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A New Ericsson Cycle Comprising a Scroll Expander and a Scroll Compressor for Power and Refrigeration Applications

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ABSTRACT

The Ericsson cycle is very much like the Stirling cycle, except that the processes of constant-volume regenerative heat transfer are replaced by constant-pressure regenerative processes. The efficiency of the Ericsson cycle is the same as that of the Carnot cycle but, as in the Stirling cycle, the net available work and the quantities of heat transferred are much greater, for given limits of pressure, volume, and temperature. In the Brayton cycle with turbomachinery, as the number of compression and expansion stages is increased, the Brayton cycle with intercooling, reheating, and regeneration will approach the Ericsson cycle and the thermal efficiency will approach the theoretical limit (the Carnot efficiency). Although the Ericsson cycle is often considered, it is usually abandoned because the added complexity of the turbomachinery and the need for many additional components, discourage its use. But in the orbiting scroll machine as a positive displacement type of machinery, it is possible to reheat the working fluid in the process of expansion and intercooling in the process of compression effectively due to the extensive surface area of the scroll wrap contacting the fluid in the working spaces. In this paper, we propose the Ericsson cycle comprising a scroll expander with heating system and a scroll compressor with cooling system. And also, heating of scroll expander uniformly can reduce differential thermal expansion over the whole scroll wrap which is critical problem in the scroll expander for high-temperature and high-pressure gas.

1. INTRODUCTION

External combustion engine including Stirling engine is a very promising heat engine with a high efficiency, multi-fuel capability, low emission, quiet operation, very low requirement and long life. Especially external combustion engine based cogen systems offer significant potential advantages over internal combustion engines in efficiency, life, noise and emission. Some companies are trying to commercialize them for a potentially great market of domestic cogen system in the near future. This paper proposes a new-type Ericsson cycle engine, one of the most efficient heat engine, comprising a scroll expander and a scroll compressor as a external combustion engine. In the Ericsson cycle engine, isothermalities of compression and expansion processes, namely the extents of cooling during compression process and heating during expansion process, are key factors on the performance. First of all, the potentialities of isothermality of scroll compressor and scroll expander will be discussed. And this paper will present the analysis of ideal gas cycle based on the Ericsson cycle and the superiorities of the cycle in comparison with the Brayton cycle. As well as for heat engine, 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. The analysis of ideal gas cycle based on the reverse Ericsson cycle will be presented and compared to the reverse Brayton cycle. The reverse Ericsson cycle is a very noticeable because of its performance and usability of a natural refrigerant, such as air.

2. REVIEW OF ERICSSON CYCLE

The Ericsson cycle is very much like the Stirling cycle in that both of two cycles consist of isothermal compression, isothermal expansion and regeneration process, except that the two constant-volume processes are replaced by two constant-pressure processes. The efficiency of the Ericsson cycle is the same as that of the Carnot cycle but, as in the Stirling cycle, the net available work and the quantities of heat transferred are much greater, for given limits of pressure, volume, and temperature. But in both cases, in spite of the high potentiality of the cycle, it is very difficult to realize the ideal cycle and the actual efficiency is much less than that of ideal cycle. The main reason is that neither the isothermal compression nor isothermal expansion in the cylinder or with the turbomachinery is practical due to the insufficient surface area for heat transfer and the short time for the process. In the Brayton cycle with turbomachinery, as the number of compression and expansion stages is increased, the Brayton cycle with intercooling, reheating, and regeneration will approach the Ericsson cycle and the thermal efficiency will approach the theoretical limit (the Carnot efficiency). Although the Ericsson cycle is often considered, it is usually abandoned because the added complexity of the turbomachinery and the need for many additional components, discourage its use. In this paper, we propose that the scroll-type compressor and expander is the most suitable for approaching the isothermal compression and expansion processes due to the extensive surface area of scroll warp contacting the working fluid. And the details about that will be presented in the following chapter.

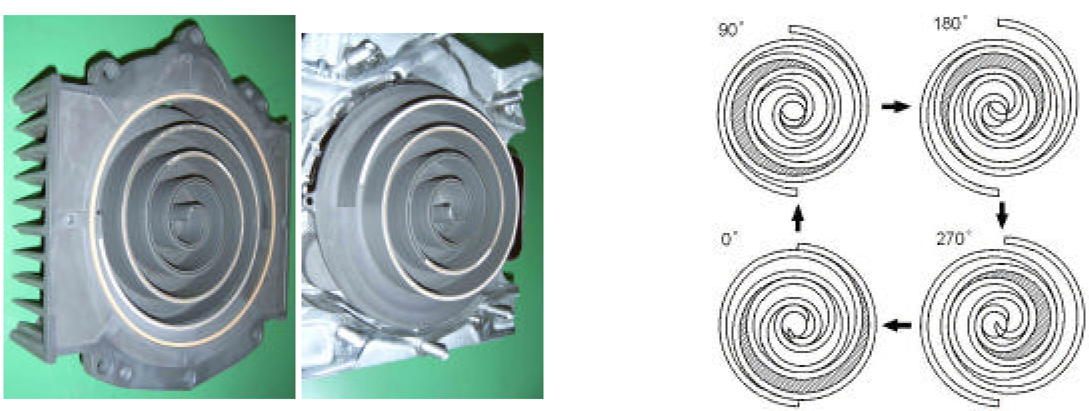
3. SCROLL-TYPE ERICSSON CYCLE ENGINE

3.1 Orbiting Scroll Mechanism

Although the orbiting scroll mechanism was patented by Leon Cruex in France as a patent named ‘Rotary Engine’, at that time, the technology to accurately make scrolls did not exist and the concept was forgotten. But in the early 1980s, as a gas compressor, it was introduced into the refrigeration industry for use mainly on small air conditioning and heat pump applications. And many units have been sold and are operating in the industrial commercial and air conditioning markets throughout the world. It is noted for its high efficiency, low noise and vibration level, smooth operation, reliability, lightweight and compactness. In configuration of an orbital scroll compressor, referring to Fig. 1, two parallel flat disks each having a raised spirally shaped strip (scroll wrap) on their flat surfaces are brought together to form an enclosed volume. One of the discs is free to orbit while the other remains stationary. The fluid is introduced into two diametrically opposite chambers simultaneously through the suction port of the compressor casing and fills the available crescent shaped chambers. Further orbiting of the moving scroll leads to a decrease in volume of the chamber containing the trapped fluid. This compression continues until the chambers containing the gas are exposed to the discharge port at the centre. In an orbital scroll compressor, continuous compression is attainable due to a series of compression of successive crescent shaped chambers. Sequence of operation for the scroll expander is reverse process of the scroll compressor. High-pressure fluid enters the machine at the centre through the inlet port and fills the crescent shaped chambers, and pushes the moving scroll to orbit. Further orbiting of the moving scroll leads to an increase in volume of the chambers containing the trapped fluid. This expansion continues until the chambers containing the gas are exposed to the outlet of the expander casing.

3.2 Ericsson Cycle Employing the Scroll Mechanism

The new-type Ericsson cycle engine is consists of one pair of scroll compressor and scroll expander, as shown in Fig. 2. In the process of compression, the heat generated in the compression space between the orbiting scroll and the stationary scroll of the scroll compressor is carried to the external dump (cooling fluid) through the inner scroll wrap



Fixed scroll Moving Scroll Compression Process

Figure 1: Scroll compressor and compression process (reverse process is expansion)

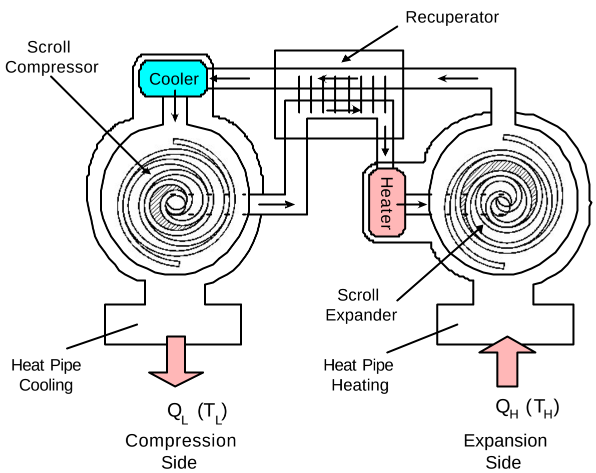


Figure 2: Scroll-type Ericsson cycle for heat engine

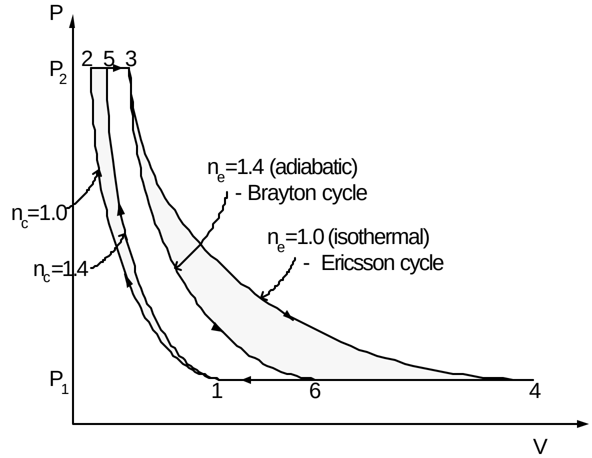


Figure 3: Brayton cycle and Ericsson cycle (P-V diagram)

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 (heating fluid) 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 new-type Ericsson cycle engine the processes of compression and expansion are not perfectly isothermal, it may have remote heater and cooler as shown in Fig. 2. 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 the heater and cooler. As well as for the heat engine, 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.

3.3 Comparison of the Ericsson cycle engine with the Brayton cycle

In the proposed scroll-type Ericsson cycle engine, as the portion of heat transfer in the compressor and expander is decreased and that in the heater and cooler is increased, the cycle approaches the Brayton cycle rather than the Ericsson cycle. The theoretical results of both cycles are obtained by the analysis of the ideal gas cycle. For the purpose of comparison high temperature $T_H = T_3 = 700$? and low temperature $T_L = T_1 = 50$? were assumed, and air (k=1.4, R=0.287 kJ/kg·K), as a working fluid, has been selected for the analysis. A comparison of P-V diagrams of the Brayton and the Ericsson cycle engine, between given limits of pressure and temperature, is shown on Fig. 3. 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 thermal efficiency of an ideal Brayton cycle and Ericsson cycle depends on the pressure ratio of the engine and the specific heat ratio of

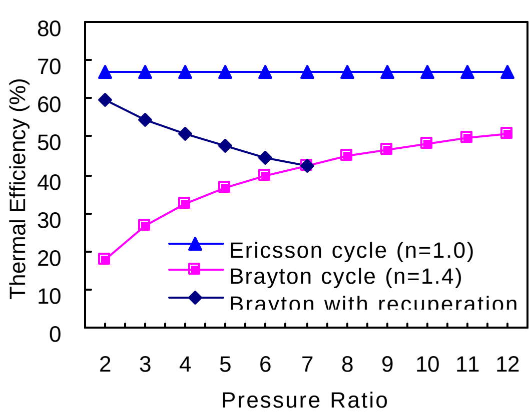


Figure 4: Thermal efficiency versus pressure ratio

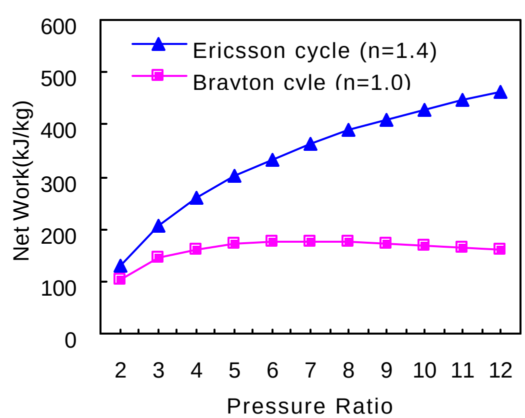


Figure 5: Net work (kJ/kg) versus pressure ratio

the working fluid (if different from air). A plot of thermal efficiency versus the pressure ratio is given in Fig. 4. Although it can be possible to improve the bad thermal efficiency of the Brayton cycle by regeneration with recuperator as shown in the Fig. 4, the thermal efficiency of the Brayton cycle is much less than that of the Ericsson cycle over the whole range of pressure ratio. And also, in the Ericsson cycle, more significant improvement in the work per unit mass was gained as the pressure ratio is increased in comparison with the Brayton cycle. The comparison of these two figures illustrates the profoundly deleterious effect on the efficiency and output that follows a departure from isothermality in the processes of compression and expansion.

3.4 Isothermality of scroll compressor

For the purpose of approaching the isothermal compression, it is necessary to cool the compressor effectively. And also, 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 30? 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 n_c, defined as polytropic index of compression, are obtained by the analysis of the ideal gas process. The inlet state of air was assumed at the same state of 50? and 0.1MPa. The outlet temperature (Tc-out) and the quantity (per unit mass) of heat rejected during compression process (Qcomp) are given in Fig. 6, and the outlet pressure (Pc-out) and the work (per unit mass) required to compress the air (Wcomp) are given in Fig. 7 as a function of n_c. In the adiabatic (n_c = k = 1.4) case, Tc-out goes up to about 470? from inlet temperature 50? , 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 compression (n_c = 1.0), 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 where the outlet temperature of compressed air is at the temperature about 30? higher than a inlet temperature, as presented in the brochure of a oil-free scroll compressor with Rc=8, corresponds to the process with 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 the scroll wrap contacting the working fluid in the scroll compressor.

3.5 Isothermality of scroll expander

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 Ericsson cycle, the scroll compressor

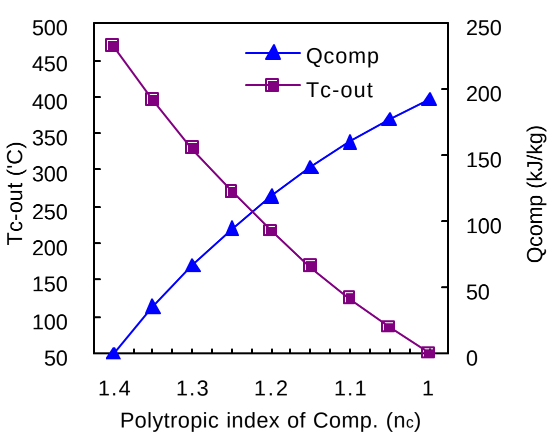


Figure 6: Tc-out and Qcomp versus n_c (Rc=8)

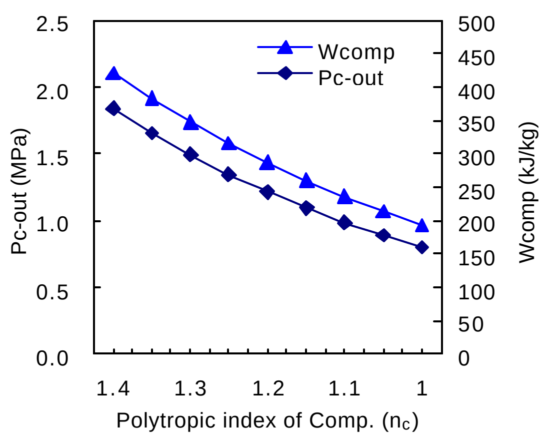


Figure 7: Pc-out and Wcomp versus n_c (Rc=8)

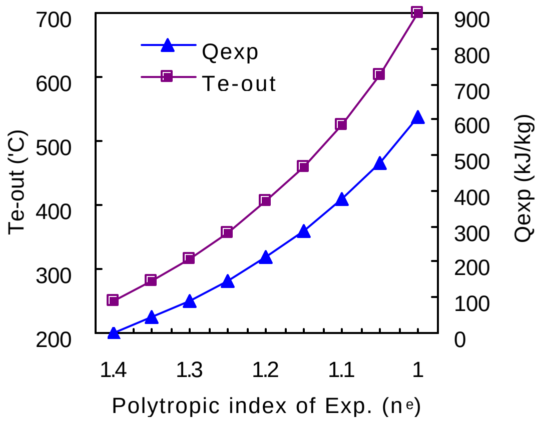


Figure 8: Te-out and Qexp versus n_e

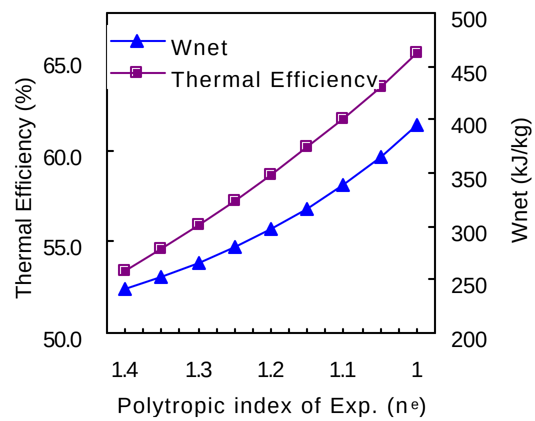


Figure 9: Cycle efficiency and Wnet versus n_e (n_c=1.043)

with Rc=8 and n_c = 1.043, this is equivalent to the previous commercial scroll compressor used for air compressor, was assumed. And all of heat exchangers including recuperator were assumed as ideally perfect: the 1.0 of efficiency and no pressure drop. The outlet pressure of the scroll expander is the same as the inlet pressure of the scroll compressor. The theoretical results for different process with n_e, defined as polytropic index of expansion, are obtained by the analysis of the ideal gas cycle. In the Fig. 8, the outlet temperature (Te-out) and the quantity (per unit mass) of heat supplied during the expansion process (Qexp) are given, and in Fig. 9, the net work (per unit mass) of the cycle (Wnet), difference between the expansion work and the compression work, and the thermal efficiency are given as a function of n_e. In the adiabatic (n_e = 1.4) case, Te-out goes down to about 250? from inlet temperature 700? and 451.1 kJ/kg of Wexp is produced. For the isothermal expansion (n_e = 1), it is necessary to add 605.4 kJ/kg of Qexp. And in that case, 605.4 kJ/kg of Wexp, which is much bigger than that in the adiabatic case, is produced. The isothermality of scroll expander of 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. 9.

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. 10. The values in parentheses in the Fig. 10 represent the values in the adiabatic case to show the cooling and heating effect obviously. In that case with n_e = 1.043, Te-out goes down to 617? from inlet temperature 700? , 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

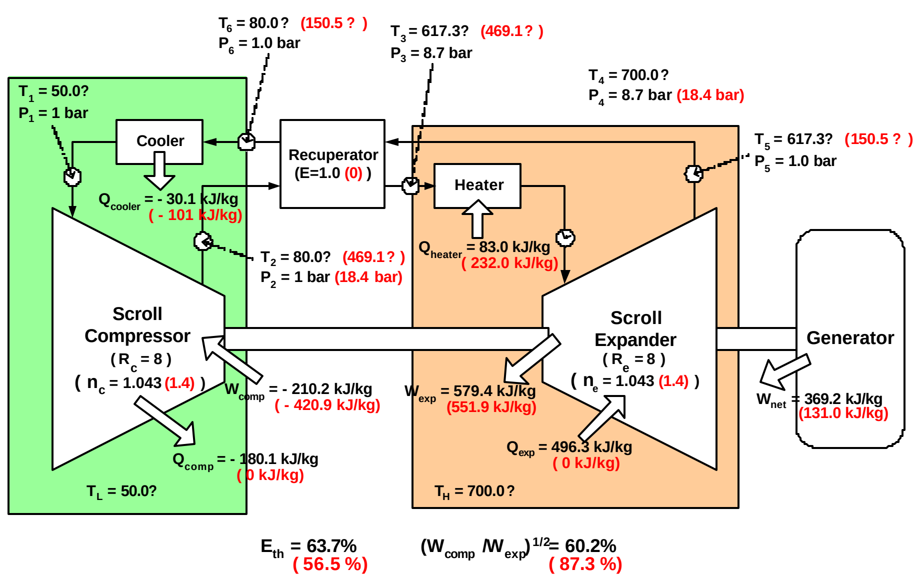


Figure 10: Scroll-type Ericsson cycle engine in comparison with Brayton cycle (Rc=8, Re=8)

practically possible to supply such quantity of heat through the scroll wrap 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. And with the proposed scroll expander, it is possible to use the phase change component with the gaseous carrier 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 a scroll expander yielding a high volumetric expansion ratio is used, 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 evaporator.
As a matter of fact, when the scroll expander is used for the expansion of high-temperature and high-pressure gas, the temperature drop of the gas in the expander causes a difference in thermal expansion between the innermost and outermost zones of the apparatus, which is a critical problem resulting in system vibration, noise and efficiency losses. In this aspect, the heating of scroll expander for the isothermal expansion can reduce the difference in thermal expansion as well as improvements in the thermal efficiency and the specific power of the engine.

4. SCROLL-TYPE REVERSE ERICSSON CYCLE REFRIGERATOR

4.1 Comparison of the reverse Ericsson cycle with reverse Brayton cycle for refrigeration

Ideally, the reverse Ericsson cycle is composed of a isothermal compressor and a isothermal expander, and the reverse Brayton cycle is composed of a adiabatic compressor and a adiabatic expander. In the proposed reverse Ericsson cycle employing the scroll mechanism as shown in Fig. 11, as the portion of heat transfer in the compressor and the expander is decreased and that in the radiator and chiller is increased, the cycle approaches the reverse Brayton cycle rather than the reverse Ericsson cycle. 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. 12. 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.

4.2 Isothermality of scroll expander in the refrigeration

The isothermality of scroll compressor in the reverse Ericsson cycle refrigerator is the same as mentioned before in the Ericsson cycle engine. The scroll compressor with Rc = 8 and n_c = 1.043 was assumed, as in the Ericsson cycle engine. Here, only the effect of the isothermality of scroll expander on the refrigeration cycle will be investigated. High temperature $T_H = T_1$ = 54.4? and low temperature $T_L = T_3$= -23.3? were assumed and air (k=1.4, R=0.287 kJ/kg·K), as a working fluid, has been selected for the analysis of ideal gas cycle. The theoretical results for different process with n_e, polytropic index of expansion, are obtained by the analysis of the ideal gas cycle. In the

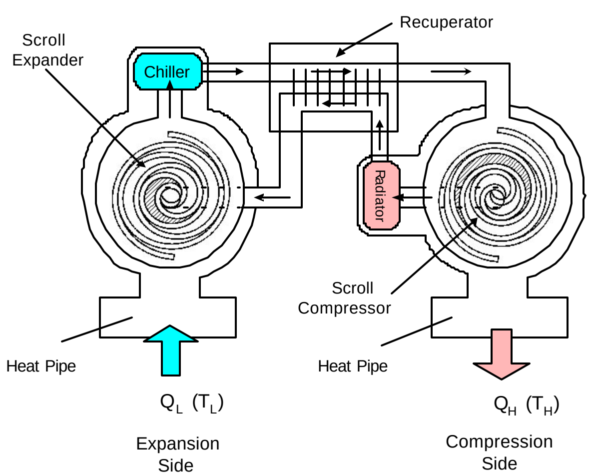


Figure 11: The reverse Ericsson cycle for refrigeration

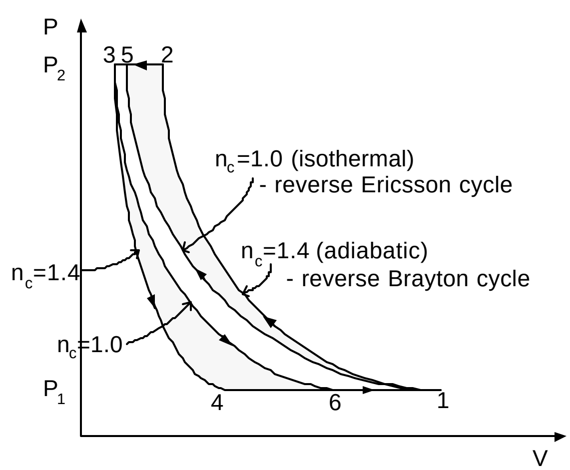


Figure 12: reverse Ericsson and reverse Brayton cycle

Fig. 13, the outlet temperature in the expander (Te-out) and the quantity (per unit mass) of heat 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. 14, the Q_L (per unit mass), defined as the magnitude of the heat removed from the refrigerated space at temperature T_L through the expander in itself and the chiller, and the net work (per unit mass) of the cycle (Wnet), difference between the expansion work and the compression work, are given as a function of n_e. In the adiabatic (n_e = 1.4) case, Te-out goes down to –138.5? from inlet temperature –23.3? and 115.8 kJ/kg of Wexp is produced. For the isothermal expansion (n_e = 1), 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 (per unit mass) as shown in the Fig. 14, which results in 126.0% of increase in the COP_R. 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 Fig. 15. The values in parentheses in the Fig. 15 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 –44.5? from inlet temperature –23.3? and it is necessary to absorb 127.4 kJ/kg of Qexp, which is about 70% of Qcomp.

As in the Ericsson cycle engine, if the phase change component with the gaseous carrier is used in the reverse Ericsson refrigerator, that contributes to improve the cooling capacity and the performance due to the phase change. In that case, it can be said that the scroll expander acts as a expansion device and partially as an evaporator.

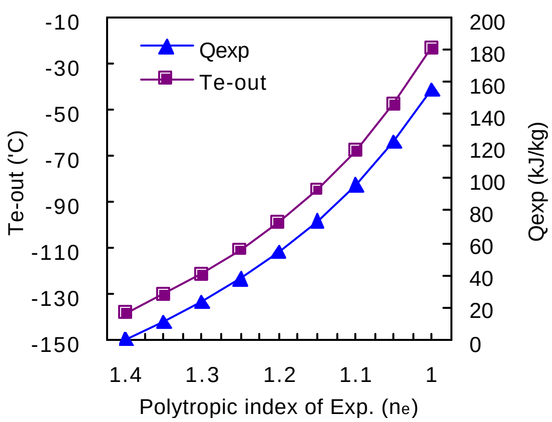


Figure 13: Te-out and Qexp versus n_e in reverse Ericsson refrigerator

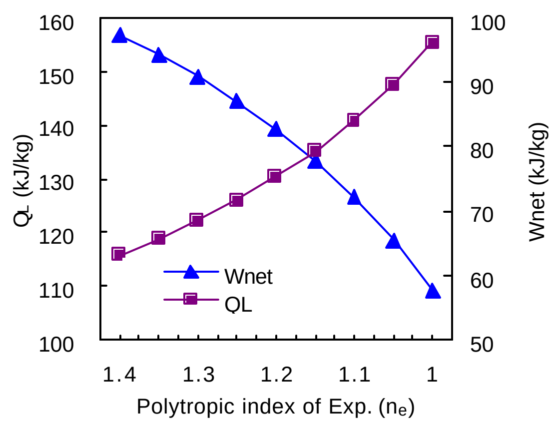


Figure 14: Q_L and Wnet versus n_e in reverse Ericsson refrigerator

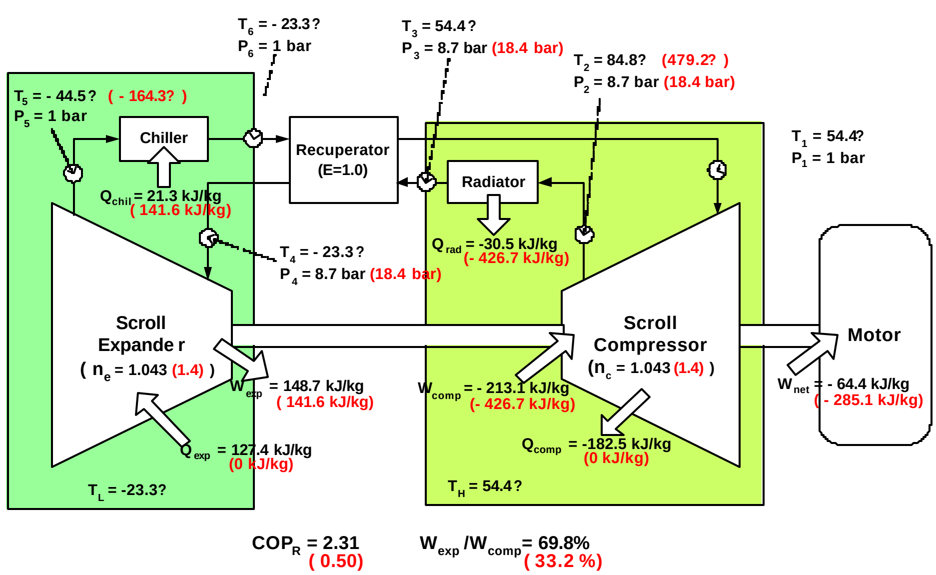


Figure 15: Scroll-type reverse Ericsson cycle refrigerator in comparison with reverse Brayton cycle (Rc=8, Re=8)

The drive power for the refrigeration cycle may be supplied from any source, including an electric motor or a heat engine, perhaps another Ericsson cycle engine. And also, corresponding to the Vuilleumier refrigerator/heat pump or duplex Stirling machines, the arrangement of a Ericsson cycle engine acting as a prime mover producing power to drive another Ericsson refrigerator acting as a heat pump, holding one scroll compressor in common, can be possible. Because the power generation and the heat pump can be controlled separately by regulating the working flow from the compressor, the system can be used as a total-energy system including a generator and a heat pump.

6. CONCLUSIONS

This paper proposes a new-type Ericsson cycle, one of the most efficient heat engine, comprising a scroll expander and a scroll compressor and heat exchangers including a recuperator. The reverse Ericsson cycle 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. Characteristics of the proposed scroll-type Ericsson cycle and the reverse cycle are summarized as follows.

- In the Ericsson cycle engine or reverse cycle refrigerator, the isothermalities of compression and expansion, namely the extents of cooling during compression process and heating during expansion process, are key factors on the performance of the engine or the refrigerator.
- From the preliminary study on a commercial scroll compressor used for oil-free air compressor, it can be said that the scroll compressor has a good potential for approaching the isothermal compression due to the extensive surface area of the scroll warp contacting the working fluid. And also, inversely it is supposed that the scroll expander has a good potential for approaching the isothermal expansion.
- In the proposed scroll-type Ericsson cycle engine, as the portion of heat transfer in the compressor and is decreased and that in the heater and cooler is increased, the cycle approaches the Brayton cycle rather than the Ericsson cycle. The more close to the Ericsson cycle (isothermal compression and expansion) the cycle become, the more improvements in the thermal efficiency and the work per unit mass were gained in comparison with the Brayton cycle (adiabatic compression and expansion).
- In the proposed reverse Ericsson cycle refrigerator, in the same way, the more close to the reverse Ericsson cycle (isothermal compression and expansion) the cycle become, the more improvements in the COP and the cooling capacity per unit mass were gained with the reverse Brayton cycle (adiabatic compression and expansion).

NOMENCLATURE

n_c , n_e	polytropic index of compression , expansion (–)
T_H , T_L	temperature of high temperature reservoir , low temperature reservoir (?)
Tc-out , Te-out	outlet temperature of compressor , expander (?)
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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