COMBUSTION CHARACTERISTICS OF HYDROTREATED VEGETABLE OIL (HVO)-DIESEL BLENDED FUELS UNDER LOW AMBIENT OXYGEN CONCENTRATION AND DIFFERENT AMBIENT TEMPERATURE

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ABSTRACT

Hydrotreated vegetable oil (HVO), a modern alternative fuels. HVO is a second generation biofuel that produce from vegetable oil by using hydrotreating process. HVO is a mixture of normal paraffin and isomerized paraffin, which has superior advantages such as low sulfur and aromatics, high cetane number, high heating value and narrow distillation temperature range. From this reason, it offers significant improvement in performance and exhaust gas emissions of diesel engines. The objective of this research is to investigate effects diesel and HVO blending percentage on characteristics of ignition and spray combustion in constant volume combustion chamber (CVCC) under low ambient oxygen concentration and different ambient temperature using heat release analysis and to visualize combustion process of diesel and HVO blends in order to describe effects of fuel properties of HVO on spray development and flame development using the shadowgraph technique. The experiment carried out using CVCC, four tested fuels were tested: commercial diesel, two commercial diesel-HVO blends by mass: 20% (H20), and 50% (H50) and 100% HVO with single-hole injector, 0.2 mm in nozzle orifice diameter, 2.5 ms in energizing time. The injection pressure was kept constant at 100 MPa. The ambient oxygen concentrations were varied between three discrete values from 21%, 15% and 10% to simulate effects of EGR conditions. The ambient temperatures were varied at 1100 K, 900 K and 700 K to study effects of ambient temperatures. The results showed that at low oxygen concentration 15% and 10%, the rate of combustion pressure decreased 9.08% and 29.58%. At low ambient temperature 900 K and 700

K, the rate of combustion pressure increases 6.77% and 6.46% because of longer ignition delay promotes available time for mixing to obtain the better mixture formation. A decrease of oxygen concentration results in an insignificant effects on spray development due to the same ambient air density and temperature, but the ignition delay is changed with oxygen concentration as seen as the luminous flame. At the 10% oxygen concentration, the flame luminosity is not observed due to the lack of oxygen, which results in a slow chemical reaction. The HVO is easier to vaporize and distribute in the chamber due to its lower density, viscosity and distillation temperature at T90. A decrease of ambient temperature effects on fuel-air mixing by providing a longer mixture formation time, resulting in later luminous flame with higher peak heat release rate. At 700 K of ambient temperature, the flame luminosity of only H50, HVO is observed because of its higher cetane number provides high reactivity at low temperature. As the results, The HVO can be recommended to use in the diesel engine with EGR application as well, but it is necessary to optimize the engine when using in low temperature combustion such as reducing the compression ratio.

Keywords: Hydrotreated vegetable oil (HVO), Constant volume combustion chamber (CVCC), Heat release rate, Shadowgraph technique, Oxygen concentration, Ambient temperature.

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CHAPTER 1
INTRODUCTION

1.1 Research background

Nowadays, the renewable energy technologies are built, developed and distributed worldwide because of the public climate change and global warming. Many countries try to make abundant contribution of clean energy into building a clean environment and protecting public health. As a result, many researchers have been developed a new clean energy for the vehicles, for instance, fuel cell, battery electric, and also plug-in electric. From this reason, in order to replace conventional vehicles by those energy technologies, it is very difficult because of the production cost, well to wheel. However, the internal combustion engines and hybrid engines still play a role of light-duty vehicles at least 30 years [1].

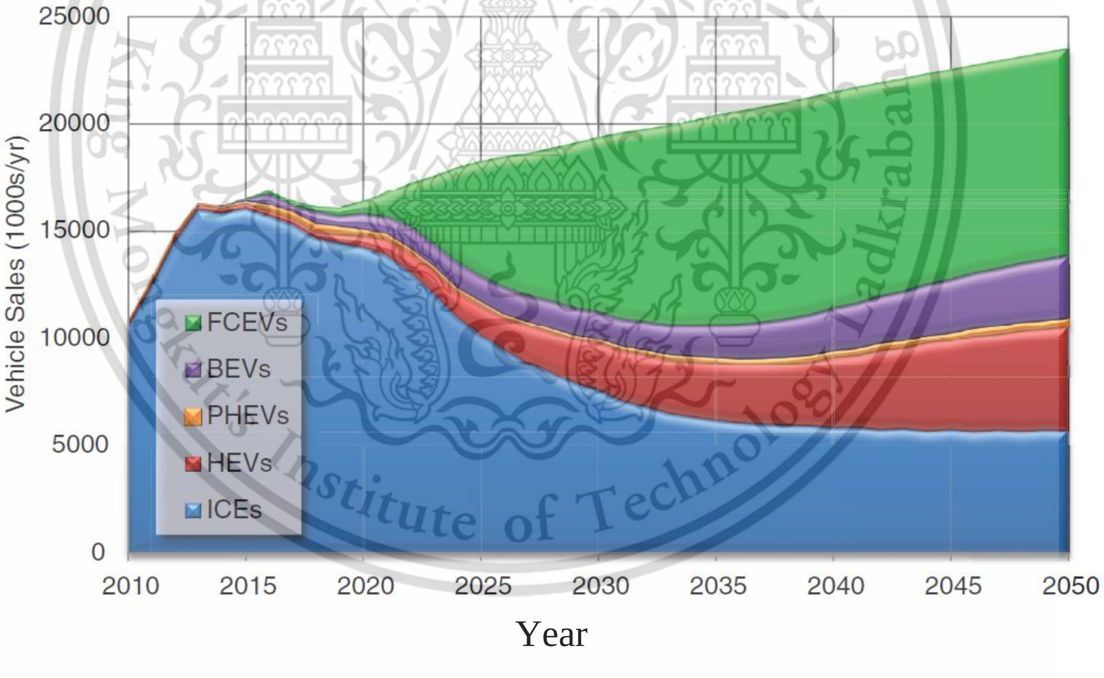


Figure 1.1 Vehicle sales by vehicle technology for midrange technologies and policies [1]

Diesel engines provided higher thermal efficiency than spark ignition engines. However, diesel engines still have serious pollutant problems. Pollution is one of the many environmental challenges facing the world today [2]. Many researchers have been employed various techniques for reducing in exhaust gas emissions such as after-treatment systems (i.e. DOC, SCR, DPF, etc.) [3], engine combustion control (i.e. common-rail, swirl mixing, EGR, etc.) [4]–[6]. On the other hand, researchers have been researched, invented and introduced alternative fuels to use in the diesel engine. An alternative fuels have not only sustainable but also friendly, respect with the environment. Therefore, a sustainable energy has been researched for applying into diesel engines such as biodiesel (i.e. FAME, SME, RME, etc.) and Hydrotreated Vegetable Oil (HVO).

Biodiesel is a first generation biofuel that produce from vegetable oil by using transesterification process [7]. Biodiesel has been widely utilized and used, as it can be directly used or blending used with diesel in the engines without no modification. However, it still has some disadvantages to the engines like high density and viscosity that make larger droplet size distribution, poor fuel-air mixing processes and mixture formation in the combustion process [8], [9], with low heating value [10], which resulted a decrease in engine power and increase fuel consumption due to loss of heating value [11]. In addition, the use of biodiesel and blends can reduce exhaust gas emissions such as Hydrocarbon (HC), Carbon monoxide (CO) due to its higher cetane number and oxygen content, and Particulate matter (PM) due to lower aromatics and sulfur content [11]. While Nitrogen Oxides (NO_x) emissions of biodiesel showed increase due to higher oxygen content that making a higher combustion temperature and flame temperature [7], [11].

From these reason, Hydrotreated vegetable oil (HVO) is attended and contributed into diesel engines from many researchers. HVO is the second generation biofuel, which can produce from many kinds of vegetable oil by using hydrotreating process. Due to HVO has potential benefits so that HVO can be promised alternative fuels as one of a candidate to replace diesel due to its more advantages in comparison with other alternative diesel fuels. The first one shows about production. It has variety of bio-feedstock, better oxidation stability. The second one is better fuel properties in the term of physical and chemical properties, for instance, low density, viscosity, low sulfur and aromatics, high cetane number, high heating value, and similar chemical composition to diesel [12], [13]. From this reason, it offers significant improvement in engine performance and exhaust gas emissions and fuel consumption. However, HVO still have limitation to use in diesel engines as it shows very high cetane number and low lubricity, so researchers suggested that HVO would not be blended over 50% [13].

However, in order to utilize HVO into diesel engines, it is necessary to understand effects of fuel properties on mixture formation, ignition and combustion as illustrated in Figure 1.2. From this figure shows the sub-processes in diesel engine mixture formation and combustion. The particular sub-processes proceed largely simultaneously and interact with each other. Generally, the conventional diesel engine combustion process is characterized by heterogeneous mixture formation and combustion. In direct-injection diesel engines, the air is induced into the combustion chamber and then compressed under compression stroke to generate high pressure and temperature, fuel is directly injected under high pressure and temperature in the combustion chamber. The liquid fuel entering the combustion chamber is atomized into small droplets, evaporated and is mixed with hot air, resulting in a heterogeneous mixture of fuel and air. Combustion is initiated by sufficient the high temperatures and pressures by an auto-ignition process. [14].

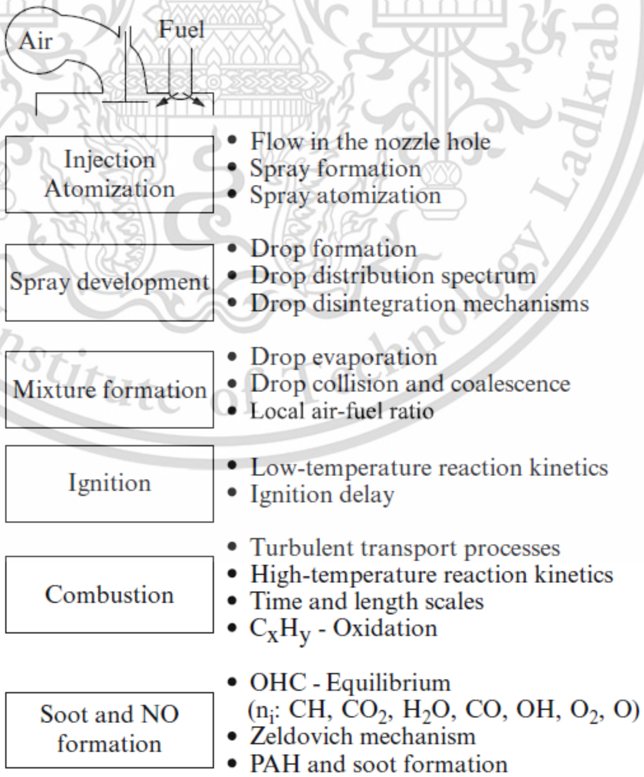


Figure 1.2 Sub-processes of mixture formation and combustion in diesel engines [14]

The employment of exhaust gas recirculation (EGR) is able to reduce NO_x emission. EGR causes a lower combustion temperature and keeps a lower flame temperature [15]–[17]. On the other hand, EGR decreases heat release due to decreased combustion pressure and combustion reaction intensity [18].

In a moment, the low temperature combustion (LTC) strategies have been widely influenced to engine performance and emission characteristics of diesel engines because LTC is effective method to reduce flame temperature and combustion pressure that related to NO_x and soot emissions [18]–[20]. LTC is very important field research to meet future regulated emissions of modern diesel engines by applying large amount of EGR ratio, low compression ratio as low ambient temperature. From previous research showed that LTC has potentially simultaneous reduction of NO_x and soot emissions under various low temperature combustion conditions [20]–[22].

Several research reported that the employment of HVO can improve engine performance and exhaust gas emissions of diesel engines. HVO has better in production and fuel properties compared to diesel. However, few researches have been performed the influence of diesel and HVO blend percentage on characteristics of the ignition delay and spray combustion by using heat release analysis under low ambient oxygen concentration and different ambient temperature. Especially, the discussion on effects of diesel and HVO blend percentage on the combustion visualization by using the shadowgraph technique, for instance, spray development and flame development have not clearly explained yet.

1.2 Research Objectives

1.2.1 To investigate effects diesel and HVO blending percentage on characteristics of ignition and spray combustion in constant volume combustion chamber under low ambient oxygen concentration and different ambient temperature.

1.2.2 To visualize combustion process of diesel and HVO blends in order to describe effects of fuel properties of HVO on spray development and flame development.

1.3 Scope of work

In this experiment investigated spray combustion characteristics under low ambient oxygen concentration and different ambient temperature in constant volume combustion chamber (CVCC) using the heat release analysis and the shadowgraph technique. The heat release analysis was applied to study the characteristics of ignition delay and combustion such as combustion pressure and temperature, heat release rate, cumulative heat release, ignition delay. The shadowgraph technique was used to visualize spray combustion process as spray penetration, flame development and to capture combustion process by using the high speed video camera. The oxygen concentrations were varied between three discrete values from 21%, 15% and 10% to simulate the effects of EGR conditions in the combustion chamber. The ambient temperatures were varied from 1100 K, 900 K and 700 K to simulate the effects of ambient temperatures.

CHAPTER 2
RESEARCH THEORY AND LITERATURE REVIEW

2.1 Conventional diesel engine

The conventional diesel engine combustion process is characterized by heterogeneous mixture formation and combustion. In direct-injection diesel engines, the air is induced into the combustion chamber and then is compressed under compression stroke to generate high pressure and temperature, fuel is directly injected under high pressure and temperature in the combustion chamber. The liquid fuel entering the combustion chamber is atomized into small droplets, evaporated and mixed with hot air, resulting in a heterogeneous mixture of fuel and air. Combustion is initiated by sufficient the high temperatures and pressures by an auto-ignition process. Therefore, the physical and chemical properties of fuel are dominated factors for the ignition process of diesel combustion [23].

2.1.1 Diesel engine operation

There are four principle state of the diesel four-stroke engine operations; firstly, the air is induced into combustion chamber during the intake stroke, then is compressed under isentropic process to a higher pressure and temperature during the compression stroke. The injector inject the liquid fuel, fuel evaporated and mixed with hot compressed air, and burnt rapidly due to the auto-ignition process during the power stroke. The exhaust gas is push out by piston movement during the exhaust stroke.

The four-stroke diesel engine consists of, the intake stroke, the compression stroke, the power stroke, and the exhaust stroke as illustrated in Figure 2.1 [23].

1. Intake stroke: The piston move down from top dead center (TDC) to bottom dead canter (BDC) during intake valve open and exhaust valve close. The fresh air is induced into cylinder due to the difference in the pressure.

2. Compression stroke: The piston move up from BDC. The air is compressed, the air temperature and pressure increase during intake valve and exhaust valve are close.

3. Power stroke: Piston moves down from TDC due to the high combustion reaction of fuel injected. The liquid fuel is injected continuously into combustion chamber during power stroke. The in-

cylinder pressure increase from combustion that converting into mechanical energy through the piston to the crankshaft.

4. Exhaust stroke: Piston move up from BDC to TDC during intake valve close and exhaust valve open. The exhaust gas is flow out of combustion chamber due to the different pressure between in-cylinder and ambient.

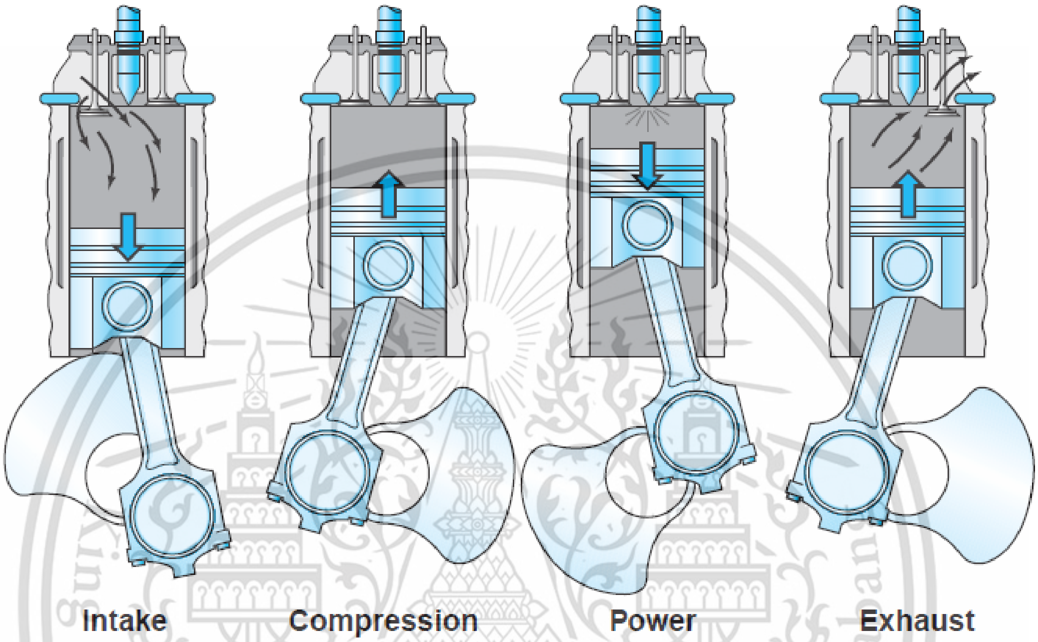


Figure 2.1 Four-Stroke diesel engines operation [23]

2.1.2 Diesel engine characteristics

Diesel engines provide higher thermal efficiency compared to other internal combustion engines. However, in order to develop the new fuel technology compatible for diesel engine, it is essential to understand the features of common diesel engines. Those features are roughly classified into six parts as described in Figure 2.2 [24].

Figure 2.2 shows the relationship of diesel engine characteristics with each other. It is show that injection and fuel spray characteristics relate to combustion, engine performance, ecology and economy. All these characteristics depend on basic parameters such as fuel type or injection system type and on various process characteristics such as the injection process, fuel spray development, atomization, mixture formation, ignition and combustion. The diesel engine characteristics may be influenced by various geometrical and setup parameters either mechanically or electronically controlled fuel injection system.

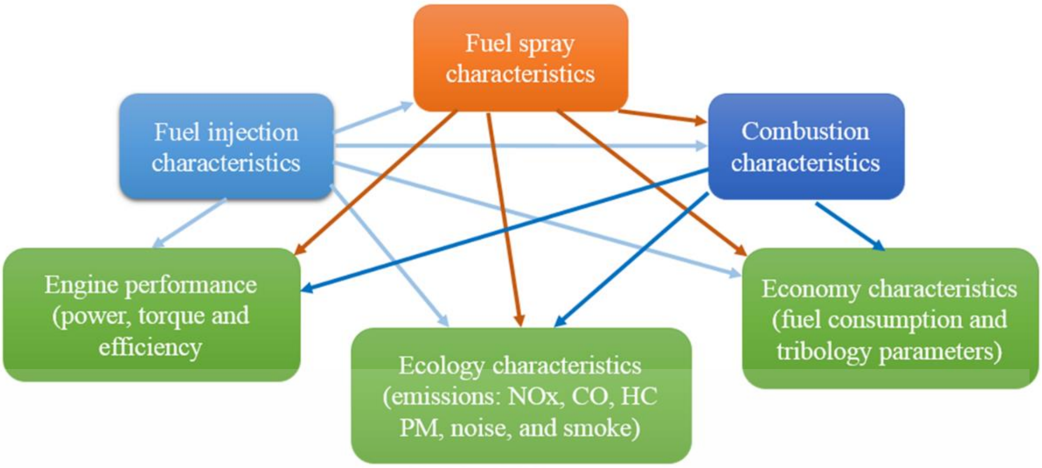


Figure 2.2 Relationship of diesel engine characteristics [24]

2.2 Combustion of diesel direct-injection

2.2.1 Diesel combustion process

In diesel engine, the liquid fuel is injected directly into combustion chamber before TDC. However the fuel will not immediately combustion. Mixing time of fuel and air are required to be a mixture formation prior the ignition and combustion process as illustrated in Figure 2.3.

Ignition delay (a to b) is defined as the interval time between the start of injection (SOI) to start of combustion (SOC) where the heat release rates recovers from negative value due to heat absorption of fuel.

Premixed combustion (b to c) is defined as a rapid heat release rate. In this phase, combustion of fuel has already mixed with air during the ignition delay period occurs rapidly in a short time.

Mixing controlled combustion (c to d). When fuel and air are mixed during the ignition delay have been consumed, the heat release rate is controlled by the rate at which mixture becomes available for burning. While several processes are involved liquid fuel atomization, vaporization, mixture formation, and chemical reaction the rate of burning is controlled in this phase primarily by the mixture of fuel and air. The heat release rate may or may not reach a second peak in this phase; it decrease as this phase progresses.

Late combustion (d to e). Heat release continues at lower rate. There are several reasons for this phenomena. A small fraction of the fuel may not yet have burned. A fraction of the fuel energy present in soot and fuel rich combustion product. The cylinder charge mixing in this period promotes more complete combustion and less dissociated gases. The kinetics of the final burnout process become slower as the temperature of the cylinder gases fall.

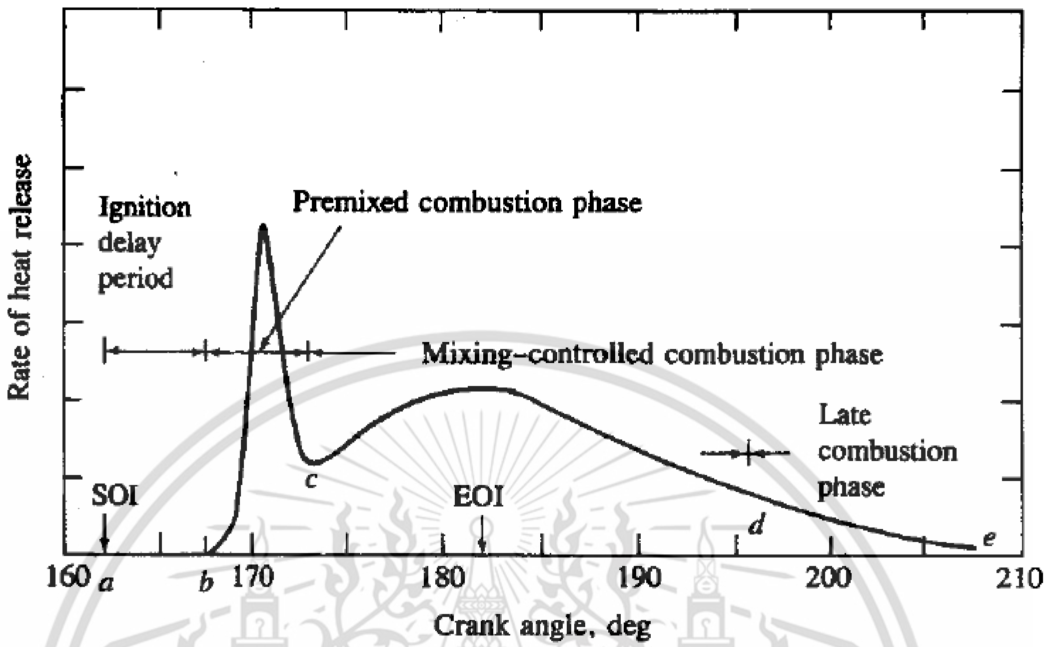


Figure 2.3 Combustion process of diesel direct-injection [25]

2.3 Simulation of diesel engine combustion condition

In order to observe the combustion phenomena and to access diesel spray flame inside the combustion chamber of diesel engine is very difficult because combustion of diesel engine is very complicated. However, researcher has used many approaches and experimental equipment to access the combustion, for instance, the optical research engine (ORE), the rapid compression expansion machine (RCEM), constant pressure flow rigs (CPFR), constant volume hot cells (CVHC) and constant volume pre-combustion cell (CVPC) or constant volume combustion chamber (CVCC). As a result, the operation rage of CVCC can cover full range of diesel combustion condition. The CVCC has been widely used to study diesel spray combustion [26]–[28]. The CVCC can simulated a wide range of ambient gas pressure and temperature prior fuel injection. The simulated ambient gas pressure and temperature of CVCC in this research are linked to the pressure and temperature of diesel engine during the end of compression stroke. The operation of the experimental equipment were summarized by Baert *et al.* [29] as shown in Figure 2.4.

Table 2.1 shows the comparison of different experimental equipment. It can concluded that even though the ORE shows that most similarity to the real diesel engine situation, the CVCC is more suited and contributed for studying basic research of free spray combustion because it can give a wide operating range and large volume for diesel spray.

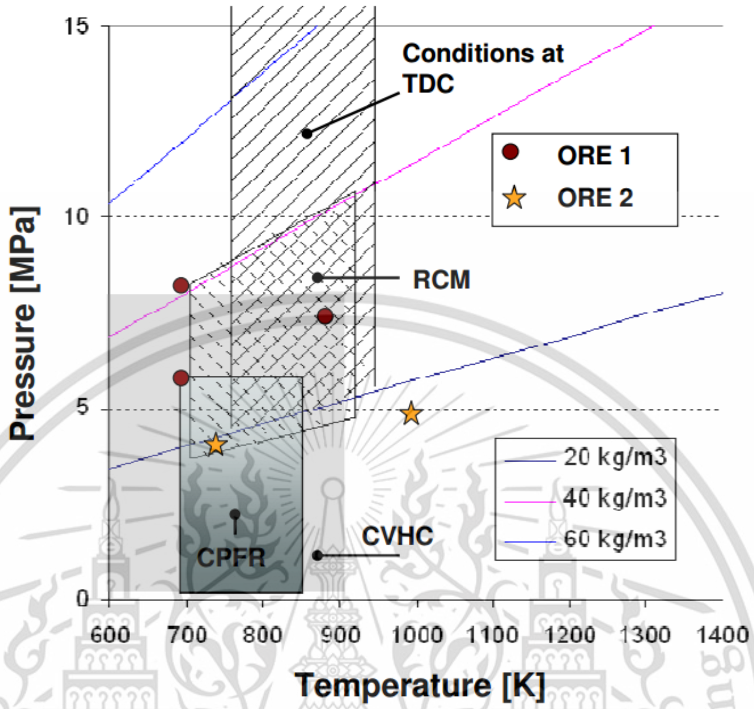


Figure 2.4 In-cylinder conditions prior injection of diesel engine compared to the operating range of different experimental equipment [29]

Table 2.1 The comparison of different experimental equipment; ORE, RCEM, CVFR and CVPC or CVCC [29].

Type of optical test rig	ORE	RCM	RCYM	CPFR	CVHC	CVPC
Optical accessibility	0	0	++	++	-	+
Similarity to the real engine situation	0	-	--	--	--	--
Free spray penetration distance	0	+	+	+++	++	++
Control on trapped gas p / T	0	+	0	++	++	++
Control on trapped gas composition (i.e. EGR)	0	-	0	+	++	+++
Flow field impact on combustion	--	-(-)	-	0	-	-
Test facility volume	0	+	0	0	++	++
Time to switch between operating conditions (i.e. T)	0	0	0	0	--	++
Time between tests [s] (*)	1	120-600	1	1-3	60	600

Note: mostly relative; 0 = neutral, + = better, - = worse

2.4 Emissions of diesel engine

Diesel engines convert the chemical energy contained in the fuel into mechanical power. Diesel fuel is injected under high pressure into the combustion chamber where it mixes with air and where the combustion occurs. The exhaust gases which are discharged from the engine contain several constituents that are harmful to human health and to the environment. By the emission of diesel engine consist of CO, HC, NO_x, SO_2 and particulate matter (PM) as shown on Equation 2.1 and Figure 2.5 show the conceptual schematic of conventional diesel combustion [30].

Carbon monoxide (CO), hydrocarbons (HC), and aldehydes are generated in the exhaust as the result of incomplete combustion of fuel. A significant portion of exhaust hydrocarbons is also derived from the engine lube oil.

Nitrogen oxides (NO_x) are generated from nitrogen and oxygen under the high pressure and temperature conditions in the engine cylinder. Nitrogen dioxide is very toxic. NO_x emissions are also a serious environmental concern because of their role in the smog formation.

Sulfur dioxide (SO_2) is generated from the sulfur present in diesel fuel. The concentration of SO_2 in the exhaust gas depends on the sulfur content of the fuel. Low sulfur fuels of less than 0.05% sulfur are being introduced for most diesel engine applications. Sulfur dioxide is a colorless toxic gas with a characteristic, irritating odor. Sulfur oxides have a profound impact on environment being the major cause of acid rains.

Particulate matter (PM) is a complex aggregate of solid and liquid material. Its origin is carbonaceous particles generated in the engine cylinder during combustion.

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

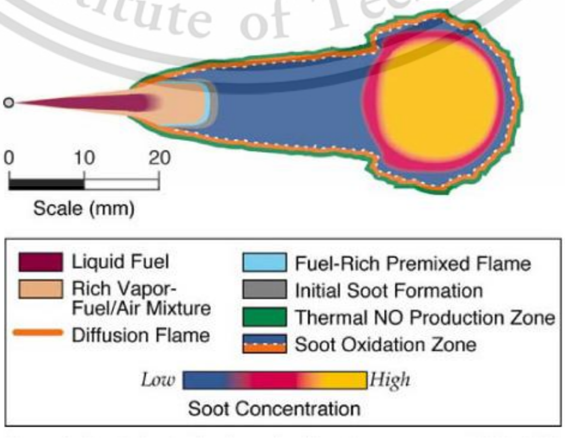


Figure 2.5 Conceptual schematic of conventional diesel combustion [30]

2.5 Fuel injection system

2.5.1 Common rail system

The high injection pressure system or called "common rail system" as shown in Figure 2.6 [31] has been developed for the modern diesel engines in order improve fuel atomization, mixture formation, combustion efficiency, and also engine-out emissions of diesel engines. There are nine main parts; (1) Supply pump is used to supply high pressure fuel into (2) the common rail, which is equipped with pressure sensors; (3) Pressure sensor is used control fuel pressure inside the rail to suit with the engine operating conditions; (4) Electronic Control Unit (ECU) is used to control operating the common rail system based on information from various sensors, such as (5) pressure limiter, the crankshaft positioning sensor, the throttle positioning sensor, the intake temperature; (6) Electric Driver Unit (EDU) receive information from ECU to control injector suit with load and speed of the engine. ECU also controls the suction control valve (SCV) of supply pumps to match with engine operation; (7) Injector is used to inject fuel; (8) Fuel filter and (9) Fuel tank.

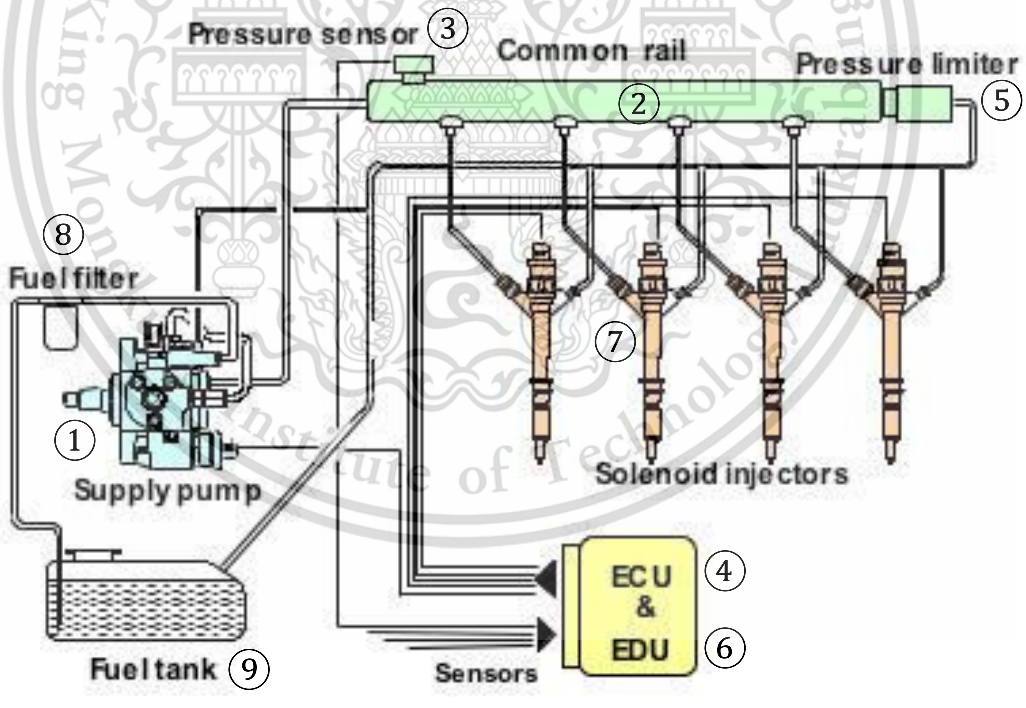


Figure 2.6 Main component of common rail system [31]

2.6 Hydrotreated Vegetable Oil

Hydrotreated vegetable oil (HVO) is the second generation biofuel, which can produce from many kinds of vegetable oil by using hydrotreating process, the triglyceride is hydrogenated in the first step and broken down into various intermediates, mainly monoglycerides, diglycerides, and carbonxylic acids. These intermediates are then converted into alkanes by different pathways: decarboxylation, decarbonylation (both removing a carbon atom from the initial intermediate), and hydrodeoxygenation (with no carbon remove) at the temperatures above 300–360 ^{o}C and pressure at least 3 MPa. Propane, water, carbon monoxide and carbon dioxide are produced as side-products [12].

HVO is a mixture of normal paraffin and iso-paraffin with straight chain-length [12], [32]. The properties of HVO are beneficial to use into the diesel engine such as high cetane number, high heating value and narrow distillation temperature range. However, there are some disadvantages that may limit to use HVO from previous study such as poor low-temperature properties, as displayed by cloud point, pour point and cold filter plugging point (CFPP) [12]. Therefore, an improvement process as isomerization process is may be use to solve that problem then HVO would be iso-HVO.

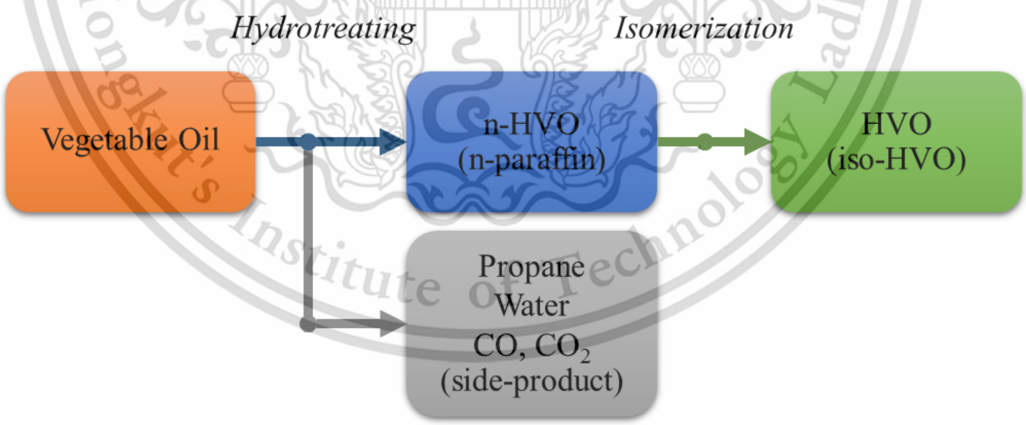


Figure 2.7 Schematic diagram of HVO production [12]

2.7 Effect of HVO on combustion and emissions characteristics

Sugiyama *et al.* [33] carried out HVO on the engine and chassis dynamometer. The engine was 4-cylinders diesel direct-injection engine, 2.2 liter turbocharged with common rail system. The results from the engine test showed that heat release rate of HVO exhibited more advance with shorten ignition delay due to the high cetane number for both single and pilot-injection cases. HVO is early start of combustion for low, medium and high engine load at engine speed of 2000 rpm. The exhaust gas emissions and fuel consumption of HVO can decrease 4.97% in nitrogen oxide (NO_x), 34.54% in hydrocarbon (HC), 37.79% in carbon monoxide (CO) and 3.21% in fuel consumption.

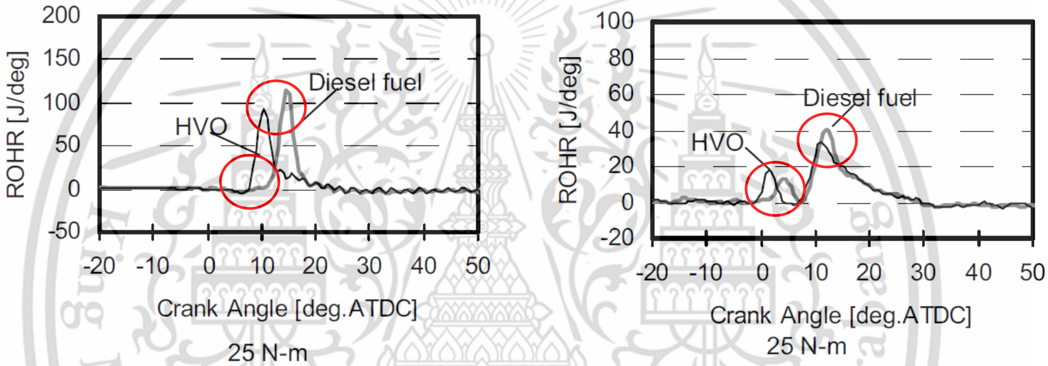


Figure 2.8 Results of rate of heat release using HVO with low engine load (a) Single-injection, (b) Pilot and main-injection [33]

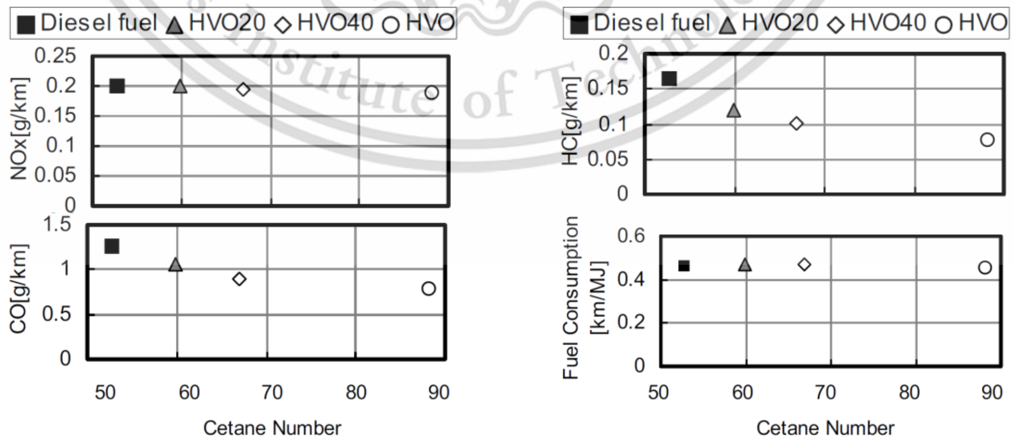


Figure 2.9 Results of exhaust gas emissions and fuel consumption using HVO in the vehicle test [33]

Jaroonjitsathian *et al*. [34] studied effects of HVO, GTL, HVO and GTL blend with diesel 50% on engine performance and emission characteristics. The experiment conducted with 4-cylinders common-rail direct injection engine, 2.5 liter. This study was performed using pilot and main injection at engine speed of 1170 rpm. The results revealed that ignition delay of HVO was shorter than diesel by 7.0% for pilot injection condition, and 7.5% for without pilot injection due to a high cetane number as shown in Figure 2.10, the shorter combustion duration by 8.46% for pilot injection, and 4.96% for without pilot injection as shown in Figure 2.11 due to narrow distillation range and lower distillation temperature at T90, which provided better vaporization and good mixture formation.

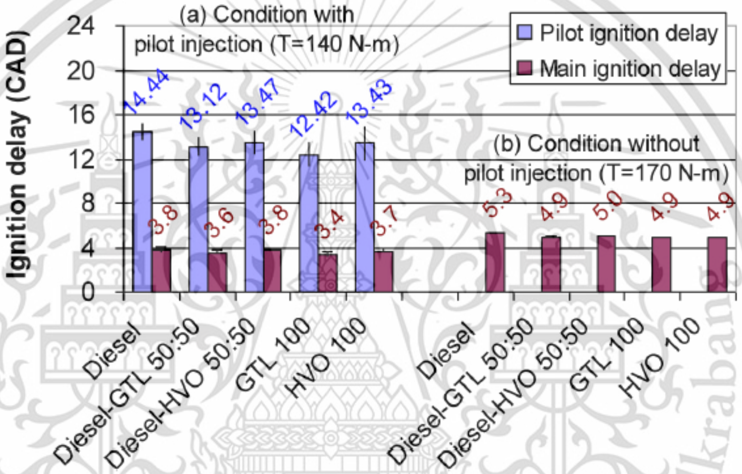


Figure 2.10 Comparison of the ignition delay [34]

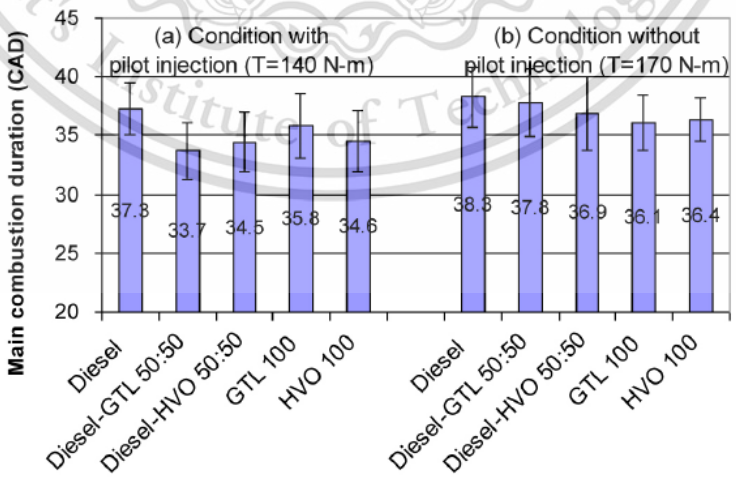
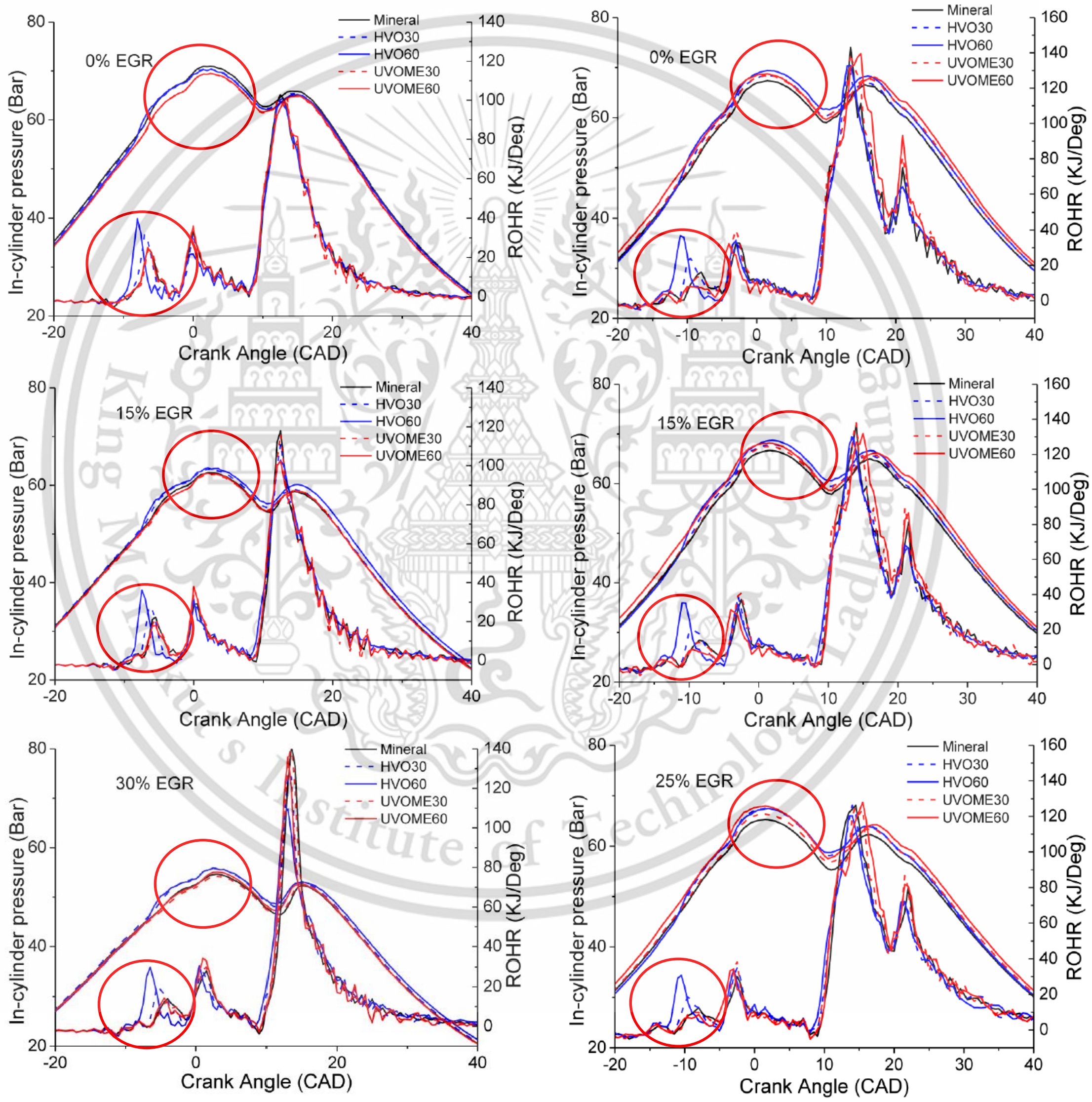


Figure 2.11 Comparison of combustion duration [34]

Liu *et al.* [6] studied effects of HVO blends on combustion characteristics by using 6-cylinders diesel engine, turbocharger 3.0 liter. The engine was performed under EGR conditions with low and high engine load at 1500 rpm. The results concluded that combustion pressure and heat release rate of HVO showed slightly higher than diesel, moreover the shorter ignition delay and combustion duration were observed with high EGR rate due to its higher cetane number, higher amount energy per unit volume, and the long-chain paraffin with straight chain-length of HVO is easier to broken up than diesel fuel.



(a) The in-cylinder pressure and rate of heat release for different fuels with various EGR rate at 1500RPM, 72Nm. (a)0% EGR, (b)15% EGR, (c)30% EGR

(b) The in-cylinder pressure and rate of heat release for different fuels with various EGR at 1500RPM, 143Nm. (a)0% EGR, (b)15% EGR, (c)25% EGR

Figure 2.12 Results of In-cylinder pressure and rate of heat release [6]

Millo *et al*. [35] investigated HVO blend by using 4-cyliders diesel direct-injection engine, turbocharger 1.2 liter at 1500 RPM@2 bar BMEP This experiment concluded that HVO blend with diesel 30% showed increase of engine performance by increasing fuel conversion efficiency and lower brake specific fuel consumption (BSFC) with various advance start of injection (SOI). In addition, HVO blend can increase fuel conversion efficiency and reduce BSFC with using EGR simultaneously at low engine load with SOI = 5.7 deg bTDC.

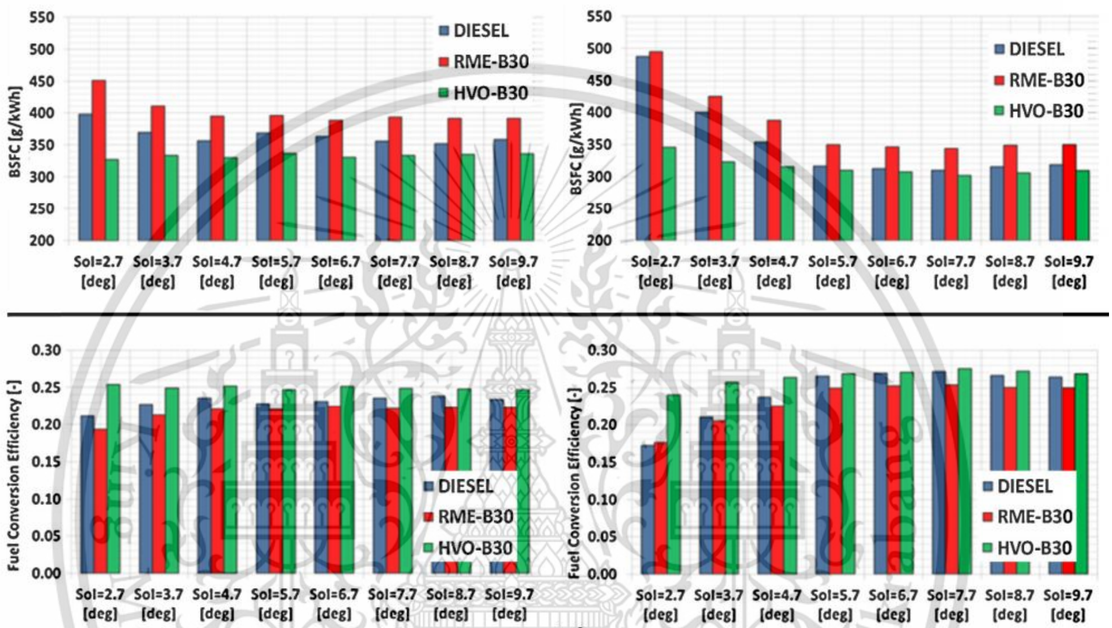


Figure 2.13 In-cylinder pressure, heat release rate, mass fraction burned, and injector signal [35]

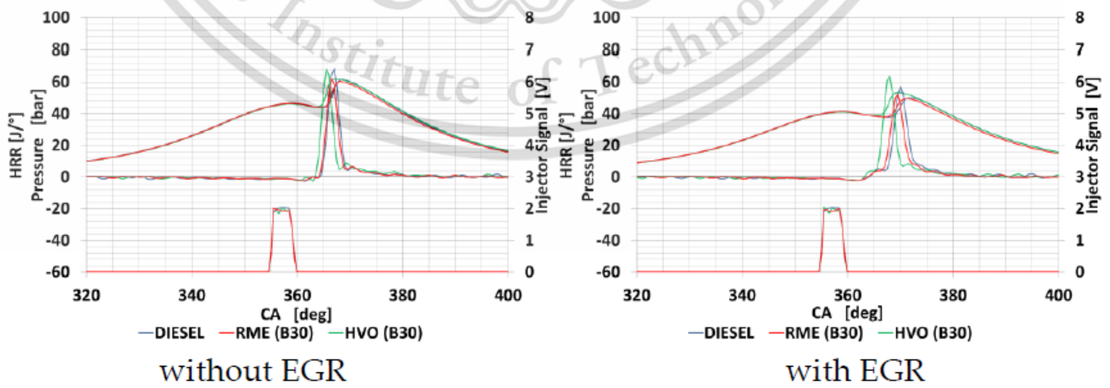


Figure 2.14 In-cylinder pressure, heat release rate, mass fraction burned, and injector signal [35]

Armas *et al*. [36] carried out HVO on chassis dynamometer to performance, combustion and emission characteristics under New European Driving Cycle (NEDC) by using 4-cylinders diesel direct-injection engine. The results showed that HVO exhibited 9.74% lower specific fuel mass consumption, 9.21% lower heat release rate, and shorter ignition delay due to its higher cetane number. In addition, exhaust gas emissions also decreased.

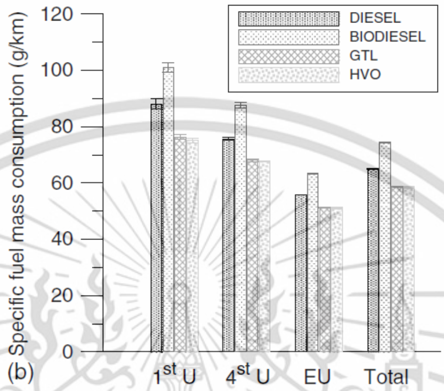


Figure 2.15 Results of specific mass consumption [36]

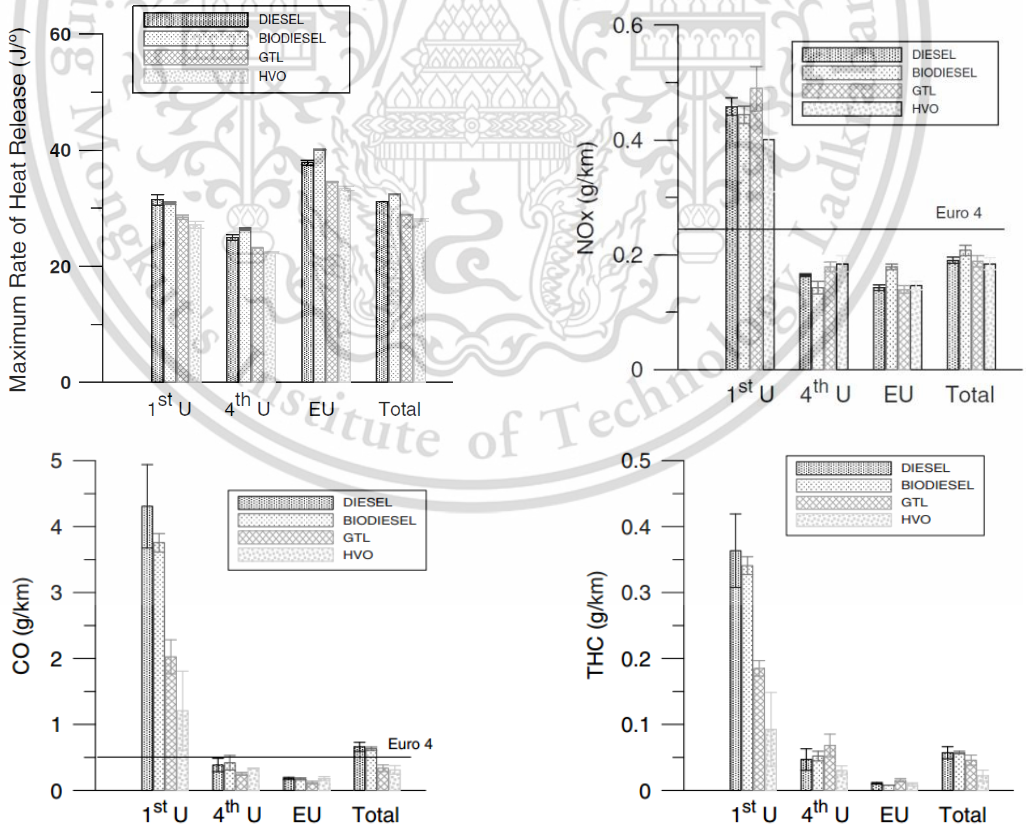


Figure 2.16 Results of heat release rate and specific NO_x, CO and HC emissions [36]

From previous researches have been done on combustion and emission characteristics, so these can be summarized that HVO provided the shorter ignition delay, similar heat release rate trend [33]–[35], and lower exhaust gas emissions [34], [37], for instance, HC and CO emissions because HVO is more complete combustion due to its better ignition quality then provide better combustion efficiency.

2.8 Exhaust gas recirculation

Exhaust gas recirculation (EGR) is one of today's most common engine control system that used to suppress NO_x formation. The exhaust gas are introduced to mix with fresh air into the combustion chamber, diluting oxygen concentration and increasing its heat capacity in combustion chamber [17], [38]. Therefore, the diluted oxygen availability caused the lower combustion temperature and reduced flame temperature, then reduced NO_x formation.

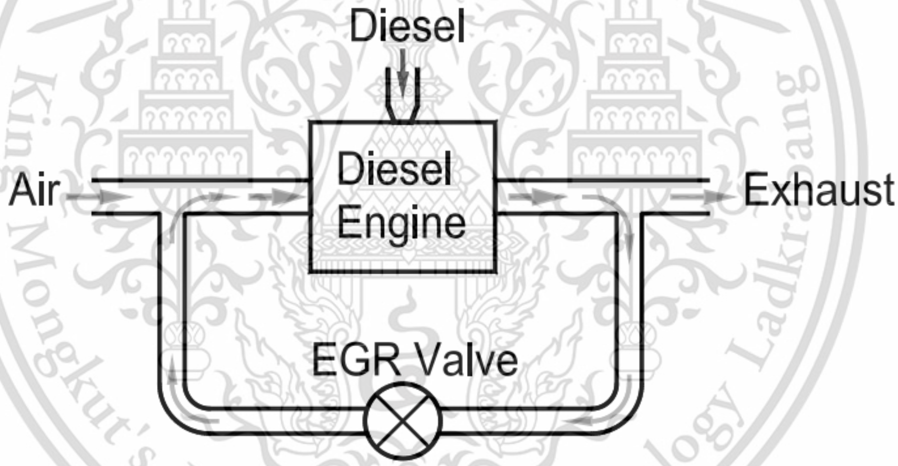


Figure 2.17 Exhaust gas recirculation (EGR) [38]

Kitamura *et al.* [28] experimentally investigated effects of oxygen concentrations on diesel combustion in constant volume combustion chamber (CVCC) under simulated diesel combustion condition. The results showed that the decreased oxygen concentration to 18% and 15% had not affected to heat release rate, but it largely affected to NO_x emission as shown in Figure 2.18.

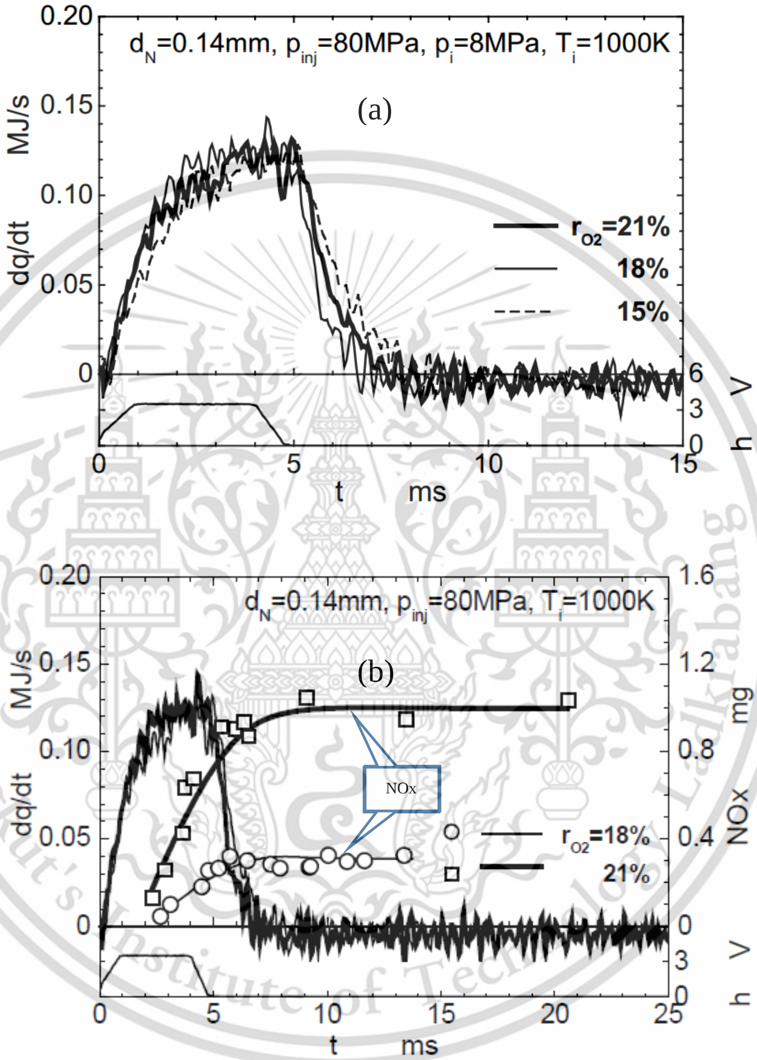


Figure 2.18 Effect of oxygen concentrations on (a) heat release rate and (b) NO_x emission [28]

Jing *et al.* [39] experimentally investigated effect of oxygen concentrations by changing oxygen concentration on flame temperature in CVCC. The results showed that a decrease in oxygen concentration effected to flame temperatures reduction as shown in Figure 2.19. In addition, EGR has potential to reduce NO_x emission by reducind the flame temperatures. However, decreasing oxygen concentration also decreased heat release due to decreasing combustion reaction intensity.

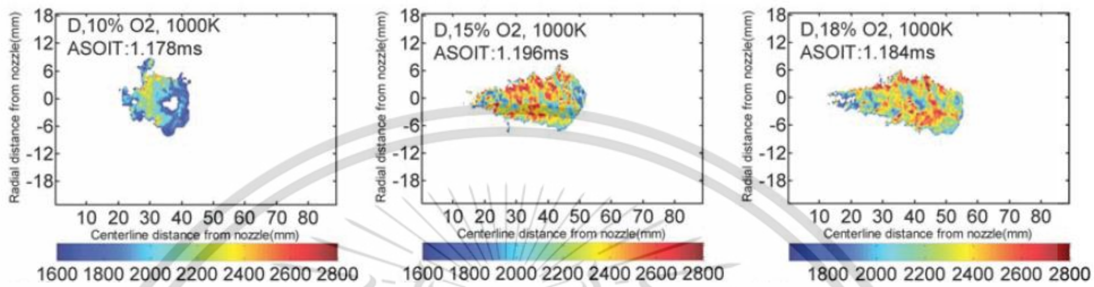


Figure 2.19 Effect of oxygen concentration on flame temperature [39]

Zhang *et al.* [18] investigated effect of oxygen concentration by change oxygen concentration 21%, 18%, 15% 12% and 10% in CVCC. The results showed that the lower oxygen concentration extended ignition delay and reduced heat release rate due to decrease in combustion reaction intensity as shown in Figure 2.20. Moreover, decreasing reaction intensity is explained in function of OH* chemiluminescence as shown in Figure 2.21.

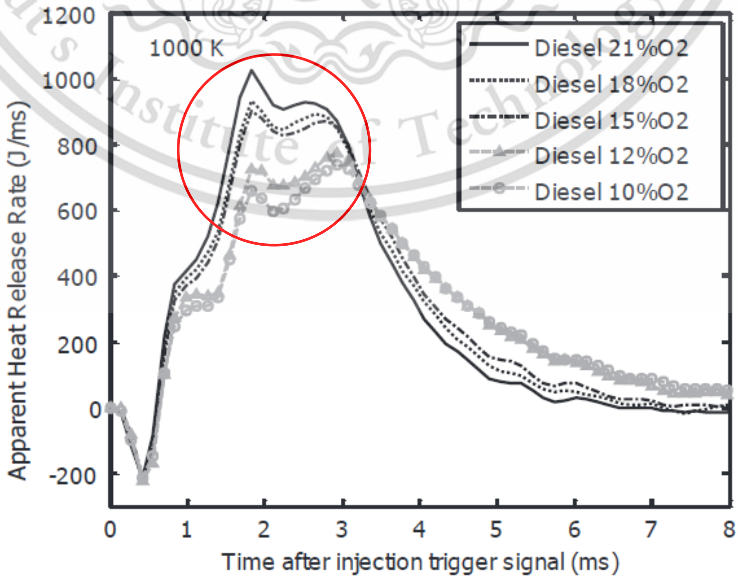


Figure 2.20 Effects of oxygen concentration on heat release rate [18]

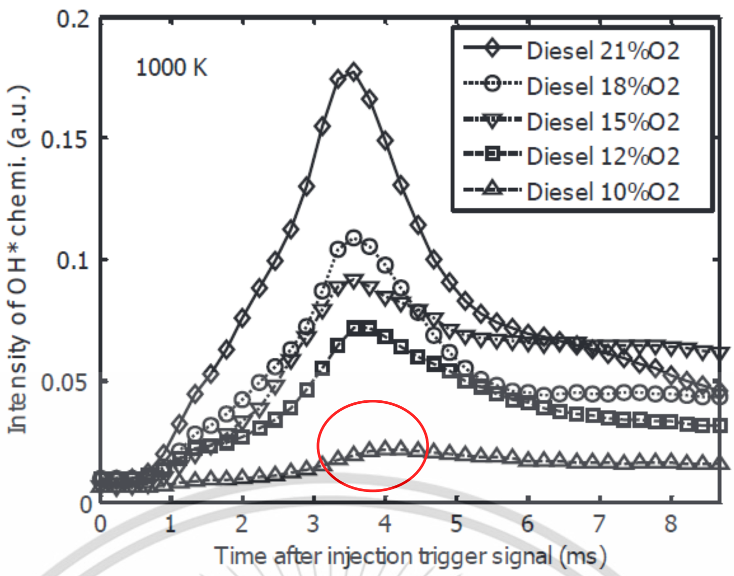


Figure 2.21 Effects of oxygen concentration on OH* chemiluminescence [18]

Zhang *et al*. [40] experimentally investigated effects of the ambient oxygen concentration on soot temperature and KL factor in CVCC. The results showed that reducing oxygen concentration can reduce soot temperature, the change of oxygen concentration greatly affected on the soot formation and flame temperature.

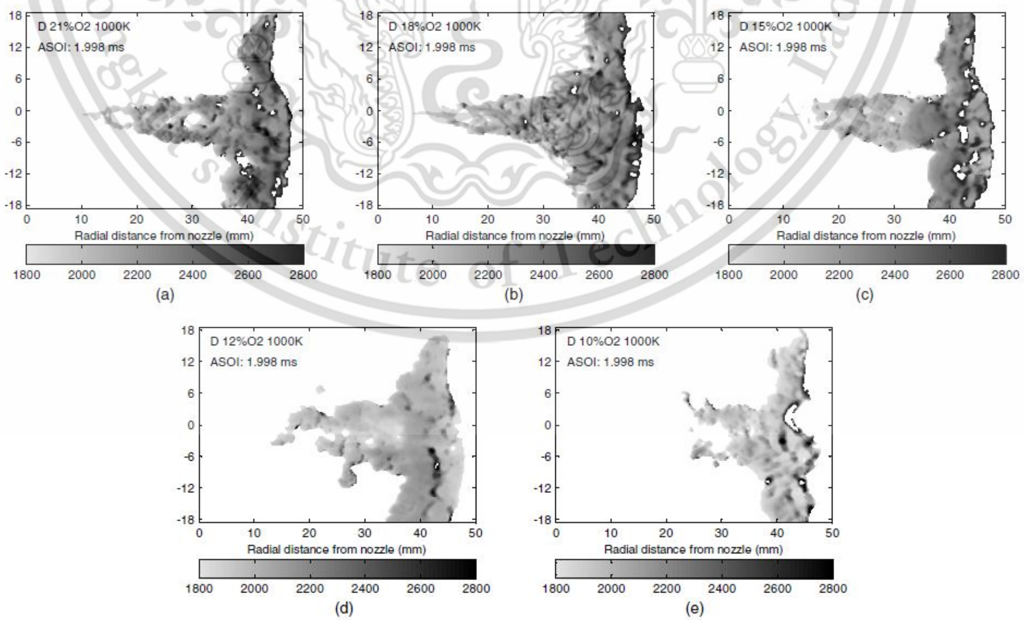


Figure 2.22 Two-color temperature of diesel combustion at 1.988 ms under O_2 concentrations of (a) 21%; (b) 18%; (c) 15%; (d) 12%; (e) 10% [40]

Ewphun *et al*. [41] investigated effects of oxygen concentration and supercharged conditions on the combustion characteristics of HVO-diesel blends in Rapid Compression Expansion Machine (RCEM). The results showed that the employment of HVO and blends decreased the ignition delay, the flame temperature, the soot concentration and the NO_x concentration simultaneously. In addition, heat release rate at oxygen concentration of 10 % dramatically dropped due to a shortened ignition delay. The heat release rate, flame temperature, and NO_x concentration were decreased when the oxygen concentration was reduced due to decreasing reaction intensity. The results of reducing in oxygen concentration and applying supercharged conditions showed that an increase in heat release rate, flame temperature, decrease in ignition delay and soot concentration.

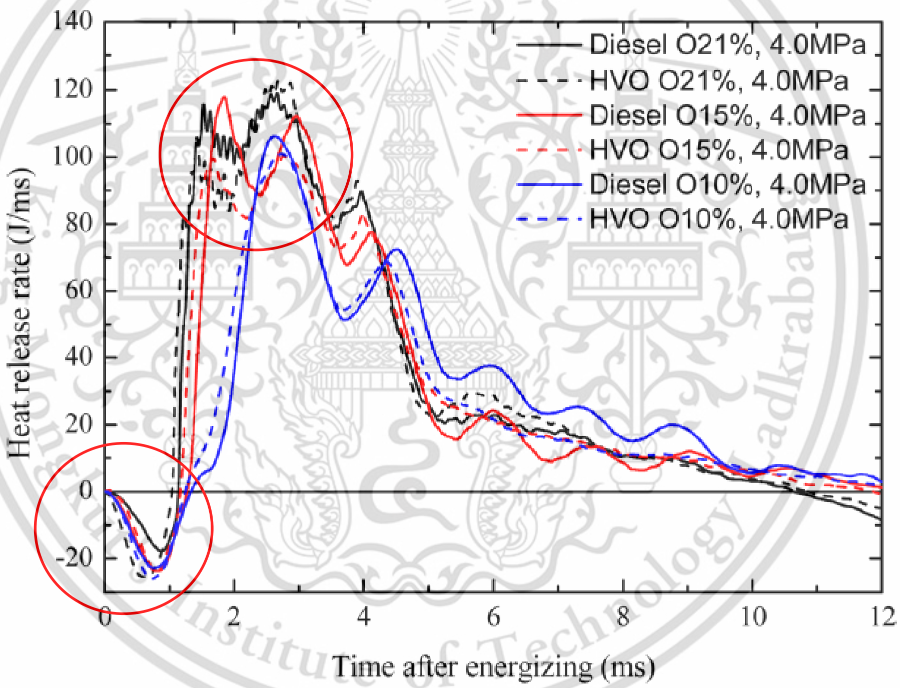


Figure 2.23 Heat release rate under reduce oxygen concentration [41]

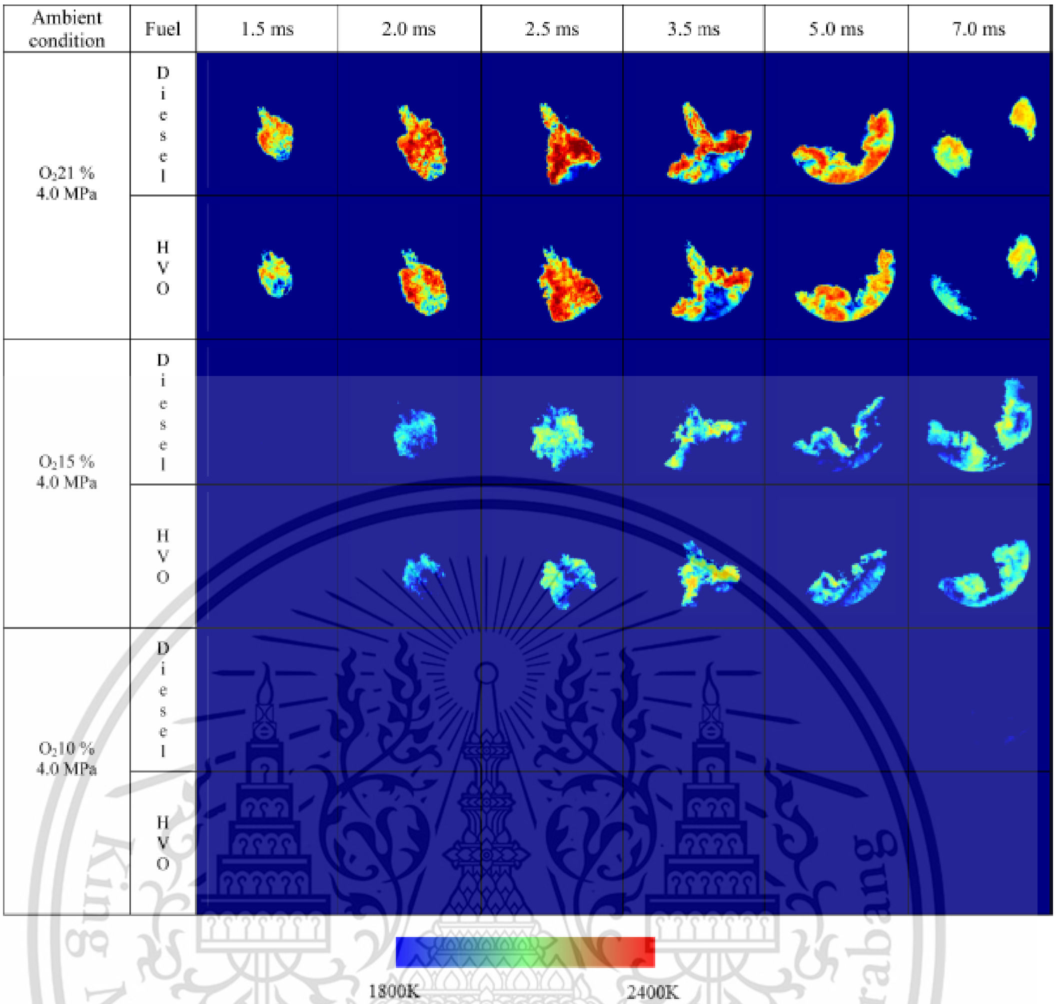
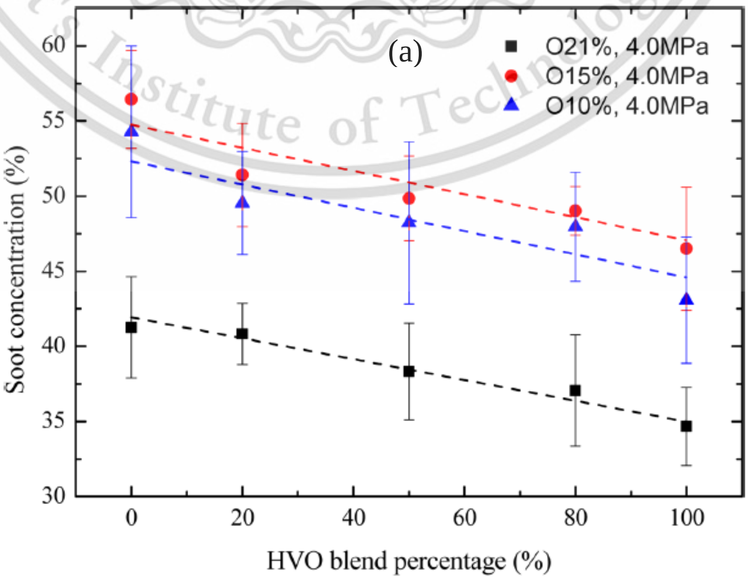


Figure 2.24 Flame temperature image under reduce oxygen concentration [41]



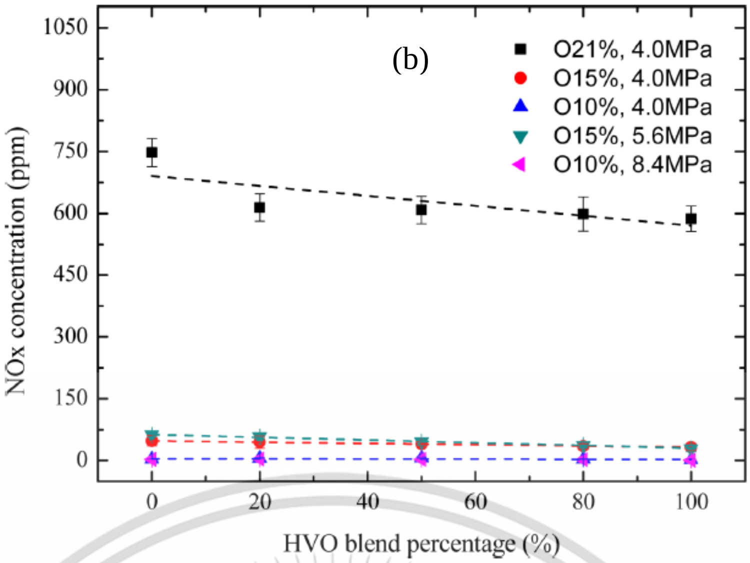


Figure 2.25 (a) Soot and (b) NO_x concentration under reduce oxygen concentration [41]

2.9 Low temperature combustion

A main benefits of using low temperature combustion (LTC) is to reduce NO_x and soot emissions simultaneously as lowering local temperature range [21], [22], [42], From previous research, there are two categories of LTC [21], [22]. First, the combustion phasing is decoupled from injection timing and combustion is dominated by the chemical reaction, the mixture is near homogenous like homogenous charge compression ignition (HCCI) as an equivalence ratio is less than 1. In the second category, using the injection event, pre-mixing occurs between the fuel injection and start of combustion event, but significant regions exist where the equivalence ratio is greater than unity at the start of the combustion like premixed charge compression ignition (PCCI).

The main objective of these strategies is to keep a flame temperature low by using low compression ratio, large amount of cooled EGR, and using retarded injection timing [22]. Therefore, LTC can achieved a lower flame temperature that leaded to decrease simultaneously in NO_x and soot emissions at expressed in Figure 2.26.

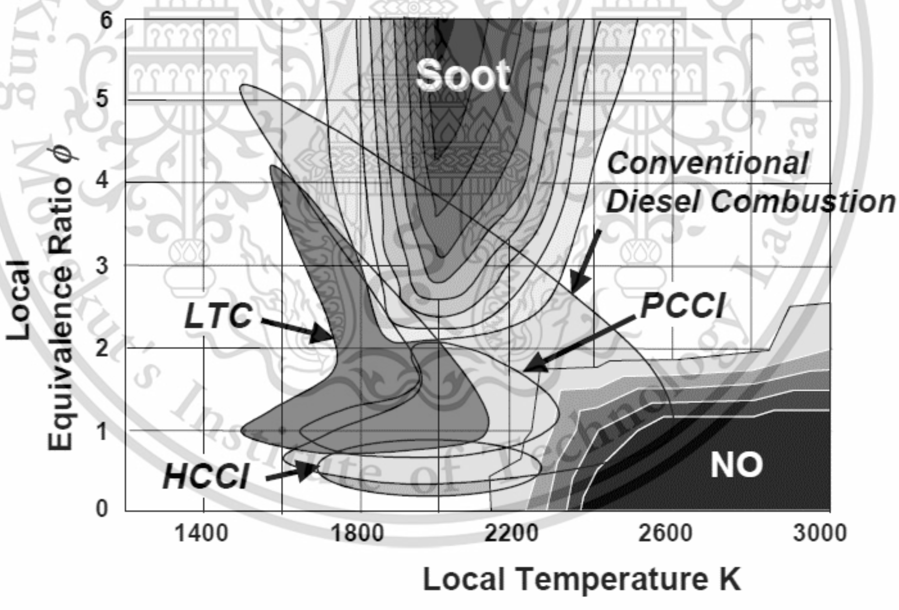
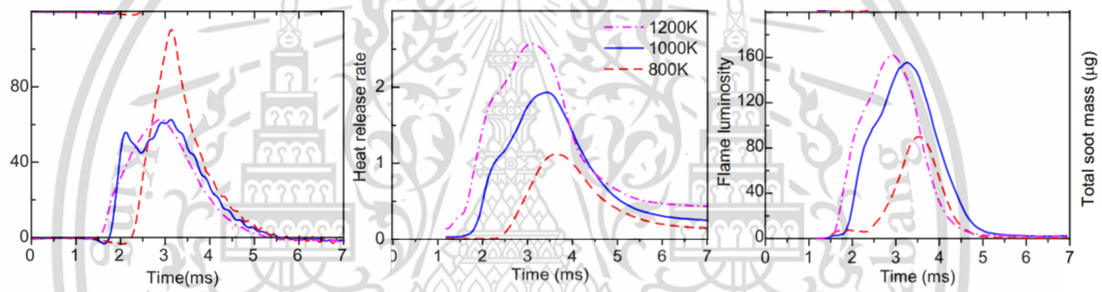


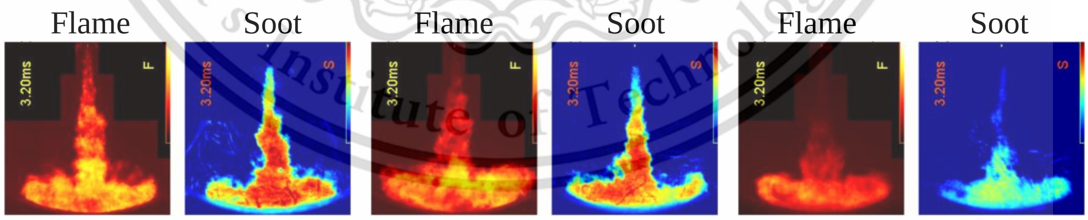
Figure 2.26 Local equivalence ratio and local temperature map [42]

Xu *et al*. [43] studied effects of ambient temperature and oxygen concentration on soot evaluation in constant volume optical spray chamber. The ambient temperature was 1200 K, 1000 K and 800 K. The oxygen concentration also was 21% to 15%. The conclusions showed that a decrease of ambient temperature caused longer ignition delay with higher heat release rate. The fuel burnt is increased with portion of premixed combustion. In addition, decreasing ambient temperature is beneficial to less soot form during combustion as shown by soot mass, and reduced flame luminosity. Lower oxygen concentration caused lower flame luminosity, while having the same soot level. In addition, the combustion process is basically not changed with changing oxygen concentration, but longer combustion duration and lower heat release rate. The lower heat release rate resulted lower flame temperature, which might contribute to reduce NO_x emission.



Heat release rate Flame luminosity Total soot mass

Figure 2.27 Heat release rate, Flame luminosity and Total soot mass under various ambient temperature [43]



At 1200 K At 1000 K At 800 K

Figure 2.28 Flame luminosity and soot distribution images under various ambient temperature [43]

Zhang *et al*. [20] studied the transient flame structure by high speed imaging of natural luminosity and OH* chemiluminescence of diesel under low oxygen concentration (15% and 10%) and low temperature combustion (1200 K, 1000 K and 800 K) in CVCC. The results showed that at the same ambient temperature, 15% O_2 concentration provides a shorter ignition delay with higher heat release rate compared to 10% O_2 concentration due to a higher oxygen availability in the chamber. At the same O_2 concentration, decreasing ambient temperature provides a longer ignition delay with higher heat release rate due to improved mixture preparation, also resulting in higher burning rate. The intensity of natural luminosity (NL) showed that decreasing O_2 concentration decreased the level NL as decrease flame temperature. Decreasing ambient temperature decreased OH* chemiluminescence as reduce combustion intensity. This lead to reduce soot form.

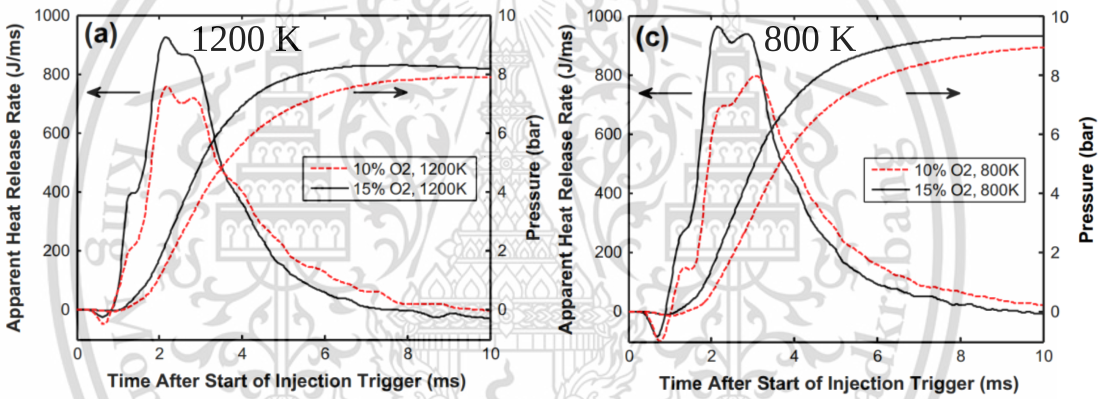


Figure 2.29 Heat release rate at 15% and 10% O_2 concentration and ambient temperature of 1200 K and 800 K [20]

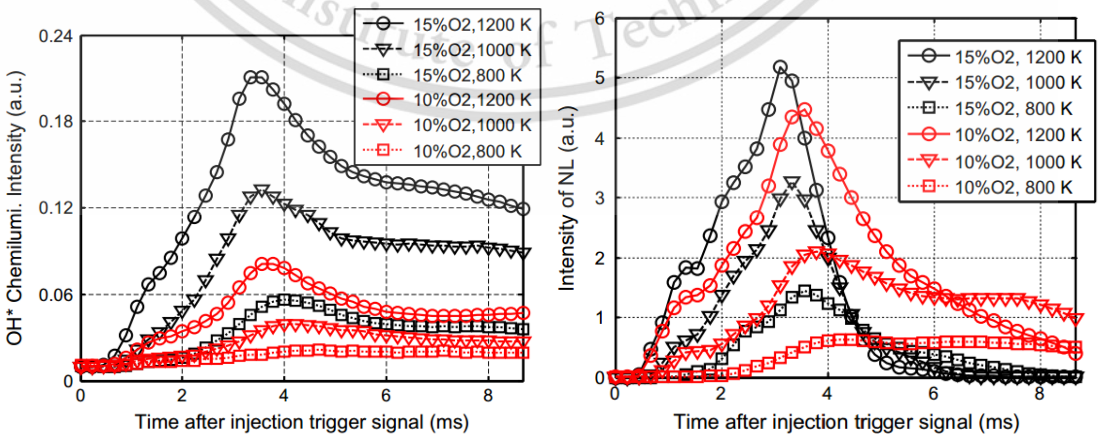


Figure 2.30 OH* chemilumi. and natural intensity at 15%, 10% O_2 concentration and ambient temperature of 1200 K, 1000 K, 800 K [20]

Jing *et al*. [44] experimentally investigated effects of ambient temperature and oxygen concentration on the natural luminosity and OH* chemiluminescence using high-speed imaging of both natural luminosity and OH* chemiluminescence in CVCC. The results showed that at 800K, lift-off length for 800K is much longer than the other two temperatures with oxygen concentration. The natural luminosity of 800 K ambient temperature has lower than other conditions. This implies that ambient conditions with low temperature and low oxygen concentration leads to a slower reaction and more mixing time before the high temperature reaction occurs with long flame penetration.

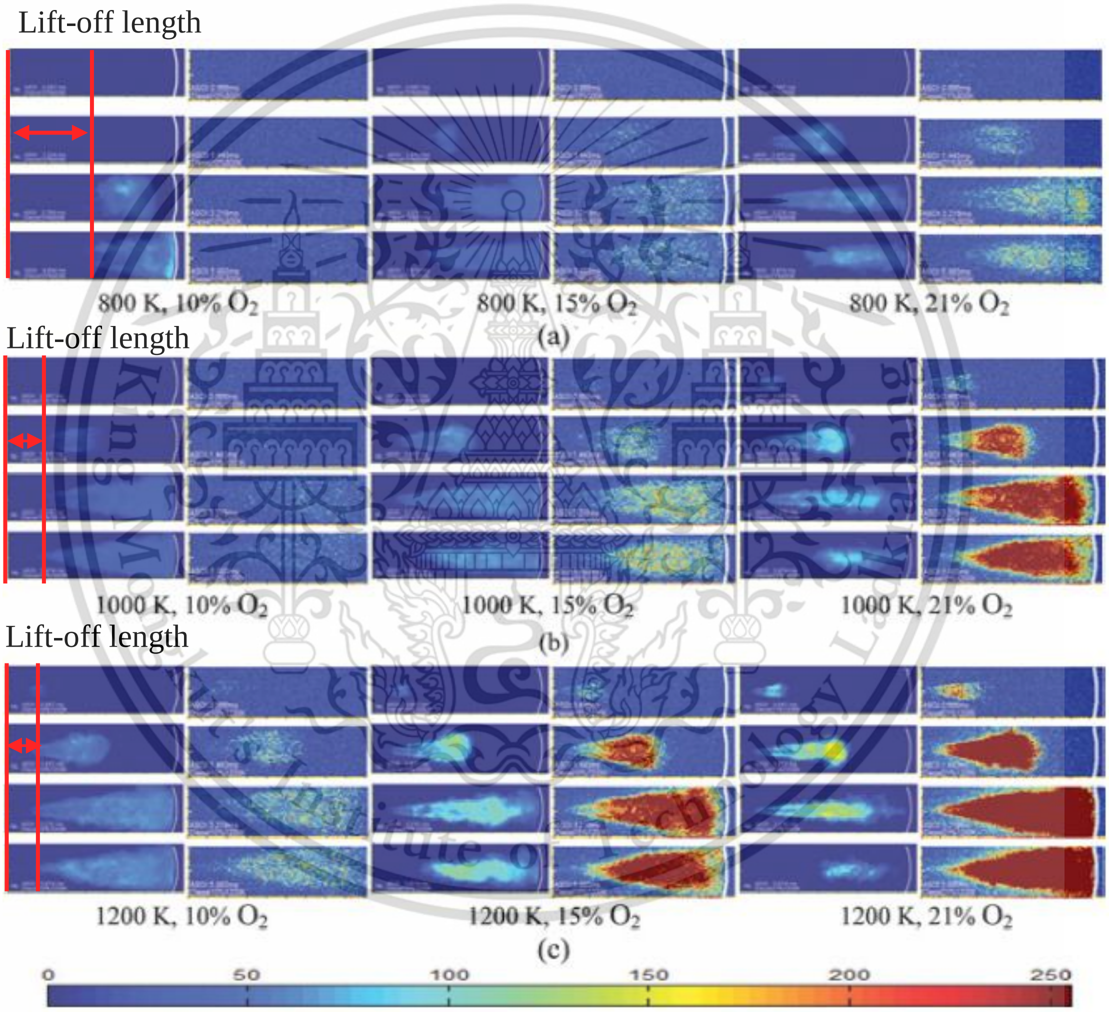


Figure 2.31 Images of natural luminosity and OH* chemiluminescence of (a) 800 K, (b) 1000 K and (c) 1200 K [44]

Cheng *et al.* [45] investigated diesel spray combustion flame and soot formation under effect of ambient temperature and pressure in CVCC by using laser-induced incandescence (LII). The results showed at higher temperature can accelerate the process atomization and evaporation of diesel droplets, thus the fuel mixture was rapidly formed that caused the decrease of ignition delay. The flame lift-off length decreased with increasing ambient temperature as shown in Figure 2.32. The increase of ambient temperature, the average LII signal intensity increased, which is proportional to soot concentration produced in the combustion process as shown in Figure 2.33.

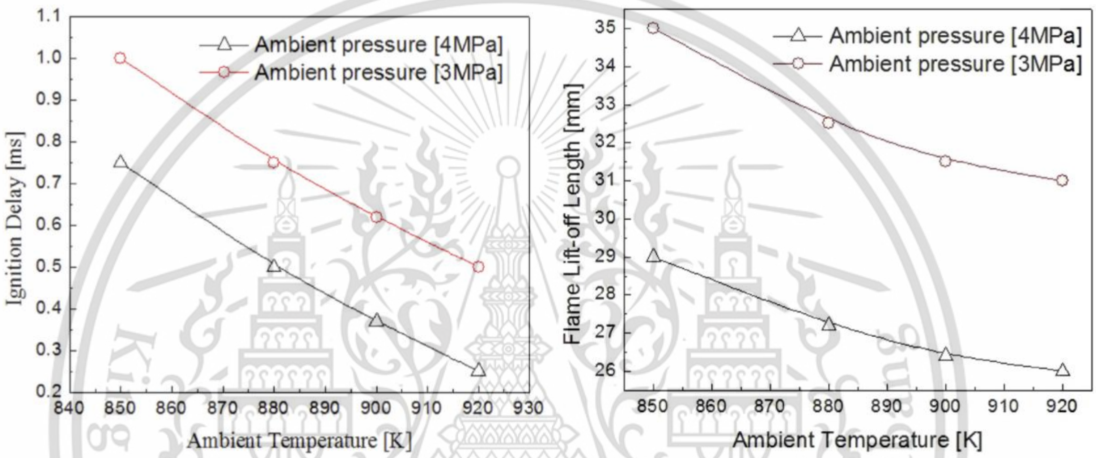


Figure 2.32 Ignition delay and flame lift-off length with various ambient temperature [45]

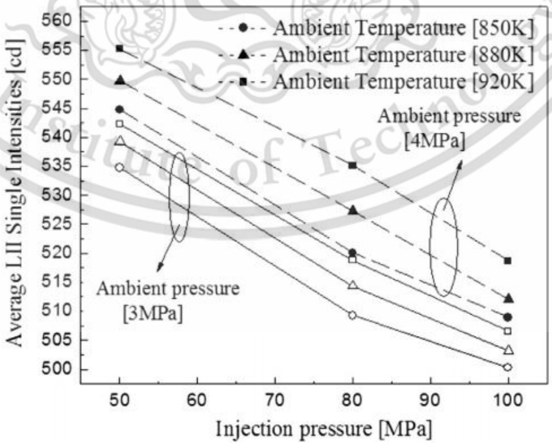


Figure 2.33 Average LII single intensity with various ambient temperature and injection pressure [45]

2.10 Research gap

The advantages of properties of HVO, as displayed by low sulfur and aromatics, narrow distillation temperature range, lower T90 distillation temperature, and higher cetane number, which might contribute to improve combustion under low ambient oxygen concentration and low temperature. In addition, to meet the further stringent emissions, the EGR method as reducing the oxygen concentration is the most effective method to control NO_x emissions. Meanwhile LTC strategy is beneficial technique to suppress the formation of NO_x and soot emissions simultaneously due to the lower flame temperature.

Currently, few researches have been concluded the influence of diesel and HVO blend percentage on the characteristics of the ignition delay and spray combustion by using the heat release analysis and the shadowgraph technique. Especially, the discussions on effects of diesel and HVO blend percentage on the combustion visualization such as spray development process and flame development have very limited data. Therefore, the fundamental data of ignition delay, combustion and the visualization of combustion are helpful information to optimize and to develop the further design in diesel operating combustion conditions by utilizing HVO.

CHAPTER 3
RESEARCH METHODOLOGY

3.1 Heat release analysis in constant volume chamber

Heat release rate [25] can be determined from pressure rise after burning injected fuel based on the first law of thermodynamics of the system, as shown in Equation (3.1).

$$\frac{dQ}{dt} = \frac{\gamma}{\gamma - 1} \cdot P \cdot \frac{dV}{dt} + \frac{1}{\gamma - 1} \cdot V \cdot \frac{dP}{dt} \tag{3.1}$$

Where,

γ	Ratio of specific heat
dV/dt	Chamber volume change with time, m^3/s
dP/dt	In-chamber pressure change with time, Pa/s
P	In-chamber pressure, Pa
V	Chamber volume, m^3

In this study, the heat release analysis was studied in constant volume chamber where the chamber volume is kept constant then the *dV/dt* term can be deleted [46]. Hence, heat release rate is expressed as follows:

$$\frac{dQ}{dt} = \frac{1}{\gamma - 1} \cdot V \cdot \frac{dP}{dt} \tag{3.2}$$

3.2 Ignition delay determination

The ignition delay characterized into physical delay, which is associated with fuel atomization, vaporization and mixing. This depend on the physical properties such as fuel viscosity, density, and distillation temperature. While chemical delay is associated with chemical composition and structure, temperature, pressure, and oxygen mole fraction. In this study, the ignition delay is defined as the interval time between start of injection (SOI) to start of combustion (SOC) where heat release rate recover from the negative value due to heat absorption. This has been widely used to determine ignition delay [25], [26], [41], [47], [48] as described in Figure 3.1.

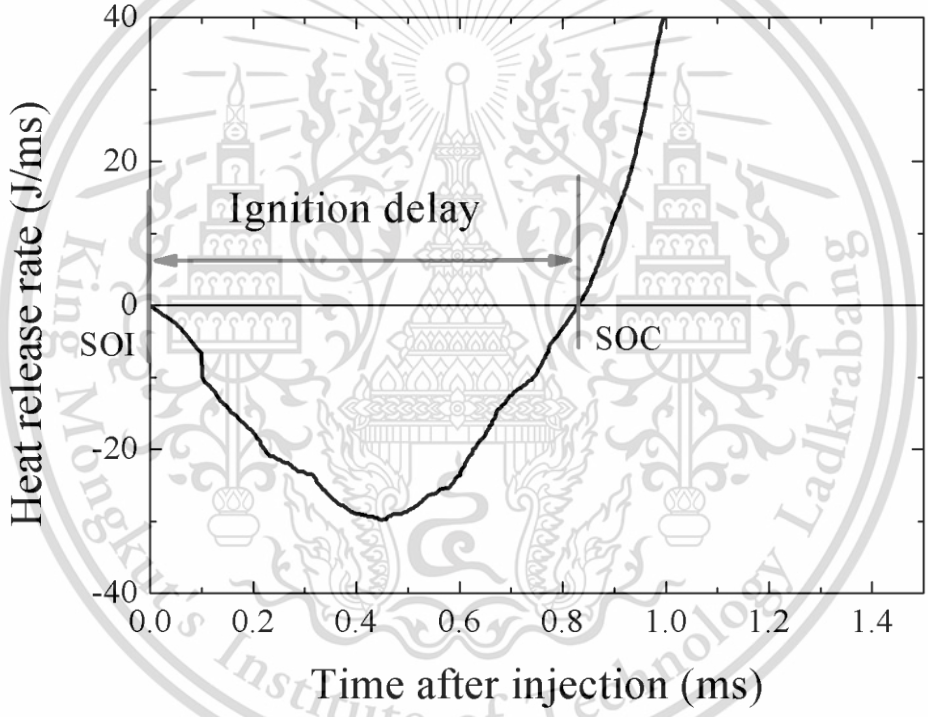


Figure 3.1 Definition of ignition delay [26]

3.3 Experimental setup

Figure 3.2 shows the schematic diagram of spray combustion characteristics experiment using CVCC. The schematic diagram in this experiment consists of three parts: (1) the high pressure fuel injection system was used to generate high injection pressure, the three-phase motor was used to drive a second generation common-rail pump to generate high fuel pressure by the inverter controller; (2) the gas system was used to allow to introduce into combustion chamber in order to simulate ambient gas condition; (3) the CVCC, the heater, the ignition system, the data acquisition and controller. The CVCC was a circular cylinder with 80 mm in diameter, 100 mm in depth. The two quartz windows were equipped in this chamber for optical assessment.

There are six main parts of combustion chamber system as shown in Figure 3.3: (1) an intake valve was used to control partial pressure of the premixed gaseous mixture of acetylene (C_2H_2), oxygen (O_2) and nitrogen, (N_2); (2) an exhaust valve was used to remove burnt-gas after combustion is ended; (3) the conventional spark plug was used to ignite gaseous mixture to generate high pressure and temperature; (4) the mixing fan was used to run 25 second before spark-ignition for maintaining uniform gas distribution throughout the combustion of fuel; (5) a dynamics pressure transducer (Kistler 6053CC60) and charge amplifier (Kistler 5011) were used to measure pressure rise of combustion, and (6) a single-hole injector was installed at the top of combustion chamber to inject tested fuel. The pressure rise of auto-ignition was recorded by oscilloscope (RIGOL DS1052E) with sampling rate 5×10^5 S/s.

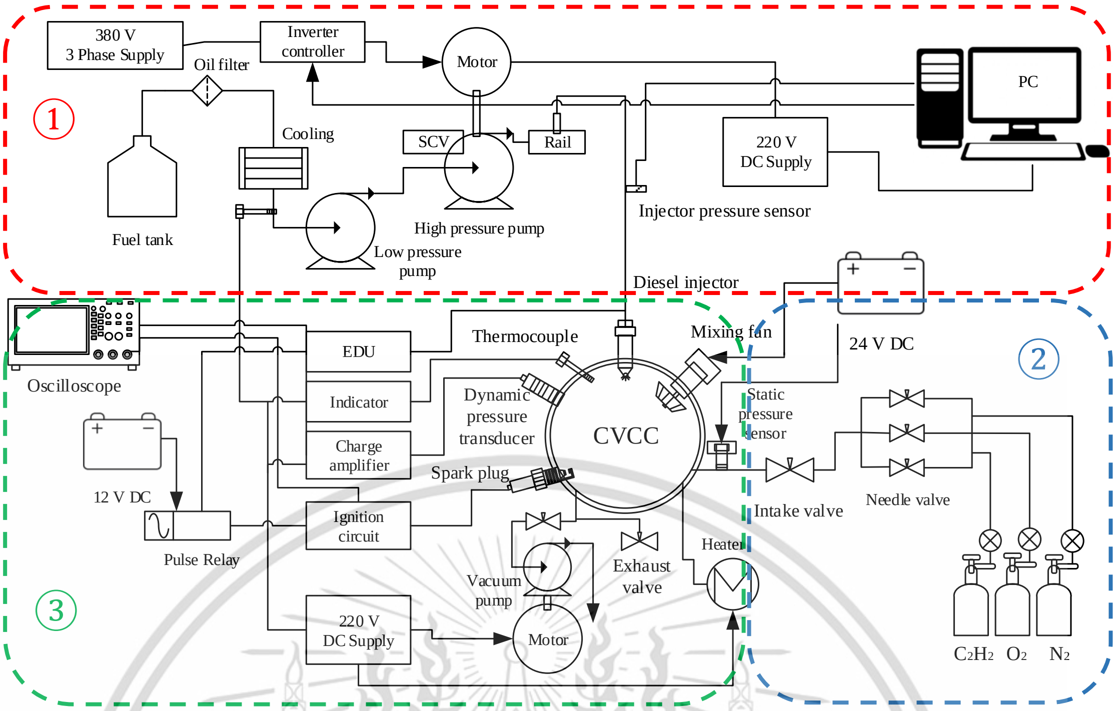


Figure 3.2 Spray combustion characteristics experiment using CVCC

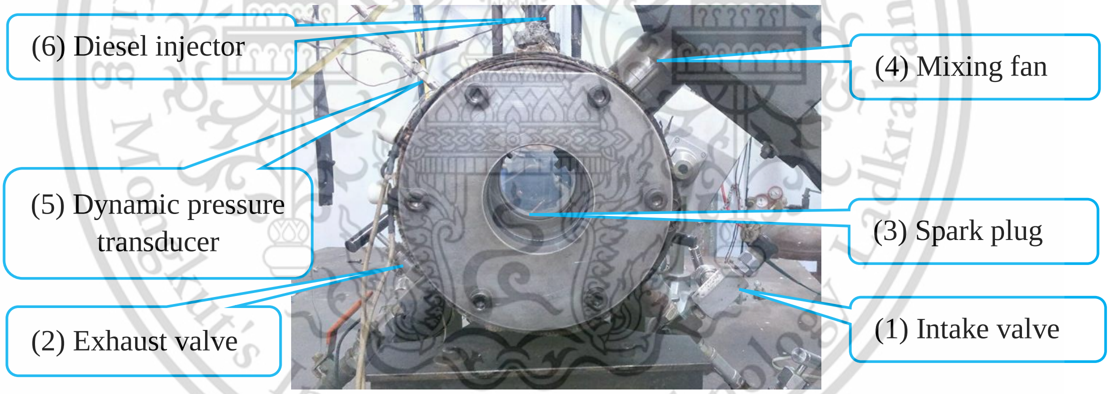


Figure 3.3 Constant volume combustion chamber system

3.4 Bulk gas temperature

The bulk gas temperature is used to calculate the relationship of ambient gas temperature and pressure as shown in Figure 3.4. From this figure, the ambient temperature is calculated based on the initial premixed gas of 1.5 MPa and the percentage of oxygen concentration of 21%. The ambient gas temperature and pressure were assumed to be uniform at each steps of ignition in this chamber. The ambient gas temperature data was used to precisely evaluate for cool down time, time after ignition where is start of injection. The equation can be expressed as follows [46]–[48]:

$$T_{bulk} = T_{init}\left(\frac{P_{bulk}}{P_{init}}\right)M \tag{3.3}$$

Where,

T_{bulk} Bulk gas temperature of combustion products, K

T_{init} Initial temperature of the premixed gas, K

P_{bulk} Bulk gas pressure of combustion products, N/m^2

P_{init} Initial pressure of the premixed gas, N/m^2

M Ratio of molecular weights of the combustion products and the premixed gas

To simulate ambient oxygen concentration of 21%, 15% and 10% and ambient temperature of 1100 K, 900 K and 700 K, it is essential to control precisely the percentages of oxygen concentration by controlling the initial pressure. The gas composition in the chamber was evaluated prior spark-ignition and at start of fuel injection. The ambient gas temperature was calculated and evaluated to obtain the time of fuel injection in the combustion chamber after spark ignition as expressed in Equation 3.4. The ambient temperature in this research is linked to the top dead center of diesel engine at the end of compression stroke. The percent of oxygen concentration remain as shown in Table 3.1.
This has been widely used to study combustion in constant volume chamber [18], [20], [44].

The reaction equation prior spark ignition and after spark ignition

$$aC_2H_2 + bO_2 + cN_2 \rightarrow dO_2 + cN_2 + aH_2O + eCO_2 \tag{3.4}$$

Table 3.1 Percent of oxygen concentration at the time of fuel injection and reactants prior to spark ignition

Percentage of oxygen concentration	Gas composition [mol%] (Composition before spark ignition)			Gas composition [mol%] (Composition at start of injection)			
	(a) C_2H_2	(b) O_2	(c) N_2	(d) O_2	(c) N_2	(a) H_2O	(e) CO_2
21%	3.5	29.3	67.2	21.0	67.2	3.5	7.1
15%	3.5	23.4	74.4	15.0	74.4	3.5	7.1
10%	3.5	18.5	79.4	10.0	79.4	3.5	7.1

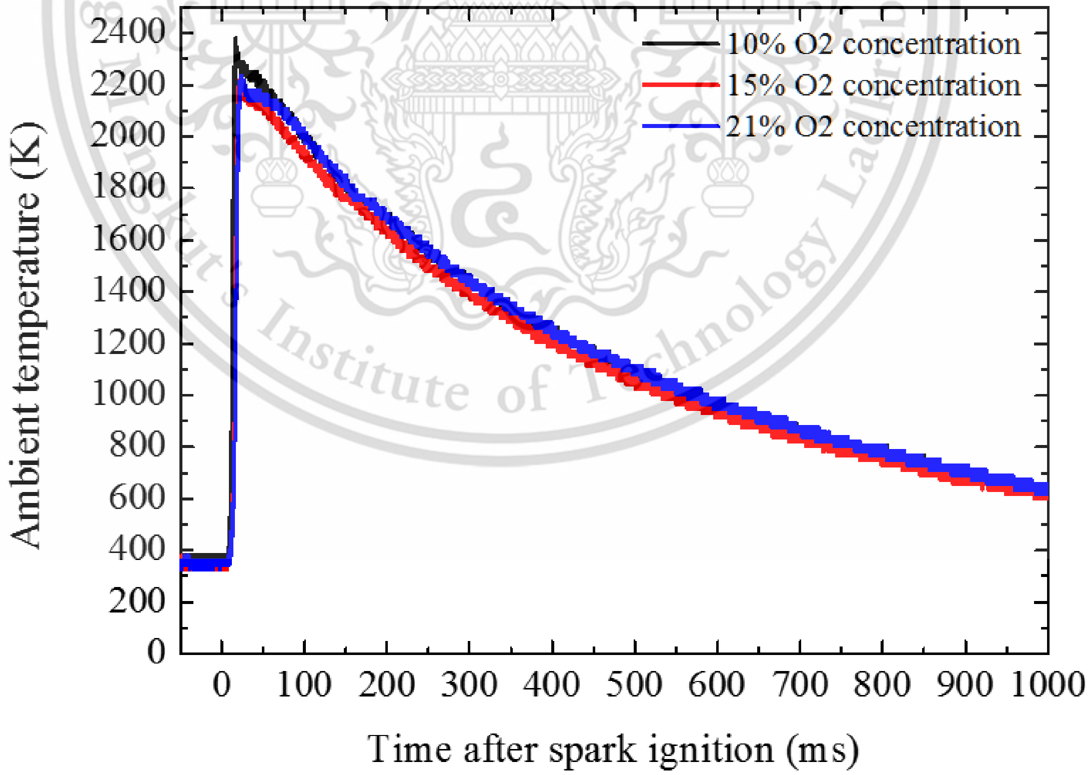


Figure 3.4 Ambient gas temperature of initial premixed gas of 1.5 MPa and 21% oxygen concentration

3.5 Experimental procedure

The spray combustion characteristics were investigated under simulated diesel combustion conditions in CVCC by using two-step combustion as illustrated in Figure 3.5. The first step was using spark plug to generate high-pressure and high-temperature ambient gas by burning the premixed gas of C_2H_2, O_2 and N_2 as premixed burning period. Before the second step, in-chamber pressure and temperature were decreased to reach diesel combustion condition due to the heat transfer as cool down process. The second step was fuel injection into combustion chamber, then injected fuel was continuously burnt as combustion of fuel.

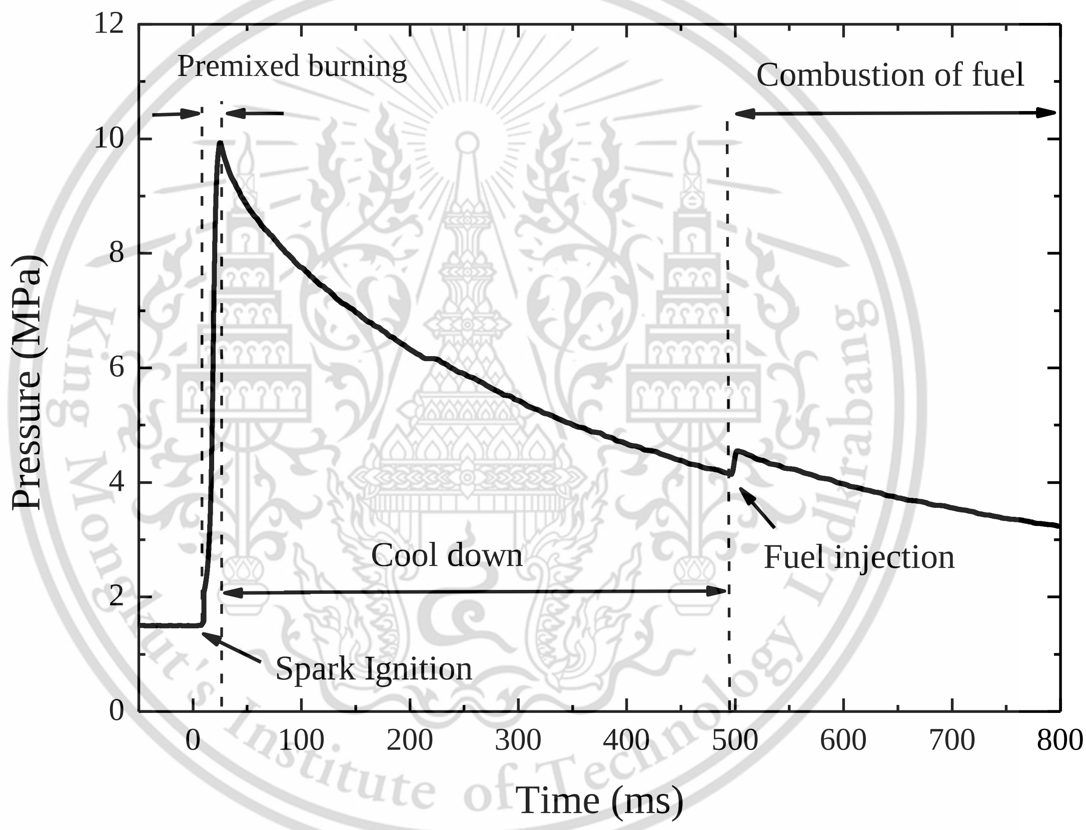


Figure 3.5 In-chamber pressure of two-step combustion

3.6 Visualization of combustion

Figure 3.6 shows schematics diagram of spray combustion experiment using the shadowgraph technique. The shadowgraph technique was used to visualize spray combustion in this research. The shadowgraph technique is a useful technique to study the macroscopic of diesel flame [50]. The concave mirror allowed to reflect the light from the Xenon lamp light source that passes through the combustion chamber. The high speed video camera (FASTCAM MINI UX-100) with 10,000 frame per second (fps) was used to capture high speed video combustion, and a resolution of 640 x 480 pixels, and shutter speed of 10.2 μs [51].

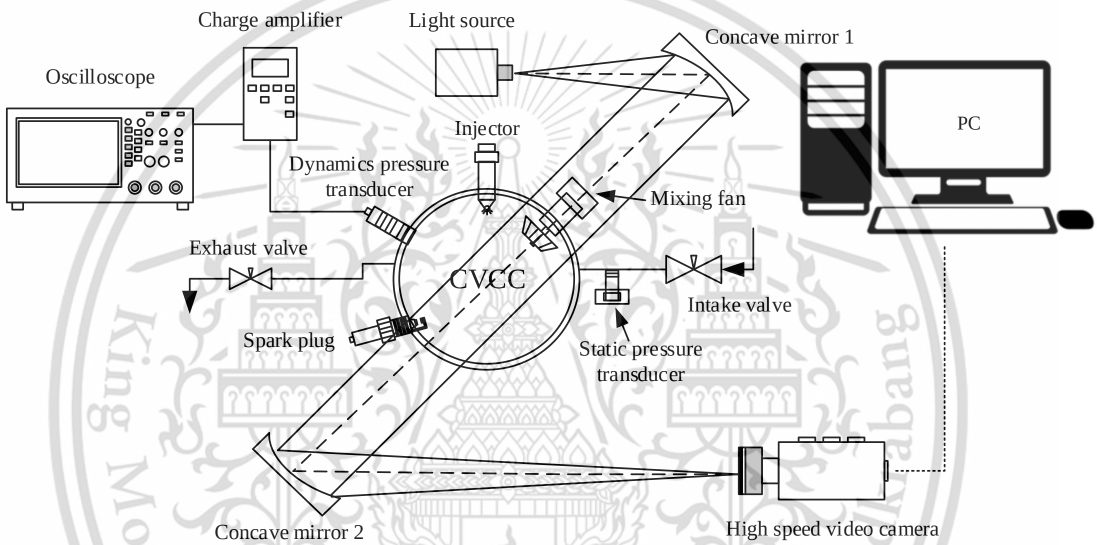


Figure 3.6 Schematics diagram of spray combustion experiment using shadowgraph technique

3.7 Test fuels

Table 3.2 shows fuel properties of test fuels. In this study, four different fuels were commercial diesel grade, commercial diesel blended HVO by mass: 20% (H20), 50% (H50) and pure HVO. The properties of test fuels were determined prior to the study of spray combustion. From this table, the viscosity of H20, H50 and HVO have lower than diesel by 4.62%, 10.49% and 18.52%, with lower density than diesel by 1.21%, 2.91% and 5.58%. But higher heating values by 0.39%, 0.11% and 2.18%. In addition, one of important property in this study is distillation temperature that describes the evaporation characteristics of fuel. At T90 distillation temperature, H20, H50 and HVO have lower than diesel by 0.21%, 0.70% and 16.77%, as shown in Figure 3.8. If fuel has lower distillation temperature, it will show fast evaporation and mixing with ambient air. The cetane index is used to estimate the cetane number of fuel, which calculated from fuel density, T10, T50 and T90 distillation temperature. According to the larger difference of density, T50 and T90 of H20, H50 and HVO compared to diesel so that the cetane index of H20, H50 and HVO are higher than diesel by 4.86%, 13.05% and 27.23% due to its lower density, T50 and T90, respectively.

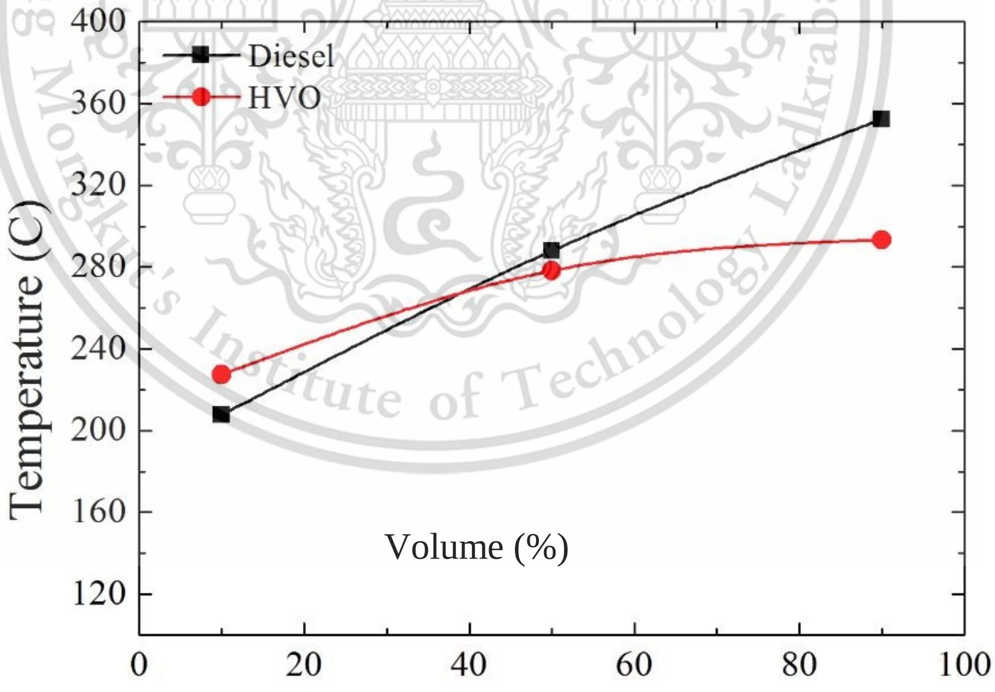


Figure 3.7 Distillation curve of diesel and HVO

Table 3.2 Fuels properties

Properties	Standard	Diesel	H20	H50	HVO
Density @ 30°C (g/cm^3)	ASTM D4052	0.824	0.814	0.800	0.778
Kinematic viscosity @ 40°C (mm^2/s)	ASTM D445	3.24	3.09	2.90	2.64
Heating value (MJ/kg)	ASTM D240	45.86	46.04	46.38	46.86
Carbon content (%)	ASTM D5291	85.73	85.43	84.98	84.24
Hydrogen content (%)	ASTM D5291	13.22	13.59	14.14	15.05
Oxygen content (%)	ASTM D5599	0.00	0.00	0.00	0.00
Distillation T10 (°C)	ASTM D86-11b	207.7	210.7	216.3	227.4
Distillation T50 (°C)	ASTM D86-11b	287.9	284.5	281.4	278.2
Distillation T90 (°C)	ASTM D86-11b	352.3	345.2	327.4	293.2
Cetane index	ASTM D4737	60.43	63.37	68.32	76.89
Auto-ignition temperature (°C)	ASTM 659	288	-	-	288

3.8 Experimental condition

Table 3.3 shows the experimental condition in this research. Four test fuels were tested: commercial diesel, two commercial diesel-HVO blends by mass: 20% (H20), and 50% (H50) and 100% HVO. The experiment was carried out using constant volume combustion chamber (CVCC) with single-hole injector, 0.2 mm in nozzle orifice diameter, 2.5 ms in energizing time. The injection pressure was kept constant at 100 MPa. The ambient temperature were varied at 1100 K, 900 K and 700 K to study effect of ambient temperatures. The oxygen concentration was varied at 21%, 15% and 10% to simulate effect of EGR conditions. All test were repeated 10 times per each test conditions.

Table 3.3 Experimental condition

Parameters	Conditions
Test fuels	Diesel, H20, H50, HVO
Nozzle orifice diameter	Single hole 0.2 mm
Energizing time	2.5 ms
Injection pressure	100 MPa
Ambient temperature	1100 K, 900 K, 700 K
Oxygen concentration	21%, 15%,10%
Repeat	10 Times / Condition

CHAPTER 4
RESULTS AND DISCUSSION

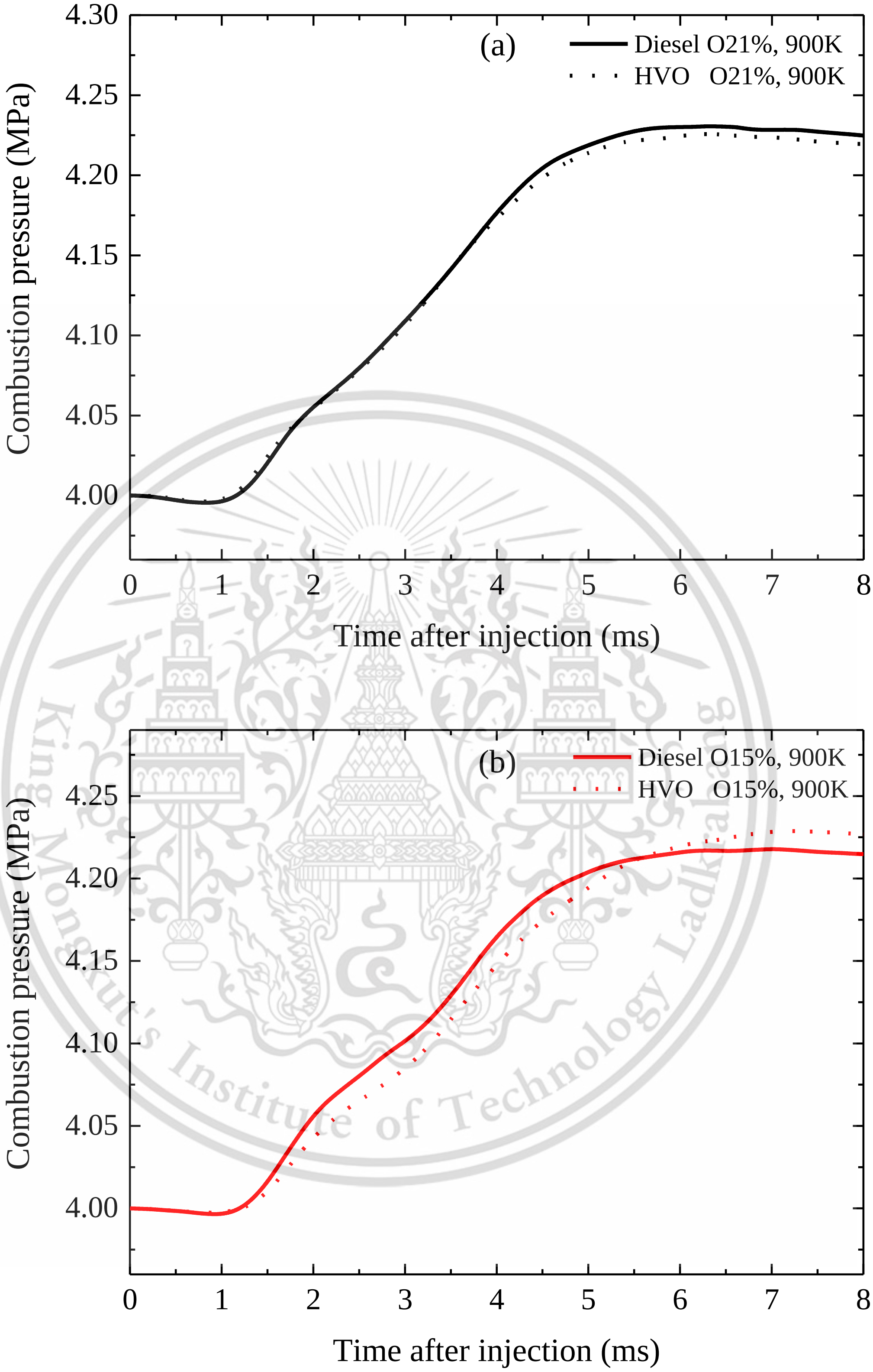
This research on the combustion characteristics of diesel and HVO blend percentage under low ambient oxygen concentration and different ambient temperature were derived into two parts. In the first part, combustion pressure, combustion temperature, heat release rate, cumulative heat release and ignition delay were presented and discussed. In the second part, focusing on the shadowgraph images, which used to describe effect of physical and chemical properties on the spray penetration and the flame development.

4.1 Combustion characteristics

The experimental results of combustion characteristics were presented in terms of combustion pressure and temperature, heat release rate, cumulative heat release and ignition delay.

4.1.1 Combustion pressure

Figure 4.1 show effect of oxygen (O_2) concentrations on the combustion pressure into diesel and HVO as representative under constant injection pressure and ambient temperature. In this experiment, combustion pressure was recorded by the dynamics pressure transducer. The combustion pressure of HVO is lower than diesel for all ambient oxygen concentration. At 15% O_2 concentration, combustion pressure of HVO becomes higher than diesel at 6 ms after injection due to its larger diffusive combustion phase. At 10% O_2 concentration, combustion pressure becomes smoother than other conditions due to the slowed combustion process. This is due to less oxygen availability in the combustion chamber [41]. Furthermore, the lower final combustion pressure had affected to a higher level of soot from combustion [18] as seen as combustion pressure dropped around 9 to 10 ms after injection at 21% O_2 concentration. The conclusion is that the decreased O_2 concentration from 21% to 15% and 10% leads to decrease in the rate of combustion pressure by 9.08% and 29.58% then this will decrease heat release rate. In addition, the decreased O_2 concentration from 21% to 15% and 10% increased the final combustion pressure by 0.12% and 0.26%, resulting in decreasing the level of soot from combustion.



(a)
Diesel O21%, 900K
HVO O21%, 900K
Combustion pressure (MPa)
Time after injection (ms)
(b)
Diesel O15%, 900K
HVO O15%, 900K
Combustion pressure (MPa)
Time after injection (ms)

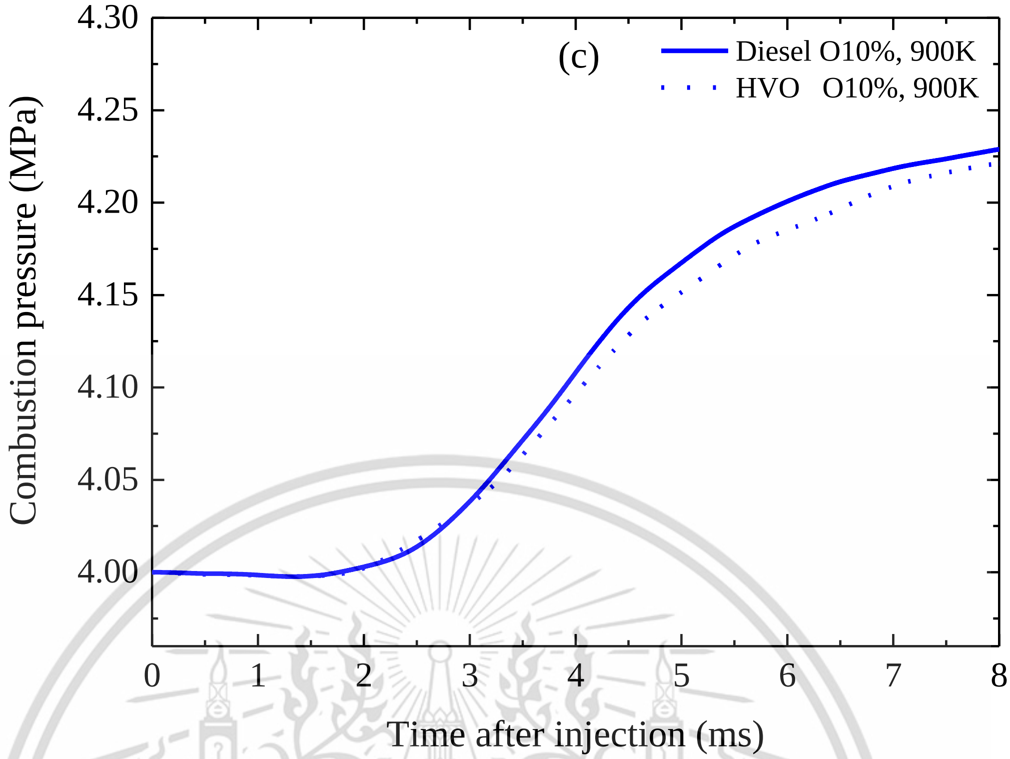
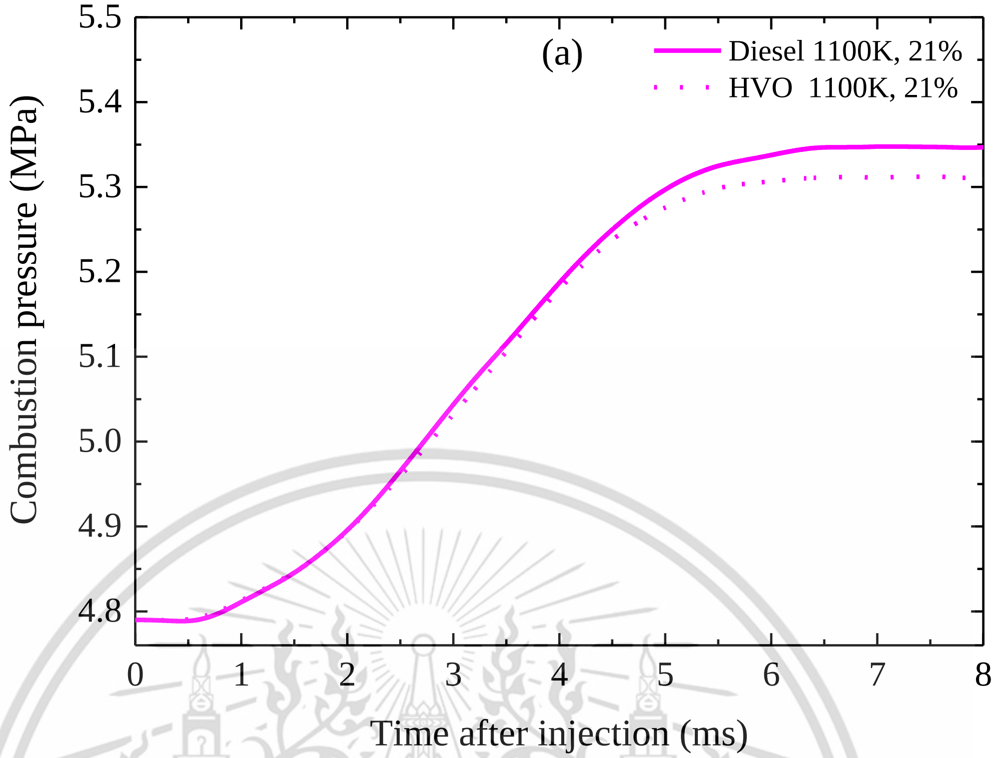
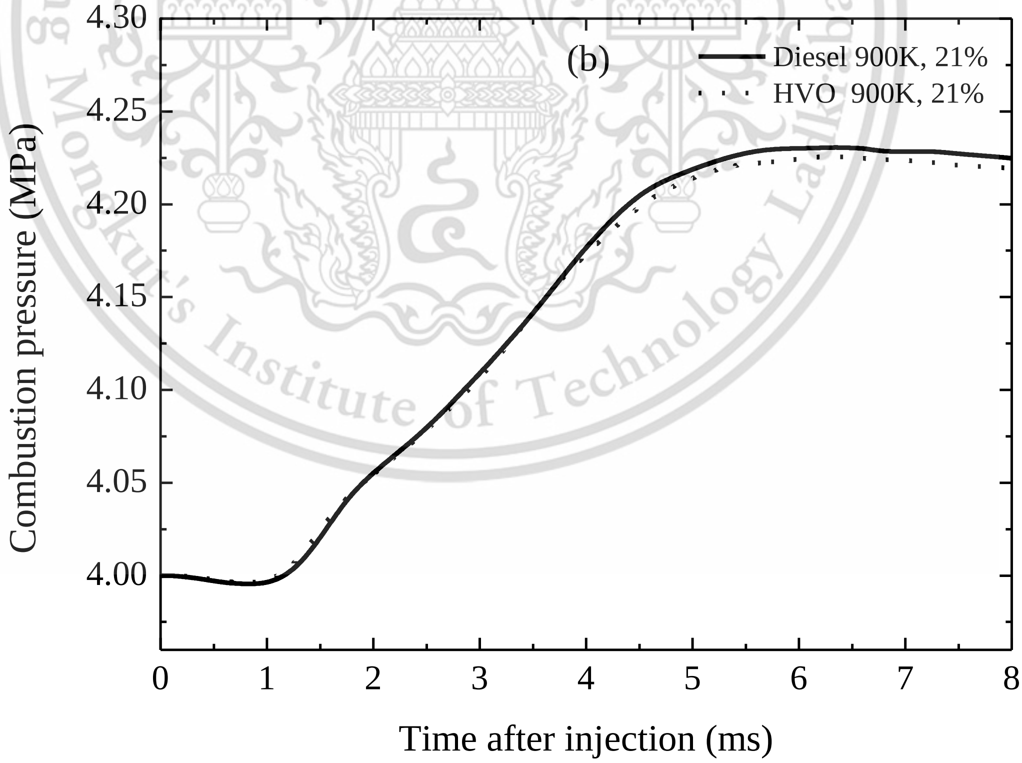


Figure 4.1 Effect of oxygen concentrations on combustion pressure (a) 21%, (b) 15% and (c) 10%

Figure 4.2 show effect of ambient temperatures on the combustion pressure into diesel and HVO as representative under constant injection pressure and oxygen concentration. Combustion pressure of 1100 K and 900 K of diesel is higher than HVO. The rate of combustion pressure of diesel is also higher than diesel at 1100 K and 900 K but at 700 K, combustion pressure of HVO is higher than diesel. Moreover, the rate of combustion pressure of HVO rapidly increases by 14.20% due to the longer ignition delay. The conclusion is that the increased ambient temperature 900 K to 1100 K leads to higher rate of combustion pressure due to incomplete fuel-air mixing [52], On the contrary, the rate of combustion at 700 K increases 6.46% compared to 900 K because the injected fuel is well mixed with ambient air in the chamber.


(a)
Diesel 1100K, 21%
HVO 1100K, 21%
Combustion pressure (MPa)
Time after injection (ms)
5.5
5.4
5.3
5.2
5.1
5.0
4.9
4.8
0
1
2
3
4
5
6
7
8


(b)
Diesel 900K, 21%
HVO 900K, 21%
Combustion pressure (MPa)
Time after injection (ms)
4.30
4.25
4.20
4.15
4.10
4.05
4.00
0
1
2
3
4
5
6
7
8

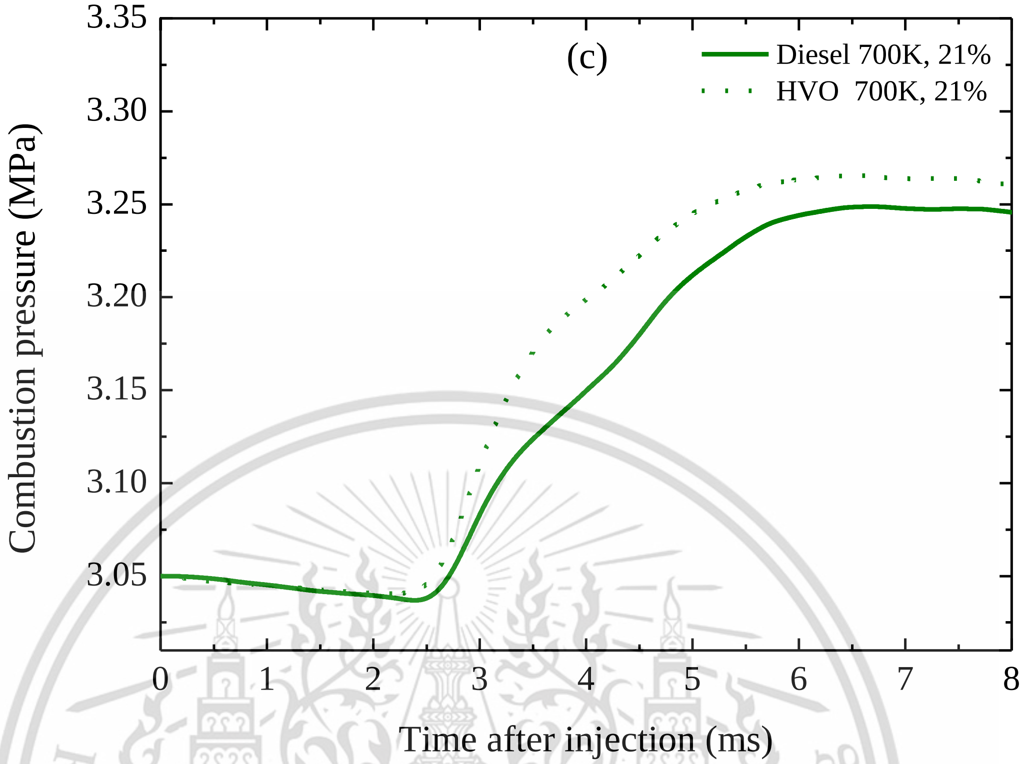
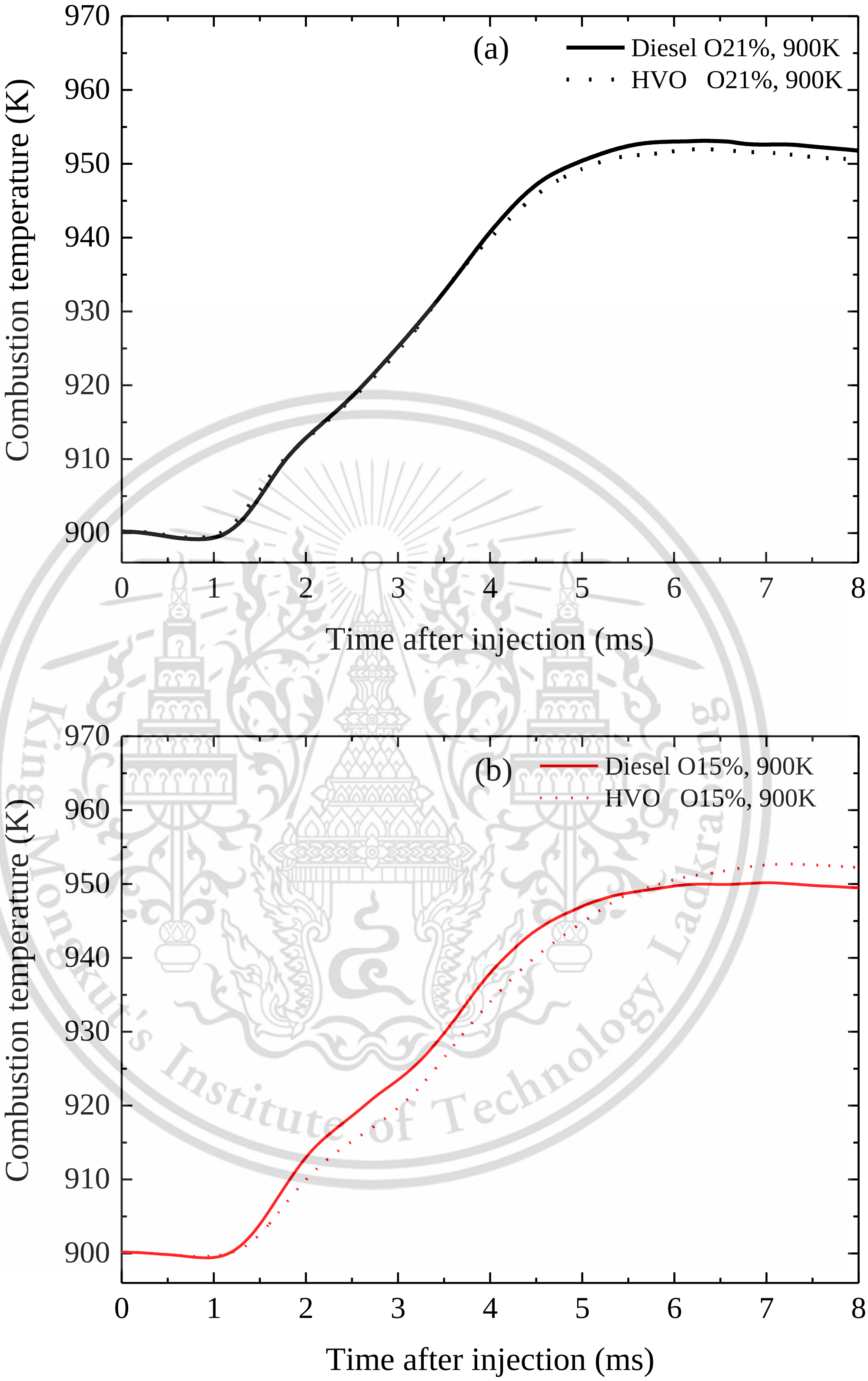


Figure 4.2 Effect of ambient temperatures on combustion pressure (a) 1100 K, (b) 900 K and (c) 700 K

4.1.2 Combustion temperature

Figure 4.3 show effect of oxygen (O_2) concentrations on the combustion temperature into diesel and HVO as representative under constant injection pressure and ambient temperature. Combustion temperature was calculated from combustion pressure by using Equation 3.3 as described in Chapter 3. Combustion temperature of 21%, 15% and 10% O_2 concentration showed the same trend as combustion pressure. The lower combustion pressure causes the lower chamber gas temperature, which might contribute to reduce NO_x emission [52]. The conclusion is that the decreased O_2 concentration from 21% to 15% and 10% results in decreasing the rate of combustion temperature by 9.08% and 29.58%. In addition, lowering combustion temperature contribute to lower NO_x emission.



970
960
950
940
930
920
910
900
Combustion temperature (K)
(a)
Diesel O21%, 900K
HVO O21%, 900K
0
1
2
3
4
5
6
7
8
Time after injection (ms)
970
960
950
940
930
920
910
900
Combustion temperature (K)
(b)
Diesel O15%, 900K
HVO O15%, 900K
0
1
2
3
4
5
6
7
8
Time after injection (ms)

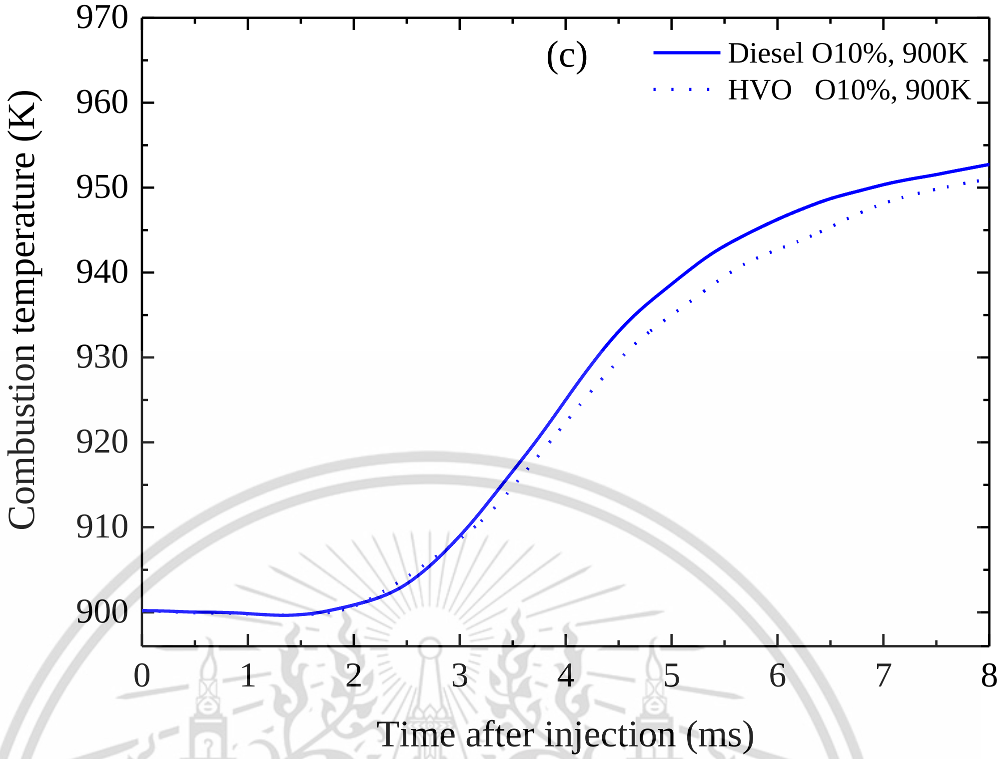
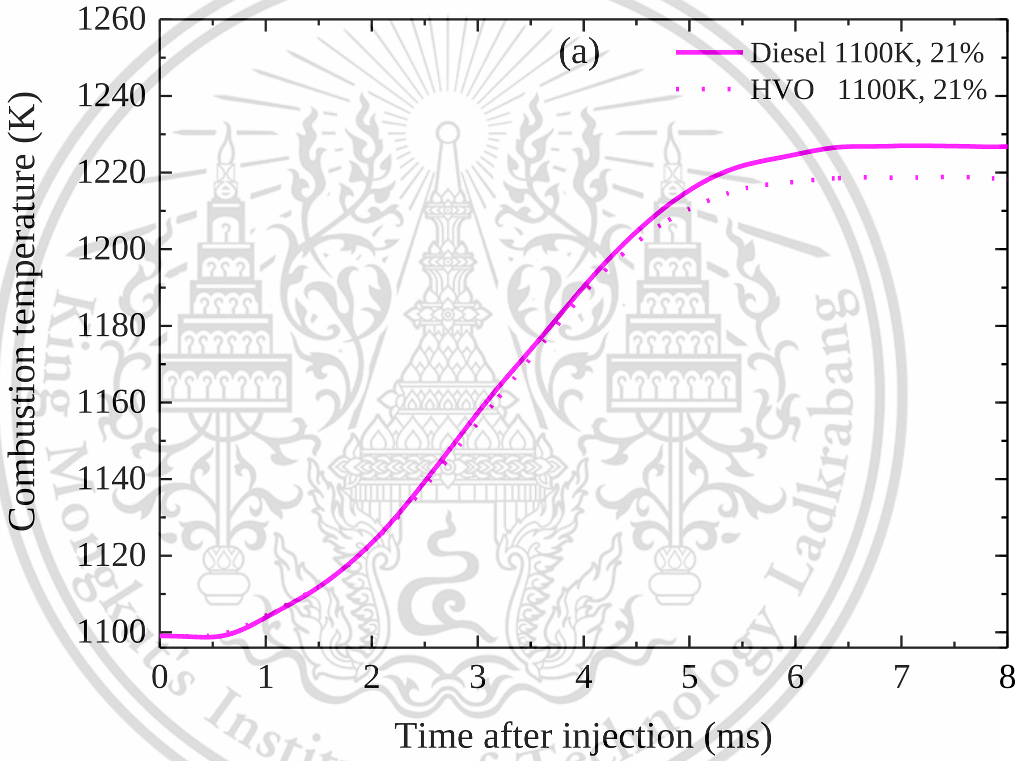


Figure 4.3 Effect of oxygen concentrations on combustion temperature (a) 21%, (b) 15% and (c) 10%

Figure 4.4 show effect of ambient temperatures on the combustion temperature into diesel and HVO as representative under constant injection pressure and oxygen concentration. Combustion temperature of 1100 K, 900 K and 700 K showed the same trend as combustion pressure. The lower combustion pressure causes the lower chamber gas temperature, which might contribute to reduce NO_x emission [52]. The conclusion is that increasing 900 K to 1100K leads to lower the rate of combustion temperature by 6.77%. On the contrary, the rate of combustion at 700 K increased 6.46% compare to 900 K because fuel and air well mixed and distributed.



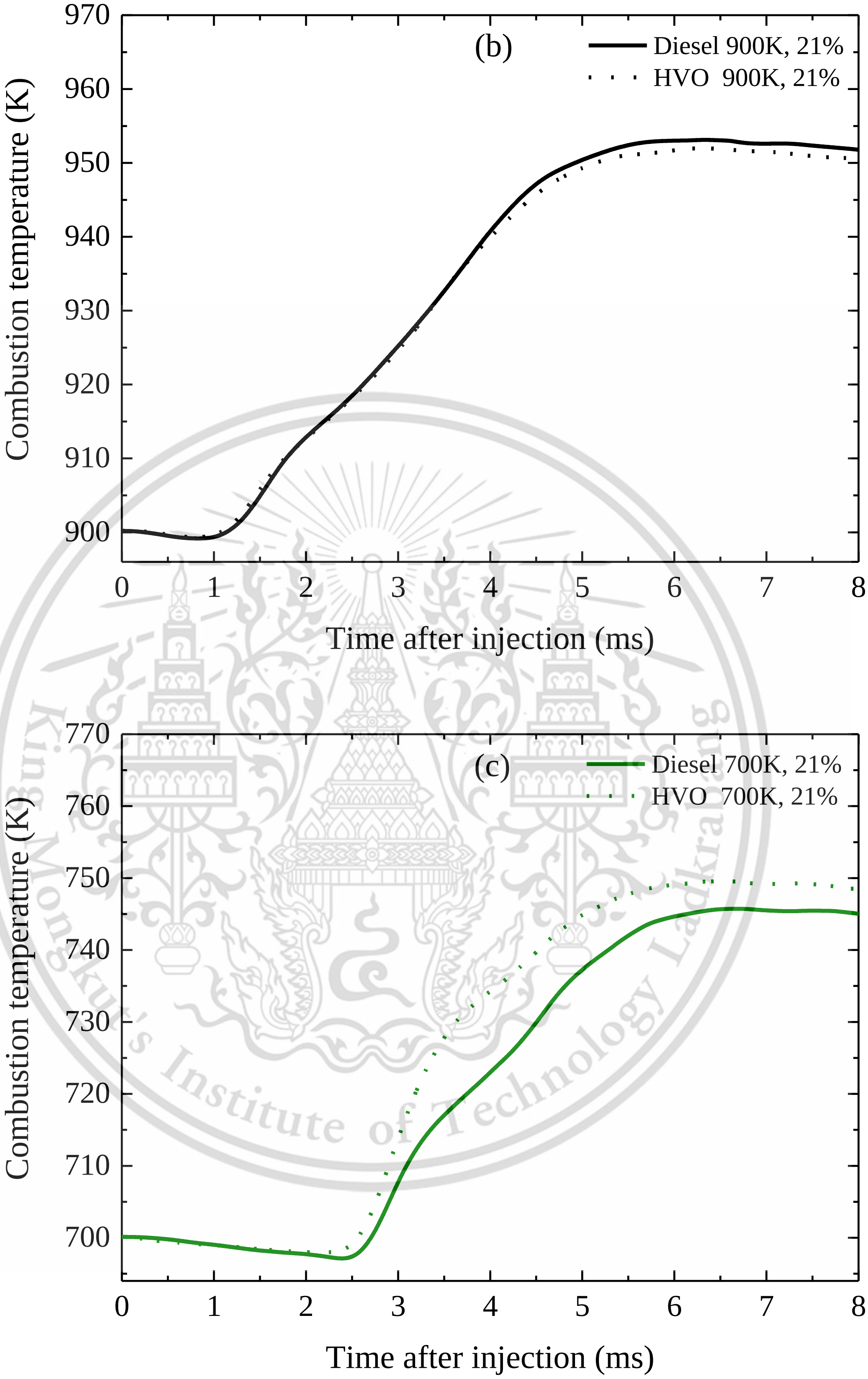


Figure 4.4 Effect of ambient temperatures on combustion temperature (a) 1100 K, (b) 900 K and (c) 700 K

4.1.3 Apparent heat release rate

Figure 4.5 shows effect of oxygen (O_2) concentrations on the apparent heat release rate into diesel and HVO as representative under constant injection pressure and ambient temperature. Heat release rate is determined from pressure rise by using Equation 3.2 as described in Chapter 3. The peak heat release rate of 21% and 15% O_2 concentration of HVO is lower than diesel due to its higher cetane number, which makes a shorter ignition delay [41]. Both 21% and 15% O_2 concentration exhibit a single-stage ignition, which is hot flame [25], premixed combustion phase, and following by diffusive combustion phase, respectively. Lower O_2 concentration produced the slowed combustion rate, indicating by the change of heat release rate slope and lower peak heat release rate [53]. Heat release rate of 10% O_2 concentration shows the lowest peak heat release due to the lower flame temperature caused by the diluted ambient gas [41], [52]. At 10% O_2 concentration exhibits the two-stage ignition. The two-stage ignition is commonly observed in the diesel combustion, the first-stage ignition is “low temperature heat release (LTHR)” and the second stage is “high temperature heat release (HTHR)” due to the slowed combustion process [25]. The LTHR were relatively similar with different O_2 concentration. On the other hand, The HTHR is shifted with different O_2 concentration [46].

Figure 4.6 shows effect of ambient temperatures on the apparent heat release rate into diesel and HVO as a representative under constant injection pressure and oxygen concentration. The heat release rate at 1100 K exhibits the lowest heat release rate because fuel and air are fast vaporized and mixed during the ignition delay period [26] as seen as the dip of heat absorption [20]. The peak heat release rate decreases with increasing ambient temperature to 1100 K due to the less time of fuel-air mixing. In addition, the shorter ignition delay is observed with 1100 K, resulting in suppress the premixed combustion period and lengthen the diffusive combustion period [19]. At 700 K, the heat release rate increases rapidly in the premixed combustion phase because of fuel and air well mixed and distributed during the ignition delay period [52], [54]. The heat release rate of HVO exhibits the highest heat release rate due to the improved mixing of fuel and air [54]. From the results, the conclusion is that HVO promotes shorter ignition delay in the combustion chamber at low ambient temperature and contributes to use in the diesel engine at low engine load, cold engine conditions [13], [35] and low compression ratio [22].

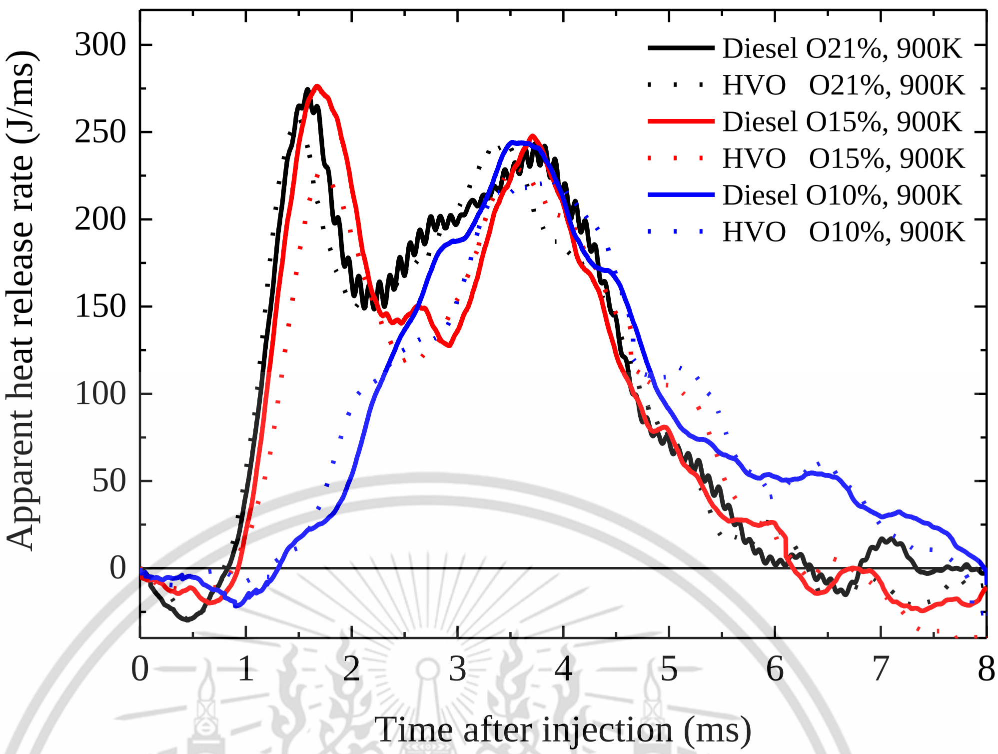


Figure 4.5 Effect of oxygen concentrations on apparent heat release rate of diesel and HVO as representative

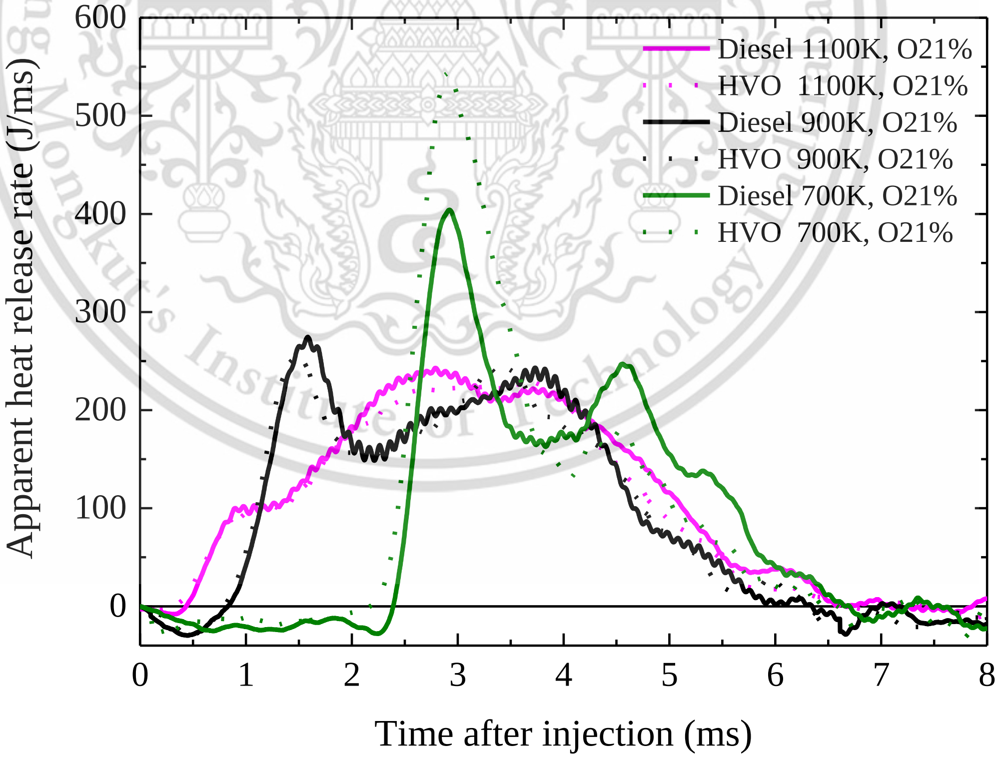


Figure 4.6 Effect of ambient temperatures on apparent heat release rate of diesel and HVO as representative

4.1.4 Cumulative heat release

Figure 4.7 shows effect of oxygen (O_2) concentrations on the cumulative heat release into diesel and HVO as a representative under constant injection pressure and ambient temperature. The 21% O_2 concentration shows the highest value of cumulative heat release at 6 ms after injection because the higher oxygen availability causes an increase in the burning rate [20], indicating by the change of cumulative heat release slope. The 21% and 15% O_2 concentration show similar cumulative heat release and begin to decrease at 7 ms time after injection. On the other hand, cumulative heat release of 10% O_2 concentration still slowly increases due to the slowed burning rate.

Figure 4.8 shows effect of ambient temperatures on the cumulative heat release into diesel and HVO as a representative under constant injection pressure and oxygen concentration. The 1100 K is observed early increase in cumulative heat release due to a shorter ignition delay. Burning rate of 1100 K exhibits similar trend to 900 K, but at 700 K exhibits the higher burning rate due to promoting long mixing time that makes better mixture formation at lower temperature [41], resulting in higher heat release rate.

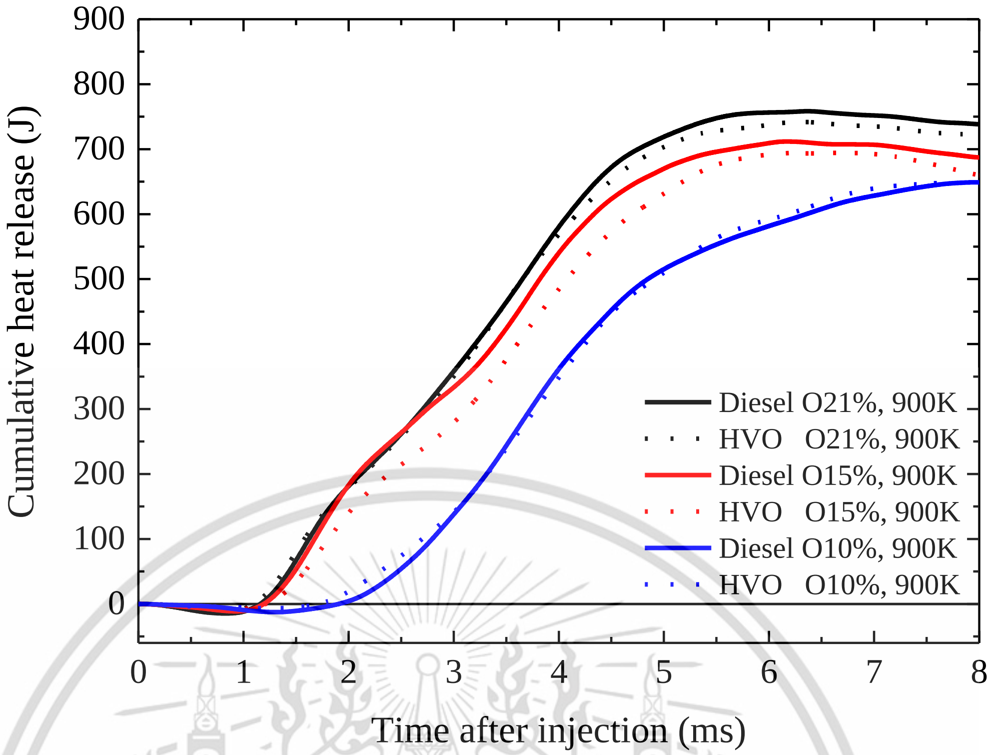


Figure 4.7 Effect of oxygen concentrations on cumulative heat release of diesel and HVO as representative

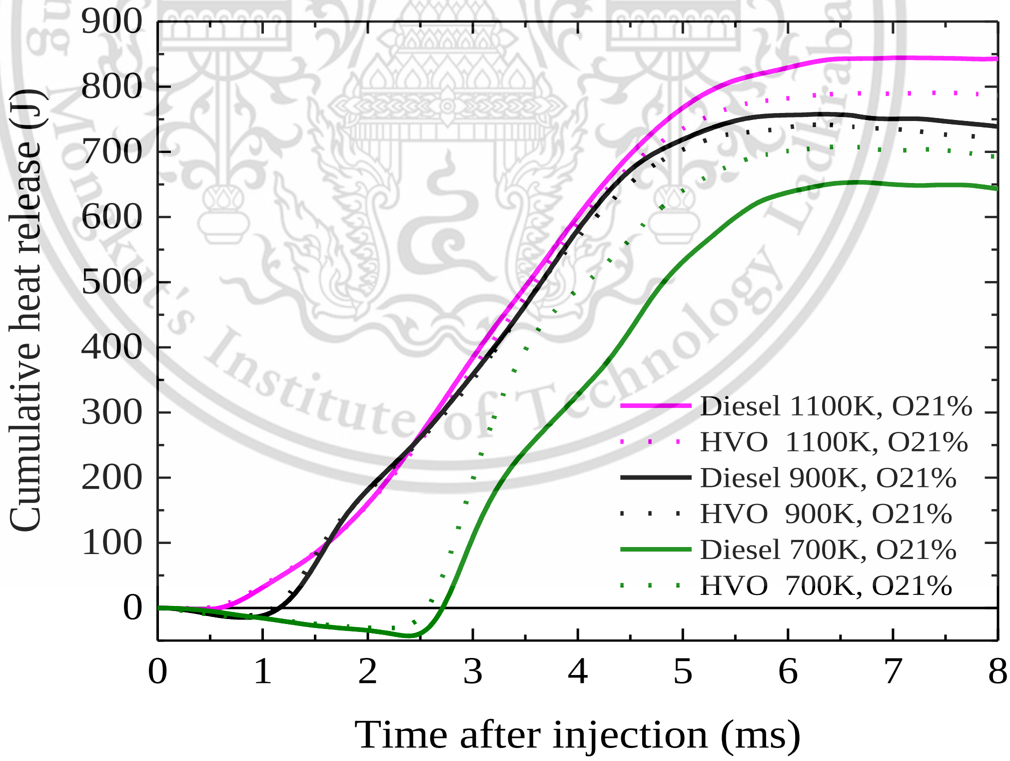


Figure 4.8 Effect of ambient temperatures on cumulative heat release of diesel and HVO as representative

4.1.5 Ignition delay

Figure 4.9 shows effect of oxygen (O_2) concentrations on the ignition delay into diesel and HVO blend percentage under constant injection pressure and ambient temperature. From this figure, the ignition delay of 21% and 15% O_2 concentration exhibit similar trend for all test fuels. On the other hand, the 10% O_2 concentration shows the longest ignition delay because of the slowed chemical kinetics due to the less of oxygen availability [41], [54]. The ignition delay of H20, H50 and HVO exhibit 0.72%, 6.0% and 6.24% shorter than diesel at O_2 concentration of 21% due to its higher cetane number [6], [33]–[35], [53] and lower T90% temperature [26], [55]. The ignition delay of HVO exhibits shorter than diesel by 6.24%, 5.42% and 9.63% at O_2 concentration of 21%, 15% and 10%, respectively. The conclusion is that decreasing oxygen concentration results in increasing ignition delay due to the less of oxygen availability. Increasing HVO blend percentage leads to shortening ignition delay for all test condition due to its higher cetane number.

Figure 4.10 shows effect of ambient temperatures on the ignition delay into diesel and HVO blend percentage under constant injection pressure and oxygen concentration. From this figure, the ignition delay of 1100 K and 900 K exhibit similar trend for all test fuels. On the other hand, the 700 K ambient temperature shows the longest ignition delay because fuel slowly evaporated, mixed with the ambient air [18], [20], [53]. The ignition delay of HVO exhibits shorter than diesel by 2.26%, 6.24% and 8.84% at 1100 K, 900 K and 700 K, respectively due to its higher cetane number [6], [33]–[35], [53]. The conclusion is that decreasing ambient temperature results in increasing ignition delay due to a lower fuel evaporation process.

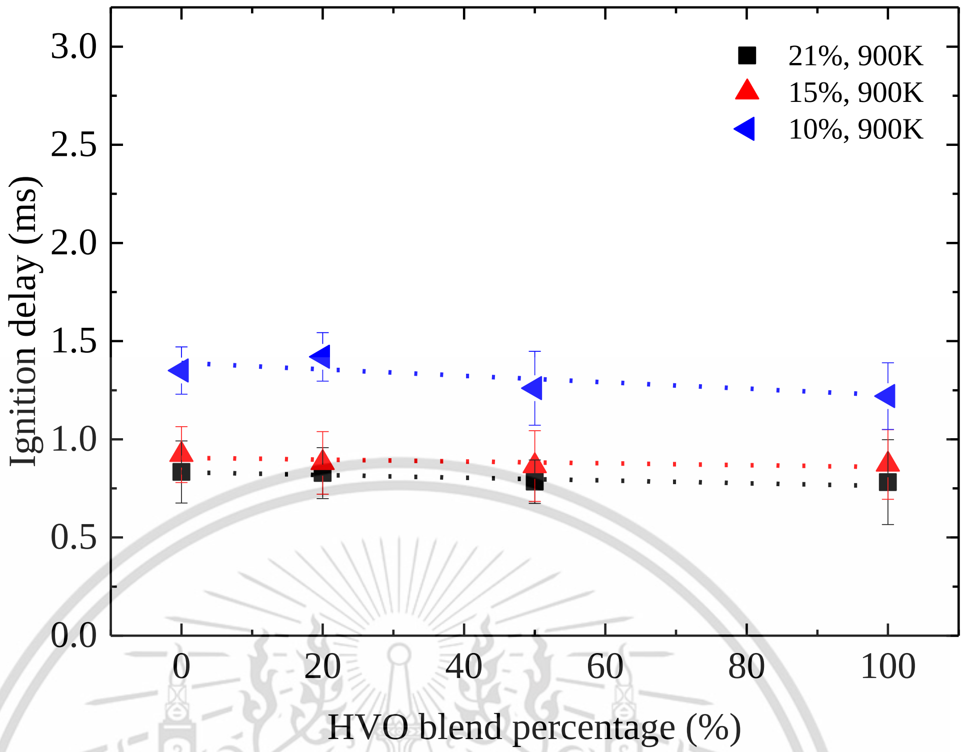


Figure 4.9 Effect of oxygen concentrations on ignition delay of diesel and HVO blend percentage

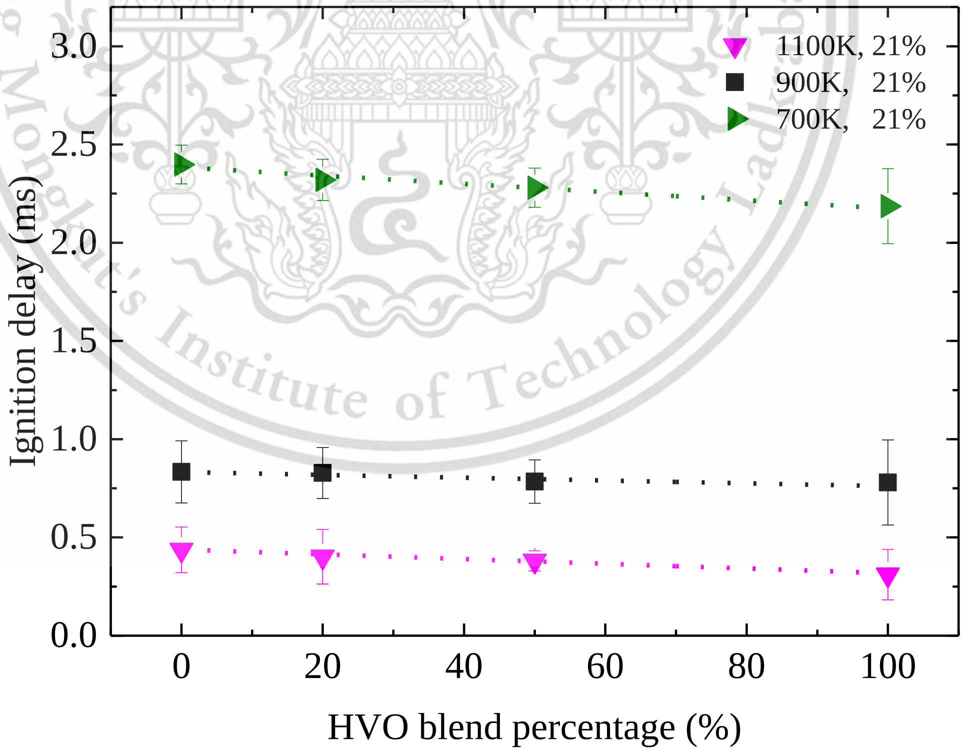


Figure 4.10 Effect of ambient temperatures on ignition delay of diesel and HVO blend percentage

4.2 Visualization of combustion

In this part, focusing on the combustion visualization of fuel. Results of combustion visualization were presented; first, the shadowgraph images of spray development describe effect of ambient conditions and fuel properties on the mixture formation and the ignition delay; second, the shadowgraph images of combustion progress describe effect of ambient conditions on the flame development. The sequence shadowgraph images of spray combustion were captured by using the high-speed video camera, and analyzed by using Photron Fastcam Viewer (PFV) program.

4.2.1 Spray development

1. Effect of oxygen concentrations

Figure 4.13, 4.14 and 4.15 illustrate the shadowgraph images of spray development of diesel, H20, H50 and HVO under effects of oxygen (O_2) concentration. All test fuels continue to penetrate, evaporate and distribute with the ambient gas in the chamber. The effect of various O_2 concentrations have an insignificant on spray penetration due to the same of ambient gas density and temperature [33], [61]. The 21% O_2 concentration exhibits early fuel evaporated at 0.9 ms to 1.0 after injection due to oxygen enhancement [26]. The 15% and 10% O_2 concentration exhibit similar spray penetration compared to 21% O_2 centration case, but it required more time of mixing, distribution, and ignition because its oxygen availability is lower than the O_2 concentration of 21% as observed during 0.8 ms to 1.0 ms after injection [41], [53]. The evaporation of HVO is quicker than diesel as it observed larger vaporized fuel around the tip of spray during 0.5 ms to 0.9 ms after injection at 21% and 15% % O_2 concentration due to its lower viscosity, density, and distillation temperature at T90, resulting in easier fuel atomized, distributed and mixed with the ambient air [26], [34], [55]. Moreover, HVO has already ignited at 0.9 ms after injection. The previous research on HVO showed that HVO is a mixture of paraffin and isomerize paraffin with straight chain-length [11] [30], [33], [57], which might contribute to easier broken up and ignition compared to diesel.

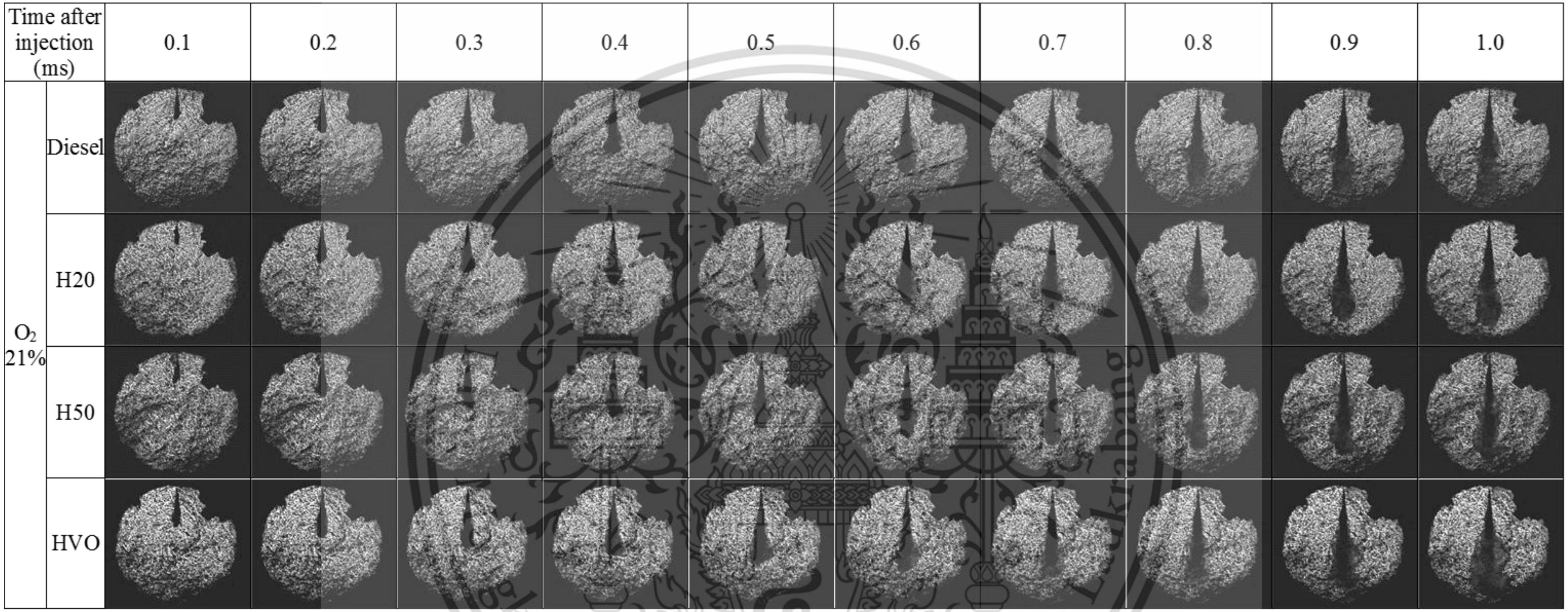


Figure 4.13 Shadowgraph images of spray development of diesel, H20, H50 and HVO at 21% oxygen concentration and 900 K ambient temperature

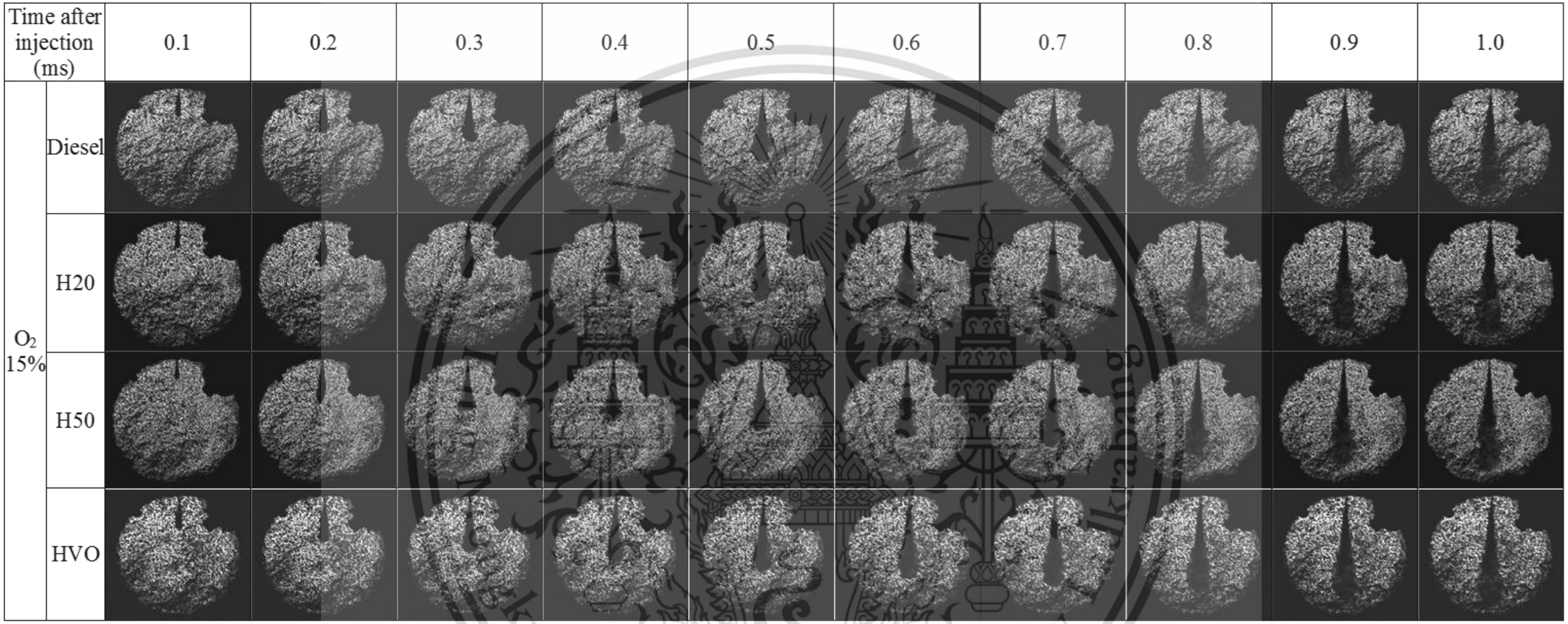


Figure 4.14 Shadowgraph images of spray development of diesel, H20, H50 and HVO at 15% oxygen concentration and 900K ambient temperature

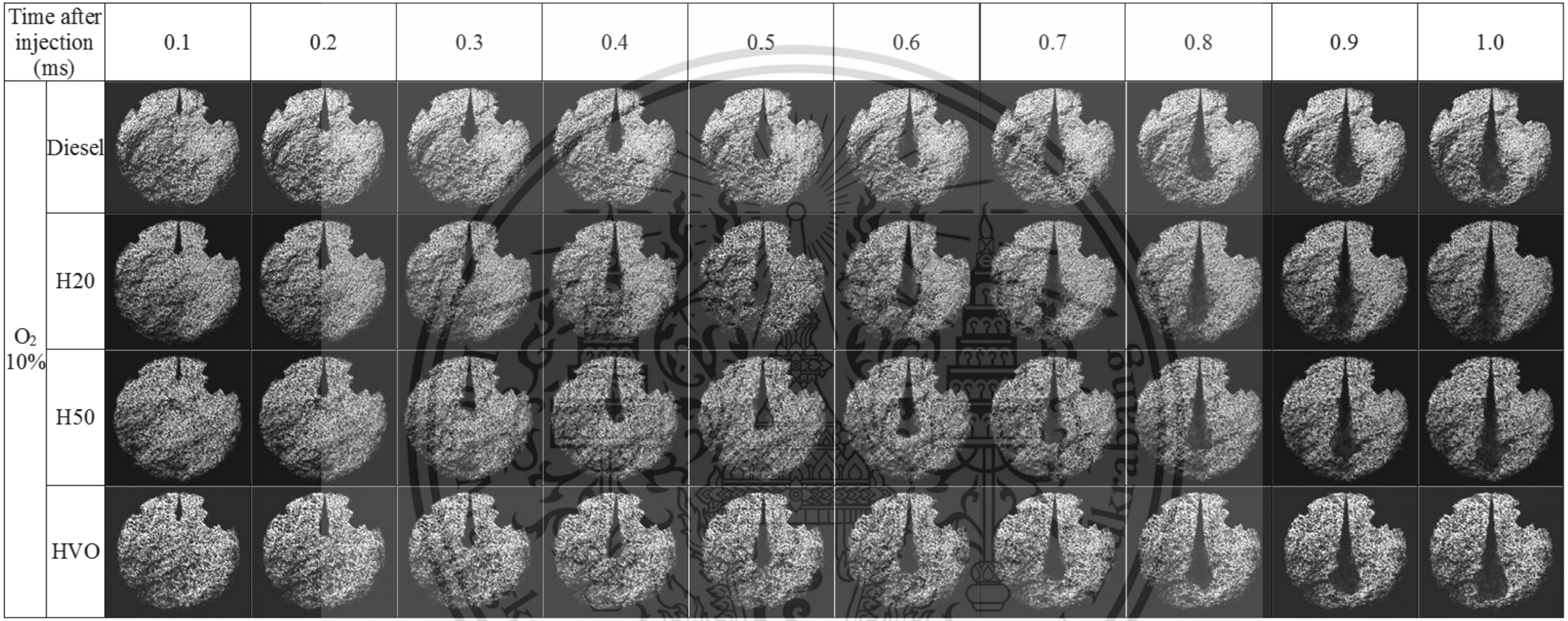


Figure 4.15 Shadowgraph images of spray development of diesel, H20, H50 and HVO at 10% oxygen concentration and 900 K ambient temperature

2. Effect of ambient temperatures

Figure 4.16, 4.17 and 4.18 illustrate the shadowgraph images of spray development diesel, H20, H50 and HVO under effect of ambient temperatures. From this figure, the effect of various ambient temperatures have largely impacted on spray penetration due to different ambient gas density and temperature [63]. The 1100 K exhibits early ignition at 0.6 ms after injection due to a rapid fuel evaporation [55], the less time of fuel-air mixing and higher combustion rate [26], [53]. Lower ambient temperature has more time for spray mixing, more ambient air entrained into fuel spray to obtain the better mixture formation, resulting in higher the peak heat release rate [43], [53], [54]. All test fuels ready ignite at 0.6 ms after injection. The comparison on spray development can be explained that HVO exhibits faster evaporate, distributed and mixed with ambient air compared to diesel for all ambient temperature as observe larger fuel evaporated around the tip of spray at 0.5 ms after injection while H20 and H50 exhibit similar to diesel.

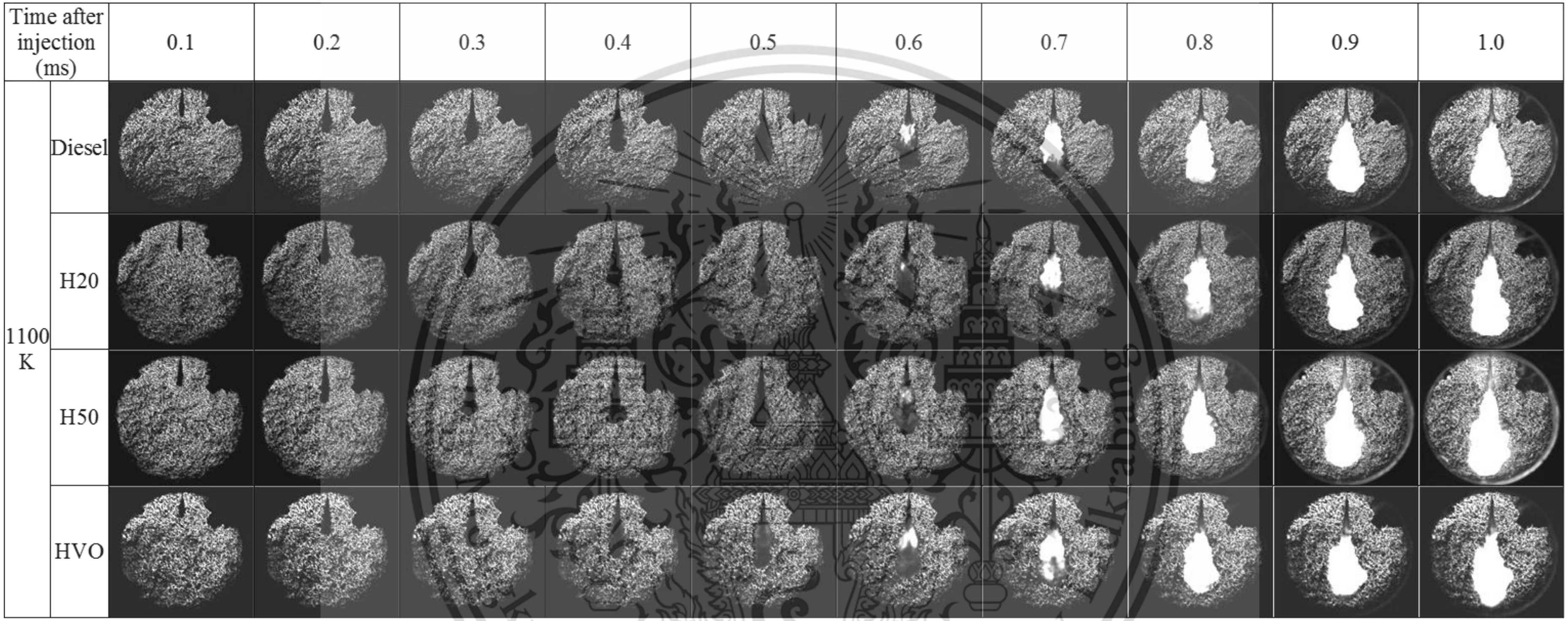


Figure 4.16 Shadowgraph images of spray development of diesel, H20, H50 and HVO at 1100 K ambient temperature and 21% oxygen concentration

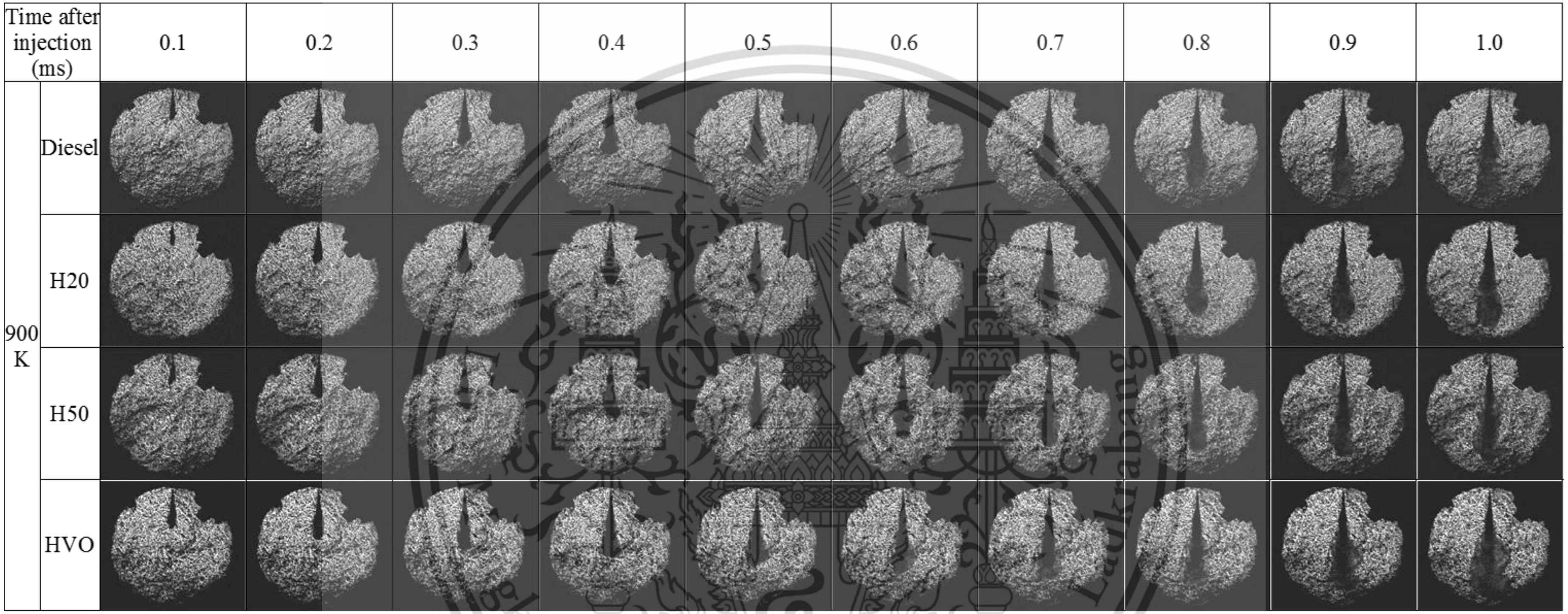


Figure 4.17 Shadowgraph images of spray development of diesel, H20, H50 and HVO at 900 K ambient temperature and 21% oxygen concentration

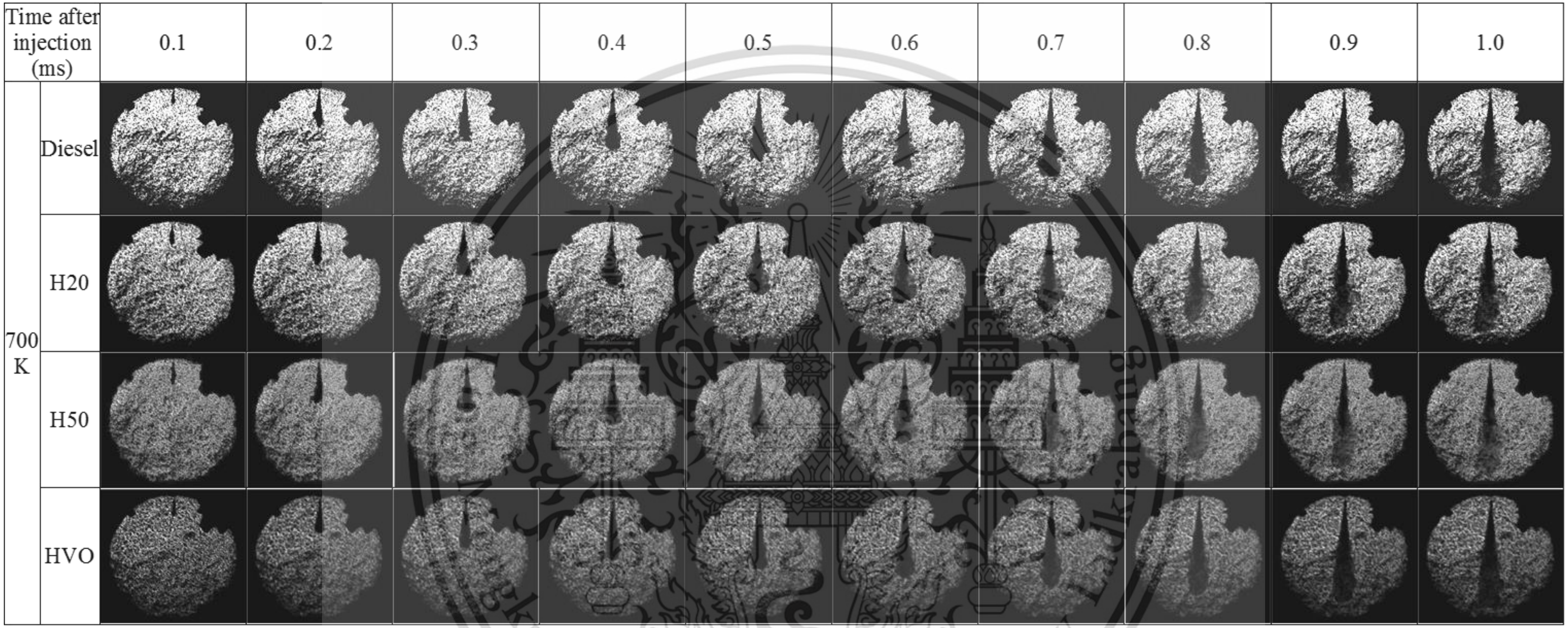


Figure 4.18 Shadowgraph images of spray development of diesel, H20, H50 and HVO at 700 K ambient temperature and 21% oxygen concentration

4.2.2 Combustion progress

1. Effect of oxygen concentrations

Figure 4.19, 4.20 and 4.21 illustrate the shadowgraph images of flame development of diesel, H20, H50 and HVO under effect of oxygen (O_2) concentrations. The flame development of test fuels were evaluated from 1.0 ms to 5.5 ms after injection. The injection was ended around 4.0 ms after injection for each conditions and each test fuels. The 21% O_2 concentration exhibits early the luminous flame at 1.5 ms after injection due to the higher oxygen availability in the chamber [53]. At 15% O_2 concentration, the luminous flame is observed around 2.5 ms after injection. The brighter flame luminosity is observed during 1.5 ms to 5.0 ms after injection for 21% O_2 concentration. Meanwhile for 15% O_2 concentration can observe during 3.5 ms to 5.5 ms after injection. On the contrary, the luminous flame is not observed at 10% O_2 concentration due to the lack of oxygen, which results in the slowed chemical reaction [54] and decreasing combustion reaction intensity [18], [20]. The comparison of flame luminosity can be explained that the flame luminosity of diesel is slightly brighter than H20, H50 and HVO. Moreover, HVO and H50 exhibits faster flame development at 21% O_2 concentration compared to H20 and diesel due to smaller droplet size distribution [64], which made fuel-air interaction very well during ignition delay through combustion process. This results is agreement with previous research [43].

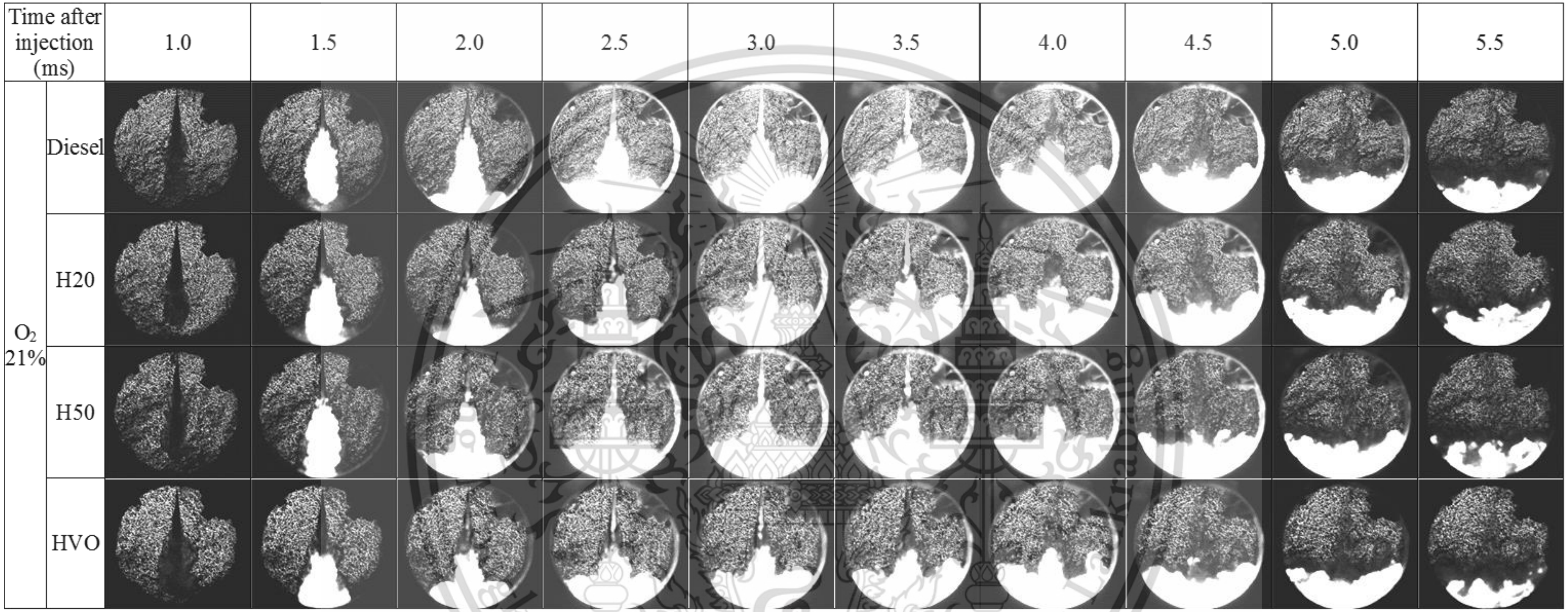


Figure 4.19 Shadowgraph images of flame development of diesel, H20, H50 and HVO at 21% oxygen concentration and 900 K ambient temperature

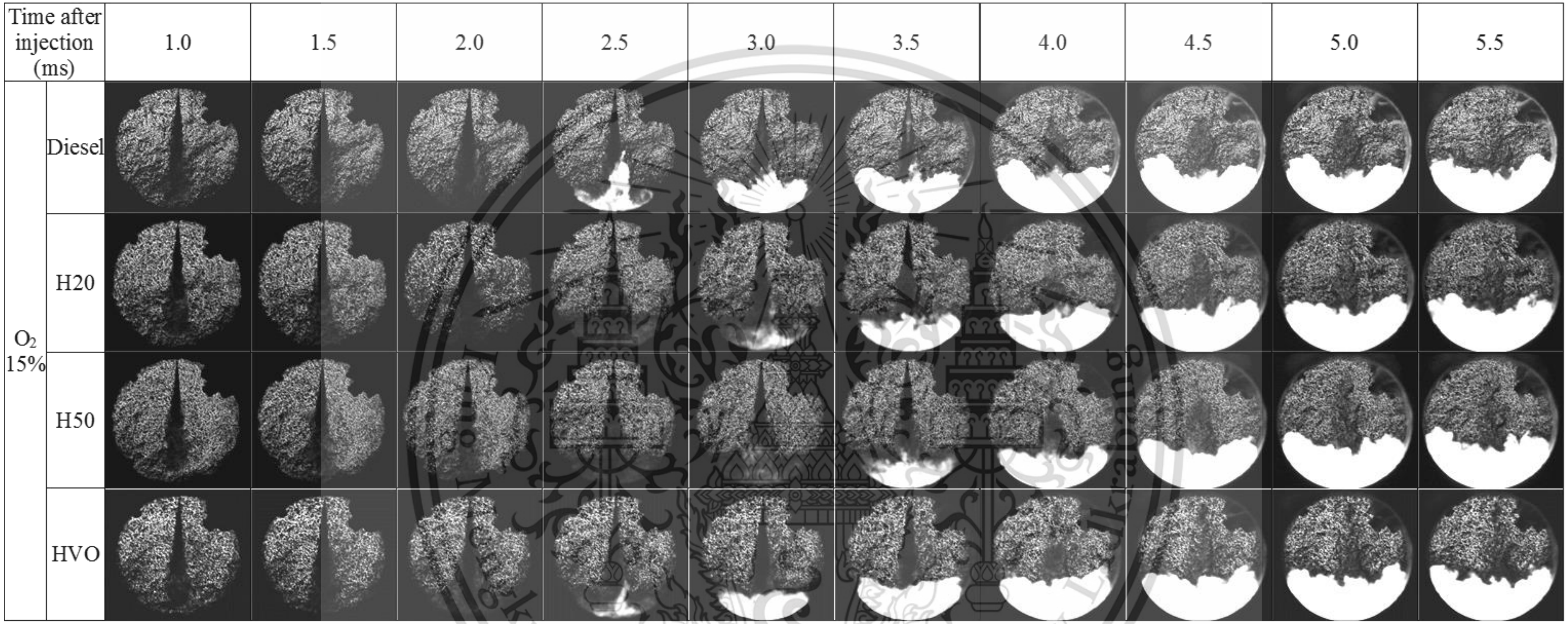


Figure 4.20 Shadowgraph images of flame development of diesel, H20, H50 and HVO at 15% oxygen concentration and 900 K ambient temperature

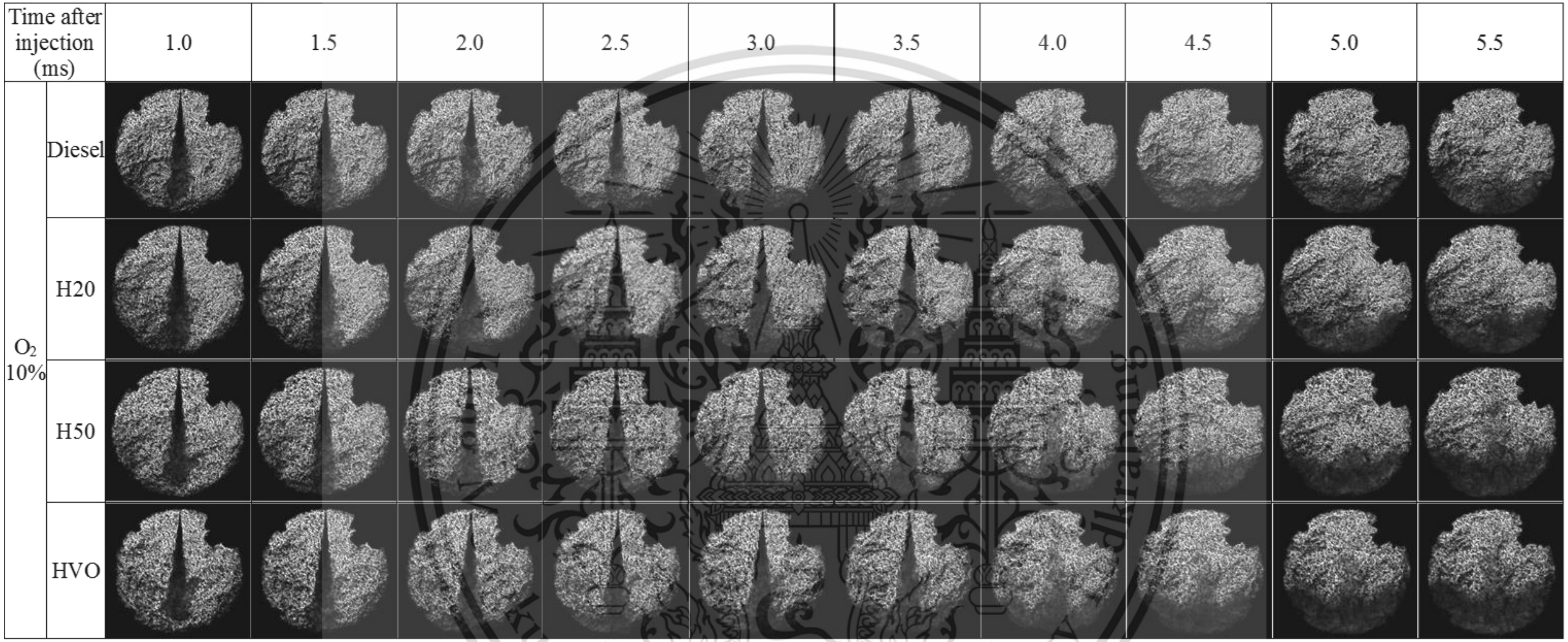


Figure 4.21 Shadowgraph images of flame development of diesel, H20, H50 and HVO at 10% oxygen concentration and 900 K ambient temperature

2. Effect of ambient temperatures

Figure 4.16 illustrates the heat release rate and shadowgraph images of flame development of diesel, H20, H50 and HVO under effect of ambient temperatures. The 1100 K is observed the brighter flame compared to 900 K and 700 K at 1.0 ms to 5.0 after injection due to the rapid fuel evaporation in the chamber. The flame luminosity of 900 K exhibits later than 1100 K due to the lower temperature. The flame luminosity can obtain for each fuels at 1100 K and 900 K. On the contrary, At 700 K, the flame luminosity appears only in case of H50 and HVO because increasing cetane number provides high reactivity at low temperature [34], [46]. In addition, HVO exhibits the luminous flame of 700 K at downstream spray because it has better mixture formation due to its higher cetane number and lower distillation temperature at T90.

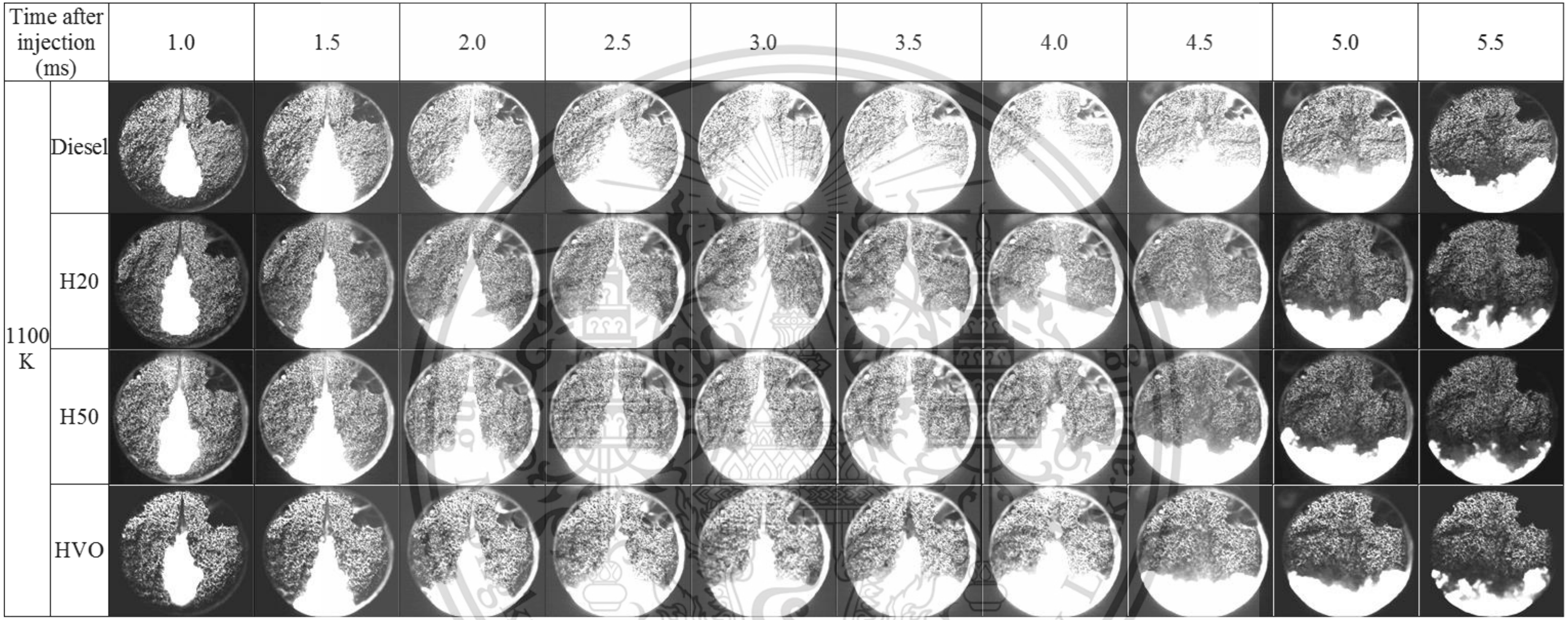


Figure 4.22 Shadowgraph images of flame development of diesel, H20, H50 and HVO at 1100 K ambient temperature and 21% oxygen concentration

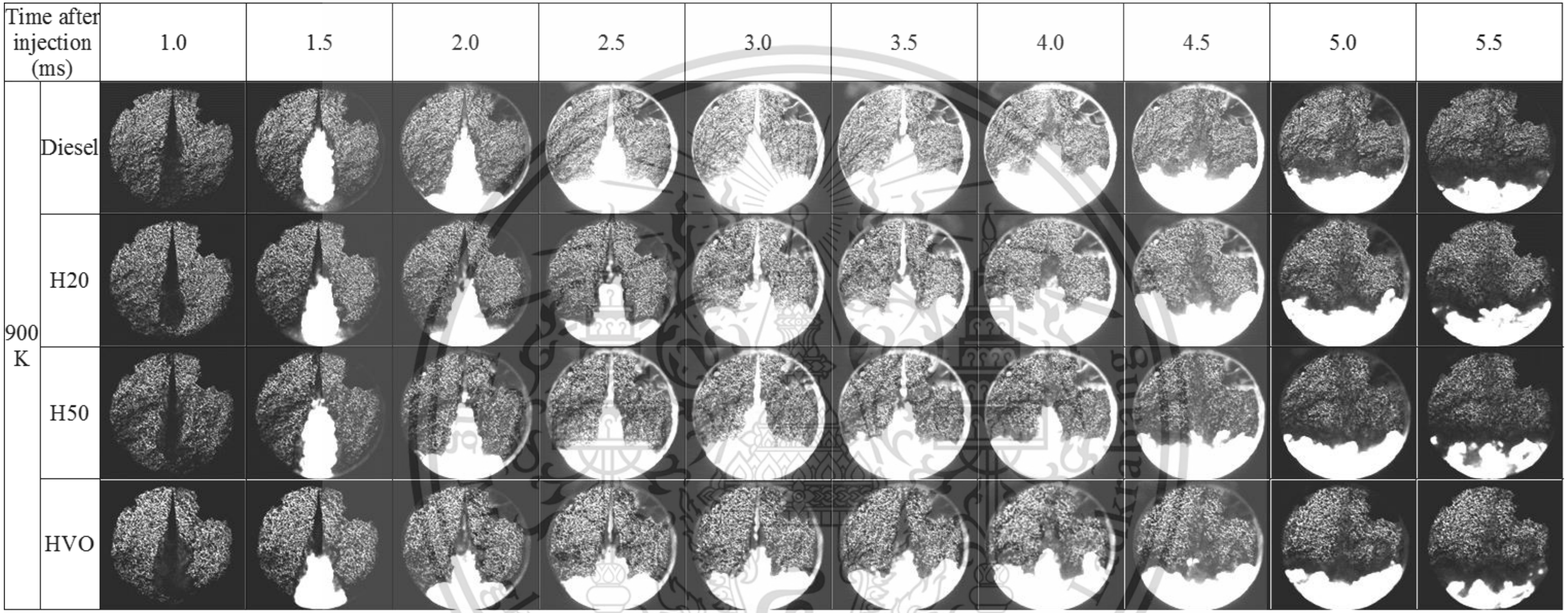


Figure 4.23 Shadowgraph images of flame development of diesel, H20, H50 and HVO at 900 K ambient temperature and 21% oxygen concentration

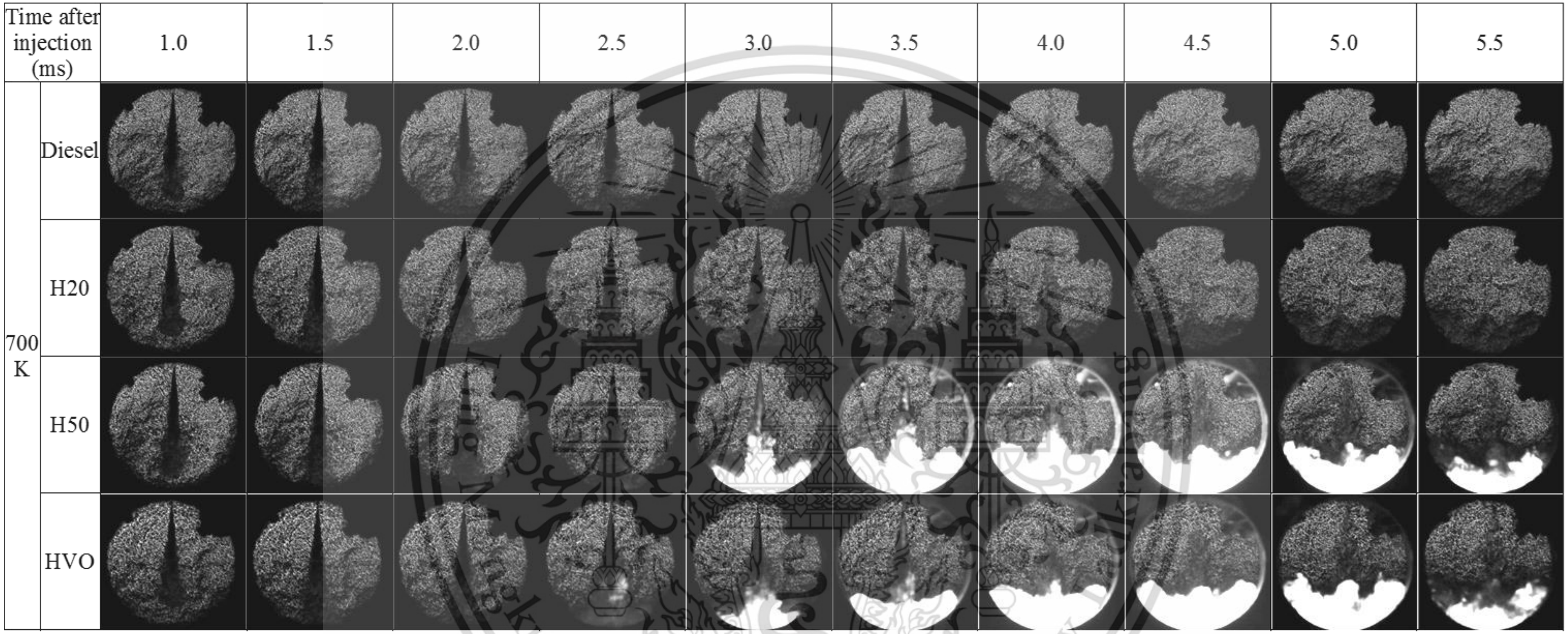


Figure 4.24 Shadowgraph images of flame development of diesel, H20, H50 and HVO at 700 K ambient temperature and 21% oxygen concentration

CHAPTER 5
CONCLUSION

5.1 Conclusion

This research investigated effect of Hydrotreated vegetable oil – diesel blend percentage on spray combustion characteristics under low ambient oxygen concentration and different ambient temperature. The conclusion were derived into two parts. In the first part, combustion pressure, combustion temperature, heat release rate, cumulative heat release and ignition delay were presented. In the second part, focusing on the shadowgraph images of spray penetration and flame development.

5.1.1 Combustion characteristics

1. The peak heat release rate of HVO was lower than diesel for all ambient condition, but an exception in ambient temperature of 700 K. A decrease of oxygen concentration resulted in lower heat release rate, longer ignition delay and more premixed combustion. Meanwhile, a decrease of ambient temperature also resulted in longer ignition delay, but rapid premixed combustion with higher heat release rate due to better mixture formation.

2. Higher cetane number of HVO and blends resulted in shorter ignition delay for all ambient condition. Ignition delay is changed with ambient oxygen concentration due to decreasing oxygen availability in the combustion chamber. Meanwhlie, a decrease of ambient temperature extened ignition delay due to the slowed fuel evaporation made slow chemical reaction to the injected fuel.

5.1.2 Combustion visualization

3. The HVO is easier to evaporate and mixed with the ambient air in the chamber due to its lower density, viscosity and distillation temperature at T90. The spray development of HVO and diesel is similarly observed at the initial fuel injection with decreasing oxygen concentration due to the same ambient air density and temperature. Lower ambient temperature have more available time to mix and more ambient air entrained into fuel spray to obtain better mixture formation.

4. A decrease of ambient temperature effect on fuel-air mixing by providing a longer mixture formation time, resulting in later luminous flame with higher peak heat release rate. At 700 K of ambient temperature, the flame luminosity of only H50 and HVO are observed because of its higher cetane number provides high reactivity at low temperature. As the results, The HVO and blends can be recommended to use in the diesel engine with EGR application as well, but it is necessary to optimize the engine when using in low temperature combustion such as reducing the compression ratio.

5.2 Suggestion and future work

1. Using needle lift sensor to clearly determine the needle lift opening and closing of injector for evaluating the injection delay and the ignition delay.

2. Investigation on effect of HVO to combustion characteristics by using High speed imaging of OH* chemiluminescence and natural luminosity to clearly understand the chemical reaction, flame luminosity.

3. Investigation on effect of HVO to combustion characteristics by using two-color method to measure flame temperature and KL factor to obtain soot concentration.

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APPENDIX A:
(Measurement specifications)

A dynamics pressure sensor specifications

Pressure

KISTLER
measure. analyze. innovate.

Miniature Measuring Probe

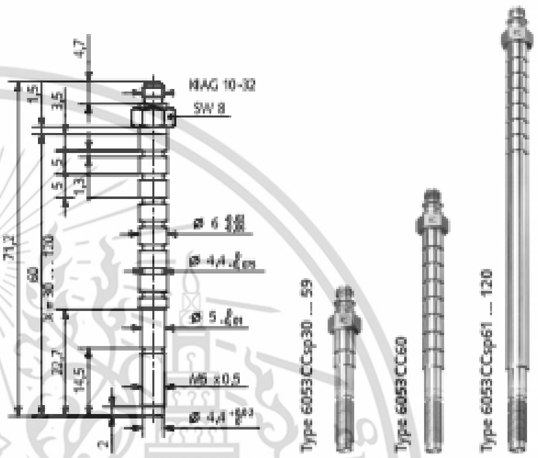
Type 6053CC...

for Non-Cooled Cylinder Pressure Measurement, M5 Thread

Patent No. US 6,105,434

The miniature measuring probe with very small dimensions and M5x0,5 mounting thread is particularly suitable for direct installation in small-capacity combustion engines with more than two valves per cylinder. The measuring element is identical to the standard sensor Type 6052C... .

- Good temperature stability of the sensitivity
- Acceleration-compensated
- Needs only 6 mm mounting bore
- Low thermal shock error and long life thanks to the front seal
- Very high sensitivity



Description

Type 6053CC... uses a new type of PiezoStar* crystal which achieves high sensitivity in conjunction with an extremely small sensor structure. The sensitivity drifts by a maximum of ±0,5 % over the temperature range of 200 ±50 °C. The passive acceleration compensation patented by Kistler keeps the influence of engine vibrations to a minimum.

The front seal allows very good heat dissipation and thus briefly a maximum operating temperature of 400 °C. The diaphragm, optimized by finite element calculation, produces good measuring results and ensures a long service life.
The shape of the probe allows mounting with a very small access bore. The cable connection has to be outside the cylinder-head, in an area free of oil mist. O-rings also permit mounting through coolant passages. The probe can be manufactured in custom lengths between 30 and 120 mm.

Application

The sensor Type 6053CC... is an excellent all-rounder. Its rugged construction makes it suitable for measurements at the knock limit as well as for thermodynamic investigations. This sensor is used mainly on multi-valve engines, motor cycle and other small engines and for combustion analysis.
This sensor should be used when there is insufficient mounting space available for the Type 6052C... .
At high speeds (vibrations), the Type 6053CC...U40 should be used. For applications mainly in the knocking range or at very high peak pressures, use of Type 6053CC...U20 with reinforced diaphragm (heavy duty version) is recommended.
The probe is supplied without a cable. See Accessories for the various cables available for different applications.

Technical Data

Measuring range	bar	0 ... 250
Calibrated sub-ranges	bar	0 ... 50, 0 ... 100, 0 ... 150, 0 ... 250
Overload	bar	300
Sensitivity	pC/bar	≈–20
Natural frequency, nominal	kHz	≈160
Linearity in all ranges (at 23 °C)	%/FSO	≤±0,3
Acceleration sensitivity		
axial	bar/g	<0,0002
radial	bar/g	<0,0005
Operating temperature range	°C	–20 ... 350
Temperature min./max	°C	–50 ... 400
Connector	°C	200

6053CC_000-571e-10.11

Charge Amplifier specifications

Electronics & Software

KISTLER
measure. analyze. innovate.

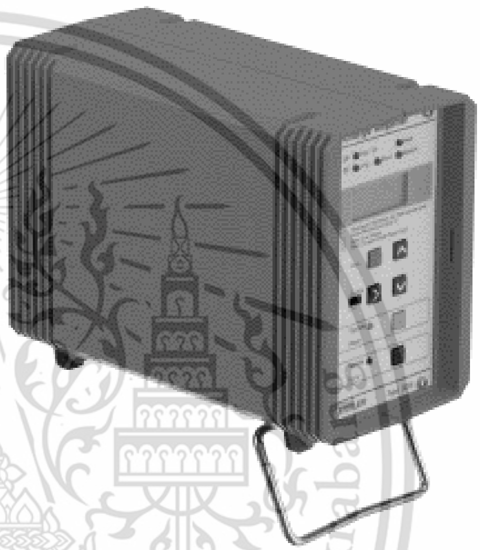
Charge Amplifier

Type 5011B...

Single-channel multi-range laboratory charge amplifier

The mains-operated, microprocessor controlled single-channel charge amplifier Type 5011B... converts the electrical charge produced by piezoelectric sensors into a proportional voltage signal.

- Large measuring range
- Wide frequency range
- Automatic zero correction
- Adjustable low-pass filter and time constant
- Various options and versions provide optimum adaptation to the measuring problem
- Conforming to CE



Description
The main features of the instrument are its continuous measuring range adjustment range from ±10 ... ±999 000 pC and convenient adjustment of the parameters with a two-line LC display. The values entered are retained in the event of an interruption in the power supply.

A built-in IEEE-488 parallel interface or a serial RS-232C interface is available as an option. This enables all set values to be entered or queried. Transmission of data measured is not possible.

The version ...Y50 additionally has a drift compensation and is used in engine measuring technology with uncooled sensors

Application
This amplifier serves mainly to measure mechanical quantities, e.g. pressure, force or acceleration.

The instrument dimensions are DIN standardized and it can be supplied in a desktop or rack mount case.

The principle measurement without calculation:
- Set sensor sensitivity
- Select display scale
- Sensor sensitivity and scale are displayed
- Set the signal output of the data acquisition unit (recorder, oscilloscope...), for example to 1 V/unit (1 V/cm)
- The display appears directly in mechanical units according to the display scale selected

5011B_000-296e-12.05

High Speed Video Camera specifications

ALAVA INGENIEROS

Preliminary Datasheet

FASTCAM Mini UX100

COMPACT HIGH-SPEED CAMERA SYSTEM

PRODUCT FEATURES:

Frame Rate Performance Examples:

- 4,000fps at 1280x1024 pixel resolution.
- 5,000fps at 1280x1000 pixel resolution.
- 6,250fps at 1280x800 pixel resolution.
- 8,000fps at 1280x624 pixel resolution.
- 20,000fps at 1280x248 pixel resolution.
- 200,000fps at 1280x24 pixel resolution.
- 800,000fps at 640x8 pixel resolution.

Compact and Lightweight :
120mm x 120mm x 90mm Camera Body
Weight: 1.5Kg

Dynamic Range (ADC):
12-bit Monochrome, 36-bit Colour.

High Light Sensitivity:
Unique image sensor technology combining high light sensitivity and small pixel size.

Global Electronic Shutter:
Minimum Shutter Speed 3.76µs to 1µs dependent on frame rate selection.

Gigabit Ethernet Interface:
Provides reliable system communication and fast image download.

Recording Memory Options:
To meet all application requirements:
4GB, 8GB and 16GB.

1 inch C-mount Compatible Sensor Size and integrated Nikon G type lens mount:
Allowing operation with both current Nikon DSLR F/G type lenses and full compatibility with 1 inch C- mount lenses at all image resolutions.

Photron
High-Speed Imaging Solutions

Compact and light weight camera system offering outstanding performance and ease of use in a wide-range of high-speed imaging applications

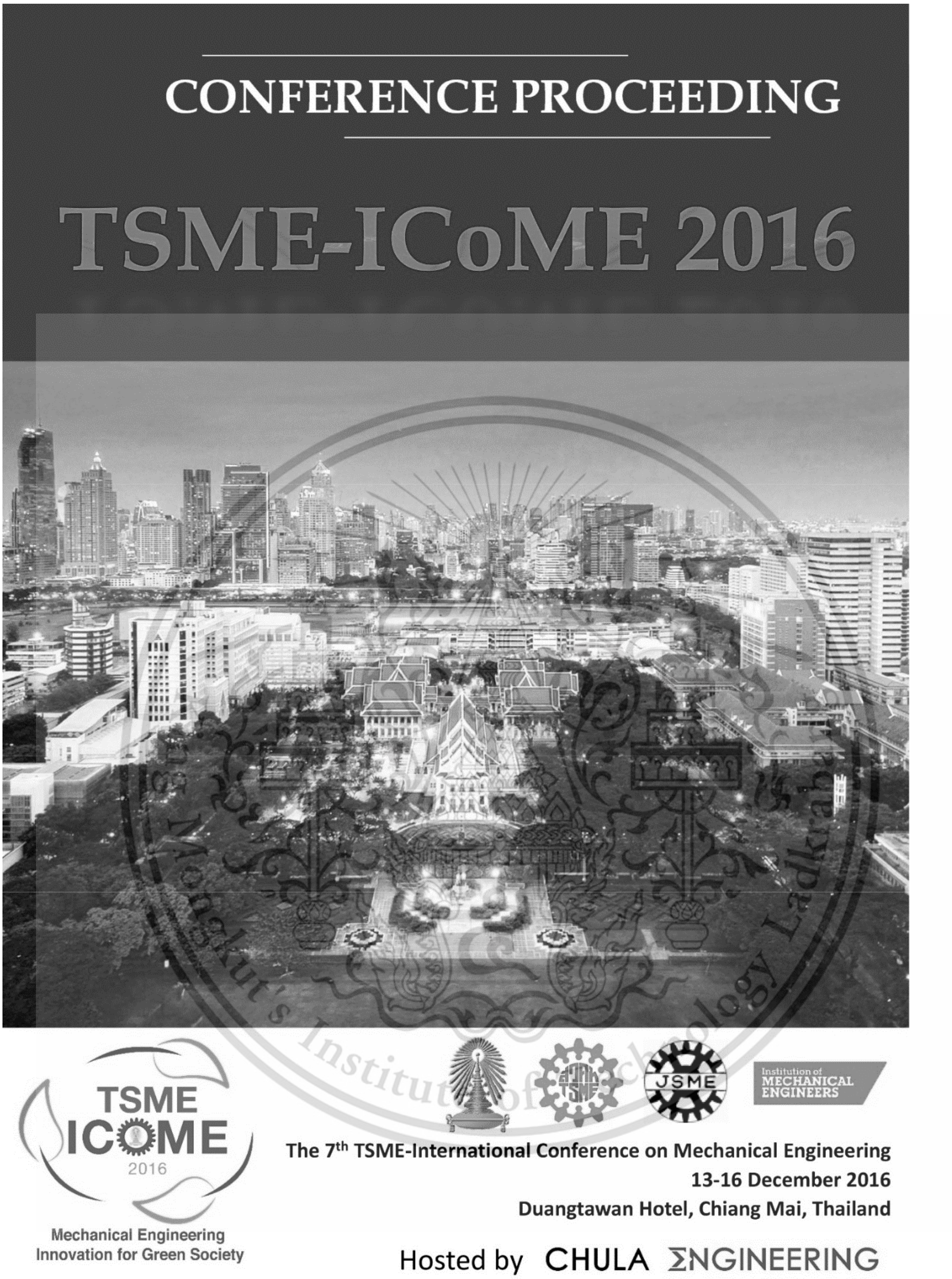
The Photron FASTCAM Mini UX100 high speed camera provides outstanding high speed imaging performance in a small and lightweight camera design. Providing 1.3 Megapixel image resolution (1280x1024 pixels) at frame rates up to 4,000fps frames per second and 1 Megapixel resolution (1280x800 pixels) at 6,250fps, the Mini UX-100 system offers both impressive high speed imaging performance and convenience of use.

Housed within a 120mm x 120mm x 90mm camera body weighing just 1.5kg, the FASTCAM Mini UX-100 is ideally suited for use in a wide range of scientific and industrial applications. The camera is available with recording memory options up to 16GB providing extended recording times and triggering flexibility.

Using innovative proprietary CMOS image sensor technology, the FASTCAM Mini UX-100 achieves high light sensitivity from a sensor having a 10µm pixel size fitted with a micro lens. At full image resolution, the image sensor size is compatible with readily available 1" C mount lenses allowing the user a wide choice of small, low cost and very high aperture (up to f0.95) objectives to use with the system and may also be coupled to scientific microscopes and borescopes. In addition, the camera is supplied with a Nikon F mount lens fitting fully compatible with G type Nikon DSLR objectives.

FASTCAM Mini UX-100 is fitted with a high speed Gigabit Ethernet interface providing reliable system control and fast image download. The system is supplied with intuitive and feature rich Photron FASTCAM Viewer (PFV) software and Photron Device Control SDK (software development kit), allowing integration with user-specific software. Alternatively, the camera can be controlled as a device within a MATLAB or LabVIEW environment.

APPENDIX B:
(Publications)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Oral Presentation

The 7th TSME International Conference on Mechanical Engineering
13-16 December 2016

AEC0021

Experimental Investigation on Fuel Injection and Spray Characteristics of Commercial Diesel and Hydrotreated Vegetable Oil (HVO) with Different Injection Pressure Using High Pressure Fuel Injection System

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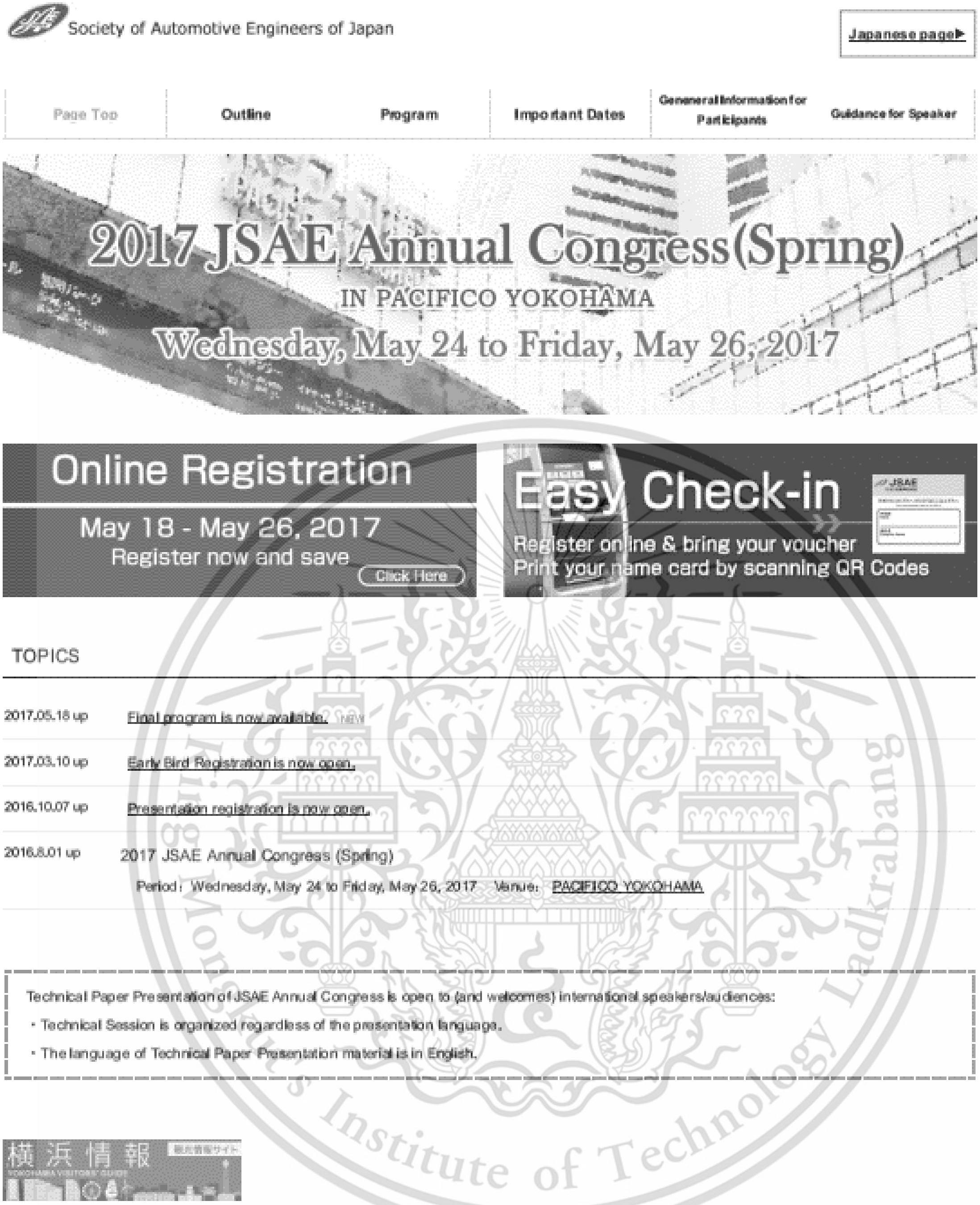
Abstract

Hydrotreated vegetable oil (HVO) is one of promising advance alternative biofuels to use instead of diesel fuel. HVO has many the beneficial fuel properties such as low sulfur, low aromatics, high cetane number, high heating value, similarity of density and viscosity with diesel. The objectives of this paper is to investigate the effects of HVO physical and chemical properties on fuel injection characteristics, spray characteristics and air entrainment. The fuel injection characteristics were investigated under Zeuch method. The fuel spray characteristics were investigated under non-vaporizing conditions using shadowgraph technique and air entrainment analysis. The fuel injection and spray results of diesel and HVO were evaluated with a single-hole solenoid injector with orifice diameter 0.2 mm, and various injection pressure 40 MPa to 140 MPa. The results shows that fuel injection rate profile, measured injection rate and injection quantity of diesel have similar results compare to HVO. The spray tip penetration and spray angle of diesel show similar results as HVO. Local equivalence ratio and the mass of air entrained of diesel and HVO show similar results. In addition, effects of high pressure fuel injection show significantly increasing fuel injection rate profile, measured injection rate, injection quantity, and the mass of air entrained but decreasing local equivalence ratio. These results relate strongly to combustion process and emissions in diesel engines due to differences of fuel properties.

Keywords: hydrotreated vegetable oil (HVO), fuel injection and spray characteristics, Zeuch method, shadowgraph technique, air entrainment analysis.

1. Introduction

Hydrotreated vegetable oil (HVO) is a second generation of biofuels which can be produced from many kinds of vegetable oil by using hydrotreating process (remove oxygen contain without removing carbon contain) [1]. HVO has many advantages to use instead of conventional diesel fuel for compression ignition (CI) engines in term of variety of bio feedstock, low aromatic, low sulfur, high cetane number and heating value, similarity of viscosity and density with diesel [1-9]. However, there are some disadvantages that may limit to use HVO from previous study such as poor low-temperature properties, as displayed by cloud point, pour point and cold filter plugging point (CFPP) [1,2]. Therefore, an improvement process as isomerization process is may be use to solve that problem then HVO would be iso-HVO. Since iso-HVO still have some disadvantage as too high cetane number and low lubricity, this work has concerned and would not be recommended to blend HVO over 50% [2]. Since HVO has been attracted many researcher for adopting it into CI engines so that many researches have done an investigation effects of fuel properties of HVO and HVO blended with diesel on engine performance and emissions [3-11]. Previous study on effects of HVO on engine performance show that HVO and HVO blend show decreasing fuel consumption [3-7], increasing brake fuel conversion efficiency [8], with brake specific fuel consumption also can decrease with optimizing start of injection for specific engine calibration [9]. Furthermore, the use HVO combined with various exhaust gas recirculation (EGR) system still show significant lower fuel consumption [6,7,9]. Previous research on combustion characteristics in CI engine show that HVO and HVO blend can significantly reduce the ignition delay because HVO show beneficial fuel properties such as high cetane number and lower distillation temperature as high ignition quality. Also, combustion duration can reduce



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講演番号 375 文献番号 20175375

Experimental Investigation on Spray Combustion Characteristics of Hydrotreated Vegetable Oil (HVO)-Diesel Blends in Constant Volume Combustion Chamber (CVCC)

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ABSTRACT: Hydrotreated vegetable oil (HVO) is one of candidate to replace diesel fuel due to its superior properties such as high cetane number and heating value. The objective of this paper is to investigate effects of HVO and diesel blends on combustion characteristics using heat release analysis and shadowgraph technique. Four test fuels were tested: diesel, four diesel-HVO blends by mass: 20%(H20), and 50%(H50) and 100% HVO(H100) in constant volume combustion chamber (CVCC) with single-hole injector. The results show that increasing HVO fraction provide shorter ignition delay and slightly lower heat release rate.

KEY WORDS: **Heat engine, Hydrotreated vegetable oil (HVO), Combustion characteristics, Heat release rate**, Shadowgraph technique. (A1)

1. INTRODUCTION

Diesel engines (DI) engines provide higher thermal efficiency than spark ignition (SI) engines, however DI engines still have serious pollutant problems. Many researchers have been employed various techniques for reducing in exhaust gas emissions such as after-treatment systems (i.e. DOC, SCR, DPF, etc.) [1], engine combustion control (i.e. common-rail, swirl mixing, EGR, etc.) [2,3]. On the other hand, research on the alternative and renewable fuels are searching to employ for proper way, which have not only sustainable, but also friendly, respect with the environment. Therefore, alternative fuels as sustainable energy have been researched for applying into diesel engines such as biodiesel (i.e. FAME, SME, RME).

Biodiesel which is a first generation of biofuel can be produced by transesterification process. Biodiesel have been widely used, as it can be directly used or blending used with diesel in the engines without no modification. However it still have some disadvantages to the engines such as low heating value [4] and high density and viscosity that make larger droplet size distribution, poor fuel-air mixing processes and mixture formation [5,6]. However, HVO is more advantage than biodiesel such as high cetane number, similar density and viscosity to diesel that can make small atomization, better mixture formation.

HVO is a second generation of biofuel that can be produced from many kinds of vegetable oil by using hydrotreating process. HVO is a mixture of normal paraffin and iso-paraffin [7]. HVO is promising alternative fuels as one of candidate to replace diesel due to its more advantages in comparison with other alternative diesel fuels. The first one shows about production. HVO has variety of bio-feedstock, better oxidation stability. The second one is better fuel properties such as low aromatics and sulfur, high cetane number, high heating value, similar density, viscosity, chemical structure to diesel [8]. From this reason, it offers significant improvement in engine performance and exhaust gas emissions. However, HVO still have limitation to use in diesel engines as it shows very high cetane number and low lubricity, so researchers suggested that HVO would not be blended over 50% [9].

Effects of HVO on combustion characteristics have been performed by many researches in the diesel engines. Sugiyama et al [10] carried out HVO on the engine and concluded that heat release rate of HVO exhibited more advance with shorten ignition delay to improve combustion which resulted decreasing

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Publications:

1) S. Marasri, P. Ewphun, S. Saeo, P. Srichai, C. Charoenphonphanich, P. Karin, M. Tongroon and H. Kasaka. "Experimental Investigation on Fuel Injection and Spray Characteristics of Comercial Diesel and Hydrotreated Vegetable Oil (HVO) with Different Injection Pressure Using High Pressure Fuel Injection System". The 7th TSME International Conference on Mechanical Engineering, Chiang Mai, Thailand, December 13th – 16th, 2016.

2) S. Marasri, P. Ewphun, P. Srichai, S. Saeo, V. T. Chau, C. Charoenphonphanich, P. Karin, M. Tongroon and H. Kasaka. "Experimental Investigation on Spray Combustion Characteristics of Hydrotreated Vegetable Oil (HVO) – Diesel Blends in Constant Volume Combustion Chamber (CVCC)". 2017. JSAE Annual Congress (Spring) No. 20175375. Yokohama, Japan, May 24th - 26th, 2017.

Publications

- **International Conference**

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2. **Sombat Marasri**, Pop-Paul Ewphun, Prathan Srichai, Sukit Saeo, Vo Tan Chau, Chinda Charoenphonphanich, Preechar Karin, Manida Tongroon and Hidenori Kosaka. Experimental investigation on spray combustion characteristics of hydrotreated vegetable oil (HVO) – diesel blends in Constant volume combustion chamber (CVCC). 2017. JSAE Annual Congress (Spring) No. 20175375. Yokohama, Japan, May 24th - 26th, 2017. (Part of Master's thesis)

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Experimental Investigation on Fuel Injection and Spray Characteristics of Commercial Diesel and Hydrotreated Vegetable Oil (HVO) with Different Injection Pressure Using High Pressure Fuel Injection System

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Abstract

Hydrotreated vegetable oil (HVO) is one of promising advance alternative biofuels to use instead of diesel fuel. HVO has many the beneficial fuel properties such as low sulfur, low aromatics, high cetane number, high heating value, similarity of density and viscosity with diesel. The objectives of this paper is to investigate the effects of HVO physical and chemical properties on fuel injection characteristics, spray characteristics and air entrainment. The fuel injection characteristics were investigated under Zeuch method. The fuel spray characteristics were investigated under non-vaporizing conditions using shadowgraph technique and air entrainment analysis. The fuel injection and spray results of diesel and HVO were evaluated with a single-hole solenoid injector with orifice diameter 0.2 mm, and various injection pressure 40 MPa to 140 MPa. The results shows that fuel injection rate profile, measured injection rate and injection quantity of diesel have similar results compare to HVO. The spray tip penetration and spray angle of diesel show similar results as HVO. Local equivalence ratio and the mass of air entrained of diesel and HVO show similar results. In addition, effects of high pressure fuel injection show significantly increasing fuel injection rate profile, measured injection rate, injection quantity, and the mass of air entrained but decreasing local equivalence ratio. These results relate strongly to combustion process and emissions in diesel engines due to differences of fuel properties.

Keywords: hydrotreated vegetable oil (HVO), fuel injection and spray characteristics, Zeuch method, shadowgraph technique, air entrainment analysis.

1. Introduction

Hydrotreated vegetable oil (HVO) is a second generation of biofuels which can be produced from many kinds of vegetable oil by using hydrotreating process (remove oxygen contain without removing carbon contain) [1]. HVO has many advantages to use instead of conventional diesel fuel for compression ignition (CI) engines in term of variety of bio feedstock, low aromatic, low sulfur, high cetane number and heating value, similarity of viscosity and density with diesel [1-9]. However, there are some disadvantages that may limit to use HVO from previous study such as poor low-temperature properties, as displayed by cloud point, pour point and cold filter plugging point (CFPP) [1,2]. Therefore, an improvement process as isomerization process is may be use to solve that problem then HVO would be iso-HVO. Since iso-HVO still have some disadvantage as too high cetane number and low lubricity, this work has concerned and would not be recommended to blend HVO over 50% [2]. Since HVO has been attracted many researcher for adopting it into CI engines so that many researches have done an investigation effects of fuel properties of HVO and HVO blended with diesel on engine performance and emissions [3-11]. Previous study on effects of HVO on engine performance show that HVO and HVO blend show decreasing fuel consumption [3-7], increasing brake fuel conversion efficiency [8], with brake specific fuel consumption also can decrease with optimizing start of injection for specific engine calibration [9]. Furthermore, the use HVO combined with various exhaust gas recirculation (EGR) system still show significant lower fuel consumption [6,7,9]. Previous research on combustion characteristics in CI engine show that HVO and HVO blend can significantly reduce the ignition delay because HVO show beneficial fuel properties such as high cetane number and lower distillation temperature as high ignition quality. Also, combustion duration can reduce because HVO can reduce time for mixture formation as more fuel vaporize and air-fuel interaction inside combustion chamber very well during combustion phase. Furthermore, HVO and HVO blend provide higher combustion pressure, heat release rate, and combustion efficiency compare to diesel [6-9] because fuel and air are very well mixed that lead to better ignition quality.

Previous study on effects of HVO properties on emissions conclude that the employment of HVO can be relevant reduced exhaust gas emissions such as hydrocarbon (HC), carbon monoxide (CO), nitrogen oxide (NO_x) and particulate matter (PM), especially decreasing in HC and CO emissions which are derived from incomplete combustion even through it combined with EGR [3-5,7-9]. Furthermore, researcher conclude that not only HC and CO decrease but also NO_x, particulate matter (PM) decrease [3-5,9], while some research conclude that PM was depended on combustion mode [7].

A few work has done HVO on fuel injection and macroscopic fuel spray characteristics. Pop-Paul et al. [10]. investigated effects of diesel, HVO and HVO blended with diesel on fuel injection characteristics under Zeuch method and concluded that injection delay was shorter with increasing HVO fraction that cause by fuel viscosity. Injection duration, injection quantity, and injection rate of diesel show similar results as HVO and HVO blend. Furthermore, discharge coefficient increase with increasing HVO fraction due to lower density and viscosity that make lower friction loss. Sugiyama et al. [6]. investigated effects of HVO properties on spray characteristics and concluded that spray penetration and spray angle of diesel show same results as HVO. This shows good agreement with the results of Hullkkonen et al. [11]. concluded that no significant difference in spray tip penetration were found between diesel and HVO with increasing injection pressure. However, spray angle of diesel shows slightly narrower than HVO with increasing injection pressure. Chen et al. [12]. investigated effects of HVO on macroscopic fuel spray using common-rail high-pressure fuel injection system and reported that spray tip penetration and spray angle of diesel have longer and narrower than HVO for all injection pressure due to its higher density. Furthermore, spray tip penetration of diesel and HVO increase with increasing injection pressure, while spray cone angle is no significant with increasing injection pressure. F. Millo et al. [9]. investigated effects of HVO blend on fuel injection system using Zeuch method and concluded that diesel show similar spray tip penetration but spray cone angle show wider than HVO blend.

From previous studies, effects of HVO properties have shown superior advantages for adapting it into CI engine as increasing engine performance, decreasing regulated and unregulated emissions. Furthermore, since its properties are similar with diesel, it also shows quite similar fuel injection and spray characteristics. However, to better understand effects of HVO advance properties on combustion, engine performance and emissions characteristics. It is required fuel characteristics which are fuel injection characteristics, fuel spray characteristics [13]. The injection and fuel spray characteristics. All the most injection characteristics such as injection delay, injection duration, injection quantity, injection rate profile and injection pressure, and spray characteristics such as spray tip penetration, spray angle, local equivalence ratio and air entrainment determine input energy, heat release, mixture formation, combustion pressure and temperature, combustion efficiency which relate strongly to engine performance and emissions.

The objectives of this paper are to investigate effects HVO physical and chemical properties on fuel injection characteristics, fuel spray characteristics, and fuel-air mixing process in various injection pressure to clearly understand the effects of injection pressure on fuel injection and spray characteristics.

2. Materials and Methods

2.1 Zeuch method for measuring injection rate

Measuring injection rate using Zeuch method [14]. Tested fuel is filled into a constant Zeuch chamber at the certain pressure then injected test fuel into the chamber. The chamber pressure rise (ΔP) proportional to injected fuel. Then pressure transducer detects the pressure rise from injected fuel. Fuel bulk modulus is fuel resistance as compressibility of liquid fuel which can be calculated from the change of volume chamber (ΔV) and pressure rise as shown in Eq. (1).

$$K = V_o \frac{\Delta P}{\Delta V} \tag{1}$$

Where K is fuel bulk modulus, V_o is Zeuch chamber volume. Using of fuel bulk modulus and pressure rise from injected fuel, then the measured injection rate can be determined by Eq. (2) [14].

$$\dot{m}_{measured} = \frac{dm}{dt} = \rho \frac{V_o}{K} \frac{dP}{dt} \tag{2}$$

The theoretical mass flow rate defined as theoretical mass flow rate of injected fuel, is derived from a combination of the continuity equation and Bernoulli's equation, assuming that the inlet velocity is negligible which can be determined from Eq. (3) [15].

$$\dot{m}_{th} = n_{orifice} \cdot S \sqrt{2 \Delta P \cdot \rho_f} \tag{3}$$

Where $n_{orifice}$ is the number of orifices on the nozzle, S is the outlet geometric cross-section area of the orifice, ΔP is the pressure difference of injection pressure and back pressure, ρ_f is fuel density.

Fig. 1 shows the definition typical fuel injection process, injection rate profile calculated by using Eq. (2). Start of energizing signal (SOE) to start of fuel injection (SOI) is defined as injection delay. Injection duration is defined as a period from SOI to end of fuel injection (EOI). Injection quantity defined as a total amount of fuel injected is calculated by integration under injection rate curve area from SOI to EOI.

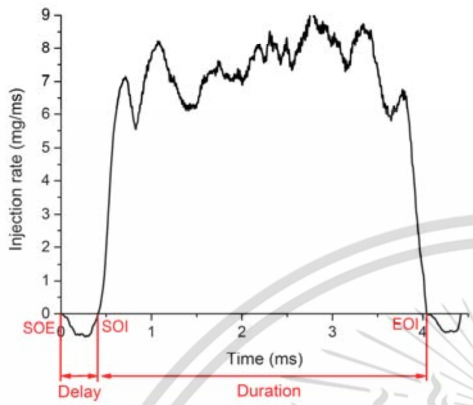

Fig. 1 Definition typical injection process

2.2 Fuel spray images analysis

The definition of fuel spray injection is presented in Fig. 2. Spray tip penetration (S) defined as the distance from injector tip to spray tip in the axial direction. Spray angle (θ) defined as the maximum angle of each side spray at ($S/2$) from the tip of injector. The sequence spray images were analyzed by image processing using MATLAB program. Spray tip penetration and spray angle data from image processing are used to determine air entrainment and local equivalence ratio.

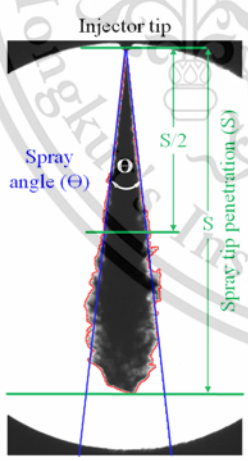

Fig. 2 Definition of fuel spray injection

2.3 Air entrainment analysis

The mass of air entrained (m_a) is presented as a function of spray angle, spray tip penetration, and ambient density can be calculated by Eq. (4) [16].

$$m_a = (\pi/3)(\tan(\theta/2))^2 S^3 \rho_a \tag{4}$$

The mass of air entrained defined as air entrained to fuel spray. For actual combustion in CI engine, it is defined as mixture formation of fuel spray inside the combustion chamber. There are many factors affecting to air entrainment such as spray tip penetration, spray angle, and ambient conditions.

2.4 Local equivalence ratio

For actual combustion in CI engine, the equivalence ratio defined as the operating combustion mode, and is a measure of air-fuel mixture relative to stoichiometric conditions. For this experiment, equivalence ratio is affected by fuel and ambient density, fuel spray, and orifice diameter.

The local equivalence ratio defined as a function of fuel density (ρ_f , kg/m^3), ambient density (ρ_a , kg/m^3), spray tip penetration (m), spray angle (rad) and orifice diameter (D_o , m), as shown in Eq. (5) [16].

$$\phi = \sqrt{\frac{\rho_f}{\rho_a}} \frac{D_o}{2S\tan(\theta/2)} \tag{5}$$

3. Experimental setup and condition

This investigation included two experiment: fuel injection experiment using Zeuch method and fuel spray experiment using shadowgraph technique as shown in Fig. 4 and Fig. 5, respectively.

3.1 Fuel injection and spray experiment

Fig. 4 shows the schematic diagram of the fuel injection experiment. A single-hole solenoid injector with orifice diameter 0.2 mm was installed at the top of 40 cm^3 Zeuch chamber capacities to clearly determine pressure signal for measuring injection rate [14]. Hand pump was used to generate back pressure by filling test fuel into the chamber via the intake valve at a certain pressure. Two pressure transducer were used in this investigation. First, a static pressure transducer was used to measure back pressure. Second, a piezoelectric pressure transducer (Kistler 6053CC60) and charge amplifier (Kistler 5011) were used to measure pressure rise from injected fuel. The pressure increase of fuel injected was recorded by oscilloscope (RICOL DS1052E) with sampling rate 2x10^8 S/s

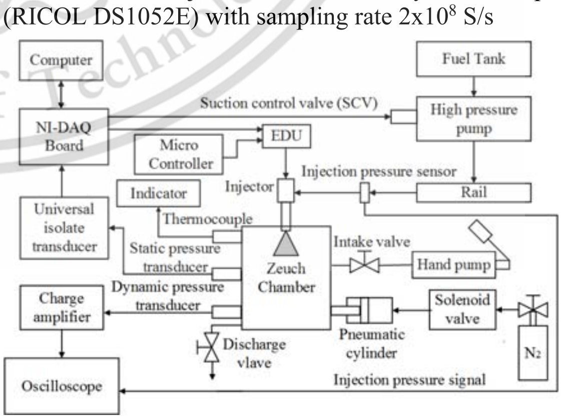

Fig. 4 Fuel injection experiment

Fig. 5 shows the schematic diagram of fuel spray experiment. A single-hole solenoid injector with orifice diameter 0.2 mm was installed at the top of constant volume combustion chamber (CVCC) to inject tested fuel. Nitrogen (N_2) gas was used to

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simulate ambient density conditions as back pressure via the intake valve. A static pressure transducer was used to measure back pressure. In addition, Shadowgraph technique was setup to visualize fuel spray and capture high speed video spray by using high speed video camera (Photron: FASTEAM MINI UX100) with frame rate of 10,000 frame per picture.

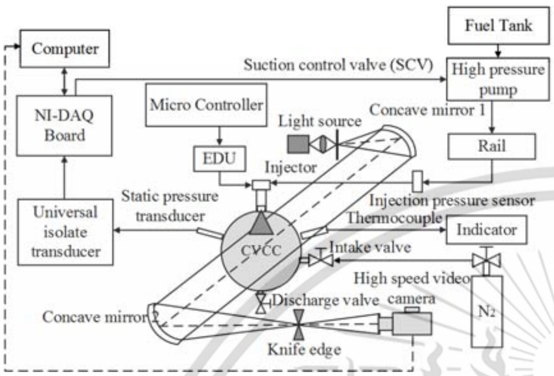

Fig. 5 Fuel spray experiment

3.1 Experimental condition

Two experiments were conducted with a second generation common-rail fuel system in order to generate high fuel injection pressure 40 to 140 MPa, a single-hole solenoid injector with orifice diameter 0.2 mm, energizing time 2.5 ms, 2.5 MPa back pressure (28 kg/m^3 ambient density) that refer to CI condition at the end of compression stroke. Fuel injection and spray experiment were performed under ambient temperature 300 ± 2 K and non-vaporizing conditions as shown in Table. 1.

Table. 1 Experimental condition for fuel injection and spray measurement

Parameters	Conditions
Test fuels	Commercial diesel, HVO
Orifice diameter	Single hole 0.2 mm
Energizing time	2.5 ms
Back pressure	2.5 MPa
Injection pressure	40-140 MPa
Ambient temperature	300 ± 2 K
Repeat	10 times / condition

3.3 Test fuels

In this study, two different fuels were hydrotreated vegetable oil (HVO) and commercial diesel. Fuel viscosity, density, and surface tension of diesel have higher 22.72%, 5.91%, and 11.26%, but air-fuel ratio and heating value have lower 2.56% and 2.13%. The fuel properties of test fuels were comparatively shown in Table. 2.

Table. 2 Fuel properties

Properties	Standard	Commercial diesel	HVO
C (%wt)	ASTM D5291	85.73	84.24
H (%wt)		13.22	15.05
O (%wt)		-	-
Chemical formula	-	$C_{14.11}H_{26.23}$	$C_{14.03}H_{29.86}$
$(A/F)_{st}$	-	14.48	14.86
Kinematic viscosity @40°C (cSt)	ASTM D445	3.24	2.64
Density @30°C (kg/m^3)	ASTM D4052	824	778
Surface tension (mN/m)	ASTM D1590	26.98	24.25
Q_{LHV} (MJ/kg)	ASTM D240	45.86	46.86

4. Results and Discussions

The effects of physical and chemical properties of commercial diesel and hydrotreated vegetable oil (HVO) on fuel injection characteristics are discussed such as injection rate profile, measured injection rate, injection quantity. The effects on spray characteristics such as spray tip penetration, spray angle, the mass of air entrained, and local equivalence ration also are discussed in this study. Both fuel injection and spray characteristics were investigated using high fuel injection system.

4.1 The comparison of the injection rate profile between diesel and HVO for different injection pressure

Fig. 6(a) and Fig. 6(b) show effects of different fuel properties and injection pressure between diesel and HVO on injection rate profile. The injection rate profile is presented as a function of fuel density, fuel bulk modulus, Zeuch chamber capacity, and the pressure increase of fuel injected which is calculated from Eq. (2). The injection rate profile shows the negative value from SOE to SOI because movement of needle lift cause changing volume of the system then chamber pressure decrease. Diesel and HVO show an increase in the injection rate profile with increasing injection pressure because increased injection pressure enhances flow capacity. The injection delay injection duration were decreased with increasing injection pressure because the increased injection pressure provides higher lift force to make opening the needle lift rapidly [17]. Therefore, the employment of an increase in injection pressure is improvement of injection rate patterns such as sufficiently increase injection rate profile, measured injection rate, and injection quantity, but decrease injection delay and injection duration.

Effects of diesel and HVO on injection rate were comparatively evaluated as a representative shown in Fig. 6(c). It can be seen that injection rate profile of diesel show the same trends as HVO. This means that there is no significant effect of fuel density on injection rate profile for same injection pressure because the mass flow rate is a function of the square root fuel density [15] then as diesel density has higher 5.91% compare to HVO, lead to increasing 2.91% in injection rate. However, clearly decrease of injection delay and injection duration were found between diesel and HVO when injection pressure increase. HVO has shown shorter injection delay and injection duration due to its lower 22.73% viscosity that lead to lower resistance force to needle lift [19].

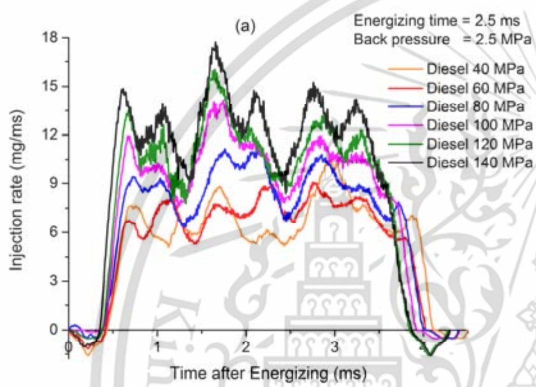

Fig. 6(a) Injection rate profile of diesel and various injection pressure

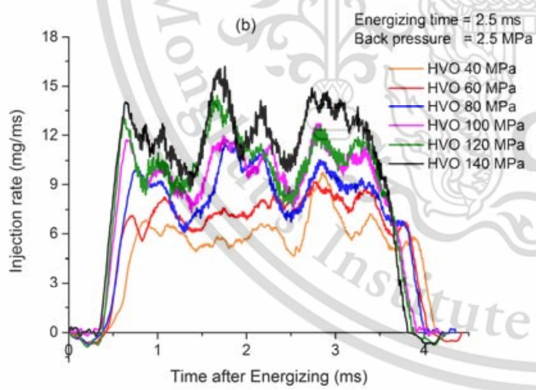

Fig. 6(b) Injection rate profile of HVO and various injection pressure

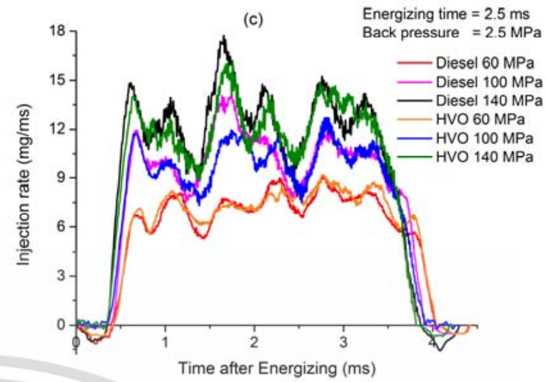

Fig. 6(c) Injection rate profile of diesel and HVO and injection pressure of 60, 100 and 140 MPa

4.2 The comparison of the measured injection rate between diesel and HVO for different injection pressure

Fig. 7 shows comparison of measured injection rate as actual injection rate between diesel and HVO for different injection pressure. Since theoretical injection rate of diesel show higher than those of HVO due to its higher 5.91% density, the measured injection rate also show the same trends as theoretical as well. The measured injection rate of diesel show higher by 13.25%, 5.42%, 2.02%, 1.68%, 4.86%, 5.36%, respectively compare to HVO because of slightly effect of diesel mass flow rate as injection rate profile results. Therefore, the main advantage of the increased injection pressure is to increase of measured injection rate and injection quantity because the higher injection pressure make larger pressure difference in Eq. (3) then it has increased theoretical mass flow rate so the measured injection rate also increase.

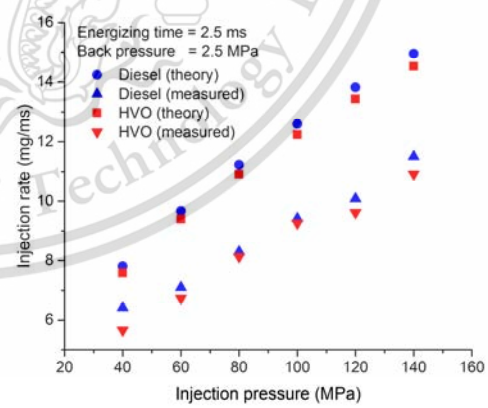

Fig.7 Measured injection rate and various injection pressure

4.3 The comparison of injection quantity between diesel and HVO for different injection pressure

Fig. 8 shows the comparison of injection quantity between diesel and HVO for different injection pressure. Injection quantity is calculated from the integration under injection rate curve area as shown in

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Fig. 1. The total amount of fuel injected as injection quantity show the same trends as the measured injection rate as well. Injection quantity of diesel show higher by 13.17%, 5.45%, 2.52%, 3.83%, 8.51%, 6.66%, respectively compare to HVO because diesel has higher injection rate profile so that the injection rate curve area has increased then lead to increasing injection quantity that supply to combustion chamber. The increased amount of fuel provides increase of an input energy as heat input in combustion process for CI engines [18].

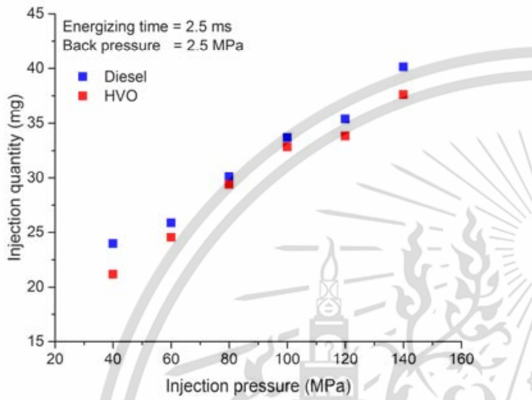

Fig. 8 Injection quantity and various injection pressure

4.4 The comparison of air entrainment between diesel and HVO for different injection pressure

Fig. 9(a) and Fig. 9(b) show comparison of air entrainment by time after energizing between diesel and HVO. Air entrainment is presented as a function of spray angle, spray tip penetration and ambient density as shown in Eq. (4). The mass of air entrained increase with increasing injection pressure. The increased injection pressure cause higher the air entrained because the wider of spray angle [16]. These mean the air can entrained to fuel spray greatly with increasing injection pressure.

Effects of diesel and HVO on mass of air entrained were comparatively evaluated as a representative shown in Fig. 9(c). It can be seen that diesel air entrainment shows the similar results as HVO for all injection pressure because similar spray tip penetration and spray angle results even through viscosity and surface tension of diesel were higher by 22.72%, 5.91%, and 11.26%, respectively compare to HVO.

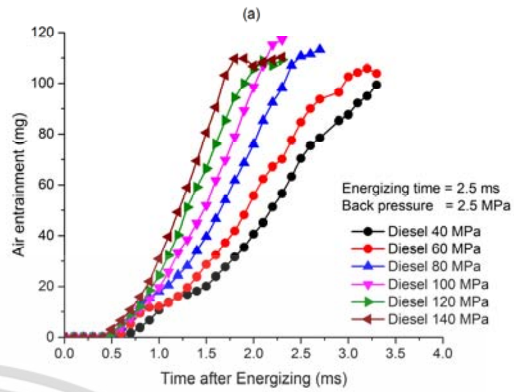

Fig. 9(a) Air entrainment of diesel and various injection pressure

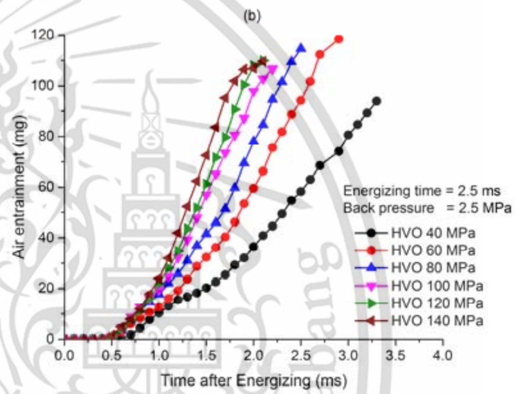

Fig. 9(b) Air entrainment of HVO and various injection pressure

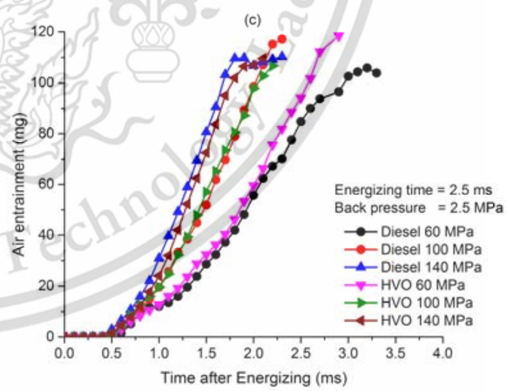

Fig. 9(c) Air entrainment of diesel and HVO and injection pressure of 60, 100 and 140 MPa

4.5 The comparison of the local equivalence ratio between diesel and HVO for different injection pressure

Fig. 10(a) and Fig. 10(b) show local equivalence ratio by time after energizing of diesel and HVO. Local equivalence ratio is presented as a function of fuel and gas density, spray tip penetration, and spray angle which is calculated from Eq. (5). Local equivalence ratio is higher initially at nearby 0.4 ms because SOE and SOI are observed around 0 ms to 0.4

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ms as shown in Fig 6. Diesel has slower increase of equivalence ratio compare to HVO for all an injection pressure due to its higher viscosity that make its slower opening the needle lift [19]. With diesel show slightly lower equivalence ratio compare to HVO due to its lower stoichiometric air- fuel ratio [16].

Effect of diesel and HVO on local equivalence ratio as a representative were shown in Fig. 10(c). Local equivalence ratio results of diesel show the same trends as HVO because the spray results as spray angle and spray tip penetration of diesel and HVO were similar. This can concluded that HVO properties have not affected to spray angle, spray tip penetration as previous research [11,12] so that local equivalence ratio were similar results. Therefore, as the spray tip penetration and spray angle between diesel and HVO show similar trends so that local equivalence ratio and air entrainment also show similar results as well. This can concluded that local equivalence ratio decrease corresponded with increasing the mass of air entrained.

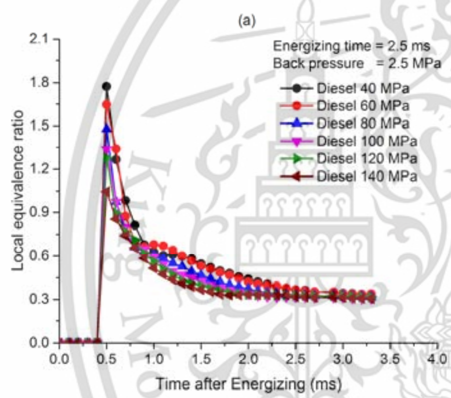

Fig. 10(a) Local equivalence ratio of diesel and various injection pressure

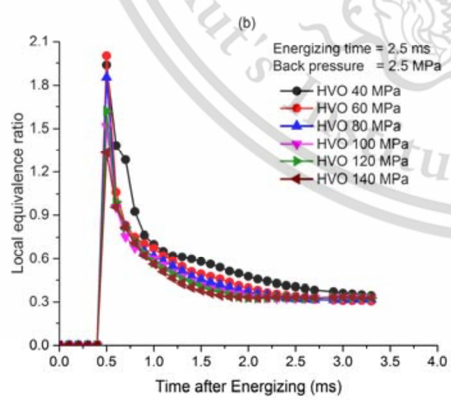

Fig. 10(b) Local equivalence ratio of HVO and various injection pressure

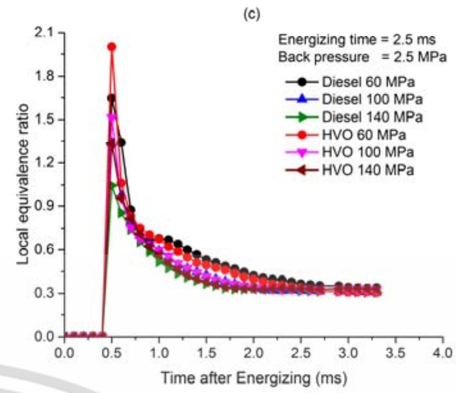

Fig. 10(c) Local equivalence ratio of diesel and HVO and injection pressure of 60, 100 and 140 MPa

5. Conclusions

The effects of diesel and hydrotreated vegetable oil (HVO) on fuel injection and spray characteristics with various injection pressure were investigated using Zeuch method, shadowgraph technique and air entrainment analysis. The injection rate profile, measured injection rate, injection quantity, the mass of air entrained, and local equivalence ratio were summarized as follows:

Fuel Injection characteristics

No significant differences of fuel injection rate profile and measured injection rate were found between diesel and HVO even through increasing fuel injection pressure.

Significant differences of injection delay and injection duration were found between diesel and HVO. The differences cause by lower viscosity of HVO. Also injection quantity of diesel has higher than HVO due to its lower density.

In addition, significant improvement of injection rate profile such as decreasing injection delay and injection duration, but increasing actual injection rate and injection quantity with increasing injection pressure.

Spray characteristics

No significant differences of the mass of air entrained and local equivalence ratio between diesel and HVO were found due to its same spray tip penetration and spray angle.

6. Acknowledgement

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Denso Corporation, Thailand and Petroleum Authority of Thailand (PTT) Co., Ltd. for providing experimental test fuels.

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Experimental Investigation on Spray Combustion Characteristics of Hydrotreated Vegetable Oil (HVO)-Diesel Blends in Constant Volume Combustion Chamber (CVCC)

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ABSTRACT: Hydrotreated vegetable oil (HVO) is one of candidate to replace diesel fuel due to its superior properties such as high cetane number and heating value. The objective of this paper is to investigate effects of HVO and diesel blends on combustion characteristics using heat release analysis and shadowgraph technique. Four test fuels were tested: diesel, four diesel-HVO blends by mass: 20%(H20), and 50%(H50) and 100% HVO(H100) in constant volume combustion chamber (CVCC) with single-hole injector. The results show that increasing HVO fraction provide shorter ignition delay and slightly lower heat release rate.

KEY WORDS: **Heat engine, Hydrotreated vegetable oil (HVO), Combustion characteristics, Heat release rate**, Shadowgraph technique. (A1)

1. INTRODUCTION

Diesel engines (DI) engines provide higher thermal efficiency than spark ignition (SI) engines, however DI engines still have serious pollutant problems. Many researchers have been employed various techniques for reducing in exhaust gas emissions such as after-treatment systems (i.e. DOC, SCR, DPF, etc.) [1], engine combustion control (i.e. common-rail, swirl mixing, EGR, etc.) [2,3]. On the other hand, research on the alternative and renewable fuels are searching to employ for proper way, which have not only sustainable, but also friendly, respect with the environment. Therefore, alternative fuels as sustainable energy have been researched for applying into diesel engines such as biodiesel (i.e. FAME, SME, RME).

Biodiesel which is a first generation of biofuel can be produced by transesterification process. Biodiesel have been widely used, as it can be directly used or blending used with diesel in the engines without no modification. However it still have some disadvantages to the engines such as low heating value [4] and high density and viscosity that make larger droplet size distribution, poor fuel-air mixing processes and mixture formation [5,6]. However, HVO is more advantage than biodiesel such as high cetane number, similar density and viscosity to diesel that can make small atomization, better mixture formation.

HVO is a second generation of biofuel that can be produced from many kinds of vegetable oil by using hydrotreating process. HVO is a mixture of normal paraffin and iso-paraffin [7]. HVO is promising alternative fuels as one of candidate to replace diesel due to its more advantages in comparison with other alternative diesel fuels. The first one shows about production. HVO has variety of bio-feedstock, better oxidation stability. The second one is better fuel properties such as low aromatics and sulfur, high cetane number, high heating value, similar density, viscosity, chemical structure to diesel [8]. From this reason, it offers significant improvement in engine performance and exhaust gas emissions. However, HVO still have limitation to use in diesel engines as it shows very high cetane number and low lubricity, so researchers suggested that HVO would not be blended over 50% [9].

Effects of HVO on combustion characteristics have been performed by many researches in the diesel engines. Sugiyama et al [10] carried out HVO on the engine and concluded that heat release rate of HVO exhibited more advance with shorten ignition delay to improve combustion which resulted decreasing in fuel consumption, Hydrocarbon (HC) and Particulate Matter (PM) emissions. Liu et al [11] concluded that combustion pressure and heat release rate of HVO showed slightly higher than diesel, moreover the shorter ignition delay and combustion duration were observed with high EGR rate due to its higher cetane number, higher amount energy per unit volume, and the long chain paraffin is easier to broken up than diesel fuel. Jaroonjitsathian et al [12] experimentally investigated HVO and HVO blended fuels on combustion characteristics and engine performance, concluded that HVO and blends provided shorter ignition delay, combustion duration, and better combustion efficiency, consequently overall engine efficiency and specific energy consumption can be improved by HVO. Sugiyama et al [10] and Armas et al [13] carried out HVO on chassis dynamometer to performance, combustion and emission characteristics under New European Driving Cycle (NEDC) and showed that HVO exhibited low fuel consumption, higher heat release rate, shorter ignition delay due to its higher cetane number. In addition, exhaust gas emissions also decreased.

From previous researches have been done on combustion so these can be summarized that HVO provided shorter ignition delay, slightly higher heat release rate, and low exhaust gas emissions due to its better ignition quality that provide better combustion efficiency. Therefore, HVO showed the better the evaporation characteristics, smaller droplet size distribution that make its better mixture formation. However, a few work have been performed the influence of diesel and HVO blend percentage on characteristics of the ignition delay and spray combustion by using heat release analysis. Especially, the discussions effects of diesel and HVO blend percentage on the combustion visualization such as spray evaporation and flame development have not clearly explain yet.

The objective of this paper is to investigate effects diesel and HVO blend percentage on the heat release rate, the ignition delay and the integral heat release in constant volume combustion chamber (CVCC) under simulated diesel combustion condition using heat release analysis. Moreover, the shadowgraph technique was to visualize combustion processes of fuel in order to describe effects of fuel properties of diesel and HVO blend percentage to fuel evaporation processes, and flame development. Therefore, the fundamental data of ignition delay, combustion and the visualization of combustion are helpful information to optimize designing in diesel engines by HVO.

2. METHODS

2.1 Heat release analysis

Heat release rate can be determined from pressure rise after burning injected fuel based on the first law of thermodynamics of the system, as shown in Equation (1) [14].

$$\frac{dQ}{dt}=\frac{\gamma}{\gamma-1}\cdot P\cdot\frac{dV}{dt}+\frac{1}{\gamma-1}\cdot V\cdot\frac{dP}{dt} \tag{1}$$

Where γ is ratio of specific heat, dV/dt is chamber volume change with time, dP/dt is in-chamber pressure change with time, P is in-chamber pressure, V is chamber volume

In this paper, the heat release analysis was studied in constant volume combustion chamber where the chamber volume was kept constant then the dV/dt term could not be considered. Hence, heat release rate for this study is express as follows:

$$\frac{dQ}{dt}=\frac{1}{\gamma-1}\cdot V\cdot\frac{dP}{dt} \tag{2}$$

The ignition delay is defined as the time from start of injection (SOI) to the start of combustion (SOC) where heat release rate recover from the negative value due to heat absorption [14, 15] as illustrated in Figure 1. The total heat release can be integrated from the heat release rate curve area from the start of combustion to the end of combustion (EOC) where heat release rate decrease to the negative value.

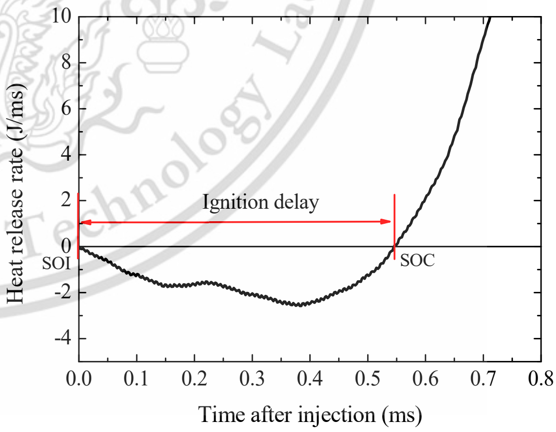

Fig. 1 The definition of the ignition delay

3. EXPERIMENTAL SETUP AND PROCEDURES

3.1 Experimental setup

Figure 2 shows the schematic diagram of spray combustion characteristics experiment using CVCC with shadowgraph technique. The CVCC was a circular cylinder with 502.65 cm^3 in volume. The combustion chamber has 80 mm in diameter, 100 mm in depth, the quartz window were equipped in this chamber for optical assessment. There are six main parts of combustion chamber: an intake valve was used to control partial pressure of the premixed gaseous mixture of acetylene (C_2H_2), oxygen (O_2) and nitrogen, (N_2), an exhaust valve was used to remove burned-gas, the spark plug was used to ignite gaseous mixture to generate high pressure and temperature, the mixing fan was used to run 25s gaseous mixture before spark-ignition for maintaining uniform gaseous distribution throughout the combustion of fuel, a dynamics pressure transducer (Kistler 6053CC60) and charge amplifier (Kistler 5011) were used to measure pressure rise of combustion of fuel , and a single-hole injector was installed at the top of combustion chamber to inject tested fuel. The 3-phase motor was used to drive a second generation common-rail pump to obtain high fuel injection pressure by the inverter controller. The pressure rise of auto-ignition was recorded by oscilloscope (RICOL DS1052E) with sampling rate $5\text{x}10^5$ S/s.

The shadowgraph technique was used to visualize spray combustion. The concave mirror allowed to reflecting the light from the Xenon lamp that passing through combustion chamber. The high speed video camera (Phantom Miro 3a10 Camera) with 10,000 frame per second (fps) was used to capture high speed video combustion, and a resolution of 512 x 464 pixels.

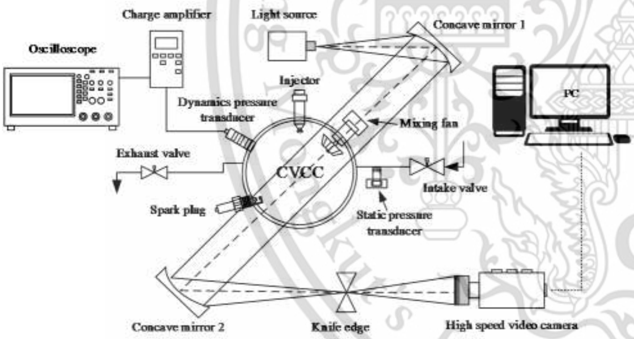

Fig. 2 Schematic diagram of spray combustion characteristics experiment using CVCC with shadowgraph technique

3.2 Experimental condition

Table 1 shows the experimental condition in this experiment. Four test fuels were tested: diesel, two HVO-diesel blends by mass: 20%(H20), and 50%(H50) and 100% HVO in constant volume combustion chamber (CVCC) with single-hole injector, 0.2 mm in nozzle orifice diameter, 2.5 ms in energizing time. The injection pressure was kept constant at 100 MPa. The ambient pressure and temperature were controlled at 4.0 MPa and 900 K that refer to diesel combustion conditions at the end of compression stroke.

Table 1. Experimental condition

Parameter	Conditions
Test fuels	Diesel, H20, H50, HVO
Nozzle orifice diameter	Single hole 0.2 mm
Energizing time	2.5 ms
Injection pressure	100 MPa
Ambient pressure	4 MPa
Ambient temperature	900 K
Oxygen concentration	21%
Repeat	10 times / condition

3.3 Test fuels

Table 2 shows fuel properties. In this study, four different fuel were diesel, H20, H50 and HVO. The properties of test fuels were determined prior to the study of spray combustion characteristics. From this table, H20, H50 and HVO have lower viscosity by 4.62%, 10.49% and 18.52%, respectively, with lower density than diesel by 1.21%, 2.91%a and 5.58%, respectively. But higher heating values by 0.39%, 0.11% and 2.18%, respectively. Moreover, one of important property in this study was distillation temperature that describes the evaporation of fuel. At T90, H20, H50 and HVO have lower distillation temperature than diesel by 0.21%, 0.70% and 16.77%, respectively. If fuel has lower distillation temperature that will be show fast evaporation and well-mixed with surrounding air dramatically. The cetane index is calculated from fuel density, T10, T50 and T90 distillation temperature that used to estimate the cetane number of fuel. The cetane index of H20, H50 and HVO are higher than diesel by 4.86%, 13.05% and 27.23% due to its lower T10, T50 and T90, respectively.

Table 2. Fuel properties

Properties	Standard	Diesel	H20	H50	HVO
Kinematic viscosity (40°C, cSt)	ASTM D445	3.24	3.09	2.90	2.64
Density (30°C, kg/m^3)	ASTM D4052	824	814	800	778
Heating value (MJ/kg)	ASTM D240	45.86	46.04	46.38	46.86
Distillation T10 (°C)	ASTM D86-11b	207.7	210.7	216..3	227.4
Distillation T50 (°C)	ASTM D86-11b	287.9	284.5	281.4	278.2
Distillation T90 (°C)	ASTM D86-11b	352.3	345.2	327.4	293.2
Cetane index	ASTM D4737	60.43	63.37	68.32	76.89

3.4 Experimental procedure

The spray combustion characteristics were investigated under simulated diesel combustion conditions in CVCC by using two-step combustion that can be illustrated in Figure 3. The first step was using spark plug to generate high-pressure and high-temperature ambient gas by burning the premixed gas of C_2H_2, O_2 and N_2 as premixed combustion period. Before the second step, in-chamber pressure and temperature were decreased to reach diesel combustion condition due to the heat transfer as cool down process. The second step was fuel injection into combustion chamber, then injected fuel was continuously burnt as combustion of fuel.

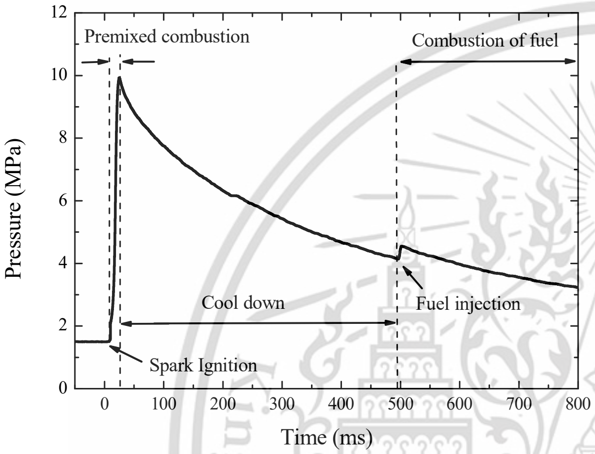

Fig. 3 Pressure history in combustion chamber

4. RESULTS AND DISCUSSIONS

The investigation on the spray combustion characteristics of diesel and HVO blend percentage under simulated diesel combustion were derived into two parts. In the first part, the heat release rate, the ignition delay, and the integral heat release were discussed. In the second part, we focused on the shadowgraph images that describe effects of fuel properties on the spray evaporation, ignition delay and flame development.

4.1 Heat release rate

Figure 4 show the effects of diesel and HVO blend percentage on the heat release rate. This study calculated the heat release rate in CVCC based on the first law of thermodynamics [14]. HVO and blends show that it early observed SOC as early observed heat release rate recover from SOI due to its high cetane number that make its short ignition delay [11,12], In addition, HVO and blends show slightly higher the peak of heat release rate than diesel due to its higher heating value Therefore, the employments of an increase in HVO blend percentage are to improve ignitability of fuel and to increase heat release rate.

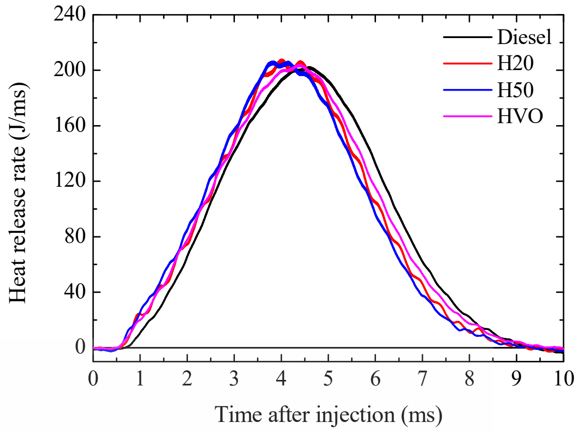

Fig. 4 Heat release rate

4.2 Ignition delay

Figure 5 show the effects of diesel and HVO blend percentage on the ignition delay. The ignition delay in this study defined as the period from SOI to SOC where heat release rate recover from the negative value due to heat absorption. The ignition delay was significantly decreased with increasing HVO percentage by 14.27%, 19.28% and 34.48%, respectively compare to diesel due to its higher cetane number. Another reason for shorter ignition delay of HVO and blends is the lower distillation temperature which make its faster evaporation with surrounding air in the chamber, and it might contribute to small droplet size distribution [15,16], which is related to a better mixture formation. The conclusion in effects of HVO and blends on the ignition delay is that increasing HVO blend percentage lead to a decrease in ignition delay so that combustion can be improved.

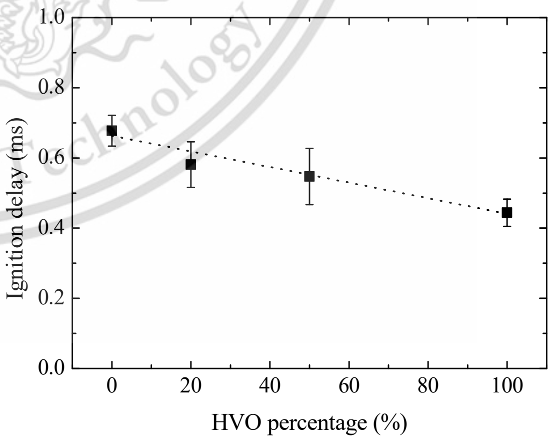

Fig. 5 Ignition delay

4.3 Integral heat release

Figure 6 shows the integral heat release as total amount of heat release where emitted from combustion of fuel, which is calculated from the integration under heat release rate curve area from SOC to EOC. The integral heat release in this study describes as the heat of combustion of injected fuel. The results showed that increasing HVO blend percentage show insignificant effect on the integral heat release due to similarity of energy input [17].

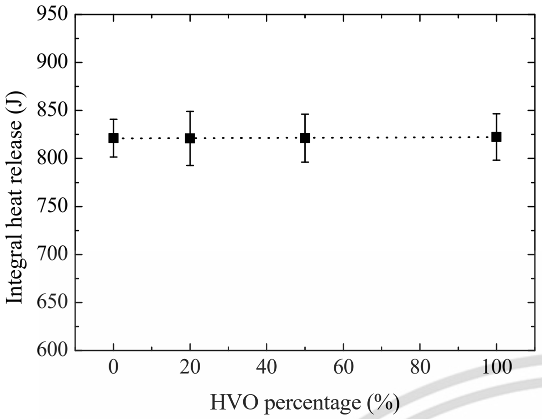

Fig. 6 Integral heat release

4.4 Combustion visualization and shadowgraph images

The sequence shadowgraph images of spray combustion of this study were capture by using the high speed video camera. The discussion on the effects of diesel and HVO as a representative on the fuel evaporation, the mixture formation and the combustion progress, such as viscosity, density and distillation temperature were illustrated in Figure 7.

In this figure, the first column indicated time after energizing, the second and the third column indicated diesel and HVO as a representative. At 0.5 ms, HVO observed slight long spray penetration and wide spray angle than diesel. At the time of 0.7 ms, HVO observed shorter liquid phase as it shows small dark spray, fast spray vaporization and well-mixing with surrounding air compare diesel as it observed larger fuel evaporated around the tip of spray due to its lower density, viscosity and distillation temperature at T90% by 18.52%, 5.58% and 16.77%, respectively, which might contribute to small droplet size distribution, lead to reducing in fuel-air mixing time. At 0.9 ms, Both diesel and HVO have already ignited as shown as flame luminosity. After 0.9 ms, diesel and HVO have continuously burnt as seen as flame development, and reach to chamber wall around 1.4 ms. The combustion progress of diesel and HVO show similar due to difference of flame luminosity area by 1.31%. But after the flame reach to the wall at 1.4 ms, the flame luminosity area of HVO show significant lower than diesel by 12.31%, 6.45%, 18.17% and 21.03%, respectively following the time after energizing of 1.9 ms, 2.9 ms 3.9 ms and 4.4 ms, respectively. This possibility is that diesel has generated more soot form than HVO, as displayed by larger flame luminosity area which emitted from the thermal radiation of soot particles [18], that agreement with previous research [7-8, 10, 12-13].

Time after energizing Diesel HVO

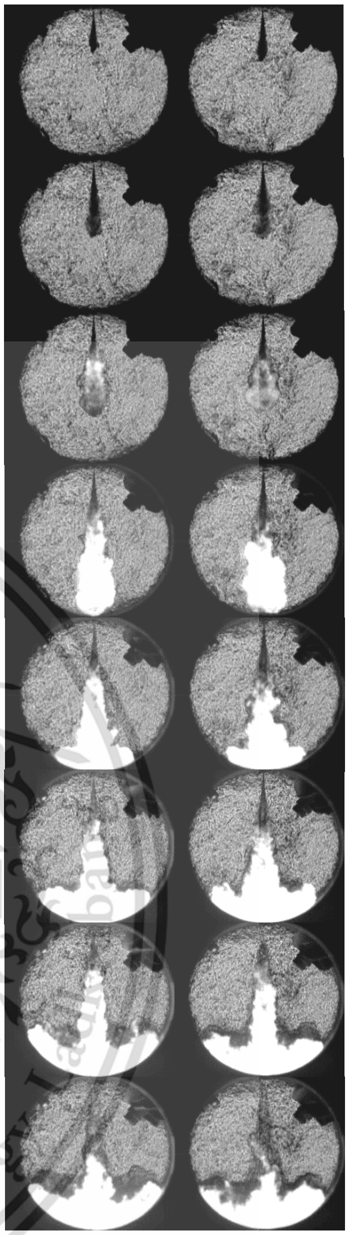

Figure 7 Shadowgraph images of combustion processes of diesel and HVO at ambient pressure of 4 MPa and ambient temperature of 900 K

5. CONCLUSIONS

The effects of HVO-diesel blend fuels on the ignition delay and spray combustion characteristics in CVCC under simulated diesel combustion condition were investigated using heat release analysis and shadowgraph images. The heat release rate, the ignition delay, the integral heat release and shadowgraph images were summarized as follows:

(1) The heat release rate of HVO and blends show similar results to diesel under constant injection pressure, ambient pressure and temperature.

(2) Increasing HVO blend percentage lead to decrease the ignition delay due to its higher cetane number and lower distillation temperature.
(3) Increasing HVO blend percentage shows insignificant improvement in the integral heat release due to the same of energy input.
(4) On the shadowgraph images of combustion processes, HVO can reduce time of mixing processes that make better mixture formation as more fuel vaporize, and fuel-air interaction inside combustion chamber very well during combustion phase due to its better evaporation characteristics.
(5) The combustion process of diesel and HVO show slight difference as seen as flame development. In addition, diesel shows larger flame luminosity area than HVO that might generate high soot formation.

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