STUDY OF DIRECT INJECTION SPARK IGNITION (DISI) ENGINE WITH HIGH COMPRESSION RATIO FOR ALTERNATIVE FUEL

RATTAPOOM KESKANGAM

A THESIS SUBMITTED IN PARTIAL FULFILLMENT OF THE
REQUIREMENT FOR THE DEGREE OF MASTER OF ENGINEERING IN
AUTOMOTIVE ENGINEERING
(INTERNATIONAL PROGRAM)
INTERNATIONAL COLLEGE
KING MONGKUT'S INSTITUTE OF TECHNOLOGY LADKRABANG
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STUDENT NAME	Mr. Rattapoom Keskangam
STUDENT ID	57610022
DEGREE	Master of Engineering
PROGRAM	Automotive Engineering (International Program)
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THESIS ADVISOR	Asst. Prof. Dr. Preechar Karin Dr. Nuwong Chollakoop Prof. Dr. Hidenori Kosaka

ABSTRACT

This research explains about the investigation on DISI (Direct Injection Spark Ignition) engine using ethanol blends and EGR (Exhaust Gas Recirculation) system for improving engine fuel consumption. With a huge number of transportations nowadays, there are global crisis with the fossil fuel usage and the limitation of fossil fuel in the world, Although, CO, HC, NOx and PM are emissions made from the fossil fuel used in the internal combustion engine. This problem is something we have to solve and make it better for human life in the present and also in the future. The combination of ethanol alternative fuel and EGR technique on part load can be directly affect to internal combustion, it would improve Brake Specific Fuel Consumption (BSFC) and emissions while power and torque remain constant accompanied with an adaptation of EGR system, mostly use in a diesel engine to DISI engine. EGR dilutes air intake in engine part load operations. The result of EGR system is an improvement of Brake Specific Fuel Consumption (BSFC), anti-knocking limit and decrease combustion temperature which leads to NOx emission with lean burn mixture in DISI engine using ethanol blends as fuel.

Keywords: Ethanol, Exhaust Gas Recirculation, Emissions.

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CHAPTER 1

INTRODUCTION

1.1 Research Background

Currently, using fossil fuel is the major factor in propelling economics into many sectors, especially in transportation. The impact of using fossil products not only in currency but also in the environment becomes an impurity as well. To seek another alternative energy for mount up clean environment together with good health and mental by plenty of researchers have been adopted into widespread likewise electrified and fuel cell vehicles. However, those of innovations are not suitable to apply on this moment owing to cost of operation for mass production, prompt of infrastructure and believing of the user from the impact to their way of life. Then, to adopt the current technology of the internal combustion engine for highly efficient and emitting low emission is the suitable solution [1].

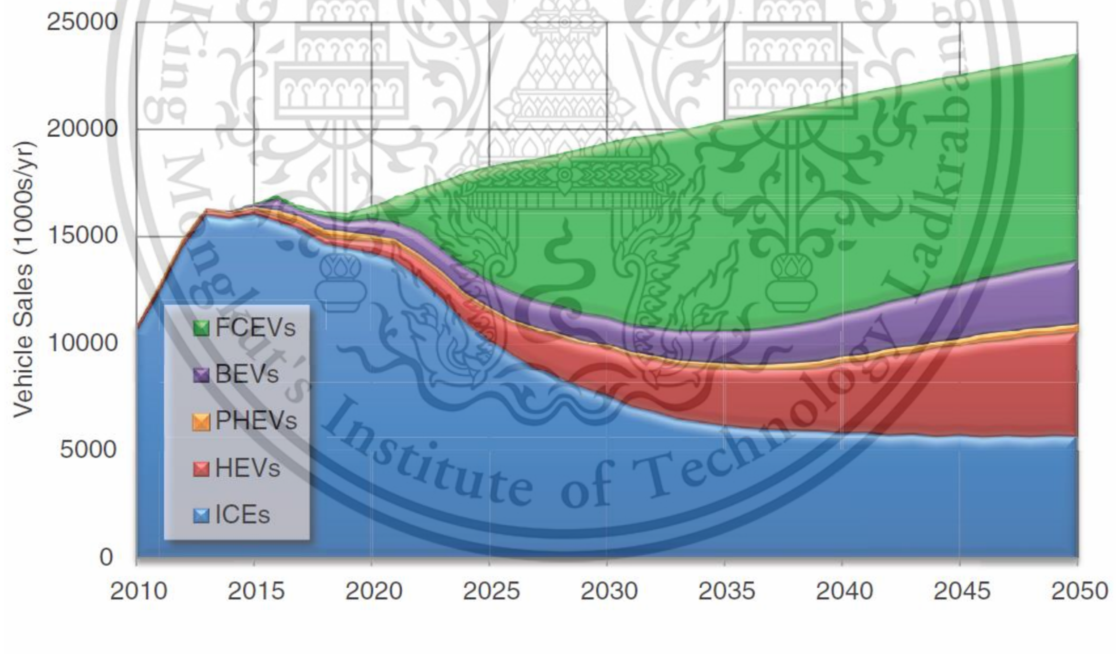


Figure 1.1 Tendency of vehicle technologies from 2010 to 2050 [1].

Regarding to renewable energy plan and policy development plan of Thailand 2010 to 2021 [2], using bio-energy form ethanol is main role from plenty of benefits such as reduction of import a crude oil by produce in their country which is highest after compared with bio-diesel and another. Hence, to research of ethanol for advance utilization is crucial.

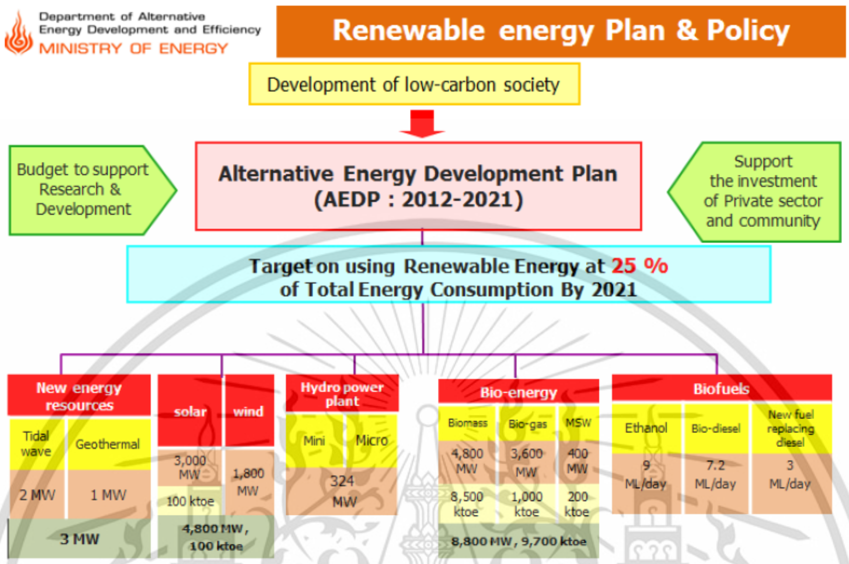


Figure 1.2 Renewable energy and policy development plan of Thailand 2010 - 2021 [2].

The direct injection spark ignition engine (DISI) become to use widespread in Thailand owing to obtain performance better than conventional port fuel injection (PFI) engine and less of fuel economy from stratified charge operation [3, 4, 5]. In contrast, the stratified charge emitted emission in particulate matter (PM) phase higher because it operates like compression ignition (CI) engine. From mentioned, fueled ethanol to mount up performance and reduce of emissions is attractive. Then to find out the proper condition between biofuel and engine operation is necessary to develop advance information for Thailand.

EGR system is a technique which recirculate exhaust gas back to dilute with fresh air inside intake manifold. It is firstly adopted in diesel engine to limit Thermal NO_x formation rate by reducing combustion temperature. But with higher issue to emission and use of energy, EGR can use as an advance technique to improve engine efficiency and emissions. It can reduce engine pumping loss in partial load condition. EGR can also has a good benefit for ultra-lean mixture in stratified charge combustion for gasoline direct injection spark ignition engine. With its advantage we can use it to make better gasoline direct injection.

1.2 Objectives

The objective of this research conducted on efficiencies and emissions from gasoline engine with direct injection technology on stratified charge mode which implied in to 3 sections as following;

1.2.1 The impact of ethanol Fueled in Gasoline blends with gasoline on performance and emissions.

1.2.2 To investigate from engine operations; the timing of injection and ignition.

1.2.3 To study the characteristics from exhaust gas recirculation system (EGR).

1.3 Scope of study

1.3.1 The characteristics of fuel on efficiencies and emissions.

Studying an effect of gasoline and ethanol on properties point of view.

1.3.2 The impact of exhaust gas recirculation system (EGR) on emissions.

Using gasoline and ethanol as fuel with different conditions of percentage of EGR input and ignition timing, it makes differences in emissions that are collected by gas analyzer equipment.

1.3.3 The impact of exhaust gas recirculation system (EGR) on ethanol blends on best condition to improve engine efficiency.

Stratified charge mode in DISI engine with ethanol blended fuel can use EGR system to improve performance and emissions by variation EGR rate and ignition timing. The results analyzed in term of performance and efficiency (brake specific fuel consumption, brake specific energy consumption and thermal efficiency) and emissions.

CHAPTER 2

LITERATURE REVIEW

2.1 Direct Injection Spark Ignition (DISI) Engine

The direct injection spark ignition engine is the next generation of the spark ignition engines to improve efficiency than conventional (multi-port fuel injection - MPI) SI engine. The engine operation of DISI type is the key to achieve by categorize into 2 strategies based on engine load condition as stratified charge and homogeneous charge condition.

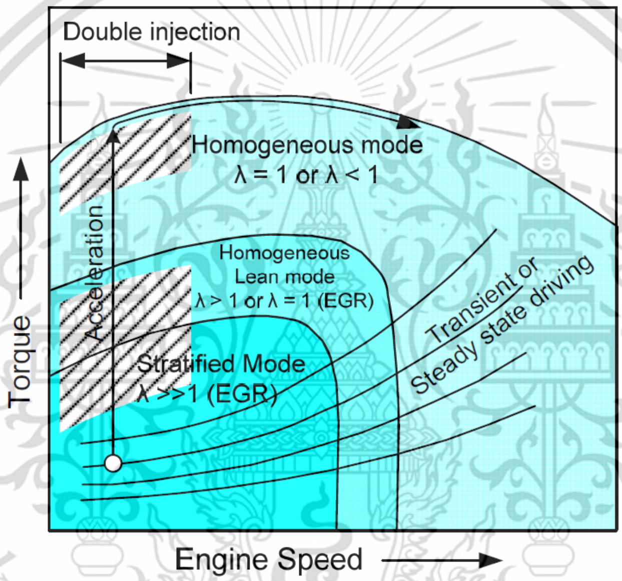


Figure 2.1 Direct Injection Spark Ignition (DISI) engine operation [6].

Regarding to figure 2.1, stratified charge mode operates on low to medium engine load condition by using late injection strategy at compression stroke where the mixture in this mode is lean equivalent ratio. The benefit of this mode can reduce pumping loss of engine results in better fuel economy. On the contrary, to require a medium to high engine load, the engine operation switches to a homogeneous charge mode for proper duration of fuel and air intake mixture. Furthermore, when a full engine load is requested. The rich mixture of air-fuel is mandatory to obtain the maximum output and complete combustion as well [4, 5, 7, 8].

2.2 The Injection Strategy of Direct Injection Spark Ignition (DISI) Engine

The Injection Strategy of direct injection spark ignition engine categorize into 3 type as the following;

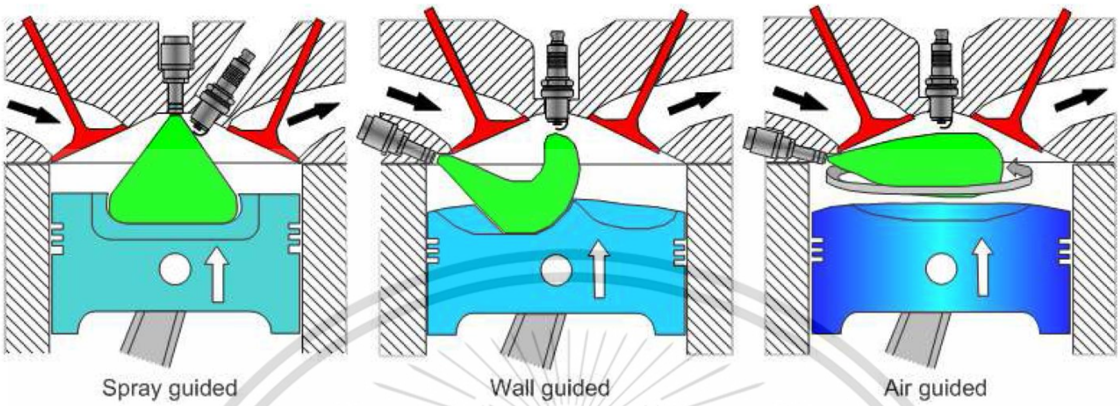


Figure 2.2 DISI engine injection strategy [9].

2.2.1 Wall Guided System

An injector is located on the side of the combustion chamber together with using a spherical piston shape for mixture preparation. This system can maintain a short range between spark gap and the tip of injector, but it relates the short time for mixture formation that generates soot and total hydrocarbon (THC). However, the wall guided strategy is reliable due to prevent the misfiring and vigorousness of combustion.

2.2.2 Air Guided System

This system uses an air flow from the intake tract as the main role to inject fuel for combustion. The shape and speed of air flow are controlled by the shape of intake port and air flap inside the intake manifold in order to compact fuel spray. From mentioned can resolve a problem of fuel wetting on cylinder wall and piston bowl.

2.2.3 Spray Guided System

Fuel is injected with the nearest speak plug than both the previous technique. This concept brings best efficiency and fuel economy as a reduction of wall wetting, extend a region of stratified charge mode operation on high engine load and speed from high excess air intake charge and multiple injection strategy in lean burn mode, low sensitive both of variation of cylinder to cylinder and in-cylinder air flow (improving the mixture formation) together with reduce the total hydrocarbon (THC) [10, 11].

Studying of optimization of DISI found value information, an effect of the injection strategy on the mixture formation and combustion characteristics in a DISI (direct injection spark ignition) optical engine was carried out by J. Song et al. [12] as the single injection, retarding the injection timing caused an increase in the combustion speed and in-cylinder pressure. In this case, the major factor that increased the in-cylinder pressure was the turbulence intensity. The latter injection produced a stronger turbulence at the spark timing, which caused a higher in-cylinder pressure. Meanwhile, the flame propagation direction was mainly affected by the equivalence ratio distribution. The flame speed toward the fuel-lean region was significantly slower than toward the fuel-rich region.

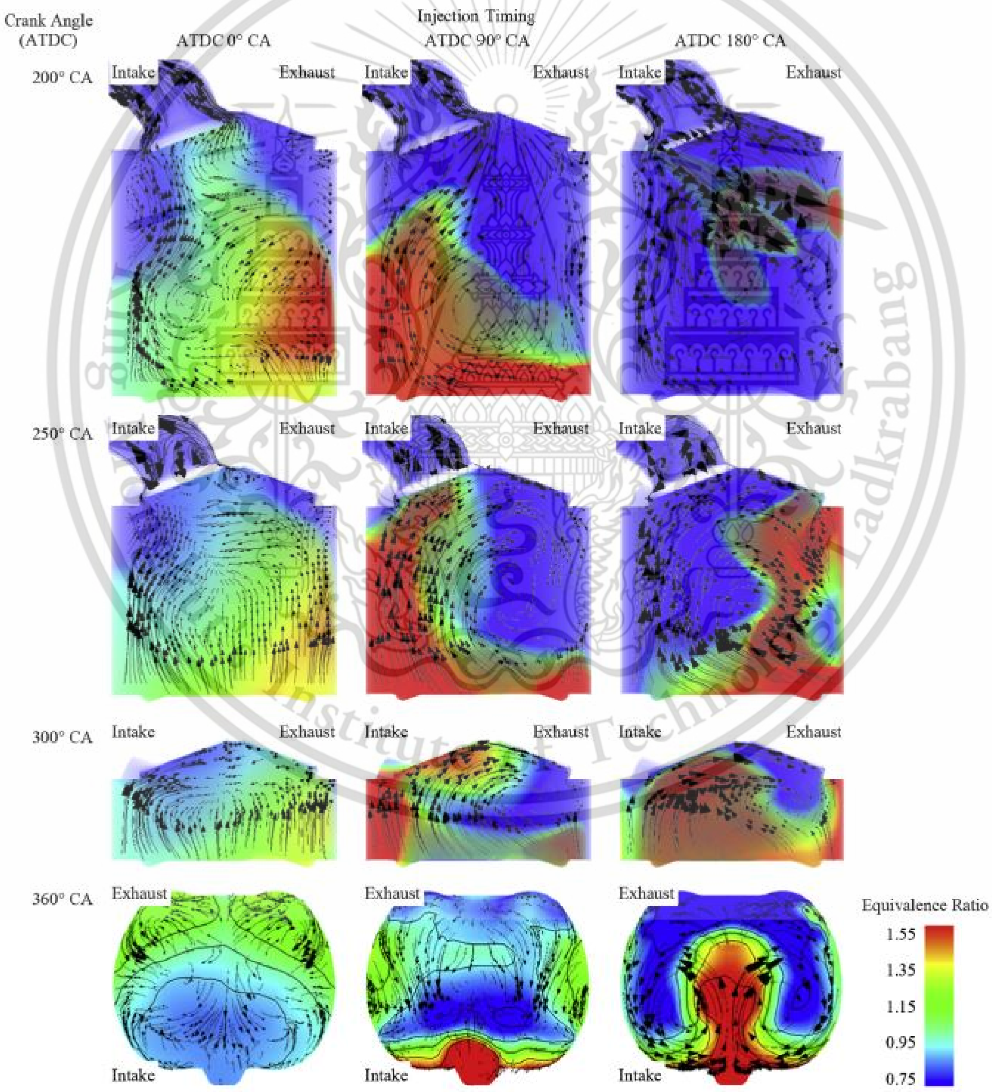


Figure 2.3 Impact of injection timings on cylinder flow.

2.3 Emission from Direct Injection Spark Ignition (DISI) Engine

Basically, the internal combustion engine (ICE) transmits chemical energy from fuel to mechanical power by pressurized fuel from high-pressure pump mixed together with air intake following with spark discharge for combustion. The product of exhaust gas consists of emissions that are harmful to human health and environment. Emissions of ICE are carbon monoxide (CO), total hydrocarbon (THC), oxide of nitrogen (NOx) and particulate matter, as shown in equation 2.1

$$C_xH_y + Air \rightarrow aCO_2 + bH_2O + cCO + dC_{(s)} + mHC_{(l)} + nNO_x \quad (2.1)$$

- Carbon monoxide (CO), total hydrocarbon (THC) and aldehydes are generated from incomplete combustion, which is the main factor of smog. The major of exhaust hydrocarbons comes from lubricant.

- Oxide of nitrogen (NO_x) occurs on high pressure and temperature conditions of nitrogen and oxygen. NO_x is one of toxic severe emission for environment from major component of smog and acid rain.

- Particulate matter (PM) is a complicated accumulation of solid and liquid matter. The PM was made from the carbonaceous particle combustion process.

2.4 Exhaust Gas Recirculation (EGR)

An exhaust gas recirculation (EGR) is main role of after treatment system to reduce emission and widely adopt in internal combustion engine (ICE). The strategy of EGR in both of spark ignition and compression ignition engine is reduce an oxide of nitrogen (NO_x) by recirculate the exhaust gas into the intake manifold for combustion in next cycle. There are benefits of recirculating gas are reduction of 3 factors as; temperature of combustion, exhaust temperature and concentration of oxygen. Furthermore, the indirect advantage of EGR can retard the ignition delay together with reduce the noise level as well. To find out the volume of EGR, the maximum mass flow rate of intake air and engine load with NO_x condition are key point for determination [13].

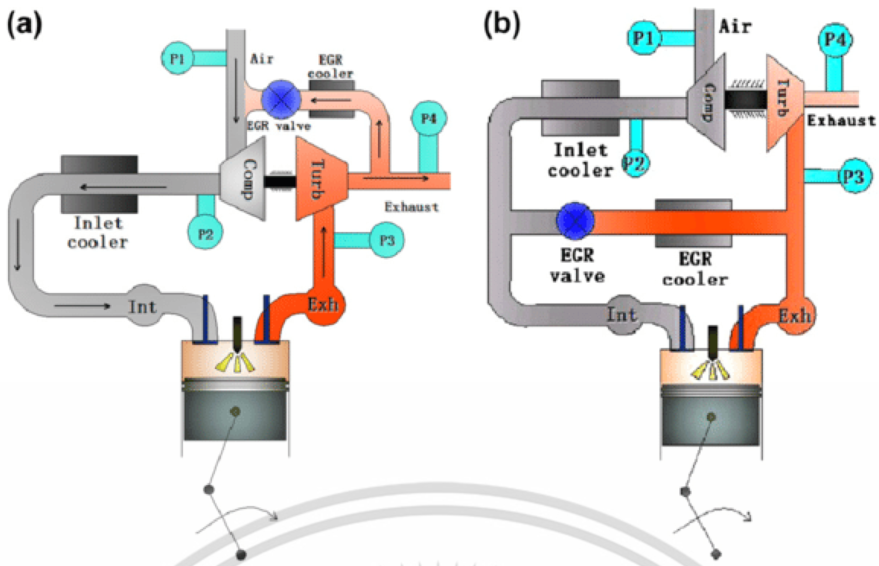


Figure 2.4 Schematic diagram of Exhaust Gas Recirculation System a) Low Pressure Loop Type b) High Pressure Loop Type [14].

The configuration of EGR can classify into 2 types as;

1. Low pressure loop (LPL) EGR

An exhaust gas from downstream side (Turbocharger condition - turbine) of after treatment system entrains to the upstream line (Turbocharger condition - compressor). To control the EGR rate, another throttle valve body is introduced for highly efficient. However, the temperature of air intake of this type is higher than HPL [15].

2. High pressure loop (HPL) EGR

This system entrains some amount of exhaust gas from exhaust manifold direct to intake manifold. EGR gas temperature is controlled by gas cooler. To control an EGR precisely, the throttle valve body on intake system is major factor to maintain rate of EGR before induces to combustion chamber [16].

Comparing both types, the HPL is more suitable owing to a simple configuration and precise control of the EGR rate. On the other hand, the LPL is less favorable because of hard to control the EGR rate from the difference between pressures of ambient and EGR at the entire of EGR circuit.

EGR system is the system mostly use in the diesel engines. It is to operate the exhaust gas to recirculate back into the intake system and the combustion chamber. When DISI engine works in stoichiometric conditions and part load throttling, EGR dilute exhaust gas with intake air and the fuel consumption (BSFC) becomes better. To remain power and torque when EGR operating, it is required to throttling more and this condition can reduce engine pumping loss as shown in figure 2.5 – 2.6 by Tianyou W et al. [17] This advantage of EGR improves fuel consumption (BSFC) when engine operate with WOT (wide open throttle) condition because WOT condition does not produce pumping loss by throttle. Nevertheless, if EGR operate at WOT condition, volumetric efficiency will reduce causing to less power and torque.

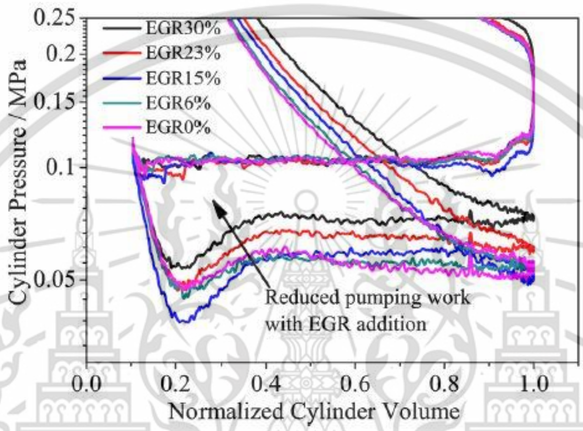


Figure 2.5 Effect of EGR ratio and pumping work.

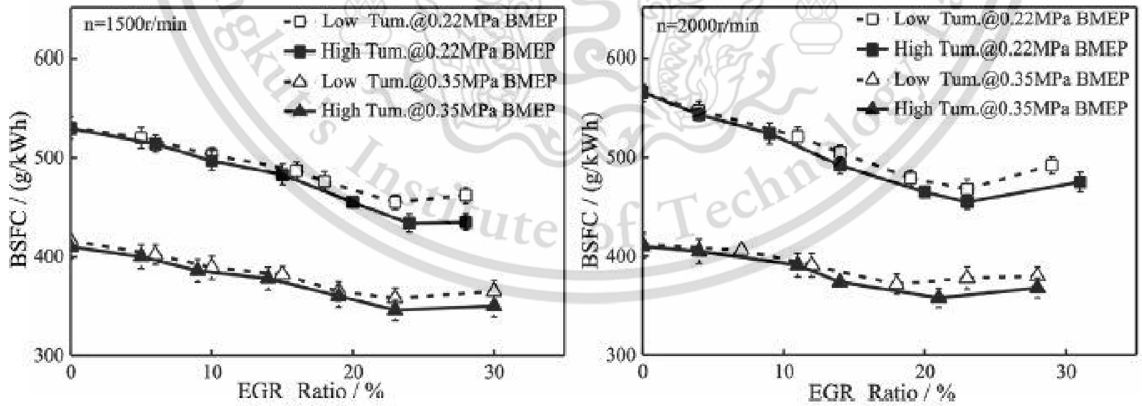


Figure 2.6 BSFC VS EGR ratio (1500 RPM, 2000 RPM, Part load).

The study of Gequn S. et al. [14] When DISI engine works with the lean mixture or ultra-lean mixture in stratified charge mode as in figure 2.7, An increasing in combustion duration causing high combustion temperature and NOx is dramatically increased. EGR is an important role to reduce NOx emissions, as shown in Figure 2.8. With a given power and torque, fuel is constant, exhaust gas will take place for fresh air, and AFR will decrease with the increase of EGR rate.

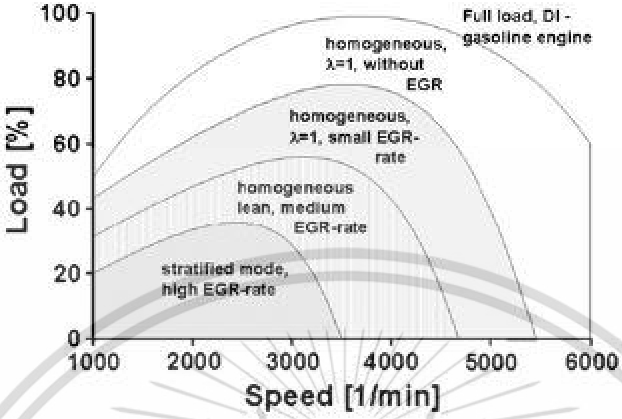


Figure 2.7 EGR operational character with DISI engine.

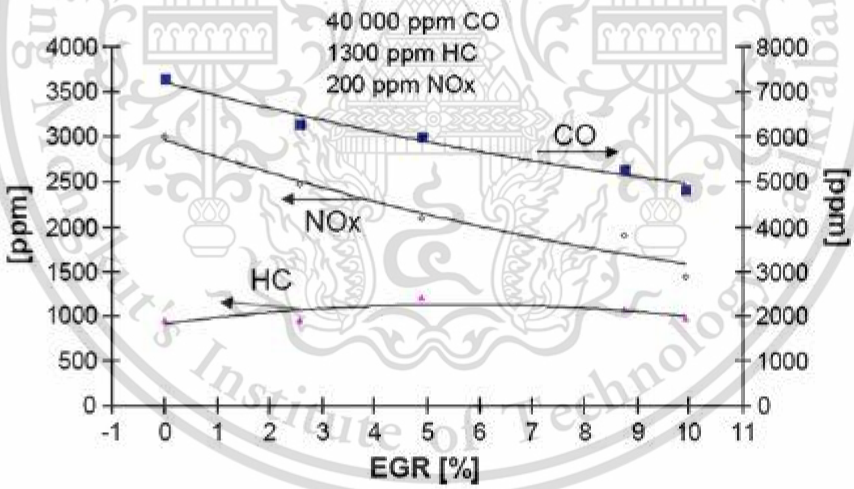


Figure 2.8 Engine emission from various EGR rate (Engine speed of 4000rpm, Stoichiometric air-fuel mixture).

Knocking is the main problem to improve BMEP of SI engine. Normally for SI engine, the method to prevent knocking is changing fuel enrichment. Excessive fuel mixture can reduce combustion temperature but catalytic converter can be effective only in the stoichiometric mixture, so excessive fuel mixture will increase emission. Using a cooled EGR is an effective way to reduce combustion temperature and inhibit knocking. The testing using cylinder pressure to collect data; the limitation are knocking combustion and combustion temperature limit in figure 2.9 shows BMEP can increase by change ignition angle but it needs EGR dilute to inhibit knocking combustion and reduce combustion temperature. Without knocking engine allowed to work with higher cylinder pressure.

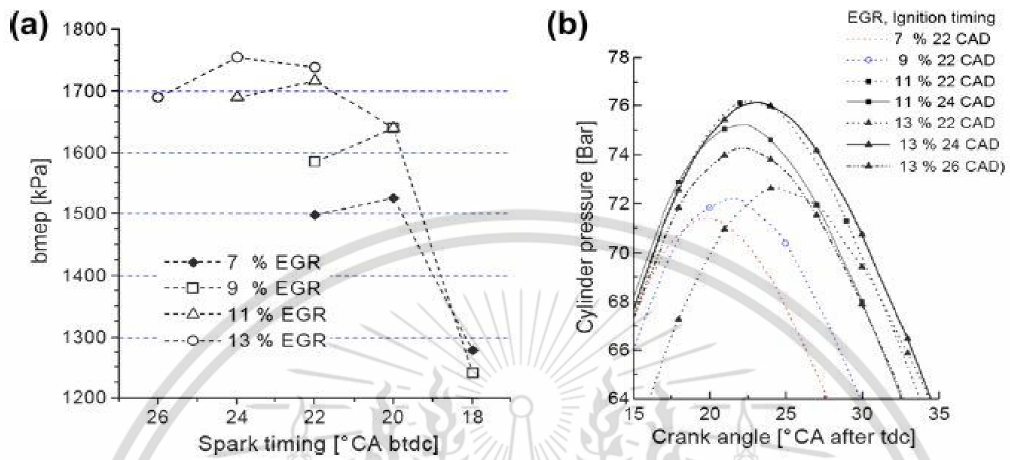


Figure 2.9 Maximum BMEP as EGR and ignition angle (4000rpm)

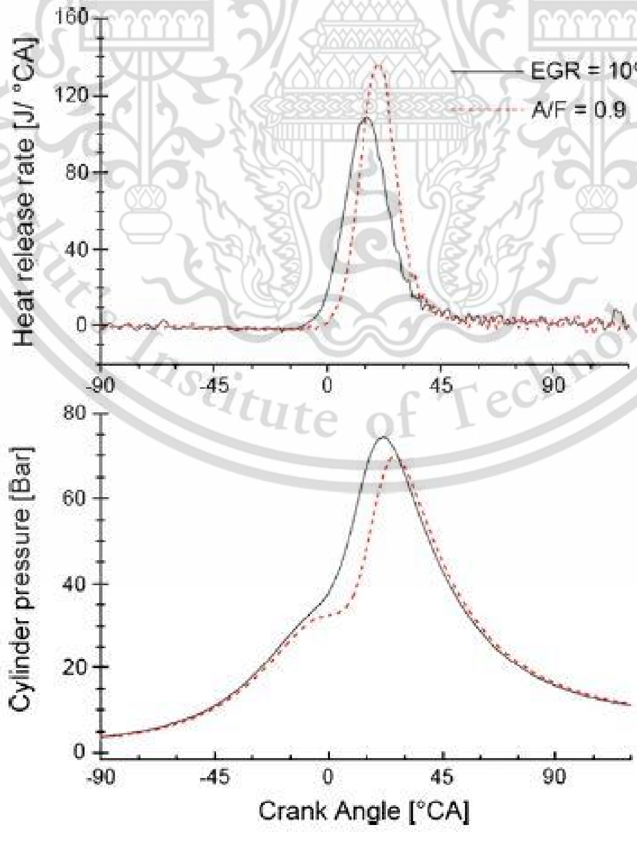


Figure 2.10 Comparison of combustion for 10% and 11% of fuel enrichment

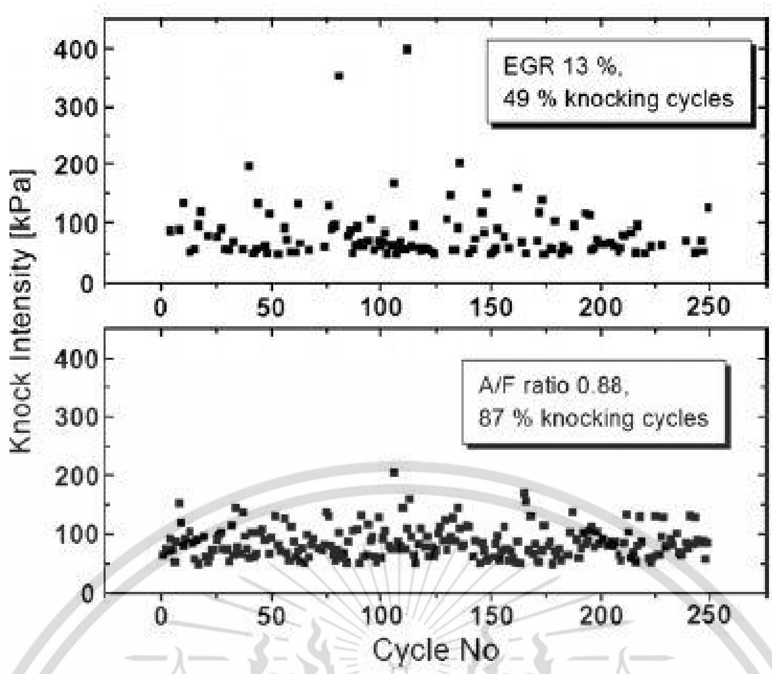


Figure 2.11 Knock intensity of knocking cycle
(5000 RPM, full load, 8 CADBTDC)

Figure 2.11 shows knocking cycle using EGR is less than knocking cycle using fuel enrichment. Using EGR instead of fuel enrichment is great way to improve fuel consumption.

For all the information shown above, the combination of the EGR system and ethanol fuel can greatly benefit the DISI engine. EGR system can achieve DISI engine better fuel consumption, NOx emission reduction and Knocking intensity reduction. Ethanol is the very competitive fuel and releases less emissions than gasoline. The conclusion will be investigated in this thesis.

3.3 Alternative Fuel: Ethanol

To achieve the mega-trend of technology for optimization and emission reduction. Not only the new generation of the internal combustion engine but also seek out alternative fuel as a supplementary choice owing to limited crude oil supply is essential. Ethanol can be considered an octane enhanced, fuel blended in crude oil and oxygenated fuel.

Regarding to various study of ethanol as fuel, the ethanol has a heating value lower than conventional gasoline that leads concern of engine in cold start. The total energy input at stoichiometric conditions is familiar with gasoline. Furthermore, an outstanding property of ethanol; The high latent heat of vaporization brings a cooling effect inside the combustion chamber. These can mount up volumetric efficiency together with better knock resistance from high octane number at high compression ratio or high boost as well.

To understand the benefits of ethanol for DISI engine application, combustion behavior and mixture formation are focused by several of researchers. To begin with spray characteristics from P.G. Aleiferis et al. [18, 19]mentioned as the spray plume shape of fuel intend to disintegration by low boiling point element unless it's not on the flash boiling point state. In case of spray optimization, the heat ethanol was conducted and results as the droplet of E85 larger than gasoline that confirm as addition of ethanol brings about the lower concentration.

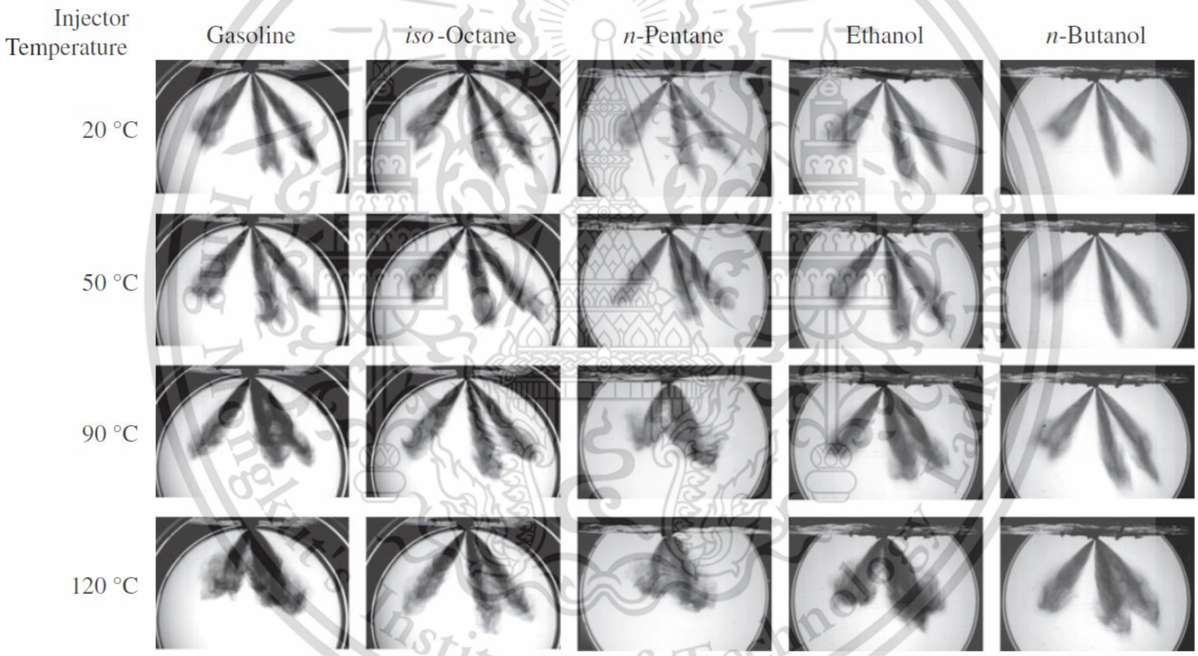


Figure 2.12 Spray images on temperatures, 0.5 bar, 777 μs ASOI. [18].

Moreover, S.H. Park et al. [20] studied the atomization and spray characteristics of bio ethanol and bioethanol blended with gasoline fuel through DISI with swirl injector reported as increasing the percentage of ethanol at lower ambient pressure decreasing the main spray tip penetration and become unostentatious when increasing an ambient pressure. [fig.5 58]

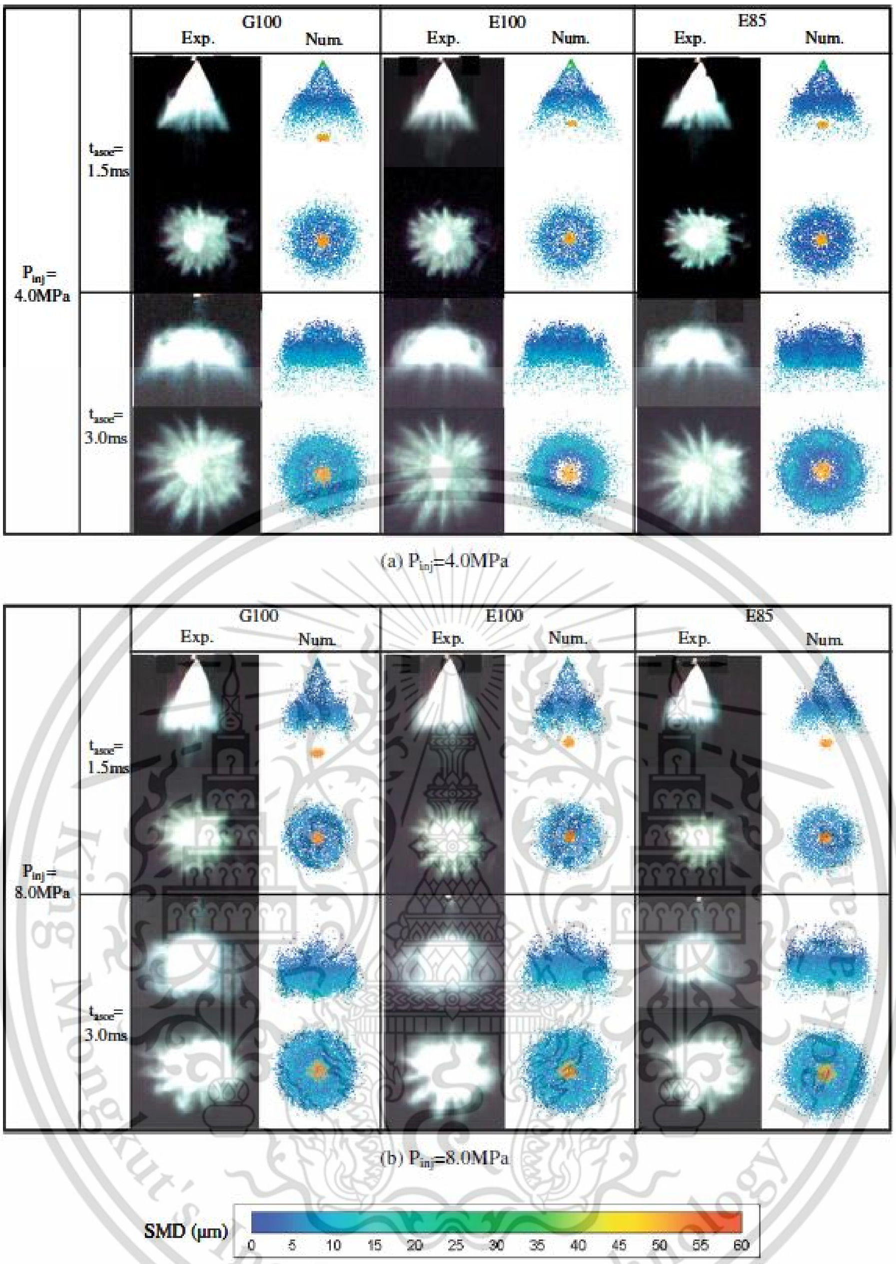


Figure 2.13 Comparison between experimental and numerical spray images as a side and bottom view [20].

Next, the combustion characteristics, Wallner et al. [21] reported as using ethanol can reduce NO_x emission from lowing combustion temperature regarding to high enthalpy of vaporization. These conform to Turner et al. [22] as ethanol can enhance combustion stability and efficiencies from better evaporation.

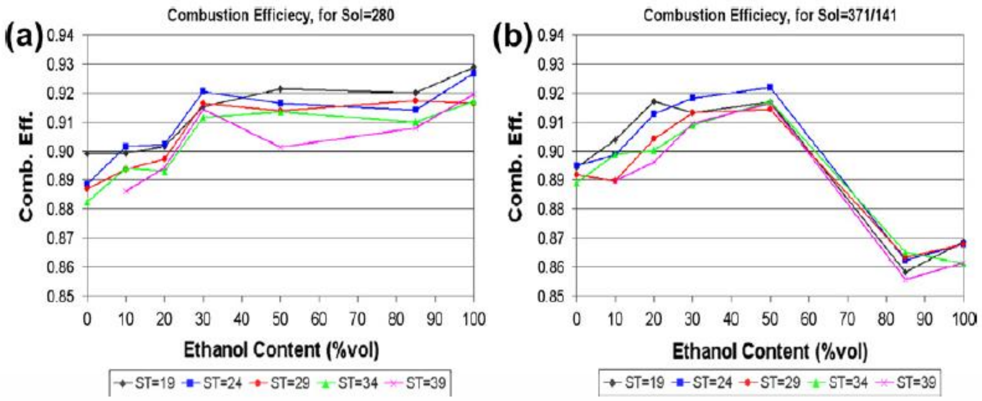


Figure 2.14 Combustion efficiency with varied spark ignition timings: (a) single and (b) split injection strategies [22].

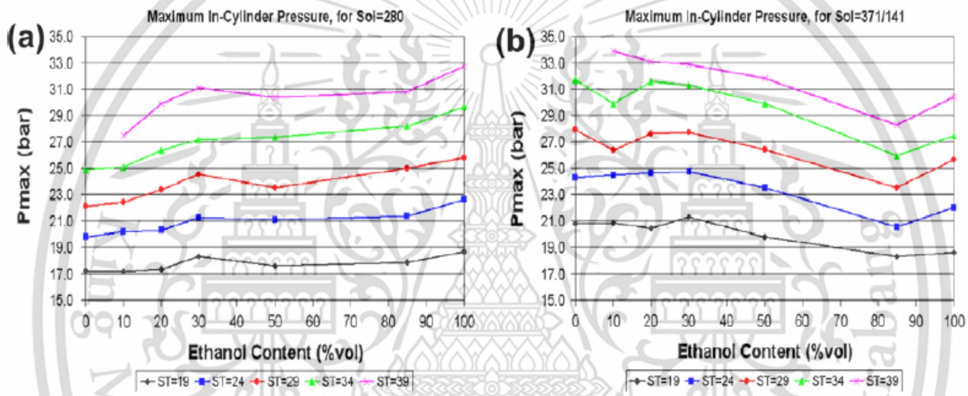


Figure 2.15 Maximum cylinder pressure with varied spark ignition timings: (a) single and (b) split injection strategies [22].

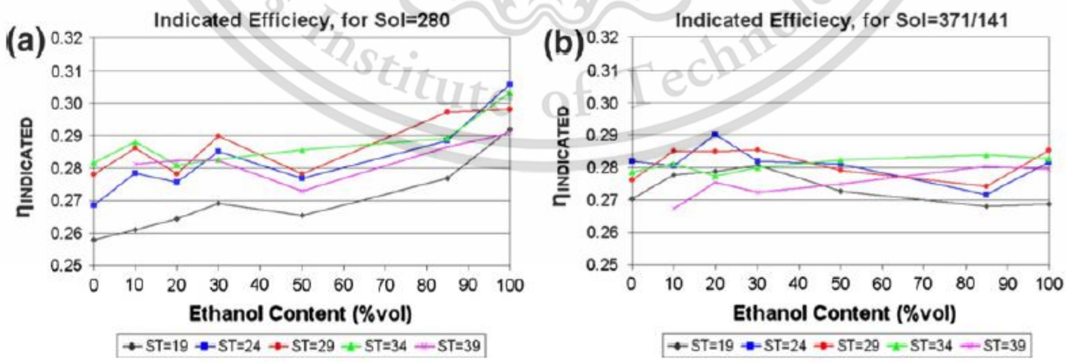


Figure 2.16 Indicated efficiency with varied spark ignition timings: (a) single and (b) split injection strategies [22].

Last in, mixture preparation and particulate matter (PM) emission from DISI engine from Longfei et al. [23] showed as at rich mixture condition can decrease PM when mount up the percentage of ethanol from oxygen inside blend fuel which can reduce intermediate species for precursors soot formation. On the contrary, PM is increased when operate on stoichiometric condition due to mixture become high heterogeneous from high heat of vaporization at the initial injection on early stroke.

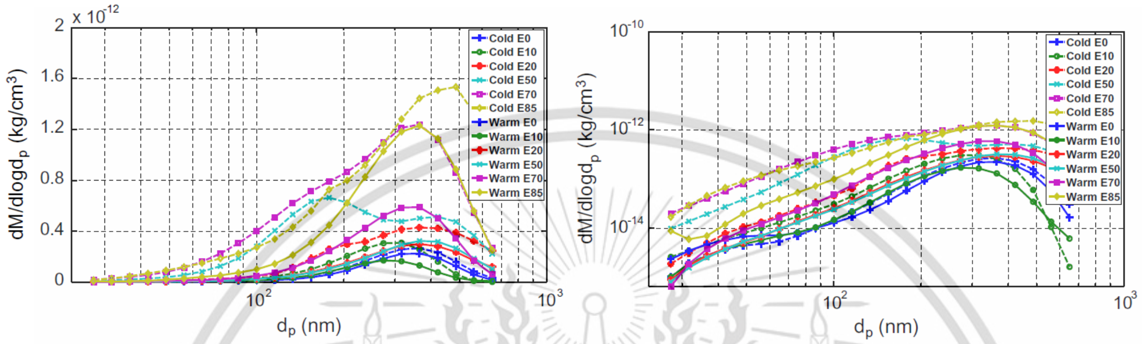


Figure 2.17 Size resolved mass concentrations of PM on a semi-logarithmic scale (top) and on a double-logarithmic scale (bottom) for different gasoline/ethanol blends in a cold (20 °C) and a warm (80 °C) engine [23].

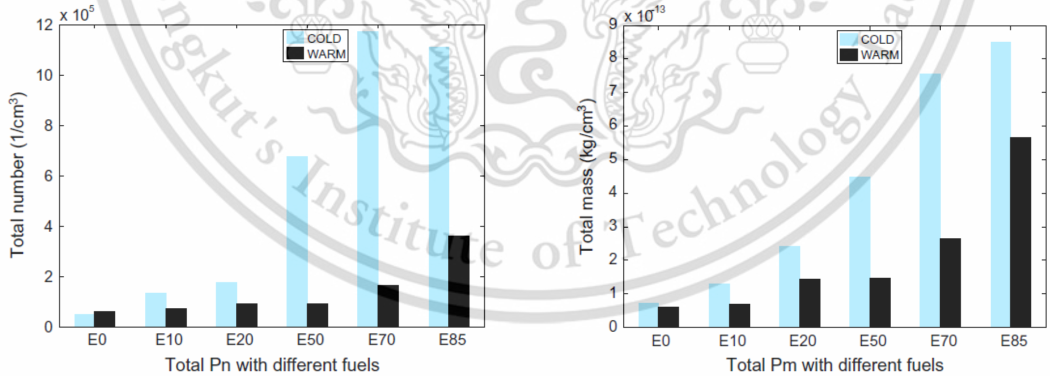


Figure 2.18 Total particulate number (top) and total particulate mass (below) emissions in a cold (20 °C) and a warm (80 °C) engine [23].

CHAPTER 3

EXPERIMENTAL AND PROCEDURE

3.1 Experimental Apparatuses and Experimental Set Up

3.1.1 Engine

The wall-guided type of direct injection spark ignition engine which fuel supply by cam-driven high-pressure pump type which pressure is approximately 0.3 MPa to maximum level (4-7 MPa). This engine is commanded by standalone engine control unit (ECU) and specification of this engine shown in table 3.1.

Table 3.1 Engine Specifications

Description	Specification
Model	MITSUBISHI 4G93 GDI
No. of Cylinder / Type	4 / In-Line,DOHC 16 Valve
Combustion Chamber	Pentroof Type
Cooling System	Water-cooled forced circulation
Displacement (CC)	1,834
Compression Ratio / Bore x Stroke (mm)	12 : 1 / 81.0 x 89.0
Intake Valve Open at BTDC	15°
Intake Valve Close at ABDC	56°
Exhaust Valve Open at BBDC	55°
Exhaust Valve Close at ATDC	15°
Maximum Output	96 kW @ 6000 rpm
Maximum Torque	177 Nm @ 3750 rpm

3.1.2 Fuel

In this research, the gasoline and ethanol blends (E20 and E85) were used. To find out the ratio between of air and fuel at stoichiometric in system is mandatory for indicate performance and efficiencies of its fuel. Those specification were indicated in table 3.2;

Table 3.2 Fuel properties [24, 25].

Fuels properties	Gasoline [24]	E10	E20	E85	Ethanol [25]
Formula	C_4 to C_{12}	$CH_{2.043}O_{0.015}$	$CH_{1.63}O_{0.065}$	$CH_{2.822}O_{0.425}$	C_2H_5OH
Molecular weight [g/mol]	100 - 105		88.12	50.60	46.70
Carbon [mass%]	85-88	86.70	79.85	55.36	52.20
Hydrogen [mass%]	12 - 15	13.2	12.88	12.89	13.1
Oxygen [mass%]	0	1.94	7.54	31.75	34.70
Density, kg/l, at 15°C	0.72-0.77	0.7608	0.7645		0.79
Vapor pres., kPa at 38°C	48-103	59.60	58.30	35-70	15.90
Lower, heating value, MJ/l	44.00	40.97	40.60	29.50	26.90
Research octane number	92.4	98.1	98.3	101.6	108.60
Motor octane number	81.2	82.3	84.6	91.1	92
$^{(R+M)}/_{2}$	86.8	90.2	91.45	96.35	100
Air/fuel ratio @ Stoichiometric	14.70	14.05	13.51	9.87	9.03
Distillation temperature, °C					
Initial boiling point, IBP	35	36.5	37.8	41.3	77.6
10 vol%	51.5	51.6	53.5	66.6	77.8
20 vol%	56.5	55.7	57.8	74.4	77.9
30 vol%	61.8	59.7	62.5	76.8	77.9
40 vol%	68.6	63.8	66.8	77.4	77.9
50 vol%	78.2	70.2	70.8	77.5	77.9
60 vol%	91.5	99.4	73.7	77.6	77.9
70 vol%	108.6	117.9	99.9	77.7	78
80 vol%	125.2	136.1	130	77.7	78
90 vol%	154	160.2	155	77.8	78
End boiling point	197.3	187.2	184.6	80.5	80
Adiabatic flame temperature, K	2266		2203.15		2197

Regarding to above table, the major difference between ethanol blends and gasoline are better antiknock quality from high research octane number (RON) and abundant of oxygen which reduce emissions. On the contrary, the blended fuel must apply more quantities for the same output due to lower heating. Furthermore, a high initial boiling point (IBP) causes a cold start problem as well.

3.1.2.1 Air-Fuel Ratio Calculation

This ratio can calculate from equation of oxidation at adiabatic flame temperature condition as the following;

$$C_xH_y + (x+\frac{y}{4})(O_2 + 3.773N_2) \rightarrow xCO_2 + \frac{y}{2}H_2O + N_2 \tag{3.1}$$

$$AFR_{Stoi} = \frac{(x+\frac{y}{4})(4.773 \cdot 29)}{(12x+y)} \tag{3.2}$$

The chemical formula of Gasoline is $C_{8.26}H_{15.5}$. Hence,

$$\left(AFR_{Stoi}\right)_{E0} = \frac{(8.26+\frac{15.5}{4})(4.773 \cdot 29)}{\left(\left(12 \cdot 8.26\right)+15.5\right)} = 14.65 \tag{3.3}$$

For E20, E85 and also E100 following equations can be used;

$$xC_2H_5OH + yC_{8.26}H_{15.5} + \left(3x+12.13y\right)(O_2 + 3.773N_2) = aCO_2 + bH_2O + cN_2 \tag{3.4}$$

Where

$x =$ mole fraction of the Ethanol

$y =$ mole fraction of the Gasoline

And

$$AFR_{Stoi} = \frac{\left(3x+12.13y\right)(4.773 \cdot 29)}{46x+114.8y} \tag{3.5}$$

The Air fuel ration of each fuel at stoichiometric shown in table 3.3 below,

Table 3.3 Air-fuel ratio at stoichiometric of each fuel

FUEL	Air Fuel Ratio	Total Molecular Weight of Fuel	LHV of Fuel ($^{MJ}/_{kg}$)
E0	14.63	114.18	44.00
E20	13.51	88.12	40.58
E85	9.87	50.60	29.47
E100	9.03	46.07	26.90

3.1.3 Exhaust Gas Recirculation System

EGR (Exhaust Gas Recirculation) Valve is controlled by National Instrument and Labview program. It is design to control percentage of EGR gas into intake pipe by analyze data from intake and exhaust oxygen sensor. The software is automatically control EGR valve.

The percentage of EGR gas is determined by “eq.1”[6]

$$\%EGR = \frac{[O_{2,amb}]-[O_{2,man}]}{[O_{2,amb}]-[O_{2,exh}]} \tag{3.6}$$

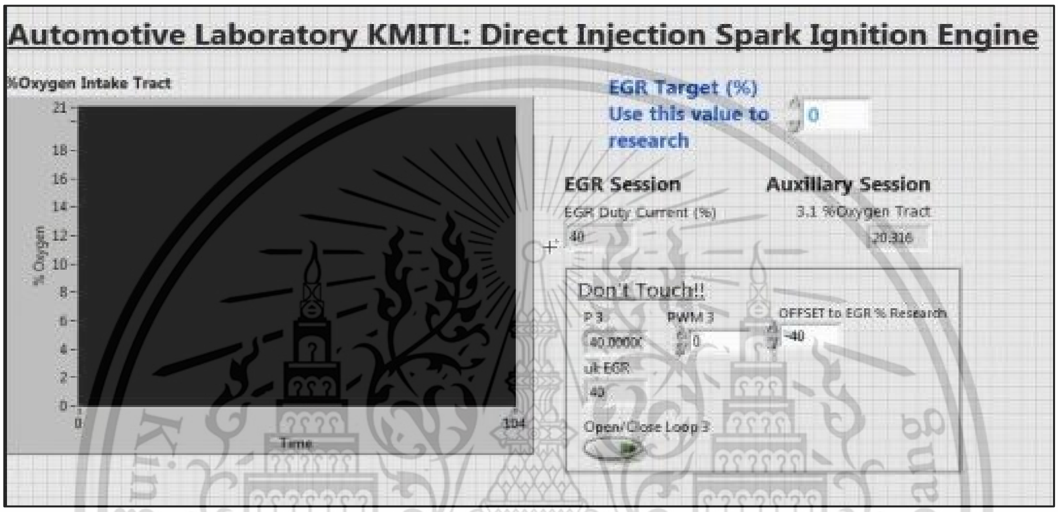


Figure 3.1 EGR controlled software

3.2 Methodology

3.2.1 Experimental System

The experimental system and schematic diagram showed in figure 3.3 and 3.4 where the specification showed in table 3.4. This engine is controlled Fuel injection and ignition of engine are controlled by DTA s60pro standalone ecu and measure real-time of performance and efficiencies together with operate in 2 mode of constant of load and speed by in-house LabVIEW program with eddy current dynamometer from Tokyo Plant ED-150-LC.

Measuring temperature on air inlet, exhaust gas, engine lubricant and coolant were conducted by type K class 1 thermocouple. The Sokken LFE25B laminar airflow meter with P277 digital manometer was measured an intake air flow is. Furthermore, the MRU - SWG200-1 gas analyzer measured emission of gas phase and OKUDA DSU-240 smoke detector was measured particle emission in term of percentage concentration (%).. All conditions were repeated for 5 times.

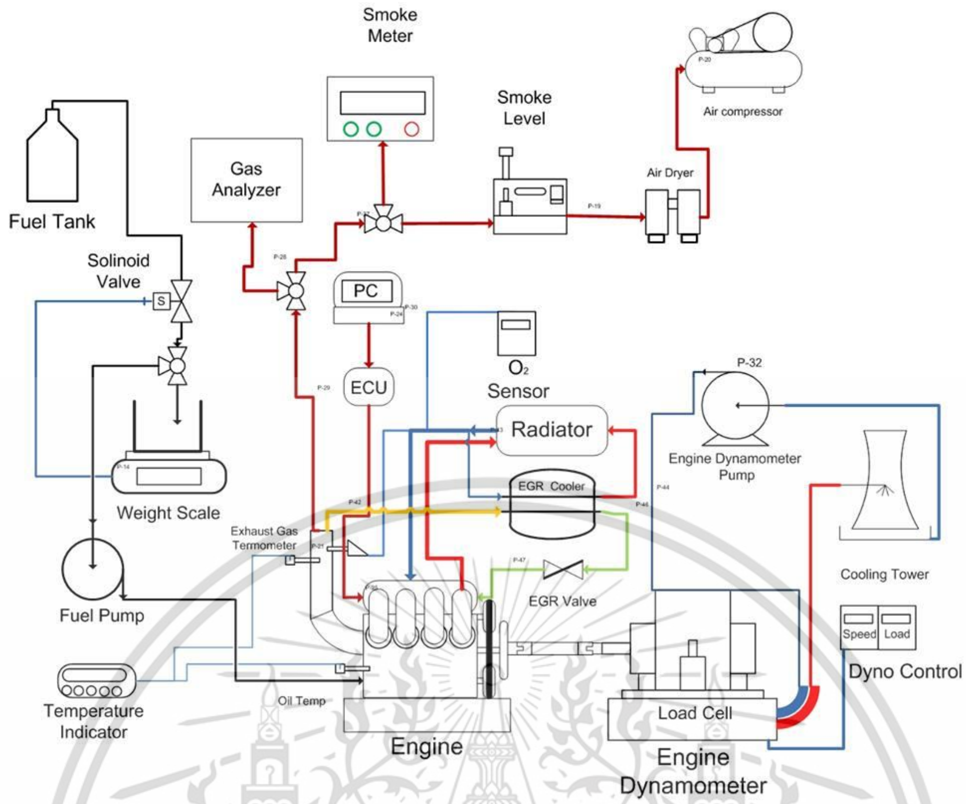


Figure 3.2 The schematic diagram of experiment

Figure 3.3 Overall display of experiment.

Table 3.4 Specification of measurement hardware

No.	Name	Unit	Description	Specification	Model
1	Engine Dynamometer	-	Generatea load, torque and power to test engine.	• Eddy current dynamometer • Max B.H.P. 150 P.S. / 3000 R.P.M. • Max Brake Torque 35.81 Kg.m • Max R.P.M. 8000 R.P.M.	Tokyo Plant ED-150-LC
2	Engine Control Unit (ECU)	-	Standalone electronic control unit of experimental engine which controls quantity, duration of fuel injection and spark timing with precision resolution.	• 20,000 rpm capability • Flexible and easily adapted to different OEM, crank, cam sensor arrangements. • Genuine four and two stroke support. • Twin spark engines. • Twin injector engines. • Sequential or grouped injection.	DTA s60pro
3	DAQ	-	To control, measure and store datum from engine dynamometer.	-	National Instrument LabVIEW system.
4	Thermocouple	^{o}C	Measurement - Ambient - Air inlet at manifold - Exhaust Pipe - Coolant - Lubricant	• Range: -40 – 375 ^{o}C • Tolerances: ±1.5 ^{o}C	Type K Class 1

No.	Name	Unit	Description	Specification	Model
5	Oxygen Sensor	-	Measures the proportion of oxygen (O_2) in the gas or liquid analyzed.	• Application: lambda 0.65 to ∞ • Fuel compatibility: gasoline/Diesel/E85 • Exhaust gas pressure: ≤ 2.5 bar (higher with decrease accuracy) • Exhaust gas temperature range (operating): < 930°C • Exhaust gas temperature range (max.) for short time: < 1,030°C • Hexagon temperature: < 600°C • Wire and protective sleeve temperature: < 250°C • Connector temperature: < 140°C • Storage temperature range: -40 to 100°C • Max. vibration (stochastic peak level): 300 m/s2	BOSCH LSU 4.9
6	Digital Air/Fuel Ratio Meter	AFR And λ	Monitoring the oxygen concentration from exhaust gas which can use to predict combustion characteristics in combustion process.	• Patented "Direct Digital" Wideband Technology • Wideband O2 Compatible with ALL fuel types • Single or Dual Channel A/F Version Available • OBD-II Scan tool- read/clear DTCs and log up to 16 channels of CAN OBD-II Data • Log directly to SD card • Playback log data on screen and/or with powerful LogWorks software (included) • Large high-contrast graphics LCD • Built-in RPM converter (direct frequency or with optional inductive clamp) • 4 fully-differential analog inputs • 2 configurable linear analog outputs • Positive lock connectors for all connections • Innovate MTS serial IN and OUT • USB connection to PC	INNOVATE LM-2

No.	Name	Unit	Description	Specification	Model
7	Laminar air flow meter	g/s	Measuring inlet rate of mass airflow.	• Pressure range : 2.5 kPa to 70 MPa • Accuracy : ±0.3% of full scale • Analogue output : 0-10V / 0- full scale • Power supply : 100V AC±10%, 30VA • Dimensions : W130 x H120 x D230	Sokken laminar air flow meter (LFE25B) with digital manometer indicator (PZ77)
8	Smoke Meter	%	Measuring the concentration of the particulate matter in the exhaust gas.	-	OKUDA DSM-240
9	Gas Analyzer	ppm	Measuring gaseous emissions.	• O_2 0 – 25.00 % • CO 0 – 4,000.00 ppm • CO_2 0 – 20.00 % • C_xH_y 0 – 5,000 ppm as C_3H_8 • NO 0 – 2,500 ppm • NO_2 0 – 500 ppm • Resolution 1 ppm respectively 0,1% • Detection limit 1 ppm respectively 0,1% • Repeatability ± 1% FS • Linearity ± 1% FS • Analog output up to 8 channel 4 - 20 mA • Digital output RS485 for long distance data transfer	MRU (SWG200-1)

3.2.2 Experimental Conditions

In this study, the experiment was measured performance, efficiency and emissions from inline 4 cylinders, 4 strokes and 1,834 cm^3 displacement of direct injection spark ignition engine by using gasoline (E0) and commercial ethanol blended with gasoline (E20 and E85) at 2000 R.P.M. under 30% loads (Brake mean effective pressure 0.3 MPa) represents as urban driving conditions.

The injection operation was controlled in stratified charge mode. The injection and ignition timing are varied for maximum brake torque (MBT) between 17° to 32° CAD BTDC with fine 3 steps of ignition timing and 80° to 110° CAD BTDC with fine 5 steps of injection timing.

The injection timing is fixed at maximum brake torque (MBT) for each fuel. The EGR gas was introduced to the engine up to 25 percent. All conditions conducted at lambda (λ) = 1. The test condition portrays in table 3.5.

Table 3.5 Testing conditions

Description\Fuel	Gasoline (E0)	E20	E85
Engine Speed (R.P.M.)		2000 ± 20	
Ambient Temperature (°C)		30 ± 5	
Coolant Temperature (°C)		87 ± 3	
Engine Load		30% Load (BMEP 0.3 MPa)	
Engine Operation		Stratified Charge	
Lambda (λ)		1	
Injection Timing (CAD BTDC)		80° - 110°	
Ignition Timing (CAD BTDC)		17° - 32°	
EGR rate (%)		0-25	

Regarding to section 3.2.2, to determine the engine load condition should base on car manufacturer which test engine is installed. The experimental engine is manufactured by Mitsubishi Motors. These is installed in Mitsubishi Lancer, model year 2003 which has specification as the following.

Table 3.6 Car manufacturer specification

MODEL	Mitsubishi Lancer Model year 2003
Dimension	
WIDTH (mm)	1,695
HEIGHT (mm)	1,430
Curb WEIGHT (kg)	1,200
Drag Co-efficient	0.3
Tire	185/65/R14
Wheel Diameter (mm)	596
Engine Specification	
Horse Power (kW) / RPM.	84/5500
Torque (Nm) / RPM.	155/4000
Transmission Specfication	
1st	2.319
2nd	1.62
3rd	1.26
4th	1.00
5th	0.7
6th	0.445
Final Drive	5.219

After car specification is purposed, the surroundings condition such as vehicle speed, pavement conditions were based on Thailand regulations which shown in table 3.7.

Table 3.7 Surroundings condition

Target	
1. Speed ($^{km}/_{h}$), [2]	90
2. Engine Efficiency (%)	90
3. Road: Fair Pavement = K_r	0.019
4. Road Gradient (%), [3]	4
5. Air Density ($^{kg}/_{m\text{^}3}$)	1.2

Remarks: Vehicle speed limit in Thailand based on [26] and road gradient refers to [27].

From table 3.6 and 3.7, total resistance consist of air resistance, rolling resistance and gradient resistance. These can determined by equation as the following and results showed in table 3.8 and 3.9 respectively.

First, Air Resistance calculated by,

$$R_a = \frac{1}{2}\rho_a \vec{v}^2 A C_d \tag{3.6}$$

$$A = W \cdot H \cdot 0.8 \tag{3.7}$$

Where

R_T	Total resistance (N)
ρ_a	Density of air
$\vec{v}$	Velocity of car ($^{m}/_{s}$)
A	Cross section of frontal area (m^2)
C_d	Drag co-efficient
W	Overall width of car (mm)
H	Overall height of car (mm)

Following with Rolling Resistance,

$$R_r = K_r \cdot w \tag{3.8}$$

$$w = m \cdot g \tag{3.9}$$

Where

R_r	Rolling resistance (N)
K_r	Co-efficient of rolling friction
w	Weight of car (N)

Then, Gradient Resistance….

$$R_g = \frac{wG}{1000} \tag{3.10}$$

Where

R_g	Gradient resistance (N)
G	Road gradient (%)
w	Weight of car (N)

Last in, Total Resistance….

$$R_T = R_a + R_r + R_g \tag{3.11}$$

Where

R_T	Total Resistance
R_a	Air Resistance
R_r	Rolling Resistance
R_g	Gradient Resistance

Table 3.8 Resistances calculation

1. Air (R_a)	
Cross-Section Area (m^2)	1.94
So, Air resistance (N)	218.15
2. Rolling Resistance (R_r)	
From Curb Weight (N)	11,772
So, Rolling Resistance	223.67
3. Gradient (R_g)	47.09
Hence, Total Resistance (R_t,N)	**488.90**

From table 3.8, an output from engine can calculated by equation as the following

Starting with, engine power

$$P_e = \frac{100 \cdot R_T \cdot \vec{v}}{\eta_e} \tag{3.12}$$

Where

P_e	Engine power (kW)
R_T	Total resistance (N)
$\vec{v}$	Velocity of car ($^{m}/_{s}$)
η_e	Engine efficiency (%)

Following with torque, these can categorize into 2 sections as torque from wheel and torque form engine as seen in equation below,

$$T_w = R_T \cdot r \tag{3.13}$$

Where

T_w	Torque from wheel (N)
R_T	Total resistance (N)
r	Wheel radius

$$T_e = \frac{100 \cdot T_w}{\eta_e \cdot i_g \cdot i_f} \tag{3.14}$$

Where

T_e	Torque from engine (N)
T_w	Torque from wheel (N)
η_e	Engine efficiency (%)
i_g	Gear ratio (In this calculation, 5th gear ratio is selected as cruising operation)
i_f	Final drive ratio

Table 3.9 Power and torque of engine determination

Power	13.58 (kW)
Torque	
1. At Wheel	145.69 (N)
2. At Engine	**44.31 (N)**

44.31 N, torque from engine, is which is validate for testing condition.

3.2.3 Experimental Methods

Performance and efficiency cope with emissions are both methods for validate in this research which classified system into 2 sections as below;

3.2.3.1 With-out EGR System

3.2.3.1.1 Performance and Efficiency

Figure 3.5 showed the controlling the ratio of air fuel become stoichiometric (lambda = 1). After that, adjusting both of timings and injection duration on maximum brake torque.

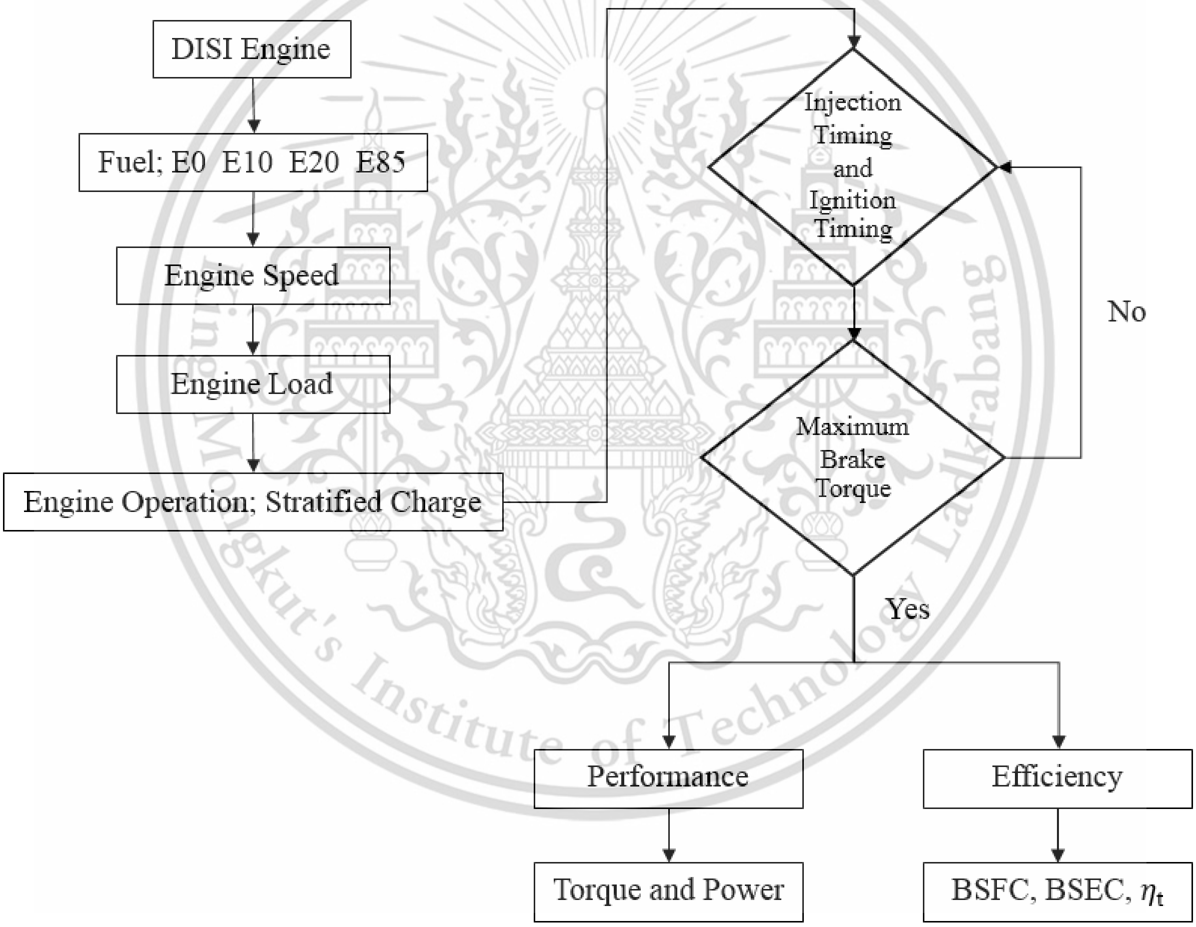


Figure 3.4 Diagram of performance and efficiency without EGR system.

3.2.3.1.2 Emissions

Operating method same as 3.2.3.1.1. Emissions of gaseous phase; Oxide of nitrogen (NO_x), Total hydrocarbons (THC) and Carbon monoxide (CO) sampled by gas analyzer indicated in figure 3.6.

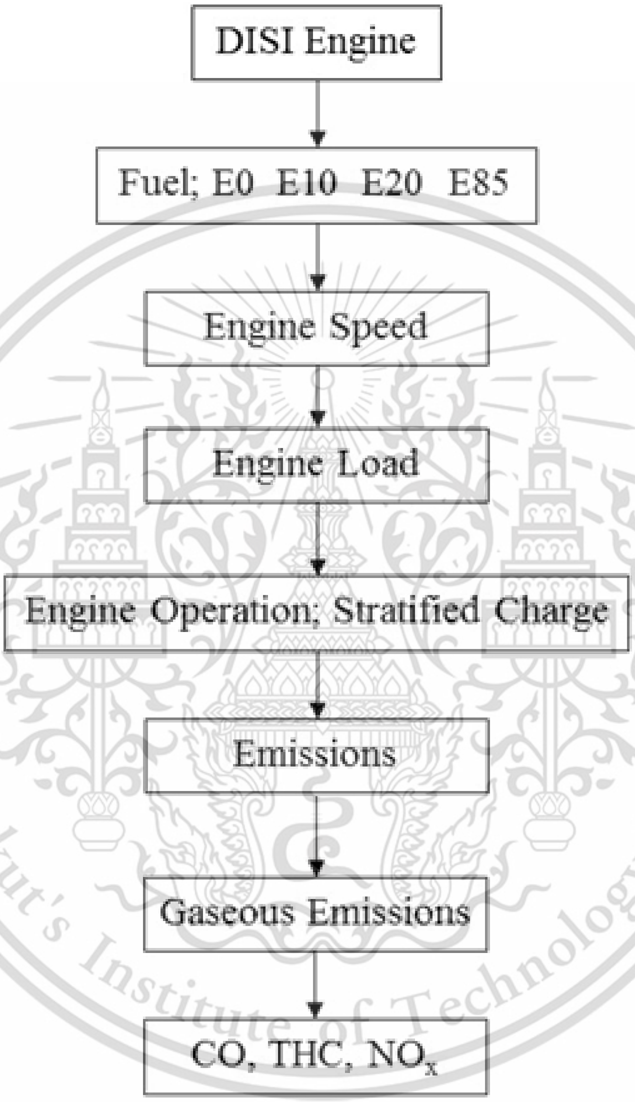


Figure 3.5 Diagram of emissions without EGR system.

3.2.3.2 With EGR System

3.2.3.2.1 Performance and Efficiency with Gaseous Emissions

The injection timing fixed at the maximum brake torque (without EGR). Then, the EGR gas was entrained into the intake manifold. The injection duration was adjusted to keep the air fuel ratio at stoichiometric. The study parameters were measured by adjust ignition timing at maximum brake torque as shown in flow chart in figure 3.7

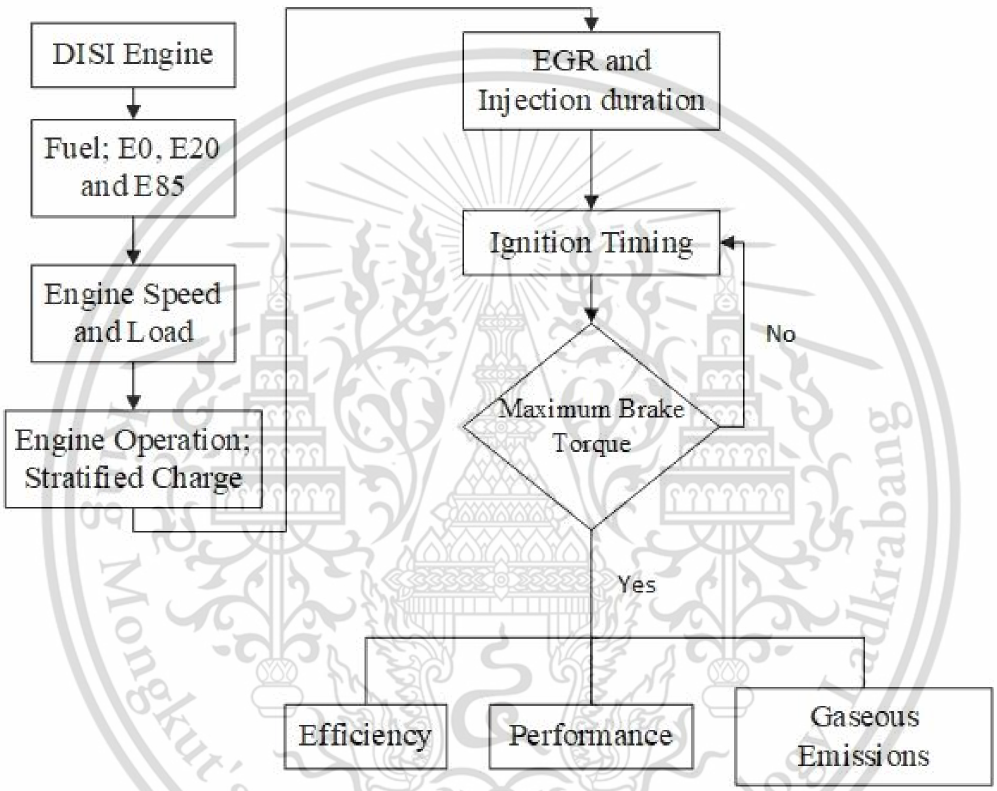


Figure 3.6 Performance, Efficiency and emissions with EGR system.

CHAPTER 4

RESULTS AND DISCUSSIONS

4.1 Impact of Injection Timing and Ignition Timing

4.1.1 Efficiencies and Emissions

The impact of injection and ignition timing explain in term of the maximum brake torque timing (MBT). The optimum timing of each tested fuel showed in table 4.1.

Table 4.10 The optimum of injection and ignition timing on each fuel

FUEL	Injection Timing (CAD BTDC)	Iginition Timing (CAD BTDC)
E0	100°	27°
E20	100°	27°
E85	105°	30°

From above table imply as the ethanol blend could adjust both of timing to advance than gasoline according to characteristics of ethanol; lower heat value—obtain amount of fuel for similar output, high heat of vaporization–benefits to high volumetric efficiency and high octane number —increase an ignition delay. The result of all tested fuel showed a familiar tendency as THC depends on the behavior of combustion, FSN is reduced by too advance and a retard from MBT, NO_x is maximum at the MBT timings and BSFC together with BSEC become fewer efficiencies when timings are too advance than a retard.

Regarding to the result from figure 4.1 to figure 4.7 can summarize the impact of injection strategies into 4 sections as following. Starting with retard of both injection and ignition timing generate the process of combustion at late of operation cycle result in cylinder peak pressure and temperature of combustion are reduced.

Following with retard of injection timing and advance of ignition timing, the lack duration of mixture causes the high concentration of mixture at the local air-fuel ratio. Then, the emissions (CO, THC and FSN) become extremely high.

Next, the advance of injection timing and retard of ignition timing. The strategy impacts as extend duration of mixture formation from high interval of end of injection (EOI) and start of spark (SOS) causes the local mixture AFR too lean. Hence, the tendency of efficiencies and emissions are similar as both advance timings.

Last in the advance of both injection and ignition timings result in an improper mixture formation—lean zone and burning at the partial zone which extend the delay angle. Hence, the oxidation rate of THC is lower that impacts to THC, CO and FSN are increased whereas NO_x is lower from combustion temperature.

The results of efficiencies on injection timing are shown in Figure 4.8. The efficiencies are most effective at optimum timing from an ideal mixture stratification, which results in the highest flame propagation and the lowest ignition delay. However, the late injection timing produces adequate mixture formation, causing the mixture around the spark plug cannot form into a rich mixture. Furthermore, the advance of injection timing brings about mixture in that zone is too lean. Hence, the effect of those timing leads to an unfavorable mixture to ignite and propagate the flame.

In case of injection timing on emission on figure 4.9, the advance timing brings mixture form too homogeneous. These effect to mount up of THC and CO but lower of NO_x from lower combustion temperature during expansion-exhaust stroke that slow down the oxidation of THC as well.

The impact of ignition timing on efficiencies is shown in Figure 4.10. Too early or late of ignition timing from MBT leads lower of efficiencies especially in retard of timing extend the formation of mixture become more homogeneous or rich than proper.

According to the impact of ignition timing on emissions in Figure 4.11. The advance of ignition timing brings about higher fuel formation from an interval of the end of fuel injection and the initial a spark. This implies that all emissions are lower. On the other hand, the retard ignition timing strategy reduces cylinder pressure and temperature in a power stroke. Hence, low combustion temperature occurred post combustion process which result in rise up of THC, CO and NO_x.

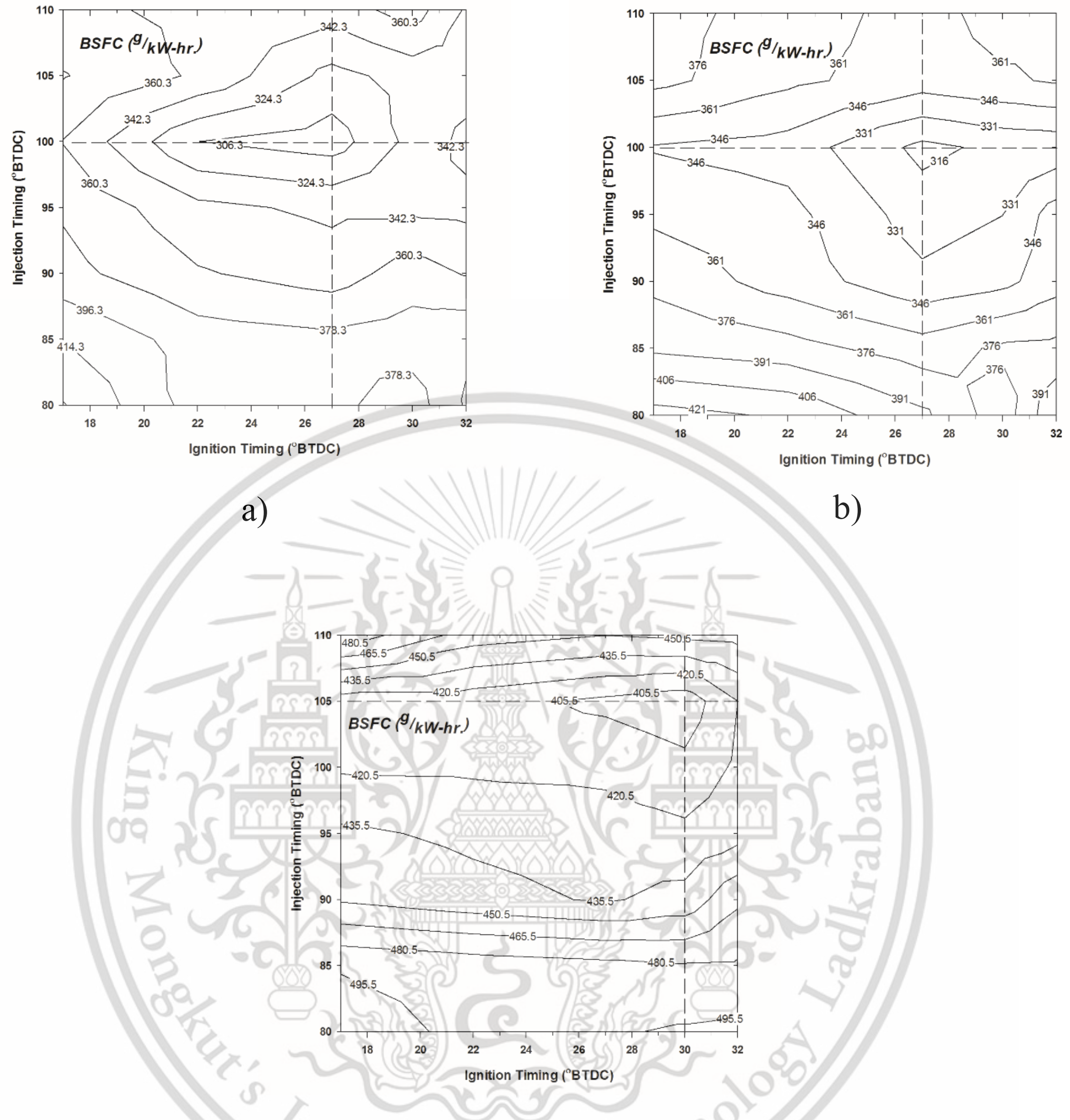


Figure 4.1 Impact of injection timing and ignition timing of BSFC of each fuel as a) E0, b) E20 and c) E85.

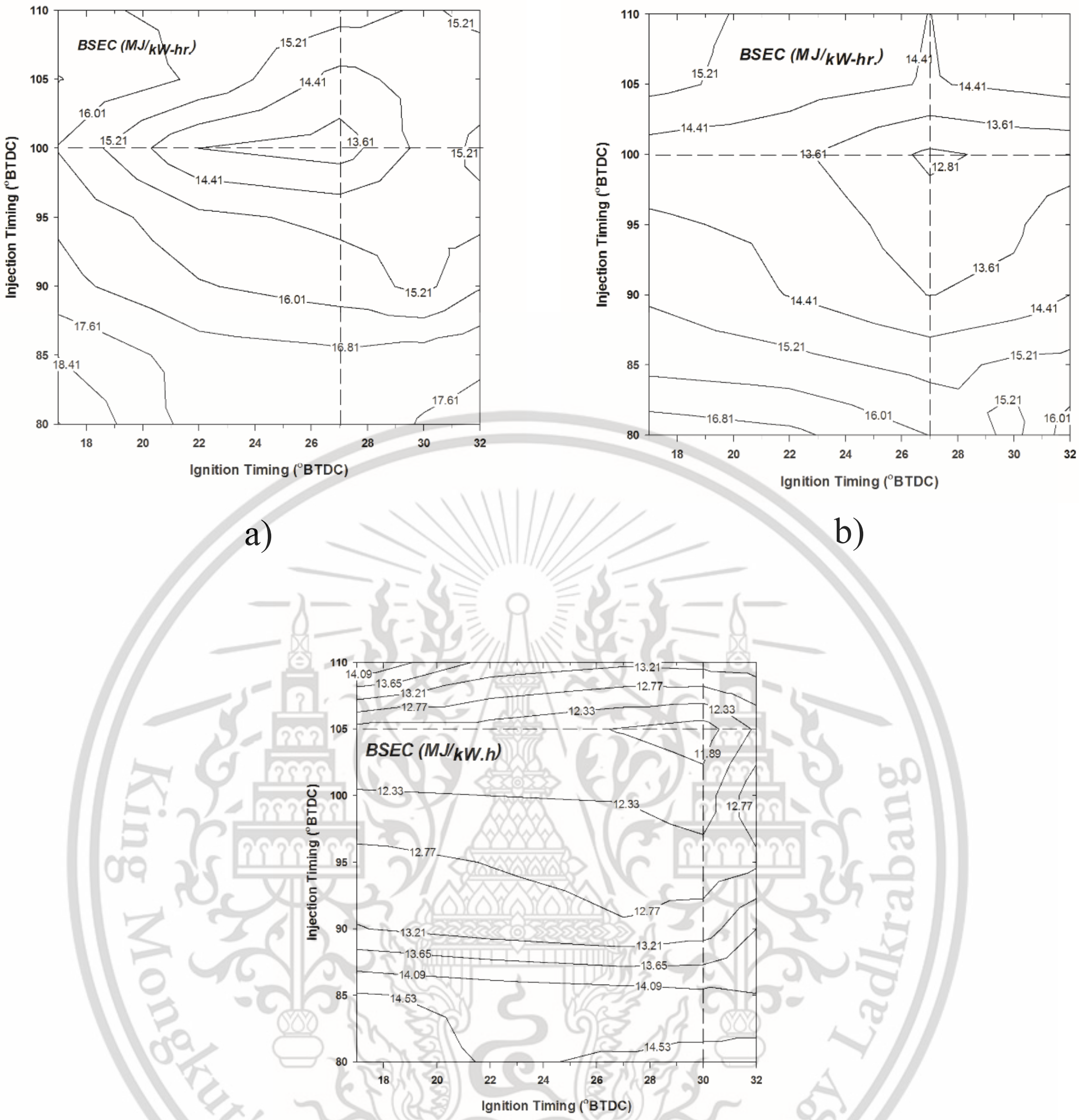


Figure 4.2 Impact of injection timing and ignition timing of BSEC of each fuel as a) E0, b) E20 and c) E85.

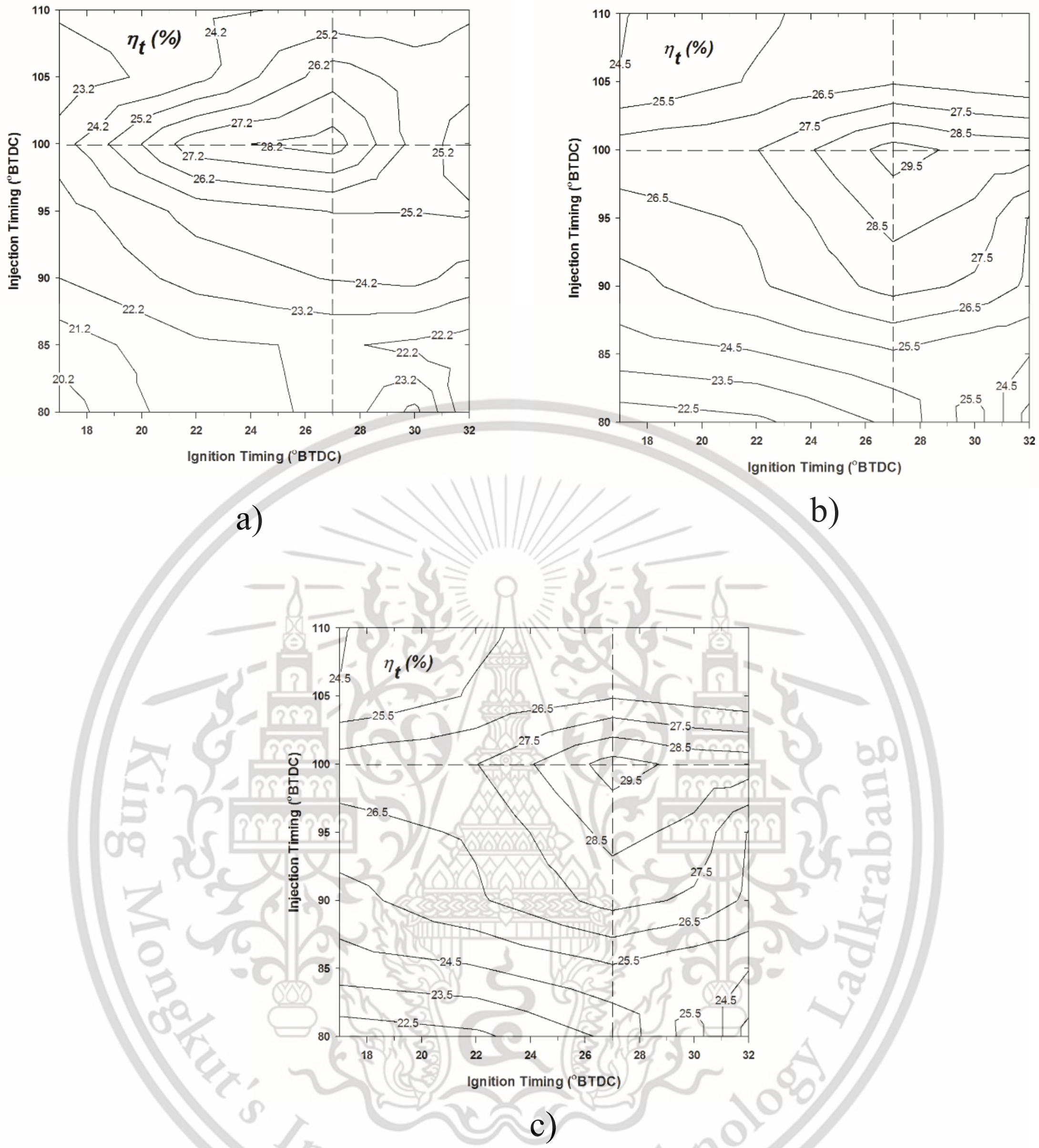


Figure 4.3 Impact of injection timing and ignition timing of thermal efficiency of each fuel as a) E0, b) E20 and c) E85.

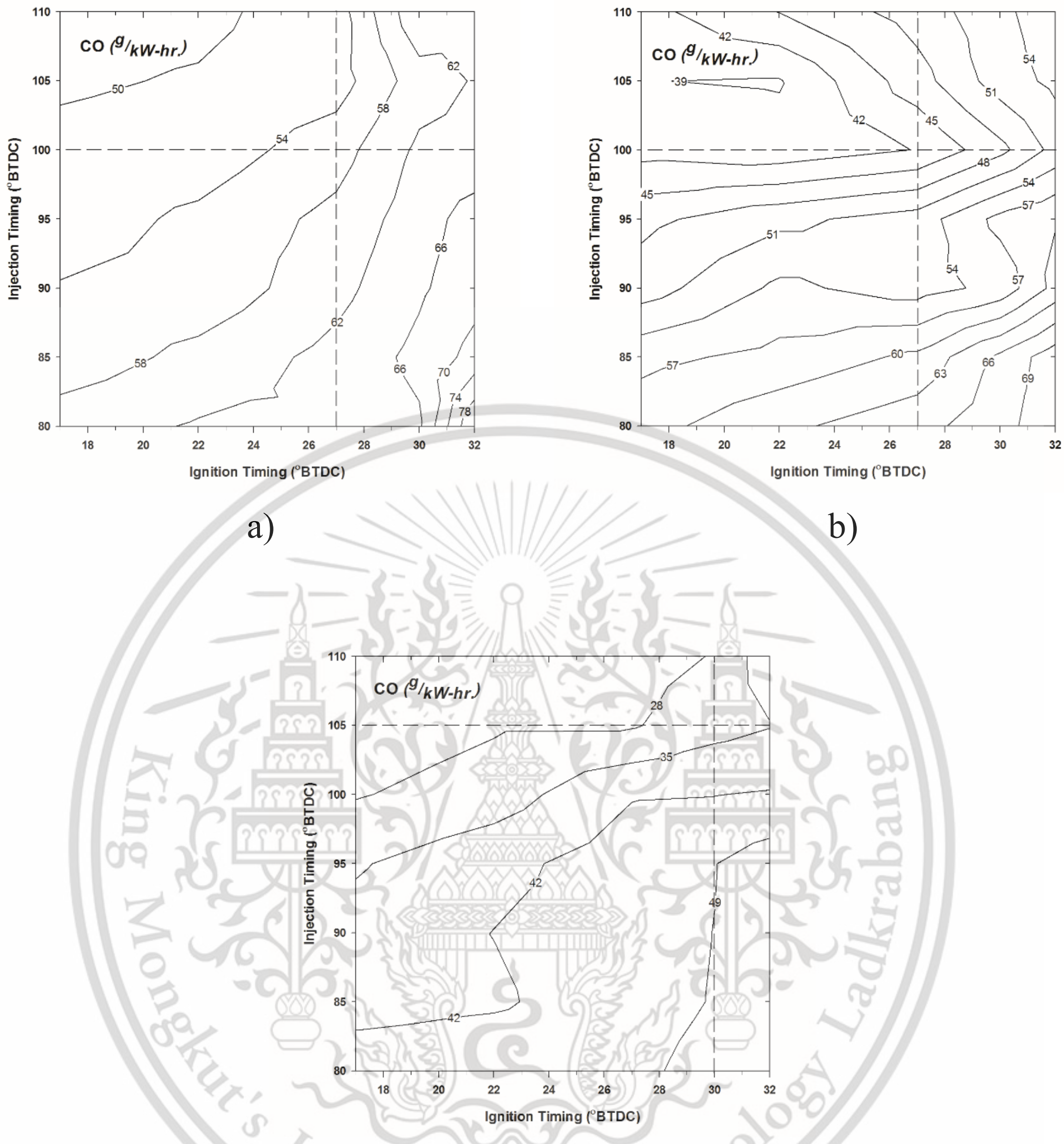


Figure 4.4 Impact of injection timing and ignition timing of emission - CO of each fuel as a) E0, b) E20 and c) E85.

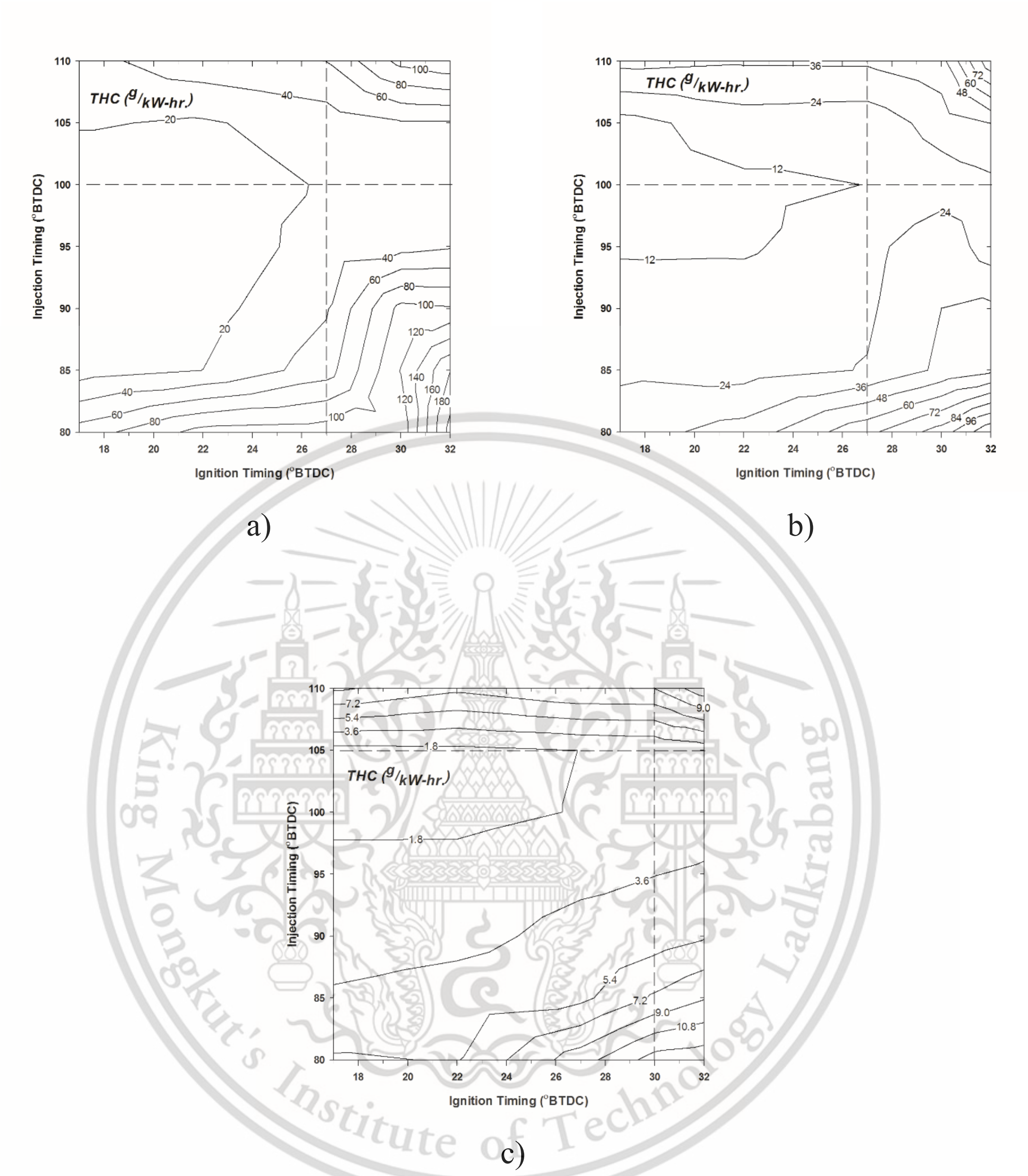


Figure 4.5 Impact of injection timing and ignition timing of emission - THC of each fuel as a) E0, b) E20 and c) E85.

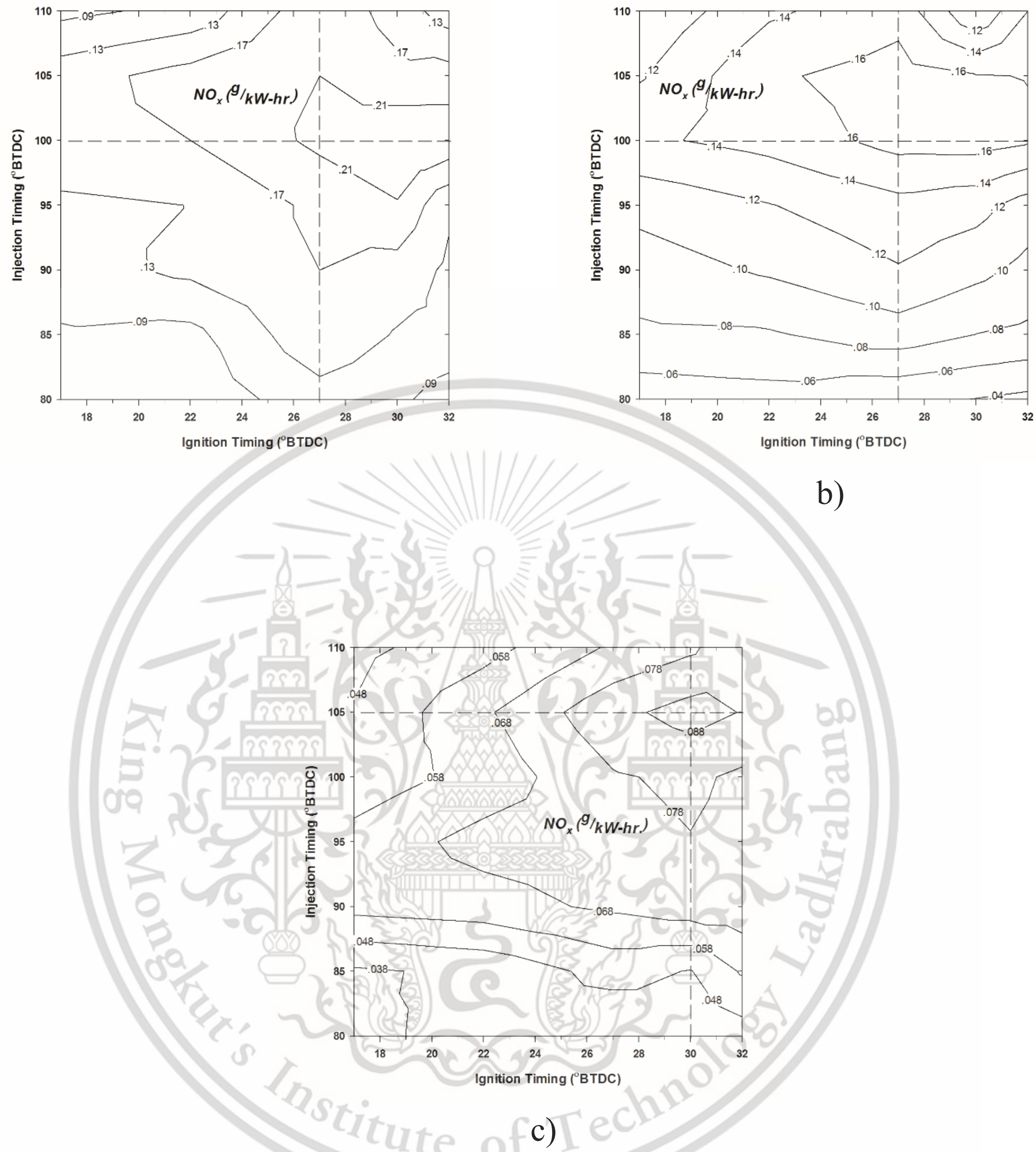
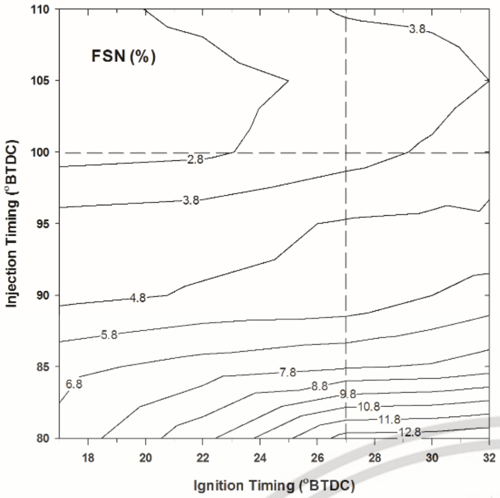
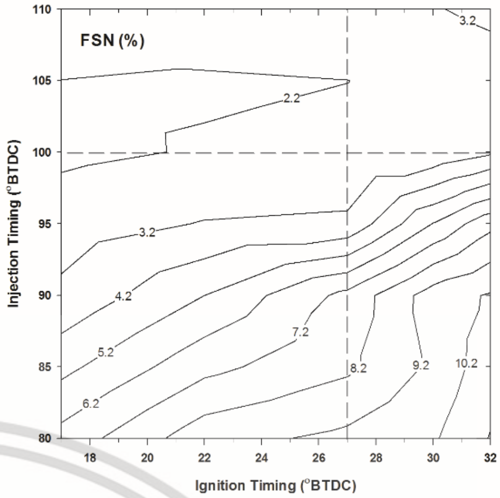


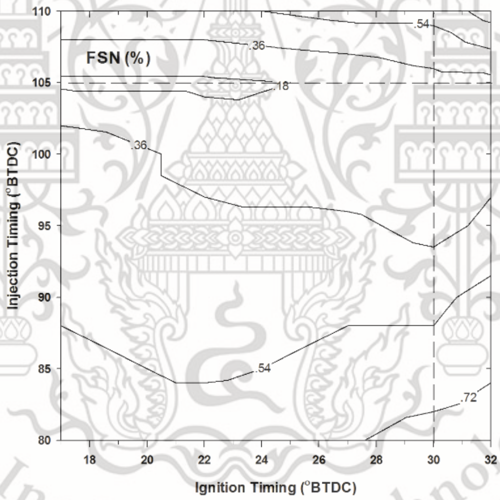
Figure 4.6 Impact of injection timing and ignition timing of emission - NO_x of each fuel as a) E0, b) E20 and c) E85.



a)



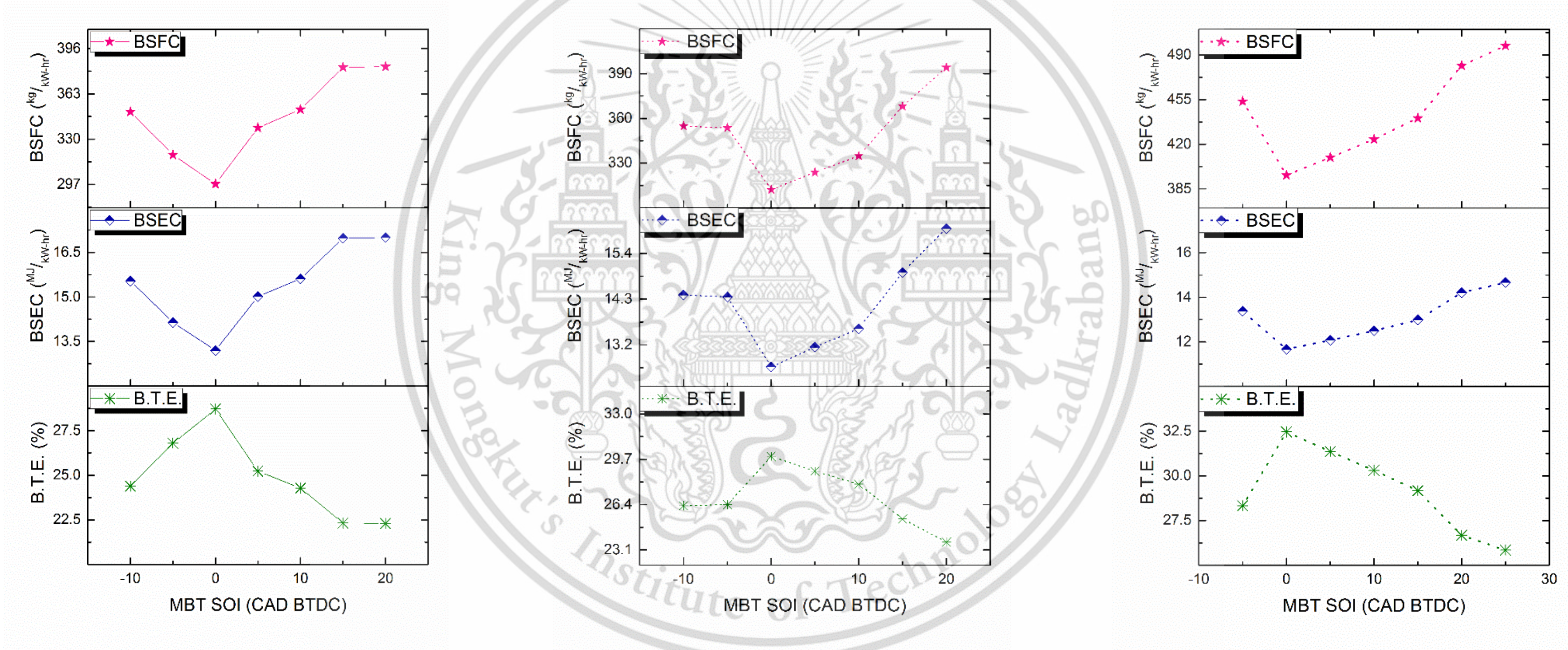
b)



c)

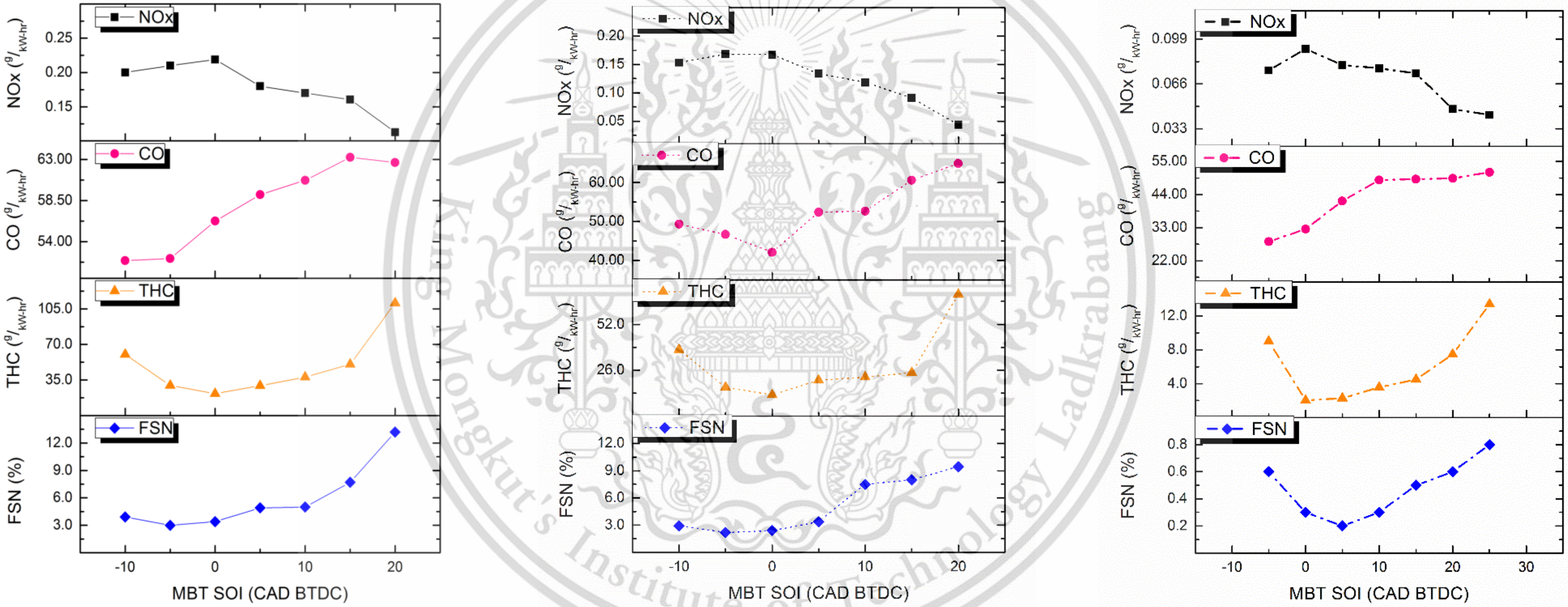
Figure 4.7 Impact of injection timing and ignition timing of emission - NO_x of each fuel as a) E0, b) E20 and c) E85

Figure 4.8 Impact of injection timing at MBT on efficiencies of each fuel as a) E0, b) E20 and c) E85.



a) b) c)

Figure 4.9 Impact of injection timing at MBT on emissions of each fuel as a) E0, b) E20 and c) E85.



a) b) c)

Figure 4.10 Impact of ignition timing at MBT on efficiencies of each fuel as a) E0, b) E20 and c) E85.

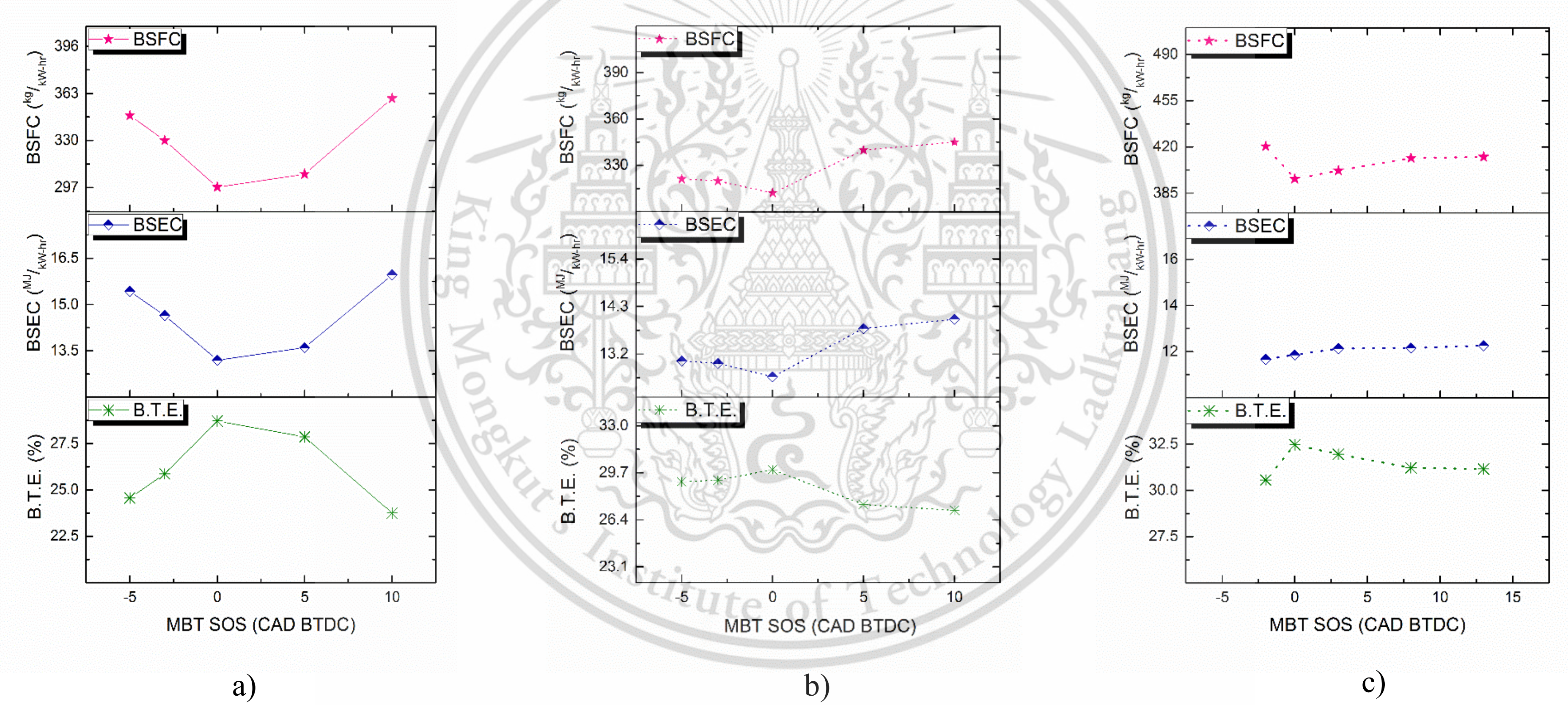
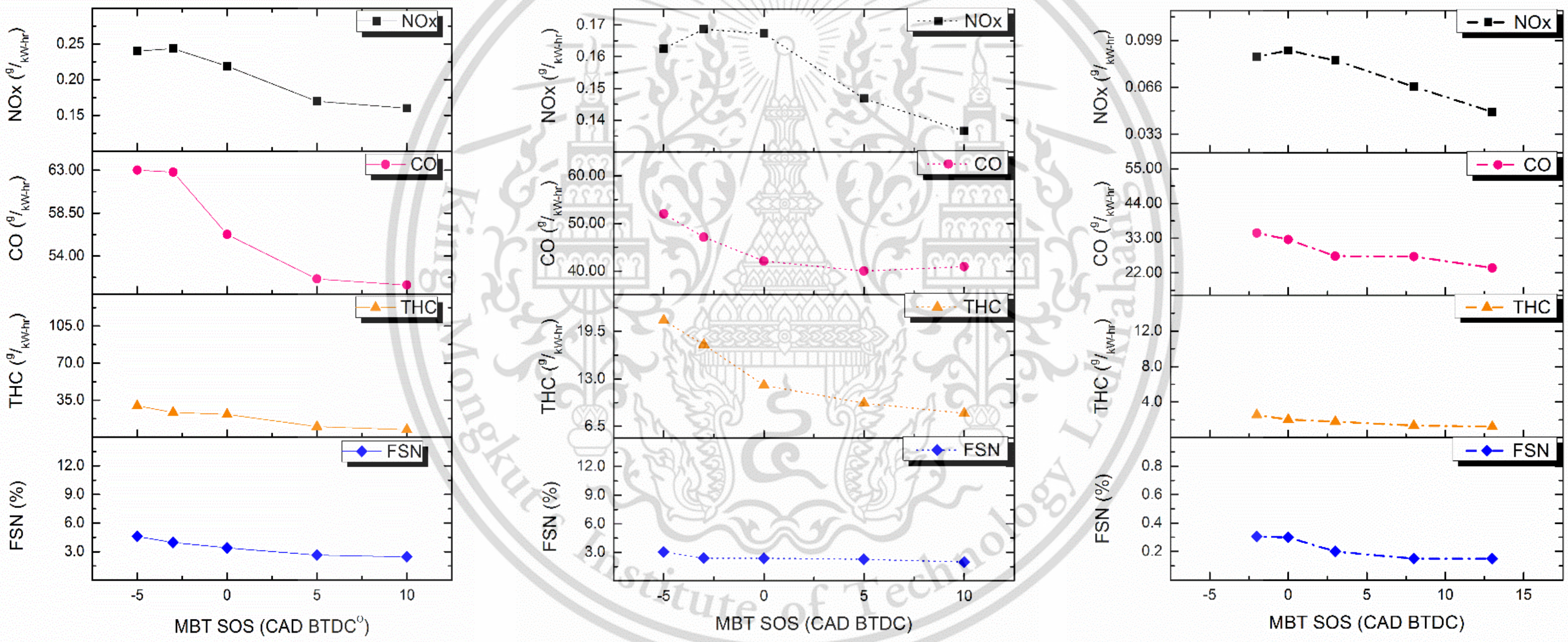


Figure 4.11 Impact of ignition timing at MBT on emissions of each fuel as a) E0, b) E20 and c) E85.



a) b) c)

4.2 Impact from Ethanol Blends

Figure 4.12 showed the results of gasoline and ethanol blends in terms of efficiencies at the optimize timings on ignition and injection of each fuel at the same ratio of air and fuel. Beginning with the brake specific fuel consumption; The high percentage of ethanol leads to more injection of fuel to compensate for the lower heating value. These leads the highest blends (E85) increased 33.12% following with 4.98% of E20 after compare with gasoline. These results conform with the basis calculation of energy from fuel supply to 1 kilogram of air as a mass of fuel multiply with low calorific value which shown in table 4.2.

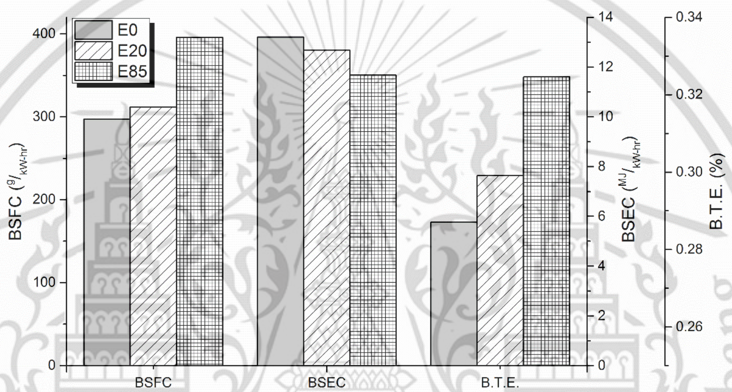


Figure 4.12 The efficiencies of ethanol blends (E20 and E85) and gasoline

Table 11 The determination of total energy input of each fuel

FUEL	Total Energy Input (J)
E0	3.009×10^6
E20	3.003×10^6
E85	2.980×10^6

However, when discuss in case of identify the efficient of energy which obtained from each fuel. The brake specific energy consumption (BSEC) is selected to clarify these phenomena. To blend a higher percentage of ethanol delivers lower brake specific energy consumption owing to a lower total energy input. These result E85 and E20 decreased -11.85%, 4.01% when compared with gasoline.

Last in the brake thermal efficiency (B.T.E.), the E85 earned highest rate following with E20 as +13.06% and 4.18% respectively according to oxygenated of blended fuel impact to combustion characteristics by enhance oxidizing rate result in the more complete combustion accompany with a higher rate of heat release which reduce loss to surroundings. This implies that ethanol obtains higher efficiencies than gasoline.

4.3 Effect from Exhaust Gas Recirculation

4.3.1 Effect on Ignition Timing for optimum point

When the exhaust gas was recirculated Figure 4.13 shows the effect of EGR ratio on ignition timing. The optimum points were selected by the minimum BSFC of each condition because of the reduction of combustion pressure and temperature, allowing the ignition timing to be advanced.

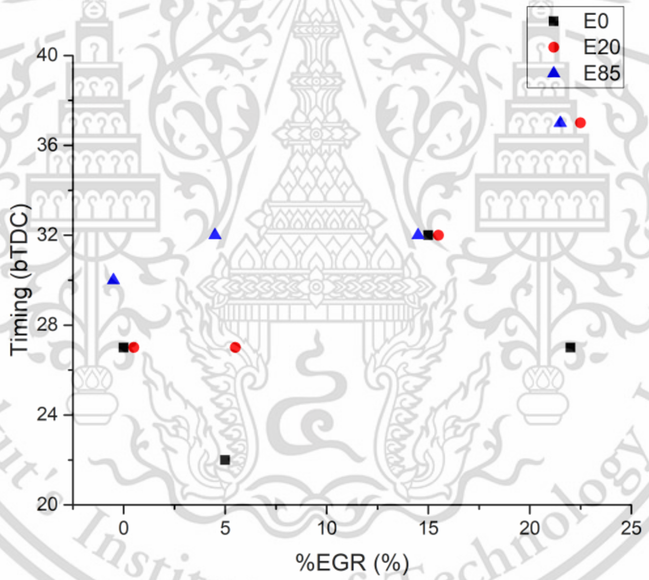


Figure 4.13 The effect of EGR ratio on ignition timing.

4.3.2 Efficiencies

4.3.2.1 Brake Specific Fuel Consumption (BSFC) and Brake Specific Energy Consumption (BSEC)

Figure 4.14 shows the effect of the EGR ratio on BSFC. The BSFC improved by 8%, 6% and 3% at 5% of EGR ratio for E0, E20 and E85 respectively. After that, it increased gradually by the EGR ratio. On the other hand, The E85 show the minimum energy consumption in every condition and 5% of EGR still consumed less energy of each fuel as shown in figure 4.23.

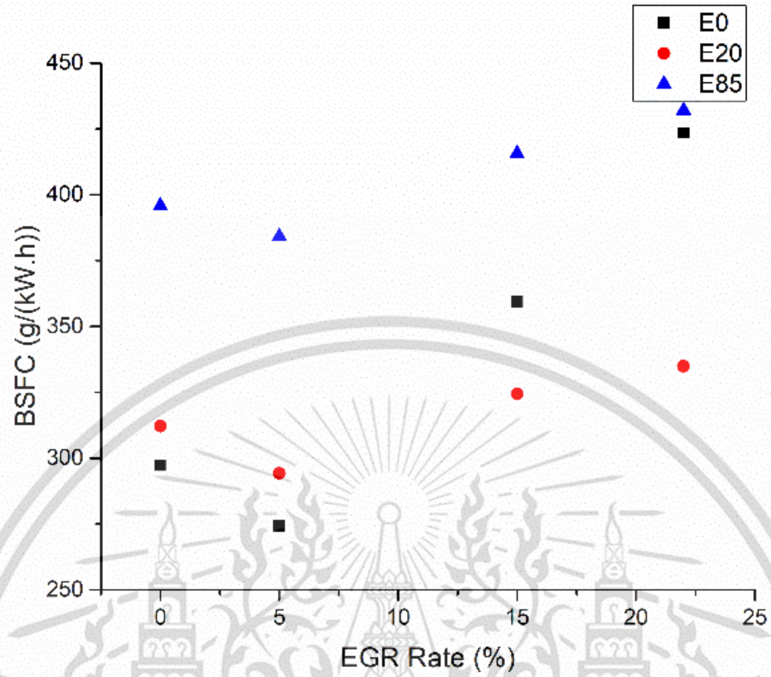


Figure 4.14 Effect of EGR on BSFC.

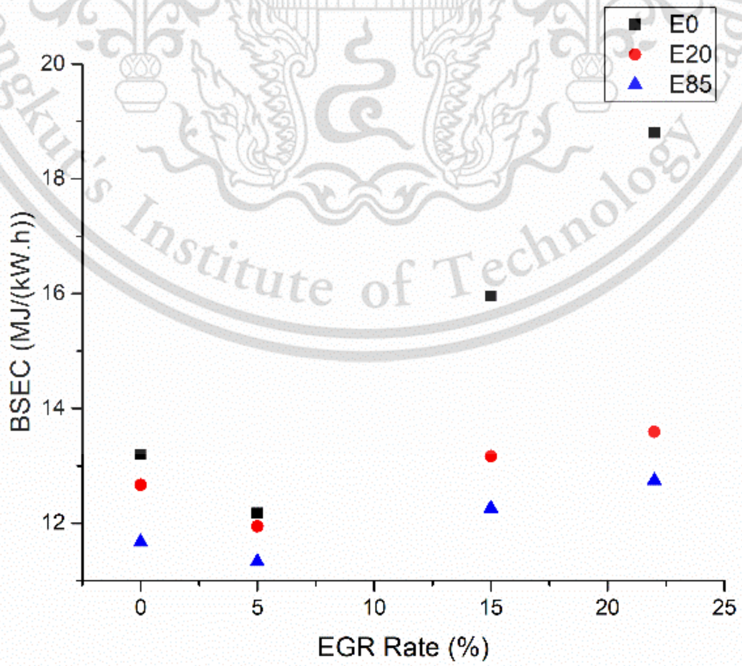


Figure 4.15 Effect of EGR on BSEC.

At 5% of EGR addition shows the minimum BSFC and BSEC, due to the reduction of pumping work decreases, which is the same result of the opening throttle valve. [17] It shows the increasing of BSFC and BSEC when the EGR addition was increased up to 15% and 22%. This because not only the pumping loss was reduced but the 50% of mass fraction burn was retarded from the optimum point [28].

4.3.2.2 Brake Thermal Efficiency (BTE)

Figure 4.16 shows the effect of EGR ratio on brake thermal efficiency (BTE.). As the EGR the flame travelled at lower speed ratio increased, thermal efficiency also slightly increased. When compared to 0% EGR ratio the BTE. Increased by 8%, 6% and 3% for E0, E20 and E85 respectively. And more EGR addition make it slightly decreased for E20 and E85 fuel. And dramatically decreased with E0 fuel.

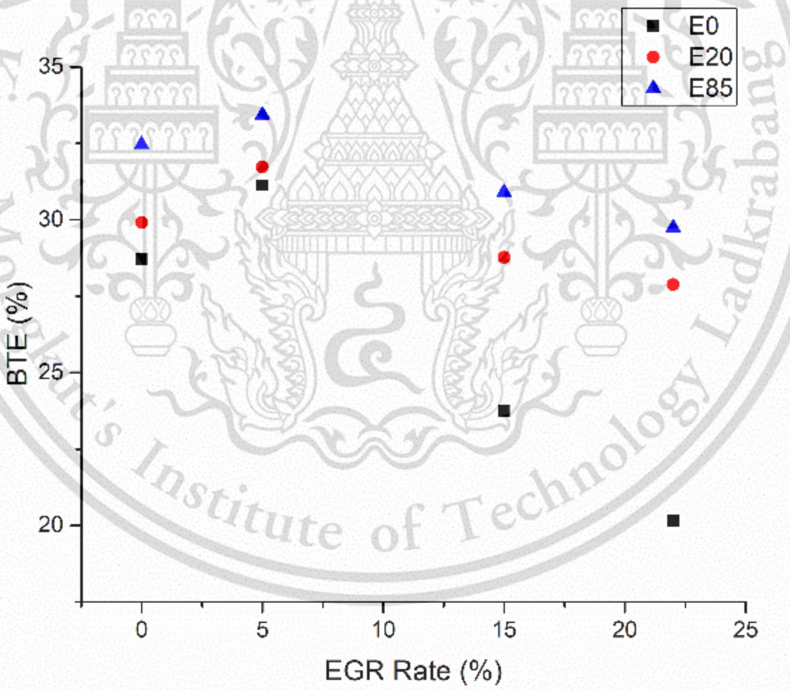


Figure 4.16 Effect of EGR on BTE.

4.3.3 Emissions

4.3.3.1 Carbon Monoxide (CO)

The CO emissions also shows the same trend as the HC emissions. The higher amount of CO emitted from the combustion process while the higher EGR gas was introduced into an intake manifold. Due to the increased of EGR gas ratio, there were not adequate amount of oxygen to oxidize CO and form CO_2 compounds [29].

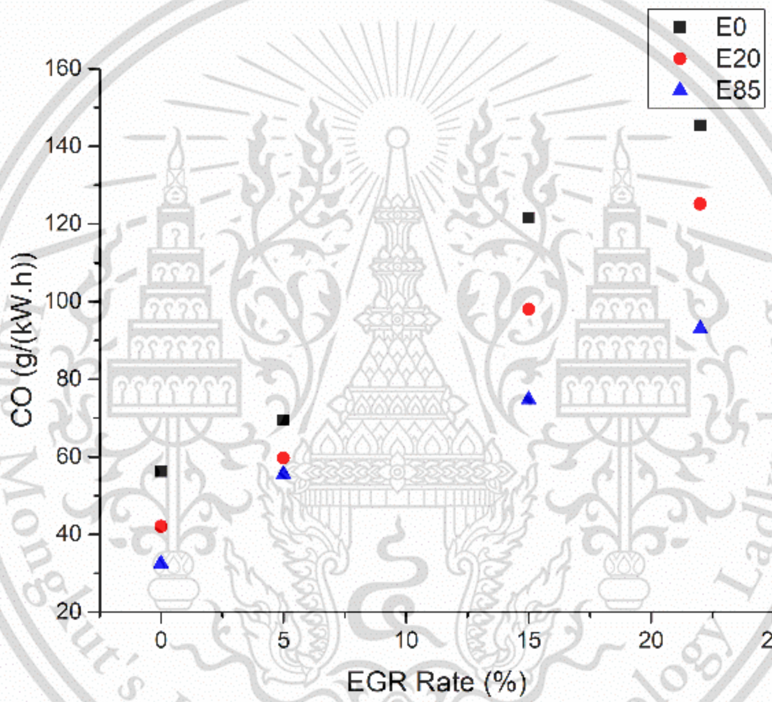


Figure 4.17 Effect of EGR on CO emissions.

4.3.3.2 Oxide of Nitrogen (NO_x)

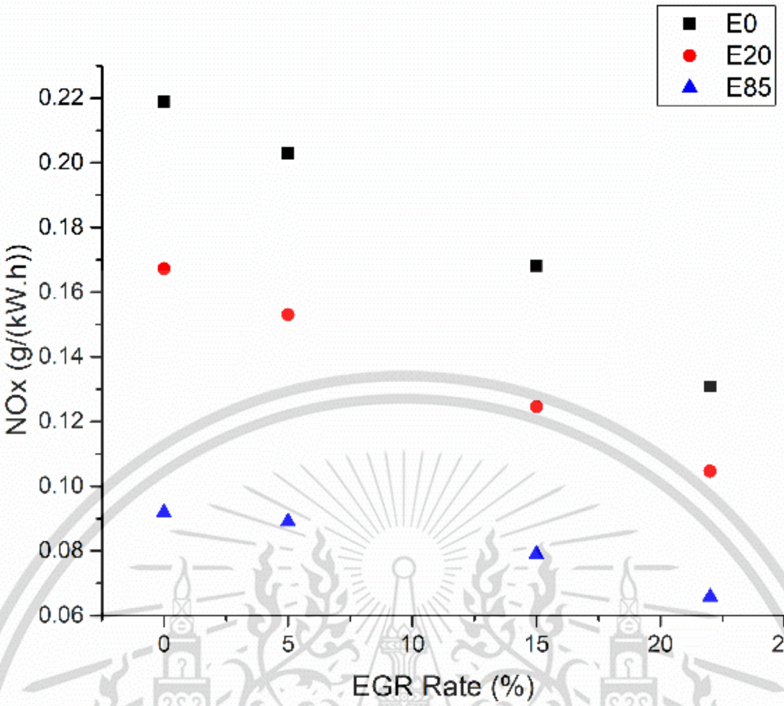


Figure 4.18 Effect of EGR on NO_x Emissions.

Figure 4.18 shows the effect of EGR ratio on NO_x emissions. NO_x emissions decreased significantly as the ethanol increased due to the higher heat vaporization. NO_x emissions also decreased when EGR gas was added. The average reduction for 5%, 15% and 22% of EGR dilution are 6%, 21% and 35% respectively. The NO_x formation decreased when EGR gas was added into an intake manifold.

The main constituents of EGR are N_2, H_2O, O_2 and CO_2. It is well known that CO_2 exerts three effects when being introduced into the combustion process [30].

Starting with, thermal effect, Exhaust gas consists of gases of two-atom and three-atom, and the heat capacity of three-atom gas increases faster in the combustion process. Due to the increase in the heat capacity of the oxidizer, the flame temperature is reduced. Secondly, Dilution effect, which results from the reduction of oxygen concentration in the mainstream of the oxidizer and from the reduction of reactive species in a combustion process, which, in turns, reduces their collision frequency. Lastly, chemical effect, since CO_2 is an active species and thus participates chemically in the combustion process.

4.3.3.3 Total Unburned Hydrocarbon (THC)

HC emissions grew rapidly when EGR gas increased. Lower THC was observed as the ethanol concentration increased. Because reduction of the heat release rate, combustion pressure and combustion temperature. Lower combustion temperatures lead to decreased reactivity resulting in a larger quenching area. Moreover, it slows down the oxygenated rate of unburned HC during expansion and exhaust process. [17, 29, 30, 31, 32].

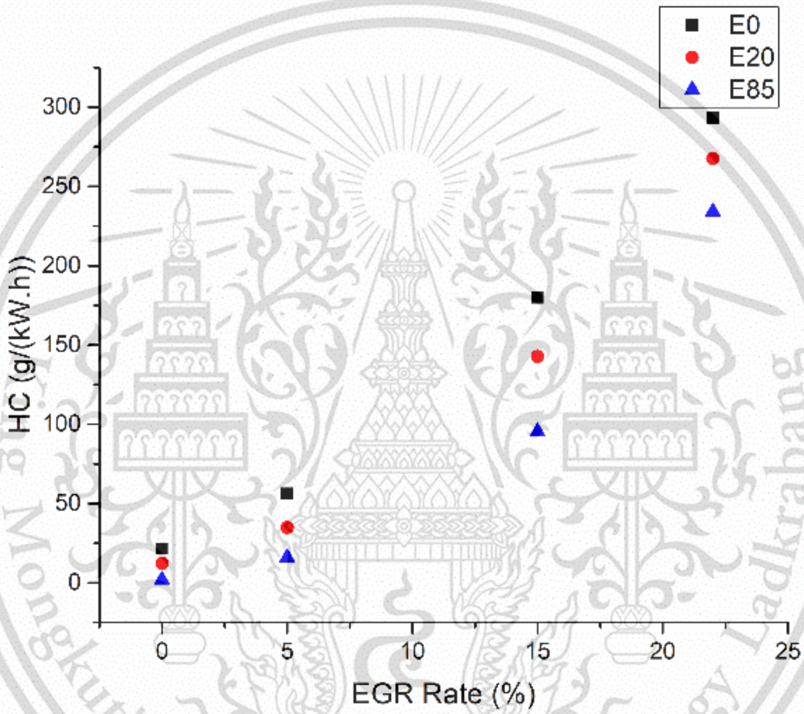


Figure 4.26 Effect of EGR on THC emissions.

CHAPTER 5

CONCLUSIONS

5.1 Ethanol Fueled in Gasoline

To compare with gasoline, fueling ethanol in gasoline impacts directly on efficiencies and emissions. Starting with efficiencies, the benefits of faster speed laminar flame – reducing duration stage of combustion accompanied with more octane number – more advance CAD of ignition and injection timings cause lower of brake specific energy consumption and greater of brake thermal efficiency.

BSEC.	:	E85 -11.85% and E20 -4.01%
BTE.	:	E85 +13.06% and E20 +4.18%

Next to emissions, the oxygenated atom inside fuel which leads to a more complete of combustion together with the characteristic of high heat of vaporization results in a reduction of combustion temperature. These factors of emission can reduce significantly after blended with ethanol.

CO	:	E85 -42.17% and E20 -25.16%
THC	:	E85 -90.64% and E20 -43.99%
NO_x	:	E85 -58.02% and E20 -23.06%

5.2 Impact of timings: Injection and Ignition

Both of timings relate directly to formation of air-fuel especially in stratified charge mode by later injection timing from optimum causes a reduction of fuel evaporate in the mixture at surrounding of a spark plug. Hence, efficiencies and emissions are severe due to remaining of fuel on the head of cylinder and wall of piston.

5.3 Exhaust Gas Recirculation

The effect of exhaust gas recirculation on fuel consumption and emissions of direct injection spark ignition engine was investigated in this paper. With 4 stroke DISI engine operated in 45 Nm at 2000 rpm. Fuel was injected at a compression stroke, stratified charge mode. The cooled exhaust gas was introduced in to an intake manifold and the fuel injection were adjusted to maintain the mixture at the stoichiometric ratio.

Ignition timing was adjusted to achieve the minimum fuel consumption the emissions data were recorded, and the following conclusions have been made:

5.3.1 Consumption and Efficiency: with the appropriate EGR ratio make the lowest both fuel consumption and energy consumption by 8%, 6% and 3% at 5% of EGR ratio for E0, E20 and E85 respectively. Due to the reduction of pumping loss. If the higher exhaust was added, the combustion duration went longer, and the engine consumed more fuel. And low consumption makes more efficiency.

5.3.2 NO_x emissions: As the EGR rate increased NO_x emissions reduced. The average reduction for 5%, 15% and 22% of EGR dilution are 6%, 21% and 35% respectively. Due to the increase in the heat capacity of the oxidizer, the flame temperature was reduced and the dilution effect reduced the oxygen concentration.

5.3.3 CO and THC emissions: Both shown the opposite trend compared to NO_x emissions. The lower combustion temperature decreased the reactivity of unburned THC during the expansion and exhaust process. THC emissions increased by 50% at the maximum BTE. With the dilution effect of EGR, the lower amount of oxygen is not enough to oxidize CO and form CO_2. The CO emissions increased up to 300% when the BTE reach the minimum point.

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APPENDICES

APPENDIX A: Fuel Distillation

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

Certificate of Analysis

Product : Gasoline E 0

Certificate No. : T-12/29294
Sample Lab No. : OP-G3H-1230331
Customer/Supplier : International College, King Mongkut's Institute of Technology
International College, King Mongkut's Institute of Technology Ladkrabang,
Chalongkrung Rd. Ladkrabang BKK 10520
Sample Location : LKB
Batch No. : -
Product Source : -

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

Sample Condition : Normal

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

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

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Certificate of Analysis

Product : E 10

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

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

Sample Condition : Normal

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

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(Phurita Pothisuk)
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Certificate of Analysis

Product : E 20

Certificate No. : T-12/29296
Sample Lab No. : OP-GSH-1230333
Customer/Supplier : International College, King Mongkut's Institute of Technology
International College, King Mongkut's Institute of Technology
Ladkrabang,
Chalongkrung Rd. Ladkrabang BKK 10520
Sample Location : LKB
Batch No. : -
Product Source : -

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

Sample Condition : Normal

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

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Product : E 85

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

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

Sample Condition : Normal

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

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TEL. +66(0)2239-7148 FAX. +66(0)2239-7149 WWW.PTTPLC.COM

Certificate of Analysis

Product : E 100

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

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

Sample Condition : Normal

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

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APPENDIX B: Publication

AEC0023
The Effect of Exhaust Gas Recirculation in Direct Injection Spark Ignition Engine Using Ethanol Blended Fuel

Rattapoom Keskangam[1*], Preechar Karin[1], Chinda Charoenphonphanich[2], Watanyoo Phirote[2], Pongsak Kummool[2], Nuwong Chollacoop[3] and Hidenori Kosaka[4]

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

Abstract

This research is explain about the investigation on DISI (Direct Injection Spark Ignition) engine using ethanol blends as fuel and EGR(Exhaust Gas Recirculation) system. With a huge number of transportation nowadays, there are global crisis with the fossil fuel usage and the limitation of fossil fuel in the world, Although, CO, HC, NOx and PM are the emission made from the fossil fuel used in the internal combustion engine. This problem is something we have to solve and make it better for human life in the present and also in the future. The combination of ethanol alternative fuel and EGR technique on part load can be directly effect to internal combustion, it improve Brake Specific Fuel Consumption (BSFC) and emission while power and torque remain constant.

This research is experiment on the adaptation of EGR system, mostly use in diesel engine to DISI engine. EGR is dilute air intake in engine part load operations. The result of EGR system is an improvement of Brake Specific Fuel Consumption(BSFC), anti-knocking limit and decrease combustion temperature which lead to NOx emission with lean burn mixture in DISI engine using ethanol blends as fuel.

Keywords: Ethanol, Exhaust Gas Recirculation, Emission.

1. Introduction

The combustion engine is the most using engine in the world which use fossil fuel as the main energy. Human have a serious problem due to the amount of fossil fuel usage for the transportation all around the world. Since the rate of decreasing of fossil fuel is a huge number. Moreover, other serious problem that effect on environment is the emission release from the internal combustion engine using fossil fuel. Internal combustion engine produce emissions called, "CO, CO2, HC, NOx and PM (particulate matter). Most of the regulations have their standard to control the amount of emission for manufacturing vehicle. Therefore, It is our responsibility to find a way to reduce this problem as much as possible.

Ethanol alternative fuel is one of the way to reduce the amount of fossil fuel usage, Also reduction of emission. Many countries try to use ethanol fuel for the vehicle as much because they can produce it themselves such as Brazil where can produce and use ethanol fuel most in the world [1]. So they can reduce the import of fossil fuel and get a better economic in their country.

This research is focusing on ethanol fuel using in DISI (Direct Injection Spark Ignition) engine with an adaptive of EGR (exhaust gas recirculation) system. Ethanol fuel has a good characteristic compare to Diesel and Gasoline with high heating value nearby to gasoline fuel.

The usage and development of SI (spark ignition) engine has been for a long time. In Thailand, people use gasoline port fuel injection spark ignition as commonly and this is an old technology now. DISI engine is a new technology engine which has better thermal efficiency. This engine can avoid self-ignition as causing in SI engine because of the direct injection technology. DISI engine can have higher compression ratio and the combustion efficiency will rise up relate to compression ratio. Furthermore with new technology DISI engine can be combustion under lean burn mixture by avoiding knocking with 'Stratified charge combustion'[2]. It can produce power with less fuel required and it can prevent knocking that cause serious damage to engine. Disadvantage of the stratified charge is, it inject fuel lately when the piston move up in compression stroke. Fuel and air mixture is not mixing well in small period of time, the combustion will not complete and will release the emissions, also lean burn combustion produce NOx emission[3]. Although factory will use after treatment method for their vehicle to reduce emission but this research use ethanol fuel which can reduce the emission because it is bio-oxygenated fuel.

2. Experiment Setup and Method

2.1 Fuel Properties

This research is about EGR System in DISI engine. E20 ethanol blended fuel is used. The study with ethanol blended fuel and DISI engine combustion character can gives further knowledge in agriculture which can produce high number of ethanol alternative fuel. Ethanol blended fuel also has some advantage such as flame speed that cause by oxygen inside the molecule of ethanol blended fuel[4]. Ethanol is made from wasted plant, the combustion can give carbon

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neutral to an environment. The properties of E20 ethanol blended fuel is shown in Table. 1

Table.1 shown E20 ethanol blended fuel properties.[5]

Fuel Properties	E20
Formula	$CH_{1.63}O_{0.065}$
Mw[g/mol]	88.12
Carbon[mass%]	79.85
Hydrogen[mass%]	12.88
Oxygen[mass%]	7.54
Density, kg/l at 15°C	0.7645
RVP at 37.8°C,kPa	58.3
LHV, MkJ/kg	40.6
RON	98.3
MON	84.6
Stoichiometric AF ratio	13.51
Initial boiling point ,IBP	42.1
End boiling point	182.8

2.2 Method

A DISI engine will test at 45Nm. engine torque and 2000 rpm which is mentioned in section 2.8. This condition is engine partial load which match to vehicle cruising speed 90kph whereas the maximum speed for Thailand local road. The testing engine will work with E20 ethanol blended fuel. Stratified mode will work in this engine operation. The optimization of DISI engine injection and ignition timing relate to E20 ethanol blended fuel will interested in 80-110 CAD(crank angle degree) which is stratified charge mode. The ignition timing will adjust from 18 to 32 CAD. Dilution of EGR rate by 0 to 25 percent will be operate at the optimized point of injection timing and ignition timing To get an improvement in BSFC and emission. Air fuel mixture is controlled by using Innovate Lm-2 oxygen wide band sensor. Brake Specific Fuel Consumption (BSFC) and gas emission will be collected and analyze. Emissions are examined from exhaust pipe directly and evaluated by gas analyzer MRU SWG 200-1 illustrated in section **2.5**. The schematic diagram of experiment setup is shown in "Fig.1"

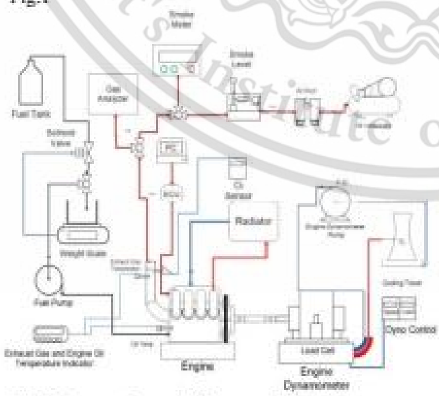

Fig.1 shown schematic diagram of experiment setup.

2.3 DISI Engine

DISI (direct injection spark ignition) engine is a variant of fuel injection employed in modern four-stroke gasoline engines. The gasoline is highly pressurized, and injected directly into the combustion chamber of each cylinder, as opposed to conventional multi-port fuel injection which fuel is injected in the intake manifolds, or cylinder port. In some applications, gasoline direct injection enables to work in stratified charged combustion mode for improving fuel efficiency, and reducing emission levels at low load. The main advantage is the increasing of engine compression ratio coped with cooling effect from the fuel direct inject to piston dome. DISI engine can operate with stratified charge combustion which can combust by lean burn mixture operation.

The in-line 4 cylinder DISI engine is operated in this research which shown in table. 2

Table 2 shown DISI engine specification.

Model	Mitsubishi 4G93 GDI
Type	In-Line, DOHC 16V.
Number of cylinder	4
Displacement(cc)	1,834
Compression Ratio	12.0:1
Bore x Stroke(mm)	81.0x89.0
Maximum Output	96kW @6000RPM
Maximum Torque	177Nm @3750RPM
Vehicle	Mitsubishi Lancer/2003
Dimension	
Width (mm)	1,695
Height (mm)	1,430
Curb Weight (kg)	1,200
Drag Co-efficient	0.3
Tire	185/65/R14
Wheel Diameter (mm)	596
Transmission Specification	
1st	2.319
2nd	1.62
3rd	1.26
4th	1.00
5th	0.7
6th	0.445
Final Drive	5.219

2.4 Engine Dynamometer

The "Tokyo Plant 150 PS@3000RPM Model" engine dynamometer interfaced with in-house program has measured the results of this experiment: power, torque, Brake Specific Fuel Consumption as shown in "Fig. 2"

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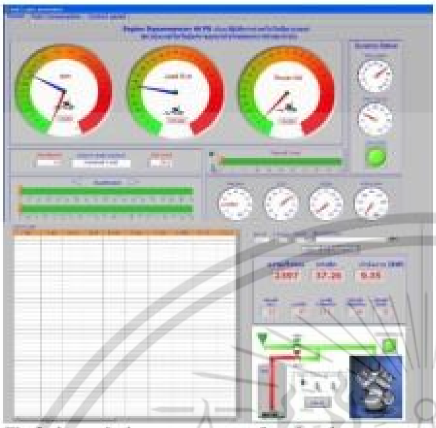

Fig.2 shown in-house program of engine dynamometer

2.5 Emission Analyzer

The gas emissions such as CO, HC and NOx are collected by Gas Analyzer. (MRU SWG 200-1). The gas analyzer measuring emission gas from exhaust pipe when engine operate at the testing condition. It can measure 0-4,000 ppm CO, 0-5,000ppm CxHy, and 0-2,500 NOx accurately.

2.6 Electronic Control Unit

The DTA fast S60 Pro is standalone electronic control unit for DISI testing engine which is able to controls Fuel injection timing, Fuel injection duration and spark ignition timing which can control testing engine conditions.

2.7 EGR Device

EGR (Exhaust Gas Recirculation) Valve is controlled by National Instrument and Labview program "Fig.3". It is design to control percentage of EGR gas into intake pipe. The program analyzes data from intake and exhaust oxygen sensor. The software is automatically control EGR valve.

The percentage of EGR gas is determined by "eq.1"[6]

$$\%EGR = \frac{[O_{2,amb}]-[O_{2,max}]}{[O_{2,amb}]-[O_{2,exh}]} \qquad (1)$$

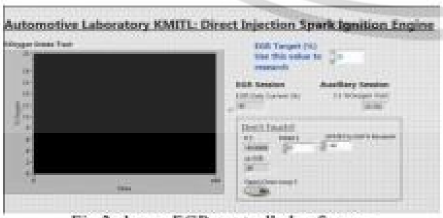

Fig.3 shown EGR controlled software

2.8 Test Conditions

DISI engine is operate on dynamometer with E20 ethanol blended fuel, 2000RPM engine speed and 45kN of brake torque which is cruising speed. testing sequence is shown in "Fig. 4".

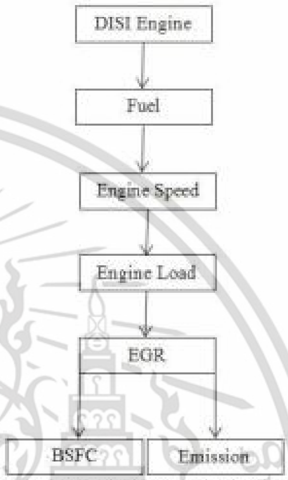

Fig.4 shown the testing sequence diagram.

Then investigate on DISI engine stratified mode characteristic by adjust fuel injection timing and ignition timing. After the optimization of fuel injection timing and ignition timing, The EGR system is operate by 0 to 25 percentage of intake air dilution as shown in Table. 3. After engine is operated in each condition, BSFC is calculated by in-house software cooperates with engine dynamometer and emission is collected by gas analyzer.

Table.3 Shown EGR testing condition

Engine Speed(RPM)	2,000
Fuel Type(%Ethanol)	20
Ambient Temperature(°C)	30
Coolant Temperature(°C)	87
Engine Torque(Nm)	45
Engine Operation	Stratified charge
Lamda(λ)	1
Injection Timing(CAD, BTDC)	80°-110°
Ignition Timing(CAD, BTDC)	17°-32°
EGR rate(%)	0-25

After vehicle specification is purposed, the surroundings condition such as vehicle speed, pavement conditions were based on Thailand regulations which shown in table.4.

Table.4 Surroundings condition

Speed (km/h),	90
Engine Efficiency (%)	90
Road: Fair Pavement = Kr	0.019
Road Gradient (%),	4
Air Density (kg/m^3)	1.2

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From table.4 and table. 5, total resistance consist of air resistance, rolling resistance and gradient resistance. With the calculation of Resistance and the following results showed in table.5 and table.6 respectively.

Table.5 Air Resistance calculated value,

1. Air Resistance (Ra)	
Cross-Section Area (m2)	1.94
So, Air resistance (N)	218.15
2. Rolling Resistance (Rr)	
From Curb Weight (N)	11,772
So, Rolling Resistance	223.67
3. Gradient Resistance (Rg)	47.09
Hence, Total Resistance (Rt,N)	488.90

Table.6 Engine calculated power and torque

Engine Power (kW)	13.58
Engine Torque (Nm)	
1. Torque at wheel	145.69
2. Torque at engine	44.31

The following table.6 shows that calculated engine torque for cruising speed of 90kph. is 44.31Nm.

3. Result and Discussion

3.1 DISI engine

An investigation of DISI engine with stratified mode made by adjust fuel injection timing and ignition timing. "Fig.5" shows BSFC of each engine condition. The best BSFC is locate at 100 CAD, BTDC, it can also shows that injection timing is really effect to engine.

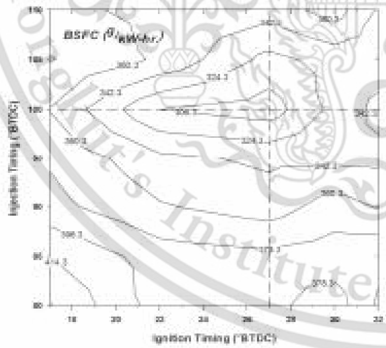

Fig.5 Effect of injection and ignition timing on BSFC

At 100CAD, BTDC injection timing and 27CAD, BTDC ignition timing can produce the highest thermal efficiency relate to BSFC. So the combustion temperature is also high which means high temperature produce NOx emission[7] as shown in "Fig.6" NOx emission is collected from exhaust gas.

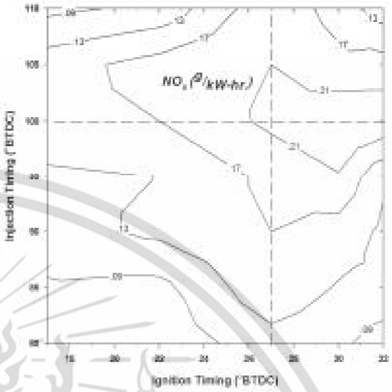

Fig.6 Effect of injection and ignition timing on NOx emission

Both "Fig.7","Fig.8" show that CO and THC reduce if inject fuel lately near TDC and will increase when inject fuel earlier. When inject fuel lately, the combustion is a late phase that will get low cylinder pressure but it can be well after burned with late phase causing low CO and THC emissions.

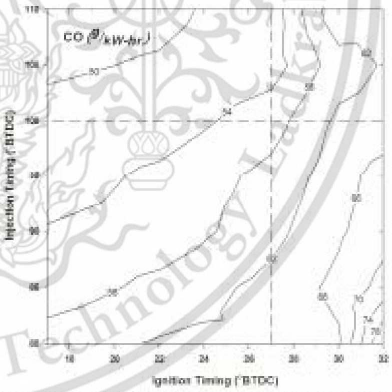

Fig.7 Effect of injection and ignition timing on CO emission

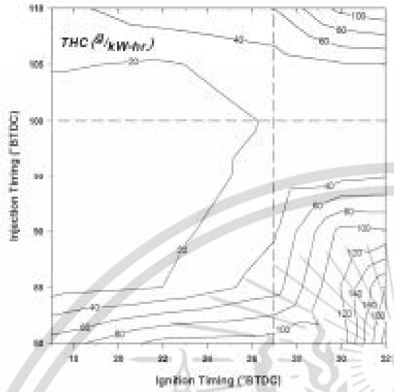

Fig.8 Effect of injection and ignition timing on THC emission

After the investigation of DISI engine with stratified mode and E20 ethanol blended fuel. It is found that at 100CAD, BTDC injection timing and 27CAD, BTDC ignition timing is the lowest number of BSFC. Furthermore, emission increase at 100CAD, BTDC injection timing and 27CAD, BTDC. This is an optimum point for Injection and Ignition timing in DISI engine using ethanol blended fuel. Some number of emission can be reduced by using EGR technic after found this optimum point.

3.2 EGR System

When the engine operate on partial load the movement of the piston suction the air make intake manifold become vacuum. EGR can reduce intake manifold vacuum which causing to reduce the pumping work[8]. It cause an improvement in BSFC. "Fig.9" shows manifold absolute pressure inside intake manifold when EGR is operate

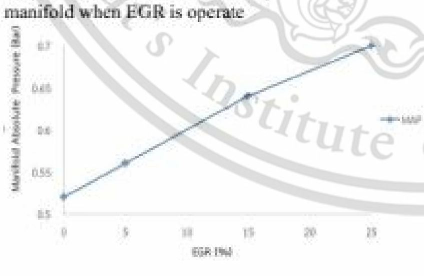

Fig.9 Effect of manifold absolute pressure on EGR rate

EGR can reduce pumping work that can improve BSFC but Exhaust gas actually harmful to combustion reaction. EGR is classified as inert gas which cause low speed flame[9]. "Fig.10" shown that EGR optimum point is at 5% EGR rate. It can improve BSFC at 5 percent of the EGR.

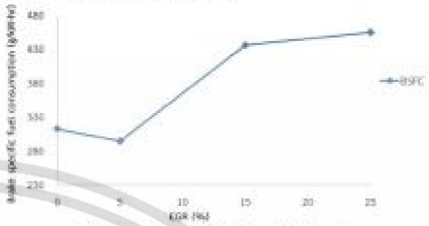

Fig.10 Effect of BSFC on EGR rate

NOx emission can reduce relate to adiabatic flame temperature as shown in "Fig.11". EGR dilution is a wall between fuel mixture molecules. It effect to exhaust temperature.

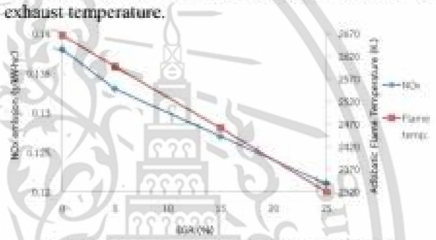

Fig.11 Effect of NOx emission on EGR rate

Although, EGR can reduce NOx emission, it increase CO and HC because too much percentage of EGR dilute to intake manifold as shown in "fig.12","fig.13"[9].

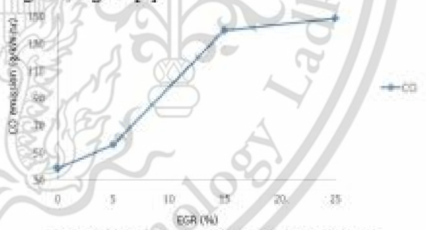

Fig.12 Effect of CO emission on EGR rate

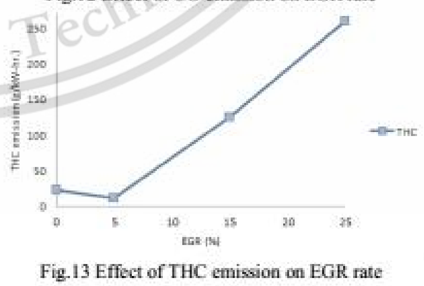

Fig.13 Effect of THC emission on EGR rate

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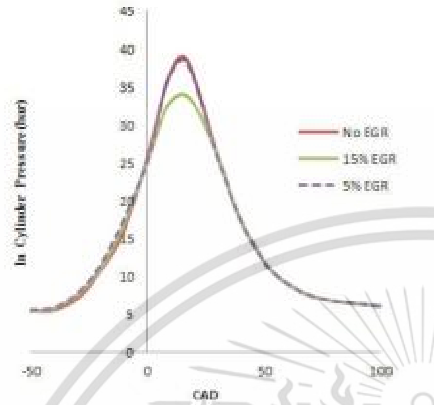

Fig.14 Effect of in cylinder pressure and CAD on EGR rate

The optimum EGR rate for DISI engine with stratified charge combustion is 5% EGR. 5% EGR can reduce pumping work and has less effect to in cylinder pressure as shown in "Fig.14". 15% EGR is highly effect to cylinder pressure which engine require more fuel to maintain engine torque. The BSFC is reduce when increasing EGR rate to engine more than 5%

4. Conclusion

4.1 DISI Engine

DISI engine operate in stratified mode with E20 ethanol blended fuel is optimized at 100CAD, BTDC injection timing and 27CAD, BTDC ignition timing.

Early injection and late ignition timing, mixture stratification is reduced by increasing duration of air fuel mixture. So, partial burns in for combustion is increase. Early injection and early ignition timing, too early of combustion phase. An over lean mixture occur rapidly in combustion phase. Late injection and late ignition timing, in-cylinder pressure is decrease[10]. This causes retarding for air fuel mixture and causing later combustion phase. Late injection and early ignition timing, In sufficient duration between the end of injection and start of spark causing over rich mixture and liquid droplet effect.

The engine can operate well at optimum point. It effect to BSFC, CO, NOx and HC emission

4.1 EGR

The optimum EGR rate for DISI engine which has the least BSFC is at 5 percent. It reduces vacuum inside intake manifold which also reduce pumping work. NOx emissions also decrease due to low flame temperature. AT optimum point 5% EGR rate HC and CO increase slightly but it increase extremely high if dilute 10 to 25 percent of EGR because of low flame temperature and highly effect to combustion pressure.

5. Acknowledge

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AUTHOR BIOGRAPHY

Author: Mr. Rattapoom Keskangam

Date of Birth: 13 Febuary 1992

Address: 26/2 Kokfaet Nongchok District Liebvaree rd. Bangkok Thailand 10530

Education:

2010-2013 bachelor's Degree in mechanical engineering, King Mongkut's Institute of Technology Ladkrabang

2014-2018 master's Degree in automotive engineering (International Program), International college King Mongkut's Institute of Technology Ladkrabang

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