DEVELOPMENT OF A 125 cc TWO-STROKE, STEP-PISTON ENGINE USING A ONE-DIMENSIONAL ENGINE CODE

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Abstract

A group of engineers from the Faculty of Mechanical Engineering, Universiti Teknologi Malaysia, undertake a project to develop a two-stroke step-piston engine – from design to prototyping. The engine is equipped with a three-port transfer system allowing the air-fuel charge to bypass the crankcase in its route to the combustion chamber. In so doing, the short-circuiting is able to minimize lubricant consumption in relation to the conventional, crankcase engine counterpart. A one-dimensional engine software the GT-Power™ is use to simulate for its performance. Subsequently, it assisted in the development and preliminary trial of the engine prototype. This paper among others highlights the methodology employed and the features of the newly developed engine.

1. Introduction

An engine design and development program was initiated in 2002 to develop R&D capabilities in small power train engineering in Universiti Teknologi Malaysia (UTM). The two-stroke engine design exercise evolved around the development of an air-cooled single cylinder step-piston reciprocating concept. The design is chosen as it provides avenues for innovative work to enhance the utilization of the two-stroke engine design [1]. The programme will eventually leads to the incorporation of features that will improve its performance as well as exhaust emission concentrations. This will eventually lead to the production of a working prototype 125 cc engine for multiple non-automotive platform applications. The step-piston engine is a relatively new type of small internal combustion engines but was not exploited of its full potential. It offers two distinct unique features. Firstly, its rotating mechanism is hybrid in nature whereby its basic operating cycle is the two-stroke engine cycle but possesses a crankcase lubrication system similar to a four-stroke engine of similar displacement size. Secondly, it has a build-in supercharger geometry that will help to improve scavenging process. Due to these features, the engine theoretically has all the attributes of a low emission, high-efficiency mobile power plant providing high power-to-weight ratio attribute.

2. The step piston design

Bernard Hooper [1] first developed the step-piston engine concept (two distinct parts fused in one). The piston works inside a set of cylinder liners, which is correspondingly machined to two different diameters. The smaller piston provides a normal piston-ported 2-stroke cylinder and combustion chamber whilst the larger piston provides the annular but variable pumping chamber. The intake charge is drawn into the annular space through a main reed valve, positioned downstream of the engine carburettor. When the second stroke commences, the charge will be compress and made to pass through a set of multi-ports prior entering the main chamber through a series of circumferential ports. These are detachable ports strategically located at the circumference of the engine bore.

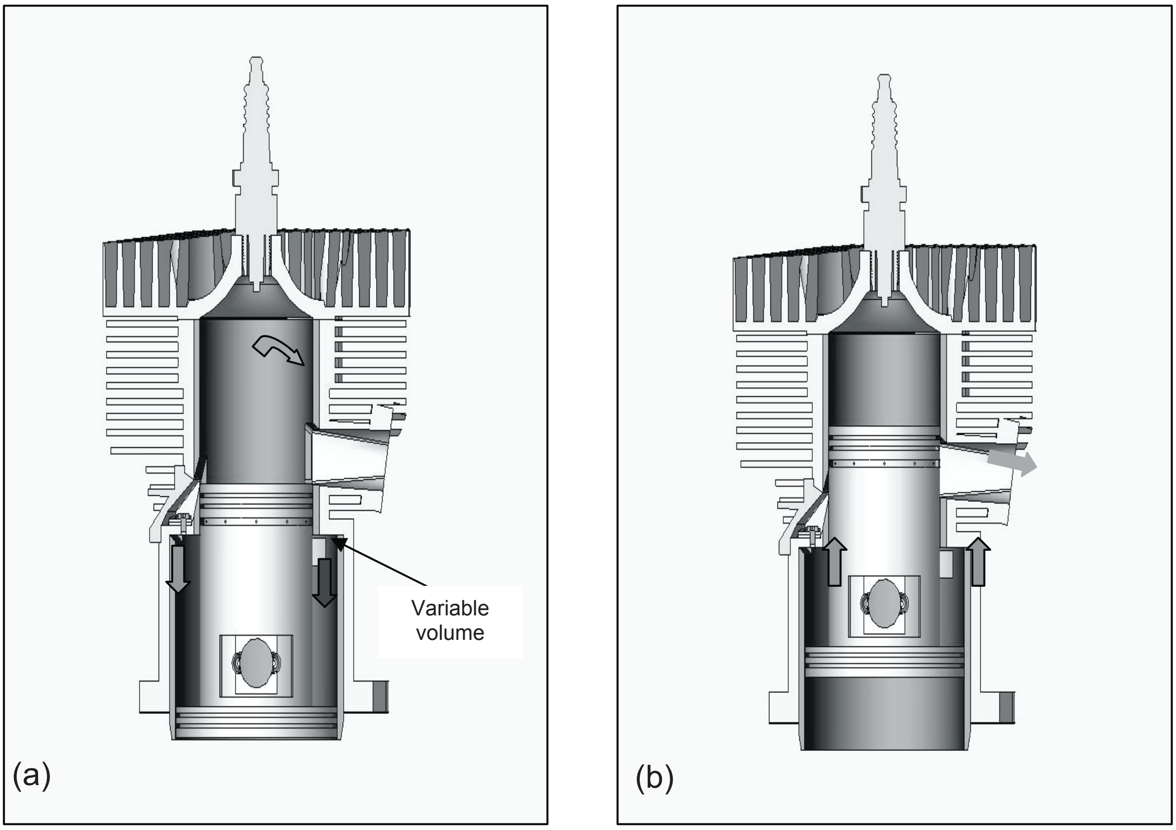

Fig. 1. Piston at the bottom dead centre position (a), and at a position approaching top dead centre (b)

Fig. 1 (a) illustrates the piston geometry with the arrows showing the path of the fresh air charge during induction and transfer. The volume bordering the lower half of the piston (inclusive of the piston skirt), will induct the fresh mixture, during the piston expansion stroke. During the compression-stroke the mixture will pass through a special set of one-way valve (reed valves) and to the transfer ports and eventually into the main combustion chamber (refer Fig. 1(b)). When the piston skirt seals the transfer port entries, the mixture will be trapped and be subjected to further compression until ignition is triggered for the commencement of the combustion cycle.

The spark plug is positioned at 30^{o} to the engine vertical axis. This is to provide an ideal configuration for stratified-charge and lean burn combustion to feature. The general layout of the engine design is shown in Fig. 2 (a). The multi port mixture transfer system will provide an effective mean of discarding the exhaust gases i.e. introducing fresh charge at three equal spacing, positioned at the circumference of the engine bore. Fig. 3 is the positions of the add-on porting units, which consist of a high temperature reed valve enclosed in metal cover. The cover acts as an accumulator in providing the transit volume to the trapped mixture prior entering the next combustion cycle.

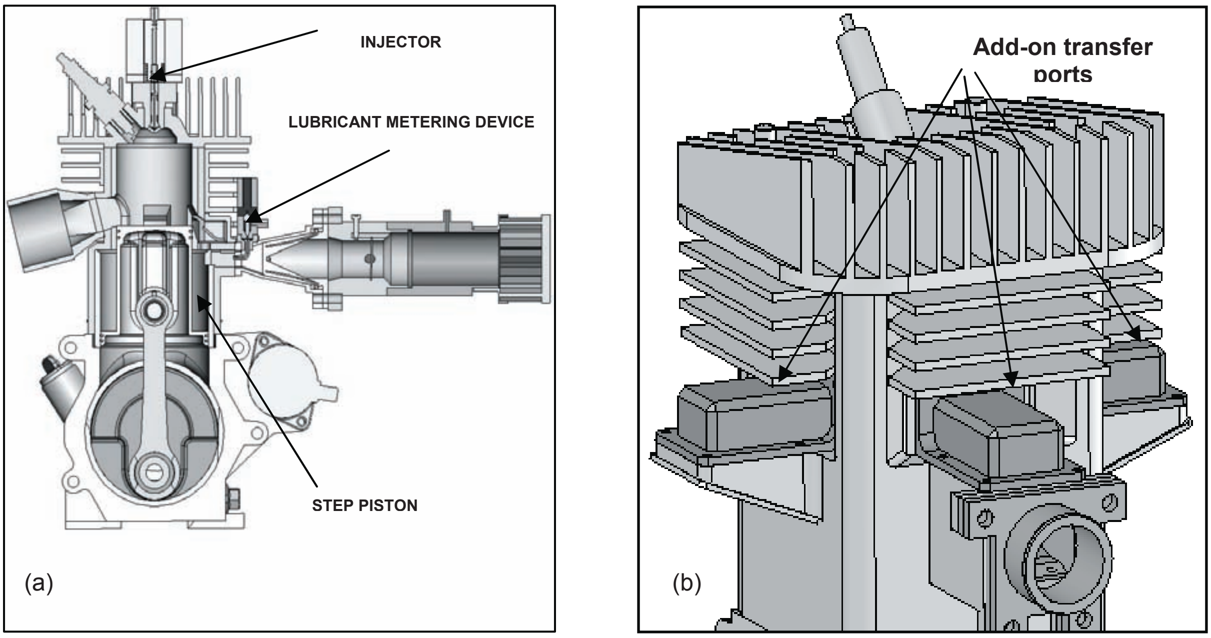

Fig. 2. (a) The engine cross-sectional area, (b) the three-port system

3. Computer simulation

The computer simulations were made using *GT-Power*™ software, a one-dimensional code to predict engine performance [3, 4]. Fig. 3 shows the two-stroke step-piston engine model created using the software. In general the system representation is built into this model using designated "components" (e.g. 'pipes', 'flowsplits', 'cylinders', 'environment' and etc.). They are connected to one another by "connections" (e.g. 'orifices', 'valves').

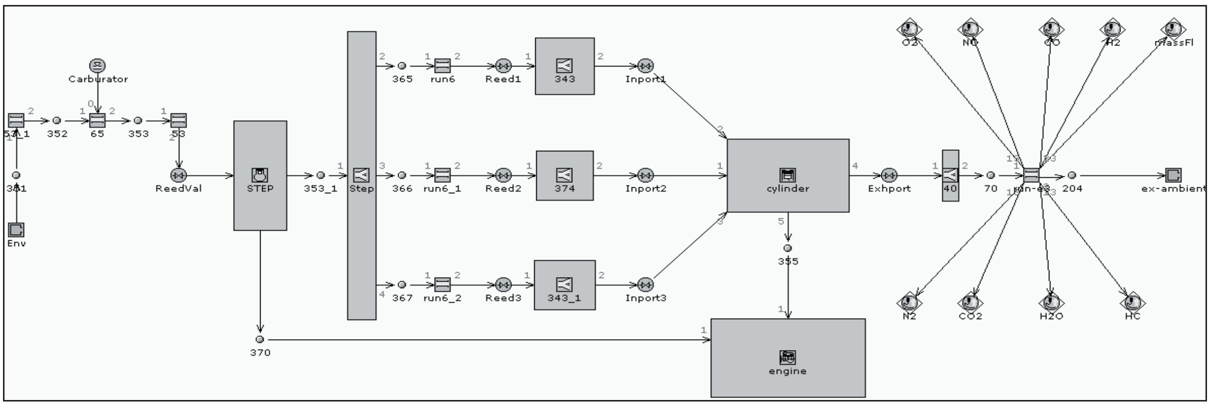

Fig. 3. The modelling of the engine using GT-Power

The intake system starts with an ambient condition designate 'Env'. It is use to describe the boundary conditions of pressure, temperature, and the mixture compositions for the intake and exhaust system. It is connected to a default orifice and subsequently to an inlet pipe.

'Pipe' template is use to model the intake and exhaust systems. It models the flow through tubes, either for constant or tapered diameter. It assumes that the pipe is round and straight. The friction multiplier, heat transfer multiplier and pressure loss coefficient are adjusted to take into account the effect of other physical geometries.

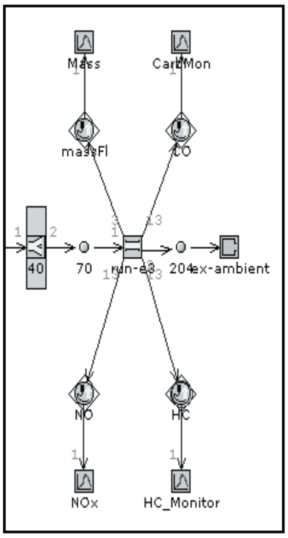

Fig. 4. The emission template

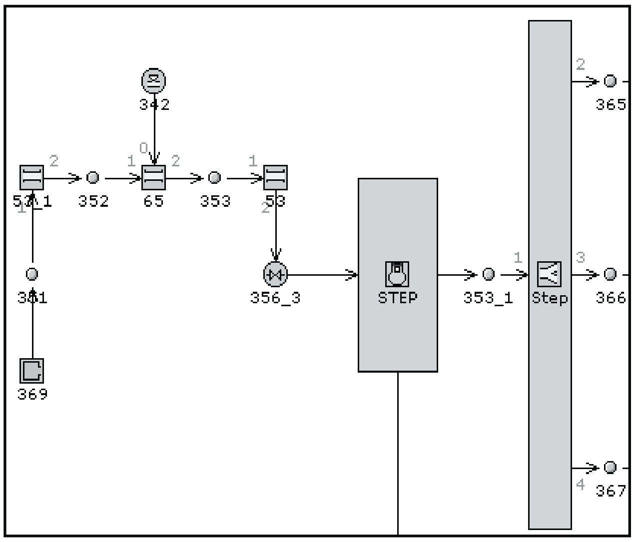

Fig. 5. The modeling of the intake manifold

The carburetor is modeled using ‘InjAF-RatioConn’ Connection. It describes an injector that injects fluid (i.e. fuel, water, etc.) at a prescribed fuel-to-air mixture entering the main pipe. Reed Valve is modeled using ‘Check Valve’ connection and is defined as ‘ReedVal’ or ‘Reed’. The connection is used to dynamically model the check valve. It is a dynamic model, i.e. which makes use of a spring, mass and damper system for analysis. The direction of the linking arrows through the 'ValveCheckConn' connection is also of particular importance. The valve is defined such that the high pressure (at the valve inlet and defined by the linking arrows) causes the valve to open and instantaneously the high pressure at the valve outlet will cause the valve to close. Therefore, the valve acts to prevent the backflow in the reverse direction of the linking arrow.

The step piston cylinder is modeled using ‘Engine Crankcase’ object and defined as *Step*. The crankcase compression ratio is also defined. The ‘flowsplit’ pipe represents the piston step section and several pipes are used to model the volume of the intake pipe (after step cylinder section), leading to the reed valve sections. The flow from the step is distributed into three different ports. As mentioned earlier, each port is equipped with a high temperature reed valve. As such, the ‘flowsplit’ object is here used to model the junction and a key in the port angle. This is base on the x, y and z-axis, of each of the port concern. There is another ‘flowsplit’ assigned after each of the reed valves. These ‘flowsplits’ are used to model the volumes of the intake ports. ‘Ported Valve Connection’ is used to model the respective intake ports. The object defines the characteristics of a ported valve for two-stroke engines, which includes the valve area and the flow coefficients respectively. The code calculates the flow through the valves in each direction using ‘effective area’ i.e. the product of the actual area and the discharge coefficient, C_D. The component called ‘Engine Cylinder’ is used to designate the attributes of engine cylinders. It is used to identify the engine parameters (e.g. bore, stroke, and connecting rod), heat transfer, combustion model, and scavenging models respectively.

The software also provides several combustion models that are suitably used for spark ignition and diesel engines [5, 6]. For this work, the Spark Ignition Turbulent Flame Combustion Model (EngCylCombSITurb) is used to predict the engine in-cylinder burn rate [7]. This prediction takes into account the cylinder’s geometry, spark timing, air motion and fuel properties. Mass entrainment rate into flame front and burn up rate are governed by the following three equations:

$$\frac{dM_e}{dt} = \rho_u A_e (S_T + S_L) , \tag{1}$$

$$\frac{dM_b}{dt} = \frac{(M_e - M_b)}{\tau} , \tag{2}$$

$$\tau = \frac{\lambda}{S_L} , \tag{3}$$

where:

M_e - Entrained mass of the unburned mixture,
t - time,
ρ_u - unburned density,
A_e - entrainment surface area at the edge of the flame front,
ST - turbulent flame speed,
SL - laminar flame speed,
Mb - burned mass,
τ - time constant,
λ - Taylor micro scale length.

The above equations state that the unburned mixture of fuel and air (entrain into the flame front through the flame zone) is at a rate proportional to the sum of the turbulent and laminar flame speeds. The burn up rate is proportional to the amount of unburned mixture behind the flame front, i.e. $(M_e - M_b)$, divided by a time constant τ. The time constant is compute by dividing the Taylor micro scale, λ by the laminar flame speed.

The computational time for the above combustion model is substantially higher than for the non-predictive combustion models (i.e. SIWiebe or Profile). Here, the simulation time has been reduced by setting the 'startup' options to allow a simple Wiebe combustion model to be used for the first several engine cycles, so that the manifolds converged mostly to steady state before the turbulent model is activated.

In-cylinder heat transfer is modelled using Woschni model [8]. The most important difference lies in the treatment of heat transfer coefficients during the period when the valves are open. The present model increases the heat transfer whenever there are large inflow velocities through the intake valves and also during backflow through the exhaust valves.

For the scavenging model, the 'Engine Cylinder Scavenging Data' template is used. This object describes the in-cylinder mixing of incoming fresh gases together with the burned gases that remain from the previous cycle. The 'Exhport' object is use to model the exhaust port. Downstream of the exhaust port (refer Figure 3) is the exhaust pipe. It is modeled using 'flowsplit' and 'pipe'. For the pipe, eight sensors are use to obtain the *Species Mass Fraction* from the exhaust products while one of the sensors is to specifically sample the mass flow rate of the exhaust gas.

The simulation of the engine model is made for 15 cases. It starts at 1500 rpm to 8500 rpm with every 500 rpm increment. Here engine performance data (e.g. brake power, bsfc, bmep, and thermal efficiency) and emissions (NOx, uHC, CO) are recorded. Fig. 4 and 5 illustrate the template used to describe the emission and the inlet manifold performances.

4. Experimental setup

For the laboratory evaluation, the engine is coupled to a *Magtrol* eddy-current type dynamometer. The fuel consumption is measured using an *Ono-Sokki* F-series fuel flow meter and the airflow rate is measured using an air box, fitted with an orifice plate (according to BS 1042). A *Toxin* (series 300) exhaust analysis system is use to analyse for the exhaust concentrations. The analyser employs a series of detectors to measure for HC, CO, CO_2, NO_X and SO_2 respectively. The gas is sample at the mid-section of the engine exhaust system via a particulate filter. A *Kistler* type 760 transducer (in conjunction with a charge amplifier) is to monitor the dynamic variation of combustion pressure in the cylinder. A *Dewetron* combustion analyser records the output signal (from the amplifier) and the top dead centre pulse (from a crankshaft optical encoder). The data is analyzed using *FlexPro*™ spreadsheet software. Throughout the program, the fuel-lubricant ratio is set at 45:1 (volume: volume), the computed performance data is corrected, taking into account the ambient temperature, pressure and relative humidity.

5. Engine specifications

This engine technical specification, based on the exhaustive laboratory evaluation of the prototype is given in Table 1 below. The emphasis of the design is to achieve high power to weight-ratio characteristic. Thus due this consideration, the selection of light materials (aluminium alloys) has been given due priority. The initial version is only equipped with conventional fuelling system whilst the upgraded version is equipped with a dedicated fuel direct injection system and an engine management system (EMS). Fig. 6 depicts the engine undergoing trial with the equipment described in Section 4 of this paper.

Table 1. Engine specifications

Parameter	Size/Feature
Capacity (cc)	125
Bore (mm)	54.2
Stroke (mm)	54.2
Compression ratio	9.6
Ignition system	Capacitive-Discharge
Fuel/scavenging systems	Carburetted/multi-port loop scavenge
Cooling	Air-cooled
Max. power, kW@rpm	8.6@7500
Max. torque, Nm@rpm	12.5@5000
Power-weight ratio (kW/kg)	0.46

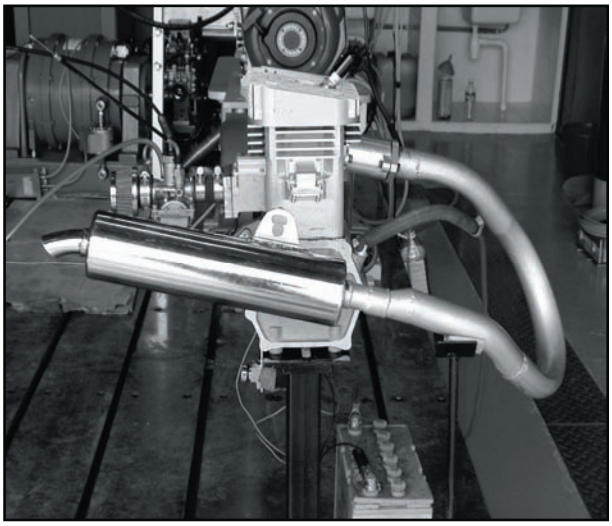

Fig. 6. The engine during trial

6. Results

The some of the comparative results (experimental versus simulation) are illustrated in Fig. 7 to Fig. 12. Referring to the data, they are confined to the in-cylinder pressure, engine performance and emission respectively. In general, the close agreement of the simulated and experimental data is very pronounced. The pressure profiles obtained through experimental means is 93% of the simulated one, indicating minor losses, either due to low mixture quantity or blowly effect.

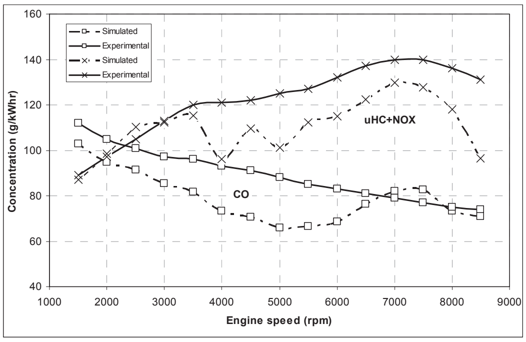

Fig. 7. NO_X + UHC concentration against engine speed

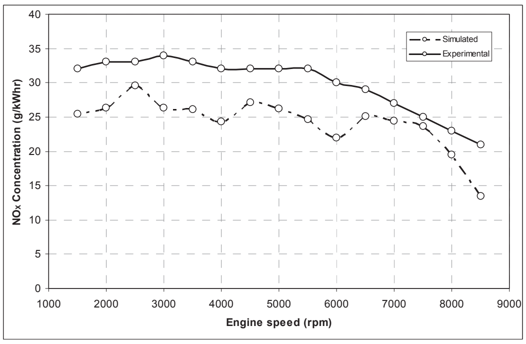

Fig. 8. CO concentration against engine speed

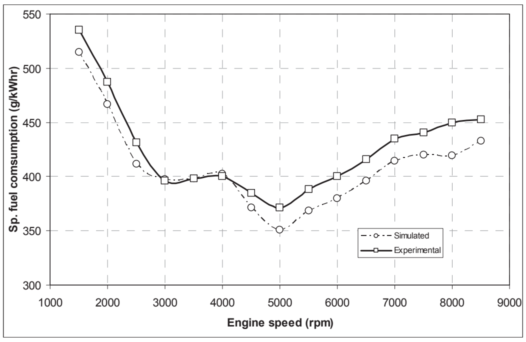

Fig. 9. The specific fuel consumption profile against engine speed

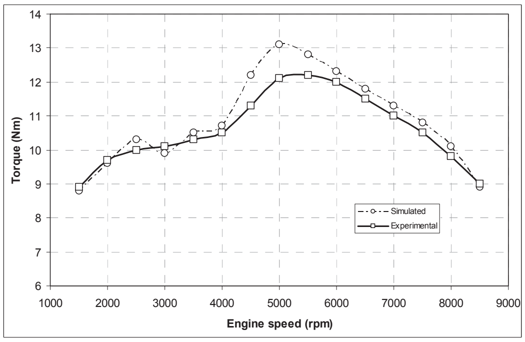

Fig. 10. Torque against engine speed

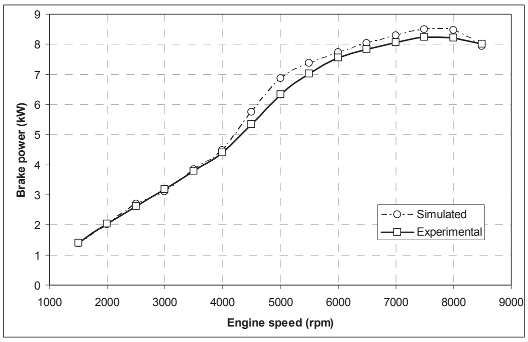

Fig. 11. The brake power output against engine speed

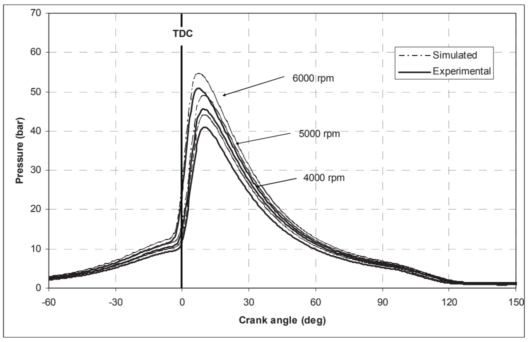

Fig. 12. Combustion pressure profiles

The emission results indicate higher values than those of the simulated ones. This is to be expected, as lower brake fuel combustion efficiency (bsfc) may be the contributing factor to this irregularity. The theoretical and experimental brake power, torque and fuel consumption curves (at maximum load) show close agreement through out the speed range.

7. Conclusions

The application of a one-dimensional engine model on a non-conventional two-stroke, air-cooled engine was successfully performed. This has led to the development of its first prototype and the initial laboratory validation within the stipulated project schedule. The experimental data is in close agreement with those of the simulated data. The simulation technique is applied not only to obtain a better insight of the unsteady flow processes but also for the development of a variable geometry of the exhaust system so as to enhance its trapping and discharge efficiency over a wide speed range.

Current work is geared towards the improvisation of the engine prototype to incorporate a dedicated fuel injection system (in conjunction with an engine management system (EMS)) to enhance the exhaust emission factor.

8. Acknowledgements

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9. References

[1] Patent Number 4,522,163, United State Patent, 11th. June 1985.
[2] One Step Beyond, Car Design & Technology, pp 29-30, October 1992.
[3] Gamma Technologies, *User's Manual of GT-Power Version 5.2*, December 2001.
[4] Christopher Ciesla, Rifat Keribar and Thomas Morel, Gamma Technologies, *Engine / Powertrain /Vehicle Modeling Tool Applicable to All Stages of the Design Process*, SAE Paper 2000-01-0934, 2000.
[5] H. Hiroyasu and T. Kadota [1976], *Models for Combustion and Formation of Nitric Oxide and Soot in Direct Injection Diesel Engines*, SAE Paper 760029, 1976.
[6] N. C. Blizard and Keck, J. C. [1974], *Experimental and Theoretical Investigation of Turbulent Burning Model for Internal Combustion Engines*, SAE Paper 740191, 1974.
[7] Gamma Technologies, *Reference Manual of GT-Power Version 5.2*, December 2001.
[8] G. Woschni, *Universally Applicable Equation for the Instantaneous Heat Transfer Coefficient in the Internal Combustion Engine*, SAE 670931, SAE Trans. Vol. 76, 1967.