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ABSTRACT

Scroll compressor has been used extensively for refrigeration since the early 1980's for its improved efficiency, greater reliability, smoother operation, lower noise and vibration. And also, nowadays, scroll mechanism is used for expander even though in niche market. But scroll expander has not been used for high-temperature and high-pressure gas, because the continuous expansion of the gas causes a wide range of temperature distribution over the whole scroll wrap that leads to differential thermal expansion of scroll elements, which results in system vibrations, noise and efficiency losses. For the scroll expander to produce power more efficiently, all of radial and axial clearances between scroll wrap must be the same. In order to reduce differential thermal expansion in addition to improvements in thermal efficiency and specific power, we propose a scroll expander with heating structure. Heat-pipe heating structure is considered as the most effective method to heat the scroll expander at a uniform temperature. This paper includes some results of preliminary study of the scroll expander with heating structure and proposals of their systems for power generation and refrigeration.

1. INTRODUCTION

Recently some companies are trying to commercialize the scroll expander for domestic cogeneration system, in which system, as a Rankine cycle system, the scroll expander is used instead of the steam turbine. The domestic cogeneration market has a potentially great growth in the near future. But until now, because any scroll expander has not been used for expansion of high-temperature and high-pressure gas and it is used in the low temperature range around 200C, the thermal efficiency of their systems is much lower than other prime movers. The main reason for that is differential thermal expansion of scroll elements due to the temperature drop of the gas in the scroll expander, which causes different clearances in radial and axial directions and results in system vibrations and efficiency losses. This paper proposes a scroll expander with external heating structure for the purpose of reducing the differential thermal expansion. And that contributes to improvements in thermal efficiency and specific power due to a reheat process. As well as for heat engine, it can be used as a replacement to the traditional throttle in refrigeration system by absorbing the heat from the refrigerated space. And then it is possible to recover more power from the expansion process and absorb more heat from the refrigerated space. Some applications of the proposed scroll expander for power generation and refrigeration system will be presented and their effects on the system will be discussed briefly.

2. A SCROLL EXPANDER WITH HEATING STRUCTURE

If the scroll expander is used for high-temperature and high-pressure gas, the continuous expansion of the gas causes a wide range of temperature distribution over the whole scroll wrap that leads to differential thermal expansion of scroll elements, which causes different clearances in radial and axial directions. For the purpose of reducing the differential thermal expansion we propose a scroll expander with external heating structure. In the process of expansion, if heat is added to the working fluid in the expansion space from an external source through the external heating fins on the expander casing and the inner scroll wrap, the fall in temperature of the working fluid will be reduced, that also contributes to improvements in thermal efficiency and specific power. To reduce the differential thermal expansion, it is desirable to heat the scroll expander as uniformly as possible so as to maintain the same temperature over the whole scroll wrap. But it is not easy to heat the scroll expander uniformly by direct heating by flames. So we preferably propose a flame-heated heat pipe coupled to the external fins of a double-sided scroll

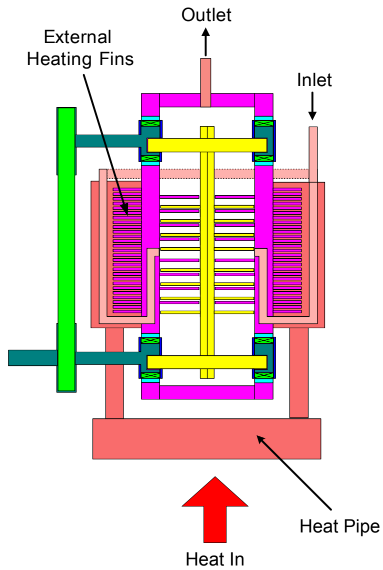

Figure 1: Scroll expander with heating structure

expander as the best way as shown in Fig. 1. The unique properties of the heat-pipe are to a large extent due to the very high rates of heat transfer by the evaporation-condensation processes at solid surfaces, far higher than the heat transfer between a gas and a solid surfaces, even with forced convection. These unique properties may be put to work as follows. A sealed chamber is constructed over the existing heating fins of the scroll expander as shown in Fig. 1. The chamber contains the heat-pipe fluid and part of its surface can be heated by the flames from a burner, or otherwise. And the flame-heated surface can be made arbitrarily large; all the heat passing through this large area cause evaporation of the heat-pipe medium which practically all condenses on the heating fins because these are the coldest objects present. The condensed fluid drops from the tubes and is transported by gravity and/or capillary action back to the large heating surface. With the heat-pipe it is possible to heat the scroll expander completely uniformly due to the most important feature of the heat-pipe. If for any reason any place of the heating fins over the external surfaces of the scroll wrap should become hotter than the rest, less vapour condenses there so that its temperature drops. If there is a cold spot, more vapour condenses to return its temperature to the norm. Temperature regulation, or at least equalization, by the laws of nature. What fluids can be considered for use in the heat-pipe can be determined according to the operating temperature. As well as for heat engine, the proposed scroll expander with heat-absorbing structure can be used as a replacement to the traditional throttle in refrigeration system.

3. CHARATERISTICS OF THE PROPOSED SCROLL EXPANDER

3.1 Isothermality of scroll compressor

First of all, it would be possible to predict the isothermality of scroll expander, as the ability to absorb heat through its surfaces, by investigating that of the scroll compressor instead. In the commercial scroll compressor used for oil-free air compressor, it is necessary to cool the compressor effectively in order to prevent excessive rise in temperatures for higher component life and an energy savings. In a case of commercial oil-free scroll compressor used for air compressor, of which volumetric compression ratio (Rc) is around 8:1, it is presented in the brochure that outlet temperature of compressed air is at the temperature about 30C higher than a inlet temperature by effective cooling. But actually, the rise in temperatures in the compressor is somewhat higher than is presented in the brochure. The compression processes, in general, can be modeled as polytropic (Pv^n = constant) process where the value of n varies between k and 1: an isentropic (n = k) process (involves no cooling), an polytropic (k < n <1) process (involves some cooling), and isothermal (n = 1) process (involves plenty of cooling). To estimate the extent of cooling of the compressor with Rc = 8 the theoretical results for different processes with the value of n_c, defined as polytropic index of compression, are obtained by the analysis of the ideal gas process with air. The inlet state of air was assumed at the same state of 50C and 0.1MPa. The outlet temperature in the compressor (Tc-out) and the heat (per unit mass) rejected during compression process (Qcomp) are given in Fig. 2, and the outlet pressure in the compressor (Pc-out) and the work (per unit mass) required to compress the air (Wcomp) are given in Fig. 3 as a function of n_c. In the adiabatic (n_c = k = 1.4) case, Tc-out goes up to 470C from inlet temperature 50C , and Pc-out goes up to 1.84MPa from inlet pressure 0.1MPa, and 420.9 kJ/kg of Wcomp is required. For the isothermal (n_c = 1.0) compression it is necessary to reject 192.8kJ/kg of Qcomp. And in that case, Pc-out goes up to 0.8MPa much less than that of adiabatic process and 192.8 kJ/kg of Wcomp, which is less than the half of that in the adiabatic case, is required. The case in the brochure (with Tc-out 30C higher than inlet temperature) corresponds to the process with the value n_c = 1.043. And in that case, it is necessary to reject 180.1kJ/kg of Qcomp. That is a fairly satisfactory result for the isothermal compression. From the result, we can convince that it is possible to reject a considerable quantity of heat during the compression process through the extensive surface area of scroll wrap contacting the working fluid in the scroll compressor.

3.2 Effects of isothermality of scroll expander on the heat engine

From the previous result about the isothermality of scroll compressor, inversely in the case of scroll expander, it is supposed that a considerable quantity of heat can be added to the working fluid through the scroll wrap. For the purpose of investigating the effects of isothermality of scroll expander on the heat engine, first of all, we propose a Ericsson cycle engine, one of the most efficient cycle, comprising a scroll-type isothermal compressor and a scroll-type isothermal scroll expander and heat exchangers as shown in Fig. 5. In the process of compression, the heat generated in the compression space is carried to the external dump through the inner scroll wrap and external cooling fins. And the compressed working fluid enters the inlet port of the scroll expander. In the process of expansion, heat is added to the working fluid in the expansion space from an external heat source through external heating fins and the inner scroll wrap. Expanded hot and compressed cold working fluid streams enter the counter-flow heat exchanger from opposite ends, and heat transfer takes place between them. But as even in the scroll-type

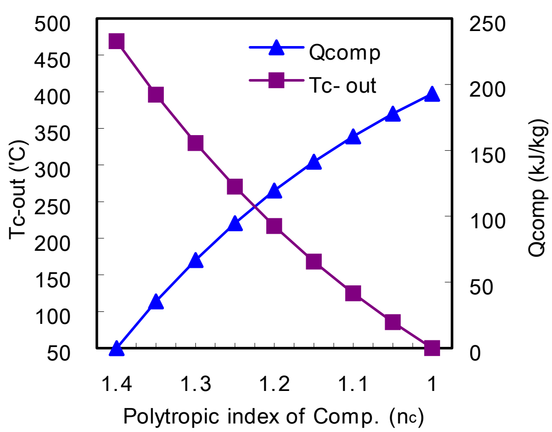

Figure 2: Tc-out and Qcomp versus n_c (Rc=8)

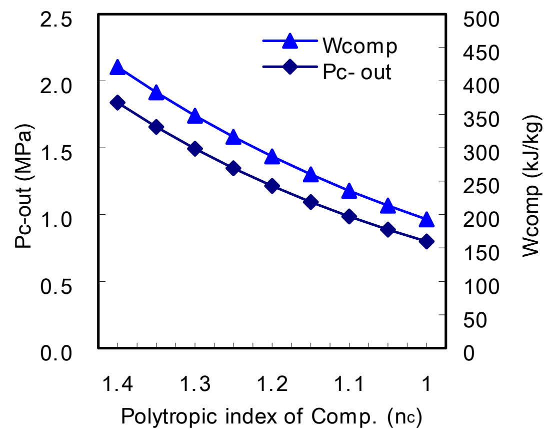

Figure 3: Pc-out and Wcomp versus n_c (Rc=8)

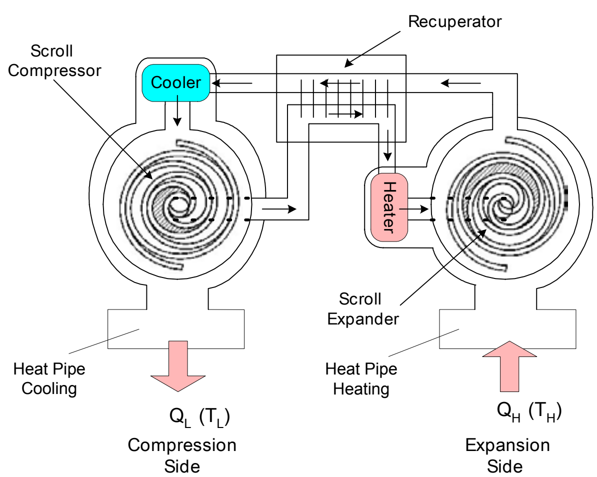

Figure 4: Scroll-type Ericsson cycle for heat engine

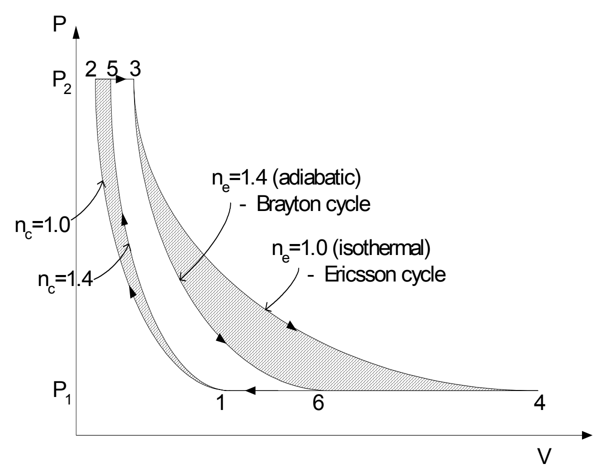

Figure 5: Brayton cycle and Ericsson cycle (P-V diagram)

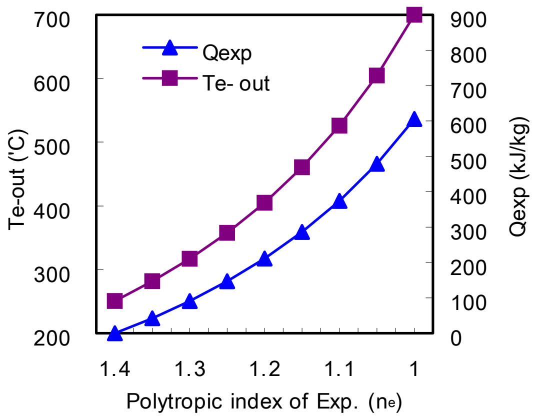

Figure 6: Te-out and Qexp versus n_e

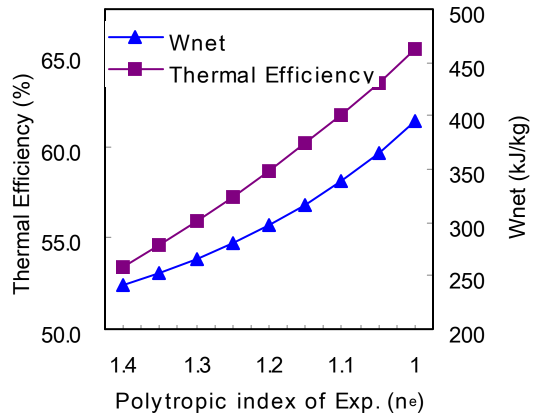

Figure 7: Cycle efficiency and Wnet versus n_e (n_c=1.043)

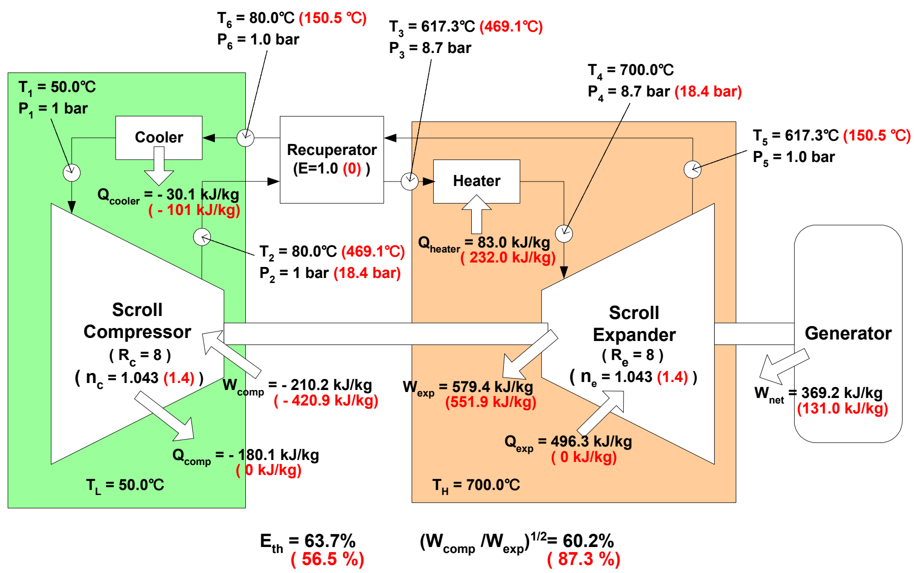

Figure 8: Scroll-type Ericsson cycle engine in comparison with Brayton cycle (Rc=8, Re=8)

Ericsson cycle engine the processes of compression and expansion are not perfectly isothermal, it may have remote heater and cooler as shown in Fig. 4. A comparison of P-V diagrams of the Ericsson cycle with the Brayton cycle (with adiabatic processes), between given limits of pressure and temperature, is shown on Fig. 5. The shaded areas 1-2-5 (decrease in compression work) and 3-4-6 (increase in expansion work) represent the additional work made available by substituting isothermal processes for adiabatic processes. The more close to isothermal process the processes of compression and expansion become, the smaller size of heater and cooler is needed and the less portion of heat transfer takes place in them. For the purpose of investigating the effects of isothermality of scroll expander on the Ericsson cycle engine, the high temperature $T_H = T_3 = 700C$ and the low temperature $T_L = T_1 = 50C$ were assumed, and air (k=1.4, R=0.287 kJ/kg·K), as a working fluid, has been selected for the analysis. And the scroll compressor with Rc=8 and n_c = 1.043, that is equivalent to the previous commercial scroll compressor used for air compressor, was assumed. All of heat exchangers including a recuperator were assumed as ideally perfect: the 1.0 of efficiency and no pressure drop. The outlet pressure in the scroll expander is the same as the inlet pressure in the scroll compressor. The theoretical results for different process with the value of n_e, defined as polytropic index of expansion, are obtained by the analysis of the ideal gas cycle. In the Fig. 6, the outlet temperature in the expander (Te-out) and the heat (per unit mass) supplied during the expansion process (Qexp) are given, and in Fig. 7, the net work (per unit mass) of the cycle (Wnet), difference between the expansion work and the compression work, and the thermal efficiency of the cycle are given as a function of n_e. In the adiabatic (n_e = k = 1.4) case, Te-out goes down to

250C from inlet temperature 700C and 451.1 kJ/kg of Wexp is produced. For the isothermal (n_e = 1) expansion it is necessary to supply 605.4 kJ/kg of Qexp. And in that case, 605.4 kJ/kg of Wexp, which is much larger than that in the adiabatic case, is produced. The isothermality of scroll expander in the cycle can result in 22.3% of increase in the thermal efficiency and 64.1% of increase in Wnet (per unit mass) as shown in the Fig. 7.

If we apply the same value of n_e = 1.043 as n_c = 1.043, the theoretical result can be obtained as shown in Fg. 8. The values in parentheses in the Fig. 8 represent the values in the adiabatic case (n_e = 1.4, n_c = 1.4) to show the cooling and heating effect obviously. In that case with the value of n_e = 1.043, Te-out goes down to 617C from inlet temperature 700C and it is necessary to supply 496.3 kJ/kg of Qexp, which is about 2.8 times larger than Qcomp. Here, we wonder if it can be practically possible to supply such quantity of heat through the scroll wrap. But luckily it is expected that it can be possible to supply such quantity of heat because the volumetric capacity of the expander is 2.8 times larger than that of the compressor (that makes the surface area contacting the working fluid 2.0 times larger than that of the compressor) and the temperature difference between the working fluid and heat source in the expander is 2.8 times larger than that of the compressor. But if we increase the pressure of the working fluid in the closed system to improve the specific power of the Ericsson cycle engine, it may be needed to improve the heat transfer rate through the scroll wrap.

3.3 Effects of isothermality of scroll expander on the refrigeration cycle

The reverse Ericsson cycle, like the reverse Brayton cycle well known, can be used for a refrigerator (or heat pump) if drive power is input then heat can be absorbed into the expansion space at a temperature lower than that of the compression space as shown in Fig. 9. A comparison of P-V diagrams of the reverse Ericsson cycle and the reverse Brayton cycle, between given limits of pressure and temperature, is shown on Fig. 10. The shaded areas 1-2-5 (decrease in compression work) and 3-4-6 (increase in expansion work) represent the reduced work for energy savings by substituting isothermal processes for adiabatic processes. For the purpose of investigating the effects of isothermality of scroll expander on the reverse Ericsson cycle refrigerator, high temperature $T_H = T_1$ = 54.4C and low temperature $T_L = T_3$= -23.3C were assumed. The isothermality of scroll compressor in the reverse Ericsson cycle refrigerator is the same as mentioned before in the Ericsson cycle engine, and the scroll compressor with Rc = 8 and n_c = 1.043 was assumed. The theoretical results for different process with the value of n_e, polytropic index of expansion, are obtained by the analysis of the ideal gas cycle. In the Fig. 11, the outlet temperature in the expander (Te-out) and the heat (per unit mass) absorbed during expansion process (Qexp) are given as a function of n_e. The volumetric expansion ratio of scroll expander has been matched to the inlet pressure of the compressor in all those cases. In the Fig. 12, Q_L (per unit mass), defined as heat removed from the refrigerated space at temperature T_L through the expander in itself and the chiller, and Wnet (per unit mass) of the cycle, difference between the expansion work and the compression work. In the adiabatic (n_e = k = 1.4) case, Te-out goes down to –138.5C from inlet temperature –23.3C and 115.8 kJ/kg of Wexp is produced. For the isothermal (n_e = 1) expansion, it is necessary to absorb 605.4 kJ/kg of Qexp. And in that case, 155.4 kJ/kg of Wexp, which is much larger than that in the adiabatic case, is produced. The isothermality of scroll expander of the cycle can result in 34.2% of increase in Q_L in spite of 40.7% of decrease in Wnet as shown in the Fig. 12, which results in 126.0% of increase in the COP.

In we apply the same value of n_e = 1.043 as n_c = 1.043, the theoretical result can be obtained as shown in Fig. 13. The values in parentheses in the Fig. 13 represent the values in the adiabatic case (n_e = 1.4, n_c = 1.4) to show the cooling and heating effect obviously. In that case with n_e = 1.043, Te-out goes down to –44.5C from inlet temperature –23.3C, and it is necessary to absorb 127.4 kJ/kg of Qexp, which is about 70% of Qcomp.

4. OTHER APPLICATION FIELDS OF THE SCROLL EXPANDER

4.1 Reheat and regenerative Rankine cycle engine for domestic cogeneration

In a power plant operating on Rankine cycle as shown in Fig. 14, reheating is a practical solution to improve the cycle efficiency. As the number of stages of reheat process increases, the expansion and reheat processes approach an isothermal processes at the maximum temperature and the cycle efficiency increases. But the reheat process inevitably adds the complexity of turbomachinery and need many additional components. Another practical solution to improve the cycle efficiency is a regeneration process by extracting steam from the turbine at various points and using it to heat the feedwater in a feedwater heater as shown in Fig. 15. Like the reheat process, as the number of stages of regeneration increases, the cycle efficiency increases. But it also adds the complexity of the machine.

At present, some companies are trying to commercialize a small size Rankine cycle engine with a scroll expander for a domestic cogen system. But, as previously stated, the thermal efficiency of their system is much lower than other prime movers because the scroll expander is used in the range of low temperature. When the proposed scroll

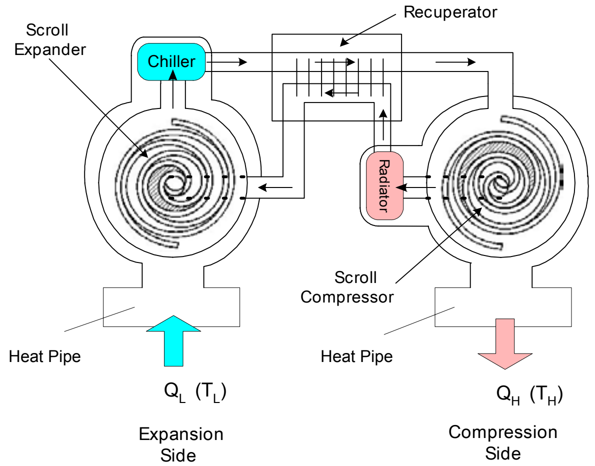

Figure 9: The reverse Ericsson cycle for refrigeration

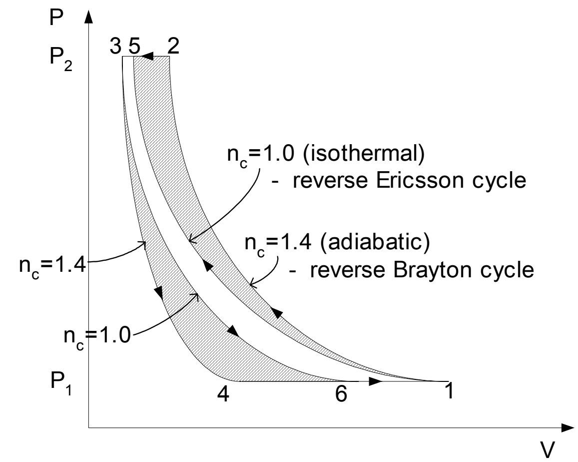

Figure 10: reverse Ericsson and reverse Brayton cycle

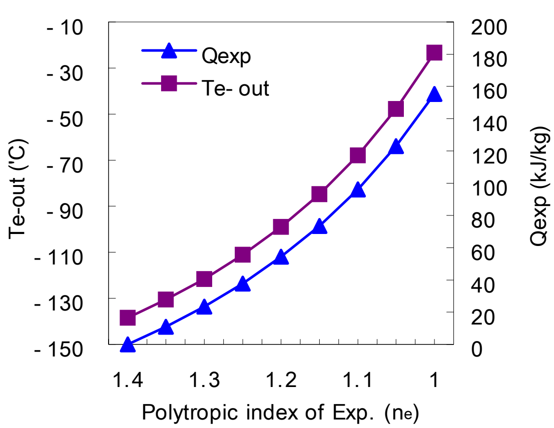

Figure 11: Te-out and Qexp versus n_e in reverse Ericsson refrigerator

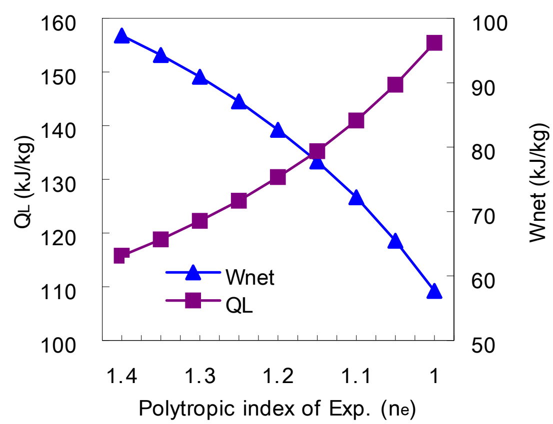

Figure 12: Q_L and Wnet versus n_e in reverse Ericsson refrigerator

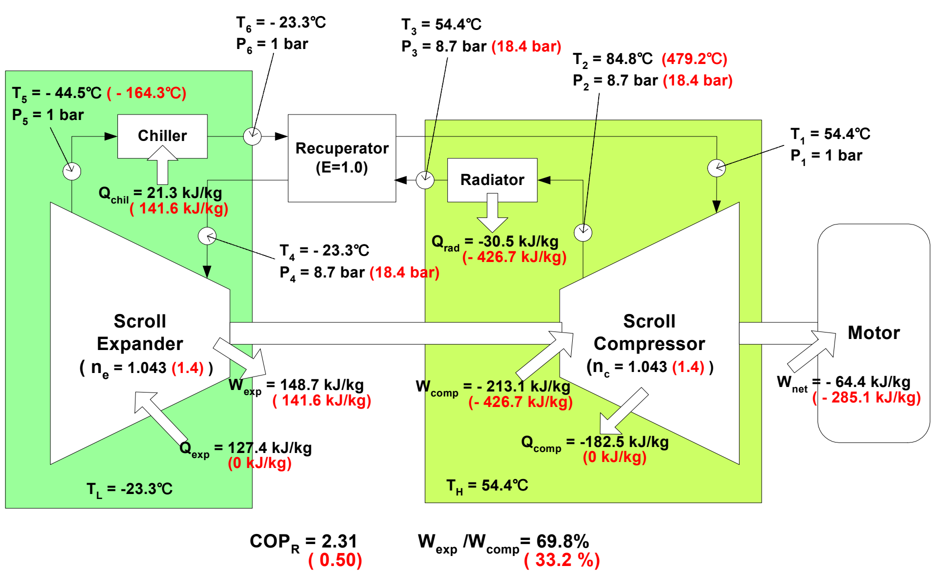

Figure 13: Scroll-type reverse Ericsson cycle refrigerator in comparison with reverse Brayton cycle (Rc=8, Re=8)

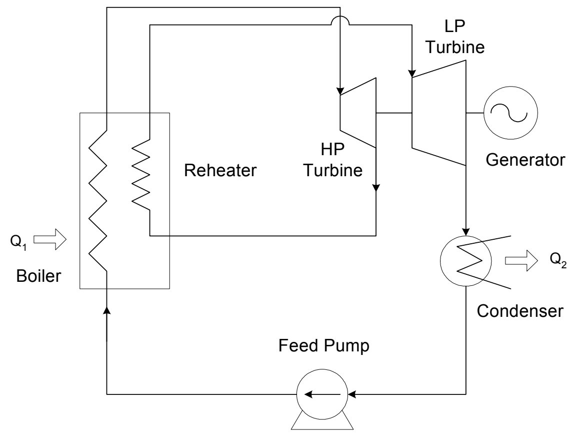

Figure 14: The two-stage reheat Rankine cycle

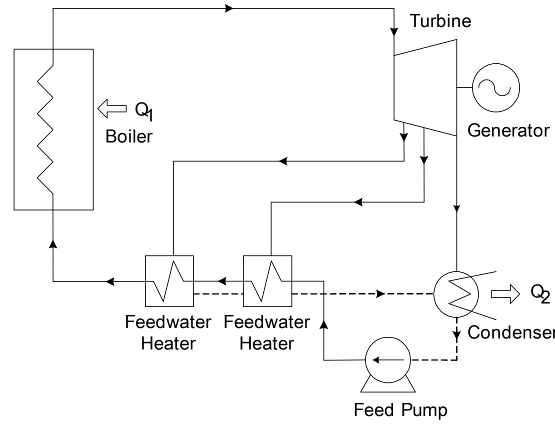

Figure 15: The two-stage regenerative Rankine cycle

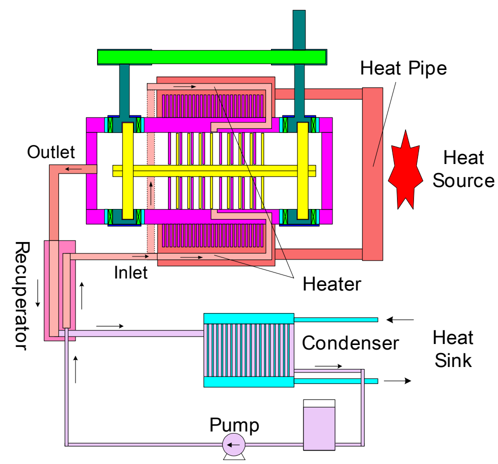

Figure 16: Reheat and regenerative Rankine cycle with a heated scroll expander

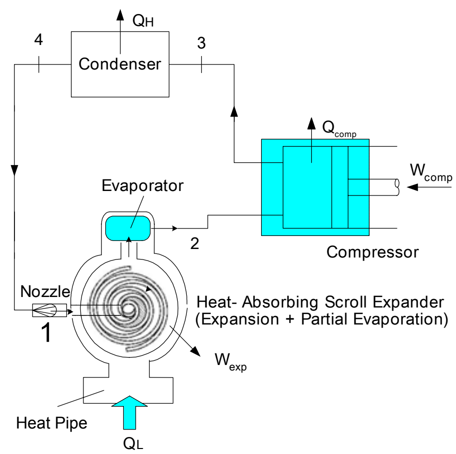

Figure 17: The use of heat-absorbing scroll expander in the refrigeration system

expander with a heating structure is applied in the Rankine cycle engine as shown in Fig. 16, the expanded hot steam is used to heat feed water from the water pump in a counter-flow heat exchanger called a 'recuperator' to improve the thermal efficiency and it is possible to increase the operating temperature of scroll expander by dissolving the differential thermal expansion of scroll elements so as to improve the thermal efficiency. Moreover both reheat process by heating scroll expander in itself and regeneration process by recuperator contribute to improve the thermal efficiency and the specific power without added complexity of machines as with a turbomachinery. And also, it is possible to use the phase change component because scroll mechanism is very effective in applications where substantial quantities of liquid enter the machine. In that case, the evaporation of liquid in the scroll expander due to the heat supplied from the heat source tends to slow the fall in pressure during the expansion process even beyond the isothermal expansion. And then if the scroll expander yields a high volumetric expansion ratio, that contributes to extract a greater amount of work resulting from the volumetric expansion due to the phase change. In that case, it can be said that the scroll expander acts as a expander and partially as a boiler.

4.2 Expansion devices for recovering power in the refrigeration system

It is well known that the use of expander as a replacement to the traditional throttle valve in vapor compression (V-C) cycle air conditioning and refrigeration systems would improve the cooling capacity and performance and reduce operating costs by recovering power from the energy wasted in the two-phase expansion process. With the proposed scroll expander capable of absorbing heat from the refrigerated space, it can be possible to recover more power from expansion process and absorb more heat from the refrigerated space, because the expansion process with it approaches isothermal process preferable to general isentropic process. Moreover, although the inherent disadvantage of V-C machines is that the saturation conditions of the working fluid define the operating pressure required for the refrigeration system at any desired operating temperature, if the proposed scroll expander is used in the two-phase expansion as shown in Fig. 17 and the working fluid is expanded down to below the saturation pressure, the evaporation of liquid in the scroll expander due to the heat supplied from the refrigerated space tends to slow the fall in pressure during the expansion, that needs for a higher volumetric expansion ratio, and then contributes to produce more power and improve cooling capacity resulting from the volumetric expansion due to the phase change. In that case, it can be said that the scroll expander acts as a expansion device and partially as a evaporator.

4. CONCLUSIONS

This paper proposes a scroll expander with external heating structure for the purpose of reducing the differential thermal expansion of scroll element due to the temperature drop of expanding gas. It can be used both in heat engine and in refrigeration system. Characteristics of the proposed scroll expander and their systems are summarized as follows.

- We preferably propose a heat-pipe coupled to the external heating fins of the scroll expander as the best way to heat the scroll expander uniformly. What fluids can be considered for use in the heat-pipe can be determined according to the operating temperature.
- From the preliminary study on the isothermality of scroll compressor, reversely in the scroll expander it is supposed that a considerable quantity of heat can be added to the working fluids through the scroll wrap.
- If the proposed scroll expander together with a recuperator is used in a heat engine, it contributes to improvements in thermal efficiency and specific power of the heat engine due to the reheat process and regeneration process.
- If the proposed scroll expander is used in refrigeration system, it can be possible to recover more power from expansion process and absorb more heat from the refrigerated space. If it is used in the two-phase expansion and the working fluid is expanded down to below the saturation pressure, the evaporation of liquid in the scroll expander due to the heat supplied from the refrigerated space tends to slow the fall in pressure during the expansion process, that needs for a higher volumetric expansion ratio, and then contributes to produce more power and improve cooling capacity.

NOMENCLATURE

n_c , n_e	polytropic index of compression , expansion (–)
T_H , T_L	temperature of high temperature reservoir , low temperature reservoir (C)
Tc-out , Te-out	outlet temperature of compressor , expander (C)
Pc-out	outlet pressure of compressor (MPa)
Q_H , Q_L	heat transfer to high temperature reservoir , low temperature reservoir (kJ/kg)
Qcomp, Qexp	heat rejected during compression , supplied during expansion (kJ/kg)
Wcomp, Wexp	work required during compression , produced during expansion (kJ/kg)
Wnet	net work of the cycle (kJ/kg)
R_c , R_e	volumetric compression ratio , expansion ratio (–)

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