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Research Article

PERFORMANCE EVALUATION OF A SMALL SPARK-IGNITED ENGINE WITH AN ELECTROMAGNETIC INTAKE VALVE

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ABSTRACT:
The aim of this project was to design and fabricated opening intake valve system of spark-ignited engine by the electromagnetic. The comparative performance of small spark-ignited engine between the electromagnetic intake valve engine and reference engine with the air mass flow rate, volumetric efficiency, indicated mean effective pressure, brake mean effective pressure, brake specific fuel consumption, indicated thermal efficiency and brake thermal efficiency at engine speed of 1200, 1500, 1800 and 2100 rpm and throttle opening of 25, 50 and 75 % wide open throttle. From experimental, it was found that the EMIV system can be used for application in an internal combustion engine. Moreover, the air mass flow rate, volumetric efficiency, indicated mean effective pressure, brake mean effective pressure, indicated thermal efficiency and brake thermal efficiency of the electromagnetic intake valve engine is lower than that from reference engine, but the brake specific fuel consumption of the electromagnetic intake valve engine is more than that from reference engine.

Keywords: *Electromagnetic intake valve, Volumetric efficiency, Brake specific fuel consumption, Brake mean effective pressure, Brake thermal efficiency*

1. INTRODUCTION

Electromagnetic actuators (EMA) was most extensively in industry because of low cost, relative high force density and easily control. Moreover EMA is being used in fuel injection actuation, exhaust gas recirculation (EGR) system, food dispensers, refrigerators, washing machines, etc. The EMV is a key technology to reduction in fuel consumption and vehicle emissions. The idea of camless engine has very interesting in the concept of variable valve timing (VVT) or fully flexible valve actuation system which is direct control of both valve timing and valve lift [1]. Several applications of EMA have been proposed. Özdalyan and Tasliyol [2] study to measure the valve profile and electrical coil current of an electromechanical system designed for small internal combustion engine at 24, 33 42 and 48 V of supply voltages, low (1200 rpm) and high engine speeds (3600 rpm), valve openings of 0, 9, 18, 27 and 36 °CA before top dead center (BTDC) and closing angles of 27, 36, 50 63 and 72 °CA after bottom dead center (ABDC). The results were found that a supply voltage of 24 V was inadequate to make valve movement for an engine with a speed of 3600 rpm. A supply voltage of 33 V was suitable for low-speed engine operations in achieving the identified valve timing. Birgul [3] was to improve an electromagnetic actuators valve (EMV) system and to investigate dynamic performance at different lifting valve operations. The supply voltage and electric current of EMV1 were 12 V and 10 A, respectively. And the supply voltage and electric current of EMV2 were 12 V and 13.33 A, respectively. The transition time for 6 mm valve lift of EMV2 was measured about 3.9 ms and the maximum engine speed was 5128 rpm.

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Sugimoto et al. [4] study on variable valve timing system using the technology of electromagnetic mechanism. The mechanism is operated by a supply voltage of 42 V. The adoption of non-throttling technology permitted a 10% improvement in brake specific fuel consumption compared to the conventional cam driven valve train. In addition, a 20% improvement in torque generation at low and medium engine speeds. Also, a 7% improvement of the 10-15 mode fuel economy was confirmed by the simulations. In the work of Parlikar et al. [5], the EMVD apparatus was designed, constructed, and integrated into a computer-controlled experimental test stand. The EMVD was operated by the electric voltage and current of 42 V and 42 A, respectively. The transition time was 3.5 ms, corresponding to sufficient for the engine speeds about 6000 rpm. Wang et al. [6] studied the valve release timing, transition times, and contact velocities for open and closed-loop control schemes of the electromechanical valve through simulations and experiments. From experiment found that a holding voltage is 100 VDC and a transition time is 3.42 ms. In this work that reduced the valve motion delay in the releasing phase from 34 ms to 1.2 ms. Xu et al. [7] study the maximum intake charging under different engine speeds and the optimal parameters of Electromagnetic intake valve train by contrasting the power consumption and pumping loss. The results show that the intake changing of the engine was increased from 7.13 to18.3% at 1000 rpm and the power consumption of the signal valve model was decreased 35.9% compared with double valve model at 2000 rpm. However, the effects of the electromagnetic intake valve (EMIV) system on volumetric efficiency, brake mean effective pressure and brake thermal efficiency of gasoline engine have not been elucidated. Therefore, the current work aims to elucidate the electromagnetic intake valve (EMIV) system on the volumetric efficiency, brake mean effective pressure, brake thermal efficiency, air mass flow rate and brake specific fuel consumption. The aim of this work was to compare performance of the electromagnetic intake valve engine and reference engine at engine speed of 1200, 1500, 1800 and 2100 rpm and throttle opening of 25, 50 and 75 % wide open throttle (WOT). The performance of engine in this work were air mass flow rate, volumetric efficiency, indicated mean effective pressure, brake mean effective pressure, brake specific fuel consumption, brake thermal efficiency, indicated thermal efficiency and brake thermal efficiency

2. EXPERIMENTAL SETUP

2.1 Experimental engine

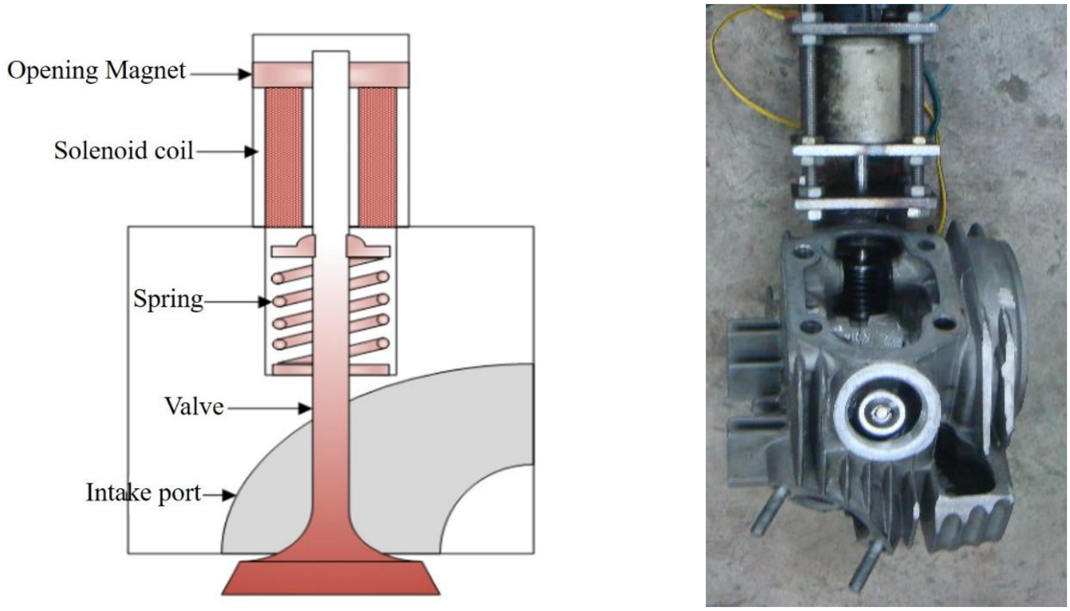
The reference and test engine used was Honda NF 100 and the technical specifications of reference engine are shown in Table 1. The first engine is the “reference engine” and the second engine is “test engine” (electromagnetic intake valve engine). In the reference engine, the intake valve and camshaft gear were standard, which were connected with a chains and driven by crankshaft. While in the electromagnetic intake valve engine, the intake camshaft was removed and replaced by the electromagnetic valve actuator which control by Electromagnetic valve control unit (EMIVCU) while the exhaust valve is still driven by a camshaft.

Table 1: Engine technical specifications.

Bore x stroke	50.0 mm x 49.5 mm
Swept volume	97.1 cm^3
Compression ratio	9.0 : 1
Intake valve opening	12 BTDC
Intake valve closing	34 ABDC
Maximum power	5.20 kW @ 8,000 rpm
Maximum torque	7.34 N-m @ 5,500 rpm
Fuel type	Gasoline (RON 91)
Number of Valves	2

2.2 The electromagnetic intake valve control mechanism

In this work, an electromagnetic intake valve single solenoid actuator is developed. The electromagnetic intake valve (EMIV) control mechanism was shown in Fig. 1. The main operation principle is a mass-spring oscillation and valve timing is achieved by control of voltage applied to the solenoid coils of electromagnetic. The EMIV control mechanism had an actuator that comprised of a solenoid coil and opening magnet that enabled the valve to open and a spring that enabled the valve to close. The opening magnet of the EMIV actuator was moving in the same direction of the intake valve. The spring was responsible for closing the intake valve. The valve core inside the solenoid coil was exposed to the magnetic field of the solenoid coil when EMIVCU supply voltage 12 VDC was applied to the solenoid coil, and moved in the direction of the opening valve. To close the valve, EMIVCU supply voltage was not applied to the solenoid coil, and the opening magnet was moved in the same direction of the valve closing to close the valve. Under that condition, no voltage was applied to the solenoid coil, when no signal is applied to the system, the moving valve core is kept in the upper position under the effect of the spring forces, and the intake valve was kept in the closed position.



(a) (b)

Fig. 1. Electromechanical valve mechanism (a) diagram and (b) electromechanical valve on the cylinder head

2.3 Experimental conditions

To test the performance of electromagnetic intake valve engines including the air mass flow rate, brake power, brake mean effective pressure and brake specific fuel consumption. The experiments were carried out at low engine speed (1200, 1,500, 1800 and 2,100 rpm) and part-load (throttle position opening of 25, 50 and 75 % WOT). The performance of the engines were compared both prototype and electromagnetic intake valve engine.

2.3.1 Intake valve profile

Figure 2 illustrates the examples of the intake valve profile with different closing phases at engine speed of 1200-2100 rpm. The electromagnetic force depends on the electric current, and is independent of the coil position [8]. The valve lift was limited to 3 mm, the maximum supply voltage and electric current were limited to 12 VDC and 24 Amp, respectively. The closing and sealing performance of the closed-loop controlled EMIVCU system can be quantified by using the following indices: the time of voltage applied to the opening coil t_0, EMIV engine valve start opening time t_1, EMIV engine full opening time t_2, EMIV engine start closing time t_3, EMIV engine seating time t_4 or 25 °CA after bottom dead centre (ABDC) and reference engine closing time t_5 or 34 °CA before bottom dead centre. The retardation time was time from t_0 to t_1 which between the release command and the start of valve motion. The transition time or valve travel time was the time from t_1 to t_2 which can be approximately calculated from the fundamental natural frequency of the mass-spring system [9].

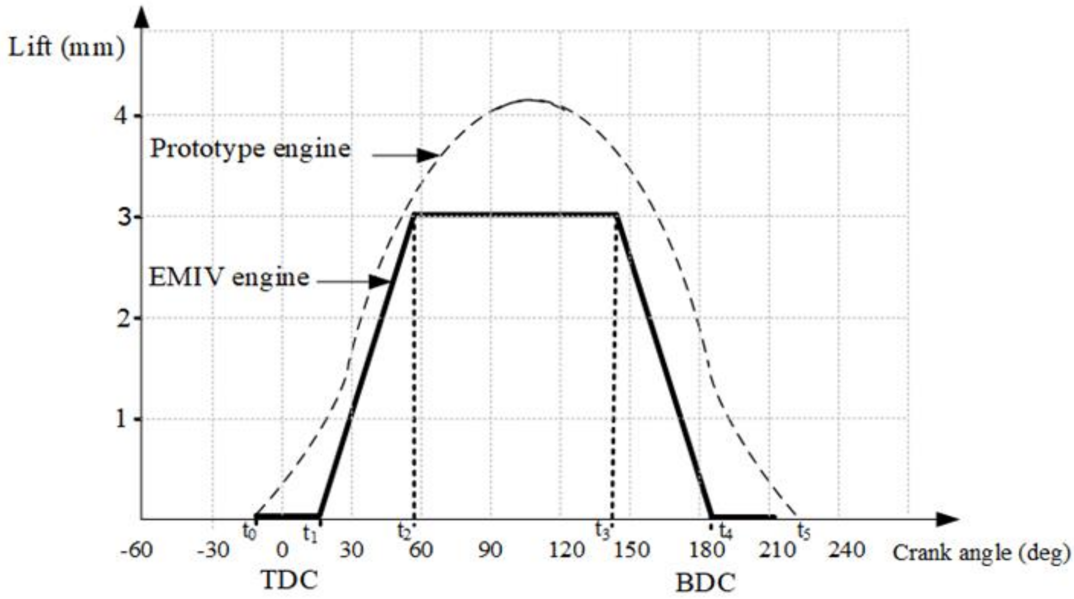


Fig. 2. Intake valve profile of engine in this work.

The maximum valve lift of the prototype and the EMIV engine were 4.2 mm and 3.0 mm, respectively. The reference engine and EMIV engine intake valve were opening at 12 °CA before top dead centre (BTDC) or at time t_0 and the reference engine exhaust valve was closed at 25 °CA after bottom dead centre while the EMIV engine exhaust valve closed at 34 °CA before bottom dead centre, which the early closing of the intake valve result in loss of flow momentum inside the cylinder required to enhance the combustion initiation and the rate of heat release [10]. The valve opening and closing profiles were square-shaped in EMIV engine. In intake valve profile, the cross-section area were under the intake valve lift curve of EMIV engine used in this work was lower than that of the reference engine which increasing the cross-section area was produces a greater flow area as the piston starts to pull in a fresh charge.

2.4.2 The electromagnetic intake valve (EMIV) engine test bench

In Fig.3. Electromagnetic valve control unit (EMIVCU) was the box for control to the intake valve, which can open the intake valve at the valve lift about 3 mm. EMIVCU controls the timing of the intake valve. It determines the position of the engine's internals using a crankshaft position sensor and camshaft position sensor and so that the intake valve is activated at precisely the correct time. Which the angle and engine speed information from the crankshaft position sensor and the camshaft position sensor was processed by the EMIVCU to create an input signal for the electromagnetic valve relay. The electromagnetic valve relay is an electrically operated switch that can be turned on the intake valve by the electromagnetic valve actuator, letting the current go through or not, and can be controlled with low voltages, like the 12V provided by the EMIVCU. In EMIV engine, the electromagnetic valve actuator can be used to open the valve and to close the valve by springs which was linear types offer strokes in about 3 mm. The electromagnetic valve actuator was provided energy based on the information transferred from the EMIVCU to the electromagnetic valve relay to open the valve. Although that actuator is well-suited to holding the engine valve open. The method of controlling the valve-spring system was to use single solenoid. The solenoid in this work was to hold the intake valve open and the spring was to close the intake valve. The crankshaft position sensor was to determine the position and rotational speed (RPM) of the crankshaft. The electromagnetic valve control unit uses the information transmitted by the sensor to control valve timing. The camshaft position sensor was used to determine the position of the exhaust camshaft for to open the intake valve which the timing of the exhaust valve is related to the timing of the intake valve. From Fig.3, The exhaust valve in the test engine was to open by the exhaust cam and closed by the spring. The electromagnetic intake valve engine in this work was shown in the Fig.1. The valve lift and the total opening time remain fixed. The valve timing can be used to influence the power output as a means to get a softer transition between the two engine operations modes discussed above.

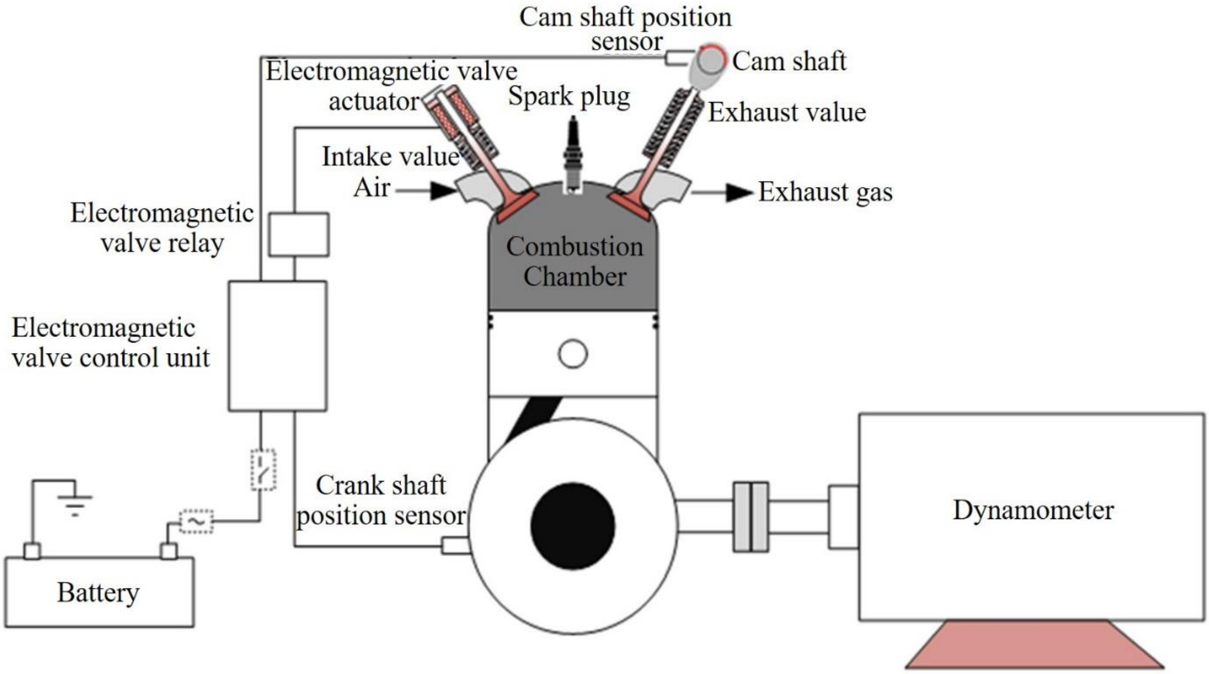


Fig. 3. Schematic diagram of single solenoid actuator in the EMIV engine.

2.4.2 Test Engine

The tests were conducted in a part-load operating condition at four engine speed (1200, 1,500, 1800 and 2,100 rpm) and under part-load (throttle position opening of 25, 50 and 75 %) for the prototype and electromagnetic intake valve engine. A Tecquipment TD 114 hydraulic dynamometer type was used to measure the brake power generated by the engine. The orifice flow rate meter was used to measure the air flow rate through the engine which according to ISO 5167-1 [11]. Measurement input signals were received from the test bed via manometer tapping from the air box. Moreover, the liquid fuel consumption per hour was calculated by measuring fuel consumed per unit time.

3. Quantitative analysis of the effect of EMIV on engine performance

The volumetric efficiency (η_v) is the ratio of the mass density of the actual air drawn into the cylinder at atmospheric pressure (during the intake stroke) to the mass density of the same volume of air in the intake manifold, which can then be calculated using the following mathematical expression [12]

$$\eta_v = \frac{2\dot{m}_a}{\rho_{a,i} V_d N} \tag{1}$$

Where $\dot{m}_a$ is the actual air mass flow rate; $\rho_{a,i}$ is the air density; N is the engine speed and V_d is the engine displacement volume.

The brake power (W_b) is the power output of any engine measured at the output shaft of an engine without the loss in power caused by the gearbox, generator, differential and other auxiliaries. The brake power of engine was varying to torque and speed engine [12]. The brake power is expressed as follows:

$$W_b = 2\pi T N \tag{2}$$

Where T is the torque produced by engine in N-m.

Brake mean effective pressure (bmep) is defined as the hypothetical pressure, which is thought to be acting on the piston. Typical values for bmep in the naturally aspirated spark ignition engines are the range 850 to 1,050 kPa at the engine speed where maximum torque is obtained (about 3,000 rpm) [12]. The brake mean effective pressure is expressed as follows:

$$bmep = \frac{2\pi T n_R}{V_d} \tag{3}$$

Where n_R is the number of crank revolutions for each power stroke per cylinder; two for four-stroke cycles and one for two-stroke cycle.

Brake specific fuel consumption (bsfc) is a measure of the fuel efficiency of the engine that burns fuel and produces shaft power. It is typically used for comparing the efficiency of the engines with a shaft output. It measures how efficient an engine is using the fuel supplied to produce work. The brake specific fuel consumption is [12]

$$bsfc = \frac{\dot{m}_f}{W_b} \tag{4}$$

Where $\dot{m}_f$ is the air mass flow rate.

The data of total motored friction mean effective pressure (fmep) for SI engines as a function of engine speed [13] are well correlated by an equation of the form:

$$fmep = 97 + 0.9N + 0.18N^2 \tag{5}$$

The indicated mean effective pressure of the standard engine can be found from:

$$imep = bmep + fmep \tag{6}$$

The first law of thermodynamic efficiency of internal combustion engine is the ratio of desired output to required input [14], particularly named thermal efficiency. The indicated thermal efficiency (η_i) is the ratio of the indicated thermal power produced per cycle to the amount of fuel energy per cycle. The indicated thermal efficiency is expressed as follows:

$$\eta_i = \frac{W_i}{\dot{m}_f q_c} \tag{7}$$

The brake thermal efficiency (η_b) is the ratio of the engine power produced per cycle to the amount of fuel energy per cycle; the brake thermal efficiency is then [15]

$$\eta_b = \frac{W_b}{\dot{m}_f q_c} \tag{8}$$

Where q_c is the heat of combustion which for gasoline fuel of 44,000 kJ/kg [12]

4. Results and Discussion

4.1 The actual air mass flow rate

The actual air mass flow rate was through the engine which that did affect the performance of the engine. Fig. 4 shows the variation of air mass flow rate with engine loads for the reference and an EMIV engine at different engine speeds. It was found that the actual air mass flow rate of the prototype and EMIV engine is increased rapidly with increasing the engine speeds due to the engine was to work at low engine speed. In calculating, ideal air mass flow rate at wide opening throttle (WOT) position and engine speeds of 1200, 1500, 1800 and 2100 rpm were 1.26, 1.58, 1.90 and 2.21 g/s, respectively. It can be seen that the air mass flow rate obtained in the engine increase when the engine speeds increased. This shows that the air mass flow rate produced in this work was rather similar to ideal air mass flow rate from calculating. At low engine speed, the rate of air flow in the intake system is low, therefore the air remains in contact with the hot intake system for a longer time cause the fuel starts to evaporate and more fuel vapor displaces the incoming air [14]. Moreover, this reduces the air density, which lowers the actual air mass flow rate at the lower engine speed. From the experiment, an increase in the engine speed implies an increase in actual air mass flow rate because of the density of air increased. Whereas increasing the throttle position opening from 25 to 50 and 75 % WOT, the actual air mass flow rate increased in both the engine. Within the scope of the parameters investigated, the maximum actual air mass flow rate of the EMIV engine was 1.47 g/s which occurred when applying the engine speeds and the throttle position opening of 2100 rpm and 75 % WOT, respectively. Whereas the maximum actual air mass flow rate of the reference engine was 1.68 g/s which occurred when applying the engine speeds and the throttle position opening of 2100 rpm and 75 % WOT, respectively. Compared with the reference engine, the EMIV engine gave lower actual air mass flow rate than the reference engine in all the engine speed and throttle position opening due to the opening of the prototype intake valve occurs faster and early closing of the EMIV intake valve, as shown in Fig. 2.

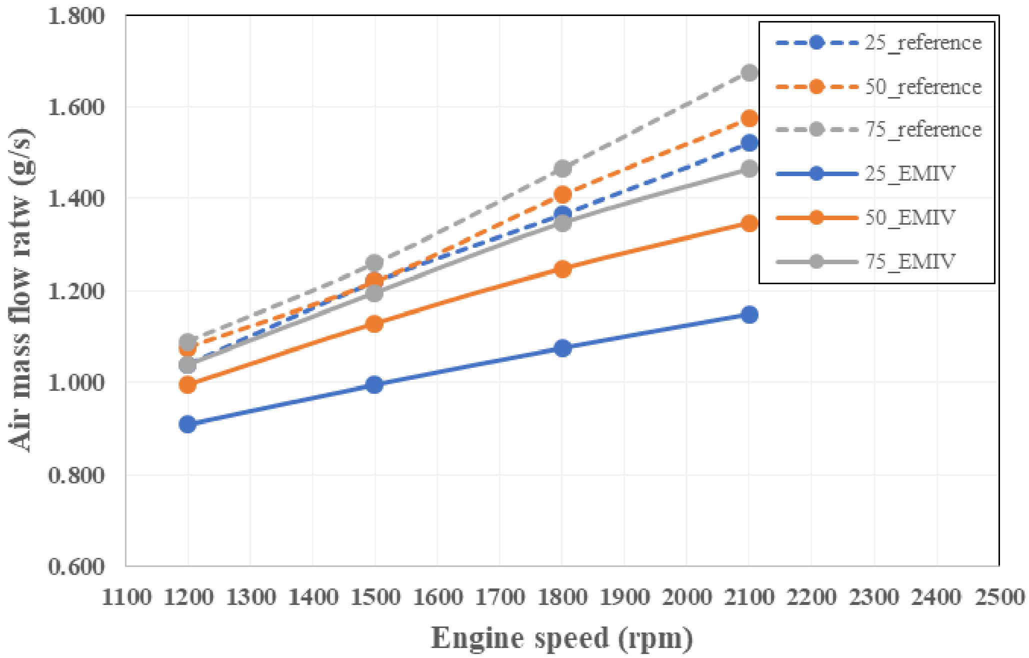


Fig. 4. The air mass flow rate of the reference engine and EMIV engine.

4.2 Volumetric efficiency

Volumetric efficiency is the ratio of actual mass flow rate into the engine's cylinder at the atmospheric pressure and temperature conditions surrounding the engine to the rate at which the mass is displaced by the piston which is a measure of the effectiveness of an engine's induction process [14]. Fig. 5 shows the variation of volumetric efficiency with engine loads for the reference and electromagnetic intake valve engine at different engine speeds. It was found that the volumetric efficiency is decreased when increasing the engine speeds and reduce the throttle position opening. Reason for this is that the cylinder intake period has been already small when the engine speed is increased and the reduction of the throttle position opening which decreasing the cross-section valve area causes to decrease the air amount into the cylinder, as shown in Fig. 4. Which ideal air capacity at all throttle position openings and engine speeds of 1200, 1500, 1800 and 2100 rpm were 1.26, 1.58, 1.90 and 2.21 g/s, respectively. In Fig. 5, the EMIV engine gave lower volumetric efficiency than the reference engine in all the engine speed and throttle position opening. Because the EMIV engine gave lower actual air mass flow rate than the reference engine which it discussed in topic 3.1.

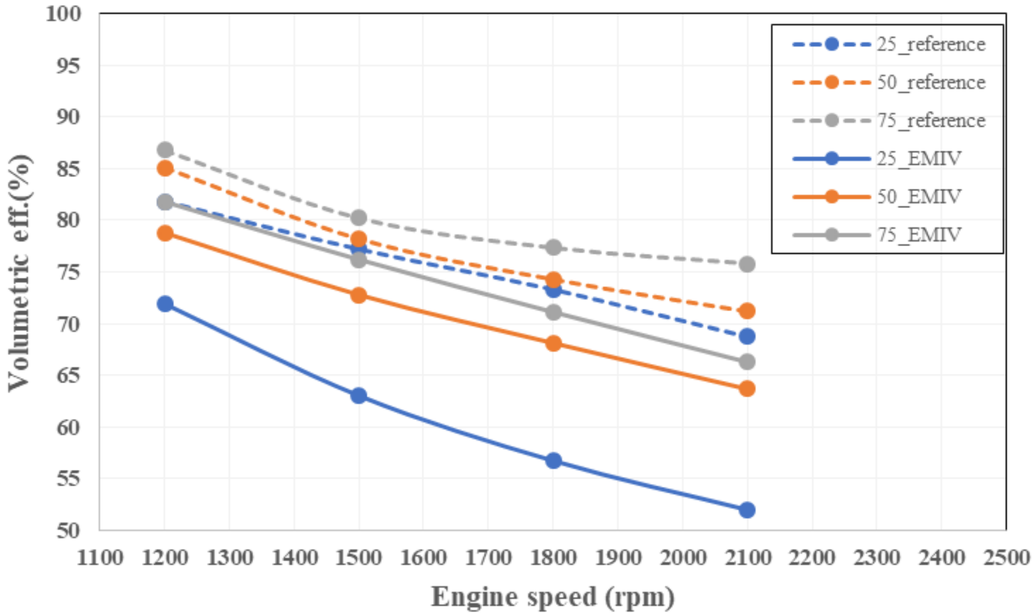


Fig. 5. The volumetric efficiency of the reference engine and EMIV engine.

The maximum volumetric efficiency of EMIV engine the found in the range of the process parameter investigated was 81.81%, which occurred when applying the engine speeds and the throttle position opening of 1200 rpm and 75 % WOT, respectively. While the maximum volumetric efficiency of reference engine was 86.81%, which occurred when applying the same engine speeds and the throttle position opening. GUPTA [14] discussed in the volumetric efficiency of an engine at variable engine speed and full load (100 % of throttle position opening) which found that the volumetric efficiency increase with increasing in low engine speed and decreasing in high engine speed.

4.3 Brake power

The brake power is the power output of the engine. Measurement of the brake power involves the determination of the torque and the engine speed of the engine output shaft [14]. Fig. 6 shows the change in the brake power for the reference and electromagnetic intake valve engine at different engine speeds and throttle position opening. It was found that the brake power is increased when increasing the engine speeds and the throttle position opening. Increasing the engine speeds can increase the air mass flow rate to cylinder to make the air intake more adequate, which it discussed in topic 3.1. While increasing the throttle position opening causes to increase the air amount into the cylinder. This is because of higher cross-section intake valve area at 50 and 75 % of full throttle position opening. From the experiment, it was found that the maximum brake power of 746.47 W occurred in the EMIV engine when applying the engine speeds of 2100 rpm and the throttle position opening of 75 %. While the maximum reference engine brake power of 1222.67 W was obtained when using the similar condition of the EMIV engine testing. Compared with the reference engine, the brake power of EMIV engine gave lower volumetric efficiency than the reference engine in all the engine speed and throttle position opening. It can be seen in Fig. 6 that the EMIV engine gave lower actual air mass flow rate than the reference engine because the duration time of charging in EMIV engine was shorter than of the reference engine as shown in Fig. 2 which the early closing of the EMIV intake valve reduces cylinder gas temperature and pressure at the end of the compression stroke [10]. In addition, the late closing of the prototype inlet valve shows an increase in brake torque for all engine speeds [16]. The lower temperature and pressure of gas in the cylinder at the initial of the combustion causes to reduce the rate of heat release and torque of the engine. Sugimoto et al. [17] conducted an experimental investigation to improve the power engine with variable valve timing system using electromagnetic mechanism in the gasoline engine and their experiment show a growth 20% when compared to conventional type.

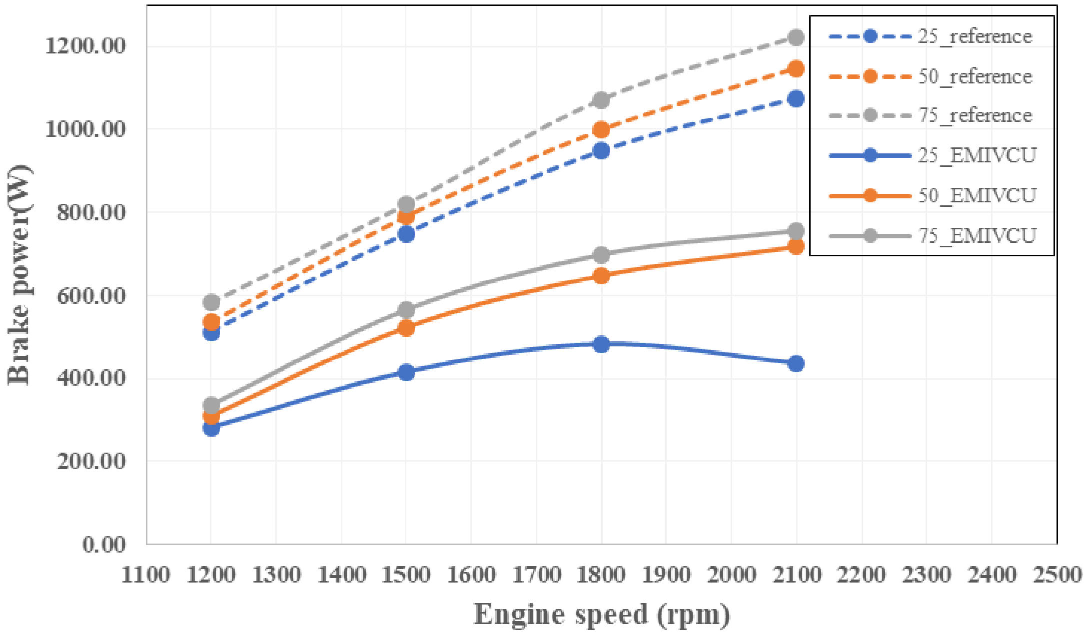


Fig. 6. The brake power of the reference engine and EMIV engine.

4.4 Indicated mean effective pressure and brake mean effective pressure

Figure 7 and 8 depict the Indicated mean effective pressure and brake mean effective pressure of engine operation with respect to engine speed for different throttle position opening. The indicated mean effective pressure (imep) does not depend on engine speed, just like torque. Whereas the brake mean effective pressure (bmep) is an indication of engine efficiency regardless of capacity or engine speed. While torque is a valuable measure of a particular engine's ability to do work, it depends on engine size. A more useful relative engine performance measure is obtained by dividing the work per cycle by the cylinder volume displaced per cycle. The parameter so obtained has units of force per unit area and is called the mean effective pressure (mep) [12]. Form Fig. 7 and 8, it was found that the imep and bmep of both engine is increased when increasing the engine speeds and the throttle position opening. Increasing of engine speeds and throttle position opening causes to increase the air amount into the cylinder and increasing the torque of engine which to increasing the brake power. The bmep of engine was reduced roughly by 17-30 % over the engine speed range of 1500-2100 rpm in comparison with imep of engine. The reduction of bmep was due to the friction lost in engine which calculated in the fmep. In eq.5, the fmep increases linearly with engine speed. The EMIV engine produce 4-18 % less fmep than the reference engine. The EMIV engine shows the maximum bmep about 493.10 kPa at 1500 rpm speed where the reference engine shows approximate 661.73 kPa at 1800 rpm. The difference of valve timing in the engine is the reason of this different trend which it's shown in Fig. 2. The bmep for EMIV engine was found to be lower as compared to the reference engine. This is because of the lower torque of EMIV engine which reduces the bmep as compared with the reference engine. For typically gasoline engines, maximum bmep is in the 850-1050 kPa [12].

4.5 Brake specific fuel consumption

In engine tests, the fuel consumption is measured as a mass flow per unit time. Besides, the specific fuel consumption is measured as a fuel flow rate per unit power output which measures how efficiently an engine is using the fuel supplied to produce work [12]. Fig. 9 shows the variation of brake specific fuel consumption (bsfc) with engine loads for the reference and EMIV engine at different engine speeds. It was found that the bsfc of EMIV engine decreases with decreasing and increasing engine speeds. This is because of the lower air mass flow rate of EMIV engine which reduces the brake power as compared with the conventional engine. The best bsfc of EMIV engine is 401.19 g/kW-hr which occurred when applying the engine speed and the throttle position opening were 1500 rpm and 75 % WOT, respectively. Whereas the best bsfc of reference engine is 239.47 g/kW-hr at the engine speed and the throttle position opening were 2100 rpm and 75 % WOT, respectively. In engine speed range of 1500-2100 rpm, the bsfc of EMIV engine was increased owing to large electric current was applied to shift the EMIV system from the opening to the closing process quickly, ensuring short valve opening duration to meet high engine speeds [18]. Sugimoto et al. [17]

shows the results of the actual engine fuel economy performance. It was confirmed that the use of variable electromagnetic valve timing permit the bsfc including electric power consumption was improved by 10% compared to the prototype cam-driven valve train engine owing to delayed intake valve closure timing. Moreover, the 10-15 mode cycle had been used in Japan for emissions and fuel economy testing for light duty vehicles, it found that the simulation result yields a 7% improvement in fuel economy. For typically gasoline engines, the best values of bsfc are about 270 g/kw.hr [12].

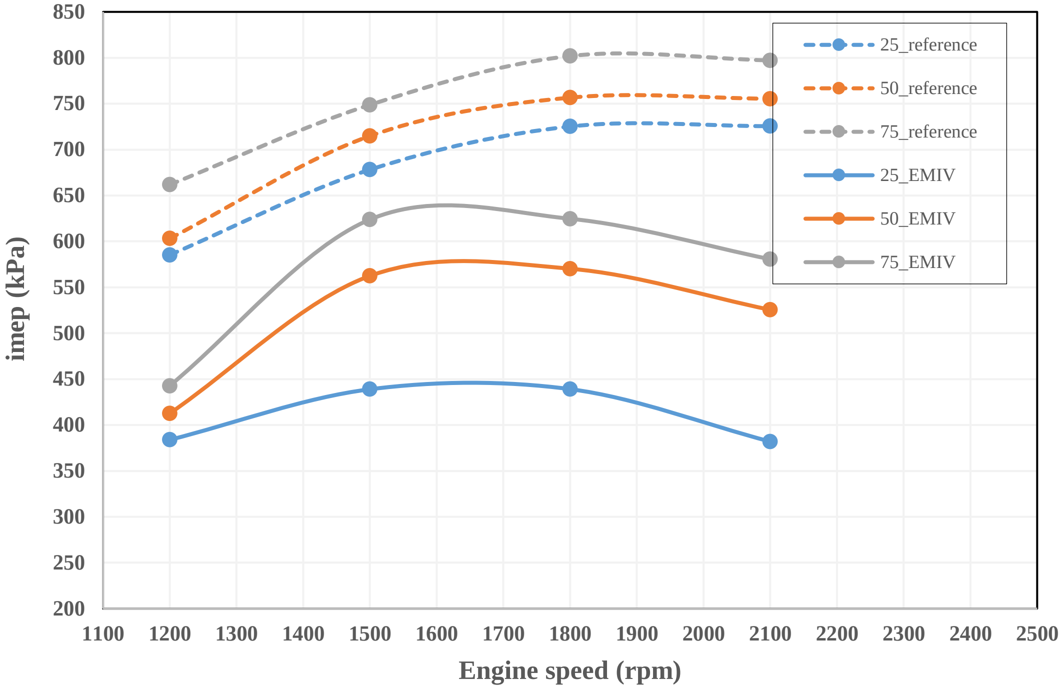


Fig. 7. The imep of the reference engine and EMIV engine.

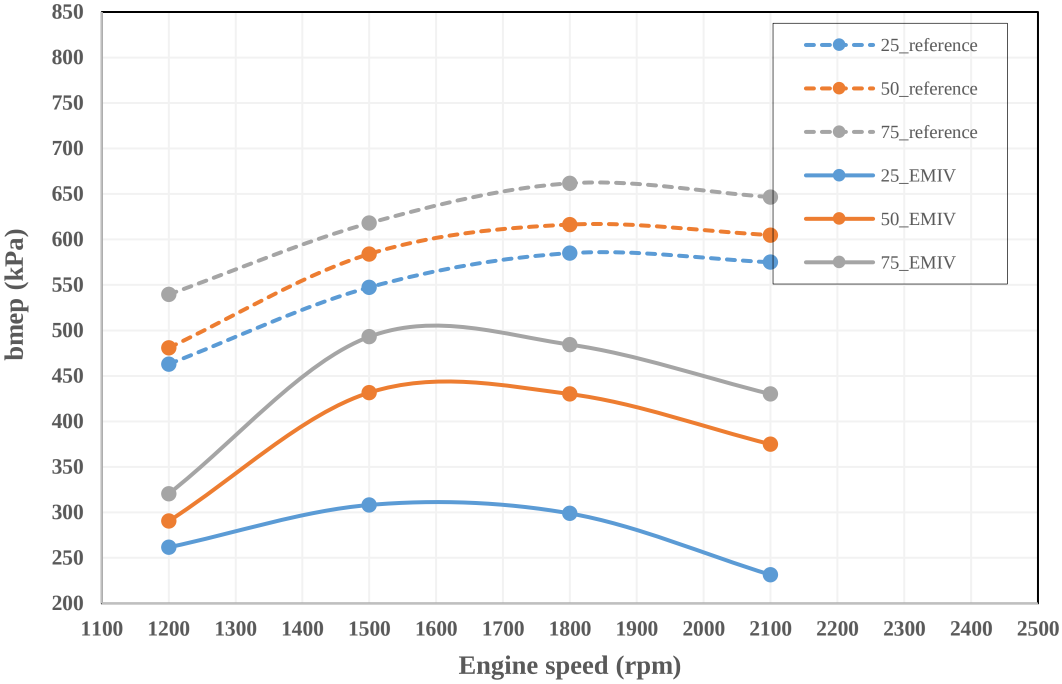


Fig. 8. The bmep of the reference engine and EMIV engine.

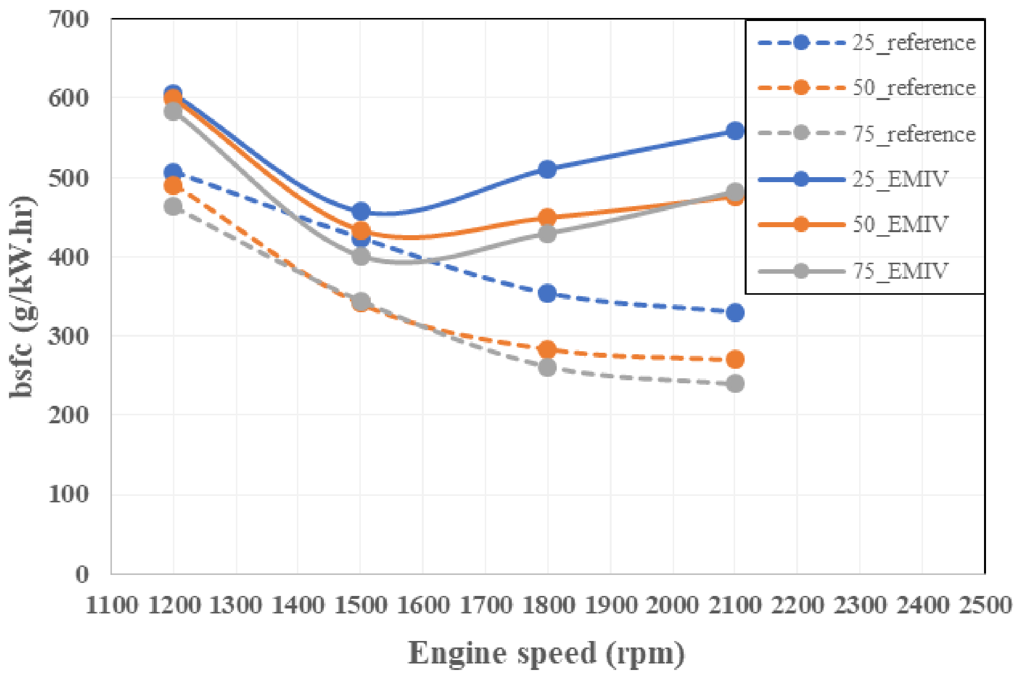


Fig. 9. The bsfc of the reference engine and EMIV engine.

4.6 Indicated thermal efficiency and brake thermal efficiency

To indicate the productivity of the system in terms of the first law of the thermodynamics, the thermal efficiency of engine is assessed as shown in the equation eq.7 and 8. In engine tests, the brake thermal efficiency is used to evaluate how well an engine converts the heat from a fuel to mechanical energy. Fig.10 and 11 shows the variation of indicated thermal efficiency (η_i) and brake brake thermal efficiency (η_b) with engine loads for the reference and EMIV engine at different engine speeds. The EMIV engine gave the maximum indicated thermal efficiency of 27.30 % followed by the engine speed of 1500 rpm and throttle position opening of 75%, which was lower than the reference engine. The brake thermal efficiency of EMIV engine was lower than the brake thermal efficiency of reference engine . The maximum brake thermal efficiency of EMIV engine was 20.4 % when the engine speed and throttle position opening were 1500 rpm and 75 %, respestively. Whereas the brake thermal efficiency of reference engine increased with increasing engine speed in all the throttle position opening. From Fig.10 and 11 shows, the indicated thermal efficiency and brake thermal efficiency of EMIV engine increased first and then decreased with incresing engine speed with different throttle position opening.

In engine speed range of 1200-1500 rpm, the indicated thermal efficiency and brake thermal efficiency of EMIV engine were increased owing to the EMIV system consumes less energy to hold the intake valve open. It is clealy seen that the bsfc of EMIV engine was decrased in the engine speed rang of 1200-1500 rpm. Whereas engine speed range of 1500-2100 rpm, the the indicated thermal efficiency and brake thermal efficiency of EMIV engine were decrased due to increasing the engine speed result in a rise in the energy consumption of EMIV system. The difference between indicated thermal efficiency and brake brake thermal efficiency of EMIV engine represents the friction loss, which was about 1-7 %.

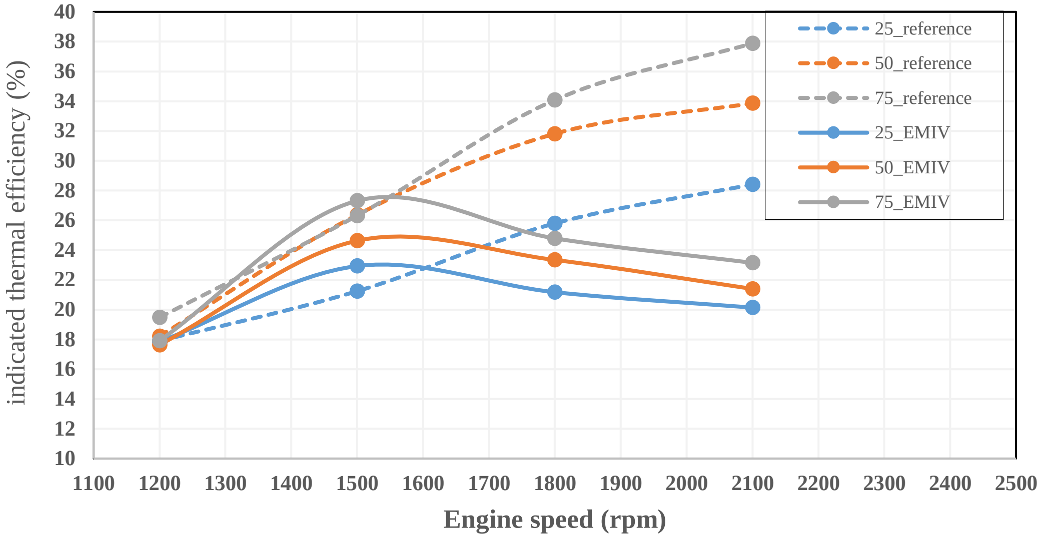


Fig. 10. The indicated thermal efficiency of the reference engine and EMIV engine.

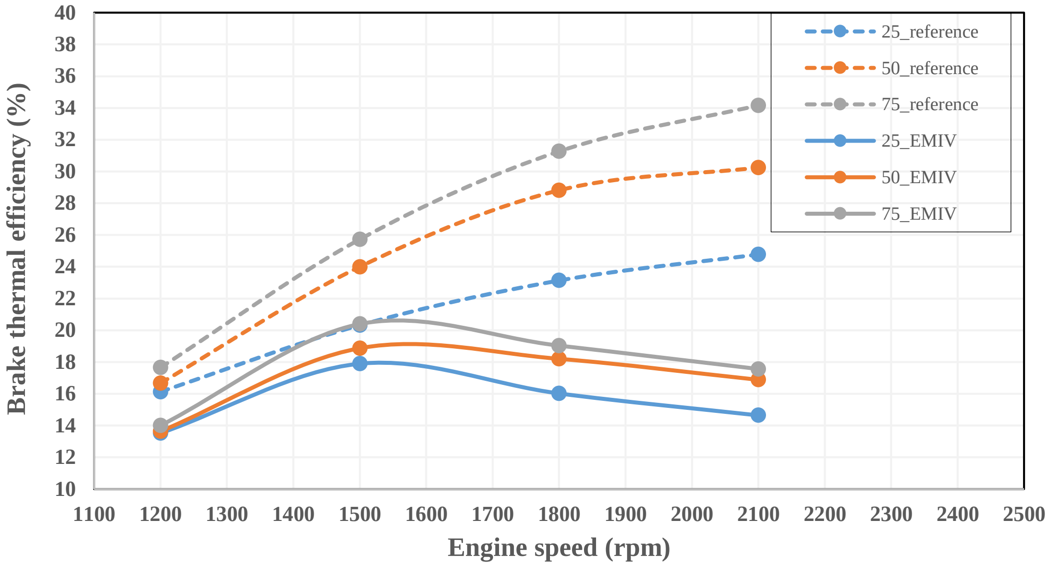


Fig. 11. The brake thermal efficiency of the reference engine and EMIV engine.

5. Conclusion

The Electromagnetic valve system designed for this study works with 12 VDC and 24 Amp. In this paper, the comparative performance of small gasoline engine between the EMIV engine and reference engine with the air mass flow rate, volumetric efficiency, brake power, indicated mean effective pressure, brake mean effective pressure, brake specific fuel consumption, indicated thermal efficiency and brake thermal efficiency. The tests were conducted for a part-load operating condition at four engine speeds (1200, 1,500, 1800 and 2,100 rpm) and under part-load (throttle position opening of 25, 50 and 75 % WOT) for the prototype and EMIV engine. Compared with reference engine, the EMIV engine gave a lower performance than reference engine, due to air mass flow into the EMIV engine less than reference engine. The transition time obtained in EMIV engine cause late intake valve timing. Moreover, the early closing of the intake valves occurred in EMIV engine. Reason for this is that the cylinder intake period has been already small. It has affected to volumetric efficiency, indicated mean effective pressure, brake mean effective pressure, brake specific fuel consumption, indicated thermal efficiency and brake thermal efficiency. Which the air mass flow rate, volumetric efficiency, brake power and brake mean effective pressure, indicated thermal efficiency and brake thermal efficiency produced with the EMIV engine had lower than the prototype engine. As a result, the EMIV engine gave higher the brake specific fuel consumption than the prototype engine.

In this work, the EMIV system can be used to opening the intake valve for replacing the camshaft in the small spark-ignited engine. The results show that the EMIV system may make this design suitable for application in an internal combustion engine. As a future study authors consider to the improvement of the EMIV system which can further development the variable valve timing technology. The new EMIV system can be used more efficiently with the use of stronger actuators, sensors and microcontrollers. The effects of new EMIV system on the engine performance and fuel consumption can be investigated.

NOMENCLATURE

bmep	brake mean effective pressure, Pa
bsfc	brake specific fuel consumption, g/kW.hr
$\dot{m}$	mass flow rate, kg/s
N	engine speed, rpm
n	number of crank revolution for each power stroke per cylinder
T	torque,N-m
W	power, watt
V	Volume, m^3
ρ	fluid density, kg/m^3
η	efficiency

Subscripts

a	air
b	brake
d	displacement
f	fuel
i	indicated
r	revolution
v	volume

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