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Conceptual design of scroll expander-compressor for Stirling engine

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ABSTRACT

Conceptual design of scroll expander and scroll compressor for 10 kW-class Stirling engine utilizing solar energy as heat source has been carried out. CO2 was chosen as the working fluid, since it is nontoxic, inflammable and it has relatively higher density among probably usable gases. Gas temperature at the expander inlet was set at 700˚C, and that at the compressor inlet was at 40°C. System efficiency and engine power output were calculated for the high side pressure ranging from 5 MPa to 8 MPa with the pressure ratio of 1.5~4. System efficiency reached maximum at the pressure ratio of about 2.5, and the peak efficiency increased with increasing high side pressure. Due to safety concern, the pressure condition of 6 MPa / 2.5 MPa was chosen as design condition. Orbiting scroll members for the expander and compressor were designed to have double-sided structure in order to reduce the overall scroll size and to cancel out the axial gas forces acting on the orbiting scroll base plate. Expander and compressor were connected by two common shafts. Timing belt were adopted to link these two shafts, functioning as anti-rotation device for the orbiting scroll members. Gear assembly was used to extract the net power output from the expander. By parametric study on the scroll profile, smaller possible size for the scroll members was obtained. With the shaft speed of 3600 rpm, the shaft output of the designed scroll expander was calculated to be 45.4 kW, while input power for the scroll compressor was 34.5 kW, yielding 10.9 kW for the output power of the Stirling engine. System efficiency was estimated to be about 7.3%, and overall efficiencies of the scroll expander and compressor were around 84.1% and 88.3%, respectively.

1. INTRODUCTION

An ideal Stirling cycle has the same theoretical efficiency as that of Carnot cycle. Efficiency of a real Stirling engine with reciprocating pistons, however, is far below the theoretical one. It is partly because lower temperature compression process and higher temperature expansion process are mechanically interfered each other so that both processes can not be perfectly completed. Another reason for the lower efficiency of the real reciprocating Stirling engine is that due to high operation speed and limited heat transfer area the compression and expansion processes can not be carried out isothermally. Rather, these processes are made closer to adiabatic ones. In order to overcome these drawbacks inherent to reciprocating Stirling engine, Stirling engine with scroll mechanism has been proposed(1). By employing scroll expander and scroll compressor, compression and expansion processes can be made completely independent from each other, and one way flow direction of the working fluid and more heat transfer area of cooling or heating can be achievable. All these effects altogether alleviate Stirling engine performance degradation from the theoretical value. In this paper, feasibility study on a Stirling engine with scroll expander and scroll compressor using solar heat as heat source has been made.

2. STIRLING CYCLE EMPLOYING SCROLL EXPANDER AND SCROLL COMPRESSOR

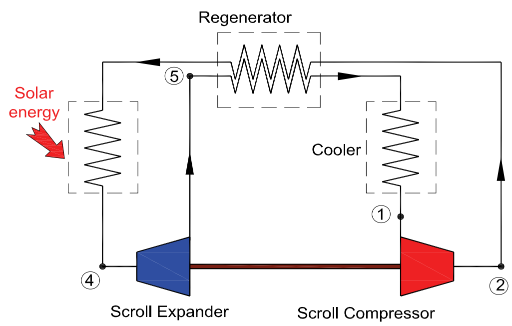


Fig. 1 Stirling cycle using scroll expander and scroll compressor utilizing solar energy as heat source

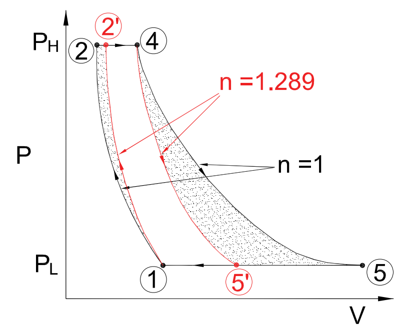


Fig. 2 Effect of compression expansion indices on P-V diagram

Fig.1 illustrates the Stirling cycle composed of scroll expander and scroll compressor. In the compression process of ①-②, the working fluid is compressed in a scroll compressor. The temperature of the working fluid is raised up firstly in a regenerator which is a heat exchanger between the working fluid of relatively lower temperature from the compressor discharge and that of relatively higher temperature from the expander outlet. Then the fluid is further heated up in a heater which receives the solar energy as heat source. In the expansion process④-⑤, the fluid of high pressure and high temperature undergoes expansion, generating power output. Some portion of the expander output is consumed to drive the compressor. The net power output of the Stirling engine is determined by the difference between the expander output and the compressor input.

Fig.2 shows the effects of the polytropic index on the P-V diagram, and thereby on the engine power output. More power output can be obtained with isothermal expansion and compression than that with adiabatic processes. For real expander and compressor devices, conditions of isothermal expansion and compression are very difficult to be met: sufficient heat transfer area is hardly available in the devices of limited size and operation speed should be extremely low, which is not realistic for practical engine. Hugenroth et al(2) proposed a way of overcoming the substantial practical difficulties of achieving isothermal compression and expansion processes by mixing a nonvolatile liquid with the noncondensable gas during the compression and expansion processes if the liquid's capacitance rate is much greater compared to the gas's capacitance rate. For this study, however, application of mixing of large amount of liquid with the working fluid was not considered because of increased complexity of the system.

Since for the presently considered Stirling engine the temperature of a working fluid entering into the expander inlet would be very high, maintaining isothermality during the expansion process seems not to be realistic. With no temperature management on the expansion process, it can be regarded as adiabatic. For compressor side, water jacket is to be provided so that the compression process can be regarded as the polytropic one with n=1.1.

Power output of the Stirling engine to be designed is 10kW and carbon dioxide is chosen for the working fluid, since it is nontoxic, inflammable and it has relatively higher density among probably usable gases. Though higher temperature at the expander inlet could produce larger power output, gas temperature was set at 700°C due to heat-resisting property of the expander material. Low side temperature was set at 40°C.

Fig.3 shows the variation of the Stirling engine efficiency with various pressure conditions. For this calculation, the following assumptions were made: $\eta_e \simeq 80\%$, $\eta_{v,e} \simeq 70\%$, $\eta_c \simeq 90\%$ $\eta_{v,c} \simeq 85\%$, and $\eta_{regn} \simeq 80\%$. At the pressure ratio of around 2.5, the Stirling engine shows maximum efficiency. The peak efficiency increases with increasing high side pressure. In this design study, however, high side pressure was set at $P_H = 6\ MPa$ because of safety consideration. Design operating conditions are summarized in Table 1. Mass flow rate for 10kW-engine under these conditions is $\dot{m} \simeq 0.332 kg/s$.

Table 1 Design operating conditions

Expander		Compressor	
P_4	5.7 [MPa]	P_1	2.5 [MPa]
P_5	2.6 [MPa]	P_2	6 [MPa]
T_4	700 [°C]	T_1	40 [°C]

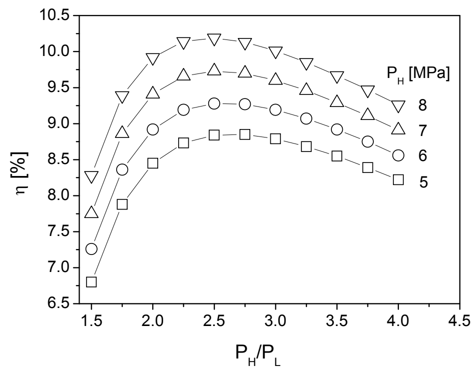


Fig. 3 Stirling engine efficiency at various pressure conditions

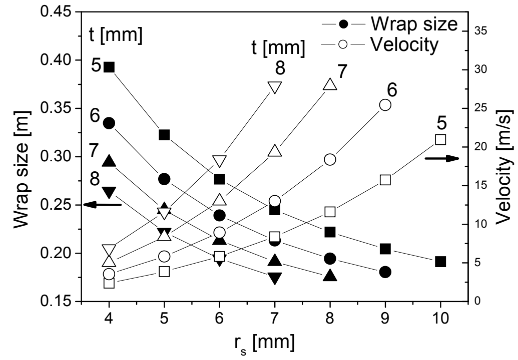


Fig. 4 Scroll wrap size and inlet gas velocity

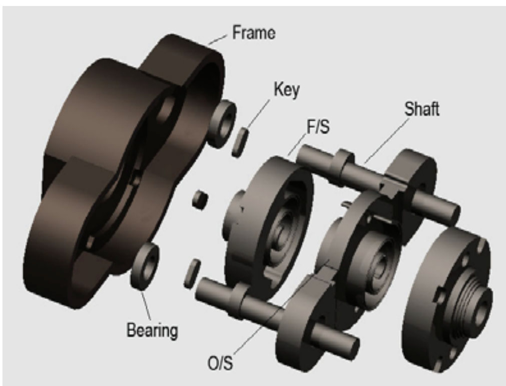


Fig. 5 Scroll expander

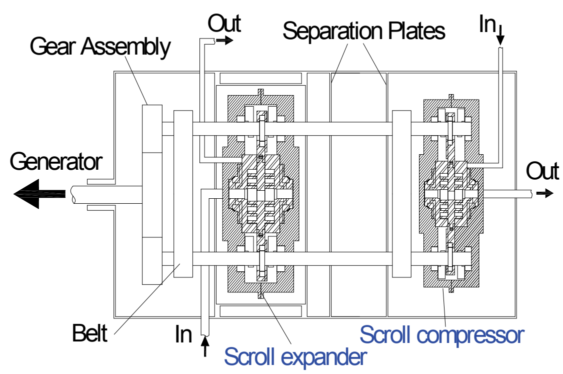
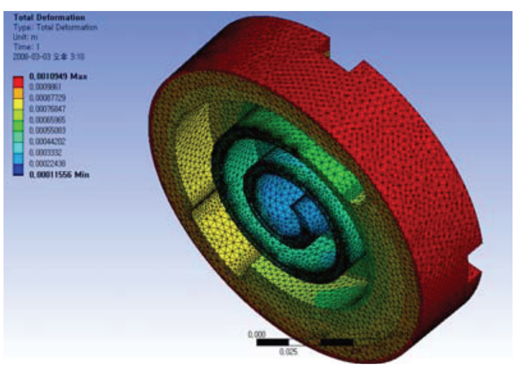
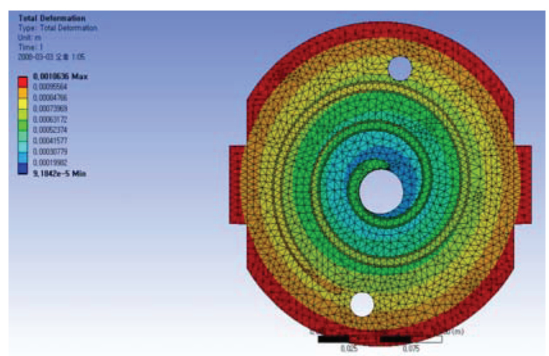


Fig. 6 Schematic layout of Stirling engine



(a)



(b)

Fig. 7 Thermal expansion on scroll expander: (a) fixed scroll, (b) orbiting scroll

3. BASIC STRUCTURE OF THE EXPANDER AND COMPRESSOR AND SCROLL SIZING

Since the expander operates in high temperature environment, availability of the high temperature material needs to be taken into consideration and there is also limitation on the size of the scroll base plate for machining. One way of reducing the scroll base plate diameter is to take the orbiting scroll structure with the scroll wraps on both sides of the base plate. With this type of double-sided scroll wrap structure, axial gas forces acting on the both sides of the base plate cancel each other out, but precision machining is required for proper performance. Another way of reducing the base plate size is to increase the operation speed, but this induces friction loss increment. Design operation speed was set at 3600rpm.

For axial compliance, back pressure chambers were made on the rear side of the fixed scroll members to produce thrust so that the fixed scroll members may be pushed towards the orbiting scroll member in between them. For radial compliance, plate type springs were inserted between orbiting scroll base plate and the drive bearing holder. These springs could also be used to absorb thermal expansion of the orbiting scroll member. Inconel was chosen as the raw material for the scroll expander, since its rigidity is acceptable at the present temperature range. The scroll expander and the compressor share the two common shafts, and timing belts connecting the two shafts function as anti-rotation device for the orbiting scroll members.

There are seven of the scroll configuration variables which are used to determine the scroll profile(3): base circle radius a, wrap thickness t, wrap height h, orbiting radius r_s, wrap end angle ϕ_e, cutter angle ϕ_a, and starting angle α, while there are 4 equations relating these variables. Relative wrap height to the wrap thickness needs to be limited because of the gas force-induced deflection and machining difficulty.

Considering that the pressure difference between the high and low sides is about 3.5 MPa, ratio of the wrap height to the wrap thickness was set to h/t=4.5. Then, two out of seven variables can be selected as independent ones, and the remaining five as dependent ones. In this study, the wrap thickness t and the orbiting radius r_s were taken as independent variables, since these two are of practical importance in machining the scroll wrap.

Fig.4 shows variation of the wrap size ($2a\phi_e$) and gas velocity at the scroll expander inlet with variation of t and r_s. With not much large inlet gas velocity, the smallest scroll size could be obtained at t=8mm and r_s= 7mm. Diameter of designed scroll base plate was about 249 mm for the expander, and it was about 218 mm for the compressor. Scroll expander designed in this way and the general layout of the Stirling engine employing the scroll expander and scroll compressor are given in Fig.5 and Fig.6, respectively.

Thermal expansion on the fixed and orbiting scrolls of the expander were calculated by using a FEM code, ANSYS, and presented in Fig.7(a) and (b). Largest expansion was 1.094mm for the fixed scroll, and it was 1.06mm for the orbiting scroll. Special care for this much large thermal expansion should be taken into account for detailed design procedure. Supporting frame and the fixed scroll were separated by a space for thermal expansion and keys were installed between them for transmitting forces from the fixed scroll to the supporting frame. Also flexible couplings were inserted between the orbiting scroll base plate and the drive bearing holders

4. PERFORMANCE ANALYSIS

Fig.8 shows the pressure curves for the expander and the compressor. Clearance between the two scroll members was assumed to be 50 μm for the expander and 30 μm for the compressor. Pressure was lower than the ideal value for the expansion process, while higher pressure for the compression process. Gas torque is shown in Fig.9. Torque variation is larger in the expander than in the compressor. Fig.10 shows shaft bearing forces and reactions on the keys of the fixed scroll for the expander. Maximum shaft bearing force reaches at 25 kN.

Loss breakdown was made at Fig.11. Power produced by the gas expansion was 54.14kW, and break shaft power was 45.35 kW due to expansion and mechanical losses. Power of 34.48kW was consumed to run the compressor, resulting in the net power output of 10.87kW for the designed Stirling engine. Table 2 shows the calculated efficiencies of the scroll expander and the compressor. Effect of polytropic index for the compression process is presented in Fig. 12, where adiabatic expansion process is assumed. With increasing the polytropic index, the net power out and the system efficiency decrease. System efficiency is 8.0% for isothermal compression, and it is 6.1% for adiabatic compression.

Table 2 Efficiencies of scroll expander and compressor

Expander		Compressor	
$\eta_{v,e}$	78 [%]	$\eta_{v,c}$	81.9 [%]
$\eta_{i,e}$	89.9 [%]	$\eta_{i,c}$	95.6 [%]
$\eta_{m,e}$	93.6 [%]	$\eta_{m,c}$	92.4 [%]
η_e	84.1 [%]	η_c	88.3 [%]

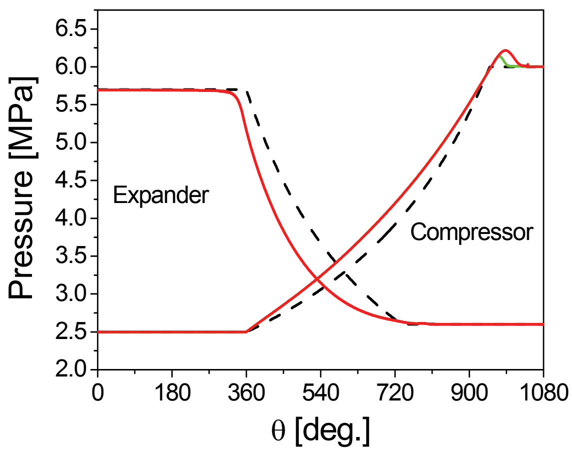


Fig. 8 P-θ diagram during expansion and compression processes

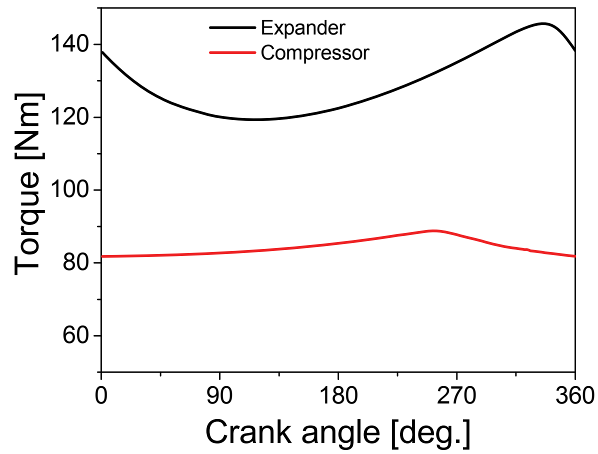


Fig.9 Gas torque variation for expander and compressor

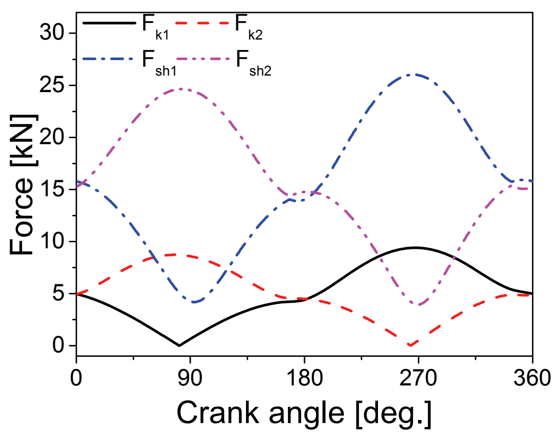


Fig. 10 Shaft bearing forced and key reactions for expander

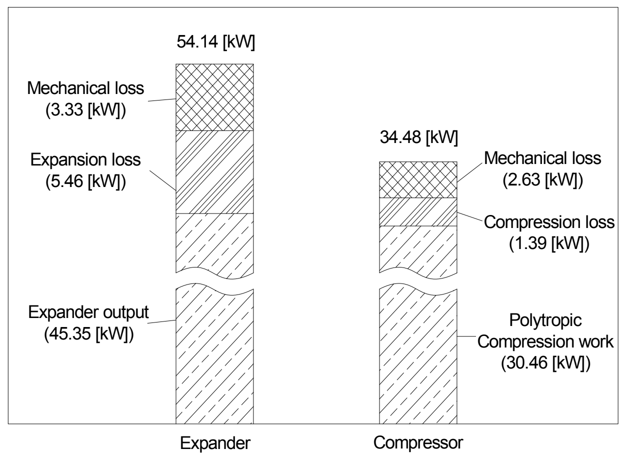


Fig. 11 Loss breakdown of expander and compressor

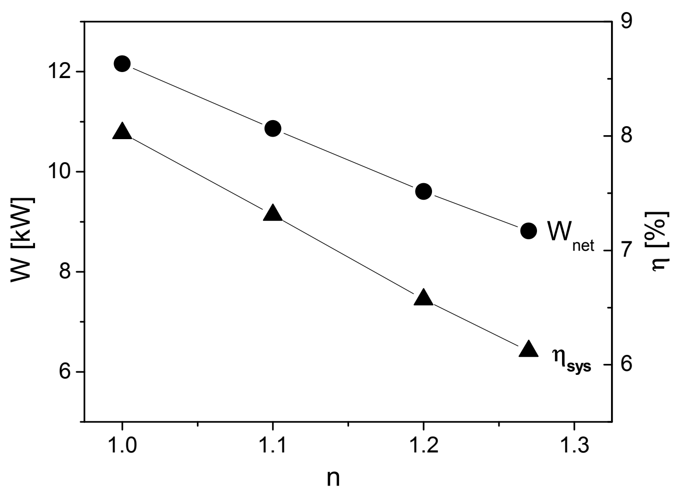


Fig. 12 Effect of polytropic index for compressor on net power output and system efficiency

5 CONCLUSIONS

In conceptual design of the 10kW-Stirling engine employing scroll expander and scroll compressor with solar energy as heat source,

- The efficiency of the Stirling cycle with CO_2 as a working fluid showed a maximum at the pressure ratio of 2.5, and the peak efficiency increased with high side pressure increment.
- Conceptually designed Stirling engine has the following characteristics: double-sided orbiting scroll structure for smaller size and for axial gas force cancellation; keys between the fixed scroll and the supporting frame for the thermal expansion; flexible coupling between the orbiting scroll base plate and the drive bearing holders; the expander and the compressor share the two common shafts with timing belts
- With design condition of t_H/ t_L =700°C/40°C and P_H/P_L=6 MPa /2.5 MPa and for the clearance between the two scroll members of 50 μm for the expander and 30 μm for the compressor, the expander and compressor efficiencies were calculated to be 84.1%, and 88.3%, respectively. Volumetric efficiencies were 78% and 81.9%, respectively. The cycle efficiency was about 7.3%.
- System efficiency can be increased from 6.1% to 8.0% by changing the compression process from adiabatic to isothermal, while maintaining the expansion process adiabatic.

REFERENCES

(1) Kim, Y. M., Shin, D. K., Lee, J. H., 2003, Noble Striling Cycle Employing the Scroll Compressor/Expander, *Proceeding of the SAREK 2003 Summer Annual Conference*, pp.911-918.

(2) Hugemroth, J., Braun, J., Groll, E., King, G, 2006, Liquid-flooded Ericsson cycle cooler: Part 1-Thermodynamic analysis, *International Refrigeration and Air Conditioning Conference at Purdue*, Paper-R168.

(3) Kwon, T. H., Ahn, J.M., Kim H.J., Shim J.H., 2008, Conceptual Design of the Scroll Air Compressor for Fuel Cell, *Journal of SAREK,* Vol. 20, No. 2, pp. 113-120.