Performance analysis for 2K heat exchanger for superfluid cryogenic system at KEK

A Kumar¹, H Nakai², K Nakanishi², H Shimizu², Y Kojima², K Hara² and T Honma²

¹SOKENDAI (The Graduate University for Advanced Studies), Tsukuba 305-0801, Japan
²High Energy Accelerator Research Organization (KEK), Tsukuba 305-0801, Japan

Email: ashish@post.kek.jp

Abstract. The niobium superconducting radio frequency cavities (1.3 GHz) operate at temperatures of 2.0 K or below, cooled with superfluid helium. A 2K heat exchanger increases the production rate of superfluid helium by reducing the inlet liquid helium temperature from 4.4 K to 2.2 K or above, recovering sensible heat of evaporated gaseous helium from the helium tanks of the cavities at 2.0 K. Effectiveness of a heat exchanger is the ratio of actual heat transfer to maximum possible heat transfer between the fluids. The required minimum effectiveness is 85% to cool liquid helium down to 2.2 K. At KEK, we have laminated-fip type 2K heat exchangers for thermal loads up to 100W. The effectiveness of the heat exchanger has been determined by computational flow dynamics, and the results are being verified experimentally using a heat exchanger test stand, specifically designed to test performance of 2K heat exchangers.

1. Introduction
The recent operating temperature of the 1.3 GHz niobium superconducting radio frequency cavity (SRF) is 2.0 K or below and such low operating temperatures are achieved with superfluid helium (T < 2.17 K). To achieve such low temperatures a 2K heat exchanger (2K HX) in series with a Joule-Thomson (JT) valve is employed in the cryogenic system. A 2K heat exchanger is required for liquid helium (LHe) temperature reduction from 4.4 K to ~2.2 K or above before the JT valve using sensible heat capacity of cold helium gas (GHe) at 2.0 K evaporating from superfluid helium tank of SRF cavities. Lower inlet temperature results in a flash loss reduction from 44% to ~10% after JT expansion and hence increased production rate of superfluid helium [1]. The JT valve maintains level and pressure of superfluid helium in the helium tank of SRF cavity. To cool liquid helium to approx. 2.2 K or above before the JT valve, minimum effectiveness of 85% is required for the 2K heat exchanger.

1.1. 2K heat exchanger at KEK
The 2K heat exchangers consists of helically coiled tubes and laminated fins made up of oxygen free copper (OFC) with two variants: Type I (old version) for heat loads until 30W and Type II (new version) for heat load up to 100W in countercflow arrangement as shown in Figure 1 and 2. The fins provide larger surface area for efficient extraction of enthalpy from LHe flowing through the helical tubes. The focus of this study will be on effectiveness determination of type II 2K HX.
2. Theoretical effectiveness for 2K heat exchanger

The effectiveness of type II 2K HX can be calculated using effectiveness-NTU (Number of transfer units) method. The known operating conditions of fluids and heat exchanger material properties for the operating range of heat exchanger are given in Table 2.

The GHe flow stream through type II 2K HX is complicated, due to the presence of helical tube and fins arranged perpendicular to flow stream. The flow nature will be perpendicular to 1st loop with six fins (Figure 3), but as the GHe flow past it, a wake region is developed behind the fins due to sharp angle of attack (90 degrees). In the wake region, the angle of attack would not be perpendicular to the next arrangement of fins, since the helical pitch is not large enough for GHe flow to develop and spatial repetition of fin arrangement causes the flow to take periodic structure.

2.1. Heat transfer coefficient of gaseous helium and liquid Helium

To get an initial estimate of heat transfer coefficient of GHe for effectiveness determination, a case of perpendicular flow over a flat plate is considered and applied for the whole heat exchanger to compare the theoretical and numerical effectiveness of type II 2K HX [2].

Nusselt number, Nu_c, is defined with the heat transfer coefficient for a flow over a flat plate, h_c, as;

\[
Nu_c = \frac{h_c D}{k} = 0.228 \times Re^{0.731} \times Pr^{0.33} ,
\]  
(1)
where D is the hydraulic diameter, Re the Reynolds number and Pr the Prandtl number.

Nusselt number for LHe in a helical tube, \(Nu_h\), is expressed, from the Schmidt relation, as;

\[
Nu_h = \frac{h_h D}{k} = \left\{1 + 3.6 \left[1 - \frac{a}{R}\right]^0.8\right\} Nu_s, \quad \text{and} \quad Nu_s = 0.021 \times Re^{0.8} \times Pr^{0.4},
\]

(2)

where \(h_h\) is the heat transfer coefficient for LHe, and \(a\) and \(R\) the radius and the curvature radius of the helical tube.

The effectiveness of the type II 2K HX for a counterflow arrangement, \(\varepsilon\), is determined by,

\[
\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\text{max}}} = \frac{1 - \exp[-NTU(1 - C^*)]}{1 - C^*\exp[-NTU(1 - C^*)]},
\]

(3)

where the total heat transfer from LHe to GHe, \(\dot{Q}\), is expressed as;

\[
\dot{Q} = \varepsilon \times \dot{Q}_{\text{max}} = \varepsilon \times (m c_{\text{min}}) \times (T_{hi} - T_{ci}) \ [W],
\]

(4)

and \(C^*\) is specific heat capacity ratio \((C_{\text{min}}/C_{\text{max}})\), NTU number of transfer units \((UA/C_{\text{min}})\) and UA the overall thermal conductance determined from the heat transfer coefficient of LHe and GHe.

Effectiveness of the heat exchanger from current relations was 83% at 37 W \((1.5 \text{ g/s})\) and would reduce to 76% for 105 W \((4.5 \text{ g/s})\) of thermal load to 2.0 K superfluid helium. This is due to heat transfer coefficient \((h_c)\) being directly proportional to Re raises to a factor ‘n’, where n is always < 1, and hence \(h_c\) will not rise with the same rate as the mass flow rate of GHe. To determine precise fluid flow through the heat exchanger, it is analysed numerically using computational flow dynamics (CFD).

3. Numerical setup for type II 2K heat exchanger in ANSYS CFX®

ANSYS CFX® for CFD is employed to determine the performance of the type II 2K HX. The fluid conditions are assumed to be steady state with properties varied with respect to pressure and temperature \([3, 4]\). The heat exchanger consists of three domains: LHe, GHe and heat exchanger domain separating the two fluid domains, as can be seen in Figure 4. Meshes were created using proximity and curvature criteria. Mesh independency tests were carried out to determine minimum mesh size, when the solution becomes independent of the mesh resolution. To accurately capture the velocity gradient near walls inflation meshes were created over fins and in the LHe flow volume to consider the \(Y^+\) values for turbulence modelling. K-\(\varepsilon\) scalable turbulence model was chosen for LHe domain due to Re >100,000 and the \(Y^+\) was around 17 for LHe flow domain. For GHe flow domain, the Reynolds number is very low, so k-\(\omega\) SST Automatic model is used to effectively resolve the boundary layer with \(Y^+ < 1\) \([5]\). The boundary conditions for the CFD analysis at 37 W of thermal load to superfluid He is given in Table 3.

| Boundary  | Material | Mass Flow [g/s] | Turbulence Intensity [%] | Static Temperature [K] | Total Pressure [kPa] |
|-----------|----------|-----------------|--------------------------|-------------------------|-----------------------|
| GHe Inlet | GHe      | 1.55            | 4.4                      | 2                       | -                     |
| GHe Outlet| GHe      | -               | -                        | -                       | 3                     |
| LHe Inlet | LHe      | 1.55            | 3.6                      | 4.4                     | -                     |
| LHe Outlet| LHe      | -               | -                        | -                       | 125                   |
| OFC       | OFC      | Interface between LHe and GHe |               |                          |                       |
| Wall      | SS304    | Adiabatic       |                          |                          |                       |

Table 3. Boundary conditions for type II 2K HX in Ansys CFX®.
To determine the performance of heat exchangers experimentally, a test stand primarily consisting of dummy load tank at 2.0 K, a LHe storage tank, a heat exchanger, a JT valve and control valves is employed. Temperature and pressure sensors at specific positions measure the state of fluids. LHe flows through the 2K HX via the JT valve to the dummy load tank. Evaporating GHe from dummy load tank flows through the 2K HX to GHe pumping system, while exchanging enthalpy with LHe and cooling it. The 2K HX is being currently tested on the test stand and the results will be presented in the future.

4. Results
The effectiveness of the type II 2K HX is calculated from the results obtained from CFD simulations and relations available from the effectiveness-NTU method. NTU is derived from effectiveness and logarithmic mean temperature difference (LMTD) obtained from CFD data due to variable UA and specific heat $C_p$ for the fluids. Due to computation restrictions, the analysis had to be tackled section by section and extrapolated to get precise effectiveness from relations available from Newton’s law of cooling and the effectiveness-NTU method [2]. The effectiveness of a heat exchanger, $\varepsilon$, and the logarithmic mean temperature difference, $\Delta T_m$, are expressed as:

$$\varepsilon = \frac{Q}{Q_{\text{max}}} = \frac{UA\Delta T_m}{(\rho c_{\text{min}}) \times \Delta T_{\text{max}}} = \text{NTU} \quad \frac{\Delta T_m}{\Delta T_{\text{max}}}, \quad (5)$$

$$\Delta T_m = \left[ (T_{hi} - T_{co}) - (T_{ho} - T_{ci}) \right] \ln \left( \frac{T_{hi} - T_{co}}{T_{ho} - T_{ci}} \right). \quad (6)$$

![Figure 5](image)

**Figure 5.** Comparison of obtained effectiveness from CFD and effectiveness-NTU method. The results had to be extrapolated to determine the precise effectiveness of type II 2K HX.

**Table 4.** Summarized results obtained from ANSYS CFX®.

| Parameters                                      | Type II 2K Heat Exchanger |
|------------------------------------------------|--------------------------|
| Effectiveness (\(\varepsilon\)-NTU) [%]       | 83                       |
| Effectiveness (CFD) [%]                        | 73                       |
| Heat transfer coefficient for GHe (\(\varepsilon\)-NTU) [Wm²K⁻¹] | 44                       |
| Heat transfer coefficient for GHe (CFD) [Wm²K⁻¹] | 30                       |
| Exit temperature for LHe (CFD) [K]             | 2.59                     |
| Flash loss for LHe (CFD) [%]                   | 14.4                     |
| Pressure drop in GHe (CFD) [Pa]                | 8                        |

| Parameters                                      | Type II 2K Heat Exchanger |
|------------------------------------------------|--------------------------|
| Effectiveness (\(\varepsilon\)-NTU) [%]       | 76                       |
| Effectiveness (CFD) [%]                        | 68                       |
| Heat transfer coefficient for GHe (\(\varepsilon\)-NTU) [Wm²K⁻¹] | 96                       |
| Heat transfer coefficient for GHe (CFD) [Wm²K⁻¹] | 68                       |
| Exit temperature for LHe (CFD) [K]             | 2.78                     |
| Flash loss for LHe (CFD) [%]                   | 16.5                     |
| Pressure drop in GHe (CFD) [Pa]                | 68                       |
The obtained effectiveness for type II 2K HX from CFD is lower than the calculated one in section 2.1 as seen in Figure 5. Increasing thermal load to superfluid helium reduces effectiveness from 73% at 1.55 g/s to 68% at 4.5 g/s flow rate and hence deteriorating the performance of type II 2K HX. Some of the obtained results from CFD and ε-NTU are summarized in Table 4. The presence of holes on the fins helps in reducing pressure drop of GHe, but it also reduces heat transfer capability of the heat exchanger. As the effectiveness reduces, the flash loss from LHe expansion through JT valve increases causing reduction in superfluid helium production.

Due to the periodic structure of flow as seen in Figure 6, the effective $h_c$ for the GHe is lower than the initial estimate hence reducing the effectiveness. When the flow rate increases to 4.5 g/s, the heat transfer coefficient decreases from 85 [W/m$^2$-K] for 1st loop to 68 [W/m$^2$-K] for the whole heat exchanger, as shown in Figure 7. Increase in mass flow rate of ~3 times yields in ~2.3 times increase in heat transfer coefficient, which will reduce effectiveness from approximately 73% at 1.55 g/s to 68% at 4.5 g/s flow rate and hence deteriorating the performance as thermal load increases. From pressure drop data, it seems that GHe flow attains stability after 8 loops and is directly proportional to square of mass flow rate ($m^2$). At 105 W of thermal load to superfluid helium, the pressure drop in GHe through the heat exchanger is approximately 68 Pa.

5. Conclusion
The effectiveness of the heat exchanger was determined from ANSYS CFX® and compared with the results obtained from initial theoretical analysis. It showed that the heat transfer capability from LHe to GHe is hindered due to presence of stacked fins perpendicular to flow, hence deteriorating the performance of the heat exchanger. The results are being verified with the heat exchanger test stand.

References
[1] Nakai H et al 2014 Superfluid helium cryogenic systems for superconducting RF cavities at KEK AIP Conference Proceedings pp 1349-1356 American Institute of Physics
[2] Shah R K and Sekulic D P Fundamentals of Heat exchanger Design (John Wiley & Sons)
[3] Hepak Software
[4] Cryogenic Technologies G. (2010, 03 02) Material Properties: OFHC Copper (UNS C10100/C10200) Retrieved from NIST
[5] ANSYS CFX-Solver theory guide