Sensitivity Analysis of Key Parameters of Stirling Converter for Small Reactor

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1. Introduction

Since its first invent in 1816, the Stirling engine has been developed to be used as a heat engine for various purposes. One of the promising power conversion systems for small or micro reactors is the combination of the heat pipe and the Stirling engine, such as HOMER series[1], LEGO-LRCs[2], KRUSTY[3] and etc.
The ideal Stirling cycle consists of two isochoric and two isotherms as seen in Fig. 1 and it is similar to the Carnot cycle. The absorbed heat at 1-2 goes through a regenerator at 2-3 and the heat is emitted at 3-4 then it goes through the regenerator again at 4-1[4].

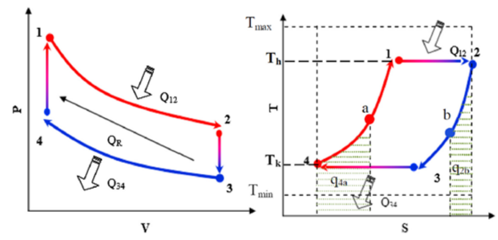

Fig. 1 P-V and T-S Diagram of Ideal Stirling Cycle[4]

To design a Stirling converter, the key parameters which determine the performance(power and efficiency) needs to be identified and analyzed. In this paper, the key design values for maximizing the performance were described and its sensitivity was evaluated.

2. Thermodynamic Analysis

For comprehensive analysis, the heat transfer of a regenerator must be addressed. The regenerative heat transfer and the heat loss during regeneration per cycle can be defined as follows[5].

$$Q_R = nC_p\varepsilon_R(T_h - T_c) \qquad (1)$$
$$\Delta Q_R = nC_p(1-\varepsilon_R)(T_h - T_c) \qquad (2)$$

where n = molar number of fluid
C_p = specific heat
ε_R = regenerator effectiveness

The piston displacement phase is determined by the time of heat transfer and this can be calculated from a regeneration time constant in finite-time thermo-dynamics[6]. The time of two isochoric processes can be described by the temperature change and a constant independent of the temperature difference but only on the property of regenerative material.

$$t_2 = {(T_h - T_c)}/{M_1} \qquad (3)$$
$$t_4 = {(T_h - T_c)}/{M_2} \qquad (4)$$
$$\pm M_i = {dT}/{dt} \qquad (5)$$

where t_j = time of each process
M_i = constant (i = 1, cooling, i = 2, heating)

The heat transfer at hot-side of heat exchanger to the working fluid and the heat rejected from the working fluid at cold-side are under convection law. Therefore, the heat absorbed and rejected by the working fluid can be described as follows.

$$\begin{aligned} Q_h &= h_h(T_{in} - T_h)t_1 \\ &= nRT_h \ln\lambda + nC_p(1-\varepsilon_R)(T_h - T_c) \qquad (6) \\ Q_c &= h_c(T_c - T_{out})t_3 \\ &= nRT_c \ln\lambda + nC_p(1-\varepsilon_R)(T_h - T_c) \qquad (7) \end{aligned}$$

where h_h = heat transfer coefficient at hot-side
h_c = heat transfer coefficient at cold-side
t_j = time of each process
R = gas constant
λ = compression ratio ($\lambda = V_1/V_2$)

The overall cycle period can be calculated as follows from Eq. (3)-(7).

$$\begin{aligned} \tau = & \; t_1 + t_2 + t_3 + t_4 \\ & = \frac{nRT_h \ln\lambda + nC_p(1-\varepsilon_R)(T_h - T_c)}{h_h(T_{in} - T_h)} \\ & + \frac{nRT_c \ln\lambda + nC_p(1-\varepsilon_R)(T_h - T_c)}{h_c(T_c - T_{out})} \\ & + (T_h - T_c)\left(\frac{1}{M_1} + \frac{1}{M_2}\right) \qquad (8) \end{aligned}$$

The heat loss due to the conduction through structure of heat exchanger is proportional to the cycle time.

$$Q_{loss} = k_{loss}\tau(T_{in} - T_{out}) \qquad (9)$$

where k_{loss} = heat leak coefficient

The total heat transferred in and out will can be defined as follows.

$$Q_{in} = Q_h + Q_{loss} \qquad (10)$$
$$Q_{out} = Q_c + Q_{loss} \qquad (11)$$

Accordingly, the power and thermal efficiency can be calculated as follows.

$$P = \frac{W}{\tau} = \frac{Q_{in}-Q_{out}}{\tau} = \frac{T_h - T_c}{\frac{T_h + a(T_h-T_c)}{h_h(T_{in}-T_h)} + \frac{T_c + a(T_h - T_c)}{h_c(T_c - T_{out})} + F(T_h - T_c)}$$

$$\eta = \frac{Q_{in}-Q_{out}}{Q_{in}} = \frac{T_h - T_c}{T_h + a(T_h-T_c) + [k_{loss}(T_{in}-T_{out})][\frac{T_h + a(T_h-T_c)}{h_h(T_{in}-T_h)} + \frac{T_c + a(T_h - T_c)}{h_c(T_c - T_{out})} + F(T_h - T_c)]}$$

where $a = \frac{C_p(1-\varepsilon_R)}{R\ln\lambda}$

$F = \frac{1}{nR\ln\lambda}(\frac{1}{M_1} + \frac{1}{M_2})$

Therefore, the design values were put into the above equations to maximize the power and thermal efficiency.

3. Sensitivity of Performance Parameter

The sensitivity of each parameter was evaluated to investigate the optimal value. The basic starting values for the parameters are as follows.

T_{in} = 600°C, T_{out} = 20°C, $h_h = h_c$ = 200 W/K, τ = 0.1 s, k_{loss} = 2.5 W/K, n = 1 mol, C_p = 20.79 J/mol-K, ε_R = 0.9, R = 8.314 J/mol-K, λ = 2.

3.1 Hot-side Temperature (T_{in})

The effect of hot-side temperature is shown in Fig. 2. The maximized efficiency occurs at about 500~600°C and the corresponding power increases as the temperature increase.

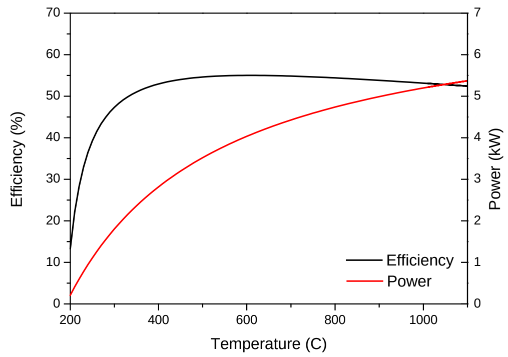

Fig. 2 Hot-side Temperature Effect

3.2 Heat Transfer Coefficient (h_h and h_c)

The effect of the heat transfer coefficient is shown in Fig. 3. With the increasing value, the power increases but the efficiency seems to saturate at certain point.

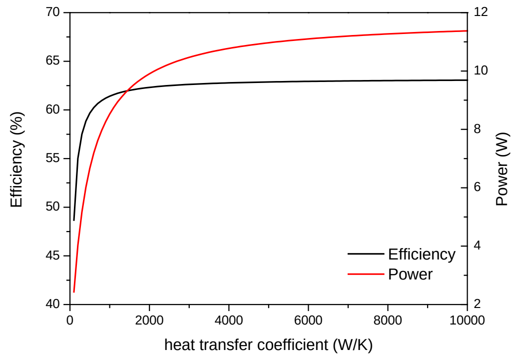

Fig. 3 Heat Transfer Coefficient Effect

3.3 Cold-side Temperature (T_{out})

The effect of cold-side temperature is shown in Fig. 4. The power and efficiency linearly drops down with the increasing temperature. It is clear that the colder temperature for cooling is the better.

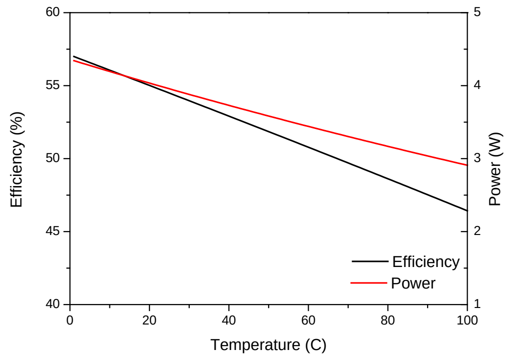

Fig. 4 Cold-side Temperature Effect

3.4 Regenerator Performance (k_{loss} and ε_R)

The performance of regenerator is defined by two parameters, which are the heat loss coefficient by conduction and the effectiveness of the regenerator. In Fig. 5, the effect of conductive heat loss is shown and in Fig. 6, the effect of the effectiveness is shown. The conductive heat loss only influences the efficiency and the effectiveness increases both the power and efficiency. Therefore, the conductive heat loss should be as small as possible and the effectiveness should be as close to 1.0 as possible.

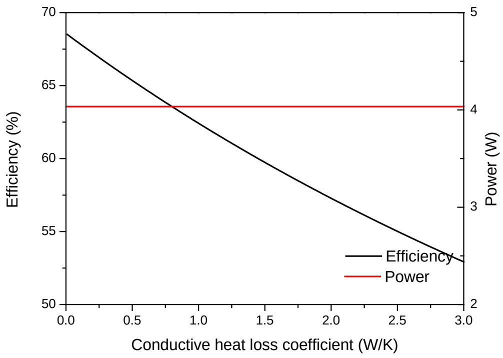

Fig. 5 Conductive Heat Loss Coefficient Effect

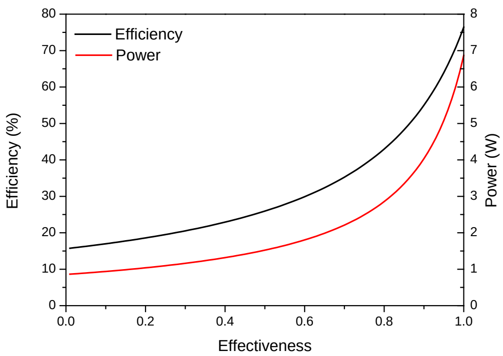

Fig. 6 Effectiveness of Regenerator Effect

3.5 Compression Ratio (λ)

The effect of compression ratio is shown in Fig. 7. As the compression ratio increases, the power and efficiency both increase. It is better to have higher compression ratio as long as allowable.

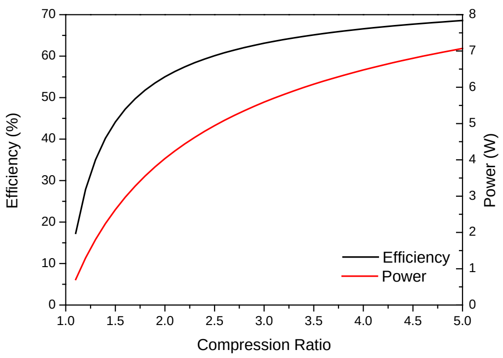

Fig. 7 Compression Ratio Effect

3.6 Cycle Period (τ)

The effect of cycle time is shown in Fig. 8. As the time increases the power and efficiency dramatically drops. This is why the Stirling cycle period is usually kept as short as possible. In this design, 10 Hz is selected but in general it is much faster than this.

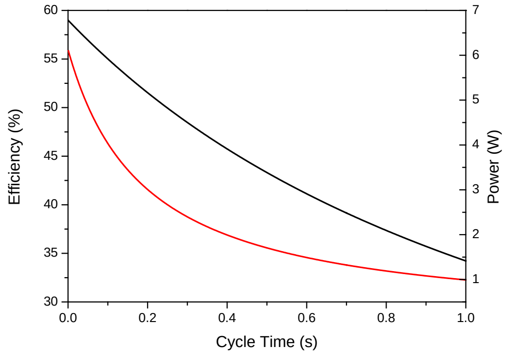

Fig. 8 Cycle Time Effect

4. Discussion

The basic layout of the Stirling converter was determined to be the free piston β-type and helium was selected to be the working gas for its desirable performance, although it has some drawbacks such as the sealing problem. The overall evaluation result clearly indicates the helium is the most promising working fluid[4].

In this analysis, the ideal gas was assumed but in the real application the real gas effect may appear. This means that the ideal gas law ($PV=nRT$) doesn't work anymore. This is also one of the reason why the helium is preferred as a working gas. The behavior of helium is the closest to the ideal gas.

Qualitatively, the real gas effect can be evaluated according to the pressure condition. At low pressure, the power and efficiency will decrease but at high pressure condition (depends on which gas is under consideration, usually over 400 atm), the power and efficiency will increase. Therefore, the inside pressure of the Stirling converter needs to be high to assure a good performance.

5. Conclusion

The Stirling converter system is considered as a tempting option for the power conversion system of small reactors owing to its various advantages. As a first step of designing an appropriate Stirling converter, the key design parameters were identified based on thermodynamic analysis of the power and efficiency. Each parameter was evaluated by sensitivity test and the results are; (1) the hot-side temperature has an optimum point, (2) the heat transfer coefficient effect saturates after certain point, (3) the cold-side temperature should be as low as possible, (4) the conductive heat loss needs to be as low as possible and the effectiveness should be as close as to 1.0, (5) the compression ratio should be highest allowable value, and (6) the cycle time needs to be as short as possible.

For further development and detailed design, these parameters should be evaluated with the actual structure and dimensions.

ACKNOWLEDGEMENT

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REFERENCES

[1] Poston D., "Nuclear Design of the HOMER-15 Mars Surface Fission Reactor", International Congress on Advances in Nuclear Power Plants (ICAPP), Hollywood FL, US, 9-13 Jun, 2002
[2] Bess J.D., "A Basic LEGO Reactor Design for the Provision of Lunar Surface Power", International Congress on Advances in Nuclear Power Plants (ICAPP), Anaheim CA, US, 8-12 Jun, US, 2008
[3] Briggs M. and Gibson M., "Electrically Heated Testing of the Kilowatt Reactor Using Stirling Technology (KRUSTY) Experiment Using a Depleted Uranium Core", NASA Report, NASA/TM-2018-219702, 2018
[4] Hachem H., et al., "Technological challenges and optimization efforts of the Stirling machine: A review", Energy Conversion and Management, Vol. 171, pp. 1365-1387, 2018
[5] Ahmadi M.H., et al., "Designing a sloar powered Stirling heat engine based on multiple criteria: maximized thermal efficiency and power", Energy Conversion and Management, Vol. 75, pp.282-291, 2013
[6] Yaqi L., et al., "Optimization of solar-powered Stirling heat engine with finite-time thermodynamics", Renewable Energy, Vol. 36, pp. 421-427, 2011