RECONDENSER PERFORMANCE: IMPACT ON THE SUPERCONDUCTING UNDULATOR MAGNET AT ARGONNE NATIONAL LABORATORY

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Abstract

The current sharing temperature of 6.5 K for the superconducting undulator magnet being developed at Argonne National Laboratory drives the thermal design of the magnet's cooling system. In order to remain below the current sharing temperature, a thermo-siphon cooling loop is being developed to sweep the anticipated heat load away from the magnet windings and deposit it in the associated liquid helium reservoir located above the magnet. Performance of the magnet's cooling system is crucially dependent on the ability of the recondenser to maintain the reservoir's saturation temperature near 4 K, despite thermal stratification and slowly varying thermal profiles within the vapor region above the liquid in the reservoir. Here we report the results of a modelling investigation regarding the impact of various heat transfer mechanisms in a recondensing system, on the time-varying saturation temperature within the helium reservoir. Future experimental work is being done to verify the modelling results.

PROJECT BACKGROUND

The project analyses the helium recondensing system comprising a part of the cryogenic cooling system for the Superconducting Undulator (SCU) assembly of the Advanced Photon Source (APS) at Argonne National Lab. The project's objective is to optimize the use of a cryocooler in order to efficiently remove heat from the thermo-siphon cooling loop that itself removes heat from the magnet windings on the SCU assembly. A liquid helium bath is included as part of the closed-loop system in order to maintain cooling in the event of a power loss to the cryocooler. Thus, this project investigates the best possible way to remove heat from the liquid helium bath. Condensation tip geometries and heat transfer paths were studied in order to determine their impact on the overall heat removal.

At the cold-tip of the cryocooler, it is best to design a surface that will allow the condensed helium film to remain thin in order to promote heat transfer. A thicker film will increase the thermal resistance between the gas and solid, thereby decreasing the condensation rate and decreasing the heat transfer ability of the condenser tip.

More significantly, this study found that other factors in the heat transfer path between the helium bath and the cold tip of the cryocooler were more important than the geometry of the condenser tip. Therefore, this paper provides a broad methodology for optimizing the cooling associated with a re-condensing liquid helium bath.

MODELLING METHODOLOGY

Figure 1 displays the thermal resistance network associated with the heat transfer paths in the system. Each resistance region was characterized separately using a different type of modeling software. The results were compared in order to determine the relative significance of the various thermal resistances. Engineering Equation Solver (EES), Finite Element Heat Transfer (FEHT), and Ansys CFX software were used to separately calculate the thermal resistances associated respectively with the film condensation, conduction through the vessel wall, and convection in the gas.

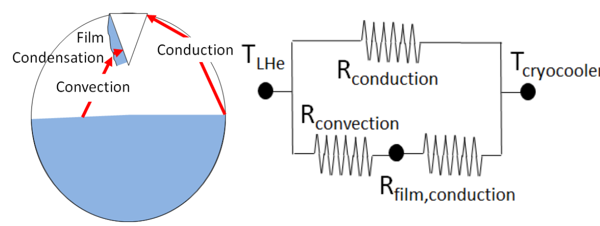

Figure 1: Thermal Resistance Network.

FILM CONDENSATION (EES)

At the cold tip of the cryocooler, gas condenses, coalesces into a film layer and runs down the condenser until it drips off the bottom of the condensing surface. As the liquid film flows toward the bottom of the condensing surface, it grows thicker. As given by theoretical considerations (see for example [1]), the film thickness at a distance x from the top of the condensing surface is calculated according to

$$\delta = \left\{ \frac{4\ x\ k_{l,sat}\ \mu_{l,sat}(T_{sat}-T_s)}{\rho_{l,sat}\ g\ \cos\theta(\rho_{l,sat}-\rho_{v,sat})\left[\Delta i_{vap}+\frac{3\ c_{l,sat}(T_{sat}-T_s)}{8}\right]} \right\}^{1/4} \quad (1)$$

where $k_{l,sat}$ is the thermal conductivity, $\mu_{l,sat}$ is the viscosity, $\rho_{l,sat}$ is the density, T_{sat} is the saturation temperature, T_s is the condenser tip surface temperature, g is the gravitational constant, θ is the condenser surface angle from vertical, Δi_{vap} is the latent heat of vaporization, and $c_{l,sat}$ is specific heat.

In order to estimate the thermal resistance of the entire film layer, the condensing surface has been divided into multiple sections of equal distance dx, each with their own individual film thickness and associated thermal resistance (or conductance), see fig. 2. The net conductance is then determined by integrating the individual section conductances over the total length (L) along the condensing surface

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$$\dot{C}_{net} = \int \left(\frac{1}{R_{net}}\right) = \int_{x=0}^{x=L} \left(\frac{k\,A(x)}{\Delta\delta(x)}\right) dx \qquad (2)$$

where *A(x)* is the surface area of the individual cone section and *k* is the thermal conduction through the film.

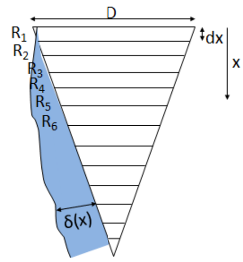

Figure 2: Film Condensation Modelling Schematic.

This integrated approach has been applied for both conical and spherical condensation tip geometries. For the same surface area of about 200 cm^2, the calculations reveal the film condensation thermal resistances given in table 1 for the cone and spherical tips.

Table 1: Condenser Geometry Resistance Comparison

Condition	Thermal Resistance
$R_{convection,cone}$	≈ 20 K/W
$R_{film\ condensation,cone}$	≈ 0.06 K/W
$R_{convection,sphere}$	≈ 17 K/W
$R_{film\ condensation,sphere}$	≈ 0.11 K/W

VESSEL CONDUCTION (FEHT)

A cylindrical vessel with approximate dimensions of 146 cm length by 31.59 cm diameter serves as the holding tank for the liquid helium reservoir. In the present model, a single condensing cryocooler is mounted at the top center of the reservoir. In order to mimic the reservoir design being used for Argonne's superconducting undulator, the reservoir wall includes a 0.475cm thickness of stainless steel and an equal thickness of copper. Finite Element Heat Transfer (FEHT) software was used to model conductive heat transfer through the curved walls of the helium reservoir between the upper surface of the liquid helium and the cryocooler. The tank is assumed to be half filled with liquid. Two different 2D geometries were used to approximate the vessel's 3D geometry.

In the first approximation, the upper half of the cylindrical helium vessel is envisioned as a flat plate with the cryocooler (shown in the center, red section of fig. 3). The meshing can be seen by the various nodes. The grey region represents the vessel wall comprised of the dual layers of copper and stainless steel. An effective thermal conductivity was calculated for the combined material at cryogenic conditions by multiplying each of the individual material's thermal conductivities by their fractional thickness in the wall and then adding these together. The boundary conditions for the analysis include a cryocooler temperature of *3.2 K* and a helium liquid temperature of *4.2 K.*

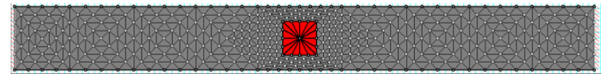

Figure 3: Cartesian Coordinate FEHT Analysis.

The second 2D approximation uses an axisymmetric approach by placing the cryocooler at the center of a round plate (see fig. 4). The upper, red portion of fig. 4 is the copper cross-section of the wall, while the lower grey portion is the stainless steel material. The left surface in fig. 4 is fixed at the cryocooler temperature while the outer right surface is the liquid helium temperature.

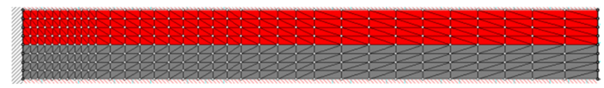

Figure 4: Cylindrical Coordinate FEHT Analysis.

The two different 2D methods give comparable values of thermal resistance with the average value appearing in table 2.

VAPOR CONVECTION (ANSYS CFX)

Convective heat transfer through the helium vapor was characterized using Ansys CFX software. The investigation focused on the influence of vessel and condenser geometry on convection. Temperature profiles for many different vessel geometries are shown.

Vessel Geometry

A variety of sizes and shapes for the helium reservoir have been included in the convection models. The thermal resistance between the liquid helium surface and the condenser tip was determined for each combination of the condenser tip and reservoir geometry. A few of the combinations are shown in fig. 6. The model calculates the time dependent heat transfer through the 3D cylindrical shape generated by rotating the images shown in fig. 6 around its vertical center axis.

The baseline geometry uses a height of 15.24 cm and width of 47.85 cm. Additional geometries varying the aspect ratio and size of the vapor region are included for comparison. Note that the graphic's scaling is different, so the sizes shown can be compared using the scaling shown on each trial.

The temperature profile of the helium vapor is shown throughout the vessel space. The results show that the equilibration time increases dramatically with the size of the vessel, an observation that is consistent with a simple scaling analysis based on the thermal or mass diffusion time constants.

$$\tau_{therm} = \frac{L^2}{\alpha_{He}} \qquad (3)$$

Figure 5 shows the characteristic times associated with thermal diffusion (τ_{therm}) for helium vapor near 4.2 K given the thermal diffusivity (α_{He}). Note that for characteristic lengths (*L*) on the order of 1 meter, the time constants grow to hundreds and even thousands of hours.

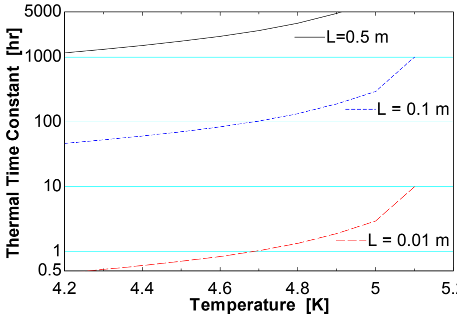

Figure 5: Thermal Time Constants for Helium.

The temperature profiles (in units of Kelvin) in the helium vapor reveal that the time dependent cooling load at the cryocooler cold tip is dominated by the convective process in the vapor, and only over very long time scales impacts the temperature of the liquid mass. Without heat transfer through the walls, the steady-state conditions for the 0.78 m characteristic length envisioned for Argonne's liquid helium reservoir will require thousands of hours.

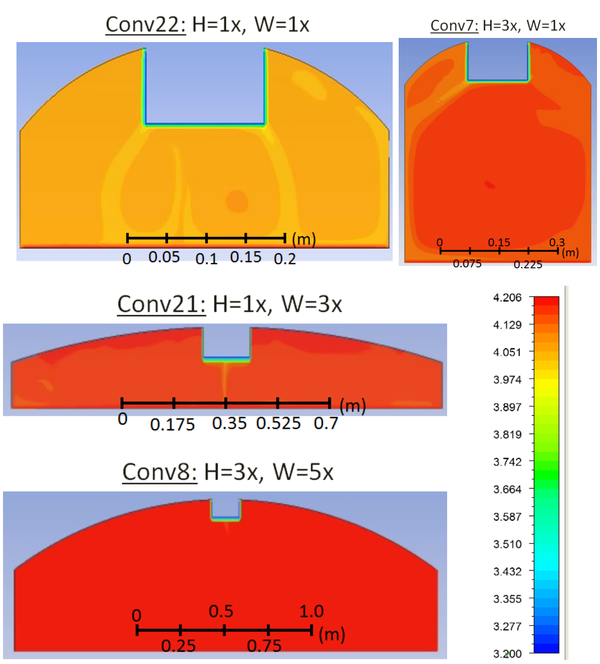

Figure 6: Ansys CFX Vapor Temperature Profiles.

Condenser Geometry

Two condenser geometries, conical and hemispherical, were included in the analyses. The thermal resistance to heat transfer between the surface of the liquid helium reservoir and the condenser tip was found. Table 1 lists the thermal resistance values for the baseline reservoir geometry and each of the condenser tips along with the condensed film resistance values determined with EES.

MODELLING RESULTS

Table 2 lists the approximate thermal resistance values for each individual heat transfer mechanism described in this paper. The thermal path through the vessel wall provides a much smaller thermal resistance than the path through the vapor region and condenser tip.

Table 2: Condenser Geometry Resistance Comparison

Condition	Thermal Resistance
$R_{wall\ conduction}$	≈ 0.5 K/W
$R_{convection}$	≈ 20 K/W (Dominant)
$R_{film\ conduction}$	≈ 0.1 K/W

EXPERIMENTAL WORK

An experimental apparatus is being assembled to verify the expected heat transfer characteristics from the model. The experimental work will investigate the time required for steady state conditions as a function of the associated length scales, and will measure the temperature profile throughout the system components. The helium boil off rate will be found for different heat loads and condenser designs.

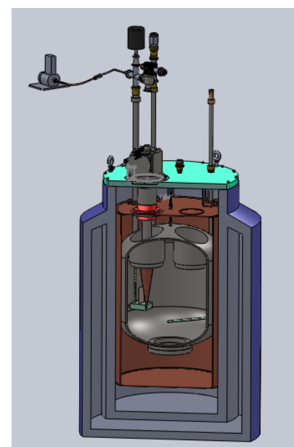

Figure 7: Experimental Setup Schematic.

CONCLUSIONS

Modelling results reveal that the thermal resistance between the liquid surface in a helium reservoir and the recondenser cold tip can be extremely large and slow to equilibrate if the primary heat path is through the helium gas. Such a constraint can be significantly reduced by using enhanced conductive paths through the reservoir wall. Reducing the characteristic length by using extended surfaces in the vapor region will also reduce the thermal resistance much more than by optimizing the geometry of a condenser surface.

ACKNOWLEDGMENT

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REFERENCES

[1] G.F. Nellis and S.A. Klein, *Heat Transfer,* (New York: Cambridge University Press, 2009), 804.