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Heat-Exchanger Design of 10 kW Brayton Cryocooler for HTS Cable Application

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Cryogenic heat exchangers are designed as next step of the thermodynamic study to develop a Brayton cryocooler in Korea for application to 1~3 km HTS cable at transmission class. The refrigeration capacity is 10 kW for continuously sub-cooling liquid nitrogen from 72 K to 65 K, and the refrigerant is helium gas. The physical dimension of standard plate-fin heat exchanger is determined for the recuperator and liquid sub-cooler, and the schemes to prevent the freezing of liquid nitrogen are also discussed.

INTRODUCTION

The high-temperature superconductor (HTS) cable project in Korea is underway with five-year plan to install and demonstrate 1~3 km cable at transmissions level in Jeju Island. In order to meet the cooling requirement of the HTS cable, a Brayton cryocooler is also under domestic development for closed-cycle refrigeration with a capacity of 10 kW at liquid-nitrogen temperatures. As the first step, an extensive thermodynamic study was carried out over the state-of-the-art in Brayton technology, and the key design parameters of the Brayton cryocooler were determined in 2011[1].

As next step towards the cryocooler development, this paper presents detailed heat-exchanger design. Among a few different options, the plate-fin heat exchanger (PFHX) has been selected because of its compactness and high effectiveness. The PFHX is widely used in a variety of cryogenic systems for midor large-scale refrigeration or liquefaction [2-4]. Lately, another governmental project in Korea has supported a company which will manufacture and supply aluminium PFHX's for a test bed of LNG plant with a capacity of 100 ton/day. In collaboration with the company, we attempt to design, fabricate, and test the PFHX of 10 kW Brayton cryocooler for HTS cable applications.

THERMODYNAMIC CYCLE

The cooling requirement is to continuously sub-cool a liquidnitrogen (LN2) stream from 72 K to 65 K at rate of 0.710 kg/s (i.e. 10 kW) with the reverse-Brayton cycle, as schematically shown in Figure 1[1]. The minimum power required for this refrigeration is 33.8 kW, and the thermodynamic performance of an actual system is evaluated in terms of the figure of merit (*FOM*), which is defined as the ratio of minimum to actual power. Helium (He) gas is selected as refrigerant, mainly because it is superior to neon in thermodynamic performance. The operating pressure is 0.5~1.25 MPa, and the flow rate of helium is 0.209 kg/s. Two counter-flow heat exchangers are employed in this system; He-He recuperator (HX1) and LN2 sub-cooler (HX2). The thermodynamic cycle is designed with an assumption that the minimum temperature difference between two streams is 5 K for HX1 and 1.5 K for HX2, and the pressure drop in each stream is 50 kPa for HX1 and 20 kPa for HX2 and after-coolers. This assumption should be verified by detailed heatexchanger design. It was further assumed that

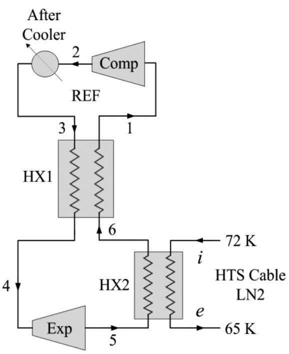


Figure 1 Schematic of reverse-Brayton cycle for sub-cooling liquid nitrogen.

the adiabatic efficiency of two-stage compressor and singlestage expander is 0.75. The expected *FOM* of this refrigeration system is approximately 0.21[1].

HEAT-EXCHANGER DESIGN AND RESULTS

The main issue is to determine the size and shape of HX1 and HX2 that are suitable for 10 kW Brayton cryocooler. As schematically shown in Figure 2, a plate-fin heat exchanger (PFHX) is composed of many layers of stacked aluminium plates and corrugated aluminium fins brazed between the plates. The fins play a role of extended surface for improving heat transfer and spacer for improving flow distribution at the same time. For detailed heat exchanger analysis, a typical offset strip (or serrated) fin is selected, and its geometrical parameters are indicated in Figure 3. The fins with a short length (l) are shifted to normal flow direction for enhancing heat transfer with slightly added pressure drop.

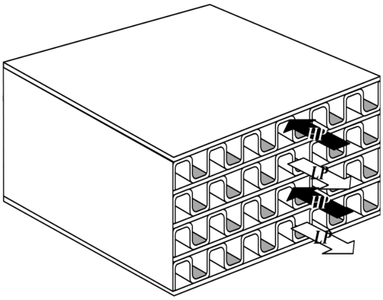


Figure 2 Schematic of counter-flow plate-fin heat exchanger.

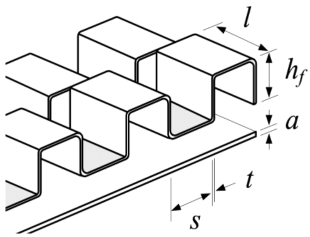


Figure 3 Geometric parameters of offset strip fin.

The inlet and exit conditions of HX1 and HX2 are given from the thermodynamic cycle, as listed in Table 1. The design process of PFHX is described in a number of references, such as Kays and London [5] or Barron [6]. The corrugated fins are considered a simple plate fin, having half height (h_f /2) and adiabatic tip because of its symmetry. The surface efficiency [5-7] is given by

$$\eta = 1 - \frac{A_f}{A}\left[1 - \frac{\tanh\left(h_f\sqrt{h/2kt}\right)}{h_f\sqrt{h/2kt}}\right] \tag{1}$$

where A_f is the surface area of fin, and k is the thermal conductivity of aluminium. The most uncertain value in this calculation is the heat transfer coefficient (h), which is estimated with the engineering correlations suggested in Wieting [8] and Rohsenow et al. [9] in terms of Reynolds number based on the hydraulic diameter of rectangular flow passage

$$d = \frac{2h_f s}{h_f + s} \tag{2}$$

The overall heat transfer coefficient (U_h) is evaluated by

$$\frac{1}{U_h} = \frac{1}{\eta_h h_h} + \frac{aA_h}{kA_w} + \frac{A_h}{\eta_c h_c A_c} \tag{3}$$

where the subscripts c and h denote the cold and hot streams, respectively. The second term of Eq. (3) is the thermal resistance of parting wall, where A_w is the surface area of plate wall. The thermodynamic and thermo-physical properties of He and LN2 are calculated with standard software (*NIST REPROP* and *Aspen HYSYS*).

Table 1 Inlet and exit conditions of heat exchangers

	HX1		HX2	
Fluid	He	He	He	LN2
***m* [kg/s]**	0.21	0.21	0.21	0.71
T_{in} [K]	300	70.5	61.4	72
T_{out} [K]	75.9	295	70.5	65
P_{in} [MPa]	1.24	0.56	0.58	0.30
P_{out} [MPa]	1.19	0.51	0.56	0.28

Table 2 Geometric parameters of fin and plate

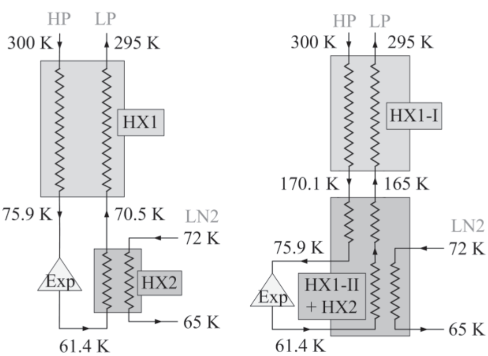
Parameters		Fin	Plate
Thickness	*t, a* [mm]	0.1	1.0
Height	h_f [mm]	3.0	-
Space	*s* [mm]	1.27	-
Unit length	*l* [mm]	3.0	-
Thermal conductivity	*k* [W/m-K]	150 (Aluminium)	

Since the cryostat for two heat exchangers and expander is installed at indoor facility of substation, the compactness is an important design issue. With typical heat transfer model and geometric parameters listed in Table 2, the required size of PFHX is calculated for HX1 and HX2. In particular, the effective width, number of layers, and length are the key dimension to determine. In general, the total heat exchange area is nearly proportional to its volume or the product of three factors. On the other hand, even though the heat exchange area may be same, its aspect ratio affects to an extent the axial conduction, the flow distribution and pressure drop, the external heat leak, and so on.

As first trial, the effective width and number of layers are taken as 0.5 m and 60, respectively. The required length of HX1 and HX2 is estimated to be 2.4 m and 0.2 m, respectively. Since the length of HX1 exceeds the limit of typical height, some modification is needed. The length may be reduced by increasing the width and number of layers, which is not preferred, however, in terms of axial conduction and flow distribution. The next trial is to split the HX1 into two pieces in series, which is not preferred either, because the pressure drop would increase due to the additional ports for inlet and exit.

It is finally decided to arrange HX1 and HX2 such that the entire heat exchanger body is made of two pieces of same size, as illustrated in Figure 4. The warm heat exchanger (HX1-I) is a simple counterflow heat exchanger that is equivalent to approximately 58% (140 kW) of HX1. The cold heat exchanger (HX1-II+HX2) is a combination of HX2 and remaining 42% (100 kW) of HX1. The design temperatures are also indicated at the inlets and exits in Figure 4.

On this modified arrangement, the dimension of two heat exchangers is determined with the same geometric parameters listed in Table 2. When the effective width and number of layers are 0.5 m and 50, respectively, the required length of heat exchanger is approximately 1.2 m for both exchangers. Figure 5 shows graphically how compactly these heat exchangers can be assembled in a cylindrical cryostat with estimated scale.



(a) First design (b) Final design

Figure 4 Arrangement of heat exchangers in Brayton cryocooler.

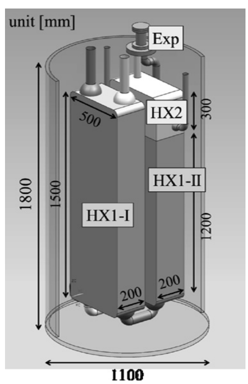


Figure 5 Assembled heat exchangers in cryostat.

DISCUSSION

Thermal performance and pressure drop of designed PFHX's can be simulated with commercial software such as *Aspen MUSE*. The dimensional factors obtained from the software are in a fairly good agreement with the presented design above. The pressure drop is estimated with a reasonable accuracy to be less than 1 kPa at the core of plate fins, but is not so easy for the headers and ports. According to the experience and sample test results by the Korean PFHX manufacturer, the total pressure drop is expected in the range of 10~20 kPa, which is well below the assumed values in thermodynamic cycle[1].

A drawback of PFHX is that the heat conduction through plates in flow direction may be excessive. The axial conduction in heat exchangers results in a heat leak to the cold end or a loss of refrigeration. With the presented design, the axial conduction is estimated about 400 W, which is 4% of the cryocooler capacity. This loss may be reduced by using a thinner plate, insofar as the mechanical strength is allowed. In addition, the loss is significantly reduced, if the plate and fins are fabricated with stainless steel instead of aluminium.

Another issue in heat-exchanger design for Brayton cryocooler is the possibility of LN2 freezing. Since the lowest temperature of refrigerant is below the freezing point of nitrogen (63.2 K), liquid may be frozen near the exit wall of HX2. Yoshida et al. [3] presented a pre-heating scheme of refrigerant before the cooling of liquid. In present design, the pressure ratio of thermodynamic cycle is determined such that at normal operation, the wall temperature at LN2 exit is above the freezing point [1].

CONCLUSIONS

Plate-fin heat exchangers (PFHX's) with offset strip fin are designed as components of 10 kW Brayton cryocooler for Korean HTS cable application. In order to achieve the required thermal performance, small pressure drop, and compactness at the same time, the dimension of PFHX's is determined for He-He recuperator (HX1) and LN2 sub-cooler (HX2) through standard methods of engineering modelling and calculation. The designed heat exchangers will be shortly fabricated and evaluated as 1/5-scale prototype.

ACKNOWLEDGMENT

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