



Combustion Characteristics of Direct Injection Stratified Charge of Gasohol Fuels

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Abstract

In this research, combustion characteristics of ethanol/ gasoline blends in direct injection stratified charge engine were carried out using a constant volume combustion chamber. To evaluate the influence of blending on the combustion behavior, the flame propagation of different blends of ethanol in gasoline (20%, 85% and 100% ethanol) as well as pure gasoline were investigated under various swirl intensities and equivalence ratios. Pressure data taken during the testing allowed for detailed analysis. The different blends of fuels were compared in terms of combustion characteristics, rate of pressure rise, combustion duration and flammability limit. In addition, stratification degree that demonstrates the stability of combustion in lean operation will be investigated and discussed.

Key words: Constant volume combustion chamber, direct injection, stratified charge, Ethanol/gasoline blends, Swirl intensities

1. Introduction

In the recent years, trend of consumption in energy has been increasing continuously. Alcohol, especially ethanol, was the new challenge candidate in alternative fuel since it can be produced from many source of biomass and is indeed the renewable energy. In addition, the raw materials for ethanol production, cassava and sugarcane, are also the main economic vegetation in Thailand. Despite the lower heating value of alcohols is lower than that of gasoline, alcohols release a little more heat than gasoline under the same equivalence ratio. Moreover, a

high octane number will allow an increase in high compression ratio; thus, an engine fueled with ethanol can have higher power output and better thermal efficiency.

For stratified charge engine, it is well know that lean, stratified combustion can reduce fuel consumption and gain some merits in gasoline spark-ignited, direct injection engines for several reasons. First, unthrottled operation allows for a significant reduction in pumping loss, especially at low loads. Second, the lean mixture being compressed has a higher ratio of specific heats. This allows for a more efficient



compression and expansion process. Third, there are lower wall heat losses in the cylinder because of the centralization of the mixture away from the walls [1].

This challenge to utilize ethanol-gasoline blends in direct injection gasoline engines is very interesting because ethanol has an anti-knock properties, as well as higher heat of vaporization, compared to pure gasoline, which allows for combustion with higher compression ratio [2]. As a result, using ethanol in stratified-charge, spark-ignition, direct-injection engines is capable of achieving significant gains in both volumetric and thermal efficiencies [3].

Although this combustion can achieve many kinds of advantage for combustion characteristics, it produces much unburned hydrocarbon and soot because of inhomogeneous charge mixture in the combustion chamber. The main problem is a misfiring under lean operation even if whole air-fuel mixture is very lean [4]. In stratification, air-fuel ratio tends to be over-rich in the middle of the mixture and over-lean in the periphery bordering surrounding air. It is essential to minimize the abovementioned air-fuel ratio difference by enhancing the stratification degree, which represents the ratio of fuel quantity involved in nearly stoichiometric mixture zone to total fuel quantity. As the stratification degree becomes low, unburned fuel amount increases and fuel economy deteriorates. Therefore, it is necessary to investigate the combustion characteristics in order to obtain the stable lean combustion.

In the combustion characteristics of stratified charge combustion, there are many factors that affect to the combustion process,

such as physical properties of the fuel, swirl intensity, ignition timing, equivalent ratio, ambient temperature and pressure etc. In this study, the effect of swirl ratio on the combustion characteristics was investigated in the ethanol /gasoline blends, which vary from pure gasoline (E0) to pure ethanol (E100) in the constant volume combustion chamber. With the pressure analysis data, the rate of pressure rise (dP/dt), combustion duration and mass fraction burned rate were analyzed. Through this study, a better understanding of stratified charge combustion run on ethanol/gasoline blends can be achieved.

2. Experimental apparatus and procedure

2.1 Experimental apparatus

Fig.1 shows the schematic diagram of the experimental apparatus used in this study. The following equipments were used: a constant volume combustion chamber, cam-driven pump providing the high pressure fuel to the injector, an air compressor and an intake swirl port with pressure regulator for making a swirl intensities and inducing the fresh air charge into the combustion chamber, a swirl nozzle injector, heating system for controlling the air temperature in combustion chamber, a conventional ignition system of the CDI (capacitor discharge ignition) type for generating ignition energy, pressure and temperature measuring equipment. Controller box was used to control the signal sequence and to interface with a data acquisition system.

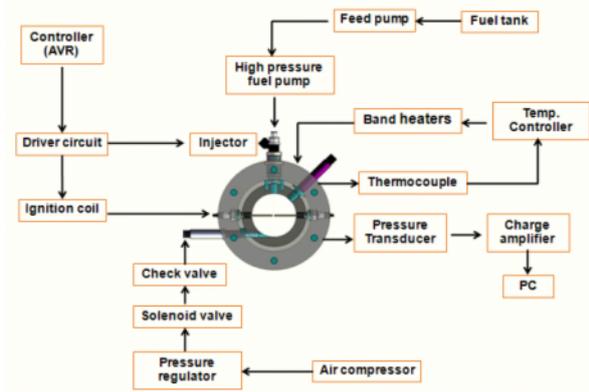


Fig. 1 Schematic diagram of experimental apparatus

Fig. 2 shows a schematic diagram of the constant volume combustion chamber. The internal diameter and width of the chamber are 70 mm and 100 mm, respectively. The volume capacity is 385cc. The quartz glass windows with 35 mm thickness are equipped at both sides of the combustion chamber for optical access and visualization of the flame propagation. Swirl port allows the induced air to flow into the combustion chamber in the tangential direction to realize the swirl flow in the chamber. Graphite gaskets are attached in both side of quartz glass to absorb impact load from combustion. The spark plugs are located to the central of the combustion chamber to generate the spherically expanding flame and prevent heat transfer to the chamber wall.

Ambient pressure (P_a) that is the initial pressure inside the chamber is controlled by adjusting the pressure regulator, which also induces pressure corresponding to the swirl intensities. A piezo-electronic pressure transducer (Kistler model 611BFD16) was used to measure the combustion pressure and the charge amplified electrical signals including high speed data acquisition system (Dewetron model DEWE-5000) were used to record the signal from the

pressure sensor. Two band heaters (1300 W) were attached to the chamber wall to control the initial temperature (T_a).

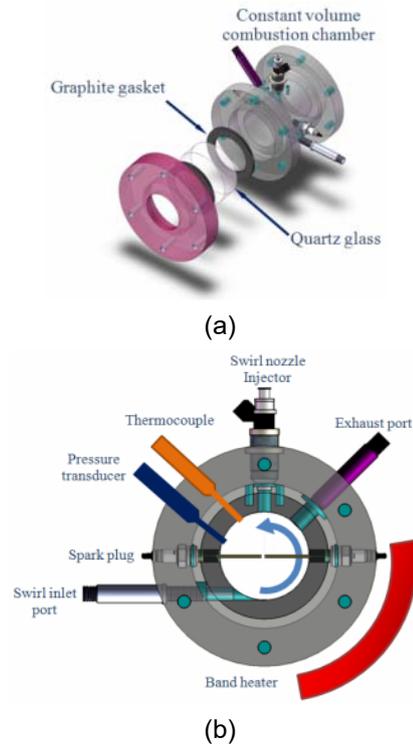


Fig. 2 Schematic diagram of the constant volume combustion chamber in (a) disassembly and (b) cross-section views.

2.2 Experimental procedure

Table. 1 shows the experimental condition

Experimental variables	Conditions
Ambient temperature (K)	450
Ambient pressure (MPa)	0.5
Ignition time (msec)	10
Injection pressure (MPa)	4.5
Inj – Ign Interval (msec)	12
Equivalent ratio	1.0,0.8,0.6
Swirl intensities or ΔP (MPa)	0.5,0.4,0.3,0.2,0.1

Since ethanol-blended gasoline fuels are in the liquid phase at the normal temperature, initial temperature in this experiment was fixed at 450K, which is higher than evaporation temperature of all tested fuel for avoid misfire when the fuel was injected into the combustion

chamber. To observe combustion behavior in early stage of combustion, the initial pressure (P_a) was set at 0.5 MPa. The total equivalence ratio of 1.0, 0.8, and 0.6 was simulating in overall lean condition.

2.2.1 Setup of equivalent ratio (ϕ)

The mass of the injected fuel was calibrated with the time of injection as shown below in Fig. 3

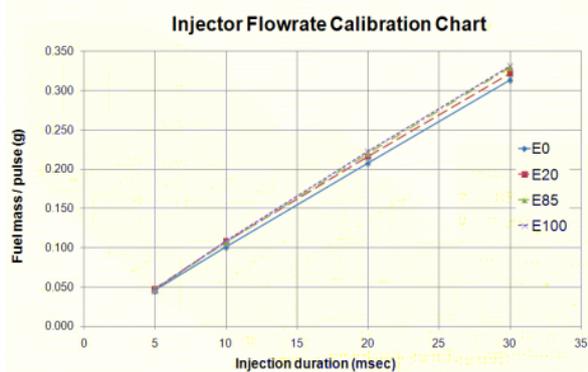


Fig. 3 Calibrated chart for injector

The injection duration can be controlled by the air mass, which is induced into the combustion chamber. Air mass in this experiment can be calculated following these equations.

$$m_{(air\ total)} = m_{initial} + m_{swirl} \quad (1)$$

where $m_{initial}$ can be calculated from ideal gas law at initial pressure and m_{swirl} can be directly measured by anemometer.

For calculation of the equivalent ratio, the following equation can be used.

$$\phi = \frac{(m_f / m_a)_{actual}}{(m_f / m_a)_{theory}} \quad (2)$$

where $(m_f / m_a)_{actual}$ is real fuel /air ratio

$(m_f / m_a)_{theory}$ is theoretical fuel /air ratio

From equation (2) theoretical air/fuel ratio can be calculated from the chemical equilibrium. The fuel properties and their air/fuel ratio is shown in Table. 2.

Table. 2 The tested fuel properties [6]

Fuel	E0	E20	E85	E100
Formula	($C_{8.26}H_{15.5}$)	-	-	(C_2H_5OH)
molecular weight	114.18	88.12	50.60	46.07
LHV (MJ/kg)	44.0	-	-	26.9
Air/fuel ratio	14.63	13.51	9.87	9.03
Heat of vaporization (kJ/kg)	305	-	-	840
Specific gravity	0.72-0.78	-	-	0.8
RON	92.0-98.0	-	-	107.0
MON	80.0-90.0	-	-	89.0

The ignition duration time was set at 10 msec. Time duration from injection to ignition in this experiment was fixed at 12 msec to ensure that the fuel spray can be fully developed. Swirl intensities were controlled by the different pressure between the air induced pressure and the chamber pressure. All of timing parameters and sequence of signal were controlled by the PC interface program.

After inducing the pressurized air into the chamber to generate swirl flow, the swirl nozzle injector will injected the fuel directly in to the combustion chamber, then spark plug will ignite the mixture at the same time as the data acquisition recording the pressure data. This signal sequences are shown in Fig. 4

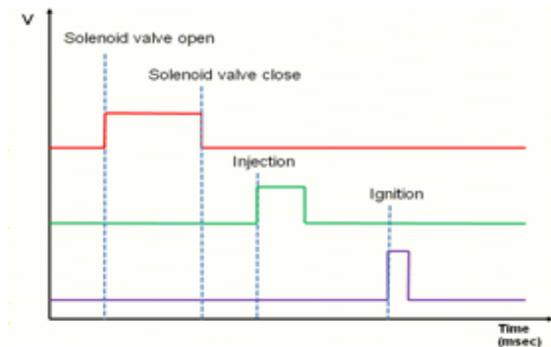


Fig. 4 Pattern of signal sequence

2.2.2 Lean limit investigation

As known, the important thing in stratified charge combustion is the stability of the flame.

Misfire will occur if the mixture and condition are not suitable to ignite. Overall lean mixture have an opportunity to deteriorate the combustion stability. Due to differences in fuel properties, lean limit of each fuel should be investigated.

The statistic data analysis was chosen in this experiment to investigate the proper lean limit or minimum equivalence ratio that mixture can be ignited at certain pressure and temperature. From pressure data, 10 data from each condition were collected for each fuel. If the misfire occurs, it will increase the coefficient of variation (CoV). Therefore, if it exceeds the acceptable value, which was set to 20% of CoV in the current study [8], it implies that mixture does not ignite. In this experimental analysis, minimum equivalence ratio at the maximum acceptable CoV is assumed as the lean limit.

2.2.3 ROHR and Mass fraction burn calculation

In this section, rate of heat release (ROHR) can be calculated by applying the first law of thermodynamics regardless of the heat transfer to the chamber wall. Then, ROHR will be obtained from the following equations.

$$\left(\frac{dQ}{dt}\right)_{\text{release}} = mc_v \frac{dT}{dt} + P \frac{dV}{dt} \quad (3)$$

Apply the ideal gas law, equation (3) becomes [8],[12]

$$\left(\frac{dQ}{dt}\right)_{\text{release}} = \left(\frac{c_v P}{R} \frac{dV}{dt}\right) + \left(\frac{c_v V}{R} \frac{dP}{dt}\right) + P \frac{dV}{dt} \quad (4)$$

with a relation between specific heat ,

$$\frac{c_p}{c_v} = k \quad \text{and} \quad c_p = c_v + R$$

Since the experiment was conducted in constant volume combustion chamber, the following equation is obtained

$$\left(\frac{dQ}{dt}\right)_{\text{release}} = \left(\frac{V}{k-1}\right) \frac{dP}{dt} \quad (5)$$

Where V is the volume of combustion chamber and k as the specific heat ratio.

As for a qualitative analysis, by neglecting the heat transfer during combustion, the deviation for turbulent combustion by this treatment is small due to the short combustion duration [14]. Thus, the rate of pressure rise is proportional to the rate of heat release; and the information of heat release rate can be reflected by the information of pressure rise rate. Hence, heat release rate can be calculated directly with the pressure history data [4-5], [8], [11]

Mass burned fraction was calculated based on the pressure data obtained from the experimental results as follow [5], [13].

$$M_f(t) = \frac{P(t) - P_i}{P_{\text{max}} - P_i} \quad (6)$$

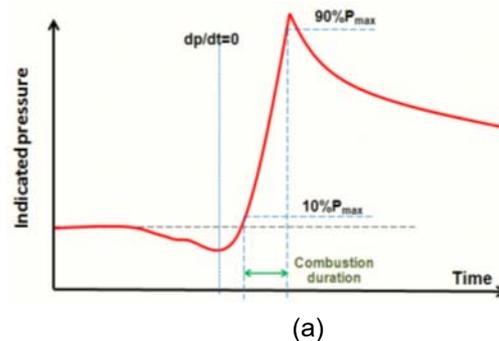
where $P(t)$ is the combustion pressure

P_i is the initial mixture pressure, and

P_{max} is the maximum combustion pressure.

If the mixture gas is fully burned when it reaches peak combustion pressure, it can be calculated according to the relative value of the combustion pressure, where M10 stands for 10 percent combustion and M100 stands for 100 percent combustion[5].

Fig. 5(a) shows the definition of combustion duration, which is expressed from 10% mass fraction burn to 90% mass fraction burn [8].



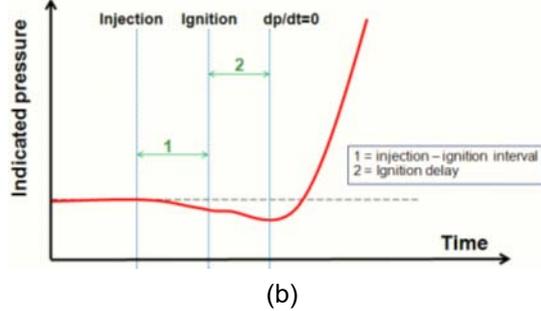


Fig. 5 Schematic behavior of indicated pressure showing (a) combustion duration and (b) ignition delay/timing

The trace of indicated pressure P is schematically drawn in Fig. 5(b), which exaggeratedly expresses the pressure drop caused by latent heat absorption by vaporized fuel just after fuel injection start [4].

Spark plug is discharged at the aimed injection-sparking interval time from injection start; and then ignition occurs at further later time by ignition delay. Ignition timing determined by combustion observation coincides with the timing when dP/dt approaches zero as shown in Fig. 5(b) [4].

3. Result and discussion

3.1 Effect of ethanol content and swirl intensity on the lean limit

Fig. 6 shows the lean limit of each tested fuel in various swirl intensities. Pure ethanol (E100) has minimum lean limit in all swirl intensities compared with other fuels. On the other hand, pure gasoline or E0 shows the highest lean limit in every swirl condition. In the low swirl flow (2600 rpm), Gasoline or E0 has the highest value of ϕ equal to 0.38; whereas, those of E20, E85 and E100 are 0.365, 0.334 and 0.310, respectively. In case of higher swirl intensities, each fuel tends to ignite in the lower equivalence ratio. At the highest swirl flow (8800 rpm), pure ethanol (E100) can be ignited at the

lowest equivalence ratio of 0.300, which corresponds to an air/fuel ratio of 30.0; while E20, E85 and E0 can be ignited at 0.34, 0.313 and 0.364, respectively.

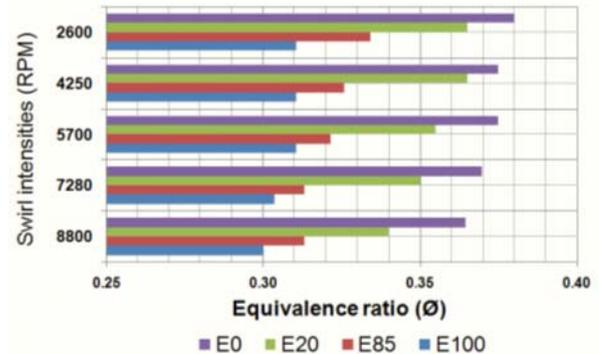


Fig. 6 Lean limit in different swirl intensities

At certain swirl value, medium swirl (5700 rpm), pure gasoline (E0) can be combusted in the highest equivalence ratio compared with each other. As the ethanol content increases, the mixtures will combust in lower equivalence ratio of 0.375. Pure ethanol reaches lowest equivalence ratio of 0.3105, which means that ethanol content can support the combustion stability in the lean condition. In other words, with a higher swirl intensity, the minimum equivalence ratio at lean condition will become less compared with the lower swirl flow. In case of E0, equivalence ratio at lean limit were 0.380, 0.375, 0.374, 0.369 and 0.364 at swirl intensity equal to 2600, 4250, 5700, 7280 rpm and the highest swirl flow was 8800 rpm, respectively. In higher ethanol content, the same trend as the pure gasoline (E0) was obtained but the result was less than expected. For pure ethanol (E100), equivalence ratio at lean limit were 0.310, 0.310, 0.310, 0.304 and 0.300 at swirl intensity equal to 2600, 4250, 5700, 7280 and the highest swirl flow 8800 rpm, respectively. The result shows little change in minimum equivalence ratio since higher percentage of the ethanol has less effect to

enhance lean limit range than the lower ethanol content. From these results, we can discuss in two aspects.

First, for stratified combustion, It was just some amount of mixture, which can be ignited, especially that of stoichiometric mixture. In higher swirl intensity, stratification degree [5,7] which is characterized by a ratio of stoichiometric mixture over the overall lean mixture became higher, Therefore, rich and lean limit of the mixture become wider [7].

According to results of M.Fujimoto[16], Mixture distribution were observed by PLIF method. It was found that the overall vapor fluorescence intensity increase with the increase of swirl rate. The fuel atomization and that the fuel evaporation was accelerated by the increase of the turbulence intensity and the decrease of the impinged fuel, both of which were caused by the swirling flow. Under the high swirling condition, the fuel vapor was distributed around the cylinder wall in a doughnut-like shape because the fuel vapor was rotated along the cylinder wall by the swirling flow. In the other hand, Under low swirl flow condition , liquid fuel was diffused and impinged to the chamber wall. This fuel droplets formed liquid film on the cylinder wall surface and was not completely vaporized. So, less stoichiometric mixture can form in low swirl condition. In other words, the swirl intensity can enhance stratification degrees that directly affect the lean limit range. Second, considering only one swirl intensity, the ethanol can be ignited and combusted at the lower equivalence ratio because an air/fuel ratio of gasoline and ethanol/gasoline blends are different. Thus, if ethanol/gasoline blends are

subjected to the same equivalence ratio as in the pure gasoline, additional amount of the fuel is needed for the same equivalence ratio. For example, at equivalence ratio equal to 1.0, the stoichiometric air/fuel ratio of each fuel in Table 2. shows that $AFR_{E0}=14.6$, $AFR_{E20}=13.5$, $AFR_{E85}=9.87$ and $AFR_{E100}=9.0$. If we need to test ethanol/gasoline blends at the same equivalence ratio as pure gasoline (E0), we should add 8.24% , 48.15% and 62.01% more fuel to E20, E85 and E100 respectively. Consequently, the mixture of ethanol/gasoline in constant volume chamber will have more chance to form mixture near stoichiometric ratio than that the pure gasoline (E0).

3.2 Effect of ethanol content and swirl intensity on the ignition delay

In Fig. 7, pure gasoline (E0) has the longest ignition delay in all swirl intensities and ignition delay decreases with increasing ethanol content. At swirl intensity equal to 2600 rpm (or $\Delta P=0.1$ MPa), ignition delay time of E0 is 4.24 msec, which becomes 3.45, 3.28 and 2.52 msec in case of E20, E85 and E100, respectively.

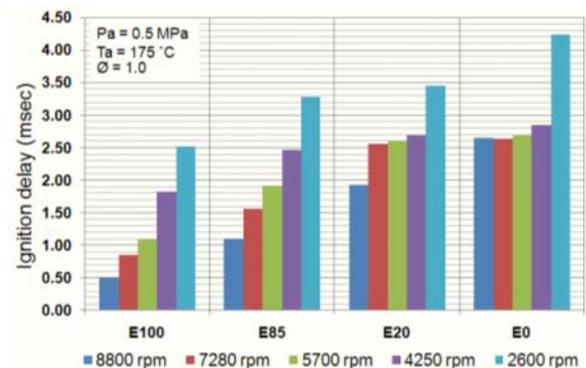


Fig. 7 Effect of ethanol content and swirl intensities on ignition delay

Fig. 8 shows the spray cone angles of various fuels. As can be seen, the spray angles of the alcohol fuel have a larger cone angle than the pure gasoline.

These phenomena demonstrate that higher ethanol content can vaporize easier than lower ethanol content. From Table 2, although ethanol has higher octane number and has low rate of vaporization, ethanol is comprised of only one component, while gasoline is a mixture of many components with various carbon atoms and structures. The components with relatively low boiling points evaporate easier and faster, while the components with higher boiling points will evaporate more slowly. Thus, gasoline has the longest ignition delay regardless of the higher vapor pressure of gasoline [9, 10]

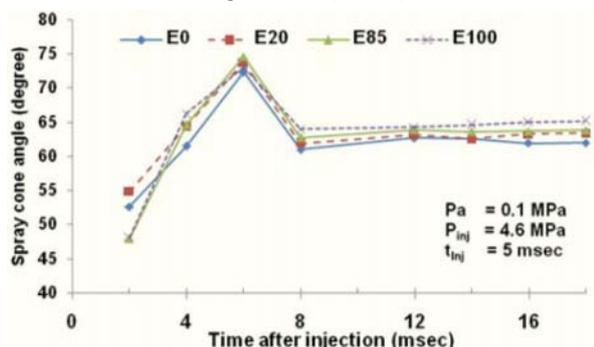


Fig. 8 Spray cone angle

As shown in the Fig. 7, ignition delay time of the ethanol (E100) will be decreased in the higher swirl flow. The result shows that the ignition delay time of E100 are 2.52 msec at 2800 rpm, 1.82 msec at 4250 rpm, 1.08 msec at 5700 rpm, 0.85 msec at 7280 rpm and reaching the shortest of 0.50 msec at 8800 rpm. These results demonstrate that swirl intensity can affect the ignition delay by diffusing the injected fuel to the wider area. Thus, the mixture near the stoichiometric ratio which can easily be combusted will be greater than the case of low swirl flow. In case of low swirl flow, the amount of fuel injected will form the very rich mixture near the center of combustion chamber. Therefore, the initial combustion would yield a long ignition delay

[7]. In the lower ethanol content such as E85, E20 also the E0, the variation in ignition delay time between each swirl intensity step will decrease as the ethanol content decrease except low swirl flow (2200 rpm) case. These may be discussed that ethanol, comprised of only one component, has lower carbon atom than the gasoline or E20 and E85. Consequently, it may be diffused by swirl flow easier than the gasoline, E20 or E85 which are comprised of various higher carbon atoms [9]. As ethanol can easily diffuse, it has more chance to form the stoichiometric mixture in broader area.

3.3 Effect of ethanol content on the rate of pressure rise and mass fraction burn

Figures 9-11 show the effect of the ethanol content on the combustion pressure at 8800 rpm swirl intensity ($\Delta P = 0.3$ MPa). As indicated in Fig. 9, the maximum pressure appears in the pure ethanol (E100) at 3.05 MPa. The peak value of combustion pressure decreases with decreasing ethanol content in the mixture due to lack of oxygen concentration of fuel. Also, the length of time required to reach the maximum value of the combustion pressure is retarded in accordance with the decrease of ethanol percentage.

The rate of pressure rise is a very important parameter because this has a very significant influence on heat release rate as described in previous section. The effects of ethanol content on the rate of pressure rise are illustrated in Fig. 10. The peak value of the pressure rise rate was rapidly increased and its timing was shorten at high percentage of ethanol. E20 has not significantly changed in maximum pressure rise rate compared to pure gasoline

(E0). However, it can release heat earlier than that of the E0. This demonstrates that ethanol content may accelerate the combustion period and reduce time of heat loss in early stage of the combustion.

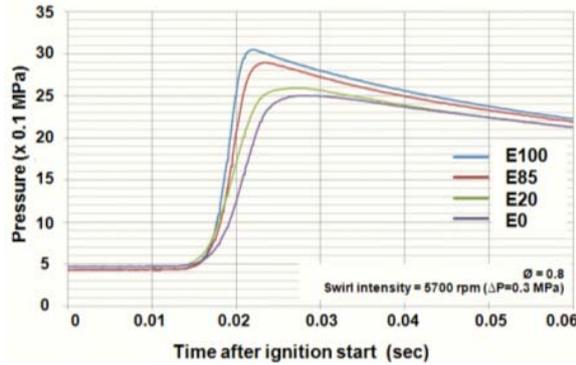


Fig. 9 Pressure data of E0, E20, E85 and E100 at equivalence ratio = 0.8

In premixed laminar flame, experimental results of C. Charoenphonphanich [13] and T.Hara [15] were shown that the laminar burning velocities in high concentration of the ethanol fuel were higher than that the pure gasoline (E0). This effect can be explained by the oxygen concentration in the fuel and phenomena of the Lewis number [13, 15].

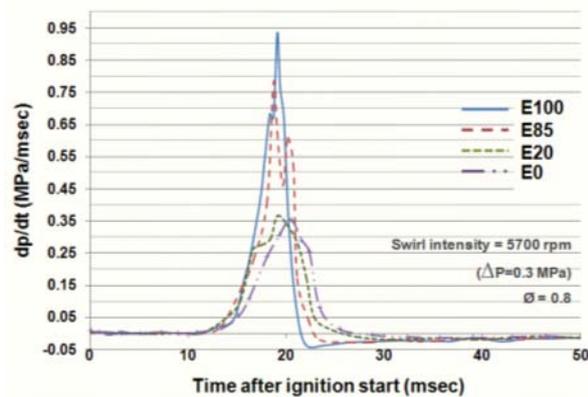
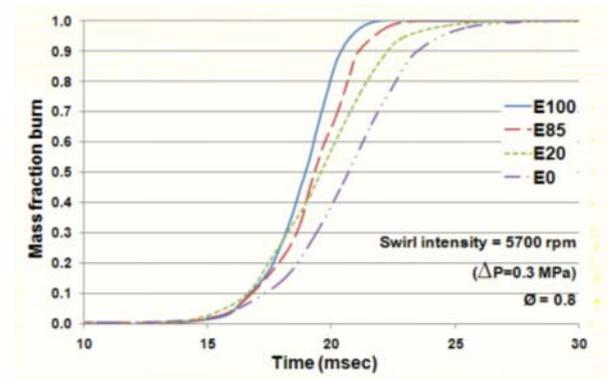


Fig. 10 Rate of pressure rise of E0, E20, E85 and E100

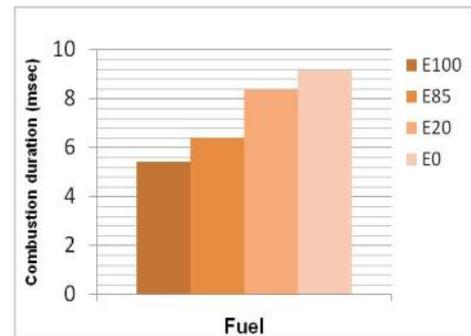
Thermodynamic analysis of measured cylinder pressure data is a very powerful tool used for quantifying combustion parameters [11]. The important one is often referred to as "burn-rate analysis". Burn-rate analysis is used mainly

to obtain the mass fraction burned, which is a normalized quantity with a scale of 0 to 1.

Fig. 11 shows the effect of ethanol content on mass fraction burned at the equivalence ratio of 0.8 and 8800 rpm swirl flow. With an increase of ethanol blend, the total combustion duration decreases, as shown in the comparison of mass fraction burned. Fig. 11(b), combustion duration of E100 was 5.44 msec; this value define as 10% to 90% of trace pressure data while E85, E20 and E0 were 6.41 msec, 8.39 msec and 9.17 msec, respectively



(a)



(b)

Fig. 11 Effect of ethanol content on (a) mass fraction burn and (b) combustion duration

Fig. 12 shows the influence of equivalence ratio and ethanol concentration on combustion pressure. As the equivalence ratio of mixture decrease, peak pressure decrease while combustion duration increase. These results were displayed the same trend in every tested-fuel. However, Low alcohol fuel-blended,

especially in E20 and E0, the combustion pressure and combustion duration were dramatically changed at low equivalence ratio.

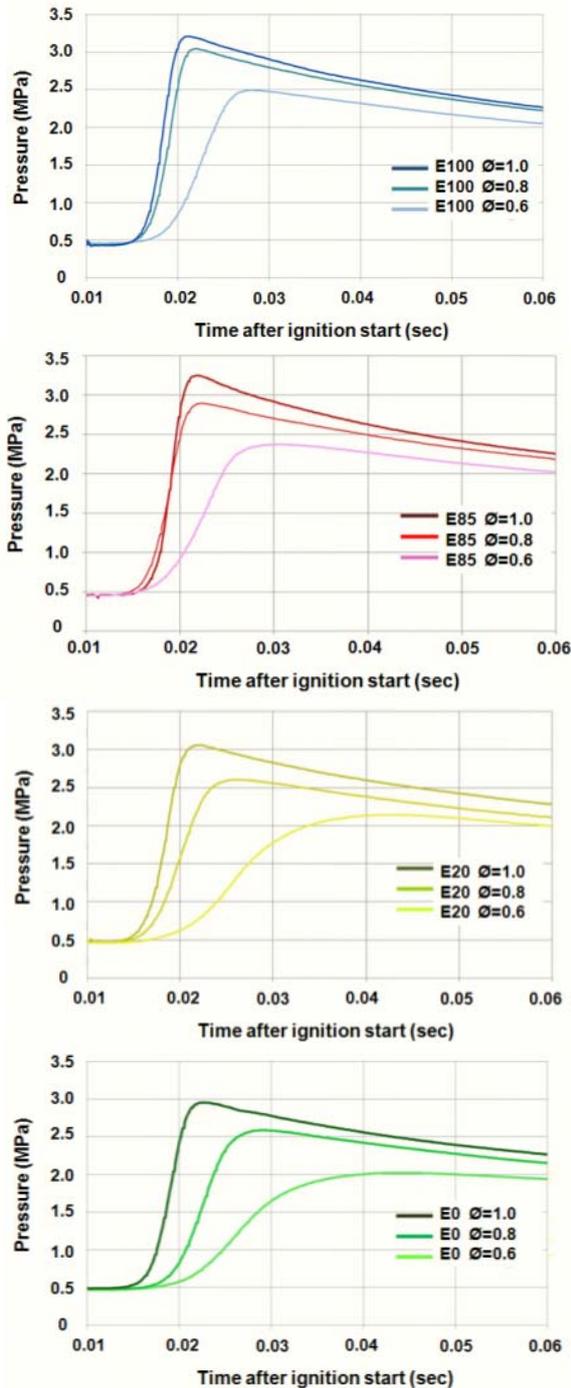


Fig. 12 Effect of equivalence ratio on combustion pressure (at swirl = 8800 rpm)

From the result in Fig. 11 and Fig. 12, it can be inferred that the decrease in ethanol fraction and equivalence ratios plays an important

role in the decrease of maximum pressure in the chamber.

4. Conclusion

1. The swirl intensity can enhance stratification degree that directly affect the lean limit range by diffusing amount of fuel to form stoichiometric mixture in the wider area.
2. Higher ethanol percentage in fuel will lead to extend the lean limit range because additional fuel injected may increase the chance of fuel to form near stoichiometric
3. Although ethanol has higher octane number and has low rate of vaporization, ethanol is comprised of only one component, while gasoline or ethanol/gasoline blends are the mixture of many components with various carbon atoms and structures. The components with relatively low boiling point evaporate easier and faster, and vice versa. Thus, pure ethanol tends to be evaporated faster than other blends.
4. In case of low swirl flow, the amount of fuel injected did not diffuse very much. Thus, the fuel and air charge will form the very rich mixture near the center of combustion chamber. Consequently, the initial combustion and the ignition delay became long.
5. The decrease in ethanol fraction and equivalence ratio plays an important role in the decrease of maximum pressure in the chamber. The combustion duration of pure ethanol (E100) is less than that of E85, E20 and E0 respectively. On the other hand, peak pressure of pure ethanol (E100) is higher than that of E85, E20 and E0 respectively.
6. Combination of both high swirl flow and high ethanol content can enhance stability of the combustion in overall lean mixture.



Further studies include advanced method to identify the consistency of the swirl intensities and mixture formation distribution, such as LIF method and flow field simulation; as well as to optimize the injection-ignition timing for the proper equivalence ratios that mixture could form near the spark plug.

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