PAPER NAME

6. IREME_VOL_15_N_2_Partial_Version.pdf

AUTHOR

Mar 5

WORD COUNT

9024 Words

CHARACTER COUNT

47125 Characters

PAGE COUNT

13 Pages

FILE SIZE

697.3KB

SUBMISSION DATE

May 5, 2023 9:10 AM GMT+8

REPORT DATE

May 5, 2023 9:11 AM GMT+8

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eISSN 2532-5655
Vol. 15 N. 2
February 2021

International Review of
Mechanical Engineering
(IREME)

Contents

Praise Worthy Prize

International Review of Mechanical Engineering (I.RE.M.E.), Vol. 15, N. 2
ISSN 1970 - 8734 February 2021

Performance Optimization of a Spark Ignition Engine Fueled with Gasoline-Bioethanol (E85) Using RSM and Non-Linear Programming Approach

M. Paloboran, H. A. Gani, Saharuna, M. I. Musa

Abstract – *The aims of this work are to obtain the characteristics and the performance optimization of the spark-ignition engine and single-cylinder by using the gasoline-bioethanol fuel blend in composition 15%-85% (E85). The Response Surface Methodology (RSM) and the Non-Linear Programming are applied in this work in order to find the area and the optimization point of the engine performances. The engine operates on different engine speeds in the range 2000–8000 RPM (increment 1000 RPM), ignition timing in interval 12–28 BTDC (increment 4 BTDC) and compression ratio in range 12–13 (increment 0.5). All the performance engine parameters of E85 fuel are better than gasoline engine performance except for specific fuel consumption and thermal efficiency that is worse than E0. Those results will be obtained when the engine parameters work on the compression ratio, 16-20 BTDC (before top dead center) of ignition timing, and higher than 4000 RPM (revolution per minute). Meanwhile, the optimization of the engine performances has been done by using Box Behnken design of response surface methodology. The methodology shows that the optimal values of the engine performance are obtained for 13:1 of compression ratio (CR), 24 BTDC of Ignition Timing (IT) and 7240 RPM of engine speed. The result of this study has revealed that at optimal parameters the values of the brake power, brake torque, thermal efficiency, mean effective pressure, specific fuel consumption, CO and HC emissions have been 12.68 HP, 12.90 Nm, 32.4%, 1083.8 kPa, 0.004224 kg/hp h, 2.6% and 89.9 ppm respectively.*

***Keywords**: Ethanol-Gasoline Blend, Response Surface Methodology, Spark Ignition, Performance Optimization*

Nomenclature

A	Cylinder area [m^2]
V_s	Swept volume of piston [m^3]
i	Number of cylinder
L	Stroke length of piston [m]
z	Number of crankshaft revolution in one cycle
$\dot{m}_f$	Mass flowrate of fuel [kg/s]
n_R	Number of crank revolutions
Q_{LHV}	Heating value of fuel [MJ/kg]
BHP	Brake horsepower [kW]
BMEP	Brake mean effective pressure [kPa]
BOE	Barrel oil equivalent
BSFC	Brake specific fuel consumption [kg/kWh]
BT	Brake torque [Nm]
BTDC	Before top dead center
BTE	Brake thermal efficiency [%]
CCD	Center composite design
CO	Carbon monoxide [%]
CR	Compression ratio
ECU	Engine control unit
EXP	Experiments
HC	Hydrocarbon [ppm]
IT	Ignition Timing [deg]
LPG	Liquid petroleum gas
PFI	Port Fuel Injection
PM	Particulate Matter
PPM	Part per Million
RON	Research Octane Number
RSM	Response Surface Methodology
UHC	Unburned Hydrocarbon [ppm]
SI	Spark Ignition
SOHC	Single Overhead Camshaft

I. Introduction

The availability and the existence continuity of fossil energy for human activity, especially in the transportation sector, should continue to improve as an anticipation for the changing of the era that is very fast.

Fossil energy reserves continue to decline in the world while energy consumption continues to increase. This is caused by several reasons, i.e. the rapid population growth, the rising income per capita, the economic growth, a surge in motor vehicles, and an increase in the standard of human life [1]-[3]. According to the data of

https://doi.org/10.15866/ireme.v15i2.19798

national energy outlook, the projected energy needs of Indonesia in the period 2016 - 2050 have increased from 795 million BOE to 4569 million BOE [3]. The biggest final energy needs are from fossil oil (40%), electricity (21%), gas (18%), coal (11%) and less than 4% from LPG, biofuel and biomass respectively. Based on the data, it has been predicted that fossil oil will be available only within 10 last years to serve the energy needs of Indonesian people [6]. Therefore, the dependence on fossil fuels as a main energy source should be ended immediately, especially in the field of transportation, not only because the fuel reserves are increasingly restricted, but also because the combustion residual gas is the main contributor to greenhouse gases [3]-[5]. Bioethanol has been a concern for the Indonesian government in order to substitute fuel with gasoline for a long time, because of its abundance raw materials. Through the presidential regulation number 5 of 2006 and number 12 of 2015 it is expected that the ethanol content in the mixed fuels until 2025 will be 20% [6], [7]. There are many alternative and renewable energies. However, bioethanol is considered the most suitable automotive fuel to replace gasoline fuel, because of its chemical and physical characteristics, which are similar to gasoline, making them easy to blend [8], [9]. The use of ethanol in spark ignition of internal combustion engines brings many advantages. The first one is that it reduces incomplete combustion products due to its high oxygen content [10]-[12]. Second, blending with gasoline will increase brake power, brake torque, and thermal efficiency. This happens because ethanol has a high octane number, enthalpy (heat of evaporation), and broader ignition boundaries than gasoline [10]. However, ethanol has a number of disadvantages, namely a lower energy content (the heating value of bioethanol is 34% lower than the caloric value of gasoline), more corrosiveness, low flame luminosity, lower vapor pressure (making cold starts difficult), miscibility with water, and toxicity for ecosystems if compared to the gasoline [10], [11], [13].

Many researchers have examined the effect of the ethanol-gasoline fuels blend on the performance characteristics of gasoline engines. Renzi [14] and Efemwenkiekie [15] have studied the effect of ethanol-gasoline blend in ethanol concentrations of 0%, 50%, and 80% (E0, E50-E80) on the performance characteristics and emissions of the gasoline engine. The result shows that brake power, brake torque, thermal efficiency, exhaust temperature, NOx and UHC have decreased if ethanol concentration has increased. Meanwhile, volumetric efficiency, BSFC, and CO emission have increased when ethanol in the fuel increased. Costa [16] has investigated the comparison of performance and emissions of SI 4 stroke engines using ethanol hydrous fuel and a blend of 78% gasoline and 22% ethanol (E22).

The result shows that torque and BMEP of E22 have been high at the low engine speed. Meanwhile, brake power, brake torque, and BMEP of hydrous ethanol are higher than E22 at high RPM of engine. However, thermal efficiency and specific fuel consumption of hydrous ethanol are higher than E22 throughout all the engine speed range. With regard to the exhaust emissions, hydrous ethanol has reduced CO and HC, but has increased CO_2 and NOx. The effect of ethanol percentage in the gasoline-ethanol blends (E0, E10, E20, E30 and E85) on emissions and performance of the 1.6 L SI-PFI engine has been carried out by Costagliola [17].

The results show that the use of ethanol-gasoline blend significantly reduces Particulate Matter (PM) by 60-90% if compared to gasoline. Likewise, gas species, which are harmful to humans such as benzene and benzoa, decrease by 50% and 70% respectively. Similar studies have been carried out by Phuangtrakul [18] in order to find out the optimal BSFC and BTE values by using E10 up to E100 fuels. The effect of variations of the gasoline-bioethanol blends on the performance and emissions of direct injection of SI engine has been carried out by Turner [19]. E0, E10, E20, E30, E50, and E100 have been used as fuels and ignition timing has been varied in interval 19-34 BTDC in order to observe the characteristic of direct injection engines. The results show that emissions have decreased while cylinder pressure, combustion stability, and engine efficiency have increased when ethanol percentage has increased.

Using variable ignition timing for the investigation of the performance of motorcycle single cylinder has been also supported by Yang Chen [20]. Nowadays, many researchers have studied strategies in order to overcome the deficiencies of bioethanol, and one of them through readjustment on engine parameters and combustion when the ethanol in high percentage is used. As known, ethanol drawbacks begin to appear if the concentration in the gasoline mixture is more than 25% [1]. Celik [21] has studied an appropriate compression ratio of spark engine to blends of gasoline-ethanol of E0, E25, E50, E75 and E100. The results show that a compression ratio of 10:1 is suitable for E50 fuels where engine power increases by 29% while BSFC, CO, CO_2, HC and NOx decrease by 3%, 53%, 10%, 12% and 19% respectively. The effect of the compression ratio on emissions and performance of a spark engine using pure ethanol has also been investigated by Paloboran [22]. The results show that the power has increased by 6% while HC and CO emissions have decreased by an average of 2% and 24% respectively, even though the energy content of E100 is lower than E0. Increasing the compression ratio when using ethanol in a gasoline engine is recommended because of the higher octane of bioethanol with respect to gasoline. This will have an impact on increasing brake power, brake torque, and thermal efficiency and reducing knocking symptoms on the engine. This is due to the more stable combustion process, increased cylinder pressure, and decreased combustion emissions [23]-[25].

The ethanol energy content is lower than gasoline (~29 MJ/kg vs ~44 MJ/kg) so, in order to get the same power as gasoline, it requires a higher injection volume of ethanol than gasoline. Paloboran [26] has investigated a suitable injection volume of pure ethanol fuels at a high compression ratios and its impact on engine performance and emissions characteristics. The results show that the maximum engine power is obtained in the range 150%-175% of the injection volume when E100 is applied on engine. This study is also supported by Celik [21] who has applied ethanol injection volumes in the interval 1.5-1.8 when using E100. Meanwhile, a research has been conducted by Ansari [27] looking for a mixture of gasoline and bioethanol that is suitable for a 4-stroke single-cylinder gasoline engine with a compression ratio of 10:1. The results show that the suitable fuel for the engine is E60. This is indicated by engine performance and emissions that are better than E0, E20, E40, E80 and E100. Evaporation pressure of ethanol is lower than gasoline; latent heat (enthalpy) of ethanol is higher than gasoline, so ignition timing of ethanol is broader than gasoline. Thus, in order to optimize these bioethanol properties, its ignition timing should be more advanced than gasoline ignition timing. Paloboran [22] has conducted a research on the effect of ignition timing on performance and emissions characteristics of a 4-stroke engine of 150cc using E100 fuel. The results show that the optimal engine performance is obtained at 13:1 of the compression ratio, 20 BTDC of ignition timing, and 4000 RPM of engine speed. The variation of ignition timing on emissions and the performance of a 4-cylinder spark engine at a compression ratio of 9.2 have been carried out also by Sakthivel [25] and Turkoz [28]. The results show that the engine performance increases and the emissions decrease if the ignition timing of ethanol is more advanced than gasoline. Performance optimization and combustion emissions of a spark ignition engines have been studied by many researchers. Yusri [29] has investigated the performance characteristics and the emissions of a Mitsubishi 4G93 SOHC engine at a 9.5:1 compression ratio using a blend of butyl alcohol-gasoline and applying the RSM to engine performances optimization. The result shows that engine speed and composition of fuel blend have a significant effect on engine performance and emission characteristics. The result of this study also shows the experiment deviation to the prediction results by using the RSM method just around 5%-10%. A similar method has been used by Najafi [30] in order to examine the effect of the bioethanol-gasoline blend of E5, E7.5, E10, E12.5 and E15 on the performance of the 4-cylinder engine at a compression ratio of 9.7. The results show that the optimum performance and emissions have been obtained on a blend of 10% of bioethanol and 90% of gasoline and 3000 RPM of engine speed. The error rate of prediction in average is less than 3% respect to the actual value.

Those studies have been also supported by Simsek [31], who has studied the performance and the emission characteristic of fusel oil and gasoline blend uses responses surface methodology. This study aims to investigate the effect of a blend of 85% of ethanol and 15% of gasoline on the performance characteristics and combustion emissions of a 1-cylinder spark ignition engine. The RSM used to state the correlation and significance impact of compression ratio, ignition timing, and speed of engine on the performance and emission of E85. The result of the method will be displayed in the regression equation. Meanwhile, the non-linear programming (LINGO) is addressed to predict the combination of compression ratio, ignition timing, and engine speed that produces high performance and low emissions of the combustion of E85 fuel.

The rest of the paper is organized as follows. Section II elaborates materials and experiment tools, research methodology, and basic theory of RSM. Furthermore, Section III will discuss the results of the research and compared to the previous study. Finally, the conclusion of the research will be given in Section IV.

II. Experiments Tools, Methods and Materials

II.1. Test Engine Set-Up

In this study, testing has been carried out using one-cylinder engine, four strokes, fuel injection port that has been designed using gasoline fuel. The schematic diagram of the test engine is showed in Figure 2.

Meanwhile, the main specifications of the test engine are detailed in Table I. This study will employ a portable JUKEN 3 ECU for adjusting the combustion variables such as ignition timing and injection volume. It is improved in order to accommodate the objective of this research. The fuel is E85, which is a blend of 15% of gasoline and 85% of bioethanol based on the volumes with RON of gasoline, and bioethanol that are 94 and 119 respectively. The improvement of the CR has been done by adding a dome on cylinder head.

II.2. Testing Procedure

The rear tire of motorcycle is placed on the dynamometer water brake roller in order to measure torque and RPM of engine. The engine testing condition is fully open throttle. Therefore, the engine speed in RPM (revolution per minute) is set based on the flow rate of water entering into the dyno test engine. Air supply is natural with a fully open throttle position while setting the engine RPM is controlled by the control valve at water pump. The time of fuel consumption of the engine has been recorded for each collecting data session, in which every process takes place after the engine consumes 25 ml of fuel. The combination of engine and the combustion variable as a testing methodology shown in Figure 1, where compression ratio, ignition timing and speed of engine denoted by CR, IT, and RPM respectively. Meanwhile, subscript 1 up to 7 is a variation of the parameter from low to high. Those are 12:1, 12.5:1, and 13:1 for the compression ratio, 12, 16, 20, 24, and 28 of BTDC (Bottom Top Dead Center) for ignition timing, as well as 1000 – 7000 RPM (increment 1000 RPM) for engine speed. The result of each test stage considered as the best performance is based on the maximum brake torque.

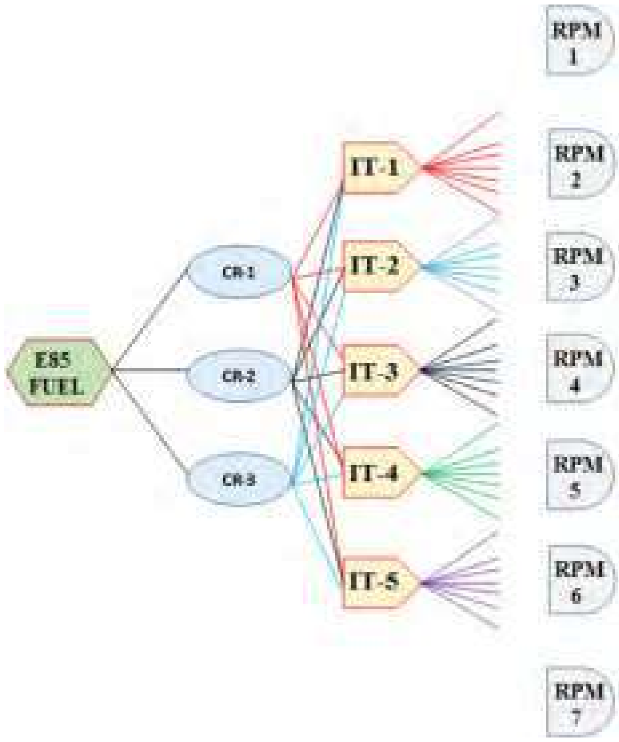

Fig. 1. Experiment methodology

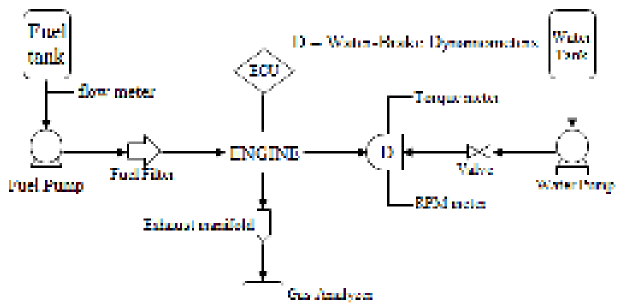

Fig. 2. Line-diagram of the test engine

TABLE I
ENGINE SPECIFICATION

Engine descriptions	
Bore×stroke	63.5 mm×47.2 mm
Displacement volume	149.5 cc
Compression ratio	11.0:1
Ignition system	Full transistorized
Fuel injection system	Electronic fuel injection system
Maximum power	12.5 kW @ 10000 RPM
Maximum torque	13.1 Nm @ 8000 RPM

The range, the accuracy, and the percentage of uncertainties of various apparatuses used in this experiment are provided in Table II.

II.3. Response Surface Methodology (RSM)

Response surface methodology is a mathematical and statistical method used to find the effect of relationships and combinations of several independent variables on response variables and aims to optimize the response variables of a study [32].

TABLE II
LIST OF INSTRUMENTS USED AND MEASUREMENT ACCURACY

Instruments	Measurements domain		Measurements accuracy
Water brakes dynamometer @ DYNO mite Land Sea	Load water: 10 – 20000 HP Engine speed: 0 – 12000 RPM Minimum flow: 10 gallon per minute		± 0.5% ± 1 RPM -
Exhaust gas analyzer @STAR GAS 898	CO	0 – 15% vol	± 0,001
	O_2	0 – 25% vol	± 0,01
	CO_2	0 – 20% vol	± 0,01
	HC	0 – 30000 ppm	± 1
	NOx	0 – 5000 ppm	± 1
Fuel meter	Liquid	0 – 50 mL	± 5%
Thermocouple probe @type K	-	-40 – 1200 °C	± 2.5%

A simple correlation between independent variable and response variable is linear form; it is called as a first-order model and written in equation:

$$y = \beta_0 + \beta_1 X_1 + \ldots + \beta_n X_n + \varepsilon_i \qquad (1)$$

where *I* is response, β is a coefficient, I is an independent variable and it represents the noise or error observed in the response (*I*). The problem solution will use a polynomial degree function if the correlation is in the quadratic form, called a second-order model, that is:

$$y = \beta_0 + \sum_{i=1}^{k} \beta_i X_i + \sum_{i=1}^{k} \beta_{ii} X_i^2 + \ldots + \sum_{i<j}\sum \beta_{ij} X_i X_j + \varepsilon_i \qquad (2)$$

The optimization of response variables has been conducted by applying the RSM method of Minitab software. The software also contains three experimental designs optionally that are factorial design, central composite design, and Behnken Box design, in which they have different characteristics. However, for optimizing response surfaces, it is common to use Center Composite Design (CCD) or Behnken Box design. The differentiation of both of them is in some operator of experiment, which is the CCD little, much more than Box Behnken design if the number of independent variables is equal. Both the designs have been developed by two-level factorial design (2^k). On the CCD design, it is required to add some points of factorial, axial and center point on the response surface, while on the Box Behnken design just need some center point.

Furthermore, the addition of the center point on the CCD method is optional and it is recommended to use 3 to 5 center points, while in the Box design it has defaulted in the 3 points. The addition of the central point is intended to maintain the stability of the variance of the predicted response value [33]. Therefore, the number of operators on the CCD design will be more than 2 to 5 compared to the Box Behnken design. Consequently, this study has used Box Behnken design. The design is simple and the prediction results are also very accurate. In addition, the use of CCD design in this study is not possible because the arrangement of engine components cannot accommodate the rotatable values (α) of the design [34]. Meanwhile, some of the formulas used in calculating the performances of spark ignition engine four-stroke and single-cylinder have been written in Equations (3) to (6).

II.3.1. The Brake Specific Fuel Consumption

The specific fuel consumption in combustion process is defined as a quantity of mass flow rate of the fuel into cylinder in order to generate a power in the one unit of the certain time [35], [36]:

$$BSFC\left[\frac{\text{kg}}{\text{hp h}}\right] = 2684.6\frac{\dot{m}\left[{}^{\text{kg}}\!/\!{}_{\text{s}}\right]}{BHP[\text{kW}]} \tag{3}$$

II.3.2. The Brake Thermal Efficiency

The brake thermal efficiency is defined as a number of work or power that is produced from combustion of every kilogram of energy in the fuel and flowing into the chamber [35], [36]:

$$\eta_{th}[\%] = \frac{BHP[\text{kW}]}{\dot{m}_f\left[{}^{\text{kg}}\!/\!{}_{\text{s}}\right] Q_{LHV}\left[{}^{\text{kJ}}\!/\!{}_{\text{kg}}\right]} \tag{4}$$

where BHP is the brake horsepower (W), $\dot{m}_f$ is the fuel mass flow rate (kg/s), and Q_{LHV} is the energy content of the fuel (MJ/kg).

II.3.3. The Brake Horse-Power

The brake horsepower is an actual power used as the vehicle's traction, and it is expressed in horsepower units. Brake horsepower is usually measured by equipment placed on the engine driveshaft. The tools could measure the forces acting on an engine and calculate the magnitude of the force's value, and various types of dynamometer are used in this work. However, a water brake dynamometer has been used in this study in order to measure the moment force on a vehicle, and then calculate the shaft power with the equation [35], [36]:

$$BHP[\text{kW}] = \frac{2\pi N\left[{}^{\text{rev}}\!/\!{}_{\text{min}}\right] T[\text{N m}]}{60} \times 10^{-3} \tag{5-a}$$

or:

$$BHP[\text{hp}] = \frac{N\left[{}^{\text{rev}}\!/\!{}_{\text{min}}\right] T[\text{lbf ft}]}{5252} \tag{5-b}$$

II.3.4. The Brake Mean Effective Pressure

The brake mean effective pressure of the engine is calculated by Equation (6) [35], [36]:

$$BMEP[\text{kPa}] = \frac{BHP \times z}{A \times i \times L \times N} = \frac{BHP[\text{kW}] \times n_R}{V_s[\text{m}^3] \times N\left[{}^{\text{rev}}\!/\!{}_{\text{s}}\right]} \tag{6}$$

where V_s is the piston swept volume, A is the cylinder area, i is the number of the cylinder, L is the stroke length of piston, z is the number of crank revolution in one cycle, and n_R is the number of crank revolutions for each power stroke per cylinder (2 for four-stroke cycles; 1 for two-stroke cycles).

III. Result and Discussion

III.1. Interactive Effect of Independent Variable on Brake Torque (BT)

Figure 3 shows the combination effect of compression ratio, ignition timing, and engine speed on brake torque both in experiment and optimization result. The graph explains that the brake torque will increase as increasing in the compression ratio when E85 fuel applied in the SI engine.

The figure also shows that the brake torque of E85 fuel is higher than E0 when the engine runs at above 5000 RPM of the engine speed, especially for the 13:1.

Meanwhile, the engine speed that continues to increase causes the brake torque to decrease on the maximum engine speed at all compression ratio and fuel.

In general, the decrease of brake torque at maximum engine speed is caused by several things; one of them is the increased engine load when engine speed increases. In this condition, the mass flow rate of fuel has increased while the mass flow rate of air is un-appropriating. This case will make the blend becomes rich and will cause incomplete combustion, producing low brake torque.

Commonly, this problem could be overcome by adding a turbocharger to the engine. The figure shows clearly that the brake torque has increased when the engine speed has increased. It will cause an increase in in-cylinder temperatures, so the evaporation pressure of ethanol increases [37].

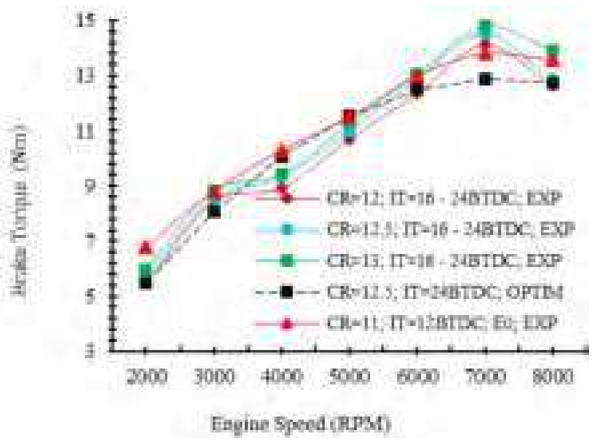

Fig. 3. Characteristics of BT on variation of CR, IT and RPM

The process will have a positive impact on the atomization of fuel, so the homogeneous mixture will be obtained. It has a significant impact on the complete combustion process to reduce emission and increase the performances. However, the low evaporation pressure of ethanol will give a cold effect on the cylinder, so the density of the blend and volumetric efficiency increased. It will generate a high combustion pressure, so the power and the brake torque of the engine increase. Figure 4 is a Contour plot of brake torque that shows the optimization result of the parameters. By the RSM method, the value of the brake torque is around 11.5 N m. It is obtained on 12.5:1, 24 BTDC, and when the engine runs at a constant speed of 5000 RPM. Meanwhile, if the compression ratio has been hold in 12.5:1, the optimum value of parameter is around 12 Nm, and it is obtained on 20-24 BTDC as well as 5000-8000 RPM. By the same method, the optimum value of BT is fixed and it has been around 11 Nm although ignition timing is held on 20 BTDC, 12:1 to 13:1, and 5000-8000 RPM. Meanwhile, by applying the non-linear programming (LINGO), the optimization result of torque is obtained on the 12.5:1, 24 BTDC, and an engine speed of 7240 RPM.

III.2. Interactive Effect of Independent Variable on BSFC

In most of the explanations of bioethanol and gasoline blend research, the heating value of the blend decreases when there is an increase of the bioethanol percentage in the mixture. It is caused by the energy content of ethanol that is lower of 35% than gasoline [38]. Consequently, the volume rate of ethanol required should be bigger than gasoline (1.75-2 that of gasoline) in order to obtain equal power with gasoline fuel if both fuels are applied in the same engine.

Figure 5 shows the effect of engine parameters on the brake specific fuel consumption of spark engine of 150cc. The figure explains that BSFC continues to decrease as engine speed increases. Increasing the engine speed will cause the cylinder temperatures to increase, having an impact on increasing the vapour pressure of bioethanol.

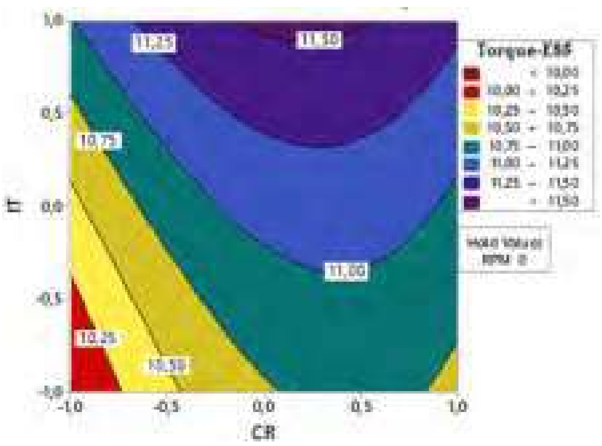

Fig. 4. Contour plot of BT on variation of CR, IT and 5000 RPM

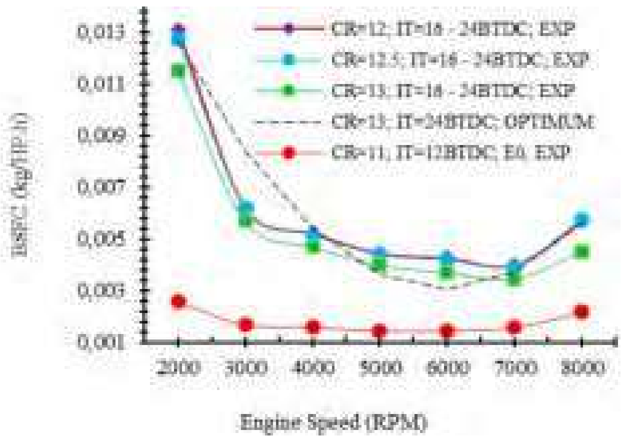

Fig. 5. Characteristics of BSFC on variation of CR, IT and RPM

Thus, the fuel will undergo more easily the atomization process so that it will burn completely [39].

Figure 5 also shows that the BSFC will be decreased when the compression ratio has increased. This is because the combustion chamber becomes smaller so that the turbulence of the flow increases and the mixture becomes more homogeneous. The states will cause the use of fuel to be more efficient so that the BSFC decreases. The BSFC value also tends to decrease if the fuel ignition is increased from 16 to 24 BTDC. This is because bioethanol has a high heat latent vaporization, so it requires more advanced ignition time than gasoline.

Figure 6 is a contour plot for the optimization area of the BSFC obtained by the RSM method. The optimum BSFC is around 0.004 kg/hp h and it will be obtained on 12.5:1 to 13:1 of CR, and 22-24 BTDC when the engine runs on 5000 RPM. The same optimum value is also obtained if the compression ratio is concentrated at 12.5:1, engine speed at 5000-6500 RPM and fuel ignition at 22-24 BTDC. The same trend will be obtained if fuel ignition is kept constant at 20 btdc, 13:1 of CR, and 5000-6500 RPM. The application of non-linear programming as a complement of the RSM method is used to predict the optimization value of BSFC. The software detects that the optimization value of BSFC is 0.0031 kg/hp h, and it is obtained on 13:1 of compression ratio, 24 BTDC of fuel ignition, and 5960 RPM of engine speed for a single response.

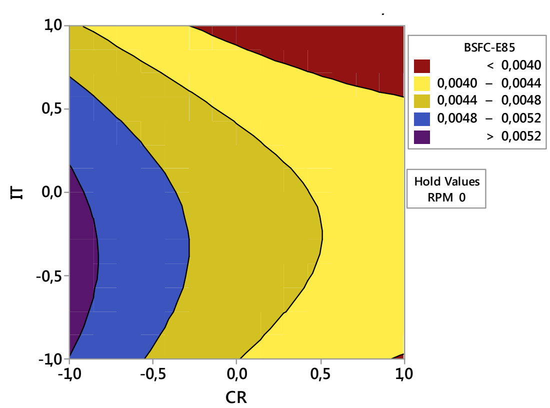

Fig. 6. Contour plot of BSFC on variation of CR, IT and 5000 RPM

However, the value will increase to 0.004224 kg/hp h if the engine parameters are referred to the optimum value of brake torque for the multi-response stage.

III.3. Interactive Effect of Independent Variable on BTE

Equation (4) has expressed that the brake thermal efficiency has a linear correlation to the power, and it is inversely proportional to the fuel mass flow rate.

Equation (4) also describes that the amount of work generated in a combustion process is dependent on the heating value and mass flow rate of fuel. The high heating value of fuel has a high potentiality to produce a high power, and it is transferred to be useful by the engine [39]. Figure 7 describes the effect of engine parameters on brake thermal efficiency. The figure shows that the BTE increases consistently with an increase in engine speed. This happens because the cylinder will produce high temperatures when the engine runs in a high-speed, so the evaporation pressure of ethanol increases and the atomization process is better. It will make the volatility of ethanol increase resulting in a combustion process that is more complete. The same process will occur if the compression ratio is increased because the combustion chamber is narrowing so that the potential energy in fuel is converted into heat energy [40]. The BTE will also increase if fuel ignition is increased from 16 to 24 BTDC, but it will decrease if ignition is continuously increased at all the compression ratios and engine speeds. Figure 8 shows a contour plot of BTE, which describes the optimum value of BTE that is resulted from the RSM method. When the engine runs on 5000 RPM, the optimum value is obtained on 16-20 BTDC of fuel ignition and 12:1 of compression ratio. Meanwhile, if the compression ratio is kept on 12.5:1, the optimum value is obtained on 24 BTDC and 5000-6500 RPM. The same tendency will occur even if the fuel ignition is constant on 20 BTDC. The non-linear programming has been found if the optimum result of BTE is around 39.06% on 13:1 of CR, 24 BTDC, and 5705 RPM for the single response. Meanwhile, the optimum value of the BTE is 32,36% for the multi-response model if the code of engine parameter is referred to the optimum torque.

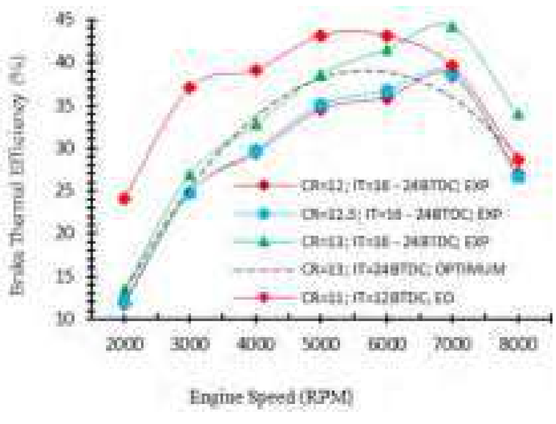

Fig. 7. Characteristics of BTE on variation of CR, IT and RPM

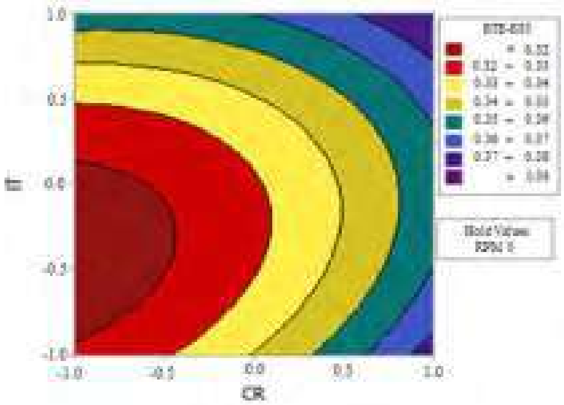

Fig. 8. Contour plot of BTE on variation of CR, IT and 5000 RPM

III.4. Interactive Effect of Independent Variable on CO Emission

Bioethanol's oxygen content is 35% greater than gasoline, so that the AFR of ethanol is lower than gasoline (~8.9 vs ~15.1). As an oxygenate fuel, bioethanol will produce lower hydrocarbon emissions than gasoline [41]. Figure 9 shows the impact of compression ratio, ignition timing, and engine speed on CO emissions of E85. The experiment result shows that carbon monoxide emissions tend to increase consistently as an increase in engine speed. It is an indication that some of the fuels do not burn completely, which can be caused by the lack of air supplied into the combustion chamber while the fuel increases with increasing engine speed. The problem in this sub-section could be overcome by employing a turbocharger on the engine when the engine runs on high load and high speed [39].

By increasing the compression ratio from 12:1 to 13:1, the carbon monoxide decreases gradually even at all fuel ignition. This happens likewise if fuel ignition is advanced from 16 to 24 BTDC for all compression ratios. Figure 10 shows a contour plot of CO emissions that is obtained from RSM method application. The figure represents an optimization area of CO emission when the engine runs on 20-24 BTDC of fuel ignition and 13:1 of CR while engine speed constant at 5000 RPM.

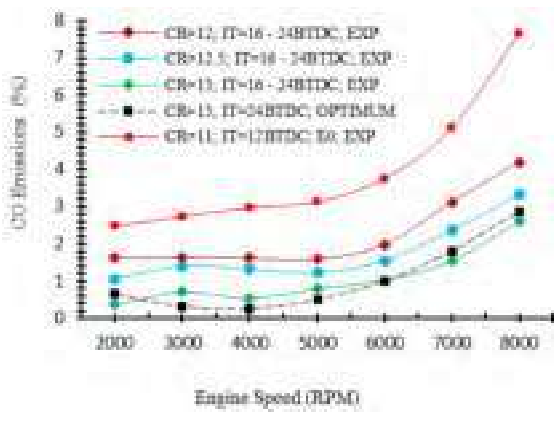

Fig. 9. Characteristics of CO on variation of CR, IT and RPM

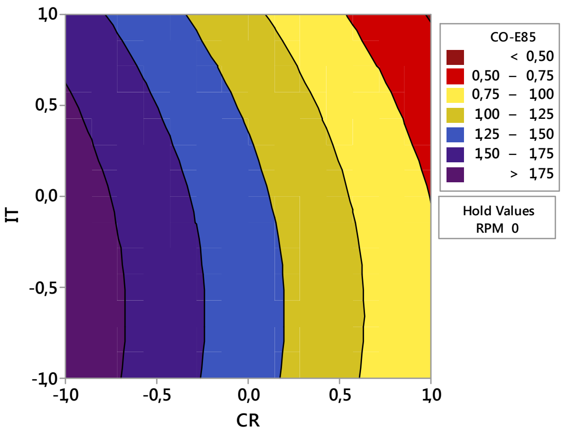

Fig. 10. Contour plot of CO on variation of CR, IT and 5000 RPM

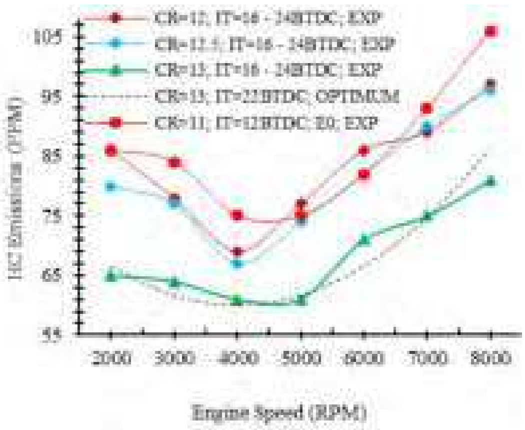

Fig. 11. Characteristics of HC on variation of CR, IT and RPM

Furthermore, if the compression ratio is kept on 12.5:1, the optimum area will move to 16-24 BTDC and 2000-4000 RPM. Meanwhile, if the fuel ignition is constant on 20 BTDC, the optimum value is obtained on 12.5:1 and 2000-4000 RPM. The application of non-linear programming has found out that the optimum point of carbon monoxide emission will be obtained around 0.484% when the engine parameters run at 13:1, 24 BTDC, and 5000 RPM. However, the optimum value of CO emission will increase to 2.55% when the engine parameters run at the multi-response mode with refers to the optimum value of the torque.

III.5. Interactive Effect of Independent Variable on HC Emission

The high hydrocarbon emissions are not only caused by the lack of air into the cylinder when the combustion process happened. They can also be caused by the interference or breakdown of the supporting components such as in the ignition systems, the wiring system, spark plug, coil, etc. Those problems have an impact on electric currents that are not properly in the complete combustion process. Moreover, too advanced or retarded fuel ignition could make the hydrocarbon fuel on incompletely combustion, so the engine produces high hydrocarbon emissions [41]-[43]. Figure 11 expresses the characteristic of HC emission on variation of CR, IT, and engine speed either in the experiment and optimization method.

The HC emissions tend to decrease at the initial engine speed and continue to increase starting from 4000 RPM to the top of RPM for the entire compression ratio mode. Meanwhile, the trend of HC will also decrease when the fuel ignition is advanced from 16-24 BTDC.

By the experiments, the minimum value of HC is obtained on 13:1, 24 BTDC, and 4000 RPM. Figure 12 shows a contour plot of HC obtained from RSM method.

The figure shows an optimum value of HC when the engine runs on 16-24 BTDC and 13:1 of CR while speed constant at 5000 RPM. Furthermore, if the compression ratio is held on 12.5:1, the optimum area will be obtained on 16-24 BTDC and 3500-5000 RPM.

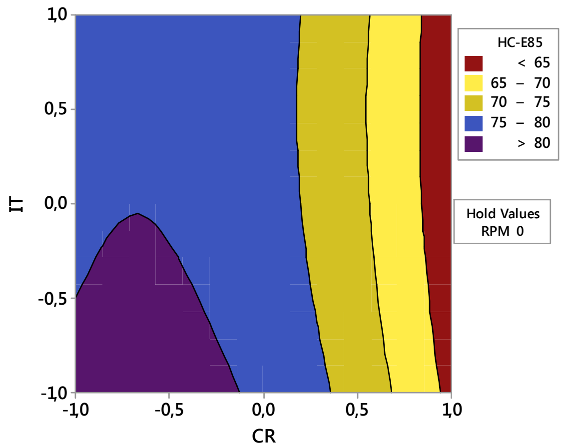

Fig. 12. Contour plot of HC on variation of CR, IT and 5000 RPM

Meanwhile, if the fuel ignition is constant on 20 BTDC, the optimum value is reached on 13:1 and 3500-5000 RPM. By referring to the non-linear analysis, the optimal value of HC is around 61.5 ppm. It is obtained at 13:1, 22 BTDC, and 5000 RPM. However, in the multi-response option that refers to optimum torque, the HC emission increases to 98.94 ppm.

III.6. Interactive Effect of Independent Variable on BHP

Figure 13 shows a graph of the effect of compression ratio, ignition, and variation of engine speed on the brake horsepower. The graph explains that the BHP of the engine increases slowly when the engine speed increases gradually. It is caused by the high cylinder temperature produced from the high speed of the engine, so the combustion process results in high power because the ethanol has a high heat latent of vaporization [44]. The figure shows that the BHP of the engine will experience an increase when the high compression ratio is applied, also if the fuel ignition is more advanced. The optimum result of BHP in experiments section is obtained on 12.5:1, 24 BTDC, and 7000 RPM. Meanwhile, Figure 14 shows the optimal result of BHP through interaction of engine parameters by using RSM method.

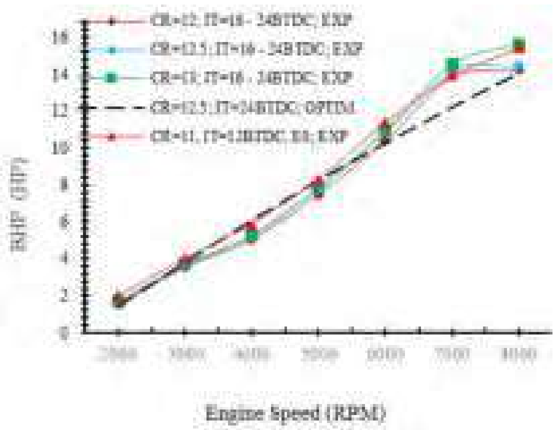

Fig. 13. Characteristics of BHP on variation of CR, IT and RPM

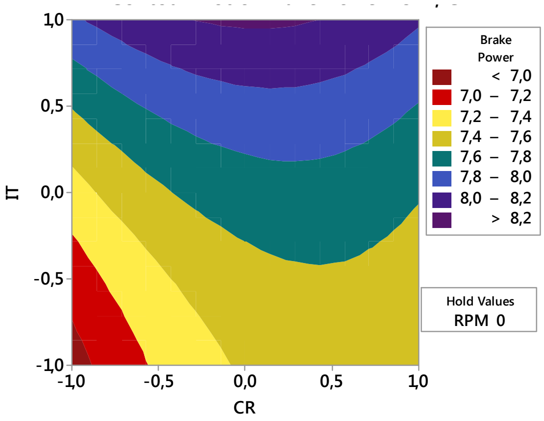

Fig. 14. Contour plot of BHP on variation of CR, IT and 5000 RPM

The optimum value will be obtained on 12-13 of CR and 24 BTDC when the engine works at 5000 RPM.

Moreover, optimum BHP is obtained on 8000 RPM and 20 to 24 BTDC when the compression ratio is fixed on 12.5:1. Furthermore, the optimum BHP will be obtained at all the engine compression ratios that are applied if the bioethanol is burned at 20 BTDC and the engine speed is 8000 RPM. Meanwhile, in the single stage response, the optimum BHP is around 14.06 HP on 12.5:1, 24 BTDC, and 8000 RPM. However, in the multi-response stage that refers to the optimum torque, the optimum value of BHP is 12.68 HP.

III.7. Interactive Effect of Independent Variable on BMEP

Brake Mean Effective Pressure (BMEP) is defined as the average pressure in the cylinder used to push the piston along the volume of piston stroke for producing the mechanical energy in each cycle. Figure 15 shows the influence of engine parameters on the BMEP of the SI engine both on the experiment and optimization result.

Figure 15 has a direct relationship to Figure 3 and Figure 13, since they have the same tendency. It is because the BMEP is a representation of the force moment required by the engine to produce power in each piston stroke, as revealed in Equation (6).

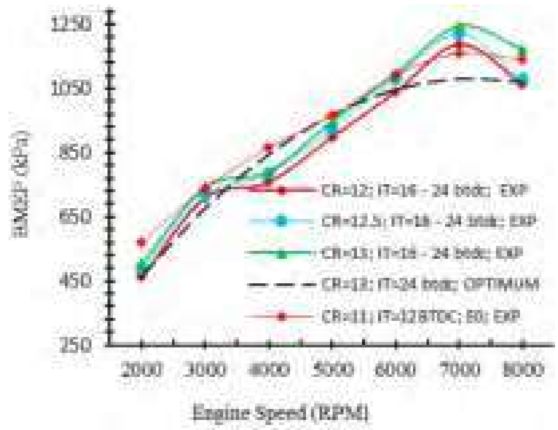

Fig. 15. Characteristics of BMEPon variation of CR, IT and RPM

The equation shows that the correlation between the BMEP, the BHP, and the torque is in a linear relation, so the effect of the independent variable on BMEP has a similar effect with BHP and torque. Furthermore, Figure 16 shows the contour plot of BMEP as an optimization response on the variation of CR and IT, while speed is constant at 5000 RPM. The optimum value is obtained on 12.5-13 of CR and 24 BTDC. Thus, the optimum value of BMEP will be obtained on 12.5 of CR and 6500 RPM if the fuel ignition is fixed at 20 BTDC. The equivalent value will be obtained on 20-24 BTDC and 6000-8000 RPM while compression ratio is constant at 12.5:1. Moreover, by using the program, the optimum BMEP in this study is around 1083.8 kPa and it will obtained at 12.5:1, 24 BTDC, and 7270 RPM for a single stage response and multi stage response.

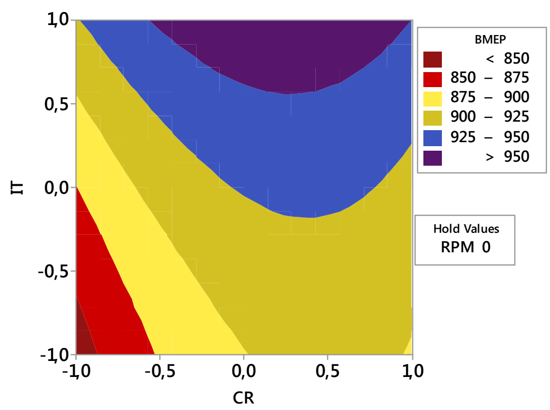

Fig. 16. Contour plot of BMEP vs CR, IT and 5000 RPM

IV. Conclusion

This study concludes several things and gives new information related to bioethanol research and its blend.

The conclusion is based on the experimentation and optimization result.

It has been found out that the RSM method increases the compression ratio gradually from 12 up to 13 (in an interval of 0.5) on the one-cylinder spark ignition engine with E85 fuelled and has not significantly impacted to increase the engine performance, except reducing the emissions of CO and HC.

However, through the result of the experiments, increasing the compression ratio will result in higher engine performance and lower hydrocarbon emissions than the compression ratio of 11 of gasoline fuel. It is recommended that the compression ratio variation interval should be greater than 0.5 in order to get a significant change in the engine performance when the optimization process uses statistical analysis of the RSM method.

Meanwhile, advancing the ignition timing of E85 fuel than ignition timing of gasoline (E0) is proven to improve engine performance and reduce combustion emissions in a spark-ignition engine. In this work, the ignition timing of E85 that gives optimal performance is 24 BTDC. It is 12 BTDC more advanced than the ignition timing of gasoline (E0). Low vapor pressure is one disadvantage of bioethanol when the fuel is applied in the spark engine, particularly in the cold weather. It is caused by the engine that is difficult to start at low temperatures because the atomization process did not properly. This condition also is seen in an experiment in which the engine performance decreases and the emissions increase when the engine runs at low speed.

Conversely, the engine performance increases and the combustion emissions decrease when the engine speed increases. It is because the cylinder temperatures increase when the engine speed increases, so the atomization process of fuel becomes better. Using the Response Surface Methodology (RSM) with the Behnken Box approach in planning an experimental design allows the experiments to be done with high quality and accuracy. It is because the response validation process is carried out in a rigorous and tiered way with statistical tests in several stages with a confidence level of 95%. The use of method also can be seen in the area of response optimization from various interactions of independent variables of research. By applying the software of non-linear programming (LINGO), the optimization result of engine performance in this study is obtained at 24 BTDC of ignition timing, 13:1 of the compression ratio, and 7240 RPM of the engine speed.

The optimization result refers to the maximum torque that is 12.9 Nm. Moreover, the other variable engine performances from that point are 0.004223 kg/hp h of BSFC, 32.36% of BTE, 2.55% of CO emissions, 89.94 ppm of HC emissions, 12.66 Hp of BHP, and 1083.8 kPa of BMEP.

For the research development, the results of this study will be tested on machines with larger capacities, such as on vehicle engines. Furthermore, the compression ratio will be reset to a rather wide range (with a difference of ≥1) so that the differentiation of the result in a variety of compression ratios more clear. The energy and exergy flow rate in this experiment will be the next research. It is required to make the process better in reducing the loss of energy, as the author has done in energy generation at PLTU Tello Makassar [45].

Acknowledgements

This research supported by the Engineering Faculty of Universitas Negeri Makassar, Indonesia through a funding non-tax state revenue in 2019, with grant number: 319/UN36.9/PL/2019.

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doi: https://doi.org/10.15866/iree.v15i5.17956

Authors' information

Department of Automotive Engineering Education, Engineering Faculty, Universitas Negeri Makassar, Jalan Dg Tata Raya Makassar, South Sulawesi, Indonesia 90224.

Dr. **Marthen Paloboran**, ST, MT (Corresponding Author), Assistant Professor & Lecturer at Universitas Negeri Makassar. He holds his doctoral from Sepuluh Nopember Institute of Technology (ITS)-Surabaya since 2018 in energy conversion engineering. Interest research in alternative energy, biofuel, exergy analysis, and combustion technology in ICE.

E-mail: marthen.paloboran@unm.ac.id

Prof. Dr. **Hamsu A. Gani**, M. Pd, Senior Professor & Post Graduate Director at Universitas Negeri Makassar. He is a doctoral graduate from the State University of Malang (UM), East Java-Indonesia. His research focussed on education technology, light vehicle technology, vocational education, and instructional media technology.

Saharuna, S. Pd, M. Pd, Assistant Professor & Head of Automotive Laboratory at Universitas Negeri Makassar. He is a doctoral candidate on Universitas Negeri Makassar. His research mainly related to improving performance and repair gasoline and diesel engine, as well as instrumentation of equipment's automotive.

Muhammad Iskandar Musa, S. Pd, MT, is an Instructor & Lecturer at Universitas Negeri Makassar. He is a graduate master's in mechanical engineering from Hasanuddin University, Makassar. His research focussed on hybrid vehicles, electric vehicles, and improving performance on gasoline and diesel engine.

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