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The effects of combustion duration on residual gas, effective release energy, engine power and engine emissions characteristics of the motorcycle engine

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HIGHLIGHTS

- Operation of a motorcycle engine is studied based on the experiment and simulation model.
- Effects of combustion duration on engine performance and emission characteristics are investigated.
- The optimal combustion duration at each engine speed is investigated.

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ABSTRACT

The purpose of this research is to study how combustion duration affects the performance and emission characteristics of motorcycle engines. From the results, the researcher is able to know the best combustion duration value that gives the target engine torque, NO_x, CO and HC emission. To achieve this goal, an experimental system was installed within a dynamo testing system, and a simulation model was established using AVL-Boost software. The simulation model was used to determine the residual gas ratio, effective release energy, and engine emission characteristics in variable combustion durations (40–110 degrees crank angle). An engine speed band of about 3000–10,000 rpm was adopted throughout the experimental process. It is worth noting that the combustion duration had a significant effect on the residual gas ratio, effective release energy, engine performance, and emission characteristics. When the engine speed was 6000 rpm at a combustion duration value of 80 degrees, the minimum residual gas ratio was 0.22%. At a combustion duration value of 60 degrees, the maximum effective release energy was 0.826 KJ. At the engine speed was 7000 rpm and 8000 rpm, the minimum residual gas ratio was 0.14% and 0.15%, respectively; the maximum effective release energy was 0.831 KJ and 0.8247 KJ at 110 degrees and 80 degrees combustion durations respectively. This research also pointed out the optimal combustion duration at each engine's speed. Also, the residual gas had a close relationship with engine emission characteristics. At 6000 rpm and 60 degrees combustion duration, the maximum engine torque was 22.7 Nm and the minimum BSFC was 320 g/(KWh).

1. Introduction

In the four-stroke SI engine, the power stroke is the most important stroke. The combustion process has been the major factor affecting engine performance with pollutant emissions associated to the conversion of the chemical energy gotten from fuel into thermal energy, and the emission of harmful gases such as NO_x, CO, HC, and PM. Because of the combustion process had a significant effect on the engine performance and emission characteristics so the parameters involve in the combustion duration process also were studied, for example: the effect of flame stretch on fuel performance of spark ignition engine [1]. Lean-burn mixture was used to improve combustion quality and reducing heat transfer losses [2]. The impact of the combustion phasing on the performance of dual-fuel engine [3]. From literatures, higher effective combustion process has been reported through newly applied technologies such as: an improvement of the thermal efficiency and knock resistance of a gasoline engine by applying a Miller cycle with split injection [4]. A multiple spark discharge using multi-coil ignition system was applied to ensure the stable, complete and fast combustion of lean SI engine operation [5]. Using direct injection with low-pressure strategies to improve efficiency and reduce emission of a small SI natural-gas two-stroke engine [6].

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Nomenclature

BDC	bottom dead centre
TDC	top dead centre
BTDC	before top dead centre
ATDC	after top dead centre
BSFC	brake specific fuel consumption, g/KWh
BMEP	brake mean effective pressure, Bar
IMEP	indicated mean effective pressure, Bar
FMEP	friction mean effective pressure, Bar
SMEP	scavenging mean effective pressure, Bar
HC	hydrocarbon
PM	particulate matter
THC	total hydrocarbon emissions
CA	crank angle, deg
ERG	exhaust gas recirculation
SI-engine	spark ignition engine
CI-engine	combustion ignition engine
η_{th}	theoretical engine efficiency
Q	total fuel heat input, W
α	crank angle, deg
α_0	start of combustion, deg
$\Delta\alpha_c$	combustion duration, deg,
m	shape parameter, (–)
a	vibe parameter
Q_T	heat lost to the wall, W/m^2
A	total surface area of cylinder head, piston and cylinder, m^2
q_{coeff}	heat transfer coefficient, W/m^2 K
T_c	combustion gas temperature, K
T_w	wall temperature of cylinder (K)
$\frac{dm}{dt}$	air mass flow rate (–)
A_{eff}	effective flow area (–)
P_1	upstream stagnation pressure, Pa
T_1	upstream stagnation temperature, K
R_0	gas constant
P_2	downstream static pressure, Pa
K	ratio of specific heats (–)
V_D	displacement volume, m^3
T_{eff}	engine effective torque, Nm
P_{eff}	engine effective power, kW
K_{cycle}	simulation cycle parameter, cycle
n_{cycle}	number of cycle per second, cycle/sec
ε	fuel consumption rate, g/s
r	compression ratio, (–
γ	specific heat ratio, (–)
C_p	specific heat capacity with constant pressure, J/(kg K)
C_v	specific heat capacity with constant volume, J/(kg K)

In the power stroke, the combustion duration is a crucial parameter that indicates the optimum burning process. Generally, for extremely short combustion duration, an incomplete conversion of fuel from chemical energy to thermal energy seems to exist. Moreover, excessive thermal energy loss tends to occur during long combustion duration due to the increase in the duration for the heat transfer process in and out of the cylinder, piston and flowing exhaust gas. Some recently published papers relating to combustion duration presented the combustion duration effect on engine performance as well as the parameters affecting the combustion duration such as: ignition timing, engine speeds, engine load conditions, and fuels etc. [7]. This research reflected on the trend of combustion duration discovered in HICE and affirms that when the equivalence ratio was greater than 0.6, the combustion duration was almost independent of engine speed. However, the low speed and leaner mixtures of the combustion duration were reduced sharply when engine speed increases [8]. An SI-engine was studied, and the effect of air-fuel ratio, engine speed, spark advance, and compression ratio in combustion duration was found based on the comparison between the experimental method and theory to determine the combustion duration, which uses the empirical correlation method [9]. The presented investigation was of the combustion duration of a diesel engine that used a dual fuel and hydrogen engine. The research shows that the combustion duration increases 2.5 degrees CA and the ignition advance decreases 2 degrees CA when the engine operated at a light load. At 80% load and 50% volume of hydrogen in the fuel, the combustion duration increased to a maximum of 9 degrees CA, and the ignition advance decreases to 6 degrees CA [10]. The engine combustion duration had a large effect on engine performance and NO_x emission of the engine with dual-fuel diesel-LNG. At a light load, the combustion duration increased from 1.7 to 6.0 degrees. After the top dead centre, the engine thermal efficiency increased to 16.7%, and the NO_x declined. However, THC and CO emissions increased. If the advance of combustion duration was before top dead centre (TDC) then the NO_x emission will increase [11]. The engine working at light load saw the combustion duration increase to 5.0 degrees CA compared to a hydrogen or diesel engine. However, the combustion duration was reduced by 5.0 degrees CA at the engine load around 80.0% with a mixing fuel of 50.0% hydrogen and 30.0% LPG [12]. This suggests to us that both the main and initial combustion durations were affected by the addition of hydrogen; when the amount of hydrogen was increased, the combustion duration decreased [13]. Haroun A.K, Maher A.R. showed that the addition of hydrogen affected combustion duration and increased flame speed, leading to complete combustion. When the amount of hydrogen increased to 10.0% in volume, the combustion duration decreased to 24.0% and the flame speed increased by 46.0%. Also, the combustion duration decreased the heat transfer rate so that engine efficiency increases [14]. The combustion duration has decreased from the norm and two stages of combustion in the same condition as the coke oven gas. In normal combustion phases, the combustion duration changes from 24.0 to 29.0 degrees of CA, while that of two-stages combustion changes from 11.0 to 14.0 degrees of CA [15]. The optimum engine combustion duration is said to vary from 24.5 to 27.5 degrees of CA upon addition of 20.0% hydrogen and from 28.5 to 32.0 degrees of CA when 13.7% hydrogen was added. However, in two-stages combustion, the combustion duration was reduced at excess amount of hydrogen [16]. Using the diesel engine for research, the effects of engine load and volume hydrogen addition on combustion duration were presented. If a small amount of hydrogen was added, the effects on combustion duration would not be high. At a 70.0% load, the hydrogen volume fraction decreases the combustion duration, by using a zero-dimension, two-zone model to study the effect of the combustion process on thermal efficiency and to determine the best thermal efficiency in each combustion phase [17]. In this research, the best thermal efficiency occurred when the maximum pressure in the cylinder was 9.0 degrees after TDC. The development of combustion duration had a higher heat transfer rate [18], A compression ignition engine using dual fuel hydrogen-ethanol was set in 1500 rpm, 100% load, and 0–80% hydrogen volume fraction. The appropriate condition in terms of engine performance was a combustion duration range of 35–42 degrees [19]. An analytical model was used to study the combustion duration effect on engine performance, and emission characteristics for the four-stroke, spark ignition engine were analyzed using propane as fuel. The results have shown that the combustion duration has a large effect on engine performance and emission characteristics, so the designer should carefully design the engine to achieve optimal performance characteristics.

All previous studies have focused on the heavy-duty engine with alternative fuel as well as used dual fuel (hydrogen-ethanol) and LPG fuel. However, these results were difficult to apply in a small SI-engine since it failed to shed light on the combustion duration effect on residual gas and effective release energy. Besides that, many previous studies reported the combustion duration effect on engine torque, engine power without elaborating on the effect on engine emission characteristics. Other studies presented the effects of combustion duration on engine emission characteristics but nothing was stressed on its effects on engine torque, BSFC, IMEP, BMEP. In our research, the combustion duration effect on the engine performance and engine emission characteristics are presented.

In the previous studies due to the lack of control and lack of ability to determine the precise combustion duration inside the cylinder. These were the main reasons for the loss of time and researching cost. However, through combined experimental and simulative methods, we have been able to eliminate these drawbacks.

Thus, this article has presented the combustion duration effect on residual gas and effective release energy, engine performance, and emission characteristic. A laboratory system was established, and the small SI-engine was studied via a simulated model to investigate the combustion duration. The investigated engine parameters were residual gas, effective release energy, BSFC, IMEP, BMEP, peak firing temperature, engine torque, engine power, NO_x, CO, and HC emissions. Also, the relationship between the optimum combustion duration at each engine speed, residual gas, effective release energy, and engine emission were discussed in this present work.

2. Experimental setup and material

2.1. Experiment setup

Figs. 1 and 2 represent the schematics diagrams of the small-SI engine testing system and experimental setup respectively. The experiment system consist of 22 parts: A controller of the dynamo testing system (1), the AVL dynamometer (MCA325MO2) (2) provide the resistant moment for the testing system and this dynamometer was connecting with the engine through the coupling (3). On the test engine (4) and the flywheel (5), the encoder (E40S8-1800-3-T-24) (6) and the temperature sensor (7) were situated. Gasoline fuel from the fuel tank (8) travels to the fuel injector (11) through the fuel pump (9) and the fuel filter (10). Air enters the intake tube through the air cleaner box (15) and throttle (16). The air heater (17) was used to control inlet air temperature. The exhaust gas analyzer, cylinder pressure transducer (19), ECU (20), data acquisition (21) and computer (22) receives signals from the oxygen sensor (12), exhaust gas temperature sensor (13), cylinder pressure sensor (14), throttle angle and air mass from the sensor (18). The experimental conditions include a compression ratio at 11.8:1.0, air-fuel ratio at 13.6, while the test environment temperature ranges from 29.5 °C to 30 °C. For the used air coolant, the oil temperature inside the engine was maintained at 80 °C, while the engine ran and the throttle was angled at 100% (opening) while the fuel system maintain a pressure range of 333–362 kPa. All the equipment was calibrated before analysis. The engine provided maximum engine torque in the range of engine speed from 6000 rpm to 8000 rpm. Also, this research noted the optimal combustion duration at the best engine torque. Moreover, we studied the effects of combustion duration on residual gas, effective release energy, engine performance, and engine emission characteristics. Based on all this indices and yard stick, the research resulted in an engine speed of 6000 rpm, 7000 rpm, and 8000 rpm respectively.

2.2. Applied engine

A small SI-engine (four strokes, V-2 cylinders, and 8-valves) was used in the experiments. This engine is equipped for a motorbike. The engine was fitted with port type fuel injection system and flat piston. This engine consists of two-cam shafts, one for controlling intake valves and the other one for controlling exhaust valves.

The applied engine specifications are shown in Table 1.

2.3. Simulation model setup

The AVL-Boost is a well-known simulation software in the internal combustion engine field. It allows researchers to simulate all types of combustion engines, such as the SI engine [20], CI engine [21,22], turbocharged diesel engine [23,24] or engine with alternative fuel [25]. Fig. 3 represents the simulation model in a small-SI engine.

The elements in the simulation model describes the parts of the engine used in the experiment. The simulation elements were used to study the characteristics of the engine parts. The test condition was set at steady state or transient state when permitted to select in the engine element E1. The monitor MNT1 was an extra element that was not present in the research engine. With the monitor element, researchers were able to observe all the selected output data such as engine torque, residual gas ratio, and effective release energy. The element SB1 and SB2 describes the system boundary of the intake and exhaust pipe. The element CL1 was used as the air cleaner for the system. In the element TH1, the opening of the throttle angle was set. In this research, the throttle was fully opened (100%), and the pressure loss in the intake and exhaust pipes (1, 2, 3…21) is described by element restriction (R1, R2, and R3). The element junctions (J1, J2, J3, J5, and J6) were used to collect or distribute the flow through the pipe. Measurements of elements MP1 and MP2 in the intake and exhaust pipes were used to determine the flow characteristic (e.g., air mass flow, flow velocity, airflow temperature). Injectors I1 and I2 were used to provide fuel to cylinders C1 and C2.

It was hard to control the combustion duration and to calculate the residual gas ratio and effective release energy through an experimental method. Therefore, a simulated method is known to be an effective and powerful tool to solve this problem. In this research, the simulated model was used to study the effect of combustion duration on residual gas and effective release energy. Based on the simulated results, it was easy to determine the residual gas ratio in the cylinder and the released energy. The simulated results show a correlation between the residual gas ratio and effective release energy. The residual gas has a significant effect on engine performance and engine emission. In diesel engines, the EGR technique was an effective way to reduce NO_x emissions [26–29]. The residual gas sometimes has a positive effect on engine efficiency [30]. The engine efficiency was calculated via equations (1) & (2) below [31].

$$\eta_{th} = 1 - \frac{1}{r^{\gamma-1}} \tag{1}$$

$$\gamma = \frac{C_p}{C_V} \tag{2}$$

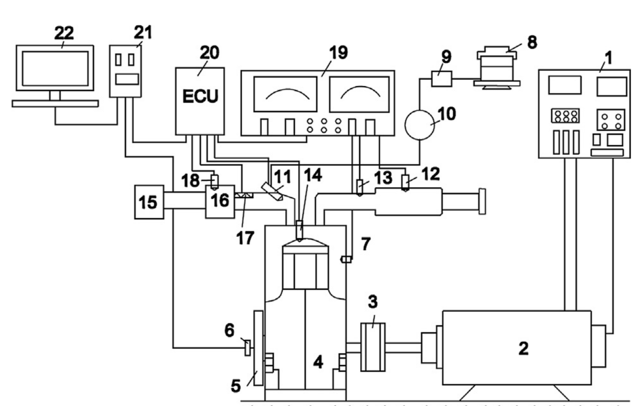

Fig. 1. Schematic diagram of the small-SI engine experimental setup.

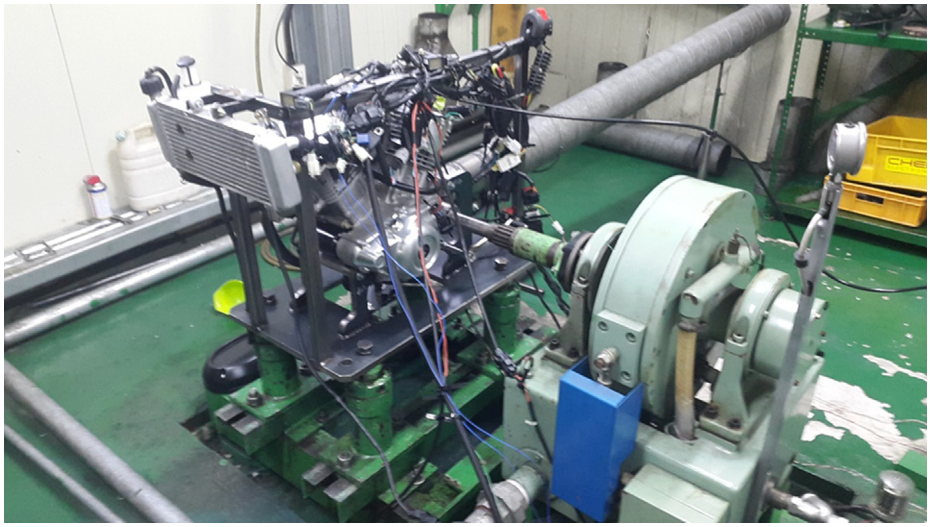

Fig. 2. The experimental system.

Table 1
Engine specifications.

Parameter	Unit	Value
Engine model	–	Four stroke, Spark ignition
Number of cylinder	–	2(V-Twin)
Compression ratio	–	11.8:1
Bore	mm	57
Stroke	mm	53.8
Connecting rod	mm	107.9
Intake valve	–	2
Exhaust valve	–	2
Cooling system	–	Air cooled

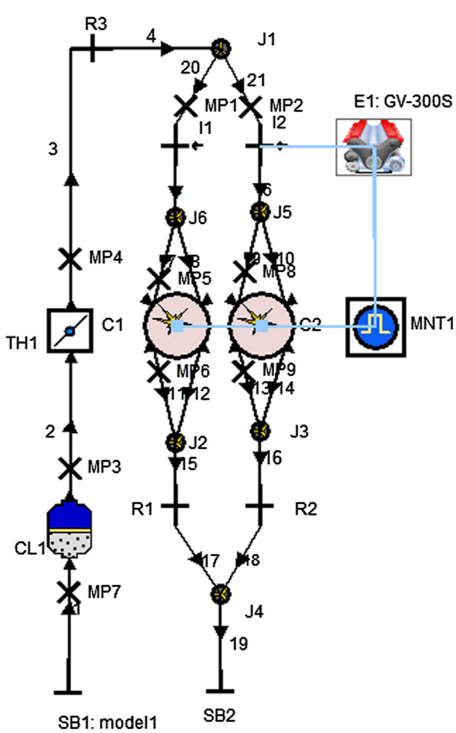

Fig. 3. Simulation model.

where

η_{th} is theoretical engine efficiency;
r is the compression ratio;
γ is the specific heat ratio (depend on monatomic gas or diatomic gas);
C_p is specific heat capacity with constant pressure (J/(kg K);
C_v is specific heat capacity with constant volume (J/kg K).

Eq. (1) shows two methods to improve engine efficiency: by increasing engine compression ratio or by increasing specific heat ratio. The residual gas is a factor that influenced the specific heat ratio because it causes N_2 gas to increase in the cylinder. Meanwhile, the specific heat ratio of N_2 gas (1.47) is greater than that of the mixed gas in the cylinder (1.2–1.4). The effects of combustion duration on residual gas ratio, effective release energy, and engine performance will be studied from the results of this research.

The simulated engine model was established based on the Vibe function to calculate heat release characteristics:

$$\frac{dx}{d\alpha} = \frac{a}{\Delta\alpha}(m+1)y^{m}\cdot e^{-a.y(m+1)} \tag{3}$$

$$dx = \frac{dQ}{Q}; \quad y = \frac{\alpha - \alpha_0}{\Delta\alpha_c}$$

where

Q is total fuel heat input (W);
α is the crank angle (deg);
α_0 is the start of combustion (deg);
$\Delta\alpha_c$ is combustion duration (deg);
m is shaped parameter (–);
a is Vibe parameter a = 6.9 (–) for complete combustion.

The mass faction burned χ was given from integral of the Vibe function:

$$x = \int \frac{dx}{d\alpha}d\alpha = 1 - e^{-a\cdot y(m+1)} \tag{4}$$

To calculate heat transfer in the combustion chamber, the Woschni's model was applied.

$$Q_T = A\cdot q_{coeff}\cdot(T_c - T_w)$$

where

Q_T is heat lost to the wall (W/m^2);
A is the total surface area of the cylinder head, piston and cylinder (m^2);
q_{coeff} is heat transfer coefficient (W/m^2 K);
T_c is combustion gas temperature (K);
T_w is the wall temperature of the cylinder (K).

The air mass flow rates through intake and exhaust ports were calculated from the equation (5) and (6):

$$\frac{dm}{dt} = A_{eff} P_1 \sqrt{\frac{2}{R_0 T_1}} \cdot C \tag{5}$$

$$C = \sqrt{\frac{K}{K-1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{k}} - \left(\frac{P_2}{P_1}\right)^{\frac{K+1}{K}}\right]} \tag{6}$$

where

$\frac{dm}{dt}$ is air mass flow rate;
A_{eff} is effective flow area (–);
P_1 is upstream stagnation pressure (Pa);
T_1 is upstream stagnation temperature (K);
R_0 is gas constant (–);
P_2 is downstream static pressure (Pa);
K is the ratio of specific heats.

3. Results and discussions

3.1. Model validation

The confidence of the simulated model was evaluated based on the comparison between the experimental results and simulated results. The black curves describe the experimental results while the red curves describe simulated results. Before using the simulated model to estimate the effect of combustion duration, this model was calibrated and validated. The values of engine's specification and experimental data (bore, stroke, connecting length, intake and exhaust tube length and diameter, experimental conditions) were used as input data for the simulated model. Since the ignition timing was the input data in the simulated model, the simulated values were very close to the experimental values (Fig. 4).

The flowing coefficient (intake tube, throttle and intake ports, and exhaust ports) was used to adjust the air mass flow of the simulated model. A comparison between experimental and simulated results in air mass flow is shown in Fig. 5. The comparison shows that most values of air mass flow were the same. The maximum difference was 1.42% at 6000 rpm, this maximum scale is acceptable because the value of air mass flow an average value during the experiment. To reduce the error of engine brake torque and power between simulated results and experimental results, besides input correct air-fuel ratio, engine friction, the parameter m (Eq. (3)) was used to reduce that error. Fig. 6 shows a comparison between experimental and simulated results for brake torque. The values for the two cases were almost the same at each engine speed. The maximum brake torque was 21.8 Nm at 7000 rpm. The maximum difference in brake torque value was 0.90% at 10,000 rpm. Fig. 7 shows a comparison between the experimental and simulated results of engine power. At 9000 rpm, the engine achieved the maximum power of 18.11 KW. The engine power in the two cases was the same at each engine speed. The maximum difference was 0.59% at 10,000 rpm.

The comparison between cases allows a steady base to support the simulated model. The simulated model had a better accuracy method for prediction of engine performance. Moreover, the effect of combustion duration on engine performance and engine emission characteristics can be studied using this simulated model.

3.2. Mass fraction burned at different combustion duration

Mass fraction burned has a value from 0.0 to 1.0 (unit). This quantity describes the process of chemical energy released in the power stroke of a four-stroke SI engine.

Fig. 8 shows the mass fraction burn, which followed the CA at various combustion duration values. The ignition timing was selected at 20 degrees of BTDC. The mass fraction of burnt gas was 1 at the larger crank angle with increased combustion duration. At 40 degrees combustion duration, the mass fraction burned was 1.0 at 15 degrees of ATDC. With 100 degrees combustion duration, the mass fraction burned was 1.0 at 70 degrees of ATDC. Thus, when combustion duration was increased, the time needed for conversion of chemical energy of fuel into thermal energy was extended. If the combustion duration is too short, the energy conversion process may be completed at BTDC. Conversely, if the combustion duration is too long, the energy converted may extend to the exhaust stroke. However, neither of these cases is expected because the engine does not show good performance.

3.3. Effect of combustion duration on engine performance and engine emission characteristic

This section described the effect of combustion duration on engine performance and engine emission characteristic. The results show the relationship between combustion duration and some other parameters as like as residual gas, effective release energy, BSFC, IMEP, BSFC, peak firing temperature, brake torque, engine power and NO_x, CO and HC emission.

Fig. 9 shows gross release energy versus combustion duration at different engine speeds. The gross release energy depends mostly on the air-fuel mass in the cylinder and the combustion conditions.

Fig. 10 shows the effect of combustion duration on the residual gas ratio. It can be seen that when combustion duration increase from 40 degrees to 110 degrees at engine speed was 7000 rpm and 8000 rpm. The residual gas ratio shows a downward trend from 0.25% to 0.13%. When the engine speed was 6000 rpm, the residual gas ratio decreases from 0.36% to 0.22% and then increase to 0.3%. The minimum residual gas ratio was 0.22% at 80 degrees combustion duration. It can be explain by that at a low and medium engine speed, an excessively long combustion duration can extend the burning process until the pistons are at BDC or even in the exhaust stroke. This process will affect the next intake stroke to reduce fresh air-fuel into the cylinder, thereby sweeping the exhausted gas outside.

Fig. 11 shows the effect of combustion duration on peak firing temperature. The peak firing temperature decreases as the combustion

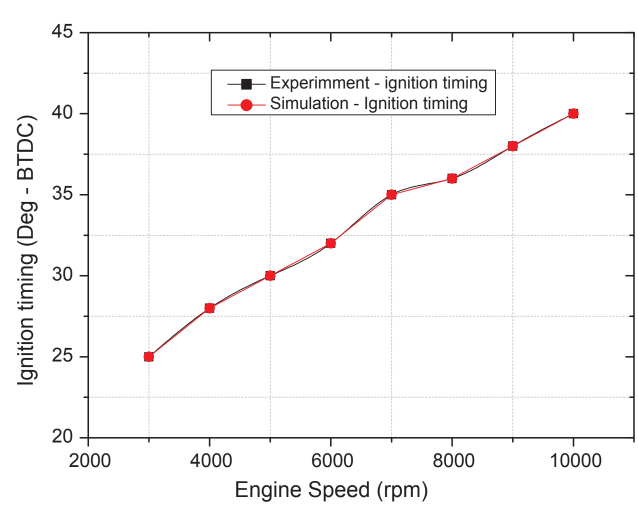

Fig. 4. Ignition timing versus engine speed.

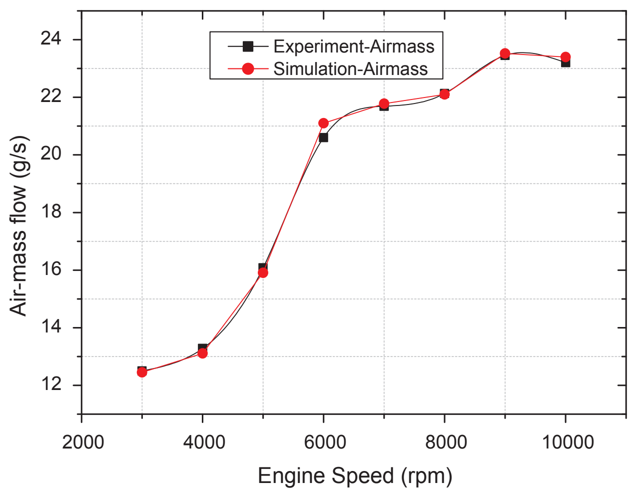

Fig. 5. Air mass flow versus engine speed.

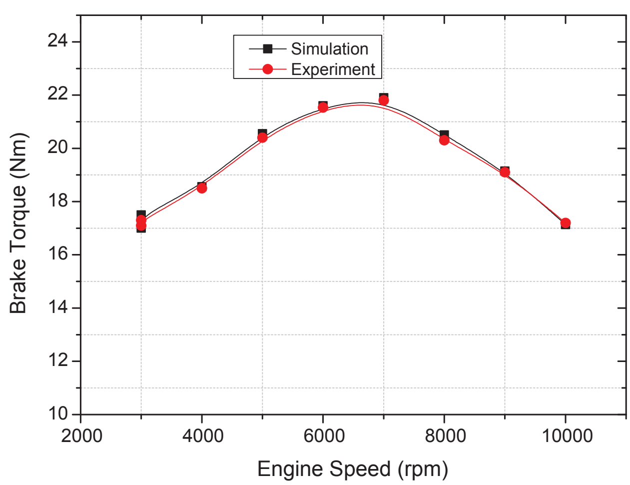

Fig. 6. Engine brake torque versus engine speed.

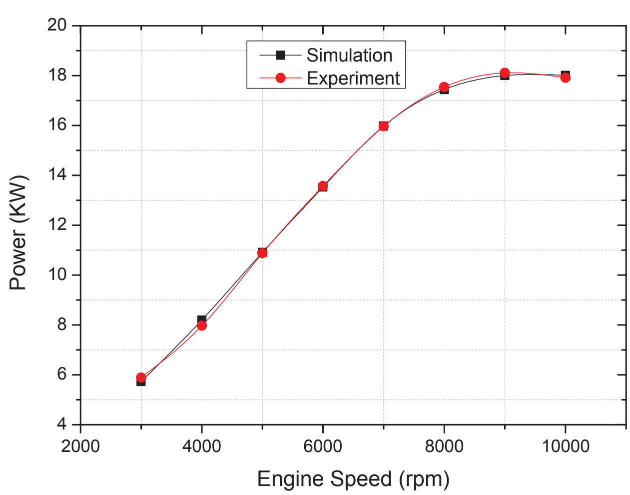

Fig. 7. Engine power versus engine speed.

duration increases. This is because the increase in combustion duration is due to the increased percentage of heat loss. On the other hand, if the combustion duration is too long then it will take more time for the heat transfer within the piston, cylinder and flowing out through the exhaust gas. This accounts for the decrease in the mean effective release energy has decreased (Fig. 12).

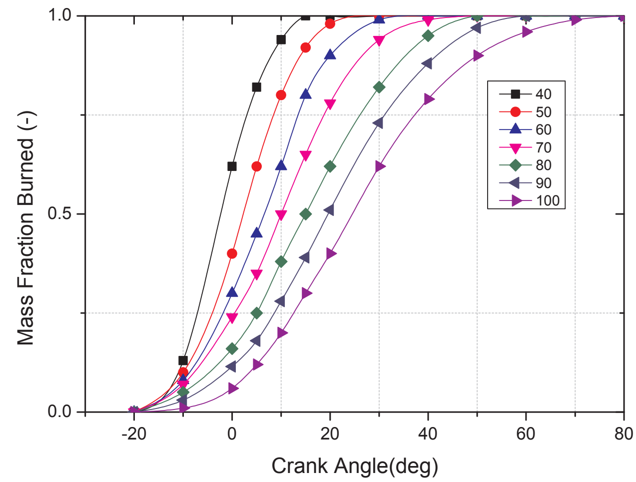

Fig. 8. Mass fraction burned in different combustion durations.

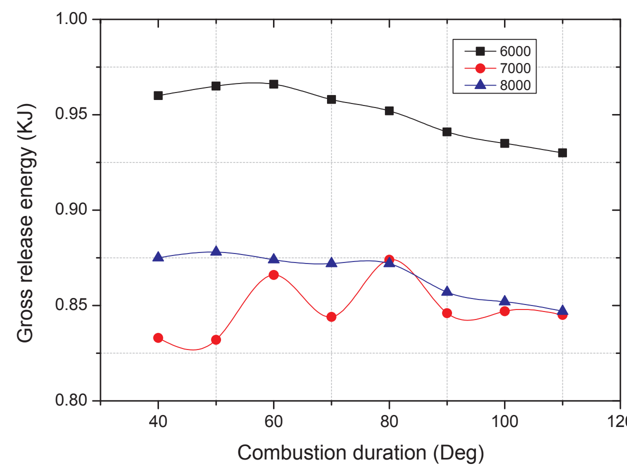

Fig. 9. Gross release energy versus combustion duration.

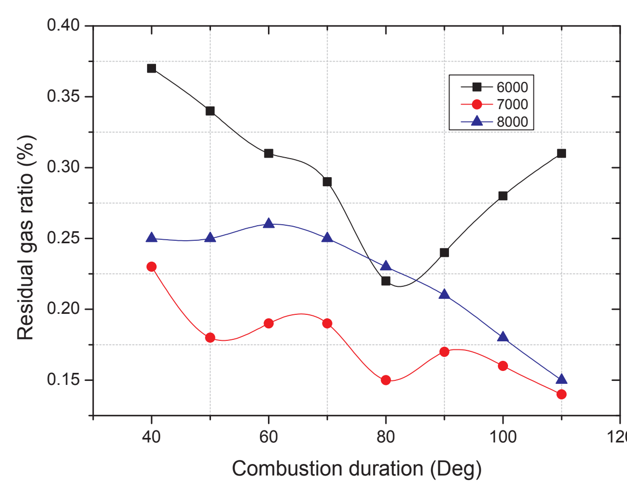

Fig. 10. Residual gas ratio versus combustion duration.

As shown in Fig. 12, as the combustion duration increases, the effective release energy increases until a maximum value was achieved after that, a subsequent decrease was observed. The maximum effective release energy was different at different engine speeds. This could be explained by following two aspect factors. Firstly, if the combustion duration is extremely short then the chemical energy of fuel could not be completely converted to thermal energy. Moreover, if the

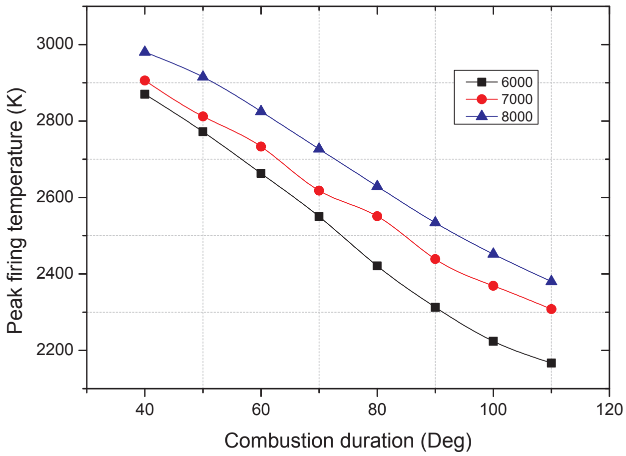

Fig. 11. Peak firing temperature versus combustion duration.

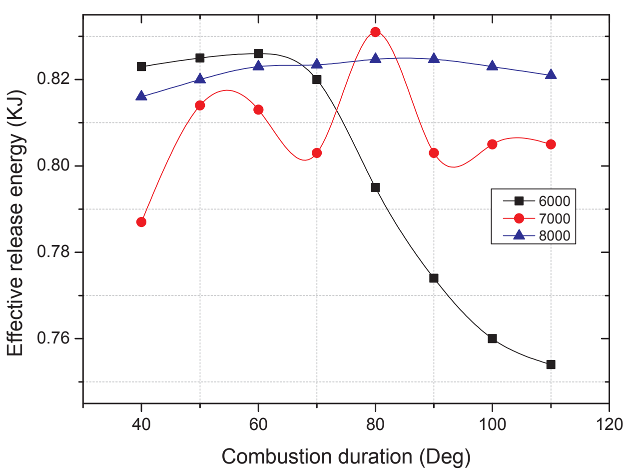

Fig. 12. Effective release energy versus combustion duration.

combustion duration is long, then more thermal energy loss will occur because of increased heat loss. That is why the effective release energy initially increased until a maximum value was achieved. After that, it was decease following the increase in combustion duration. Secondly, in a real SI engine, because the combustion reaction always takes some time, the heat release is conducted instantaneously at TDC [24]. To increase thermal efficiency, a short duration is required for the total heat release energy to occur. Because it increases the frequency of heat release energy to make the real cycle resembles an Otto cycle better. The engine speed band has a strong effect on the time of total heat release energy which accounts for the different maximum release energy at gotten different engine speeds. In this research, at 6000 rpm, the maximum effective release energy was 0.826 KJ at 60 degrees combustion duration. At 80 degrees combustion duration, the maximum effective release energy at 7000 rpm and 8000 rpm was 0.831 KJ and 0.825 KJ, respectively.

Figs. 13 and 14 show the BMEP and IMEP versus combustion duration. The BMEP and IMEP increases until a maximum value was achieved before further decrease began to occur. At each engine speed, there was an optimal combustion duration value, at that value, the engine showed a higher BMEP and IMEP. Besides that, the engine achieves a maximum BMEP and IMEP at the same combustion duration. This is because the effect of combustion duration on BMEP and IMEP was same with effect on effective release energy. As shown in Fig. 12, Figs. 13 and 14 similar trend of effective release energy, BMEP and IMEP was presented. The Eq. (7) shows the relationship between BMEP and IMEP. This equation explains why the maximum BMEP and maximum IMEP was dropped at the same combustion duration value.

$$BMEP = IMEP - FMEP - SMEP \tag{7}$$

where

FMEP is friction mean effective pressure, Bar;
SMEP is scavenging mean effective pressure, Bar.

In this research, at 6000 rpm and 60 degrees combustion duration, the maximum BMEP was 10.55 bar and maximum IMEP was 12.6 bar. At 7000 rpm and 80 degrees combustion duration, the maximum BMEP was 10 bar, and the maximum IMEP was 12.20 bar. At 8000 rpm, the maximum BMEP was 9.24 bar, and maximum IMEP was 11.85 bar.

Fig. 15 shows the combustion duration effect on BSFC. Initially, the BSFC was decreasing until a minimum value was achieved after that, an increase began to occur. At each engine speed, a combustion duration value for optimal fuel consumption was identified. At 6000 rpm and 60 degrees combustion duration, the minimum BSCF was 319.8 g/KWh. At 7000 rpm and 80 degrees combustion duration, the minimum BSFC was 373.5 g/KWh. At 8000 rpm and 80 degrees combustion duration, the minimum BSFC was 385.7 g/KWh. The effect of combustion duration on BSFC, engine effective torque and engine effective power all suggest that the BMEP has a strong effect on BSFC, engine effective torque, and engine effective power.

Eq. (8) shows the relationship between BMEP and engine effective torque.

$$T_{eff} = \frac{BMEP \cdot V_D}{k_{cycle} . \pi} \tag{8}$$

where

T_{eff} is engine effective torque (Nm);
$V_D$is displacement (m^3);
k_{cycle} is a simulation cycle parameter, cycle.

The relationship between BMEP and engine effective power

$$P_{eff} = BMEP \cdot V_D \cdot n_{cycle} \tag{9}$$

where

P_{eff} is engine effective power (kW);
n_{cycle} is a number of cycle per second, cycle/sec.

The relationship between BSFC and engine power

$$BSCF = \frac{\varepsilon}{P_{eff}} \tag{10}$$

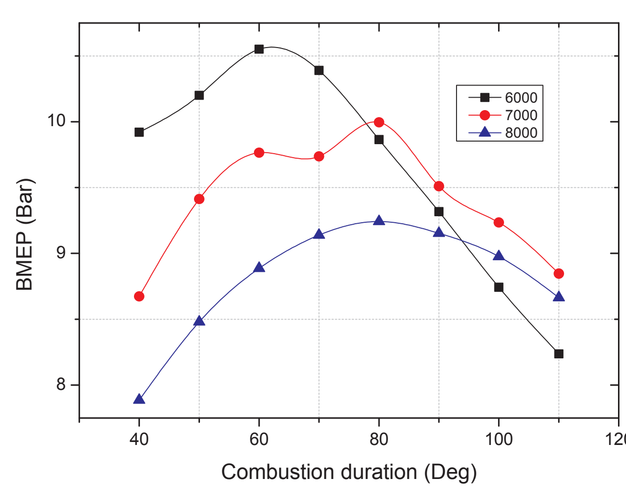

Fig. 13. BMEP versus combustion duration.

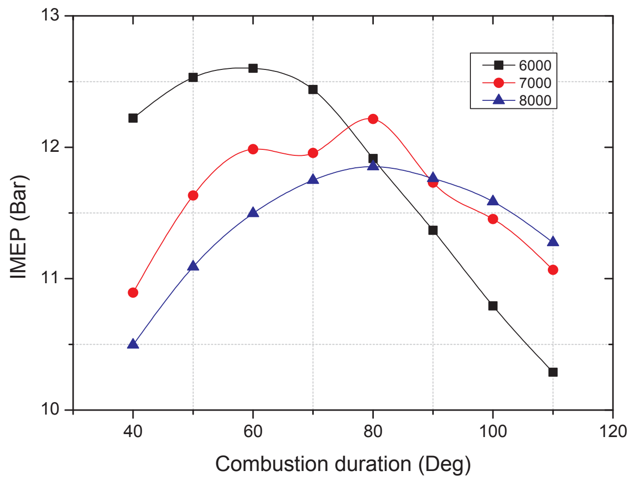

Fig. 14. IMEP versus combustion duration.

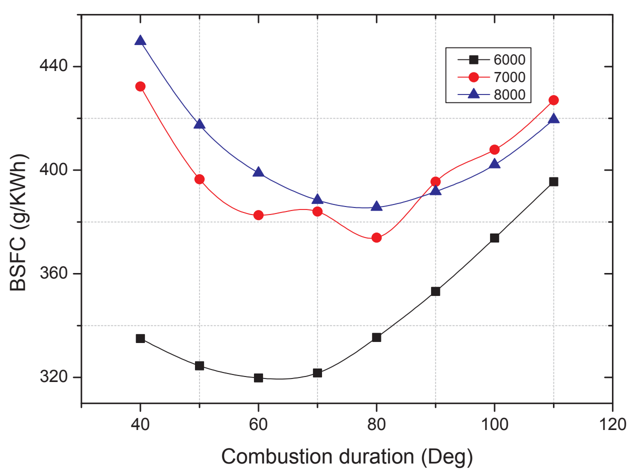

Fig. 15. BSFC versus combustion duration.

where

ε is the fuel consumption rate in grams per second (g/s).

From Eqs. (8)–(10) it can be seen that the increase of BMEP helps improve engine effective torque, engine effective power, and BSFC.

Figs. 16 and 17 show the combustion duration effect on engine power and engine brake torque. Following the increased combustion duration, the BMEP, engine brake torque and engine power had the same fluctuating trend. The engine brake torque and engine power increases until a maximum value was achieved after that, a decrease was observed as the combustion duration keeps increasing. At each engine speed, the optimal combustion duration value at that the engine shows an optimal performance. The maximum engine brake torque and engine power dropped at the same combustion duration value. At 6000 rpm and 60 degrees combustion duration, the maximum brake torque was 22.7 Nm and maximum power 14.48 KW. At 7000 rpm and 80 degrees combustion duration, the maximum brake torque was 21.55 Nm, and maximum power was 16.01 KW. At 8000 rpm and 80 degrees combustion duration, the maximum brake torque was 20.25 Nm, and maximum power was 16.92 kW.

In case of the engine speed at 7000 rpm and 70 degrees combustion duration, the values fluctuated because of the combustion duration on the mixing air-fuel homogeneity at 70 degrees was not better than that at 60 degrees and 80 degrees. The homogeneous air-fuel mixing in the combustion chamber was reflected through the residual gas (Fig. 11) and CO emission (Fig. 20). With a combustion duration of 70 degrees, the residual gas was bigger than that at 60 degrees and 80 degrees.

Fig. 18 shows the optimal combustion duration at each engine speed. The optimal combustion duration has increased trend with increasing engine speed. That suggest to us that when engine run at higher speed, parameter such as air-fuel ratio, spark advance, and compression ratio [15] should be adjusted to extend the combustion duration. When the engine is operated at the optimal combustion durations, the engine torque is at maximum value and BSFC is at a minimum value. The optimal combustion duration at 5000 rpm and 4000 rpm was the same value 55 degrees. Because it was difficult to determine the exact combustion duration value with error is 1 degree to 3 degrees at high engine speed. So, the optimal combustion duration should be selected from the average value. This explains the reason why the optimal combustion duration was the same at 4000 rpm and 5000 rpm.

Fig. 19 shows the effect of combustion duration on the NO_x emission. The NO_x emission decreases with increasing combustion duration because of the decreased peak firing temperature (Fig. 11). The peak firing temperature has the greatest effect on NO_x emission. The decrease in peak firing temperature is said to reduce NO_x emission and vice versa.

Fig. 20 shows that the trend of CO emission was not maintained, this trend was like that of the residual gas ratio. At 6000 rpm, the CO emission decreased while the combustion duration increased from 40 degrees to 80 degrees. At 7000 rpm and 8000 rpm, the CO emission trend decreases while engine combustion duration increases from 40 degrees to 80 degrees. This suggests that the residual gas had a huge effect on the CO emission characteristics because when the residual gas increases, the amount of fresh air in the next intake stroke decreases. Thus, the CO emission increases with an increase in the residual gas.

Fig. 21 shows that HC emission decreases as the combustion duration increases because of the increase in the time needed for complete combustion of the fuel to occur.

The above result reveals the importance of combustion duration in a small SI-engine. In addition, how this parameter influences the residual gas ratio, effective release energy, BSFC, engine torque, and engine emission characteristics were discussed.

4. Conclusions

This paper has presented a solution for solving the lack of control and the inability to precisely determine the combustion duration. For the very first time, the effect of combustion duration was totally presented in a motorcycle engine. With an engine testing speed band ranging from 3000 rpm to 10,000 rpm, the combustion duration increased from 40 degrees to 110 degrees crank angle. Through combined

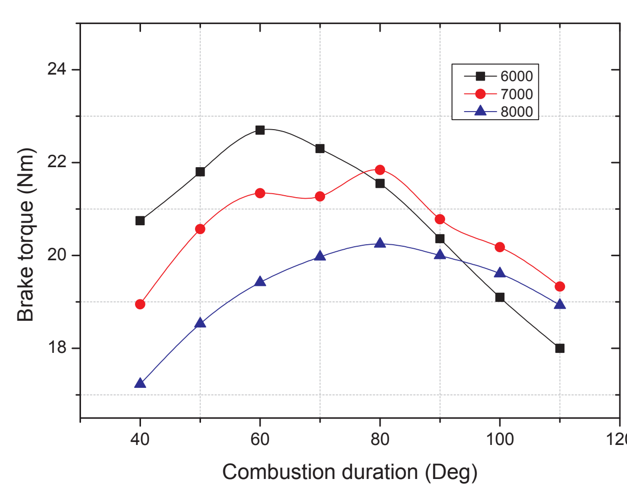

Fig. 16. Brake torque versus combustion duration.

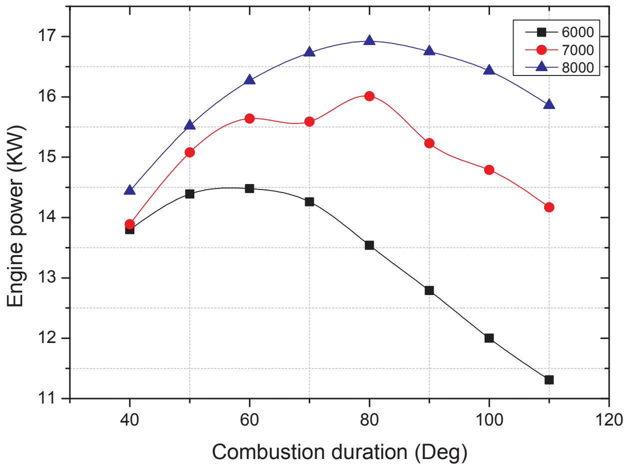

Fig. 17. Engine power versus combustion duration.

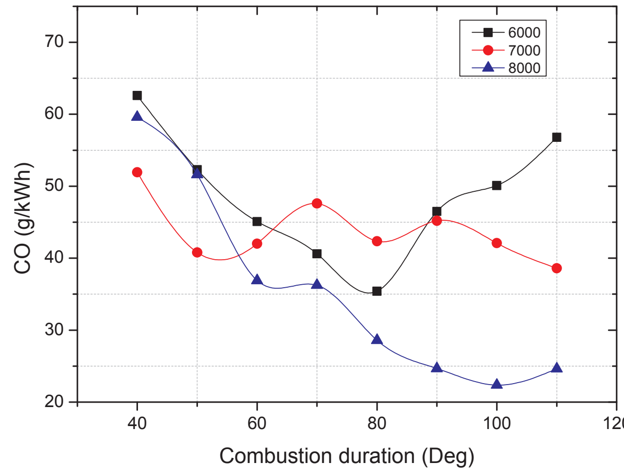

Fig. 20. CO emission versus combustion duration.

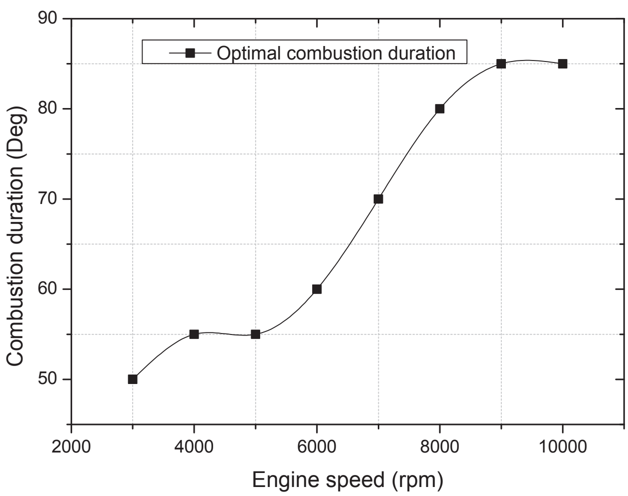

Fig. 18. Optimal combustion duration at each engine speed.

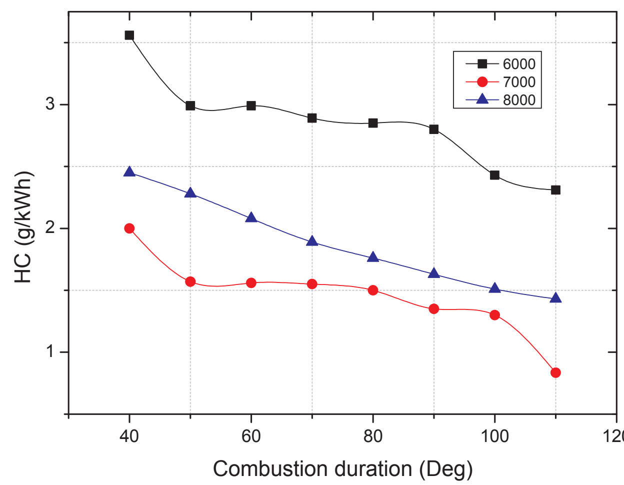

Fig. 21. HC emission versus combustion duration.

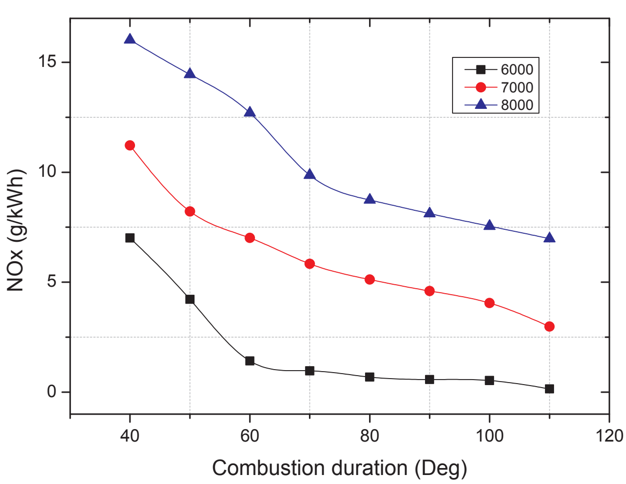

Fig. 19. NO_X emission versus combustion duration.

experimental and simulated methods, this is an accurate method for predicting the effect of combustion duration on engine performance and engine emission characteristic. The optimal combustion duration at each engine speed was also determined. At that optimal combustion duration value, the engine displays superior performance. By varying the combustion duration, we were able to control both the NO_x and CO emission in the limited band.

The results of our research were summarized below:

(1) The combustion duration has a significant effect on residual gas and the effective release energy. In various engine speeds, the effect of combustion duration on residual gas, and effective release energy varied. Following increased of combustion duration, at an engine speed was 7000 rpm and 8000 rpm the residual gas and effective release energy decreased. The minimum residual gas ratio was 0.14% and 0.15%, respectively. When engine speed was 6000 rpm, the residual gas decreases initially and increases thereafter. Likewise, at 80 degrees combustion duration, the minimum residual gas ratio was 0.22% while the maximum effective release energy was 0.826 KJ at 60 degrees combustion duration.
(2) The optimal value of effective release energy, BMEP, IMEP, and BSFC was achieved at the same combustion duration value of each engine speed. At an engine speed of 6000 rpm and 8000 rpm, the optimal value of effective release energy, BMEP, IMEP, and BSFC drops at 60 degrees and 80 degrees combustion duration.
(3) At each engine speed, an optimal combustion duration value was found. According to the result of the research, the engine gives the best performance at 6000 rpm and 60 degrees combustion duration. In this case, the engine was able to achieve the maximum brake torque at 22.7 Nm and minimum BSCF at 319.8 g/KWh.
(4) The combustion duration affected heat loss in the cylinder because the increased combustion duration gave rise to an increased heat

transfer within the cylinder and piston. As the combustion duration increases from 40 degrees to 110.0 degrees, the peak firing temperature decreases from 2900 K to 220 K.

(5) The residual gas affected the released energy and CO emission. The trend of residual gas was opposite to trend of effective release energy. As the residual ratio increases, the effective release energy decreases while the CO emission increase, and vice versa.

(6) The NO_x and HC emission decrease when the combustion duration increases from 40 degrees to 110.0 degrees. At engine speed of 6000 rpm, the NO_x and HC decreases from 7 g/kWh to 0.2 g/kWh and 3.56 g/kWh to 3.1 g/Kwh respectively. When engine speed was 7000 rpm, the NO_x and HC decrease from 11.22 g/kWh to 2.98 g/kWh and 2 g/kWh to 0.835 g/Kwh respectively. When the engine speed was 8000 rpm, the NO_x and HC decrease from 16.2 g/kWh to 6.9 g/kWh and 2.45 g/kWh to 1.43 g/Kwh in respectively.

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