

TEST RESULTS OF APPLICATIVE 100 W STIRLING ENGINE

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ABSTRACT

A small 100 W displacer-type Stirling engine, 'Ecoboy-SCM81' has been developed by a committee of the Japan Society of Mechanical Engineers (JSME). The engine contains unique features, including an expansion cylinder which is heated by either combustion gas or direct solar energy. Also, a simple cooling system rejects heat from the working fluid. A displacer piston has both heating and cooling inner tubes for the working fluid which flows to and from outer tubes. The outer tubes for heating were located at the top of the expansion cylinder and the outer tubes for cooling were located in the middle of the cylinder. A regenerator is located in the displacer piston.

The components of the engine adopted some new technologies. For instance, a porous type matrix consisting of pressed zigzag stainless steel wires was adopted for the regenerator. The matrix is practical for Stirling engines because it can be made at low cost and the assembling process is simplified.

INTRODUCTION

A committee called RC110, was organized in JSME from 1992 to 1994 to collect and organize both the knowledge and design methods for conventional Stirling machines. Its final goal was to make a draft of a design manual which was intended to guide Stirling developers (JSME, 1994). From 1994 to 1996, another JSME committee, RC127, was established to confirm the previous design methods and to organize the information for applications which use the Stirling machines. One of the projects of RC127 was to develop a small 100 W displacer-type Stirling engine using the RC110 manual.

After many discussions, a prototype engine will be used as a prime mover for a small generator. The goal was to achieve an output of 100 W with 20 % thermal efficiency. Many unique ideas for the engine configuration and its components were proposed by the committee. During the initial design process, the performances of major components were calculated individually and the total engine performance was estimated by several models (Kagawa 1995_a). The results implied that the engine could achieve the target goal.

The engine was constructed based on detailed

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TABLE 1 SPECIFICATIONS AND TARGET PERFORMANCE

Engine type	Displacer type (gamma configuration)
Working gas	Helium/Air
Mean engine pressure	1 Mpa
Expansion space temp.	923 K
Compression space temp.	343 K (Water/Air cooling)
Engine speed	1000 rpm
Bore x Stroke	72 x 20mm
Output power	100 W
Thermal efficiency	20 %

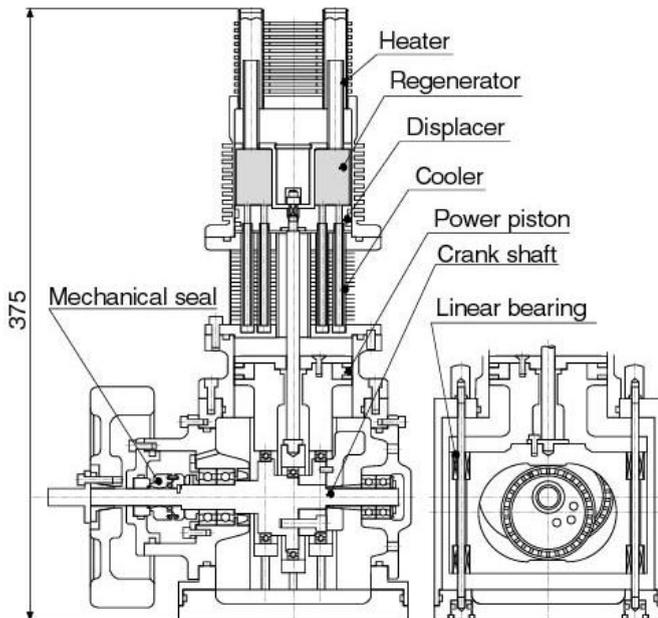


FIGURE 1 SCHEMATIC VIEW OF ECOBOY-SCM81

blueprints and was named Ecoboy-SCM81. In order to collect adequate information for completing the design manual, evaluation of the engine's performance and analysis methods of the experimental results of the engine and its components will be discussed in this paper.

CONFIGURATION OF ECOBOY-SCM81 ENGINE

The engine was designed to be applied as a portable generator with a sealed container filled with liquefied propane and butane as fuel.

The engine's goal was to achieve an output power of 100 W at a mean pressure of 1 MPa operating at 1000 rpm as shown in Table 1. The mean pressure and temperature, 1 MPa and 923 K, although relatively low values when compared with most high-performance engines, were chosen in order to make the engine compact and easy to construct. The low temperature of the heat exchanger was set to 343 K since the generator will be cooled by ambient air.

TABLE 2 SPECIFICATIONS OF HEAT EXCHANGER SYSTEM

	Heater	Cooler
Type	Double tube	Double tube
No. tubes	10	24
Diameters of tubes (OD, ID) (mm)		
inner	9.5, 7.0	5.0, 4.0
outer	13.8, 12.4	8.0, 7.4
Length of tubes (mm)	54	56
Material	Stainless steel	Copper

	Regenerator
Type	Annular tube
Unit No.	1
Diameters (mm)	
inner	27
outer	66
Length of tubes (mm)	35
Matrix material	Stainless steel

TABLE 3 DETAILS OF REGENERATOR MATRICES

	Wire dia. (mm)	Weight (g)	Surface area (m ²)	Porosity
Spring Mesh	0.100	219	1.10	0.754
Stacked mesh	0.101	280	1.33	0.686

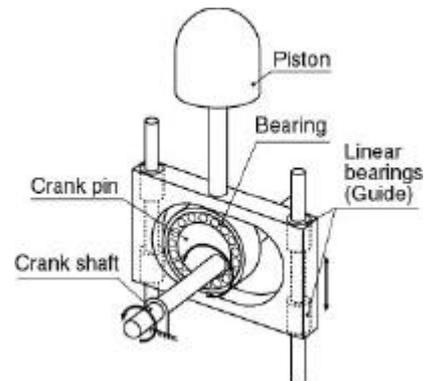


FIGURE 2 SCOTCH-YOKE MECHANISM

To construct a small and light engine easily, the heat exchanger design was simplified. Also, we adopted a unique gamma configuration with a specialized heat exchanger system as shown in Fig. 1. This can be called a 'moving-type heat exchanger'. Its specifications are shown in Table 2. The displacer piston integrates an annular regenerator and has the heating and cooling tubes at the top and bottom sides which function as heat exchangers. These tubes are reciprocated in both outer tubes which receive heat form the high-temperature source at the upper side of the engine and reject heat to the low-temperature sink in the middle of the cylinder. Using an in-line layout, the displacer and power piston are integrated in a cylinder which has outer tubes for heating located at the top.

Two different matrix types were chosen for the regenerator. One is a conventional stacked mesh made of

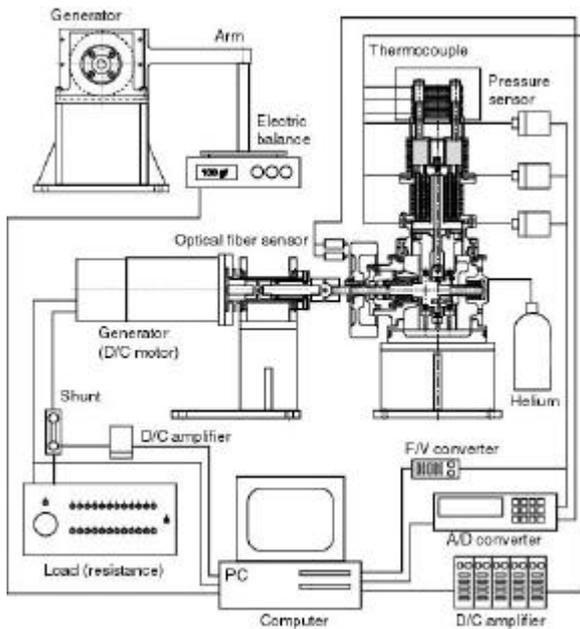


FIGURE 3 MEASURING SYSTEM OF PROTOTYPE ENGINE

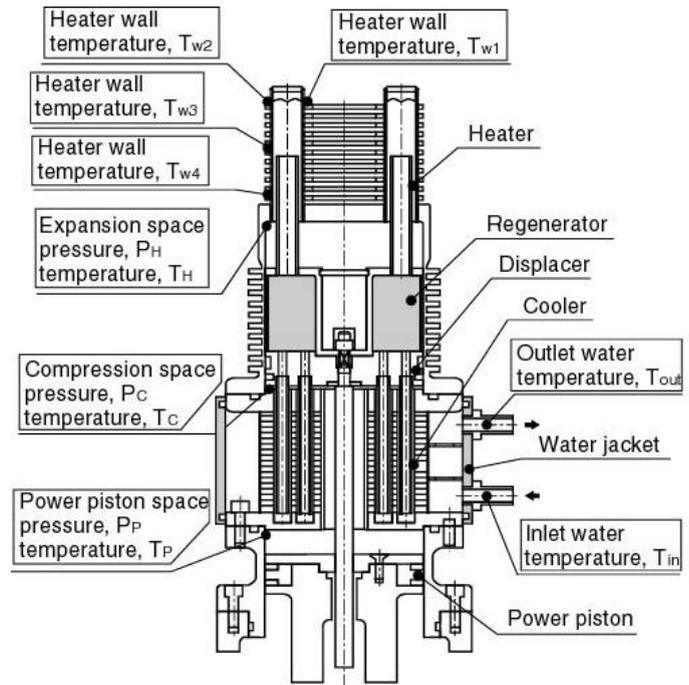


FIGURE 4 MEASURING POINTS OF PRESSURE AND TEMPERATURE

TABLE 4 EXPERIMENTAL CONDITIONS

Heat source	Electric heater
Heater wall temp.	973 K
Working gas	Helium
Mean engine pressure	0.8 MPa / 0.6 MPa / 0.4 MPa
Regenerator	Spring Mesh / Stacked mesh
Cooling type	Water cooling
Piston ring	4-piece type
Engine speed	700 ~ 1400 rpm

stainless steel. Another is a porous type matrix consisting of pressed zigzag stainless steel wires called 'Spring Mesh'. Because the matrix can be made at low cost and the assembling process is simplified, the matrix is practical for Stirling engines. Details of the matrices are given in Table 3. The stacked mesh is composed of stacking 100 wire mesh. The Spring Mesh is composed of stacked and pressed zigzag wire.

For the mechanism, a Scotch-yoke design was adopted as shown in Fig. 2. No lubricant system was required for this crank mechanism. The Scotch-yoke mechanism can use compact radial bearings. Their mechanical dimensions were carefully determined based on calculated forces at loads on the mechanical parts (Kagawa, 1995_a). In order to keep the displacer piston from contacting the outer tubes of the heat exchangers during reciprocation, the mechanical parts carefully maintain strict tolerances

(Kagawa, 1995_b).

EXPERIMENTAL METHOD

Figure 3 shows the measuring system for the prototype engine. Figure 4 shows measured pressure and temperature points. The engine's pressure changes were measured in three spaces; an expansion space, a compression space and a power piston space. These pressure measurements were triggered and arranged by pulse signals generated at the top dead point of the displacer piston and at every 2 degree crank angle by using two optical fiber sensors. From the pressure data and pulse signals which show each piston position, the compression power, L_c , was calculated as the sum of indicated powers in the compression and power piston space in Fig. 4. The indicated power, L_i , was calculated as the sum of the expansion power, L_h , and the compression power, L_c .

Four heater wall temperatures, three working gas temperatures, and the inlet and outlet water temperature of the water jacket were measured by thermocouples.

The load to the engine output was adjusted by changing a value of a resistance which is connected to a generator. The generator used a D/C motor with an arm attached as shown in Fig. 3. The arm was connected to an electric balance to measure the engine torque.

EXPERIMENTAL RESULTS

Table 4 shows experimental conditions. A 1 kW (max.)

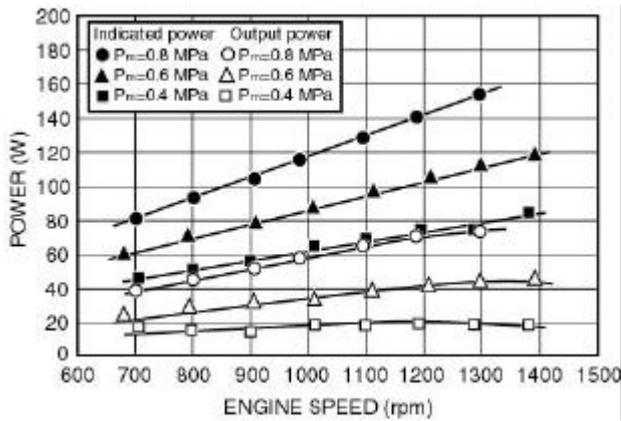


FIGURE 5 POWER AS A FUNCTION OF ENGINE SPEED

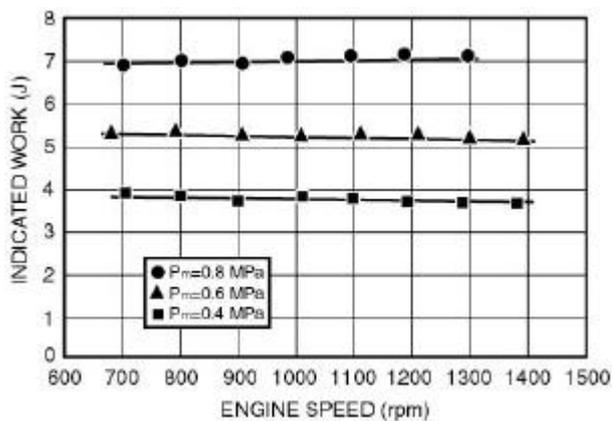


FIGURE 6 INDICATED WORK AS A FUNCTION OF ENGINE SPEED

electric heater was used to heat the engine. Heat input from the electric heater was adjusted to maintain the heater wall temperature, T_{w2} , to 973 K. T_{w2} indicates the highest heater wall temperatures in the engine system. A constant flow rate of water was used in the experiments to cool the engine. Conventional four-piece-type piston rings were used at the power piston. Helium was used as the working gas and its mean pressure was adjusted to either 0.4 MPa, 0.6 MPa or 0.8 MPa. Both the Mesh Spring and the stacked mesh were used under the above conditions.

Engine Performance

Figure 5 shows the relation between the engine speed

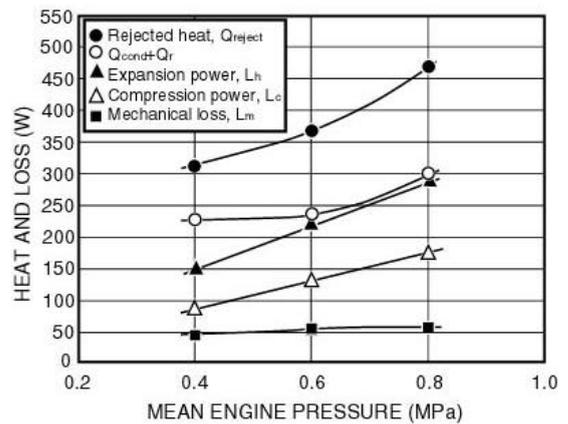


FIGURE 8 HEATS AND LOSSES AS A FUNCTION OF MEAN ENGINE PRESSURE

and the indicated and output power using the Spring Mesh as the regenerator matrix. During the experiment, the expansion gas temperature, T_H , reached approximately 763 K. This temperature was definitely lower than the target temperature from Table 1. There was 250 K temperature difference between T_{w2} and the bottom heater wall temperature, T_{w4} . This large temperature difference may be caused by the shape of the electric heater and furnace. From Fig. 5, the indicated and output power increase proportionally with higher engine speed. The maximum output power was 74 W at 0.8 MPa mean engine pressure operating at 1300 rpm engine speed. For these conditions, the engine generated an electric power of 52 W.

Figure 6 shows the relation between the engine speed and the indicated work under the same conditions. The indicated work was constant even when the engine speed changed.

Figure 7 shows the relation between the engine speed and the mechanical loss, $L_m (=L_i - L_s)$. From this figure, the mechanical loss increases proportionally with higher engine speed. These mechanical loss values are two or three times as large as the calculated results of the initial design process (Kagawa, 1995_a). This large mechanical loss is one of the reasons why this engine has not been able to reach the target goal. A more detailed analysis on the mechanical loss is required to increase the engine performance.

Efficiency and Heat Loss

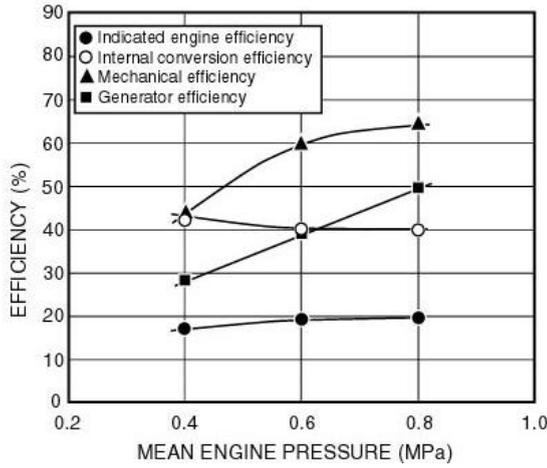


FIGURE 9 EFFICIENCY AS A FUNCTION OF MEAN ENGINE PRESSURE

Figure 8 shows results using the Spring Mesh as the regenerator matrix and operating at 1000 rpm engine speed. The rejected heat, the sum of the conduction loss and the regenerator loss, the expansion power, the compression power and the mechanical loss are plotted as function of the mean engine pressure. In this figure, the rejected heat, Q_{reject} , was calculated from the difference between the cooling water inlet and outlet temperatures, the water flow rate of the cooling water and the specific heat of the cooling water. The sum of the conduction loss and the regenerator loss was calculated as the difference of Q_{reject} and L_c approximately as shown in Eq. 1. From this figure, these heats and losses increase with higher mean engine pressure.

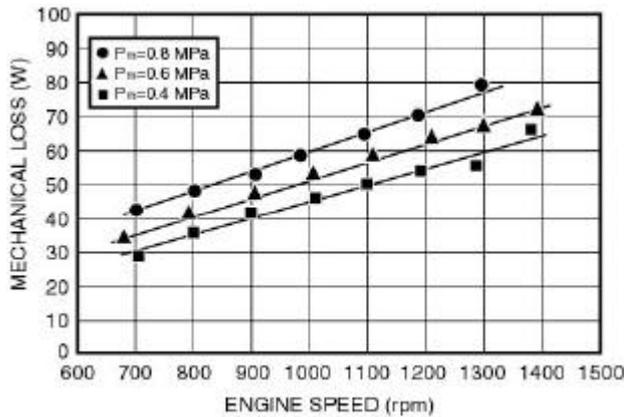


FIGURE 7 MECHANICAL LOSS AS A FUNCTION OF ENGINE SPEED

$$Q_{cond} + Q_r \cong Q_{reject} - L_c \quad (1)$$

Figure 9 shows the indicated engine efficiency (η_{it}), the internal power conversion efficiency (η_{int}), the mechanical efficiency (η_m), and the generator efficiency (η_g) as a function of the mean engine pressure under these same conditions.

The indicated engine efficiency, η_{it} is defined as the indicated power, L_i , and the rejected heat, Q_{reject} as shown in Eq. 2.

$$h_{it} = \frac{L_i}{L_i + Q_{reject}} \quad (2)$$

Other efficiencies are derived from the indicated power, L_i , the expansion power, L_h , the output power, L_s , and the generator power, L_g , as follows.

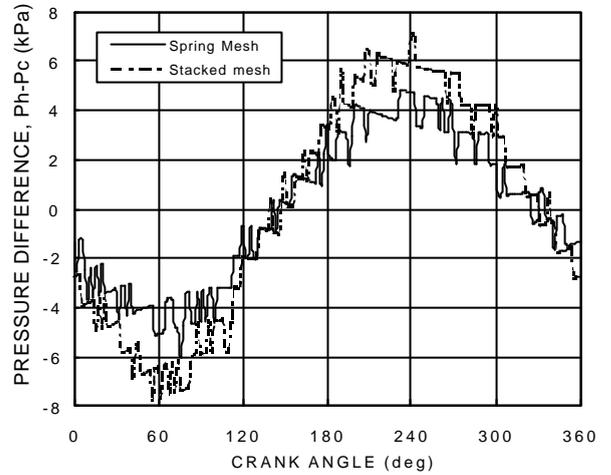


FIGURE 11 PRESSURE DIFFERENCE AS A FUNCTION OF CRANK ANGLE

$$h_{int} = \frac{L_i}{L_h} \quad (3)$$

$$h_m = \frac{L_s}{L_i} \quad (4)$$

$$h_g = \frac{L_g}{L_s} \quad (5)$$

From this Fig. 9, the mechanical efficiency, η_m , and the generator efficiency, η_g , increase with higher mean engine pressure. On the other hand, the internal power conversion efficiency, η_{int} decreases with higher mean engine pressure. The indicated engine efficiency, η_{it} , is approximately constant. The mechanical efficiency is lower than that of general high performance Stirling engines.

Matrix of the Regenerator

Figure 10 shows the relation between the engine speed and the power with 0.8 MPa mean engine pressure for both the Spring Mesh and the stacked mesh. From this figure, the indicated power using the Spring Mesh was slightly lower than that of the stacked mesh. But the output power using the Spring Mesh is higher than that of the stacked mesh. The reason for this is that the larger flow resistance and the inertia of the heavier mass by the matrix increased the force loading on the mechanical parts. Although the mechanical loss was increased by these items, the increasing ratio is small compared to the overall mechanical loss. To find the reason for the large mechanical loss requires more detailed analysis of the mechanical parts.

Figure 11 shows the pressure difference between the expansion and compression spaces as a function of the crank angle. In this figure, the crank angle 0 degree corresponds to the angle of the top dead point of the displacer piston. The pressure difference using the Spring Mesh was smaller than that using the stacked mesh.

Table 3 shows that the surface area and weight of the stacked mesh are larger than those of Spring Mesh. However, the porosity of the stacked mesh is smaller than that of Spring Mesh. The stacked mesh has better heat transfer characteristics than those of the Spring Mesh. On the other hand, the Spring Mesh has better flow characteristics. Figures 10 and 11 show the effects of heat transfer and flow loss on regenerator performance.

CONCLUSION

The prototype engine reached a maximum output power of 74 W at 0.8 MPa mean engine pressure operating at 1300 rpm engine speed. The reasons that the engine did not reach the target goal are:

- (i) the mechanical loss was larger than the calculated results of the initial design process
- (ii) the expansion gas temperature was lower than the calculated results of the initial design process.

The engine will reach the target goal after finding the solution to these problems.

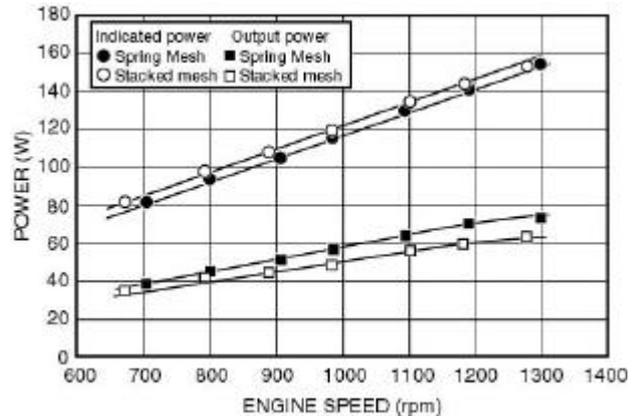


FIGURE 10 EFFECT OF MATRIX OF THE REGENERATOR

The output power of the prototype engine using the Spring Mesh was larger than that using the stacked mesh under the same conditions. The Spring Mesh is therefore a practical alternative for Stirling engines.

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