Designing of Experimental Stirling Engine and its Performance Test*

By Keiichi NISHIMOTO** and Hiroaki KITAMOTO***

With a target of developing a Stirling engine to be driven by waste heat of refuse incinerators etc. with the working gas such as air or nitrogen, a displacer type engine was made on experimental basis, and its performance was tested. Air and nitrogen are cheaper and easier to handle than helium or hydrogen which is generally used as the working gas, but the flow resistance of the gas of such a kind is fairly higher than that of helium. In order to reduce the flow resistance, special perforated copper heat-transmitters were adopted as heater elements and regenerator matrices. Conveniently, the welding job required for making the heater of this type is comparatively little.

The test showed almost satisfactory performance of the engine. However, it became clear that in order to raise the mean working pressure still more, the gas-seal device for the power piston should be much improved.

Key Words : Stirling Engine, External Combustion Engine, Heat Engine, Thermodynamics, Thermal Engineering Equipment

1. Introduction

As the regenerator matrix for Stirling engines, packs of wire gauze have been widely used in general. The wire gauze matrix gives large heat transfer area per unit volume, but its flow resistance and void ratio are fairly large, and when the number of gauze layers is comparatively large or clogging occurs due to lubricant leaking into the regenerator, the performance of the engine falls remarkably.

On the other hand, the regenerator should be designed long enough to reduce the heat conduction loss from the hot end to the cold end. These circumstances make the designing of the regenerator highly difficult.

In this study, a Stirling engine using as the working gas air or nitrogen which is cheap and easy to handle, and driven by the waste heat from the refuse incinerator etc. was assumed, and a displacer type engine was made on experimental basis using a regenerator matrix and a heater element which caused less flow resistance and were comparatively easy to make, and its performance was investigated.

2. Engine made on Experimental Basis

The principal specifications of the experimental engine are shown in Table 1, and its schematic construction is illustrated in Fig.1.

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** Assistant Professor, Faculty of Technology, Kanazawa University (2-40-20 Kodatsuno, Kanazawa, Japan).

*** Graduate Student, Faculty of Technology, Kanazawa University (2-40-20 Kodatsuno, Kanazawa, Japan).

Table 1 Principal specifications of the test engine

Type of Engine	Displacer Type
Displacer Bore-Stroke	119 mm × 70 mm
Power Piston Bore-Stroke	75 mm × 60 mm
Working Gas	Air
Mean Working Pressure	100 kPa ~ 500 kPa
Cylinder Arrangement	Tandem Cylinder
Piston-ring	PTFE Ring
Gas-seal of Shaft-end	Mechanical Seal
Fuel	Town Gas

① Expansion Space	⑦ Power Piston
② Heater	⑧ Connecting Rod
③ Regenerator	⑨ Crankshaft
④ Cooler	⑩ Flexible Joint
⑤ Compression Space	⑪ Mechanical Seal
⑥ Displacer	⑫ Gas Burner

Fig.1 Configuration of the test engine

Generally, the heater employs a number of tubes of small diameter made of stainless steel or heat resisting alloy laid in parallel. But in this engine, 12 heater cylinders of stainless steel which hold copper blocks having the same cross section as that of the heater element B as illustrated in Fig.3 are fitted around the displacer cylinder, in order to secure the necessary area of heat transfer with as little welding job as possible. As shown in Fig.1, lower-half of the displacer cylinder and the stainless steel tubes connecting the heater cylinders and the displacer cylinder are also heated by a gas burner. The cooler consists of 8 copper tubes, but the power cylinder made of aluminium with cooling fins and the outer wall of the compression space made of copper are both surrounded with cooling water and take part in the function of the cooler. Annular type regenerator of 157 mm diameter and 60 mm height surrounds the upper-half of the displacer cylinder. And as the matrix to be held in it, stainless steel balls of 6 mm diameter and perforated copper rings as illustrated in Fig.7 are tested.

The displacer and the power piston are arranged tandem and connected to the W-type crankshaft with connecting rods. The phase angle between them is tentatively set at 90° . In order to reduce the frictional resistance at various bearings due to a gas pressure acting on the power piston, the crank case is pressurized up to the mean working pressure and mechanical seals are adopted for gas sealing at the shaft ends.

To prevent the deterioration of sealing due to the deflection of the crankshaft and a possible error in assembling, the crankshaft and the output shafts are connected with flexible joints made of stainless steel bellows. As the gas seal for the power piston, endless rings made of PTFE (polytetrafluoroethylene) having U-shape cross-section with support of plate springs are fitted along outside and inside of the piston head. The piston head is made of graphite. As the coefficient of thermal expansion of graphite is so small that its adoption makes it possible to keep the clearance between the piston and the cylinder wall small enough, the sealing function of the rings is improved.

3. Performance Test of Heater Elements

In order to obtain the data for making choice of kind of the heater elements to be held in the heater cylinders and that of the regenerator matrix, samples of various heater elements listed in Table 2 were placed in a stainless steel cylinder having the same cross-section as the heater of the engine, and the heat transfer performance at steady heating and the flow resistance against steady air flow were measured. The dimensions of the heater cylinder used in the test and the positions where the temperature was measured are shown in Fig.2, and the shape of the heater element B in Table 2 is illustrated in Fig.3. The heat transfer test was done by putting the heat-

er cylinder which holds various elements into an electric furnace and at the air flow rates of 18, 24, 30, 36 and 42 L/min at the inlet end, while keeping the point A in Fig.2 at a constant temperature of 823 K. From these tests, it was known that the temperature at the points A, B and C presents an approximately linear distribution.

And so, the thickness of the insulating walls at both ends of the furnace was so adjusted as to set the temperature of these three points on one straight line as shown in Fig.4. This diagram shows the temperature distribution for 30 L/min of air flow rate $\dot{V}$, in which the abscissa of each point is taken proportionally to the interval of the positions indicated by alphabetical letters. The air temperature within the sections D~A' and C'~E is re-

Table 2 Model heater elements

Mark	Type and Size
$\phi 3$	3 mm Stainless Steel Ball
$\phi 5$	5 mm Stainless Steel Ball
M	60 Mesh Stainless Steel Wire-gauze
B	15mm Dia. Perforated Copper Cylinder

Fig. 2 Model heater cylinder

Fig. 3 Model heater element B

Fig. 4 Temperature distribution in the model heater ($\dot{V} = 30 \text{ L/min}$)

presented by horizontal lines, expediently assuming they are constant. But actually, there is a certain temperature gradient within these sections. The relation between the temperature rise of air within the section D-E for each heater element and the air flow rate is shown in Fig.5.

According to this diagram, the heat transfer performance of the steel ball is higher than that of other matrices, and the temperature rise with B is about 88~94% of that with $\phi 3$ steel ball at the same condition. However, it must be considered that in this heat transfer test, heat is transferred from the cylinder wall to air passing through the heater element, and in the case of M, contact between the wire gauze and the inner surface of the cylinder wall might not possibly be close enough. When these elements are used as the regenerator matrix, heat exchange takes place only between the matrix and air, therefore the result about the wire gauze is supposed to be much better than that shown in this diagram. Then, a steady air flow was sent through an identical heater cylinder as used for the heat transfer test, and the pressure difference between inlet and outlet of the heater cylinder was measured.

The result of this test is shown in Fig.6. The flow resistance of the element B is far smaller than those of others, and its value is less than 1/10 of the resistance of $\phi 3$ steel ball at the same flow rate.

4. Selection of Heater Element and Regenerator Matrix

In this study, engines that use air or nitrogen, as the working gas, whose flow

Fig. 5 Temperature rise of air by each heater element

Fig. 6 Flow resistance of each heater element

resistance is much higher than that of hydrogen are aimed at. Therefore, the element B was adopted as the heater element, emphasizing the importance of flow resistance in particular. Restricted by the dimensions of the engine body, the length of the heater cylinder becomes 110 mm, thus 4 copper blocks of 27.5 mm length were placed in it. For the regenerator matrix, copper rings which are provided with 108 perforations of 3.2 mm diameter which are similar to the test element B were adopted. The shape of these rings is shown in Fig.7.

A pile of 10 sheets of these rings was held in the regenerator and small washers of stainless steel were inserted between the rings at every two sheets and by means of four through-pins the position of the perforations was brought into line in order to prevent the flow resistance and the heat conduction loss through the matrix.

Besides these copper rings, stainless steel balls were used as the regenerator matrix for the purpose of comparison. In this test, the diameter of balls was changed to 6 mm allowing for some reduction of flow resistance compared with the test mentioned in the previous section. Fig.8 shows the result of measurement of the pressure difference generated between both ends of the regenerator when a steady air flow was sent through it at normal temperature. Although the steel ball matrix is supposed to be superior in the heat transfer, its flow resistance is considerably large, reaching about 6 times that of the copper ring matrix.

5. Running Test

5.1 Power test and motoring test

The power test was performed from the

Fig. 7 Perforated copper ring matrix

Fig. 8 Flow resistance of each regenerator

minimum load up to the maximum load at which continuous running was possible at the mean working pressures $P_m = 100, 300$ and 500 kPa, using a small D.C. electric dynamometer of 440 W. As a very small torque remains after the electric current is cut off due to the friction of bearings of the dynamometer, a perfect no-load running can not be done. Chromel-alumel thermocouples of 0.1 mm wire diameter are fitted to the expansion space, the compression space and hot and cold ends of the regenerator, and the air temperature at each position is measured in digital millivoltmeters. At the time of the minimum load running for each P_m , the mean temperature in the expansion space was adjusted to 960 K and the measurement at each loading was done, while the flow rates of fuel gas and cooling water were kept constant at the original values. The motoring test was done using the dynamometer as a D.C. motor and the driving torque at several running speeds was measured in the same three steps of P_m as in the power test. In these cases, the torque due to the flow resistance of air is included in the driving torque.

5.2 Observation of pressure curve

The wave form of pressure variation in the expansion and the compression spaces at the minimum load running for each P_m was observed by the synchroscope through the pressure converters of variable inductance type.

6. Result of Tests

6.1 Comparison of two kinds of regenerators

The relation between the frictional work L_f and the net work L_e per one cycle and the engine speed n obtained from the running tests which were performed using two kinds of regenerators is shown in Fig. 9. The scale of the values of L_f is taken downward from the origin on the ordinate.

The figures written along the curves indicate the mean working pressure P_m in kPa, which is also applicable to Fig. 10, Fig. 11 and Fig. 15 shown hereunder. The curves plotted with black symbols show the characteristics with the steel ball matrix,

Fig. 9 L_f and L_e by each regenerator

and those plotted with white symbols show the characteristics with the copper ring matrix. L_f increases as P_m rises, but the value with the steel ball matrix is larger than that with the copper ring matrix for every P_m , because of the prominence of flow resistance. With regard to L_e , the value with the steel ball matrix is smaller, and the running speed at the minimum load is much lower than that with the copper ring matrix. This is supposed to be caused not only by the difference of flow resistance but also by the fact that the void volume in the regenerator is larger with steel balls than with copper rings. The void volume is about 198 cm^3 for steel balls and about 64 cm^3 for copper rings. Since it is thus made clear that a considerably higher output is obtained by the copper ring matrix, further discussion will be made only on the case of the copper ring matrix.

Fig. 10 L_f , L_e and L_i by copper ring regenerator

Fig. 11 Temperature variation due to running speed

6.2 Indicated work and air temperature at each location

The indicated work L_i obtained as the sum of the values of L_f and L_e at each running speed from the curves with the copper ring matrix in Fig.9 is shown in Fig.10 together with L_f and L_e . And Fig.11 shows the mean air temperature during one cycle at the expansion and the compression spaces and both ends of the regenerator which were measured in the power test. As the running speed increases, the temperature at the expansion space and the hot end of the regenerator falls, and that of the compression space and the cold end of the regenerator rises. And also, as P_m rises, the temperature at the compression space and the cold end of the regenerator rises, and the inclination of the curve of temperature at the expansion space and the hot end of the regenerator becomes steeper.

6.3 Pressure curves at expansion and compression spaces

Fig.12 shows the pressure curves at the expansion and the compression spaces observed during the minimum load running at each P_m . The pulse marks displayed along the bottom of each diagram represent the crank angle of 5° before or after top and bottom dead centers (TDC and BDC) and of every 10° . The pressure amplitude at the compression space is slightly larger than that at the expansion space for every P_m .

This is due to the flow resistance of air that flows reciprocally between the both spaces.

7. Model for Calculation

In order to examine the test results mentioned in the previous section, the authors partially modified the formulae representing the performance of the Stirling engine which were derived in their previous report⁽¹⁾ based upon the assumption of Schmidt's theory, and put the data of the test engine and of the test running condition into these formulae to obtain the theoretical values of pressure and the work done per one cycle.

Nomenclature

- T_E : gas temperature in expansion space
- T_C : gas temperature in compression space
- τ : temperature ratio = T_C / T_E
- V_A : average volume of expansion space
- V_B : average volume of compression space
- V_Z : total volume of heater, cooler, regenerator and their connecting ducts
- V_D : swept volume of displacer
- V_P : swept volume of power piston
- σ : swept volume ratio = V_P / V_D
- ϑ : crank angle of power piston
(put $\vartheta = 0^\circ$ at BDC of power piston)
- α : phase angle of displacer
- M : total mass of working gas
- R : gas constant of working gas
- P : pressure of working gas
- P_m : mean working pressure
- L : work done per one cycle
- $x = V_A / V_D$
- $y = V_B / V_D$
- $z = V_Z / V_D$

P_E : Pressure of Expansion Space
 P_C : Pressure of Compression Space

Fig. 12 Pressure curves at expansion and compression spaces

Relation among the volumes of various parts at $\alpha = 90^\circ$ is shown in Fig.13. In the previous report, the volume V_Z shown in this figure was divided in two parts. One half of the regenerator volume and the volume relating to the heater were included in V_A , and the other half of the regenerator volume and the volume relating to the cooler were included in V_B . Then, the temperatures in V_A and V_B were assumed to be T_E and T_C respectively. But in this study, V_Z is distinguished from V_A and V_B , and dealt with independently for the reasons mentioned below. Out of the air temperatures at various locations that are shown in Fig.11, if those at the minimum load running for every step of P_m are plotted referring to the abscissa with the scale taken proportionally to the cumulative volume up to the measuring position, Fig.14 is obtained.

While the temperature distribution between the measuring positions is not known, and at the hot end of the regenerator, the temperature varies periodically with an amplitude of about 65 K at the minimum load and about 80 K at the maximum load, both for $P_m = 500$ kPa, the temperatures at these four points may be considered to fall into one straight line anyway. The amplitude of temperature fluctuation at the cold end of the regenerator is about 60% of that at the hot end, and in the case of lower P_m , the amplitude becomes smaller.

Even if the running speed comes down towards the maximum load operation and the temperature of each position changes, this linear relation does not change. Therefore, this problem is dealt with tentatively on the assumption that the temperature distributes approximately linear to the cumulative volume from T_C to T_E over the range of V_Z . If the distribution of the gas densities due to this temperature distribution is taken into consideration, we get

$$T_z = \frac{T_E - T_C}{\ln(T_E/T_C)} = -\frac{1-r}{r \ln r} T_C \dots\dots\dots(1)$$

as the thermodynamical mean temperature of the gas in the volume V_Z . Transforming the formulae for the pressure and for the work done per one cycle in the previous report by applying this formula, following formulae are obtained.

Fig. 13 Designation of volumes in the engine (in the case of $\alpha = 90^\circ$)

$$A = rx + y - \frac{r \ln r}{1-r} z \dots\dots\dots(2)$$

$$P = \frac{2MRT_c/V_D}{2A + (1-r) \cos(\theta + \alpha) - \sigma \cos \theta} \dots\dots\dots(3)$$

$$L = \int \frac{\sigma MRT_c \sin \theta d\theta}{2A + (1-r) \cos(\theta + \alpha) - \sigma \cos \theta} \dots\dots\dots(4)$$

Then, putting them as Eq.(5), and integrating Eq.(4), we get Eq.(6).

$$\delta = \frac{\sqrt{\sigma^2 - 2\sigma(1-r) \cos \alpha + (1-r)^2}}{2A} \dots\dots\dots(5)$$

$$L = \frac{\pi MRT_c(1-r)\sigma}{2A^2\delta^2} \left(\frac{1}{\sqrt{1-\delta^2}} - 1 \right) \sin \alpha \dots\dots\dots(6)$$

And the mean working pressure P_m is given by Eq.(7), and Eq.(8) should hold.

$$P_m = \frac{1}{2\pi} \oint P d\theta = \frac{MRT_c}{V_D A \sqrt{1-\delta^2}} \dots\dots\dots(7)$$

$$MRT_c = A \sqrt{1-\delta^2} V_D P_m \dots\dots\dots(8)$$

Using the volumes of various spaces of the test engine and the values of r and P_m at the operation, the value of MRT_c is obtained. By substituting this value into Eq.(3) and Eq.(6), P and L are calculated.

ES : Expansion Space
H : Heater
R : Regenerator
C : Cooler
CS : Compression Space

Fig. 14 Temperature distribution against cumulative volume in the engine

⊙ : Indicated Work Obtained from
Observed P-V Diagram

Fig. 15 Comparison of L and L_i

8. Comparison of Theoretical and Experimental Values

The work done per one cycle L which was calculated using the temperature ratio τ obtained from the temperature of the expansion and the compression spaces in the power test as shown in Fig.11 and the indicated work L_i shown in Fig.10 are superposed and illustrated in Fig.15. The values of L are calculated at the points plotted along each curve.

In Fig.16, the P - V diagrams drawn from the calculated pressure by Eq.(3) using the temperature ratio τ at the minimum load operation for each P_m and from the observed pressure in the compression space P_C shown in Fig.12 are superposed and presented. The value of τ is indicated at the top of each diagram. The volume V on the abscissa represents the total volume of working space which is expressed by Eq.(9).

$$V = V_D(x+y+z-0.5\sigma \cos \theta) \dots\dots\dots (9)$$

Corresponding to the fact that frictional work that was obtained in the motoring test includes the flow resistance of air, the indicated work includes this resistance.

Therefore, the authors decided to use, as the ordinate in the P - V diagram of the observed pressure, the pressure of the compression space P_C which has an amplitude larger than that of P_E by the amount of flow resistance. In Fig.15, the difference between L and L_i gradually grows as P_m rises. This would be caused by the fact that the gas leakage at the PTFE ring of the power piston increases as P_m rises and the pressure amplitude becomes smaller than the theoretical value. The value of L comes down as the running speed increases.

This is caused by the increase of τ .

At 300 and 500 kPa, L_i shows the same downward tendency as L , but at 100 kPa, L_i does not show this tendency. As the cause of this, the gas leakage which might have begun in the low speed range is suspected.

In the P - V diagrams in Fig.16, the difference in the area between the calculated and the observed values increases as P_m rises corresponding to the difference between L and L_i in Fig.15.

9. Conclusions

With the target of developing a Stirling engine that utilizes waste heat with cheap and easy-to-handle gases such as air or nitrogen as the working gas, the authors built an engine of displacer type on experimental basis with particular stress on the reduction of flow resistance and tested its performances.

As the result of the tests, the following facts were made clear:

(1) The tested heater using stainless steel cylinders holding perforated copper blocks causes a small flow resistance and requires no difficult welding job as does the conventional heater fitted with numerous fine parallel tubes.

(2) As the regenerator matrix, two kinds of them, namely, stainless steel balls of 6 mm diameter and a pile of perforated copper rings were tested. While the steel ball matrix shows good heat transfer performance, its flow resistance is high, and it has a large void ratio. Thus, high overall performance as expected was not gained. The matrix of copper rings causes a small flow resistance and gives almost satisfactory result.

(3) The diameter of the fine perforation of the copper blocks for the heater and of the copper rings for the regenerator matrix was 3.2 mm. But if the leakage of lubricant from the power piston could be prevented, the area of heat transfer surface could be increased further by choosing a smaller diameter and by increasing the number of perforations.

(4) The heat resisting properties of copper that was used as the heater element and the matrix have not been fully investigated.

It would be necessary to search for more excellent heat resisting and heat conducting materials in order to improve their durability.

(5) In order to get an engine for practical use with increased performance, the mean working pressure should be raised considerably. However, the seal ring used for the power piston did not function perfectly.

Further development of this part is required most urgently among other things.

Fig. 16 Comparison of P - V diagrams for calculated and observed pressure

(6) As the result of the measurement, it was found that the temperature at the hot end of the regenerator is fairly lower than that of the expansion space. Though this may suggest that the function of the regenerator is not sufficient, the fact is attributable to the heater being heated by the gas burner from its bottom. If viewed from another angle, the copper blocks in the heater cylinder have a similar structure to that of the regenerator matrix, and have a considerably large heat capacity.

Therefore, it may be construed that these copper blocks are performing the function of the heater and the regenerator simultaneously. The authors intend to make a closer investigation about this respect from now on.

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