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Design, Fabrication and Testing of a Swirl Burner for Alpha V-Shaped Stirling Engine

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Abstract: This study presents the design, fabrication and testing of a swirl burner that was used as part of the heater head section for a 194 cc. V-Shaped alpha stirling engine. The incorporation of a swirl burner with a stirling engine fulfilled its multi-fuel characteristic, since a hot producer gas from a gasification or combustion of any source of fuel can be utilized including biomass. The swirl burner with two heat input channels was designed based on the swirl number, S, which was calculated as 19.6. The swirl burner was made of 4 mm of mild steel and internally covered with 10 mm of cement. The flare from the mixture of air and Liquefied Petroleum Gas (LPG) was torched through the primary inlet and swirled uniformly through the area of an annulus in between the hot working cylinder and the outer swirl burner. The flame temperature inside the swirl burner was found to exceed 1000°C and produced the hot temperature up to 770°C. The required hot temperature inside the stainless steel expansion-working cylinder of 550 and 650°C was realized by the swirling effect of the flare inside the burner. Like a swirl combustor, the mathematical equation of the swirl number, S was found to be applicable to the swirl burner with the swirling effect only at the area of an annulus. The hot temperature increased with the increase of heater temperature inside the swirl burner and improved the expansion process.

Key words: Gasification, combustion, biomass, multi-fuel, regenerative, converted engine, power generation

INTRODUCTION

The Stirling engine is an externally heated, closed-cycle regenerative machine (Walker, 1980) with the salient advantages of high efficiency, low emission, low pollution, low noise, long life and fuel flexibility (Yuan, 1993; Scott *et al*., 2003). The shortage of fossil fuels nowadays has demanded other alternative fuels be explored and utilized for power generation and the Stirling engine is one of the technologies that can utilize these alternative fuels, since it can safely and silently run with any combustible fuel (Senft, 1993). Many applications can be expected in small companies and communities if alternative fuels such as biomass or agricultural waste are used as a source of fuel (Podesser, 1999).

In the development of multi-fueled stirling engines, the design consideration of an efficient fuel burning system is very important. The efficiency of the fuel burning system will be determined by the capability of the external heat source system to provide sufficient heat input and the capability of the engine heater head to store the heat supply for the working cylinder and to minimize heat loss. Many

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researchers had incorporated an electrical heater as part of the engine pre-heating or heating head section, particularly for the alpha v-shaped Stirling engine, since it is easier to assemble the electrical heater to the engine body as compared to other means of heating systems due to its sloped position. The electrical heater acts as a heat interface between the fuel burner and the engine hot working cylinder. Undeniably, it is good for continuous, stable and easy to regulate heating purposes but there are a few other alternatives that can also potentially be utilized especially in the development of a low cost, biomass-fueled stirling engine in rural areas. Due to the cost, the alpha-type stirling engine is typically selected because many parts from the industrial mass production can be used. The necessary maintenance and repair work of this engine can also be done by a standard car workshop (Podesser, 1999).

Karabulut *et al*. (1998) manufactured and tested the alpha v-type stirling engine at different set up values of pressure and hot source temperature. An electrical furnace was used to heat the engine and air as the working gas. The engine produced a maximum power output 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. Podesser (1999) developed a 3 kW alpha v-type stirling engine for electricity production in rural villages. A hot flue gas from a biomass furnace heated the engine with an electrical heater used as part of the engine pre-heat section. The engine generated the electrical power output of 3 kW at the thermal power output of 12.5 kW, flue gas temperature of 1000°C and engine cooler temperatures of 30 to 70°C. Air and nitrogen were used as the working gas. Batmaz and Üstün (2008) developed and tested a prototype alpha v-type stirling engine using double heaters at the hot working cylinder. The electrical heating system heated the engine at laboratory conditions. The engine produced the maximum power output of 118 W at the charge pressure of 1.0 bar and heater temperature of 9502°C. The range of heater temperature tested was within 650-1000°C and the pressure ranged from the ambient value to 2 bar. Helium was used as the working medium. Yusof *et al*. (2009) developed a converted four-stroke diesel on the alpha v-shaped stirling engine. 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. The theoretical power output was estimated at about 25 W. During the preliminary investigation, the engine produced friction torque ranges from 0.80 to 1.30 Nm at the engine speed range of 300 to 1200 rpm. With the effect of engine speed, hot temperature and air as the working gas, the maximum mean pressure obtained was approximately 1.41 bar.

The main objectives of the present study were to design, fabricate and test a swirl burner that was incorporated as part of alpha v-shaped stirling engine heater head section and to test experimentally its heating performance with the targeted heater temperature of 1000°C and hot temperature inside the working cylinder within a range of 550 to 650°C. The engine as described in this study is the same as in Yusof *et al*. (2009) with certain modifications on the hot working cylinder whereby a new water cooling jacket made of aluminum is incorporated into the bottom half of the cylinder.

GENERAL SYSTEM DESCRIPTION

The schematic view of the experimental engine is shown in Fig. 1. Basically, the engine consists of a crankcase with two cylinders positioned at 90° angles to each other, two working cylinders with a hot cylinder having an extension referred as an extension cylinder, two pistons with a hot piston having an extension referred as a piston crown, two connecting rods linked in between the pistons and a crank journal, a swirl burner for the heating of the surrounding of the extension cylinder, a water cooling jacket for the cooling of the hot cylinder block, a regenerator and a connecting tube. The water-cooling jacket is the new component added to the engine to smoothen both the piston and the piston ring motion inside the hot working cylinder specifically when the hot temperature inside the cylinder exceeds 600°C. Figure 2 shows a picture of the water-cooling jacket that was assembled to the hot cylinder block.

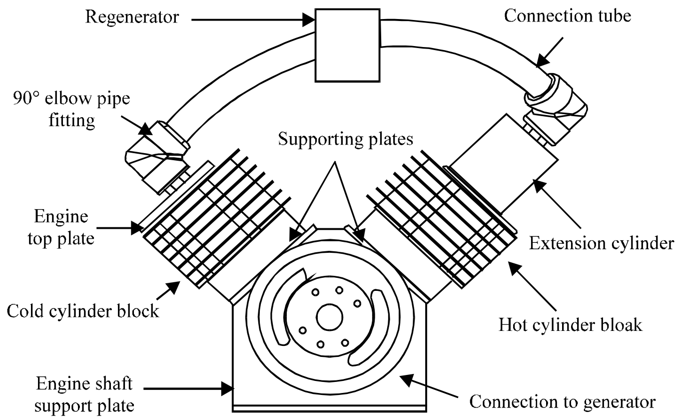

Fig. 1: A schematic view of the alpha v-shaped stirling engine

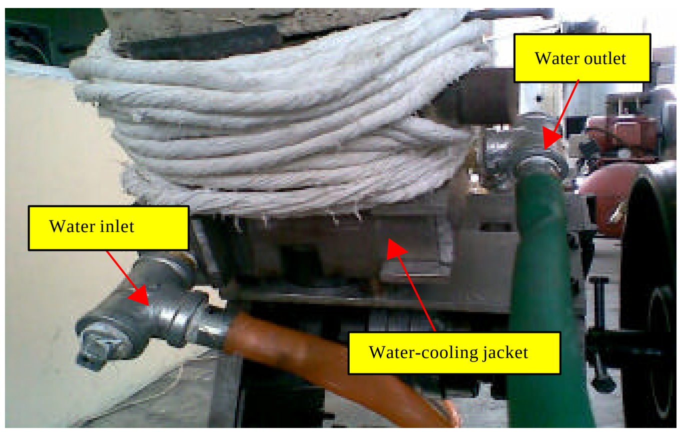

Fig. 2: Water-cooling jacket incorporated into the engine body

The design specifications of the engine are presented in Table 1. As reported in Yusof *et al*. (2009), all the technical features of the engine are within the acceptable range of the stirling engine design requirement. The design requirements of both the heater and hot temperature are set based on the type of material used for the construction of the expansion-working cylinder. The incorporation of the water-cooling jacket has no effect on the total dead volume of the engine.

DESIGN CONSIDERATIONS

Swirl Number

The swirl number S is normally used to characterize the swirling flow in a swirl burner (a similar kind of combustion device as the cyclone combustor). The main effects of swirl are to improve flame stability as a result of the formation of toroidal recirculation zones and to reduce combustion lengths by producing high rates of entrainment of the ambient fluid and fast mixing particularly near the boundaries of the recirculation zones. For a cyclone combustor or a swirl burner, the swirl number S is defined as:

Table 1: Technical specifications of the engine

Engine type	Alpha (v-type, 90° phase angle)
Cylinder bore	53 mm
Cylinder stroke	44 mm
Swept volume	194 cm^3
Total dead volume	115 cm^3
Expansion dead volume (hot)	13 cm^3
Compression dead volume (cold)	15 cm^3
Regenerator dead volume	87 cm^3
Compression ratio	1.8:1
Working gas	Air
Max. engine speed	1000 rpm
Heater temperature	650-750°C (max. target of 1000°C)
Hot temperature (inside cylinder)	550-650°C
Cold temperature (inside cylinder)	30-70°C
Regenerator	4 mm stainless steel ball bearings

$$S = \frac{2.G_\phi}{D_e.G_x} \tag{1}$$

where, G_ϕ is the axial flux of angular momentum and G_x is the axial flux of linear momentum.

A simpler variant is usually calculated based on the geometry of the cyclone or the swirl burner, since the swirl number calculation is not straightforward without detailed information on the flow. So, the geometric swirl number S_g refers to the isothermal operation as:

$$S_g = \frac{\pi.D_e.D}{4.a.b} \tag{2}$$

For a non-isothermal operation where the inlet and outlet gas temperatures are known, a non-isothermal geometric swirl number is used as follows:

$$S_{gT} = S_g \times \frac{T_{inlet}[K]}{T_{outlet}[K]} \tag{3}$$

The exit Reynolds number was defined as:

$$Re = U^*D_e/v$$

where, U is the average exit velocity based on the total mass flow rate, the exit area and the exit gas temperature. D_e is the exit diameter and v is the kinematics viscosity of the gases at the exit based on the exit temperature. The swirl number for a swirl burner or cyclone combustor was recommended to be typically at $8 < S < 20$ (Syred and Beér, 1974).

Flow Characteristics

The swirl number can be used to characterize and compare different swirls or cyclone combustor configurations. By using Eq. 2 and 3 and with the value of S between 8 and 20 as recommended, the basic dimensions of a swirl burner or a cyclone combustor can be found. If the inlets are circular with diameter D_i, Eq. 3 can be written as:

$$S_{gT} = \frac{D_e.D}{n.D_i^2} \times \frac{T_{inlet}[K]}{T_{outlet}[K]} \tag{4}$$

where, n is the number of inlets.

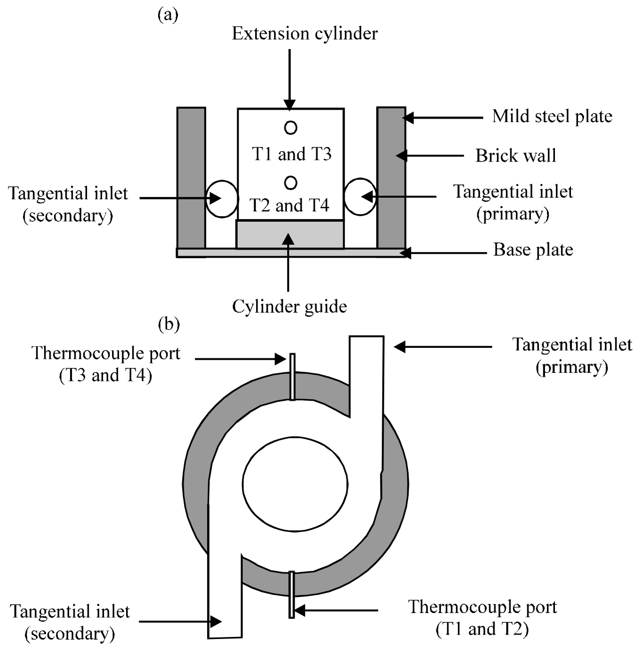

Fig. 3: Burner layout, (a) side view and (b) top view

In reference to Eq. 4 and taking D_e as the equivalent diameter of annulus with D = 102 mm, D_i = 22 mm, n = 2 (primary and secondary heat input channels), T_{inlet} = 1000°C and T_{outlet} = 300°C, the swirl number, S, obtained was 19.6. Theoretically, it was proven that the desired swirl burner with the area of an annulus that was derived from the difference in between the outer area of the swirl burner and the area of the hot working cylinder for alpha v-shaped stirling engine could be manufactured in compliance with the recommended swirl number. Experimentally, the swirl burner must be tested for its capability of supplying the heat input to the engine with the maximum heater temperature of 1000°C that was considered as the inlet temperature, T_{inlet} from Eq. 4.

Fabrication

A swirl burner with two tangential inlets of 22 mm was designed and fabricated to cater for various heat sources including natural gases and producer gas from biomass gasification and/or combustion. The layout of the burner covered the overall diameter of 130 mm and a height of 65 mm. The inner wall of the swirl burner was covered with 20 mm of cement and the outer wall of the burner was made of 8 mm mild steel. The cylinder guide was designed at the bottom of the swirl burner in order to fix the position of the swirl burner when it was inserted into the extension cylinder. Without the cylinder guide, the swirl burner might tilt due to the sloped position of the engine cylinder. Figure 3a and b show the schematic layout of the swirl burner for the engine heating head section.

MATERIALS AND METHODS

This study was conducted at the Biomass Energy Laboratory of Universiti Sains Malaysia, Penang in March, 2009, where the manufactured swirl burner was assembled to the engine heater head section, well-insulated with an asbestos layer insulation rope and tested for its heating capability. The

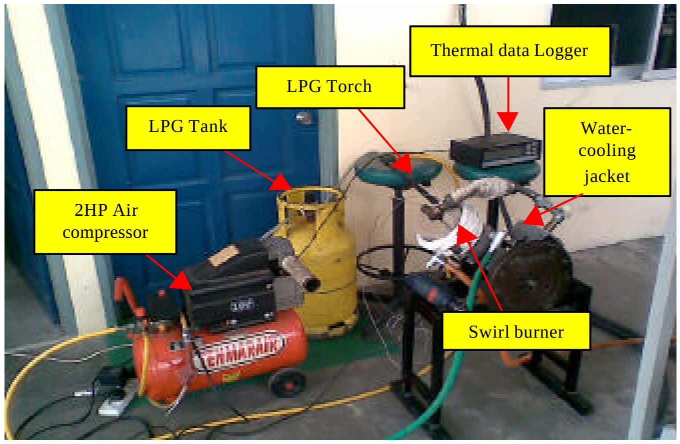

Fig. 4: Test set-up of the swirl burner heating capability test

heating capability of the swirl burner was influenced by the swirling effect of the flare from the fuel source. Figure 4 shows the test set-up of the engine with the incorporation of the swirl burner and a water-cooling jacket onto the hot working cylinder. The hot part of the regenerator tube was insulated with an asbestos layer insulation rope as well to prevent heat loss. The temperatures of the engine critical parts including the interior section of the swirl burner were measured by K-typed thermocouples and the temperature profile was logged and recorded using a Thermal Data Logger ScanLink2.0. The heat source to heat up the engine heater head section was Liquefied Petroleum Gas (LPG) mixed with compressed air supply from a 2 hp air compressor. A mixture of LPG from the tank and compressed air from a 2 hp compressor was ignited at the mouth tip of the LPG torch using a cookware lighter. A blue flame was obtained by adjusting the amount of both LPG and air inputs. The LPG torch with a blue flame was then inserted into the primary inlet of the swirl burner. This was done to make sure that the flame swirled nicely by adjusting the positioning of the LPG torch. The top of the swirl burner was then covered to prevent unnecessary heat loss during the test. The secondary inlet of the swirl burner was blocked. The intention was to justify if a single heat input into the swirl burner was sufficient in providing the hot temperature range of 550-650°C inside the working cylinder. Using the data logger, the temperature profiles of all the engine critical sections consisting of hot and cold working cylinders, internal section of the swirl burner, water inlet and outlet were monitored.

RESULTS AND DISCUSSION

Temperature Profile of the Engine

The temperature profile of the engine is essential to determine the actual temperature distribution at each of the engine critical section with respect to time. Figure 5 indicates that the temperature inside the swirl burner (for both facing and opposite the LPG torch) exceeded 1000°C. The maximum temperature for the location that was facing the LPG torch was recorded as 1100°C. For the location that was opposite the LPG torch, the maximum temperature achieved was approximately 1070°C. This showed that the swirling flow of the flare inside the swirl burner was stable and consistent surrounding the engine hot cylinder. As shown in Fig. 5, the hot temperature inside the expansion cylinder was up to 770°C.

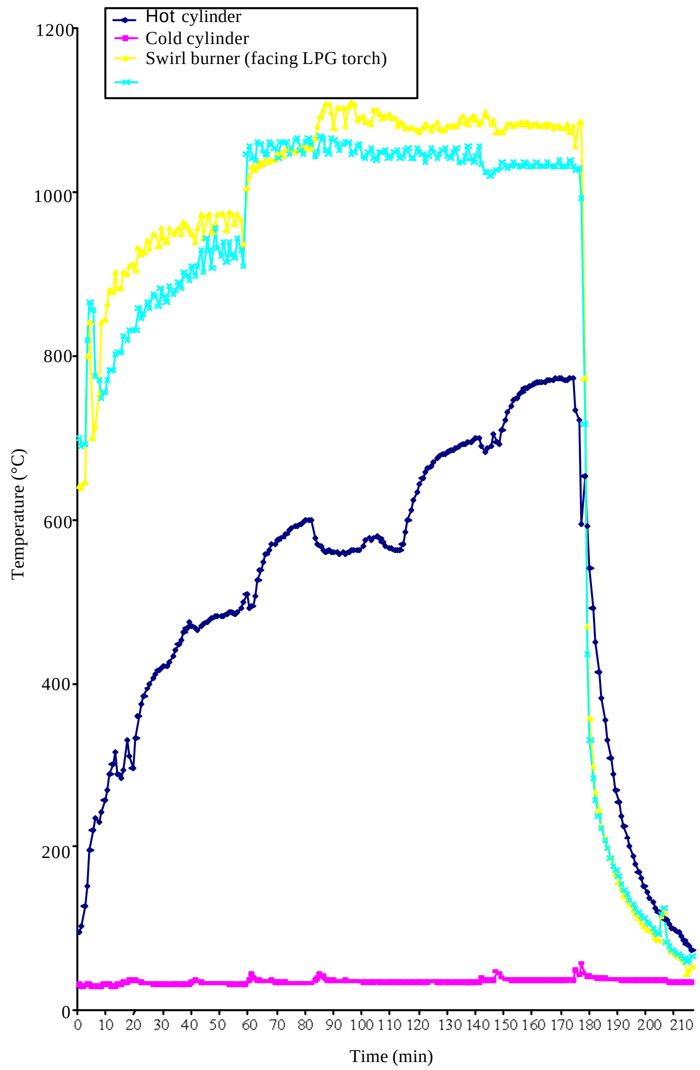

Fig. 5: Temperature profiles of Alpha engine

Temperature Profiles of the Water-cooling Jacket

The temperature profiles of the water inlet and outlet from the water-cooling jacket were also taken during the test. From Fig. 6, the water outlet temperature was found to fluctuate slightly between 30 and 31°C and the water inlet temperature also fluctuated slightly between 29 and 30°C along the 120 min heat load test.

Engine Performance

A preliminary test of the engine performance is as shown in Fig. 7 and 8. It was found that the engine started to rotate c ontinuously at the hot temperature of approximately 300°C and air pressure of 1.5 bar. Figure 7 showed that the engine speed increased from 50 to 120 rpm when the hot

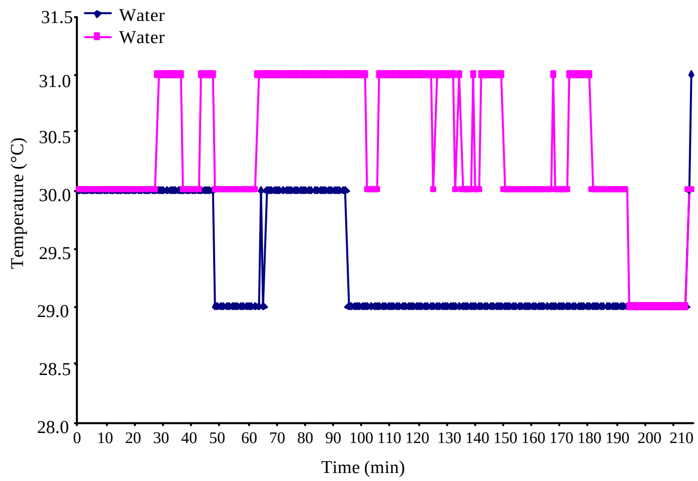

Fig. 6: Temperature profiles of the water-cooling jacket

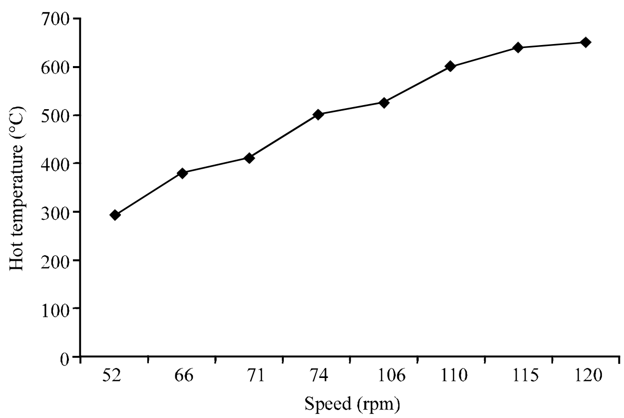

Fig. 7: Alpha 194cc. engine speed performance with respect to hot temperature (no load)

temperature increased from 300 to 650°C. The maximum speed of 120 rpm was achievable at the hot temperature of 650°C. From Fig. 5, it shows that the hot temperature of 650°C was met at the heater temperature of approximately 1040 to 1080°C.

The engine net power output was approximated using Beale formula:

$$P_{NET} = B_N \text{ x } P_{mean} \text{ x } V_{swept} \text{ x } f$$

where, P_{NET} is the engine net power output, B_N is Beale Number, P_{mean} is the mean pressure and f is the engine rotational speed. The Beale Number, B_N was taken from Beale Chart as shown in Fig. 9. The Beale Number of 0.015 was approximately correct for all types and sizes of stirling engines (Walker, 1980). And from Fig. 9, it was shown that for normal machining engine, the heater temperature required was approximately 900 K or 630°C at the Beale Number of 0.015.

Based on the experimental results, the required heater temperature of 900 K or 630°C was realized by the swirling effect of the flare from the mixture of air and LPG inside the swirl burner. The flame temperature of air and LPG mixture inside the annulus of the swirl burner was found to exceed 1000°C

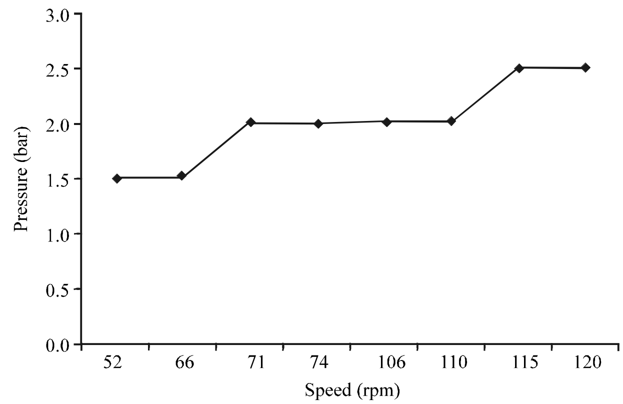

Fig. 8: Alpha 194cc. engine speed performance with respect to air pressure (no load)

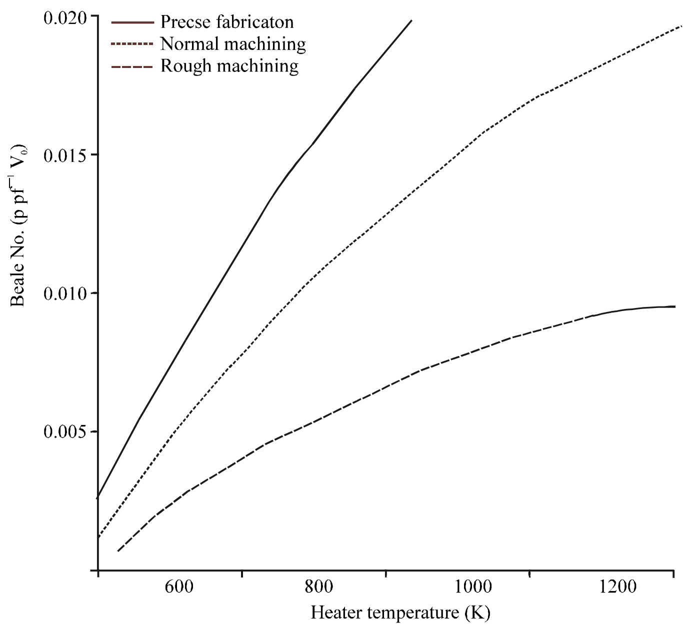

Fig. 9: Beale chart showing the relationship of beale number and heater temperature

at the burner inlet and its opposite location (distance of approximately 51 mm from the burner inlet). This was in good agreement with the experimental study by Solero *et al*. (2000) on a swirl burner with a single annulus area where the temperature profile at a distance of 45 mm from the burner inlet indicated two temperature peaks of approximately 1100°C.

The increase of actual heater temperature above 1000°C would possibly increase the Beale Number from 0.015 to 0.020 and eventually, increase the engine net power output. The hot temperature of 770°C inside the stainless steel expansion-working cylinder was also found to be higher than the design requirement, which was within 550 to 650°C as reported by Raggi *et al*. (1997). Ideally, the increase of hot temperature inside the expansion-working cylinder would significantly impact the Stirling cycle processes, thermal efficiency and net power output. This study proved that the mathematical equation of the swirl number, S was applicable to the swirl burner with the swirling effect only at the area of an annulus in between the hot cylinder and the outer swirl burner. The recommended values of swirl number for the swirl burners in between 8 and 20 by Syred and Beer (1974) were valid for this study. The maximum cold temperature inside the compression-working cylinder was recorded as 58°C, which mostly fluctuated within 30-45°C during the test. Podesser (1999) controlled the cold temperature of the compression-working cylinder within the range of 30-70°C for the heater temperature of 1000°C. Thus, the range of operating cold temperature for the engine was acceptable.

The incorporation of the water-cooling jacket was found to be efficient since both piston and piston ring motion at the expansion-working cylinder were smooth at the heater temperatures of 1040 and 1080°C inside the swirl burner. As compared to the previous test of the engine without the water-cooling jacket, both piston and piston ring were hardly moved at the heater temperature of 600°C and above. It was suspected that the water-cooling at the bottom section of the hot working cylinder prevented a quick dry-up of the lubrication oil inside the cylinder and also prevented an over-expansion of both piston and piston ring materials during the heating of the engine heater head.

The engine performance was mainly affected by the air leakage problem throughout the working cylinder during the experiment. It was found that an additional of 1 bar air supply was needed to sustain the engine rotary motion, which was from 1.5 to 2.5 bar as shown in Fig. 8. Based on physical inspection, certain amount of air leaked out from the bottom of the working cylinder. It was suspected that the air passed through the piston sealer or ring during the test. Therefore, further improvement actions were needed to overcome the sealing problem. Among them is the introduction of Teflon piston with a closer gap clearance in between the piston and the cylinder wall from 0.3 to 0.1 mm.

CONCLUSION

The swirl number S is essential to characterize the swirling flow in a swirl burner. A good swirling flow will contribute to flame stability and a uniformly surrounded heating zone inside the burner. The design requirements of a heater temperature range of 650-750°C and a hot temperature range of 550-650°C inside the expansion-working cylinder of an alpha v-shaped engine were realized throughout the use of the swirl burner with the swirl number of 19.6. The targeted heater temperature of 1000°C inside the swirl burner was achieved by a single channel heat input (primary inlet). The preliminary test of the engine performance showed that the engine started to rotate continuously at the speed of 50 rpm at the hot temperature of 300°C and air pressure of 1.5 bar. The engine speed increased up to 120 rpm with the increase of hot temperature from 300 to 650°C. Further improvement action of the engine sealing is needed to overcome the air leakage problem and to improve the engine performance. The use of a swirl burner enhanced the potential of using other sources of fuel for the alpha stirling engines such as biomass, agricultural wastes or domestic wastes that are widely and freely available. A hot producer gas from the gasification or combustion of these wastes can be the input into the swirl burner as a replacement for LPG. Hence, this type of Stirling engine will become practically multi-fueled and cost effective.

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NOMENCLATURES

D = Outer diameter
D_e = Exit diameter (mm)
D_i = Inlet diameter (mm)
G_ϕ = Axial flux of angular momentum
G_x = Axial flux of linear momentum.
n = Number of inlet
Re = Reynolds number
S = Swirl number
S_g = Geometric swirl number for isothermal operation
S_{gT} = Geometric swirl number for non-isothermal operation
T_{inlet} = Inlet temperature (K)
T_{outle} = Outlet temperature (K)
U = Average exit velocity (m sec^{-1})
v = Kinematics velocity of gas (m sec^{-1})

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