В даний час основне джерело енергії сильно залежить від викопної енергії. Сучасна технологія транспортування також використовує викопні джерела енергії для запалювання двигунів автомобілів. Крім того, електрика, якою в даний час користуються мільярди людей, є результатом використання викопної енергії. Обмеження існуючих джерел викопної енергії і проблема глобального потепління змусили набагато розширити використання поновлюваних джерел енергії та енергозбереження для підтримки доступності енергії. Одним з альтернативних джерел енергії, який в даний час розробляється, є використання біоетанолу в якості суміші або заміни викопного палива. Використання біоетанолу (C_2H_5OH) в якості замінника суміші викопного палива впливає на ефективність двигуна, виробленої паливом. У цьому дослідженні розглядається вплив суміші біоетанолового бензину (RON 80) на одноциліндровий двигун з іскровим запалюванням (ІЗ) 125 куб.м., який виконується зі змінами в паливних сумішах (E0, E5, E10 і E15) з додаванням 0,5 об. % оксигенованого циклогексанолу, і це експериментальне випробування проводиться до 800 циклів для кожної паливної суміші з відкриттям дросельної заслінки на 100 % , і змінами швидкості обертання двигуна при 4000 об/хв до 8500 об/хв зі збільшенням частоти обертання двигуна кожні 500 об/хв. Робочі характеристики двигуна вимірюються шляхом підключення машини з динамометром, а зміна тиску згоряння в циліндрі вимірюється датчиком тиску. Очікується, що результати випробувань доведуть, що суміш палива з оксигенованим циклогексанолом може знизити COVIMEP в варіаціях циклу до циклу (E10 ++, що становить 4,24 %), так що коливання крутного моменту не відбуваються, що призводить до надійної роботи двигуна або підвищення керованості транспортного засобу, крім того, як продуктивність так і потужність крутного моменту стають кращими

Ключові слова: біоетанол, оксигенований, тиск в циліндрі, коефіцієнт варіації, варіація від циклу до циклу, потужність, крутний момент

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THE EFFECT OF ADDITIVE ON COMBUSTION CHARACTERISTICS AND CYCLE TO CYCLE VARIATIONS ON SI ENGINE FUELED BY GASOLINE AND BIOETHANOL

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1. Introduction

Research and development of spark ignition engines are currently more focused on improving engine performance and reducing exhaust emissions. Those are important to find the substitution or at least additional fuel that can reduce the problems caused by the continuous fossil fuels used [1]. Bioethanol (C_2H_5OH) is a renewable fuel source derived from biomass derivative product from the plant's fermentation containing starch. Bioethanol has a simple molecular structure easily defined chemical and physical properties. Bioethanol can be used as fuel either directly or as a mixture of other fuel, such as gasoline.

To apply bioethanol as a replacement or mixture fuel for the engine, bioethanol must have a high content at least 99 % (anhydrous ethanol). If used entirely as fuel, engine modifications are necessary, but if mixed with gasoline, engine modifications are not necessary. Anhydrous ethanol is used because the water content is very little and can even be said to be pure so that when mixed directly with gasoline, it is possible to get a homogeneous mixture and can directly enter the combustion chamber. While hydrous ethanol with a low concentration and water content in it, it can't be directly mixed with gasoline. Usually, for this hydrous ethanol, the water content used is around 4.9–5 % while to be used as a mixture with gasoline the water content is a maximum of 7.4 %. Therefore simple technology is necessary that can accommodate low-grade bioethanol produced by the community to be converted into high-grade bioethanol, and the results can be directly applied as a mixture of fuel in the engine.

2. Literature review and problem statement

Paper [2] experimented on a 250 cc one cylinder engine equipped with ethanol direct injection (EDI) and gasoline port injection (GPI). Experimental results showed less effective on LEDI (late ethanol direct injection) because heat transfer increased from the cylinder wall. The quality of the mixture could deteriorate under LEDI conditions which resulted in low engine efficiency and high emissions. Volumetric efficiency increased and the duration of combustion decreased at EEDI (early ethanol direct injection). The combined effect of increasing volumetric efficiency, reducing the duration of combustion and ignition timing were good enough to increase the thermal efficiency of the engine at EEDI. The maximum Lambda achieved in EEDI conditions was 1.29 when the ethanol energy ratio was 24 %, and SOI was 2,900 CAD BTDC. The LEDI was only slightly increased than the stoichiometric air-fuel ratio. In EEDI conditions, IMEP was greater, and combustion stability (COV) was better than LEDI. Emissions under EEDI conditions were also lower than LEDI conditions.

Article [3] experimented to carry out on 2 single cylinder engines 4-stroke volume 125 cc, each with a carburetor and engine with fuel injection. The used fuel was E15 and G95 commercial gasoline. The experimental results showed that a mixture of ethanol and gasoline with 15 % ethanol could be used on the engine with carburetors and fuel injection without adjusting the engine. Reduced emissions were CO, alkane, alkene, and aromatic groups, compared to unleaded gasoline on the market both in the carburetor and in the fuel injection. But acetaldehyde emissions increased sharply. Ozone-formation in the exhaust also decreased. In general, the variation of emissions from the engine with fuel injection was lower than the engine with the carburetor.

In study [4] conducted an experimental method by utilizing the effects of direct injection which carried out in 2 stages; the performance of the machine was checked at high compression ratio and constant speed. The first injection at the suction step and the second injection at the end of the compression step used various injection ratios (gasoline, E10, E20, M10, and M20). The results showed that the first injection timing has a significant effect on the gas pressure on the cylinder and heat release to fuel gasoline-ethanol but both effects were reduced for gasoline fuel only. Meanwhile, the second stage injection also produced a significant effect on combustion and performance compared to the first injection, even though the injection ratio was changed. Maximum cylinder gas pressure which showed effective IMEP and thermal efficiency was able to be controlled by using the second injection timing. The E10 and M20 ignition timings were occured earlier than gasoline. Increased ethanol levels reduced Pmax compared to an increase in methanol levels which increased Pmax.

In research [5] experimented on a 4-stroke single cylinder engine Yamaha YBR250. Direct injection were used for ethanol , but port injection were used for gasoline. Port injection pressure was around 250 kPa, while direct injection pressure was around 3–13 Mpa. Tests were carried out at light loads and heavy loads, with engine speed variations at 3,500 rpm to 5,000 rpm with interval 500 rpm. Test results showed that the effects of ethanol used could increase BMEP, volumetric efficiency, thermal efficiency and reduce NO, by increasing the ethanol energy ratio (EER). Another effect of including ethanol, it could improve engine performance with the cooling effect where ethanol was injected directly into the combustion chamber, the stoichiometric LHV mixture per unit air mass increased with EER, and also the combustion speed increased.

Paper [6] conducted tests to perform on a single cylinder engine, at high speed with a full load and half load at 6,500 rpm and 8,500 rpm with pure gasoline fuel, and 30–35 % mixture of butanol and gasoline. This experiment also aimed to find the effect of combustion with variations in ignition timing, the ratio of butanol mixture and engine load. Test results showed that the butanol gasoline mixture produces high knocking resistance by advancing ignition timing on the engine SI so that combustion was more efficient. With the butanol ratio increased, the combustion process was perfect. With the engine load increased, heat release was faster and closer to the maximum both for pure gasoline or for a mixture of butanol and gasoline. Power engine, torque, specific brake energy consumption, HC, CO, and O_2 were better than using pure gasoline. But NO_x and CO_2 were higher than pure gasoline fuel.

In article [7] conducted a test with the aim to reveal a quantitative analysis of exhaust gas emissions and engine performance with a volume of 0–40 % hydrous ethanol as fuel, used as a substitute for the anhydrous ethanol-gasoline mixture. Tests were carried out on a single cylinder engine with a ratio of air/fuel mixtures varying from 0.9 to 1.1. The results showed that high pressure and lower temperatures were obtained with a hydrated ethanol mixture. Also, the exhaust gas heat level, combustion efficiency, and thermal combustion efficiency were not affected by water content. The practical consequences of burning fuel hydro were reduced, nitrogen oxide emissions appear to be reduced, and the water content in fuel was increased.

Paper [8] conducted a comparative experiment, carried out on gasoline engine port injection with hydrous ethanol gasoline (E10W) fuel, ethanol gasoline (E10) and pure gasoline (E0). The effect of engine load and the addition of ethanol and air on combustion and emission characteristics were analyzed in depth. According to the experimental results, compared to E0, E10W shows higher results for pressure in the cylinder at high loads. The use of E10W increased NO_x emissions at high loads. However, at low loads, HC, CO, and CO_2 conditions were significantly reduced. The E10W also produced less HC and CO, while CO_2 emissions were not significantly affected in higher operations. Compared to E10, E10W showed higher results in-cylinder pressure and heat release rates in the tested operating conditions. Also, a decrease in NO_x emissions was observed for E10W from 5nm to 100nm, while HC, CO, and CO_2 were slightly higher in low and medium load conditions. From the results, it could be concluded that the E10W fuel was able to be considered as a potential alternative fuel which was applied to the gasoline engine.

In study [9] conducted a simulation to examine heat release cycle-to-cycle variation (CCV) on SI machines fueled by the gasoline-ethanol mixture. The mixture ratio was changed from 0.7–0.9 and 1.0 from the thin mixture to the stoichiometric mixture. Ethanol was added proportionally 5–25 %. This simulation was done to calculate the coefficient of variation (COV) of heat release at each additional volume of ethanol. From COV values, they found that at the fixed mixture ratio CCV heat release decreased at each addition of ethanol. They also found that in mixtures with a fixed ethanol volume, CCV increased with a thinner mixture, using continuous wavelet transform to analyze heat release. The results based on COV showed a fixed mixture ratio, CCV decreased significantly according to the increase in ethanol, when the mixture was thin. When the mixture approached stoichiometry, CCV decreased not significant to change in ethanol levels. Also, COV results showed that in fixed ethanol levels, CCV decreased according to the increase in the mixture ratio, and this increase occured specifically in the thin mixture. The results of the wavelet analysis showed that the CCV heat release on the SI engine fueled a highly dynamic gasoline-ethanol mixture consisting of intermittent high-frequency fluctuations and low-frequency oscillations. For gasoline engines (without the addition of ethanol), CCV decreased according to changes in the composition of the mixture from the thin mixture to the stoichiometric mixture. Also, at a fixed mixture ratio, CCV could be reduced by mixing gasoline with ethanol. The addition of ethanol to gasoline was expected to improve engine performance regarding IMEP, COV_{IMEP}, and emissions. For E50, a better performance mixed would be obtained when the initial evaporation during injection and spray were compact with higher penetration especially at 100 bars, which avoided the formation of liquid deposits on the piston and pollutants in the exhaust. Meanwhile, for E85, the characteristics of spray evaporation were slower because they were less homogeneous and the reduction of flame propagation in the final phase of propagation that allowed faster and more efficient combustion of higher gasoline content.

Paper [10] conducted experiments to minimize cyclic variations on the SI engine, by controlling the spark timing for the entire cycle in a row. A stochastic model was performed between spark timing and maximum cylinder pressure using system identification techniques. The maximum cylinder pressure from the next cycle is estimated with this model. Control algorithms generated from LabView and installed into the Field Programmable Gate Array chassis. The test results, the maximum cylinder pressure of the next cycle could be predicted quite well, and the spark timing could be adjusted to maintain the maximum cylinder pressure desired to reduce cyclic variations. In the fixed spark timing experiments, COVPmax and COVimep were 3.764 and 0.677 %, while the results decreased to 3.208 and 0.533 % when the GMV controller was applied.

According to Heywood [11], cyclic variations can be identified with parameters in the four main categories, namely the pressure, combustion parameters, related to the front flame and exhaust gas. The corresponding pressure parameters are the maximum cylinder pressure (Pmax), the crank angle where the maximum pressure of the cylinder occurs (qPmax), the maximum pressure rate rises (dP/dq) max, the crank angle where the maximum level of pressure increase occurs q (dP/dq) max, shows the average effective pressure (IMEP) of the individual cycle. The parameters are related to combustion about heat release, burning mass fraction and duration of combustion characteristics. The parameters are related to front flame about the formation, development, and velocity of the flame. The last category is in the subject of exhaust gas concentration in the muffler.

Article [12] tested a 4-stroke single cylinder engine; the M380 MINSEL engine was cooled and coupled to an asynchronous engine with constant engine speed. This engine was originally designed to be a compression ignition engine, with a flat head cylinder and piston bowl combustion chamber. Some changes were made to turn it into an SI engine. The original injector was replaced with a spark plug and modification on the piston was done to change the combustion chamber and to reduce the compression ratio. Cylinder specifications were 80 mm bore, 75 mm stroke, and compression ratio 11.4. Experiments had been carried out in two different stages. In the first stage, the engine rotation speed was maintained at 1,500 rpm, and the intake pressure was set at 0.5 bar, the spark ignition angle was set to get the maximum brake torque for each test, and the fuel/air ratio varied from 0.63 to 1.0. In the second stage of the experiment, the intake pressure was about 0.5 bars and the fuel/air ratio varied from 0.7 to 1.0, while the engine rotation speed was modified from 1,000 rpm to 2,500 rpm, to determine the effect of speed engine rotation on cyclic dispersion. In the experimental setup, the pressure inside the cylinder was measured using a piezo-electric AVL GU21D sensor (maximum calibration error of 0.06 %). This sensor was connected to KISTLER Chargers 5018A1000 (maximum calibration error 0.3 %). The charge amplifier output signal was recorded on a Yokogawa DL750 Scopecorder (16 bit AD converter). The estimated pressure gain was 0.36 % of the measurement range). The angle of the crankshaft measured using the free end AVL 360C.03 encoder angle. Encoder had 600 marks per revolution: I resolution of 0.6 degrees and also single pulses per revolution signal. The fuel was being used during the experiment was natural gas (NG). The NG inlet mixture and air were made using two BROOKS thermal mass flow controllers. This controller was equipped with a proportional valve and actuator. Therefore, mass flow rates could be measured and controlled at once. The 5853S model was used for air, and the 5851S model was used for NG. The mixture of NG and air were formed in the inlet manifold, so the level was premixed high and almost constant.

In paper [13] tested the properties of fuel from various gasoline-ethanol mixtures. In this study, nine types of ethanol fuel mixtures had been prepared based on the variable mixing ratio between 87.5-octane gasoline and 99.5 % ethanol quality by volume. The main components of the experimental setup consisted of SI gasoline engine, engine control unit (ECU), engine performance testing unit, air flow meter, air-fuel ratio analysis, fuel weight scale, and temperature measuring unit. The experiment was carried out on a Toyota 3ZZ-FE DOHC, 16-valve, 1.6 L (1,598 cc) engine, 100 PS Nishishiba NEDD-130H spark ignition engine. Eddy dynamometer was used to measure torque and engine power. Cussun P7204 with a maximum engine driving capacity of 50 L/s was used to measure air consumption. Horiba MEXA-730λ was used as an air-fuel ratio analysis by installing a lambda sensor (oxygen sensor) in the exhaust manifold. Mettler Toledo-ML4001 with a maximum capacity of 4,200 g and 0.01 g legibility was used as a balance scale for measuring fuel quantity. To control fuel injection and ignition systems as desired, the original ECU was modified to standalone type, which allowed fuel injection timing and ignition timing (CA, BTDC) to be adjusted with maximum torque brake. The temperature sensor unit was used to monitor the temperature of the exhaust gas, air intake, fuel, lubricants, and coolants. This study was to test the optimal level of combination of ethanol-gasoline fuel to maximize the efficiency of commercial SI engine thermal brakes. A comparison between the different levels of ethanol-gasoline fuel combination and fuel with 10 % ethanol with volume (E10) was investigated with the wide open throttle (% WOT), engine speed, and a combination of ethanol-gasoline fuel rate. The results of the experiment showed that the right ethanol-gasoline mixing ratio could improve engine torque output performance, especially at low engine speeds. The E40 and E50 fuels provided maximum thermal brake efficiency at 58–73 % WOT and 2,000–2,500 rpm. The E20-E40 fuel provided the highest MBT at WOT 70–100 % and

1,000–4,000 rpm. For applications, this study provided the possibility of using multi-blend rates of ethanol fuel equipped with real-time adaptive controllers in sequence to obtain optimal thermal brake efficiency of all times at a certain engine speed and the percentage of throttle intake opening. While the engine was assembled with E0 and E100 fuel tanks were running at 2,000–2,500 rpm and 58–73 % WOT, adaptive fuel. The controller might be integrated with E0 and E100 to E40 or E50 so that maximum thermal brake efficiency was produced.

Paper [14] conducted the test by using distilled bioethanol low-grade autonomously that utilized the exhaust heat on compact destilator to produce high-grade bioethanol which was ready to use as fuel mixture. From the test it was obtained that wheel power and wheel torque generated from a mixture of gasoline and bioethanol had a higher value than pure gasoline as fuel. A mixture of gasoline and bioethanol was able to increase the power by 15 %. While the torque value was generated at a mixture of E5, E10, and E 15 respectively amounted to 6.92 Nm, 6.64 Nm, and 6.92 Nm, where the value was higher than pure gasoline by 6.1 Nm. The torque value was generated at a mixture of E5, E10, and E 15 with additive oxygenated respectively amounted to 7.5 Nm, 7.6 Nm, and 7.53 Nm [15].

Thus, the use of ethanol as a mixture of fuel in gasoline is very influential on COV, which in turn is very influential on engine performance (torque), but which mixture can reduce the problem. Apart from that, how about the combustion characteristics and the cycle to cycle variations with the addition of oxygenated additives to the fuel mixture.

3. The aim and objectives of research

The aim of research is reducing the cyclic variation on SI 125 cc engine fueled by a mixture of gasoline bioethanol (E0, E5, E10, and E15), and in the mixture with the addition of cyclohexanol additive. An impact on both power and torque performance is also tested.

To achieve the set aim, the following objectives are accomplished:

– determine the combustion characteristics of each mixture by comparing combustion pressure to the crank shaft rotation at each variation of engine speed, and calculates the cyclic variation of the deviation that occurs in the combustion pressure value of the average value;

– the engine used is equipped with an ECU which can automatically adjust the ignition timing according to the change in oxygen content of the exhaust gas sent by the O_2 sensor. This variation of combustion pressure can affect the power and torque fluctuations that are produced, for this purpose a measurement of power and torque of the engine with engine dynamometer;

– the mixture of gasoline and ethanol used is only up to 15 % so that engine modifications are not needed so that the fuel mixture can be used directly. Mixing gasoline, ethanol, and oxygenated additive fuels according to the desired mixture in the fuel tank;

– determine the properties and characteristics of gasoline, ethanol, and oxygenat (RON, oxygen content, vapor pressure, and specific gravity at 15 °C) other than those already known from the reference.

4. Experimental setup

4. 1. Materials

The engine used in this study was the SI Honda type AFX12U21C07 single cylinder 125 cc SOHC with electronically controlled fuel injection system. General specifications of the test engine as in Table 1.

Table 1

Test Engine Specifications

General Specifications	Parameter
Engine Type	4 stroke, SOHC, single cylinder
Displacement	125 cc
Bore × stroke	52.4 mm × 57.9 mm
Compression ratio	9.3:1
Max Output	7.4 kW/8,000 rpm
Max Torque	9.3 Nm/4,000 rpm
Fuel System	Fuel Injection (PGM-FI)
Lubricant Capacity	0.7 L at periodic maintenance
Clutch type	Multiple wet Clutch with Coil Spring
Transmission type	4 Speed Manual, Rotary
Starter type	Electrical and Kick Starter

The fuel used is 7 types of the gasoline-bioethanol mixture prepared based on variable mixing ratio from RON 88 to RON 96, ethanol quality by volume, with a mixture of E5, E10, and E15, as well as the addition of cyclohexanol ($C_6H_{12}O$) additive with a composition of 0.5 % on each fuel mixture. The mixture is formed in the fuel tank and inlet manifold. So the level premix is quite high and almost constant. Therefore, mass flow rates can be measured and controlled at once. Testing the properties of fuel from various gasoline-bioethanol mixtures is carried out. Characteristics of gasoline and bioethanol are shown in Table 2.

Table 2

Characteristics of Gasoline and Ethanol

Property	Ethanol	Gasoline
Chemical formula	C_2H_5OH	C_5–C_{11}
Relative molecular mass	46	95–120
Density (kg/L)	0.79	0.700–0.750
Boiling point (°C)	78.4	25–215
Flash point (°C)	13	–40
Latent heat of vaporation (kJ/kg)	840	373
Stoichiometric heat of vaporation (kJ/kg)	93.9	25.8
Stoichiometric air-fuel ratio	8.95	14.7
Auto-ignition temperature (°C)	363	300–400
LCV (MJ/kg)	26.9	42.9
LCV (MJ/L)	21.3	31.9
Lower heating value (kJ/kg)	27,000	44,000
Mixture heating value (kJ/m^3)	3593	3750
RON	108	88
Laminar flame speed (m/s)	0.5	0.38

To complement the properties of the fuel, each fuel mixture is tested for its characteristics. The results of testing various fuels are as shown in Table 3.

Cylinder combustion pressure is measured using a Kistler 6617B piezo-electric sensor (maximum pressure up to

200 bars) and recorded by the LabVIEW acquisition system. The crank position angle (up to 720 crank angles) is acquired with the shaft encoder; the sequence is adjusted to synchronize the cylinder combustion pressure signal with the crankshaft angle. The temperature sensor unit with the K type thermocouple is used to monitor the temperature of the exhaust gas, fuel, lubricant, and spark plug. This machine is connected to the engine dynamometer for power, torque and fuel consumption analysis, and is connected to the QROTECH-401 (4/5 gas analyzer) to measure the content in exhaust gases such as Carbon Monoxide (CO), Hydrocarbons (HC), and Nitrogen Oxides (NO_x). Analysis of the air-fuel ratio is done by installing a lambda sensor (oxygen sensor) in the exhaust manifold.

Table 3

Fuel Characteristics Test

Parameter	E0	E5	E10	E15	Method
RON	87.9	90.5	93.6	96.5	ASTM D 2699
Oxygen Content (% m/m)	0	1.8	4	5.9	ASTM D 4815
Vapor Pressure (kPa)	48.6	38.8	68.7	65.8	ASTM D 323
Specific Gravity on 15 °C (kg/m^3)	718.4	728.1	745.8	748.7	ASTM D 4052

The following is an experimental set-up chart on a 125 cc SI engine connected to other supporting components (Fig. 1). One of them is a pressure transducer that is directly attached to the spark plug.

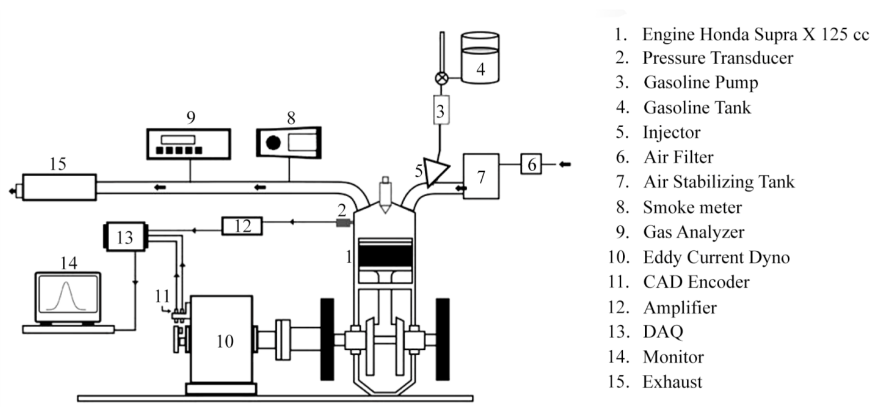

Fig. 1. Experimental SI Set-up Engines

4. 2. Methods using in investigation

In the operation of the Spark Ignition engine, the peak pressure of the combustion varies greatly depending on the operating conditions. Variations in-cylinder combustion pressure occur from one cycle to the next, called cyclic variations, while the using of leaner mixtures and exhaust gas recirculation, and increasing operation occur due to various conditions in idling stop system. The cyclic variations in the Spark Ignition engine are identified as very fundamental combustion condition. They are able to cause torque fluctuations, which are ultimately resulted in poor engine operation. By reducing cyclic variations, engine output can increase up to 10 % in the same fuel consumption conditions, and can reduce unusually noisy engine and vibration.

The combustion pressure in the cylinder is an important indicator of cyclic variations, which are measured in each cycle at each rotation angle of the crankshaft. Some important parameters are related to the pressure in the cylinder namely; the peak pressure in the cylinder (Pmax), the crankshaft angle where the peak pressure occurs (CA Pmax) and the Indicated Mean Effective Pressure (IMEP) in one cycle. Engine performance is power and torque depending on IMEP, and variations in IMEP cause torque fluctuations.

Pmax is a measure of the pressure levels that increases due to combustion. If burning is faster, a higher rate of increase in pressure will occur. Pmax is shown depending on changes in the combustion phase and the level of combustion. The amount of variation depends on whether the combustion is faster or slower. Faster combustion will be resulted in a higher Pmax. Pmax will tend to occur closer to the Top Dead Center (TDC), while the slower combustion cycle will have a lower Pmax and the PMax CA will move away from TDC.

The variation coefficient of effective mean pressure (COV_{IMEP}) is widely used to evaluate cycle to cycle variations (CCV). COV_{IMEP} is the most commonly accepted parameter for analyzing CCV. COV_{IMEP} is defined as follows:

$$COVimep = \frac{\sigma imep}{imep} \times 100. \tag{1}$$

The Indicated Mean Effective Pressure (IMEP) is easily calculated and describes the size of the work results of the engine cycle. IMEP is defined as follows:

$$imep = \frac{Wc}{Vd}, \tag{2}$$

where V_d is the volume of the machining step and W_c is the work generated from each machine cycle, which is defined as follows:

$$Wc = \oint PdV. \tag{3}$$

This experimental test is carried out as much as 800 cycles for each mixture of gasoline and bioethanol fuel, after running the engine until it reached a steady state, where the oil and cooling water temperatures were at 50 °C. The throttle valve opening is maintained at 100 %, and the ignition timing is controlled according to the ignition system in the fuel injection control. As for engine speed variations at 4,000 rpm up to 8,500 rpm with engine speed increases every 500 rpm. This engine speed variation is seeing conditions from low, medium, to high speed.

5. Results

Fig. 2 below is a variation of the combustion cycle that occurs in different cycles for each mixture of fuels. Combustion pressure is measured at each change in the crank angle so the variation will be very clear.

The mixtures of fuels tested were E0, E5, E10, and E15 displaying COV at different ratios of 7.05 %, 4.90 %, 7.02 %, and 5.96 %, respectively, at 6,000 rpm. E5 shows the best results for COV_{IMEP}. As for adding oxygenated cyclohexanol additives in each mixture of fuels, COV_{IMEP} for E5++, E10++, and E15++ respectively were 4.67 %, 4.24 %, and 5.56 %. The addition of cyclohexanol at E10 shows that COV_{IMEP} is the most optimal at 4.24 %, which previously was 7.02 %.

Variations in the measured cylinder pressure are shown in Fig. 3. This measurement result is obtained from the average cylinder pressure every variation of the gasoline-bioethanol fuel mixture and is carried out from 4,000 rpm to 8,500 rpm, with a total of 800 cycles. From the picture can be seen in a certain cycle the peak pressure in the cylinder (Pmax) is higher than the other cycles.

As shown in Fig. 3 above, the fuel mixture E10++ at 6,000 rpm produces a cylinder max pressure of 49.86 bar. This mixture is the most optimum mixture compared to mixtures of E5, E5++, E10, E15, and E15++, in fact the mixture is E0 (pure gasoline). Then in Fig. 4 below, at 8,500 rpm, the cylinder max pressure is 46.84 bar produced from E5 mixture. This mixture is the most optimum mixture compared to mixtures of E5++, E10, E10++, E15, and E15++, even from a mixture of E0.

In the second part, the Otto engine performance analysis in this study includes torque and power. The test results in the form of a ratio of torque and power value on the variation of engine speed. For each tested fuel variation and the results in this study are presented in the form graphs that aim to facilitate data analysis. In this test, the amount of torque produced by each variable can be seen in Fig. 5.

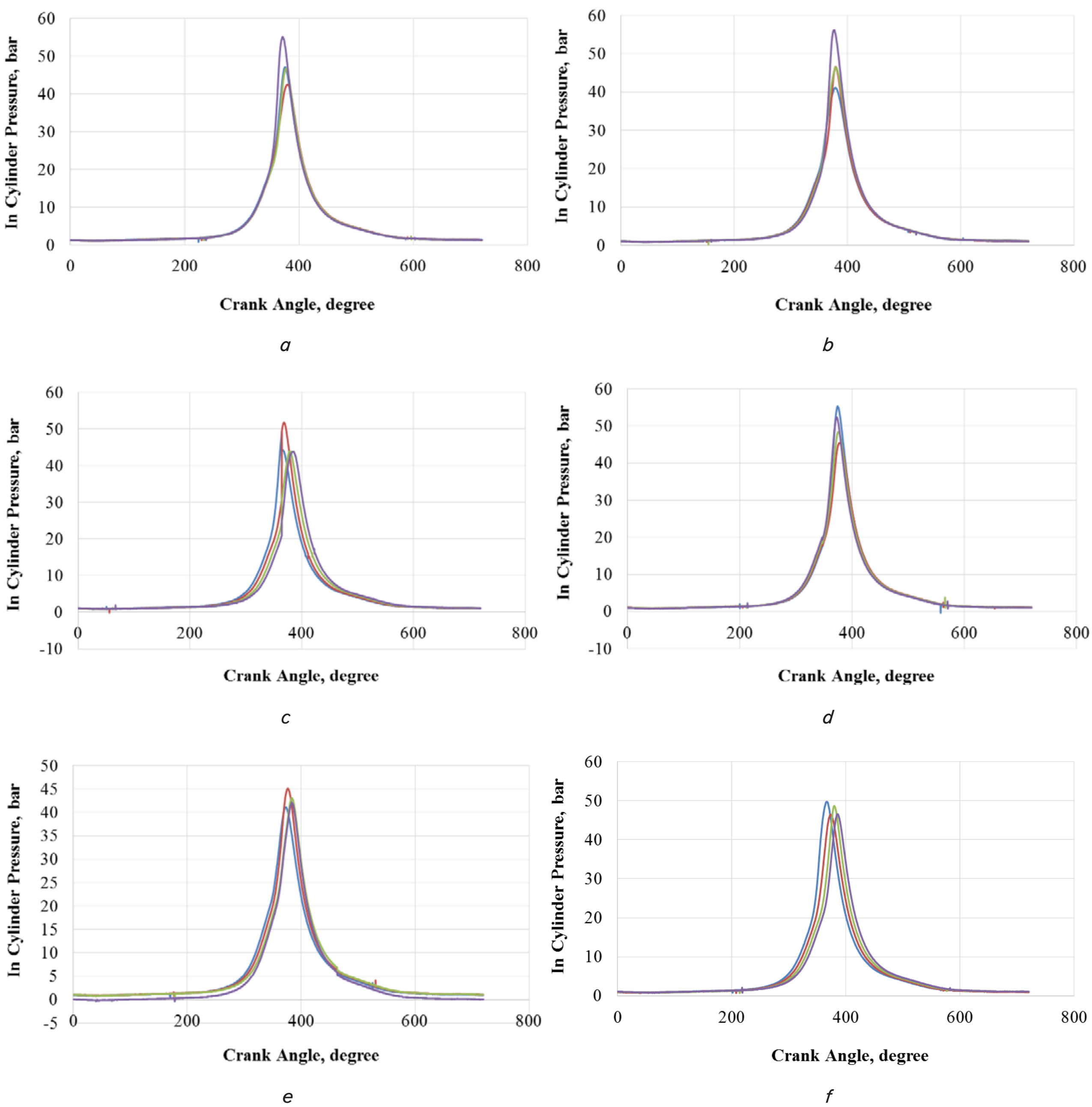

Fig. 2. In-Cylinder Pressure VS Crank Angle on Fuel Mixture Variations at 6,000 rpm:
a – E5; *b* – E5++; *c* – E10; *d* – E10++; *e* –E15; *f* – E15++

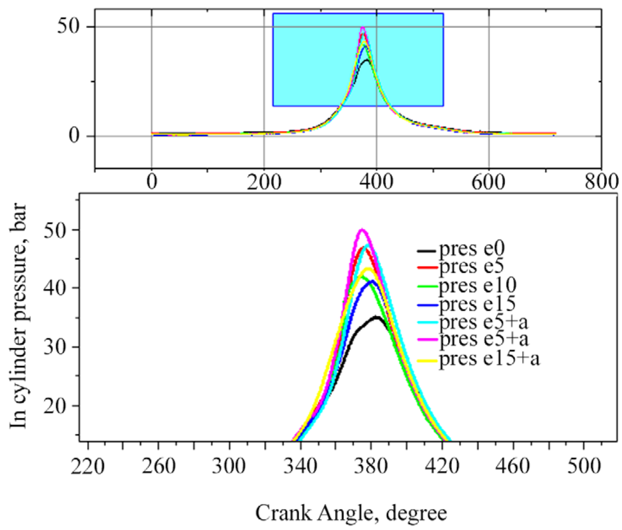

Fig. 3. In-Cylinder Pressure VS Crank Angle on Fuel Mixture Variations at 6,000 rpm

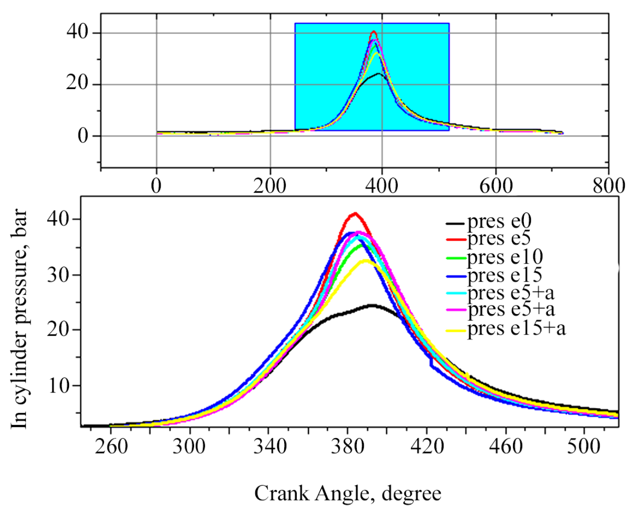

Fig. 4. In-Cylinder Pressure VS Crank Angle on Fuel Mixture Variations at 8,500 rpm

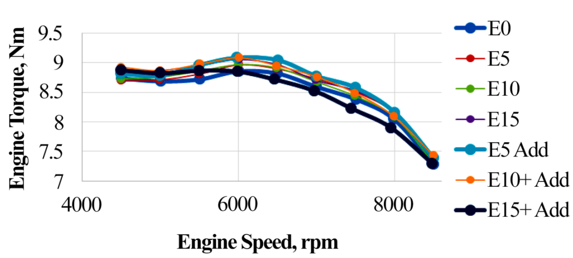

Fig. 5. Torque VS Engine Speed

Based on the results of research that has been done using a mixture of gasoline with bioethanol, the torque value has increased along with the increase in the percentage of ethanol mixed. Addition of additives also results in higher maximum torque when compared to pure gasoline (E0) or with the addition of ethanol by 5 % to 15 % without the use of additives.

It can be seen in the Fig. 6 that by adding ethanol to the fuel, the torque produced will also be higher. The most optimal fuel mixture in producing torque is in the fuel mixture E10 which is added with an oxygenated cyclohexanol additive of 5 ml of fuel per liter; this is due to the advantages of additives which can also increase the latent heat of evaporation and as an anti-knock performance. The torque value produced in this optimum mixture is 9.09 Nm. While the lowest torque is produced by pure gasoline (E0) of 8.86 Nm, it can be seen that the resulting torque increase is 2.6 %. But with the use of additives, along with the addition of ethanol, the maximum torque will decrease.

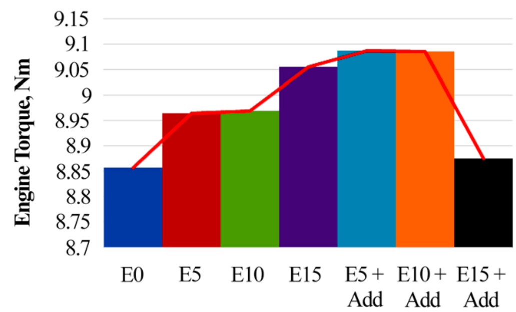

Fig. 6. Maximum Torques of 7 Fuel Variations

Fig. 7 shows a graph of the results of the power testing on the rotation variation from 4,500 rpm to 8,500 rpm for various fuel variables tested. The test results on each fuel variation have a similar trend line. It appears that maximum power occurs at 8,000 rpm engine speed. The maximum power that can be produced when testing using fuel mixture E15 is 6.84 kW.

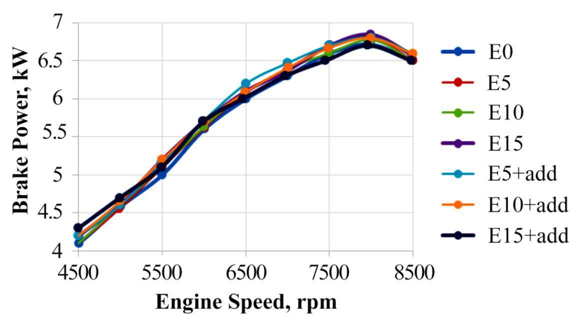

Fig. 7. Power VS Engine Speed

Fig. 8 shows that the most effective brake power is generated by E15 with the resulting value of 6.84 kW, an increase of 1.94 % of the brake power produced by pure gasoline (E0) this is due to the addition of more oxygen (34.7 % by weight), and the smallest brake power is found in the variation of E15 fuel which is added with additives, the decrease that occurs in 0.22 % of pure gasoline.

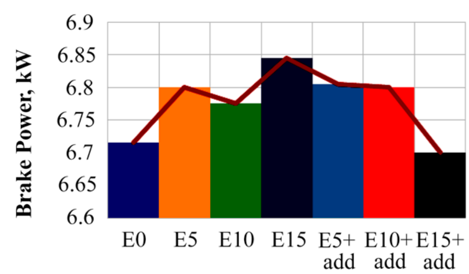

Fig. 8. Maximum Powers of 7 Fuel Variations

6. Discussion of experimental results

COV_{IMEP} changes occur after the mixture of gasoline-bioethanol ratio exceeds 5 vol %, COV_{IMEP} increases, this indicates a tendency to increase the intake manifold temperature and decrease volumetric efficiency. However, there is no problem when the engine is operated at 15 vol % ethanol ratio. COV_{IMEP} in the E10++ mixture at 6,000 rpm is the most optimum, on the other hand at the same rpm, this mixture also produces the most maximal cylinder pressure, it shows that the addition of cyclohexanol additives greatly affects the combustion process which occurs including the use advantage ethanol, in addition to influencing the intake manifold temperature which causes volumetric efficiency to increase, additives also have a higher heating value than ethanol, and ultimately are resulted in a higher peak combustion pressure in the cylinder.

The varying cylinder pressure has a correlation with the torque, which is directly related to the comfort of the vehicle. Reducing cyclic variations in combustion and in setting pressure in a cylinder to a certain extent can help the level of fuel use and emissions. If variations in cyclic combustion can be controlled so that all combustion in the cycle becomes good, fuel savings may be felt. Another very important benefit that comes from cyclic variation control is that the engine can be improved in stability, especially with automatic transmission or manual transmission, which dampens variations in engine torque. Machine roughness may also be reduced.

Engine performance can be known from the power value or commonly called Brake Horse Power (BHP). BHP is a parameter that indicates the performance of an engine obtained from the crankshaft. The term brake referred to the load applied to the engine and held it at a certain rpm. During testing, the output torque and rotational speed are measured to determine BHP. The power obtained at BHP measurements is higher than the power obtained on the wheels. BHP itself provides an actual engine power picture before losing power through components such as gearboxes and so on.

The beneficial effect of ethanol fuel is that the hydrogen bridge formation contains more oxygen (34.7 % by weight) which allows complete combustion. Besides that the ethanol density (0.79 kg/L) is higher than the gasoline (0.73 kg/L) which causes the same volume of fuel to be higher, and finally the latent heat of vaporation (840 kJ/L) is higher than the gasoline (373 kJ/L), thus providing a lower temperature intake for the intake manifold and increasing its volumetric efficiency, this results in an increase in BHP. At higher engine speed combustion that occurs faster along with the speed of ignition time, the available time is short enough to complete combustion in one cycle, where in this case the speed of fire propagation is very necessary. Ethanol has a higher laminar flame speed (0.5 m/s) so that the flame propagation speed also increases, and it is also possible to reduce engine knock, besides the result of high octane values on ethanol.

In E15 fuel with additives having the highest octane value, this results in longer ignition timing. Because of the long ignition timing, the engine will lose the power that can be produced. Also, the addition of additives to the E15 mixture increases oxygen content so that the air-fuel ratio conditions become leaner.

Adding oxygenated cyclohexanol additives to the fuel mixture will increase the oxygen content of the mixture; this makes the air and fuel ratio in lean conditions or lean mixture. Lean combustion produces less power because of fewer fuel enters than air. The ratio between fuel and air that is not optimal will reduce the BHP engine.

Increased torque due to increased ethanol content, this is due to the latent heat of evaporating ethanol which is higher than gasoline to also provide a lower temperature intake in the intake manifold and provide a positive impact on increasing engine power because it increases volumetric efficiency, the incoming ethanol becomes more. Also, the propagation velocity of the flame is also very necessary, which is influenced by several parameters such as air condition, fuel mixture, temperature and pressure of the fuel mixture.

In this paper, the discussion is only about the cyclic variation of combustion pressure due to the use of varied fuels and their effect on the power and torque produced. This paper can be an additional recommendation material when you want to use bioethanol as an alternative fuel to reduce the use of fossil fuels.

This study uses a 125 cc gasoline injection engine that is widely used in Indonesia, besides using bioethanol as well as cyclohexanol oxygenate, the current research only uses laboratory scale engines, diesel or carburetor gasoline engines. Difficulties faced because the dimensions of the combustion chamber are very small, so the input of bioethanol cannot be done directly into the combustion chamber, and high ethanol concentration is relatively more expensive.

7. Conclusion

1. One important measure of cyclic variability is measuring the COV indicated mean effective pressure. The lowest COV_{IMEP} produced from 800 cycles is in the fuel mixture E10++ which is 4.24 %, it defines the cyclic variability in indicated work per cycle, and it has have found that fluctuating torque (vehicle driveability) problems decrease because COV_{IMEP} is less than about 10 %.

2. Addition of bioethanol and oxygenated cyclohexanol generally can improve the performance (torque and power) produced by the fuel engine. Torque and brake power increase after engine speed above 5,000 rpm. The highest torque value is obtained from the variation of fuel E10++ of 9.09 Nm at engine speed 6,000 rpm, 2.6 % higher than pure gasoline fuel (E0). The most optimum power (brake power) is produced by the E15 variable of 6.84 kW at 8,000 rpm engine speed which increases 1.94 % from E0.

3. Fuel grade ethanol allows a more homogeneous mixture of fuels so that during the fuel testing process it does not require a mixer for mixing fuel.

4. Addition of ethanol to gasoline resulted in increased RON value, oxygen content, vapor pressure, and specific gravity.

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