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ENERGY AND EXERGY ANALYSIS ON SPARK IGNITION ENGINES UNDER VARYING IGNITION TIMING WITH PURE BIOETHANOL FUEL

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Abstract

The first and second laws of thermodynamics analysis used to show the rate of energy and exergy as a performance of SI-PFI engine with the variation of fuel ignition between 10 and 26 BTDC (before top dead Centre) at interval 4 BTDC. The engine was performed on eight levels of speed in intervals 2000–8000 RPM (increment of 1000 RPM) with pure bioethanol fuel (E100) and 13:1 of compression ratio. The effect of fuel ignition on energy and exergy analysis of E100 fuel will be compared with E0 as reference fuel that performed in 11:1 of compression ratio and 10 BTDC. The results show that the maximum efficiency of energy and exergy for the E100 are 46.59 and 41.90 % at 18 BTDC and 6000 RPM. Meanwhile, the maximum efficiency of energy and exergy for E0 were 43.33 and 31.76 % at 5000 RPM. Moreover, the minimum BSFC for the E100 is 0.2867 kg/(kW·h) at 6000 RPM and 18 BTDC while for the E0 is 0.1960 kg/(kW·h) at 5000 RPM. These results indicate that E100 is more effective in transferring heat into useful work although it is 30 % more wasteful than E0

Keywords

Energy, exergy, spark ignition, bioethanol, gasoline, thermodynamics law

Received 02.08.2022
Accepted 29.11.2022

Introduction. Fossil fuel consumption continues to increase from year to year along with the times and the increase in world population. It will cause increasing carbon emissions in the air that trigger global warming. It will continue to damage the environment and reduce oil reserves that threaten human survival. Therefore, the use of bio-hydrocarbon fuels cannot be avoided because it will be one of the solutions to these problems. Biofuel is one type of alternative fuel that is considered capable of restraining the rate of production of carbon emissions through the use of bioethanol for gasoline and biodiesel for diesel fuel [1]. Both fuels have similar combustion characteristics with the dedicated fuel of engines so that their use does not need a significant adjustment on the machine [2–5]. Many studies showed that the carbon emission produced by biofuel is lower than fossil fuel when it is applied both to a spark-ignition engine or diesel engine. [6, 7].

Based on the properties analysis, there are some advantages of bioethanol compared to gasoline fuel when applied to a spark engine. Bioethanol has approximately 35 % greater oxygen compounds from gasoline, as shown in Table [8, 9]. The existence of oxygen compounds in bioethanol is a primary contributor to reducing carbon emissions in the combustion process, so the production of carbon emissions is lower, and it was done by Chansauria [10]. The research also shows that the use of E5, E10, M5, and M10 fuels produced hydrocarbon emissions lower than gasoline, even for NO_x emissions. The oxygen content in bioethanol has also had a contribution in reducing the air-fuel ratio, so the AFR of bioethanol is lower than gasoline.

The comparison of fuel properties of gasoline and bioethanol

Properties	Fuel	
	Gasoline C_8H_{15}	Bioethanol C_2H_5OH
Composition of C, H, O, mass %	86, 14, 0	52, 13, 35
Research octane number	90–100	110
Oxygen content, mass %	0	34.7
Lower heating value (LHV), MJ/kg	42.4	29.6
Heat of evaporation, kJ/kg	223.2–349	725.4–920
Stoichiometric A/F ratio	14.6	8.9 9
C / H ratio, %	0.44–0.55	0.33
Density, kg/m^3	730	794
Solubility in water, ml/100 ml H_2O	< 0.1	Fully miscible
Flash point, °C	–45 to –38	21.1
Auto ignition temperature, °C	olum434	434
Boiling point, °C	25–225	78.4
Laminar flame speed, cm/s	33–34	39–42
Reid vapour pressure, kPa	53–102.4	15.8–17

The octane number of bioethanol is higher than gasoline and led to bioethanol could be applied in engines with having a higher compression ratio. Sakthivel's research [11] proves that carbon dioxide and hydrocarbon emissions will decrease when the compression ratio of engines is increasing from its standard size. The study result was obtained when the engine using E30 on the spark-ignition engine. The results of this study also show that the brake power increases if the compression ratio increase from the standard compression ratio to a ratio of 9.4, 10.9, and 11.5, although at the same time, the NOx emission increases. Moreover, the BSFC decreases while the BTE and volumetric efficiency increase if the compression ratio increases in the range 8, 9, and 10, when using gasoline and methanol blends of M50 [12].

Meanwhile, there are many shortcomings of bioethanol when compared to gasoline when applied to SI engines. The lower calorific value of bioethanol than gasoline causes bioethanol consumption (BSFC) to increase in the same injection volume [13]. The lower calorific value of bioethanol than gasoline also causes engine power and torque to be lower than gasoline if the bioethanol is used in the engine, as noted in [14]. A simple indicator that shows the calorific value of bioethanol is lower than gasoline can be confirmed from the number of carbon atoms in fuels or through the compound names of the fuel (see Table). The latent heat of vaporization of bioethanol is higher than fossil fuel. Therefore, bioethanol will have produce heat energy three times higher than gasoline. It has been proven in many studies, one of which was conducted by Reddy [15]. His research shows that the heat released from the orange oil and diesel fuel blend is higher than diesel fuel. Likewise, the heat released by the orange oil-diesel fuel blend is higher with the increasing percentage of biofuel in the blends.

The high solubility of bioethanol in water will facilitate the oxidation process in materials that use these fuels. It will cause the metal material to corrode while the rubber material will stretch so that it does not function properly. Therefore, the usage of bioethanol fuel requires tighter and more close handling of atmospheric air [16]. The lower vapor pressure of bioethanol than gasoline is a problem in the SI engine, especially in cold weather which the engine is difficult to start. However, research [17] shows there is no evidence of engine malfunction even when it is started at 50 °C when the percentage of methanol increases in the gasoline mixture. Moreover, flame propagation and fast burning of bioethanol are shorter than gasoline, so that the combustion process will increase in the first 50 cycles if the percentage of methanol increase to 30 % in the blend.

Exergy analysis or availability analysis used to determine the quantity and quality of energy in a heat generation system using the thermodynamic second law [18]. This method applicated is one of the ways to save energy, especially in the fossil energy crisis that is faced in all the countries in the world today. All the results mentioned in the introduction section above are analyzed by the energy analyses by referring to the thermodynamic first law. The advantages of exergy analysis compared to energy analysis include; exergy analysis is more accurate in making optimal designs for thermal and energy processes, is more accurate in calculating the energy loss in the process and wasted into the environment, and to identify the quality of energy in the process [19].

The use of the second law of thermodynamics to analyze exergy rates in internal combustion engines in various types of fuels has been done by many researchers. Ibrahim [20] has calculated the performance of an SI engine with a mixture of gasoline-bioethanol, and gasoline-methanol blend using energy and exergy analysis. The results show that engine speed has a high impact on fuel exergy, exergy loss, exergy destruction, and brake power when both of the fuel blends are applied. In [21] analyzed the effect of fuel ignition on the performance of hydrogen-fueled SI engines using energy and exergy analysis. The result shows that the efficiency of exergy decreases if the fuel ignition continued to advance.

This study will test the performance of energy and exergy single-cylinder spark-ignition engine fueled pure bioethanol (E100) at a high compression ratio and variable fuel ignition. Engine performance evaluated at wide-open throttle, fuel ignition varies from 10–26 (increment 4 BTDC), engine speed from 2000–8000 RPM (increment 1000 RPM), and 13:1 of compression ratio. Moreover, at the same time, it has obtained testing data of engines on standard compression ratios which 11:1 using gasoline fuel. All the experiment's results will be compared between one and another to describe their characteristics. Unlike previous studies, this study applies a high compression ratio in engine speed range widely with variations in fuel ignition. These parameter settings are conditioned to reduce the deficiency of the E100 in the combustion process and optimize its advantages.

Energy and exergy analysis on bioethanol combustion under various ignition timing and compression ratio has never been conducted by researchers before. Advancing the ignition timing of burning bioethanol is intended to extend the ignition delay to increase the evaporation pressure of bioethanol so that it is easily atomized and mixed with air to produce stoichiometric combustion. Increasing the compression ratio in the bioethanol combustion is needed to accommodate the octane number of bioethanol is higher than gasoline so that there is no knocking in the combustion process. Therefore, this research has a strategic position for researchers in the future as one method to obtain high performance both in energy and exergy analysis.

Materials and methods. This research will show the differentiation between energy and exergy flow for a single-cylinder spark-ignition engine using gasoline

(E0) and pure bioethanol (E100). The engine compression ratio has been improving to 13:1 to accommodate the objective of this study when using E100. Meanwhile, the engine remains conditioned on the standard size when using gasoline fuel.

The airflow rate is allowed to flow normally because the engine throttle is fully open wide, so the test starts at maximum engine speed. Meanwhile, the volume rate of fuel is measured using a stopwatch and its time is recorded after fuel consumption reaches 15 mL. The engine speed was measured using an Omega hht12 tachometer. Type K thermocouples were placed on several parts of the engine to measure engine coolant oil temperature, exhaust gas temperature, and engine temperature. Product emissions of the combustion process were obtained from a gas analyzer of STARGAS type. The specification of spark engine is given below:

Bore × stroke, mm	63.5 × 47.2
Displacement volume, cm^3	149.5
Compression ratio	11.0:1
Ignition system	Full transistorized
Fuel injection system	Electronic fuel injection system
Maximum power, kW	12.5
Maximum torque, N · m	13.1

The data obtained in the experiment are engine temperature, coolant temperature, exhaust temperature, engine rpm, fuel consumption rate, and engine torque. Meanwhile, other data was obtained through calculating using suitable mathematic equations. All the data above obtained when the engine speed in 2000 up to 8000 RPM (increment 1000 RPM) and on fuel ignition of 10 to 26 BTDC (increment 4 BTDC) for all fuel tests. Torque data was obtained by applying a water break dynamometer coupled to the rear tire axle. Likewise, the engine rpm is obtained by adjusting the water flow to the dynamometer using the Omega hht12 tachometer (Fig. 1).

Energy analysis. The main principle of energy analysis is that energy is eternal so that energy changes from one state to another are the same. Thus the energy changes in a system during adiabatic and non-adiabatic processes are the same [22]. Energy can move in two modes, namely heat (Q) and work (W), so that the energy change in a closed system must be the same as the accumulation of the amount of heat energy transferred into the system and the work generated by the system, or written in the equation below:

$$E_2 - E_1 = Q - W. \tag{1}$$

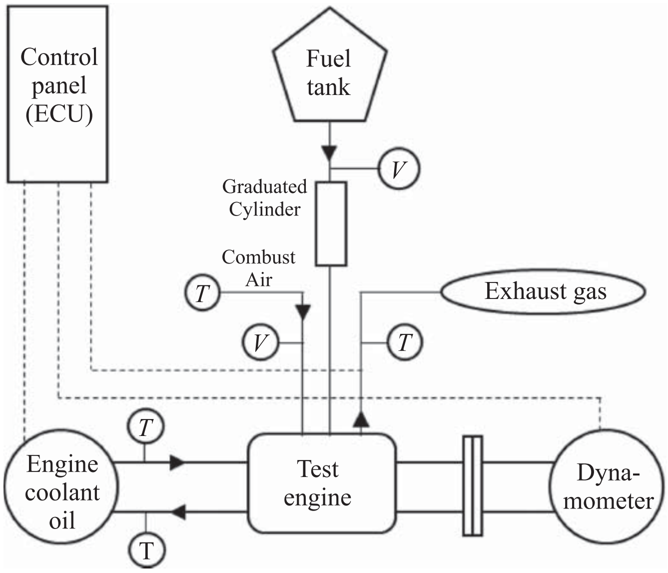


Fig. 1. Schematic diagram of temperature measurement and fuel flow rate
(V — velocity; T — temperature)

The energy change in Eq. (1) can be associated with the change in internal energy (ΔU) in which the kinetic and potential energies are neglected. In an internal combustion engine, the initial work is put into the system so that the piston can do the work to flow the air and fuel mixture into the cylinder and then compress it before combustion occurs. The combustion process is a manifestation of changing chemical energy into heat energy. Therefore, the energy balance of a closed system at steady-state conditions is:

$$\sum u + pv = \frac{Q}{m} - \frac{W}{m}$$

or $\sum \dot{m}_{out} h_{out} - \sum \dot{m}_{in} h_{in} = \dot{Q} - \dot{W}$, where $\dot{m}$ is the mass flow rate of fluid; $\dot{h}$ is the specific enthalpy of fluid; $\dot{Q}$ is the net heat transfer rate; $\dot{W}$ is the rate of brake power. Meanwhile, the subscript in and out is indicated the mass or heat flow into and out of the system. In energy analysis, the mass conservation law also applies to volume control under steady-state conditions, namely:

$$\sum \dot{m}_{out} = \sum \dot{m}_{in}. \tag{2}$$

From Eq. (2), it can be seen that the mass flow rate of the gas and the air would be calculated from the AFR formula as written below:

$$\text{AFR} = \frac{\dot{m}_{air}}{\dot{m}_{fuel}},$$

where the AFR of each fuel is 8.99 for bioethanol and 14.6 for gasoline. Furthermore, the potential energy in the fuel will be converted into heat energy through a thermo-chemical process. Then, the heat is converted into the work through the crankshaft. However, the heat generated does not convert entirely into useful work, but some of it will be lost to the environment. Therefore, the quantity of heat and work produced from the fuel energy conversion process is

$$\dot{E}_{fuel} = \dot{W} + \dot{Q}_{exh} + \dot{Q}_{loss},$$

where E_{fuel} is the fuel energy; $\dot{Q}_{exh}$ is fuel energy that converted into heat of combustion gases; $\dot{Q}_{loss}$ is heat energy loss to the environment. Assuming that combustion occurs at a pressure of $P_0 = 1$ atm and a temperature of $T_0 = 298$ K as a reference, then the energy of combustion air is negligible. Thus, the rate of heat energy entering the system is $\dot{E}_{fuel} = \dot{m}_f LHV$.

The caloric value of gasoline and bioethanol presented in Table, and the mass flow rates obtained in the experiment. The brake power value in Eq. (3) could be calculated by the equation:

$$\dot{W} = \frac{2\pi n T}{60}.$$

Here $2\pi n/60$ is the conversion of the engine angular speed ω from rev / min to rad/s; T is the engine torque. Meanwhile, the heat rate of exhaust gas is an accumulation of the enthalpy of gas produced as a combustion product. The balance of the combustion reaction for gasoline and bioethanol is as follows:

$$\mathrm{C_xH_y} + a\left(\mathrm{O_2} + 3,76\mathrm{N_2}\right) \rightarrow b\mathrm{CO_2} + d\mathrm{H_2O} + e\mathrm{N_2}, \quad (4)$$

$$\mathrm{C_2H_5OH} + a\left(\mathrm{O_2} + 3,76\mathrm{N_2}\right) \rightarrow b\mathrm{CO_2} + d\mathrm{H_2O} + e\mathrm{N_2}. \quad (5)$$

The type of gasoline used in this study is C_8H_{15} known in Indonesia as premium. The main product of combustion gases from the reaction equation ((4) and (5)) is CO_2, H_2O and N_2, while other compounds are ignored, so the heat rate of the combustion gas exhaust is

$$\dot{Q}_{exh} = \dot{m}_g \Delta h_{\mathrm{CO_2}} + \dot{m}_g \Delta h_{\mathrm{H_2O}} + \dot{m}_g \Delta h_{\mathrm{N_2}}.$$

Heat energy loss from the system to environment could be calculated by modifying Eq. (3) to be

$$\dot{Q}_{loss} = \dot{E}_f - \left(\dot{W} + \dot{Q}_{exh}\right).$$

The engine performance is usually measured from two important parameters; thermal efficiency and specific fuel consumption (BSFC), and both could be calculated using the following formulas:

$$\eta = \frac{\dot{W}}{\dot{E}_f} \cdot 100\ \%, \quad \text{BSFC} = \frac{\dot{m}_f}{\dot{W}}.$$

Exergy analysis. Exergy analysis is a method of calculating the rate of energy in a closed system that undergoes an irreversible process. The irreversible process occurs caused by the system in contact with the environment so that heat energy generated by the system tends to be released into the environment as destructive exergy. The release of heat to the surrounded will accompanied by the production of entropy by the system and will reduce the performance of the system itself. Therefore, the greater the heat released into the environment, the greater the production of entropy and increase the exergy destruction [23–24]. Thus exergy is defined as the theoretical maximum work that can be achieved by a system when it interacts with its environment in achieving equilibrium [25].

The concept of exergy analysis introduced through the conception of reversible work for energy analysis. Therefore, with ignore the kinetic energy, potential energy, electric energy and the nature of the magnetization of the material, then the exergy balance in the control volume of the steady-state is

$$\sum \dot{m}_{in}\varepsilon_{in} = \sum \dot{m}_{out}\varepsilon_{out} + \sum\left(1-\frac{T_0}{T_w}\right)\dot{Q}_{loss} + \dot{W} + \dot{E}_d, \tag{6}$$

where $\sum \dot{m}_{in}\varepsilon_{in}$ is the rate of mass transfer and exergy entering the combustion chamber that consisting fuel and air; $\sum \dot{m}_{out}\varepsilon_{out}$ is the flow rate of gas as a product of combustion with specific exergy. However, the exergy effect of combustion air is neglected by assuming that the combustion air enters the engine at ambient conditions. The second term to the right of Eq. (6) is the rate of exergy displacement accompanying the heat transfer or written as:

$$\dot{E}x_Q = \sum\left(1-\frac{T_0}{T_w}\right)\dot{Q}_{loss}. \tag{7}$$

Here $T_0 = 298$ K is the reference temperature; T_w is the temperature of the coolant oil of the engine. The third term to the right of Eq. (6) is the rate of exergy that transfer through the work and calculated by the equation: $\dot{E}x_W = \dot{W}$.

Therefore, the exergy rate transferred through the work is equal to the engine brake power. Meanwhile, the specific exergy of combustion gas products is calculated based on the specifics of enthalpy and entropy of each compound present in the combustion gas products, namely H_2O, CO_2, and N_2 by the equation $\varepsilon_{out} = (h-h_0) - T_0(s-s_0)$, where the specific enthalpy h and entropy s of each compound contained in the combustion gas product are calculated based on temperature, and it is assumed as an ideal gas. The enthalpy h_0 and entropy s_0 of each compound are in a dead state. The fuel exergy calculated by equation $\dot{E}x_{in} = \dot{m}_f \varepsilon_f$, $\dot{m}_f$ is the flow rate of fuel and ε_f is the specific exergy rate of fuel and calculated by the equation $\varepsilon_f = \mathrm{LHV}\phi$. The chemical exergy factor for liquid fuel calculated by equation below [20, 26]:

$$\phi = 1.0401 + 0.1728\frac{h}{c} + 0.0432\frac{o}{c} + 0.2169\frac{\alpha}{c}\left(1 - 2.0628\frac{h}{c}\right),$$

where h, c, o, and $\alpha$are the mass fractions for H, C, O and S, respectively. The rate of exergy transfers through flue gas which consists of physical and chemical exergy specific is calculated by the equation $\dot{E}x_{exh} = \dot{m}_{exh}\left(\varepsilon_{phi} + \varepsilon_{chem}\right)$.

The physical and chemical exergy specific calculated in two formulas as written below respectively $\varepsilon_{phi} = \left(h - h_0\right) - T_0\left(s - s_0\right)$ and $\varepsilon_{chem} = \overline{R}T_0 \ln\left(y_i / y_0\right)$. Here $\overline{R}$ is the universal gas constant; y_i is the mole fraction of the exhaust gas component; y_0 is the mole fraction of the exhaust gas component at the reference state. Similar to energy efficiency, exergy efficiency is a quantity to express system performance. However, exergy efficiency is more accurate in measuring system performance than the thermal efficiency of the first law of thermodynamics. Exergy efficiency is the ratio between exergy transfer through work and the fuel exergy, or $\eta_{Ex} = \dot{W} / (\dot{E}x_{in})$.

Results and discussions. *Energy and exergy analysis for fuel. Figure 2, a shows the energy quantity obtained from combustion E0 and E100 fuel, while Fig. 2, b shows the fuel exergy of E0 and E100. Energy and exergy of E0 are produced from a compression ratio of 11:1 and fuel ignition of 10 BTDC, while the fuel energy and exergy of E100 are generated from a compression ratio of 13:1 and in varying fuel ignition.*

Figure 2, *a* shows the energy E100 is higher than the energy E0 at all engine speeds and fuel ignition. Furthermore, Fig. 2, *b* shows that the tendency of the fuel exergy of E100, especially at fuel ignition of 22 and 26 BTDC is higher than E0 except at 6000 RPM. It caused by two reasons. First, the heat of vaporization of E100 is three times greater than that of E0, so the E100 requires higher heat in its atomization process. Therefore, the fuel ignition must be advanced so that the ignition delay will be longer, and E100 fuel will receive a higher heat of evaporation to produce a better combustion process. Second, the combustion of E100 was performed at a compression ratio of 13:1 so that the thermal efficiency would be much better than E0 [27, 28].

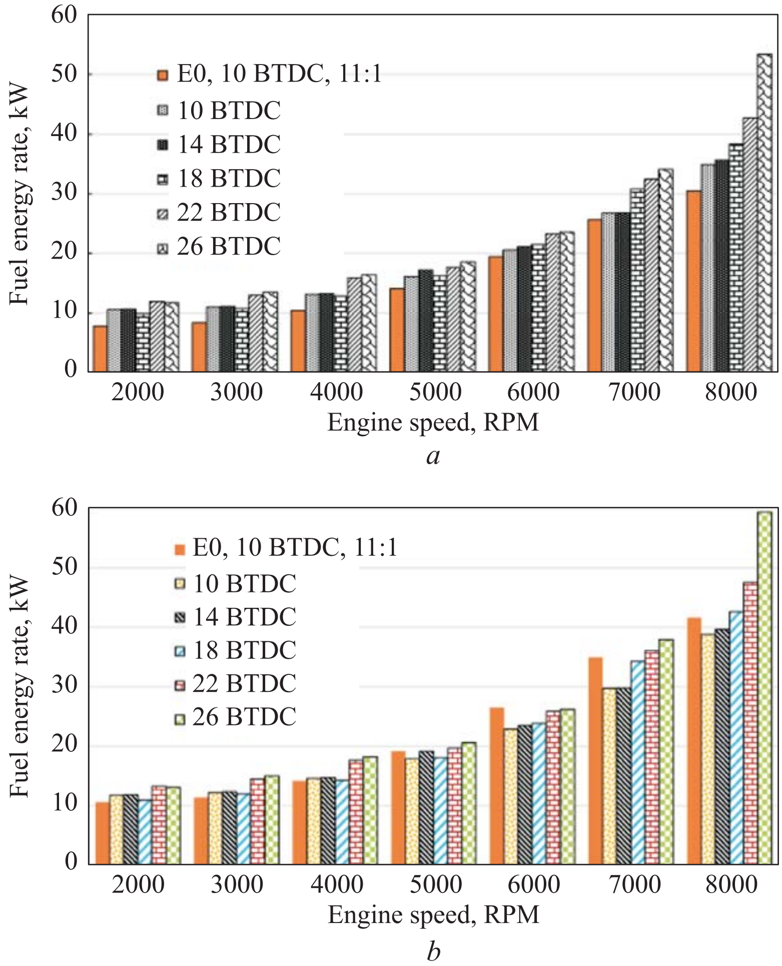


Fig. 2. Fuel energy rate on varying ignition timing and speed engine (*a*) and fuel exergy rate on ignition timing and speed engine variations (*b*)

Whereas Fig. 2, *b* shows the heating value of E0 has a dominant effect on improving fuel exergy of E0, especially at low engine rpm. These results were linear with Bhatti's study [29] that the input exergy by the fuel will be greater than its input energy, namely ≈ 18.0 vs 17.1 kW, and it obtained at CR 7 and 1800 RPM. In this study, the maximum input exergy of fuel is 59.37 kW while 53.39 kW for maximum input energy, and those obtained at the ignition of 26 BTDC and 8000 RPM.

Brake power analysis. Brake power is analyzed based on the energy concept whose value is proportional to the multiplication of engine torque and RPM. Figure 3 shows the brake power of E100 and E0 in the SI-PFI engine. It was clear that the increase of engine RPM will generate a high brake power, either by E100 or E0. This figure also shows that the brake power of the E100 is higher than the E0, especially after the engine speed exceeds 4000 RPM and at 18 BTDC of fuel ignition.

The brake power increases with increasing engine rpm due to the piston movement increased, so the turbulence and homogeneity of the blend to be better. It will lead to a better combustion process and produce high heat energy. Furthermore, the high temperature will generate a high enthalpy and produce a high of work. Meanwhile, the advanced fuel ignition will cause the ignition delay to be longer. It will be advantageous for biofuels to increase their evaporation pressure through heat absorption are larger, so it's the atomization and combustion process easily. Figure 3 shows the maximum engine brake power generated by fuel ignition of 18 BTDC. Meanwhile, the SI engine brake power will decrease if the fuel ignition is too advanced [28].

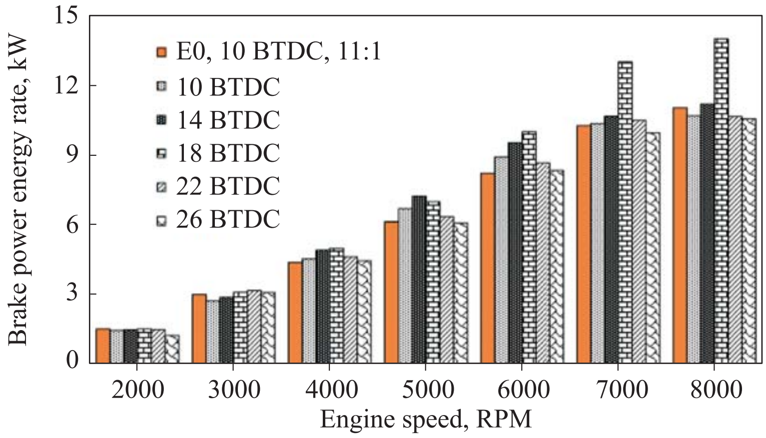


Fig. 3. Brake power energy rate on ignition timing and speed engine variations

Exhaust gas energy and exergy analysis. Figure 4 shows the energy transfer of fuel E100 and E0 to the combustion gas products calculated by energy and exergy analysis. Figure 4 explain that the transfer of the fuel energy to gas increases with increasing engine speed, both E0, and E100 fuels. The ignition effect of E100 fuel on its energy flow shows an increasing trend as the ignition degree and the engine speed increase.

Figure 4 also expressed increasing the compression ratio and advancing the fuel ignition will increase fuel energy conversion of E100 to gas, either in energy or exergy analysis. The figure describes that conversion fuel energy produces exhaust gas energy three times greater than its exhaust exergy. These results indicate that the environmental contribution has a high impact on the amount of useful exergy generated in the thermal systems. This result is in-line with many other researchers, one of which is Kumar Sharma [30]. His research

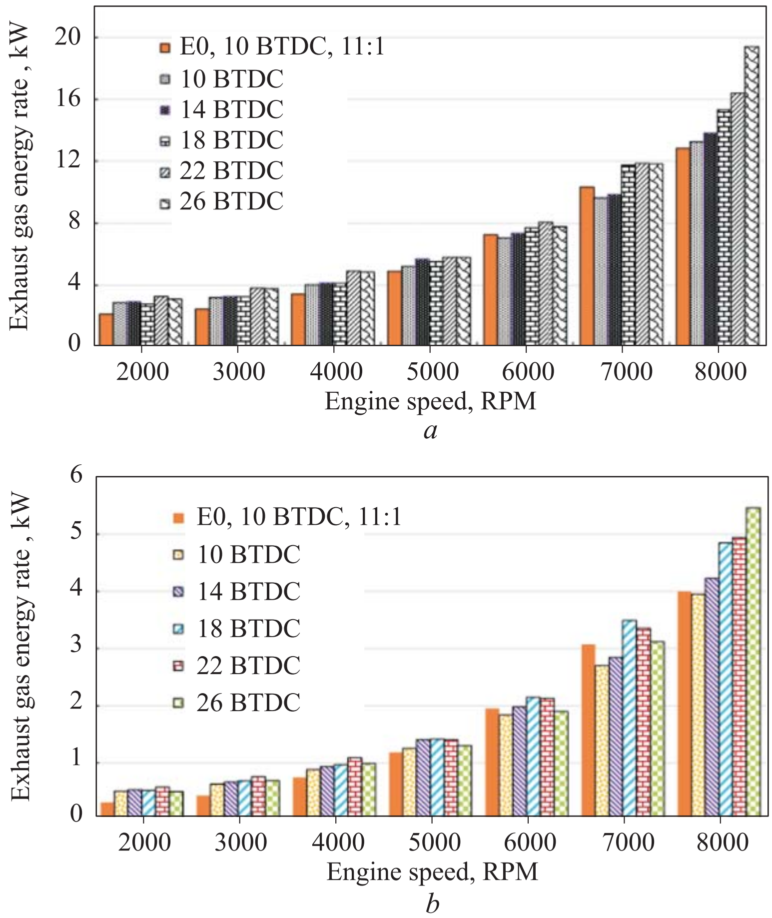


Fig. 4. Exhaust gas energy rate (*a*) and exhaust gas exergy rate (*b*) on ignition timing and speed engine variations

shows that the transfer of exhaust gas energy is three times greater than the transfer of exhaust gas exergy for each mass flow rate of HHO fuel.

Brake specific fuel consumption analysis. The BSFC is the volume rate of fuel required to produce 1 kW of engine power. Figure 5 shows the BSFC decreases with increasing engine RPM and reaches its minimum point at 6000 RPM. However, E0 slightly increased while E100 significantly increased at an engine speed of 7000–8000 RPM. That Figure also shows advancing fuel ignition will decrease the engine's BSFC, but too advanced ignition will increase the engine's BSFC. This study found the minimum BSFC obtained at the ignition of 18 BTDC and engine speed of 6000 RPM. At that point known that usage of E100 requires a 42.6 % higher than E0, which the injection of E0 and E100 fuel is 0.20107 kg/(kW · h) and 0.28669 kg/(kW · h), respectively.

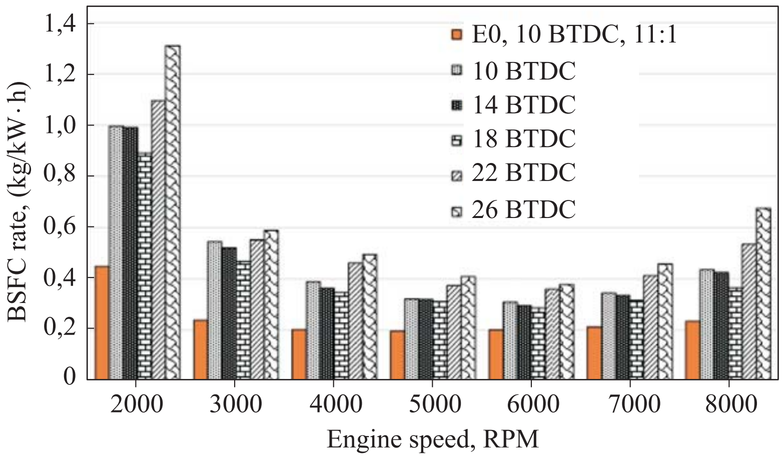


Fig. 5. Specific fuel consumption rate on ignition timing and speed engine variations

Figure 5 concludes that the fuel consumption of the E100 is more wasteful than the E0. this is because the calorific value of E100 is lower than E0 so it takes more fuel injection to produce the same power produced by E0.

Rate of energy loss, heat loss of exergy transfer and exergy destruction analysis. Energy loss is the accumulation of fuel energy minus the number of combustion gases and shaft power (Eq. (5)). Meanwhile, the exergy heat transfer rate is the exergy flow rate that accompanies the heat transfer from the system to the environment (cooling water or engine oil) or calculated by Eq. (7). Figures 6, *a, b* show that the energy loss and heat transfer exergy fluctuate from 2000 to 5000 RPM. However, it will increase consistently once the engine speed is over 5000 RPM. These experimental results confirm research [23] which has the same trend.

Figures 6, *a, b* show the loss of energy and exergy of E0 through heat transfer is lower than the E100. It is because the heat enthalpy produced by E0 lower than E100. It will cause differentiation between the temperature of gas enthalpy of E0 of the system and the environment temperature to be smaller so that the potential energy loss and exergy loss by heat transfer becomes small [31]. Figures 6, *a, b* also show advancing fuel ignition to 18 BTDC results in energy loss and exergy loss by heat transfer lower than the other fuel ignition when E100 is applied on an SI-PFI 150cc engine. These results reinforce the research conducted by Sohret [21] on the effect of ignition timing and compression ratio on the energy rate and exergy of hydrogen-fuelled SI engines.

Figure 6, *c* shows the exergy destruction of SI-PFI engines using E0 and E100 fuels at various fuel ignition. The trends seen in Fig. 6, *c* are similar

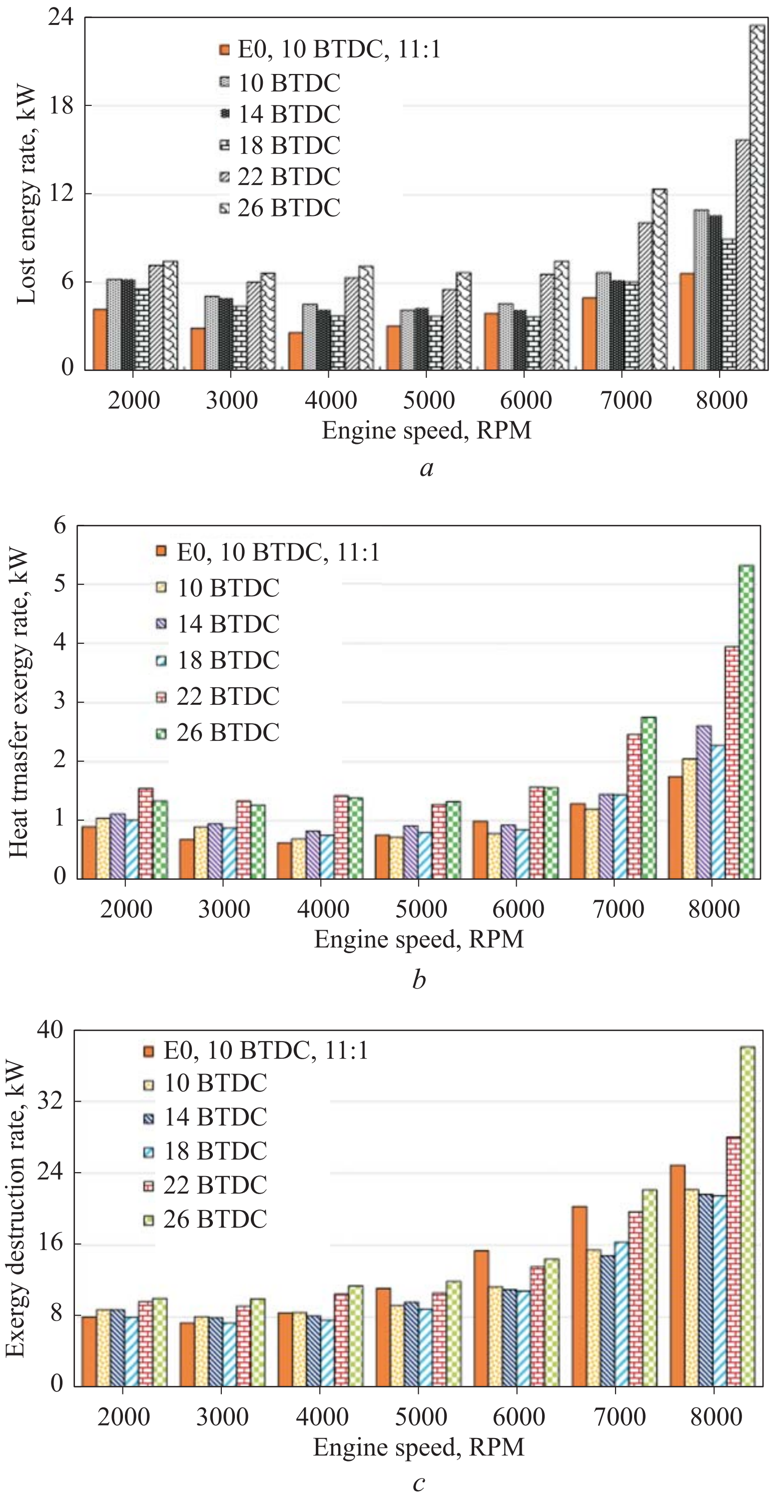
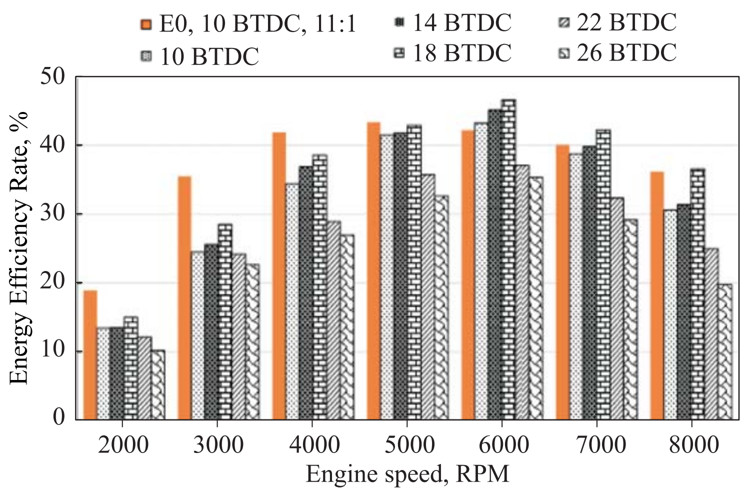


Fig. 6. Loss of energy rate (*a*), heat transfer exergy rate (*b*) and exergy destruction rate (*c*) on ignition timing and speed engine variations

to those shown in fuel exergy (see Fig. 2, *b*). It explained the higher the heat produced in the thermal system, the higher the potential for the system to generate destructive exergy. It is caused by the higher differentiation between the system temperature and the environment temperature so it will generate higher entropy for the irreversibility process. Irreversibility is caused by chemical reactions, heat transfer, mixing processes, friction, etc and whose magnitude is proportional to the exergy destruction [32].

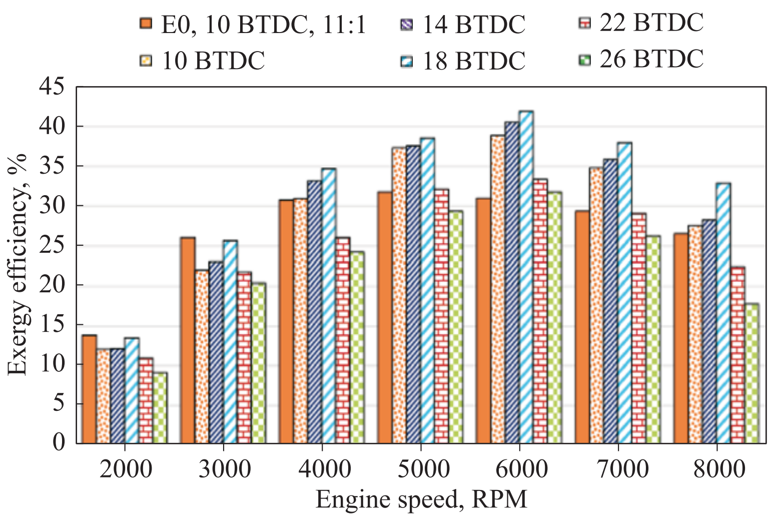
Figure 6, *c* proves that the higher the engine speed will increase cylinder temperature, so it will potentially generate higher exergy destruction, as mentioned by many researchers, and one of which is Yesilyurt [33]. Advancing fuel ignition to 18 BTDC will reduce destructive exergy, but overly advanced fuel ignition will ignite increase exergy destruction. The lowest exergy destruction is 7,348 kW obtained at the ignition of 18 BTDC and engine speed of 3000 RPM, while the largest is 38,041 kW reached at the ignition of 26 BTDC and an engine speed of 8000 RPM for bioethanol fuel.

Energy and exergy efficiency. Figure 7 shows the energy and exergy efficiency of gasoline and bioethanol in the spark-ignition engine. In both pictures show the energy and exergy potential of fuel that is converted into useful work has the same tendency. Both of them will increase as increasing engine speed but decreases over the engine speed of 6000 RPM. The results of data analysis inform the energy and exergy efficiency maximum efficiencies are 46.59 and 41.90 %, respectively. Those are obtained at an engine speed of 6000 RPM and fuel ignition of 18 BTDC for E100, while the energy and exergy maximum of E0 is obtained at an engine speed of 5000 RPM.



a

Fig. 7 (beginning). Energy efficiency rate (*a*) on ignition timing and speed engine variations



b

Fig. 7 (end). Exergy efficiency rate (*b*) on ignition timing and speed engine variations

These results prove that exergy efficiency is always lower than energy efficiency, as shown by the results of several researchers' studies [21, 26]. It is caused exergy process analyzed based on the second law of thermodynamics is an irreversible process in which environment contribution has a high effect on degradation the exergy quality. It is due to the system continuing to produce entropy for the system still in a un-equilibrium state with the environment [25].

Conclusion. This work presents energy and exergy analysis of an SI-PFI engine using gasoline (E0) and bioethanol (E100) as tested fuel in spark timing and engine speed variable. This study shows that the fuel performance of E100 can be better than E0 when the compression ratio increased, and fuel ignition advanced from the standard setting of E0 fuel. Therefore, the most suitable fuel ignition and compression ratio for E100 fuel in the spark-ignition engine are 18 BTDC and 13:1. At this point, the engine was in the best performance for all engine parameters.

The exergy of E0 and E100 is higher than its energy value, as shown in Figs. 1 and 2, *a*. This result has been proven by many kinds of research as mentioned above [17, 19–34]. However, the exergy destruction will be greater than the energy loss, especially in the combustion chamber, so the exhaust gas exergy rate will be lower than the exhaust gas energy rate, as shown in Figs. 3 and 4, *a*.

The heat transfer exergy rate is lower than the loss energy rate indicates that the temperature difference between the combustion product gas and engine cooling oil is lower than the temperature difference between the combustion gas and the environment. This condition will cause the production of entropy in the cooling oil to be lower, and it will reduce the production of exergy destruction.

The maximum energy efficiency of E0 is higher than E100 in the engine speed of 2000–5000 RPM. Otherwise, the other hand, the maximum energy efficiency of E100 will be higher than E0 in the engine speed of 6000–8000 RPM.

Exergy efficiency in Fig. 7, *b* is evidence that it turns out that bioethanol (E100) is more effective in transferring fuel exergy into useful engine power compared to gasoline (E0).

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Please cite this article as:

Paloboran M., Syam H., Yahya M., et al. Energy and exergy analysis on spark ignition engines under varying ignition timing with pure bioethanol fuel. *Herald of the Bauman Moscow State Technical University, Series Natural Sciences*, 2023, no. 2 (107), pp. 140–159. DOI: https://doi.org/10.18698/1812-3368-2023-2-140-159

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