Cycle Engine Modelling Of Spark Ignition Engine Processes during Wide-Open Throttle (WOT) Engine Operation Running By Gasoline Fuel

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Cycle Engine Modelling Of Spark Ignition Engine Processes during Wide-Open Throttle (WOT) Engine Operation Running By Gasoline Fuel

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Abstract: One-dimensional engine model is developed to simulate spark ignition engine processes in a 4-stroke, 4 cylinders gasoline engine. Physically, the baseline engine is inline cylinder engine with 3-valves per cylinder. Currently, the engine's mixture is formed by external mixture formation using piston-type carburettor. The model of the engine is based on one-dimensional equation of the gas exchange process, isentropic compression and expansion, progressive engine combustion process, and accounting for the heat transfer and frictional losses as well as the effect of valves overlapping. The model is tested for 2000, 3000 and 4000 rpm of engine speed and validated using experimental engine data. Results showed that the engine is able to simulate engine's combustion process and produce reasonable prediction. However, by comparing with experimental data, major discrepancy is noticeable especially on the 2000 and 4000 rpm prediction. At low and high engine speed, simulated cylinder pressures tend to under predict the measured data. Whereas the cylinder temperatures always tend to over predict the measured data at all engine speed. The most accurate prediction is obtained at medium engine speed of 3000 rpm. Appropriate wall heat transfer setup is vital for more precise calculation of cylinder pressure and temperature. More heat loss to the wall can lower cylinder temperature. On the hand, more heat converted to the useful work mean an increase in cylinder pressure. Thus, instead of wall heat transfer setup, the Wiebe combustion parameters are needed to be carefully evaluated for better results.

1. Introduction

Internal combustion engine has become major powertrain to drive modern transportation. The engine design and development process has emerged progressively with the finding of newer experimental method and instrumentation to gain insight information of the in-cylinder processes. Most of the experimental approach required huge expenses with considerably high lead time. The advent of digital computers allowed engine's performance prediction prior to the hardware development through engine modelling. Three dimensional engine modelling allowed detail spatial and temporal characterization of flow and combustion with high computing time. A cheaper means is to use zero or one-dimensional engine models which solve basic thermodynamic and gas dynamic equation. This type of engine model is also called cycle engine model which permit prediction of numerous engine performance parameter such as engine torque, brake power, volumetric efficiency and thermal efficiency. But the governing equation of such model is the combustion or energy equation where its solution will determine the rate of heat release, subsequently the rate of temperature and pressure rise. These parameters then will determine the engine indicated work, power and efficiency. One dimensional engine model has been used in current study to simulate the 4 stroke cycle engine operation. The model is developed using *GT-SUITE*^{TM} software. The model fundamental equations comprise of one-

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dimensional equation of the gas exchange process, isentropic compression and expansion model and progressive engine combustion model based on Wiebe approach. This engine model also accounts for the heat transfer and frictional losses as well as the effect of valves overlapping.

2.0 Objective of the Study

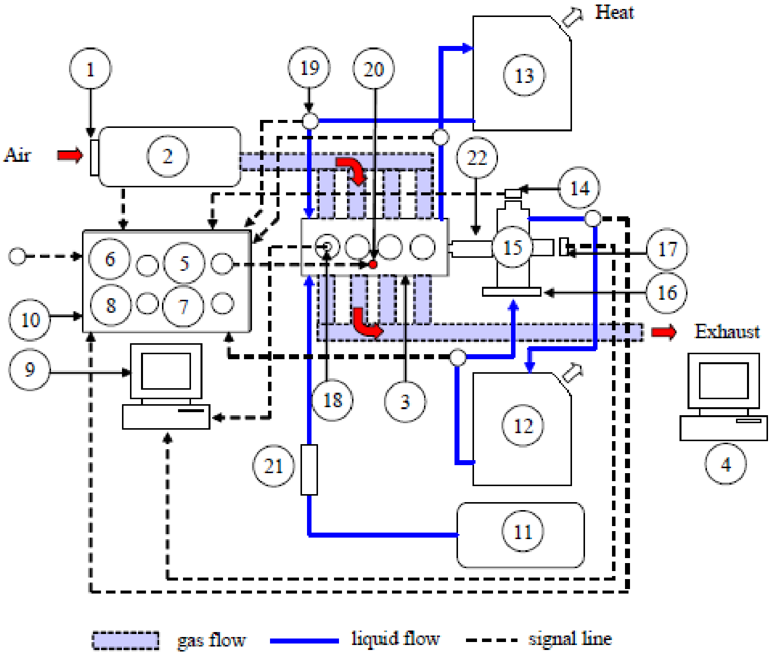
The model is developed as a means to acquire engine performance prediction solely for spark ignition engine. The parameters of interest are engine cylinder pressure and temperature. The engine performance predictions need to be validated with experimental engine data in order to pledge trustworthiness and reliability. The engine which is currently running on gasoline is expected to be used for alternative fuel study namely compressed natural gas and hydrogen. The introduction of gaseous fuels such as hydrogen into existing engine will require the addition of gaseous fuel injector which is expected to be completed in the latter stage of the study.

3. Experimental Engine Setup

The baseline engine is a 4 cylinder, 12 valves engine which is currently running on gasoline fuel. The engine is naturally aspirated and the fresh mixture is carburetted in design. Each cylinder equipped with two intake valves of different valve diameter and one exhaust valve which has larger valve diameter. The ignition is control by mechanical distributor. For the purpose of static dynamometer experimentation, the original cooling loop has been made external and increased in terms of water capacity to improve the cooling capability during wide open throttle condition. Other modified system is the intake and exhaust system. The intake air filter is removed while the other parts are retained and attached to the surge tank with orifice airflow measurement. The exhaust pipe is made longer to connect with an exhaust gas calorimeter. The baseline engine has been tested for experimental data on a stationary test-bed coupled with hydraulic dynamometer. All required engine geometrical and operational data have been measured and embedded into this model. The following table provides detail engine specifications.

Table 1: Baseline engine specifications

Parameter	Size and Feature
Valve train type	In-line OHV, SOHC
Number of cylinders	4 cylinders
valves per cylinder	3 valves per cylinder
Combustion chamber type	Pent-Roof type
Total displacement (cm^3)	1,488
Cylinder bore (mm)	75.5
Piston stroke (mm)	82.0
Compression ratio	9.2
Intake valves open/close	15^o BTDC/63^o ABDC
Exhaust valve open/close	57^o BBDC/ 13^o ATDC
Lubrication system	Pressure load, full – flow filtration
Oil pump type	Trochoid type
Cooling system	Water – cooled forced circulation
Water pump type	Centrifugal impeller type
Maximum output	66 kW@6000 rpm
Maximum Torque	124 Nm@3000 rpm



1.	Orifice plate meter	12.	Dynamometer water supply
2.	Surge tank	13.	Engine cooling system
3.	Tested engine	14.	Load sensor
4.	Emission analyzer	15.	Hydraulic dynamometer
5.	Throttle controller	16.	Dynamometer controller
6.	Pressure manometer	17.	Crank angle encoder
7.	Load indicator	18.	Pressure Transducer
8.	Temperature scanner	19.	Temperature sensor
9.	Combustion analyzer	20.	Throttle body
10.	Instrumentation panel	21.	Fuel flow meter
11.	Fuel supply tank	22.	Engine-dynamometer coupling

Figure 1: Schematic of the hydraulic dynamometer engine test rig

4. Engine Modelling

The engine model is developed using $GT\text{-}SUITE^{TM}$ package which has special capability to predict engine cycle performance. Based on the physical appearance of an actual engine, the engine model can be also divided as follow (1) the intake manifold system (2) the engine combustion chamber and cylinder (3) the engine crankcase, and (4) the exhaust manifold system. The quasi one-dimensional approach is mostly adopted for the solution of the gas dynamic flow within the intake and exhaust manifold whereas the engine cylinder represent the zero dimensional part of the model. Basically, the engine model is governed by the equation of mass continuity, momentum and energy equation. The combustion or rate of heat release model is commonly the mathematical equation that determines the rate of temperature and pressure rise which will further affect the engine performance parameter. Wiebe model of spark ignition engine combustion is among the simplest. The mass fraction burn rate (MFB) using Wiebe function can be expressed as Eq. (1):

$$MFB(\theta) = 1 - \exp\left[-a\left(\frac{\theta-\theta_0}{\Delta\theta}\right)^{m+1}\right] \tag{1}$$

where θ is crank angle degrees with θ_0 corresponding to the initialization of heat release and $\Delta\theta$ corresponding to the duration of burn. The equation is also defined by two constants a and m which have typical values of 5 and 3 respectively. The $\boldsymbol{a}$ and $\boldsymbol{m}$ are also adjustable parameters that can be fine-tune to fit the model with experimental data. Hence, it can be said that the Wiebe function is useful in a state where the cylinder data properties are available and it is not predictive in nature.

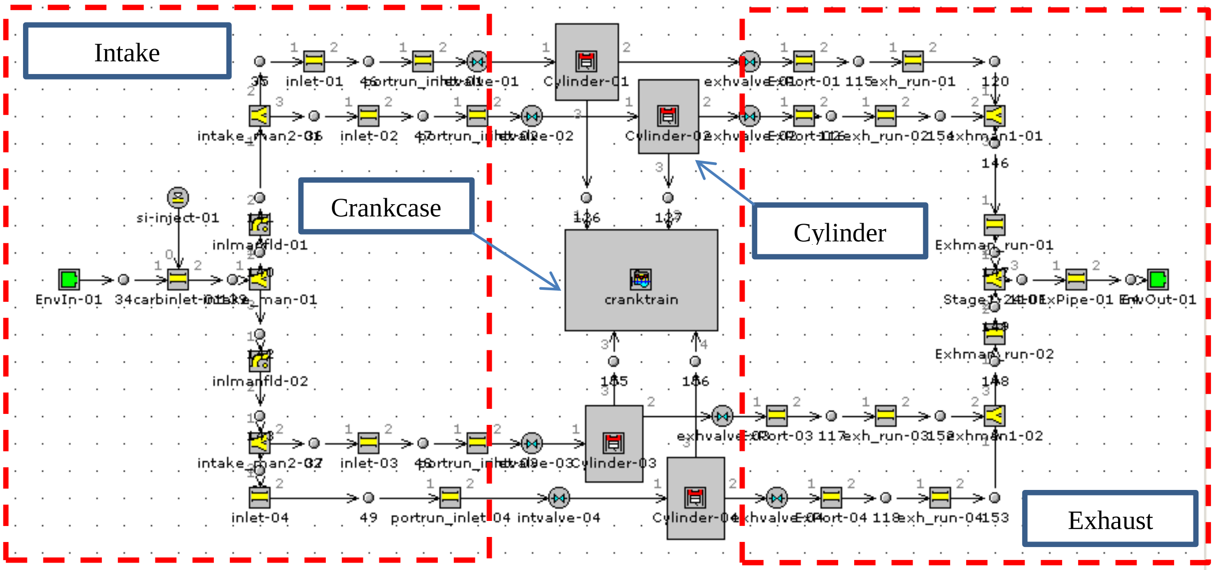


Figure 2: The layout of the engine model

5. Results and Discussion

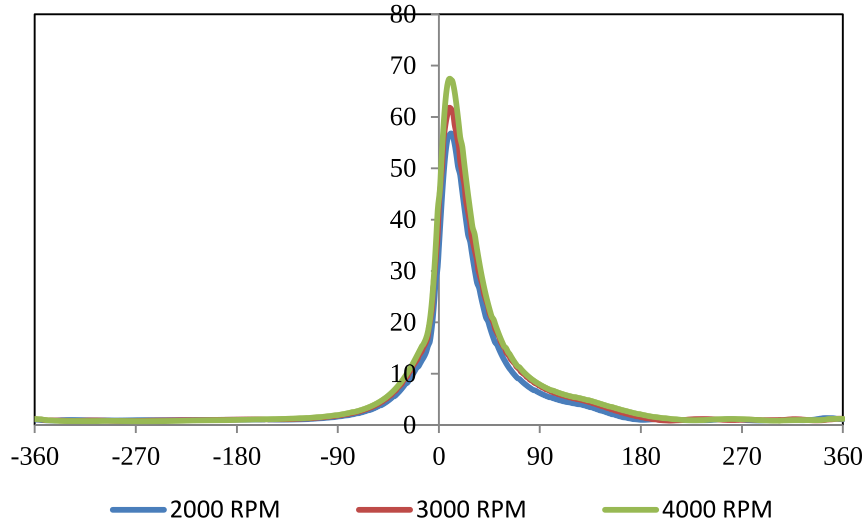
5.1 Effect of Different Engine Speed

Figure 3 (a) and (b) present the simulated pressure-crank angle and temperature crank-angle diagram for engine rotational speed of 2000, 3000 and 4000 rpm. The engine model simulates the wide open throttle operation of the engine. Important inputs for the model are presented by Table 2. The air to fuel ratio, the burning duration (10% to 90% burning), the 50% burning crank angle values are all based on the experimental data of the engine tested at wide open throttle condition. The value of ***a*** and ***m*** (the adjustable parameters) are retained as default since the curve-fitting process required heuristic approach that will increased the time allocation for this stage of the study.

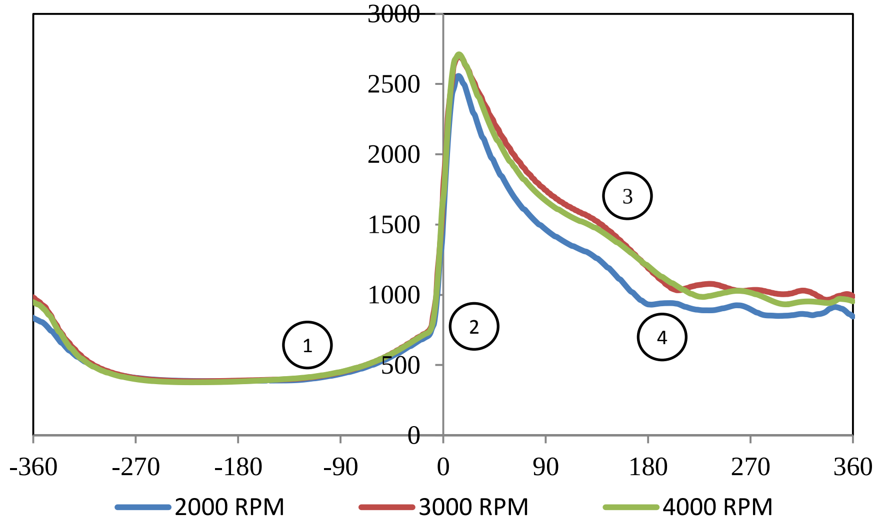
Table 2: Input for the simulated cases

Engine speed (rpm)	Air fuel ratio	Burning duration, $\Delta\theta$, *(CA unit)*	50% burning crank angle (*CA*)	a	m
2000	12.07	13.99	-1.19		
3000	13.99	14.39	-0.40		
4000	12.06	15.99	2.40		

Based on the pressure-crank angle diagram, the simulated results present an increment trend of cylinder peak pressure. The maximum peak for 2000, 3000 and 4000 rpm are 56.8 bar, 61.7 bar and 67.4 bar. All peak pressure timing is obtained at about 10^{o} CA after top dead centre which corresponds to the optimum peak pressure timing of a natural aspirated spark ignition engine. These are also reflected the measured engine data that produce the same results. Based on the data specified by Table 2, the decisive factor that may differentiate the maximum peak pressure of each case is the air to fuel ratio, the burning duration, $\Delta\theta$ and the heat release curve anchor angle, the 50% burning crank angle. The richest air to fuel ratio input for the model are 12.07 and 12.06 for 2000 rpm and 4000 rpm respectively whereas the air fuel ratio input at medium engine speed is about 13.99, approaching the stoichiometric value for gasoline fuels. However, the air fuel ratio is not fully reflecting the trend of peak pressure. Examination on the burning duration provides a more justified reason for the increment of peak pressure. As the engine speed increase, the burning duration is also increase which initially showed an inverse proportional. However, if the unit of crank angle is converted to the time scale, the corresponding value of burning duration for 2000, 3000 and 4000 rpm are 1.16 ms, 0.8 ms and 0.67 ms respectively. Theoretically, the faster burning duration will enhance higher cylinder pressure and these results support the trend of peak cylinder pressure.



(a)



(b)

Figure 3(a): pressure-crank angle diagram for cylinder no. 1**(b):** temperature-crank angle diagram for cylinder no.1

The temperature-crank angle diagram presents a normal trend of temperature variation in engine cylinder. By overall, the trend is in agreement with the cylinder pressure data. As the engine speed increase, the cylinder pressure increase as well as the cylinder temperature. However, it is noticeable that the cylinder temperature at 4000 rpm is just slightly higher than the data at 3000 rpm. Point 1 showing the start of temperature increased corresponding to the inlet valve closing at -117^{o} before top dead centre. Point 2 corresponding to the ignition point and the start of major heat release event. Point

3 corresponding to the start of exhaust blow down process during the start of exhaust valve opening at 123^{o} CA after top dead centre. While point 4 corresponding to the start of exhaust stroke where the piston moved upwards expelling the burned gas within the engine cylinder.

5.2 Validation Based on Cylinder Pressure

The model is first validated by using the cylinder pressure data. Figure 4 (a) - (c) present the comparison of measured cylinder pressure (measured at cylinder no.1) with the simulated cylinder pressure. In general, the simulated cylinder pressures replicate the trend of measured data. The peak pressure timing is coincident in most cases. The simulated rate of pressure rise and it's reduce follow the trend of experimental data. The discrepancy is noticeable in terms of the cylinder peak pressure. Table 3 presents the deviation comparison between the simulated and measured cylinder peak pressure.

Table 3: Deviation of simulated cylinder peak pressure against measured data

Engine speed (rpm)	**Simulated peak pressure (bar)**	**Measured peak pressure (bar)**	**Deviation (%)**
2000	56.8	61.19	-7.17
3000	61.7	62.27	-0.92
4000	67.4	67.05	0.52

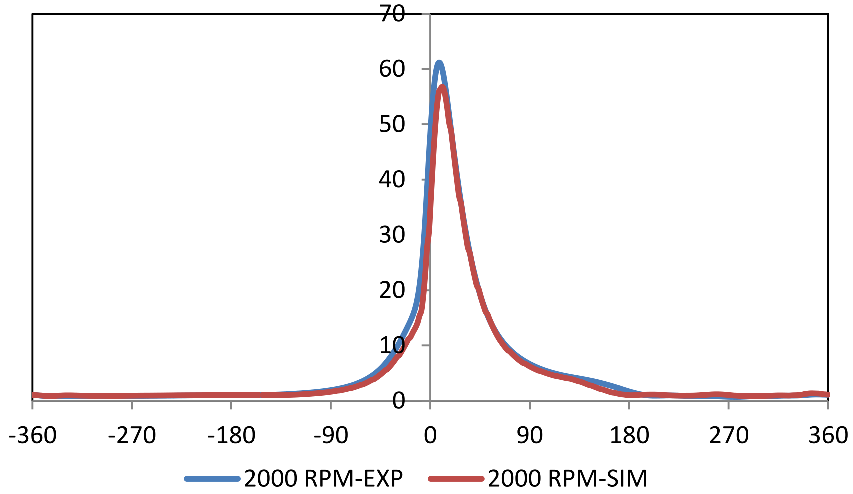
The largest deviation occurred at low engine speed where the model under predicted the cylinder peak pressure. In other two cases, the deviation is less than 1%. Hence, it can be concluded that the current model accuracy is adequate for the prediction of cylinder peak pressure. This means that all other related performance parameter based on cylinder pressure such as the indicated work indicated power are accurate enough for further evaluation.

5.2 Validation Based on Cylinder Temperature

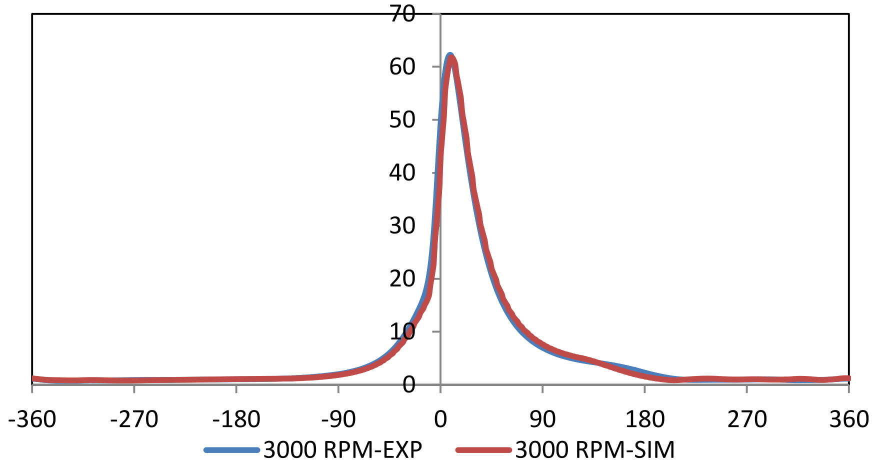
The second validation is based on the cylinder temperature data. Figure 5 (a) - (c) present the comparison of measured cylinder temperature (measured at cylinder no. 1) with the simulated cylinder temperature. For the purpose of comparison, the measured cylinder temperature is not directly measured from the engine cylinder, however the values is derived based on the cylinder pressure using the ideal gas relationship within a closed volume at the instance of intake valve closing. Hence only the temperatures values during all valves closed are considered. The expression of the ideal gas is given by Eq. 2:

$$T_{cyl} = \frac{P_{cyl} \times V_{cyl} \times T_{amb}}{P_{amb} \times V_{amb}} \quad \text{(Eq.2)}$$

In general, the simulated cylinder temperatures also replicate the trend of measured data. From the instance of intake valve closing, followed by the compression and ignition process, the simulated temperature profile fitted the measured data. The discrepancy is then obvious in terms of cylinder peak temperature and its subsequent instantaneous values. Table 4 presents the deviation comparison between the simulated and measured cylinder peak temperature. Based on the comparison, the deviation of cylinder peak cylinder temperatures are about 10%. Due to the difference in cylinder peak temperature, the subsequent instantaneous temperature values also over predicted the measured data. However, the rate of temperature reduction are seems to be comparable with the measured data except for the case of 2000 rpm where the measured data showing a moderating rate of temperature reduction. This clearly cannot be captured by the model.



(a)



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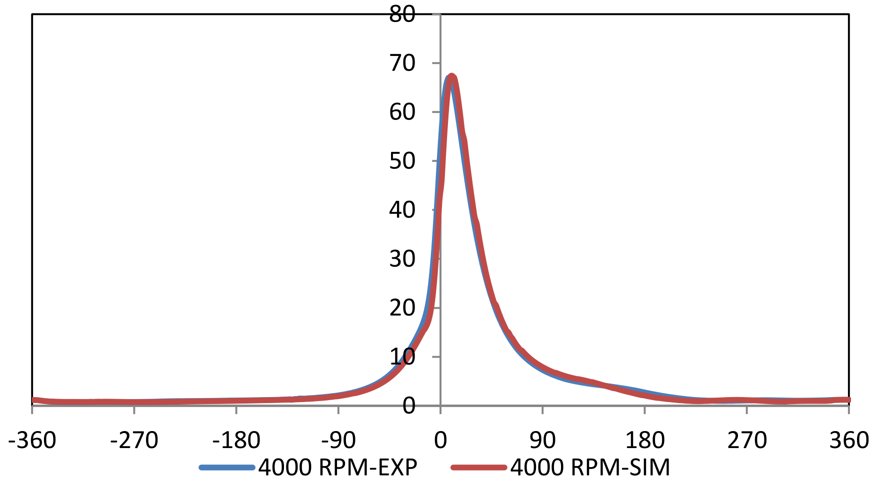

(c)

Figure 4(a)-(c): Comparison between measured and simulated cylinder pressure for engine speed of (a) 2000 rpm (b) 3000 rpm and (c) 4000 rpm.

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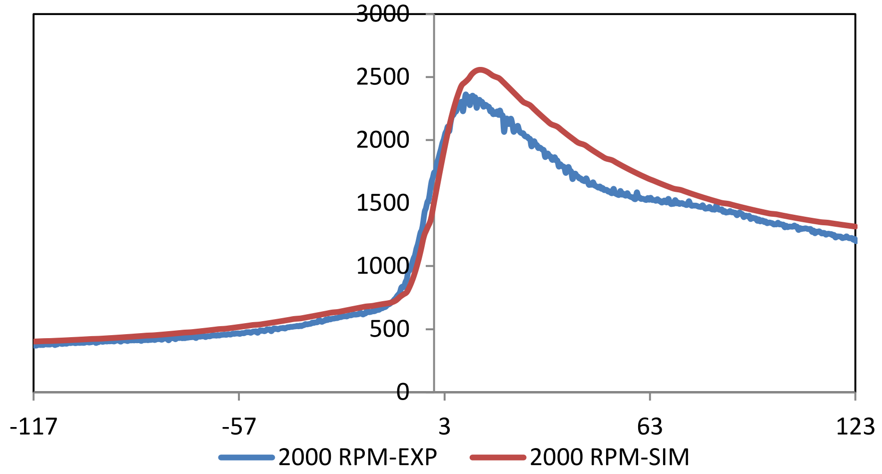
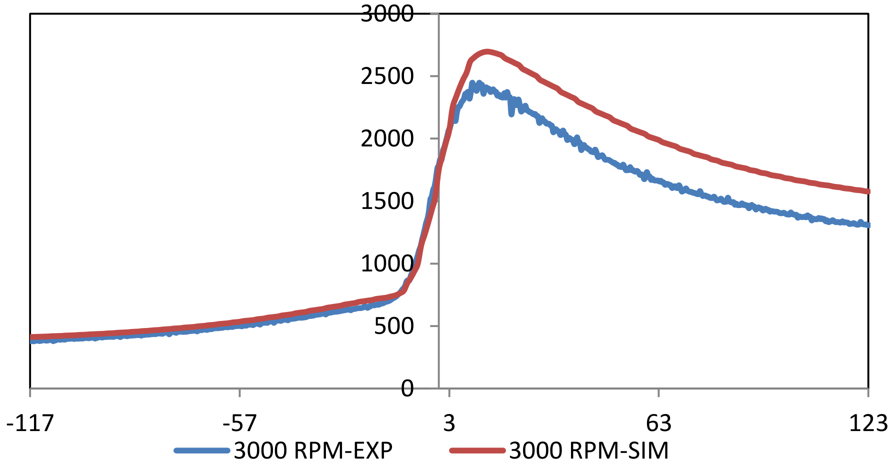


(a)



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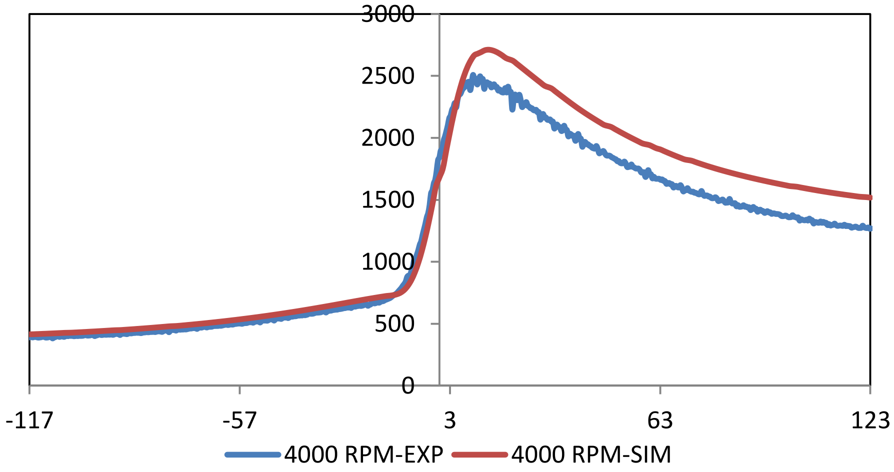

(c)

Figure 5(a)-(c): Comparison between measured and simulated cylinder temperature for engine speed of (a) 2000 rpm (b) 3000 rpm and (c) 4000 rpm.

It was believed that, the moderating rate of temperature reduction is due to the burning of end gas in actual engine combustion. Additional heat was deliberated and causing the rate of temperature reduction becoming lessened. All simulated cases showed an over prediction of cylinder peak temperature.

Table 3: Deviation of simulated cylinder peak temperature against measured data

Engine speed (rpm)	Simulated peak temperature (^oC)	Measured peak temperature (^oC)	Deviation (%)
2000	2556	2362	8.21
3000	2696	2447	10.17
4000	2711	2508	8.09

The difference between the simulated and measured cylinder peak temperatures are expected as the model assumed a complete combustion whereas in actual condition, the dissociation process imposed significant impact by forming intermediate combustion products. Current model also neglect the formation of nitrous oxide, carbon monoxide and other rich combustion products that will absolutely alter the maximum heat deliberated during combustion.

6. CONCLUSION AND RECOMMENDATION

Cycle engine modelling has been carried out and it was proven that this type of model capable of delivering adequate performance prediction for spark ignition engine provided that realistic input and detail geometrical configurations are available. But noticeable discrepancies are still occurred and that required further improvement on the model. Suggestion for further improvement includes the following work:

1. Optimisation of the adjustable Wiebe parameter ***a*** and ***m*** that may be affected by the type of fuel, engine geometrical details and engine operating condition.
2. Inclusion of knock and NO_x model and other rich combustion products that maybe alter the cylinder peak temperature.
3. Investigation of other engine performance parameter such as the brake quantities and engine efficiencies.
4. Switching the combustion model that is more predictive in nature which is not totally rely on existing cylinder data and includes the effect of ignition or injection timing.

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