Fabrication and Test of Gamma-Type Stirling Engine

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***Abstract*— In this investigation, a gamma-configuration Stirling engine is designed and constructed. The single and twin power cylinder engines are tested with air at atmospheric pressure by using an electric heater as a heat source. The engine is tested with heater input of 156.3W, 187.6W, and 223.2W. Variations of engine torque, shaft power and brake thermal efficiency with engine speed and engine performance at various heat inputs are presented. The results indicate that at the maximum heater input of 223.2 W, the heater temperature for single power cylinder and twin power cylinder are 612℃ and 574℃. The two engines produce a maximum torque of 0.13 Nm at 405 rpm and 0.15 Nm at 412 rpm; a maximum shaft power of 5.73 W at 456 rpm and 6.47 W at 412 rpm; a maximum brake thermal efficiency of 2.57 % at 456 rpm and 2.9 % at 412 rpm, respectively.**

Keywords - Stirling engine, Hot-air engine, Regeneration, Shaft power, Torque, Brake thermal efficiency

I. INTRODUCTION

The Stirling engine is a simple type of external-combustion engine that can operate with different heat sources. The Stirling engine was the first invented regenerative cycle heat engine Robert Stirling patented the Stirling engine in 1816 [1]. Stirling engines were simple and safe to operate, ran almost silently on different heat sources [2]. The era of the Stirling engine was terminated by the rapid development of the internal-combustion engine and the electric motor.

In 1937 [2], the Stirling engine was brought to a high state of technological development by the Philips Research Laboratory in Eindhoven, Holland, and has progressed continuously. Initial work focused on the development of small thermal-power electric generators for radios and similar equipment used in remote areas [2,3].

New materials were one of the keys to the Stirling engine success, such as stainless steel. Another key to success was having a better knowledge of thermal and fluid physics.

Kolin [1,2] experimented with a number of low temperature differential (LTD) Stirling engines. In 1983, he presented a model that worked on a temperature difference between the hot and cold ends of the displacer cylinder as low as 15℃.

Senft's [4] research in LTD Stirling engines which had an ultra-low temperature difference of 0.5℃.

In 1997, Iwamoto et al. [5] compared the performance of a LTD Stirling engine with a high-temperature differential Stirling engine. They concluded that the LTD Stirling engine efficiency at its rated speed was approximately 50 % of Carnot efficiency.

In 2003, Kongtragool and Wongwises [6] made a theoretical investigation on the Beale number for LTD Stirling engines. The Beale number data for various engine specifications were collected from literatures.

In 2005, Kongtragool and Wongwises [7] investigated, theoretically, the power output of the gamma-configuration LTD Stirling engine. They pointed out that the mean-pressure power formula was the most appropriate for LTD Stirling engine power output estimation.

In 2007, Kongtragool and Wongwises [8] reported the performance of two LTD Stirling engines tested using LPG gas burners as heat sources. The first engine was a twin-power-piston engine and the second one was a four-power-piston engine. They presented the engine performances, thermal performances, including the Beale's numbers.

Recently, Kongtragool and Wongwises [9] presented the performance of a twin-power-piston Stirling engine powered by a solar simulator. The heat source was a solar simulator made from a 1000 W halogen lamp.

Because little research in this area, therefore, a gamma-configuration Stirling engine is fabricated and tested in this study. The single and twin power cylinder engines are tested with air at atmospheric pressure by using an electric heater as a heat source. Variations of engine torque, shaft power and brake thermal efficiency with engine speed and engine performance at various heat inputs are presented.

II. EXPERIMENTAL APPARATUS

Details of the experimental apparatus are as follows:

A. Test engine

The schematic diagram of the test engine is show in Fig. 1. Main engine design parameters are listed in Table 1. The engine of gamma-configuration is designed with a twin power piston. The power pistons are removable. Since the gamma configuration provides a large regenerator heat transfer area [5] and is easy to be constructed, this configuration is used in the present study. This configuration shows good self-pressurization [10]. The engine cylinder should be designed in vertical type rather than horizontal in order to reduce bushing friction [11].Two power cylinder are directly connected to the cooler plate to minimize the cold-space and transfer-port dead volume.

The crankshaft is made from steel since steel's strength is better. The crankshaft is supported by two ball bearings. It is connected by ball bearings between crankshaft and the connecting rod to reduce friction.

The power cylinders are made from bronze. The power pistons are made from aluminum since its weight is less than steel and strength enough. The clearance between piston and cylinder is 0.02 mm, approximately.

The displacer cylinder and displacer cylinder head are made from stainless steel. It can maintain bigger temperature gradient because stainless steel's thermal conductivity is low. Its thick is 0.5 mm. The displacer is made from aluminum since its weight is less. The displacer swept volume is 31.53 cm^3 with 3.56 cm bore and 3.18 cm stroke. The single power cylinder's swept volume ratio of this engine is 1.55. The twin power cylinders' swept volume ratio of this engine is 0.77.

The displacer rod is made from steel. The displacer rod is guided by aluminum bushings. The power piston connecting rods are made from aluminum. Both ends of the connecting rod are attached to two ball bearing housing.

B. Testing facilities

The schematic diagram of the experimental apparatus is shown in Fig. 2. The photograph of engine with testing facilities is show in Fig. 3. Details of the testing facilities are as follows:

1. Heating system: Using Extech-electronics' 6910 power supply and a heating ring.
2. Cooling water system: Using a constant temperature water trough and a flow meter that control temperature and flow.
3. Temperature data logger: Water cooling inlet and outlet are measured by two T-type thermocouples. The heater wall temperature is measured by a K-type thermocouple. The accuracy of the temperature measurement is ± 0.1℃.
4. AC server motor: It contains torque sensor with rotation rate sensor. It can load different load by control panel.
5. Pressure sensor: Using PHYWE technology's pressure sensor connect cooler plate's compression space. Measuring range from 20~250 kPa (200 to 2500 mbar). Working temperature is from -40~125℃. There is max error that is 1.5% when working temperature is from 0~85℃.
6. Computer: Connecting pressure sensor and temperature data logger. And record the data.

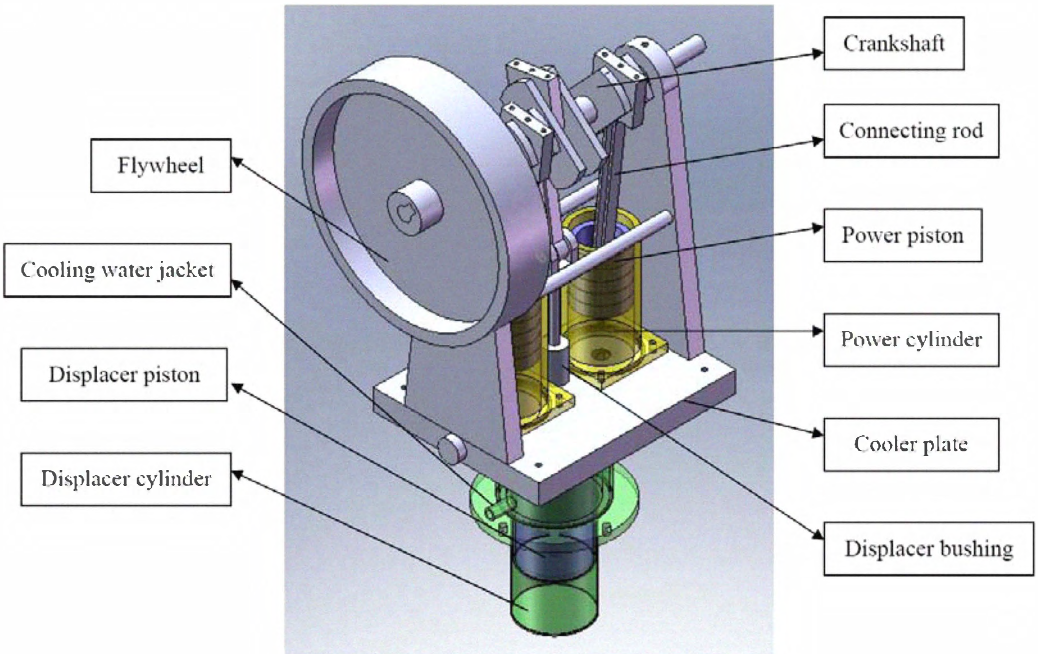

Figure 1. Schematic diagram of the Stirling engine.

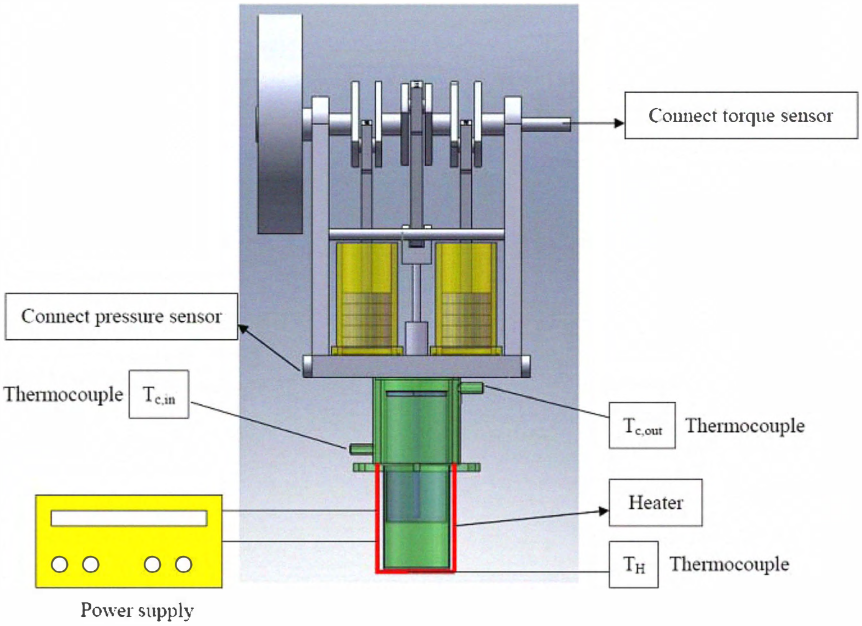

Figure 2. Schematic diagram of the performance test for the twin power piston engine.

TABLE I. MAIN ENGINE DESIGN PARAMETERS.

		Single power cylinder	Twin power cylinder
Power piston	Bore	2.86 cm	2.86 cm
	Stroke	3.18 cm	3.18 cm
	Swept volume	20.36 cm^3	40.72 cm^3
Displacer piston	Bore	3.56 cm	3.56 cm
	Stroke	3.18 cm	3.18 cm
	Swept volume	31.53 cm^3	31.53 cm^3
Swept volume ratio		1.55	0.77
Compression ratio		1.4	1.8
Phase angle		90°	90°

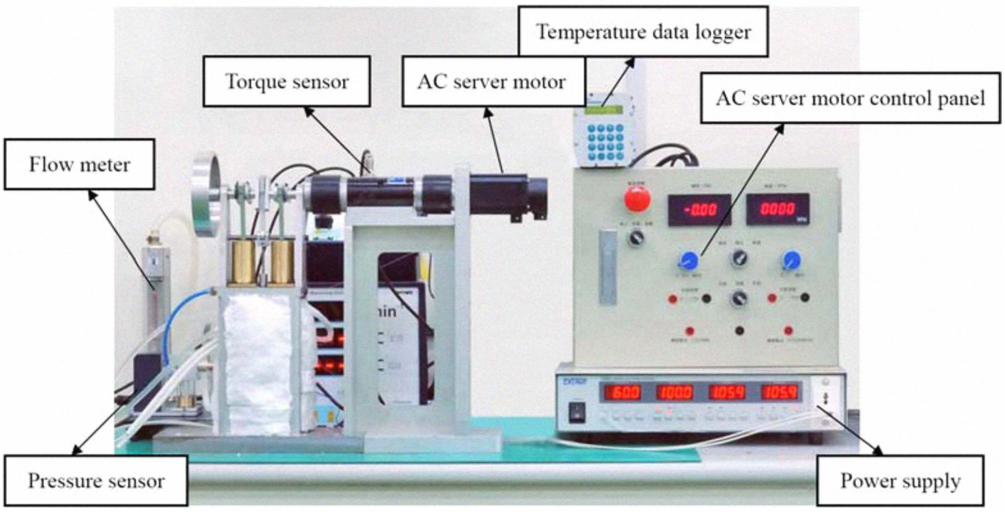

Figure 3. Engine with testing facilities.

III. Experimental Procedures

A. Heat source test

The actual heat input can not be determined directly while the engine is running because of the difficulties caused by instrumentation. This experiment has been carried out before the performance test, to determine the actual heat input to the engine.

The displacer cylinder insulated with insulator. One litre per a minute of water is put into the displacer cylinder. This water is used to absorb heat from the electric heating. The absorbed heat is the useful heat input to the displacer cylinder.

The thermocouples for measuring the displacer head wall temperature and the water temperature are installed. Two T-type thermocouples are used to measure the temperature of water cooling inlet and outlet. One K-type thermocouple is used to measure the displacer cylinder head wall temperature.

The actual heat input is calculated from:

$$q_{in} = \dot{m} \times C_P \times \Delta T \qquad (1)$$

where q_{in} is the actual heat input to the engine, $\dot{m}$ is the mass of water to absorb heat, C_P is the specific heat of water at constant pressure, ΔT is the temperature of water cooling outlet subtract inlet.

The power supply inputs 200 W, 250 W, and 300 W, respectively. The actual heat inputs are 156.3 W, 187.6 W, and 223.2 W. The heat source efficiencies are 78 %, 75 %, and 74 %. It seems the more heat inputs, the heat source efficiencies are low.

B. Performance test

All thermocouples are connected to the data logger and computer and calibrated before the engine is started. And calibrate the pressure sensor. The cooling water system is connected to the engine cooling pan. The cooling water flow rate is adjusted at a liter per a minute. The power supply turned on. The displacer cylinder head is heated up until it reaches the operating temperature. The engine is started and run until the steady condition is reached.

The engine is loaded by AC serve motor control panel. The engine speed reading, torque, and all temperature from the thermocouples are collected. Another loading is added until the engine is stopped.

Testing can be repeated with another heat input by changing the power supply input. In this study, engine tests are performed by using three heat inputs. They are 156.3 W, 187.6 W, and 223.2 W, respectively.

VI. Experimental Results and Discussion

The shaft power is calculated from:

$$P = 2 \times \pi \times T \times N \qquad (2)$$

where P is the shaft power, T is the torque, N is the engine speed.

E_{BT} is the brake thermal efficiency calculate from:

$$E_{BT} = P/q_{in} \qquad (3)$$

where q_{in} is the actual heat input to the engine.

N_B is the Beale number calculate from:

$$N_B = P/(P_m V_P f) \qquad (4)$$

where P_m is engine mean pressure in bar, V_P is the power piston swept volume (cc), and f is the engine frequency (Hz). For a non-pressurized engine, Pm = 1 bar is used in the calculation, as described by Senft [2].

A. The performance of single power cylinder

In the engine test, as the load is gradually applied to the engine, its speed is gradually reduced, until eventually it stops.

Figs. 4-6 show the variations of engine torque, shaft power and brake thermal efficiency with engine speed at various heat inputs. An increase of the engine torque, shaft power is show to depend on the heater temperature. The brake thermal efficiency of the engine decreases with increasing the heat inputs. Because the increasing the heat inputs, the increasing heat loss.

The variations in engine heater temperature at various actual heat inputs are shown in Fig. 7. The heater temperature of the engine increases with increasing actual heat inputs. The heater temperature of the engine decreases with increasing engine speed.

Fig. 8, the maximum shaft power and Beale number at various heat inputs are plotted against the heater temperature. The shaft power and Beale number increase with increasing heater temperature.

The shaft power increases with increasing engine speed until the maximum shaft power is reached and then decreases with increasing engine speed. The decreasing in shaft power after the maximum point, results from the friction that increases with increasing speed together with inadequate heat transfers at higher speed.

The brake thermal efficiency increases with increasing the torque until the maximum brake thermal efficiency is reached and then decreases with increasing the torque. The decreasing in brake thermal efficiency after the maximum point, results from the shaft power decrease with decreasing engine speed.

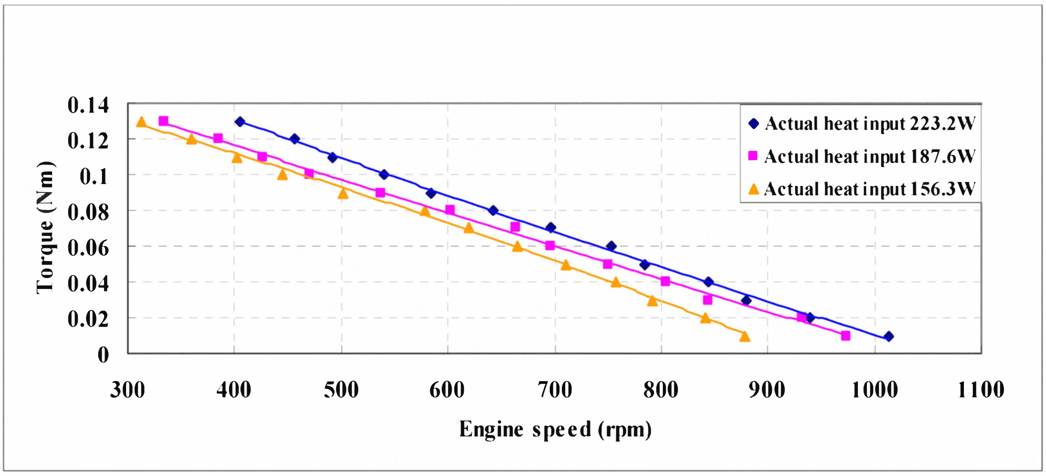

Figure4. Variations of engine torque at various actual heat inputs for single power.

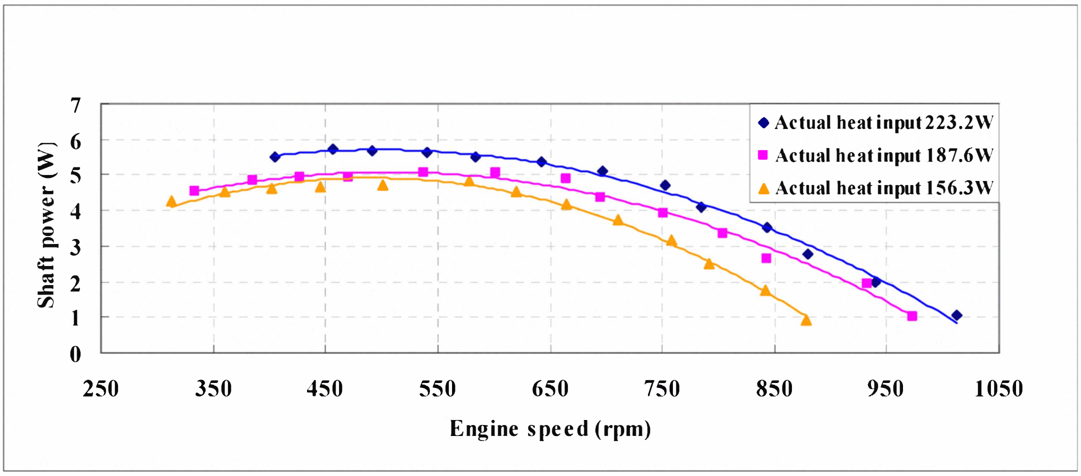

Figure 5. Variations of engine shaft power at various actual heat inputs for single power.

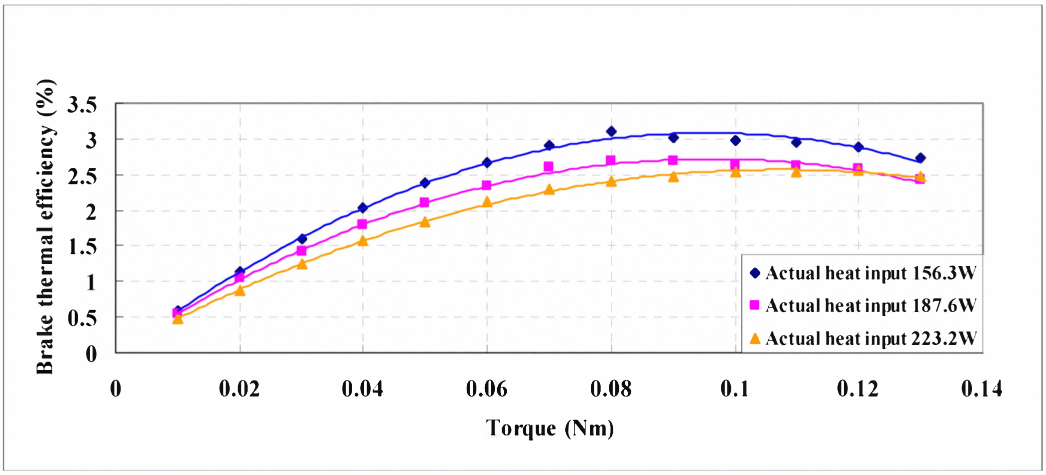

Figure 6 Variations of brake thermal efficiency at various actual heat inputs for single power cylinder.

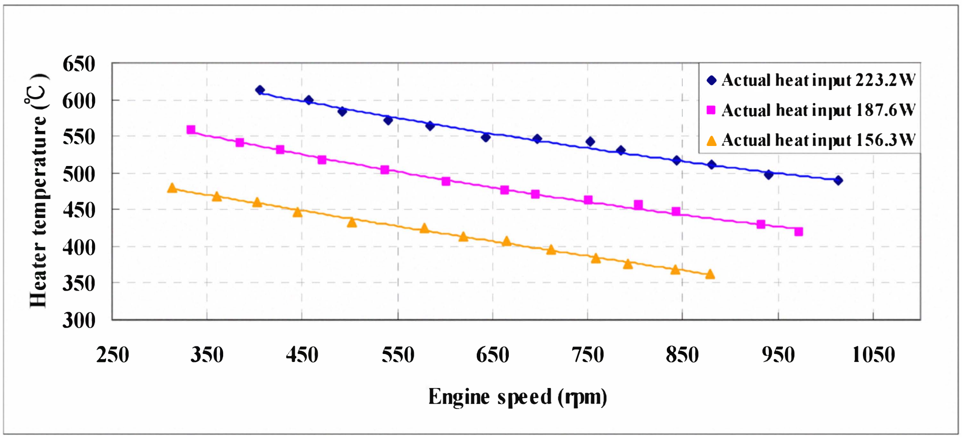

Figure 7. Variations of engine heater temperature at various actual heat inputs for single power.

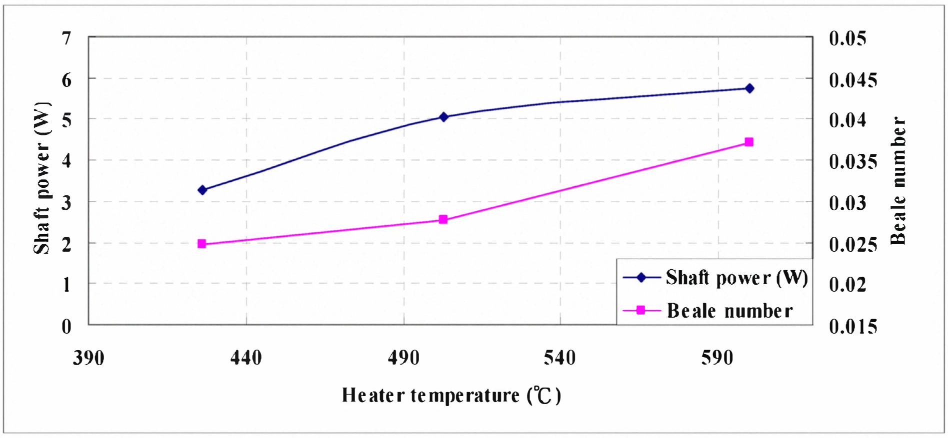

Figure 8. Variations of engine maximum shaft power and Beale number with heater temperature for single power.

B. The performance of twin power cylinders

Figure. 9-11 show the variations of engine torque, shaft power and brake thermal efficiency with engine speed at various heat inputs. An increase of the engine torque, shaft power is show to depend on the heater temperature. The brake thermal efficiency of the engine decreases with increasing the heat inputs. Because the increasing the heat inputs, the increasing heat loss.

The variations in engine heater temperature at various actual heat inputs are shown in Fig. 12. The heater temperature of the engine increases with increasing actual heat inputs. The heater temperature of the engine decreases with increasing engine speed.

Fig. 13, the maximum shaft power and Beale number at various heat inputs are plotted against the heater temperature. The shaft power and Beale number increase with increasing heater temperature.

The shaft power increases with increasing engine speed until the maximum shaft power is reached and then decreases with increasing engine speed. The decreasing in shaft power after the maximum point, results from the friction that increases with increasing speed together with inadequate heat transfers at higher speed.

The brake thermal efficiency increases with increasing the torque until the maximum brake thermal efficiency is reached and then decreases with increasing the torque. The decreasing in brake thermal efficiency after the maximum point, results from the shaft power decrease with decreasing engine speed.

TABLE II. COMPARE WITH SINGLE POWER CYLINDER AND TWIN POWER CYLINDERS.

q_{in}(W)		T_H(℃)	T_{max}(Nm)	P_{max}(W)	E_{BTmax}(%)	N_B(W/bar×cm^3×Hz)
156.3	single	426	0.13	4.84	3.10	0.025
	twin	424	0.10	4.76	3.05	0.015
187.6	single	503	0.13	5.06	2.70	0.028
	twin	491	0.12	5.64	3.00	0.019
223.2	single	600	0.13	5.73	2.57	0.037
	twin	574	0.15	6.47	2.90	0.023

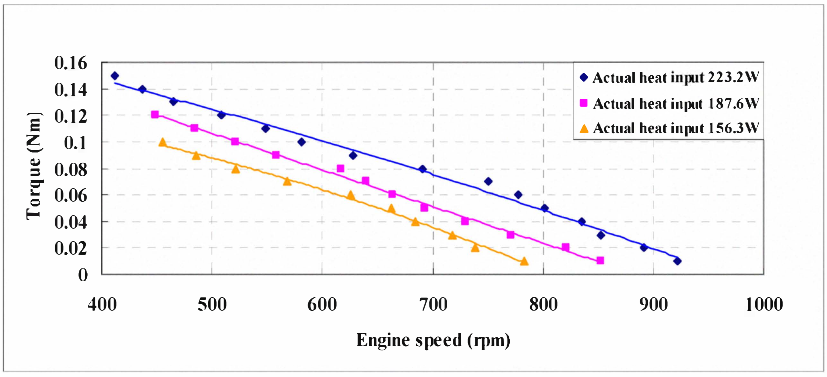

Figure 9. Variations of engine torque at various actual heat inputs for twin power cylinders.

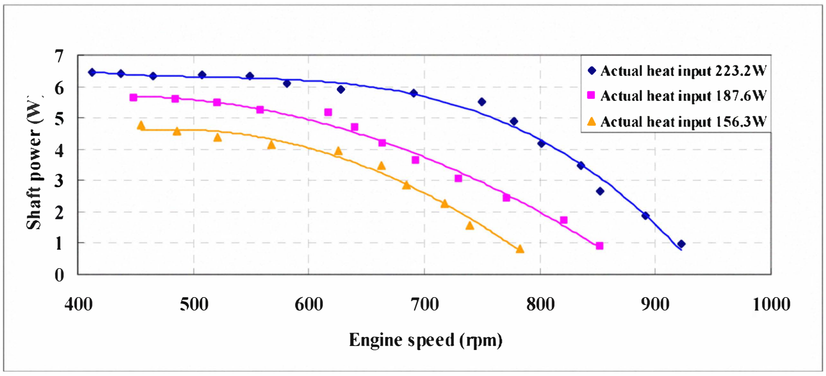

Figure 10. Variations of engine shaft power at various actual heat inputs for twin power cylinders.

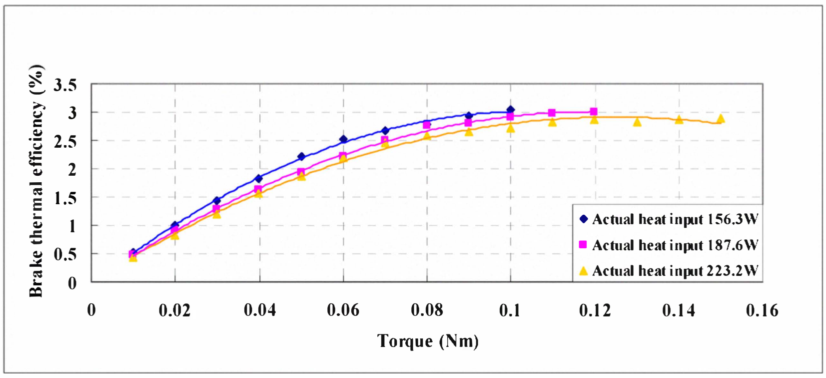

Figure 11. Variations of brake thermal efficiency at various actual heat inputs for twin power cylinders.

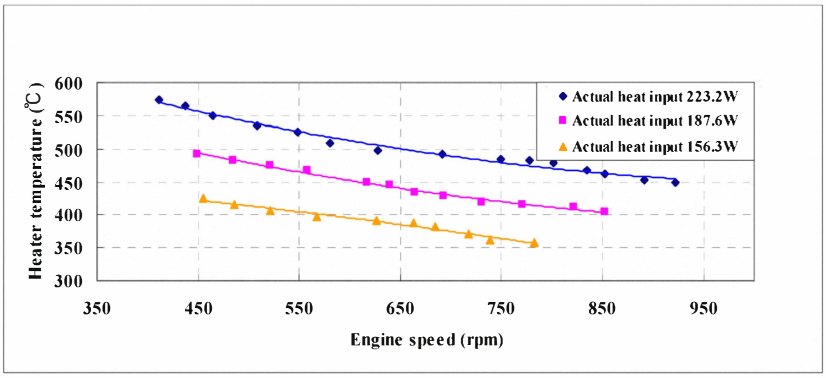

Figure 12. Variations of engine heater temperature at various actual heat inputs for twin power cylinders.

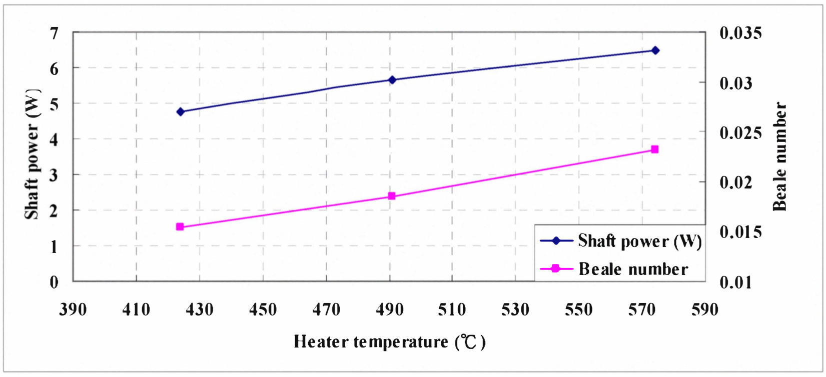

Figure 13. Variations of engine maximum shaft power and Beale number with heater temperature for twin power cylinders.

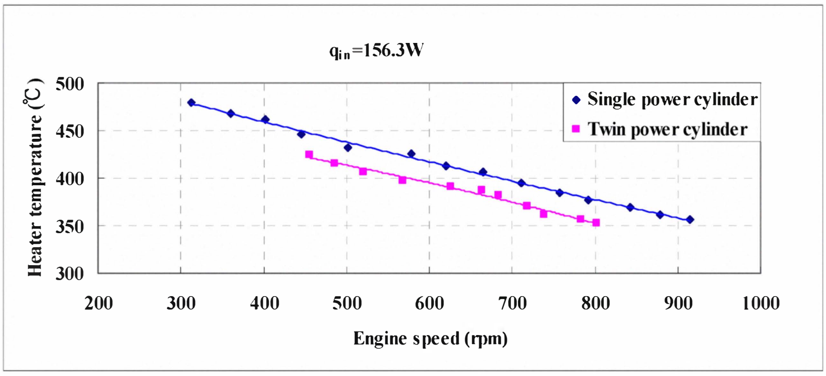

Figure 14. Variations of engine heater temperature at various cylinders.

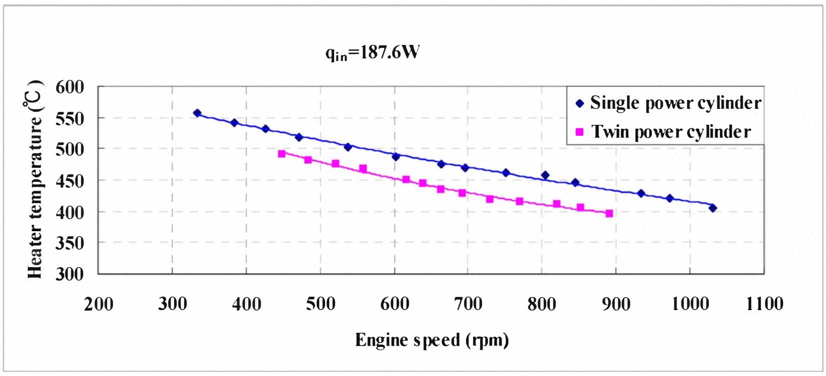

Figure 15. Variations of engine heater temperature at various cylinders.

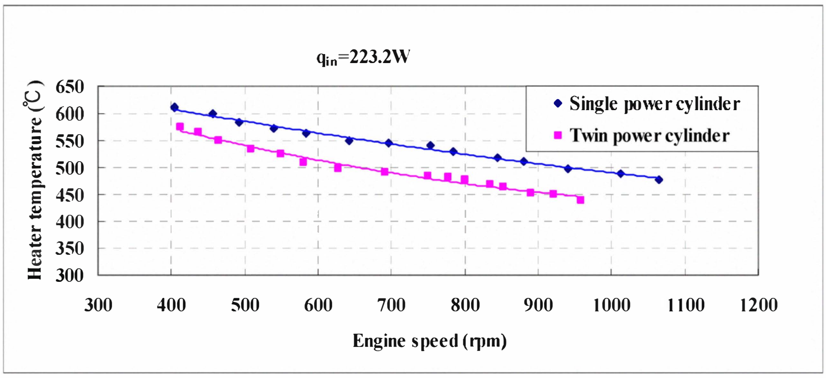

Figure 16. Variations in engine heater temperature at various cylinders.

C. Compare performance with single power cylinder and twin power cylinders

From Table 2, at actual heat input is 156.3 W, the performance of the single power cylinder is better than twin. It results from the single power cylinder's friction is less than twin. At other actual heat inputs, the performance of twin power cylinder is better than single. It results from the other heat inputs enough to overcome the friction's influence.

The variations in engine heater temperature at various cylinders are shown in Figs. 14-16. It seems the heater temperature of twin power cylinders is lower than single power cylinder.

V. CONCLUSION

A gamma-configuration Stirling engine is fabricated and tested. Results from this study indicate that the engine torque, shaft power, brake thermal efficiency, speed, and heater wall temperature increase with increasing the heat inputs. In fact, it can be said that the maximum engine torque, shaft power and the Beale number of the engine increase with increasing heater wall temperature.

The results indicate that at the maximum heater input of 223.2 W, the heater temperature for single power cylinder and twin power cylinder are 612℃ and 574℃. The two engines produce a maximum torque of 0.13 Nm at 405 rpm and 0.15 Nm at 412 rpm; a maximum shaft power of 5.73 W at 456 rpm and 6.47 W at 412 rpm; a maximum brake thermal efficiency of 2.57 % at 456 rpm and 2.9 % at 412 rpm, respectively.

Further research is undergoing to improve engine performance by adjusting the precision of the engine parts and reducing the friction from the flywheel.

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