

**THE EFFECTS OF ETHANOL INJECTION IN EGR ON PERFORMANCE  
AND EMISSIONS OF AN ETHANOL FUMIGATED DIESEL ENGINE**



**A THESIS REPORT SUBMITTED IN PARTIAL FULFILLMENT  
OF THE REQUIREMENTS FOR THE DEGREE OF  
MASTER OF ENGINEERING IN AUTOMOTIVE ENGINEERING  
INTERNATIONAL COLLEGE  
KING MONGKUT'S INSTITUTE OF TECHNOLOGY LADKRABANG  
ACADEMIC YEAR 2018  
KMITL-2018-IC-M-004-002**

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**Kontorn Thammakul**



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|---------------------|--|
| <b>Thesis Title</b> | The effects of ethanol injection in EGR on performance and emissions of an ethanol fumigated diesel engine |
| <b>Student Name</b> | Mr.Kontorn Thammakul   |
| <b>Student ID</b>   | 58610004   |
| <b>Degree</b>       | Master of Engineering  |
| <b>Program</b>      | Automotive Engineering   |
| <b>Advisor</b>      | Asst.Prof.Chinda Charoenphonphanich  |
| <b>CO-Advisor</b>   | Dr.Manida Tongroon   |
| <b>CO-Advisor</b>   | Prof.Dr.Hidenori Kosaka  |

### ABSTRACT

Fossil fuel is the main source of energy for human to be used in the number of applications. That is the reason why it is undeniable that the need of fossil fuel is increasing. However, fossil fuel is very difficult to be produced. It needs very long time for organic material to be buried and formed into crude oil. Consequently, the amount of fossil fuel seems to be decreasing, unavoidably. Furthermore, the emissions from the usage of fossil fuel are needed to be concerned. They can cause many unhealthy problems to living things and environment. Many organizations have enacted an abundant of emission regulations to control the level of emissions and try to reduce their numbers. Another way to help reduce using of fossil fuel is finding the new source of energy. The alternative energy such as renewable energy is more sustainable than the energy from fossil fuel and it is cleaner because it is mostly produced from plant called bioenergy. Ethanol is one of bioenergy and has been used to fuel the engine for years. The usage of alcohol with diesel engine can be applied by many ways such as blended fuel, dual injection and Fumigation system. Ethanol Fumigation requires simple modifications on the engine. That reason makes this system become more attractive for using alcohol as the secondary energy to fuel the engine. This system could help reduce NO<sub>x</sub> which is the main emissions that concern to diesel engine. In contrary, the amounts of HC and CO are increasing because of the lowering of the combustion temperature caused by the ethanol. The reduction of NO<sub>x</sub> can be caused by other technique, EGR. Most of diesel vehicles already install an Exhaust Gas Recirculation system which will dilute the amount of Oxygen presence in the intake charge to reduce the combustion temperature and NO<sub>x</sub>. This research will focus on the effect of ethanol fumigation and EGR temperature on the diesel engine. The temperature of EGR will influence in vaporization of the ethanol and it is supposed to promote the combustion temperature and reduce more emissions as the expected result.

## **Acknowledgement**

I would like to send all of my respect and gratitude to my advisor, Dr.Manida Tongroon, for encouraging me and bringing me to the right path of work. All of my knowledge, both from studying and working in the Renewable Energy Laboratory at National Metal and Materials Technology Center (MTEC), it could not be found from anywhere else. She always provides a good advice to me in both work and life view and always backs me up all the time whenever I am depressed. I also would like to pay a grateful thank to my co-advisors, Asst.Prof.Dr.Chinda Charoenphonphanich and Dr.Preechar Karin from King Mongkut's Institute of Technology Ladkrabang and Prof.Hidenori Kosaka from Tokyo Institute of Technology for guidance and advice of this research.

I would like to pay a sincere gratitude to TAIST-Tokyo tech for providing the scholarship of this program and also Hino Motors for the daily expense scholarship through my first two years. I also would like to thank NSK, Japan for selected me to become an internship student during late 2016 which was my first time abroad and opened the world to me.

I would like to thank to all of my colleagues, friends, trainee, technicians and lab staffs for helping me through the all related technical stuffs of this work. This work would be a lot more difficult without those helps.

Finally, I would like to express my profound gratitude to my family and my girlfriend for supporting and giving an encouragement to me throughout these years of study and doing the research. They are also the reasons of my ability to fight through these years and also the reasons to never stop learning. Thank you all for everything.

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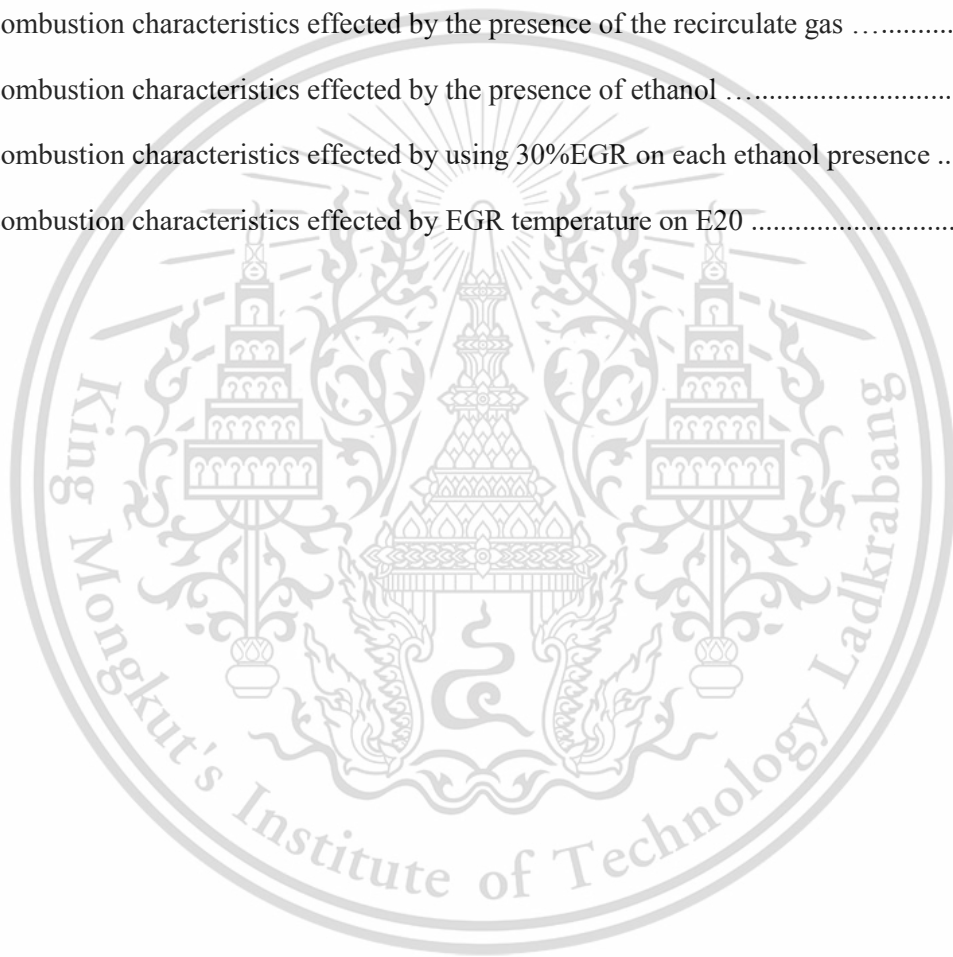
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## CHAPTER 1

### Introduction

Fossil fuel has been used as the main source of energy for very long time. It is undeniable that fossil fuel is necessary. People use fossil fuel in many ways such as in transportation system, agriculture and industrial fields. However, fossil fuel is decreasing because of increasing high demand. Many organizations have urged and convinced people to solve this problem by using the sustainable and renewable energy which is somehow cleaner and friendlier to an environment than fossil fuel. Some sustainable or renewable fuels can be completely used as a main fuel instead of fossil fuel. They have been applied for vehicles in automotive industry, household generator, agricultural engine or even a power plant in an industry over the last decade. Alcohol is one of the most interesting biofuels that has been used as an alternative source for years. Alcohol like ethanol is the one that people have used regularly. It is usually volatile-colorless liquid, flammable and has slightly chemical odor. It can be made from many kinds of sugary and starchy plants such as sugar cane, cassava, rice and corn which could be grown up in Thailand. Ethanol is often used as a blend in fuel, called gasohol, especially in gasoline fuel due to its high octane number and a good miscibility with gasoline fuel. Hence, ethanol properties are similar to gasoline fuel.

On the other hand, ethanol has been mixed with diesel fuel in the form of diesohol. Ethanol has very low miscibility with diesel. Therefore, high amount of ethanol cannot be blended with diesel homogeneously. As the result, diesel and ethanol will be obviously separated. Ethanol has low cetane number which indicates the ability of auto ignition. High cetane number is preferable for the diesel engine. This means that auto ignition may not be achieved when using high concentration of ethanol in the blends. In addition, the use of alcohol in high percentage with diesel fuel has tendency to lower the combustion temperature due to its high level of heat of vaporization. Ethanol will absorb the heat in the combustion chamber to vaporize itself prior to the combustion process. This lower combustion temperature leads to the reduction of NO<sub>x</sub> emissions. Ethanol also has the low lubricity which can cause problem on the diesel fuel supply system, particularly common rail injection parts that require high pressure of injection.

According to the Alternative Energy Development Plan (AEDP 2015), issued by Ministry of Energy [1], ethanol has been aimed to be used for 11.3 million liter per day by 2036. However, ethanol, mainly in a form of gasohol, has been consumed only 3.35 million liter per day, recently [2]. This number is lower than the expectation even though the ethanol is mixed in the high percentage such as E10, E20 and E85. Hence, the method of using ethanol in other applications like using with diesel, which has about three time usage level higher than gasoline [3] as shown in figure below, seems to be interesting and conforms to the AEDP plan.

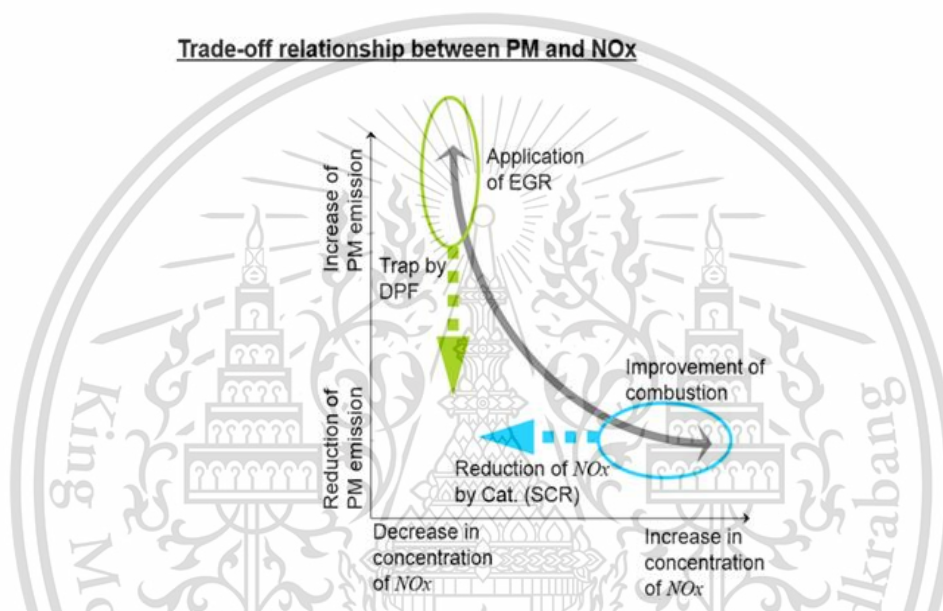
**Table 1 Demand and supply of fuel of year 2017**

| 2017                    | Amount (Thousand barrels/day) |          |        |        | Change (%) |          |        |        |
|-------------------------|-------------------------------|----------|--------|--------|------------|----------|--------|--------|
|                         | Used                          | Produced | Import | Export | Used       | Produced | Import | Export |
| Gasoline                | 189                           | 218      | 30     | 28     | 3.4        | 4.1      | 13.8   | 15.4   |
| Gasoline                | 8                             | 9        | -      | 1      | -8         | -15.5    | -      | -12.6  |
| Gasohol 91              | 67                            | 95       | -      | 26     | -4.6       | 0.2      | -      | 16.2   |
| Gasohol 95              | 114                           | 114      | -      | 0.05   | 9.8        | 9.7      | -      | 578.9  |
| Diesel                  | 401                           | 464      | 14     | 90     | 2.7        | 4.7      | -1.9   | 9      |
| Kerosene                | 0.1                           | 34       | -      | 0.2    | -29.7      | -11.4    | -      | -54    |
| Aviation fuel           | 116                           | 128      | 1      | 13     | 4.3        | 5.9      | -44.8  | 36.4   |
| Heavy fuel oil          | 37                            | 101      | 1      | 60     | -6.8       | 3.8      | -50.9  | 9.2    |
| Liquified petroleum gas | 202                           | 192      | 20     | 8      | 3.3        | 5.9      | 40.9   | 192.6  |

Diesel engine is one kind of internal combustion engine. It starts the combustion process by compressing the air to the high pressure and temperature state. When the piston approaches to the top dead center, the reactive fuel is injected directly in to the chamber. Then, the combustion will begin, without the need of any assisted ignition devices such as a spark plug. This combustion process is called compression ignition. Diesel engine has higher thermal efficiency than gasoline engine. It can produce more work with the same amount of energy input. It also has high durability. Due to these benefits, diesel engine is often used for industrial and agricultural fields as well as transportation sector.

Although diesel engine seems interesting and has been widely used, it emits hazardous emissions. The main emissions from diesel engine are hydrocarbon (HC), nitrogen oxide (NO<sub>x</sub>), carbon monoxide (CO) and particulate matters or soot. The factors that affect all these main emissions are the amount of oxygen available in the combustion chamber and the combustion temperature. The hydrocarbon and soot can be found in the low temperature combustion and rich mixture condition. This condition of combustion cannot burn the carbon and hydrogen efficiently and also causes the formation of CO due to the incomplete combustion. The unburned carbon and hydrogen will form as the hydrocarbon in exhaust gas and the small solid particles called soot or particulate matter (PM). These emissions are irritants and carcinogen when they are in atmosphere. From these reasons, it seem like the way to reduce these emissions can be achieved by increasing the combustion temperature. But the higher combustion temperature will increase the tendency for nitrogen in the combustion chamber to dissociate from each other and participate with oxygen atoms, thus forming nitrogen oxide (NO<sub>x</sub>). NO<sub>x</sub> is hazardous to respiratory and skins. It also causes an acid rain that can corrode houses and buildings, worsens an environment. This ends up as the trade-off relationship between NO<sub>x</sub> and PM as shown in Figure 1.

Exhaust Gas Recirculation (EGR) is the exhaust gas reduction technique, mainly used in the diesel engine. Its operation principle is using the non-reacting gas in the exhaust emissions to dilute the oxygen content in the intake mixture. The lack of oxygen available in the intake charge will deteriorate the combustion, resulting in the lower in-cylinder temperature. As the result, the tendency of  $\text{NO}_x$  formation will be reduced. This system is a good approach to reduce  $\text{NO}_x$  emission. However, by using this system, the performance of the engine such as thermal efficiency and specific fuel consumption might become worse. So the EGR system will not always be operating. It will operate in just some of proper operating conditions to maintain the trade-off between  $\text{NO}_x$  and soot emissions.



**Figure 1 Trade-off relationship between PM and  $\text{NO}_x$**

There are many researches about the ways to use ethanol in diesel engines [4]. Since ethanol can be produced from Thai economic starchy plants (Figure 2), ethanol can be used as the blends, dual injection and fumigation technique [5, 6, 7, 8]. Each approach have different drawbacks such as a phase separation between diesel and ethanol in blended fuels. The separation can be prevented by adding an emulsifier or a co-solvent. Finding the proper kind of emulsifier/co-solvent and the appropriate amount are maybe the new concerns. Because of the limited area on the cylinder head, the installation of the secondary injector to inject the ethanol as the secondary fuel in dual injection technique has to be carefully considered. For the fumigation technique, ethanol is injected inside the intake port through the port fuel injector (PFI) and mixed with an intake air. Without blending the fuel, the fumigation method allows more amount of ethanol to be used with diesel fuel. Due to high heat of vaporization of ethanol, the heat from the combustion would be reduced, thus lowering  $\text{NO}_x$  emission. However, the  $\text{NO}_x$  formation is also depended on the operating conditions. Not only combustion temperature decreases, but also more excess oxygen from the ethanol molecule could promote the  $\text{NO}_x$  formation.



**Figure 2 Starch plant for producing ethanol**

The effects of EGR in conjunction with ethanol fumigation system on the exhaust emissions have been studied [6, 7]. It is found that EGR and ethanol decrease the temperature of the combustion. This clearly could help to reduce the NO<sub>x</sub> in the exhaust gas. On the other hand, the high amount of EGR and ethanol causes the increase of other emissions, especially HC and CO. Thus, the optimal amount of both EGR and ethanol should be determined to prevent an unwanted increase of emissions. Moreover, the temperature of EGR may alter the formation of exhaust gas concentration and should be investigated.

For combustion characteristics such as in-cylinder pressure and heat release rate, there is less data available from the past researches. Therefore, an idea to study about the effect of EGR temperature and ethanol fumigation technique on the combustion characteristics, engine performance and exhaust gas emissions has been conducted. The hypothesis of this research is if the EGR is fed to the intake port at the place where ethanol injector is installed, the temperature of EGR will assist in vaporization of ethanol. Ethanol will be better to vaporize itself and mixed with air properly. It may cause ethanol reforming process and has the effect to the combustion characteristics and emissions.

### 1.1 Research objectives

- To study in combustion characteristics that effected from fumigated ethanol and the temperature of the recirculate exhaust gas at intake port.
- To compare and study on the performance and emissions from ethanol fumigated diesel engine with the different temperature of the recirculate exhaust gas.
- To develop the technical knowledge in renewable energy and internal combustion engine and introduce the use of alcohol-type alternative fuels on diesel engine.

## 1.2 Expected results

- EGR temperature will show trends of influence on the effect of emission, performance and the combustion characteristics.
- The Ethanol fumigation system can be compatible with EGR system and has more effectiveness to reduce the emissions.
- The usage of fossil fuel, diesel fuel, will be reduced. The ethanol usage will be promoted to complete the national renewable energy plan and balance the usage of gasoline and diesel fuel.
- This research can be used as a reference to develop the diesel engine in the field of using the flexible fuel by maintaining the effectiveness close to present diesel engine while reducing the emissions, simultaneously.
- This research can produce more skilled researchers in the field of internal combustion engine and renewable energy and can develop more knowledge to automotive field and produce technical data to define the national energy plan.

## 1.3 Thesis Outline

This thesis dissertation will be split in to 5 Chapters, starting by the introduction, this part, is Chapter 1.

Chapter 2 will be covering the literature reviews of past relevant past works, mostly focus on the performance and emissions. It will be divided into 2 parts which is the works based on the use of ethanol as the secondary fuel on diesel engine in the view point of both ethanol fumigation and blends, and another part is the effect of using the exhaust recirculate gas system.

Chapter 3 presents the testing apparatus which includes the tools and equipment that is necessary for the experiment and the testing conditions of this research.

Chapter 4 will be discussing on the experimental results. This part is divided into 4 parts. It will start by introducing the effect of using exhaust recirculate gas on the diesel engine by varying the rate of recirculate gas of 10%, 30% and 50% following by discussion on the effect of ethanol fumigation system by 10%-30% of energy portion from ethanol on both non-EGR and with EGR operating conditions. The effect of the temperature of the recirculate gas will also be discussed on the different temperature as follows: 60°C, 80°C and 100°C and finally the combustion characteristics on ethanol fumigation and EGR temperature influences.

The summary and conclusion of this research will be provided in Chapter 5, in the scope of emissions, performance and combustion characteristics.

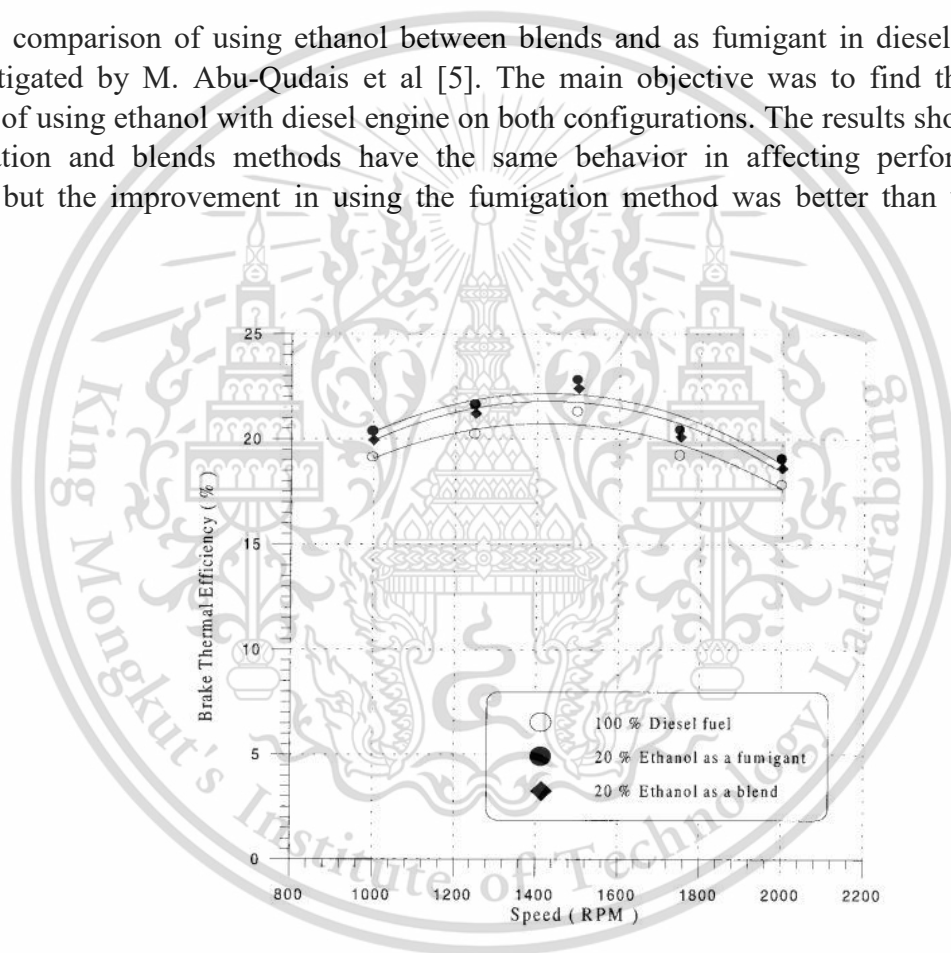
## CHAPTER 2

### LITERATURE REVIEW

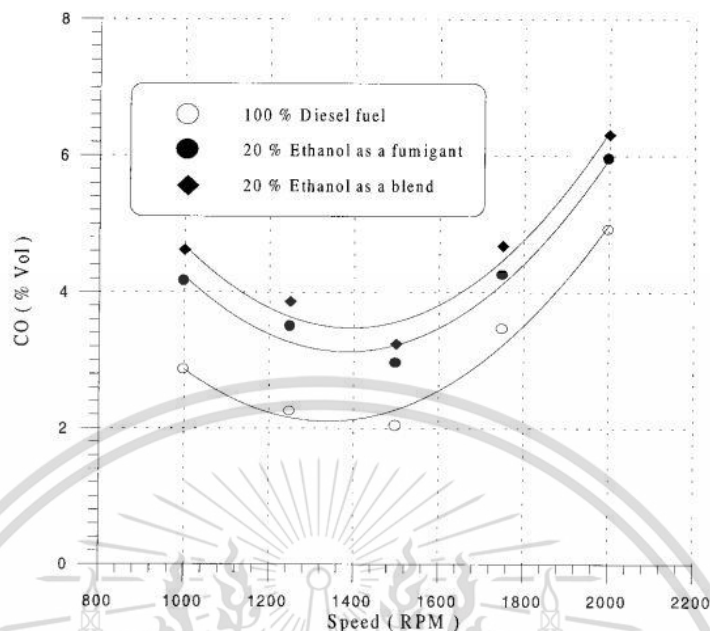
This section will be divided into two parts. The first part will show the past works that have been conducted on the diesel engine with the use of ethanol as the secondary fuel by both blend and the fumigation to refer to the effect of the ethanol on the diesel engine following by the section that will show the effect of using EGR in the diesel engine. The effect of both parameters will be concluded at the latest of this chapter.

## 2.1 Use of ethanol in the diesel engine

The comparison of using ethanol between blends and as fumigant in diesel engine has been investigated by M. Abu-Qudais et al [5]. The main objective was to find the optimum percentage of using ethanol with diesel engine on both configurations. The results show that both the fumigation and blends methods have the same behavior in affecting performance and emissions, but the improvement in using the fumigation method was better than when using blends.

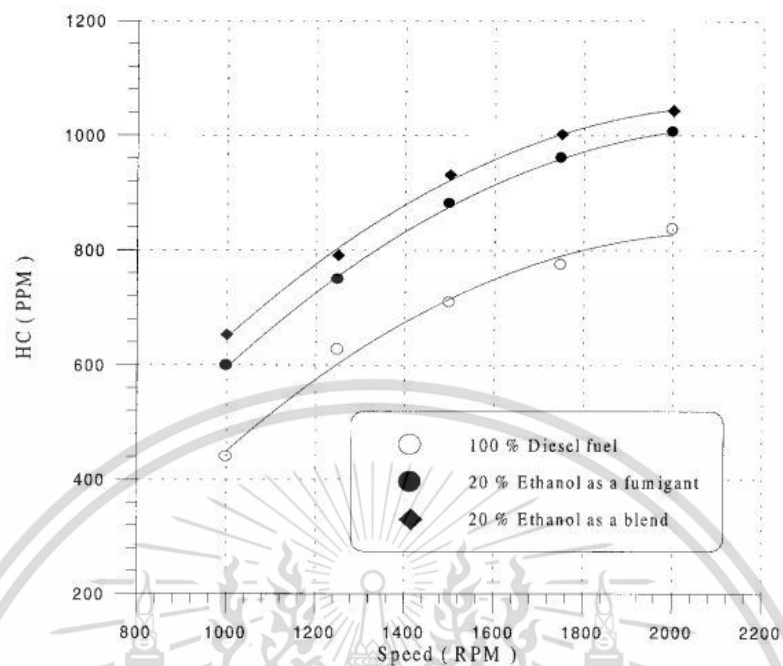


**Figure 3 Brake thermal efficiency versus speed for ethanol fumigation and for fuel blends**

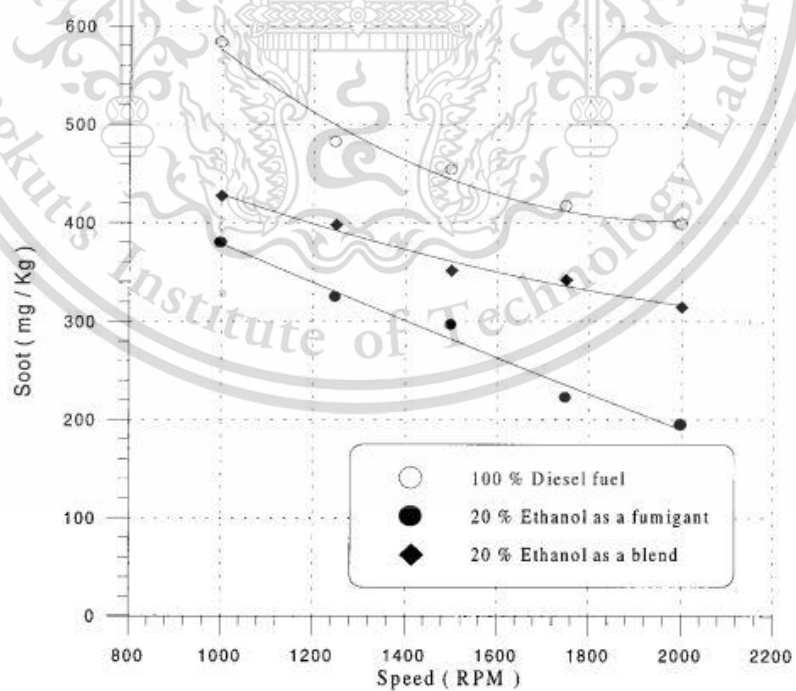


**Figure 4 CO emissions versus speed for ethanol fumigation and for fuel blends**

Thermal efficiency is slightly increased with increased ethanol substitution as depicted in Figure 3. It may be attributed to the increase in the ignition delay, so a rapid rate of energy is released which reduces the heat loss from the engine. The blended fuels compared least favorably with fumigation at all speeds and approach the diesel fuel in some cases. The author suggests that the physical properties of diesel fuel are changed when ethanol is added in solution (blend). The addition of ethanol causes the viscosity of diesel fuel to decrease. Also, the addition of ethanol in solutions with diesel fuel causes the cetane rating to drop and the heating values to be lower and also the evaporation of ethanol in the intake air (fumigation case) lowers the intake mixture temperature and increases the density. Thus, as more air is made available in the cylinder, greater amounts of power can be generated if the right proportion of fuel is added.



**Figure 5 HC emissions versus speed for ethanol fumigation and for fuels blends**



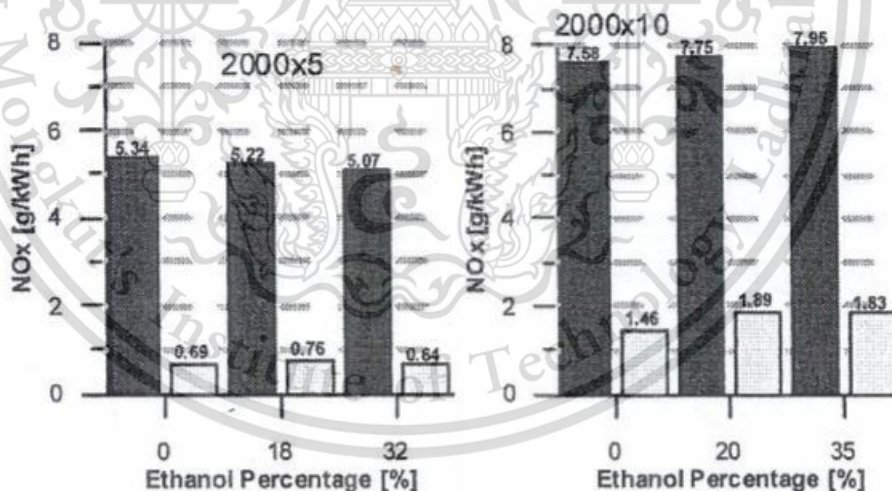
**Figure 6 Soot emissions versus speed for ethanol fumigation and for fuels blends**

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Figure 4 and Figure 5 show that the blends fuel produces the higher CO and HC emissions than the fumigation configuration. These results are caused by the incomplete combustion of the fuel. The higher heat of vaporization of ethanol might responsible for the combustion deterioration by the cooler combustion temperature. The fumigation system might have the better air utilization and led to more homogeneous charge mixture. But in the term of soot, both configurations show the lower level as show in Figure 6. The fumigation configuration had the least level of soot concentration. The charge cooling increases ignition delay and enhances the mixing of diesel fuel with the ethanol–air mixture and causing the better air utilization and less smoke.

The effect of ethanol port injection on combustion characteristics in light duty diesel engine has been studied by Gabriel Di Blasio et al [9]. This research has been conducted on a single cylinder dual fuel diesel engine. The objectives are to study about the effect of ethanol-air mixture in an intake port and to optimize the operation range and also finding out what the main control factor is. The results of this research show the reduction of NO<sub>x</sub> and soot by using ethanol port injection obviously in low to medium engine speed range (Figure 7). Thermal efficiency is increased and also the amount of HC and CO (Figure 8 and Figure 9). The rate of heat release also changes when different ethanol presence is introduced as shown in Figure 10. From the result, the main control factor is the pilot injection of ethanol. But with too much of pilot ethanol, above 30%, will make the ignition become uncontrollable. The dual fuel such as ethanol with diesel still has some limit and inferior to conventional diesel engine.



**Figure 7 NO<sub>x</sub> result for fumigation percentage without EGR (dark bars) and with EGR (white bars)**

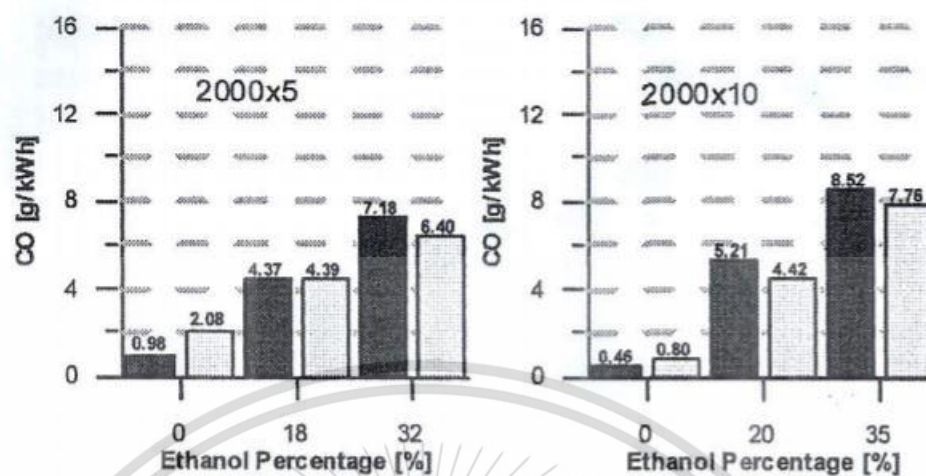


Figure 8 CO result for fumigation rate without EGR (dark bars) and with EGR (white bars)

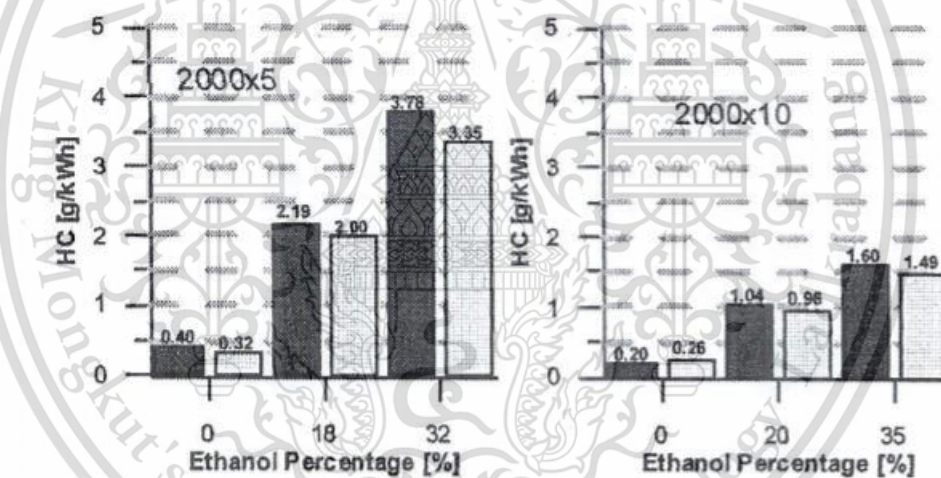
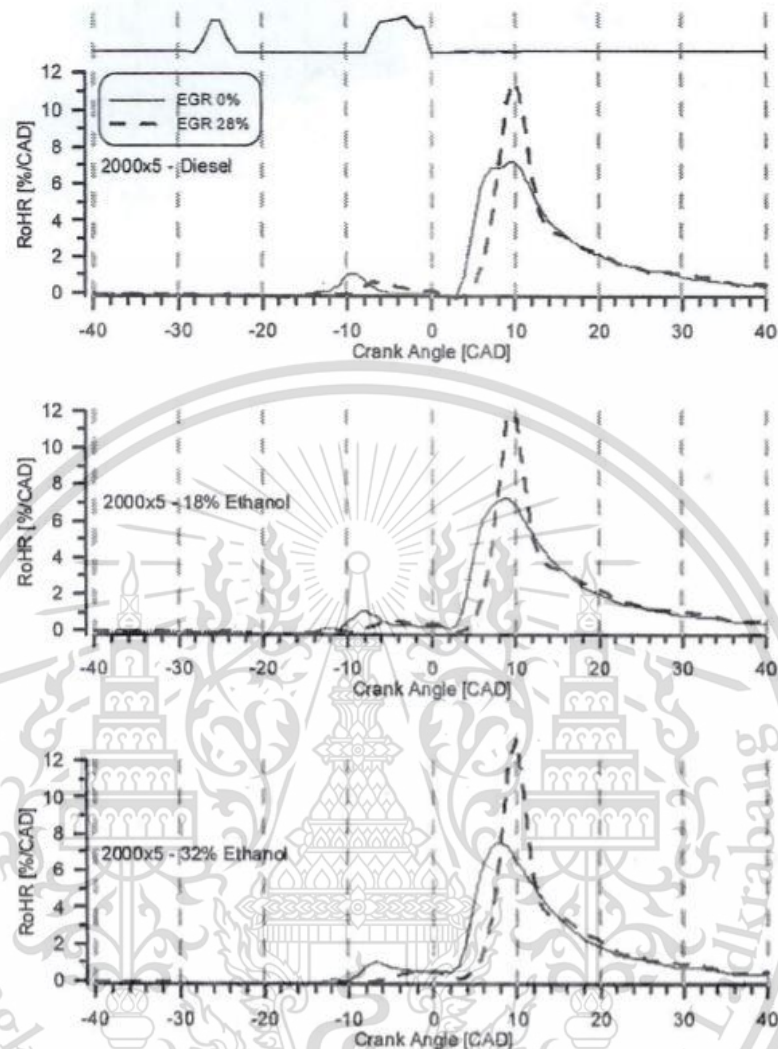
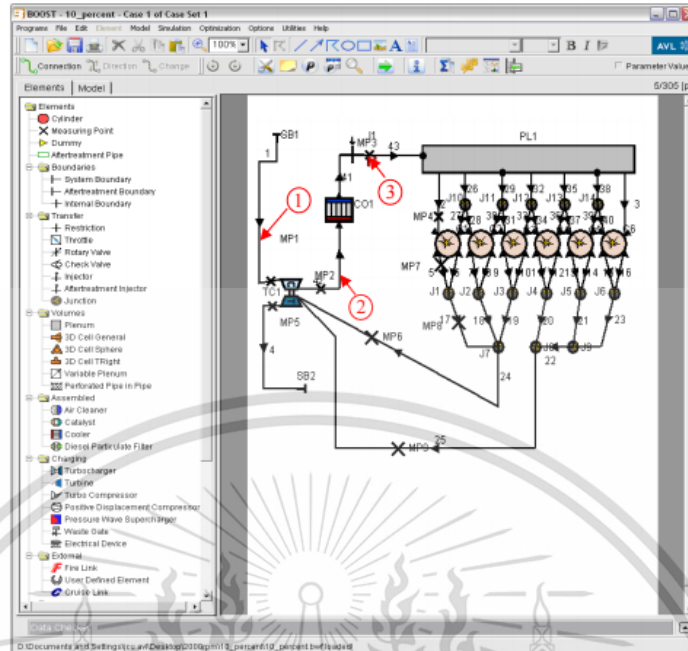


Figure 9 HC result for fumigation rate without EGR (dark bars) and with EGR (white bar)



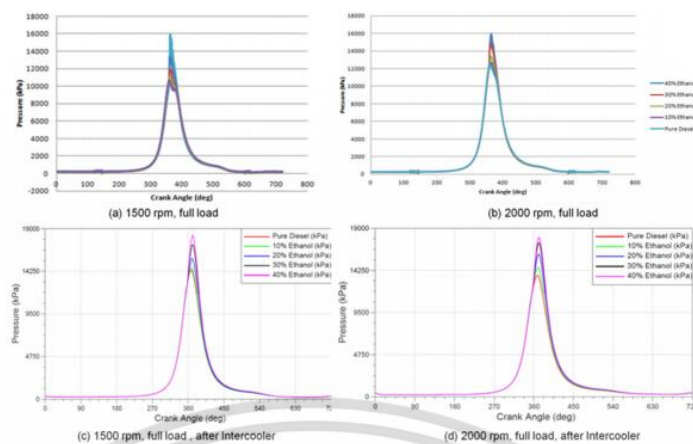
**Figure 10 Different rate of heat release from different ethanol pilot injection compared with neat diesel**

In addition, thermodynamics of diesel engine with ethanol fumigation system has been compared between experimental result and simulation result by Rong Situ et al. from Queensland University of Technology, Australia [10]. The simulation was conducted in AVL Boost engine simulation program (Figure 11) under Double Vibe combustion math model.



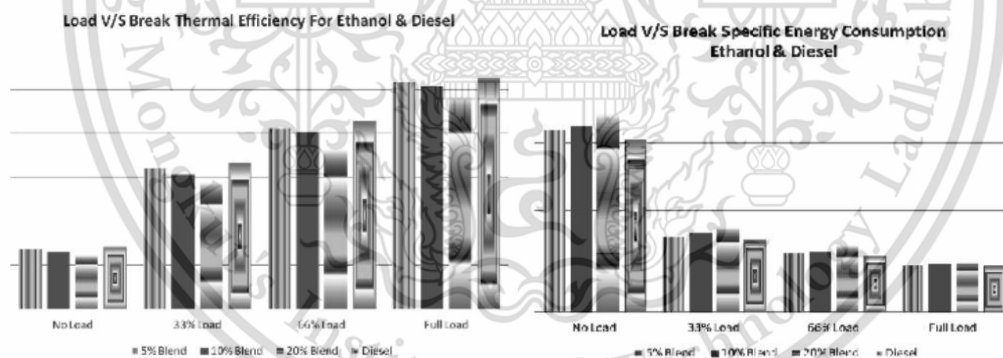
**Figure 11 Simulation model in AVL Boost**

The result indicators of this comparison are peak pressure and mean effective pressure. A six cylinder turbocharged engine is used. The ethanol fumigation system can help increase the combustion peak pressure and cause the increase of mean effective pressure as the result. Both simulation and experimental result show the same trend. But with the limitations of software calculation, the simulation result gives higher number than the experimental result (Figure 12). The ethanol injection position also has been observed. The researchers have varied the injection position as follow 1.before turbocharger 2.between turbocharger and intercooler and 3.after intercooler. The results were not different between those three cases and it cannot be concluded clearly without experiment. This may be caused from the limitations of the thermodynamics calculation of the software in some features which need to be developed further.



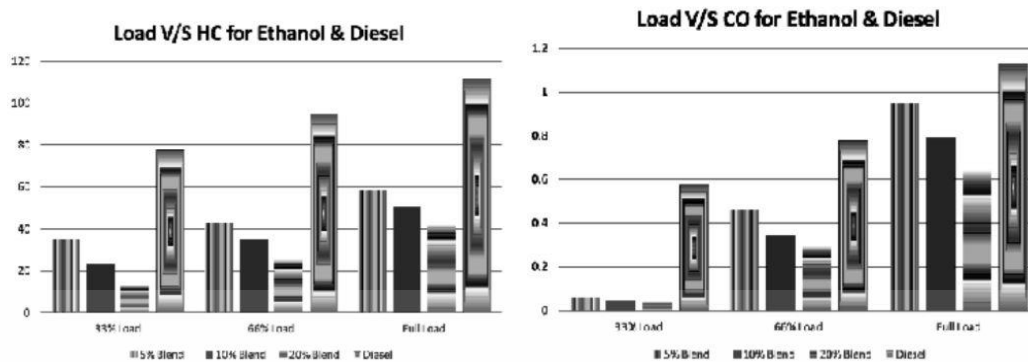
**Figure 12 Pressure comparison between simulation (upper part) and experiment (lower part)**

S.Gomasta from Department of Mechanical Engineering, Kanpur Institute of Technology, Kanpur, (U.P.) India [11] have investigated the effect of blended ethanol with diesel fuel in the view point performance and emission. The work suggests that the total fuel consumption and break specific fuel consumption of ethanol diesel blended fuel increased for the reason that low heating value of ethanol is about half of that of diesel, and it is increasing with increase in blend ratio (Figure 13). Ethanol blend show the lower brake thermal efficiency as the load is increased the difference in efficiency also increases.

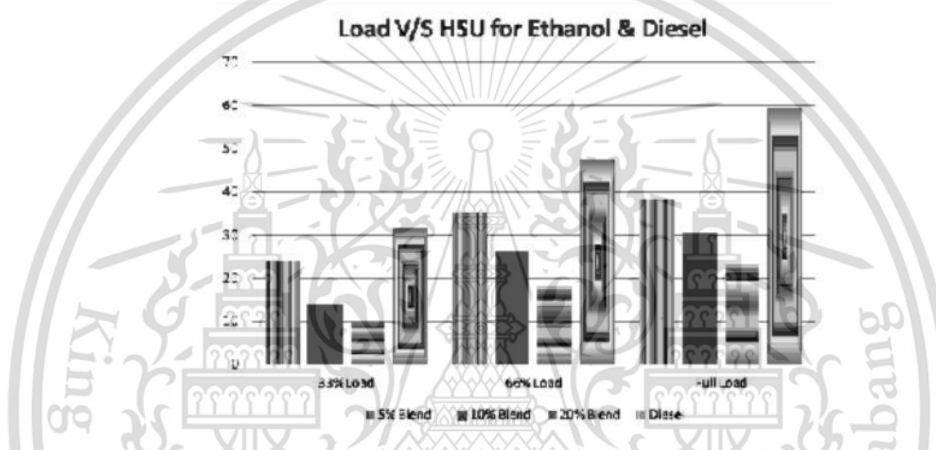


**Figure 13 Thermal efficiency and brake specific energy on different load with different percent ethanol blends**

The emission trends show the significant reduction of CO with increase in blend ratio (Figure 14), the same pattern is followed in particulate matter as well (Figure 15). It is attributed to the same physical and chemical characteristic but HC emissions were reduced with the use of the ethanol-diesel fuel blends with respect to that of neat diesel.



**Figure 14 HC and CO on different load with different percent ethanol blends**



**Figure 15 smoke opacity on different load with different percent ethanol blends**

The study of fumigation of alcohol in the multi cylinder diesel engine was conducted by Rodica A. Baranescu [12]. Three alcohol were injected and compared; methanol, ethanol and aqueous ethanol. The work suggested that the fumigation of alcohol increased the pressure rise rate comparatively with pure diesel operation (Figure 16). Alcohol-air mixture provided the faster combustion rate. All alcohol types had knocking operation occurred in the lower alcohol energy content. The fumigation of alcohol increased the peak cylinder pressure. The higher portion of alcohol led to the higher peak cylinder pressure. The increase of the cylinder pressure should be observed closely due to the prevention of the mechanical failure of the engine. Thus, that would be another concern limit of the alcohol fumigation amount.

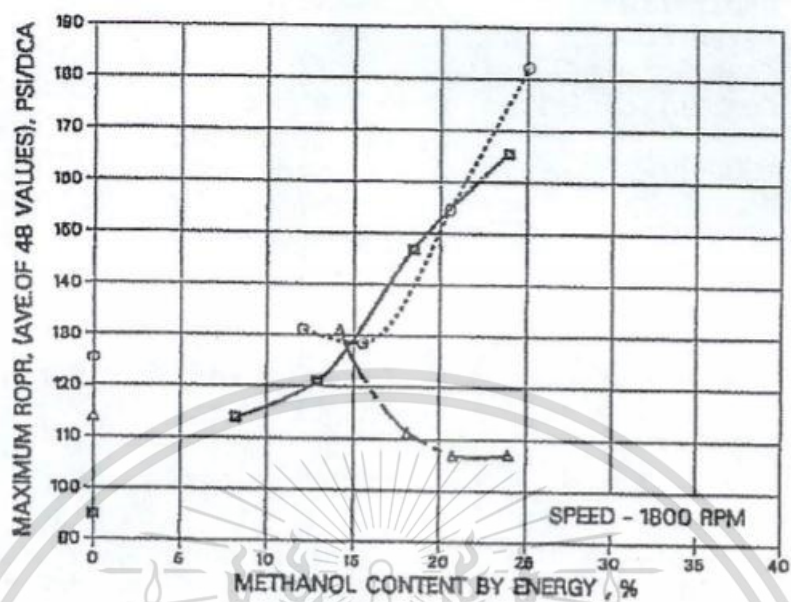


Figure 16 Rate of pressure rise of methanol fumigation (PSI/degree crank angle)

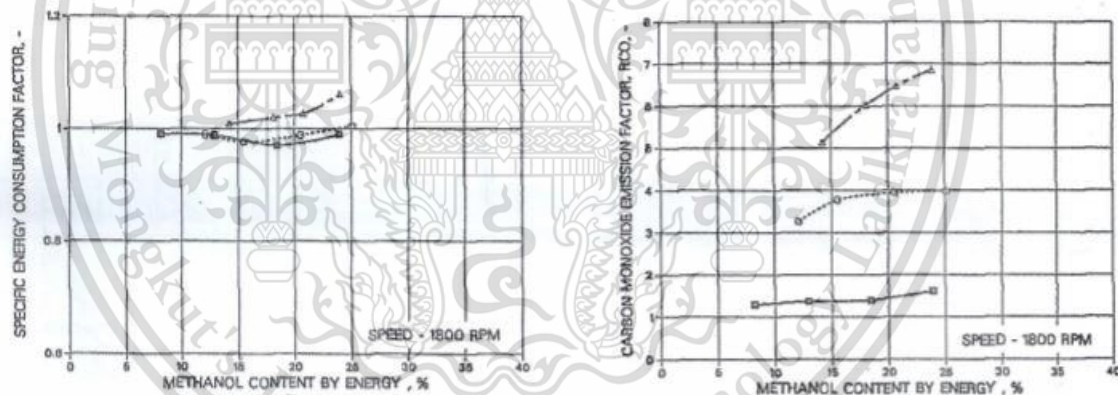
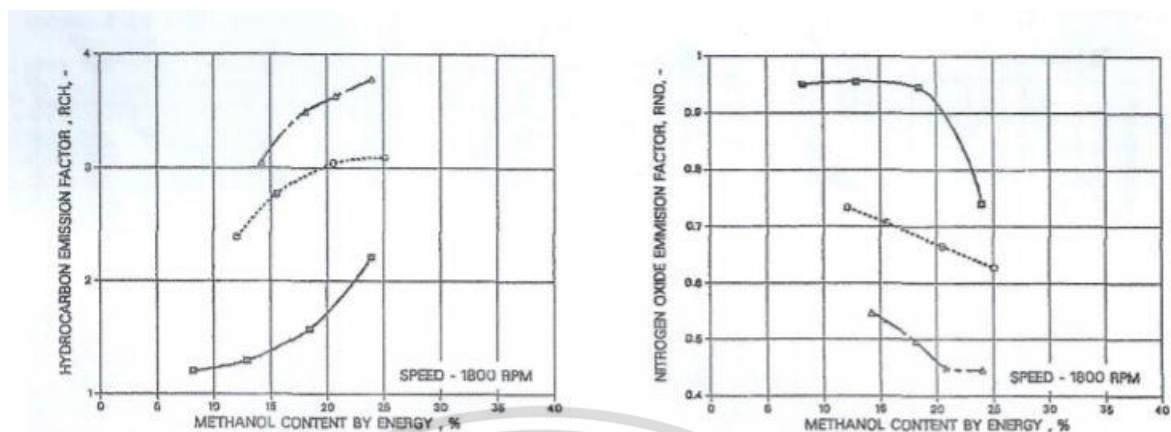
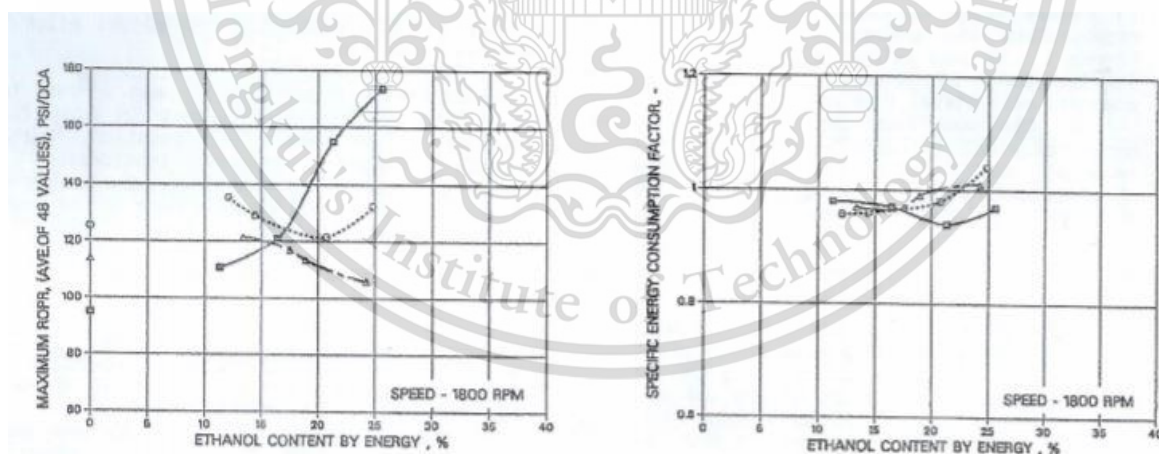


Figure 17 specific energy consumption ratio and CO emission factor with methanol fumigation

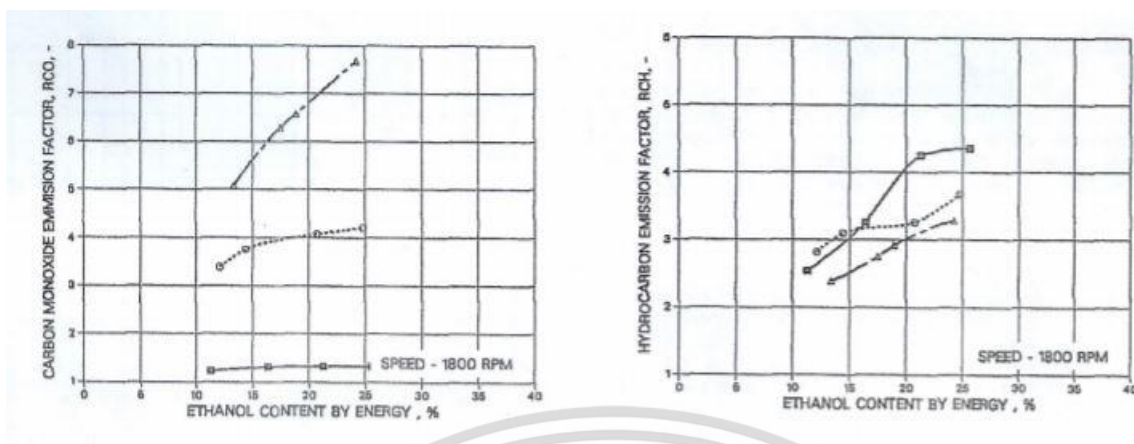


**Figure 18 HC and NOx emission factor with methanol fumigation**

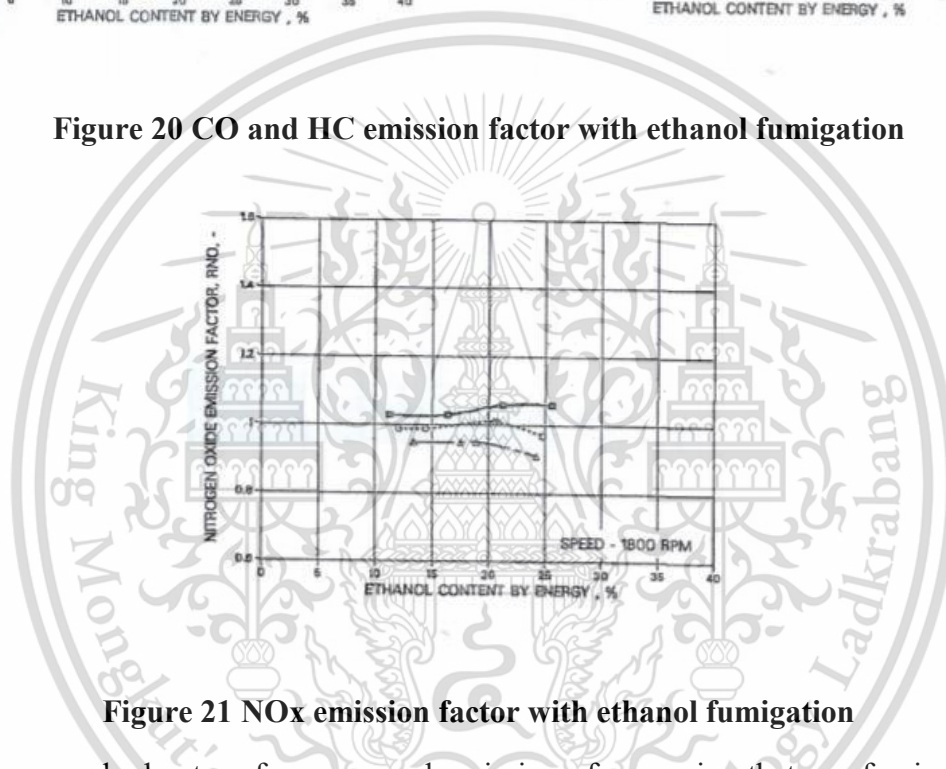
The emission with carbon-content; CO and HC were dramatically increased by the using of alcohol fumigation (Figure 17 and Figure 18). It is due to the incomplete combustion when operate on the dual-fuel, ethanol fumigation operation as shown in the increase in pressure rise rate in Figure 19. And also, the incomplete combustion leads to the less formation of NOx. The emissions problem is another limitation of using ethanol fumigation configuration. The results of the emission were shown as emission ratio between emissions generated from dual-fuel operation and pure diesel operation (Figure 20 and Figure 21).



**Figure 19 Rate of pressure rise of ethanol fumigation (PSI/degree crank angle) and specific energy consumption ratio with ethanol fumigation**

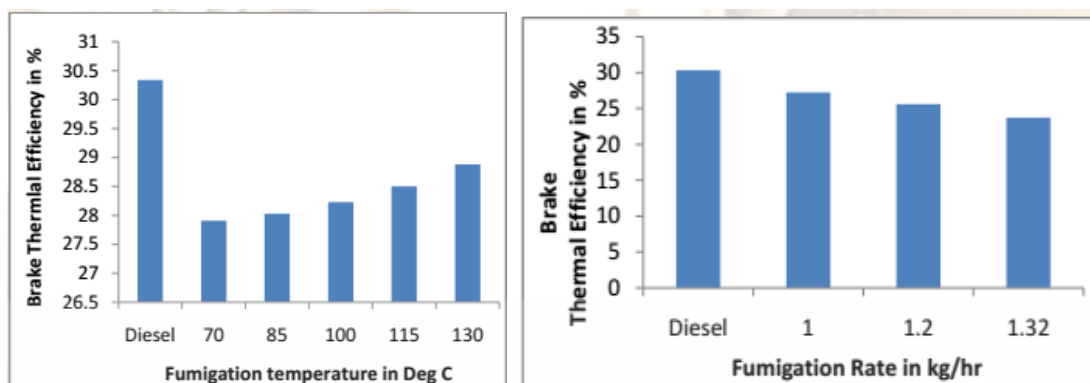


**Figure 20 CO and HC emission factor with ethanol fumigation**



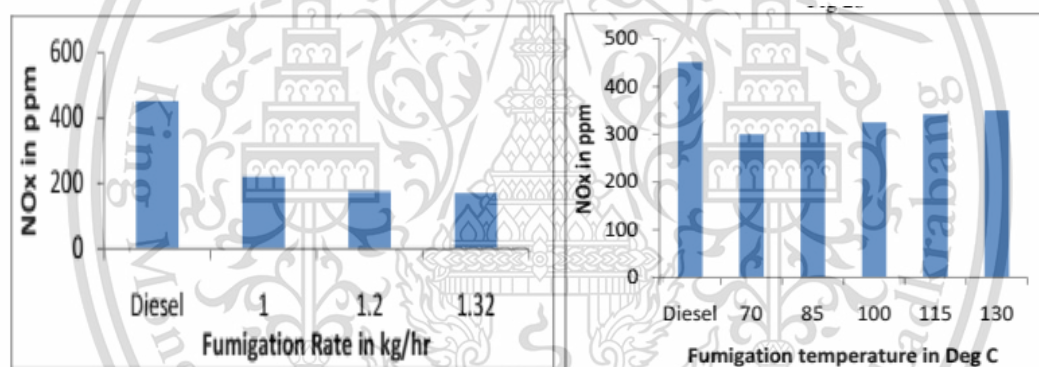
**Figure 21 NOx emission factor with ethanol fumigation**

The research about performance and emission of an engine that use fumigation system has also been conducted by A.Pannirselvam et al [8]. A single cylinder diesel engine was used in this experiment. Loads of the engine have been varied for 5 levels (0,25,50,75 and 85%). 4 rates of ethanol fumigation have been used in this experiment (0.4,0.6,0.8 and 1 kg/hr). Ethanol was heated before injection at 5 point (70, 85, 100, 115 and 130 °C). The result shows that both ethanol fumigation rate and the ethanol temperature effect to the performance of the engine. The higher rate of ethanol will decrease the thermal efficiency (Figure 22) but on the other hand, thermal efficiency will increase with respect to the temperature of heated ethanol-air mixture.

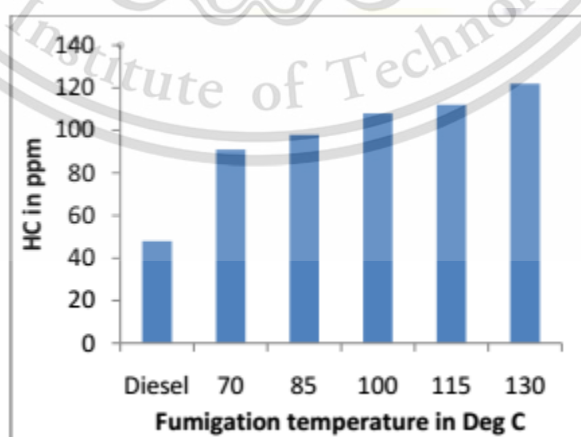


**Figure 22 BTE increases with higher temperature but decreases with higher rate of EGR**

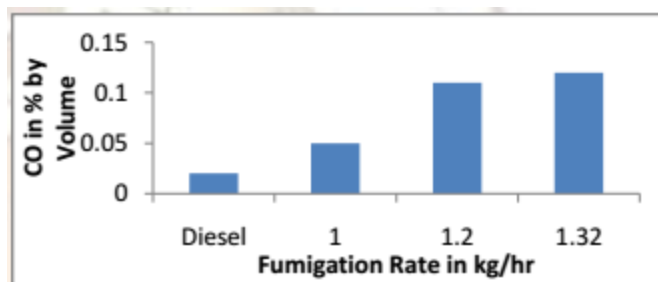
However, both cases show the lower thermal efficiency than neat diesel engine. For the emissions, NO<sub>x</sub> is lower than occurrence in neat diesel engine and can be decreased with higher rate of ethanol (Figure 23). But the higher temperature of heated mixture will make NO<sub>x</sub> higher. HC and CO are increased with higher ethanol rate and higher temperature of heated mixture (Figure 24 and Figure 25).



**Figure 23 NO<sub>x</sub> reduces with higher rate of fumigation rate but increases with higher temperature**

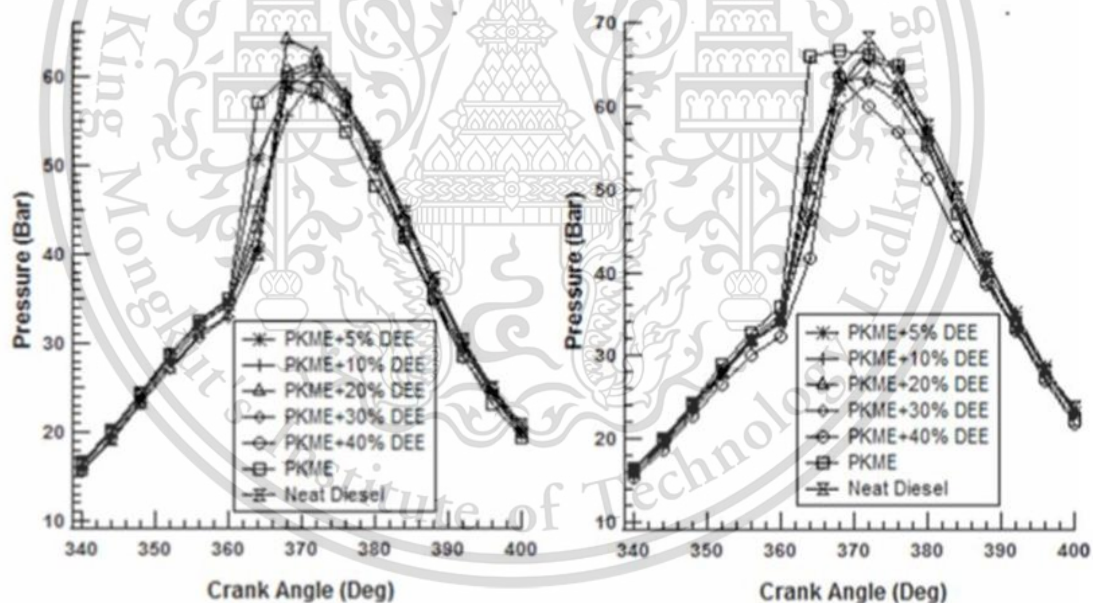


**Figure 24 HC increases with higher temperature of heated ethanol**



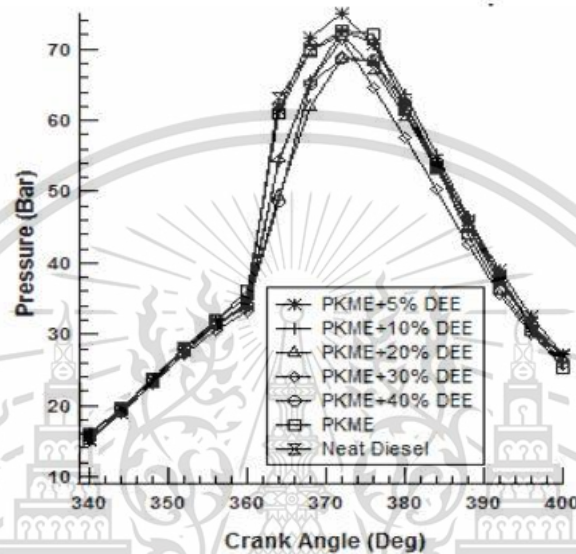
**Figure 25 CO increases with higher rate of fumigation**

Diethyl ether (DEE) is another kind of fuel that could be used with the diesel fuel. It is the by-product from the production of ethanol. Diethyl ether has high volatility and high cetane number and is usually used as the starting fuel with the combination of gasoline or diesel fuel. The heat release rate of C.I. engine with using of co-injection of DEE-H<sub>2</sub>O solution has been studied by Y.V.V.Satyanarayana Murthy, R.Rajeswara Reddy Mechanical engineering department, Gitam University [13]. This work fumigated water and DEE into C.I. engine that uses neat diesel fuel and PKME as a primary fuel.



**Figure 26 Pressure vs crank angle for half load and 75% load condition.**

The result shows that in the half load, the peak pressure of neat diesel is higher than PKME (Figure 26). By adding the DEE and H<sub>2</sub>O as the co-injected fuel, the ignition delay is become longer and causing an increase of peak pressure. DEE dominates the heat release from the PKME and caused the sharp rise of the pressure. In higher load, the decrease of pressure rise is occurred.



**Figure 27 Pressure vs crank angle for full load condition**

The decrease of pressure rise in higher load is caused by more burning occurs in the diffusion phase rather than in premixed phase (Figure 27). Peak pressure mainly depends on the premixed combustion. The latent heat of vaporization of DEE+H<sub>2</sub>O is absorbed during the combustion of PKME keeping the cylinder temperature lower creating cooler combustion. premixed combustion is improved and better diffused combustion takes place with betterment of torque.

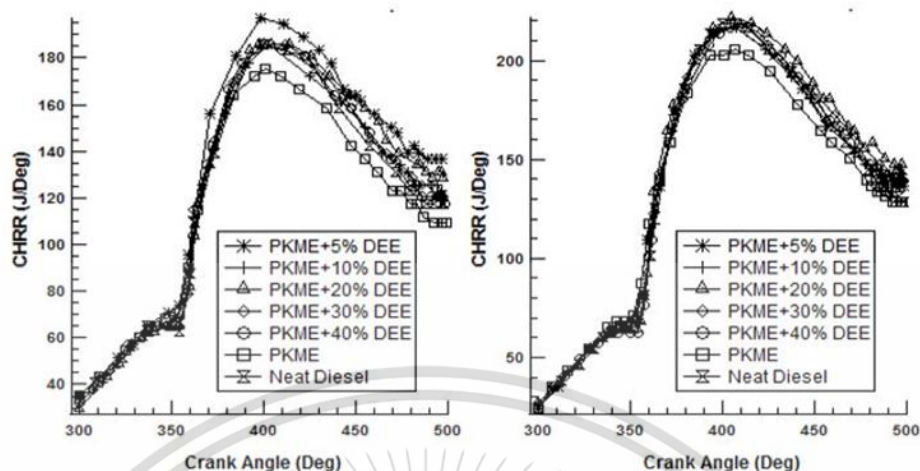


Fig. 5. CHRR vs. Crank Angle for 1/4 full load condition.

Fig. 6. CHRR vs. Crank Angle for 1/2 full load condition.

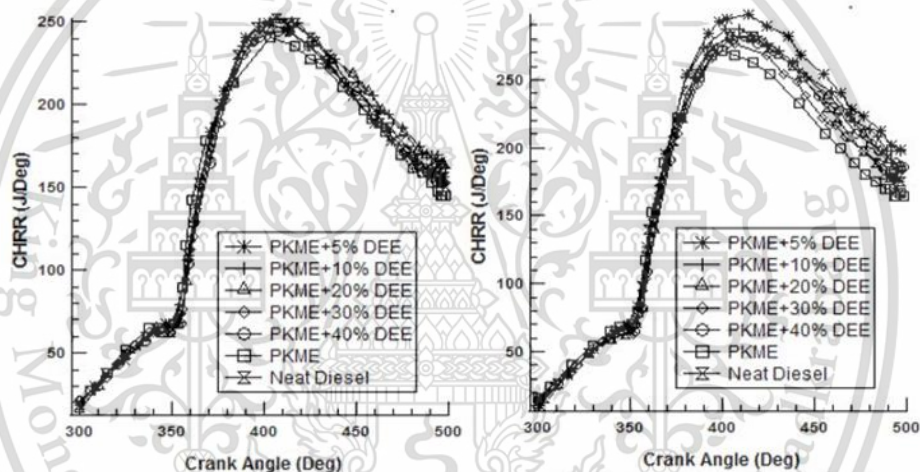


Fig. 7. CHRR vs. Crank Angle for 3/4 full load condition.

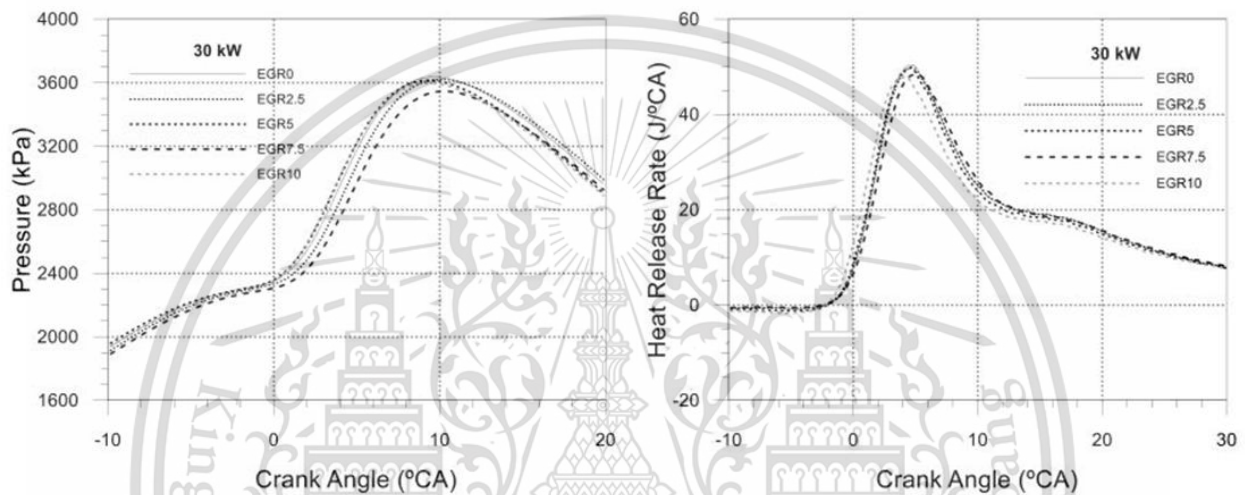
Fig. 8. CHRR vs. Crank Angle for full load condition.

### Figure 28 Net heat release vs crank angle of each load

Figure 28, the net heat release shows the neat diesel has higher heat release rate higher than neat PKME fuel. In the lower load, 5%DEE and 20%DEE has highest HRR caused by lower gas temperature in cylinder and poor vaporization (for 20%DEE), more accumulation of the PKME before ignite. Higher load, it shows the less premixed phase, better diffusion phase. DEE is completed burnt and less ignition delayed causing more HRR.

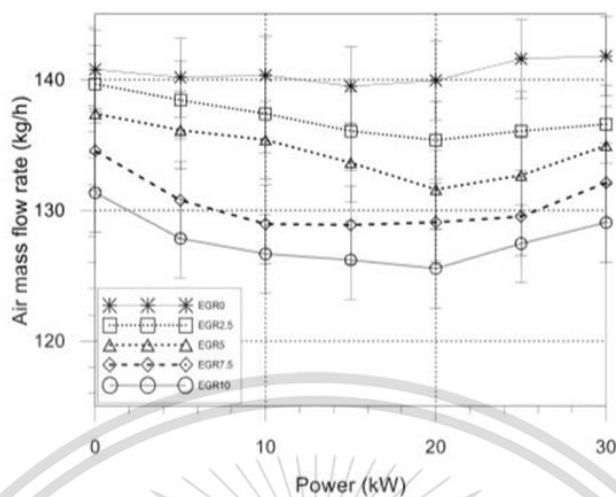
## 2.2 Use of EGR in the diesel engine

The reduction in NO<sub>x</sub> emission by using EGR was also studied by D De Serio et al[14] from Polytechnic University of Turin. It was aimed to analyze the application of an EGR system to a 49 kW diesel power generator operating with diesel oil containing 7% biodiesel (B7) in the view point of performance, emissions and combustion characteristics and finding the feasibility of using EGR system for the abatement of NO<sub>x</sub> emissions.



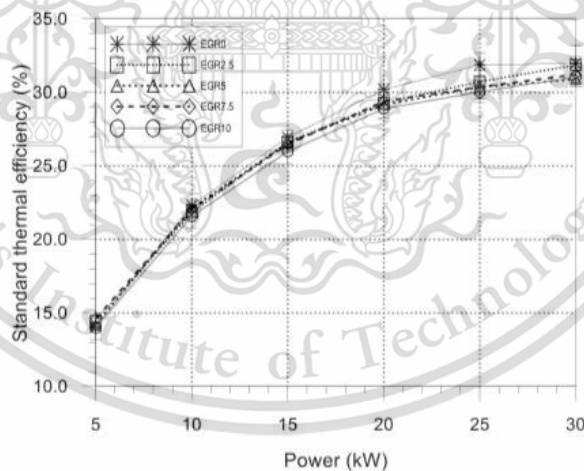
**Figure 29 In-cylinder pressure and heat release rate of 30 kW for different EGR rates**

The results show that the peak pressure was slightly decreased for all EGR rates and causing the decrease in combustion temperature (Figure 29). Therefore, it is leading to the reduction of NO<sub>x</sub> formation. The peak heat release was reduced with the use of EGR. The higher reductions were at 7.5% and 10%.



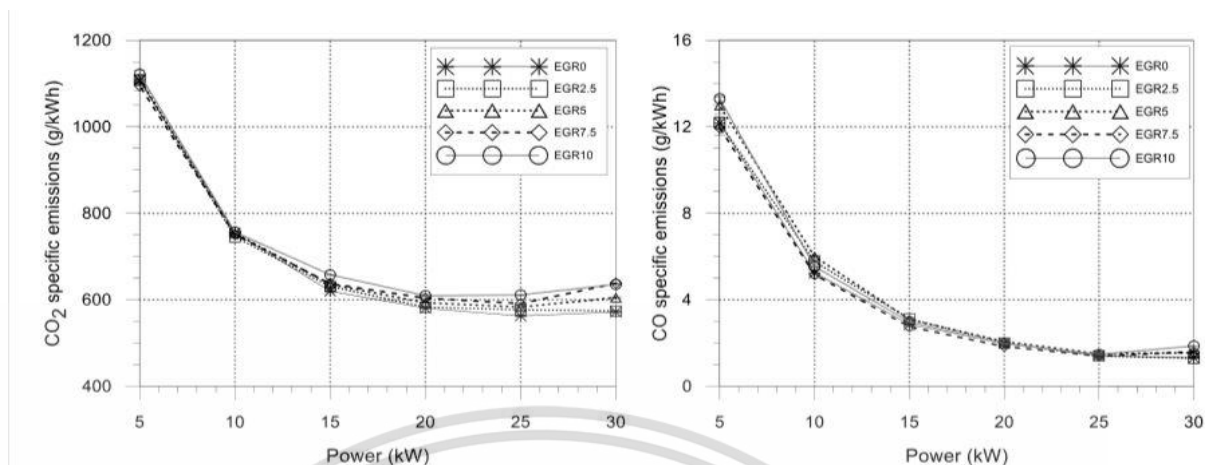
**Figure 30 Air mass flow rate for different EGR rates**

Mass flow rate was decreased as depicted in Figure 30. The number of decrease in mass is nearly same as the number of percentage of EGR calculated by  $\text{CO}_2$ . Thus, the calculation of volumetric flow rate of air intake and  $\text{CO}_2$  concentration of air intake can be used to estimate the EGR flow rate.



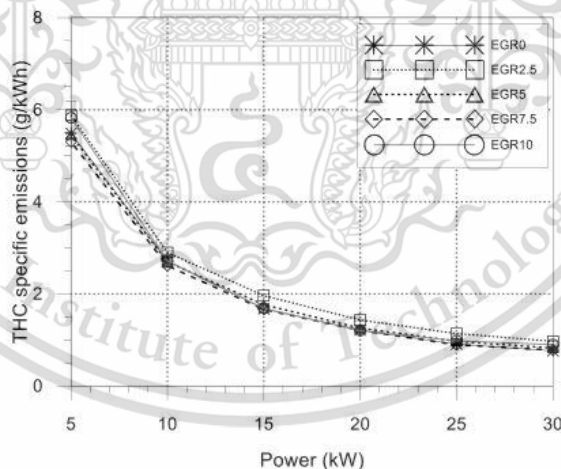
**Figure 31 Thermal efficiency for different EGR rate.**

Thermal efficiency was decreased but not in the significant trend as shown Figure 31. There is no suggestion for using EGR in the high load if concerning about Thermal efficiency due to the decrease trend in higher load.



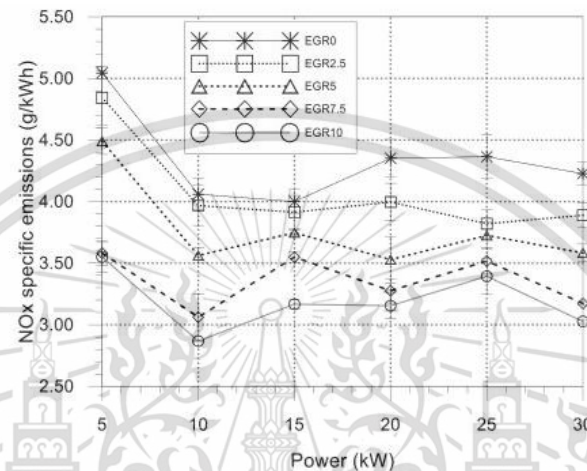
**Figure 32 CO<sub>2</sub> and CO specific emission for the different EGR rate**

Figure 32, CO<sub>2</sub> and CO emissions were gradually increased with the use of EGR. However, EGR carries some amount of CO<sub>2</sub> from previous combustion and feeding back into the combustion chamber causing higher CO<sub>2</sub> and CO emission and also causing the deceleration of the reaction rate of air-fuel mixture, leading to more difficulty to be completely combusted.



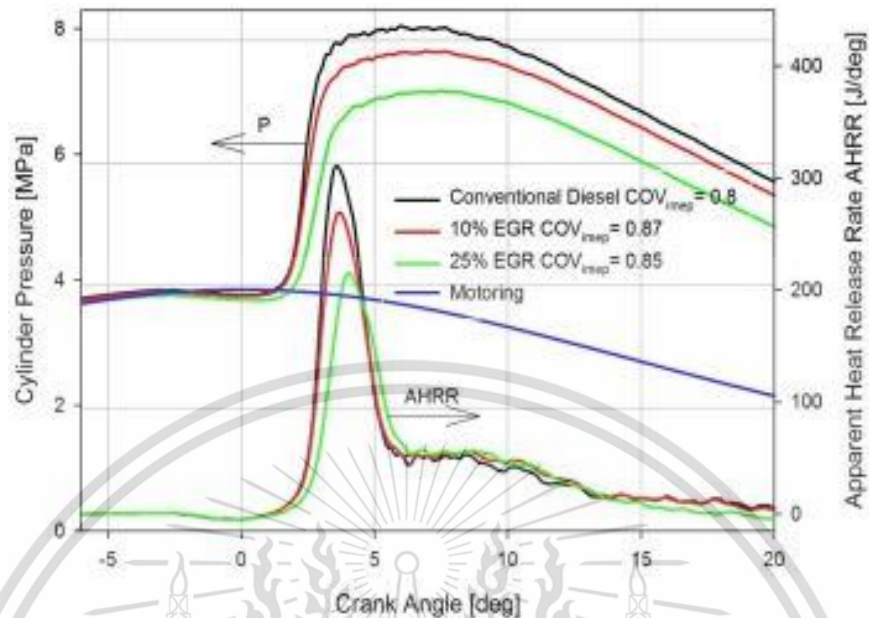
**Figure 33 THC specific emission for different EGR rate**

Total hydrocarbon was increase but was not in the significant trend. This is caused by the less combustion capability due to the lack of fresh oxygen in each different EGR rates (Figure 33). While NO<sub>x</sub> is obviously decreased, the local flame temperature and the higher specific heat of the gas in the combustion zone were getting reduced (Figure 34).



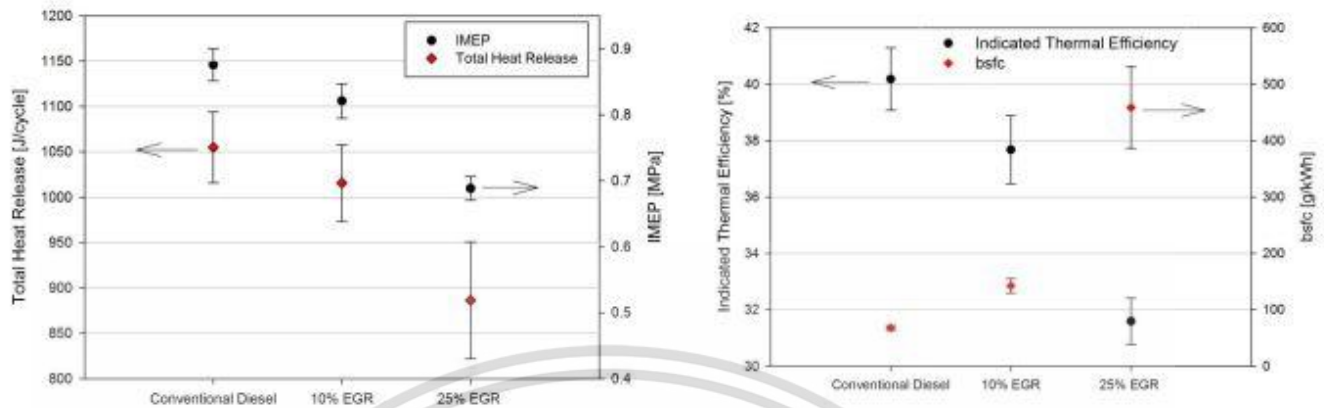
**Figure 34 NO<sub>x</sub> specific emissions for different EGR rates.**

The investigation of the effect of ethanol injection into the exhaust manifold on the combustion and exhaust emission of the direct injection diesel engine has been conducted. M-Nour [15] from E-JUST, Egypt-Japan University of Science and Technology conducted the experiment by based on the hypothesis of injecting ethanol into the intake manifold would cause the charge cooling and combustion quenching . The conductor focused on the injection of ethanol into the exhaust manifold for waste heat recovery and early evaporation of ethanol to enhance the combustion in the next cycle. The effect of exhaust gases recirculation (EGR) ratio, ethanol injection timing and ethanol injection amount on combustion and emissions are studied. And the conductor has also compared the injection position between at the intake manifold and the exhaust manifold.



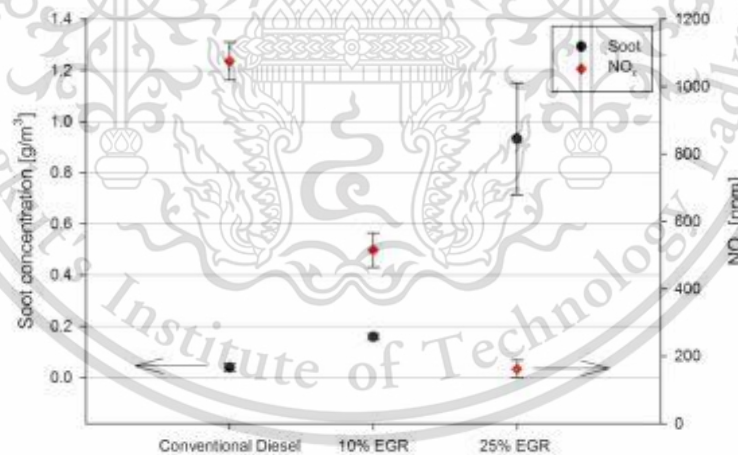
**Figure 35 The effect of the different EGR ratios on the cylinder pressure and the rate of heat release**

The cylinder pressure is affected by EGR where the cylinder pressure is decreased with increasing EGR ratio (Figure 35). The peak cylinder pressure decreased by 6.5% for EGR10 and by 13.6% for EGR25 compared to conventional diesel combustion. It could be attributed to the fact that increasing EGR ratio decreases the inlet  $O_2$  concentration which decelerates mixing between  $O_2$  and fuel resulting in the extension of the flame region. Thus, the quantity of  $CO_2$  and  $H_2O$  gases that absorbs the heat released is increased resulting in lower flame temperature and heat release rate. EGR decreases the premixed phase of the heat release but increase the diffusion phase instead. The combustion ends earlier with the increasing EGR due to the lack of oxygen and also means the incomplete combustion.



**Figure 36** The effect of different EGR ratios on the engine performance parameters ; Heat release rate and IMEP (left), Thermal efficiency and bsfc (right)

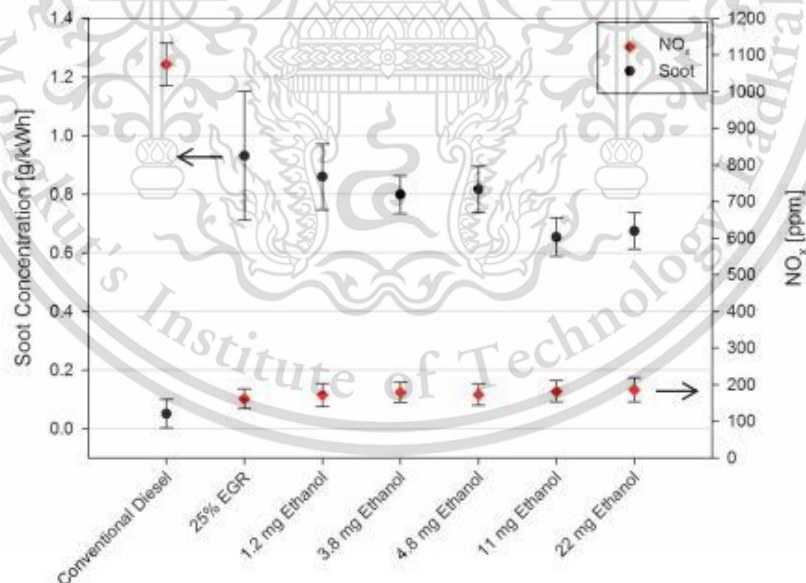
Figure 36, the lack of the combustion causes IMEP becomes declined while enhancing EGR ratios. Consequently, the bsfc increased dramatically for EGR10 and EGR25 and the indicated thermal efficiency reduced by 6% and 21% for EGR10 and EGR25 compared to conventional diesel.



**Figure 37** Effect of the different EGR ratios on soon and NOx concentrations

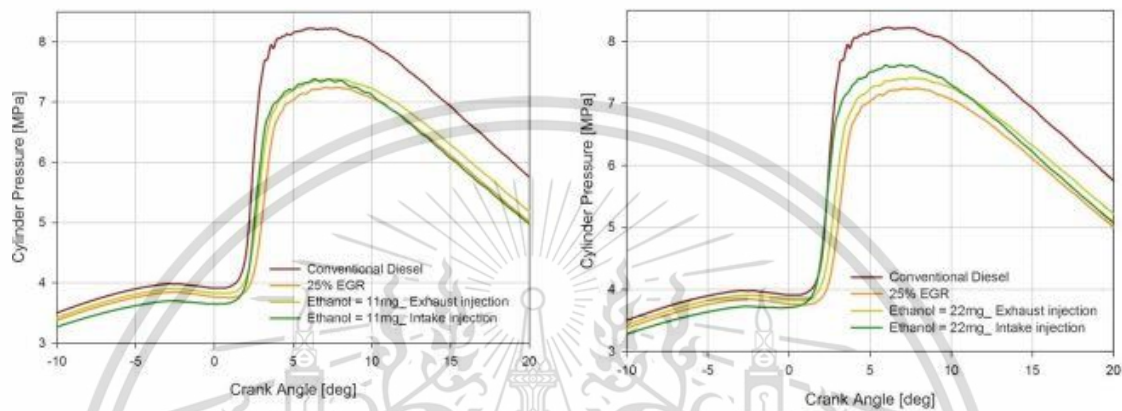
The reduction of flame temperature due to the decrease in the inlet O<sub>2</sub> concentration and consequently the mixing efficiency with the fuel led to decrease the NO<sub>x</sub> emission and increase in the soot emission. NO<sub>x</sub> was drastically decreased with the increase of EGR ratios as depicted in Figure 37. However, soot increased three times and five times with the 10%EGR and 25%ERG, respectively.

Ethanol was injected in the different timing of  $350^\circ$ ,  $400^\circ$  and  $450^\circ$ . The appropriate ethanol injection timing and EGR ratio that gives good combustion characteristics, lower soot and NO<sub>x</sub> emissions can be obtained (Figure 38). The high-delivered amount of ethanol together with high EGR ratio reduces the effect of ethanol on the cylinder peak pressure. For ethanol injection cases, the highest cylinder pressure was attained at ethanol injection timing of  $400^\circ$  CA at EGR10 compared to the other injection timings. Ethanol injection increases the amount of fuel burned under premixed condition compared with EGR and also more ignition delay due to the worse quality of auto ignition point. However, this effect was reduced in the current study due to the reduction in ethanol evaporation time as the ethanol was vaporized before entering the engine cylinder. Consequently, ethanol injection in exhaust manifold gives similar ignition delay with the cases without ethanol injection. The combustion duration is reduced with the injection of ethanol and 25%EGR but not show the same case with the 10%EGR. High EGR rate leads to more incomplete combustion and causes lower time available for the combustion process. Using of EGR is obviously increasing soot. But with ethanol injection soot becomes lower due to the enhancement of soot oxidation by adding the oxygenated ethanol fuel while causing the NO<sub>x</sub> formation due to the increase of the burned fuel amount in premixed phase and the increase in-cylinder pressure.

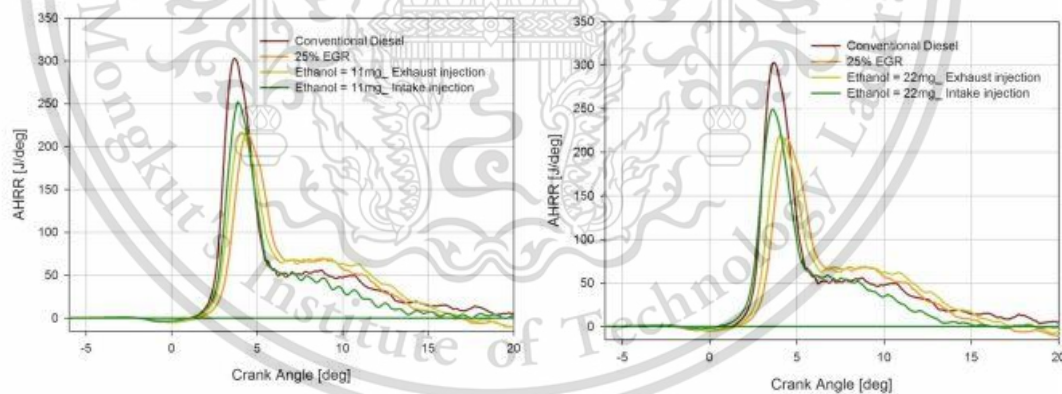


**Figure 38 The effect of various ethanol injection amount with EGR on soot and NO<sub>x</sub> emission**

The author has chosen the 25% EGR at 400° CA of ethanol injection timing to investigate on the further experiments such as the effect of ethanol injection amount. An increase of the amount of injected ethanol leads to increase in the total heat release of a maximum of 14% and IMEP of a maximum of 8% (Figure 39 and Figure 40). Also, NO<sub>x</sub> emission slightly increases to a maximum of 14%, and soot concentration reduces to a maximum of 29% (Figure 41). However increasing the ethanol injection reduces the combustion efficiency.



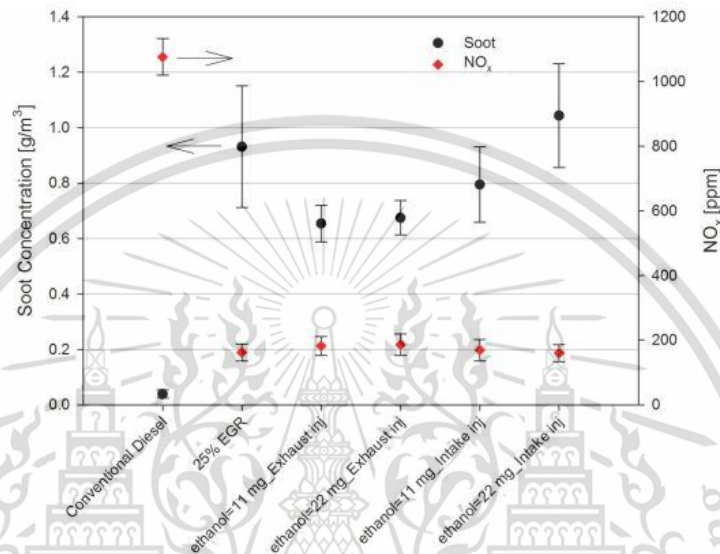
**Figure 39 . The in-cylinder pressure comparison between ethanol injection in the intake manifold and exhaust manifold for 11 mg of ethanol (left) and 22 mg (right) of ethanol per cycle.**



**Figure 40 The heat release rate pressure comparison between ethanol injection in the intake manifold and exhaust manifold for 11 mg of ethanol (left) and 22 mg (right) of ethanol per cycle.**

For comparison between intake and exhaust manifold injection of ethanol, the soot emissions are lower with ethanol injection into exhaust manifold by 24% and 53% for ethanol amount of 11 mg/cycle and 22 mg/cycle, respectively compared to ethanol injection into the intake manifold. Ethanol injection in the intake manifold achieves a higher cylinder pressure. However, it causes the heat release to become negative through the ignition delay for all the

tested fuels due to the vaporization of the accumulated fuel in the flame propagation process. The fuel took the heat from the EGR, hot air in the compression stroke and the chamber walls. The period of the negative AHRR for ethanol injection into the intake manifold is shorter than ethanol injection into the exhaust manifold, which means a shorter ignition delay for the intake manifold injection compared to the exhaust manifold injection.



**Figure 41 The comparison of soot and NOx emission of the different injection strategies**

Exhaust manifold injection compared to the injection strategies used in the previous studies for ethanol injection such as injection in the intake port or direct injection, the injection through exhaust manifold is an effective way as ethanol is evaporated before introducing it into the combustion chamber. Also, the enthalpy of exhaust gases is reused to enhance the combustion process in the next cycle. The combination of ethanol injection, high EGR rates and late fuel injection timing enabled low-temperature combustion in diesel engines that can simultaneously reduce soot and NOx emissions with improvement in engine efficiency.

Ravindra Sadashiv Deashpande from M. Tech Student Agnos College of Technology (R.K.D.F. University), Bhopal (M.P.) et al[16] has studied on the performance of diesel engine with EGR system as well. It is known that EGR can help to reduce NOx emission by limiting oxygen supplied into intake manifold, lowers the combustion temperature. However, it also reduces engine performance. The investigation was conducted on 0%EGR, 5%EGR, 10%EGR and 15%EGR rate for studying the performance & emission characteristics of the engine.

The results suggest that when using EGR, specific fuel consumption is increased and causes the partially decrease in brake thermal efficiency. The gas emissions such as HC, CO and

NO<sub>x</sub> were all decreased while soot was increased in conversely as the other works have suggested.

The effect of using EGR in the diesel engine was studied in many works. Mingfa Yao et al[17] have also studied on the using EGR with diesel engine. With the reason that EGR help reduce NO<sub>x</sub> due to it dilutes the fresh air intake and causing the poorer combustion, EGR is now used in many engine these days. The authors have done the experiment on the heavy-duty diesel engine with the fixed engine speed of 1400 RPM and inject the fuel for 50 mg/cycle to investigate the effect of changing the combustion parameters such as injection pressure, injection timing, boost, compression ratio and etc.

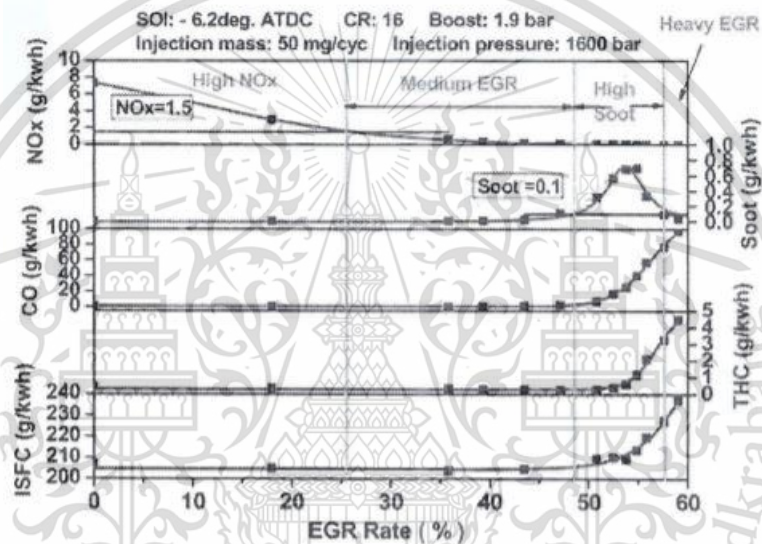
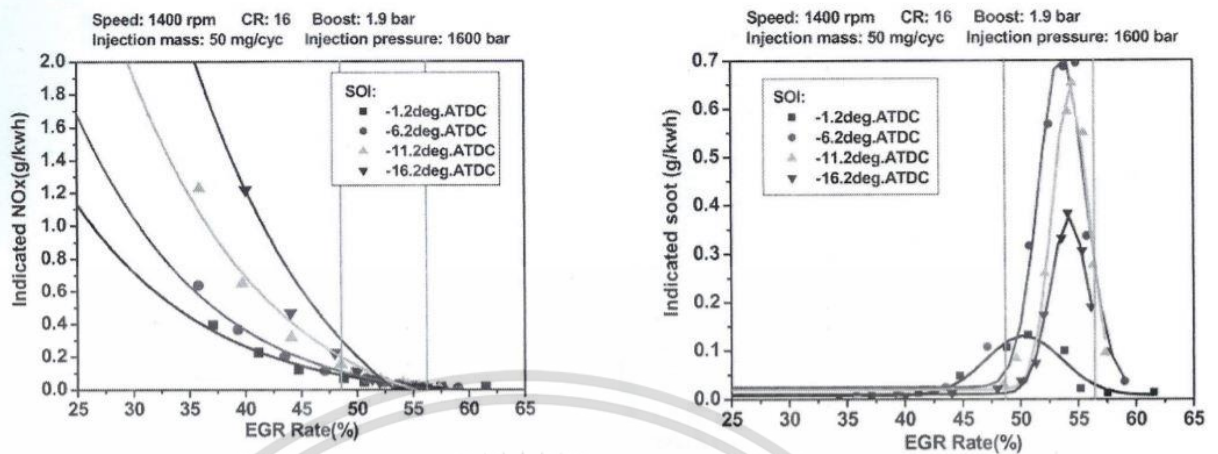


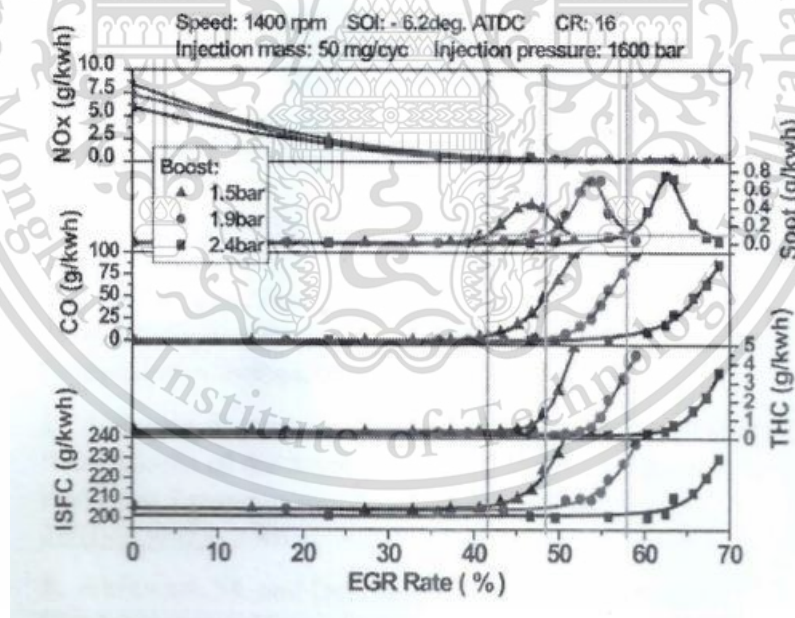
Figure 42 Effect of EGR rate on emissions and ISFC

The effect of conventional combustion parameters such as injection pressure, injection timing and boost on NO<sub>x</sub> emission become small as the EGR rate is increased while soot is strongly depend on the ignition delay and the oxygen concentration or EGR rate (Figure 42). Decreasing the compression ratio and increasing the injection pressure will reduce the soot level as shown in Figure 43. With the increase of boost, soot emissions decreased below medium EGR rate (Figure 44). However, this work suggests the low-temperature combustion in the heavy EGR operation. Low NO<sub>x</sub> and soot can be achieved simultaneously while producing a lot of CO and HC emission.



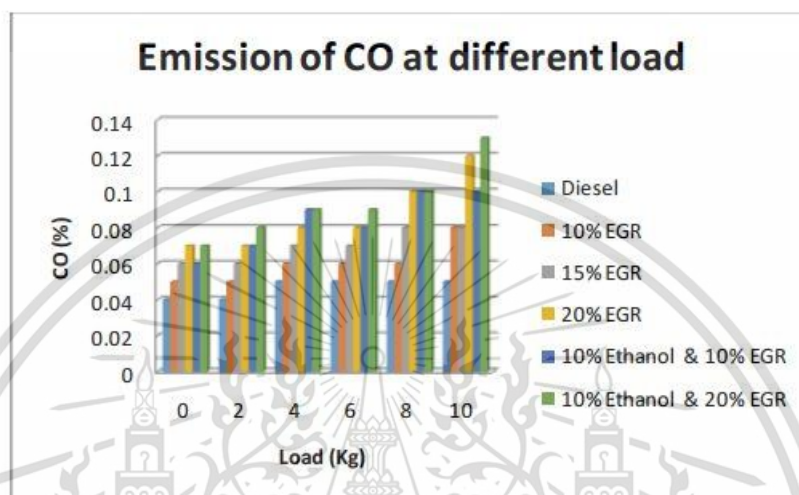
**Figure 43 Effect of compression ratio on NO<sub>x</sub> and Soot**

The trade-off between NO<sub>x</sub>-soot of the conventional combustion is turned into the soot-fuel efficiency (HC and CO emissions) when operating in heavy EGR rate. Nevertheless, all gaseous emissions and soot are low in medium EGR operation, causing least fuel penalty. Medium EGR is a promising method to meet future emissions regulation with higher fuel efficiency.

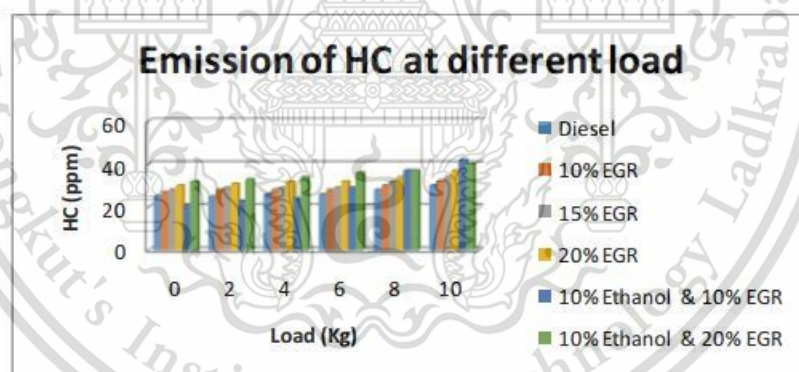


**Figure 44 Effect of boost on emissions and ISFC**

The performance and analysis of ethanol and EGR effect on the exhaust gasses in compression ignition has been studied by Binyamin Chritian [18] from Dept. Thermal in Mechanical Engineering, LDRP Institute of Technology and Research, Gandhinagar. The Experiment is done on a single cylinder, 4-stroke, water cooled, Kirloskar made diesel engine by mixing of ethanol in diesel by 10% and EGR at 10%, 15% and 20%.

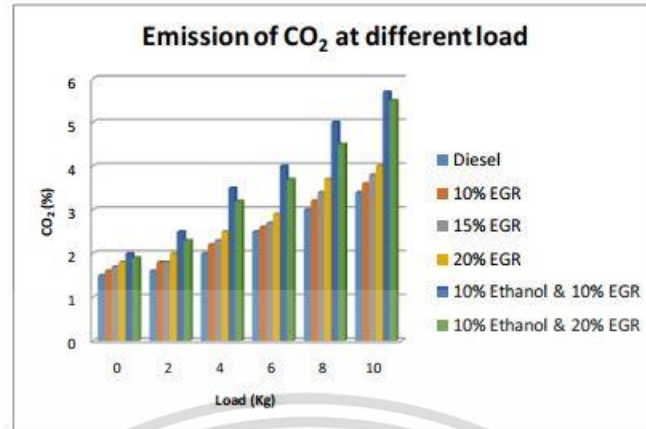


**Figure 45 Comparison of CO emission in the different engine load**

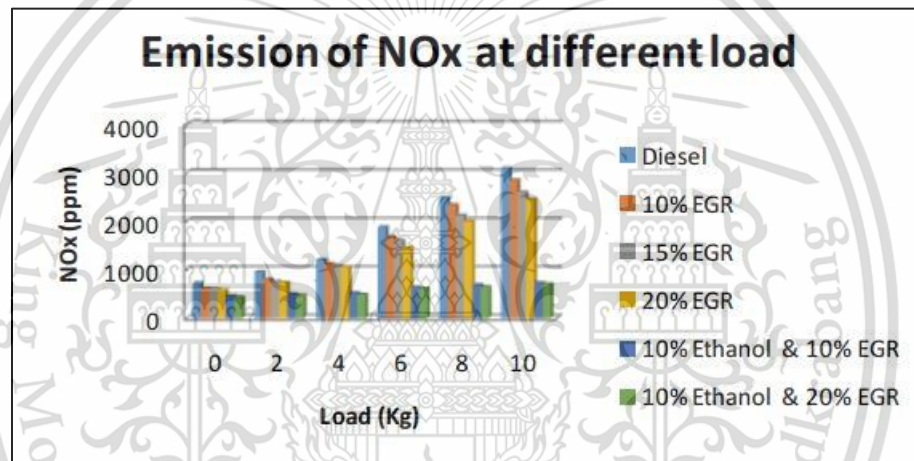


**Figure 46 Comparison of HC emission in the different engine load**

The result suggests that at the part-load CO emission is gradually increased and becomes lower at the higher load. The use of ethanol for 10% with the different EGR rate of 10% and 20% EGR also influence the formation of CO emission respectively (Figure 45). HC emission is becoming higher with respect to the EGR rate. But when using 10% ethanol fraction with 10% EGR in part-load, HC emission was decreased. These results conform to other works that also suggest the increase of HC and CO when EGR is operating (Figure 46). Emission of CO<sub>2</sub> gradually increases with increased EGR while higher percent of CO<sub>2</sub> was observed with additive of 10% Ethanol (Figure 47).

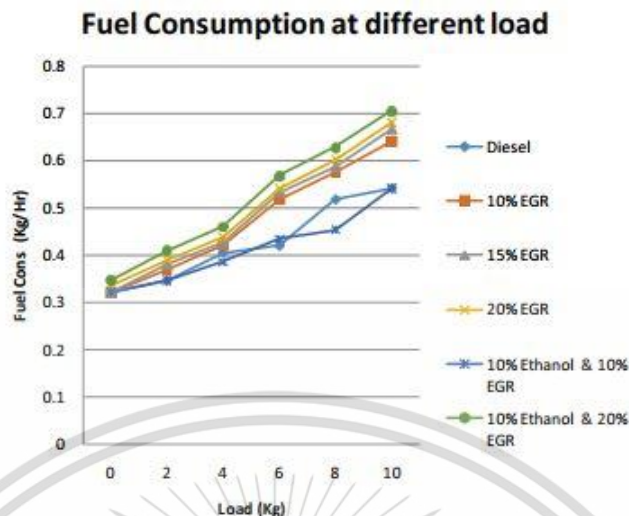


**Figure 47 Comparison of CO<sub>2</sub> emission in the different engine load**



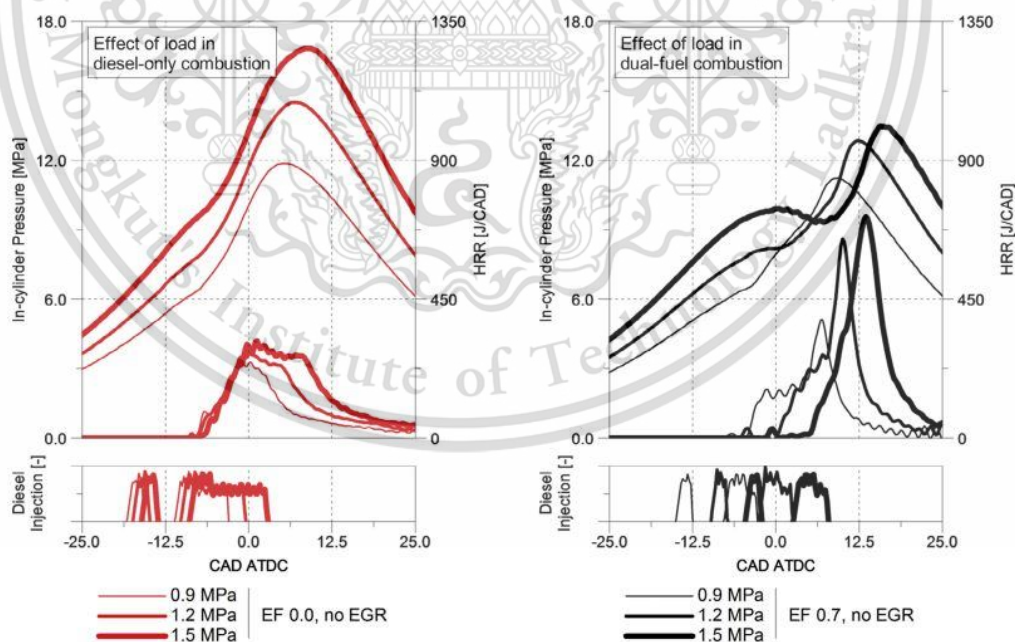
**Figure 48 Comparison of NO<sub>x</sub> emission in the different engine load**

Figure 48 shows the comparison of NO<sub>x</sub> emission that is changed by EGR rate at different load. The feeding rate of EGR obviously decreases the NO<sub>x</sub> emission on each load and still shows the similar level although the engine load is increased when using 10%ethanol with 10%EGR and 20%EGR. Fuel consumption becomes little higher for the different EGR percent and it was comparatively lower for 10%ethanol mixture at different EGR percent. The results is depicted in Figure 49.

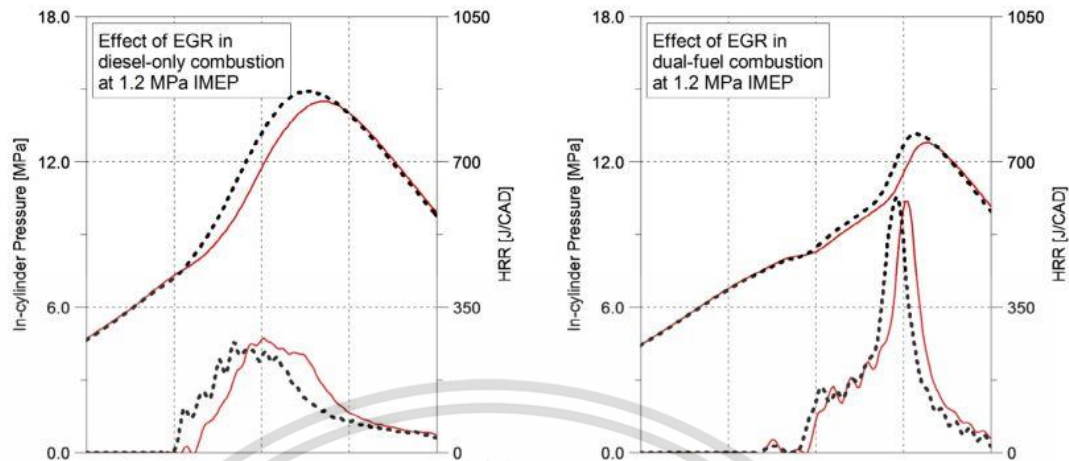


**Figure 49 Comparison of fuel consumption at different engine loads**

Vinícius B. Pedrozo, Ian May, and Hua Zhao have studied on the mid-load potential of ethanol-diesel dual-fuel combustion with and without EGR [19]. This work aimed to optimize the use of ethanol as a partial substitute for diesel fuel with the operation of EGR system. The experiment was conducted on the single cylinder C.I. engine with three loads of 0.9, 1.2 and 1.5 MPA of IMEP with the different ethanol portion of 0% to 80%.

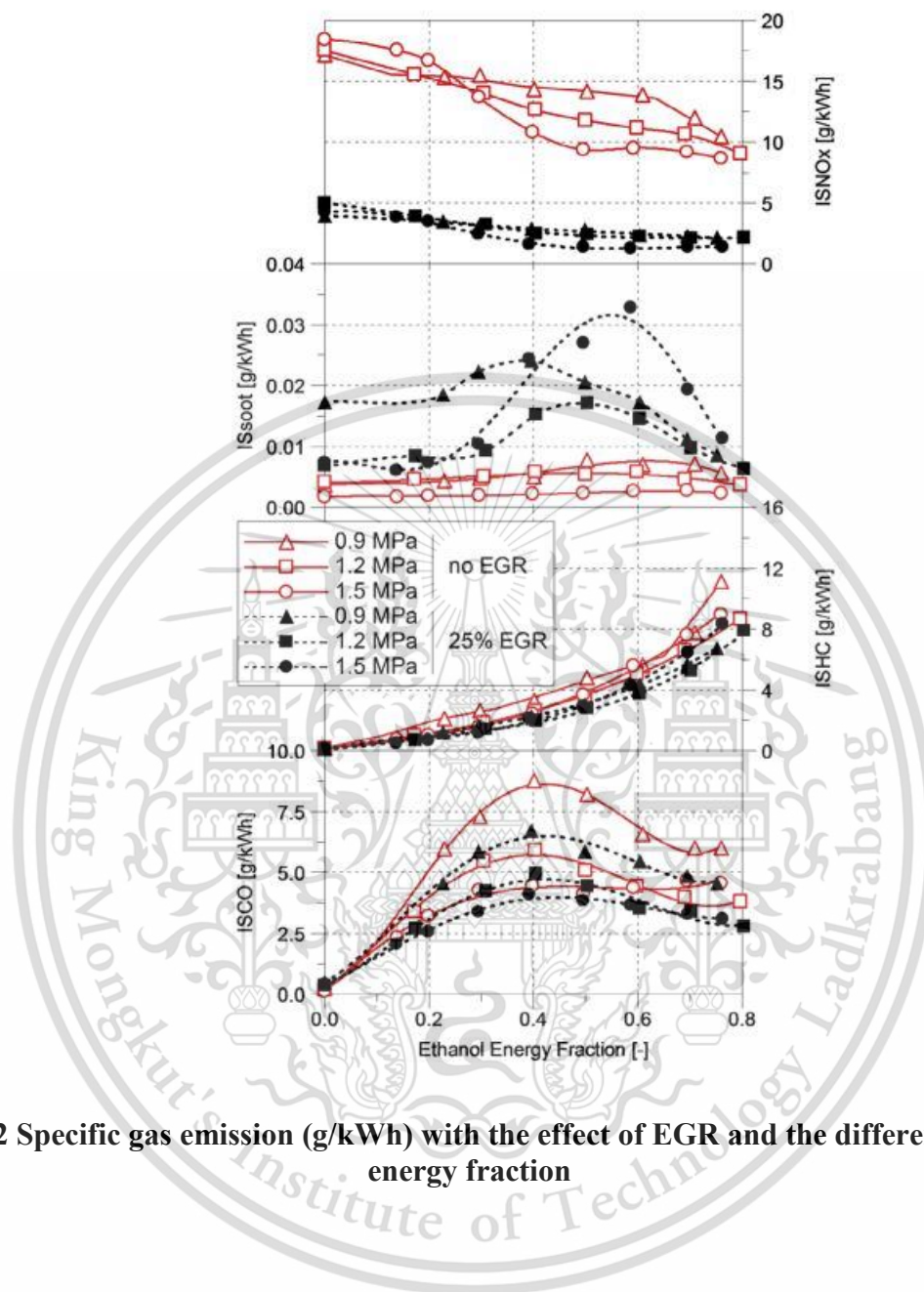


**Figure 50 The effect of engine load on in-cylinder pressure and rate of heat release of diesel-only (left) and dual-fuel combustion process (right)**



**Figure 51 The effect of EGR on in-cylinder pressure and rate of heat release on diesel only (left) and dual fuel combustion (right)**

The results show that the increase in engine load from 0.9 to 1.5 MPa IMEP in dual fuel mode led to earlier ignition of the premixed ethanol and shorter combustion durations (Figure 50). This required retarded diesel injection timings to lower the in-cylinder pressure rise rates. Higher ethanol percentages reacted similarly to the effect of increased load, results in faster burn durations and requires later diesel injection timings while introduce 20%EGR into the combustion process also retards the starting of the combustion (Figure 51). Despite the retarded combustion, net indicated efficiency was maintained essentially constant due the more thermodynamically optimum heat release. The use of high ethanol fractions resulted in shorter and delayed combustion process, similar indicated efficiency, and up to 68% lower NO<sub>x</sub> emissions than conventional diesel combustion. Soot levels varied with different ethanol percentages. Unburned hydrocarbon (HC) and carbon monoxide (CO) emissions increased with higher amounts of premixed ethanol fuel (Figure 52). The introduction of 25% EGR led to further NO<sub>x</sub> reductions, decreasing the nitrogen oxides levels of the non-EGR cases by 80%.



**Figure 52 Specific gas emission (g/kWh) with the effect of EGR and the different ethanol energy fraction**

Gurumoorthy S.Hebbar et al [20] has conducted an experimental research about diesel emission control by hot EGR and ethanol fumigation in a single cylinder DI diesel engine. The main objectives are reduces the amount of emissions in exhaust gas and study about the effect on thermal efficiency by from this system. The result shows that EGR has effected on the reduction of NOx emission influentially (Figure 53). And the too much number in percentage of ethanol will worsen the combustion performance (Figure 54)The higher rate of EGR that feed to the engine will increase the amount of HC and CO (Figure 55). 10% of ethanol is the optimum percentage for using in higher rate of EGR as the conclusion of this experiment.

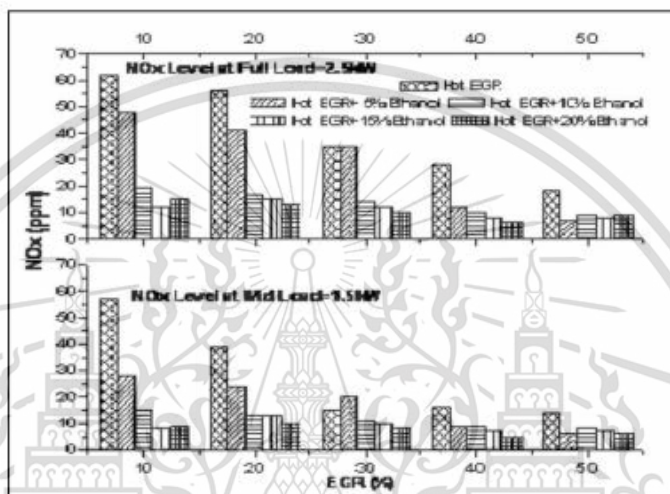


Figure 53 NOx emission result

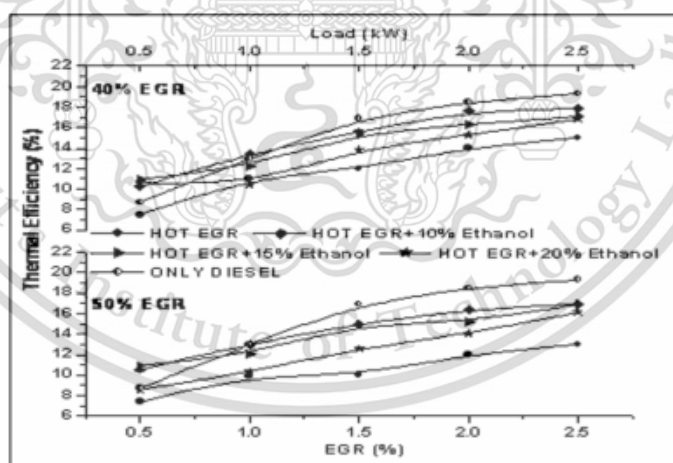
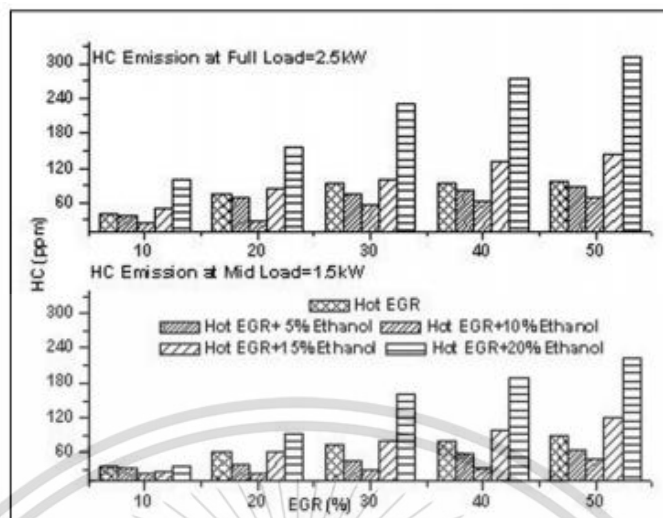
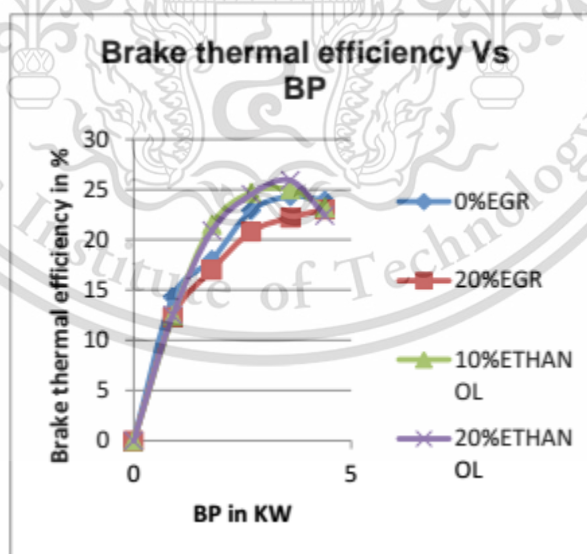


Figure 54 Thermal efficiency between 40% EGR and 50% EGR



**Figure 55 HC emission result**

The study of diesel engine that using EGR and ethanol fumigation system also has been conducted by CK college of Engineering and Technology, India by T.Vivek Asalan and Dr.G Sasikumar [21] have aimed to study about the performance of EGR installed diesel engine. EGR system will reduce the combustion temperature and worsen the performance of the engine. The result shows that ethanol fumigation system will improve the efficiency of the EGR installed engine. The higher rate of ethanol in the system shows the trend of increase of thermal efficiency (Figure 56).



**Figure 56 BTE with respect to Brake power for 20%EGR with 10 and 20% ethanol percentage**

The fuel consumption has been increased with respect to the amount of exhaust gas in the system because the engine needs to inject more fuel to maintain the level of performance. Ethanol fumigation will cause the reduction of fuel consumption from the combustible property of ethanol (Figure 57). The using of ethanol fumigation and EGR can reduce the NO<sub>x</sub> effectively (Figure 58). In contrary, the diluted air and lower combustion temperature from these two systems cause higher rate of HC emission.

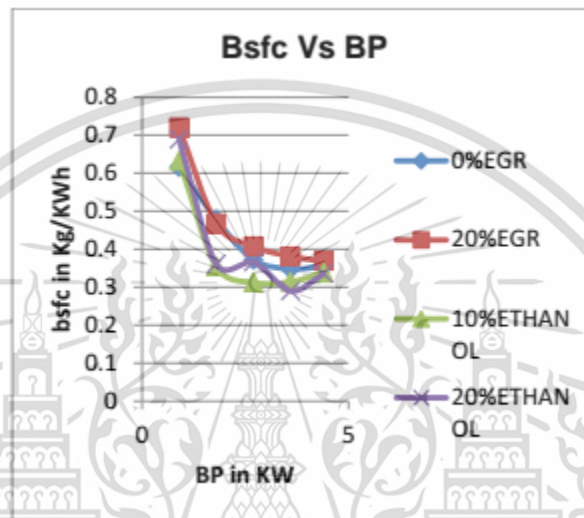


Figure 57 specific fuel consumption result

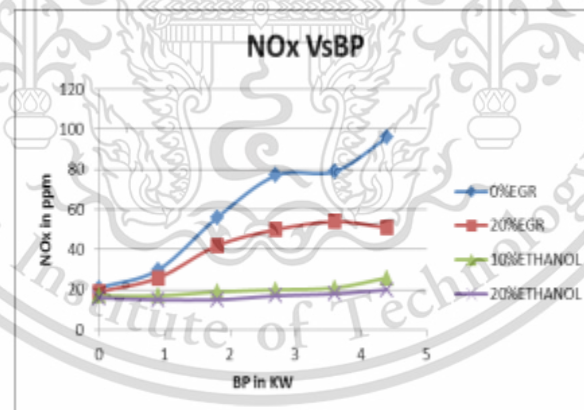
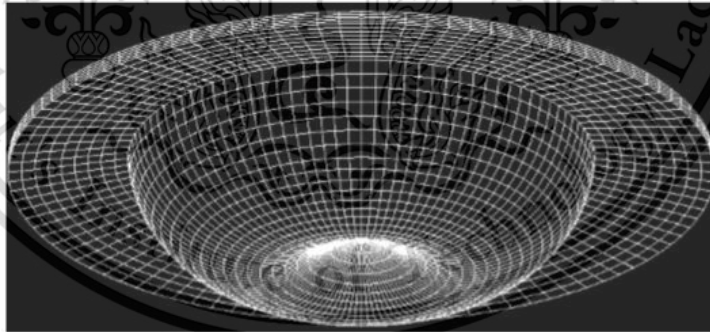


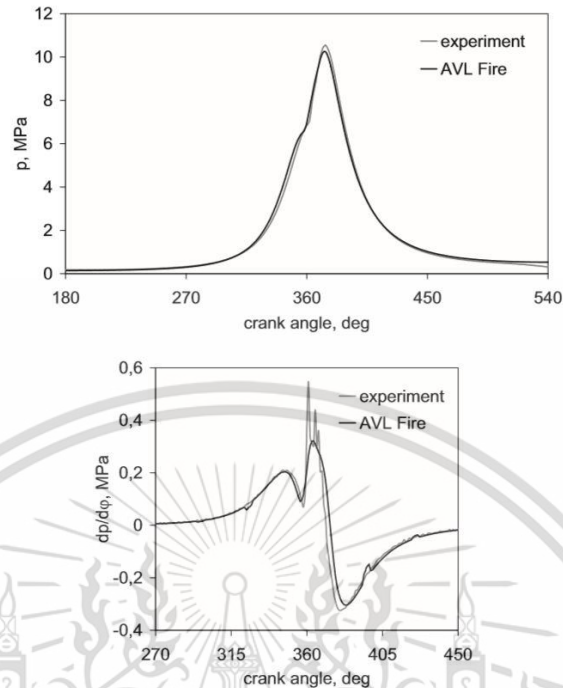
Figure 58 NO<sub>x</sub> emission reduction result

One of the most important parameters in diesel engine is the ignition delay. It will effect on the performance of the engine, emissions and also the patterns of combustion characteristics. This parameter is the time between the start of injection and the start of the combustion and technically has both physical delay and chemical delay. The fuel needs time to atomize and vaporize itself before mixing with the air. The time that is used in that period is the physical delay. And the chemical delay is the reaction between fuel and air in pre-combustion period. Ignition delay in diesel engines has a direct effect on engine efficiency, noise and exhaust emissions. The delay can be changed by the change of some parameters such as cylinder pressure, temperature, types of fuel and etc. Many researchers such as Hardenberg & Hase, Wolfer & Watson and Assanis have proposed the correlations of the ignition delay, mathematically.

W.Tutak et al. have conducted the research about the effect from EGR on ignition delay in supercharged CI engine [22]. This research determined the ignition delay that effected by portion of EGR. The experiment was modeled in six-cylinder supercharged diesel engine with a constant rotational speed. The combustion model of coherent flame ECFM-3Z was used. Injection timing and load were kept constant also. The results of modeling and experiments are compatible together. The results were shown the MEP and the indicate efficiency first. By the increase of EGR mass in the system, the trends of those parameters are decreased. The rate of heat release curves were compared together between modeling and test engine experiment which provided nearly same patterns. The patterns were used to determine the ignition delay by using the proposed algorithm from Hardenberg and Hase, Watson and Assanis and AVL Fire modeling program.

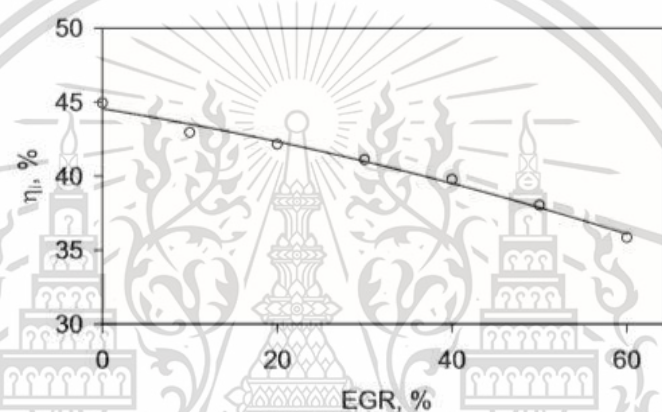
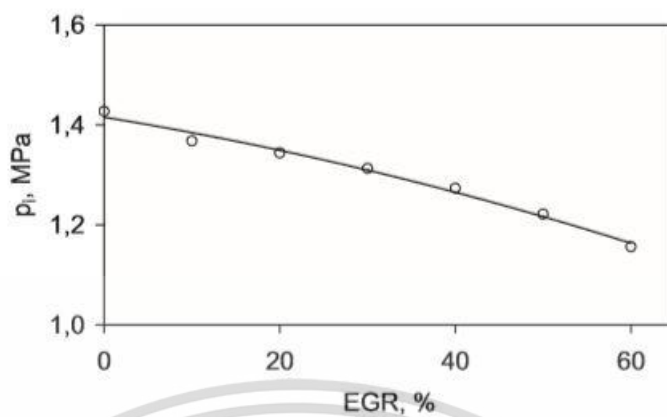


**Figure 59** The computational grid for combustion chamber modeling in TDC

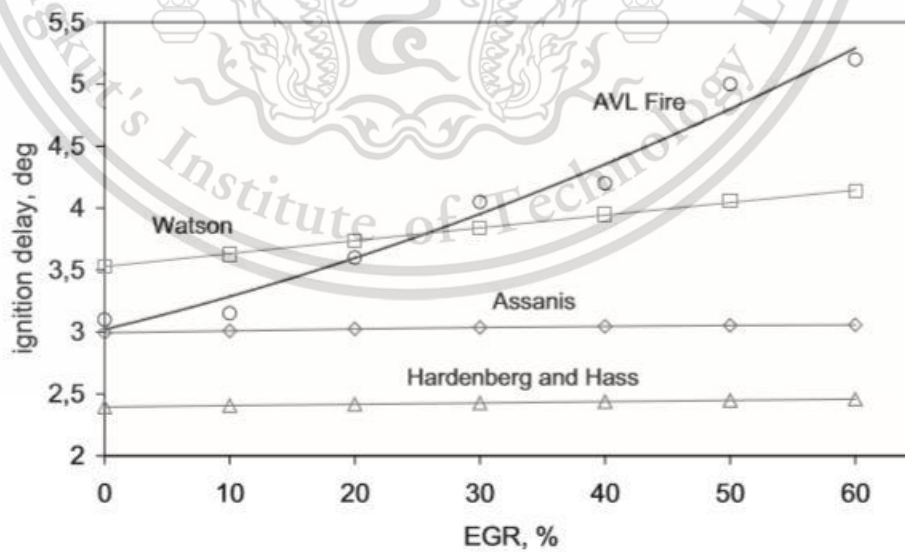


**Figure 60 Result of model validation, traces of pressure and pressure different on each crank angle**

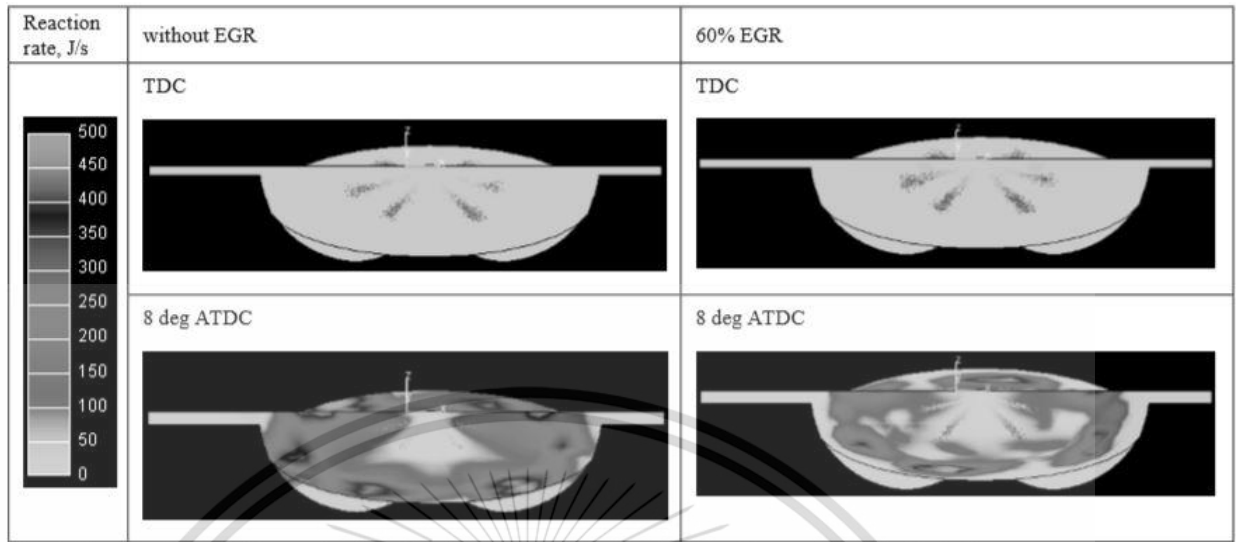
The comparison on ignition delay from each algorithm shows that in conditions without EGR, the closest value to AVL Fire of ignition delay was obtained by using of the Assanis correlation (Figure 60). With the increase of the recirculated exhaust gas portion, the ignition delay increased with the drops of mean effective pressure and thermal efficiency as shown in Figure 61 and Figure 62. These results conform to other works. The results obtained with the use of Hardenberg and Hase and Assanis correlation did not give satisfactory results. The cross sections of the combustion chamber when combustion was occurred also shows that in the case with EGR the space covered by combustion process in the combustion chamber is smaller than in the case without recirculation (Figure 63).



**Figure 61 Mean indicated pressure and indicated efficiency**



**Figure 62 Model result of EGR effect on ignition delay**



**Figure 63 In-cylinder simulation of the combustion b the effect of EGR**

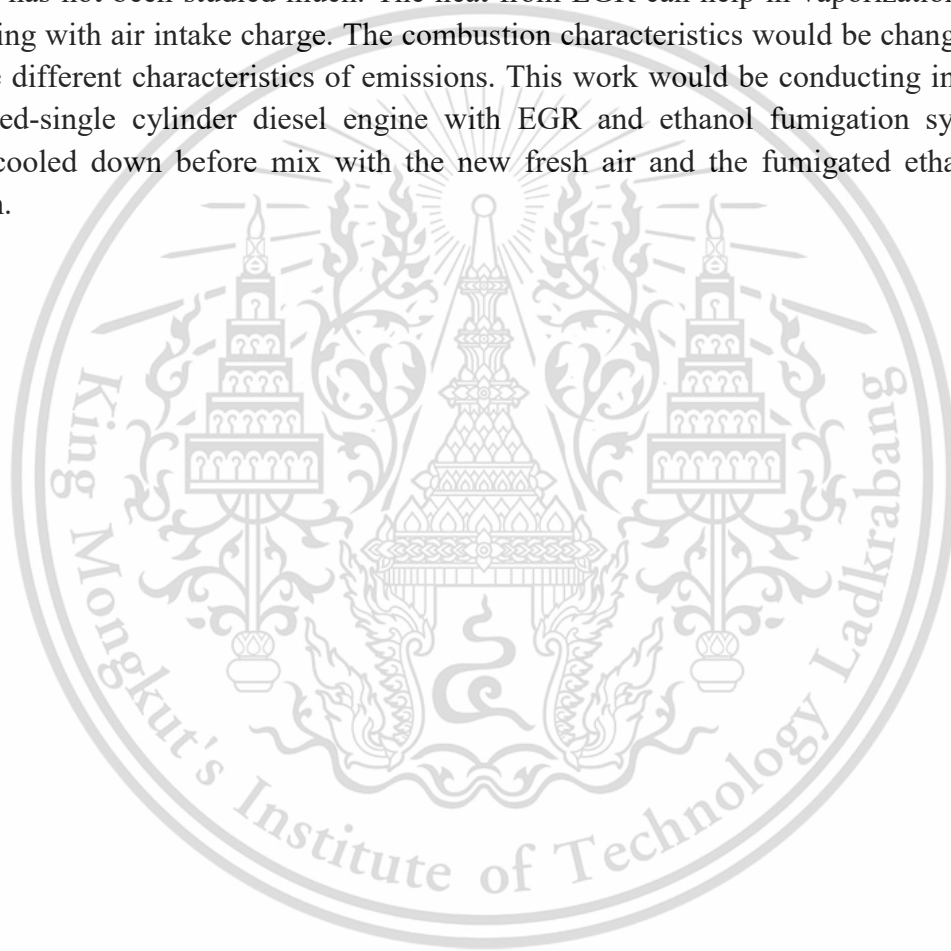
The comparison of the ignition delay from the proposed correlations and AVL Fire engine simulation program showed that with the increase of EGR portion in the system, ignition delay was also increased. But some of the proposed correlation did not give the satisfactory results by remains in the nearly same period of the ignition delay when increase the EGR participation. The cross section reaction simulation results also show the influence of EGR on the ignition delay, smaller reaction area is provided.

From above reviews, using ethanol fumigation system with diesel engine will lead to the increase of carbon-content emissions such as CO and THC [23, 24, 25] due to more carbon and hydrogen atom provided from ethanol ( $C_2H_5OH$ ). For the formation of NO<sub>x</sub>, it could be both increased and decreased depends on the condition. The increase of NO<sub>x</sub> could come from more oxygen atom available to be formed from ethanol while the main reason of the decrease trend is from the cooler combustion temperature due to the heat of vaporization of ethanol. Soot or PM mostly depends on the diesel fuel injection amount. Less use of diesel fuel could lead to the reduction of soot, nevertheless, the completeness of the combustion is also the main factor.

In the view point of performance, brake thermal efficiency or brake specific energy consumption can have both possibilities. It depends on the operating condition and each set up configurations of the engine. Fumigating ethanol into diesel engine causes an increase in-cylinder pressure. The starting of the combustion becomes more retard due to its low cetane number. The pattern of the heat release is changed with higher premixed phase but ends faster than pure diesel operation.

EGR mostly retards the performance of the engine[26]. It causes lower brake thermal efficiency and the higher brake specific energy consumption. Because the combustion cannot be done efficiently with the lack of fresh intake air and with the fed-back exhaust gas in the system. And also, it increases CO, THC and soot due to the poorer combustion. However, it is undisputable that EGR obviously reduces NO<sub>x</sub> [27, 28, 29]. Ignition delay of the combustion has become longer with the higher portion of EGR.

Although the works and experiments about EGR and ethanol fumigation have been done, but there is still the area of how the EGR temperature will effect on the ethanol fumigated diesel engine that has not been studied much. The heat from EGR can help in vaporization of ethanol before mixing with air intake charge. The combustion characteristics would be changed and also provide the different characteristics of emissions. This work would be conducting in a common rail modified-single cylinder diesel engine with EGR and ethanol fumigation system. EGR would be cooled down before mix with the new fresh air and the fumigated ethanol for the comparison.





### 3.1 Testing components

#### 3.1.1 Engine and Dynamometer

The experiment was tested on a single cylinder-direct injection diesel engine. The engine capacity and details of its performance were shown in Table 2. The engine was coupled with radiator for cooling ability. K-type thermocouples are used for measuring the temperature for six parts of the engine i.e. 1) Engine oil temperature 2) Inlet cooling water temperature 3) Outlet cooling water temperature 4) Air intake temperature 5) Exhaust temperature and 6) EGR temperature.

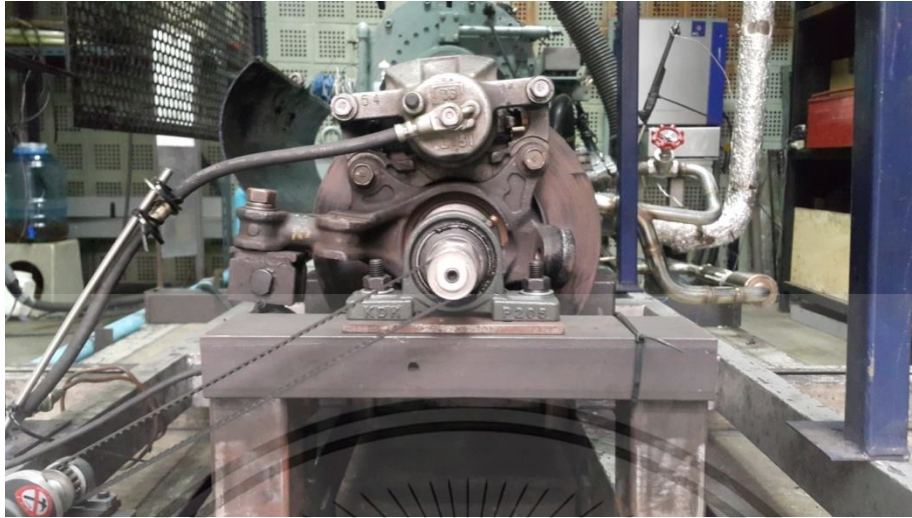
**Table 2 Test engine specifications**

|                     |                     |
|---------------------|---------------------|
| Displacement volume | 709 cm <sup>3</sup> |
| Stroke              | 96 mm               |
| Bore                | 97 mm               |
| Connecting rod      | 144.5 mm            |
| Compression ratio   | 18:1                |
| Injection Timing    | 20 bTDC             |
| Injection Pressure  | 500 Bar             |
| Maximum torque      | 45 N.m at 1600 RPM  |
| Maximum power       | 8 kW at 2000 RPM    |

The engine will be loaded by coupling with an adapted Prony brake dynamometer as shown in Figure 65. The vehicle brake has been adapted. The shaft was used to connect the flywheel of the engine and brake. Loading force was measured by S-type tension/ compression load cell with the capability of 1961 N from Minebea, model U3B1-200K-B as shown in Figure 66. The specification of this load cell is shown in Table 3.

**Table 3 Minebae U3B1-200K-B load cell specification**

| Specification            |   |
|--------------------------|---|
| Rated capacity           | 1.961 kN  |
| Safe overload            | 150 %R.C.   |
| Ultimate overload        | 200 %R.C.   |
| Rated output             | 3 mV/V±0.015 mV/V   |
| Non-linearity            | 0.05 %R.O.  |
| Hysteresis               | 0.05 %R.O.  |
| Repeatability            | 0.03 %R.O.  |
| Excitation, recommended  | 10 V or less  |
| Excitation, maximum      | 15 V  |
| Zero balance             | ±0.03 mV/V  |
| Input resistance         | 350 Ω±3.5 Ω   |
| Output resistance        | 350 Ω±5 Ω   |
| Insulation resistance    | 2 000 MΩ or more (DC50 V) (between bridge and main body)  |
| Temp. range, compensated | -10 °C ~ 70 °C  |
| Temp. range, safe        | -20 °C ~ 80 °C  |
| Temp. effect on zero     | 0.05 %R.O./10 °C  |
| Temp. effect on output   | 0.1 %LOAD/10 °C   |
| Cable                    | φ6.3, 4-cores shielded 3 m cable 3 m directly attached, Y-crimp type terminal lugs are attached at cable end. |
| Class of protection      | IP6 4   |
| Material of element      | U3B1-20K~200K-B : Alluminium alloy  |
|                          | U3B1-500K~1T-B : Alloy steel  |
| Durability               | 1 000 000 times with rated load applied.  |



**Figure 65 Prony brake dynamometer**



**Figure 66 Minebae load cell U3B1-200K-B**





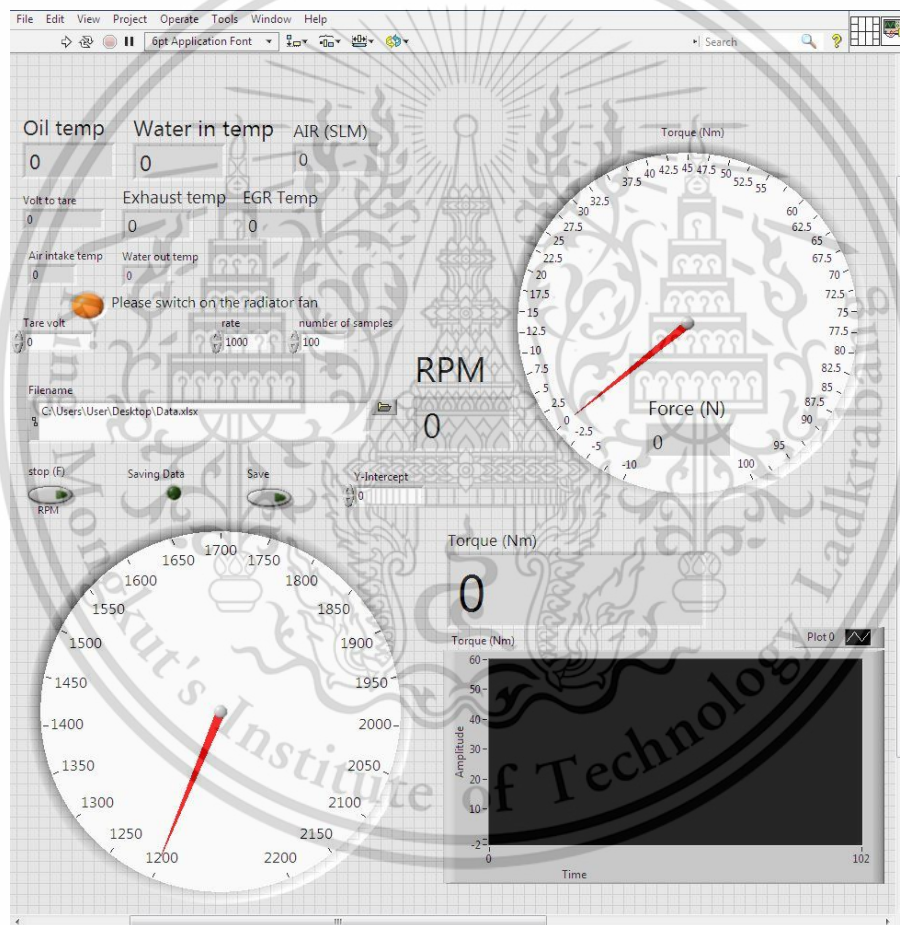
Figure 67 Autonic E40HB Encoder

### 3.1.2 Data Acquisition system

The data from load cell and thermocouples would be recorded by using Labview, a data acquisition program. The data were obtained with the frequency of 1000 Hz and the rate of sampling of 100 samples per second. National Instrument DAQ USB-6363 (Figure 68) was used for transferring the data. The interface of the program and the data channels are shown in figure Figure 69 and Figure 70.



**Figure 68 DAQ USB-6363 Data acquisition**



**Figure 69 Labview interface**



**Table 5 Kistler 6052C Specifications**

| Kistler Type 6052 C                                 |        |   |
|---|--------|---|
| Measuring range                                     | bar    | 0 ... 250                                 |
| Calibrated partial ranges                           | bar    | 0 ... 50, 0 ... 100, 0 ... 150, 0 ... 250 |
| Overload  | bar    | 300                                       |
| Sensitivity   | pC/bar | ≈ - 20                                    |
| Natural Frequency (measuring element)               | kHz    | ≈ 160                                     |
| Linearity, all ranges ( at 23 °C )                  | %/FSO  | < ± 0.3                                   |
| Axial acc. sensitivity                              | bar/g  | < 0.0002                                  |
| Radial acc. sensitivity                             | bar/g  | < 0.0005                                  |
| Operating temperature range                         | °C     | -20 ... 350                               |
| Temperature min./max.                               |        | -50 ... 400                               |
| Sensitivity change                                  |        |   |
| 200 °C ± 50 °C                                      | %      | < ± 0.5                                   |
| 23 ... 350 °C                                       | %      | < ± 2                                     |
| Thermal shock error<br>(at 1500 1/min, pmi = 9 bar) |        |   |
| Δp (short time drift)                               | bar    | < ± 0.5                                   |
| Δpmi  | %      | < ± 2                                     |
| Δpmax   | %      | < ± 1.0                                   |
| Insulation resistance at 23 °C                      | Ω      | ≥ 10 <sup>13</sup>                        |
| Shock resistance                                    | g      | 2 000                                     |
| Capacitance, without cable                          | pF     | 5   |

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### 3.1.4 Fuel Supply System

The amount of fuel consumption was measured by calculating the amount of fuel injected from the injectors. By knowing the characteristics of the injectors, the amount of injecting fuel could be obtained for each injection duration as depicted in Figure 72 and Figure 73. The diesel injector is from 4JJ-TC engine from Izusu D-Max pick-up truck with the injection pressure of 500 Bar at 20 bTDC, while the injector of Denso 23250 is used for ethanol injection for fumigation method. The injection timing of ethanol is at 160 bTDC with the injection pressure of 3 Bar. The characteristic of the diesel injector is shown below.

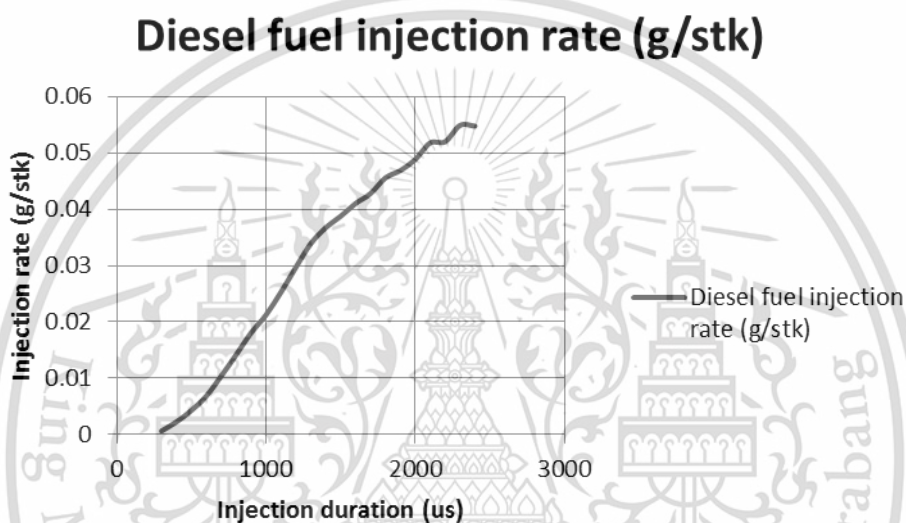


Figure 72 Diesel fuel injection rate (g/stk)

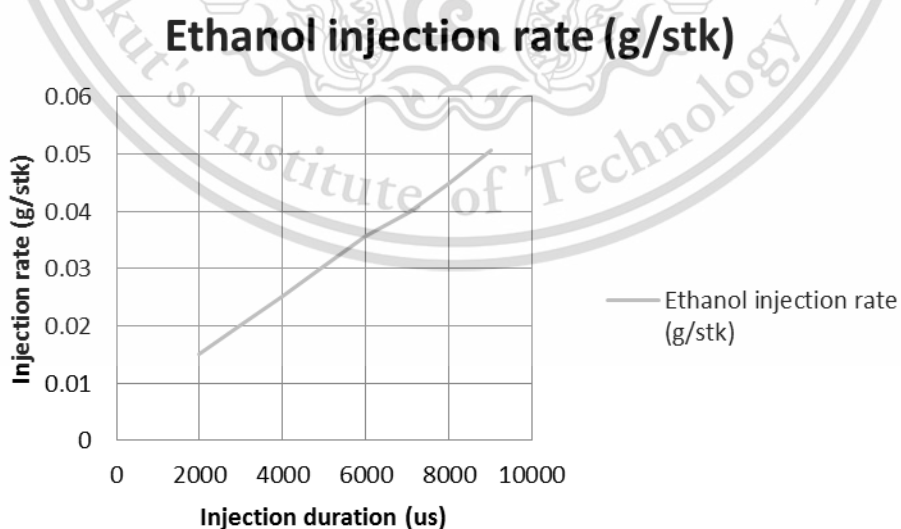


Figure 73 Ethanol injection rate (g/stk)

### 3.1.5 Air Intake and Exhaust Recirculate Gas System

Air flow intake is measured by using the air flow meter from Teledyne Hastings HFM200 (Figure 74). The scale of flow rate is 0-750 litre per minute. This flow meter is used for measuring the flow rate of air intake as volumetric flow rate and also used for measuring the rate of EGR being fed into the system. The signal of the flow meter would be sent to data acquisition system.



Figure 74 Teledyne Hastings HFM200

Exhaust gas recirculation system has been installed on the engine. The valve and piping system was created and connected the exhaust port to the intake port (Figure 75). The amount of the feeding back exhaust gas is measured by the volumetric flow as the equation (1) or as the proportion of the difference of  $\text{CO}_2$  gas in intake and exhaust as the equation (2). This experiment would define the portion of EGR as a volumetric flow portion. The variation of EGR temperature was controlled by ice bath and installed-heater as shown in Figure 76. The heater was used to heat up the water before feeding into the heat exchanger for maintaining the temperature and help in increase the temperature of exhaust gas to achieve the desire point while the ice bath was used to cooling down the exhaust gas temperature. The data would be recorded when EGR temperature reached the desire point with the deviation of  $\pm 2^\circ\text{C}$ .



Figure 75 EGR system

$$\%EGR = \frac{\text{Recirculated exhaust gas flow rate}}{\text{Flow rate of intake}} \times 100 \quad (1)$$

$$EGR \text{ ratio} = \frac{[CO_2]_{\text{intake}} - [CO_2]_{\text{ambient}}}{[CO_2]_{\text{exhaust}} - [CO_2]_{\text{ambient}}} \quad (2)$$

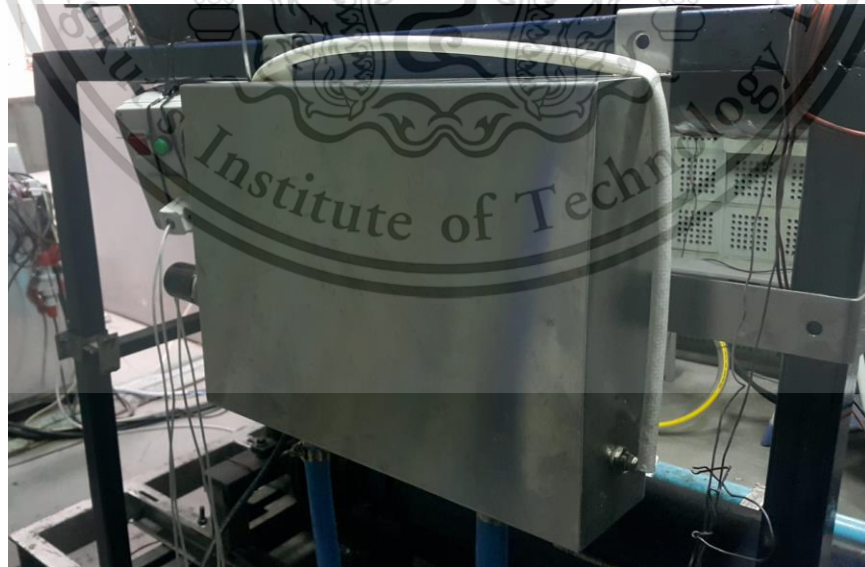


Figure 76 Heater and Ice Bath set for EGR cooling system

### 3.1.6 Gas Emissions Analyzer and Smoke Meter Measurement

The gas emissions from the engine were measured by using the Exhaust Gas Analyzer from Horiba, model MEXA-1600D as shown in Figure 77 and Figure 78. The specifications were shown in Table 6. Mexa-1600D has three compartments for measuring each kind of emission. FCA-266 is used for measuring THC by hot flame ionization method, detecting ions formed by organic compound under the combustion of hydrogen flame and  $\text{NO}_x$  by detecting an emitted light from the reaction between  $\text{NO}$  and  $\text{O}_3$  or Chemiluminescence detector method. IMA-262 was used to detect  $\text{O}_2$  by Magneto Pneumatic Detection (MPD) system. And AIA-260 is used for detecting  $\text{CO}$  and  $\text{CO}_2$  by NDIR-Non dispersive infrared principle.



Figure 77 Horiba Mexa-1600D



Figure 78 Horiba Mexa-1600D interface

Table 6 Horiba Mexa-1600D specifications

| Model   | Component                       | Method               | Range        | T90     | Noise    |
|---------|---------------------------------|----------------------|--------------|---------|----------|
| AIA-260 | NDIR<br>(250-mm cell)           | 100-3K ppm           | 3.5 s        | ±1.0 FS |          |
|         | CO-H<br>(10-mm cell)            |                      |              |         |          |
|         | CO <sub>2</sub><br>(10-mm cell) | NDIR<br>(10-mm cell) | 1-16 vol%    | 2.0 s   | ±1.0 FS% |
| FCA-266 | THC                             | Hot-FID              | 100-20K ppmC | 2.0 s   | ±1.0 FS% |
|         | NO <sub>x</sub>                 | CLD<br>(atmospheric) | 100-5K ppm   | 3.5 s   | ±1.0 FS% |
| IMA-262 | O <sub>2</sub>                  | MPD                  | 10-25 vol%   | 2.5 s   | ±1.0 FS% |
|         | EGR-CO <sub>2</sub>             | NDIR<br>(10-mm cell) | 1-10 vol%    | 2.0 s   | ±1.0 FS% |

For the smoke or the particle emission, is has been measured by AVL Smoke Meter (Figure 79). The method of measurement is detecting the black filtered soot on the filter tape by using photoelectric probe. The microprocessor will evaluate the level of soot in the FSN or  $\text{mg}/\text{m}^3$ . The resolution of the smoke meter is 0.001 FSN or  $0.01 \text{ mg}/\text{m}^3$ . The specification of the smoke meter was shown in Table 7.



Figure 79 AVL Smoke Meter

**Table 7 AVL Smoke Meter specifications**

|   |   |
|---|---|
| Measurement principle:                      | Measurement of filter paper blackening  |
| Measured value output:                      | FSN (filter smoke number) or mg/m <sup>3</sup> (soot concentration)   |
| Measurement range:                          | 0 to 10 FSN   |
| Detection limit:                            | 0.002 FSN or ~ 0.02 mg/m <sup>3</sup>   |
| Resolution:                                 | 0.001 FSN or 0.01 mg/m <sup>3</sup>   |
| Exhaust pressure ranges:                    | (-300*) -100 to 400 mbars<br>(-500*) -200 to 750 mbars with the special sampling option<br>0 to 3000 mbars with the high-pressure option<br>(* ) with activated altitude simulation |
| Maximum exhaust temperature:                | 600 °C with standard 340 mm sample probe (800 °C with 780 mm long sample probe)   |
| Power supply:                               | 100 – 115 VAC or 230 VAC, 50/60 Hz  |
| Power consumption:                          | 700 VA  |
| Compressed air (for compressed air option): | ~150l/min during purge  |
| Sample flow:                                | ~ 10 l/min  |
| Ambient conditions:                         | 5 to 55 °C / max.95 RH; without condensation<br>Sea level -500 to + 5000 m  |
| Repeatability:                              | Standard deviation 1 s = ± (0.005 FSN + 3 % of the measured value @ 10sec intake time)  |

### 3.2 Testing conditions

This experiment was conducted on fixed engine speed at 1600 RPM. To see effect of EGR on each load, the engine loads were described as low, medium and high load for 25%, 50% and 75% of maximum BMEP respectively. The engine, Kubota RT140 single cylinder diesel direct injection has been modified to be common rail electronic-control injection instead of mechanical injection. The injection pressure of 500 Bar was chosen to be used in this experiment with the injection timing at 20 degree before top dead center. The maximum BMEP of the engine is 7.97 Bar which leads to 5.97 Bar, 3.98 Bar and 1.99 Bar for high load, medium load and low load of brake mean effective pressure. These BMEPs are equal to 30 Nm, 20 Nm and 10 Nm of torque respectively. For the fumigation, ethanol has been injected into the intake port where the injector was located about 15 cm away from intake valve with the injection timing at 160 before top dead center which is the period of intake valve opening. Fuel portions were divided by energy content, 10%-30% of energy were come from ethanol as shown in tables below and were fixed on both non-EGR and EGR operation.

**Table 8 Energy portion for the injection of high load**

| High load             | E0     | E10      | E20      | E30     |
|-----------------------|--------|----------|----------|---------|
| Diesel fuel (kJ/stk)  | 1.2249 | 1.10241  | 0.9799   | 0.85743 |
| Ethanol (kJ/stk)      | 0      | 0.122491 | 0.244981 | 0.36747 |
| Total energy (kJ/stk) | 1.2249 | 1.2249   | 1.2249   | 1.2249  |

**Table 9 Energy portion for the injection of medium load**

| Medium load           | E0     | E10    | E20    | E30    |
|-----------------------|--------|--------|--------|--------|
| Diesel fuel (kJ/stk)  | 0.8614 | 0.7753 | 0.6892 | 0.6030 |
| Ethanol (kJ/stk)      | 0      | 0.0861 | 0.1722 | 0.2584 |
| Total energy (kJ/stk) | 0.8614 | 0.8614 | 0.8614 | 0.8614 |

**Table 10 Energy portion for the injection of low load**

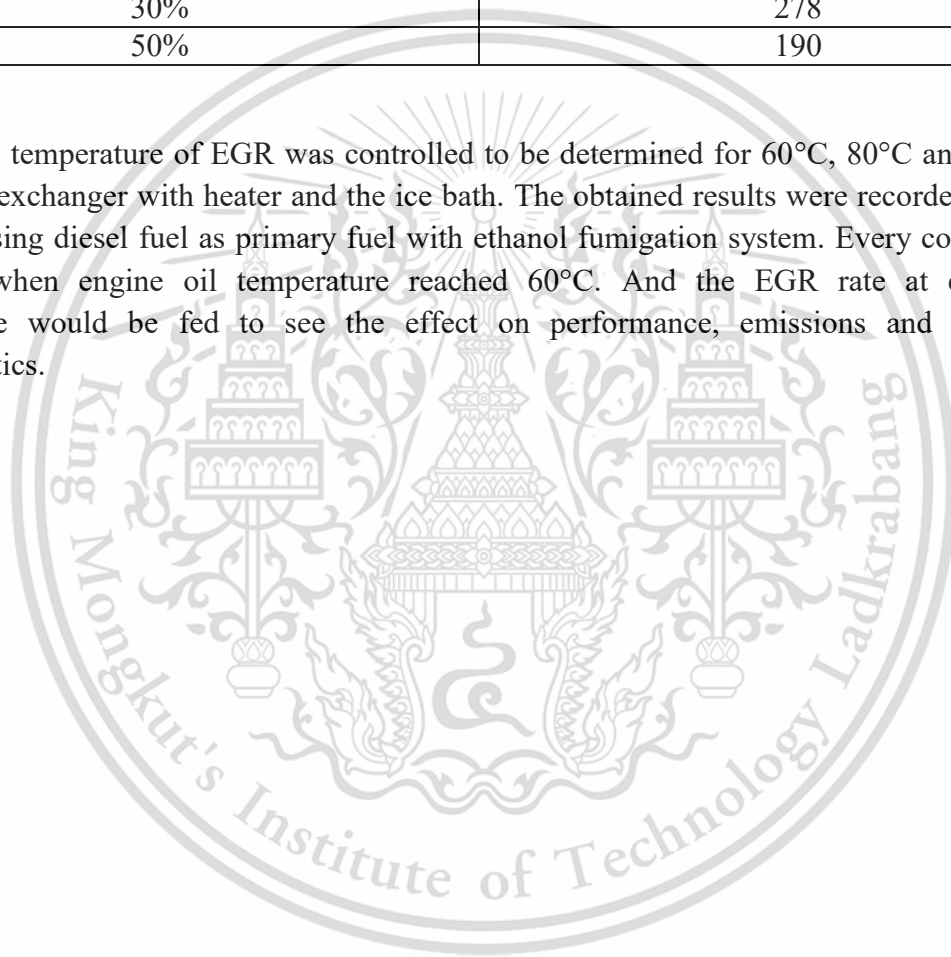
| Low load              | E0     | E10    | E20    | E30    |
|-----------------------|--------|--------|--------|--------|
| Diesel fuel (kJ/stk)  | 0.6780 | 0.6102 | 0.5424 | 0.4746 |
| Ethanol (kJ/stk)      | 0      | 0.0678 | 0.1356 | 0.2034 |
| Total energy (kJ/stk) | 0.6780 | 0.6780 | 0.6780 | 0.6780 |

Exhaust gas recirculation (EGR) would be introduced to the intake port by valve. The dilution of fresh air by EGR in this experiment would be measured by volumetric flow rate. 10%, 30% and 50% of EGR is used as EGR portion and would be described for low EGR, medium EGR and high EGR.

**Table 11 EGR operation rate**

| %EGR operation rate | Volumetric flow rate (l/min) |
|---------------------|------------------------------|
| 0                   | 380                          |
| 10%                 | 343                          |
| 30%                 | 278                          |
| 50%                 | 190                          |

The temperature of EGR was controlled to be determined for 60°C, 80°C and 100°C by using heat exchanger with heater and the ice bath. The obtained results were recorded to see the effect of using diesel fuel as primary fuel with ethanol fumigation system. Every condition was observed when engine oil temperature reached 60°C. And the EGR rate at each desire temperature would be fed to see the effect on performance, emissions and combustion characteristics.



## CHAPTER 4

### RESULTS AND DISCUSSION

To analyze the effect of exhaust gas recirculation temperature that is used in the ethanol fumigated diesel engine, the experimental results would be focused into three scopes i.e. emission, performance and combustion characteristics. Each scope will be observed through this section starting by the effect of the dilution from the recirculate gas, the effect of using ethanol fumigation method and finally the influence of the recirculate gas temperature on the diesel fumigated engine at 3 different temperature of 60°C, 80°C, 100°C.

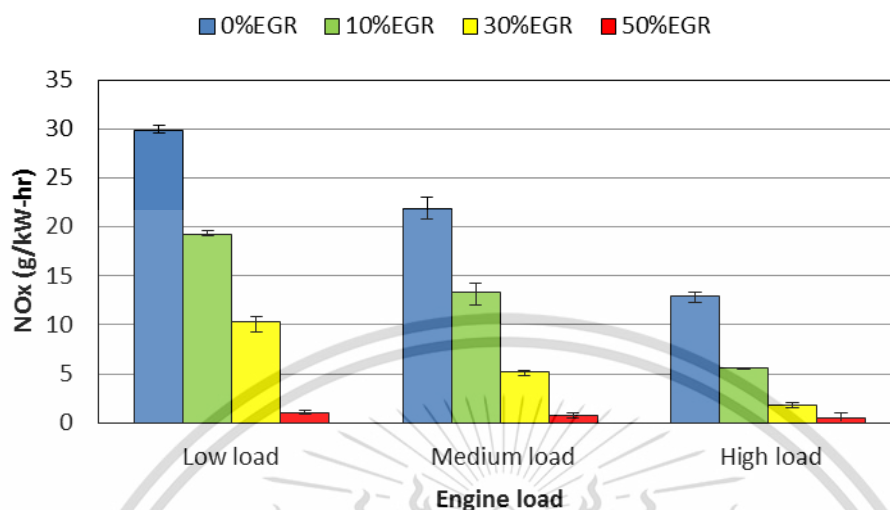
#### 4.1 Effect of the recirculate gas

The figures below show the experimental results of exhaust gas emissions i.e. NO<sub>x</sub>, CO and THC, particulate matters or soot and brake thermal efficiency with the influence of exhaust gas recirculation at each engine load. Each bar color represents each EGR condition as follows: blue is for 0%EGR, green is for 10%EGR, yellow is for 30%EGR and finally red is for 50%EGR.

##### 4.1.1 Specific NO<sub>x</sub> emission

Figure 80 shows the effect of the recirculate gas on NO<sub>x</sub> emissions at each operating load conditions. The results show that NO<sub>x</sub> emission is decreased with the increase of engine load. The availability of oxygen and nitrogen atoms in the intake air charge is compulsory for the formation of NO<sub>x</sub>. This research engine is generally operated in the lean side of air-fuel equivalence ratio which means there are lots of oxygen atoms contained in the intake air charge. Since this experiment was conducted at the constant speed, without any air restriction system, more fuel was required to achieve the higher load. The air-fuel equivalence ratio of the combustible charge has become richer and also, the available atoms of oxygen are now taken for the oxidation with carbon atoms from the injected diesel fuel as CO and CO<sub>2</sub> emissions instead of NO<sub>x</sub> due to the better combustion, results in the lower formation of NO<sub>x</sub> in the higher load.

The formation of NO<sub>x</sub> emission also decreases with respect to the higher rate of the recirculate gas. As mentioned above, the available atoms of oxygen and nitrogen are factors that affect the formation of NO<sub>x</sub>. When the recirculate gas was presented in the combustion process, the fresh air intake was substituted by the exhaust gas. The less availability of both oxygen and nitrogen atoms were shown by this dilution effect from the recirculate gas. Moreover, NO<sub>x</sub> formation is quite thermal-dependent. NO<sub>x</sub> emission has dramatic formation rate when the combustion temperature is high. The reduced amount of oxygen atoms during the combustion from using the recirculate gas also leads to the worsened combustion. The deteriorated reaction between air-fuel mixture was occurred [14], and with the high specific heat of the recirculate gas, the less combustion temperature was provided which also results in the lower formation of NO<sub>x</sub> emission.



**Figure 80 Specific NO<sub>x</sub> emission effected by the recirculate gas**

#### 4.1.2 Specific CO emission

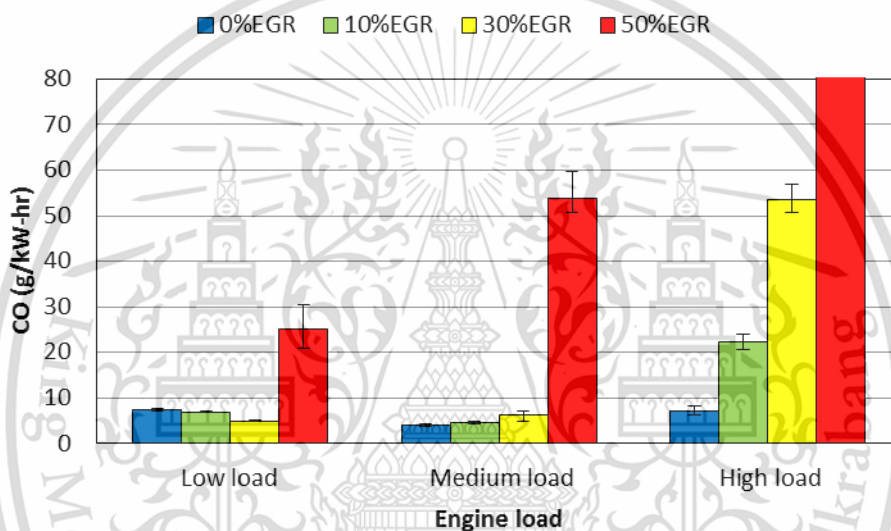
The result of carbon monoxide emission that is affected by EGR rate at each operating load is depicted in Figure 81. The level of CO emission increases with respect to the recirculate gas presence at the medium and high load operating condition. But for low load, the level of carbon monoxide shows the different tendency.

The increase of carbon monoxide level with respect to the higher amount of recirculate gas in the medium and high load operating condition is caused by the lack of oxygen. The higher amount of the recirculate gas that introduced to the combustion chamber leads to the less available area of fresh intake air, and also, air-fuel equivalence ratio becomes richer. The combustion was deteriorated. The oxidation of hydrocarbon atoms from fuel becomes more incomplete to be oxidized to CO<sub>2</sub>. At the high load operating condition, the effect of the recirculate gas shows the most tendency. The large amount of fuel was injected while the amount of fresh intake air was diluted. With the less intake air available, the incomplete combustion was possibly occurred and contributed to the higher amount of CO emission.

For the low load operating condition, carbon monoxide is decreased with the higher presence of the recirculate gas. However, when the recirculate gas was increased up to 50%, CO emission dramatically increased and had the highest level. Because the low load operating condition requires the less amount of injected fuel during the combustion process, the air-fuel equivalence ratio of this operating condition is in very lean. The combustion cannot be generated effectively with the possibly misfiring, results in the higher amount of CO emission. When the exhaust gas was recirculated into the intake air, the heat from EGR could help promote the

combustion. CO was easily oxidized to CO<sub>2</sub>. At 50%EGR, however, the heating effect was dominated by the dilution effect of the excess amount of the unburned gas.

With the same EGR concentration, CO emission tend to increase with the higher engine load. Since this emission is primarily controlled by the air-fuel equivalence ratio, generally, it will be formed most on the rich side of the air-fuel equivalence ratio from the lack of oxygen to oxidize with the carbon atoms from the fuel into CO<sub>2</sub>. In medium and high load operation range, large amount of fuel was injected. More carbon atoms were introduced through the fuel and leads the air-fuel charge becomes richer. The less oxidation reaction was provided, CO emission was increased.



**Figure 81 Specific CO emission effected by the recirculate gas**

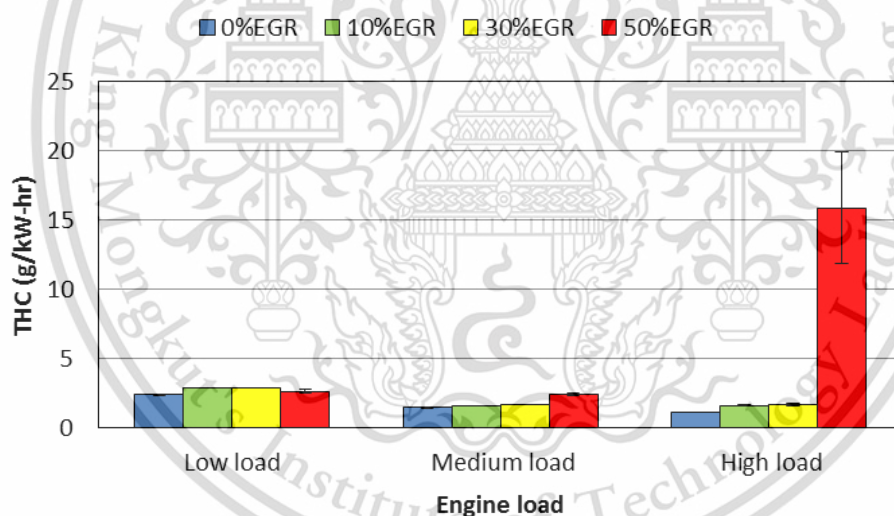
#### 4.1.3 Specific THC emission

Total hydrocarbon emission is increased with respect to the higher amount of the recirculate gas. Moreover, based on the same EGR rate, the level of THC is slightly increased during medium load to high load while low load condition shows the same decrease tendency with the result of CO emission. The result depicts in Figure 82.

Without the effect of the recirculate gas, the higher load seems to provide less formation of THC even though the more fuel injected could lead to more carbon and hydrogen atom available for the formation of total hydrocarbon but the combustion rate and temperature also becomes higher which suggest more thermal reaction and burns more carbon and hydrogen atom. The heat transfer between gas flow around crevices and cylinder walls become smaller, the less heat loss, total hydrocarbon is burned effectively.

When feeding the recirculate gas into the combustion process, it deteriorates the combustion from the dilution of the fresh intake air. The less oxygen presence from air intake charge is provided. Although the recirculate exhaust gas is non-reactive, but it still contains hydrogen and carbon atoms which could be fed back and burned in the combustion process, provides the higher amount of THC emission.

The tendency of increasing in THC emission with respect to the presence of the recirculate gas occurs differently during low load operation. In conversely, adding more EGR causes the lower THC formation for this case. Generally, the air-fuel mixture during the combustion in the diesel engine is heterogeneous. Non-uniformity of the mixture is provided for the combustion. Some part of the charge could be either locally overrich or locally overlean mixture which cannot be ignited or even support the propagation of the combustion flame. For this case, in low load operation, the mixture inside the combustion chamber becomes locally overlean which cannot be burned effectively and, thus, results in the amount of THC remains unburned. But adding the recirculate gas can influence this overlean mixture region by reducing the air presence in the charge. Some part of the charge becomes locally richer, a better combustion is provided, more hydrocarbon can be consumed through the combustion process.



**Figure 82 Specific THC emission effected by the recirculate gas**

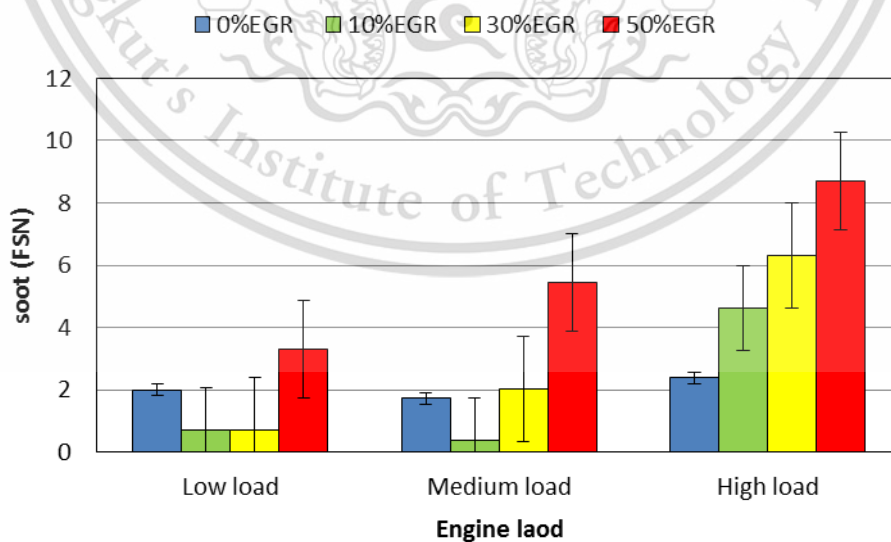
#### 4.1.4 Particulate matters

Figure 83 shows the results of soot that is affected by the amount of the recirculate gas on each operating load. The level of this emission seems to increase with respect to both load and the higher rate of the recirculate gas.

Soot level basically depends on the amount of diesel fuel injected into the combustion process for the unthrottled engine. It mostly results from the incomplete combustion of fuel hydrocarbon, which carbon, the based component of soot, is contained in the fuel. To achieve the higher load, more fuel has to be injected to the system. More carbon content is introduced to the combustion and results in the higher level of soot formation.

By feeding more of the recirculate gas, soot is more generated from the worsened combustion by the dilution of the exhaust gas. The decreased level of oxygen from the intake air results in the poorer combustion. And it also results in the less complete oxidation from soot to CO or CO<sub>2</sub>. Moreover, the fuel molecules in the current combustion process meet the precursor molecules of the previous cycle and the recirculate gas, larger amount of soot nucleation occurs.

For low load and medium load, the combustible mixture still has the large amount of air comparatively with the fuel portion. By the dilution of exhaust gas of 10%-30% could suppress the level of particulate matters by diluting the air content and brought the overlean region to become richer but still also results high NO<sub>x</sub> formation. However, when exhaust gas is fed back for 50% the combustion becomes heavily deteriorated, all emissions levels have been raised except NO<sub>x</sub>.



**Figure 83 Soot effected by the recirculate gas**

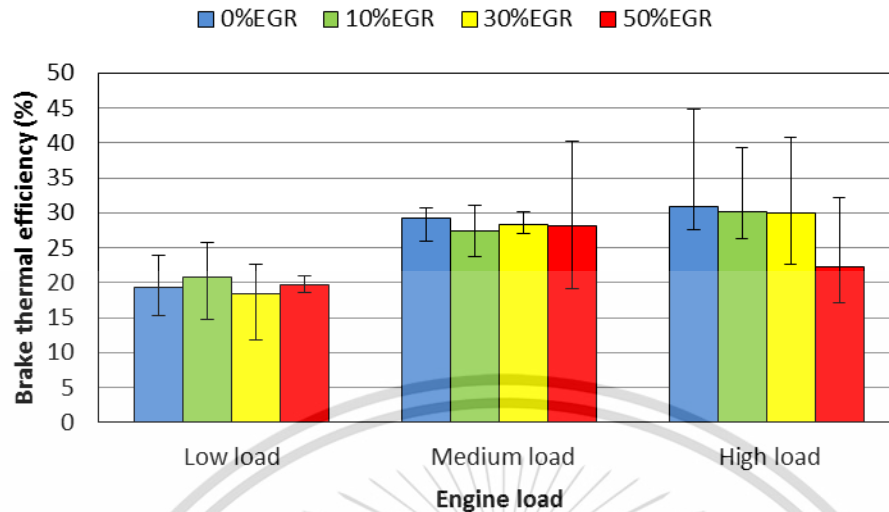
#### 4.1.5 Brake thermal efficiency

Brake thermal efficiency or BTE shows the capability of work generation that can be obtained from the heat energy of the combusted fuel. Higher BTE means the less energy wasted, more complete combustion is occurred to generate the work. Figure 84 shows the results of brake thermal efficiency of each load affected by the recirculate gas.

By increasing the load, brake thermal efficiency is indicated to become higher. It shows dramatically low level on the low load operating condition. The fuel injected in this operating condition cannot be combusted with the large amount of excess air to achieve the target operating load and also results in high specific NO<sub>x</sub>, CO and THC. When higher amount of fuel is introduced in higher load, the mixture becomes more appropriate to combust. The higher temperature of the combustion on these operating conditions also results in the less heat energy loss through the cylinder wall. Thus, brake thermal efficiency shows more effective results.

With the operating of the recirculate gas, the result suggests that the recirculate exhaust gas has the slight tendency to reduce the brake thermal efficiency. Although the results of emissions can be implied to the exacerbated combustion from the EGR but with the conditions of the same energy input of this experiment for each load, the target work still could be closely achieved. But in high load shows the large decrease in brake thermal efficiency at 50%EGR. A large starvation of fresh air is responsible for this case. The large amount of injected fuel cannot generate the combustion with the less fresh air to achieve the target load, results in the low thermal efficiency and high CO, THC emissions.

At some conditions also show the increase of brake thermal efficiency from EGR. This is because in the recirculate exhaust gas contains some amount of hydrogen and carbon atoms from contained CO<sub>2</sub> and H<sub>2</sub>O in the recirculate gas, which could be reburned as provide some energy during the combustion process [33].



**Figure 84 Brake thermal efficiency effected by the recirculate gas**

## 4.2 Effect of ethanol injection

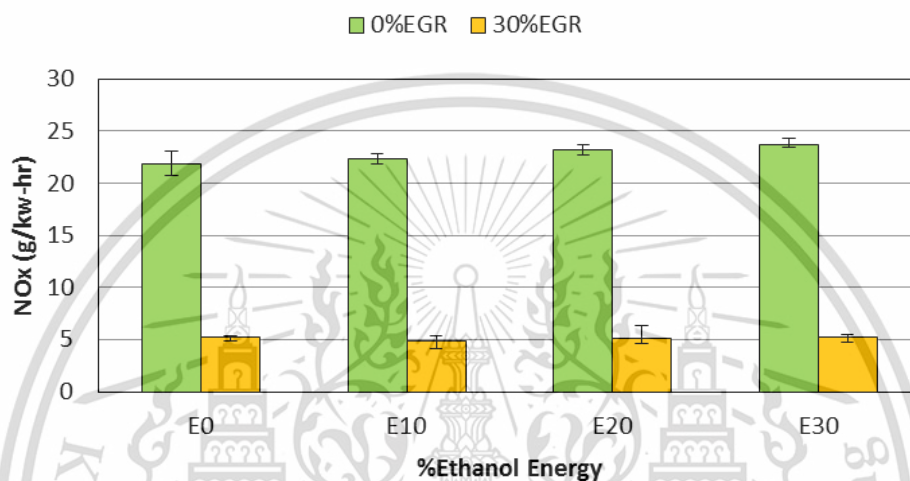
This section will be showing the experimental result of this work with the effect of ethanol. The focusing operating condition is at medium load from the former section with 30%EGR due to it is the typical medium BMEP operating range (about 4 Bar) of naturally aspirated diesel engine and 30% of EGR still shows a good brake thermal efficiency which is not far from the non-EGR operation and also provide better result when compare with 50%EGR. Ethanol will be injected by the portion of energy from 0% to 30% which the results will be reported in the view point of emissions and performance.

### 4.2.1 Specific NOx emission

Specific NOx emission of medium load with 0%EGR and 30%EGR results are depicted in the Figure 85. The result suggests that without the recirculate gas, injecting more ethanol shows the slight tendency of increasing NOx formation. Injecting of ethanol into the combustion process causes longer ignition delay and more aggressive combustion during the premixed phase [5]. Since NOx is normally formed under the high temperature of the combustion, the premixed phase of the combustion is ignited by the very –close to stoichiometric air-fuel charge which suggests the highest temperature region during the combustion process. More heat from the combustion of diesel-ethanol spurs the formation of NOx.

When operating on 30%EGR condition, it shows almost no-effect from ethanol on NOx emission. The recirculate gas can suppress the formation of NOx from the dilution effect. With

the lack of oxygen and nitrogen from intake air, the available atoms for the formation are less. Which also means, the combustion cannot be generated effectively from the lack of reactant, the lower combustion temperature is provided when operates with the recirculate gas. Although ethanol can provide more oxygen atom in the intake air charge, but in this case, injection of ethanol has no influence on NO<sub>x</sub> emission comparatively with the recirculate gas.



**Figure 85 Specific NO<sub>x</sub> emission effected by the increase of ethanol**

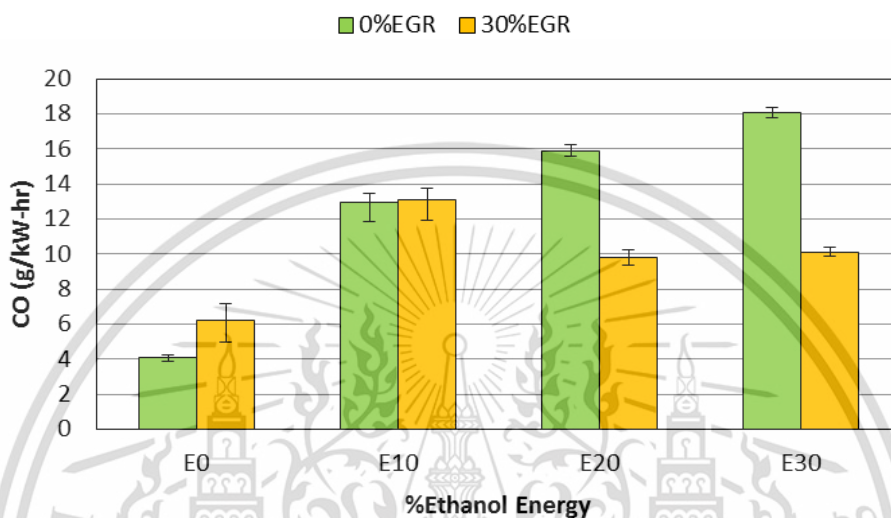
#### 4.2.2 Specific CO emission

Ethanol shows different effects with 0%EGR and 30%EGR conditions. Figure 86 depicts the tendency of increase of CO emission with respect to the higher ethanol portion.

Without the influence of the recirculate gas, ethanol causes faster rate [12] of the combustion for this case and brings more formation of carbon monoxide emissions from incomplete oxidation of CO<sub>2</sub>. Nonetheless, ethanol still has some cooling effect from its heat of vaporization which can absorb the heat from the intake air charge. With cooler air charge, the combustion temperature becomes lower. The less heat for the oxidation of CO to CO<sub>2</sub> is provided.

Applying 30%EGR also shows the same tendency of increase in CO. The highest value is on E10 while E20 and E30 results in the CO formation level lower than non-EGR operation condition. Although using EGR causes more incomplete combustion and provide higher formation of CO from its dilution effect, but using EGR with ethanol fumigation system can provide the less formation of CO emission. The recirculate gas has some heat to influence the combustion. Since ethanol has been injected into the combustion at the intake port by mixing

with the air charge and EGR, ethanol can absorb the heat from EGR to vaporize itself before mixing with air intake charge before getting into combustion process. The vaporized ethanol is mixed better and causes more homogeneity of the intake charge which improves the combustion to be more complete, provides less CO formation within the longer reaction time from EGR influence.



**Figure 86 Specific CO emission effected by the increase of ethanol**

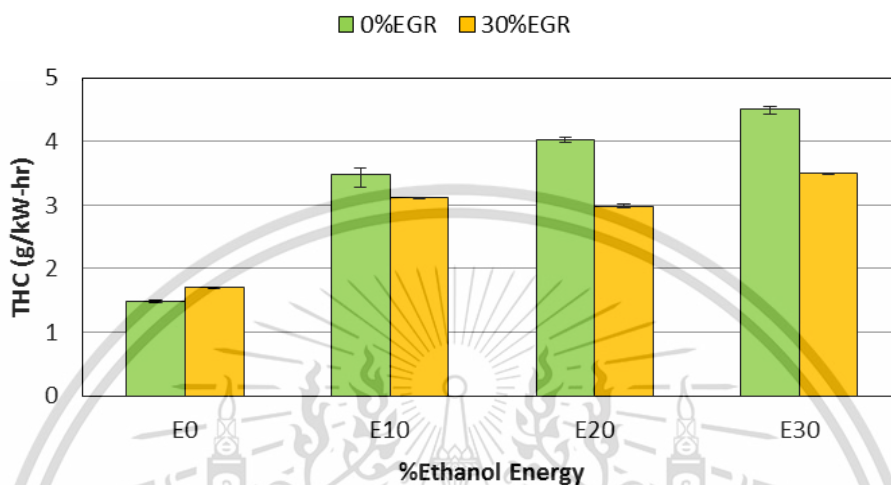
#### 4.2.3 Specific THC emission

The result of total hydrocarbon emission that is effected by ethanol is shown in the Figure 87. As in the same manner of CO emission result above, THC is increased with the higher ethanol presence without the influence of the recirculate gas. When the recirculate gas is approached, the lower THC emission is shown.

Since THC emission is resulted by the incomplete burning of hydrocarbon emission from fuel, as mention above in CO result emission, the cooling effect from ethanol has brought the combustion temperature down with the higher presence of ethanol. Also, ethanol contains oxygen, carbon and hydrogen atom, injecting ethanol at the intake port also provide more hydrocarbon atoms in the intake charge before getting into the combustion chamber. Some part of the intake charge becomes more heterogeneous to combust. These result in the higher level of THC with respect to the higher ethanol presence.

Once the recirculate gas is fed to the system, THC level becomes lower than non-EGR operating condition but still increase with higher ethanol presence. The cooling effect of ethanol is now compensated by the heat from EGR. Ethanol absorbs the heat from EGR for the vaporization before mixing with the intake air and also brings more homogeneity to the charge.

More aggressive combustion from using ethanol with the recirculate gas brings higher thermal oxidation reaction. Hydrogen and carbon atoms were burned more effective, results in the lower level of THC in comparison with non-EGR operation. But from the containment of hydrogen and carbon atoms of ethanol, the higher level of THC still shows in the higher ethanol presence.



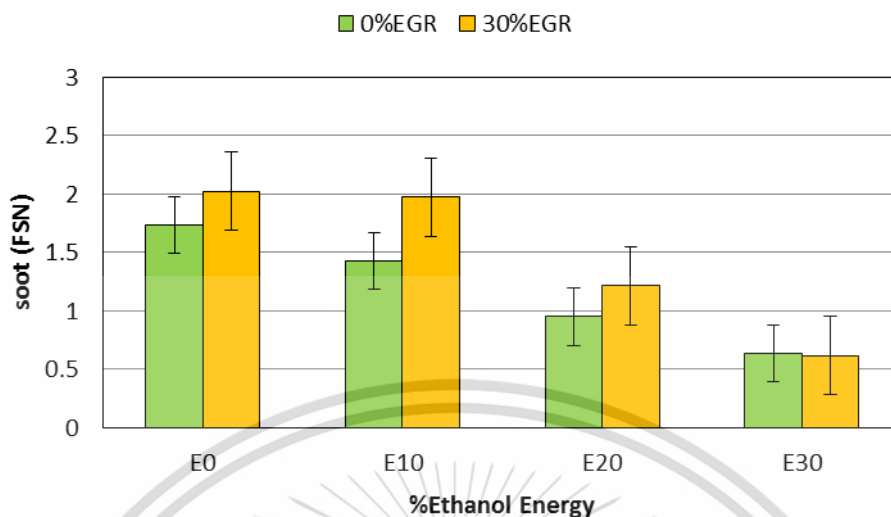
**Figure 87 Specific THC emission effected by the increase of ethanol**

#### 4.2.4 Particulate matters

Figure 88 shows the particulate matters result from the effect of using ethanol fumigation system. The level of soot has the tendency to decrease with respect to the higher ethanol presence in the system on both non-EGR and 30%EGR operation.

Particulate matters are generally results from the amount of diesel fuel. Since ethanol was added into the system by the energy percentage, the amount of diesel fuel using of this experiment has to be decrease and injecting more ethanol in order to achieve the target total energy input. The charge cooling and the lower cetane number increase ignition delay and enhance the mixing of diesel fuel with the ethanol-air mixture and causes the better air utilization [5]. Thus, the less available carbon molecules from diesel fuel were present and led to lower soot generation.

Soot still shows the higher level when the engine operates with the recirculate gas even using the ethanol fumigation system. Although it shows the better result of CO and THC when using ethanol fumigation with EGR, but the combustion was still being worsen by the recirculate gas. EGR effect still dominates the combustion. The lack of oxygen from using the recirculate gas is still responsible for the lack of completeness of burning fuel and also provides the lower oxidation of soot to CO and CO<sub>2</sub>.



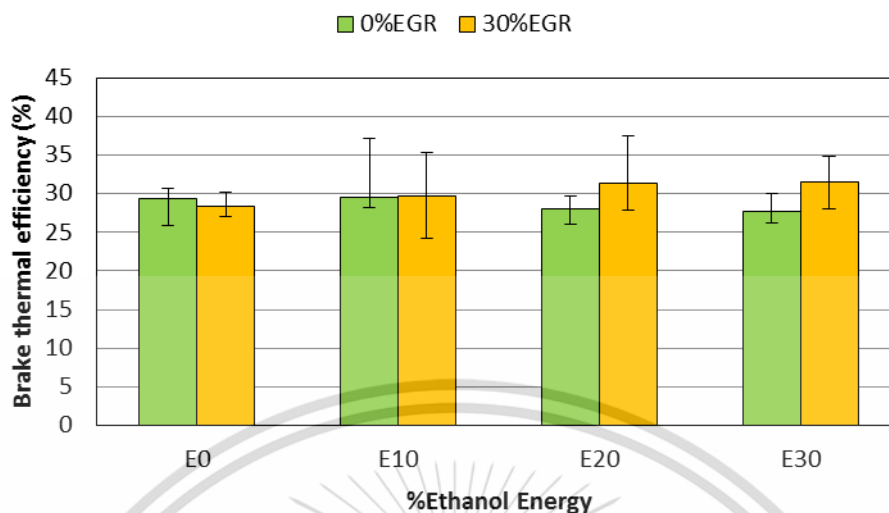
**Figure 88 Soot effected by the increase of ethanol**

#### 4.2.5 Brake thermal efficiency

Without the recirculate gas, brake thermal efficiency is effected by ethanol fumigation by slightly increase at E10 and becomes lower than non-ethanol condition at E20-E30. But with 30%EGR condition, brake thermal efficiency slightly increase with respect to ethanol portion. The highest value is E20 and then drops down at E30 as shown in Figure 89.

Using ethanol in the combustion process contributes the longer ignition delay and causes the higher rate of energy release which leads to the improved in thermal efficiency of E10. But the higher ethanol presence cools down the combustion from its high heat of vaporization property. The heat from the intake charge was absorbed to vaporize ethanol which leads to the cooler charge and the cooler combustion. More heat energy has to be provided to achieve the targeted load.

The recirculate gas changes this tendency. The heat from EGR was absorbed by ethanol for the vaporization instead of the heat from air intake for this case. The intake charge might be formed more homogeneous before the combustion occurs. And also the charge and in-cylinder environment is warmer than the former operating condition. The better combustion is provided with respect to the higher ethanol presence, results in some of lower emission above. But it is finally dominated by the cooling effect of ethanol which slightly drops down at E30.



**Figure 89 Brake thermal efficiency effected by the increase of ethanol**

### 4.3 Effect of EGR temperature

This section will be discussing about the effect of EGR temperature on ethanol fumigated diesel engine. The interesting operating condition is medium load with 30%EGR. The recirculate gas will be fed with temperature varies of 60°C, 80°C and 100°C. The result of emissions and performance is shown below.

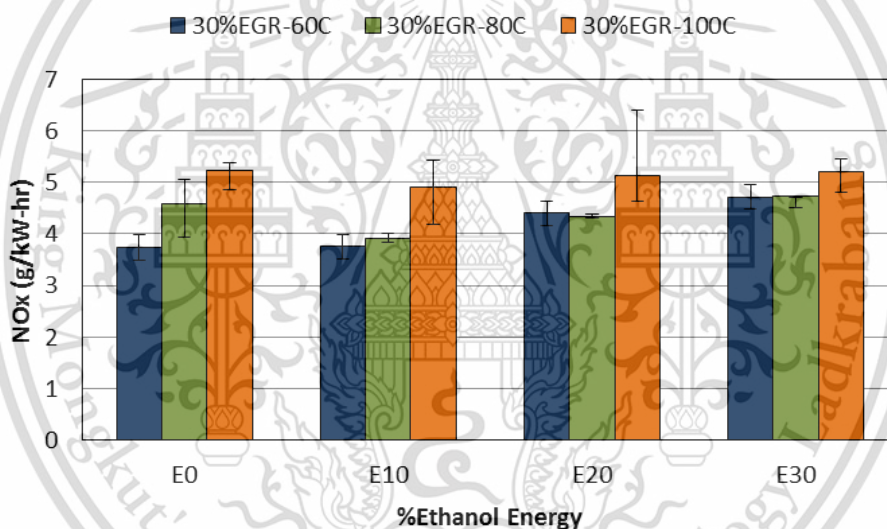
#### 4.3.1 Specific NOx emission

The behavior of the increase of NOx level with respect to the recirculate gas temperature for 30%EGR at medium load of each ethanol presence is shown in the Figure 90 below. But this tendency of increase is slightly changed within the higher ethanol presence in the combustion process.

Comparing on the same recirculate gas temperature, injecting ethanol tends to increase the level of NOx emission with respect to the higher ethanol presence. It drops down firstly at E10 due to the lack of homogeneity and the cooling effect of ethanol which also results in high CO and THC. But it keeps increasing on E20 and E30. Ethanol still causes the longer ignition delay. When diesel fuel is approached to the compressed high pressure-high temperature state of air-ethanol mixture, the combustion starts spontaneously. But this process takes longer time to start the combustion comparing with the charge that contains only air. Ethanol has low cetane number which is not appropriate to auto-ignition. During the delay period, fuel is getting accumulated. Once the appropriate air-fuel mixture, close to stoichiometric, reaches the high temperature state, the ignition can occur with the higher premixed combustion flame.

With using ethanol fumigation system, the hotter EGR generally results in the higher formation of NO<sub>x</sub> emission, as it was mentioned in above section, NO<sub>x</sub> formation is quite a temperature dependency reaction in the combustion. Although EGR is inert, non-reactant gas, its heat still can help in the vaporization of ethanol and maintaining the in-cylinder temperature during the combustion process and leads to less heat loss. With more aggressive combustion during premixed phase from using of ethanol fumigation and the lower amount of heat loss, the higher combustion temperature is resulted. More formation rate of NO<sub>x</sub> occurs with the hotter recirculate gas due to this reason.

However, in the higher presence of ethanol such as E20, at 60°C and 80°C of the recirculate gas temperature result slightly change for the formation of NO<sub>x</sub>. The hotter recirculate gas also has the higher specific heat which could absorb the heat in nearby area. The combustion could become cooler by this phenomenon which causes the slightly lower NO<sub>x</sub> formation and also insignificant change in CO result.



**Figure 90 Specific NO<sub>x</sub> emission effected by the recirculate gas temperature**

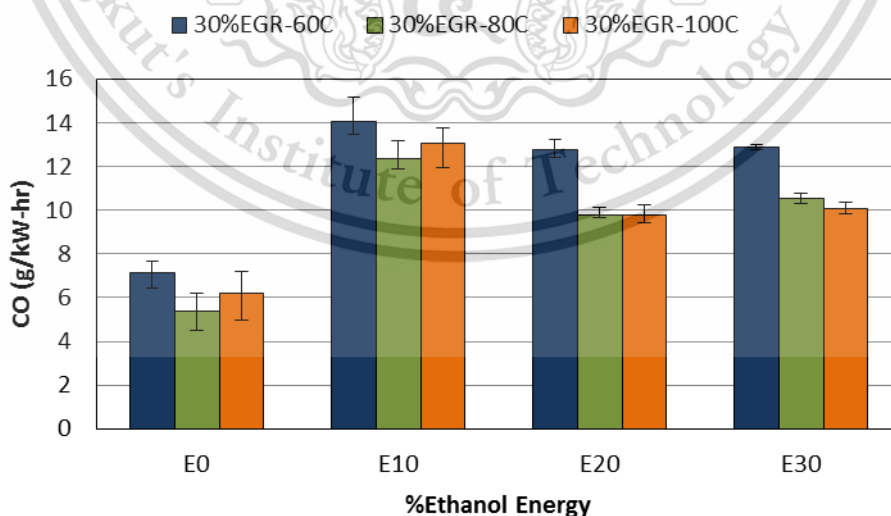
### 4.3.2 Specific CO emission

The higher temperature of the recirculate gas reduces the level of CO formation due to more complete oxidation of CO<sub>2</sub> emission. Injecting ethanol in the higher presence also leads to the less formation of CO as shown in Figure 91.

Carbon monoxide emission shows the same tendency on each EGR temperature, which is increasing in CO level at E10 and then drops down with the higher ethanol presence. The property of high heat of vaporization of ethanol still influences the charge to become cooler at E10 condition. The vaporized ethanol also cannot be mixed uniformly for this case, results in the incomplete oxidation of CO to CO<sub>2</sub>. But with more ethanol injected, this effect is dominated by more homogeneity of vaporized ethanol in the intake air charge the combustion and oxidation is improved, provides the less CO emission level.

The higher recirculate gas temperature helps preventing heat loss during the combustion from the warmer intake charge. It also leads to more sufficient heat energy in the cylinder for letting hydrocarbon atoms from fuels use to oxidize from CO to CO<sub>2</sub> as it can be observed in the lower level of CO when the recirculate gas has higher temperature.

But for some cases, such as E0 and E10, at 80°C of EGR provides the lowest CO formation and increase the level of formation at 100°C. The effect of the higher specific heat of the hotter gas is now dominating these cases. The heat from the recirculate gas might help in maintaining the in-cylinder temperature but the hotter exhaust gas also has the higher specific heat which can reduce local flame temperature [14]. CO cannot be oxidized to CO<sub>2</sub> in the colder region. Less complete thermal oxidation occurs and has brought up the level of CO for these cases.



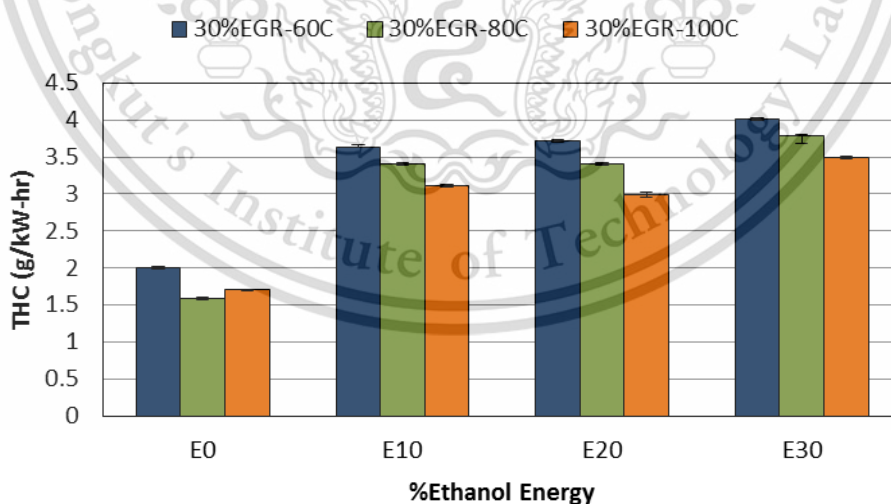
**Figure 91 Specific CO emission effected by the recirculate gas temperature**

### 4.3.3 Specific THC

The level of THC emission with the influence of using different recirculate gas temperature is shown in Figure 92. With the higher recirculate gas temperature, total hydrocarbon is getting decrease while more ethanol presence still results in the increase this emission.

The injection of ethanol into the intake port to be mixed with air as an air-ethanol intake charge provide carbon and hydrogen atom in the intake charge. The compression stroke has to compress air-ethanol which some carbon and hydrogen atoms can flow and stuck into some of the crevices. The combustion flame cannot reach and burn these hydrocarbon atoms, results in the increase of THC emission with respect to the higher ethanol presence based on the same EGR temperature.

From E10 to E30 conditions, the hotter EGR decreases the level of THC emission. This is caused by the heat from EGR helps in the combustion process in the same manner as CO emission result. The higher heat and more completeness can be achieved through these ethanol-influenced conditions. The effect of higher heat amount from the recirculate gas provides the better utilization of the fuels in the overlean region, and also the less heat transfer through cylinder wall and crevices with the longer duration before flame quenching can occur, providing the less formation of total hydrocarbon under the higher temperature of the recirculate gas. However, this result sometime shows the sign of less efficient combustion through the cooler local flame temperature on E0 at 100°C EGR condition caused by the higher specific heat of hotter recirculate gas similar to CO emission.



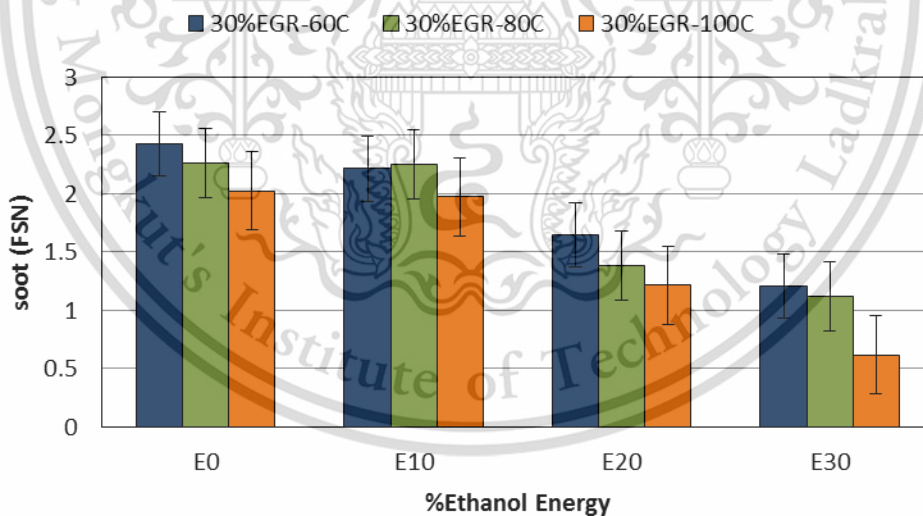
**Figure 92 Specific THC emission effected by the recirculate gas temperature**

#### 4.3.4 Particulate Matters

The concentration of soot also shows the tendency to be decreasing with respect to the EGR temperature as shown in Figure 93. Soot is getting reduced with respect to the higher recirculate gas temperature on all ethanol presence.

The reduction of soot with respect to the higher ethanol presence was mentioned in section 4.2.4 above. The decrease amount of diesel fuel which contains approximately 12 carbon atoms was substitute by ethanol which contains 2 of carbon atoms. The less carbon atoms were provided for soot generation. Also with the longer ignition delay, there was more available time for the air-ethanol charge to be mixed with the injected diesel fuel for the better utilization. These result in the lower soot generation with the higher ethanol presence.

By observing on the same ethanol presence, the higher EGR temperature causes the higher heat from the combustion. The fuel-air mixture can be burned effectively by its better maintained in-cylinder temperature from the heat of EGR. The less available molecules for soot generation were shown. Since soot particles are clusters of solid carbon spheres which is a solid carbon with HC and other absorbed component [32], the more complete burning of hydrocarbons atoms leads to the less generation of the particles. Thus, apart from the lowered-level of particulate matters that is affected by substitute ethanol to the decreased diesel fuel amount. Using the hotter exhaust gas also provides the lowered particulate matters emission.



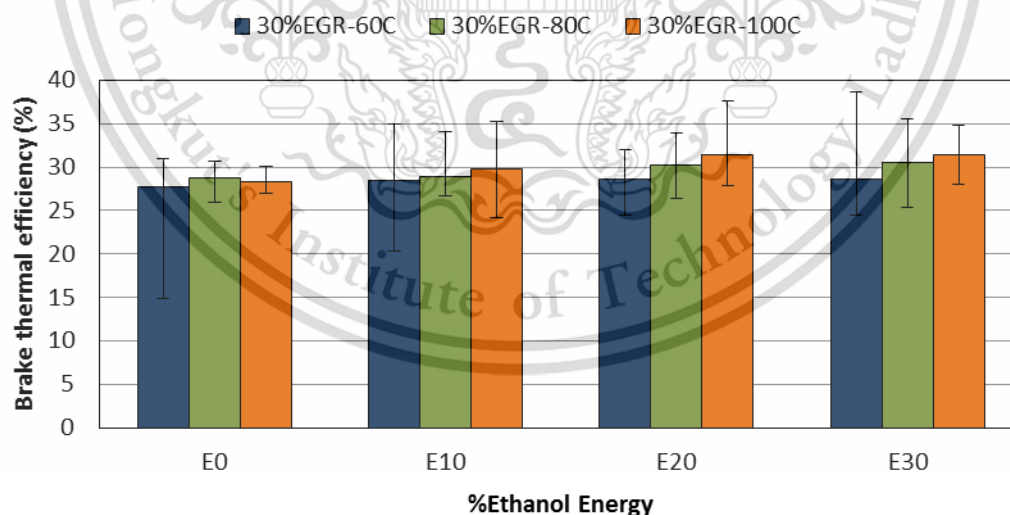
**Figure 93 Soot effected by the recirculate gas temperature**

#### 4.3.4 Brake thermal efficiency

The influence of the recirculate gas temperature of the medium load operating condition depicts in the Figure 94. On every ethanol presence, brake thermal efficiency slightly increase with respect to the higher recirculate gas temperature.

Although ethanol can absorb heat from the intake charge and leads to the lower combustion temperature, but for this case, using ethanol fumigation with EGR, the intake charge and in-cylinder environment becomes warmer. The less heat loss is provided through the combustion which results in the high thermal efficiency. Also, the heat from EGR was absorbed by the ethanol for its self-vaporization. With the longer ignition delay caused by ethanol, the mixture has more available time to be mix uniformly, a better combustion can occur with the higher ethanol presence and provide the less amount of some emissions.

Feeding the recirculate gas with the different temperature, the higher recirculate gas temperature contributes the higher thermal efficiency. The less heat energy wasted during the combustion is provided from the warmer in-cylinder environment and the warmer intake charge. The lower amount of heat loss is occurred with the higher recirculate gas temperature, more heat energy can be used as the desired output load. The highest brake thermal efficiency is at E30 condition. At some condition, E0, shows the lower brake thermal efficiency when EGR temperature rises up from 80°C to 100°C. The higher specific heat of the hotter gas dominates the effect. The exacerbated combustion was generated in the higher EGR temperature. This result also conforms to CO and THC results.



**Figure 94 Brake thermal efficiency effected by the recirculate gas temperature**

#### 4.4 Combustion Characteristics

The combustion characteristics or the combustion patterns generally are shown in the term of rate of heat release (RoHR). This result could be observed by obtaining the in-cylinder pressure with respect to the engine crank angle degree and convert them in to the rate of heat release according to equation (3). For this section, the specific heat ratio was set to be constant for 1.3 as suggested [31] although in particular situation the specific heat ratio could be changed because it is the function of various factors such as temperature, pressure, air-fuel ratio, etc. But some works have already suggested that the influence of the charge composition on the specific heat ratio value is very small [34, 35].

$$\frac{dQ_{net}}{d\theta} = \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dp}{d\theta} \quad (3)$$

Where  $Q_{net}$  is the net heat energy release.

$\gamma$  is the ratio of specific heats.

$p$  is the pressure on each volume changes (Pa).

$V$  is the displacement volume (m<sup>3</sup>).

$\theta$  is the crank angle position (degree).

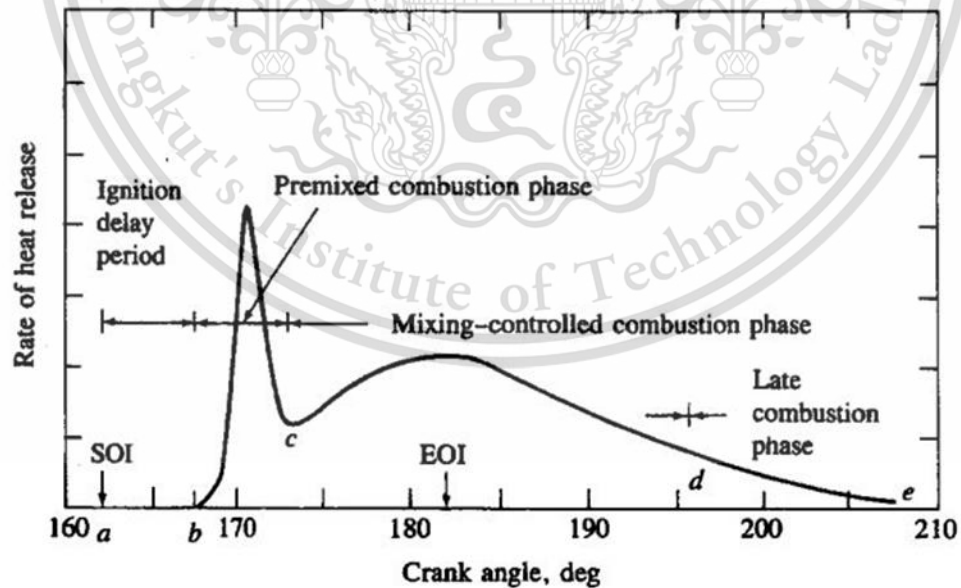


Figure 95 Rate of heat release patterns of typical diesel combustion [Heywood, 1988]

The phase of diesel combustion can be distinguished into three phases which is premixed or the phase when the heat is rapidly generated in few crank angle by the mixture of fuel and air within the flammability limits that is accumulated during the ignition delay and can spontaneously ignite, mixing controlled or the phase of the rate of the combustion is controlled by the availability of the mixture during the combustion and late combustion phase which can be seen when the rate of heat release becomes near zero as shown in Figure 95 above.

This section will be discussing the rate of heat release that obtained from the in-cylinder pressure data that can be affected by the influence of the recirculate gas on each EGR rate, the effect of each ethanol presence and finally the effect of the temperature of recirculate gas. The focusing operating condition is the medium load operation range with BMEP of 4 Bar which is the general operating range of natural-aspirated diesel engine. The data will be shown in this section as obtained in-cylinder pressure plotted on each engine crank angle and the rate of heat release.

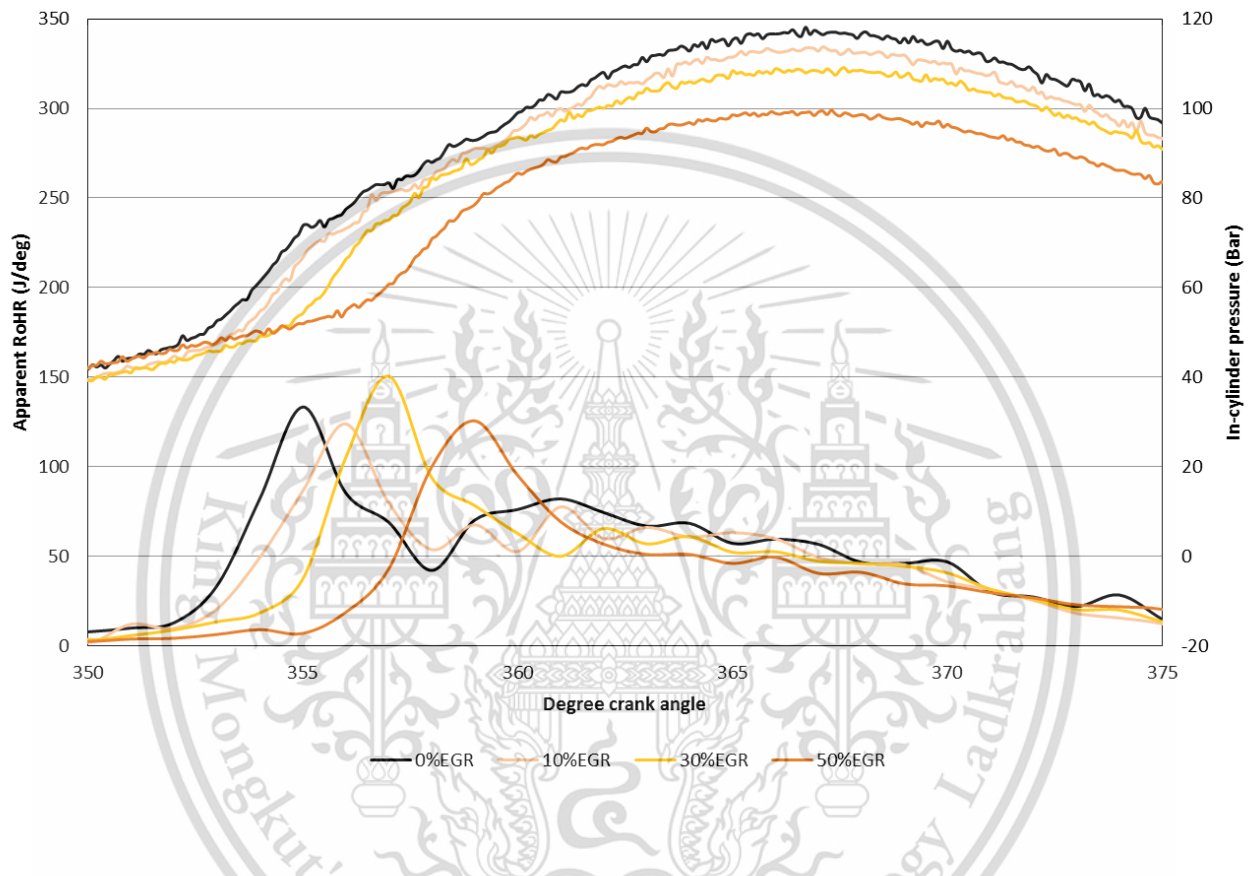
#### 4.4.1 Effect of the recirculate gas

The combustion characteristics of medium load operating condition with the effect of the different rate of the recirculate gas is depicted as in-cylinder pressure and the rate of heat release in Figure 96. Each color will represent as follows: black is for 0%EGR condition, pink for 10%EGR, orange for 30%EGR while 50% will be present in brown color.

The patterns of in-cylinder pressure show that, with the higher presence of the recirculate gas, the peak pressure drops down. Moreover, the higher rate of EGR also results in the longer ignition delay which can be observed as the crank angle that the change of the slope on pressure( $\theta$ ) curve.

Converting the data from in-cylinder pressure into the rate of heat release, the combustion patterns show that the presence of the recirculate changes the characteristics of the combustion. In non-EGR operating condition, the combustion patterns can be distinguished into three phases obviously following the ignition delay period. But when the recirculate gas is introduced to the system, the patterns are becoming more difficult to define, especially on the mixing controlled-phase. The recirculate gas delays the auto-ignition due to it has high specific heat. It absorbs the heat from air intake charge which also results in the lower peak of the in-cylinder pressure. The temperature of the intake charge becomes inappropriate to ignite. And also by the substitution of the recirculate gas on the intake air, the less fresh air was provided during the combustion. The local air-fuel equivalence ratio becomes difficult to predict, some part of the charge might become closer to the stoichiometric ratio with the higher presence of the recirculate gas. During the ignition delay, more fuel is also getting accumulated. By these two phenomena, the ignition can be generated more aggressive. This can suggest to the higher premixed phase of 30%EGR condition. However, with the dilution effect on fresh oxygen content by the recirculate gas, the combustion is difficult to be complete. The available mixture to get combust during the mixing-

controlled phase becomes lower and leads to the lower level of energy released during this phase. These result in the lower combustion temperature. More formation of CO, THC and soot was provided from the less oxidation reaction and also provides the less NO<sub>x</sub> formation due to the lower combustion temperature and the lack of nitrogen and oxygen atoms.



**Figure 96 Combustion characteristics effected by the presence of the recirculate gas**

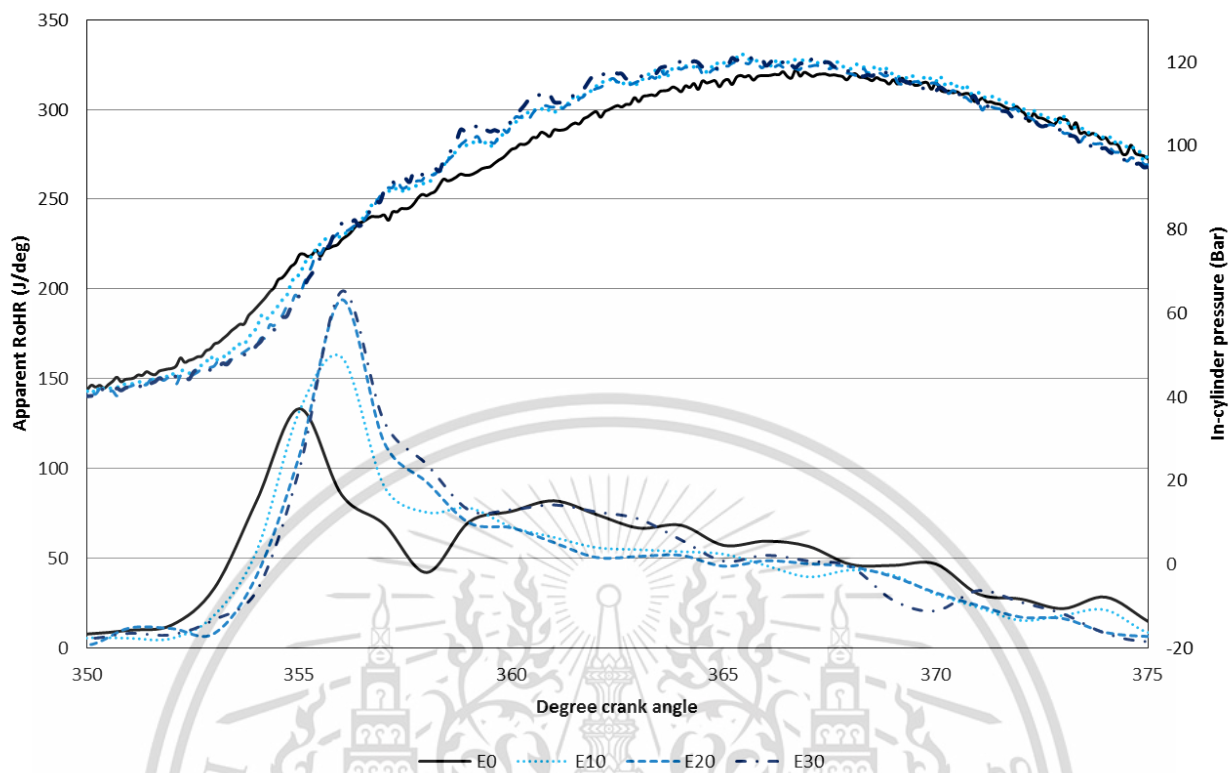
#### 4.4.2 Effect of ethanol

Figure 97 shows the combustion characteristics when the engine is operating on the ethanol fumigation system. The darker color means the higher presence of ethanol during the combustion process while the black color represents the non-ethanol condition.

The obtained in-cylinder pressure shows the longer ignition delay when more ethanol is present in the combustion process. The peak pressure also becomes slightly higher with the higher presence of ethanol.

The rate of heat release patterns are changed when the engine operates in the ethanol fumigation mode. Comparatively with non-ethanol operating condition, the higher energy was released during the premixed phase is shown with the higher ethanol presence in the system. Since the primary fuel of this combustion is diesel fuel which generally ignites by the auto-ignition from the higher cetane number property, the introduction of ethanol into the combustion process can lead to the longer ignition delay from its lower cetane number. Ethanol has been fumigated in the intake port to mix with the air intake. The engine has to compress the charge that contains both air and ethanol. But it still cannot auto-ignite even in the high temperature state in the engine unless the diesel fuel approaches. During this period, the large amount of fuel is now gathered. Once the diesel fuel is injected into the high pressure and high temperature charge, the charge region that is close-to stoichiometric ratio ignites, the dramatic heat energy was released from the burning of accumulated ethanol in the diesel burning flame. Thus, results in the very high premixed phase and high peak pressure during the ethanol fumigation operation which results in the increase of NO<sub>x</sub>. However, the lower heat energy was released during mixing-controlled phase and the late combustion phase since the main part of the mixture is already burned. The cooler intake charge also results in the lower combustion temperature. Ethanol has high heat of vaporization property which can absorb the heat from the air intake and bring the charge temperature becomes lower. The lower temperature was also provided from the combustion which leads to the higher CO and THC.

Some case shows slightly different effect on E20 and E30 operating condition. Both of these conditions almost ignite on the same angle with the almost same peak of the premixed combustion phase. For this case, the homogeneity of the charge might be responsible. During the ethanol fumigation operating condition, the combustible charge contains both air and ethanol. The higher presence of ethanol might not be mixed uniformly. Some part of the charge of E30 condition might have the local area similar to E20 condition which can ignite in the similar manner. With the possible same period of ignition delay, the amount of accumulated fuel is quite on the close amount. This also results in the same peak of energy released in the premixed phase. However, for E30 condition, it still provides higher amount of heat energy develops over the mixing-controlled phase in comparison with E20.



**Figure 97 Combustion characteristics effected by the presence of ethanol**

#### 4.4.3 Effect of the recirculate gas on ethanol

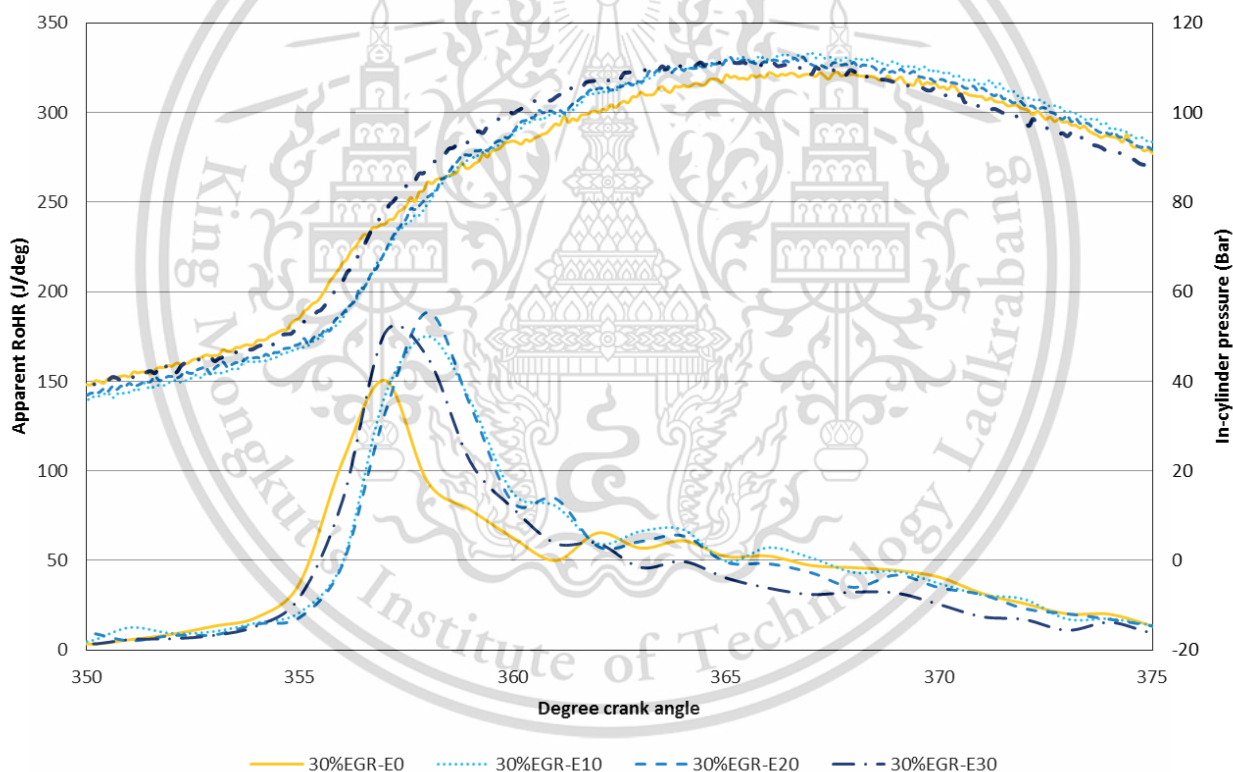
To analyze the effect of the recirculate gas on ethanol, 30%EGR rate was chosen to be observed since it has good thermal efficiency close to non-EGR operating condition. The obtained in-cylinder pressure data and the rate of heat release of the combustion when engine operates in the ethanol fumigation mode and with 30%EGR presence is shown in Figure 98. Same as the former section above, the darker of blue color means the higher presence of ethanol while the orange color represents 30%EGR-E0 condition.

Ethanol still shows effect on the ignition delay and the higher of peak pressure of the in-cylinder pressure data even the combustion is getting deteriorated by the influence of the recirculate gas. But at the highest presence of ethanol in this case, E30, the combustion starts earliest among all ethanol operating conditions.

The combustion patterns still show the higher premixed phase with the higher ethanol presence same as it was mentioned the section 4.4.2 above. The lower cetane number of ethanol

is still responsible in the longer ignition delay that leads to the more gathering of fuel during the delay period, ends up as the rapid rate of energy released in the few crank angle during the premixed phase. But with the dilution effect of the recirculate gas, the less oxygen atom cannot contribute the ignition effectively. The less heat energy was released during the premixed combustion phase and the mixing-controlled phase comparing non-EGR operating condition.

However, the heat from the recirculate gas can compensate the cooling effect of ethanol. Since ethanol needs the heat from its environment for the vaporization and mixing with the air intake charge, the heat from the recirculate is available for the absorption from ethanol to vaporize before mixing. The vaporized ethanol is now mixed better in the higher presence with the air intake charge. More completeness is brought to the combustion and results in the lower CO and THC emissions. And also, the charge becomes more appropriate to ignite due to the locally proper air-fuel equivalence ratio. The earlier starting of the combustion is occurred for the case of E30.



**Figure 98 Combustion characteristics effected by using 30%EGR on each ethanol presence**

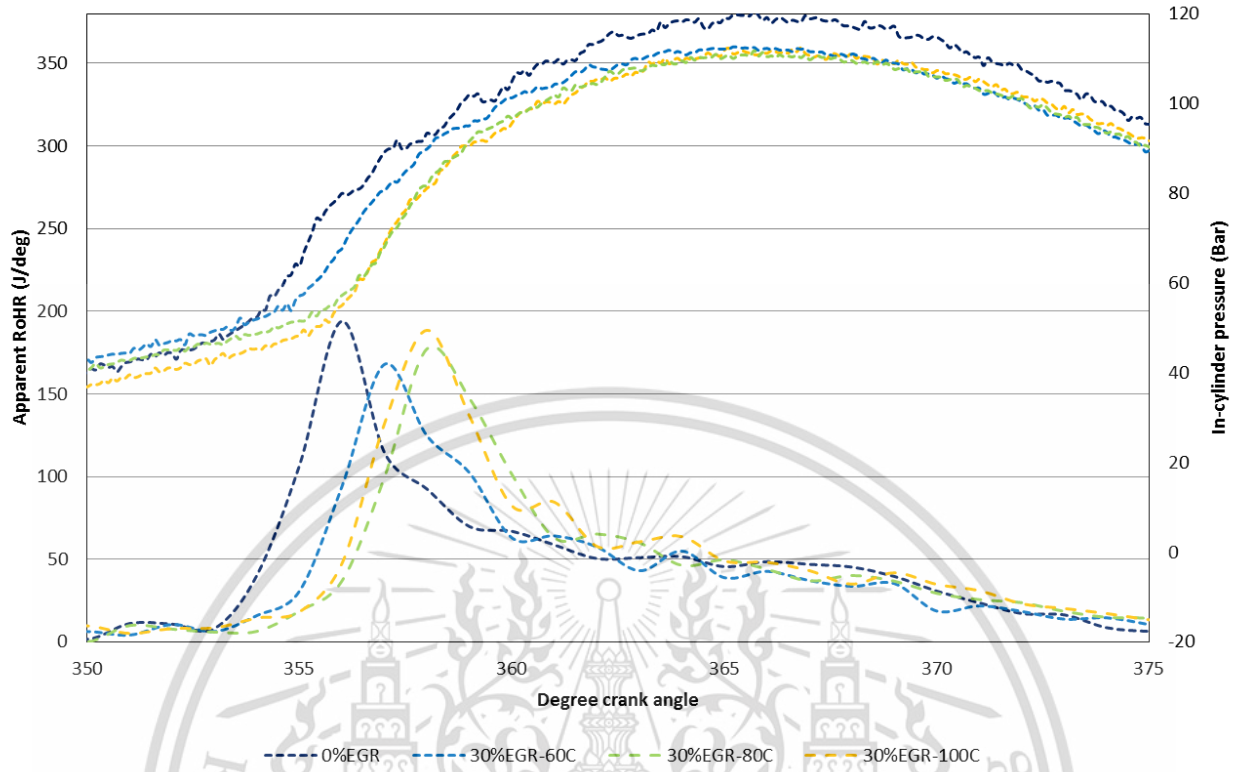
#### 4.4.4 Effect of the recirculate gas temperature

The patterns of the combustion characteristics of the engine that is operating on 30%EGR with the presence of ethanol of 20% by energy is depicted as the in-cylinder pressure and the rate of heat release in Figure 99. Since E20 suggests in the low level of all emission, it is chosen to be an interesting scope. The navy-dashed represent the condition of E20 without the influence of the recirculate gas. The blue, green and orange represent the recirculate gas temperature of 60°C, 80°C and 100°C respectively.

The in-cylinder pressure data shows the tendency of lowering in the in-cylinder peak pressure when the different recirculate gas temperature is introduced to the combustion process. Moreover, the higher temperature of the recirculate also results in the longer ignition delay. But from the in-cylinder pressure data, between 80°C and 100°C EGR condition is unnoticeable changed.

Without the effect of the recirculate gas of E20 operating condition, the earliest combustion is provided for this case with also the highest level of heat energy released during the premixed phase. By introducing the recirculate gas, the starting of the combustion becomes late due to the lack of oxygen content from the dilution effect and the specific heat of the recirculate gas. The ignition delay is getting retarder with the hotter recirculate gas approaches. Besides from the reason that ethanol can utilize the heat from the recirculate gas to vaporize itself and cause more homogeneity through the better mixing, the higher heat from the hotter recirculate gas may also help in maintaining the combustion temperature through the warmer intake charge presents in the combustion process. Which it also means the less heat wasted through the heat transfer during the combustion. More completeness of the combustion can be generated and gradually causes the lower carbon emissions like CO and THC through more available heat for the oxidation and also the leads to the higher NO<sub>x</sub> formation level from the higher temperature influenced.

For some case, the effect of the higher specific heat of the higher recirculate gas temperature is shown at 80°C condition. The starting of the combustion of 80°C recirculate gas is slightly after the ignition of 100°C which the effect of the specific heat is dominated by the higher temperature environment inside the cylinder. At 80°C EGR operating condition, the heat of the air-ethanol intake charge is absorbed. The charge becomes cooler and having the longer ignition delay. More heterogeneity of the charge may also be provided since this temperature is close to the boiling point of ethanol (78°C). These reasons cause the reduction of heat energy released for 80°C EGR case. This also conforms to the slightly lower NO<sub>x</sub> formation and unchanged level of CO at 80°C of the recirculate gas condition.



**Figure 99 Combustion characteristics effected by EGR temperature on E20**

## CHAPTER 5

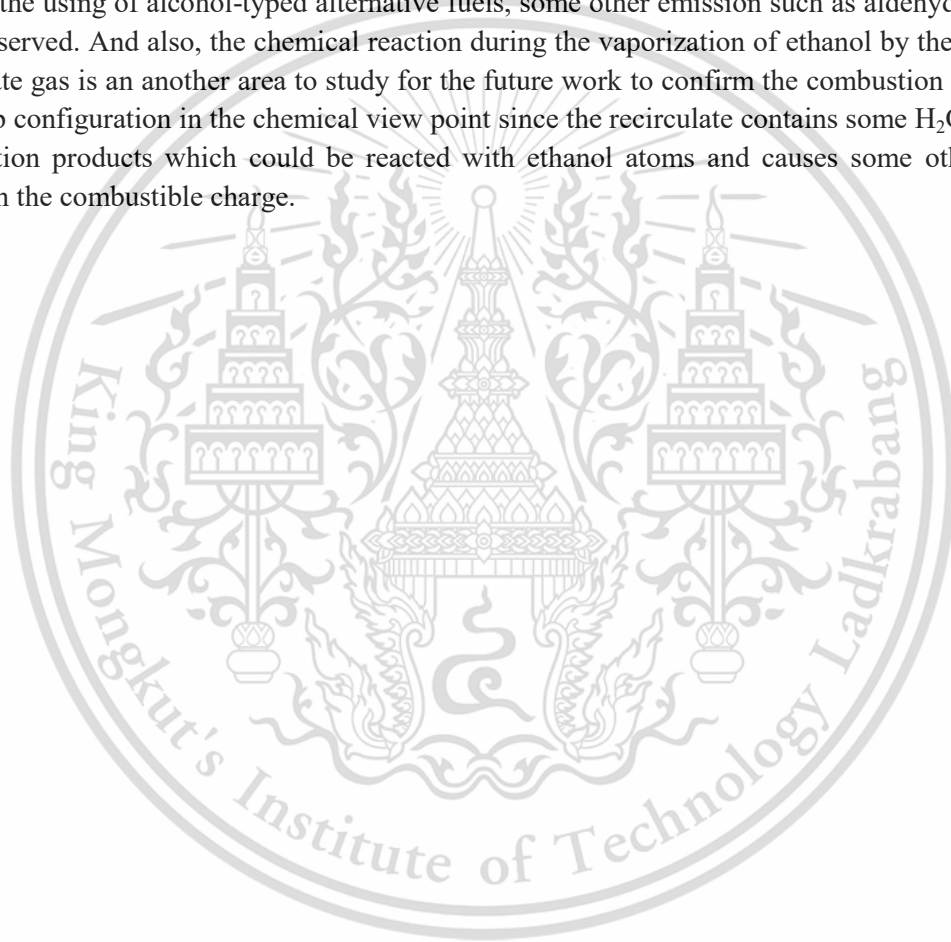
### CONCLUSION

The effects of ethanol injection in EGR on performance and emissions of an ethanol fumigated diesel engine has investigated the influence of the recirculate gas (EGR) temperature on the diesel engine that is used ethanol fumigation method to promote the use of the alternative fuels. The experiment was divided into three parts to analyze the effect of the recirculate gas by feeding the recirculate gas of 10%, 30% and 50% onto each engine operating load conditions, effects of using an ethanol fumigation method by choosing medium load operating conditions with 30%EGR feed to fumigate ethanol by portion of energy for 10%-30% and finally, the effect of the recirculate gas temperature by varying the recirculate gas temperature of 60°C, 80°C and 100°C. The experimental results can be concluded as follows:

1. The recirculate gas mainly effect on the combustion by its dilution effect. With the higher presence of the recirculate gas, the lower of fresh oxygen content is provided. The combustion was deteriorated, results in the lower brake thermal efficiency and the higher of CO, THC and soot emissions while its abatement of fresh nitrogen and oxygen provides the less formation of NO<sub>x</sub> with the lower combustion temperature.
2. The presence of ethanol in the combustion system leads to the cooler combustion temperature from its high heat of vaporization property. The higher level of CO and THC was provided through the incomplete thermal oxidation. Ethanol also results in higher NO<sub>x</sub> formation which also conforms to the higher premixed phase from more rapid energy released. However, for this research, brake thermal efficiency is decreased in the higher ethanol presence. Soot was decrease from the lessened amount of the diesel fuel.
3. Adding ethanol on EGR operation can slightly improve the brake thermal efficiency by absorbing heat from the recirculate gas for the vaporization and causes more uniform charge for more complete combustion. With the higher ethanol presence on 30%EGR condition, the level of CO and THC are also suppressed.
4. The different temperature of the recirculate gas shows significant effects on the combustion. The increase in EGR temperature lead to the higher brake thermal efficiency. The less heat energy wasted was generated from using the hotter recirculate gas. It also brings the higher formation of NO<sub>x</sub> with the decrease of CO, THC and soot while sometimes this effect is dominated by the higher specific heat of the hotter recirculate gas. It would be shown as the increase of CO and THC and the lower formation of NO<sub>x</sub> in some case.
5. The combustion patterns show that the use of ethanol and recirculate gas retards the starting of the combustion. The phases of the combustion becomes more difficult to distinguish when there is ethanol presence, the higher premixed phase was shown with an unobvious mixing-controlled phase and the late combustion phase. The use of recirculate gas also slightly changes the patterns with the longer ignition delay and lower energy

provided. The higher temperature of the exhaust gas provides the slightly longer ignition delay from the higher specific heat of the hotter gas. But it sometimes gets dominated by the higher temperature environment inside the cylinder such as E20 100°C EGR case.

From above results it can be conclude that the alternative fuel such as ethanol can be applied to the diesel engine that uses the exhaust gas recirculation technique to reduce NO<sub>x</sub> emission. By the fumigation method, the miscibility problem can be overcome, the higher presence of ethanol can be used, leads to the less diesel fuel consumption and also provide the less CO and THC emission with the better brake thermal efficiency. However, the temperature of the recirculate gas should be controlled appropriately to avoid any unwanted disadvantages that can be caused such as the higher generation of emissions in some cases. With the using of alcohol-typed alternative fuels, some other emission such as aldehydes should be carefully observed. And also, the chemical reaction during the vaporization of ethanol by the influence of the recirculate gas is an another area to study for the future work to confirm the combustion phenomenon of this set up configuration in the chemical view point since the recirculate contains some H<sub>2</sub>O and CO<sub>2</sub> as the combustion products which could be reacted with ethanol atoms and causes some other chemical compound in the combustible charge.



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# The Effects of Ethanol Injection in EGR on Performance and Emissions of an Ethanol Fumigated Diesel Engine

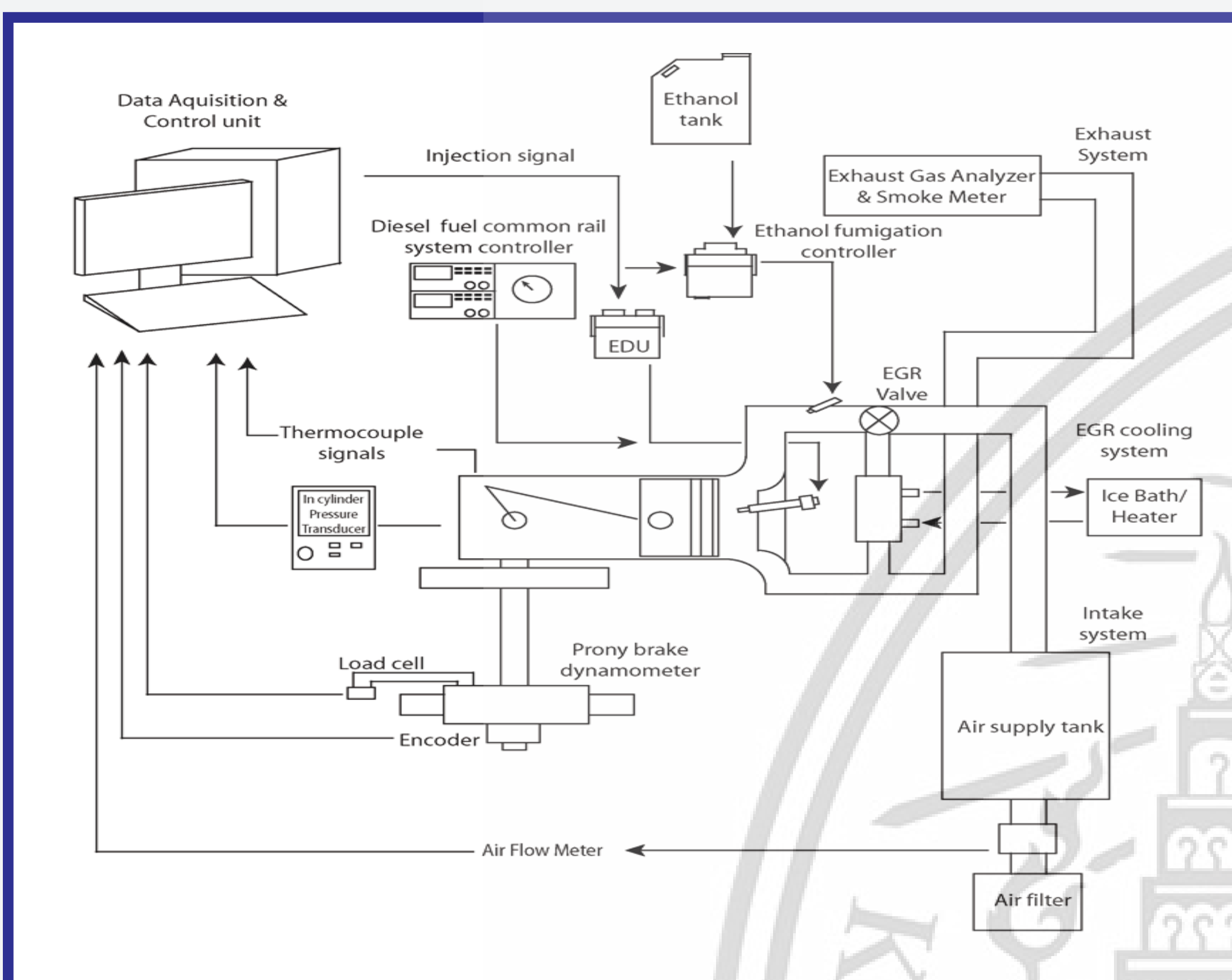
Kontorn Thammakul (TAIST-Tokyo Tech Automotive Engineering #9)  
Advisors : Dr.Manida Tongroon, Asst. Prof. Dr. Chinda Chareonpornpanich, Prof. Hidenori Kosaka

## Abstract

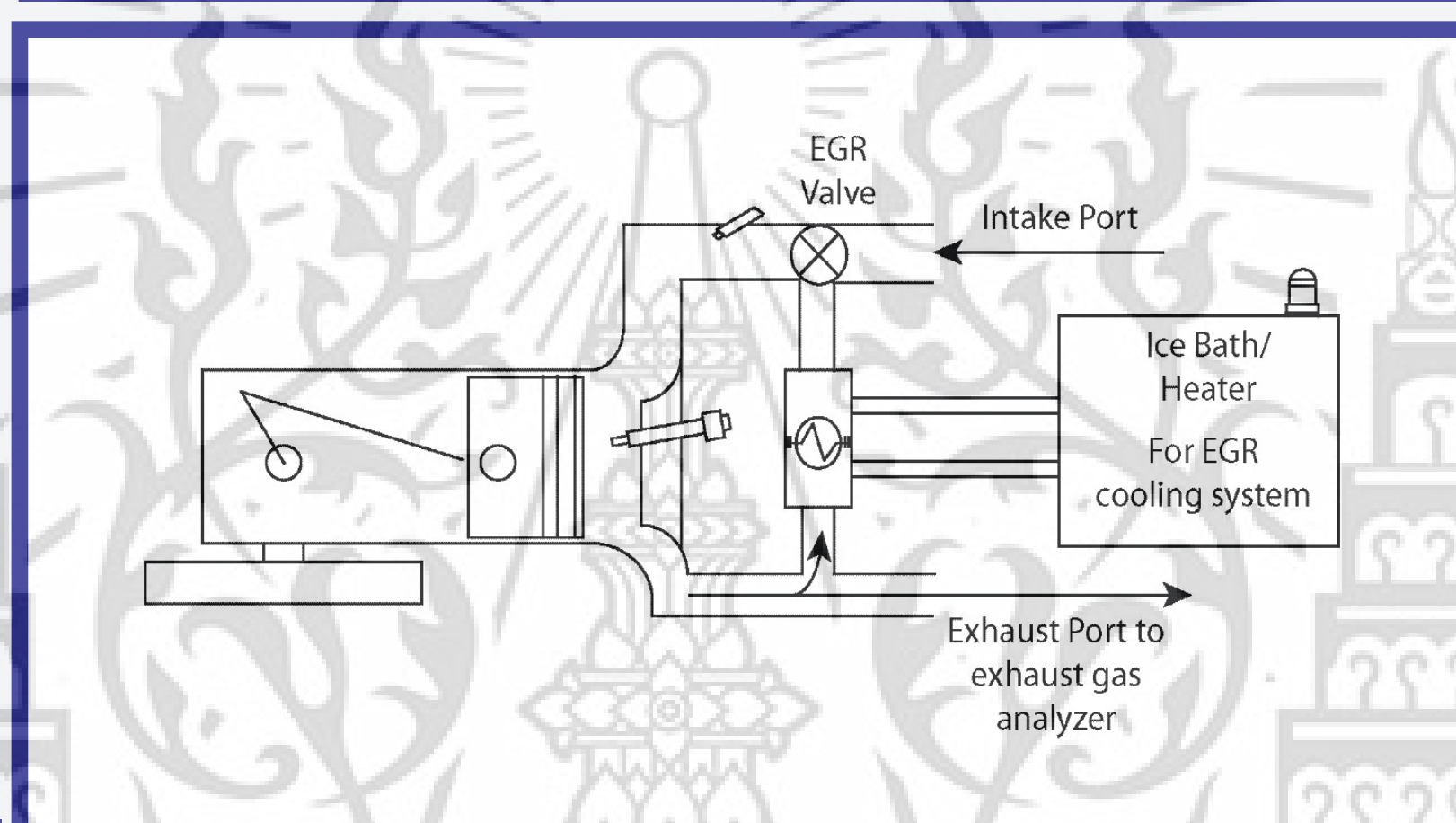
Ethanol could be used with diesel fuel in many ways. Fumigation system is more complicated but could overcome the ethanol miscibility problem. Exhaust gas recirculation (EGR) is now installed with commercial diesel engine for treating NOx emission problem. Using EGR with ethanol fumigation system would help compensating the combustion when using EGR while keeping the low level of NOx generation. By injecting ethanol into the intake port after EGR inlet, ethanol will vaporize and dissociate by the heat from exhaust gas. This research will focus on the effect of ethanol fumigation and EGR temperature on the diesel engine. The different temperature of EGR will influence in vaporization of the ethanol and is supposed to promote the combustion temperature and reduce more emissions as the expected result.



## Test Apparatus



|                            |                     |
|----------------------------|---------------------|
| Displacement volume        | 709 cm <sup>3</sup> |
| Stroke                     | 96 mm               |
| Bore                       | 97 mm               |
| Connecting rod             | 144.5 mm            |
| Compression ratio          | 18:1                |
| Diesel Injection Timing    | 20 bTDC             |
| Diesel Injection Pressure  | 500 Bar             |
| Ethanol Injection Timing   | 160 bTDC            |
| Ethanol Injection Pressure | 3 Bar               |
| Maximum torque             | 45 N.m at 1600 RPM  |
| Maximum power              | 8 kW at 2000 RPM    |

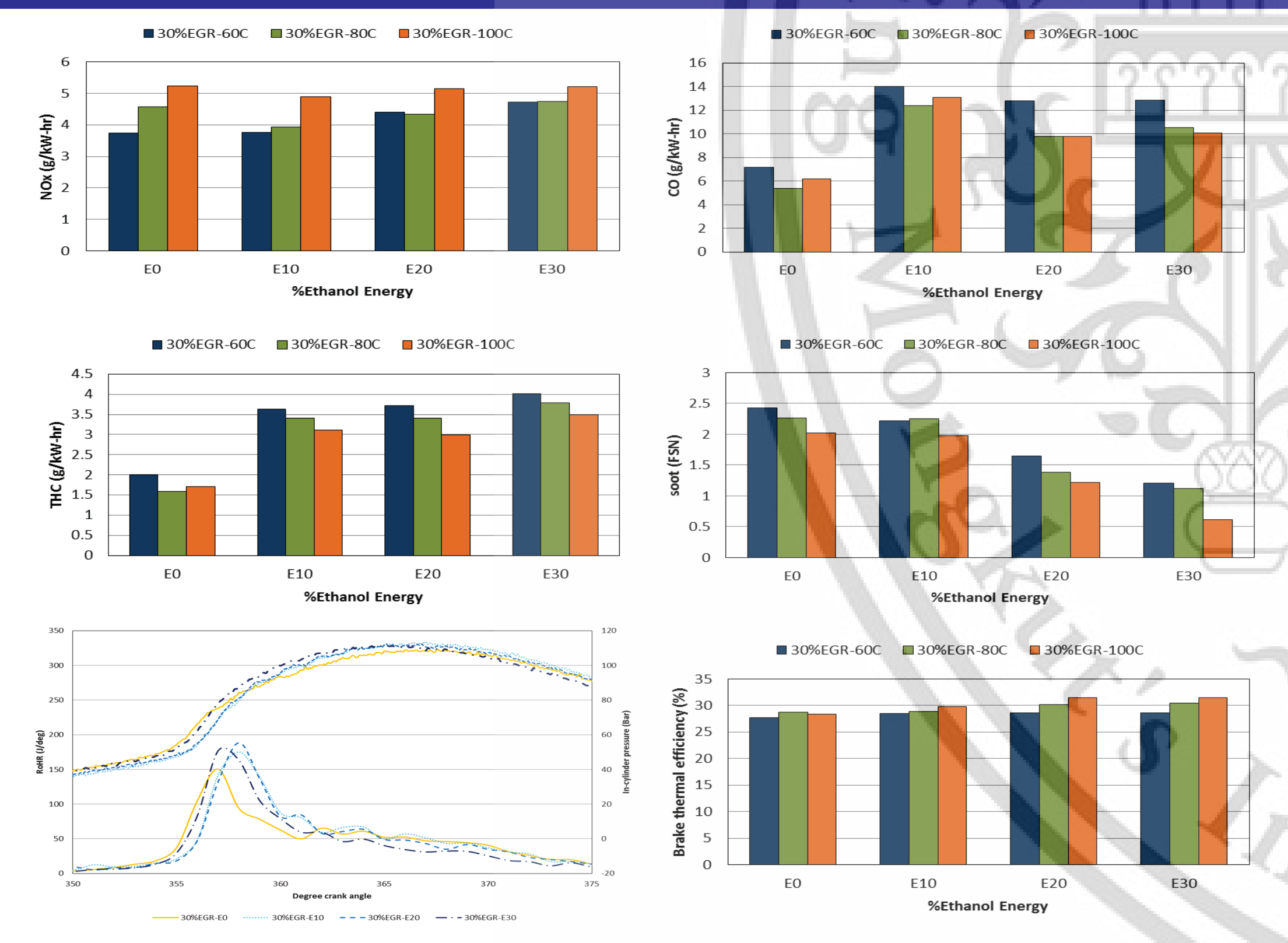


The experiment was conducted on a common rail-modified-single cylinder diesel engine coupled with the exhaust recirculation piping and intake port ethanol injection. The capability of the engine is shown in the table on the left.

The recirculate gas was varied for 10%, 30% and 50% of the volumetric flow to see the effect of the exhaust recirculate gas on the combustion, emission and performance on each engine load. Then, the experiment of ethanol fumigation was tested on both without EGR operation and with 30%EGR operation on the medium load of the engine. Ethanol was injected as energy portion of 0-30%. The ethanol injector was located at the intake port after the EGR port, with distance of 15 cm away from the intake valve. For the effect of EGR temperature, it was tested by feeding 30% of the recirculate gas and varying its temperatures for 60°C, 80°C and 100°C. Its temperature was controlled by using both ice bath and heater.

The performance data and the in-cylinder pressure was obtained by using Labview data acquisition program. The gas emission was measured and recorded by Horiba Mexa-1600D exhaust gas analyzer while soot emission level was measured by AVL smoke meter.

## Results



When the recirculate gas is fed into the combustion process, NOx emission was decreased since the recirculate gas directly substitute the fresh air intake which contains O<sub>2</sub> and N<sub>2</sub>. This substitution dilutes the amount of the available atoms of the O<sub>2</sub> and N<sub>2</sub> for NOx formation. Moreover, the lack of oxygen also caused the deteriorated combustion. The incompleteness of the combustion was brought and caused the higher level of CO, THC with the lower brake thermal efficiency. The combustion was also started later than non-EGR operation. The less heat energy was released when using the exhaust gas recirculation system.

The injection of ethanol provided the higher level of CO and THC through the less complete oxidation during the combustion. The combustion became cooler since ethanol has high heat of vaporization which it has to absorb more heat to vaporize itself before mixing with the air intake charge. Also, ethanol has low cetane number, brings the longer ignition delay. During the delay period, more fuels were accumulated. When the combustible charge reached the appropriate state of high pressure and temperature with the approaching of the diesel fuel, the heat energy was rapidly released. It can be seen as the higher amount of energy released during the premixed phase and also caused the higher formation of NOx emission. However, using EGR can suppress the formation of emissions in the higher ethanol presence with the slightly higher brake thermal efficiency.

The different recirculate gas affects to the emission and performance differently. Apart from helping in the vaporization of ethanol the hotter recirculate gas can help maintain in-cylinder temperature. The hotter environment inside the cylinder contributes more complete combustion and leads to the lower level of CO, THC and soot while NOx and the brake thermal efficiency were increased. At some conditions, this effect was dominated by the higher specific heat of the hotter recirculate gas. CO and THC were increased again in the higher temperature of the recirculate gas. And it also starts the combustion slightly earlier than the cooler recirculate gas temperature condition.

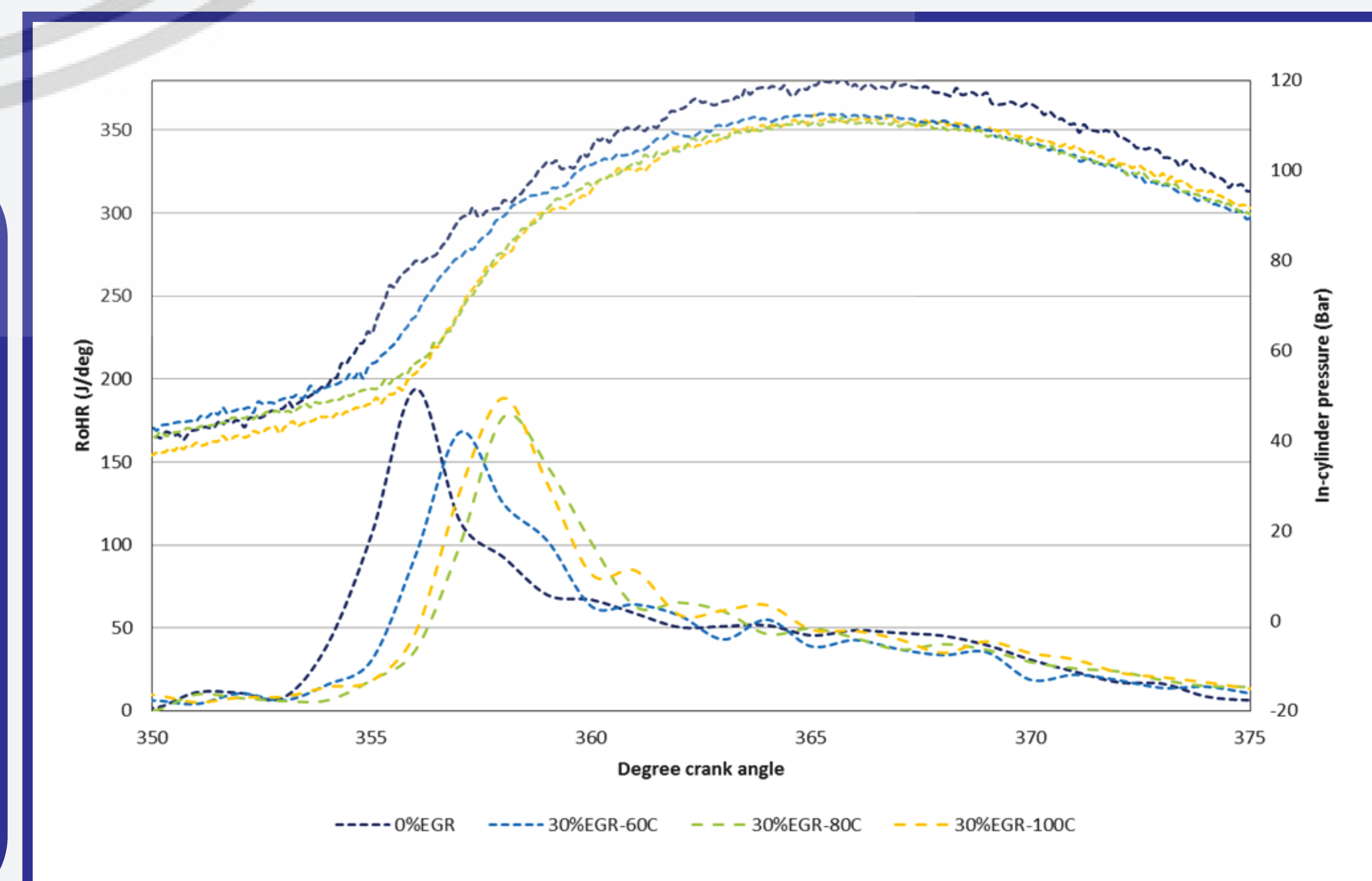
## Conclusions

The recirculate gas mainly effect on the combustion by its dilution effect. With the higher presence of the recirculate gas, the lower of fresh oxygen content is provided. The combustion was deteriorated, results in the lower brake thermal efficiency and the higher of CO, THC and soot. Its abatement of fresh nitrogen and oxygen provides the less formation of NOx with the lower combustion temperature.

The presence of ethanol in the combustion system leads to the cooler combustion temperature from its high heat of vaporization property. The higher level of CO and THC was provided through the incomplete thermal oxidation. Ethanol also results in higher NOx formation which also conforms to the higher premixed phase from more rapid energy released. Brake thermal efficiency is decreased in the higher ethanol presence.

Adding ethanol on EGR operation can slightly improve the brake thermal efficiency by absorbing heat from the recirculate gas for the vaporization. More uniform charge was provided through more complete combustion. With the higher ethanol presence on 30%EGR condition, the level of CO and THC are also suppressed. The different temperature of the recirculate gas shows significant effects on the combustion. The increase in EGR temperature lead to the higher brake thermal efficiency. The less heat energy wasted was generated from using the hotter recirculate gas. It also brings the higher formation of NOx with the decrease of CO, THC and soot. Sometimes this effect is dominated by the higher specific heat of the hotter recirculate gas increase of CO and THC and the lower formation of NOx in some case.

The combustion patterns show that the use of ethanol and recirculate gas retards the starting of the combustion. The higher temperature of the exhaust gas provides the slightly longer ignition delay from the higher specific heat of the hotter gas. But it sometimes gets dominated by the higher temperature environment inside the cylinder such as E20 100°C EGR case.



### Acknowledgement :

I would like to send all of my respect and gratitude to my advisor, Dr.Manida Tongroon, for encouraging me and bringing me to the right path of work, always provides a good advice to me in both work and life view and always backs me up all the time whenever I am depressed. I also would like to pay a grateful thank to my co-advisors, Asst.Prof.Dr.Chinda Chareonphonpanich and Dr.Preechar Karin from King Mongkut's Institute of Technology Ladkrabang and Prof.Hidenori Kosaka from Tokyo Institute of Technology for guidance and advice of this research. I would like to send a sincere gratitude to TAIST-Tokyo tech for providing the scholarship of this program and also Hino Motors for the daily expense scholarship through my first two years. I also would like to thank NSK, Japan for selected me to become an internship student during late 2016 which was my first time abroad and opened the world to me. Thanks to all of my colleagues, friends, trainees, technicians and lab staffs for helping me through the all related technical stuffs of this work. Finally, I would like to express my profound gratitude to my family and my girlfriend for supporting and giving an encouragement to me throughout these years of study and doing the research. They are also the reasons of my ability to fight through these years and also the reasons to never stop learning.

## **AUTHOR BIOGRAPHY**

**Author:** Mr.Kontorn Thammakul  
**Degree:** Master of Engineering  
**Date of Graduation:** 15th August 2018  
**Date of Birth:** 17<sup>th</sup> August 1992  
**Place of Birth:** Nakhon Rachasima, Thailand

### **Undergraduate and Graduate Education:**

Master of Engineering in Automotive Engineering,  
King Mongkut's Institute of Technology Ladkrabang, Bangkok, 2018.

Bachelor Degree of Engineering (Automotive Engineering),  
Suranaree University of Technology, Nakhon Rachasima, 2014.

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