THE IMPACT OF INJECTION TIMING IN ETHANOL FUEL ON DIRECT INJECTION SPARK IGNITION (DISI) ENGINE

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ABSTRACT

Nowadays the major problem of automobile is extravagant of fuel consumption in transportation affected in many ways as currency and fuel economy dissipative including with environmental impurity. Those belong to environmental quality declination and lack of fossil fuel especially in transportation that uses lots of fuel mostly consumed by motor vehicle. So the opportunity to save currency and improve environmental quality through fuel use reductions is clear by using ethanol as renewable resources from benefits its properties together with direct injection spark ignition (DISI) engines.

The objective of this research presents effect of injection timing on performance and emissions in stratified charge mode when fueled with ethanol. The four cylinder and four stroke is tested and controlled by standalone electronic control unit. To test efficiencies and emissions were conducted by using difference purity ethanol blended in gasoline (E20 and E85) and gasoline (E0), variation of spark and injection timing to maximum brake torque power at stoichiometric A/F ratio which measured break specific fuel consumption (BSFC) and break specific energy consumption (BSEC) at 2000 rpm 3 bar of nominal BMEP. The results showed that when compare with gasoline, Fuel blends with higher content of ethanol causes injection and ignition timing to advance which obtained 13.06% more thermal efficiency together with emissions have reduced likewise; carbon monoxide-42.17%, total hydrocarbon-43.99% and oxide of nitrogen-23.60%. However, in case of morphology, the size of particle emission is larger due to local rich mixture from long injection duration which causes fuel impingement.

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Panuwat Kangkaya.

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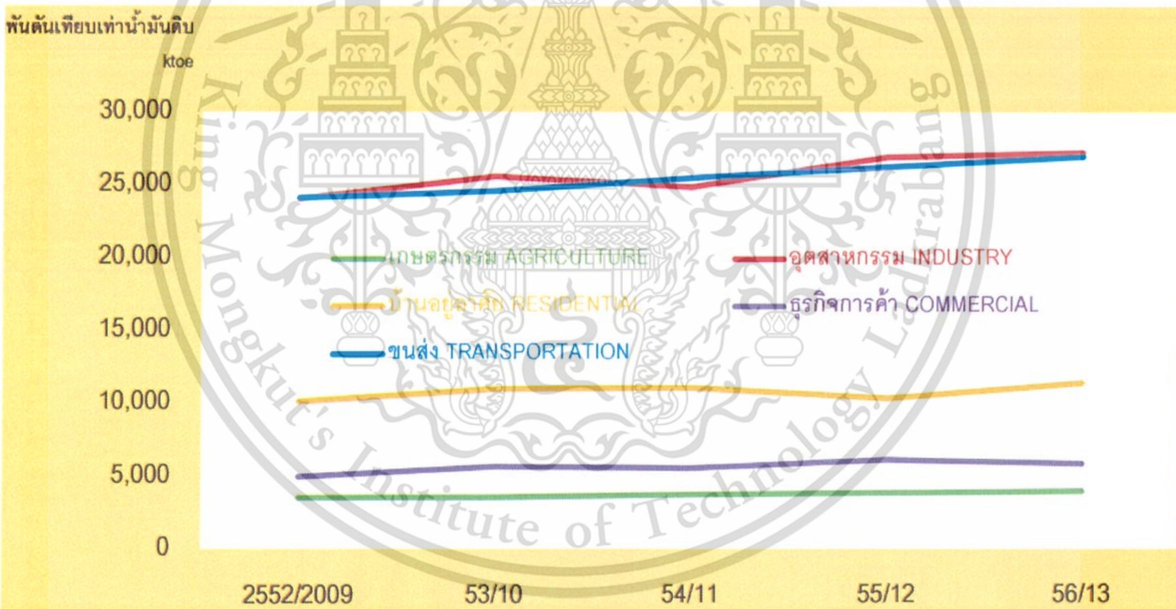
Chapter 1

INTRODUCTION

1.1 Background

Nowadays the major problem of automobile is extravagant of fuel consumption in transportation affected in many ways as currency and fuel economy dissipative including with environmental impurity. Focusing on Thailand, the economics is driven by 2 major factors as transportation and industry which consumes 35 percent of overall energy usage in this country as showed in figure 1.1. Considering in transportation section, 22.8 percent is consumed by gasoline with the rest is ethanol blended with gasoline or called "Gasohol" with shown on figure 1.2.



Remarks: Industrial Section comprise of production, mining and construction industry.

Figure 1.1 Using byproduct of energy production classified by economy sections 2552 – 2556 B.E [1].

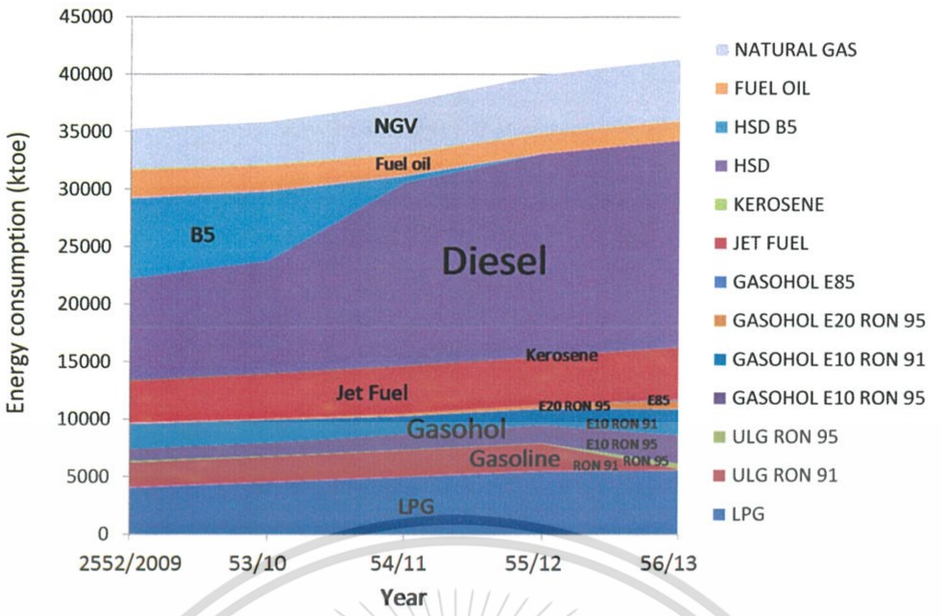


Figure 1.2 Using byproduct of energy production classified by sources 2009 – 2013 [2].

From the energy crisis which tend to high rise of cost causes increasingly pecuniary loss in term of import crude oil which portrays in table 1.1. It is imperative to find out renewable energy instead of fossil fuel.

Table 1.1 Quantity value and average import crude oil price 2007–2013 [3].

A.D	Quantity (million liter)	Value (billion baht)	Aver. Price (baht/bbl.)
2007	46,333	715,354	2,458.26
2008	47,112	999,851	3,377.88
2009	46,608	623,013	2,126.32
2010	47,365	753,648	2,529.02
2011	46,090	976,789	3,369.50
2012	50,055	1,119,564	3,658.20
2013	50,373	1,072,412	3,506.45

Ethanol is suitable alternative fuel. It broadly promoted by the government due to interested properties of ethanol and it can decline dependency of imported crude oil [4]. Furthermore, ethanol can be produced from many source of biomass and were the renewable energy. From figure 1.3, the overall of ethanol production is 2-3 million-liter per day and mount up to 9 million-liter per day [5]. Then, to research and development this fuel is necessary.

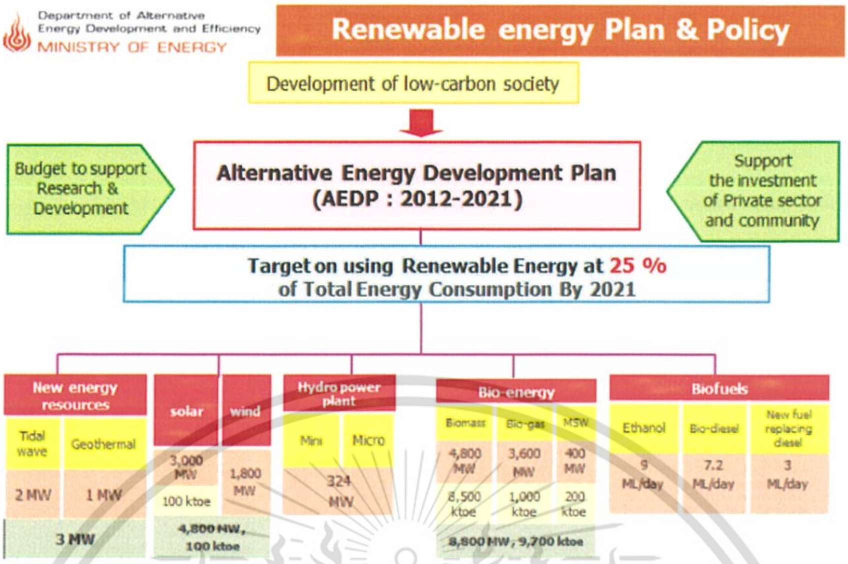


Figure 1.3 Alternative energy development plan (AEDP: 2012-2021) [5].

Using new technologies of spark ignition engine for high performance and low emissions, direct injection spark ignition engine (DISI) is appropriate solution due to fuel is injected directly into combustion chamber brings about high thermal efficiency conform with higher output than conventional engine which operates in port fuel injection (PFI). Moreover, energy consumption is lower from extra operation which called "Ultra-lean combustion" or "Stratified mode" [6, 7, 8]. However, the stratified charge causes particle emissions higher than conventional operation due to characterization of this mode resemble like compression ignition engine (CI) as diffusion flame. Particle emissions include with soluble organic fraction (SOF) and solid fraction (SOL), those rely on operating conditions and composition of fuel [9, 10, 11, 12].

Then, using ethanol blends with gasoline together with a direct injection spark ignition engine to better performance and reduction of emissions is interesting. But most of engine generally are not suitable to using biofuel hence to optimization parameter of its engine is necessary. Hence, investigation the effect from properties of ethanol and DISI engine operations is essential to develop insightful information.

1.2 Objectives

1.2.1 To investigate the impact of injection timing and ignition timing on efficiency (maximum torque advance, brake specific fuel consumption, brake specific energy consumption and thermal efficiency) and emissions of direct injection spark ignition engine in stratified charge mode.

1.2.2 To study the effect of ethanol blends with gasoline on performance and emissions in direct injection spark ignition engine in stratified charge mode.

1.2.3 To study the particulate matter on physical characteristics (morphology) from variation of injection timing, ignition timing and gasoline-ethanol of direct injection spark ignition engine in stratified charge mode.

1.3 Scope of work

1.3.1 The characterization of fuel composition.

To study fuel properties of gasoline, ethanol blends with gasoline and ethanol; how different of fuel composition effects to performance and emission characteristics.

1.3.2 The effect of fuel on emissions.

The impact of fuel; gasoline, ethanol blends with gasoline and ethanol on emission as particulate matter on morphology which measuring by smoke meter and scanning electron microscope (SEM) image coped with gaseous phase by gas analyzer.

1.3.3 The impact of injection and ignition timing on ethanol blends with gasoline to best operation.

Performance and emissions of gasoline-ethanol on DISI engine in stratified charge mode to improve performance and reduce emissions by variation of parameters (ignition timing, injection timing and injection duration). This results analyzed in term of performance and efficiency (maximum torque advance, brake specific fuel consumption, brake specific energy consumption and thermal efficiency) and emissions (emissions concentration, morphology of particulate matter).

Chapter 2

RESEARCH BACKGROUND

2.1 Engine

2.1.1 Introduction of Internal Combustion Engine

2.1.1.1 Performance and Efficiency

There are 3 parameters that impact to overall thermal efficiency of internal combustion engine. Starting with compression ratio following with air-fuel ratio or specific heat ratio and the last one is the duration of combustion. "Air-standard analysis" is used to determine the ideal efficiency of internal combustion engine by pure air which represented as working fluid. The efficiency of simple thermodynamic analysis of the ideal air-standard Otto cycle is calculated by

$$\eta = 1 - \frac{1}{{r_v}^{\gamma-1}} \tag{2.1}$$

Where r_v = compression ratio

$\gamma = C_p / C_v$; Specific heat ratio

First, compression ratio (r_v) or described as volumetric compression ratio; is the ratio between maximum volume of chamber at bottom dead center (BDC) and minimum volume of chamber at top dead center (TDC). All of heat presume to be added to the cycle at constant volume at TDC. The common analysis portrays as the compression ratio should be high as much as possible for high efficiency.

From Figure 2.1 showed the ideal thermal efficiency as function of compression ratio of air-standard Otto cycle equate to the variation of cut-off ratio of air-standard diesel cycle. It reveals that at the compression ratio, the efficiency of air-standard Otto cycle is more than air-standard diesel cycle, but in fact the efficiency from diesel engine is higher due to it operates higher compression ratio when compose with spark ignition engine.

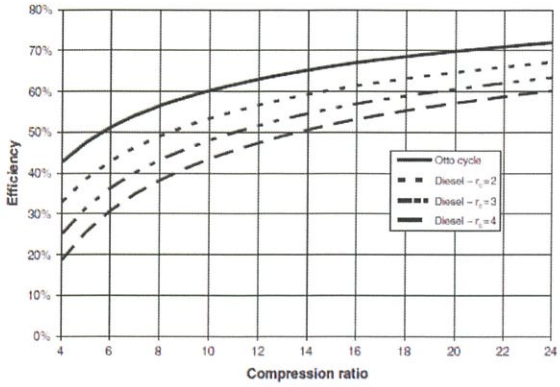


Figure 2.1 The efficiency with several of compression ratio for a constant volume air standard cycle.

From above, indicates as the specific heat ratio should be as much as possible to be high efficiency. In fact, the specific heat ratio of air (γ =1.4) is greater than the regular hydrocarbon fuel air-fuel mixture. It discusses as the mixture with surplus of air (such as lean mixture) has higher specific heat ratio than rich mixture that conform as the analysis from Heywood (1988) which shown in Figure 2.2. The figure showed the theoretical efficiency of ideal Otto cycle engine by using fuel and air as working fluid as the function of fuel-air equivalence ratio (ϕ). By value of equivalence ratio is greater than 1.0 known as rich mixture, below 1.0 as lean mixture following with equal 1.0 as stoichiometric mixture.

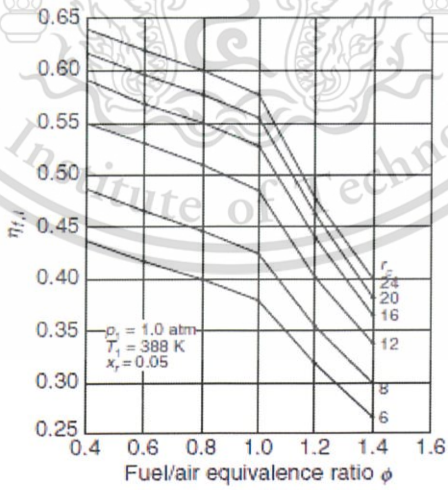


Figure 2.2 Fuel –air equivalence ratio versus thermal efficiency of constant fuel-air cycle with 1-octane fuel [7].

From this figure, clarify as the mixture leaner leads to high thermal efficiency. The efficiency drops dramatically when the equivalence ratio of mixture over than 1.0 due to the available unburned fuel in the mixture. In the other words, the oxygen is not enough to make complete combustion of all fuel when operate in fuel-rich mixture. This figure describes another reason for diesel efficiency which is greater than Otto cycle efficiency when the un-throttled diesel engine always operate at very lean overall air-fuel ratio, generally on part-load.

Last, in the duration of combustion. The idealistic of duration of combustion wound be release all of the energy into the cylinder immediately at TDC of the compression…due to the fuels have a finite burning rate, that is impossible and the output obtained from a fuel burned is lower when equate to ideal cycle, so this results in declination of thermal efficiency. From the figure 2.3 showed these effects token by Campbell (1979) which showed the relation between the power outputs with spark advance by $\Delta\theta_c$ showed the duration of combustion. The combustion duration and the spark advance represent in term as crank angle degree and degree before the top dead center. When the combustion duration increased causes the optimum value of spark timing to increase as well. From the figure indicates that higher output and increase in thermal efficiency belongs to reduction of combustion duration.

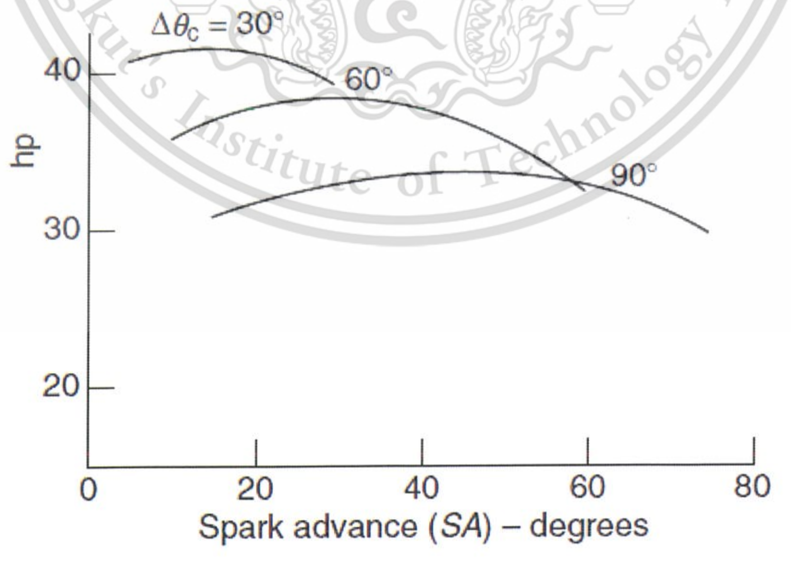


Figure 2.3 Effect of spark advance and the duration of combustion to the power output.

As the maximum burning rates is occurs near stoichiometric air-fuel ratio, when operating spark ignition engine to lean (ϕ <1.0) extends the duration of combustion. When refers to the figure 2.3, this operation reduces power output and thermal efficiency. Whereby tending to neutralize the increased efficiency of lean mode due to an increase of specific heat ratio as seen in figure 2.2 that is essential. Hence, when operate a spark ignition engine under lean burn conditions, to design the combustion operates in this mode should regard to high burning rate.

2.1.1.2 Emission in DISI Engine

Direct injection spark ignition engine converts chemical energy from fuel to mechanical power. Fuel which injected from high pressure pump in to combustion chamber mixes with intake air results in combustion process after spark discharged. The exhaust gas from combustion process consists of emissions which harmful to environment and human health. Emissions composes of carbon monoxide (CO), total hydrocarbon (THC), oxide of nitrogen (NO_x) and Particulate matter (PM) as shown in eq.2.1

$$C_xH_y + Air \rightarrow aCO_2 + bH_2O + cCO + dC_{(s)} + mHC_{(2)} + nNO_x \quad (2.2)$$

2.1.1.2.1 Carbon monoxide (CO), Total hydrocarbon (THC) and Oxide of nitrogen (NO_x)

Carbon monoxide (CO), Total hydrocarbon (THC) and aldehyde are generated from incomplete combustion. Most of exhaust hydrocarbons came from the lubrication. Accumulate of carbon monoxide are formed by engine operates in worse space which causes headaches, lethargy and eye irritation. Furthermore, total hydrocarbon has negative effect to environment as dominant component of smog.

Oxide of nitrogen (NO_x) formed by nitrogen and oxygen under high temperature and pressure. Actually, NO_x formed by nitric oxide (NO) and nitrogen dioxide (NO_2) in a small fraction. NO_x is one of severe emission to environment due to this factor of smog generation.

The portion of spark ignition engine emission based on air-fuel ratio. As shown in figure 2.4 from Heywood (1988) can discuss by 2 mixture operations. First, lean mixture where fuel-air equivalence ratio (ϕ) is less than 1.0. Surplus of air always operate in this mode emitted carbon monoxide slightly. Whereas total hydrocarbon is decrease when ϕ become from 0.7 to 0.9 and rises up after ϕ = 0.9 due to misfiring is occurred from the proximity to the lean misfire limit bring about rise up of total hydrocarbon for ϕ less than 0.9. Focus on oxide of nitrogen generation, combustion temperature is a dominant factor of production; the higher temperature, the higher opportunity of oxide nitrogen compounds to oxide nitrogen. When combustion temperature is at maximum near stoichiometric (ϕ =1) and propel down for lean and rich mixtures, the oxide of nitrogen trend line is shown in concave down.

At rich mixture, where ϕ is greater than 1.0. The level of total hydrocarbon is risen up by the fuel that have not enough intake air to completely burn as well as the high of carbon monoxide due to enough present oxygen to oxidize carbon monoxide to carbon dioxide (CO_2).

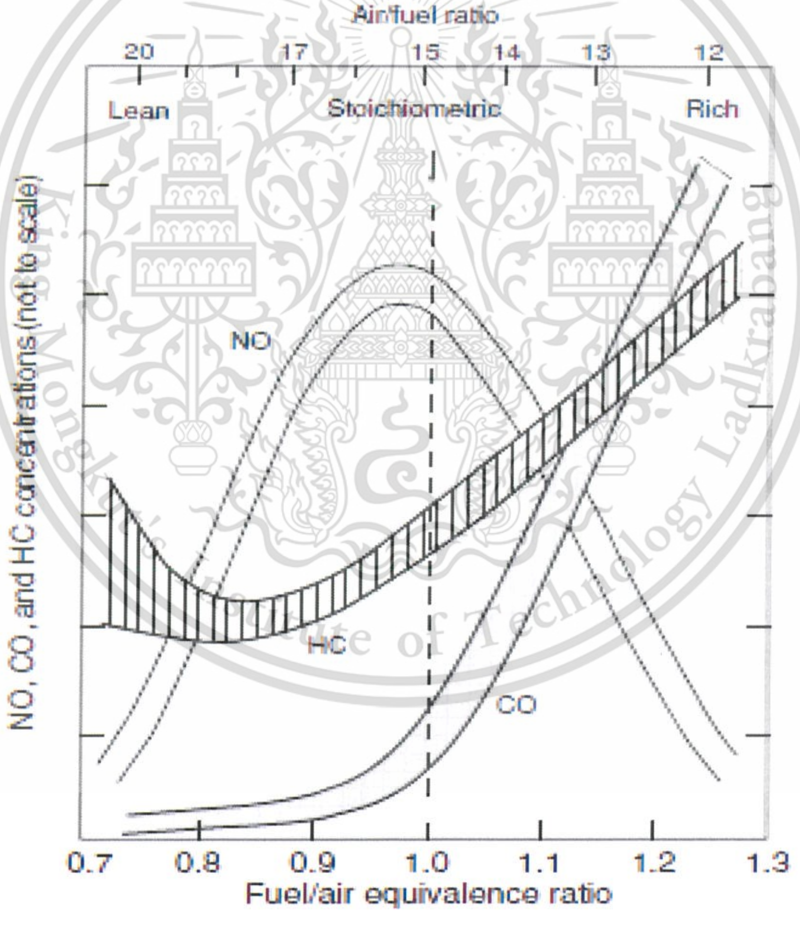


Figure 2.4 Emissions concentration as a function of equivalence ratio ϕ [7].

2.1.1.2.2 Particulate Emissions

The particulate matter is interested of DISI emission which formed into black smoke. The particulate matter also called "PM". The particulate matter was classified by size into 3 sections as nucleation mode, accumulation mode and coarse mode. First, nucleation mode consists of more arcane which formed from nucleated volatiles. Second, accumulation mode particle formed by a solid core of carbonaceous blocks known as "spherules". This spherules is constant in size (20-50 nm), and formation of plenty carbonaceous building blocks also called "agglomerate" which the size range are 60-100 nm. Last in coarse mode particle are formed by different composition. Mainly made by attaching the predecessors lodge within exhaust system. These particles join each other and make larger particle which go out by exhaust stream. This trend has random manner and unpredictable. As shown in figure 2.5, they may consist of solid core and volatile organic fraction (VOF) layer on the surface.

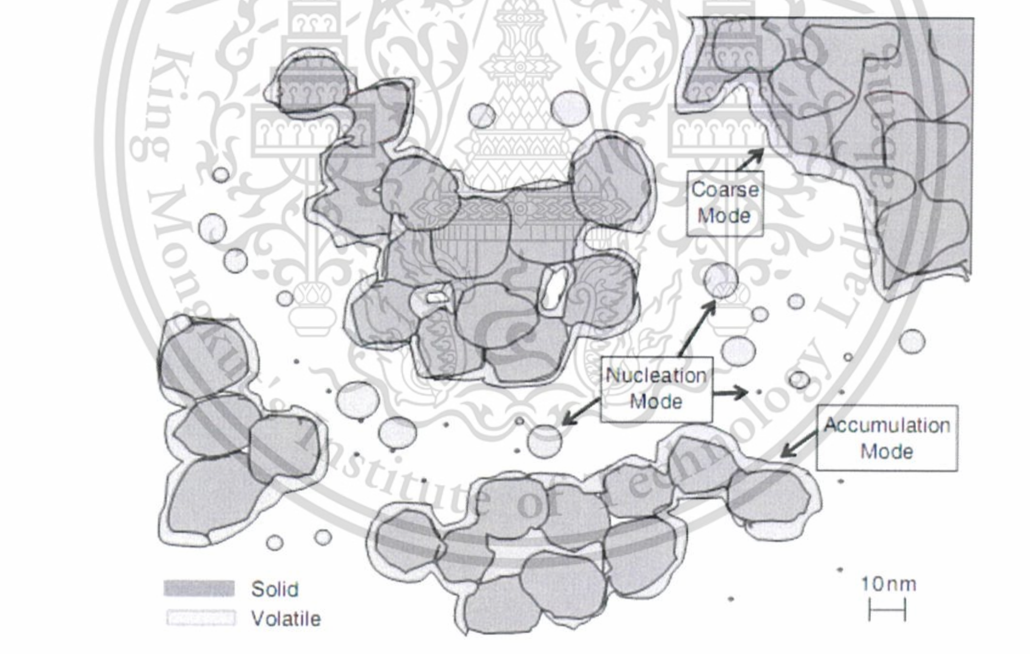


Figure 2.5 Schematic diagram of particulate matter classification [13].

There are 2 main part of particulate matter. Starting with Solid Organic Fraction (SOL) consists of hydrocarbon. Figure 2.6 portrays the definition of atmosphere particle size as PM_{10} (where D (diameter) less than 10 μm), fine particles (D<2.5 μm), ultrafine particle (D<0.1 μm) and nano-particle (D<0.05 μm or 50 nm).

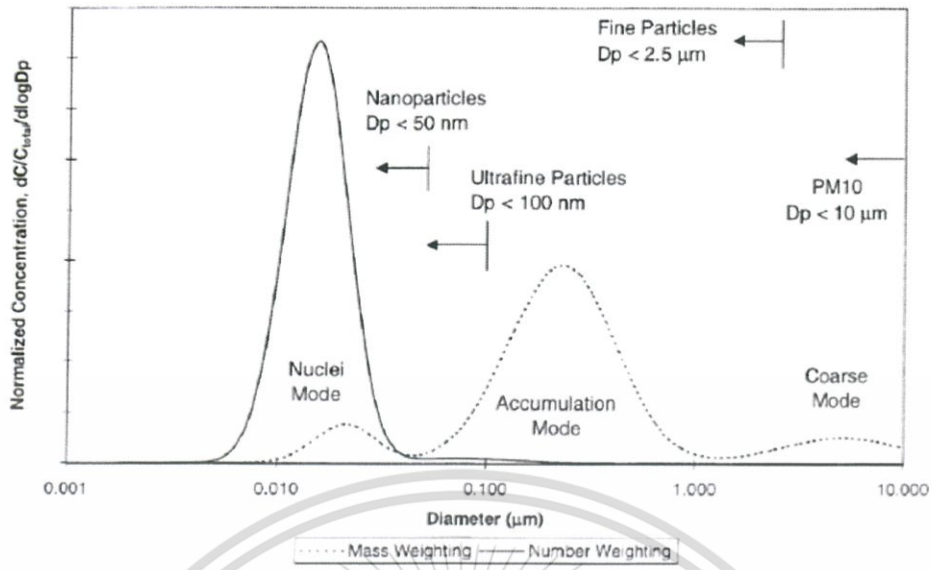


Figure 2.6 Particle size distribution of particulate matter [14].

Particulate matter (PM) is 1 of 6 pollutants which emitted to atmosphere and was harmful to human health [15, 16, 17]. Many studies founded that as 10 microns (PM_{10}) of particulate matter size has strong effect to inhalation humanity. PM consists of solid particles and liquid droplets which came from many sources such as industry and automobile. These mixtures are formed in atmosphere by transformation of gaseous emissions. From figure 2.7, M.M. Maricq et al. [15] classified particulate matter into 2 sections as

1. Nucleation Particles; formed during combustion and dilution which diameter less than 30 nm. This mode consists of hydrocarbons and sulfate.

2. Accumulation mode formed by agglomeration of nucleation mode which diameter is larger than 30 nm. This mode composes of condensed or adsorbed volatile material as carbon together with remaining metallic ash. Then, cover with heavier condensed compounds and sulfate.

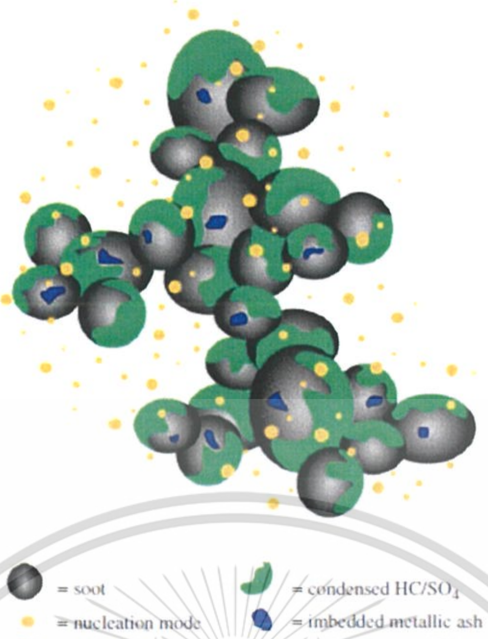


Figure 2.7 The concept of particulate matter [15].

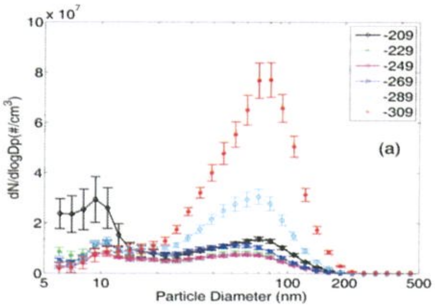
X. He et al. [18] studied "Effect of Gasoline Direct Injection Engine Operating Parameters on Particle Number Emissions" by using E0 and E20 with wall-guided DISI engine in order to clarify the operating parameters (star of injection, injection pressure, spark timing, exhaust and intake cam phase and air-fuel ratio) influenced to particulate (PN) emissions and size distribution. The results are shown as the following;

1. Injection timing is the most impact parameter to PN and size distribution due to fuel impingement in piston bowl and cylinder wall.

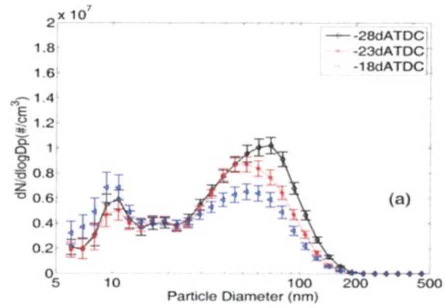
2. Cam phasing and Spark timing are insensitive to PN especially in spark timing. It is not recommended because of disadvantage effect causing to engine efficiency and other pollutants emissions.

3. Increasing fuel pressure reduces droplet size for simple evaporation that results in faster air fuel mixing.

4. Ethanol has impact to PN emission. Using ethanol blends produce large particulate size and number due to long injection duration to obtain same heating output when compare to gasoline which causes fuel impingement including high heat of vaporization that requires more energy to evaporate. Hence, these increase the possibility of local rich mixture which presented in higher PN and larger size.



a)



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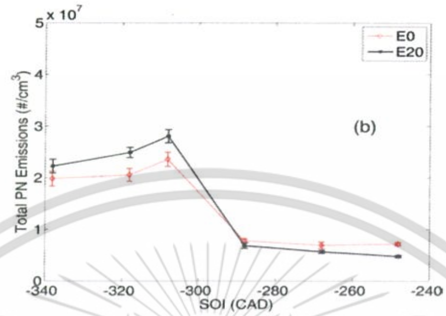


c)

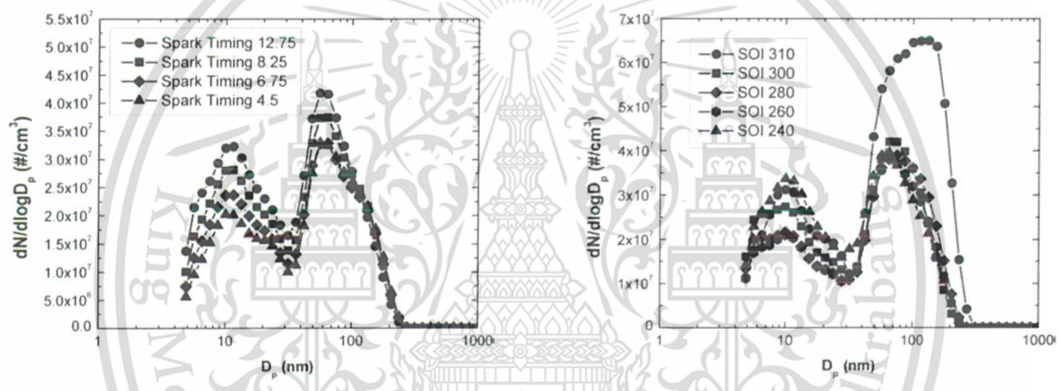
Figure 2.8 a) Impact of SOI on PN emission, b) Impact of spark timing on PN emissions and c) Impact of ethanol on PN emissions [18].

Chen et al. [19] studied "Measurement of Enthalpies of Vaporization of Isooctane and Ethanol Blends and Their Effect on PM Emissions from a GDI Single Cylinder Optical Engine" for a range of isooctane/ethanol blends at part load operating condition. The results shown that as increasing percentage of ethanol in gasoline effects directly to the quantity of particulate number due to the homogeneity of air-fuel is lower as result of higher specific enthalpy of vaporization of ethanol.

Particulate emissions from a 2009 gasoline direct injection engine with 3 difference fuel by Khalek et al. [20] examined as particulate matter formation come from 2 dominant factors as slow droplet evaporation and the presence of fuel enrichment zones during combustion process.

The effect from fuel type, engine strategy and fuel strategy on particle emissions from natural aspired direct injection spark ignition engine by Szybist et al. [21] proposed as first, injection timing of fuel is the most important factor to particle emission. Last in single direct injection operation, E85 produce 1-2 order of magnitudes lower particle emissions when compare with E0.

The particulate matter emission comparison of spark ignition direct injection (SIDI) and port fuel injection (PFI) operation of a boosted gasoline engine by J. Su et al. [22] with boosted 2.0 liter gasoline is used to compare particulate matter emissions between 2 modes as DISI and PFI operated with 1500 rpm and 8 bar BMEP. The experiment founded as too retard injection timing reduced duration of fuel evaporation and mixing formation that results in increasing charge stratification and also the localized fuel region is rich. Then, particle emission is high and larger size. In case of spark timing, retarding spark timing reduce particle emission and size distribution from the limitation formation of particle due to lower peak combustion temperature. Furthermore, it is not recommended to reduce particulate matter by this solution due to the particulate matter reduction and negative effect to fuel efficiency.



a)

Figure 2.9 a) Effect of spark timing (°bTDC) and b) Effect of injection timing (°bTDC) on SIDI particle size distribution at 1500 rpm, 8 bar nominal BMEP [22].

Y. Q. Pei et al [23] study the number of exhaust particles, the particle size distribution and the geometric mean diameter from 1.5 gasoline direct injection engine with interested parameters on same engine load and engine speed due to understand how mixture before ignition affect to particulate emission and size distribution, the results are discuss as;

1. The particulate concentration can be reduced by retarding spark timing due to lower formation rate from the lower combustion temperature.

2. To compare with rich mixture, leaner mixture can reduce more than 90% of number concentration from lower combustion temperature and higher oxygen in chamber can enhance particle's oxidation rate.

3. Injection timing effects directly to the mixing process. To advance of injection timing prolong mixing duration which results in more homogeneous charge before ignition. Thus, it can reduce number concentration and size distribution of accumulation mode effectively.

The above described outline the engine parameters on performance, efficiencies and emissions from DISI engine but impact of high ethanol blends rate together with injection timing is less information available especially in PM size distribution comparison and how biofuel can maximized performance and efficiencies coupled with minimized emission in advance details. Then the objective of this study is to conduct a comparison those parameters by using wall-guided DISI engine and provide evaluation of particulate matter for further development of gasoline particulate filter.

2.2 DISI Engine

2.2.1 DISI Engine Operation

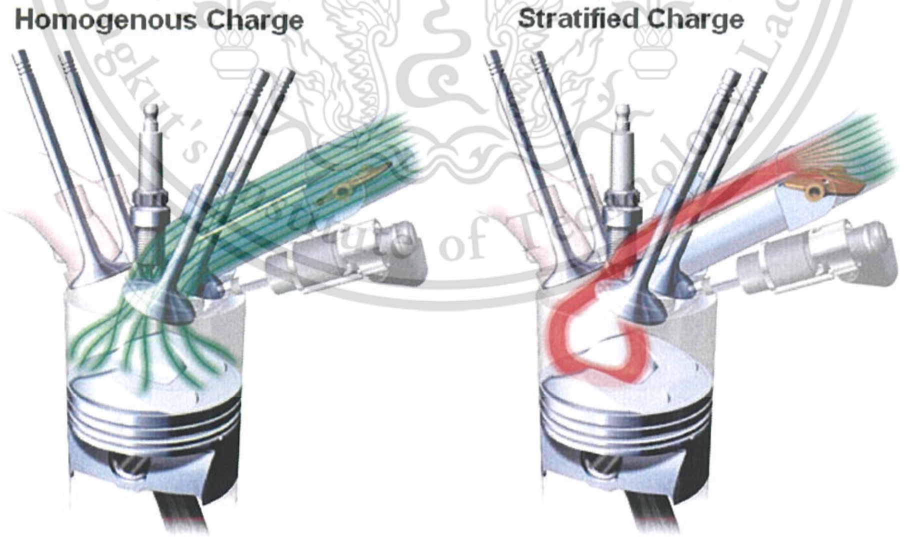


Figure 2.10 The injection strategy of DISI engine; Homogeneous charge (Left) and Stratified charge (Right) [24].

As seen the prior introduction of DISI engine which has efficiency higher than conventional SI engine. Injection strategy is crucial factor to achieve. The DISI engine operation categorizes into 2 sections as homogeneous charge or early injection where operates in intake stroke by make uniform mixture while stratified charge operates in compression stroke or late injection where the mixture has overall lean equivalence ratio which shown in figure 2.10

Both of injection strategies operated by engine load and engine speed as seen in figure 2.11. Starting with low and medium load, stratified charge mode is operated due to reduction of pumping loss that results in better in fuel economy. However, this mode is limited or operated in narrow engine range in term of engine load and speed. In case of medium to full load are responsible by homogeneous mode in order to have sufficient duration between abundance quantities of fuel and air to homogeneous mixture. Moreover, the fuel-air rich mixture is important to attain the maximum torque for complete combustion process when full load is required.

In additional from this figure, the in-cylinder flow become more turbulence when increases an engine speed is over than 3,000 rpm and brake mean effective pressure is exceed 5.0 bar. Then, fuel requires large quantities to proper of mixture formation that results in generation of soot.

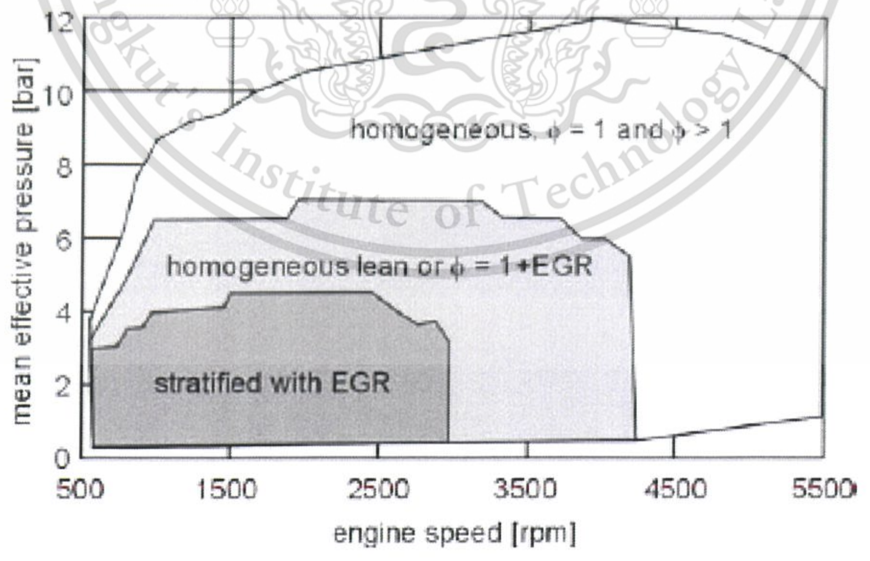


Figure 2.11 DISI engine operation mode [8].

2.2.2 Stratified Charge Strategy

- Wall Guided

This technique uses a spherical piston surface due to transmit fuel to the central of spark plug. This strategy obtained an increasing of emissions of carbon monoxide and unburned hydrocarbon due to the fuel is injected on the piston surface which cannot completely evaporate until ignition occurs. Then reduction of fuel consumption cannot achieve. However, this strategy is reliable regarding to robustness of combustion concept and the prevention of misfiring.

- Air Guided

Wall wetting by using this concept is neglected due to fuel is injected into an in-cylinder airflow that transmits compact spray plume to the spark plug. The production of stable air motions leads spray plume to compact and deliver to spark plug whilst enhance a homogenous mixing inside the cloud. Hence, the efficiency and reliability from this mode is achieved. Swirl and Tumble air motion are possible to air motion in cylinder. Swirl flow is benefit as cylinder head is flat type whereas the strong tumble flow is allowed in pent roof cylinder.

Even though, the generation of stable flow that enhance mixture formation inside the spray cloud together with keeps it compact at the same time and transport it into the spark plug such that ignition can occur at a thermodynamically optimum timing is nearly impossible to realize for all speed and load points in the stratified charge operation range. Moreover, to throttle engine increases losses from production of swirl or tumble. Hence, economy is reduced.

- Spray Guided

This technique allows the highest fuel economy. The spray-guided classified by a narrow arrangement of spark plug and injector. Spray from injector is directly delivered to the spark plug by its kinetic energy. Hence, the special design of combustion chamber and piston is not necessary. From the narrow arrangement of injector and spark plug leads the duration between injection and ignition (duration of mixture formation) is too short. Then, high injection pressure is essential to deserve sufficient energy for mixture formation and evade the generation of scot. On the other hand, low lubricity of gasoline causes problem to high injection pressure in term of friction and wear as well.

Due to the duration of spray arrives to the spark plug is rely on injection timing and not complicated air motion, there are no limitation in spark timing and thermodynamically optimal timing can be realized easier than 2 techniques as started before. So, this technique allow highest opportunity to decrease fuel consumption at part-load due to the spray not impact on a cylinder wall because strong in-cylinder air motion is not required, pumping loss and heat loss to the engine are lowest of all 3 techniques.

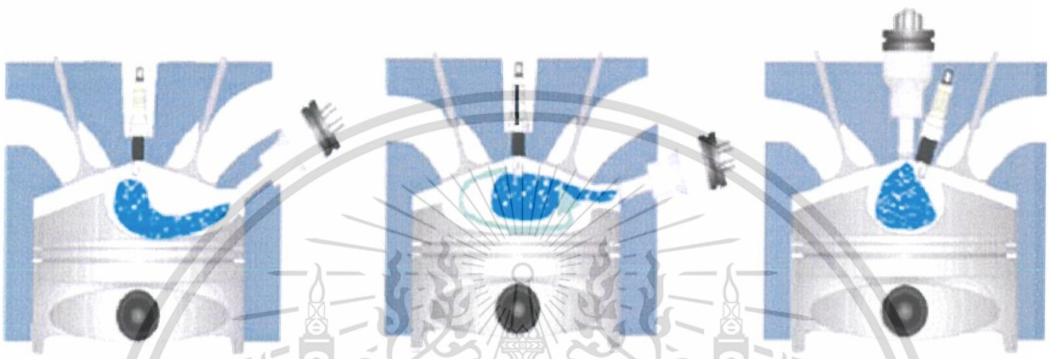


Figure 2.12 Classification the operation of DISI- stratified charge combustion, (a) Wall-guide technique, (b) Air-guide technique and (c) Spray-guide technique.

2.2.3 DISI Combustion

Both of improving efficiency of engine and lowering exhaust emissions, stratified charge combustion has been recognized. There are many parameters impact to the process of combustion likewise ignition timing, injection timing, temperature, properties of fuel and vice verso. So, to control the stable combustion of this mode is challenging. To improve the combustion stability and efficiency, the…will base on the impact of fuel as ethanol with stratified combustion and effect of injection timing and ignition timing to the combustion characteristic.

Figure 2.13 showed stratified charge engine combustion chamber by Benson and Whitehouse, 1979. The mixture burns near the sparkplug and downstream of spark plug as premixed flame, where as in the remain of the cylinder; the mixture is very lean due to the fact that fuel is introduced at one in the combustion chamber and burnt immediately means the overall air-fuel ratio is very lean, while the local is near and is a direct result of the stratified nature of the charge. The stratified charge engine can operate over a much of wider range of air-fuel ratio than a conventional spark-ignition engine. This results in higher efficiency due to leaner overall air-fuel ratios and reduced pumping loss when less throttling is required.

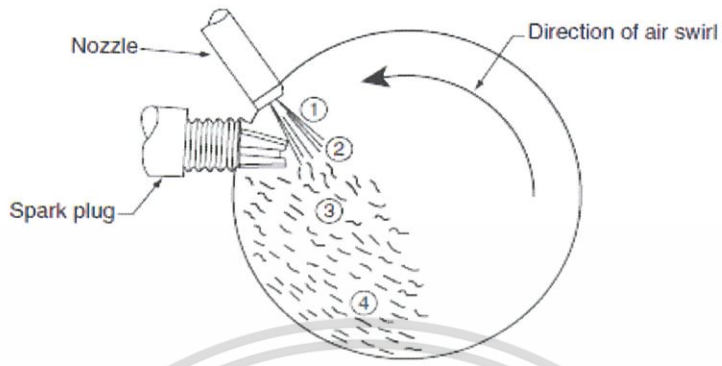


Figure 2.13 Stratified charge engine combustion chamber.

2.2.4 Effect of Parameter in DISI Engine to Combustion

- Duration between Start of Injection and Start of Spark

The combustion efficiency of gasoline direct injection engine by using constant volume combustion chamber from Naoki et al [25] discussed as changing interval between end of injection and start of spark affected to pressure rise rate and heating efficiency by both factors were decreased when the interval was extended.

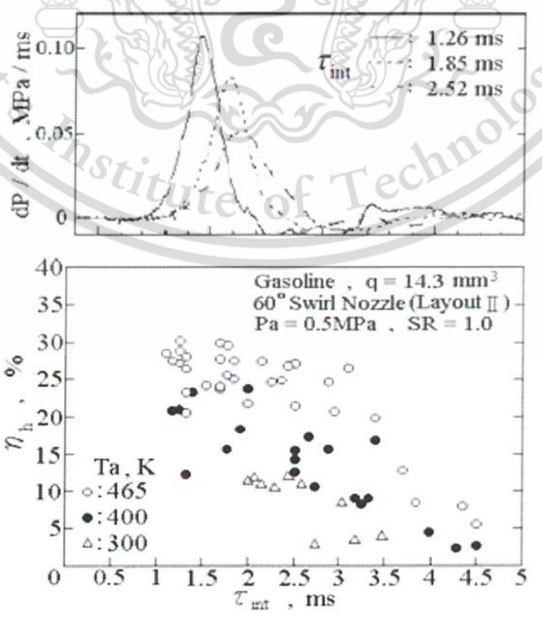


Figure 2.14 Duration between start of injection and start of spark to pressure rise rate (top) and heating efficiency (bottom) [25].

- Impact of Injection Timing

Z. Huang et al [26] concluded as when injection early, heat release pattern showed slow rate at initial and faster burning at late stage. Degree of charge stratification and remaining of turbulence from the fuel injection are both parameters to cause this phenomena. When injection timing is early, the decay of forming turbulence and weaker charge stratification occurs in cylinder results in heat release pattern and flame propagation in homogeneous mixture while retarding injection timing showed faster burn in initial stage and slow burn in later stage. This pattern seems like diesel combustion which 2 types of heat release rate can identify as initial stage showed pre-mixed phase following with diffusion phase in later stage. Hence, the retard injection can burn rapidly in the initial stage from the strong turbulence and later stage operates as diffusive flame similar to diesel combustion

J. Song et al [27] investigated the effect of the injection strategy on the mixture quality and combustion characteristics based on 2 overall equivalence ratio (0.8 and 1.0) with variation of injection timing (0° - 180° ATDC) results are shown as the following;

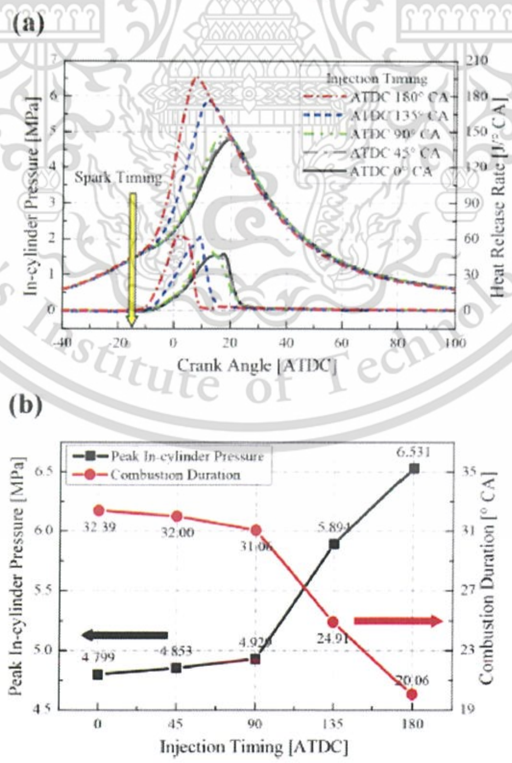


Figure 2.15 Combustion characteristics for various injection timings (a) in-cylinder pressure history & HRR and (b) peak in-cylinder pressure & combustion duration [27].

First, combustion characteristics, the in-cylinder pressure and combustion duration are affected to injection timing. Peak in-cylinder pressure at 180° ATDC is early and greater than 1 MPa when compare to early injection timing (0° ATDC) together with the combustion duration become lower when injection timing is retard as seen in figure 2.15. These caused by the difference in combustion speed, the faster flame can burn more fuel before cylinder expand and obtain higher energy from the combustion increased the pressure in the cylinder.

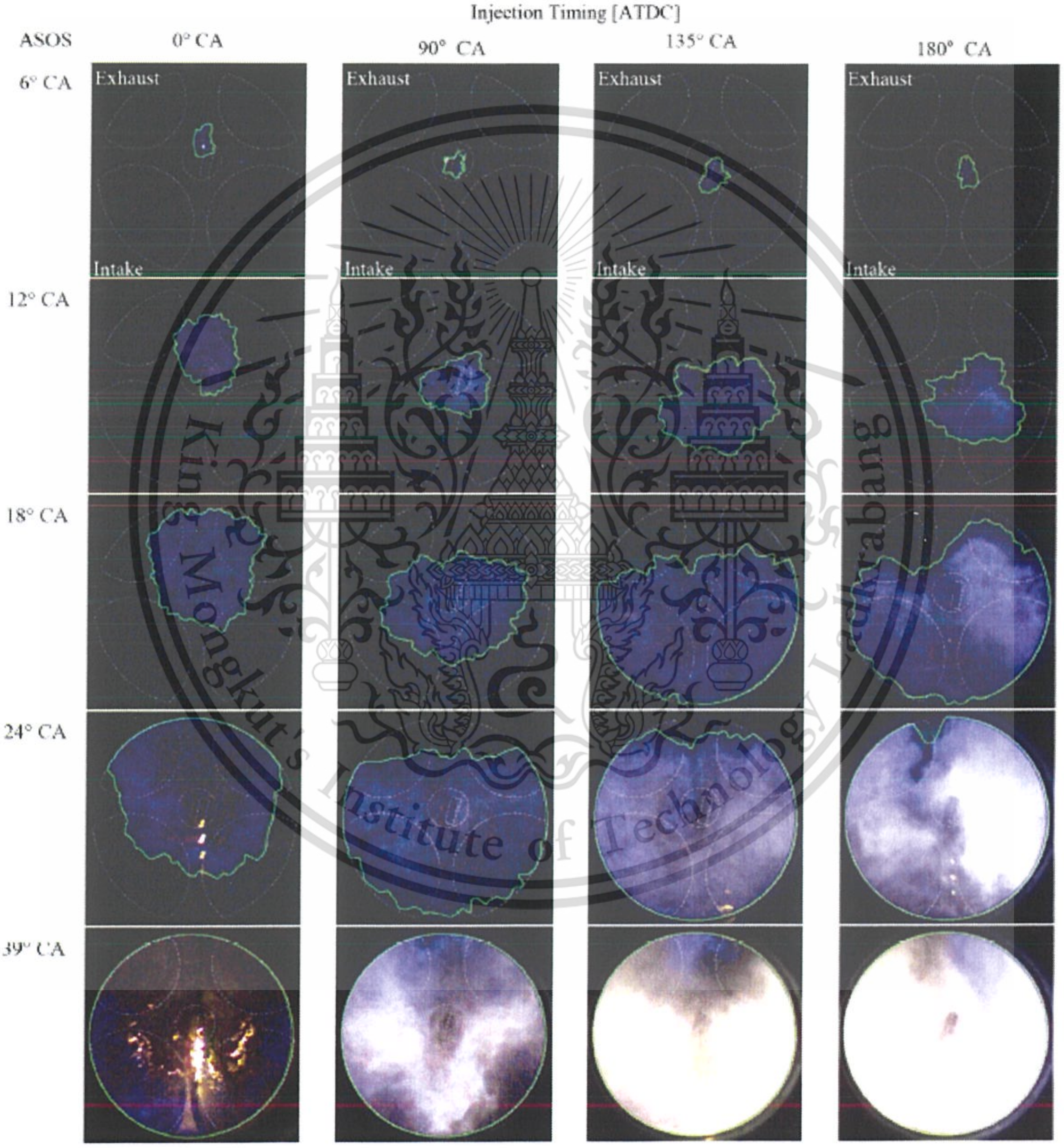


Figure 2.16 Flame propagation for various injection timings at equivalence ratio = 1 and ignition timing 15° BTDC [27].

Next, flame propagation image, as the in-cylinder pressure increased when retard injection timing that caused by increasing the combustion speed. To identify the relation between in cylinder pressure and combustion speed by using flame develop in a spherical shape is appropriate even though the real engine did not develop in spherical shape, this method is valid due to it is represents as the amount of fuel burned as shown in figure 2.16 flame growth speed sensitive to injection timing by flame growth was decreased when injection timing was advanced conform with figure 2.17; the pressure data, flame speed, flame radius showed similar trend that can imply as faster flame speed leads higher pressure. From this result, it can analyze that there are 4 factors influences to the flame speed; temperature, pressure, turbulence and equivalence ratio. Faster flame speed caused by high temperature with low pressure but both factors are not relate to injection timing. Hence, injection timing is independent variable. Only the turbulence and equivalence ratio are affected to flame speed.

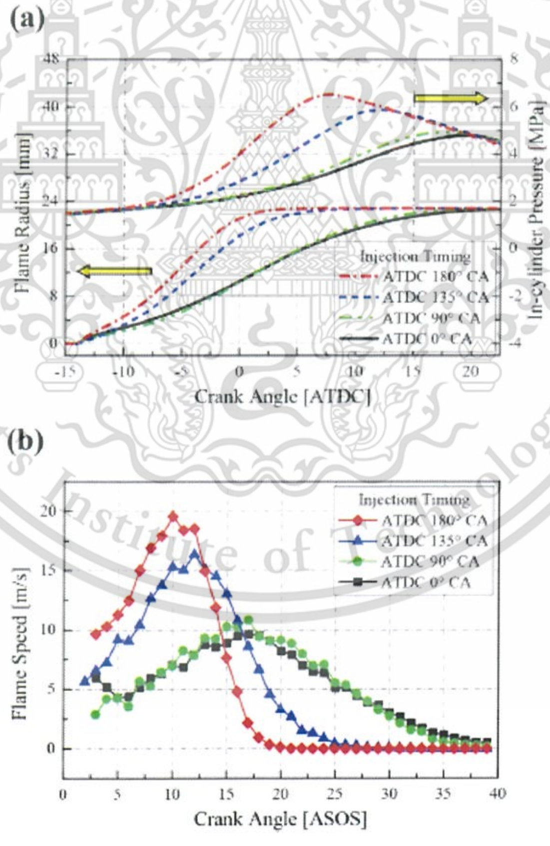


Figure 2.17 Flame speed and flame image from flame propagation analysis for various injection timings at equivalence ratio = 1 and ignition timing 15° BTDC [27].

Following with turbulence in the in-cylinder temperature, turbulence intensity supplied by intake flow is maximum at 45° ATDC due to fast intake flow occurs at initial stage of intake stroke and become slower at compression stroke due to turbulence was dissipated by friction. In case of DISI engine, fuel injected at high pressure causes the spray motion gain strong momentum supplied additional turbulence in the cylinder and strong enough to increase turbulence kinetic energy (TKE). From figure 2.18, the strong TKE caused by later injection due to shorter duration between injection and ignition. This support the reason why in-cylinder pressure and flame speed are dependent upon the injection timing. The stronger turbulence intensity from later injection timing causes faster combustion and high peak in-cylinder pressure.

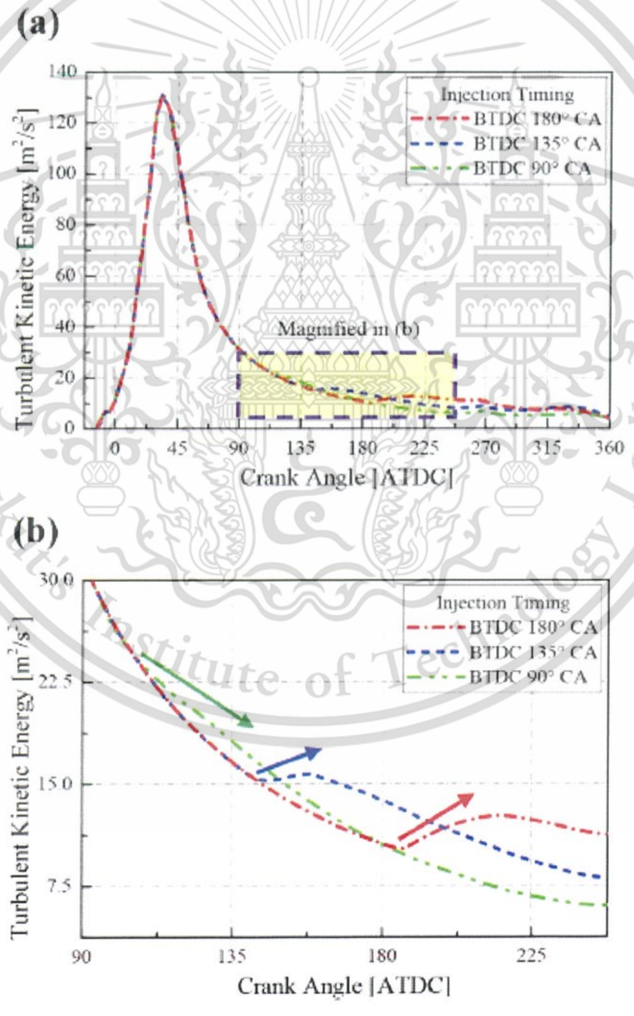


Figure 2.18 Turbulence Kinetic Energy from various injection timings at equivalence ratio = 1 and ignition timing 15° BTDC [27].

Last in equivalence ratio effects to the distribution on the flame propagation direction as seen in figure 2.19. The gasoline laminar flame speed was fastest at a slightly rich mixture equivalence ratio (around 1.15); combustion cannot occur in a lean mixture below the flammability limit. The flame propagation direction was determined by equivalence ratio distribution at early injection timing, the mixture at exhaust side richer than intake side. Hence, flame propagates from exhaust side whereas retard injection timing, flame propagates from intake side due to mixture is richer caused by strong tumble motion.

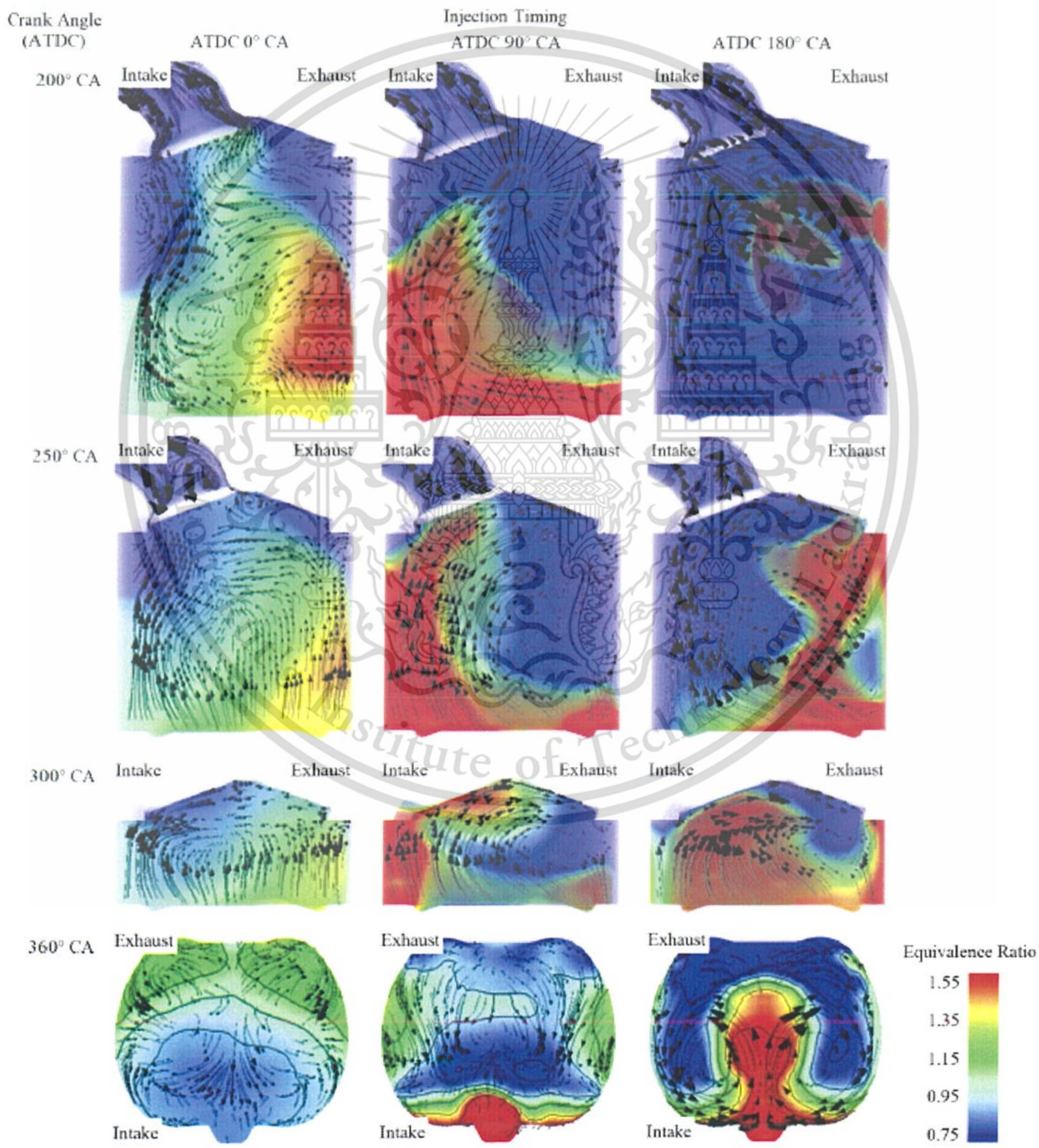


Figure 2.19 Equivalence ratio distribution and cylinder flow from various injection timings [27].

2.3 Ethanol

C_2H_5OH is the chemical formula of ethanol, consists of polar and non-polar which the polarity is important factor in physical and chemical characteristics of substances. The OH radical or hydroxyl group allows joining with other polar substances. The bond is relatively weak but strong enough to let ethanol more viscous and less volatile than other similar but less polar substances. From the fact that ethanol has both polar and non-polar substances. Hence, polar end can dissolve with polar substance likewise water where as non-polar end can dissolve with non-polar compounds such as diesel or gasoline.

To utilize ethanol as fuel in spark ignition engine which can be used by addictive (likewise ETBE) blended with commercial gasoline and neat fuel. The dominant characteristics of ethanol are oxygen content inside fuel called "oxygenated fuel" and higher of vaporization heat which can improve the combustion process by mount up volumetric efficiency from cooling effect. Moreover, the octane number of ethanol is higher than gasoline leads engine to operate with higher compression ratio or more degree of ignition timing. These can obtain higher of thermal efficiency. In case of emission, using ethanol was reduced. Oxide of nitrogen (NO_X) is lightening from low combustion temperature. Carbon monoxide and total hydrocarbon is lower than gasoline from oxygen content and lower carbon atom of ethanol.

Even though, ethanol is benefit to the engine combustion but the emission from aldehyde and formaldehyde conform to cold start stability are challenging to use ethanol in internal combustion engine.

From the effects to the combustion when using ethanol man studies were researched and found out many useful information to clarify factors of this fuel to combustion. Starting with the visualization of flame propagation and combustion characteristics of ethanol blended gasoline fuels from Prathan et al [28] by using gasoline, E20, E85, and E100 in constant volume combustion chamber. The result shown as at the same equivalence ratio (ø= 1), E100 has highest peak pressure of combustion and gasoline (E0) is lowest, mass fraction burned that determined from pressure of combustion is faster when blend higher percentage of ethanol. The factors which lead ethanol blended combust faster than gasoline due to oxygen content inside blended fuel enhance faster combustion together with the percentage of carbon in ethanol lower than gasoline causes faster flame propagation. Next, adiabatic flame temperature of ethanol is lower it can reduce heat loss to surroundings that improve combustion efficiencies. From above, those can accelerate flame speed which results in widespread of flame area and reduce combustion duration. Moreover, higher peak combustion pressure is dominant factor to extend area under the curve of otto-cycle. Hence, the power is increased along with emission is lessened from short duration of combustion.

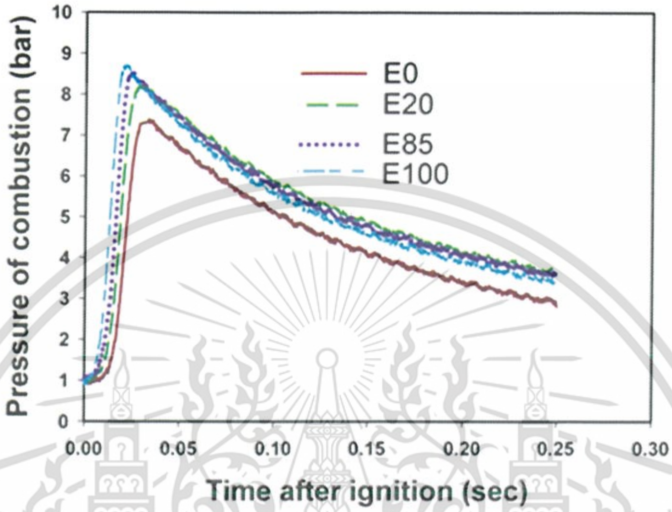


Figure 2.20 Pressure of combustion of difference ethanol blends [28].

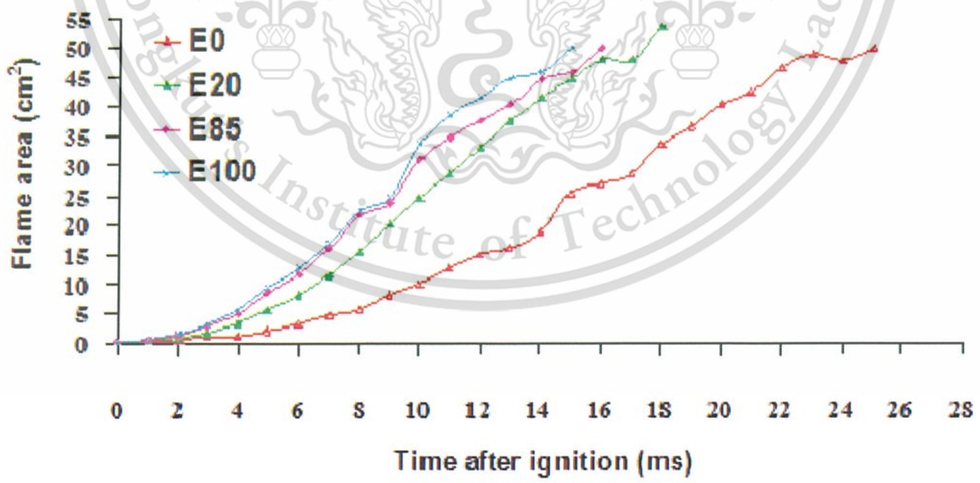


Figure 2.21 Flame area versus time after ignition of difference ethanol blends at the equivalence ratio = 1 [28].

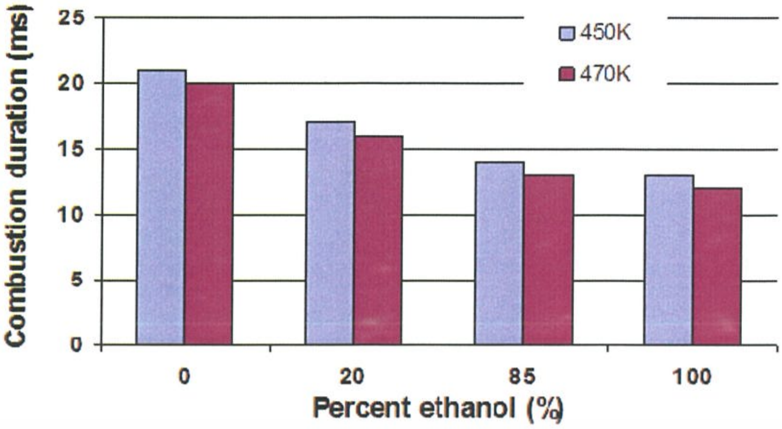


Figure 2.22 Effect of ethanol blends on combustion duration with different initial temperature [28].

Next, Piyaboot et al [29] studied the combustion characteristics of ethanol-gasoline blends for stratified charge engine by using constant volume combustion chamber. This experiment conducted on variation of ethanol blended to swirl intensity, lean limit investigation, mass fraction burn and pressure rise rate. The results indicated that

1. Swirl intensity enhances stratification degree which directly impact to lean limit range by diffuse amount of fuel to form stoichiometric in wider area.

2. Higher percentage of ethanol can extend lean limit due to more fuel injected may increase the change of fuel to form near stoichiometric. Furthermore, the peak pressure is higher and duration of combustion is shorter as well.

3. In case of low swirl flow, the fuel and air will be form in very rich mixture near center of combustion chamber sue to fuel injected did not diffuse quite much. So, the initial combustion and ignition delay is long.

4. Even though, ethanol has higher octane number and low rate of vaporization, ethanol composes of single component whereas gasoline or gasoline blended with ethanol consists of many components with various carbon atoms and structures. Then, component with relatively low boiling point which evaporate easier and faster while components with high boiling point will evaporate slower. Hence, ethanol tends to evaporate fastest.

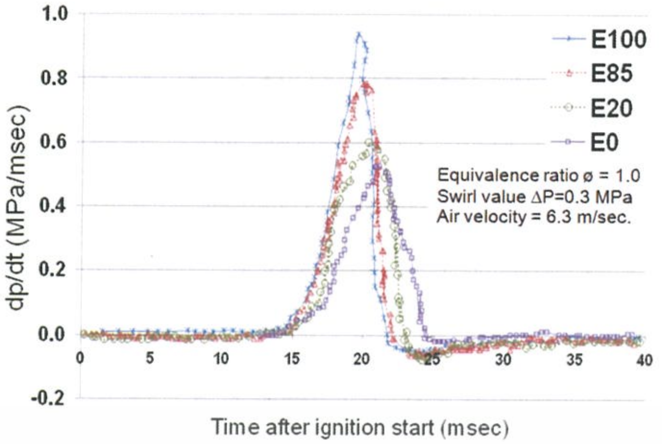


Figure 2.23 Effect of ethanol blends on rate of pressure rise with same equivalence ratio, swirl ratio and air velocity [29].

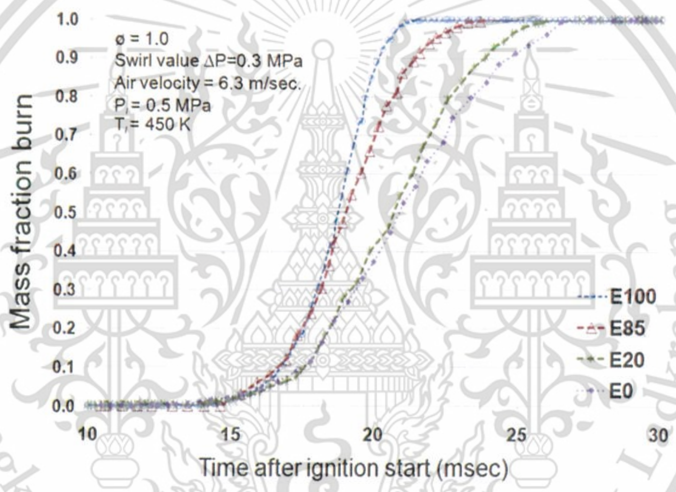


Figure 2.24 Effect of ethanol blends on mass fraction burn [29].

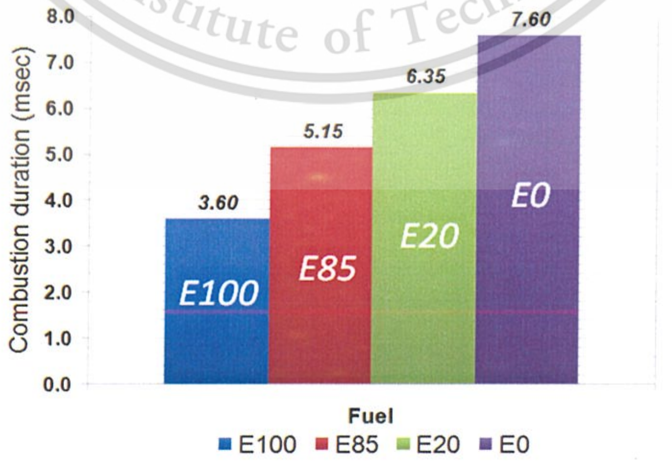


Figure 2.25 Effect of ethanol blends on combustion duration with equivalence ratio = 1, 0.3 MPa (6.3 $^{m}/_{s}$) swirl intensity [29].

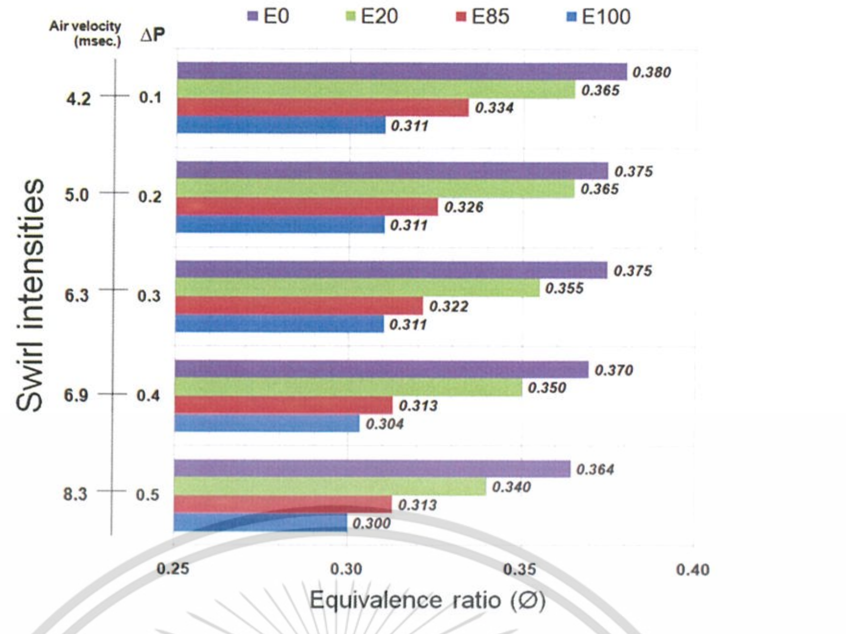


Figure 2.26 Effect of ethanol and various swirl intensity on lean limit [29].

Investigation of ethanol DISI engine performance and emission from Athiwat et al [30] focused on the wall guided DISI engine fuelled with gasoline and gasoline blended with ethanol. The results shown that remaining of particle emission of gasoline is higher than that ethanol in all cases due to ethanol oxygen molecule in ethanol can improve more complete combustion of the engine additional with particle emission in stratified mode is higher than homogeneous mode from the late injection causes less fuel propagation from rich fuel region around spark plug. In case of CO, HC, and CO_2, ethanol can reduce emissions more than gasoline due to lower carbon content of ethanol but in term of fuel consumption is higher from much amount of fuel injected from low heating value. Last in performance, ethanol should advance ignition timing because of long ignition delay. Maximum brake torque is increases at optimum ignition timing.

In combustion behavior, to study the effect of fuel to spray characteristic is essential due to the vaporization and spray pattern are dominant effect to the mixture distribution. Especially in DISI engine, the spark timing should be proper to the equivalence ratio at the tip of electrode. So, the spray pattern is important factor to preparation of mixture.

The spray characteristics of swirl injector fuelled with methanol and ethanol from Xibin Wang et al [31] shown that the spray pattern of gasoline, methanol and ethanol are similar. At low back pressure ambient condition, the spray pattern is hollow cone with wide angle and initial spray slug at the tip where as high back pressure is formed in the middle part of spray. These can conclude as increasing of injection pressure can enhance swirl intensity.

However, according to fuel properties, the gasoline has higher vapor pressure than methanol and ethanol. The gasoline spray should be shorter penetration and has wider cone angle but the experimental result is in contrast. These can explained by methanol and ethanol are consist of single component while gasoline contains many components with various carbon atoms and structures and it has largely of vapor pressure from component with low boiling point that evaporate faster and easier whereas the component with higher boiling point evaporate slowly. So, the initial spray of gasoline is different from both fuel and the main spray of gasoline has long penetration and small cone angle.

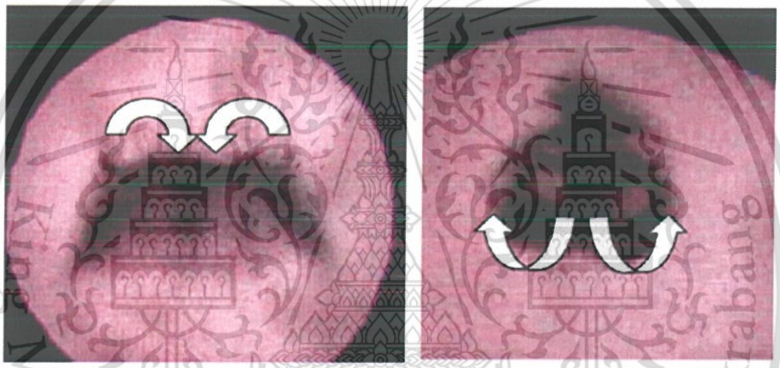


Figure 2.27 Spray pattern and entrainment of air with different back pressure (0.1 MPa on left side and 1.0MPa on right hand side) [31].

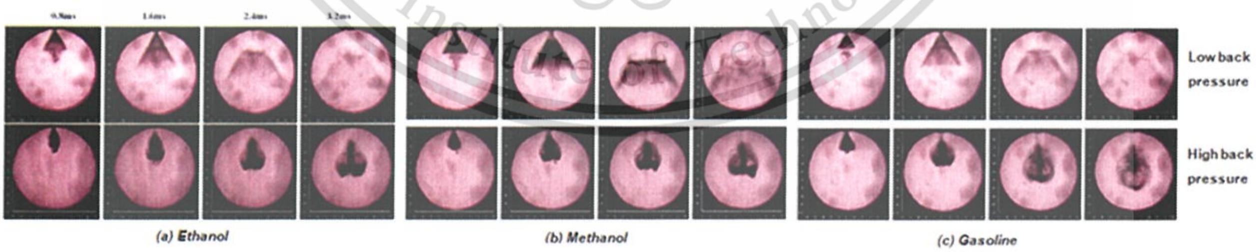


Figure 2.28 Spray pattern and entrainment of air with different back pressure (0.1MPa on left side and 1.0MPa on right hand side) [31].

Chapter 3

EXPERIMENTAL AND PROCEDURE

3.1 Experimental Apparatuses and Experimental Set Up

3.1.1 Engine

Mitsubishi 4G93 GDI with the wall-guided direct injection spark ignition engine is used in this experiment and controlled by standalone engine control unit (ECU) and fuel supply system is feed test fuel from tank to high pressure pump which is cam-driven type and located beside intake camshaft. The plunger in high pressure pump pressurizes fuel from tank which pressure is approximately 0.3 MPa to high pressure level (4-7 MPa). Then, the pressurized fuel delivers to injector and discharged to the combustion chamber. The specification of this engine shown in table 3.1.

Table 3.1 Direct Injection Spark Ignition Engine Specifications

Description	Specification
Model	MITSUBISHI 4G93 GDI
Type	In-Line OHV, DOHC 16 Valve
Number of Cylinder	4
Combustion Chamber	Pentroof Type
Cooling System	Water-cooled forced circulation
Displacement (CC)	1834
Compression Ratio	12 : 1
Bore x Stroke (mm)	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

Compared with gasoline, using ethanol as a fuel leads to benefits by outstanding properties for instance high research octane number (RON) which causes better anti-knock quality from increased of compression ratio that obtained better volumetric efficiency, the stoichiometric air fuel ratio is lower due to the fact that it has oxygen content inside its molecule leads to complete of combustion and reduce emissions from surplus of oxygen to react remains. However, ethanol must injected more quantities because of lower heating value to achieve same of output energy and cold start problem from high initial boiling point (IBP) when compare with gasoline as shown in table 3.3.

3.1.2.1 Air-Fuel Ratio Calculation

Air-fuel ratio calculation is determined from oxidation equation at adiabatic flame temperature condition. Stoichiometric ratio for hydrocarbon fuel showed as.

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

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

Pure gasoline has the chemical formula as $C_{8.26}H_{15.5}$ .thus,

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

For gasoline blended with ethanol blend such as E20, E85 and also E100 following equations can be used;

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

Where

$x=$ mole fraction of the Ethanol

$y=$ mole fraction of the Gasoline

And

$$AFR_{Stoi} = \frac{(3x + 12.13y)(4.773 \times 29)}{46x + 114.8y} \quad (3.5)$$

All of stoichiometric calculation can portrays in table 3.2 below,

Table 3.2 Determination of air-fuel ratio at stoichiometric condition

FUEL	Mass (kg)	Molecular Weight (MW)	C_p ($^{kJ}/_{kg.k}$)	LHV ($^{MJ}/_{kg}$)	Moles	Mole Fraction	AFR (STOICH)	Total Molecular Weight of Fuel	LHV of Fuel ($^{MJ}/_{kg}$)
E0									
$C_{8.26}H_{15.5}$	1.00	114.18	1.93	44.00	0.00876	1.000	14.63	114.18	44.00
C_2H_5OH	0.00	46.07	1.70	26.90	0.00000	0.000			
Total	1.00				0.00876	1.000			
E20									
$C_{8.26}H_{15.5}$	0.80	114.18	1.93	44.00	0.00701	0.617	13.51	88.12	40.58
C_2H_5OH	0.20	46.07	1.70	26.90	0.00434	0.383			
Total	1.00				0.01135	1.000			
E85									
$C_{8.26}H_{15.5}$	0.15	114.18	1.93	44.00	0.00131	0.066	9.87	50.60	29.47
C_2H_5OH	0.85	46.07	1.70	26.90	0.01845	0.934			
Total	1.00				0.01976	1.000			
E100									
$C_{8.26}H_{15.5}$	0.00	114.18	1.93	44.00	0.00000	0.000	9.03	46.07	26.90
C_2H_5OH	1.00	46.07	1.70	26.90	0.02171	1.000			
Total	1.00				0.02171	1.000			

3.1.2.2 Fuel Distillation

Fig. 3.1 and appendix A shows Distillation curves for ethanol blends which are provided in the supporting information. The ethanol- gasoline blends' trends are similar to the methanol-gasoline blends. A same type effect that has been observed with vapor pressures, adding small percentages of alcohol to gasoline increases the pressure more than the large ones. The observations are consistent with the well-known phenomenon of azeotropic behavior of mixtures of methanol and the hydrocarbons of the base gasoline. For true azeotrope, the liquid molar composition is identical to the composition of the vapors formed.

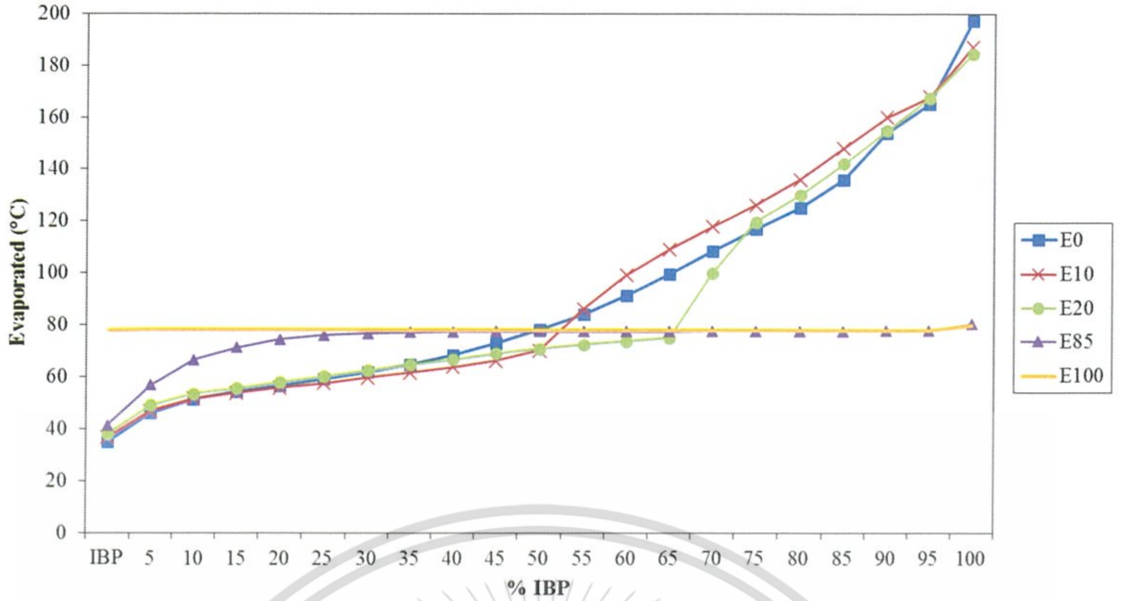


Figure 3.1 Gasoline (E0), Gasohol (E10 E20 E85) and Ethanol distillation curve.

During distillation of an azeotrope, both compositions remain stable until the fluid is totally evaporated. Although, the distillation curve for a true azeotrope would become flat and remain in that way until the distillation is complete. With the methanol- gasoline blends, the distillation curve never becomes flat and shows a steady rise in the late period. According these considerations, it may be more appropriate to describe that is a "near-azeotropic mixture" of the alcohol and the gasoline hydrocarbons. While the addition of 10 and 20% ethanol leads to little, or no, discernible change in IBP (initial boiling point), there is a substantial decrease in distillation temperature (i.e., increase in volatility) over the middle portion of the distillation curve. Observing with methanol blends, the addition of small amounts of ethanol (E10) gives the maximum increase rate in volatility for the first approximately 30% of the distillation curve. As the fraction of ethanol increases (from E10 to E20), each distillation curve moves progressively closer to that of gasoline over the first approximately 30% of volume distilled. The IBPs for E85 blends are substantially much more than those of the base gasoline and trend toward that for pure ethanol (78 °C). Similarly observed for methanol blends, the extent of the deviation in the initial portion of the distillation is substantially larger than expected to simply from the amount of ethanol in the blend. For example, the distillation temperature for E5 increases slower than the base gasoline from T_{10} to T_{30}, but then increases sharply at T_{50} and thereafter, approaching that of gasoline.

These observations indicate that ethanol forms a near-azeotropic mixture with the gasoline hydrocarbons and consequently the impact of ethanol on fuel volatility is substantially greater than expected for ideal mixtures.

Table 3.3 Fuel properties [32, 33]

Fuels properties	Gasoline [32]	E10	E20	E85	Ethanol [33]
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
Specific heat, kJkg-1K-1	2				2.40
Lower, hheating val.,, 103 MkJ/lkg	44.00	40.97	40.60	29.50	26.90
Research octane number	92.4	98.1	98.3	101.6	108.60
Motor octane number	81.2	82.3	84.6	91.1	92
$(R+M)/2$	86.8	90.2	91.45	96.35	100
Stoichiometric air/fuel ratio	14.70	14.05	13.51	9.87	9.03
Distillation temperature, °C					
Initial boiling point, IBP	35	36.5	37.8	41.3	77.6
10 vol%	51.5	51.6	53.5	66.6	77.8
20 vol%	56.5	55.7	57.8	74.4	77.9
30 vol%	61.8	59.7	62.5	76.8	77.9
40 vol%	68.6	63.8	66.8	77.4	77.9
50 vol%	78.2	70.2	70.8	77.5	77.9
60 vol%	91.5	99.4	73.7	77.6	77.9
70 vol%	108.6	117.9	99.9	77.7	78
80 vol%	125.2	136.1	130	77.7	78
90 vol%	154	160.2	155	77.8	78
End boiling point	197.3	187.2	184.6	80.5	80
Adiabatic flame temperature, K	2266		2203.15		2197

3.2 Methodology

3.2.1 Measurement and Command System

The measurement and command system in this experiment and schematic diagram showed in figure 3.2 and 3.3 where the specification showed in table 3.4. This engine is commanded by Tokyo Plant ED-150-LC eddy current dynamometer with in-house LabVIEW program. The program can measure and store information in real-time such as torque, power and brake specific fuel consumption which includes 2 operation modes as constant speed and constant load.

The intake air flow is measured by Sokken LFE25B laminar airflow meter with P277 digital manometer. To measure temperature, the type K class 1 thermocouple is used by located on air inlet tract, exhaust gas, lubrication at oil-pan and coolant. Moreover, emissions had measured by 2 systems; gaseous emission by MRU model SWG200-1 gas analyzer which measured in PPM and resulted into emission index ($^{g}/_{kW\text{-}hr}$), particle emission is sampled by OKUDA DSU-240 smoke detector in term of percentage concentration (%). Fuel injection and ignition of engine are controlled by DTA s60pro standalone ecu. All conditions were repeated for 5 times.

Figure 3.2 Measurement and command system of this experiment.

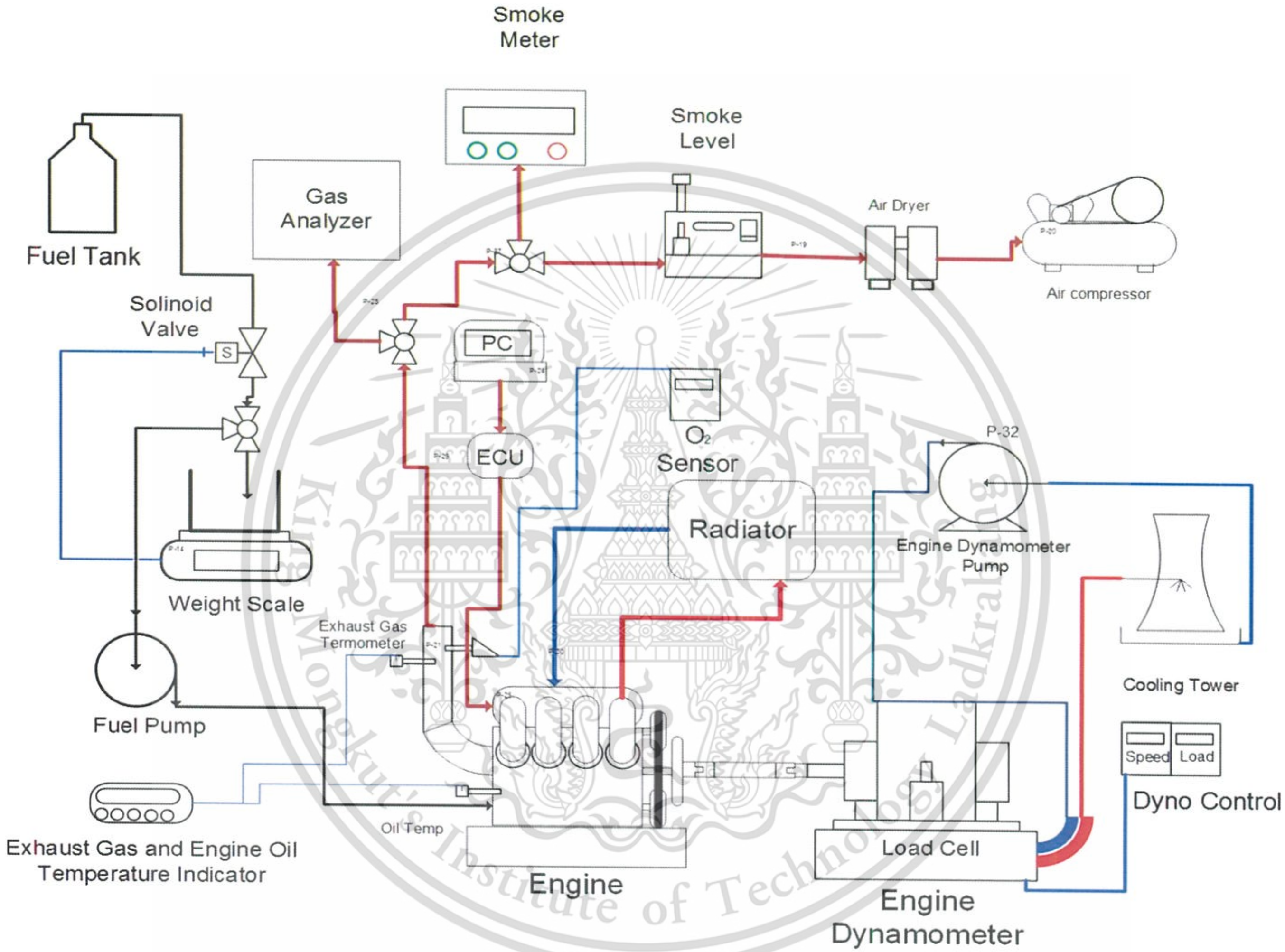


Figure 3.3 Schematic diagram of experimental setup

Table 3.4 Specification of measurement hardware

No.	Name	Unit	Description	Specification	Model
1	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
2	Thermocouple	°C	Temperature measurement - Ambient - Air inlet at manifold - Exhaust Pipe - Coolant - Lubricant	• Range: -40 – 375 °C • Tolerances: ±1.5 °C	Type K Class 1
3	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)
4	DAQ	-	To control, measure and store datum from engine dynamometer.	-	National Instrument LabVIEW system.

Table 3.4 Specification of measurement hardware (Cont'd)

No.	Name	Unit	Description	Specification	Model
5	Engine Dynamometer	-	Applying a 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
6	Smoke Meter	%	Measuring the concentration of the particulate matter in the exhaust gas.	-	OKUDA DSM-240
7	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)

No.	Name	Unit	Description	Specification	Model
8	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
9	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

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 which shown calculation in appendix B. 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. All conditions conducts 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°	

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 ($^{kg}/_{m^3}$)	1.2

Remarks: Vehicle speed limit in Thailand based on [34] and road gradient refers to [35].

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 Resistance (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 Resistance (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 categorized 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 Engine power and torque calculation

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

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

3.2.3 Experimental Methods

There are 2 main parts to classify experimental methods as performance together with efficiency and emissions. To describe the impact from variation of injection and ignition timing and benefits from using ethanol as oxygenated fuel. Those can explained by flow chart.

3.2.3.1 Performance and Efficiency

The experimental engine fueled with gasoline and gasohol fuels. All test was controlled air fuel ratio to be stoichiometric or lambda = 1. Then, the study parameters were measured by adjust injection timing, injection duration and ignition timing at maximum brake torque as shown in flow chart in figure 3.4.

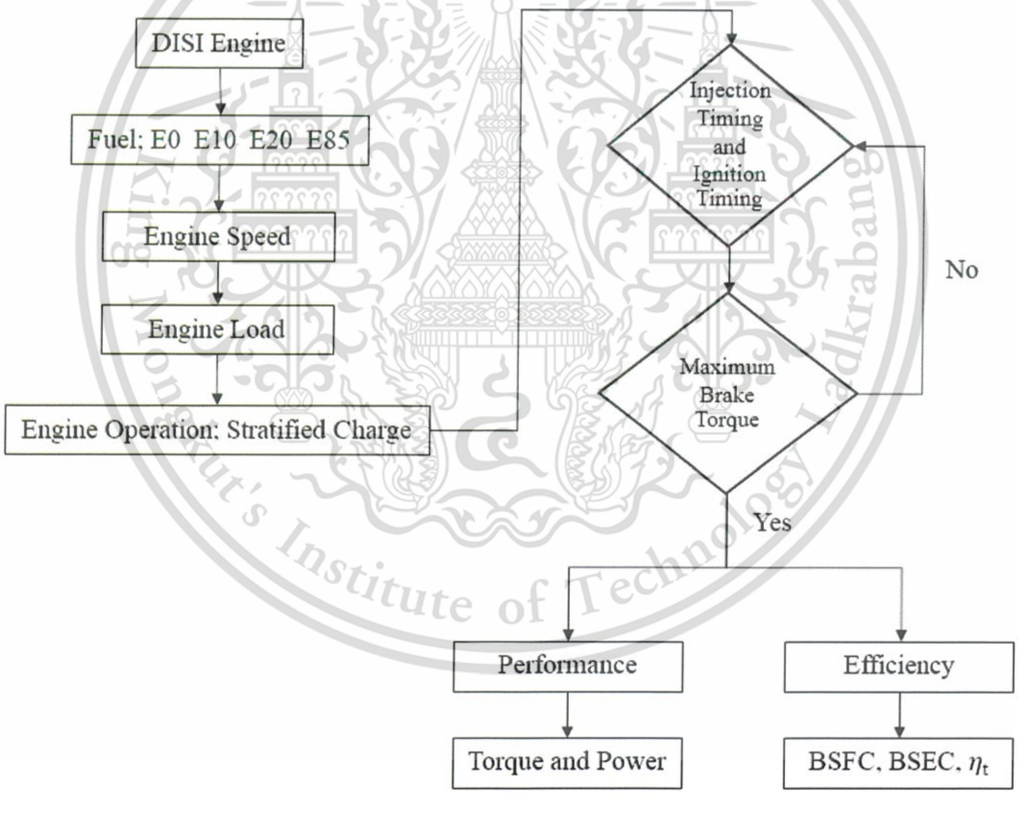


Figure 3.4 Performance and efficiency experimental method.

3.2.3.2 Emissions

Emissions from DISI engine had operated in same conditions as previous section. Emissions from exhaust pipe were trapped directly and categorize in 2 section as Emissions in gaseous phase; Carbon monoxide, Total hydrocarbons (THC) and Oxide of nitrogen (NO_x) were measured by gas analyzer coped with emissions in solid phase; particle emissions were sampled from filter paper by smoke meter. After that, particle emissions in filter paper had analyzed into micro-scale by Scanning Electron Microscope (SEM) in order to investigate accumulated size from the impact of injection timing and ignition timing and the effect from fuel between gasoline and gasohol as showed in figure 3.5.

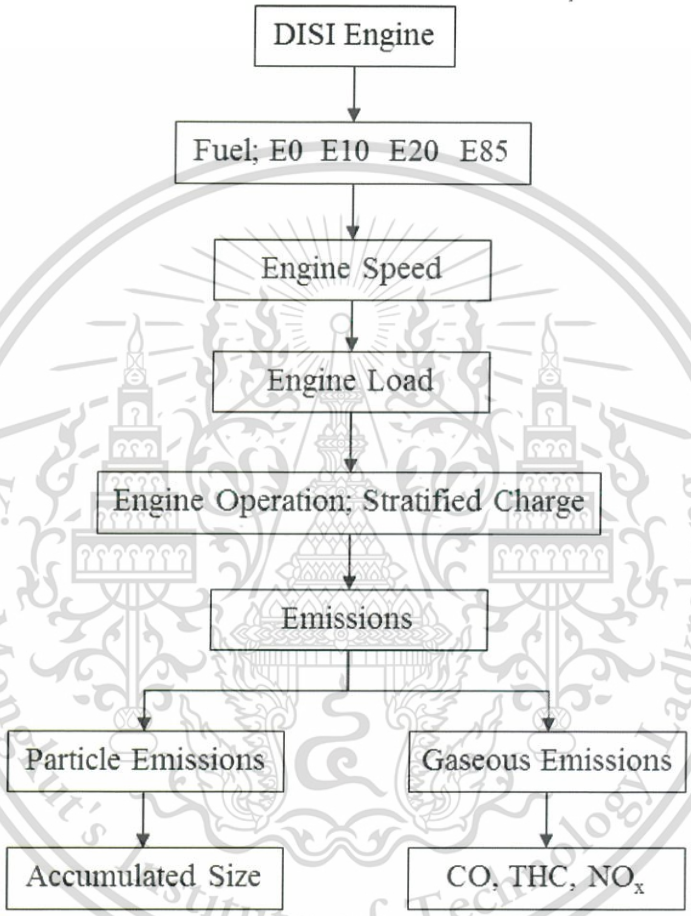


Figure 3.5 Emissions experimental method.

3.2.3.2.1 Emissions Analysis: Scanning electron microscope

The operation of scanning electron microscope (SEM) is beams of electrons to initiate a several of signals to surface of specimens. These send signal and convert information into chemical composition, crystalline structure, morphology (texture) and vice versa. In most applications, data were sampled on selection area of the surface. Areas ranging from approximately 1 cm to 5 microns of width can be sampled by conventional SEM techniques. The SEM components contains; The generation of electrons, Column down where electrons move along with electromagnetic lenses, Electron detector, Sample chamber and personal computer which figure 3.6.

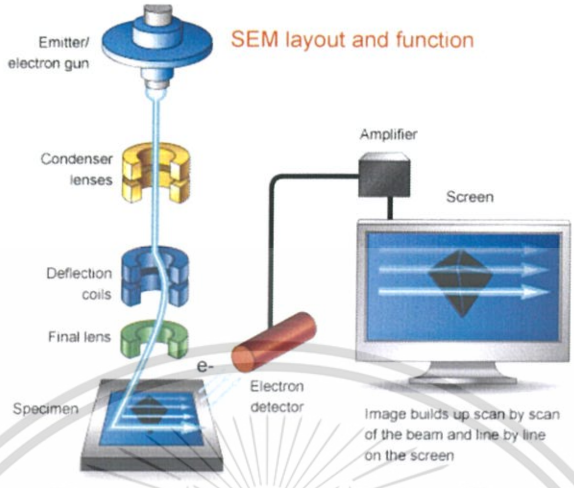
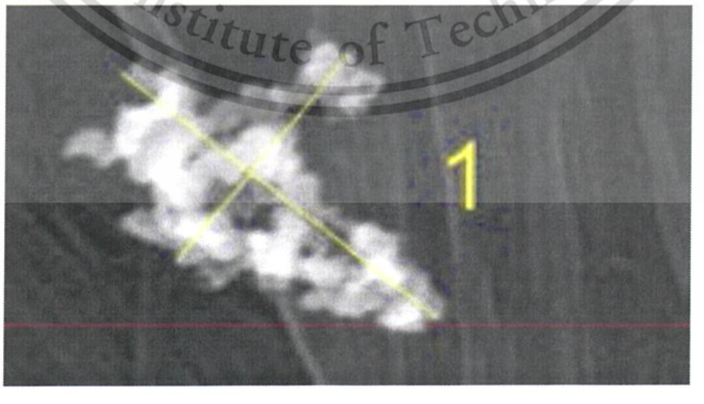


Figure 3.6 Scanning electron microscopy schematic diagram [36].

3.2.3.2.2 Emissions Analysis: Particulate Matter Size Distribution

The accumulated particulate matters from direct injection spark ignition engine are trapped by paper filter from smoke meter. A paper filter is taken image and analyzed in morphology into micro - scale with scanning electron microscope. After that, the images from SEM analysis are measured dimension by average of two diagonal lengths as shown in figure 3.7. 100 particles of each condition had measured for accumulated particulate matter size distribution analysis.



$$Average\ Diameter = \frac{D_1 + D_2}{2}$$

Figure 3.7 Accumulated particulate matter size distribution analysis.

Chapter 4

RESULTS AND DISCUSSIONS

4.1 Effect of Injection Timing and Ignition Timing

4.1.1 Efficiencies and Emissions

On stratified operation, the fuel is transmitted to the cylinder bowl during the compression stroke. The air motion called reverse tumble motion delivers fuel towards the top of the combustion chamber to generate mixture cloud at the center around the vicinity of spark plug and enclose with excess air. This Strategy high AFR and then improved better fuel economy when compare with homogeneous operation. Fuel could formation and combustion depends on injection and ignition timing.

Figure 4.2 – 4.13 illustrated the impact of injection timing and ignition timing on efficiencies and emissions of tested fuel in term of contours. The results are discussed at the maximum break torque (MBT), the optimum timing which explained as the minimum advance of timing. The optimum injection timing of gasoline (E0) was found at CAD (crank angle degree) of 100° followed with E20 and E85 at 100° and 105° bTDC together with the optimum ignition timing at 27° of E0 followed with E20 and E85 at 27° and 30° bTDC, respectively. These shown that to blend fuel with ethanol causes both injection and ignition timing become advance than that gasoline due to the low heating value of fuel blended with ethanol is lower than that of gasoline. Hence, plenty of blended fuel should injected into the engine for achieving the same heat energy output. Moreover, higher latent heat of vaporization causes air intake temperature in combustion chamber lower because of evaporation of alcohol which leads to increased volumetric efficiency together with too delay of ignition timing from high octane number. Hence, advanced both timings are required. Focusing on the tendency of tested fuels on efficiencies and emissions parameters are similar, advancing ignition and injection timing is harmful to the fuel economy than retarding timings from maximum brake torque timing (MBT). Oxides of nitrogen (NO_X) emissions are located on MBT as maximum and decreased filter smoke number when both timings retard away from MBT. The

Tendency of total hydrocarbon and carbon monoxide depend on phasing and quality of combustion. This results can refer some interested information from P.J. Shayler et al. [37] as the relation of injection timing and spark timing to the mass-fraction burn. At 50% of mass fraction burn (MFB) location is not effect to too advance ignition timing due to flame development period (defined as 0 – 10 % of mass fraction burn) occurred in over-rich region and slightly vaporized of mixture cloud as illustrated in figure 4.1a. As shown in figure 4.1b, the increase of ignition timing to maintain 50% mass fraction burn equalize the increase of flame development angle by too advance ignition timing from MBT, the increasing of spark advance and flame development angle is similar (ie; 40° BTDC of spark advance equal to approximate 40° of flame development angle). On the other hand, when ignition timing is too retard from MBT, the combustion in leaner mixture cloud is generated from later phasing that prolong burning duration and retard the 50% of mass fraction burn as well.

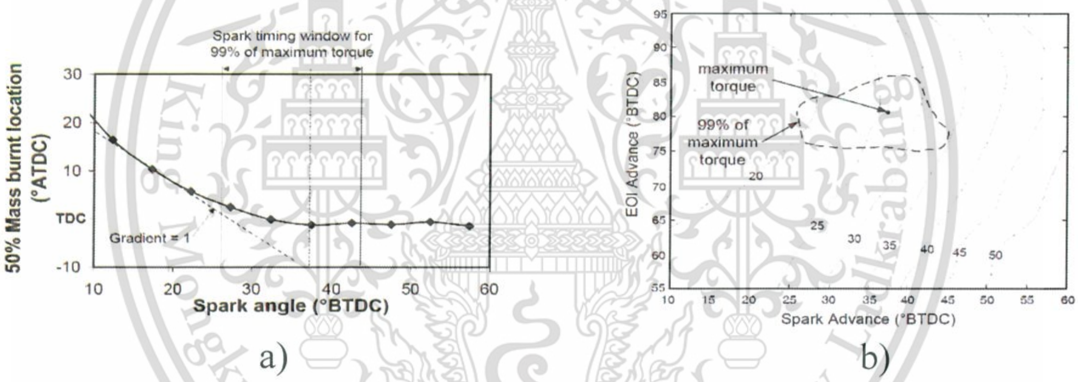


Figure 4.1 P.J. Shayler et al. stratified operation at low load analysis by a) variation of 50% MFB location against ignition timing, b) ignition timing and injection timings effects on flame development angle [37].

From the result contours, there are 4 main sections to define the characteristics of efficiencies and emissions. Starting with both injection timing and ignition timing are retarded causes efficiencies drop because of combustion process occurs at late phase of cycle. Hence, flame development angle and combustion process are generated at similar rate which leads to decreased of in-cylinder temperature and peak pressure. So, the quantity of NO_X is decreased and efficiencies become deteriorate.

Next, when injection timing and ignition are too advanced causes the poorer mixture stratification from inadequate duration of air fuel formation.

So, lean zones and partial burns are occurred by flame development happened on much lower cylinder pressure at early phase of combustion process which contributes to delay angle is extended and increase duration of fuel cloud to grow and diffuse to surroundings. So, the main part of combustion is over lean mixture that leads to low temperature of combustion. From mentioned this results in less oxidation rate of unburned hydrocarbon along the cycle. Hence, THC, CO and FSN are raised and NO_X is lower. Furthermore, if both timings are too early causes local AFR during combustion is too lean. Then, it results in misfire, partial burns and surplus of THC.

Next, when engine operates on advance injection timing and retard the ignition timing means the duration between end of injection and start of spark is prolonged, the duration of fuel clod formation is also increases. Referring to the reason same as both timings advanced, local AFR is too lean. So, the engine efficiencies is dropped while emissions are lessened as well.

Last, in injection timing retard and ignition timing advance known as the duration between end of injection and start of spark is too short. This causes insufficient duration for fuel cloud formation and diffuses to air surroundings. Hence, the mixture at spark discharge is too rich and leads CO, THC and FSN extremely high. Moreover, if ignition timing is extremely advanced, misfire is occurred because of spark discharged before fuel clod arrives to the vicinity of spark plug.

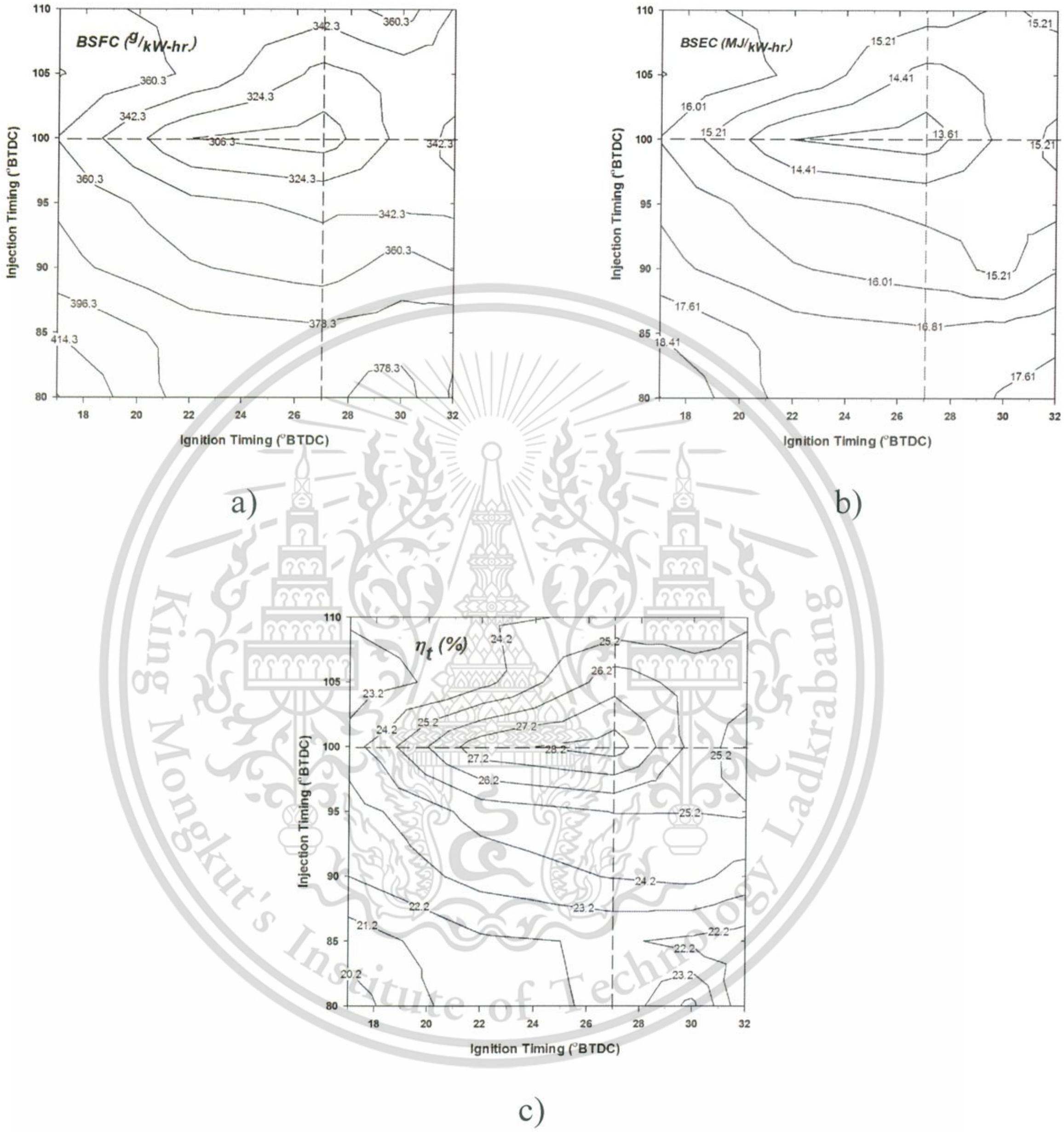


Figure 4.2 Effect of injection timing and ignition timing of E0 on efficiencies as a) BSFC, b) BSEC and c) Thermal efficiency.

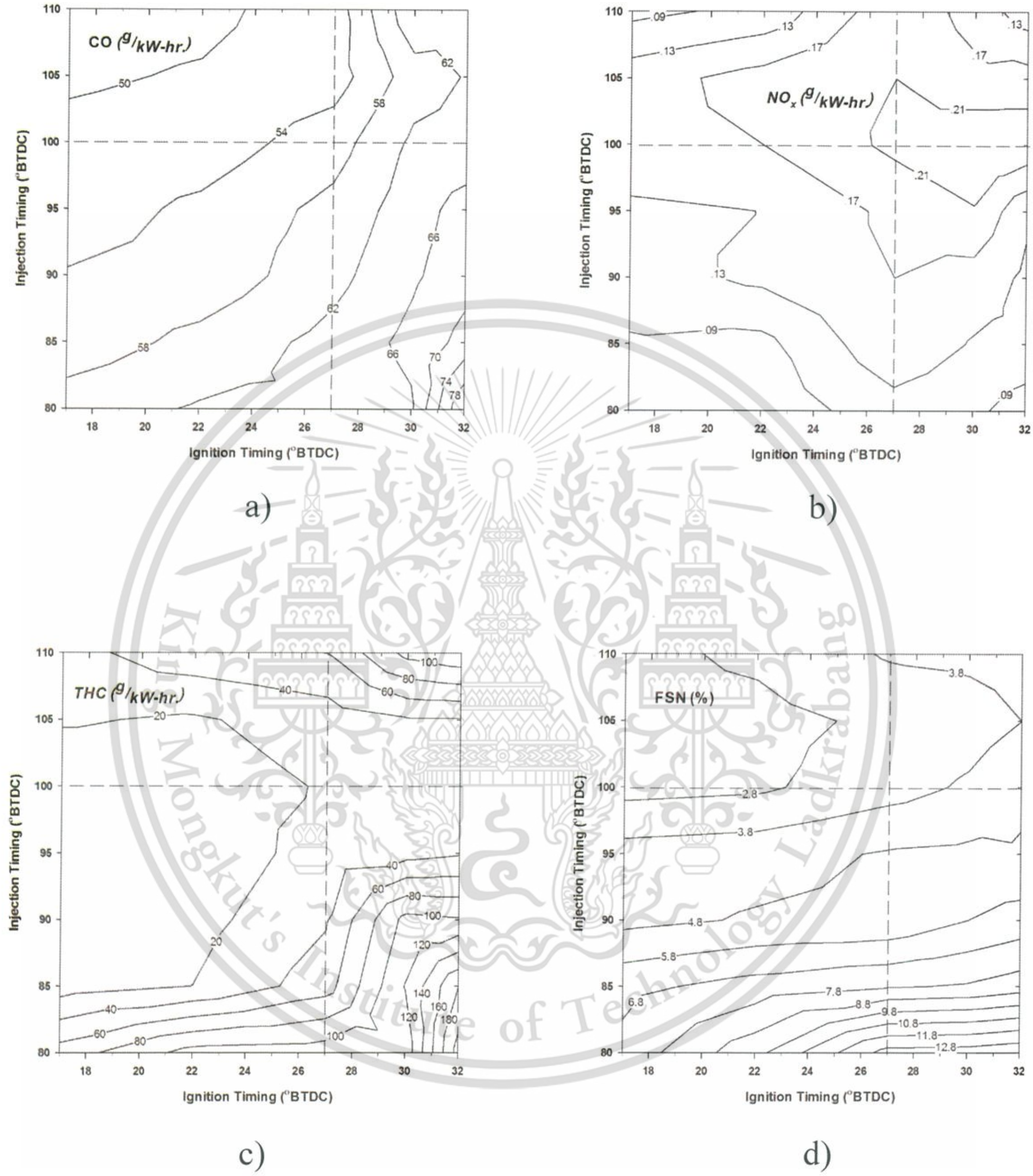


Figure 4.3 Effect of injection timing and ignition timing of E0 on emissions as a) CO, b) NO_x, c) THC and d) Filter smoke number (FSN).

Figure 4.3a, 410a and 4.14a showed the effect of injection timing on efficiencies. The trends of tested fuel are similar where summarize result shown in table 4.1. The maximum brake thermal efficiency coped with minimum brake specific fuel consumption and brake specific consumption is lowest at optimum injection timing due to this timing occurs the peak in-cylinder pressure and maximum heat release and their locations are approach to top dead center. This leads stratified mixture distribution become to ideal. Moreover. It obtains shortest ignition delay and fastest flame propagation that reduce heat loss in combustion process as well. To advance of injection timing produces long duration for mixture preparation, this causes mixture around the vicinity of spark plug becomes lean. Furthermore, too much of uniform mixture which leads to lean bring about unfavorable for ignition of the mixture and propagation of flame. Hence, the efficiencies is lower than that optimum. When injection timing injects too late, the duration of fuel formation is not adequate to mixing. So, the fuel cannot completely evaporate and diffuse to the spark plug and it cannot form rich mixture rich mixture near the spark plug that would favor for flame propagation [38]. In this case, efficiencies are not satisfied.

Figure 4.3b, 4.10b and 4.14b illustrated the effect of injection timing on exhaust emissions where summarize result show in table 4.2. It can be seen that NO_x slightly decrease, CO emission decrease significantly and THC rise up as advance injection timing from MBT. This can discuss that using early injection timing leads to more homogeneous mixture, these causes higher peak in cylinder pressure will increase the fuel injected moving into the top land of crevice region. So, it obtains more THC whereas the expansion and exhaust stroke bring about lower combustion temperature from this injection strategy will show down the THC oxidation. Then, both factors lead to rise up of THC. Meanwhile, the higher burning rate, high in-cylinder pressure and temperature are the main factor to increasing of NO_x. So, NO_x is reach maximum at MBT and slightly drop because of the location of peak in cylinder pressure and rate of heat release are not approach to top dead center (TDC) . Consequently, CO become lower due to more homogeneous mixture from early injection timing that leads high oxygen concentration available in the mixture cloud [39].

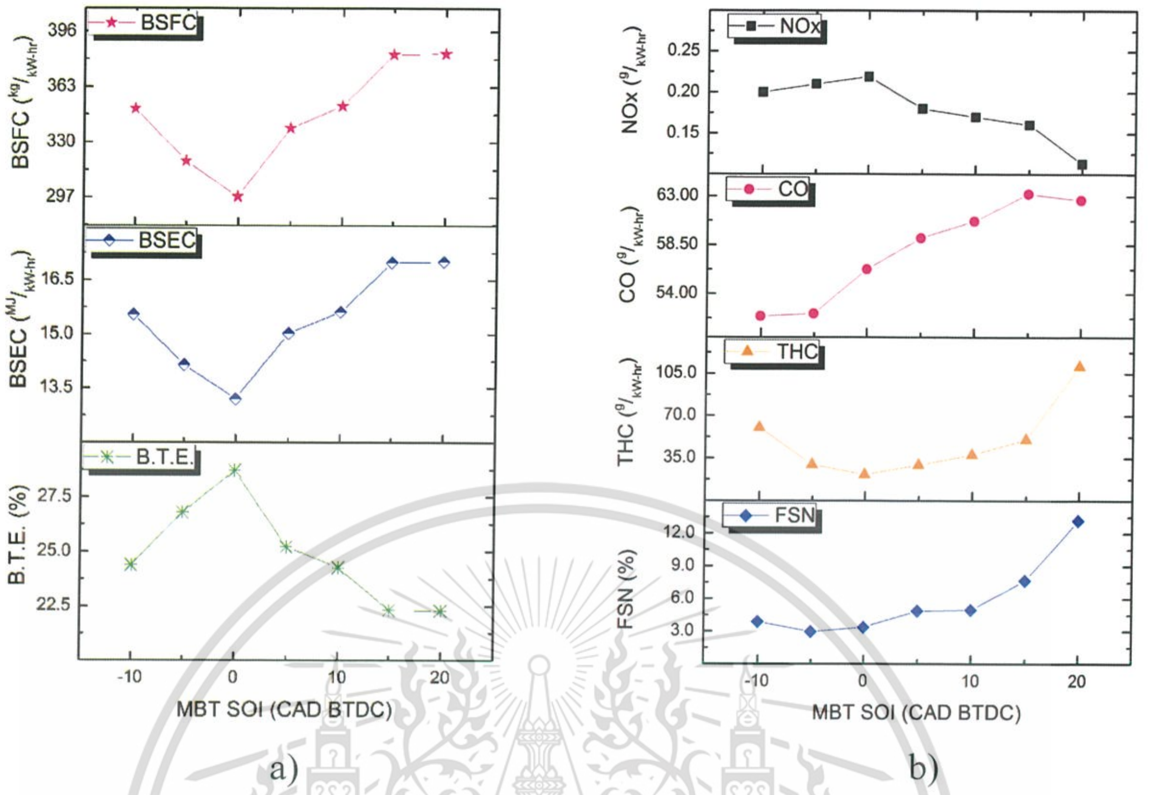


Figure 4.4 Effect of injection timing at MBT of E0 on efficiencies and emissions as a) efficiencies and b) emissions.

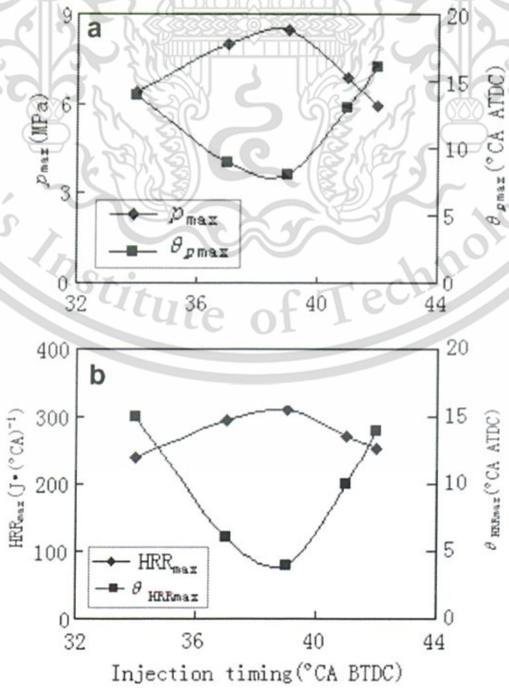


Figure 4.6 Effect of injection timing on the maximum in-cylinder pressure, the maximum heat release rate, and their corresponding crank angles from J. Li et al. analysis [38].

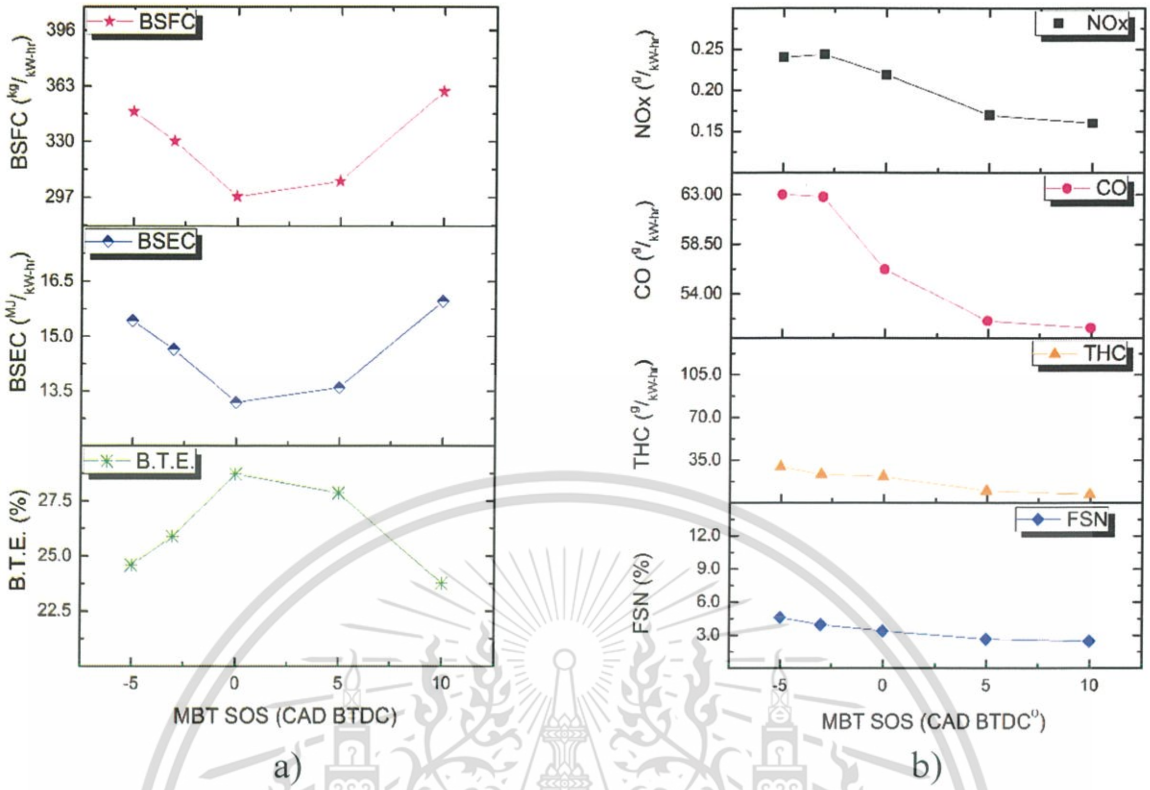


Figure 4.7 Effect of ignition timing at MBT of E0 on efficiencies and emissions as a) efficiencies and b) emissions.

Figure 4.7a, 4.11a and 4.15a show effect of ignition timing on efficiencies, the tendency of tested fuel is similar. The result shown that early or late timing from optimum point causes the distribution of mixture concentration in the chamber not ideal leads to propel down in efficiencies. Retarding ignition timing prolongs the duration of mixture formation that result too much homogeneous or more uniform than proper. It results in deteriorate of efficiency.

Regarding to effect of ignition timing on emission as shown in figure 4.7b, 4.11b and 4.15b. All emission are decreased as retard injection timing because of the duration between end of fuel injection and start of spark is increased that results in greater fuel formation. Moreover, too late of ignition timing from MBT decreases peak in cylinder pressure and temperature in power stroke. So, THC can oxidized in expansion stroke and exhaust stroke from post-combustion process together with NO_X and CO concentration from low combustion temperature as well.

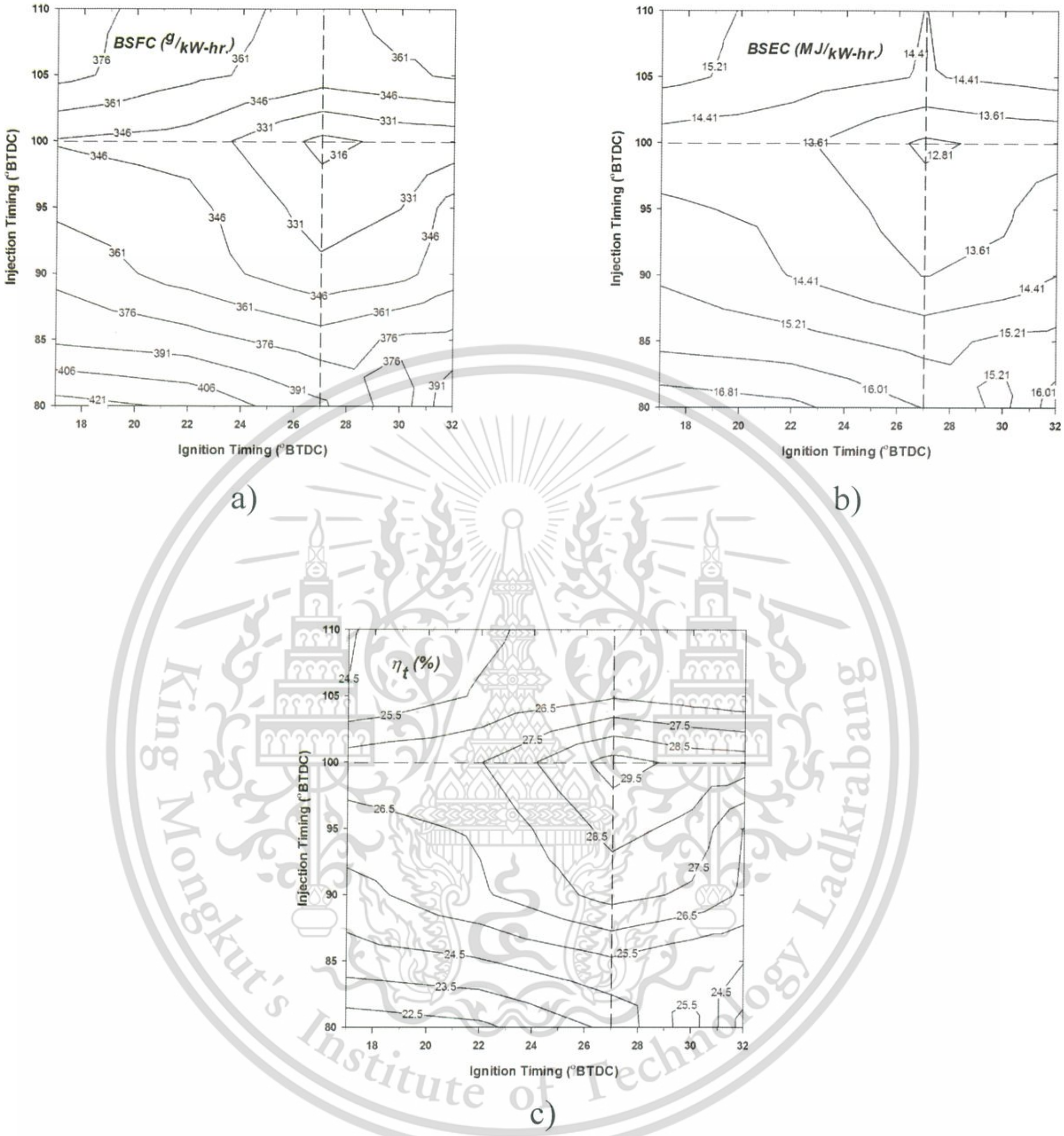


Figure 4.8 Effect of injection timing and ignition timing of E20 on efficiencies as a) BSFC, b) BSEC and c) Thermal efficiency.

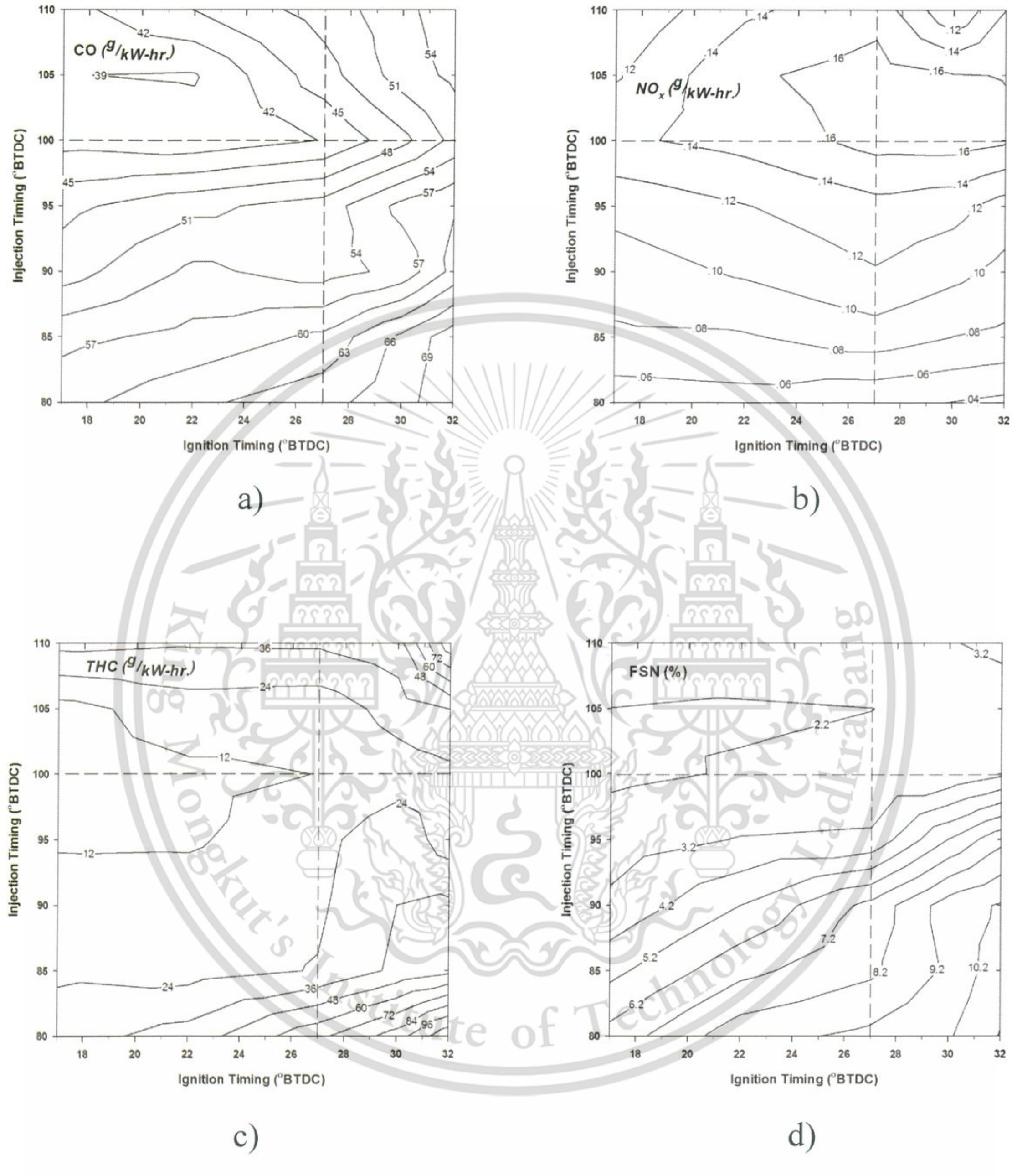
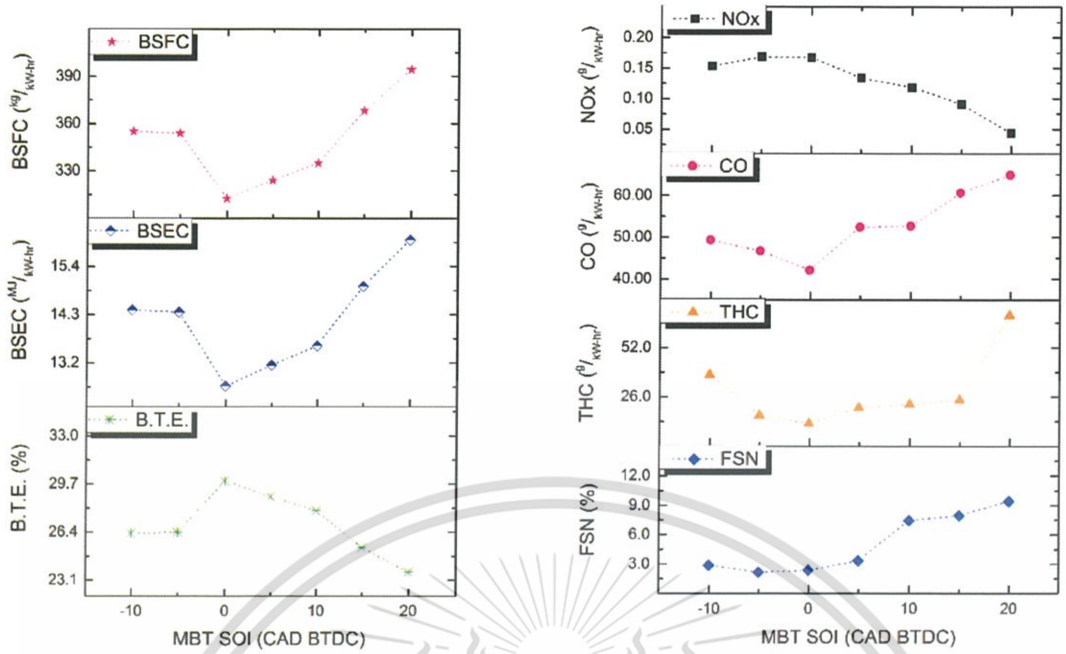
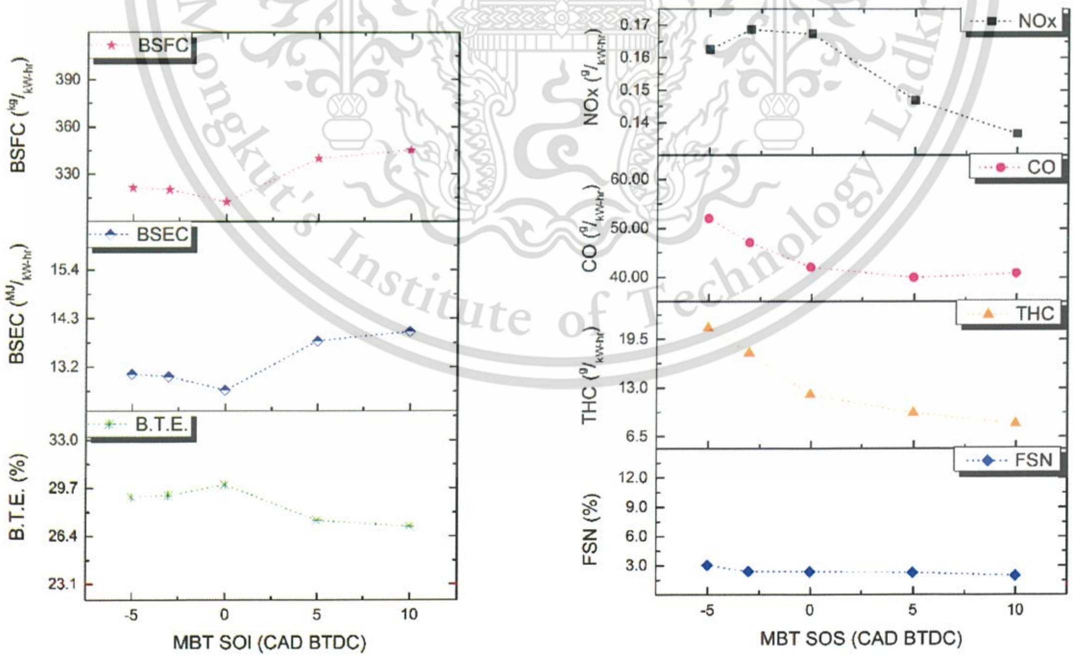


Figure 4.9 Effect of injection timing and ignition timing of E20 on emissions as a) CO, b) NO_x, c) THC and d) Filter smoke number (FSN).



a) b)

Figure 4.10 Effect of injection timing at MBT of E20 on efficiencies and emissions as a) efficiencies and b) emissions.



a) b)

Figure 4.11 Effect of ignition timing at MBT of E20 on efficiencies and emissions as a) efficiencies and b) emissions.

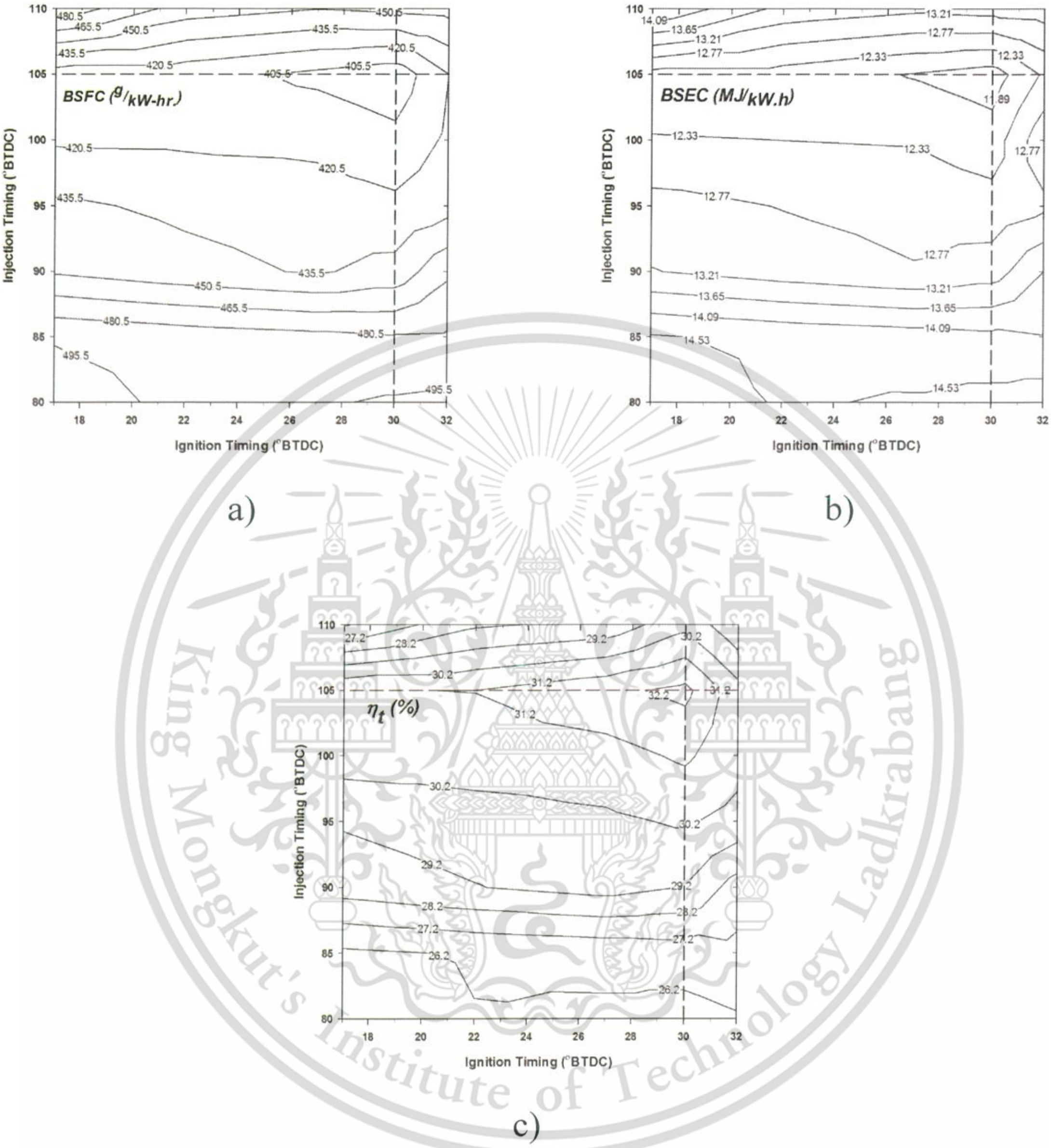


Figure 4.12 Effect of injection timing and ignition timing of E85 on efficiencies as a) BSFC, b) BSEC and c) Thermal efficiency.

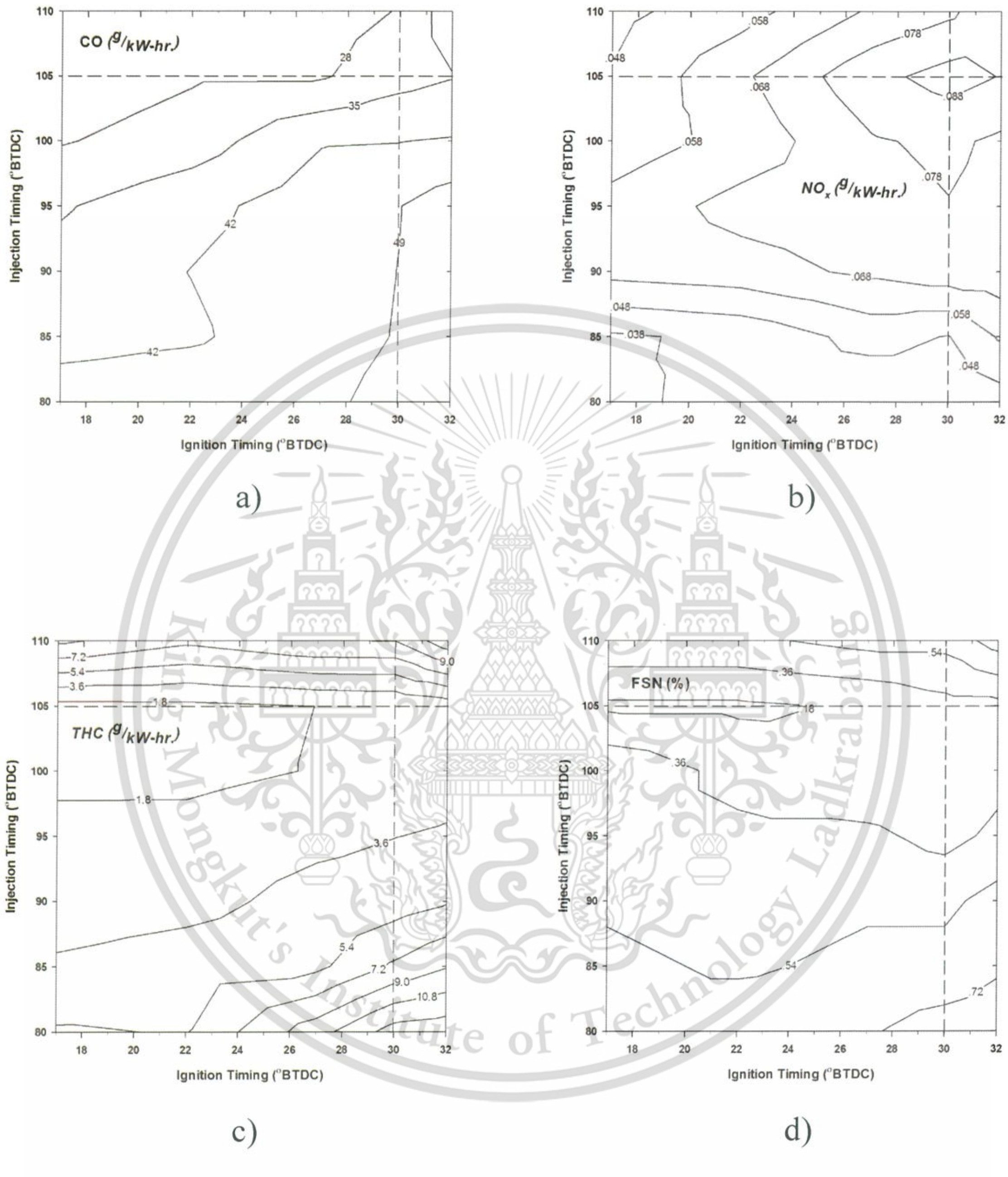
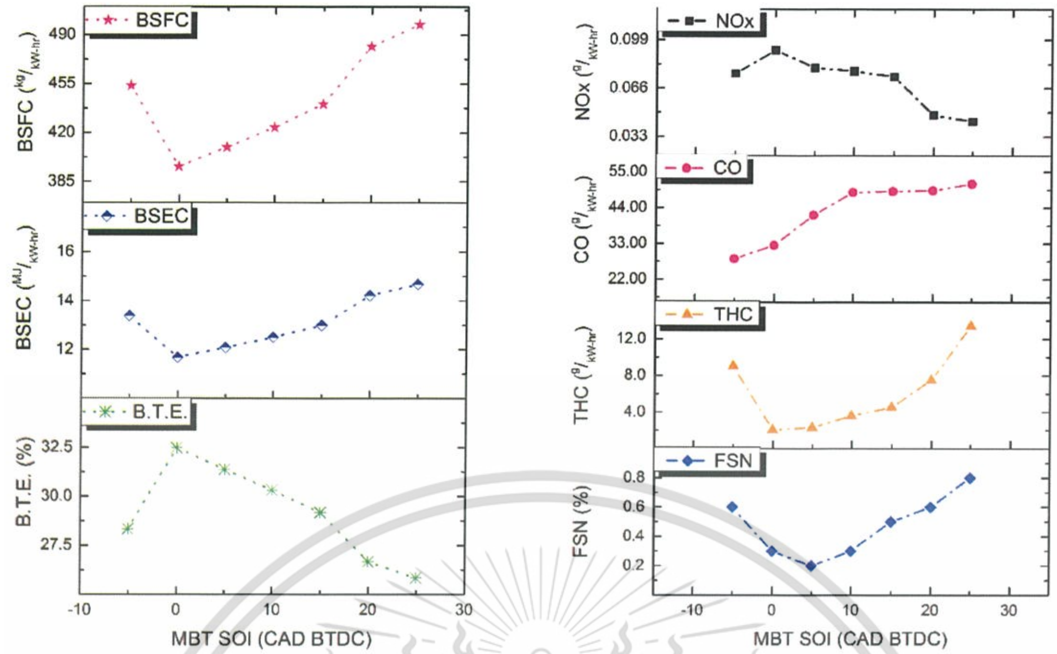
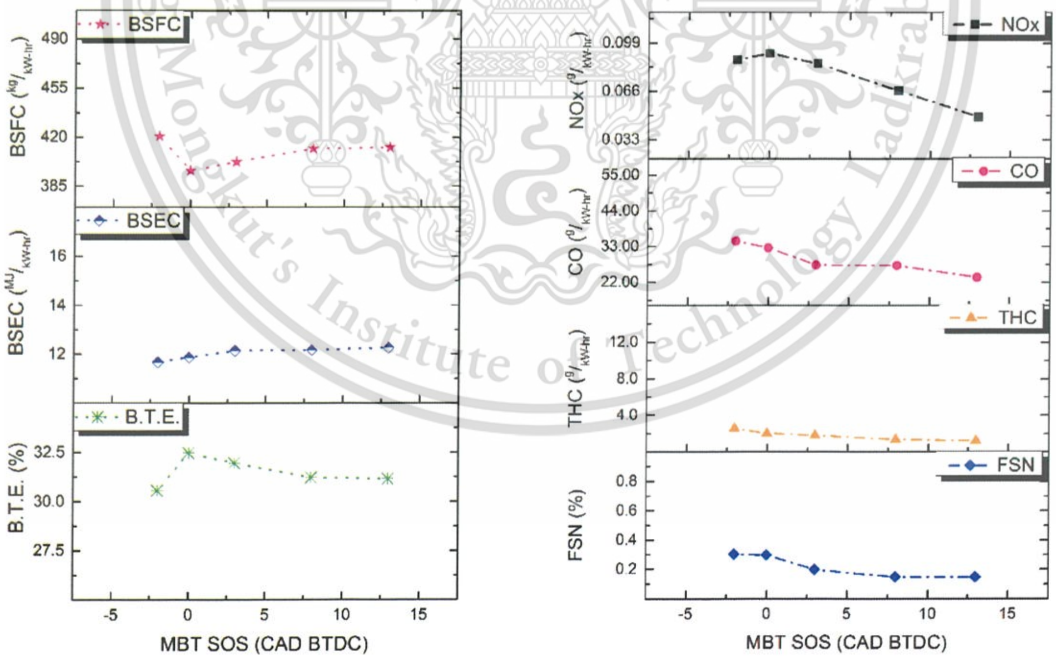


Figure 4.13 Effect of injection timing and ignition timing of E85 on emissions as a) CO, b) NO_x, c) THC and d) Filter smoke number (FSN).



a) b)

Figure 4.14 Effect of injection timing at MBT of E85 on efficiencies and emissions as a) efficiencies and b) emissions.



a) b)

Figure 4.15 Effect of ignition timing at MBT of E85 on efficiencies and emissions as a) efficiencies and b) emissions.

Table 4.1 Comparison of injection timing between MBT with retard and advance timing in term of efficiencies

		BSFC. ($^{g}/_{kW\text{-}hr.}$)		BSEC. ($^{MJ}/_{kW\text{-}hr.}$)		B.T.E. (%)	
		VALUE	COMPARE WITH MBT	VALUE	COMPARE WITH MBT	VALUE	COMPARE WITH MBT
E0	INJ. MBT.	297.28	0.00%	13.20	0.00%	28.71%	0.00%
	TOO RETARD	382.85	28.78%	17.00	28.78%	22.29%	-22.35%
	TOO ADVANCE	349.95	17.71%	15.54	17.71%	24.39%	-15.05%
E20	INJ. MBT.	312.08	0.00%	12.67	0.00%	29.91%	0.00%
	TOO RETARD	394.15	26.30%	16.00	26.30%	23.68%	-20.82%
	TOO ADVANCE	354.71	13.66%	14.40	13.66%	26.31%	-12.02%
E85	INJ. MBT.	395.74	0.00%	11.67	0.00%	32.46%	0.00%
	TOO RETARD	497.22	25.64%	14.67	25.64%	25.83%	-20.41%
	TOO ADVANCE	453.53	14.60%	13.38	14.60%	28.32%	-12.74%

Table 4.2 Comparison of injection timing between MBT with retard and advance timing in term of emissions

		NO_x ($^g/_{kW\text{-}hr.}$)		CO ($^g/_{kW\text{-}hr.}$)		THC ($^g/_{kW\text{-}hr.}$)		FSN (%)	
		VALUE	COMPARE WITH MBT	VALUE	COMPARE WITH MBT	VALUE	COMPARE WITH MBT	VALUE	COMPARE WITH MBT
E0	INJ. MBT.	0.22	0.00%	56.26	0.00%	21.68	0.00%	3.40	0.00%
	TOO RETARD	0.11	-48.38%	62.65	11.36%	110.80	411.06%	13.20	288.24%
	TOO ADVANCE	0.20	-8.63%	51.94	-7.68%	60.27	177.98%	3.91	15.00%
E20	INJ. MBT.	0.17	0.00%	42.10	0.00%	12.14	0.00%	2.38	0.00%
	TOO RETARD	0.04	-73.96%	64.93	54.22%	69.03	468.45%	9.45	296.63%
	TOO ADVANCE	0.15	-8.43%	49.34	17.18%	37.69	210.39%	2.93	22.81%
E85	INJ. MBT.	0.09	0.00%	32.53	0.00%	2.03	0.00%	0.30	0.00%
	TOO RETARD	0.04	-52.99%	51.40	57.98%	13.39	559.45%	0.80	166.67%
	TOO ADVANCE	0.08	-17.23%	28.33	-12.91%	9.01	344.15%	0.60	100.00%

Table 4.3 Comparison of ignition timing between MBT with retard and advance timing in term of efficiencies

		BSFC. ($^{g}/_{kW\text{-}hr.}$)		BSEC. ($^{MJ}/_{kW\text{-}hr.}$)		B.T.E. (%)	
		VALUE	COMPARE WITH MBT	VALUE	COMPARE WITH MBT	VALUE	COMPARE WITH MBT
E0	IGN. MBT.	297.28	0.00%	13.20	0.00%	28.71%	0.00%
	TOO RETARD	359.49	20.92%	15.96	20.92%	23.74%	-17.30%
	TOO ADVANCE	350.75	17.99%	15.57	17.99%	24.33%	-15.24%
E20	IGN. MBT.	312.08	0.00%	12.67	0.00%	29.91%	0.00%
	TOO RETARD	344.90	10.52%	14.00	10.52%	27.06%	-9.51%
	TOO ADVANCE	321.10	2.89%	13.04	2.89%	29.07%	-2.81%
E85	IGN. MBT.	395.74	0.00%	11.86	0.00%	32.46%	0.00%
	TOO RETARD	412.35	4.20%	12.27	3.41%	31.15%	-4.03%
	TOO ADVANCE	442.30	11.76%	12.40	4.57%	29.04%	-10.53%

Table 4.4 Comparison of ignition timing between MBT with retard and advance timing in term of emissions

		NO_x ($^g/_{kW\text{-}hr.}$)		CO ($^g/_{kW\text{-}hr.}$)		THC ($^g/_{kW\text{-}hr.}$)		FSN (%)	
		VALUE	COMPARE WITH MBT	VALUE	COMPARE WITH MBT	VALUE	COMPARE WITH MBT	VALUE	COMPARE WITH MBT
E0	IGN. MBT.	0.22	0.00%	56.26	0.00%	21.68	0.00%	3.40	0.00%
	TOO RETARD	0.16	-26.91%	50.95	-9.43%	7.22	-66.69%	2.44	-28.12%
	TOO ADVANCE	0.25	14.21%	64.02	13.79%	53.68	147.62%	4.80	41.18%
E20	IGN. MBT.	0.17	0.00%	42.10	0.00%	12.14	0.00%	2.38	0.00%
	TOO RETARD	0.14	-18.35%	40.93	-2.78%	8.29	-31.76%	2.00	-16.27%
	TOO ADVANCE	0.16	-2.78%	52.00	23.51%	21.05	73.31%	3.04	27.59%
E85	IGN. MBT.	0.09	0.00%	32.53	0.00%	2.03	0.00%	0.30	0.00%
	TOO RETARD	0.05	-46.99%	23.60	-27.45%	1.24	-38.97%	0.15	-50.00%
	TOO ADVANCE	0.08	-8.27%	44.16	35.72%	3.07	51.45%	0.40	33.33%

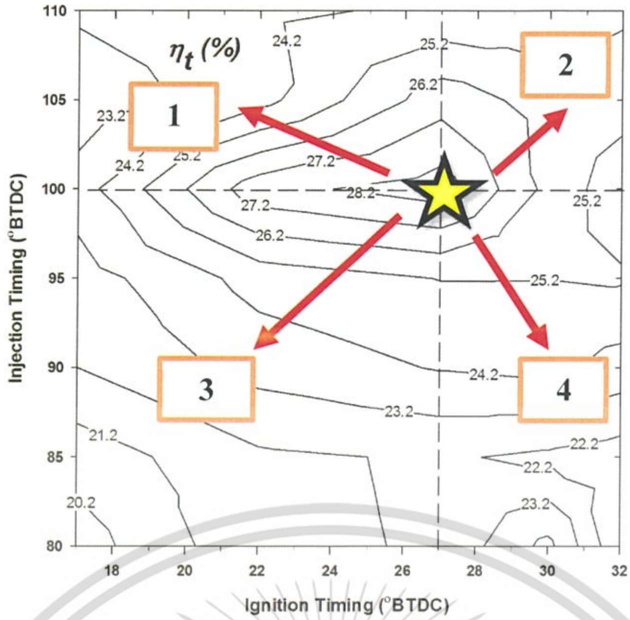


Figure 4.16 The conclusion of effect from injection timing and ignition timing shift from MBT on stratified charge operation.

Last in, figure 4.16 portrays the conclusion from injection and ignition timing impact to the formation of fuel cloud. This information can imply as indicated in table 4.5

Table 4.5 the conclusion of injection timing and ignition timing strategy on stratified operation.

Condition	Injection Timing	Iginition Timing	Description
1	A	R	Mixture stratification is reduced by increasing duration of cloud information. So, partial burns is increased.
2	A	A	Too early of combustion phase lengthened the delay angle. Hence, an over lean mixture occurred rapidly in combustion phase.
3	R	R	In-cylinder peak pressure is decreased. This causes retarding cloud formation and later combustion phasing.
4	R	A	Insufficient duration between end of injection and start of spark causes over-rich mixture and liquid droplet effect.

NOTE: A = Advance from MBT and B = Retard from MBT

4.2 Effect of Ethanol Blends

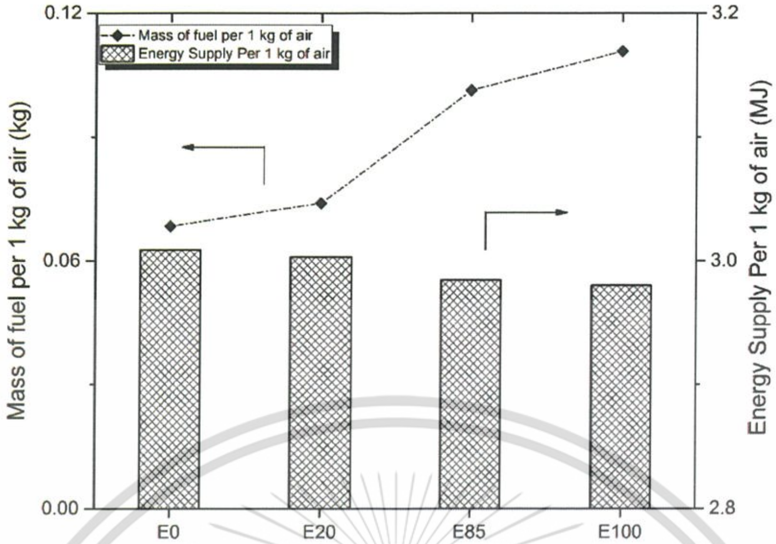


Figure 4.17 Mass of fuel and energy supply per 1 kg_{air} of each fuel.

In this study, the results of tested fuels were comparing at same lambda. So, the amount of fuel injected into combustion chamber is essential to calculate an energy input. Figure 4.17 illustrated energy supply per 1 kilogram of air at the same lambda of tested fuel.

From table 3.2, the ethanol blends (E20, E85) has lower calorific value than gasoline (E0). This causes difference in air fuel ratio. Hence, the ethanol blends should injected more fuel to compensate for same lambda of gasoline. When compare with gasoline, the quality of E20 and E85 should be added as 8.85% and 48.99% respectively.

To determine total energy supply in term of energy supply per 1 kilogram of air The quantity of fuel for lambda = 1 is the first step for determination that are 68.35g of E0, 79.02g of E20 and 101.31g of E85. Then, the total energy input of each fuel can calculated by multiplication of low calorific value and mass of fuel from step that are 3.009 MJ of E0, 3.003 of E20 and 2.98 of E85 consecutively.

As shown in figure 4.17 including with calculation from last paragraph, the energy input of tested fuel are not different but energy input become lower as increase percentage of ethanol conform with the pressure history data from Piyaboot et al. that the peak pressure decreases as ethanol content decrease and duration to reach peak pressure us advanced by increase concentration of ethanol. These can mention as oxygen concentration in ethanol impact to combustion characteristics in term if oxidizing. The high percentage of ethanol, the high ability to oxide with oxygen will be.

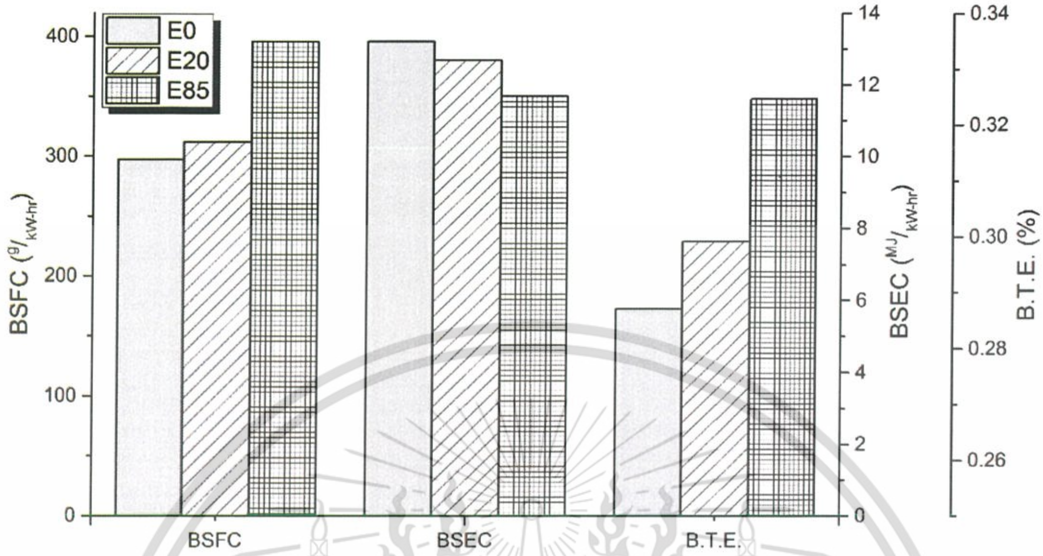


Figure 4.18 Comparison of efficiencies between gasoline and ethanol blends (E20 and E85).

From figure 4.18 shown the comparison of Brake Specific Fuel Consumption (BSFC), Brake Specific Energy Consumption (BSEC) and Brake Thermal Efficiency (B.T.E.) on gasoline and ethanol blends (E20 and E85) can discuss below as the following;

Regarding to optimum injection timing and ignition timing, gasoline obtains lowest BSFC and highest at E85 due to the fact that ethanol has lower heating value than that gasoline (approximately 30%). Hence, the engine fueled with ethanol should inject more quantity than gasoline for the same output which causes high BSFC. At E20 and E85, BSFC of both fuel increased by 4.98% and 33.12% respectively.

Due to low heating value of ethanol is lower than gasoline. To obtain same engine output, ethanol blends should require more quantity which described in above passage. Hence, to find out specific energy consumption of each fuel always selected "Brake Specific Energy Consumption (BSEC)" that described as the quantity of energy consumed per unit power developed in a unit of time. Described in briefly, BSEC is how efficiently of energy obtained from its fuel. From figure shown as BSEC of gasoline is highest and slightly decrease when blends higher percentage of ethanol (4.01% of E20 and 11.55% of E85). These can imply as fuel blend with ethanol uses lower heating energy to obtain same engine output lower than gasoline even though the quantity of ethanol blends is higher, but the total energy input (the multiplication between mass of fuel and low heating value) is lower because of ethanol has lower calorific value as described prior. Hence, BSEC is lower when blended more of ethanol.

Last in Brake Thermal Efficiency (B.T.E.), to blend higher percentage of ethanol causes higher B.T.E. as 4.18% of E20 and 13.06% of E85. These can discusses as when gasoline and blended fuel obtain similar heating output, the blended fuel obtains higher engine output. B.T.E. is inverse variation of BSFC and relate of complete of combustion. Hence, it can imply that blended fuel causes more complete combustion than gasoline due to oxygen content inside ethanol can enhance oxidation rate by accelerate flame development process and increase the rate of heat release which helps to reduce heat loss throughout the surroundings. Furthermore, the ethanol blended has higher octane number and ignition delay which can advance more degree of ignition timing. These results in higher efficiencies.

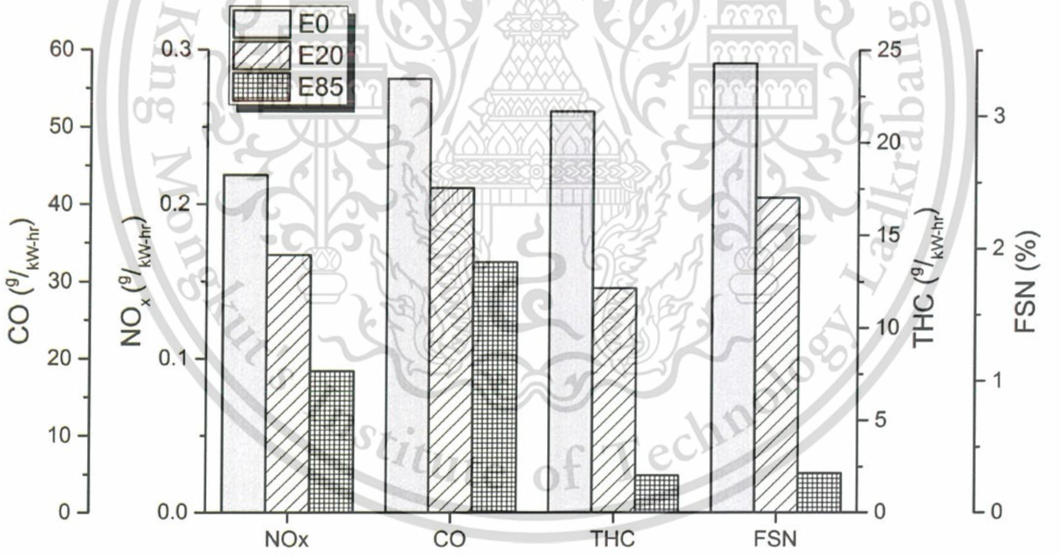


Figure 4.19 Comparison of emissions between gasoline and ethanol blends.

Figure 4.19 shown Comparison of emissions; oxide of nitrogen (NO_x), Carbon monoxide (CO), total hydrocarbon (THC) and filter smoke number (FSN) between gasoline and ethanol blends optimum injection timing and ignition timing. Focusing on carbon monoxide (CO) is occurred from inadequate of oxygen to combust fuel in combustion chamber. Mostly, the amount of CO relies on air-fuel ratio of mixture. From figure 4.19, the specific emission of CO at 2000 rpm at 3.0 bar nominal BMEP portrayed as gasoline emitted highest and decreased when blending more percentage of ethanol. The decreases of E20 and E85 are 25.16% and 42.17% respectively. The reduction of CO when using ethanol can be explained by the oxygen content inside fuel which obtains plenty of oxygen in rich mixture that benefits to more complete of combustion.

In case of total hydrocarbon, the specific emission of THC at experimental condition found that the percentage of ethanol effect to the concentration of THC. The THC is decreased by blending more of ethanol, the reduction in THC of E20 and E85 when compare with gasoline are 43.99% and 90.64% respectively which can imply as to blend more of ethanol. It consists of higher oxygen, lower carbon atom together with hydrogen than that gasoline. Hence, it can enhance combustion process to more complete that results in reduction of THC.

Filter smoke number (FSN), the percentage of filter smoke number measured from light opacity. The 0% shown a little portion of particle emissions whereas 100% means full of particle emissions consecutively. The results expressed as gasoline has highest percentage of FSN as 3.40% and decreases dramatically when blending higher percentage of ethanol as 2.38% of E20 and 0.30% of E85. This result can simplify by particulate emissions formation of gasoline was higher that contributed to more of remain of particulate emissions during combustion process than ethanol. Furthermore, oxygen content inside ethanol can enhance combustion process by oxidize with available of oxygen in the flame zone.

Last in oxide of nitrogen (NO_x), blending higher percentage of ethanol, the decreases of E20 and E85 are 23.60% and 58.02% respectively. This result can imply as the higher of heat of vaporization in ethanol reduces the combustion temperature which conforms to the reduction of exhaust temperature. Hence, NO_x was reduced. [40, 41]

From experimental results, there are many discussions to analyze the benefits when using ethanol as fuel. Starting with discussion in term of flame speed from combustion duration, from figure 2.22 and 2.25, combustion duration is decreased when percentage of ethanol higher. As piyaboot et. Al., analysis combustion duration is determined from mass fraction burned by 0 – 10 % from MFB refers to flame development period and 10 – 90% is combustion duration period. From his analysis can imply as ethanol content may speed up the initial of combustion that leads to reduce duration of heat loss in early stage of combustion period.

Moreover, as known that flame speed is associated to air-fuel ratio and lambda at stoichiometric (λ =1) causes maximum flame speed. From analysis result in figure 2.22 and 2.25 can imply in further detail of using ethanol blends that it can make the stoichiometric mixture in wider area when equate to gasoline due to ethanol blends (E20 and E85) required more quantity if fuel to same output and completely evaporate easier than gasoline. Hence, E20 and E85 have ability to join stoichiometric mixture in wider area. Furthermore, oxygen content in ethanol blends is strongly impact to flame speed in term of increasing oxidation rate as well.

Following in discussion by chemical reaction session, particularly in combustion process has many radical reactions before reactant transform into products. Focused on each step of chemical reaction it has specific activation energy (E_a) but the minimum of each specific of activation energy also known as "Rate determining step – $E_{a\text{-}min}$".

Due to the OH – radical is available in ethanol blends causes rate step lower than gasoline. From this reason, the OH-radical is readily to react with another species. Hence, OH-species in ethanol blends can enhance reaction rate faster than gasoline. Because, gasoline doesn't contain OH molecule and it consists of large amount of aromatic rings when equate to ethanol blends. These aromatic rings are hard to react with another substance likewise oxygen. From reason, causes high $E_{a\text{-}min}$ of gasoline and leads to slower reaction rate than ethanol blends.

Furthermore, from C.K. Westbrook and F. L. Dyer [42]can explain the reaction rate of hydrocarbon in single step by using general equation of single step reactor that fulfill to above discussion session as shown in equation 4.1, figure 4.20 and 4.21.

$$C_xH_y + (x+\frac{y}{4})O_2 \longrightarrow xCO_2 + (\frac{y}{2})H_2O$$
$$\frac{d\left[C_xH_y\right]}{dt} = -Ae^{(-\frac{E_a}{R_uT})}\left[C_xH_y\right]^m\left[O_2\right]^n \tag{4.1}$$

Where

A	Total number of collision per second
$e^{(-\frac{E_a}{R_uT})}$	Possibility that any given collision will result in reaction
m , n	Reaction order of C_xH_y and O_2

Fuel	Pre-exponential Factor, A^a	Activation Temperature, E_a/R_u (K)	m	n
CH_4	$1.3 \cdot 10^8$	24,358[b]	−0.3	1.3
CH_4	$8.3 \cdot 10^5$	15,098[c]	−0.3	1.3
C_2H_6	$1.1 \cdot 10^{12}$	15,098	0.1	1.65
C_3H_8	$8.6 \cdot 10^{11}$	15,098	0.1	1.65
C_4H_{10}	$7.4 \cdot 10^{11}$	15,098	0.15	1.6
C_5H_{12}	$6.4 \cdot 10^{11}$	15,098	0.25	1.5
C_6H_{14}	$5.7 \cdot 10^{11}$	15,098	0.25	1.5
C_7H_{16}	$5.1 \cdot 10^{11}$	15,098	0.25	1.5
C_8H_{18}	$4.6 \cdot 10^{11}$	15,098	0.25	1.5
C_8H_{18}	$7.2 \cdot 10^{12}$	20,131[d]	0.25	1.5
C_9H_{20}	$4.2 \cdot 10^{11}$	15,098	0.25	1.5
$C_{10}H_{22}$	$3.8 \cdot 10^{11}$	15,098	0.25	1.5
CH_3OH	$3.2 \cdot 10^{12}$	15,098	0.25	1.5
C_2H_5OH	$1.5 \cdot 10^{12}$	15,098	0.15	1.6
C_6H_6	$2.0 \cdot 10^{11}$	15,098	−0.1	1.85
C_7H_8	$1.6 \cdot 10^{11}$	15,098	−0.1	1.85
C_2H_4	$2.0 \cdot 10^{12}$	15,098	0.1	1.65
C_3H_6	$4.2 \cdot 10^{11}$	15,098	−0.1	1.85
C_2H_2	$6.5 \cdot 10^{12}$	15,098	0.5	1.25

[a] Units of A are consistent with concentrations in Eqn. 5.2 expressed in units of gmol/cm³, i.e., A[−] (gmol/cm³)$^{1-m-n}$/s.

[b] E_a = 48.4 kcal/gmol.

[c] E_a = 30 kcal/gmol.

[d] E_a = 40 kcal/gmol.

Figure 4.20 Parameters use to calculate the reaction rate of gasoline (isooctane) and ethanol.

$$C_8H_{18} + (12.5)O_2 \longrightarrow 8CO_2 + 9H_2O$$

$$M_{fuel} = (1)[(8\times12)+(1\times18)] = 114$$

$$M_{O_2} = 12.5[2\times16] = 400$$

$$M_{total} = 514$$

$$[C_8H_{18}] = \frac{114}{514} = 0.22 \qquad [O_2] = \frac{400}{514} = 0.78$$

$$A = 4.6\times10^{11}\ gmol/cm^3-s \qquad T = 450\ K$$

$$\frac{E_a}{R_u} = 15,098\ K \qquad m = 0.25 \qquad n = 1.5$$

$$-\frac{d[C_8H_{18}]}{dt} = (4.6\times10^{11})\ e^{(-\frac{15,098}{450})}[0.22]^{0.25}[0.78]^{1.5}$$

$$-\frac{d[C_8H_{18}]}{dt} = 0.000583$$

$$C_2H_5OH + (2)O_2 \longrightarrow 2CO_2 + 3H_2O$$

$$M_{fuel} = (1)[(2\times12)+(6\times1)+(1\times16)] = 46$$

$$M_{O_2} = 2[2\times16] = 64$$

$$M_{total} = 110$$

$$[C_2H_5OH] = \frac{46}{110} = 0.42 \qquad [O_2] = \frac{64}{110} = 0.58$$

$$A = 1.5\times10^{12}\ gmol/cm^3-s \qquad T = 450\ K$$

$$\frac{E_a}{R_u} = 15,098\ K \qquad m = 0.15 \qquad n = 1.6$$

$$-\frac{d[C_2H_5OH]}{dt} = (1.5\times10^{12})\ e^{(-\frac{15,098}{450})}[0.42]^{0.15}[0.58]^{1.6}$$

$$-\frac{d[C_2H_5OH]}{dt} = 0.00148$$

Figure 4.21 The determination of reaction rate of gasoline (isooctane) and ethanol.

According to the determination of reduction rate of hydrocarbon in gasoline ($\frac{d\,[C_8H_{18}]}{dt}$) is lower than ethanol ($\frac{d\,[C_2H_5OH]}{dt}$) which are 0.000583 sec^{-1} and 0.00148 sec^{-1} respectively. This difference of reaction rate is impacted by 2 dominant factors as physical properties ad amount of collision of fuel due to the total number of collision – A of ethanol is higher than that gasoline whereas $e^{(-\frac{E_a}{R_uT})}$ is quite similar. Furthermore, the reaction order – M of ethanol is lower than gasoline but the concentration of ethanol is more than gasoline. So, when compare the multiplication of mass concentration and reaction order, $[C_xH_y]^m[O_2]^n$, of both fuel is not different essentially.

From discussion of chemical reaction, to imply with flame speed in order to support above is essential. The relation between in-cylinder pressure and combustion speed were conducted by flame area because of this refers to amount of fuel burned. From results of [28, 29] the in-cylinder pressure and combustion speed faster when blended higher percentage of ethanol due to flame area of ethanol blended spread faster than gasoline. Hence, this can explain the reason of ethanol have reaction rate faster than gasoline as the molecule of ethanol has smaller size than gasoline. Then, regarding to combustion process in engine that operate in finite volume, the larger size of molecule have less probability to react with surrounding itself when compare with smaller molecule that leads to slower reaction rate.

Next, the discussion spray image of both fuel from shadow graph image comparing between gasoline and ethanol on figure 2.28 found that at the initial of evaporation, gasoline can vaporize faster than ethanol as shown in spray characteristics of this fuel gasoline has wider cone angle with short penetration. As time passes, ethanol is completely evaporated to vapor phase faster than gasoline. This can imply that gasoline composes of light and heavy component more than ethanol. So, the lighter component owing to evaporate in beginning period where as ethanol does not start. With the passing of time, the evaporation rate of gasoline is slower and still remains whereas ethanol which consists of single component is completely transformed to vapor phase. This information can discuss with distillation curve as shown in figure 3.1.

Last in discussion in case of ignition delay, this is a dominant factor to combustion process. Not only indicate the duration formation of mixture, but also describe the waste time in the initial of combustion. Especially in spark-ignition engine, the less of ignition delay, the combustion pressure close to ideal will be. So, the reduction of heat loss can reduce by decreasing of ignition delay.

Regarding to general definition of ignition delay, gasoline has lower ignition delay than ethanol blends and pure ethanol even though the Reid – Vapor – Pressure (RVP) of gasoline is highest. So, gasoline is fully evaporated early than ethanol. In the end of this process, ethanol is completely evaporated and shifts to vapor phase faster than gasoline. This phenomenon can express with azeotropic behavior. So, it can improve performance of mixture vaporization process by using ethanol blends in gasoline.

The evaporation of gasoline is slower than ethanol blends. Then, when fuel is injected into the chamber, the decay of fuel is still in liquid phase and not evaporates. These over – rich mixture is hard to ignite with flame kernel. Hence, at the beginning stage of combustion spends long duration when compare with ethanol whereas ethanol fuel is completely evaporate. So, the mixture of ethanol blends has more capability to join in stoichiometric in wider are than gasoline. Consequently using ethanol blends can reduce duration of combustion leads to complete combustion which reduces emissions and improve efficiencies when compare with gasoline.

4.3 Particulate Emissions

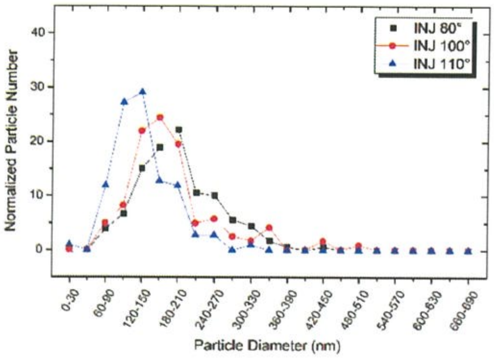
4.3.1 Effect from Injection Timing

The injection timing of DISI engine effects to the mixture of air-fuel before spark discharged. The interval between start of injection and start of spark is dominant to homogeneity of mixture in combustion process and emission formation. The particulate matter formation comes from the local equivalence change in ratio and temperature. Therefore, the results can indicate information below;

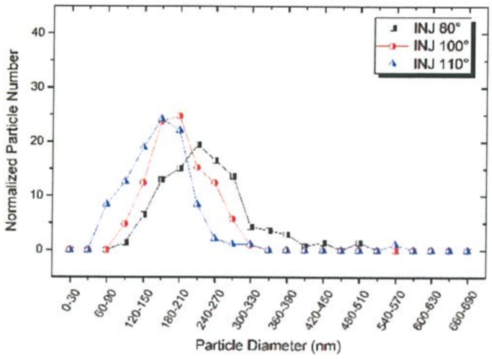
As shown in figure 4.23 - 4.24 and table 4.2 - 4.4 , the particulate size distribution is larger when retard the injection timing as 13.26% of E0, 20.76% of E20 and 11.75% of E85 whereas advancing injection timing causes smaller particulate size distribution as 19.25% of E0, 15.77% of E20 and 30.86% of E85 compare to MBT Timing of each fuel. There are 2 major factors cause larger in size distribution as the following;

First, Duration of mixture preparation consists of time for evaporate and mix to form homogeneous mixture is shortened when injection timing is too late. Hence, it occurs local-rich mixture (inhomogeneity) in combustion process.

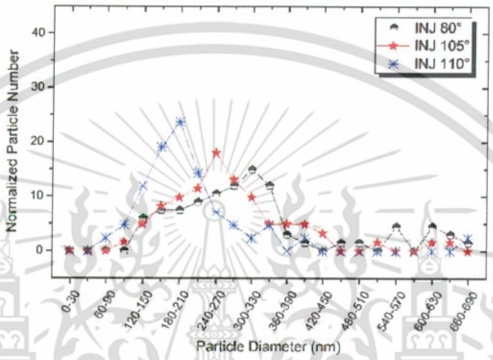
Last in, Fuel impingement which located on wall wetting and piston bowl are not completely vaporized and well mixed with intake air at the start of combustion. So it creates the fuel film and this film can burn diffusively and as a pool fire near the piston. Both factors results in generating large size of particulate matter [43].



a)



b)



c)

Figure 4.22 Effect of injection timing (°bTDC) of a) E0, b) E20 and c) E85 at 2000 rpm, 3.0 bar nominal BMEP.

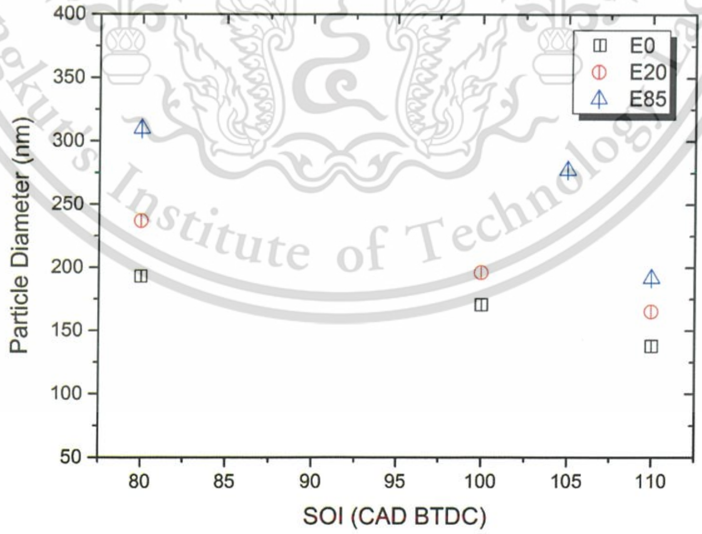


Figure 4.23 Average particle size distribution from effect of injection timing (°bTDC) of tested fuel at 2000 rpm, 3.0 bar nominal BMEP.

Table 4.6 Particulate matter from scanning electron microscope technique of E0 at each injection timing at 2000 rpm, 3.0 bar nominal BMEP.

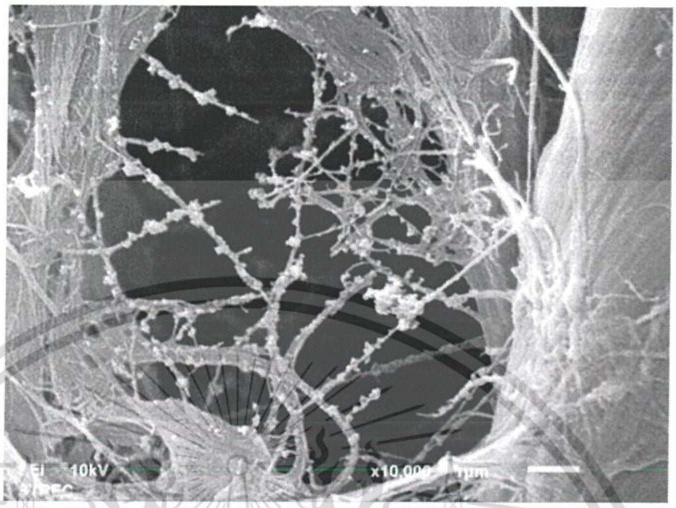
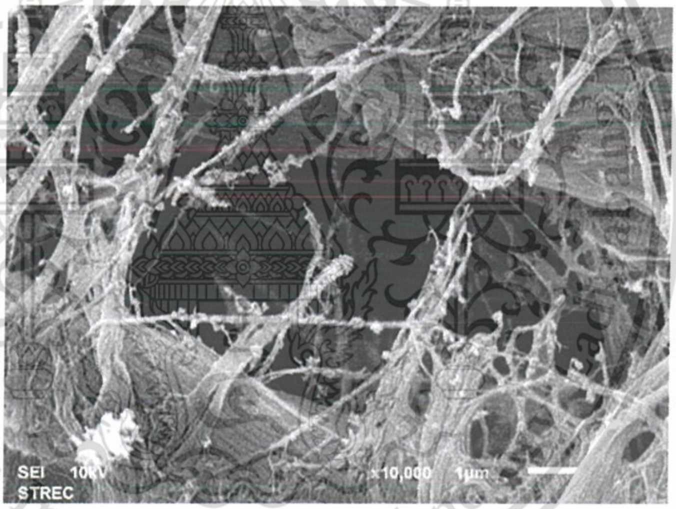
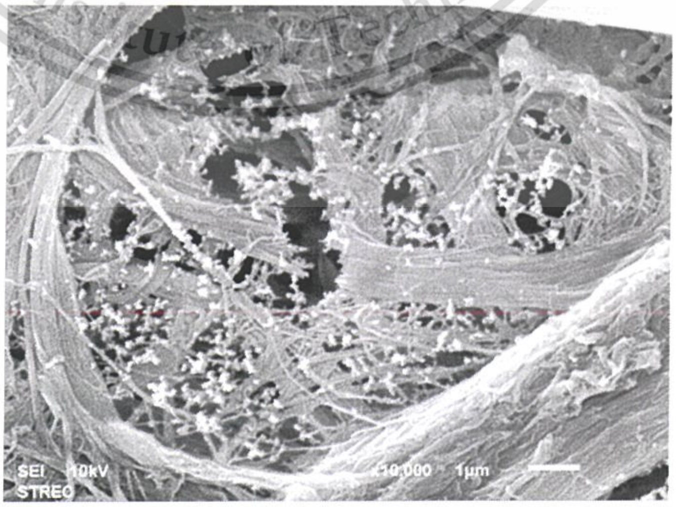
TYPE / FUEL.	Particulate Matter (PMs)
Fuel: E0 INJ T. 80°	
Fuel: E0 INJ T. 100°	
Fuel: E0 INJ T. 110°	

Table 4.7 Particulate matter from scanning electron microscope technique of E20 at each injection timing at 2000 rpm, 3.0 bar nominal BMEP.

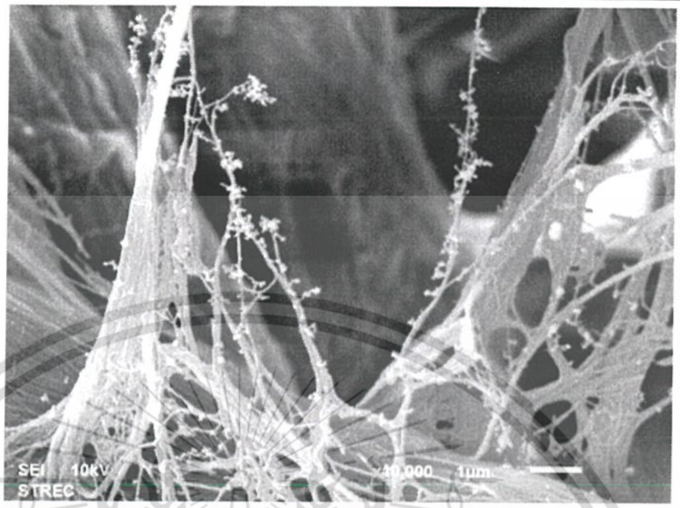
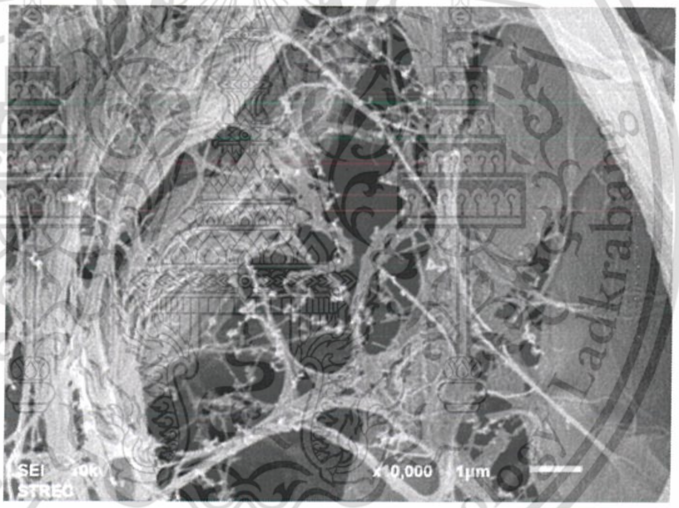
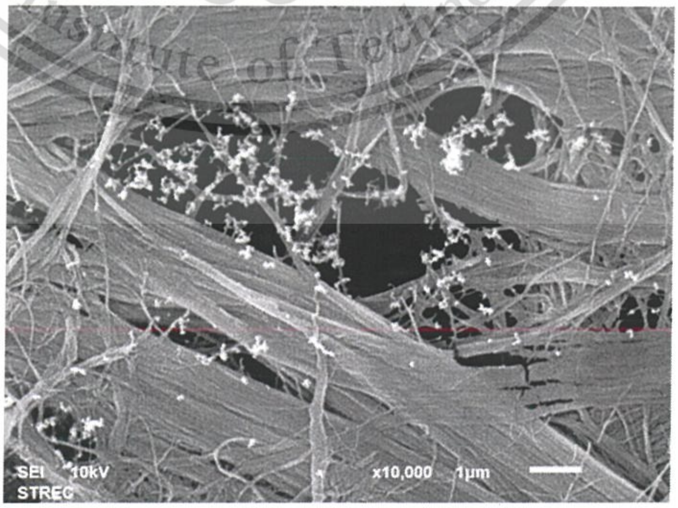
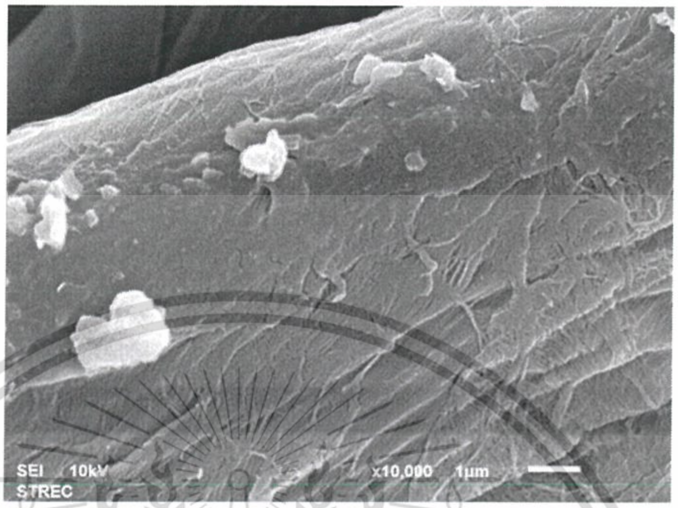
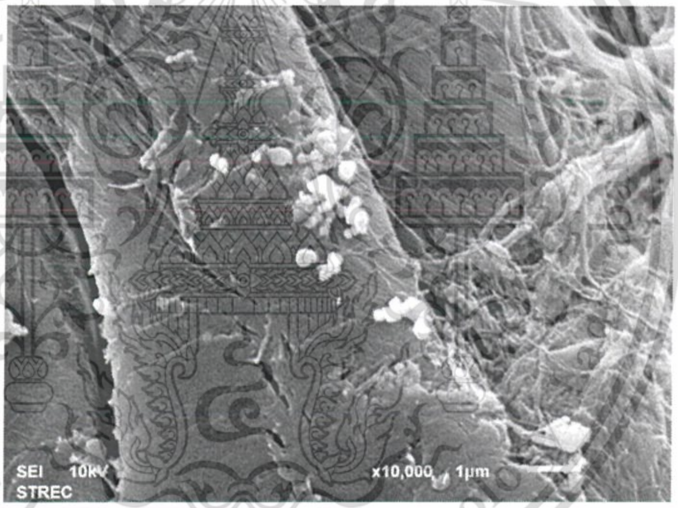
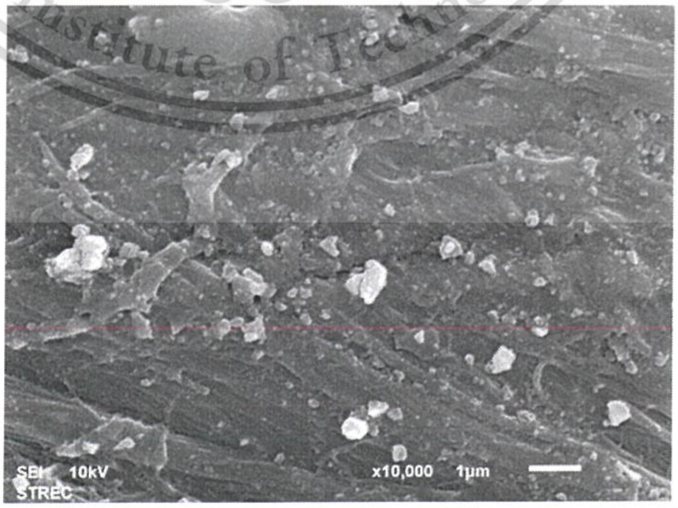
TYPE / FUEL.	Particulate Matter (PMs)
Fuel: E20 INJ T. 80°	
Fuel: E20 INJ T. 100°	
Fuel: E20 INJ T. 110°	

Table 4.8 Particulate matter from scanning electron microscope technique of E85 at each injection timing at 2000 rpm, 3.0 bar nominal BMEP.

TYPE / FUEL.	Particulate Matter (PMs)
Fuel: E85 **INJ T. 80°**	
Fuel: E85 **INJ T. 105°**	
Fuel: E85 **INJ T. 110°**	

4.3.2 Effect from Ignition Timing

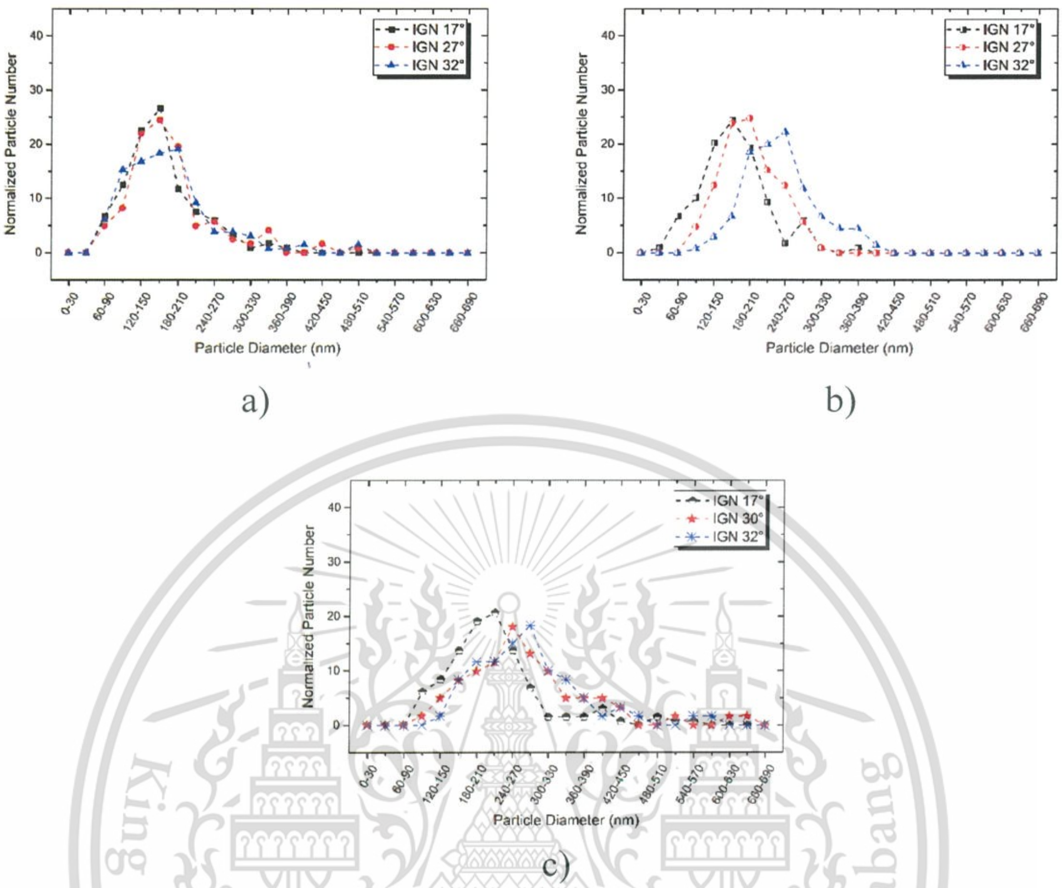


Figure 4.24 Effect of ignition timing (°bTDC) of a) E0, b) E20 and c) E85 at 2000 rpm, 3.0 bar nominal BMEP.

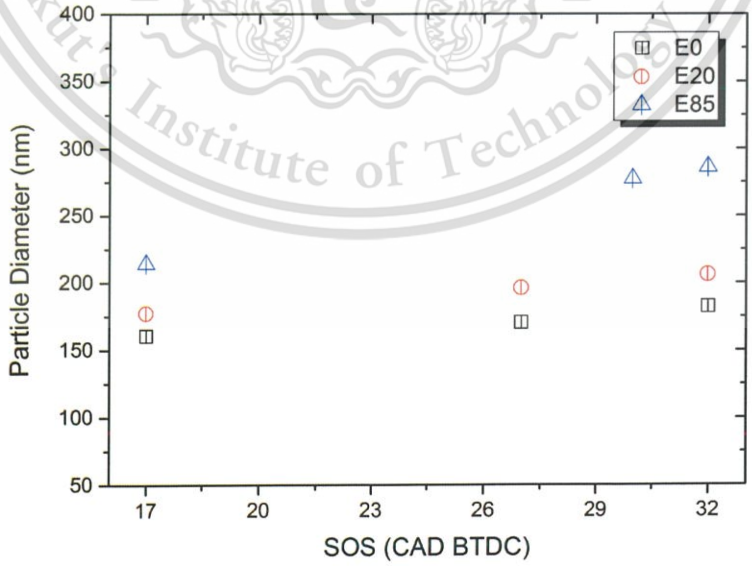


Figure 4.25 Average particle size distribution from effect of ignition timing (°bTDC) of tested fuel at 2000 rpm, 3.0 bar nominal BMEP.

From figure 4.25 - 4.26 and table 4.5 - 4.7, Particulate size distribution become smaller when retard the spark timing but the value did not change much as the effect of injection timing. 5.85% of E0, 9.75% of E20 and 22.81% of E85 are the percentage of size reduction from retard spark timing when compare with MBT timing due to

1. The interval between start of spark and end of fuel injection is increased. Then, it extends duration between fuel and intake charge which obtain uniform of air mixture or more homogeneity that leads to reduce size of particulate matter.

2. Retarding spark timing effects to the peak of heat release rate and the peak temperature in cylinder become lower. So, the corresponding crank angle move to the right hand side and the average temperature during expansion stroke become higher.

From discussion above, the result is conform to Y. Q. Pei et al. analysis [23] as shown in figure 4.27. This mentioned as the lower peak cylinder pressure and the combustion temperature from retarding spark timing results in post-combustion in expansion stroke and exhaust pipe that inhibits the generation of particle during combustion process including with enhance oxidation rate of particulate matter to smaller size as well. On the other hand, advancing spark timing causes larger of particulate matter size because of higher of peak pressure in cylinder impacts to fuel entrains to the top land of crevice region. Thus, large dimension of particulate matter. Additional with low exhaust temperature at expansion stroke and exhaust stroke lead to slow down particulate matter oxidation [44]. So, both factors result in larger size distribution of particulate matter.

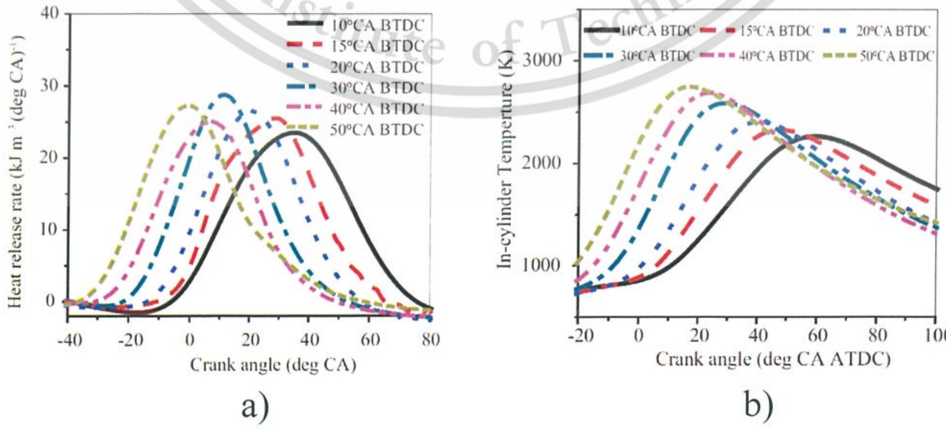


Figure 4.26 a) Heat release rate and b) In-cylinder temperatures with various spark timings of 3.0 bar BMEP from Y. Q. Pei et al. [23].

Table 4.9 Particulate matter from scanning electron microscope technique of E0 at each ignition timing at 2000 rpm, 3.0 bar nominal BMEP.

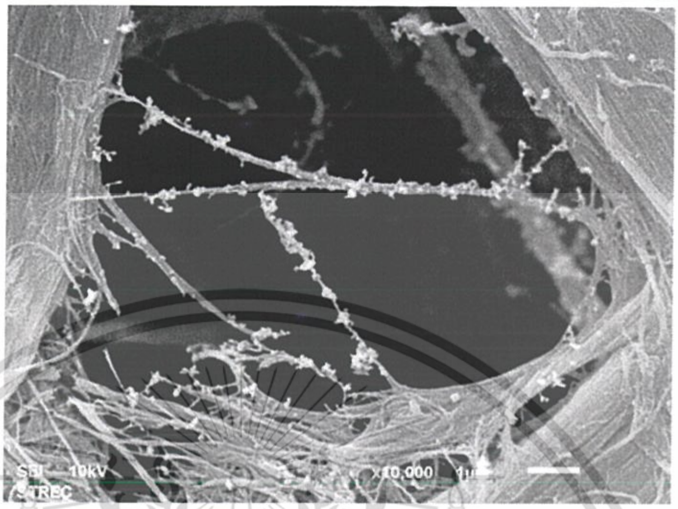
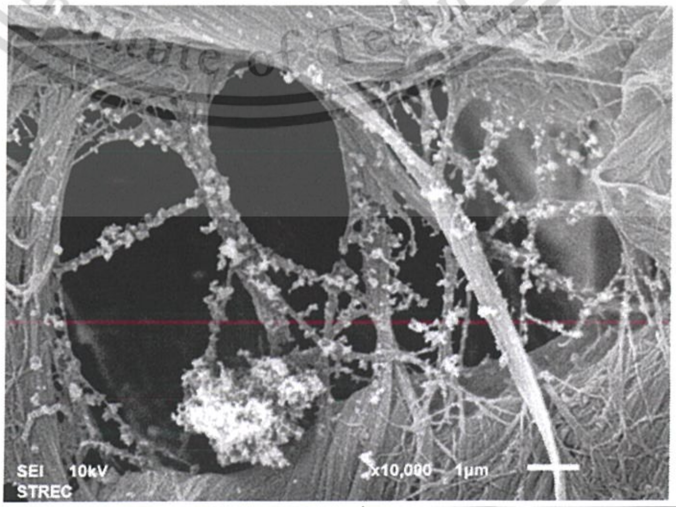
TYPE / FUEL.	Particulate Matter (PMs)
Fuel: E0 IGN T. 17°	
Fuel: E0 IGN T. 27°	
Fuel: E0 IGN T. 32°	

Table 4.10 Particulate matter from scanning electron microscope technique of E20 at each ignition timing at 2000 rpm, 3.0 bar nominal BMEP.

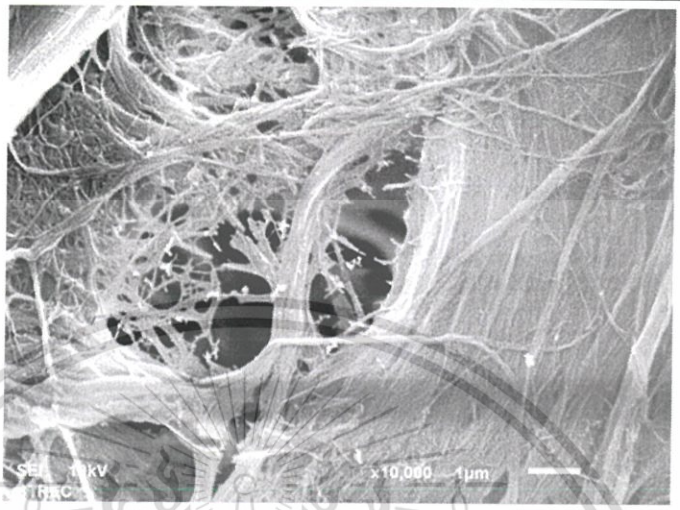
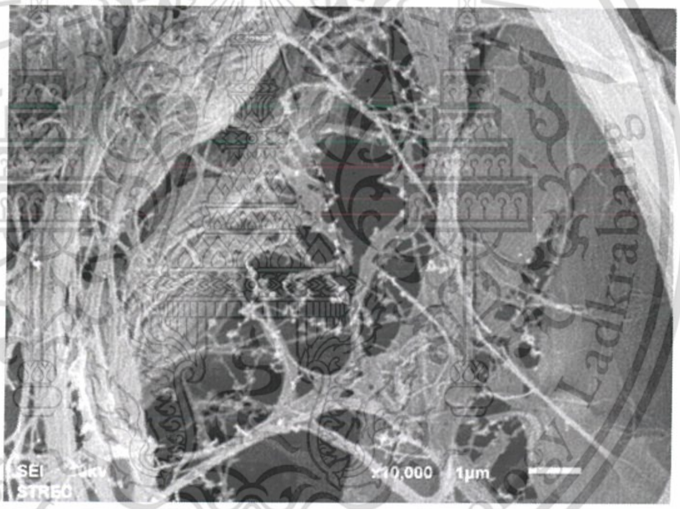
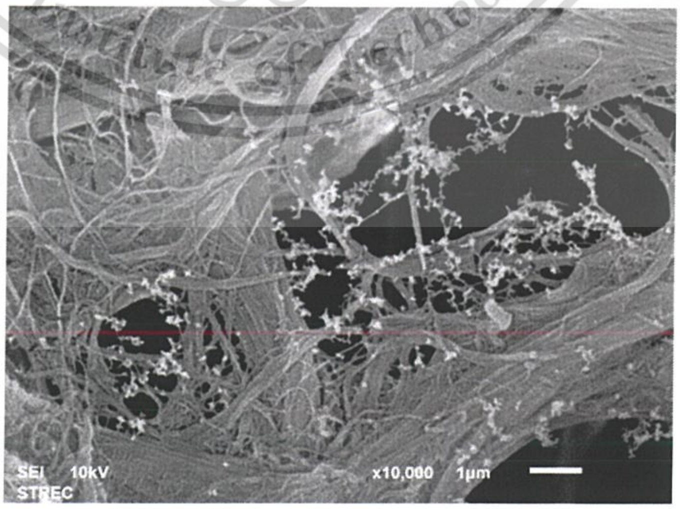
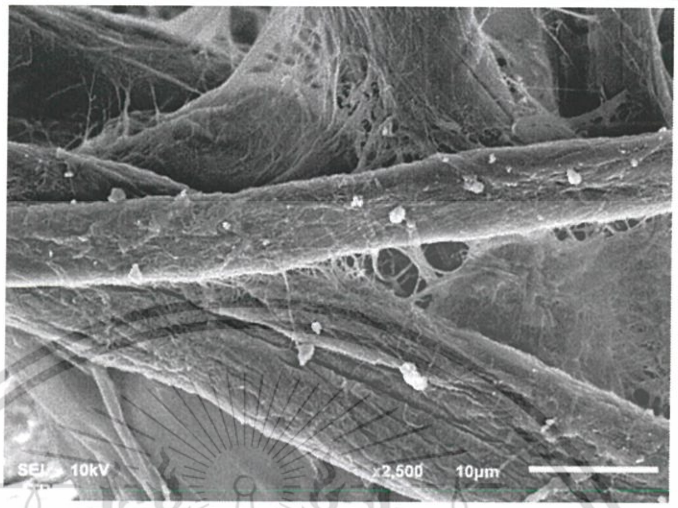
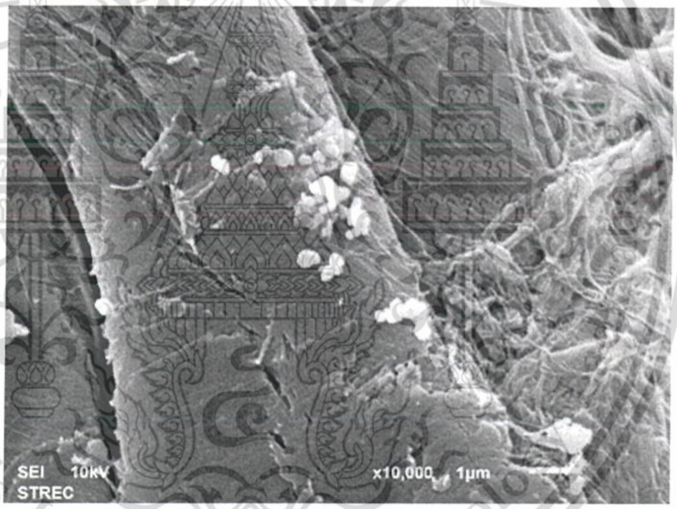
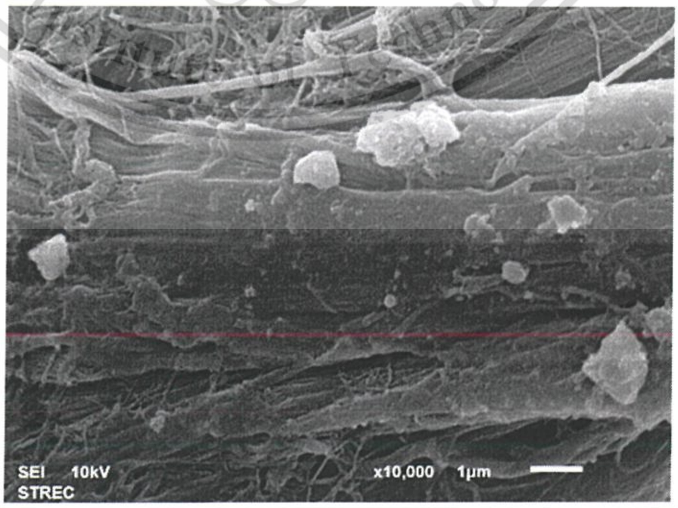
TYPE / FUEL.	Particulate Matter (PMs)
Fuel: E20 IGN T. 17°	
Fuel: E20 IGN T. 27°	
Fuel: E20 IGN T. 32°	

Table 4.11 Particulate matter from scanning electron microscope technique of E85 at each ignition timing at 2000 rpm, 3.0 bar nominal BMEP.

TYPE / FUEL.	Particulate Matter (PMs)
Fuel: E85 IGN T. 17°	
Fuel: E85 IGN T. 30°	
Fuel: E85 IGN T. 32°	

4.3.3 Effect from Ethanol Blends

The effect of ethanol blends on particle size distribution portrays in figure 4.27 and table 4.8. To blend more of ethanol cause size distribution shift to larger. The percentage of average size of E85 is largest (E85: 62.81%) following with E20: 15.49% when compare with E0 due to the fact that first, the injection duration of ethanol blends is longer that gasoline from lower calorific value which cause fuel impingement on the piston and combustion surface that induces pool fire to this region. Second, higher of heat vaporization of ethanol leads slow fuel evaporation. Hence, it produces more heterogeneous of combustion.

From above reasons, many hydrocarbons which evaporate from fuel pool after combustion were condense on the surface of primary soot particles formed in cylinder together with pool fire of blend fuel evaporate slower than gasoline from high heat of vaporization and lower vapor pressure. So, the particle of ethanol blends fuel emitted higher volatile organic content particle [45] which result in higher mass fraction volatile compound and larger size distribution when compare with gasoline.

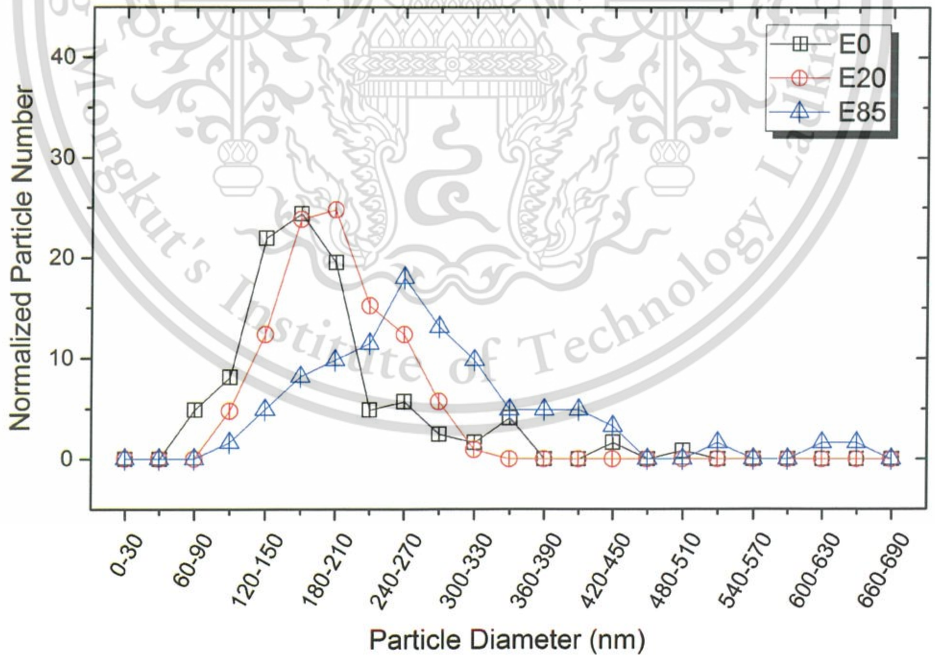
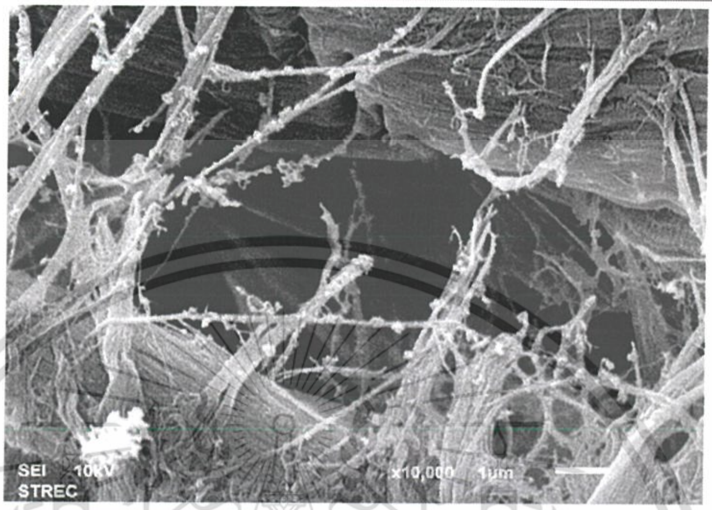
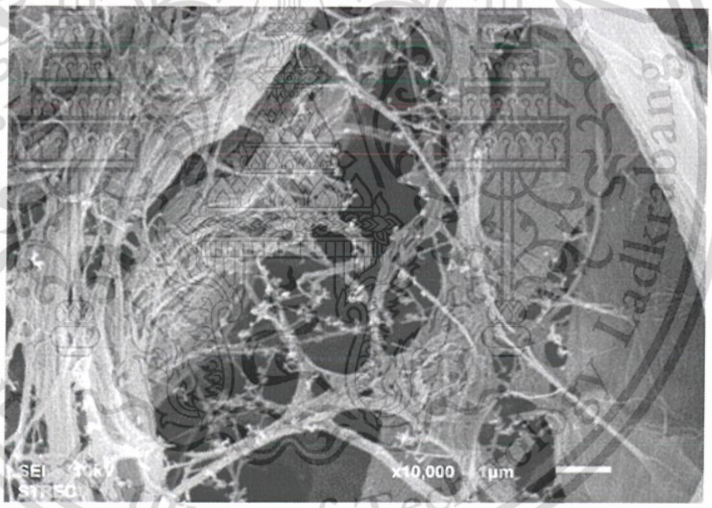
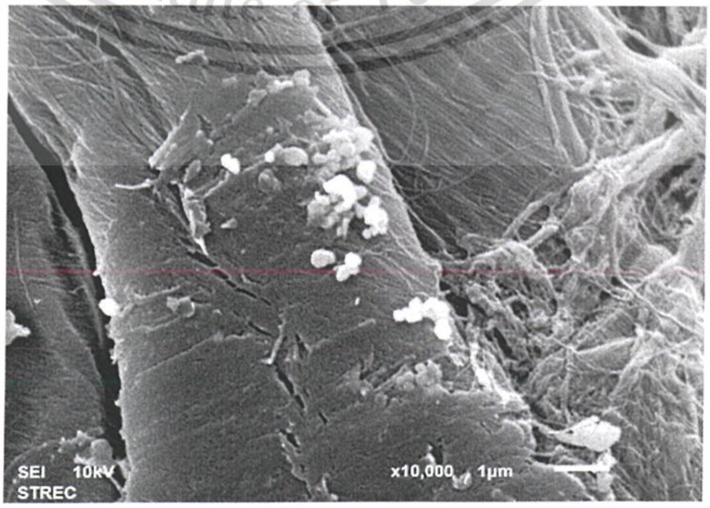


Figure 4.27 Effect of ethanol blends on particle size distribution at 2000 rpm, 3.0 bar nominal BMEP.

Table 4.12 Particulate matter from scanning electron microscope technique of tested fuel at maximum brake torque timing at 2000 rpm, 3.0 bar nominal BMEP.

TYPE / FUEL.	Particulate Matter (PMs)
E0	
E20	
E85	

Chapter 5

CONCLUSIONS

5.1 Effect of Injection Timing and Ignition Timing

5.1.1 The impact of injection and spark timing are relate to the fuel cloud formation. To change both timings over the optimum cause high sensitivity to emissions which is disadvantage of stratified charge operation and results in deteriorate of efficiencies.

5.1.2 When injection timing is retarded from optimum point causes the mixture around the spark plug due to reduce duration of fuel to evaporate and mix with air. Furthermore, these reduces gas flow rate in compression stroke from severe fuel impingement at cylinder and piston wall.

5.1.3 At late injection timing, the piston and wall impingement and fuel remains causes THC increased. The NO_x is reduced from low combustion temperature. On the other hand, CO is increased by poorer mixture formation process.

5.2 Effect of Ethanol Blends

5.2.1 The efficiencies of direct injection spark ignition engine fueled with gasoline, gasoline-ethanol shows the brake specific fuel consumption is increased 4.98% of E20 and 33.12% of E85 respectively due to low heating value of gasoline-ethanol is approximately 0.6 times of gasoline which should inject more quantity than gasoline. However, the brake specific energy consumption is lower (4.01% of E20 and 11.55% of E85) and higher in brake thermal efficiency as 4.18% of E20 and 13.06% of E85. It can imply as higher laminar flame speed of gasoline-ethanol reduces heat loss throughout the surroundings from enhancing initial stage of combustion. Furthermore, the ethanol blended has higher octane number and ignition delay which can advance more degree of injection and ignition timing. These results in higher efficiencies.

5.2.2 Gasoline-ethanol can reduce emissions in direct injection spark ignition engine when operates in stratified charge mode as the decreases of carbon monoxide; E20 and E85 are 25.16% and 42.17% and the reduction in THC of E20 and E85 are 43.99% and 90.64% respectively are lower by oxygen atom inside molecule bring about complete combustion that conform to brake thermal efficiency tendency when fueled more of ethanol.

5.2.3 In case of oxide of nitrogen, blending higher percentage of ethanol can reduce NO_x emissions as 23.60% of E20 and 58.02% of E85. The temperature of combustion is major factor to reduction. The result of exhaust temperature in gasoline-ethanol are lower than that gasoline due to heat of vaporization in ethanol reduces the combustion temperature which conforms to the reduction of exhaust temperature.

5.3 Particulate Emissions;

5.3.1 Injection timing is outstanding factor to impact the particulate matter. It's directly effect to the mixing process between air and fuel. Longer duration led more homogeneous mixture before ignition that reduce size of particle whereas fuel-rich region from liquid fuel impingement by retard injection timing results in larger size of particulate matter.

5.3.2 Spark timing is insensitive to particulate matter size distribution because it not significant in size distribution and causes disadvantage on engine efficiency or other pollutants.

5.3.4 Particulate matter of gasoline blended with ethanol obtains from higher volatile organic compounds mainly due to the decreasing of carbon formation by ethanol combustion, higher of heat vaporization and lower calorific value causes longer injection duration that leads to fuel impingement and slow vaporization. These contributes to larger size distribution as 15.09% of E20 and 62.81% of E85 when compare with gasoline.

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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-GSH-1230331
Customer/Supplier : International College, King Mongkut's Institute of Technology
International College, King Mongkut's Institute of Technology Ladkrabang,
Chalongkrung Rd. Ladkrabang BKK 10520
Sample Location : LKB
Batch No. : -
Product Source : -

Received Date : 18 Dec 2012
Date of Test : 18 Dec 2012
Date of Sampling : 18 Dec 2012
Sample Condition : Normal

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

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

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Page 1 of 1

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

Approved by

(Phunta Pothisuk)

Position Title : Vice President in Quality Analysis Department

Date of Issue : 25 Dec 2012

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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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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,
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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

Approved by : Phurita
(Phurita Pothisuk)
Position Title : Vice President in Quality Analysis Department
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APPENDIX B: Publication

The 6th TSME International Conference on Mechanical Engineering
16-18 December 2015

Harmonized Engineering Technologies

AEC004

Investigation of Performance and Emissions in a DISI Engine when Using Ethanol Blends as Fuel

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Abstract

Nowadays the major problem of automobile is extravagant of fuel consumption in transportation affected in many ways as currency and fuel economy dissipative including with environmental impurity. Those belong to environmental quality declination and lack of fossil fuel especially in transportation that uses lots of fuel mostly consumed by motor vehicle. So the opportunity to save currency and improve environmental quality through fuel use reductions is clear by using ethanol as renewable resources from benefits its properties together with direct injection spark ignition (DISI) engines.

The objective of this research presents effect of ignition timing on performance and emissions when fueled with ethanol. The four cylinder and four stroke of direct spark ignition engine (DISI engine) is tested and controlled spark timing by standalone electronic control unit. To test performance and emissions were conducted by using difference purity ethanol blended in gasoline (E10, E20 and E85) and gasoline (E0), variation of spark timing and stoichiometric A/F ratio which measured power, torque, break specific fuel consumption (BSFC) and break specific energy consumption (BSEC) at 2000 rpm of low and medium loads. The results showed that when compare with gasoline, Fuel blends with higher content of ethanol causes ignition timing to advance 3° CAD which obtained 26.79% more torque together with emissions have reduced likewise; carbon monoxide-31.03% and hydrocarbon-42.86%. However, carbon dioxide is higher than 4.86% due to quantities of fuel blended with ethanol has lower heating value.

Keywords: Ethanol, Ignition Timing, Direct Injection Spark Ignition Engine, DISI

1. Introduction

With contribution to performance of engine, the fuel consumption in transportation field is an important reason to realize for propelling the motor vehicles. In particular, the total energy consumption in the world depends on the remaining fossil fuels. Using new development technologies from a direct injection spark ignition engine and renewable oxygenated fuels such as ethanol are considered as the most suitable solution for the future.

To investigate the ethanol characterization and optimization to an engine have been studied by many researchers. P. Ornman [1] found that ethanol has started to vaporize slower than ethanol but when terminated, ethanol was fully evaporated in to vapor phase faster than that of gasoline because of gasoline has more light and heavy fraction that ethanol. Hence, the lighter components leads to evaporated early at the beginning stage while ethanol wasn't start. After that, heavy fraction in gasoline which are comprised of various higher carbon atoms regardless to RVP properties still remained while ethanol which is comprised of only one component, lower carbon atom than the gasoline, already completely changed into the vapor phase.

The effect of ethanol-unleaded gasoline blends on engine performance and exhaust emission was studied by Hasan [2], 4-stroke SI-engine was fueled with gasoline and gasohol with various load and speed (1000 to 4000 rpm) at 75% of full throttle. The results founded higher percentage of ethanol increased the brake power, brake specific fuel consumption, thermal and volumetric efficiency. Carbon-monoxide (CO) and hydrocarbon (HC) were decreased 46.5% when compared with gasoline. However, carbon-dioxide was increased up to 7.5%.

Türköz and et al. [3] studied the use of E85 in the engine with 9.2:1 of compression ratio and advancing ignition timing from original gasoline timing. This study operated at wide open throttle ranging speeds from 2250, 2500, 3000 3500 to 4000 rpm, respectively. The results found that engine torque and power were increased when using E85 advanced ignition timing of 4 degree crank angle and causes minor effect of carbon-dioxide and carbon-monoxide reduction. On the contrary, nitrogen oxide, hydrocarbon and brake specific consumption were increased [4, 5].

However, using ethanol blends together with a DISI engine is interesting due to high octane number fuel and the optimization of spark advance which influence to better performance and reduction of emissions. Hence, investigation the effect from properties of ethanol and DISI engine operations is essential to develop insightful information.

The objective of this study has focused on the sensitivity of gasoline compare with ethanol as alternative fuel together with engine modification by adjust an ignition timing and control a lambda. The advantages from the results are discussed on performance and emissions in order to optimize the DISI engines when using ethanol as a fuel.

2. Experimental Apparatus

2.1 Fuel

Compared with gasoline, using ethanol as a fuel leads to benefits by outstanding properties for instance high research octane number (RON) which causes better anti-knock quality from increased of compression ratio that obtained better volumetric efficiency, the stoichiometric air fuel ratio is lower due to the fact that it has oxygen content inside its molecule leads to complete of combustion and reduce emissions from surplus of oxygen to react remains. However, ethanol must injected more quantities because of lower heating value to achieve same of output energy and cold start problem from high initial boiling point (IBP) when compare with gasoline as shown in table 1.

Table. 1 Fuel Properties [6, 7]

Fuels properties	Gasoline	E10	E20	E85	Ethanol
Formula	C_4 to C_{12}	[illegible]	[illegible]	[illegible]	C_2H_5OH
MW. [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
RVP at 37.8°C, kPa	48-103	59.60	58.30	35-70	15.90
LHV, MkJ/kg	44.00	40.97	40.60	29.50	26.90
HOV, kJ/kg	305.00	-	-	610-762.5	840.0
RON	92.0-98.0	96.0	98.3	101.6	107.0
MON	80.0-90.0	84.1	84.6	91.1	89
Stoichiometric air/fuel ratio	14.70	14.05	13.51	9.87	9.03
Distillation temperature, °C					
Initial boiling point, IBP	35	39.7	42.1	46.9	77.6
10 vol%	51.5	52.5	54.3	68.6	77.8
50 vol%	78.2	79.1	70.5	77.6	77.9
90 vol%	154	153.6	150.9	78.5	78
End boiling point	197.3	186.1	182.8	79.7	80

2.2 Engine

Gasoline direct injection (GDI) or direct injection spark ignition (DISI) engine is a variant of fuel injection employed in modern engines. The gasoline is highly pressurized and injected directly into the combustion chamber of each cylinder, as opposed to conventional multi-point fuel injection in which fuel is injected in the intake port. In some applications, gasoline direct injection enables a stratified fuel charged (ultra-lean burn) combustion for improving fuel efficiency and reducing emission levels at low load.

The in-line 4 cylinders with 1,834 cm^3 displacement of DISI engine is operated in this experiment which shown in table 2.

Table. 2 Engine Specifications

Model	MITSUBISHI 4G93 GDI
Type	In-Line OHV, DOHC 16 Valve
Number of Cylinder	4
Displacement (cm^3)	1,834
Compression Ratio	12 : 1
Bore x Stroke (mm)	81.0 x 89.0
Maximum Output	96 kW @ 6000 rpm
Maximum Torque	177 Nm @ 3750 rpm

2.3 Engine Dynamometer

The "Tokyo Plant 150 PS Model" engine dynamometer interfaced with in-house program has measured the results of this experiment likewise; power, torque, brake specific fuel consumption as shown in Fig. 1.

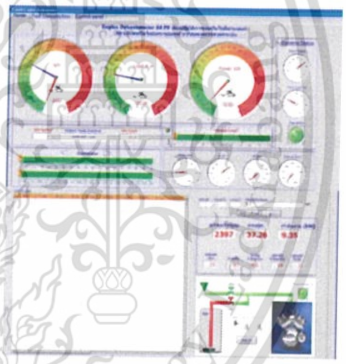

Fig. 1 Engine dynamometer with in-house controller

2.4 Electronic Control Unit (ECU)

The DTA fast S60 Pro is standalone electronic control unit of experimental engine which controls quantity, duration of fuel injection and spark timing with precision resolution as presented in Fig. 2.

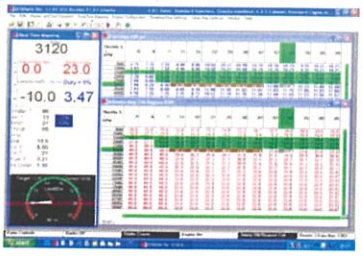

Fig. 2 DTA fast S60 Pro user interface

3. Methodology

A DISI engine has tested at 2000 rpm with 20% and 40% of load which represent as light and middle load condition. Gasoline and ethanol blended fuels (E10, E20 and E85) are used in this experiment which runs on homogeneous charge by variation of ignition timing from 18° to 36° bTDC. All conditions are controlled at $\lambda \approx 1$ by Innovate Lm-2 oxygen wide band sensor and 87°C of engine coolant. Emissions are examined from exhaust pipe directly and evaluated by KOENG KEG-500 gas analyzer illustrated in Fig. 3 Moreover, performances for instance power, torque, brake specific fuel consumption (BSFC) and brake specific energy consumption (BSEC) have measured due to understanding the effect from different fuel and conditions of engine.

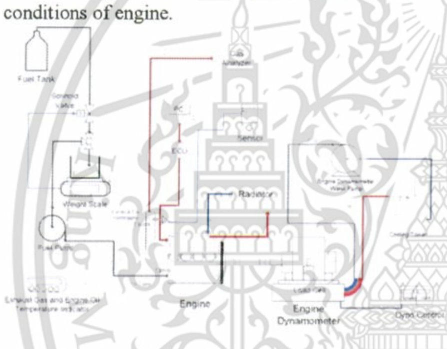

Fig. 3 Schematic diagram of experimental setup

4. Results and Discussion

4.1 Engine Performance Characteristics

4.1.1 Brake Torque

The results are discussed at the maximum break torque (MBT), the optimum timing which explained as the minimum advance of ignition timing. The gasoline (E0) was found at CAD (crank angle degree) of 27° followed with E10, E20 and E85 at 30° 30° and 33° bTDC, respectively.

When compared with E0, using ethanol blended with gasoline E85, E20, and E10 increased the engine torque 19.53%, 6.09% and 2.09% at 20% of load and 26.79% , 14.70% and 6.46% at 40% of load respectively regarding to fuel mass at the same air-fuel ratio where illustrated in Fig. 4-6. The increment tendency of brake torque is discussed as follow.

Firstly, ignition timing impacts to break torque as advanced ignition timing leads to early process of combustion in the cycle. This extends the residence time and the reaction of partial oxidation during the combustion stroke. Peak pressure and high temperature are resulted from the combustion of most fuel when the piston reaches closer to top dead center while the volume of cylinder becomes smaller which is an idealistic crank angle of combustion. Hence, thermal efficiency is increased [8]. Nevertheless, too advanced spark timing lets the abundance of gas burnt by the time of piston going up. Then, the net power will be lessened from the work used to compress gas.

Conversely, the highest pressure and temperature is decreased with retarded ignition timing due to inadequate time between top dead center and ignition timing to complete chemical reaction. Then, several of fuels are burnt after top dead center known as post-reaction [9]. Hence, incomplete combustion is assorted with retarded ignition timing.

Lastly, the LHV of fuel blended with ethanol is lower than that of gasoline. The plenty of ethanol injected into the engine test should be about double amount of gasoline for achieving the same heat energy output which mounted up in torque and power [9]. Moreover, higher latent heat of vaporization causes air intake temperature in combustion chamber lower because of evaporation of alcohol which leads to increased volumetric efficiency together with too delay of ignition timing from high octane number. Hence, advanced ignition timing is required [3]. A higher flame speed of oxygenated fuels reduce combustion duration equate to gasoline which causes shorter flame propagation. Thus, the rate of energy releases rapidly which decreases heat loss from the engine due to insufficient time for heat to depart the cylinder through heat transfer to engine coolant [10].

From this result founded that to increase 12.63% of ethanol's RON causes 3 degrees of advance spark timing. This experimental result is identical with other researches [11] and [12] as augmentation of octane number or conversely ignition delay. Then, using ethanol with early ignition is essential.

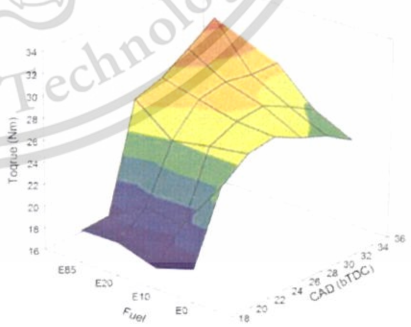

Fig. 4 Torque from DISI Engine at 20% load

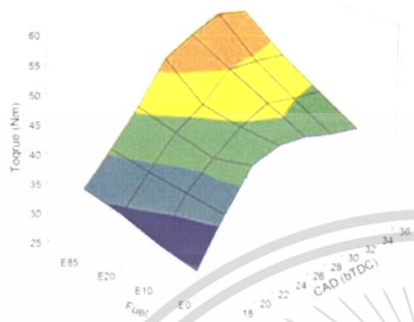

Fig. 5 Torque from DISI Engine at 40% load

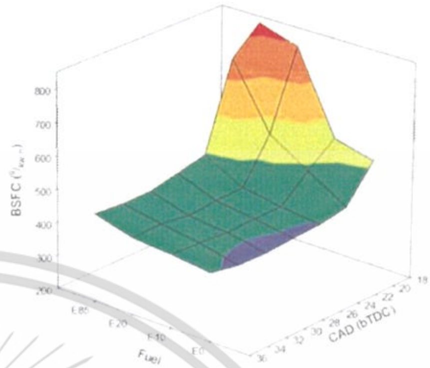

Fig. 7 Brake specific fuel consumption from engine at 20% load.

Fig. 6 Torque trends at optimum ignition timing

Fig. 8 Brake specific fuel consumption from engine at 40% load.

4.1.2 Brake Specific Consumption and Brake Specific Energy Consumption

Fig. 7-8 portrays the BSFC of the test fuels at diverse of engine load. It showed that at optimal timing, gasoline has the lowest BSFC when compared to other ethanol gasoline blends. At the 20% of load BSFC increased with E10, E20 and E85 by 4.00%, 7.12% and 15.92% whereas at 40% of load ethanol additions in gasoline increased 4.48%, 6.70% and 34.98% with E10, E20 and E85 respectively.

The heating value of fuel is an important factor which affects brake specific fuel consumption (BSFC) of engine. This increasing relies on the percentage of ethanol in fuel where ethanol has heating value less than gasoline around 30% [13]. More ethanol blended in the fuel to produce same heat energy at similar condition resulted to the increment of BSFC.

Fig. 9 shows the comparison of fuel consumption at different load. The result indicates that higher load causes less fuel consumption due to lower of pumping loss from throttle plate is open wider.

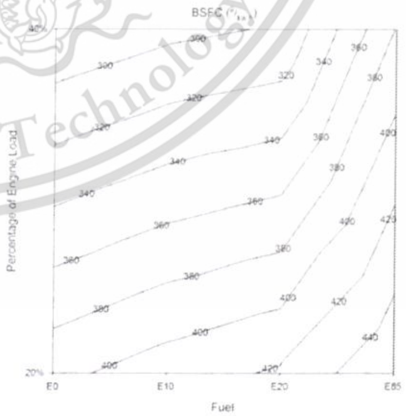

Fig. 9 Tendency of brake specific fuel consumption at optimum ignition timing

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Brake specific energy consumption (BSEC) is described as the quantity of energy consumed per unit power developed in a unit of time. Described in briefly, BSEC is how efficiently of energy obtained from its fuel. With the same condition as above BSFC, Fig. 10-12 depict that ethanol blends cause the decreased brake specific energy consumption by 3.16%, 1.16% and 22.28% with E10, E20 and E85 at 20% load and around 2.71%, 1.27% and 9.50% with E10, E20 and E85 respectively of 40% load. This reverse trend was investigated due to lower calorific value with increasing in the percentage of ethanol blends.

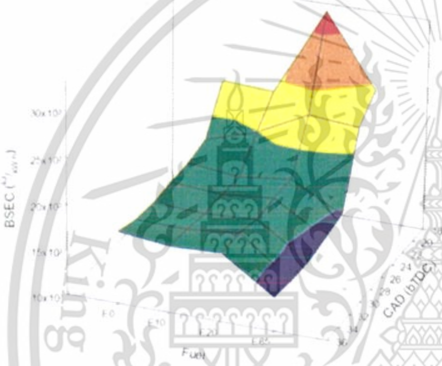

Fig. 10 Brake specific energy consumption from engine at 20% load.

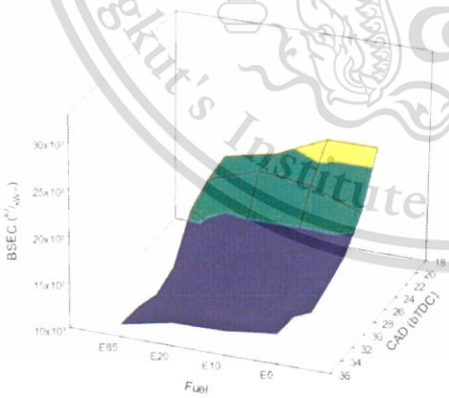

Fig. 11 Brake specific energy consumption from engine at 40% load.

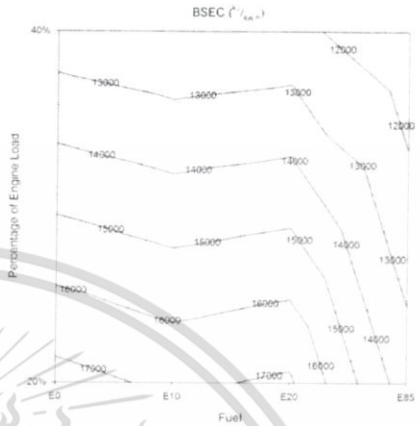

Fig. 12 Brake specific energy consumption trends from engine at optimum ignition timing

4.2 Emissions

4.2.1 CO and HC emission

Carbon monoxide (CO) and hydrocarbon (HC) are influenced by oxygen which available during combustion. So, there are anticipated to reduce as the mixture becomes leaner.

Using ethanol blends reduce HC as 38.97%, 53.31% and 59.19% of 20% load along with 21.73%, 34.18% and 42.86% of 40% load at optimal ignition timing with E10, E20 and E85 respectively as shown in Fig. 11(a). Fig. 11(b) shows the reduced CO of 26.47%, 44.12% and 50.00% of 20% load along with 8.62%, 27.59% and 31.03% of 40% load at optimal ignition timing with E10 E20 and E85 respectively.

The lower CO and HC than those of gasoline due to oxygen content of ethanol in blended fuel mount up the ratio of oxygen to fuel in over-rich zone. The actual of air-fuel ratio becomes stoichiometric whereas increasing of ethanol content in fuel contributes to more complete combustion which leads to decrease soot formation [14].

To consider the impact of timing, advance in ignition timing is important strategy when using ethanol due to higher of octane number and latent heat of vaporization compare to gasoline which cause prolong of ignition delay from heat absorption. Then, reduction of those emissions from the short duration between the terminate of fuel injection and ignition timing contributes to improvement of flame propagation from increasing of turbulence intensity and high mixture stratification which defined as the mixture near the spark plug which is locally rich and retains a constant overall lean air-fuel ratio [15-17].

However, to retard ignition timing from optimum timing decreases flow intensity in the cylinder and lessens in the mixture stratification which mount up the fraction of unburned hydrocarbon in the region of lean mixture as well as CO is increased from incomplete combustion.

Lastly in term of engine load, those emissions are increased. The more of engine load, the more fuel is injected. Thus, flame speed is important factor for completing combustion of rich mixture conditions together with high engine load. This study showed outstanding property of ethanol that lets higher flame speed to assist this process complete and to reduce the duration of combustion. The high blend ratio of ethanol can reduce emissions effectively.

Fig. 11a Carbon monoxide (CO) and Fig.11b Hydrocarbon (HC) trends from DISI engine affected by variation of fuel from E0 to E85 and ignition timing from 18° to 36° bTDC

4.2.2 CO_2 Emission and O_2

As seen in fig. 12a, ethanol blended with gasoline increase CO_2 emissions as 0.08%, 0.64% and 2.25% of 20% load coped with 3.84%, 4.70% and 4.86% of 40% load as shown in fig. 12a following with O_2 as 18.39%, 27.59% and 32.18% of 20% load together with 17.14%, 28.57% and 32.38% of 40% load as shown in fig. 12b. Even though, the fuel quantities of ethanol blended is more than gasoline but, the oxygen content in ethanol lets excess air to combust residue emissions and enhance complete combustion which leads higher of CO_2 [9].

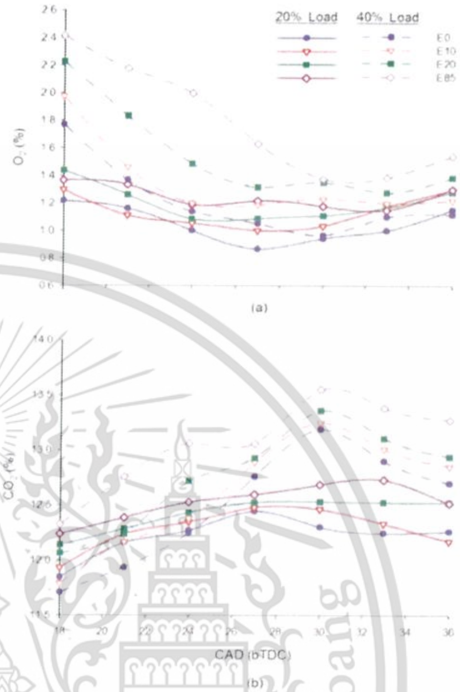

Fig. 12a Carbon dioxide (CO_2) and Fig. 12b oxygen (O_2) trends from DISI engine by variation of fuel and ignition timing

5. Conclusion

1. To advance ignition timing brings higher torque and lower emission from complete combustion whereas retard of ignition timing is contrary from post-reaction and lower peak pressure which occurred in expansion process.

2. Gasoline blended with ethanol increases brake torque from quantity that injected and reduces carbon-monoxide and hydrocarbon by excess air of oxygen content in its fuels. However, brake specific consumption is higher than gasoline due to lower heating value. Conversely, brake specific energy consumption is lower because more energy is released than gasoline.

3. High octane number of ethanol causes too ignition delay from higher heat absorption. Then, higher flame speed of ethanol together with advancing ignition timing is suitable from fuel blended with ethanol for complete combustion.

This experiment will conduct the DISI engine suitably using ethanol blended with gasoline and utilizing the benefits of spark timing which adjusted together with outstanding properties of ethanol to operate on stratified mode with less emission in the next study.

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Publication:

1) Kangkaya P., Karin P., Phairote W., Charoenphonphanich C., Tongroon M. and Kosaka H., "Investigation of Performance and Emissions in a DISI Engine when Using Ethanol Blends as Fuel", The 6th TSME International Conference on Mechanical Engineering, Hua-Hin, Thailand, December 16 to 18, 2015.