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Research paper

The Influence of Air-Fuel Equivalence Ratio on the Performance and Emission Characteristics of a Crank-rocker Engine

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Abstract

This paper investigates the effect of air-fuel equivalence ratio on the crank-rocker engine performance and emission. The engine is a four-stroke single curved-cylinder spark ignition gasoline engine. The experiment was conducted at different equivalence ratios of 0.7, 0.96, 1.15, and 1.21 with each one operating at 50% throttle. The entire tests were carried out at an engine speed of 2000 rpm and 6.5° CA BTDC ignition timing. The performance data such as brake torque, brake power, brake specific fuel consumption, and brake thermal efficiency were calculated. The engine exhaust gas emission such as CO, CO2, HC, and NOx have also been measured. The results showed that the maximum values for the brake torque and power occurred when air-fuel equivalence ratio was equal to 0.91. While the minimum brake specific fuel consumption occurred when the equivalence ratio was equal to 1.15; which is the point of the maximum brake thermal efficiency. From the experimental results, it was found that the concentration of CO and HC emissions decreased while CO2 and NOx emissions increased with the increase in air-fuel equivalence ratio. To conclude, the crank-rocker engine may become an alternative to the conventional engine in the future but further research work will be required.

Keywords: *Air-fuel equivalence ratio; Curved-cylinder; Crank-rocker engine; Performance and emissions*

1. Introduction

Improving internal combustion engine performance and efficiency is the main concern for automotive engineers nowadays. In general, most contemporary researchers and automotive companies are working on the optimizations of the ICEs, and have intensified efforts at improving the performance and efficiency ever since the first engine was manufactured. As a result, many researchers have proposed different techniques and optimization methods for controlling the engine operating parameters in order to gain high efficiency. Engine control parameters, such as compression ratio, ignition timing, air-fuel equivalence ratio, and engine throttle position are the most significant parameters in the spark ignition (SI) engine. Many studies had been conducted experimentally and numerically in order to investigate the effects of these parameters on the engine performance and exhaust emissions [1-4].

Air-fuel equivalence ratio is the most significant parameters which influence the engine combustion behaviour in the spark ignition engine, mainly because there are restrictions for rich and lean regions. Super rich mixture causes the engine to misfire, while super lean mixture makes the engine hard to start or run at all. Previous studies had been conducted in order to investigate the effect of air-fuel equivalence ratio on spark-ignition engine fueled either by pure gasoline or by other blended fuels [5-9].

Rahman et al. [3] studied the effect of air-fuel equivalence ratio (λ) on SI engine performance. They found that brake thermal efficiency (BTE), and brake mean effective pressure (BMEP) decreased with the increasing of air-fuel equivalence ratio, whereas brake specific fuel combustion (BSFC) increased with the increase in air-fuel equivalence ratio. Salimi et al. [10] had carried out a similar work numerically and experimentally on a four-cylinder SI engine. Analyzed data from the numerical model engine were compared with experimental data. The results showed that BMEP increased almost linearly with the increase in air-fuel equivalence ratio from rich to lean limit. Furthermore, the effect of air-fuel equivalence ratio on engine brake power had been discussed in this work. From the result, it was shown that maximum value of brake power occurred when the air-fuel equivalence ratio was about 1, (stoichiometric), and this happened for all the cases of engine speeds [7]. The peak value of NO_x emission occurred when the air-fuel equivalence ratio was about 0.8, and after this point, it started to decrease.

One of the major problems associated with the current conventional engine configuration is that the piston is slapping against the cylinder wall during engine operation, which causes wear. Thus, this may result in loss of compression pressure and engine efficiency due to leakage. Another inherent problem of the conventional slider-crank engine is that, when the piston reaches the top dead centre, the gas force is at the peak when the piston in this position. However, the engine could not deliver high torque to the crankshaft due to the poor transmission angle. As a result, many engineers, inventors, and automobile manufacturers have put intensified efforts in developing a more powerful and efficient reliable engine since the first ICE was manufactured. They were trying to make the ICEs much better but different from the norms, by looking for alternative engine configurations in order to increase engine performance, efficiency, and power-to-weight ratio. The examples of new engine configurations which have been addressed and advanced are the radial engine, opposed piston engine, duke engine, rotary engine (Wankel), toroidal engine (curved cylinder) and etc.

For many years, researchers had more interest in replacing the conventional reciprocating engines with oscillating curved cylinder engines, and some of them patented their work [11, 12]. However, their researches did not continue could be due to complexity in designs, difficulty in manufacturing and lack of fabrication facilities. Recently, the toroidal engines have gained more attention and concern by automobile manufacturers. Some inventors had attempted to design engines incorporating oscillating curved pistons in order to increase engine efficiency.

Farrell [13], Hoose [14] and Morgado[15], invented and designed different type of curved cylinder toroidal engines, however, the proposed designs were noted to be very complex. Oscillating curved cylinder engine with opposed piston was also patented and developed by Hüttlin[16]. Despite the advantages of this engine, it also has few drawbacks such as the rotation speed was limited at 3000 rpm, very complicated, and it had low power density [17]. It can be noted that most of the toroidal curved cylinder engines were used for two-stroke engines. A four-stroke toroidal curved-piston engine was proposed and invented by Taurozzi [18]. Although this engine could be fully balanced, the engine concept was quite complicated, and this engine had low efficiency [17]. As a conclusion, previous inventors wanted to improve engine efficiency by changing engine configurations. Many types of different mechanisms with oscillating motions have been proposed, but the potential problems associated with these mechanisms are that they are very complicated. An alternative way of producing a crank output motion with oscillating curved-piston is through utilizing a four-bar or crank-rocker mechanism.

In this study, a new engine is introduced and referred to as the Crank-Rocker (CR) engine. The most unique feature of this engine is that it is very simple in design and easy to be manufactured. In addition to that, the special feature of this engine is that it is the combination of both the conventional and toroidal engines. The engine can be modified easily to work on alternative fuels such as compressed natural gas (CNG), hydrogen, ethanol and biodiesel. The proposed design for the new engine is shown in Figure 1. As can be seen from the figure, a single-curved piston assembly travels within a curved engine cylinder.

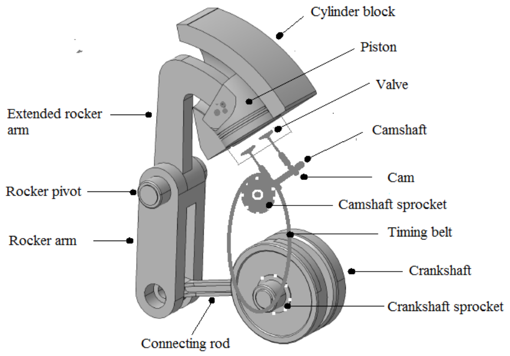


Fig. 1: A proposed design for the Crank-rocker engine geometry.

In this work, the effect of air-fuel equivalence ratio on the crank-rocker engine performance and exhaust emissions was experimentally investigated. The experiment was performed at a constant engine speed of 2000 rpm and 50% open throttle.

2. Methodology

This section describes the experimental setup and procedures adopted for collecting and analyzing the needed data for this project. The equipment which was used, engine parameters and data collection systems are described. The engine used in the present work was a crank-rocker, single curved-cylinder, four-stroke spark-ignition with a swept volume of 120 cc and a compression ratio of 8:1. The engine specifications are given in Table 1.

Table 1: The crank-rocker engine main data

NO	Parameters	Crank-rocker
1	Cylinder Displacement	120 cc
2	Cylinder Diameter	55 mm
3	Number of Strokes	4
4	Number of Cylinders	1
5	Stroke	50.6 mm
6	Compression Ratio	8:1
7	Connecting Rod	100 mm
8	Rocker and Extended-rocker Length	138.9 and 138.9 mm
9	Throw Angle	21°
10	Fuel	gasoline

The schematic diagram of the experimental setup is shown in Figure 2. All experimental studies were conducted at the Centre for Automotive Research and Electric Mobility (CAREM) located at the Department of Mechanical Engineering, Universiti Teknologi Petronas (UTP), Malaysia.

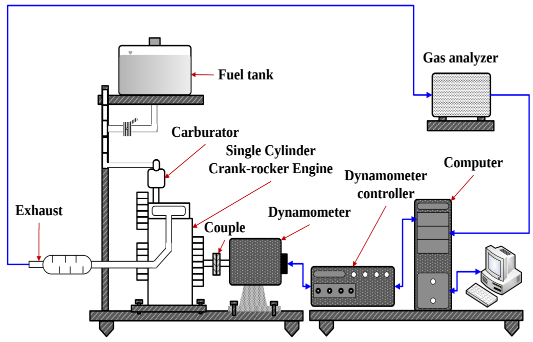


Fig. 2: The schematic diagram of the experimental setup.

The crank-rocker engine was originally designed for gasoline fuel but with a little modification and it could be used for any type of fuels. The engine was coupled to an eddy current dynamometer that allowed the engine braking while the performance parameters were measured. Table 2 shows the specifications of the dynamometer.

Table 2: The dynamometer specifications

NO	Iake and Model	Focus Applied Technologies
1	Type	Eddy current
2	Capacity	5 kW
3	Iaximum Speed	4500 rpm
4	aximum Torque	20 Nm
5	orque Accuracy	± 0.1 Nm
6	peed Accuracy	± 10 rpm

For each operating point, the data was recorded once the engine had stabilized, after 4-5 minutes. The engine performance parameters such as BP, BSFC, and BTE were calculated from the following equations:

$$BP = {}^{2\pi NT}\!/_{60}\,(W) \tag{1}$$

$$BSFC = {}^{\dot{m}_f}\!/_{BP}\left({}^{g}\!/_{kWh}\right) \tag{2}$$

$$BTE = {}^{BP}\!/_{\dot{m}_f\, CV_f}\quad(\%) \tag{3}$$

A BEA 460® Bosch analyzer is used to measure the exhaust gases in the engine. The gas analyzer is capable of measuring about 5 gas species in the exhaust and providing the reading in percentage and parts per million (ppm). The gas analyzer measures the CO, CO_2, O2, HC and NO_x exhaust gas components for gasoline engines with high measuring accuracy for each gas. Figure 3.31

shows the BEA 460® Bosch analyzer. The analyzer is able to calculate the lambda values based on the oxygen concentration. By using lambda and stoichiometric air to fuel ratio, it is capable to calculate the actual air to fuel ratio. The BEA 460® can be connected to standard laptops via cable or Bluetooth in order to run the test sequence and display measurement values. The measuring system is controlled by the specified software from Bosch. The program automatically optimizes exhaust gas analysis and therefore ensures a rapid and economic test sequence. Table 3 shows the specifications of the Bosch analyzer.

Table 3: Bosch exhaust gas analyzer specifications

NO	Standard method	Measure	Range	Accuracy
1	Non –dispersive infrared	CO	0-10 vol%	± 0.001 vol%
2	Flame ionization detector	HC	0-9999 ppm	± 1ppm
3	Non –dispersive infrared	CO2	0- 18 vol%	± 0.01 vol%
4	Electro- chemical transmitter	NOx	0-5000 ppm	± 1ppm
5	Electro- chemical	O2	0-22 vol%	0.01 vol%

The engine test bed unit for testing of the new engine has also been designed and installed to fit the engine requirements. The crank-rocker engine which is completely assembled with all components and accessories on the test bed is shown in Figure 3.

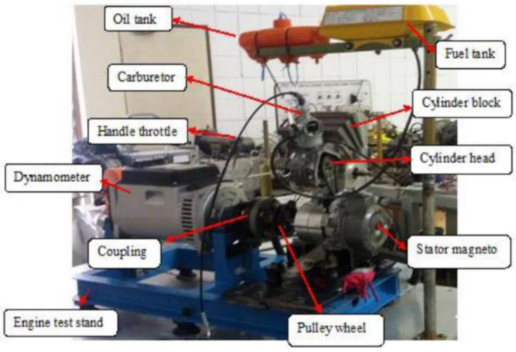


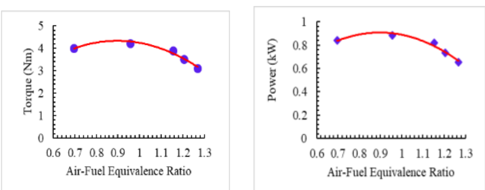
Fig. 3: The crank-rocker engine and the accessories on the test bed.

3. Results and Discussion

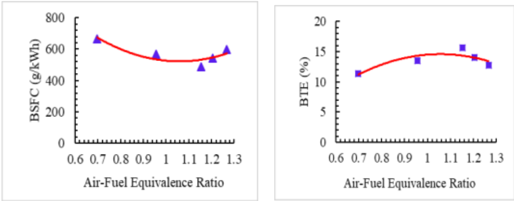
The effects of air-fuel equivalence ratio (λ) on the engine performance and emission characteristics are investigated. The experiment was conducted at different air-fuel equivalence ratios of 0.7, 0.96, 1.15, and 1.21 with each one operating at half open throttle (50%) conditions. The entire tests were carried out at an engine speed of 2000 rpm.

3.1 Engine Performance

Figure 4a and 4b show the engine brake torque and brake power versus the air-fuel equivalence ratio at 50% throttle conditions. In general, it can be clearly seen that both brake torque and power curves increase up to the maximum value at an air-fuel equivalence ratio of 0.91 and then start to decrease with the increase in air-fuel equivalence ratio. The maximum values for the brake torque and power are found to be 4.23 Nm and 0.96 kW respectively.



(a) (b)



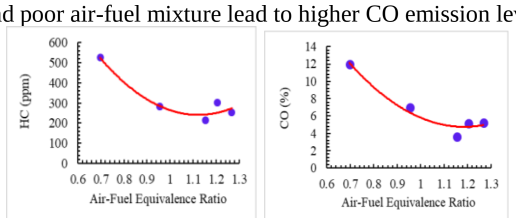
(c) (d)

Fig. 4: (a-c). (a) BT (b) BP (c) BSFC (d) BTE vs air-fuel equivalence ratio at 50% throttle and 2000 rpm

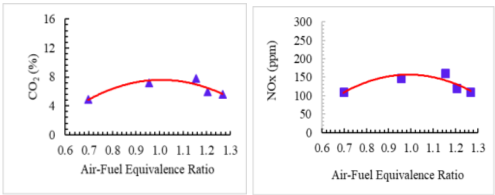
The reason behind this is that at this point (i.e. air-fuel equivalence ratio 0.91) the mixtures burn faster than any other points of air-fuel equivalence ratio, and this leads to the increase in mixture temperature. Higher temperature results in higher pressure inside the cylinder. The higher the pressure inside the cylinder, the more force will act on piston providing higher torque as well as higher power output.
Figure 4c shows the BSFC plotted against air-fuel equivalence ratio at 50% throttle and 2000 rpm. In general, the BSFC decreases with the increase in the air-fuel equivalence ratio until 1.15 and then starts to increase with the increase in the air-fuel equivalence ratio. The BSFC for all the cases is found to drop to the lowest value (i.e. 486.94 g/kWh) when air-fuel equivalence ratio is about 1.15. At richer and leaner mixtures, the BSFC is high mainly due to the high amount of fuel injected at the richer region, while at leaner one is due to the lower power output. The brake thermal efficiency (BTE) characteristics for different air-fuel equivalence ratio at 50% throttle is shown in figure 4d. As expected, the characteristics of the BTE plot is the inverse of the plot of BSFC. It can be observed from the figure that the BTE values are found to increase with the increase in air-fuel equivalence ratio up to 1.15 and then starts to decrease.

3.2 Engine Emission

The influence of air-fuel equivalence ratio on the HC emission is shown in Figure 5a. As can be seen from the figure, at rich mixtures (i.e. when air-fuel equivalence ratio is 0.7) the engine produces greater HC concentrations due to incomplete combustion. On the other hand, the HC emissions decrease as the air-fuel equivalence ratio increases.
Extra air in the fuel mixture also causes the HC concentrations to rise. The explanation to this phenomena is owing to the uneven distribution of the air-fuel mixture inside the combustion chamber and this leads to miss-ignition in the lean combustion chamber regions.
Figure 5b shows the effects of air-fuel equivalence ratio on the CO concentrations. As a response to variations in lambda, CO shows similar trend curves as HC emissions. It can be clearly seen in this figure that the CO emissions decrease as the air-fuel equivalence ratio increases from rich to lean. For rich mixtures, the CO emissions show the highest value which results in incomplete carbon oxidation during the combustion process due to the insufficiency of air. In general, it can be concluded that the incomplete combustion and poor air-fuel mixture lead to higher CO emission level.



(a) (b)



(c) (d)

Fig.5: (a-d). (a) HC (b) CO (c) CO_2 (d) NO_x emissions vs air-fuel equivalence ratio at 50% throttle and 2000 rpm

The effect of lambda on CO_2 emission is represented in Figure 5c. It is observed from figure 5c that CO_2 increases when λ increases. CO_2 formation depends upon the carbon-hydrogen ratio of the fuel and the combustion process, so CO_2 concentrations increase with the increase in lambda as a result of good combustion. At leaner mixture, more oxygen is a presence in the combustion chamber and this leads to incomplete combustion resulting in the decrease of CO_2 concentrations.
Figure 5d shows the effect of air-fuel equivalence ratio on NO_x emission at WOT and 2000 rpm. In general, NO_x emission is strongly dependent on the air-fuel equivalence ratio [19]. As λ increases, NO_x emission increases until λ equals 1.2, after that it starts to decrease with the increase in λ. To explain this, more air in the mixture leads to lower combustion temperature that results in the NO_x concentrations to decrease.

4. Conclusion

Based on the investigation of the effect of air-fuel equivalence ratio, it was found that the maximum values for the brake torque and power were 6 Nm and 1 kW respectively when air-fuel equivalence ratio was equal to 0.91. The BSFC decreased with the increase in the air-fuel equivalence ratio until λ equal 1.15 and then starts to increase with the increase in the air-fuel equivalence ratio. The BTE for all the cases increased to the highest value (i.e. 11.5%) when λ equal 1.15; which is the point of the minimum BSFC. In general, CO and HC emissions decreased while CO_2 and NO_x emissions increased with the increase in air-fuel equivalence ratio. It can be said that this engine behaves similarly to the conventional engine in term of performance and emission with slight differences.

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