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Preliminary Investigation of a Converted Four-Stroke Diesel to Alpha V-Shaped Stirling Engine

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Abstract: This study presents a preliminary investigation on a converted four-stroke diesel to an air charged V-shaped Alpha Stirling engine. The engine was manufactured with a total swept volume of 194 cc., volume compression ratio of 1.84, 90° phase angle and air as a working gas. The engine was designed to fulfill the requirements of hot end temperature up to 1000°C, cold end temperature of 20 to 30°C, charge pressure of minimum 1 bar or above and engine speed up to 1200 rpm. Design considerations of developing a simple and low cost Alpha-typed Stirling engine using major components of both commercial diesel engine and industrial mass production were discussed. Major modifications were done on the engine heater head design in order to cater for both natural gases and biomass fueled heating process as the external heat source. The net power output was estimated about 25 W at minimum charge pressure of 1 bar using Beale formula. The preliminary results show that the friction torque increases with the increase of engine speed. The minimum friction torque of 0.80 Nm was obtained at minimum speed of 300 rpm and the maximum friction torque obtained was approximately 1.30 Nm at the speed of 1200 rpm. The gas pressure inside the working cylinder also increases with the increase of engine speed and the maximum mean pressure obtained with the effect of engine speed and hot temperature was approximately 1.41 bar.

Key words: Stirling engine, alpha-typed, design considerations, low cost, heater head, multi-fuel heating

INTRODUCTION

Stirling engines are eminent for their prospect of high efficiency, safe operation, long life, fuel flexibility, low emissions, low pollution, low vibration and noise level (Scott *et al*., 2003) compared to internal combustion engine. However, a wide spread utilization of Stirling engines has not yet become a reality due to commercial and economical factors (Raggi *et al*., 1997). Issues such as low engine power to weight ratio and high manufacturing costs remain as main challenges that makes a mass production of Stirling engines is not feasible at present.

Modifying an internal combustion engine into the Stirling engine has been the preferred alternative especially for academic and experimental purposes (Raggi *et al*., 1997). Apparently, the manufacturing cost of the modified engine can be very much reduced since the ready engine components have the quality in terms of material strength and parts precision. The engineering time (design and fabrication) can be shortened and the spare parts can be easily sourced if the engine is subjected to wear and tear.

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The numerous investigations made by scientists and engineers since from the invention of the engine have made good base line information for designing engine system, but a more insight is essential to design systems together for thermo-fluid-mechanical approach. It is seen that for successful operation of such system a careful selection of drive mechanism and engine configuration is essential. An additional development is needed to produce a practical engine by selection of suitable configuration; adoption of good working fluid and development of better seal may make Stirling engine a real practical alternative for power generation (Thombare and Verma, 2006). A number of promising research study involving design, manufacture and performance test of the Stirling engines are as follow:

Karabulut *et al*. (1998) manufactured and tested Alpha V-type Stirling engine at different set up values of pressure and hot source temperature. The electrical furnace is used as a heat supply and air as a working gas. The engine produces maximum output power of 65 W at the operating parameters of 1100°C hot source temperature, 2.5 bars charge pressure, 500 rpm speed and 0.4 Nm torque. Lane and Beale (1999) developed and tested a free-piston biomass-fired 1 kWe Stirling engine generator known as Biowatt™. The engine utilizes thermal output of 4 kW to produce 1 kwe electrical power output. Working medium is helium and heat sources are propane and flue gas from chunk wood and pellet burner. Design life of the engine is greater than 40,000 h and to date; over 2000 full thermal start-stop cycles have been completed. The pellet-fueled system has achieved over 1 kWe consuming of 1.5 kg h^{-1} of pellets (8-10% moisture). Podesser (1999) developed a 3 kW Alpha Stirling engine for electricity production at rural villages. The engine produces the electrical power output of 3 kWe at the thermal power input of 12.5 kW, flue gas temp. of 1000°C and engine cooler temperatures of 30 to 70°C. Air and nitrogen are used as the working gas. Cinar *et al*. (2004) manufactured and tested a 192 cc. Beta-type Stirling engine operating at atmospheric pressure. The engine receives heat from the electrical heater at the temperatures of 800, 900 and 1000°C. The engine generates maximum power output of 5.98 W at 208 rpm, with the hot-source temperature of 1000°C. Kongtragool and Wongwises (2007) experimented a twin power piston, Gamma-configuration and low temperature differential Stirling engine using a solar simulator from LPG gas burners as a heat source. At the maximum simulated solar intensity of 7145 W m^{-2} or heat input of 261.9 J sec^{-1}, with a heater temperature of 436 K, the engine generates a maximum torque of 0.352 Nm at 23.8 rpm, a maximum shaft power of 1.69 W at 52.1 rpm and a maximum brake thermal efficiency of 0.645% at 52.1 rpm. Kongtragool and Wongwises (2008) developed and tested a four power-pistons, Gamma-configuration and low-temperature differential Stirling engine. The engine obtains heat from a solar simulator made of 100 W halogen lamps. At a maximum energy input of 1378 W and a heater temperature of 439 K, the engine produces a maximum torque of 2.91 Nm, a maximum shaft power of 6.1 W and a maximum brake thermal efficiency of 0.44% at 20 rpm.

The purposes of this study are to develop a simple, low-cost and multi-fueled Alpha V-shape Stirling engine throughout the minimization of custom-made components by utilizing major parts of four-stroke diesel engine and industrial mass production and to perform a preliminary investigation on the engine assembly to obtain friction torque and pressure variations for different engine speeds.

GENERAL SYSTEM DESCRIPTION

The Stirling engine was developed using critical components of a four-stroke diesel engine. The original parts such as piston, piston rod and pin, piston ring, cylinder head, crankshaft, semi round shaft key and thrust bearings are the ready engine components. The rest of the engine components comprising of the regenerator, connecting tube, cylinder and cylinder platform are custom-made. Major modifications were done on the engine heater head in order to cater for both natural gases and biomass fueled heating process. An extension piston known as piston crown was fabricated and coupled to the top of the original piston of the hot cylinder. The intentions are to prevent the lubrication of original

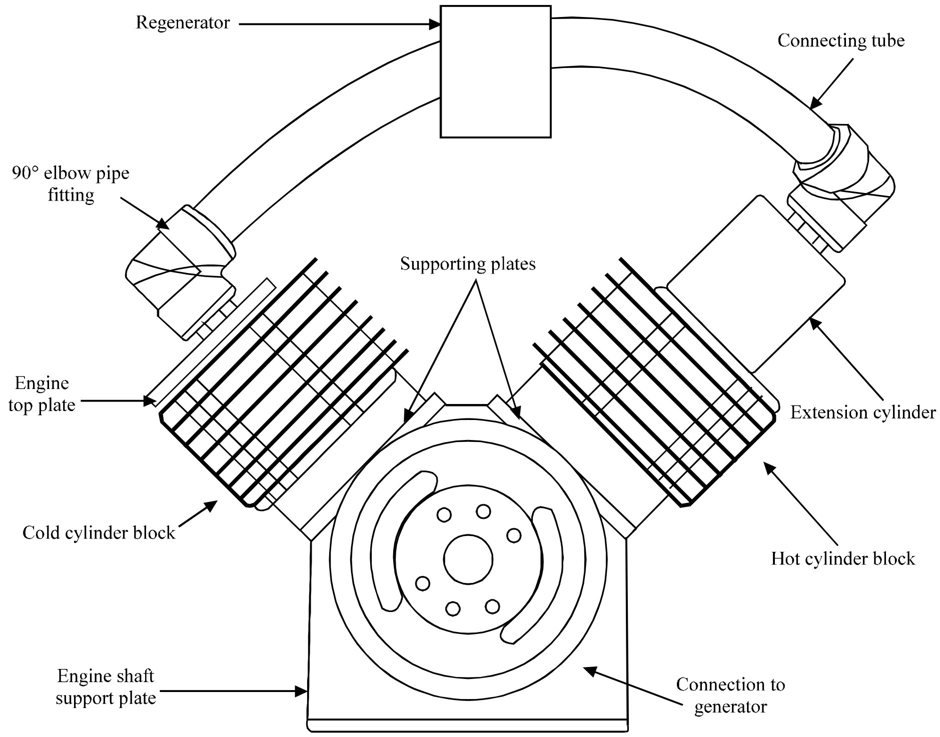

Fig. 1: A schematic view of alpha V-shape stirling engine

expansion piston and cylinder to be completely dried up and cause friction during the heating process and also to prevent further stress to the piston ring during the engine movement. The original hot cylinder was also extended to cater for the extension of piston. A simple cylindrical shape regenerator filled with metal balls was designed and fabricated as a heat exchanger. The cold side of the cylinder has no extension piston and it operates at the ambient temperature. The rectangular fins surrounding the cylinder block help to remove heat out from the cold section. A schematic view of the developed engine is as shown in Fig. 1.

DESIGN CONSIDERATIONS

Engine Overall Design

Essentially, the engine consists of a crankcase with two cylinders situated at 90° angles to each other or in V-shape orientation, a crankshaft holds flywheel from one end to the crankcase, two connecting rods and two pistons which are both connected to the same crank journal. A stainless steel connecting tube with a cylindrical-bed regenerator filled with metal balls was positioned in between the hot and cold cylinders. Majority of the parts were made of stainless steel such as cylinders, connecting tube and regenerator. Main advantages of stainless steel are its resistance to high temperature and superior strength characteristic. Pistons were made of aluminum due to its lightweight and its combined characteristics of moderate strength at elevated temperature and thermal conductivity. And the pipe elbows at the top head of both hot and cold cylinders were made of brass.

The original cylinder blocks at both hot and cold sections were with external fins. The rectangular fins help to remove out heat at the bottom dead space of the hot cylinder and to maintain low temperature at the boundary surface of the cold cylinder. The extension cylinder was the primary section, which is subjected to external heating process. The original expansion cylinder was not subjected to direct heating in order to avoid the lubrication of original expansion piston and cylinder to be completely dried up and cause friction during the heat-up process and also to prevent further stress to the piston ring during the engine movement.

Heater Head

The extension cylinder with a total height of 65 mm covers the primary heating section of the engine. The base of extension cylinder was connected to the original cylinder block with external fins. The design of the extension cylinder was compatible with the incorporation of a swirl burner to the engine heater head. The extension cylinder was made of stainless steel with a thickness of 4 mm.

Regenerator

Regenerators in Stirling engines are designed to provide sufficient thermal contact to the working gas to minimize loss due to irreversible heat transfer while generating as little viscous loss as possible (Backhaus and Swift, 2001). A cylindrical bed regenerator made of stainless steel containing a large amount of commercially available ball bearing was utilized for the engine. The cylindrical shape regenerator was selected because it is easy to manufacture using common parts from industrial mass production. And the ball bearings with 4 mm diameter are cheap and easily available.

Burner

The swirl burner with two tangential inlets was designed to cater for various heat sources such as propane, butane and liquefied petroleum gas (LPG) from natural gases and producer gas from biomass gasification and/or combustion process. The layout of the burner covers the overall diameter of 130 mm and height of 65 mm. Figure 2 portrays the layout of the swirl burner for the engine heater head.

ENGINE SPECIFICATIONS

Engine Configuration

The engine was Alpha-configured which means that both original hot and cold pistons and cylinders are identical. The engine is in V-shaped orientation and the phase angle is 90°.

Engine Speed

The engine was projected to work with a nominal speed of 1000 rpm. The determination of the engine speed is mainly influenced by the regenerator effectiveness during the engine operation.

Bore and Stroke

The cylinder bore and stroke were fixed at 53 mm and 44 mm, respectively resulting in swept volume of 97 cc.

Dead Volume

The total dead volume of the engine was 115 cc., which covers 13 cc. of expansion cylinder, 15 cc. of compression cylinder and 87 cc. of regenerator tube. The total dead volume for the engine was about 54% of the total volume. In normal Stirling engine design practice, the total dead volume is approximately 58% of the total volume (Martini, 1979).

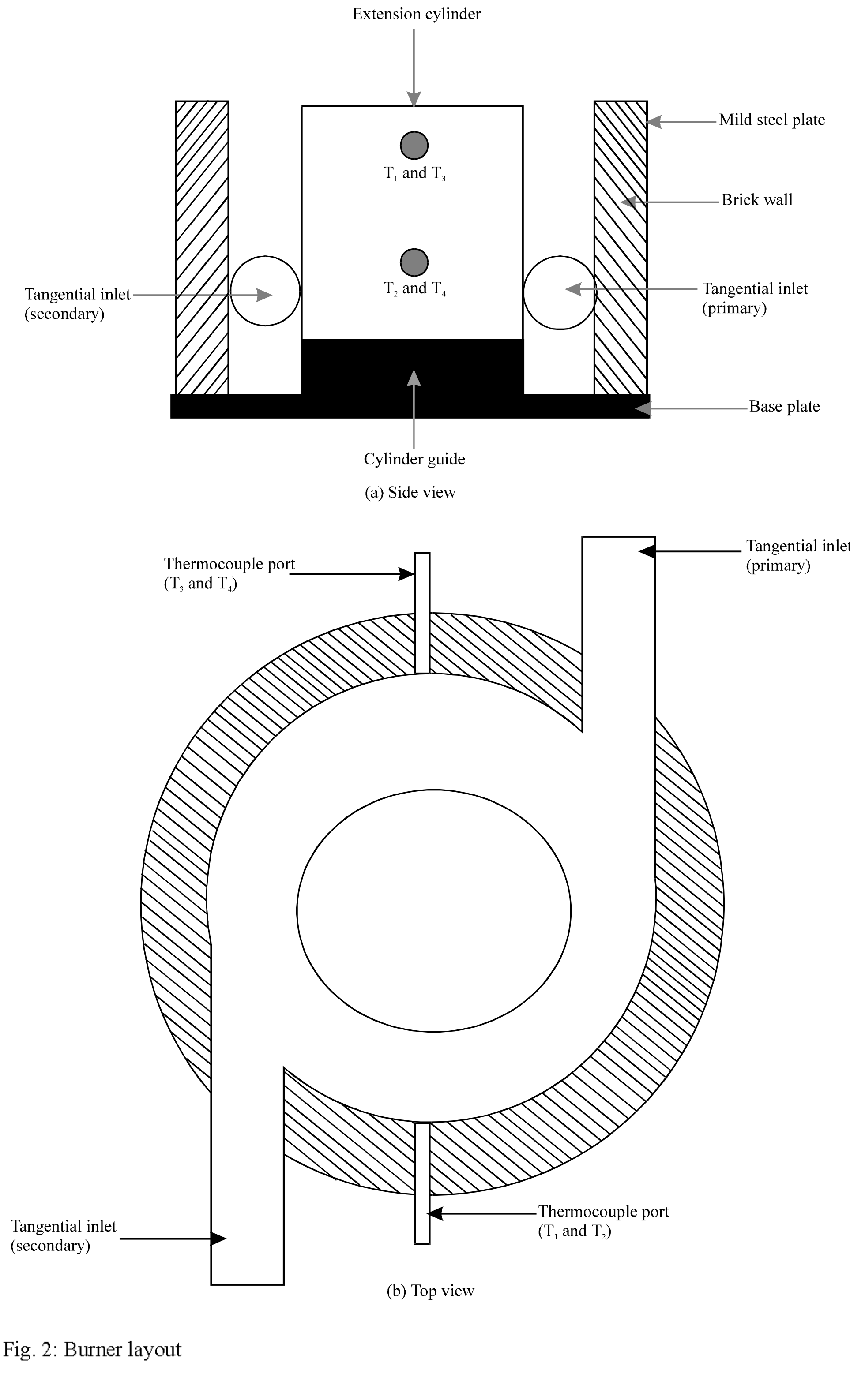

Fig. 2: Burner layout

Volume Compression Ratio

The volume compression ratio was calculated as 1.80, which is within the acceptable range for Stirling engines. Whatever type of engine configuration, it is difficult to increase the compression ratio above 2.5 (Walker, 1980). Attempt to increase this compression ratio will result in having inadequate surface area for heat exchange.

Heater Temperature

The hot cylinder was made of stainless steel. Thus, it implies a range of wall temperature within 650 to 750°C and for this reason; a working gas temperature inside the hot cylinder of about 550 to 650°C was expected (Raggi *et al*., 1997).

Cooler Temperature

The engine was naturally cooled at the ambient temperature of 30°C. The external fins of the cold cylinder promote heat removal from the working gas.

Working Gas

Air was used as a working gas due to its free availability and low risk of explosion. Besides, at low engine speeds and low power densities, there is no significant difference between air and natural gases such as hydrogen and helium in terms of the engine performance (Beale, 1984).

PERFORMANCES ESTIMATION

Heat Input

The heat input was theoretically derived from thermodynamic analysis:

$$Q = \dot{m}\, Cp(T_H - T_L) \tag{1}$$

with an assumption that T_H is at maximum of 650° C and T_L is at ambient temperature of 30°C. Mass flow rate of air as the working gas, $\dot{m}$ was determined from the multiplication of the engine volume flow rate, $\dot{V}$ in m^3 sec^{-1} and air density, ρ_{air} in kg m^{-3}. And Cp is the specific heat of air in kJ kg^{-1} K. The theoretical heat input for the engine was estimated approximately 1120 J sec^{-1}.

Power Output

The engine power output was approximated using Beale formula:

$$P_{NET} = B_N \times P_{mean} \times V_{swept} \times f \tag{2}$$

The Beale number, B_N is taken at 0.015, which is approximately, correct for all types and sizes of Stirling engines (Walker, 1980). The engine swept volume, V_{swept} in m^3 and mean pressure, P_{mean} in bar are obtained from theoretical calculations. The engine speed, f was taken at the nominal value of 17 Hz. The net power output of the engine, P_{NET} was estimated about 25 W at minimum charge pressure.

Thermal Efficiency

The thermal efficiency is estimated about 33% at a probable value of 50% of the Carnot efficiency (Walker, 1980). This is based on the assumption of both maximum and minimum temperature of 600 and 30°C.

$$\eta = \frac{\text{Power output}}{\text{Heat supplied}} = 0.5 \times \left[\frac{T_{max} - T_{min}}{T_{max}}\right] \tag{3}$$

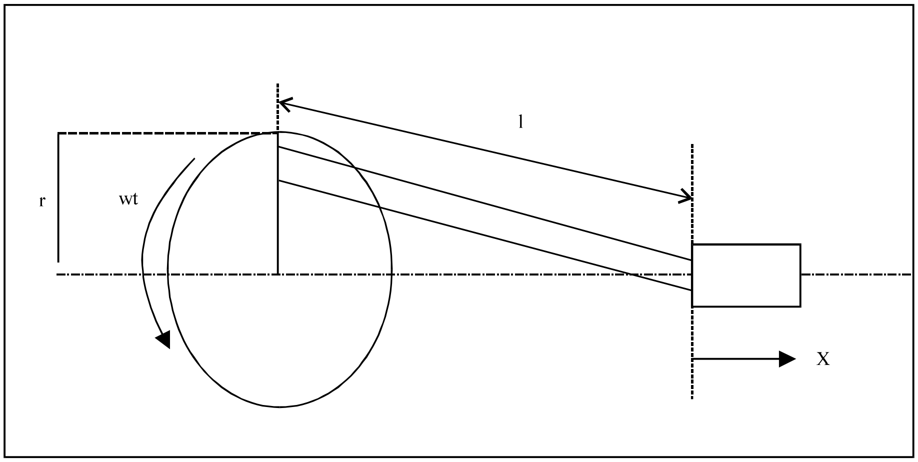

Fig. 3: Slider crank mechanism

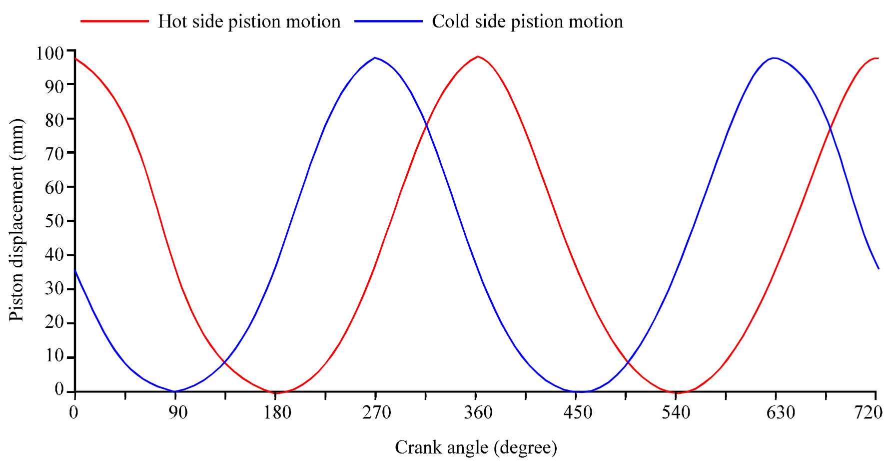

Fig. 4: Piston displacement (x) with respect to crank angle (wt)

Kinematics Analysis of Slider Crank Mechanism

As shown in Fig. 3, let the crank radius as r, the connecting rod length as l and the crank angle as wt. The crank radius is 48.5 mm and connecting rod length is 97 mm. Crank angle will be observed from 0 to 720 degrees.

The relationship of piston displacement, x and crank radius, angle and connecting rod is given as,

$$x \cong l - \frac{r^2}{4l} + r\left(\cos wt + \frac{r}{4l}\cos 2wt\right) \tag{4}$$

Figure 4 shows the plot of piston displacement curves for both hot and cold pistons with respect to crank angle, which is in sinusoidal motion.

Work Diagram

The work diagram (PV diagram) is useful in interpreting the pressure variation with respect to the swept volume inside both hot and cold cylinders. The cycle is based on Schmidt-Cycle design equations, which ideally used for the Stirling cycle (Thombare and Verma, 2006).
Volume of expansion space (hot cylinder):

$$V_e = \frac{1}{2} V_E (1 + \cos\phi) \tag{5}$$

Volume of compression space (cold cylinder):

$$V_C = \frac{1}{2} k V_E \left[1 + \cos(\phi - a)\right] \tag{6}$$

Instantaneous pressure:

$$p = p_{mean} \frac{1 - \delta}{1 + \delta\cos(\phi - \theta)} \tag{7}$$

$$\delta = \frac{\left(\tau^2 + 2tk\cos\alpha + k^2\right)^{\frac{1}{2}}}{t + k + 2S} \tag{8}$$

$$\tan\theta = \frac{k\sin\alpha}{\tau + k\cos\alpha} \tag{9}$$

Where:

$$S = \frac{2X\tau}{\tau + 1}$$

The work diagram of the engine comprising of the pressure variation in the expansion volume (hot cylinder), compression volume (cold cylinder) and total working volume (hot and cold cylinders) is presented in Fig. 5.

PRELIMINARY INVESTIGATION

Static Friction Test

The test was done to determine the static friction power generated from the engine. The engine was loaded with standard masses to initiate the movement. The static friction power was calculated by using the following equation:

$$P_s = 2\pi NT \tag{10}$$

Torque, T was derived from the multiplication of masses to initiate the movement converted into force and the radius of the flywheel, r. And the engine speed, N is assumed at nominal 1000 rpm. The static friction power, P_s was found to be approximately 36 W.

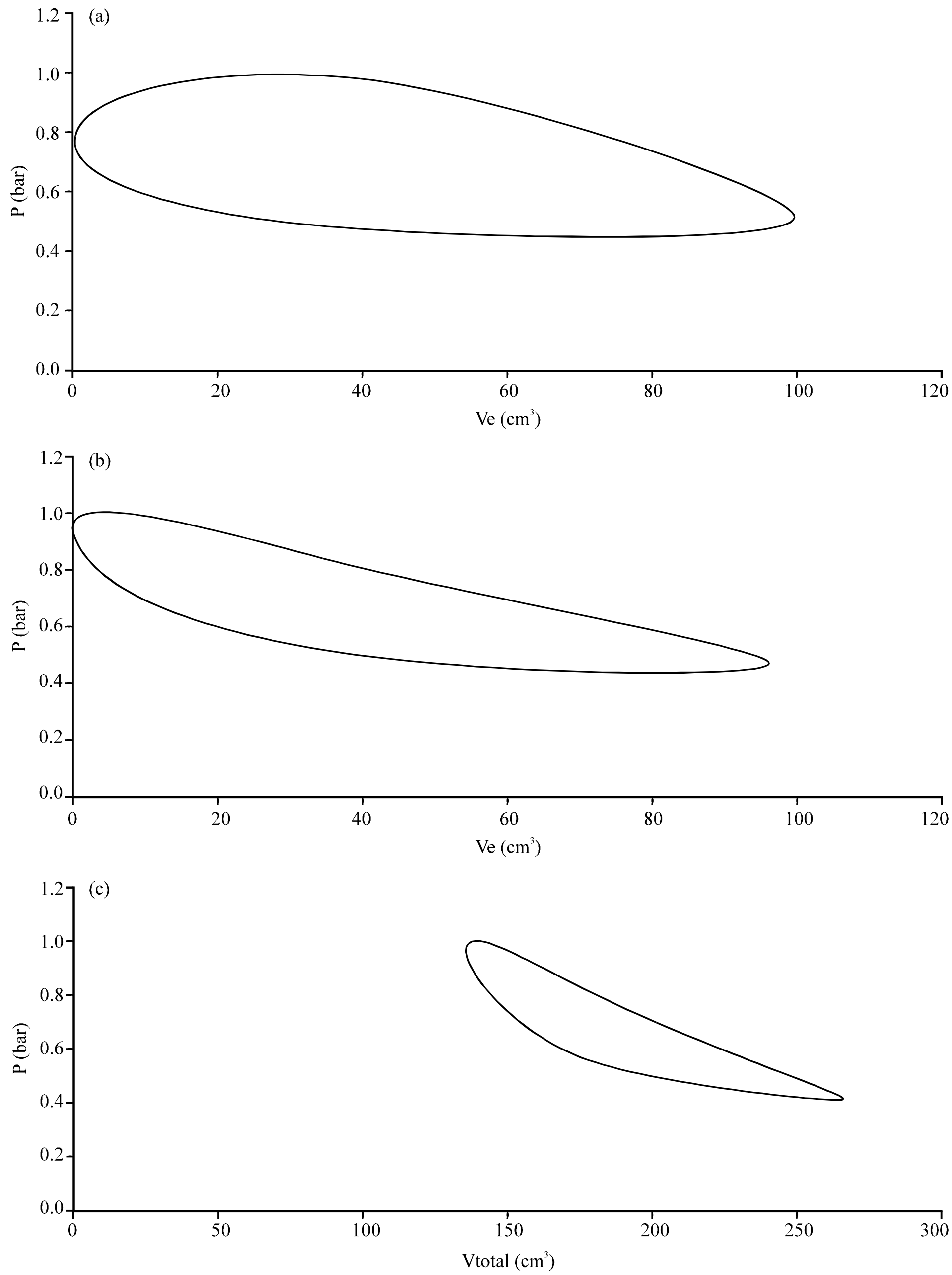

Fig. 5: Work diagram (PV diagram) of Alpha engine for (a) expansion volume, (b) compression volume and (c) total volume

Dynamic Friction Test

Dynamic friction test was done to determine the friction torque variation of the engine during rotary motion. The engine was coupled to the dynamometer and rotated at different speeds. The test set-up is as shown in Fig. 6. The result of friction torque variation of the engine with different engine speeds is presented in Fig. 7.

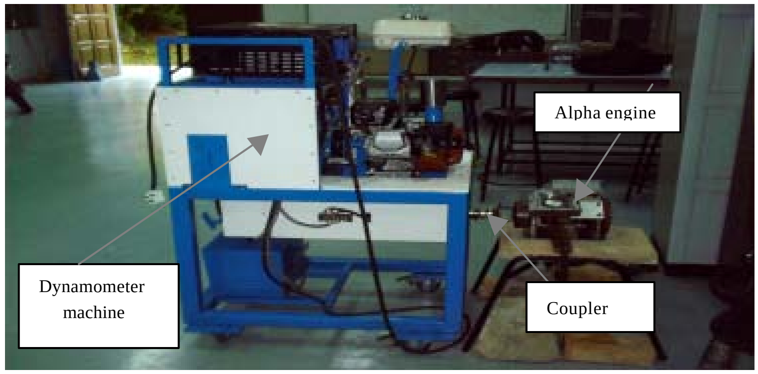

Fig. 6: Test set-up for dynamic friction test

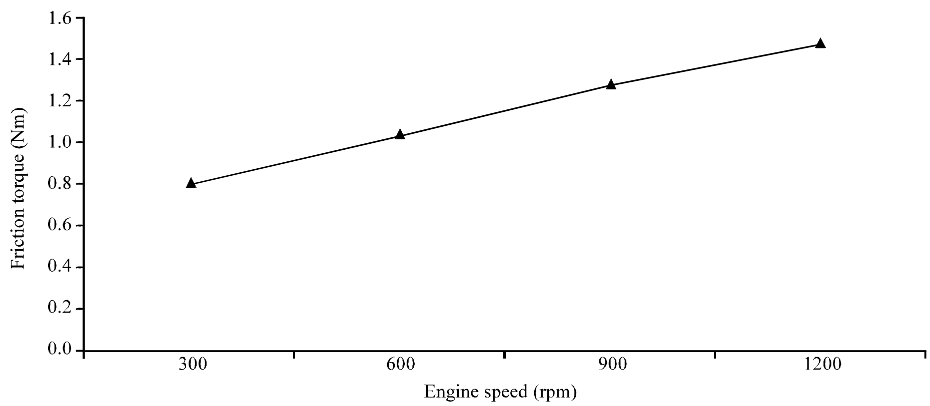

Fig. 7: Friction torque variation of the engine with different speeds

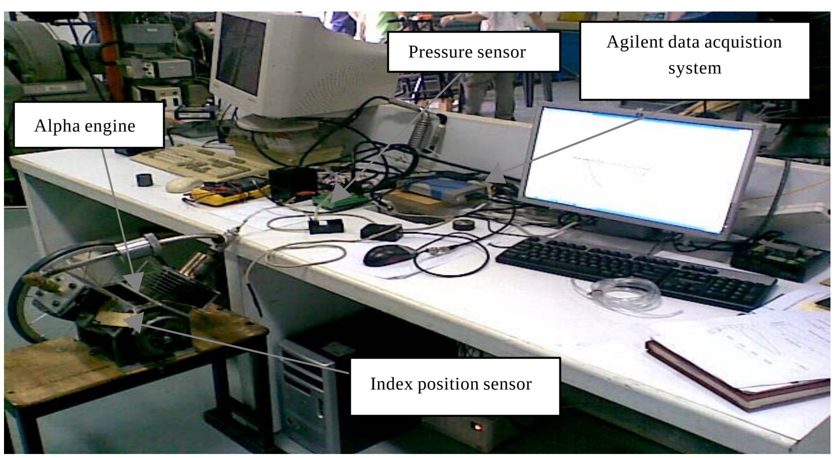

Fig. 8: Test set-up for pressure versus speed measurement at ambient temperature

Pressure Variation with Different Engine Speeds at Ambient Temperature

The actual pressure variation with different engine speeds was done using the test set-up as shown in Fig. 8. The pressure-crank angle measurement was performed using Agilent Measurement manager (AMM data acquisition system). The raw data of crank positions and time were manipulated to obtain speed in revolution per minute. Figure 9 depicts the plot of pressure variation with different engine speeds at ambient temperature.

Pressure Variation with Different Engine Speeds at the Elevated Temperature

The next stage was to perform the pressure versus engine speed measurement when Bunsen burners heat the hot cylinder. For this particular test, the regenerator tube was insulated with insulation rope to prevent heat loss. The test set up is as shown in Fig. 10.

Figure 11 shows the mean pressure variation with different engine speeds at the elevated temperature within a range of 50 to 300°C. The mean pressure was determined from $P_{mean} = (P_{max} + P_{min})/2$, where both maximum and minimum pressures were obtained from the test.

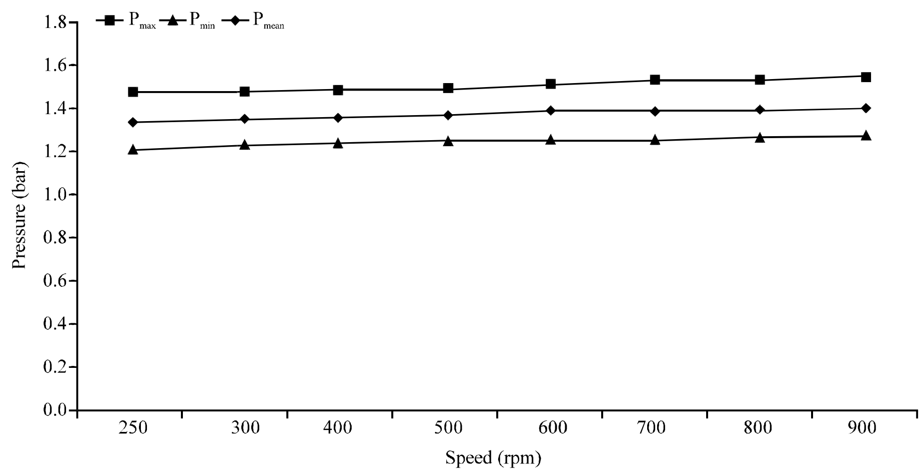

Fig. 9: Pressure variation with different engine speeds at ambient temperature

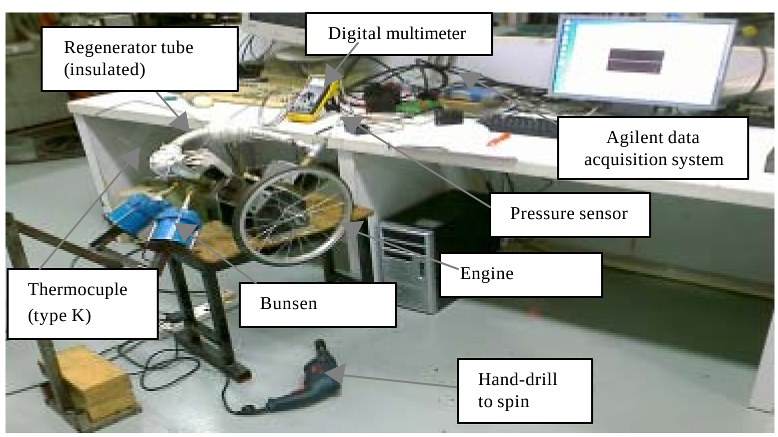

Fig. 10: Test set-up for pressure versus speed measurement at the elevated temperature

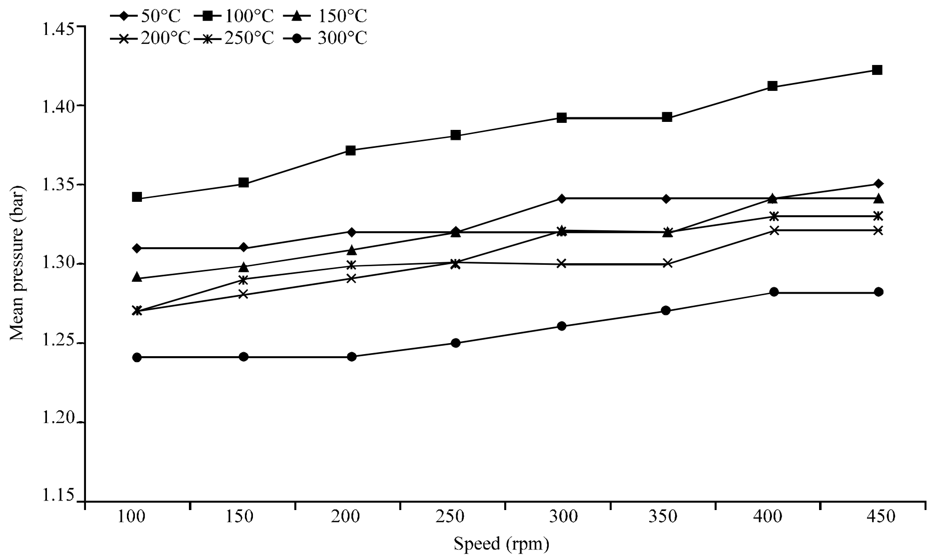

Fig. 11: Mean pressure variation with different engine speeds at the elevated temperature

RESULTS AND DISCUSSION

The design specifications of the fabricated engine were within the acceptable range. The theoretical heat input of 1120 J sec^{-1} was achievable at the hot temperature of 650°C and cold temperature of 30°C. For the engine heater head made of ordinary stainless steel, it was possible to have the hot temperature of 650°C inside the engine at the heat source temperature of 700°C and above. Thus, the use of swirl burner for heating the engine heater head was expected to be practical since heat sources such as natural gases and producer gas of biomass gasification and/or combustion are capable of producing heat up to 1000°C.

The estimated power output for the engine was 25 W using the Beale formula. The Beale number of 0.015 was derived from the Beale chart at the heater temperature of about 630°C. And the charge pressure for estimating the net power output was taken at minimum 1 bar. Based on Beale formula, the net power output is a function of Beale number, mean pressure, swept volume and engine speed and Beale number is derived from the heater temperature. Therefore, it is important to increase the heater temperature, mean pressure and engine speed in order to overcome the friction torque generated by the engine and to improve the engine performance.

The minimum friction torque of 0.80 Nm was obtained at minimum speed of 300 rpm throughout the dynamometer test and the maximum friction torque obtained was approximately 1.30 Nm at the speed of 1200 rpm. In terms of power, both minimum and maximum friction power were determined as 25 W and 160 W respectively. It was suspected that the increase of friction torque as the engine speed increases was contributed by the increase of tension of the original piston ring used as the hot piston sealer with the effect of high temperature. The use of different material for the piston ring such as Teflon is recommended for future experimental test.

Based on the plots of pressure versus speed, pressure increases when the engine speed increases at both ambient and elevated temperature. However, there was a very slight increase of pressure when the hot cylinder was heated. The maximum mean pressure was approximately 1.41 bar at 100°C compared to 1.38 bar at ambient temperature. It was slightly higher than the typical requirement of 1 bar. The pressure started to drop after the hot cylinder was heated above 100°C. Further investigation is required to determine any possibility of working gas leakage to atmosphere during heating process.

CONCLUSION

The design and manufacture of Alpha Stirling engine using major components of four-stroke diesel engine and industrial mass production were realized. The main intention was to reduce the manufacturing cost and therefore, the engine design should be as simple as possible. The use of swirl burner can fulfill the multi-fuel characteristic of the engine since it was dedicated for various heat sources including biomass, which is widely available and cheap. Based on theoretical calculations, the design parameters were found to be within the acceptable range. Detail analysis is required to determine possible causes of considerably high friction and low gas pressure inside the working cylinder. In reference to Beale formula, the net power output is a function of Beale number, mean pressure and engine speed. And Beale number is a function of heater temperature. Thus, heater temperature, mean pressure and engine speed are critical factors that influence the engine performance in terms of net power output.

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