Design and test of the 4.5 K supercritical Helium coiled sub-cooler

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Abstract. Helium forced flow cooling was demonstrated as a reliable way for superconducting (SC) magnets. For the large Cable-in-Conduit Conductors (CICC) in high magnetic field, supercritical helium forced flow cooling is utilized. A coiled sub-cooler for 4.5 K supercritical helium, which is the vital component of the system, has been designed and manufactured to supply required helium for the SC magnet. The coiled sub-cooler installed in the valve box of the SC magnet has been tested during the experiment process of SC magnet. In this paper, the design details and the heat transfer performance of the coiled sub-cooler are presented. The analysis of test results reflects the design principles for the coiled sub-coolers.

1. Introduction

The 40 T hybrid magnet which has been built at High Magnetic Field Laboratory of the Chinese Academy of Sciences, consists of the outsert superconducting magnet and the insert resistance magnet. The superconducting magnet with a bore diameter of 580 mm at room temperature can provide 11 T and the resistive magnet with a bore diameter 32 mm can produce 29 T.[1] The outsert superconducting magnet is constructed by cable-in-conduit conductor (CICC). The inner layer is wound by Nb\textsubscript{3}Sn conductor while the outer layer is wound by NbTi conductor.[2] The superconducting magnet which was cooled by 4.5 K forced-flow supercritical helium contains 11 tons total cold mass and will store 102 MJ energy at the nominal operation current of 14.5 kA.[3] A sub-cooler is used to supply the satisfactory supercritical helium to ensure the relatively stable working condition for superconducting magnet.

The sub-cooler locates inside the valve box system of the 40 T hybrid magnet and is immersed in the saturated liquid helium in the sub-cooling tank. Helium from the cold box of the cryogenic system enters the valve box system, approximately 7 K @ 4.5 barA. After exchanging heat with the return cold helium gas, it cools down to about 6 K. Subsequently the incoming flow enters the sub-cooler, causing the temperature to decrease to about 4.5 K. The sub-cooled helium is delivered to the superconducting magnet system as a coolant. The cryogenic system can provide a cold mass flowrate about 30 g/s to the superconducting magnet to eliminate heat leakage load and self-heating load. Along the sub-cooler, the temperature change of the helium is less than 2 K, and the mean temperature difference is less than 1 K.

Conventional heat exchanger design is usually carried out by logarithmic mean temperature difference (LMTD) method or effectiveness-number of transfer unit method.[4] Usually, the fluid properties and heat transfer coefficient are assumed to be constant. Actually, the thermal property changes rapidly as the pre-condition changes. Meanwhile, the Reynolds number of the fluid are very large beyond the application range of many empirical formulas. So, the sub-cooler cannot be simply designed using conventional methods.
In summary, this paper focuses on following points. (1) Based on the distributed parameter differential method, the calculation model of the coiled sub-cooler is proposed. (2) The sub-cooler is manufactured according to the design scheme and tested under different working conditions. (3) According to the test results, the design method of the sub-cooler is summarized.

2. Design of the sub-cooler

2.1. Model theory

The lumped parameters method (LPM), the distributed parameters method (MDT), and the stream evolution method (SEM) are often used to design heat exchangers.[5] In order to improve the calculation accuracy and simplify the calculation, the distributed parameter differential method is adopted for design. It is assumed that the heat exchanger design is carried out under steady condition while neglecting the wall longitudinal heat conduction and the environmental heat load. The sub-cooler is divided into elements of some given physical length, in a geometry-oriented approach. Moreover, the thermal hydraulic calculations are separately done in each element. The temperature field of the whole heat exchanger is obtained through iteration.

2.2. Distributed parameter differential method description

As shown in the figure 1, the sub-cooler is divided into n elements, and each of which contains cold fluid, hot fluid and heat exchange wall. It is assumed that the outer wall of the hot fluid is adiabatic boundary, the outer wall of the liquid bath is isothermal boundary. Each element is independent from the outside environment.

![Figure 1. The distributed parameter differential method.](image)

In the equations, the subscript h and c represent the hot fluid and the cold fluid, the subscript i represents the element number. The Re, Pr and Nu respectively represent the Renolds number, Prandtl number and Nusselt number. The T and P represent the temperature and pressure.

For each element, the heat flux is obtained by heat balance. In the following equation, h and A represent the heat transfer coefficient. The Q and H represent the quantity of heat and enthalpy.

\[ h_{i-1} A_{i-1} (T_{h,i-1} - T_{c,i-1}) = Q_{i} = m_{h}(H_{h,i} - H_{h,i-1}) \]  

For hot fluid, as the forced convection heat transfer in the tube, the Nusselt number is calculated by Gnielinski equation:[6]

\[ Nu_{h,i} = \frac{\left(\frac{d}{l}\right)(Re_{h,i} - 1000) Pr_{h,i}}{1 + 12.7 \left(\frac{d}{l}\right)^{\frac{2}{3}} \left(Pr_{h,i}^\frac{3}{2} - 1\right)} \]  

In equation (2), \( c_i \) is the bend-correcting factor. \( f \) is the Darcy drag coefficient of turbulent flow, which calculated according to the Filonenko equation. The \( d \) and \( l \) represent the equivalent diameter and the tube length.

As \( \zeta \) is the drag coefficient that related to Re, \( \kappa \) is the correction coefficient of coiled tube, \( d_e \) and \( g \) represent the equivalent diameter and the acceleration of gravity. The pressure drop \( \Delta P \) is calculated: [7]
\[ \Delta P_{h,i,j} = \zeta \Delta x \frac{\gamma u_{h,i,j}^2}{d_c} \frac{K}{2g} \quad (3) \]

Considering different regimes of heat transfer, the heat transfer in the saturated liquid helium bath sits in the nucleate boiling regime. The heat transfer in nucleate boiling regime is described by Kutateladze correlation. [8] And the convective heat transfer coefficient could be calculated by the above-mentioned complex Kutateladze correlation.

The thermal conductivity of the tube wall is selected according to the temperature of the tube wall.[9] And the helium properties are obtained from HEPAK. The sub-cooler is immersed in the saturated liquid helium bath, so the pressure and temperature in the liquid bath is identical everywhere.

\[ T_{h,0} = T_{h,\text{in}}, T_{h,n} = T_{h,\text{out}}, P_{h,0} = P_{h,\text{in}} \quad (4) \]
\[ T_{c,n} = T_{c,\text{in}}, P_{c,n} = P_{c,\text{in}} \quad (5) \]

The calculation preconditions are as following table:

| Table 1. Precondition of the sub-cooler design. |
|-------------|-------------|-------------|-------------|-------------|-------------|
| $T_{h,\text{in}}$ | $P_{h,\text{in}}$ | $T_{h,\text{out}}$ | $P_c$ | $T_c$ | Mass flowrate |
| 6 K          | 5 barA      | 4.52 K      | 1.3 barA | 4.5 K      | 30 g/s       |

2.3. **Sub-cooler Design results**

According to the assumptions, the sub-cooler is simplified to the quasi-one-dimensional problem by the distributed parameter differential method. The calculation program is written in Python. During the iteration, the local physical property is obtained according to the local temperature and pressure.

**Figure 2.** The sub-cooler schematic diagram. **Figure 3.** The temperatures distribution along the length.

The coiled tube sub-cooler is adopted due to its simple structure, low cost and easy operation. Combined the sub-cooling tank structural requirement and the mass flowrate requirement, the sub-cooler is set to two coils structure as showing in figure 2. And copper is selected as the tube material. The wall thickness of the sub-cooler tubes is 2 mm according to the pressure requirement.

The diameters of the tubes will influence the convective heat transfer coefficient inside the tubes. Therefore, the calculation results of sub-cooler length vary with the tube diameter. Meanwhile, the pressure drop between the inlet and outlet will also be affected. Calculation results show that the tube with 18 mm outer diameter gets the overall best performance.

The sub-cooler with 18 mm outer diameter is about 25 m. According to the size of the sub-cooling tank, the tubes will wind 8 layers with coiling diameters about 1040 mm and 1100 mm. The temperature distribution along the length is as shown in figure 3.
3. The sub-cooler test results and analysis

3.1 Test results
The sub-cooler is tested when it is completely submerged in the liquid helium. The sub-cooler is tested by measuring the state of the inlet and outlet under different flowrate and pressure conditions.

![Figure 4. The sub-cooler temperatures change with mass flowrate.](image1)

![Figure 5. The sub-cooler outlet pressure changes with the mass flowrate.](image2)

Figure 4 shows the effect of the sub-cooler inlet and outlet temperature changes as the mass flowrate changes. The sub-cooler inlet temperature increases as the mass flowrate increases. As the inlet valve opens, the mass flowrate of the sub-cooler inlet increases and the inlet pressure increases. As shown in figure 5, in the experimental verification range, the sub-cooler outlet pressure, that is, the magnet inlet pressure, increases monotonically with the increase of the flowrate. At a flowrate of 30 g/s, the sub-cooler outlet pressure is about 5 barA. The increase of pressure in the magnet cooling passage causes the pressure increase in the sub-cooling tank, and the saturation temperature of the liquid in the subcooling tank increases. The change of the liquid helium bath temperature results in the change of the outlet temperature of the sub-cooler. In figure 4, $T_{\text{bath}}$ basically keeps pace with $T_{\text{bath}}$ revealing that the increase of the sub-cooling tank pressure will cause the rise of outlet temperature. Therefore, it is necessary to pay extra attention to controlling the pressure of the sub-cooling tank.

Referring to the definition equation of heat exchanger effectiveness, the sub-cooler effectiveness $\varepsilon$ is calculated as following equation:

$$\varepsilon = \frac{T_{\text{in}} - T_{\text{out}}}{T_{\text{in}} - T_{\text{bath}}}$$

As showing in figure 6, the sub-cooler effectiveness increases with the mass flowrate. When the flowrate is small, the Reynolds number in the tube increases as the flowrate increases, and the turbulent disturbance in the sub-cooler gradually develops, which optimizes the heat exchange between the hot fluid and the wall surface. The heat exchange effectiveness increase. When the flowrate is bigger than

![Figure 6. The sub-cooler effectiveness $\varepsilon$ changes with the mass flowrate.](image3)
25 g/s, as the flowrate increases, the effectiveness growth of the sub-cooler slows down with the increase of the flowrate. This is because the effect of reducing the total thermal resistance is not obvious by promoting the forced convection heat transfer. In this scenario, the thermal resistance is dominated by the natural convection heat transfer outside the tube. As shown in the figure 6, the sub-cooler gets high effectiveness when the flowrate is between 25 g/s and 32 g/s.

3.2 Analysis of test results
In the test result, when the flowrate is 30 g/s, the outlet temperature of the sub-cooler is 4.71 K, which is 0.19 K higher than the design result. The experimental results show that under the design condition, the effectiveness of the sub-cooler reaches 88.74%.

There are several reasons why the experimental results deviate from the design values. (1) The pressure in the sub-cooling tank is about 1.35 barA, which is higher than the design working condition of 1.3 barA. (2) The is axial heat conduction is ignored during the design process. The axial heat conduction can cause internal energy loss and weaken the heat transfer. [10] (3) The ambient heat load, which increases the outlet temperature of the sub-cooler, is ignored in assumptions. (4) Temperature and pressure measure errors will have impact on the test results, and the thermometer self-heating will influence the results.

3.3 Guide for the design of the sub-cooler
The test result of the sub-cooler, which designed by the distributed parameter differential method, deviates from the design condition. In the application, it is necessary to compensate the calculation results for the impact of the assumptions.

To further optimize the effect of the sub-cooler, more attention should be paid to decrease the thermal resistance of the natural convection heat transfer outside the tube. Improvement can be made by adding fins outside the tube.

4. Conclusion
The distributed parameter differential method is proposed to design the sub-cooler, and the test result shows that the sub-cooler can supply satisfied helium to magnet. The fluid properties get dramatic change near the critical region, so the accurate properties are need during design. The deviations between the design results and the experimental results are explained. In addition, some suggestions to improve the heat transfer effect of the sub-cooler are proposed.

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