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AN EXPERIMENTAL STUDY TO EVALUATE THE PERFORMANCE OF A TWO-STROKE DIRECT INJECTION SPARK IGNITION ENGINE

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Abstract

The solution to increasing the efficiency of the small two-stroke spark-ignition engine is to use direct injection system technology. An experimental study was carried out by varying the fuel pressure, then an electronic module through an optical sensor was used to set the injection timing and injection duration. Performance is measured by loading the engine using Prony brakes to get engine speed, brake torque, and brake power, then the rate of fuel consumption is also measured for the Brake-specific fuel consumption (BSFC) calculations. The results indicate that injection timing affects torque, while power is influenced by the injection duration and pressure, while the BSFC is closely related to power and injection timing. Therefore, by combining existing parameters, certain conditions produce an optimal performance which occurs at the injection timing of 85°, injection duration of 55° to 65° at 8 bar fuel pressure.

Keywords: Direct injection, Performance, Spark-ignition engine, Two-stroke.

1. Introduction

The problem of air pollution due to motor vehicle emissions has become a worrying problem, especially in urban areas in Indonesia. Therefore, the Ministry of Environment of the Republic of Indonesia issued regulation no. 05 of 2006 which regulates the threshold for motor vehicle exhaust emissions [1]. This regulation requires increasing the combustion efficiency of internal combustion (*IC*) engines which will produce environmentally friendly emissions.

Several experimental studies into combustion characteristics of *IC* engines operated with blended fuels were conducted to improve the combustion efficiency of existing *IC* engines. In a 4-stroke spark-ignition engine, a mixture of bioethanol can improve the combustion process to reduce Hydrocarbon (HC) and carbon monoxide (CO) emissions, even in a 20% ethanol-gasoline mixture, the level of Nitrogen oxide (NO_x) emissions decreases by 50% without any change in engine power [2]. Another experimental study of a methanol-gasoline fuel mixture in a 4-stroke spark-ignition engine also showed increased engine performance and reduced HC and CO exhaust emissions [3]. In compression ignition engines, the results of an experimental study of blended fuels between diesel and vegetable oils can improve performance and reduce emissions, especially CO and CO_2 although there is an increase in NO_x emissions when biodiesel content increases [4]. Methods to reduce NO_X emission levels in biofuel fuel mixtures include injecting a water-diethyl ether solution through the inlet manifold [5]. Another method of reducing NO_X levels that is better without affecting engine efficiency and other emissions in compression ignition engines with biofuels can be done by installing Exhaust Gas Recirculation (EGR) [6].

However, many cities in Indonesia motorcycle population more than any other motor vehicle. The problem in air pollution is the emission of motorcycles, especially motorcycles using a two-stroke spark-ignition engine act as a significant contributor to pollution on air quality and health. Currently, two-stroke engines with carburettor fuel systems are increasingly reduced production in the automotive industry. This is large because the two-stroke spark ignition engine has high exhaust emissions and poor fuel efficiency [7, 8]. To overcome this high emission, direct injection is now one of the requirements for modern automotive two-stroke spark-ignition engines [9]. The introduction of direct injection technology applied to two-stroke spark-ignition engines enable the right mixture of fuel and air, giving complete control of combustion and emissions. As a result of which power output and efficiency increase. Direct fuel injection technology has also been used to reduce emissions. Another significant advantage of using direct fuel injection at two-stroke spark-ignition engines is that it is economical too as it provides a correct estimation of the quantity of fuel required at the proper time and providing control over combustion [10].

Research and application of technology direct injection fuel systems today are generally used in compression ignition engines [11-19]. As for the spark-ignition engine technology direct injection fuel systems are still focused on the four-stroke engine [20-24]. In the four-stroke direct injection spark ignition engine, the ability to ignite the mixture is affected by the choice of injection timing and injection duration [25, 26]. However, only a few studies have focused on the use of fuel injection systems as a fuel mixing system on a two-stroke spark-ignition engine and generally still use conventional fuel injection systems [27, 28]. The two-stroke

engine is an engine that completes its cycle in one crankshaft revolution [29]. Meanwhile, the shortcomings in this engine are producing high air pollution. This happens because of the combustion of lubricants and fuel during the scavenging process that escapes are discharged directly through the exhaust port [30]. To overcome these problems, then one way is to change the two-stroke spark-ignition engine into an engine that uses a direct injection fuel system, then this engine does not occur in the waste of fuel wasted during scavenging so that this engine does not cause excessive exhaust emissions [31]. The solution to produce good engine performance is to use a direct injection device for mixing the fuel system into the engine [32].

Preliminary studies using simulations of two-stroke spark-ignition engines with direct injection with gas fuel promise to increase fuel energy conversion and reduce exhaust emissions [33, 34]. Indeed, the results of research on 4-stroke spark-ignition engines using CNG gas type fuel coupled with hydrogen resulted in a significant reduction in emissions and an increase in specific energy consumption [35]. Furthermore, a two-stroke direct injection spark ignition engine experiment using gas fuel has also been carried out [36, 37]. However, these studies have drawbacks that the gas fuels used are difficult to obtain at gas stations, unlike liquid fuels. Therefore, there needs to be research that uses liquid fuel that is easy to obtain is liquid fuel. Numerical modeling studies that simulate the operation of a two-stroke direct injection spark ignition engine with liquid fuel promised outputs and fuel economy that are on par with the latest four-stroke spark-ignition engine [38]. Whereas the other two-stroke direct injection spark ignition engine experiment that uses liquid fuels is focused on exhaust emissions which when combined with the installation of a catalytic converter can reduce CO and HC emissions [39]. Studies on the direct injection compression ignition engine show injection pressure affects performance, emissions, and combustion characteristics [40]. It is validated in two-stroke direct injection spark ignition engine experiments using liquid fuel, where Initial studies were limited to being able to operate the engine known variations in injection pressure and injection timing influenced engine speed [41].

In engines with a fuel injection system, the injection time and injection duration greatly affect the ability of the mixture to burn, engine performance, and fuel consumption. this is especially the case in small equivalent operations for the period between the injection timing, injection duration, and ignition timing would influence the formation of the mixture in the cylinder [42-44]. This initial experimental study on two-stroke direct-injection spark-ignition engines found that the higher the injection pressure and the closer to the top dead center (TDC) the injection timing, the higher the engine speed. This happens because the increase in injection pressure and injection timing causes more fuel to be burned and converted into mechanical energy [41]. Furthermore, experimental studies on injection pressure and injection duration have shown that the higher the injection pressure and the longer the injection duration, the higher the peak power generated in this two-stroke direct-injection spark-ignition engine [45].

Therefore, this study will concentrate on gaining a thorough understanding of the effects of injection timing, injection duration, and fuel pressure which are related to the performance of a two-stroke direct-injection spark-ignition engine such as engine speed, fuel consumption, torque, and power.

2. Method

The method used in this research is experimental. The test equipment scheme used can be seen in Fig. 1. The engine tested is a two-stroke spark-ignition engine with specifications that are cylinder diameter 52 mm, stroke 51 mm, and displacement volume 110 cm^3. The engine specifications are shown in Table 1.

Table 1. Engine specifications of 1997 Yamaha F1ZR motorcycle.

Type	2-stroke, single-cylinder
Fuel system	carburettor modified to direct injection system
Bore x stroke	52 mm × 52 mm
Ignition system	CDI AC
Cylinder capacity	110.4 cm^3
Compression ratio	7.1: 1
Fuel during testing	RON 90

It was modified to be a two-stroke direct injection spark ignition engine. At the cylinder head, the Keihin brand injector mounting hole is adjacent to the exhaust port and placed next to the spark plug hole. Busi in position as far as possible from the wall in the center of the combustion chamber so that the flames spread to the wall uniform combustion to minimize heat loss through the walls. This is what causes the injector to be positioned vertically against the piston has a 45° angle from the spark plug, where around this position the pressure direction is still uniform [46]. Then the solenoid valve is placed in the fuel line right in front of the injector, where the nozzle hole is 0.3 mm in diameter as the inlet valve for pressurized fuel into the engine. The injection timing and injection duration settings are regulated by a simple injector drive module programmed with Proteus 8 software, in which the injector drive module processes translucent infra-red light through the disc board when the light starts to be received as injection timing then the duration of the light the interrupted light is regulated by the angle on the disc board as input, and open-close the injector solenoid as output.

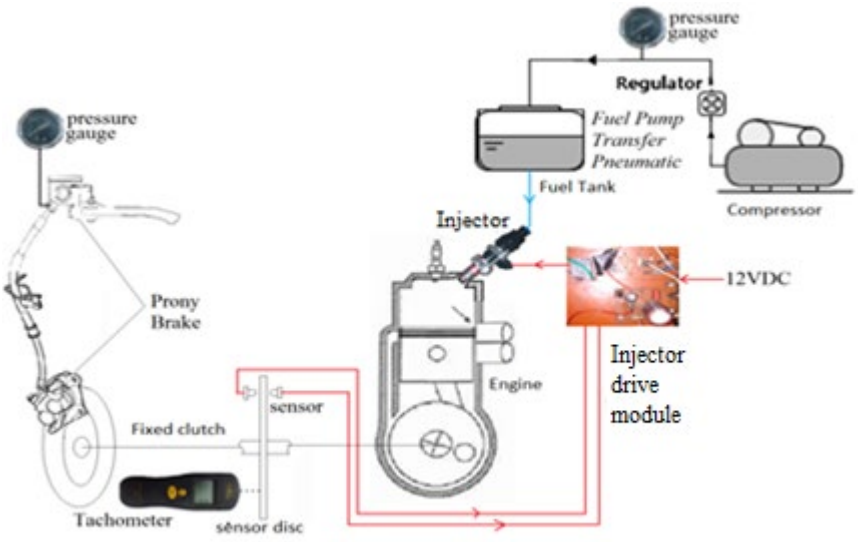

Fig. 1. Experimental setup.

The parameters measured from the test engine in the form of engine revolution speed are measured using a noncontact tachometer type Tachometersmart sensor AR926 whose sensor is placed on the crankshaft shaft. The tachometersmart sensor AR926 has an Accuracy of ± (0.05+1 digits). Sample time 0.8 seconds (more than 60 rpm) when selecting test range automatically and detection distance 50-500 mm measuring range 2.5-99999 rpm have a resolution 0.1 rpm (2.5-999.9 rpm), 1 rpm (above 1000 rpm). Torque is measured using a Prony brake in the form of a conventional hydraulic dynamometer employing a hydraulic disc brake with 2 piston callipers. Torque T is measured as brake lining friction coefficient, effective arm radius $\bar{r}$ and force transducer F. Effective arm length is $\bar{r}$ = 0.0835 m. Assuming the brake lining friction coefficient is μ = 0.36 at a temperature of 50°C based on the code SAE16FF. while the force F as a result of measuring the pressure on the hydraulic pressure gauge and the surface area of the brake lining is 13,295 cm^2. The pressure gauge is a bourdon tube analog pressure gauge Cl. 1.6, EN 837-1 NG 63. Furthermore, the pressure gauge is calibrated with the standard BS EN 837-1. Accuracy class 1.6. The indication of the number of errors at the ambient temperature of -20°C to +60°C on the gauge does not exceed the allowable error limit value of ± 1.6%. The torque measured on the Prony brake dynamometer is calibrated to ensure that there is a minimum error in the entire measurement range and obtain a calculation error value of ± 1.75%. The engine power is calculated based on the engine speed measurement and Torque calculation data from the Prony brake dynamometer.

The selected variations in injection timing and injection duration are shown as a diagram in Fig. 2. The injection timing angle variations are selected at the end of the scavenging process to obtain the optimal angle between minimizing the fuel wasted to the exhaust port and giving the fuel as long as possible to mix. with air. Therefore, the variations in the injection timing angles taken are 85º, 90º, and 95 before TDC. the variation of the injection duration angle is useful for finding the air-fuel ratio that results in an efficient combustion process. The variations in the injection duration angle taken are 45º, 55º, and 65º while the fuel pressure used is 6 bar, 7 bar, and 8 bar. The test is carried out when the engine reaches the optimum revolution and then is given a loading (brake pressure) by the Prony Brake. Then measure the speed of the revolution on the crankshaft due to the brake torque that is carried out continuously before the engine turns off. In the fuel consumption test, a burette is used to measure 100 ml of fuel, then the 100 ml fuel is filled into the tank. it is assumed that the density of gasoline at 30º C is 750 kg/m^3, so 100 ml of gasoline equals 75 grams. Then in a certain variation of injection timing, injection duration, and fuel pressure are measured how much time is needed until 100 mL of fuel is used up. Furthermore, this data is used to calculate the Brake-specific fuel consumption (BSFC) which is defined as the ratio of the mass rate of fuel consumption to engine power (BSFC = m_f / P_b).

3. Results and Discussion

The test results using the Prony brake dynamometer obtained torque, power, and BSFC along with the increase in the engine revolution. Furthermore, the maximum value of brake torque, brake power, and fuel consumption is selected from each combination of injection timing, injection duration, and fuel pressure. The following is Table 2 data collected from each adjustment with peak values of torque and power and the others are the lowest values of BSFC.

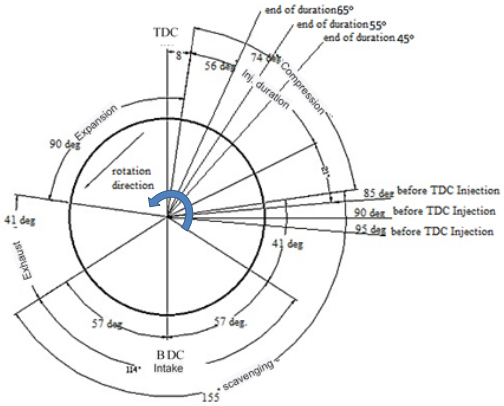

Fig. 2. Port timing diagram of the test engine.

Table 2. Parameters of the results of the peak operation of the test engine.

No.	injection timing BTDC	injection duration	fuel pressure, bar	peak torque / engine speed, Nm/rpm	Peak power / engine speed, kW/rpm	BSFC/ engine speed, (kg/kWh)/rpm
1	85°	45°	6	12.55/1292	1.7/1292	0.36/1578
2	85°	45°	7	12.55/1379	1.81/1379	0.4/1619
3	85°	45°	8	12.55/1300	2.15/1868	0.32/2021
4	85°	55°	6	12.55/1383	1.93/1683	0.32/1833
5	85°	55°	7	12.55/1791	2.35/1791	0.35/1983
6	85°	55°	8	12.55/1888	2.61/2268	0.307/2415
7	85°	65°	6	12.55/1740	2.36/2058	0.37/2105
8	85°	65°	7	12.55/1887	2.48/1887	0.34/2345
9	85°	65°	8	12.55/1940	2.67/2327	0.31/2552
10	90°	45°	6	10.97/1353	1.55/1353	0.42/1520
11	90°	45°	7	10.97/1457	1.67/1457	0.43/1618
12	90°	45°	8	10.97/1642	1.89/1642	0.44/1765
13	90°	55°	6	10.97/1466	1.72/1743	0.39/1783
14	90°	55°	7	10.97/1517	1.8/1831	0.43/1831
15	90°	55°	8	10.97/1803	2.09/2124	0.41/2124
16	90°	65°	6	10.97/1747	2.01/2043	0.44/2043
17	90°	65°	7	10.97/1837	2.11/1837	0.42/2094
18	90°	65°	8	10.97/1930	2.22/1930	0.42/2216
19	95°	45°	6	9.4/1466	1.44/1466	0.49/1466
20	95°	45°	7	10.97/1321	1.52/1321	0.55/1535
21	95°	45°	8	10.97/1513	1.74/1513	0.55/1619
22	95°	55°	6	10.97/1151	1.69/1712	0.46/1712
23	95°	55°	7	10.97/1423	1.9/1931	0.49/1931
24	95°	55°	8	10.97/1762	2.02/1762	0.51/1971
25	95°	65°	6	10.97/1376	1.96/1986	0.45/1986
26	95°	65°	7	10.97/1508	2.05/2083	0.47/2083
27	95°	65°	8	10.97/1841	2.24/2270	0.46/2270

Figure 3 shows the peak torque at each setting. It can be seen that in setting the injection timing of 85° and the fuel pressure of 6 bar, 7 bar or 8 bar has the highest peak brake torque of 12.55 Nm, whereas in a setting of the injection timing-injection duration-fuel pressure 95° - 45° - 6 bar has the results the lowest parameter where the peak brake torque is 9.41 Nm at the engine speed of 1466 rpm. If observed by comparing the torque between setting the injection timing, duration, and pressure of fuel injection is more dominant then affect the engine torque is the injection timing. when setting the injection timing at 85° has a higher torque value

than at an angle of 90º or 95º before TDC. This relates to trapped volume for injection timing because at an injection timing angle of 90º or more before TDC, more fuel will be wasted through the exhaust port.

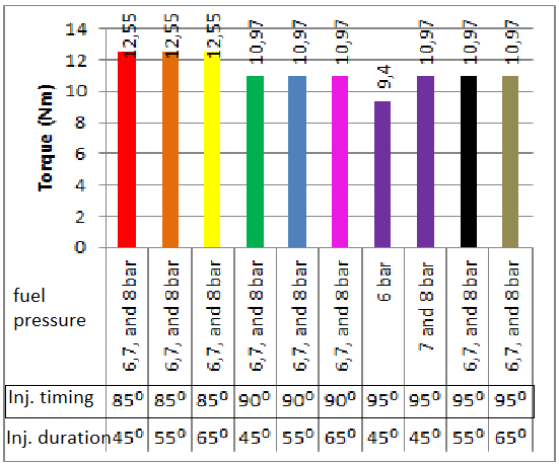

Fig. 3. Peak torque.

As the injection duration setting increases from 45º to 65º, the peak torque will be achieved at a higher engine speed (Fig. 4). This happens because an increase in the injection duration will increase the mass flow rate of the fuel so that there will be a rich mixture of fuel from the air-fuel ratio (AFR) in the combustion chamber. When more fuel is burned the more energy is released, so this will overcome the change in load on the engine. Likewise, fuel injection pressure also influences torque, which is almost in most variations, with the higher fuel injection pressure, the engine speed that produces peak torque also increases. If the injection duration is combined with the fuel pressure that occurs is an increase in the injection duration will increase the amount of fuel injected and the greater the fuel pressure will improve fuel atomization thereby causing an increase in engine speed which results in peak torque.

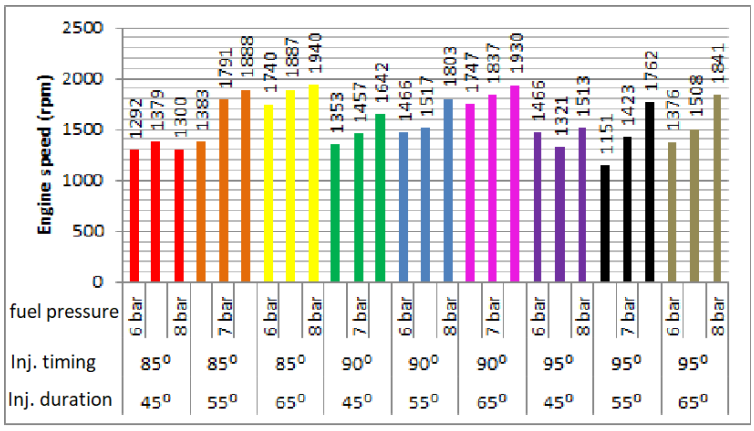

Fig. 4. Engine speed at peak torque.

In the variation of injection duration angle and fuel pressure at the same injection timing angle, it mostly does not affect the peak torque produced by the engine (Fig. 3) but, the variation of the injection duration angle and fuel pressure affect the engine speed that produces peak torque (Fig. 4). Engine power is proportional to the product of torque and engine speed. This is what affects the peak power produced by the engine, i.e. the higher the injection duration angle and fuel pressure, the greater the power produced. This is because even though the torque produced is always the same, the higher the angle of injection duration and fuel pressure causes the highest torque to occur at higher engine speeds so that the power produced is also increased. The peak power of a combination of injection timing (85°, 90°, and 95°) with injection duration (45°, 55°, and 65°) and fuel pressure (6, 7, and 8 bar) which have been tested from Table 1 can be seen in Fig. 5.

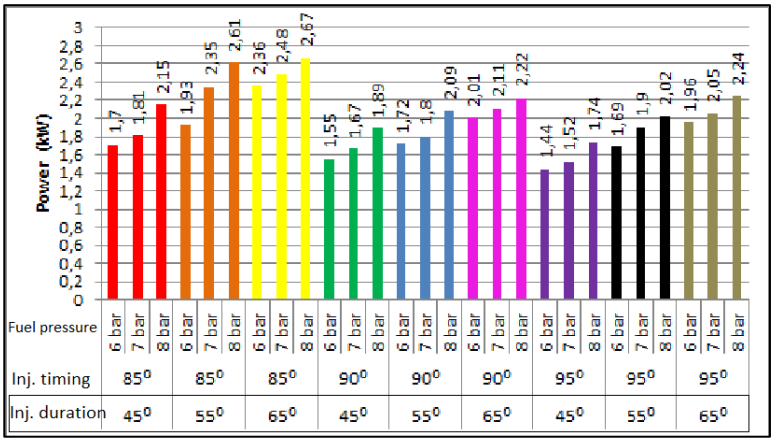

Fig. 5. Peak power.

Power is developed in the engine cylinder by combustion from charging. The charging is in the form of a fuel mass flow rate in the variable injection duration and an increase in pressure as shown in Fig. 5. The useful power is known as brake power output because the power can be absorbed and measured on the test bench (in this test using the Prony brake) utilizing brake friction. The result is shown in Fig. 5 that the peak power rises with increasing fuel injection duration, the highest is at a setting at 65° and the fuel pressure that increases is the greatest at 8 bar fuel pressure. In Fig. 5 the maximum peak power is reached at the injection timing setting 85° before TDC, 65° duration, and 8 bar fuel pressure with a result of 2.67 kW at 2327 rpm.

Figure 6 shows the lowest values of brake-specific fuel consumption (BSFC) of the entire adjustment. Specific Fuel Consumption (SFC) is the mass of fuel consumed by the engine to produce an effective power of 1 kW for 1 hour. Specific fuel consumption is very closely related to engine power. It is clear that by advancing injection time in direct injection mode, for example in setting the injection time of 95°, then when the injector starts to spray fuel the piston position has not closed the exhaust port so that there is wasted fuel which is not completely converted into power compared to the injection time of 85° which produces power greater range from 2.36 kW - 2.67 kW. This is shown in Fig. 6, which is at the

setting of 85º - 55º - 8 bar at the most efficient fuel consumption of 0.307 kg/kWh at 2415 rpm with power reaching 2.61 kW at 2415 rpm.

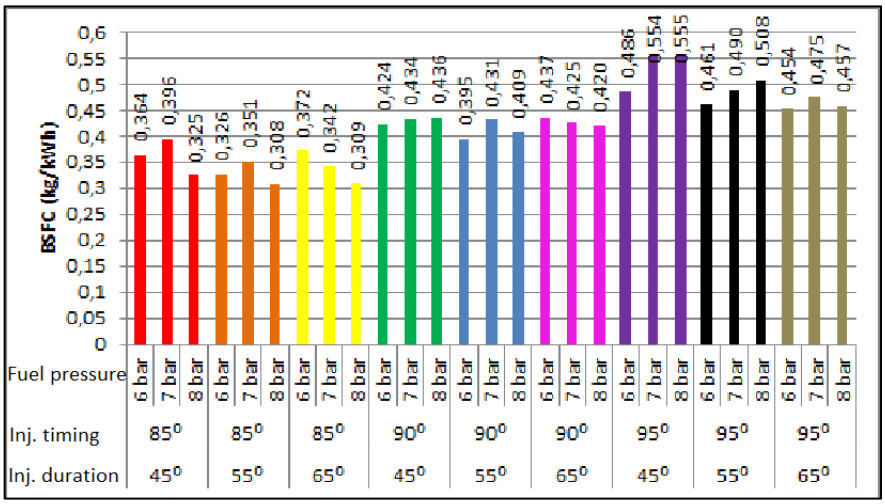

Fig. 6. Lowest values of brake-specific fuel consumption (BSFC).

Based on the BSFC equation = m_f/P_b the engine can be said to be fuel efficient if more time is needed to spend a certain amount of fuel and produce a large effective power, so the BSFC value generated is small. For the largest BSFC value, which is 0.555 kg/kWh in the 95º- 45º- 8 bar setting, it is mainly due to the value of the power generated is not too large while the mass flow rate of fuel that occurs is large. It should be noted that any injection duration and increased fuel pressure for this two-stroke engine if fuel is injected just before the trapped stroke, will make wasteful fuel consumption resulting in a lack of engine capacity to produce effective power.

The air-fuel ratio (AFR) does not change much, and the lowest fuel consumption is obtained at approximately the speed at which the maximum torque starts to increase i.e. at the time of the engine speed where the optimum value of volumetric efficiency occurs. This volumetric efficiency is closely related to the Brake Mean Effective Pressure (BMEP).

Figure 7 shows the performance curve of a two-stroke direct injection spark ignition engine at the best setting which results in optimum performance. The torque produced is directly related to the mean effective pressure (MEP). As shown in Fig. 7, in the setting of 85º-65º-8 bar, when the engine speed rises from low rpm to high rpm BMEP continues to rise, the torque also increases to reach peak torque of 12.55 Nm at 1940 rpm. Torque will drop after reaching the peak value at high rpm because the mean effective pressure above the piston is reduced. The peak power in the character of a reciprocating engine always occurs after peak torque in the higher rpm range, as shown in Fig. 7, the peak power obtained is 2.67 kW at 2327 rpm.

As for fuel consumption, shown in Fig. 8, that the BSFC value increases along with the addition of a higher load. This indicates that the greater the load received

by the engine, the fuel required increases. The optimal value is obtained at the setting of 85º-55º-8 bar adjustment occurs near after peak power at the engine speed of 2415 rpm with BSFC 0.307 kg/kWh.

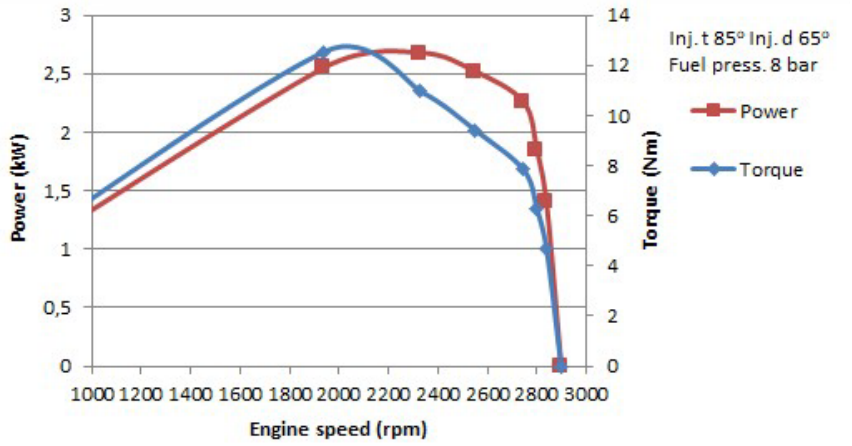

Fig. 7. Optimal performance curve for setting injection timing 85º BTDC.

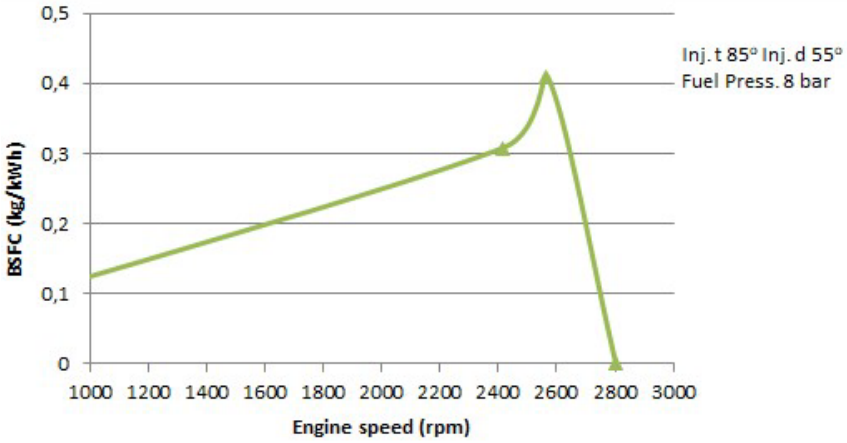

Fig. 8. The curve of BSFC at 85º-55º-8 bar setting.

4. Conclusions

Torque increases when the injection timing angle gets closer to TDC, likewise with power, when the injection timing angle gets closer to TDC, and the longer the injection duration at higher fuel pressure, the more power generated. Therefore, the highest peak torque and peak power occur at injection timing of 85º before TDC and injection duration of 65º at 8 bar fuel pressure, which is the value for the torque of 12.55 Nm at 1940 rpm and power of 2.67 kW at 2327 rpm. The lowest BSFC occurs at the injection timing setting of 85º before TDC, the injection duration is 55º at 8 bar fuel pressure, which is as much as 0.307 kg/kWh at 2415 rpm. Based on this, it is concluded that injection timing of 85º and injection duration of 55º to 65º at 8 bar fuel pressure is the most optimal.

This study only takes three parameters with limited variations, to understand the characteristics of this engine further, more parameters are needed, especially exhaust emission data for a wider variety. In addition, it is recommended to modify the injector drive module with a more sophisticated design of a microcontroller-based electronic control module so that in the future it can be added including throttle body and input sensors parameters to determine and regulate the air-fuel flow.

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Nomenclatures

AC	Alternating current
AFR	Air to fuel ratio
BMEP	Brake mean effective pressure, kPa
BSFC	Brake-specific fuel consumption, kg/kWh
BTDC	Before top dead center, ^{o} (deg.)
CDI	Capacitor discharge ignition
EGR	Exhaust gas recirculation
F	Force, N
MEP	Mean effective pressure, kPa
m_f	Mass rate of fuel consumption, kg
NO_X	Nitrogen oxide
P_b	Engine power, kW
$\bar{r}$	Effective arm radius, m
SFC	Specific Fuel Consumption, kg/kWh
T	Torque, Nm
TDC	Top dead center, ^{o} (deg.)

Greek Symbols

μ	Friction coefficient

Abbreviations

BS	British Standards
CNG	Compressed natural gas
IC	Internal combustion
RON	Research octane number
SAE	Society of Automotive Engineers

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