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THE DEVELOPMENT OF COMBUSTION STRATEGY IN IMPROVING THE PERFORMANCES OF SI-PFI ENGINE USING E50 OF GASOLINE-BIOETHANOL FUEL BLEND

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Abstract

This research aims to improve the combustion performance of gasoline-bioethanol fuel blended in the ratio of 50:50 (E50) on the spark-ignition engine by employing a new combustion strategy. The Box Behnken Design of Response Surface Methodology and Non-Linear Programming was employed to optimize the performance of the engine and create some engine parameters. The performance of the engine consists of power, torque, thermal efficiency, fuel consumption, and the emission of CO and HC, while the engine and combustion parameters are compression ratio, ignition timing, and engine speed. A new combustion strategy will be applied in this study with a tiered mapping process for each engine parameter based on the MBT. The brake torque increased by 13.5 % while HC and CO emissions decreased by 15 % and 71 % respectively when the combustion strategy applied if compared o the pure gasoline in engine standard condition. Furthermore, the BSFC increased by 33 % while BTE decreased by 15 % towards the gasoline fuel. The non-linear programming applied in this study intended to figure out the best combination of the engine parameters in obtaining optimum engine performances. In the RSM analysis, the codes –1, 0, 1 represented 12, 12.5, and 13 of compression ratio, 16, 20, and 24 BTDC of ignition timing and 2000, 5000, and 8000 rpm of engine speed. Therefore, 20 BTDC of ignition timing and 13:1 of compression ratio is the optimum engine

Keywords

Spark ignition engine, gasoline-bioethanol blend, Response Surface Methodology, combustion strategy, compression ratio

parameters used in gaining the optimal performance of the engine when E50 runs in SI-PFI engine of 150 cm^3

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Introduction. The conversion of fossil fuel into renewable energy must be done immediately, especially in transportation sector. It is an effort to avoid the use of fossil fuel as its availability has decreased and its combustion emissions contribute to air pollution and environmental damage. Nowadays, about 25 % of world oil consumption spent in automotive. Therefore, more than 70 % of air pollution in the atmosphere produced by the combustion of fossil fuel [1] and [2]. So, according to the data, the pollutants had been estimated to continue to increase by about 4 times along with the increase in the motorcycle and people population of the world in 2050 [3] and [4].

The lack of fossil fuel is not because its reserves have declined, but technologies for exploitation and exploration have not compatible to take up the oil from its location [5]. Thus, it is required a high technology to overcome this problem. However, it has an impact on a high-cost economy consequently. The strict regulations in vehicle emissions aimed to reduce greenhouse gas emissions and restrict the usage of fossil fuel. Therefore, it is necessary to encourage all countries around the world to use alternative fuel immediately [5–7].

Currently, bioethanol is one of the most popular fuels to replace gasoline as a vehicle fuel. It is because many combustion properties of bioethanol are similar to the gasoline in the spark engine. Moreover, the usage of bioethanol in gasoline engines will reduce carbon emissions significantly. Furthermore, the raw material of bioethanol is abundant and renewable. Bioethanol could be made from cellulose, corn, sugar, carbohydrates, or ethylene of gas in fuel oil. The materials will go through various processes such as galatination, hydrolysis, fermentation, distillation, or dehydration before turning into fuel grade [8–11].

As a fuel, bioethanol has several advantages over gasoline such as having high oxygen content (35 %). The oxygen molecules will make a better combustion process for decreasing carbon emissions and increasing torque and power, especially at high engine speed [12–15]. The high oxygen content of bioethanol also has an impact on the shorter propagation of laminar flame speed so that the combustion process will be more stable and the cycle variation value will decrease [16] and [17]. Therefore, these processes reduce the loss of pressure and increased combustion efficiency [18] and [19].

The octane number of bioethanol higher than gasoline, so this fuel could be applied to the engine with a high compression ratio. A study revealed that every increased octane number by 5 can boost up the compression ratio of the engine by 1 [20–23]. The bioethanol with a high octane number is effective to reduce detonation and avoid the engine breakdown [24] and [25]. So, the fuel could be applied in engine with a high compression ratio to increase the brake thermal efficiency [26] and [27]. Latent heat of vaporization of bioethanol is higher than gasoline, thus the properties will give a cold effect in the cylinder. Therefore, it will cause decreased in-cylinder pressure, so that the airflow rate into the intake valve increases and has an impact on higher volumetric efficiency [28]. Moreover, the cold effect in the cylinder will decrease the peak cylinder temperature so the production of NO_x could be minimized [29] and [30].

In contrast, there are several drawbacks to bioethanol if compared to gasoline. The Reid vapour pressure of bioethanol is lower than gasoline, so that bioethanol requires higher temperatures to facilitate the atomization process to be more easily. The drawback of this property has an impact on the engine that is difficult to run in the cold weather [30–32]. The use of bioethanol on the spark-ignition engines requires more advanced ignition timing compared to the gasoline [33] and [34]. It is due to the laminar flame speed of bioethanol faster than gasoline, so the ignition timing of bioethanol should be done more in advance to increase the temperature and pressure of evaporation of E50 so that it will be atomized easily. Moreover, the ignition delay will be longer if the ignition timing is specified earlier. This condition will be an advantage for the bioethanol to increase its volatility so that it is easy to be evaporated [35]. Also, by advanced ignition timing, the peak pressure of the cylinder can be maximized when bioethanol applied in a spark-ignition engine [36].

The cold start problem could be overcome by increasing the fuel temperature before injected into the cylinder by using the pre-heater process when the bioethanol applied [23]. The bioethanol is heated in the second reservoir, while the engine runs with gasoline in the primary tank in advance. The system could be switched to bioethanol when the fuel temperature increases and the operational temperature of the engine achieved. This problem also could be overcome by installing a heater element on the injector section [36].

The solubility of bioethanol in water is very high and it causes the oxidation process to be more easily in producing oxide compounds of $Fe_2O_3 \cdot H_2O$ [23]. These compounds will damage engine components made of metal or rubber. Therefore, it is necessary to develop the materials resistant to these compounds when the bioethanol is applied [22]. As a single compound, bioethanol has a boiling point in the range of 78–79 °C. Meanwhile, the gasoline is formed by the benzene group that has a boiling point in the range of 27–225 °C [24]. The difference in the boiling point of both fuels causes very low homogeneity of mixture and will produce an azeotropic effect in the blend [37–38]. It will have an impact on the low combustion stability, which is characterized by the increase of cycle variations, especially at high engine speed. This problem also will have an impact on the increased losses of torque and the decrease of mean effective pressure that the energy efficiency decreases [19], [26], and [39].

In many research, the use of bioethanol up to 20 % in gasoline is the optimum fuel proportion to obtain high engine performance without any adjustment on the engine. In this formation, the power and efficiency of the engine increased, while carbon emissions decreased compared to the gasoline [40]. It is because the bioethanol will act as an octane booster caused by the oxygen content of the bioethanol effective to produce a stoichiometric blend. However, in this composition, there is an increase in the production of NO_x emissions [41] and [42]. Meanwhile, the application of gasoline-bioethanol blend in a range of 25 % to 40 % required an adjustment on the engine either in ignition timing to be more advanced even with an increase of compression ratio. Furthermore, usage of gasoline-ethanol blend in concentration between 50 % and 100 % required an adjustment on several engine parameters [12–14]. It should be done to restrain the rate of decrease in power and torque as well as increases in fuel consumption if the concentration of bioethanol increases in the blend [37]. However, the combustion strategy applied in this study is considered capable to minimize the drawbacks of bioethanol, especially on the last concentration.

This study will improve the performance of E50 in the spark-ignition engine of 150 cm^3 by integrating three engine parameters, i.e., the compression ratio, ignition timing, and engine speed. The strategy will expect to produce high performance and low fuel consumption as well as less carbon emission. *Response Surface Methodology* (RSM) with the box Behnken design will be applied in this study. This methodology used to determine and predict the optimization area of the engine performances and emissions that will state through a correlation of varied engine parameters in a regression equation. This method was very useful to recognize how the significant impact of each combustion and engine parameters to influence engine performance. Moreover, this statistical method is strongly compatible with predicting the optimization of the dependent variable of the research towards the produced independent variable [43]. Overall, the method of RSM has successfully applied in this study with an error of less than 4 % from the results of the experiments.

Finally, the purpose of this research is to overcome the drawback of gasoline-ethanol fuel blend in composition 50:50 (E50) compared to pure gasoline by implementing a new combustion strategy. The combustion strategy is proven to produce higher power and torque rather than gasoline. Moreover, the adjustment of compression ratio and fuel ignition timing to the engine speed simultaneously can reduce fuel consumption and carbon emissions compared to the adjustment of engine parameters partially. All of the results was obtained will be compared to the engine performances by using gasoline fuel, even when the results are produced by RSM method and NLP software. However, the results obtained by using the RSM and NLP methods such as regression equation are not discussed detailedly in this article.

Methodology and experiment set-up. The test engine used in this experiment has detailed specifications as shown in Table 1. The standard compression ratio of the engine is 11:1, but it has developed to 12, 12.5, and 13 by applying a gasket and dome on the cylinder head to accommodate the methodology of this experiment. The statistical analysis is used codes of –1, 0, and 1 as a mark for compression ratios of 12, 12.5, and 13 (Fig. 1).

Table 1

Specification of test engine

Parameters	Standard
Engine type	4 stroke, 4 valve, 1 cylinder
Bore, mm	63.5
Stroke, mm	47.2
Displacement volume, cm^3	149.5
Compression ratio	11:1
Ignition system	Full transistorized
Maximum power	12,5 kW / 10000 rpm
Maximum torque	13,1 Nm / 8000 rpm
Intake valve open	5° BTDC, lifting 1 mm
Intake valve close	35° ABDC, lifting 1 mm
Exhaust valve open	35° BBDC, lifting 1 mm
Exhaust valve close	5° ATDC, lifting 1 mm
Valve train	Chain, DOHC

The air supply is fixed in a fully open throttle while the speed of the engine is varied from 2000 up to 8000 rpm. Furthermore, codes of –1, 0, 1 in the speed of the engine chosen to replace 2000, 5000, and 8000 rpm for application of RSM methodology. Meanwhile, 16, 20, and 24 BTDC (before top dead centre) of ignition timing has been used and will be represented by codes of –1, 0, and 1 respectively in statistical analysis. The codes of engine parameters can be placed on the *x* or *y*-axis as desired.

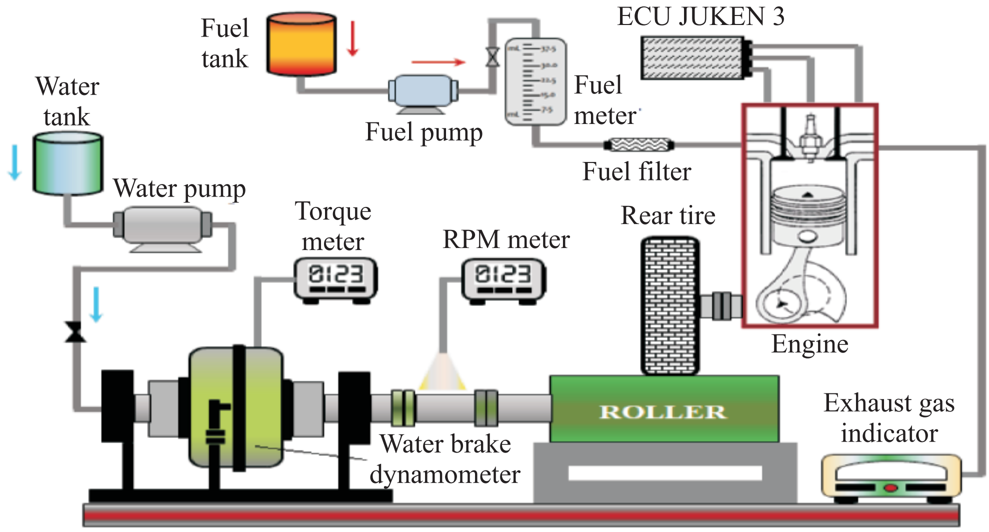


Fig. 1. The experiment tools set-up

The engine speed (rpm) was controlled with a load of Dynamite-type of water brakes dynamometer by opening a water intake valve from a reservoir. Then, torque noted based on the torque gauge that placed on the dynamometer after the engine reaches a specified speed. Some thermocouples are mounted at any engine equipment to record the temperature of coolant oil, cylinder temperature, and exhaust gas. A STARGAS 898 IND placed on the exhaust manifold to measure the concentration of carbon emissions, while a type of engine injector used CB150R street fire with part code of 16450K15921.

Gasoline with 94 of octane numbers is a comparing fuel in this work, while bioethanol with RON 119 the other one. Both of these fuels are blended in a ratio of 50:50 by volume. The properties of the fuel blend taken from various literatures as listed in Table 2. The bioethanol in this test produced by *Energy Agro Nusantara (ENERO) Co. Ltd*. The timing of fuel consumption will note when the engine had spent 25 mL of fuel in each testing section. In previous experiments had been noted that the best injection volume of E50 is in the range of 175–125 % or 1.75–1.25 that greater than gasoline on engine speed of 2000 up to 8000 rpm. It means that the injection volume of E50 will decreases with increasing the engine speed. The test based on the maximum brake torque (MBT), thus this value would be used in this work. The entire testing process controlled by an electronic management tool (ECU). The ECU of JUKEN-3 modified and created by the motor racing team Surabaya for more compatible with this work.

Table 2

Properties of fuels

Properties	E0	E50
Density, kg/m^3	757.5	763.4
RVP (at 37 °C), kPa	103.4	51.4
MON	88	88.1
RON	94	103.2
HHV, MJ/kg	44.2	33.34
LHV, MJ/kg	44	33.72
Distillation, °C: initial boiling point	42.1	329.2
10 %	61.5	57.3
50 %	102.3	75.1
90 %	163.1	79.4
end of boiling point	207.4	157.7

Results and discussion. The engine performances in this study consist of power, torque, thermal efficiency, specific fuel consumption, emissions of CO, and HC. Meanwhile, brake means effective pressure not revealed in this paper. However, the BMEP value will be reflected in the brake torque appearance because both of them have a linear correlation (see [12]–[14] and [27]):

$$\mathrm{BT} = \frac{\mathrm{BMEP}ALi}{2\pi z} = \frac{Pz}{ALNi},$$

where A is area of cylinder; L is length of stroke; i is indicates a number of cylinder on engine; z is a coefficient of motors based on the engine stroke ($z = 1$ for 2 stroke and $z = 2$ for 4 stroke); P is engine power; N is engine speed (rpm).

The use of bioethanol is very useful to reduce the production of NO_x emissions especially when the percentage of bioethanol in gasoline is over 50 %. It is due to the latent heat of vaporization of ethanol increases when its concentration gets higher in the gasoline fuel blend. The high latent heat of bioethanol has an impact on the decreased cylinder temperature so that the NO_x emissions could be minimized. However, the analysis of NO_x emissions not presented in this study due to the lack of testing tools.

The other side of ethanol usage has been expressed by Green [44], either as pure fuel and as a fuel blended. The result indicates that ethanol is the potential to produce a high greenhouse gas emission than gasoline. According to the environment protection agency (EPA), the utilization of ethanol can produce a high chemical emission and lead to the production of ozone. Ethanol farming systems have the potential to reduce the availability of land and freshwater sources. This is due to the cultivation of ethanol materials such as corn requires large areas of land and excessive water. In short, more corns cultivated for ethanol means more fertilizers, pesticides, and herbicides in waterways and it potentially contaminates the water [45].

Brake torque. The maximum torque obtained when ignition timing is between 16 and 24 BTDC (before top dead centre) as shown in Fig. 2, *a*. The ignition timing of E50 is more advance compared to the ignition timing of E0. It is because the latent heat of vaporization of bioethanol is higher than gasoline [43] and [46]. The result shows that on low engine speed, the torque of gasoline is higher than E50. However, above 3000 rpm, the torque of E50 continued to intensify as an increase of speed and it is higher than the torque of E0. The torque of E50 is lower than E0 on the low engine speed because of two reasons. The first is the energy content of E50 is lower than E0. The last is the vaporization pressure of E50 lower than E0. It caused the atomization process of E50 is less than E0. In this study, it was also found that the torque of E50 has a similar trend between experiments (exp.) result and optimization result (opt.).

By setting up the fuel ignition at 20 BTDC, the torque of E50 will get higher than gasoline, especially on the compression ratio of 12.5 and 13 (Fig. 2, *b*). The figure also shows that the torque of E50 gets higher as an increased compression ratio. It is because of the octane number of E50 higher than E0 so that it is allowed for the application of a higher compression ratio [47].

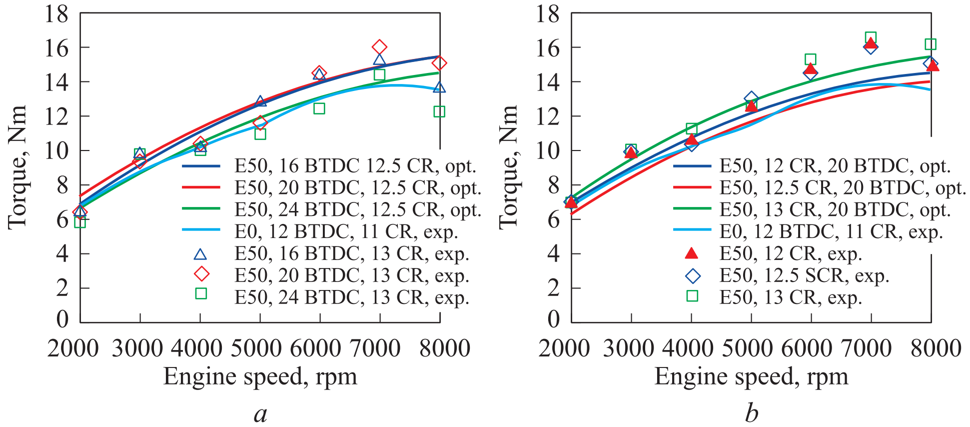
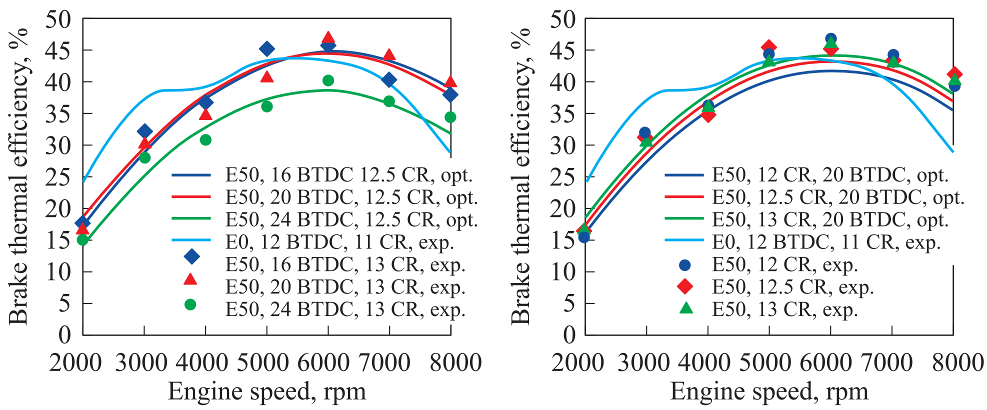


Fig. 2. Torque vs variation of fuel ignition; rpm on opt. and exp. (*a*), torque vs variation of compression ratio; rpm on opt. and exp. (*b*)

The high compression ratio will generate high pressure in the cylinder and it will have an impact on increased torque of the engine. It can occur as the ratio between cylinder volume and the piston stroke getting smaller if the compression ratio increases [44] and [48]. Therefore, it caused an increase in the turbulence flow among air and fuel as well as increasing the density of mixture in the cylinder. This process will produce high pressure and torque.

Based on the RSM analysis, the optimization value of torque obtained when the engine works at high speed in all compression ratios and ignition timing is fixed on 20 BTDC. It is due to the intensified cylinder temperature as an increase of engine speed, so the E50 easy to evaporate. The non-linear programming found that the optimum of torque would be obtained on 8000 rpm of engine speed, 12.5 of compression ratio, and 20 BTDC of ignition timing.

Brake thermal efficiency. The thermal efficiency of E50 will increase if the ignition advanced compared to the ignition timing of E0, particularly on engine speed the over 5000 rpm (Fig. 3, *a*). The standard ignition timing of E0 is 12 BTDC while the optimum value of BTE of E50 obtained on 20 BTDC. However, the change in fuel ignition of E50 from 16 to 20 BTDC has no significant effect on the BTE. Otherwise, the BTE of E50 will decline if the ignition timing is too advanced to 20–24 BTDC and it lower than BTE of E0. Meanwhile, the BTE of E0 is higher than E50 on the low engine speed because the E0 is more volatile and has a higher heating value compared to the E50.



a *b*

Fig. 3. BTE vs fuel ignition; rpm on opt. and exp. (*a*), BTE vs CR; rpm on opt. and exp. (*b*)

The ignition timing of E50 should be more advanced if compared to the ignition timing of E0. It is because the latent heat of vaporization of E50 higher than E0, so the E50 has longer ignition delay than E0 to absorb more heat in-cylinder [35]. Moreover, the advanced ignition timing for E50 intended to accommodate an increase of the compression ratio due to an increase of the octane number of E50. In this condition, the thermal efficiency of E50 would intensify due to the latent heat of E50 to be more useful in producing the work. The values of the optimization of thermal efficiency by RSM analysis obtained at a compression ratio of 12.5:1.

The thermal efficiency tends to decrease when the engine speed over 6000 rpm as shown in Fig. 3, *b*, even though at that point the brake torque still increases (see Fig. 2). It is because the generated brake torque is lower than the mass flow rate of fuel at the same speed. The thermal efficiency of gasoline is higher than E50, especially when engine speeds lower than 5000 rpm (see Fig. 3). It is because the vapour pressure of bioethanol is lower than E0 so that it requires high temperature and it will be obtained when the engine runs at a high speed [49]. On the cylinder temperature is low, the E50 difficult to evaporate so the air-fuel blend on inhomogeneous conditions. Thus, it will cause an increase in energy loss and influence on decreased thermal efficiency. The result of the RSM method indicated that the BTE optimization area of E50 is in the compression ratio by 12 up to 13. It obtained when the engine speed in range 5000–7000 rpm and ignition timing is fixed at 20 BTDC.

Brake specific fuel consumption. BSFC is a parameter that reflects the mass of fuel consumed to generate an engine power at a certain time. Fuel consumption influenced by many parameters including the heating value of fuel that reflected on the flow rate of fuel in the combustion process [50]. The high energy content of fuel has a potential to produce high power and low fuel consumption, as has been proven in Fig. 4, *a*. The BSFC of E0 lower than E50 especially on engine speed under 6000 rpm, although E0 burned on the lower compression ratio that is 11:1. Nevertheless, the increase in BSFC of E50 could be reduced by an advance of ignition timing and increases the compression ratio as a result of this study.

The application of the testing strategy in this research has been successful to reduce the flow rate of E50 to be 1.35 that still higher than E0. This result is better than previous research that is on 1.60–1.80. The decrease of BSFC of E50 at high engine speed due to the atomization of the fuel is better when the cylinder temperature intensifies as an increase of speed. Moreover, the high oxygen content of E50 also contributes to the decrease of BSFC. It is because of the oxygen compound in E50 more effective to make a combustion process complete [41, 51–53]. The fuel consumption for all types of fuel would increase when the engine works at a speed of 6000–8000 rpm. It is because injected of air into cylinder decreases so the combustion process becomes rich. Moreover, the cylinder temperature will increase when the engine speed increases so that it would cause friction in a cylinder and has an impact on decreased BSFC [54].

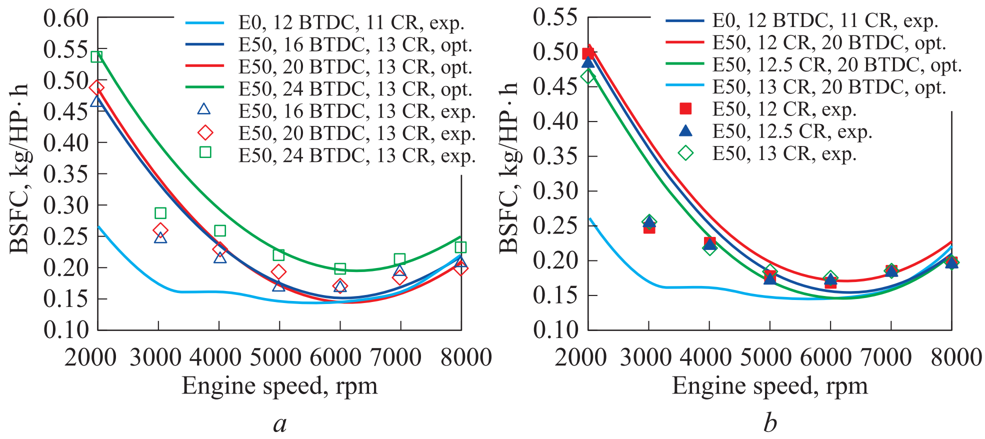


Fig. 4. BSFC vs ignition timing; rpm on opt. and exp. (*a*), BSFC vs compression ratio; rpm on opt. and exp. (*b*)

Advance of ignition timing that higher than 12 BTDC would increase the BSFC of E50 both in experimental and in RSM method (Fig. 4, *b*).

The RSM method found the optimization area of BSFC obtained when the engine parameters in position; compression ratio of 12–13, engine speed of 5000–7300 rpm, and ignition timing fixed at 20 BTDC. Meanwhile, the application of non-linear programming predicted that the optimum result of BSFC would be obtained on 6051 rpm, the compression ratio of 13 and 20 BTDC on ignition timing.

Emissions of Carbon-monoxide (CO). The most popular advantage of bioethanol is to produce lower hydrocarbon emissions than fossil fuels. In many studies, the usage of bioethanol could reduce CO emissions by around 40–80 % and HC emission of 18–40 % compared to gasoline [12–14, 55] and [56]. As an oxygenated fuel, the oxygen content in bioethanol by 35 % and it is proven effective for making the air-fuel ratio close to the stoichiometric state. It would influence for better combustion process thus it decreases carbon emissions [35] and [52].

The changes in ignition timing from 16 up to 24 BTDC have no significant impact on CO emissions-reducing (Fig. 5, *a*). However, the CO would decrease when the engine works over 5000 rpm, while ignition timing on 24 BTDC. It was obtained both in the RSM method and in the experiment. The increase in compression ratio would have a significant impact on the decreased CO emissions (Fig. 5, *b*).

In the variation of the compression ratio, the CO emission of E50 is lower at around 71.62 % than gasoline. Meanwhile, in a variation of ignition timing, CO emissions are lower approximately 71.55 % than gasoline. The minimum

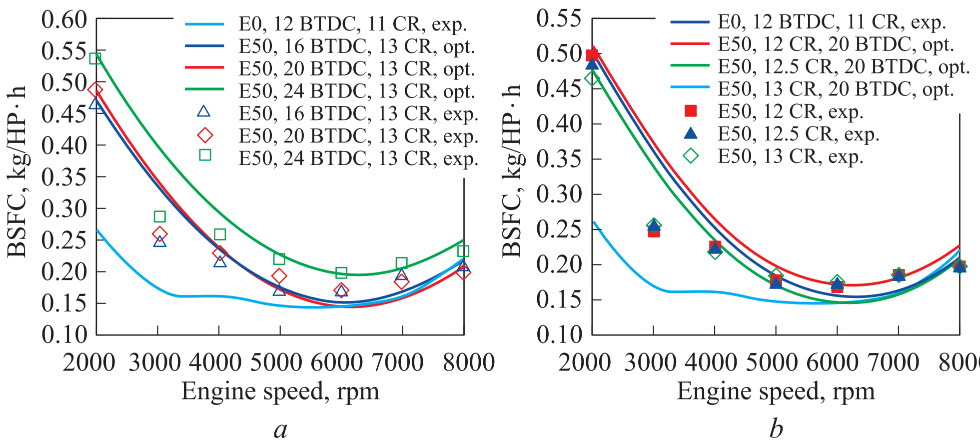


Fig. 5. CO emissions vs IT; rpm on opt. and exp. (*a*), CO vs CR; rpm on opt. and exp. (*b*)

CO emissions were obtained on the compression ratio of 12.5–13.0 when the engine runs on 2000–5000 rpm and ignition timing is constant at 20 BTDC. Furthermore, the optimization area of CO will be changed if the compression ratio constant is 12.5 while ignition timing and engine speed sets on 16–24 BTDC and 2000–5000 rpm respectively.

Emissions of Hydro-carbon (HC). The high production of HC emissions in a combustion process does not only caused by the non-stoichiometric state of air and fuel blend. It also could be caused by interference in the ignition system consisting of non-standardized wire of spark plugs, coils, and spark plugs. These problems would have an impact on generated electric current as a trigger for a spark plug becomes poor, even distribution of the electric system does not work properly. Besides, excessively retarded or advanced in ignition timing has a contribution to the misfiring of fuel and increase of hydrocarbon emissions [57] and [58].

The compression ratio on 13:1 and the ignition timing on 16 BTDC should be installed on SI-PFI of 150 cm^3 to produce lower HC emission as suggested in Fig. 6, *a*. Nevertheless, the fuel ignition exceeding 24 BTDC has a contribution to the increased HC emissions. It is influenced by the fast flame propagation of E50, thus if the ignition sparked in overly advanced then the explosions of combustion of air-fuel blend would occur earlier. In this condition, a part of hydrocarbon fuel does not completely burn yet that it would be HC emissions and going out to the environment.

The high compression ratio would have an impact on the reduced HC emissions (Fig. 6, *b*). It is due to the increase in the compression ratio that

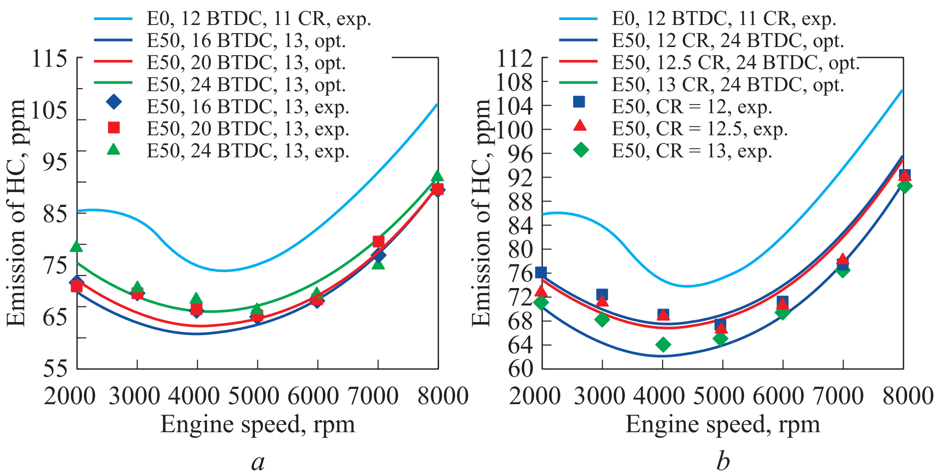


Fig. 6. HC emissions vs fuel ignition; rpm on opt. and exp. (*a*), HC vs compression ratio; rpm on opt. and exp. (*b*)

would boost up cylinder temperature [51] and [52]. Also, the increase on compression ratio would intensify homogeneity of the air-fuel blend and it would have an impact in the improved combustion efficiency so that it decreases HC emissions. In non-linear programming found that the ignition timing of fuel in interval between 12 and 13 has no significant effect on reducing HC emissions. Otherwise, the compression ratio and engine speed influence the decrease in HC emissions. The optimum area of HC emission is obtained if the engine compression ratio is 13:1, the fuel ignition in interval 16–18 BTDC and the engine speed is 5000 rpm. Meanwhile, the non-linear programming indicates that the optimum of HC will obtain on 5000 rpm of engine speed, the compression ratio of 13, and ignition timing of 24 BTDC (see Fig. 6, *a*).

Brake power (BP). The brake power of E50 will continues to increase with increasing engine speed and higher than the brake power of E0 (Fig. 7). By the RSM method and nonlinear programming, the brake power optimization of E50 around 15.48 HP obtained at the ignition timing of 20, the compression ratio of 12.5, and the engine speed of 8000 rpm. Meanwhile, under the same conditions, the brake power of E50 in the experiment noted at 16.90 HP. In many previous studies, the brake power of E0 will show a tendency higher than E50. This result is very reasonable because the energy content of E0 is greater than E50 and does not any adjustment neither in engine parameters nor combustion parameters [59] and [60–62].

On the other hand, the testing also is done on the gasoline engine without making changes to the engine. However, advancing fuel ignition and increasing engine compression ratio as well as adding injection volume following E50

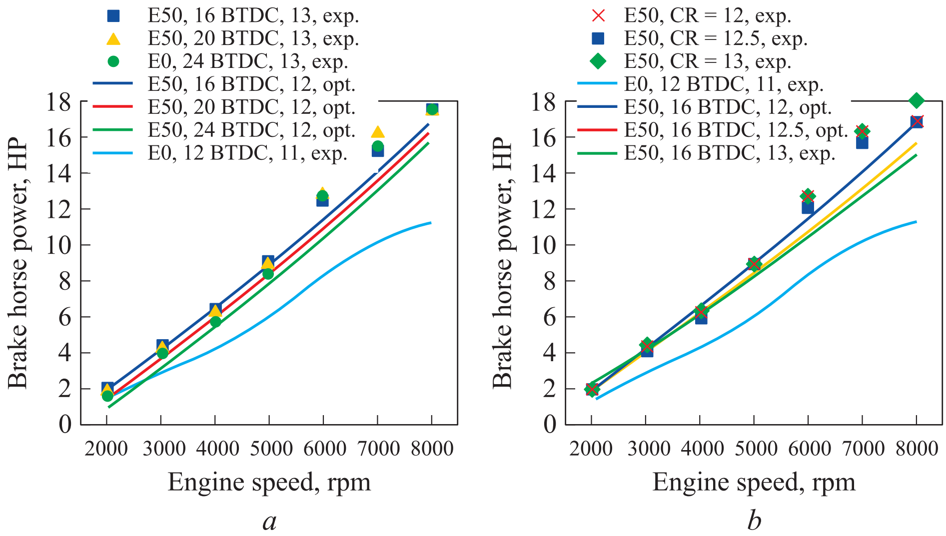


Fig. 7. Power vs fuel ignition; rpm on opt. and exp. (*a*), power vs CR; rpm on opt. and exp. (*b*)

characteristics, will result in better engine performance as shown by the results of this study. The optimization values of the brake power is obtained at a compression ratio of 12 and 12.5 and an engine speed of 7000–8000 rpm if the ignition timing set at 20 BTDC. Based on the results of non-linear programming the optimization values of the brake power of E50 obtained at a compression ratio of 12.5 and ignition of 20 BTDC as well as on engine speed of 8000 rpm.

The maximum engine torque of E50 obtained at 7000 rpm as shown in Fig. 8, *a*. The torque of the engine continued to intensify as an increase in engine speed. However, it would tend to decrease when the ignition timing is advanced. Also, the unsteady torque in a variation of ignition timing will get stronger but tend to decrease when the engine speed exceeded 5000 rpm. It can be caused by exceedingly huge differentiation of boiling point, the heat of vaporization, and vapour pressure of the fuel. By increasing the homogeneity of the fuel blend is considered as one solution to reduce the phenomenon. The fluctuation of the brake torque on a variation of ignition timing also can be caused by interference in the ignition system. However, this problem still requires advanced research as evidence.

The suitable injection volume of E50 for SI-PFI engine was in interval 1.25–1.75 if compared to E0 (Fig. 8, *b*) and E50 injection volume decreases as engine speed increases. The figure also describes that at an engine speed of 2000 rpm is required an injection volume of 1.75 is greater than E0, while at 8000 rpm is only 1.25 greater than E0. It is due to by the vapour pressure

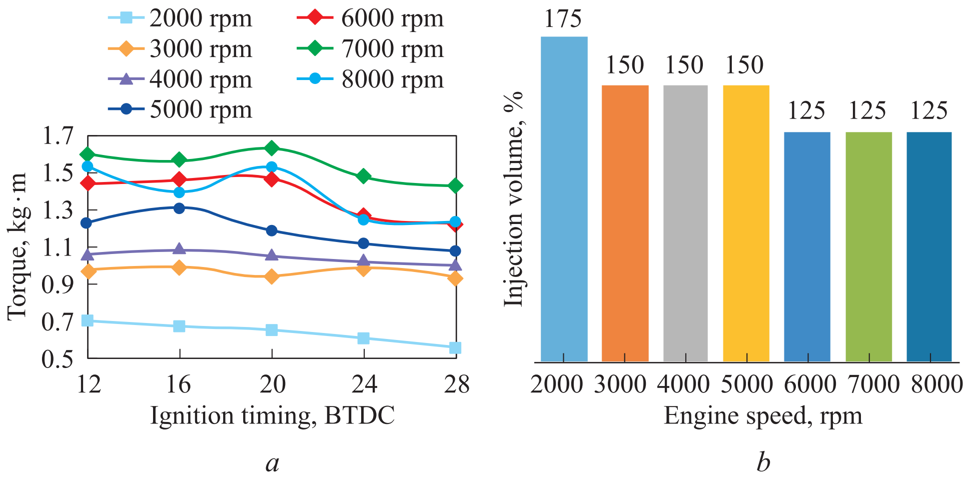


Fig. 8. Effect of fuel ignition and rpm on the torque of engine (*a*), injection duration along the engine rpm (*b*)

of E50 intensifies caused by the increased of cylinder temperature and the engine speed boosts up.

Conclusion. Increasing the compression ratio is more effective to improve the performance of the spark-ignition engine than advancing the ignition timing if using E50 fuel. This research recorded that BSFC increased by 36.64 % on 20 BTDC when advancing ignition timing while boosting up the compression ratio, the BSFC of E50 increased only around 32.8 % compared to the BSFC of E0. It indicates that the fuel consumption of E50 saved more significantly when improving the compression ratio than advancing the ignition timing of fuel. Although in practice, raising the engine compression ratio must also advance the fuel ignition. this study also found that the average E50 injection was 1.35 greater than E0 but it better than the results of previous studies which were in the average range of 1.6–1.8.

The BTE of E50 decreased by 14.28 % if the fuel ignition advanced, while only around 10.09 % decreased when the compression ratio increased compared to the E0 on an interval of engine speed of 2000–5000 rpm. Meanwhile, at high engine speed, the thermal efficiency would increase on average by 17.30–17.87 % compared to gasoline. It occurs when the ignition timing advanced and the compression ratio increased. It is proof that bioethanol is superior to gasoline, especially at high speed because bioethanol needs a high temperature more than gasoline to obtain high performance.

The increased BSFC and decreased BTE of E50 fuels compared to gasoline in this study could be still minimized by reducing the interval of injection volume and ignition timing. The reasons are to explore the heating value and vaporization heat of the fuel on a suitable point with their characteristic. As information, those parameters in this work have been done very well in the experiment for pure bioethanol (E100). Hence, for the next research, the internal mapping of injection volume and ignition timing should be smaller than 25 % and 4 BTDC when the E50 is applied.

On the other parameters, HC and CO emissions decreased by 14.64 % and 71.55 % when the ignition timing of E50 advanced from the ignition timing of E0. Those values are similar when the compression ratio of the engine increased from 11:1 to 13:1. Meanwhile, the torque of E50 boosts up around 13.40 % compared to gasoline, on a high compression ratio, and the optimum ignition timing. This study found that required an injection volume of E50 by 1.35 greater than injection volume of gasoline.

Table 3

Optimization of single response

Performance	Engine parameters			Values
	Compression ratio	Ignition timing	rpm	
Torque, Nm	12.5	20	8000	14.1
BSFC, kg/(HP·min)	13	20	6251	$2.5 \cdot 10^{-3}$
BTE,	12.5	20	6032	41.83
CO, %	13	20	5000	0.757
HC, ppm	13	24	5000	64.63
Brake power, HP	12.5	20	8000	15.48

Table 3 denotes the optimum performance in a combination of engine parameters and the combustion parameters. Table 3 expresses the optimization values of all response variables in a single response method by using non-linear programming. It shows that the optimum of brake torque obtained in a combination of factor variable, i.e., compression ratio of 12.5, the ignition timing of 20 BTDC, and an engine speed of 8000 rpm. However, the statistical analysis by using the RSM method found that the variation of the compression ratio has no significant impact on the performance of the engine, and the statement has been shown in Table 3. The result of the RSM method and non-linear programming in Table 3 could be used as standard data for the development of the electronic control unit (ECU) when the engine runs with E50.

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