Helium Recondensing System Utilizing Cascade Roebuck Refrigerators

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Key Words : Roebuck refrigerator, cascade, recondensing, Roebuck compressor, bleeding, isothermal compression

Abstract

This paper describes a design of the helium-recondensing system utilizing cascade Roebuck refrigerators. Superconducting generator or motor has the superconducting field winding in its rotor that should be continuously cooled by cryogen. Since liquid helium transfer from the stationary system to the rotor is problematic, cumbersome, and inefficient, the novel concept of a rotating helium-recondensing system is contrived. The vaporized cold helium inside the rotor is isothermally compressed by centrifugal force and expanded sequentially in cascade refrigerators until the helium is recondensed at 4.2 K. There is no helium coupling between the rotor and the stationary liquid helium storage. Thermodynamic analysis of the cascade refrigeration system is performed to determine the key design parameters. The loss mechanisms are explained to identify entropy generation that degrades the performance of the system.

Nomenclature

- *COP* : Coefficient of performance of the refrigeration system
- *FOM* : Figure of merit
- h : Enthalpy (J/kg)
- h_{fg} : Enthalpy of vaporization or latent heat of vaporization (J/kg)
- $\dot{m}$: Mass flow rate (J/kg · K)
- P : Pressure (atm)
- Q : Heat transfer rate (W)
- Q_L : Cooling load (W)
- q : Amount of heat transferred per unit mass flow rate (J/kg)
- R : Radius of the rotor (m)
- s : Entropy (J/kg · K)
- T : Temperature (K)

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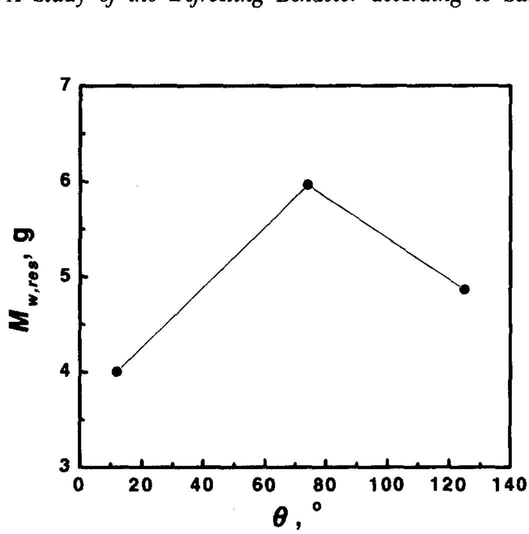

Fig. 7 Weight of residual water with contact angle.

4. Conclusions

In the present study, the effects of surface characteristics on the defrosting behavior of a fin-tube heat exchanger are investigated by successive experiments on the frosting/defrosting. The conclusions from the present work are as follows:

(1) The rate of draining water is evenly dispersed during defrosting in the hydrophilic and hydrophobic heat exchangers because of high density frost and a hydrophobic characteristic, respectively.

(2) The hydrophilic and hydrophobic heat exchangers show higher melting efficiencies during defrosting and lower draining water ratios during the rest period when compared to those of the bare one.

(3) The residual water in the hydrophilic and hydrophobic heat exchangers are about 20% less than that of the bare one.

Acknowledgement

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x : Quality of cryogen (helium or neon)

Greek letters

ω : Rotating speed (rpm)

Subscripts

1 : Inlet state before compression at Roebuck refrigerator
2 : State after isothermal compression
3 : State after isentropic expansion
b : Bleeding flow
i : Inlet state of each refrigeration stage
Hxi : Inlet state of the heat exchanger
Hxo : Exit state of the heat exchanger
LP : State of low pressure gas stream

1. Introduction

Motor and generator are the most power-consuming electric machines in industry. There has been a lot of improvement in the efficiency of generator and motor in recent years. The development of superconducting generator has been also continued with its inherent advantage of high efficiency[1]. Even though the large-scale generator has more than 98 % efficiency, it is always desirable to improve its efficiency. Furthermore, as HTS (High Temperature Superconductor) wire has been manufactured, more attention has been paid to the practical HTS motor development[2]. The strong magnetic field can increase the power density of the machine or for a given power rating, a superconducting machine can be made to be smaller and lighter. However, the superconducting generator or motor has the field winding in its rotor that should be continuously cooled by liquid helium or liquid neon below the critical temperature of the used superconductor. The cryogenic cooling system of the superconducting generator has provided liquid helium into the rotor and recovered the vaporized helium from the rotor. It is, therefore, absolutely necessary to have an efficient cryogenic transfer coupling that connects the rotating and stationary frame. Novel techniques such as mechanical face sealing and ferrofluidic sealing were developed and successfully installed in the previous superconducting generators to keep a good vacuum for thermal insulation. A combination of ferrofluidic and mechanical seals was also used as the cryogenic transfer coupling of the recent 1000 hp HTS motor design[2]. It is, however, still questionable how reliable this rotating liquid helium transfer system is, in order for the superconducting machines to be commercially available. The cryogenic transfer coupling has complicated the whole system and added additional inefficiency. The vaporized helium from the rotor is finally either wasted or recovered, but with a great deal of expense and effort[3]. The sensible enthalpy of cold helium gas is wasted in the external recovery process, which results in thermodynamic inefficiency of cryogenic system. The horizontal steep temperature gradient in the axial liquid helium transfer line and the vent line under the centrifugal force can also cause flow instability. Previous research showed that the flow instability in the liquid helium flow passage resulted in very large heat leak at a certain rotating speed[4, 5]. If the superconducting rotor operates without the liquid helium transfer from the stationary frame, the rotating sealing component would not be necessary. The previous paper of the authors introduced an idea of the rotating helium-recondensing system utilizing the cascaded Roebuck refrigerators[6]. This paper describes the main idea of the rotating recondensing system for helium and explains how such a refrigeration system can be optimized. The design of the prototype machine including the cryogenic mechanical compressor is also discussed. Additionally, instead of using mechanical compressor, a modified Roebuck thermal compressor is also explained in this paper as a novel compression device of a cryogen in a rapidly rotating system.

2. Thermodynamic analysis

2.1. Basic cycle

The rotating recondensing system utilizes a number of Roebuck refrigerators that are connected in series at different temperature stages. The Roebuck refrigerator was originally conceived in 1945[7]. As shown in Fig. 1, Roebuck refrigerator is a U-shape rotating tube. It is ideally composed of an isothermal compression part and an adiabatic expansion part as shown in Fig. 2. The outward gas flow from 1 to 2 is compressed while the compression heat is rejected to the heat exchanger isothermally and expanded in the section from 2 to 3 to produce low-temperature gas stream. The high-pressure helium is generated from the vaporized helium by centrifugal compression in the rotating frame. If the temperature of the gas is low enough, a significant cooling effect is obtained when the gas re-expands. This expansion process is different from Joule-Thomson expansion in that the pressure energy does work to the system as it increases the potential energy of the gas. If the expansion process is ideally adiabatic, it is an isentropic process. The pressure rise of the gas by the centrifugal force becomes large as the density of the gas increases. Thus, the Roebuck refrigerator is very effective at cryogenic temperature rather than at room temperature and capable of being

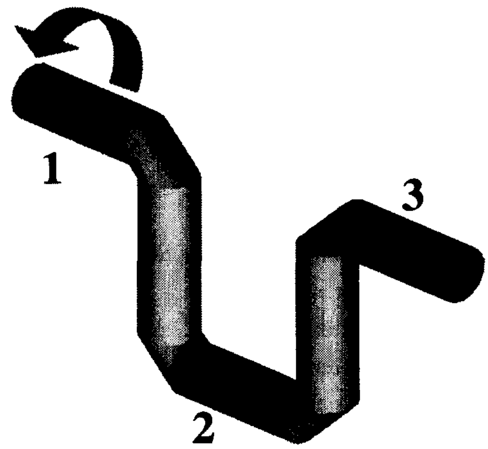


Fig. 1 U-shape tube required for Roebuck refrigerator.

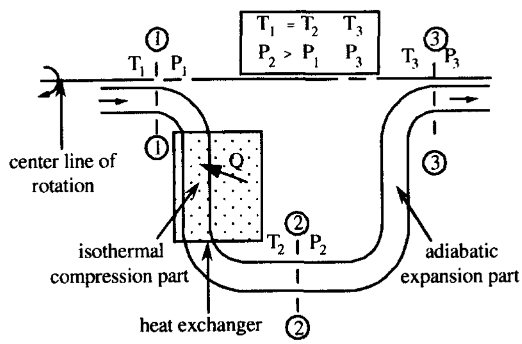


Fig. 2 Schematic diagram of the basic Roebuck refrigerator

used for a recondensing system of liquid helium in the superconducting generator. The first order design of the helium recondensing system does not consider the detailed part of the heat exchanger where the isothermal compression should be obtained by a proper cooling method. The following equations are solved to determine the thermodynamic states in the refrigerator:

$$q = h_2 - h_1 - \frac{1}{2} R^2 \omega^2 \qquad (1)$$

$$\frac{q}{T} = s_2 - s_1, T = T_1 = T_2 \quad (2)$$

$$0 = h_3 - h_2 + \frac{1}{2} R^2 \omega^2 \quad (3)$$

$$0 = s_3 - s_2 \quad (4)$$

where r and ω are respectively the radius and the angular velocity of the Roebuck refrigerator. If the gas were assumed to be an ideal gas, the temperature drop at a single stage of the refrigerator would be calculated as follows:

$$\triangle T = \frac{r^2 \omega^2}{2 C_P} \quad (5)$$

where C_P is the average specific heat of the gas during the expansion process. Since the state of helium in the recondensing system, however, is far from the condition of ideal gas, the real thermodynamic properties must be calculated by computer[8, 9] to find the states more accurately. The next section describes how to connect the Roebuck refrigerator in series to construct the cascade system.

2.2. Cascade system

Based on the basic Roebuck refrigerator, several stages of the cascaded Roebuck refrigerators can be designed. Fig. 3 is the schematic block diagram of the rotating helium recondensing refrigeration system. The design has two separate cryogenic refrigeration system; one between liquid helium and liquid neon reservoirs, and the other between liquid neon and liquid nitrogen reservoirs. The design of the rotating liquid helium recondenser in this paper includes the heat exchange section at liquid nitrogen temperature, which ultimately rejects heat (Q_{N2}) to the surrounding stationary system and the cryogenic mechanical compressor which boosts the pressure of the system. The penetration through the interface of the rotating system and the stationary environment is, therefore, only mechanical torque and heat at 77 K instead of the liquid helium as in the case of conventional superconducting generators. Helium is used as a first refrigerant below liquid neon temperature and neon as a second refrigerant above liquid neon temperature. As shown in Fig. 3, the helium gas at 32 K is isothermally compressed by a mechanical cryogenic compressor, transferring its compression heat (Q_{Ne}) to the liquid neon reservoir. The compressed helium at 32 K is sequentially cooled-down at the 8-stage helium recondensing system. The final stage produces liquid helium at 4.2 K for cooling superconducting field winding in the rotor. The vaporized helium is re-

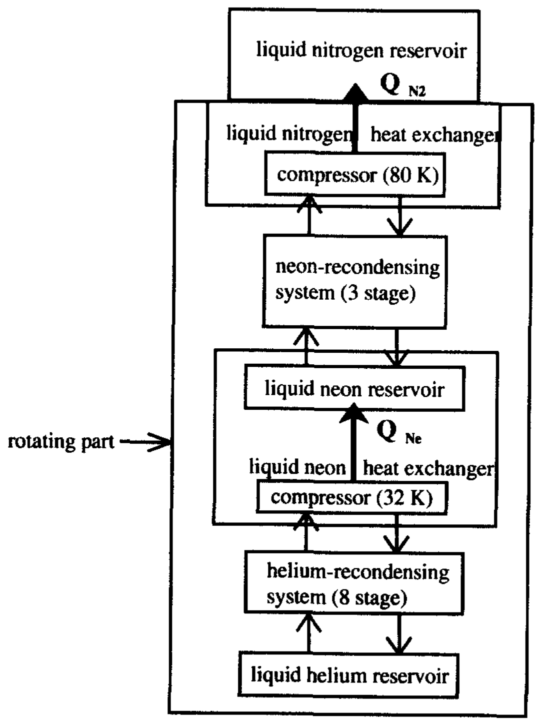


Fig. 3 Block diagram of the rotating helium-recondensing system

turned to the 32 K compressor while it is used for cooling the high-pressure incoming helium at the cascaded Roebuck refrigerators. The neon gas is compressed by another mechanical compressor at 80 K, giving off its heat to the stationary liquid nitrogen reservoir via the heat exchanger which is made with copper disk fins on the rotor. The compressed neon is similarly cooled-down at the 3-stage neon recondensing system.

The cascade refrigeration system incorporates many counterflow heat exchangers between the compressed gas and the low-pressure gas (essentially 1 atm in the case of helium stages) in each unit as shown in Fig. 4. The low-pressure gas is actually the vaporized helium coming from the superconducting field winding and being used for cooling in the counterflow heat exchanger. As the detailed design is carried out, however, two important problems are encountered immediately. The first one is the heat capacity mismatch between the high-and the low-pressure helium streams, which is similar to the well-known characteristic of the helium liquefier design[10]. Since the cooling capacity of the vaporized low-pressure helium is not big enough to cool down the high-pressure compressed helium, the bleeding process from the high-pressure stream is sometimes necessary. Some portion of the high-pressure helium is, therefore, expanded isenthalpically instead of further compression as shown in Fig. 4. This is just a simple Joule-Thomson expansion process. The bled helium is expanded to 1 atm and mixed with the low-pressure helium stream. As the bleed-ratio increases, the cooling capacity of the low-pressure helium is virtually increased. The second problem emerges when the modified Roebuck refrigerator with a bleeding process operates above 45 K where the isenthalpically bled helium cannot be used to

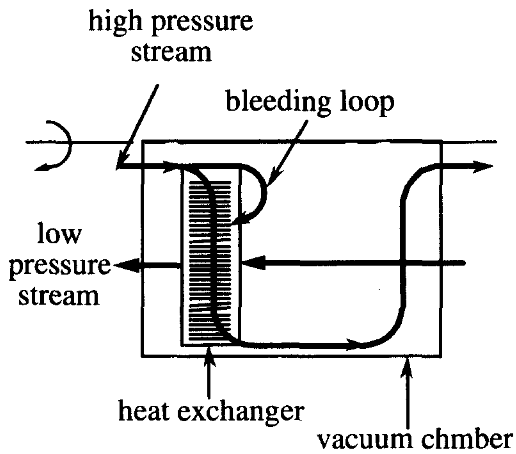


Fig. 4 Schematic diagram of each stage with bleeding process

cool down the high-pressure stream any more. Since the temperature exceeds the Joule-Thomson inversion temperature of helium[11], the temperature of the expanded helium increases rather than decreases. The single-component (helium only) cascade system, therefore, cannot operate between 40 K and 80 K with a bleeding process. It is why the dual-component system with helium and neon is designed. Using neon has another advantage above 30 K because the dense neon has more compression-expansion effect by centrifugal force than helium at this temperature range.

2.3. Optimization of the cascade system

The helium recondensing refrigeration system as described in the previous section can be actually designed in many ways with different number of temperature stages for helium or neon side. Since the vaporized helium should be first pressurized at the highest temperature stage by a mechanical compressor, the compressor size is a significant factor in the system optimization. The compressor load depends on not only the pressure ratio but also the mass flow rate it handles. The whole refrigeration system should be optimized to produce a reasonable amount of liquid helium for the demanded cooling capacity, but without too much flow rate of the recirculating helium or neon. The first design presented in the previous paper[6], had a FOM of 0.0007, which means a very inefficient system. For a cooling load of 1 W, it required too large mass flow rate of helium and neon, resulting in 205 W of heat rejection to the liquid neon and 23.7 kW of heat rejection to the liquid nitrogen reservoirs. The main idea of the system optimization in this paper is the selection of the optimum liquid amount ratio at the final stage and the proper distribution of the bleeding mass flow rate at each stage.

The following result is obtained for the helium recondensing system operating between 4.2 K and 80 K. The rotation speed and the radius of the refrigeration system is assumed as 3600 rpm and 0.5 m respectively. Fig. 4 shows the heat exchanger of the isothermal compression process where the heat exchange occurs between the high-pressure stream and the low-pressure stream. Since the gas stream recirculates, the actually compressed mass flow rate of the high-pressure stream without the bleeding mass is the same as that of the low-pressure stream. The mass flow rate of the low-pressure stream for cooling is increased due to the bled stream. Therefore, the energy balance in the heat exchange process is written as follows.

$$\begin{aligned} Q &= \dot{m}_i T(s_1 - s_2) \\ &= \dot{m}_i(h_{HXo} - h_{HXi}) + \dot{m}_b(h_{HXo} - h_1) \end{aligned} \tag{6}$$

In Eq.(6), S_1, S_2, T, and h_1 are determined from the basic design of the previous section. The liquid production rate of this helium recondensing system is determined by the quality of the finally expanded stream. The mass flow rate at the lowest temperature stage is calculated as follows.

$$Q = \dot{m}_i h_{fg}(1 - x) \tag{7}$$

The optimization of each stage starts from the low temperature side. The first stage at the lowest temperature immediately determines h_{HXi} because it is the enthalpy of the saturation vapor at the given pressure of the liquid helium reservoir. Since the isothermal compression temperature and the pressure of the stage has been calculated from the Eq. (1)-(4) of the basic design section, the enthalpy of the low-pressure stream after the heat exchange, h_{HXo} is obtained from Eq. (6) if the bleeding mass flow rate is determined. The second law of thermodynamics requires that the temperature of the low-pressure stream must be lower than that of the isothermal compression process. This is the necessary condition for selecting the amount of the bleeding mass flow rate. Since the enthalpy of the low-pressure stream is almost proportional to the temperature, it is desirable for the bleeding mass to lower the exit enthalpy of the low-pressure stream effectively when the cooling capability of the low-pressure stream itself is not enough. The bleeding process is assumed as an isenthalpic process and the design is considered to minimize the amount of the bleeding mass flow rate to reduce the circulation mass flow rate of the whole refrigeration system. Eq. (6) is rewritten as follows.

$$h_{HXo} = \frac{T(s_1 - s_2) + h_{HXi} + \frac{\dot{m}_b}{\dot{m}_i} h_1}{1 + \frac{\dot{m}_b}{\dot{m}_i}} \tag{8}$$

Since the temperature is pre-determined at each stage, the equation (8) can be written as follows by replacing each term, $T(s_1-S_2)+h_{HXi}$ with A and h1 with B.

$$h_{HXo}=B+\frac{A-B}{1+r} \tag{9}$$

It is evident that the reduction of h_{HXo} by increasing the bleeding mass flow rate ratio is more significant as A - B is bigger. It is the temperature stage with large value of A - B that maximizes the effect of the bleeding process. A - B is calculated from the real thermodynamic property of cryogen at various temperature and pressure. For the optimization process of 8 stages of helium and 3 stages of neon, the calculated value of A - B, is compared at each stage and the optimum amount of bleeding mass is determined. The optimization is to minimize the circulation mass flow rate of the system and the work input for a given cooling load at the lowest temperature stage.

2.4. Calculation results

The calculation results are shown in Table 1. The cooling load of the system is assumed as 1 W and the mass flow rate of the cryogen is determined to optimize the FOM. The recondensing temperature of helium is chosen as 4.2 K in consideration of cooling the superconducting field winding made with NbTi. The expansion pressure at the final stage is, therefore, 1 atm. According to the quality of the expanded helium at the final recondensing stage, the temperature of the upper stage is determined. The thermodynamic analysis is actually done from the final lowest-temperature stage to the upper higher temperature stages even though the intuitive physical process looks to occur from the high-temperature stage to the low-temperature stage. The thermodynamic state of the first warmest stage is resultantly dependent on the low-temperature stages. Since the temperature of the first helium stage immediately influences the temperature of the liquid neon reservoir, the optimization of the neon recondensing refrigeration system is made after the helium system. The most important factors to consider in the optimization process are the mass flow rate of the helium or the neon at the warmest stage, the heat transfer rate to the liquid nitrogen reservoir, and the number of the bleeding flows. The design case with excessively high pressure or large mass flow rate is excluded because it is unrealistic to be constructed. The system pressure tends to increase as the quality of helium at the coldest expansion process decreases. On the other hand, if the quality increases, the required mass flow rate increases. It is, therefore, concluded that there exists an optimum quality to minimize the work input of the recondensing refrigeration system. Since the thermodynamic efficien-

Table 1 Thermodynamic calculation results

Stage	P_1 (atm)	$T_1 = T_2$ (K)	$\dot{m}_i$ (g/s)	$\dot{m}_b$ (g/s)	P_2 (atm)	P_3 (atm)	T_3 (K)
He1*	4.0	7.1	0.06	0.66	27.0	1.0	4.2
He2	5.7	11.3	0.72	0.00	13.0	4.0	7.1
He3	6.4	15.0	0.72	0.00	11.6	5.7	11.3
He4	6.8	18.5	0.72	0.00	10.8	6.4	15.0
He5	7.0	21.9	0.72	0.00	10.4	6.8	18.5
He6	7.2	25.3	0.72	0.00	10.1	7.0	21.9
He7	7.4	28.7	0.72	0.00	9.9	7.2	25.3
He8	7.6	32.0	0.72	0.00	9.7	7.4	28.7
Ne1*	4.8	43.3	6.15	27.8	15.2	2.8	31.0
Ne2	5.6	61.9	33.9	34.6	11.5	4.8	43.3
Ne3	6.0	79.7	68.5	0.00	10.4	5.6	61.9

* The quality of the recondensed mixture at the He1 and Ne1 stage is 0.2 and 0.8 respectively.

cy of the whole recondensing system is dominated by the low-temperature one, the optimization is first considered for the helium stages. The quality of the helium at the final stage is first varied to see the optimized states at each temperature stage. From the calculations for the quality of 0.1 through 1.0 with the interval of 0.1, the quality of 0.2 shows the maximum thermodynamic efficiency. As shown in Table 1, the helium system requires 8 stages with the circulation mass flow rate of 0.72 g/s and the highest temperature of 32 K. The next step is to optimize the neon system. The critical pressure of neon is 26.2 atm, which is much bigger than that of helium, 2.26 atm. If we set the pressure of the low-pressure neon stream as 1 atm as in the case of helium stages, the isothermal compression process at the final lowest-temperature stage should accommodate the two-phase neon. Since the isothermal compression with the saturated two-phase mixture is cumbersome and there is no need to keep the low-pressure side as 1 atm, the analysis proceeds for higher pressures. The recondensing temperature of neon is varied as 28 K, 29 K, 30 K, and 31 K, each of which corresponds to the saturation pressure of 1.3 atm, 1.7 atm, 2.2 atm, and 2.8 atm. The optimization result of the neon system is obtained for the quality of 0.8 at 2.8 atm. The neon system is designed to have total 3 stages with the circulation mass flow rate of 68.5 g/s and the highest-temperature of 79.7 K.

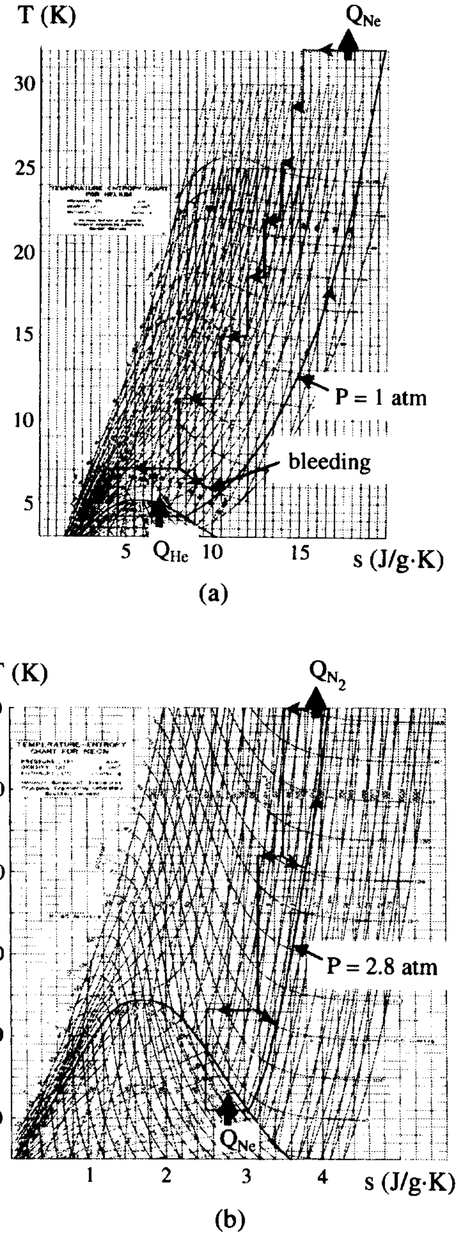


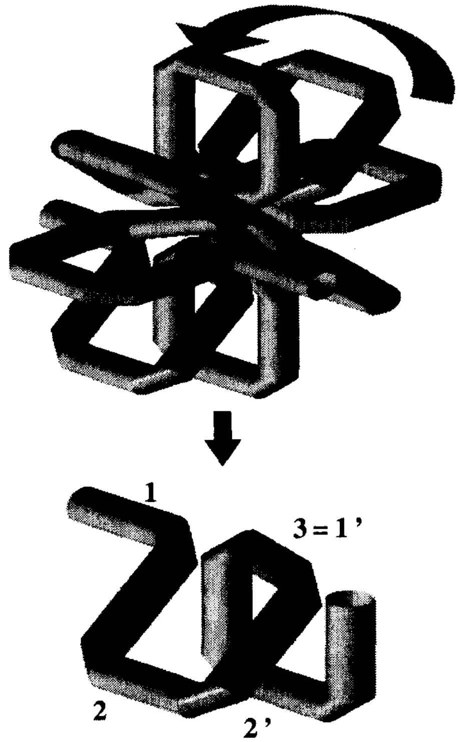
Fig. 5 T-s diagram of (a) helium (b) neon cascade refrigeration cycle with bleeding processes

Fig. 5 (a) shows the optimized helium recondensing system at the T-s diagram of helium. Total number of refrigerator staging is 7. As shown in Fig. 5 (b), the optimized neon system consists of total 3 stages with two bleeding processes. The whole helium recondensing system generates liquid helium for 1 W cooling at 4.2 K by rejecting heat of 98 W to the liquid neon reservoir and 1787 W to the liquid nitrogen reservoir. The FOM is calculated as 0.01.

3. Design consideration for the prototype machine

The prototype machine of the optimized helium recondensing system is discussed in this section. The main expansion process occurs adiabatically (or isentropically) while the bleeding stream expands isenthalpically and is mixed with the low-pressure helium inside the heat exchanger. The heat exchanger can be an one-pass shell and tube type, surrounding the compression part as shown in Fig. 4, which is the schematic diagram of each stage. The arrangement of all 11 cascade Roebuck refrigerators in series can be done peripherally. Figure 6 shows some idea of this kind of arrangement in 3-D diagram. The system is initially cooled down to 4.2 K by liquid helium transfer when the rotor is stationary after precooling. The further cooling load of the rotor at its steady operation is to be taken by the rotating helium recondensing system.

There are several things that have not been considered in the previous design section. First, the pressure drop of the flow generates entropy that is acting as an additional cooling load in the next stage. The entropy generation at the low temperature stage, however, can be kept smaller than that of the high temperature one by bleeding some flow and reducing the mass flow rate for the next stage. The pressure drop of the low-pressure stream may not be negligible due to its increased specific volume. This fact would ultimately require a larger mechanical compressor than the one designed without considering the flow pressure drop. Second, the mixing entropy generation would occur if the temperature of the bled helium or neon were slightly different from that of the low-pressure side stream in the heat exchanger where they are mixed. Third, the iso-



1 - 2 - 3(=1') - 2' - 3' - ...

Fig. 6 Schematic diagram of the cascade arrangement of Roebuck refrigerators.

thermal compression requires very slow process and large heat transfer area to be accomplished. Since there is a space limitation in the real rotor, the size of the heat exchanger cannot be infinitely large, which may cause non-isothermal compression and degradation of the system performance. Fourth, even though the isentropic process is assumed in the main expansion flow by vacuum insulation and small axial conduction along the expansion tube, some parasitic heat leak would prevent the flow from reaching the possible lowest temperature. Fifth, even though the main compression process is possible by the centrifugal force of the rotor, the additional two mechanical compressors are necessary at 32 K and 80 K. Since both compressors operate at cryogenic temperature with relatively small pressure ratio, a metal bellows-type compressor may be a good choice for this application. It can be driven by a linear motor at cryogenic temperature. The 80 K stage is the final heat rejection part from the rotating system to the environment. The compressed neon by the mechanical compressor should be ultimately cooled by stationary liquid nitrogen in the outside of the rotor. The cryogenic compressors were utilized in many applications at liquid helium temperature and higher than that. Most applications were for producing lower temperature than 4.2 K[12] or increasing the efficiency of helium liquefier[13]. The report of Asakura et al.[14] about 80 K centrifugal compressor shows very promising result even though it is not a linear motor-driven bellows-type compressor. Despite all these efforts, the efficient operation of cryogenic compressor in rotating frame still remains as a challenge.

Therefore, a novel method of utilizing thermal compressor instead of mechanical compressor in the refrigeration system is explored. Similar to Roebuck expansion device, the reversible Roebuck compressor can be devised if the fluid flow is reversed. The Roebuck compressor utilizes centrifugal force to compress the working fluid with the assistance of the heat transfer. The idea of Roebuck compressor and its application for rotating cryogenic refrigeration system is being pursued by the authors of this paper.

4. Summary

A cascade Roebuck expansion devices are designed for recondensing vaporized helium in rotating superconducting magnet system. One of the great advantages is that the introduced design idea does not involve any liquid-cryogen transfer between the rotor and the outside stationary cryogen storage. The detailed thermodynamic analysis was performed between the temperature range of 4.2 K and 90 K to determine the key design parameters. The loss mechanisms are also explained to identify entropy generation that degrades the performance of the system.

Acknowledgments

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