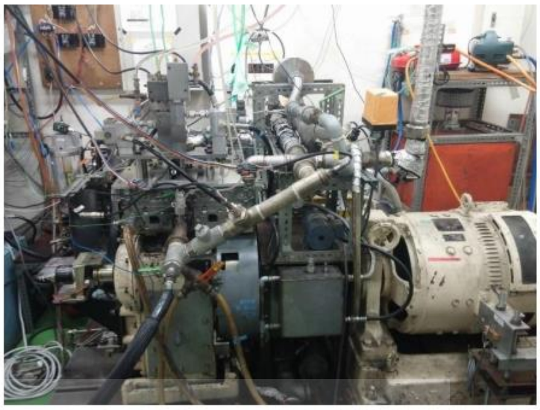

Figure 2. Tested engine

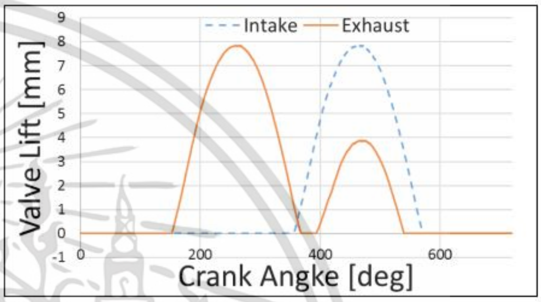

Figure 3. Valve lift curve

Analysis Method

To measure the output power, IMEP shows the engine capacity to generate work which is relate to engine displacement. It is calculated by the integration of the cylinder pressure and volume in the expansion stroke.

$$IMEP = \frac{\int pdV}{V_{displacement}} \tag{1}$$

p is the cylinder pressure

V is the cylinder volume

$V_{displacement}$ is the cylinder displacement volume

To analyze the cylinder pressure, the RoHR was used and calculated from the measured cylinder pressure and cylinder volume as the following equation

$$dQ_{net} = \frac{1}{K-1}V\frac{\partial P}{\partial \theta} + \frac{K}{K-1}P\frac{\partial V}{\partial \theta} - \frac{PV}{(K-1)^2}\frac{\partial K}{\partial \theta} \tag{2}$$

Q_{net} is the apparent rate of heat release

K is the specific heat ratio

θ is the crank angle

The specific heat ration (K) is depend on the gas temperature, and was calculated by equation (3)

$$K = 1.386 + 1.778\times10^{-4}T - 5.293\times10^{-7}T^2 + 4.004\times10^{-10}T^3 - 9.932\times10^{-14}T^4 \quad (3)$$

T is the gas temperature

The combustion phasing is determined by CA03, CA50 and CA90. CA03 is the crank angle at which the 3% of the total heat is released as illustrated in Fig.4. The heat release amount was calculated according to equation (4). The CA03 calculation was used as an indication of the start of the combustion process and used for ignition delay estimation. Similarly, CA50 and CA90 are defined as the crank angle at which 50% and 90% of the total heat is released and CA90 indicates the end of the combustion process.

$$Q_{net} = \int_{\theta_{IVC}}^{\theta} \left(\frac{dQ_{net}}{d\theta}\right) d\theta \quad (4)$$

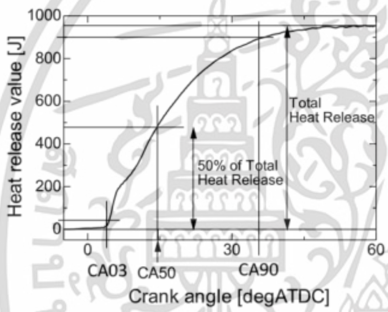

Figure 4. Definition of CA03, CA50 ad CA90

From the available literature, it can be said that the manner in which ethanol is introduced into the engine cylinder influences diesel engine combustion characteristics and effects of both NO_x and smoke formation. Although ethanol fumigation and PFI are effective ways to prevent ethanol and diesel blends separation and enables the easy change of ethanol to diesel ratio, the used injection technique in this study depends mainly on ethanol injection into the intake manifold. This injection strategy is used to vaporize the injected ethanol by utilizing the waste heat in exhaust gases. Then, by controlling the valve lift and timing, the exhaust valve is opened during the intake stroke. Part of the remaining exhaust gases also introduced to the combustion chamber as internal EGR. An electrically controlled pressure control valve is installed to allow boosting of exhaust. The ethanol injector was fixed 0.2 m approximately from the intake valve with the angle of 45°. Internal exhaust gas recirculation was used in every testing conditions.

Result and Discussion

The first part of this experiment will vary 3 different diesel injection timing and 3 different diesel injection pressure as presented in table 2. which will also include with other essential parameters like injection amount of diesel and ethanol. And the property of diesel fuel and ethanol will be shown in table 3. The heating value in the table is needed to calculate the proportion of ethanol fumigation to show the percentage of the total energy input. The exhaust valve will be reopened and lift 4mm. as shown in table 4 as an internal EGR.

Table 2. Experimental conditions

Engine Speed [rpm]	1000
Fuel injection pressure [MPa]	80,100,120
Fuel injection amount [mg/cycle]	28,32,36
Fuel injection timing [°aTDC]	-14, -10, -6
Intake air temperature [°C]	65
Cooling water temperature [°C]	85
Lubricating oil temperature [°C]	70
Intake valve lift [mm]	8
Exhaust valve lift [mm]	8
Exhaust valve reopen lift [mm]	4

Table 3. Fuel property

Fuel properties	Diesel fuel	Ethanol
Specific gravity [kg/m^3] at 15 °C	0.84	0.785
Viscosity [cP] at 20 °C and 1 atm	3.35	1.2
Molecular weight	170	46.07
Higher heating value [kJ/kg]	46100	29700
Lower heating value [kJ/kg]	43200	26900
Heat of vaporization [kJ/kg]	270	840
Cetane number	50	8

Alternative fuel such as ethanol is a good choice for this experiment. It has some advantage like it can be growth, reduce CO_2 by photosynthesis process. Lower intake air temperature and may reduce smoke when inject into the combustion process. Reduce smoke production from decrease diesel proportion of the fuel. But it also has some disadvantage such as lower energy content, not able to blend with diesel in high proportion. Low cetane number, so knocking or auto-ignition may occur easily. It also less lubricating ability compare to diesel. More alcohol fuel than diesel fuel is required by mass and volume because of its energy content.

And the second part of this project is going to fumigate ethanol into the intake manifold port. Ethanol will be injected up to 10, 20, and 30% by energy substitute the total energy input from the diesel fuel. Ethanol will be injected at 400°aTDC with the pressure of 2MPa. The exhaust valve will be reopened and lift 4mm. as shown in table 4 as an internal EGR.

Table 4. Fumigation testing condition

Diesel	100%	90%	80%	70%
Amount [mg]	40	36	32	28
Timing [°ATDC]	-14	-14	-14	-14
Pressure [MPa]	120	120	120	120
Duration [us]	1074	1036	1003	964.3
Ethanol	0%	10%	20%	30%
Amount [mg]	-	6.424	12.85	19.27
Duration [us]	-	3690	3935	4287
Timing [°ATDC]	-	400	400	400
Pressure [MPa]	-	2	2	2
Exhaust valve reopen lift [mm]	4mm	4mm	4mm	4mm

Effect of varying injection timing and pressure

This experiment was conducted by vary the injection pressure and injection timing of diesel fuel to find the condition which generate least NO_x which is the main concern emission from diesel engine to do further experiment for the ethanol fumigation. By advancing the injection timing and increase the injection pressure. The combustion pressure become higher respectively and will be illustrated as Fig.5.

Fig.6 show the ignition delay for all testing condition. When advancing the injection timing, the ignition delay was increased and this gives more time for the mixing of fuel and air. The combustion process of compression ignition needs enough pressure and temperature to ignite itself. So advance injection need more time for the self-ignition process. The ignition delay was decrease when the injection pressure was increased. This is due to higher pressure and more atomization of the fuel makes itself become readily for ignite. Formation of mixing of fuel to air becomes better. So, it need less time to present self-ignition. The CA50 in Fig 7. Gives same phenomena to the CA03. 50% of total heat release generate more rapid when advancing the injection timing and increase the injection pressure. Fig 8. Which is CA90 shows the end of combustion process. At low injection pressure (80MPa). The combustion ends faster when advancing the injection timing. This may cause by incomplete combustion from the combustion process. But at higher injection pressure (100,120MPa). The combustion ends slower when injection timing was -14°aTDC. Due to the internal EGR the temperature requires for the self-ignition become less. Then The combustion starts faster with lower temperature while better mix of the air and fuel, some part of the mixture still remain for the diffusion phase. So, the combustion ends slower.

Fig 9. Shows the output performance of the given condition as IMEP. The highest IMEP was given at lowest injection pressure and most retard injection timing. Rate of heat release from the experiment was shown in Fig 10. Highest injection pressure and most advance injection timing gives the highest peak for this RoHR which is in premixed combustion phase. But the duration for this phase in this condition is shortest compare to other given testing condition. And in mixing-controlled combustion phase, it generates very low RoHR. This cause by the readily self-ignition condition before the combustion occur. By using internal EGR with high injection pressure and advance injection timing.

Longer mixing times create the possibility for partially premixed air-fuel mixtures to combust at lean and low temperature conditions. In the low temperature combustion (LTC) concept, the majority of the injected fuel is premixed and thus is able to combust under such conditions. This is the advantage of lean low temperature combustion. As mixing time is increased, NO_x levels decrease dramatically. Like for NO_x emissions, smoke emissions are also known to benefit from locally less rich conditions. Which were illustrated as Fig 11. And Fig 12. Respectively.

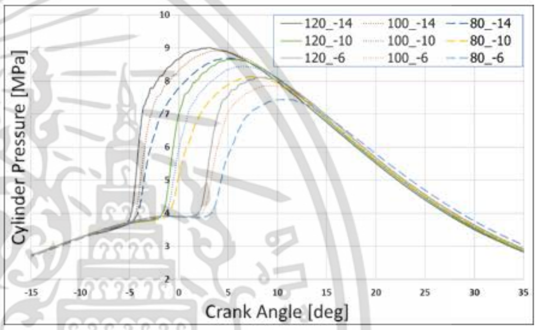

Figure 5. Cylinder pressure

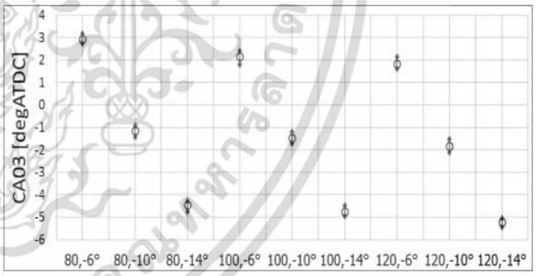

Figure 6. CA03

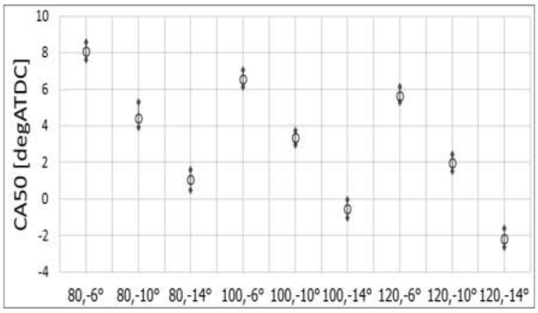

Figure 7. CA50

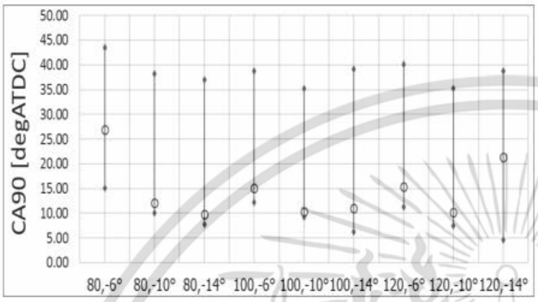

Figure 8. CA90

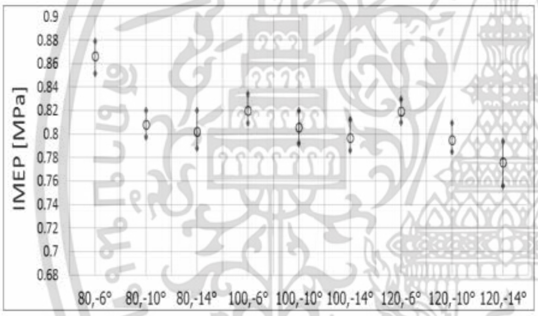

Figure 9. IMEP

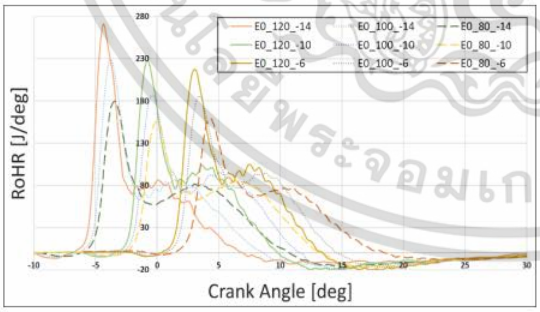

Figure 10. Rate of heat release

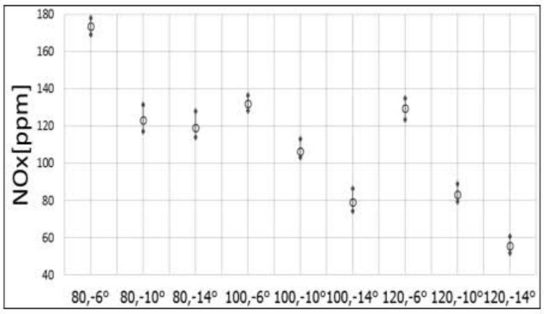

Figure 11. NO_x emission

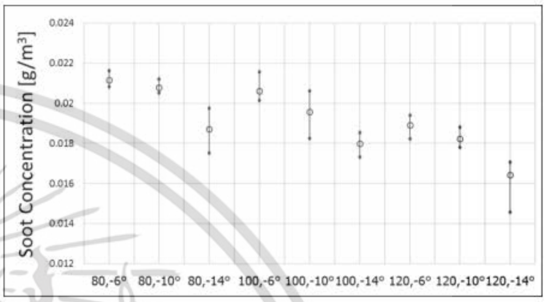

Figure 12. Smoke concentration

When advancing the injection timing, and increasing the injection pressure of diesel. And with the internal EGR, ignition delay was increased. The time consumed in premix phase become short. Because the temperature in the combustion chamber still high before the compression ignition occur. The combustion process generates high pressure and temperature in a shorter time compare to other injection timing and pressure. But for the diffusion phase. When increasing the injection pressure and advancing the injection timing. The result shows that the combustion in mixing-controlled combustion phase has lower temperature and longer time. So, it gives lower NO_x and smoke. In this test, -14°aTDC and injection pressure of 120MPa gives the best result for smoke, and NO_x reduction. So, this condition was selected to do the ethanol fumigation test.

Effect of ethanol fumigation

From the increasing of O_2 content in the combustion process, and the cetane number is lower than diesel fuel in the mixture by adding ethanol into the combustion chamber. The combustion process gives higher temperature in the cylinder as shown in Fig 13. Peak cylinder temperature is respectively high from E0 to E30, but the combustion tends to ends faster when ethanol was increase such as using E30 condition. The combustion pressure will be shown in Fig 14. And because of the O_2 concentration in the mixture is increased. The combustion become more complete. So, the peak cylinder temperature and RoHR were increased.

The combustion starts a bit slower around 0.1°CA compare with E0 and E30. The ignition delay was increased due to the low cetane number which was determine by using CA03 diagram in Fig 15. CA50 was shown in Fig 16. Which show that when ethanol was added up to 30% by energy, the 50% of total heat release become earlier compare to E10. The combustion ends much earlier when using E20 and E30 (around 20°CA) which shown in Fig 17. respectively, compare to no ethanol fumigation.

IMEP was higher when increase the proportion of ethanol as show in Fig 18. The performance of the engine becomes better due to ethanol was quickly evaporate, but no self-ignite. And will be ignited by burning diesel. That generate higher power output. Rate of heat release was shown in Fig 19. For the ignition delay phase. Adding ethanol increase the ignition delay phase. High ethanol proportion gives high peak RoHR in the premixed phase, and also increase the duration of diffusion phase. Because ethanol was injected 3 times.

NO_x emission also increased due to high combustion temperature, early or rapid combustion, and more O_2 concentration as shown in Fig 20. Smoke concentration decrease due to the low proportion of diesel injected. From the increased of ignition delay enhance the mixing of diesel fuel, ethanol and air mixture as show in Fig 21. Using ethanol increase the hydrogen content in the mixture which lead to the reduction of smoke.

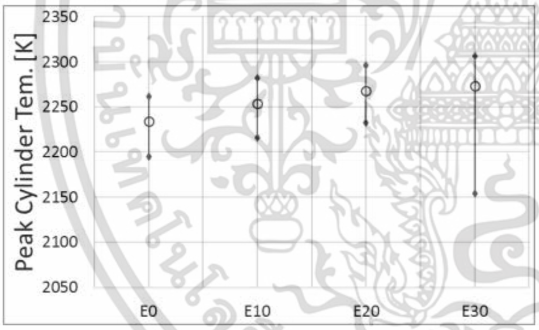

Figure 13. Peak Cylinder Temperature

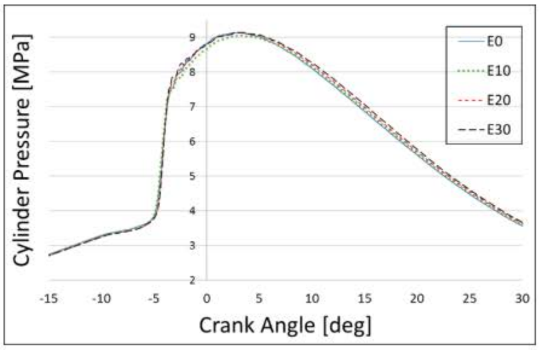

Figure 14. Cylinder pressure

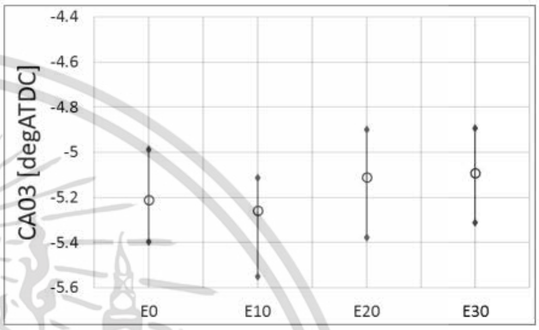

Figure 15. CA03

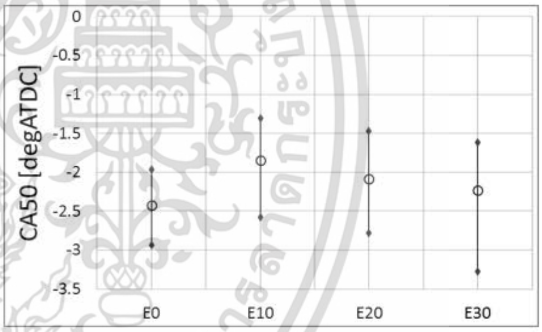

Figure 16. CA50

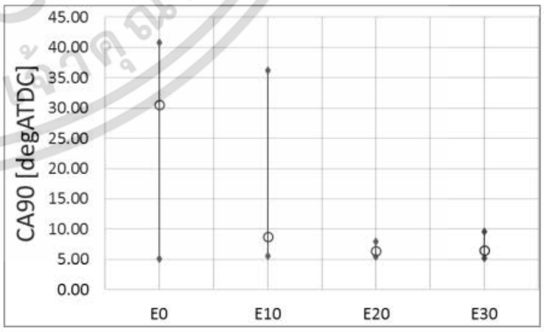

Figure 17. CA90

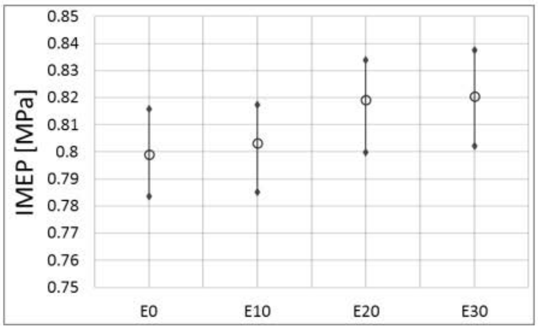

Figure 18. IMEP

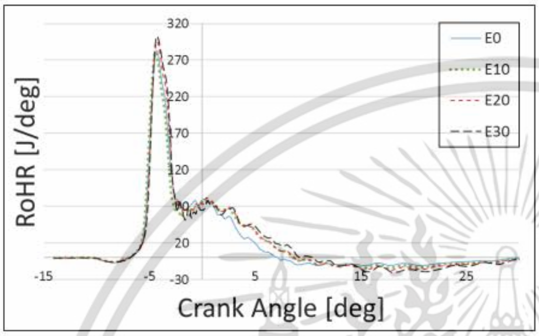

Figure 19. Rate of heat release.

Figure 20. NO_x emission

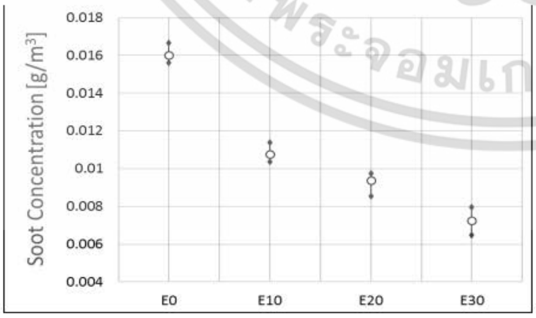

Figure 21. Smoke concentration

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Conclusion

1. Effect from injection timing - when advancing the injection timing, the ignition delay was increased. The combustion pressure and heat release rate become higher. Advancing injection timing also generate lower IMEP, NO_x and smoke.

2. Effect on injection pressure - when increase the injection pressure, the ignition delay was decreased. The combustion pressure and heat release rate become higher. and also generate low NO_x and smoke.

3. Effect on ethanol fumigation - when ethanol was added into the intake manifold at the maximum amount as E30. The highest combustion pressure was found and it is also gives highest heat release rate. Due to the proportion of diesel fuel was reduced, the smoke concentration was decreased. While NOx was increased due to the increase of oxygen concentration in the fuel. The ignition delay was increased due to the lower cetane number of ethanol.

For the further experiment. To find the appropriate parameter for various working condition of the engine. For example, different engine speed, different engine load. All parameters need to be controlled and trial for many different values. To give the best performance, emission and cover the demand from the user and manufacturer. So, mapping is very essential for applying the project to the road use condition and for the commercial section.

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ABBREVIATIONS

bTDC	before top dead center
aTDC	after top dead center
EGR	exhaust gas recirculation
BSFC	brake specific fuel consumption
BMEP	break mean effective pressure
BTE	break thermal efficiency
P_{max}	peak combustion pressure
CO	Carbon monoxide
CO_2	Carbon dioxide
HC	hydro carbon
PM	particulate matter
NO_x	nitrogen oxide
IMEP	indicated mean effective pressure
RoHR	rate of heat release
FIP	fuel injection pressure

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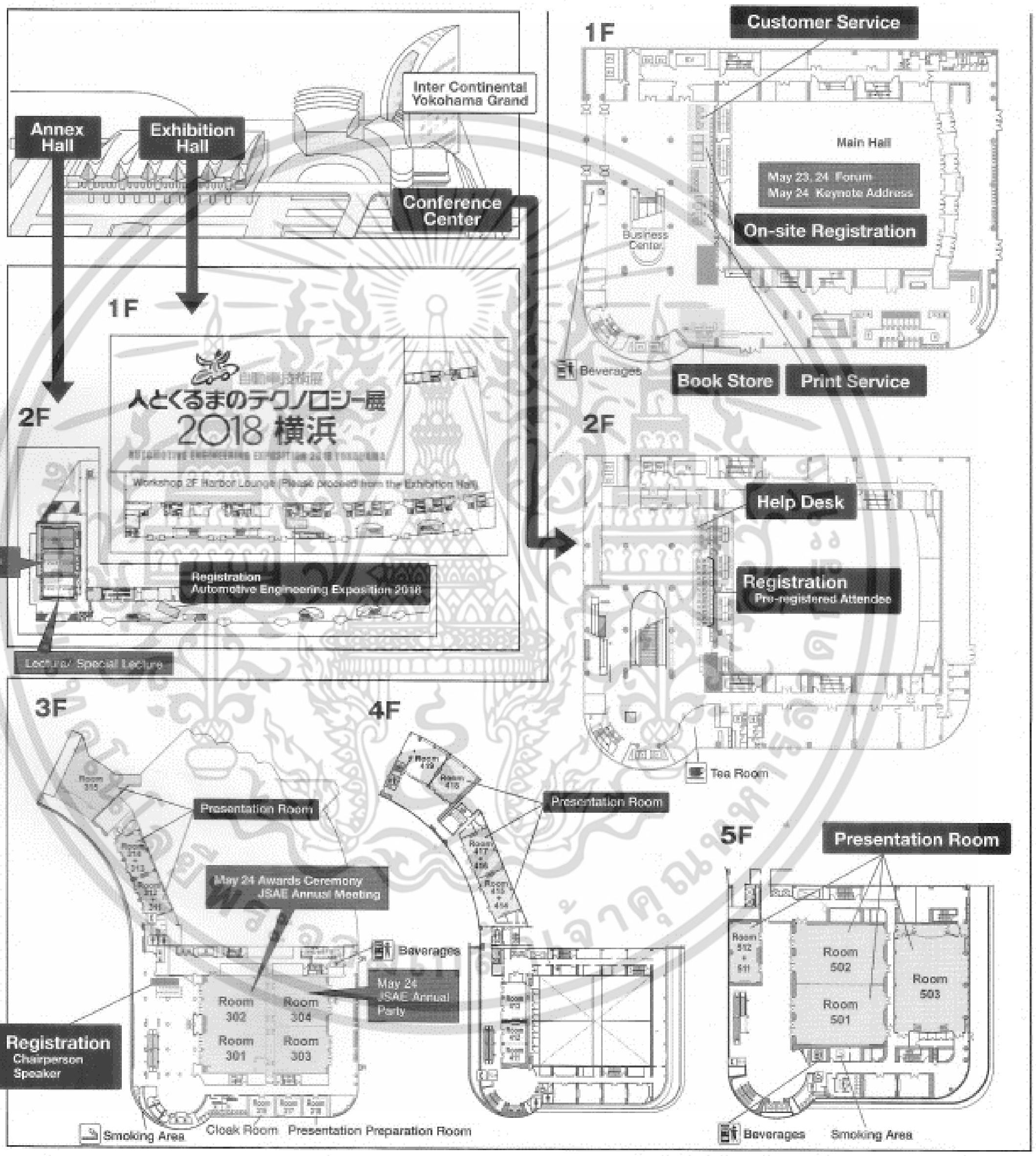

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Injection strategy of Diesel fuel with Ethanol Fumigation in Diesel engine

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ABSTRACT: Diesel engine gives high torque at low engine speed. While fossil fuel will going to be finishe in the future, and it also high pollution at the same time. To improve these problems. This experiment will use ethanol as an alternative fuel which will be inject into the intake manifold during the suction stroke. The injection timing and injection pressure of diesel fuel will be varied. Ethanol will be inject to substitute the amount of energy while reducing the injection of diesel fuel. Brake specific energy consumption, smoke, and gas emission were collected during the experiment.

KEY WORDS:**Injectino timming, Injection pressure, Dual fuel engine, Ethanol fumigation, Injection timing, NO_x**

1. INTRODUCTION

Increasing the amount of ethanol in the fuel mixture produced higher peak temperature in the cylinder. This effect increased NO_x emissions and leads to increase the BSFC and decrease BTE. Advancing the injection timing, CO and unburned HC emissions decreased while NO_x and CO_2 emissions increased. Retarding the injection timing presented the minimum results of NO_x and CO_2 emissions. The original injection timing gave the best results of BSFC and BTE. The result is quite similar [1], which is similar to [2] BSEC and BTE increased on advancing the injection timing while reduced on retarding. Knocking was observed. Advanced injection timings led to rapid combustion [3]. Engine performance was superior at low FIP leading to lower BSFC and higher BTE at all engine loads. Lower mass emission of CO_2, CO, HC and NO_x was observed at lower FIP. Particulate number concentration in a CI engine increased with increasing engine load. But [4] shows that the best performance of the pressure injection has been obtained at 220 bar, specific fuel consumption has been obtained at 200 bar for fixed load – variation speeds and at 180 bar for variation loads – fixed speed, when changing the FIP from 180 to 220 bar.

Low temperature combustion (LTC) decrease NO_x and PM simultaneously but increase HC and CO emissions. Recent attempts to attain LTC by biodiesel have created a hope for reduced HC and CO emissions. Decreased performance issue during LTC is also being taken care of by latest ideas [5]. Fumigation and blends methods have the same behavior in affecting performance and emissions, but the improvement in using the fumigation method was better [6]. The optimum percentage for ethanol fumigation is 20%. This percentage produces an increase of 7.5% in brake thermal efficiency, 55% in CO emissions, 36% in HC emissions and reduction of 51% in soot mass concentration. This also decrease of 48% in engine smoke and 51% in soot mass concentration. The optimum percentage for ethanol diesel fuel blends is 15%. This produces an increase of 3.6% in brake thermal efficiency, 43.3% in CO emissions, 34% in HC, it can also produce a reduction of 33.3% in engine smoke and 32.5% in the soot mass concentration. [7] The emission reductions are mainly associated with the reduction of diesel fuel burned in the diffusion mode. The reductions have been found between a wider range of 14–57% at over all engine load conditions. [8], [9] conclude that Ethanol may be a good additional fuel to CI engines when being injected into inlet port in a proper proportion to diesel fuel. High efficiency enabled by hydrous ethanol use in dual-fuel engines are studied. [10], [11] The purpose of these project are to use hydrous ethanol to demonstrate high efficiency with reduced emissions in a modified diesel engine where ethanol provides up to 80% of the fuel energy input.

Diesel engine has lower fuel consumption and produce lower CO_2 compare to gasoline engine [12]. While diesel fuel price is rising [13], [14] and the reserving amount become lower. [15], [16], [17]. Alternative fuel such as ethanol which was good for economic state and CO_2 reduction was produced in Thailand [18], [19]. But ethanol cannot mix with diesel properly [20]. Diesel engine has incomplete combustion [21]. The major concerns are NO_x, CO, HC, Smoke (Particles and soot) and in the recent past CO_2 as its accumulation add to greenhouse gases in the atmosphere [22]. To reduce emissions, many car company try to understanding the nature and quantity of vehicle emissions and developing control technologies [23]. EURO 6 was instituted in 2014 and imposed in 2015 in Europe [24]. NO_x which will reacts to the atmosphere and cause acid rain [25] and soot are depended on both combustion temperature and fuel mixture concentration [26].

2. PROCEDURE AND METHODS

2.1. Experimental Setup

In this study, base engine is Kubota RT140 which is single cylinder direct injection water cooled 4-stroke engine. And the engine specification shows in table 1.

Table 1 Type the Caption Here.

Model	Kubota RT-140
Number of cylinder	1
Bore x stroke (mm)	97x96
Displacement (cc)	709
Max Output (HP (kW) / rpm)	14/2,400 (10.3kW/2,400)
Continuous Rated Output (HP (kW) / rpm)	12.5/2,400 (9.2kW/2,400)
Specific Fuel Consumption (at continuous rated output) (g/HP-hr)	170 (231 g/HP-hr)
Compression Ratio	18.1
Valve Clearance (mm)	0.195 - 0.235
Max Torque (Kg-m / rpm)	5.0/1,600
Crankcase Oil Capacity (L)	2.8
Lubricating Oil	SAE 40 API CF
Combustion System	Direct Injection
Cooling System	Radiator
Lubricating System	Forced Lubrication with Trochoid Pump
Air Cleaner Type	Wet/Dry Type
Starting System	Electric
Battery (ES Model)	12 V, 30 Amp up
Direction of Revolution	Counter-clockwise Facing Fly Wheel
Dry Weight (V)	116

The engine will be equipped with common-rail system for the test to control the injection pressure and injection timing properly. By using common-rail system together with direct injection system, then one more injector will be fixed at the intake manifold pipe to inject ethanol in the intake manifold. All injectors will be controlled by FPGA system. The engine will be equipped with performance evaluation system by disc brake system (to measure torque from the tested engine). Convert mechanical energy from the engine to be force then generate to the Load cell Minebea U3B1-20K-1T-B. To measure the emission from of the experiment, Horiba Mexa 1600D emission analyzer was use. This analyzer is able to measure carbon dioxide, carbon monoxide, oxide of nitrogen, hydrocarbon compound, and oxygen. While smoke, amount will be measured by AVL Smoke Meter. The crank shaft angle will be measure by Rotary Encoder incremental type Autonics E40HB. This shaft encoder resolution is 3600 pulses per revolution. Crank shaft angle signal will be collect and recorded in to a computer called Data Acquisition. During the test, many parameters will be collected from various sensors which is consist of crank angle position sensor, ethanol and diesel injection controller. Temperature will be measured by K Type Thermocouple. The measurement range is -20 to 1350 C°. The injection system of diesel fuel will be replaced by a common-rail system from the conventional mechanical pump of the original Kubota engine. This system composes of high pressure, suction control valve, high pressure rail and high-pressure injector. The injection system of ethanol will compose of ethanol pump, pressure relive valve, pressure gauge, and the ethanol injector. The cooling system of the engine will be replaced to be as a conventional cooling system in the commercial car to increase the cooling capacity of the engine and to control the coolant temperature. The water cooling system will compose of water pump and the radiator with fan to blow out heat from cooling water. To measure the mass of air flow into the engine. Pressure sensor was used to measure the pressure different from the ventury. Figure 1 shows engine overview and operation system.

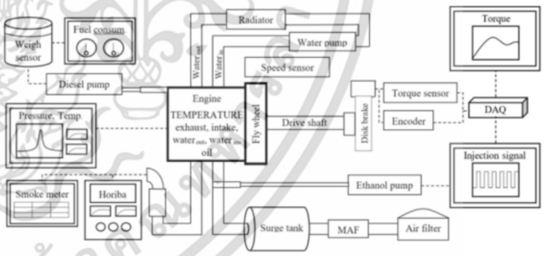

Fig. 1 Engine overview and operation system.

2.2. Experiment Procedure

Test were conducted by giving same energy input in every test condition for each engine speed. Decrease the energy input from diesel fuel and substitute by adding more ethanol injection. The total energy input is same. So the result of every testing condition in each engine speed can be compared. Ethanol will be injected into the intake manifold during the intake stroke. The injection pressure of ethanol is 3 bar. And will be injected by 10% increment by energy substitute the total energy from the diesel until the engine can not run (knocking or misfiring) start from pure diesel or e0. The injection pressure of diesel fuel will be 300, 500, and 700 bar. The injection timing of diesel fuel is 334, 337, 340, 343, 346, 350, and 355 degree crank angle. The original injection timing is 340 degree crank angle. The injection timing of ethanol will be limited by intake valve lifting character. Ethanol will be injected during intake stroke while the intake valve lifting up. The condition of the experiment will be shown in table 2. And the property of diesel fuel and ethanol will be shown in table 3. The heating value in the table is needed to calculate the proportion of ethanol fumigation to show the percentage of the total energy input. The injection duration of both fuel will be varied by engine speed and testing condition.

Table 2. Experimental conditions

Engine Speed [rpm]	1400,1600,1800,2000
Diesel injection pressure [Bar]	300,500,700
Diesel injection timing [^{o}aTDC]	-26,-23,20,-16,-14,-10, -5
Ethanol injection pressure [Bar]	3
Ethanol injection timing [^{o}aTDC]	-160

Table 3. Fuel property

Fuel properties	Diesel fuel	Ethanol
Specific gravity [kg/m^3] at 15 ^{o}C	0.84	0.785
Viscosity [cP] at 20 ^{o}C and 1 atm	3.35	1.2
Molecular weight	170	46.07
Higher heating value [kJ/kg]	46100	29700
Lower heating value [kJ/kg]	43200	26900
Heat of vaporization [kJ/kg]	270	840
Cetane number	50	8

3. RESULT AND DISCUSSION

3.1. Performance

At 1400rpm, high injection pressure does not increase the engine torque. For 300bar FIP, adding more ethanol tend to give higher torque. But for e40 to e80, the engine cannot run at 334-346 ^{o}CA. For e0, retarding gives lower torque. But when fumigate more ethanol, retarding injection timing tend to give higher torque. For 300bar, e60 with 355^{o}CA gives highest torque. For 500 and 700bar FIP, retarding injection timing gives higher torque. For 500bar FIP, adding more ethanol gives a little bit higher torque. When increasing ethanol fumigation, the engine can run just only in more retard injection timing. For 500bar FIP, the ethanol can add up to e50 but can run only at 355^{o}CA and gives higest torque for 500bar. For 700 bar FIP, the result was similar to 500bar FIP. But gives highest torque at e20 with 355^{o}CA. At 1400rpm, most retard injection timing gives highest torque. Highest torque was generated at e60 300FIP and 355^{o}CA as shown in Fig.2

For 1600rpm, retarding injection timing from original injection timing gives higher torque except at e0, e10, e20 with 300bar FIP. For those conditions, 340^{o}CA gives best torque. Increasing the ethanol fumigation amount gives higher torque. Higher FIP tend to give higher torque, but highest torque was generated at 300bar e80 with 355^{o}CA which is quite similar to 500bar e40 with 355^{o}CA. For 500bar, 355^{o}CA with e40. For 700bar FIP. E20 with 355^{o}CA gives highest torque. Fig. 3 shows the result for torque at 1600rpm.

For 1800rpm, retarding injection timing from original injection timing gives similar or higher torque except at e0, e10, e20, and e30 with 300bar FIP. For those conditions, 340^{o}CA gives best torque. For 300bar FIP, adding more ethanol gives higher torque until e70. Highest torque was given at e70 and 355^{o}CA. For 500bar FIP, e0, e10, and e20 do not give high torque when retarding injection timing. But adding ethanol result in increasing the engine torque. Highest torque at 500bar FIP was generated at e40 with 355^{o}CA, which is the highest one for 1800rpm. For 700bar FIP, ethanol fumigation increase the engine torque until e20. Advancing injection timing from 340^{o}CA decrease engine torque. Best torque was generated at e20 and 355^{o}CA as shown in Fig. 4.

At 2000rpm, Higher FIP and higher ethanol fumigation gives higher engine torque for most of the testing condition. But for 300bar FIP, when the ethanol fumigation was increase higher, the engine torque was decrease. Which is similar to the effect of advancing injection timing, when ethanol was fumigated at e0 to e60 and at e80, advancing injection timing gives lower torque. Best torque was given at e70 with 355^{o}CA. When FIP was 500bar, adding more ethanol fumigation gives higher torque. But at low ethanol amount from e0 to e20, advancing injection timing does not increase the engine torque. Highest torque at 500bar FIP was found at e50 with 355^{o}CA which is highest torque for 2000rpm. For 700bar FIP, adding more ethanol fumigation gives higher torque. Highest torque was found around 343 and 346^{o}CA. Highest torque was generated at e20 with 346^{o}CA for 700bar FIP as shown in Fig.5

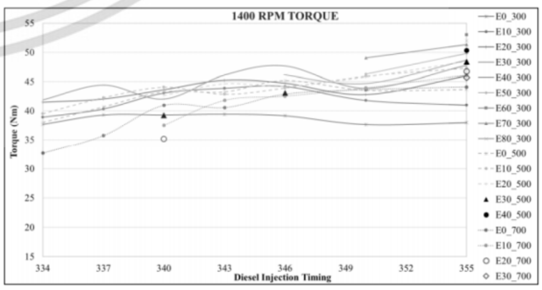

Fig.2 TORQUE @ 1400 rpm

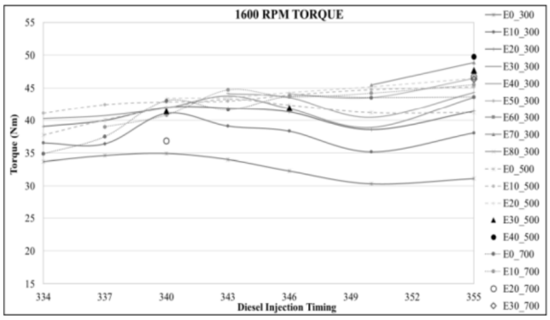

Fig.3 TORQUE @ 1600 rpm

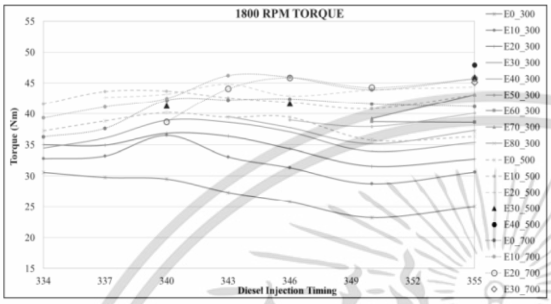

Fig.4 TORQUE @ 1800 rpm

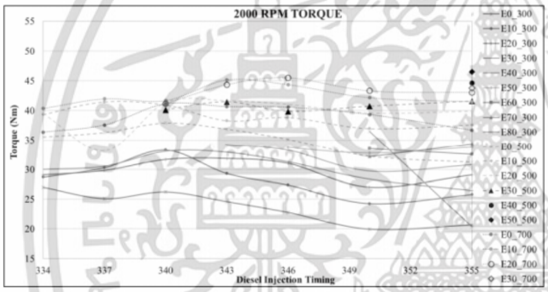

Fig.5 TORQUE @ 2000 rpm

3.2. Brake Specific Energy Consumption

For Brake Specific Energy Consumption, 300bar with low amount of ethanol fumigation gives the lowest BSEC for every engine speed except at 340°CA. And higher pressure will increase the BSEC for every condition.

At 1400rpm, advancing injection timing tend to decrease the BSEC and will also effect more at higher FIP. More ethanol fumigation tend to gives higher BSEC for 500 and 700bar FIP. Lowest BSEC for 1400rpm was found at 300bar e10 with 346°CA as shown in Fig. 6. The result was similar to 1600rpm. The lowest BSEC for this engine speed was generated at 300bar e10 with 340°CA which will show in Fig.7. And the result for 1800rpm is the same, at 300bar e10 with 340°CA, lowest BSEC as shown in Fig.8. And at 2000rpm, the result still shows that, at 300bar e10 with 340°CA gives lowest BSEC as Fig.9.

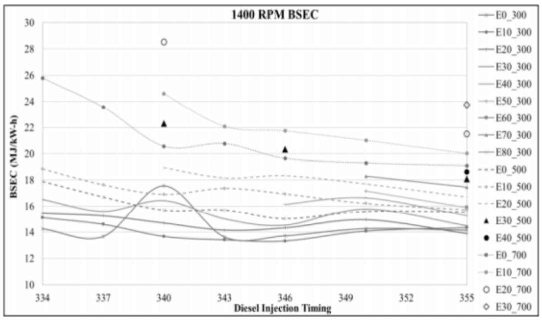

Fig.6 BSEC @ 1400 rpm

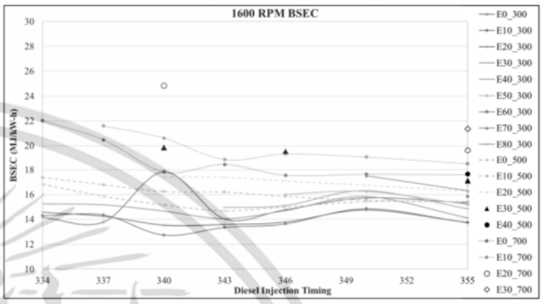

Fig.7 BSEC @ 1600 rpm

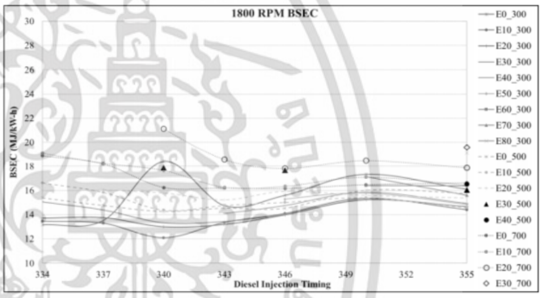

Fig.8 BSEC @ 1800 rpm

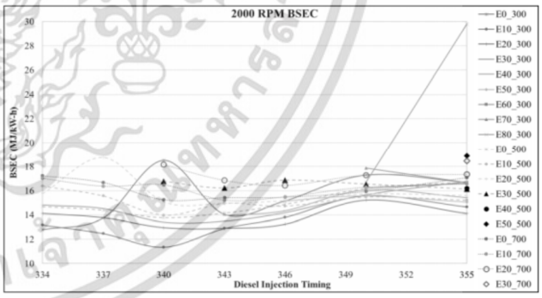

Fig.9 BSEC @ 2000 rpm

3.3. NO_x Emission

NO_x was decrease when the injection timing was retard for all testing conditioin. For 1400rpm, 300bar FIP gives high NO_x for advancing injection timing 334 to 340°CA and it become lower. Adding high amount of ethanol in advancing injection timing gives high NO_x, but from e10 to e30, NO_x was low from 343 to 355°CA. For 1400rpm, 300bar FIP e20 and 350°CA gives lowest NO_x emission. For 500bar FIP, NO_x was lower when retarding injection timing in e0 to e20. But NO_x will increase when fumigate up to e30 and highest at e40 for 355°CA. For 700bar FIP, NO_x was decrease when advancing injection timing and NO_x increase when adding more ethanol. Fig.10.

For 1600rpm, 300bar FIP also gives high NO_x when advancing injection timing. Retarding to 355°CA decrease NO_x until e50 and e70. E80 gives highest NO_x at 355°CA. Increase ethanol fumigation amount also gives high NO_x. For 500bar FIP, NO_x was increase when adding ethanol fumigation from 340°CA to 355°CA. For 700bar FIP, NO_x was very low when running with only diesel fuel. Adding more ethanol gives higher NO_x. NO_x was lowest at 700bar e0 334°CA for 1600rpm as Fig.11.

At 1800rpm, NO_x emission characteristic was similar to at 1600rpm. Retarding injection timing gives lower NO_x for all ethanol fumigation amount and every injection pressure. 700bar FIP e0 with 346°CA gives lowest NO_x emission for 1800rpm as shown in Fig.12.

For 2000rpm, Retarding injection timing gives lower NO_x same as 1600, and 1800rpm. 300bar FIP gives high NO_x from 334°CA to 340°CA. 500bar FIP tend to give lower NO_x at this injectin timing. Lowest NO_x emission was given at 500bar e0 with 350°CA for 2000rpm as shown in Fig.13

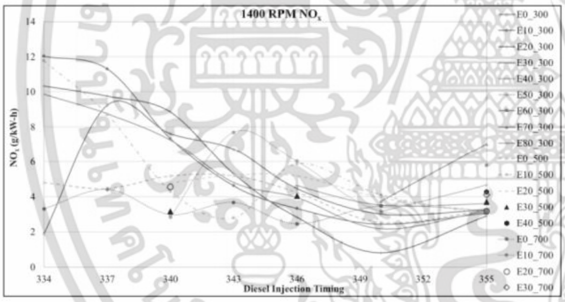

Fig.10 NO_x Emission @ 1400 rpm

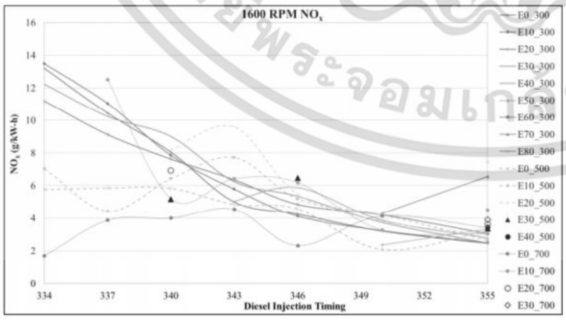

Fig.11 NO_x Emission @ 1600 rpm

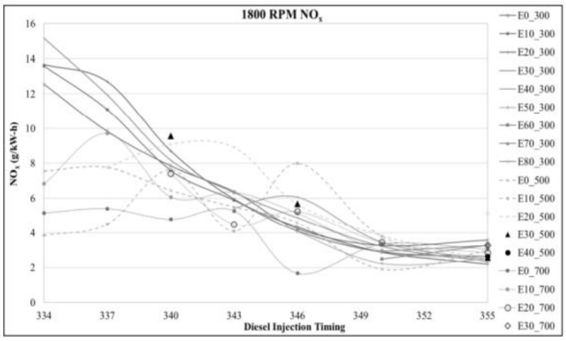

Fig.12 NO_x Emission @ 1800 rpm

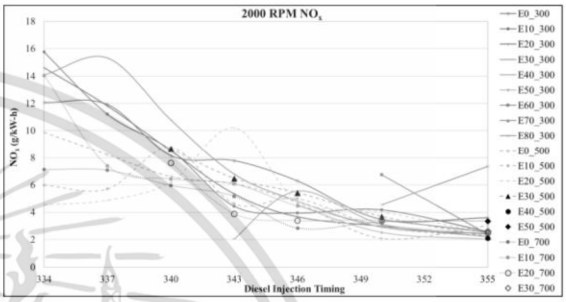

Fig.13 NO_x Emission @ 2000 rpm

3.4. Smoke Emission

For smoke emission, it clearly see that when reducing diesel injection, smoke emission was lower. For 1400rpm, high FIP gives high smoke emission. Reducing amount of diesel injected into the combustion chamber will generate low somke respectively. Lowest somke emission was given at 300bar FIP e80 and 355°CA. Advancing injection timing tend to give higher smoke emission as show in Fig.14. At 1600rpm. The result was similar to 1400rpm. But somke was least generated at 300bar e70 with 355°CA as shown in Fig.15. which was same as at 1800rpm. Smoke emission characteristic was same, and lowest smoke was generate at the same testing condition at 300bar e70 with 355°CA as shown in Fig.16. For 2000rpm, The smoke emission characteristic was still similar to other engine speed. Lower diesel injected means lower smoke emission. But lowest smoke emission was given at 300bar e70 with 350°CA as shown in Fig.17. All smoke emission characteristic was decrease due to increasing ethanol fumigation. And most of smoke was increase due to retarding diesel injection timing as well.

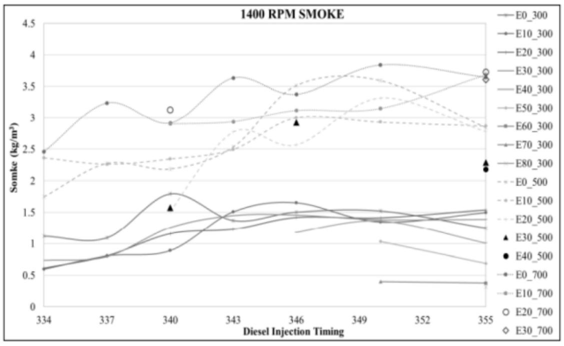

Fig.14 Smoke Emission @ 1400 rpm

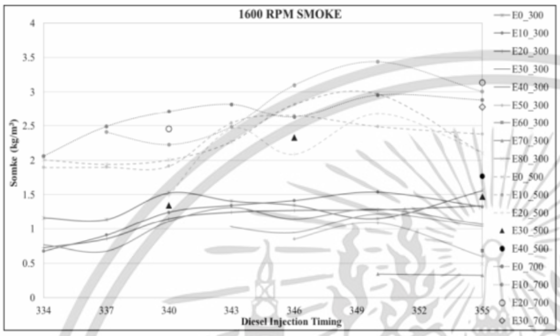

Fig.15 Smoke Emission @ 1600 rpm

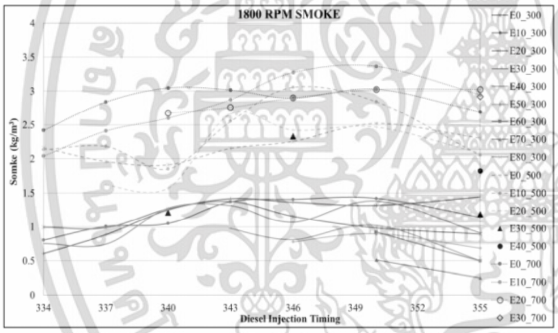

Fig.16 Smoke Emission @ 1800 rpm

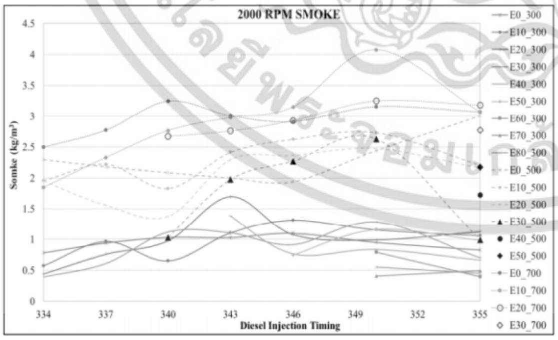

Fig.17 Smoke Emission @ 2000 rpm

4. CONCLUSION

From the experiment. Engine need different injection pressure, injection timing and injection amount to get the best result for performance and emission and it also depend on engine speed. To get highest torque from the engine at 1400rpm, e60 300bar FIP and 355°CA, at 1600rpm, 300bar FIP e80 with 355°CA, at 1800rpm, 500bar FIP e40 with 355°CA, and at 2000rpm, 500bar FIP e50 with 355°CA will give the best result for torque. The result shows that highest torque will generated at most retard injection timing. Because when the piston rotate up near to the top dead center, the pressure and temperature will be very high compare to lower position. The combustion will happen rapidly and will gives very high torque.

For BSEC at 1400rpm, 300bar e10 with 346°CA, at 1600rpm, 300bar e10 with 340°CA, at 1800rpm, 300bar e10 with 340°CA, and at 2000rpm, 300bar e10 with 340°CA gives the best result. Retarding injection timing can increase the ethanol fumigation amount, but in very high ethanol amount. The engine needs to run at high engine speed too. Because the temperature of the combustion is high enough to burn ethanol. Which will result in low BSEC.

For NO_x emission, at 1400rpm, 300bar FIP e20 and 350°CA, at 1600rpm, 700bar e0 334°CA, at 1800rpm, 700bar FIP e0 with 346°CA, and at 2000rpm, 500bar e0 with 350°CA gives the best result. To reduce NOx at low engine speed, ethanol must be inject in low amount. Because the cooling ability of itself can lower the combustion temperature. But for higher engine speed, ethanol fumigation will cause NOx emission because of excess oxygen in the combustion process.

For Smoke emission, at 1400rpm, 300bar FIP e80 and 355°CA, at 1600rpm, 300bar e70 with 355°CA, at 1800rpm, 300bar e70 with 355°CA, and at 2000rpm, 300bar e70 with 350°CA gives the best result. Smoke will reduce due to injection amount of diesel fuel. When increase the ethanol fumigation and reducing diesel injection, smoke will be lower.

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ABBREVIATIONS

bTDC	before top dead center
aTDC	after top dead center
EGR	exhaust gas recirculation
BSFC	brake specific fuel consumption
BSFC	brake specific energy consumption
BMEP	break mean effective pressure
BTE	break thermal efficiency
P_{max}	peak combustion pressure
CO	Carbon monoxide
CO_2	Carbon dioxide
HC	hydrocarbon
PM	particulate matter
NO_x	nitrogen oxide
IMEP	indicated mean effective pressure
RoHR	rate of heat release
FIP	fuel injection pressure

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APPENDIX B

EQUIPMENT SPECIFICATION

Table 0.1 Kubota RT140 engine specification

Model	**Kubota RT-140**
Number of cylinder	1
Bore x stroke (mm)	97x96
Displacement (cc)	709
Max Output [HP (kW) / rpm]	14/2,400 (10.3kW/2,400)
Continuous Rated Output [HP (kW) / rpm]	12.5/2,400 (9.2kW/2,400)
Specific Fuel Consumption (at continuous rated output) (g/HP-hr)	170 (231 g/HP-hr)
Compression Ratio	18.1
Valve Clearance (mm)	0.195 - 0.235
Max Torque (Kg-m / rpm)	5.0/1,600
Cooling Water Capacity (L)	2.1
Fuel Tank Capacity (L)	11
Crankcase Oil Capacity (L)	2.8
Fuel	Light Diesel Oil (SAE No. 2-D)
Lubricating Oil	SAE 40 API CF
Combustion System	Direct Injection
Cooling System	Radiator
Lubricating System	Forced Lubrication with Trochoid Pump
Air Cleaner Type	Wet/Dry Type
Starting System	Electric
Battery (ES Model)	12 V. 30 Amp up
Direction of Revolution	Counter-clockwise Facing Fly Wheel
Dry Weight (V)	116

Table 0.2 Load cell specification

Minebea U3B1-20K-1T-B	
Rated capacity	196.1 N – 9.807 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.
Temperature range	-20°C to 80°C
Class of protection	Equivalent to IP67
Durability	10^6 times with rated load

Table 0.3 Specification of Horiba Mexa 1600D

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%

Table 0.4 Specification of AVL smoke meter

Measurement principle:	Measurement of filter paper blackening
Measured value output:	FSN (filter smoke number) or mg/m³ (soot concentration)
Measurement range:	0 to 10 FSN
Detection limit:	0.002 FSN or ~ 0.02 mg/m³
Resolution:	0.001 FSN or 0.01 mg/m³
Exhaust pressure ranges:	(-300*) – 100 to 400 mbars (-500*) - 200 to 750 mbars with the special sampling option0 to 3000 mbars with the high-pressure option (*) with activated altitude simulation
Maximum exhaust temperature:	600 °C with standard 340 mm sample probe (800 °C with 780 mm long sample probe)
Interfaces:	2 serial RS232 interfaces with AK protocol Digital via Instrument Controller 4210 1 Ethernet interface with InPort option installed with AK protocol
Power supply:	100 – 115 VAC or 230 VAC, 50/60 Hz
Compressed air (for compressed air option):	~150l/min during purge
Compressed air quality required:	Grades 1.1.1 to 1.4.1 according to ISO8573.1:2001(E) Recommended connection pressure on the AVL Smoke Meter: 5 to 8 bars at the measurement device input
Sample flow:	~ 10 l/min
Ambient conditions:	5 to 55 °C / max.95 RH; without condensation Sea level -500 to + 5000 m
Repeatability:	Standard deviation 1 s = ± (0.005 FSN + 3 % of the measured value @ 10sec intake time)

Table 0.5 Specification 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

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