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STUDIES ON A HEAT EXCHANGER PRODUCING SUBCOOLED LIQUID HELIUM

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ABSTRACT

A heat exchanger for cooling the TRISTAN final focusing superconducting quadrupole magnet was studied. This heat exchanger could produce subcooled liquid helium of 20 g/s and 0.16 MPa below 4.4 K with saturated liquid helium at 0.11 MPa. For the heat transfer of single phase flow in the heat exchanger, the calculations with the Dittus-Boelter and Kutateladze equations agreed well with the measurements. For the heat transfer during the condensation process, the Shah equation including vapor quality was applied. The calculated heat transfer energy was twice as large as the measured value and the vapor effect on the heat transfer coefficient was not as large as that calculated by the Shah equation.

INTRODUCTION

As a cooling method for a large superconducting magnet system, though forced-flow cooling of helium is mainly applied, subcooled liquid helium is also one possibility, except for supercritical helium. Producing subcooled liquid helium, however, involves a condensation process of which the thermal characteristics have not been studied enough. In order to understand the process, we studied a heat exchanger which can supply subcooled liquid helium of 20 g/s and 0.16 MPa at 4.4 K. In the experiment, we produced subcooled liquid helium and measured the helium temperature at the outlet while changing the heat exchanging surface area. In calculating the temperature profile along the heat exchanger, heat transfer coefficients of forced-flow helium in the condensation process were estimated with the Shah equation¹. Earlier this equation was examined with R-11, R-12 and ethanol etc., but it was never applied to liquid helium. We tried to analyze the temperature profile along the heat exchanger with the Kutateladze², Dittus-Boelter and Shah equations.

EXPERIMENTAL SYSTEM

The heat exchanger of which the performance was estimated is shown schematically in Fig. 1. The heat exchanger was used in a prototype cryogenic system for TRISTAN final focusing insertion quadrupole magnets³. The heat exchanger was designed to produce subcooled liquid helium of 20 g/s and 0.16 MPa at 4.4 K. The heat exchanger was made of copper tubing 43 meters long and with inner and outer diameters of 17.05 mm and 19.05 mm, respectively. The material was phosphorized copper. The copper tubing was wound in a diameter of 140 mm at the first layer increasing to 300 mm at the outer layer. The total

Figure 1. Schematic view of the studied heat exchanger.

Figure 2. Flow diagram of the experimental system. TCGR and TPtCo represent the carbon glass resister thermometer and the platinum cobalt resister thermometer, respectively. P and PA represent the pressure gauge and the absolute pressure gauge, respectively.

number of turns was 63. The heat exchanger was located in a helium vessel whose height and inner diameter were 622 mm and 350 mm, respectively. The pressurized helium inside the tubing was designed to be heat exchanged with the saturated liquid helium outside the tubing.

INSTRUMENT

The temperatures of the helium which passed through the tubing were measured by carbon-glass resister thermometers just before the inlet and at the outlet of the heat exchanger. The saturated helium temperature outside the tubing was measured by a carbon-glass resister thermometer too. These thermometers had already been calibrated by the supplier, and we estimated that the measurement error of the temperatures was within 30 mK by immersing these thermometers in saturated helium and comparing these measured temperatures and the saturated temperature, which was measured with a precise pressure gauge. The pressure of saturated helium in the vessel was measured by an absolute pressure gauge. This pressure gauge is accurate to 7.8×10^2 Pa within the full range of 4.1×10^5 Pa. The mass flow rate of helium through the heat exchanger was calculated by subtracting the mass flow rate through the expansion turbines from the total mass flow rate through the refrigerator. The total mass flow rate was measured by an orifice flow-meter at room temperature, and the flow rate through the turbine was calculated from the pressure and temperature of helium at the inlet of the second turbine. The temperature was measured by a platinum-cobalt resister thermometer. The mass flow rate of helium through turbines was measured by quickly closing the control valve of the turbine circuit. This measurement was carried out 9 times. The difference between the measured mass flow rate through the turbines and the calculated one was 4.46 %, on an average, of the measured mass flow rate, and the standard deviation was 8.45 %. Important sensors are shown in Fig. 2. The liquid helium level sensor was installed in the vessel and the wetted surface of the heat exchanger was calculated at that level.

MEASUREMENT RESULTS

Figure 3 shows the temperature changes of helium at the tubing outlet and in the vessel. The helium pressures inside the tubing were 0.1602 ~ 0.1612 MPa and the pressures in the vessel were 0.1098 ~ 0.1135 MPa. The lateral axis represents a wetted length of the tubing calculated by the liquid helium level sensor. Figure 4 shows the mass flow rate and an enthalpy of helium at the inlet of the heat exchanger which was calculated from the temperature and the pressure of helium just before the inlet valve. Since the helium is in a supercritical condition before it reaches the inlet valve, we can estimate the enthalpy of the helium at the inlet of the heat exchanger. The measured mass flow rate varied from 20 to 23 g/s. The Reynolds number of helium in the tubing was greater than 5×10^5 , and the helium was in turbulent flow. The vapor qualities at the inlet were estimated from 35 % to 85 %. The helium condition at the inlet was dependent on the thermal condition of the cryogenic system, and the enthalpy and vapor quality changed with the liquid helium level in the vessel. The cryogenic system, however, was thermally stable when the measurement was carried out at each liquid helium level.

From Fig. 3, the helium temperature in the tubing at the outlet began to fall sharply at a length of 10 ~ 12 meters. At positions up to this point, the temperatures were constant and the liquid helium was considered to be in two phase flow. The average temperature and pressure in this region were 4.727 K and 0.1606 MPa, respectively. The saturated helium temperature of 0.1606 MPa was 4.752 K from a helium property⁴. From these results, it is considered that the systematic error was within 30 mK for this measurement. In this region, the heat transfer mechanism involves a condensation process. At positions beyond this point, the liquid helium was in single phase flow. The helium temperature at the outlet of the tubing was almost constant over 20 meters of the wetted length and the temperature reached around 4.39 K. For the 43 meters of wetted length, the helium temperature inside the tubing was 4.391 K at a helium temperature of 4.356 K in the vessel. Figures 5 and 6 show the results for helium at a pressure of 0.15 MPa inside the tubing. The helium temperatures show the same changes as the results of 0.16 MPa. The temperature difference between the helium in the tubing and in the vessel was 40 mK for the wetted length of 43 meters.

Figure 3. Temperature changes for 0.16 MPa helium in the tubing. T_{sat} and T_{sub} represent the temperature of saturated helium in the vessel and helium in the tubing, respectively. Up and down correspond to the conditions of increasing and decreasing liquid helium level in the vessel.

Figure 4. Enthalpy and mass flow rate changes for 0.16 MPa helium in the tubing. M and H represent the mass flow rate and the enthalpy of helium in the tubing, respectively.

NUMERICAL ANALYSIS

In order to calculate the temperature change of the helium inside the tubing, we used the following equations. The heat flux from the outer surface of the tubing to the liquid helium in the vessel was estimated by the Kutateladze equation;

$$Q_{out} = A_{out} \times 4.87 \times 10^{-11} \times \left\{ \frac{C_{pl}}{\epsilon \times \rho_v} \right\}^{1.5} \times \frac{\lambda_l \times \rho_l^{1.282} \times P^{1.75}}{\sigma^{0.906} \times \eta_l^{0.626}} \times (T_w - T_{hv})^{2.5}$$

Figure 5. Temperature changes for 0.15 MPa helium in the tubing.

Figure 6. Enthalpy and mass flow rate changes for 0.15 MPa helium in the tubing.

The heat flux from the helium inside the tubing to the inner surface of the tubing was estimated by the following equations;

$$Q_{in} = A_{in} \times H_{in} \times (T_{ht} - T_w)$$

For liquid helium in a single phase, the forced-flow heat transfer coefficient in the tubing was defined by the Dittus-Boelter equation;

$$H_{in}(S) = 0.023 \times Re^{0.8} \times Pr^{0.4} \times \frac{\lambda_l}{D}$$

For the two phase helium during condensation, the heat transfer coefficient was defined by the Shah equation;

$$H_{in}(T) = H_{in}(S) \times \left[(1 - x)^{0.8} + \frac{3.8 \times x^{0.76} (1 - x)^{0.04}}{P_{rd}^{0.38}} \right]$$

On the heat transfer surface in the steady state, $Q_{out} = Q_{in}$. In the thermal model, we neglected the temperature distribution in the copper wall and the heat exchange at the dry wall. Figure 7 shows the calculated results. The calculated helium condition inside the tubing at the inlet was 0.1606 MPa, 20.8 g/s of mass flow rate and 10.77 J/g of enthalpy. The helium pressure outside the tubing was 0.1103 MPa. The temperatures were calculated from the saturated pressure. This condition corresponds to the experimental results which are marked by arrows in Figs. 3 and 4. The entire coil of copper tubing is assumed to be wetted by the saturated liquid helium. As shown in Fig. 7, the helium temperature inside the tubing began to decrease 5 meters from the tubing inlet. Experimental results show that the temperature began to decrease at around 11.5 meters. From a comparison between the calculations and the experimental results, the calculated heat flux is two times larger than the heat flux estimated by the experiment. The calculations show that the wall temperature of the copper tubing decreased in the two phase region in spite of the constant helium temperature in the tubing and in the vessel. These are due to the vapor effect on the heat transfer coefficient. In this calculation, the vapor quality at the tubing inlet was 77.3 % and the heat transfer coefficient was enhanced to 3.6 times larger than the heat transfer coefficient of the single phase helium by the Shah equation.

DISCUSSION

For calculating the heat transfer between the subcooled helium and the saturated helium via the copper tubing, we applied the Dittus-Boelter and Kutateladze equations. With the 0.15 MPa data, we estimated the difference between the calculations and the measurements. As shown in Figs. 5 and 6, the enthalpy of helium at the inlet from 7 to 20 meters along the wetted length is almost constant and around 6 J/g. In this case, the helium temperature in the tubing began to decrease at 10.9 meters along the wetted length and reached a constant temperature of 4.38 K at 18.9 meters. The temperature decrease and the exchanged heat energy within this length were 0.31 K and 39.6 J, respectively. In the calculations beyond the temperature decreasing point, the temperature decrease and the exchanged heat energy within 8 meters were 0.29 K and 44.7 J, respectively. The difference between the calculations and the measurements was around 13 %.

As shown in Fig. 7, the calculated length of the two phase region was almost half of the measured length. Comparing with the data from E. Sauvage-Bouter et al.⁵, the flow pattern is estimated to be annular at the inlet and to change into intermittent billowing flow at 4 meters from the inlet. It is considered that the surface inside the tubing was wetted by the saturated liquid helium at 0.16 MPa. In order to estimate the heat transfer coefficient of the two phase region, the Dittus-Boelter equation was applied instead of the Shah equation. The helium was assumed to be single phase liquid. The calculated results are shown in Fig. 8.

Figure 7. Calculated temperature changes with the Shah, Dittus-Boelter and Kutateladze equations.

Figure 8. Calculated temperature changes with the Dittus-Boelter equation instead of the Shah equation.

The calculated length was almost the same as the experimental results. In this kind of heat exchanger with the condensation process, vapor can not flow faster than liquid because the crest touches the upper wall of the tubing in the case of intermittent flow of low vapor quality. It is then considered that the heat transfer coefficient is not enhanced as much as that calculated by the Shah equation.

CONCLUSION

A heat exchanger which can produce subcooled liquid helium was studied. It was made of phosphorized copper tubing 43 meters long and with inner and outer diameters of 17.05 mm and 19.05 mm, respectively. The results are as follows;

1. For helium conditions of which enthalpy was around 6 J/g at the inlet, the heat exchanger could produce subcooled liquid helium at 20 g/s, 0.16 MPa and below 4.4 K with saturated liquid helium of 0.11 MPa. In this case, the wetted length of 20 meters was enough to produce the subcooled liquid helium.
2. Heat transfer between the subcooled liquid helium and the saturated helium via copper tubing could be calculated with the Kutateladze and Dittus-Boelter equations. The calculations agreed well with the measurements.
3. In order to estimate the heat transfer during the condensation process, the Shah equation was applied but the measured heat transfer energy was half as much as the calculated energy. In this kind of heat exchanger, it is possible that the vapor effect on the heat transfer coefficient was not as much as that calculated by the Shah equation.

NOMENCLATURE

- A_{out} = outer area of heat transfer
 A_{in} = inner area of heat transfer
 C_{pl} = specific heat at constant pressure of the liquid
 D = inside diameter of the tubing
 H_{in} = heat transfer coefficient inside the tubing; (s) represents a single phase, (T) represents a two phase
 P = pressure of the boiling system
 P_{rd} = reduced pressure
 T_w = wall temperature of the tubing
 T_{hv} = helium temperature in the vessel

T_{ht} = helium temperature inside the tubing
 x = vapor quality
 ϵ = latent heat of vaporization at saturation
 ρ_v = density of the vapor
 ρ_l = density of the liquid
 λ_l = thermal conductivity of the liquid
 σ = surface tension between the liquid and its own vapor
 η_l = viscosity of the liquid
 Re = Reynolds number
 Pr = Prandtl number

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