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INFLUENCE OF THE COMPRESSION RATIO AND IGNITION TIMING ON SINJAI ENGINE PERFORMANCE WITH 50% BIOETHANOL-GASOLINE BLENDED FUEL

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ABSTRACT

Influence of the compression ratio and injection timing on Sinjai engine performance with 50% bioethanol-gasoline blended fuel were investigated on water brake dynamometer. The properties of bioethanol were measured based on American Society for Testing Materials (ASTM) standards. Fuel consumption was measured by the time of fuel consumption per 25 cc of fuel in a measuring glass, whereas combustion air consumption was measured using an air flow meter. The emission parameters, exhaust gas temperature and air fuel ratio were measured using STARGAS exhaust gas analyzer. The compression ratio was increased from 9, 6; 10, 6 to11, 6 dan ignition timing was set for minimum advanced of spark ignition for best torque, MBT with limited by knocking. Ignition timing was adjusted for maximum torque and thermal efficiency. Engine performance testing conducted in a variable speed, starting the engine speed 2000 to 5000 rpm with intervals of an increase of 500 rpm. The engine performance parameters evaluated were torque, brake mean effective pressure, power, specific fuel consumption, thermal efficiency, exhaust gas composition and volumetric efficiency. The results showed that the increase of compression ratios improved engine performance for 50% bioethanol-gasoline blended (E50) fuel throughout all the speed range investigated. The both fuel fuel has same the tendency that the degree of the ignition timing is more advanced due to higher engine speed but the value of degree of ignition timing advanced, from 18 ^{0}BTDC at engine speed 2000 rpm until 26 ^{0}BTDC at engine speed 5000 rpm. The addition of compression ratio requiring retarded ignition timing to avoid detonation. The use of E50 fuel, at the compression ratio is 11.6 can improve brake torque, power and mep respectively by 3, 68%, 4.58% and 3.68% as compared to using pure gasoline at a compression ratio of 9, 6. While the influence of adding compression ratio at the E50 can reduce bsfc by 13, 42 % and increase thermal efficiency by 14, 67 %.

Keywords: compression ratio, ignition timing, bioethanol, gasololine, engine performance.

INTRODUCTION

Developing renewable energy has become an important part of worldwide energy policy to reduce green house gas emissions caused by fossil fuels. Renewable alternative fuels, as defined by the Energy Policy ACT of 1992 (EPACT), include alcohols (biomethanol, bioethanol and others), natural gas, liquified petroleum gas, hydrogen, coal derived liquid fuels, biofuels (include biodiesel, syntesis gas and others), and electricity. These fuels are being used wordwide in a variety of stasioner and mobile applications [1]. They are a major force in the effort to reduce fossil fuels consumption and environmental impact (include harmful pollutans and exhaust emissions) in the transportation sectors.

Bioethanol is sustainable fuel that it can be produced from suger cane or other renewable energy sources so it employed most widely. Besides, bioethanol is made up of a group of chemical compounds whose molecules contain a hydroxyl group (OH) that bonded to a carbon atom. The oxygen content of bioethanol fuel favors the further combustion of gasoline [2]. Also, bioethanol is used to increase octane number of gasoline. It can be concluded that using bioethanol–gasoline blended fuels can reduced air pollution and the depletion of gasoline fuels simultaneously. Many researcher were studying the effect of bioethanol fuel on the performance and pollutant emissions of an engine by experimentally [3-6] and artificial neural network [7].

According to these issues, the development of new clean-gasoline engines, such as injection system fueled with bioethanol fuel is important. Because it has the advantages of higher thermal efficiency and power output due to fuel injection than conventional engines [8]. The injection system gasoline engine has also a better transient response, more precise control of the air–fuel ratio, an improvement of fuel economy and a reduction of exhaust emissions. Moreover, the higher compression ratio due to the reduced possibility of knocking, leads to an improvement of the output performance by using gasoline bioetanol blended fuel [9].

Biethanol has a heating value about 60% lower and higher research octane number than gasoline fuel. These parameters allow higher compression ratio, higher boost in turbocharged engine and require synchronization between the timing of injection and ignition. For the purpose of synchronizing the time of injection and ignition then need a programmable ECU to adjust the degree of suitability of the injection and ignition. Mapping the ignition timing to optimize performance and emissions of port injection gasoline engine has been done by Sudarmanta and Baniantoro [10]. In the port injection system, fuel is injected into the intake port of each cylinder, and there is an associated time lag between the injection event and the induction of the fuel and air into the cylinder. During cranking and cold starting, a transient film of liquid fuel forms in the intake valve area of the port. This causes a fuel delivery delay and an associated inherent metering error due to partial vaporization, making it necessary to supply amounts of fuel that significantly exceed that required for the ideal stoichiometric ratio. This puddling and time lag may cause the engine to either misfire or experience a partial burn on the first 4-10 cycles, with an associated significant increase in the UBHC emissions [11].

The purpose of the present study is to investigate the influence of the compression ratio and ignition timing on Sinjai engine performance with 50% bioethanol–gasoline blended fuel, furthermore reffered to as E50. This engine is equipped with a programmable engine control unit (ECU) which has the facility to adjust the suitability of injection and ignition timings. ECU function is to control the quantity of fuel, injection timing, ignition timing and engine speed by receiving signals from six sensors. These sensors are oxygen sensor, manifold air pressure sensor, intake air temperature sensor, throttle position sensor, cooled water temperature sensor and engine speed sensor. A multi port fuel injection system is used to inject the fuel into intake valve area of the port to the combustion chamber.

LITERATURE REVIEW

According to Jeuland *et al* [12], some interesting properties of bioethanol to be used as a fuel for spark-ignition engines:

- A very high octane number, which induces a strong resistance to knock and consequently the ability to optimize the engine (compression ratio, spark-advance).
- A density close to the gasoline one.
- The presence of oxygen in the formula, which can provide a more homogeneous fuel/air mixing and consequently a decrease in unburned or partially burned molecule emissions (HC and CO).
- A high latent heat of vaporization enabling a “cooling
- effect” of air and consequently can enhance the filling
- efficiency.

On the opposite, some disadvantages have to be considered:

- The oxygen included in the molecule (30%wt) induces an increase in the fuel volumetric consumption.
- The high latent heat of vaporization can induce running
- difficulties in cold conditions, especially cold start.
- Ethanol leads to azeotropes with light hydrocarbon fractions and can lead to volatility issues.
- Ethanol is miscible with water, which can cause demixing issues when blended with hydrocarbons.
- The high oxygen content of ethanol and its ability to oxidize into acetic acid induce compatibility issues with some materials used in the engine, such as metals or polymers.
- Ethanol combustion in engines induces aldehydes emissions, which can have a negative impact on health

Ethanol and water are miscible in any proportions, while hydrocarbons and ethanol are not miscible. With conventional gasolines containing hydrocarbons, or ethers, the presence of water in the fuel is not a serious concern. In fact, up to 50 ppm at ambient temperature (of course this value depends of the gasoline chemical composition), water remains completely soluble [12]. Over this level, water separates without affecting the hydrocarbon portion and the water layer can be extracted if necessary.

At the internal combustion engine, the improvement of mean effective pressure with increasing compression ratio is a consequence of increased peak cylinder pressure. Pressure increase is followed by combustion temperature increase, requiring retarded ignition timing to avoid detonation. High compression ratio allows for improved fuel conversion efficiency, as the engine thermal efficiency is increased. Therefore, specific fuel consumption is reduced [9; 13].

Ignition timing is very important operating parameter that affects spark ignition engine performance and efficiency. and not much previous work was done on hydrogen–ethanol dual fuel engine, thus this study concentrates on investigation of ignition timings on performance characteristics of engine fuelled with hydrogen–ethanol mixtures and in determining the optimum value of spark timing for maximum BTE and minimum bsfc

METHODS

In this study, the experiments were performed by SINJAI engine 20 kW, a four stroke two cylinder with spark plug ignition. The engine specifications are given in Table-1.

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Table-1. SINJAI engine specifications.

Engine type	SINJAI 20
Number of cylinder	2
Bore x stroke	76 x 71 mm
Displacement volume	650 cc
Compression ratio	9
Control system	Programmable ECU
Fuel intake system	Multi port injection
Maximum torque	57 Nm / 3000 rpm
Maximum power	20 kW/4500 rpm
Coolant system	Liquid with radiator

Waterbrake dynamometer with power capacity 120 hp used in these experiments. The fuel consumption was measured by the time fuel consumption per 25 cc of fuel in a measuring glass, whereas combustion air consumption was measured using an air flow meter. The emission parameters, exhaust gas temperature and air fuel ratio were measured using STARGAS exhaust gas analyzer. The fuel measuring glass was fitted to Sinjai engine and it contained E50 fuel. A schematic diagram of experimental setup is shown in Figure-1.

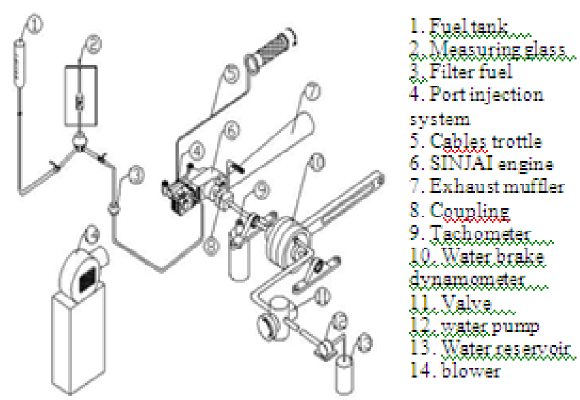

Figure-1. Schematic diagram of the experimental setup.

In order to perform the tests E50 and gasoline fuel were used and three compression ratios were used, 9,6; 10,6 and 11,6. Each configuration used flat surface pistons. The cylinder head was milled to obtain the desired compression ratio. The combustion chamber volume for each compression ratio was verified by filling it with a liquid through a burette of resolution 0.1 ml, with access through the spark plug orifice.

This engine is equipped with a programmable electronic control unit which has the facility to adjust the suitability of injection and ignition timings. ECU function is to control the quantity of fuel, injection timing, ignition timing and engine speed by receiving signals from six sensors. These sensors are oxygen sensor, manifold air pressure sensor, intake air temperature sensor, throttle position sensor, cooled water temperature sensor and engine speed sensor. A multi port fuel injection system with is used to inject the fuel into intake valve area of the port to the combustion chamber.

ECU engine was employed to optimize air fuel ratio, equivalence ratio and ignition timing for all engine configurations and fuel types. For all conditions fuel/air mixture equivalence ratio was set to 1 ~1,1, to minimize the fuel amount necessary to obtain the maximum torque[14;15]. The ignition timing was set for minimum advance for best torque (MBT) or limited by knock whichever occurred first.

A series of experiments were carried out using gasoline, and continued with the E50. All fuels were tested with variable engine speed method. The engine was started using gasoline fuel and it was operated until it reached the steady state condition. The engine speed, fuel consumption, waterbrake load engine, emission parameters such as CO, HC, CO_2, O_2 and exhaust temperature were measured, while the brake power, brake specific fuel consumption, brake mean effective pressure and brake thermal efficiency were computed. All experiments have been carried out at full open throttle setting.

Table-2. The properties of fuels.

No	Properties	Gasoline	E50	Bioethanol
1	Research Octane Number, RON	88	96	105
2	Density, kg/m^3	0,76	0,79	0,81
3	Latent heat of vaporization, kJ/kg	289	644	854
4	Lower Heating Value, kJ/kg	42690	34600	26805
5	Laminer Flame Speed, cm/s	43	41	39
6	Molar Weight, kg/kmol	102,5	74	46,07
7	Air fuel Ratio Stoikiometry	14,7	12,4	9

RESULTS AND DISCUSSIONS

The engine performance tests were carried out to study the influence of compression ratio and ignition timing on the performance parameters of the engine. These tests were conducted with E50 fuel at three different compression ratios of 9, 6; 10, 6 and 10, 6 and varied of ignition timing advanced until 28^0 btdc. Full open trottle test method was conducted variable engine speed test runs from 2000 rpm to 5000 rpm, in 500 rpm engine speed intervals with adjusting of the brake water loading switches.

The ignition timing was set for minimum advanced of spark ignition for best torque, MBT with limited by knocking. Ignition timing was adjusted for maximum torque and thermal efficiency. Thermal efficiency and mean effective pressure were improved with increasing compression ratio, until reaching a maximum value. Thereafter, thermal efficiency and mean effective pressure have decreased slightly due to higher compression ratio. Improvement of mean effective pressure with increasing compression ratio is a consequence of increased peak cylinder pressure. Pressure

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increase is followed by combustion temperature increase, requiring retarded ignition timing to avoid detonation. High compression ratio allows for improved fuel conversion efficiency, as the engine thermal efficiency is increased. Therefore, specific fuel consumption is reduced.

The firstly is done mapping the degree of ignition timing on the condition of the best torque, MBT condition. Furthermore graphic mapping the degree of ignition timing at MBT condition shown in Figure-2.

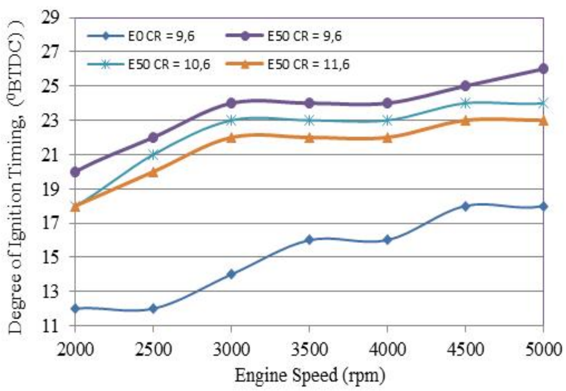

Figure-2. Mapping degree of ignition timing at MBT condition.

Mapping degree of ignition timing at MBT condition was showed by Figure 2. The trends observed in Figure-2 are explained that for all variable testing indicate that the degree of ignition timing is advanced due to the increase in speed engine. For gasoline fuel (E0) the degree of ignition timing gradually rises, ranging from 12^0 BTDC at engine speed 2000 rpm up to 18^0 BTDC at engine speed 5000 rpm. While the E50 has the same tendency that the degree of the ignition timing is more advanced due to higher engine speed but the value of degree of ignition timing advanced, from 18 ^{0}BTDC at engine speed 2000 rpm until 26 ^{0}BTDC at engine speed 5000 rpm. The addition of compression ratio requiring retarded ignition timing to avoid detonation.

Figure-3. shows the influence of compression ratio on brake torque. Torque is a measure of the ability to produce a working engine and is used to overcome the obstacles in the way or to raise engine speed. Figure-3 showed a tendency that the torque starts to rise from the lower engine speed (2000 rpm) to achieve maximum torque at a certain engine speed (3000 ~ 3500 rpm) and further decreased at higher engine speed (5000 rpm).

The trend of torque is due to that the higher the engine speed, the turbulence of the flow entering the combustion chamber will be higher thus increasing the air mixing with the fuel and flame propagation so that the output torque will increase. Along with the increase in engine speed will be greater the friction losses, heat losses and uncomplete combustion process so torque will be reduced. The highest brake torque of 49,58 Nm occurs at engine speed 3000 rpm and produced by compression ratio 11, 6. Overall, compression ratio 11, 6 on 50% bioethanol-gsoline blended fuel can increase the brake torque of 3, 68% compared to using pure gasoline at compression ratio 9,6. This increase is due to that high compression ratio increases cylinder pressure. This is will increasing the work done on the moved piston so consequently increasing of torque.

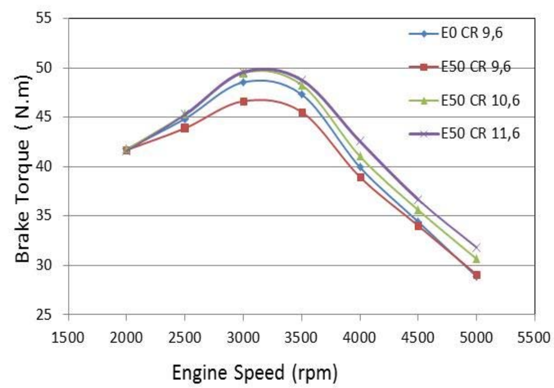

Figure-3. Brake torque at variation compression ratio and engine speed.

The amount of power is proportional to the torque that occurs, because it is related to the braking load by waterbrake dynamometer. The greater the braking load increased torque occurs. Theoretically, when the engine speed increases, the power will also increase. Figure-4 shows the influence of influence of compression ratio on power output.

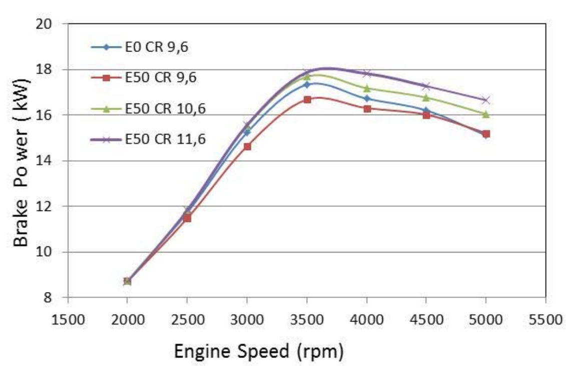

Figure-4. Brake power at variation compression ratio and engine speed.

The increase of compression ratio will increases the power generated by the engine. The highest brake power output of 17, 88 kW occurs at high speeds especially at 3500 rpm and produced by compression ratio

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11, 6. Compression ratios 9,6 and 10, 6 produced similar power in the whole range studied. As the output power is obtained from the product of torque and engine speed, the results shown by Figure 4 are a direct consequence of the ignition timing results shown by Figure-2. In comparison with gasoline fuel, E50 produced a peak brake power 4,58% higher, at 3500 rpm. As same with torque, this increase is due to that high compression ratio increases cylinder pressure so will increasing the work done on the moved piston and consequently increasing of power.

A parameter used to describe the performance of engines with reciprocating piston is the brake mean effective pressure, bmep. Bmep is the theoretical constant pressure that, if it acted on the piston during the power stroke, would produce the same net work as actually developed in one cycle. Figure-5. shows the influence of compression ratio on brake mean effective pressure, bmep.

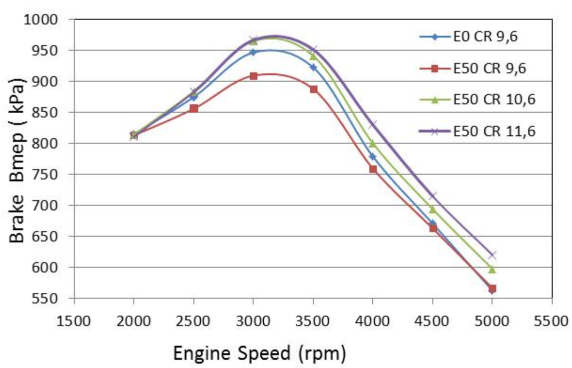

Figure-5. Bmep at variation compression ratio and engine speed.

For E50 and compression ratio 9,6, the ignition timing was also set to MBT produce slightly lower bmep than gasoline in the whole speed range investigated. In this case, the lower energy content of E50 caused to produce slightly lower torque, power and bmep than gasoline. However, when E50 was used with compression ratios 10,6 and 11,6, the ignition timing had to be advanced to avoid knock, especially in the region from moderate to high engine speed. That is the reason why torque, power and bmep were higher for high compression ratios and engine speeds when E50 was used instead of gasoline fuel. The highest bmep of 966,94 kPa or with increased until 3,68% occurs at engine speed 3000 rpm and produced by compression ratio 11, 6.

As shown from Figure-2, bmep increases with advanced ignition timings. This is caused by the time intervals between the admission of fuel mixture into cylinder and ignition timing decreases with advancing ignition timing, and this can ensure a high turbulence at the timing of ignition and increases the burning velocity of mixture. The shortening of time intervals between the end of fuel admission and ignition timing can form a better-stratified mixture, promoting mixture combustion and increasing bmep [13;16]. The mixture stratification is favorable to improve the burning velocity at lean mixture combustion of E50 fuel.

The brake specific fuel consumption, bsfc illustrates the flow rate of fuel required by the engine per unit of power generated. Due to the low calorific values, LHV E50 fuel is lower than gasoline (as show in Table-1), the amount of mass that is higher than E50 fuel required per unit of power generated, compared to gasoline. It was explained why bsfc E50 fuel is higher than gasoline, as shown by Figure-6.

From Figure-6, for E50 fuel shows that bsfc is decreased with increasing compression ratio. The compression ratio 9, 6 produced the highest sfc for most of the engine speed range investigated, while compression ratio 11, 6 produced the lowest bsfc values in the whole engine speed range. For all compression ratios, the bsfc levels obtained from E50 fuel is much higher than gasoline fuel. This is caused by LHV value of E50 fuel is smaller than gasoline. Besides the compression ratio of an engine is limited by fuel knocking resistance. For E50 fuel, increased compression ratio from 9, 6; 10,6 until 11,6 still showed a decrease in the amount of bsfc. This indicates that the compression ratio is still below that of knocking ability is limited by the compression ratio for the E50. This indicates that a given the compression ratio is still below the knock resistance of fuel used. While the influence of adding compression ratio at the E50 can reduce bsfc by 13, 42 %.

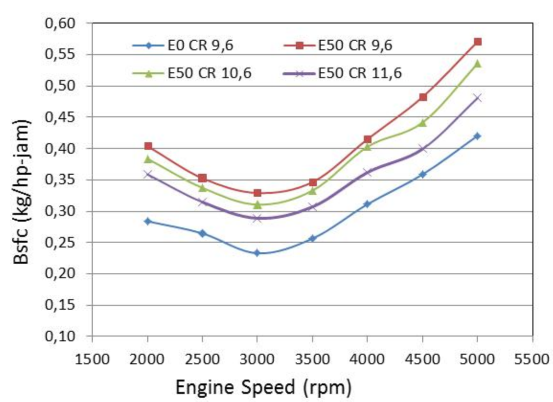

Figure-6. Bsfc at variation compression ratio and engine speed.

Thermal efficiency is a measure of the amount of the thermal energy stored in the fuel is converted into power for the engine. Generally, it is calculated by the ratio between power generated and thermal energy stored in the fuel. Figure-7 shows the influence of compression ratio on thermal efficiency of engine. For E50 fuel, increased compression ratio from 9, 6; 10, 6 until 11,6 still showed a increase in the amount of thermal

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efficiency. It is a consequence of still below of resistance knocking fuel to a given compression ratio.

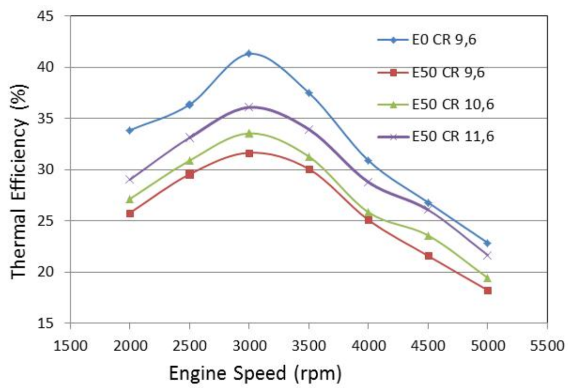

Figure-7. Thermal Efficiency at variation compression ratio and engine speed.

Thermal efficiency value depends on the quality of the air and fuel mixture is burned in the combustion chamber. As shown from Figure-7, the highest thermal efficiency of 44,10% occurs at engine speed 3000 rpm and was produced by E15 blended fuel. Overall, the addition of 15% bioethanol in gasoline fuel can increase the brake thermal efficiency of 1.81% compared to using pure gasoline. While the influence of adding compression ratio at the E50 can increase thermal efficiency by 14, 67 %.

CONCLUSIONS

The results showed that the increase of compression ratios improved engine performance for 50% bioethanol-gasoline blended (E50) fuel throughout all the speed range investigated. The both fuel fuel has same the tendency that the degree of the ignition timing is more advanced due to higher engine speed but the value of degree of ignition timing advanced, from 18 ^{0}BTDC at engine speed 2000 rpm until 26 ^{0}BTDC at engine speed 5000 rpm. The addition of compression ratio requiring retarded ignition timing to avoid detonation.

The use of E50 fuel, at the compression ratio is 11.6 can improve brake torque, power and mep respectively by 3, 68%, 4.58% and 3.68% as compared to using pure gasoline at a compression ratio of 9,6. While the influence of adding compression ratio at the E50 can reduce bsfc by 13, 42 % and increase thermal efficiency by 14, 67 %.

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