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HEAT EXCHANGER DESIGN STUDIES FOR AN LHC INNER TRIPLET UPGRADE

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ABSTRACT

A luminosity upgrade of the CERN Large Hadron Collider (LHC) is planned to coincide with the expected end of life of the existing inner triplet quadrupole magnets. The upgraded inner triplet will have much larger heat loads to be removed from the magnets by the cryogenics system. A number of cryogenics design studies have been completed under the LHC Accelerator Research Program (LARP), including investigations of required heat exchanger characteristics to transfer this heat from the pressurized He II bath to the saturated He II system. This paper discusses heat exchangers both external to the magnet cold mass and internal to the magnet cold mass. A possible design for a heat exchanger external to the magnet cold mass is also presented.

KEYWORDS: He II heat exchanger, LHC upgrade, quadrupole

INTRODUCTION

After several years of operation, the final focusing inner triplet quadrupoles of the LHC will need to be replaced. This replacement is planned to coincide with a luminosity upgrade of the machine. A number of such scenarios are under study, including large-aperture, high-gradient Nb_3Sn quadrupoles; large-aperture, low-gradient NbTi quadrupoles; and forward quadrupoles and/or dipoles placed near the interaction regions.

An important consideration for any luminosity upgrade scenario is the removal of heat deposited in the magnets, as much as 1200 W at the 1.9 K temperature level. For comparison, the estimated beam induced heat load to the current LHC inner triplet quads is approximately 200 W. One facet of this is the sizing of cold mass cooling channels and cryostat piping to maintain acceptable temperature drops within the cold mass pressurized He II bath. A second facet, the topic of this paper, is the design of a heat exchanger to transfer this heat from the cold mass pressurized He II bath to the saturated He II system with an acceptable temperature drop.

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One way to remove heat from the magnet cold masses is the use of a heat exchanger external to the cold mass. Two heat exchanger designs are discussed here: the bayonet heat exchanger and the shell and tube heat exchanger. A second way to remove heat from the magnet cold masses is the use of a heat exchanger internal to the cold mass.

EXTERNAL BAYONET HEAT EXCHANGER

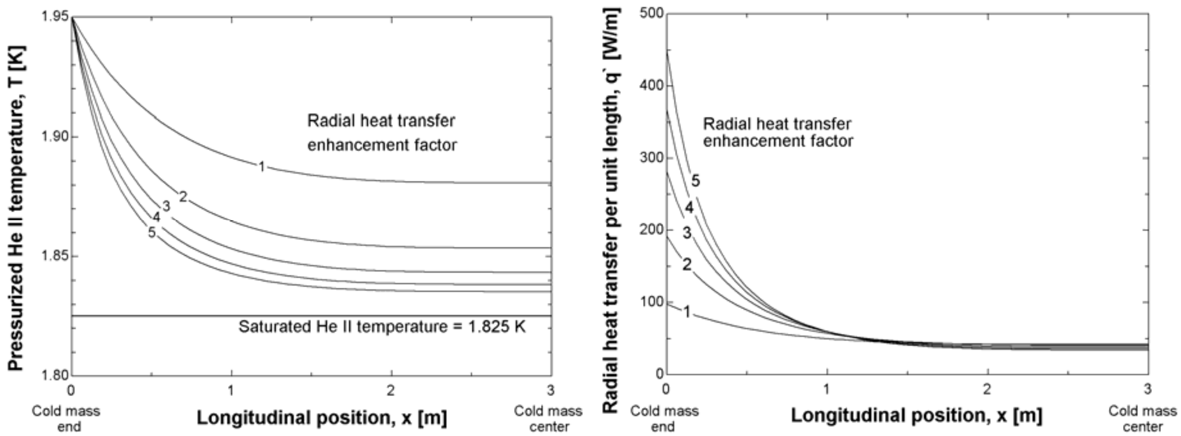
A bayonet heat exchanger, conceptually described elsewhere [1], is used to cool the existing inner triplet. The heat exchanger for the current LHC inner triplet quads is a copper corrugated pipe of 9.6 cm outer diameter and 8.5 cm inner diameter. Saturated He II flows inside of this corrugated pipe. Pressurized He II in an annular volume outside the corrugated pipe is contained by a 16 cm inner diameter pipe. It is estimated that approximately 22% of the corrugated pipe inner surface will be wetted by liquid He II when the existing inner triplets are in operation. Analysis has been completed to look at how much heat could be removed by a bayonet-style heat exchanger with a larger fraction of its surface wetted. The location the highest heat load is at the non-IP end of Q1 with a heat load of 329 W, 250 W from the quadrupole cold mass and 79 W from the corrector cold mass.

A finite difference model of a 3 m length of bayonet heat exchanger was constructed. The temperature of the pressurized He II at the inlet of the heat exchanger pipe was specified as 1.950 K. The temperature of the saturated He II inside the corrugated pipe was specified as 1.825 K. The final boundary condition was a thermal gradient of zero at the other end of the 3 m length due to symmetry.

Figures 1a shows the calculated longitudinal temperature profile in the pressurized He II as a function of radial conduction enhancement with 90% of the corrugated pipe internal surface wetted. Figure 1b shows the calculated longitudinal distribution of radial heat transfer per unit length as a function of radial conduction enhancement, where 1 signifies no enhancement, 2 is a two-fold increase in heat transfer conductance both inside and outside the corrugated pipe, etc. Large temperature gradients in the pressurized He II as a result of longitudinal conduction resistance results in a large fraction of the heat transfer occurring at the end of the cold mass. The heat transfer conductance enhancement significantly improves the heat transfer at the end of the cold mass but has little effect toward the longitudinal center of the cold mass.

However, nearly the entire surface one the saturated He II side of the corrugated pipe must be wetted to even approach the required heat transfer rate of 329 W over a 3 m length. Vapor velocity and liquid carryover would certainly become an issue. The use of many fins on both the inside and outside surfaces of the corrugated pipe or other significant heat transfer enhancement would also be required, making manufacturing and handling difficult and impractical.

A single bayonet heat exchanger as used in the existing inner triplet would not be sufficient for the upgraded inner triplet. The thoroughly-studied bayonet heat exchanger design could be used by cooling the inner triplet with multiple bayonet heat exchangers or a single, scaled-up version of the existing design.



(a) (b)

FIGURE 1. (a) Pressurized He II temperature and (b) radial heat transfer per unit length vs. longitudinal for a bayonet heat exchanger with a 90% wetted surface and varying degrees of heat transfer enhancement.

EXTERNAL SHELL AND TUBE HEAT EXCHANGER

A possible new heat exchanger for the LARP inner triplet quadrupoles is shown in Figure 2. The heat exchanger is alongside the magnets. The tube side is connected to the magnet pressurized He II bath, and the shell side contains saturated He II with a smooth-walled pumping line above. The heat exchanger consists of 37 12.7 mm outer diameter smooth copper tubes. The bundles of tubes are brazed to the end plate and inserted into a 250 mm stainless steel pipe so that the shell side will be filled with saturated He II while flowing. Each heat exchanger is approximately 6 m long, the same as an individual magnet. The pumping pipe is about 30 m long, the length of the inner triplet.

The proposed heat exchanger has several advantages over the corrugated pipe heat exchanger currently used: it is easy to manufacture; heat transfer surface area is predefined if the heat exchanger shell side is filled with saturated He II; and the pumping line is a smooth pipe so that a relatively smaller pipe is needed to meet the strict requirement of small pressure drop. The pumping line will have the same slope as the LHC tunnel slope to allow flowing saturated helium to flow from a higher point to a lower point.

Pumping Line Pipe Design

The pressure drop of saturated He II flowing in the pumping line can be calculated by equation (1).

$$\Delta P = f \frac{L}{D} \frac{\rho v^2}{2} \tag{1}$$

where f is the friction factor, L the pipe length, D the pipe inner diameter, ρ the helium vapor density, and v the vapor flow velocity. The required helium mass flow is determined by the estimated heat load to the inner triplet quadrupoles and vapor quality after the Joule-Thomson (J-T) valve. Assuming the heat load is 1200 W and liquid fraction after the J-T valve is 87%, the required maximum vapor helium flow is 60 g/s at the pumping line pipe exit. The vapor pressure drop-induced equivalent temperature drop as a function of pipe dimensions and helium properties at 1.8 K is shown in Figure 3.

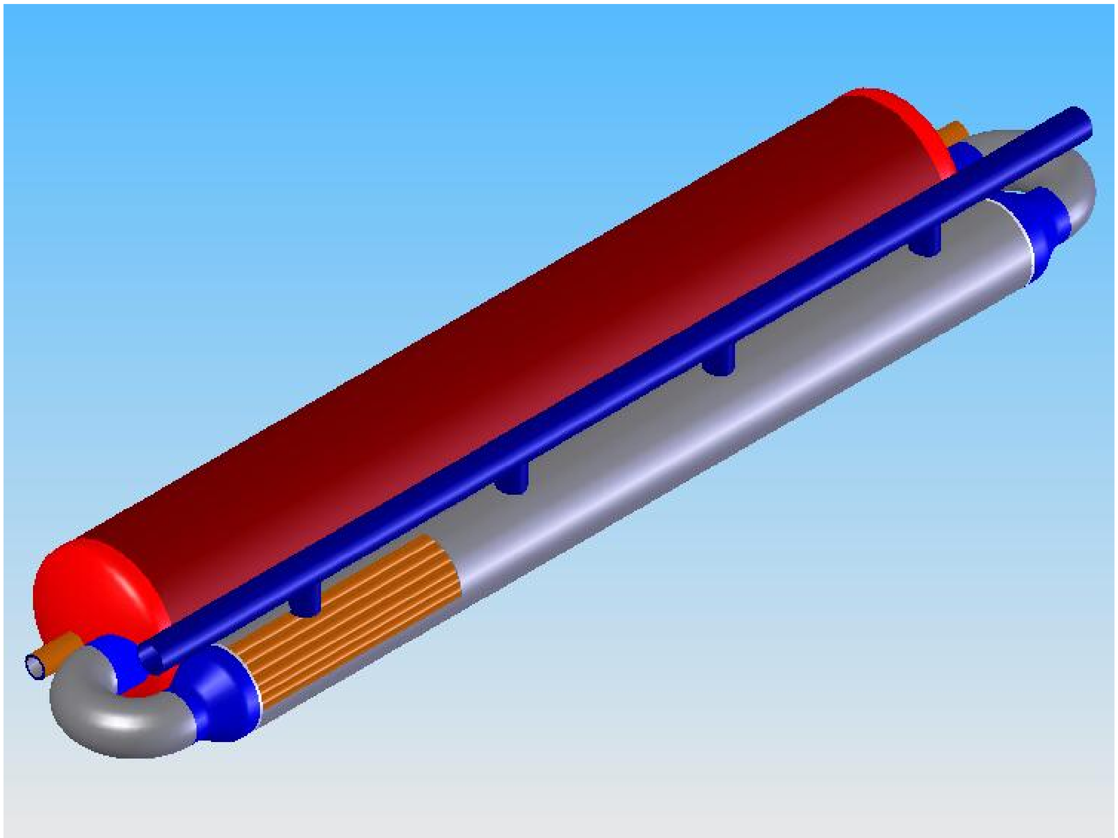


FIGURE 2. A possible heat exchanger for the upgraded inner triplet.

For a 100 mm diameter, 30 m long pipe, the calculated pressure drop is only 1.5 Torr if the helium vapor flow is 60 g/s in the whole pipe length. The corresponding temperature drop is 33 mK.

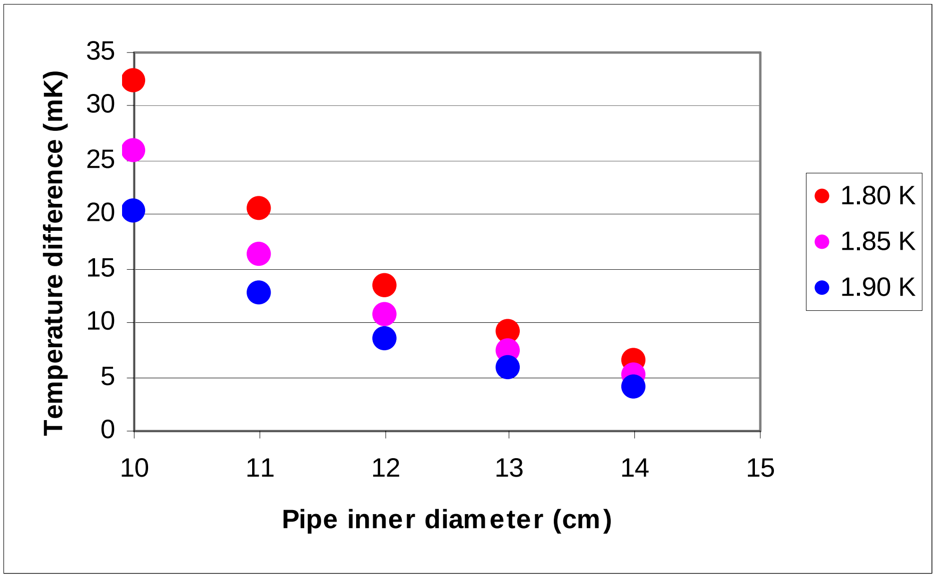


FIGURE 3. The pumping line pressure drop-induced equivalent temperature drop as a function of pipe dimensions and helium properties at 1.8 K.

Heat transfer within pressurized He II on tube side

The inner triplet quadrupole coils are cooled with pressurized He II. Beam-induced heat loads are conducted away by the internal convection of stagnant helium. The analysis presented here focuses on the heat exchanger thermal analysis, sizing the required He II heat transfer cross-sectional area and the heat transfer surface area required between the pressurized He II and saturated He II. For a constant heat flux along the heat exchanger length, the temperature difference is proportional to the heat flux raised to the third power and the heat exchanger length as shown in equation 2.

$$\Delta T = \frac{q^3 L}{1.2} \tag{2}$$

where q is the heat flux in W/cm^2, L is half length of heat exchanger, ΔT is the required temperature difference in mK.

If heat is carried away linearly by the heat exchanger surface along its length, then the temperature difference will be a quarter of the value calculated from equation (2) as shown in equation (3).

$$\Delta T = \frac{q^3 L}{1.2 \times 4} \tag{3}$$

The calculated thermal performance of the shell and tube heat exchanger is summarized in Table 1.

Heat transfer across heat exchanger surface wall

The thermal barrier between He II and heat exchanger surface wall is dominated by Kapitza thermal resistance. From the engineering point of view, the temperature drop across the heat exchanger wall can be estimated by equation (4).

$$\Delta T = \frac{q}{h} \tag{4}$$

where q is the heat flux across the heat exchanger surface in W/m^2 and h is the effective heat transfer coefficient in W/m^{2}K. The calculated temperature drop on one side of the heat exchanger is only 5.2 mK for a given heat flux of 13.6 W/m^2 and Kapitza thermal conductance of 2610 W/m^{2}K. The total temperature drop between the pressurized He II and the saturated He II will be 10.4 mK.

TABLE 1. Calculated thermal performance of the shell and tube heat exchanger pressurized He II side.

Heat Load in each heat exchanger	300 W
Heat exchanger tube outer diameter	27.9 mm
Heat exchanger tube wall	0.86 mm
Heat exchanger tube inner diameter	26.2 mm
Number of heat exchanger tubes	37
He II cross sectional area	200 cm^2
Half length of heat exchanger	350 cm
Heat flux in the He II channel	0.75 W/cm^2
Temperature difference	31 mK

TABLE 2. Calculated thermal performance of the shell and tube heat exchanger saturated He II side.

Heat Load in each heat exchanger	300 W
Heat exchanger shell outer diameter	273.05 mm
Heat exchanger shell wall	4.191 mm
Heat exchanger tube inner diameter	264.7 mm
Cross sectional area for saturated He II	323 cm^2
Half length of heat exchanger	350 cm
Heat flux in the He II channel	0.464 W/cm^2
Temperature difference	7.3 mK

Heat transfer within saturated He II on shell side

The thermal analysis on the saturated He II side of the heat exchanger is the same as that of the pressurized He II side. The calculated temperature drop on the saturated He II side under the required heat load and the dimensions of the heat exchanger is listed in Table 2. Due to the relatively larger cross sectional area, the temperature drop on the saturated He II side is only 7.3 mK.

Other considerations

The maximum slope of the LHC tunnel is 1.26%, which translates into an elevation change of 88 mm over a horizontal length of 7 m. The shell side of the heat exchanger is filled as flowing saturated He II is supplied from the higher end of the triplet. As long as the vertical pipe connecting the pumping line and the heat exchanger shell is at least 88 mm long, the entire tube bundle will be submerged in liquid helium with only saturated helium vapor flowing in the pumping line.

Bellows will be located on the heat exchanger shell to absorb the thermal contraction during the cool down and warm up of the inner triplets.

The material of the heat exchanger will be stainless steel except the copper tubes to transfer the heat load between the two helium media. A brazed transition joint will connect the heat exchanger end plate and the copper tubes. Since the pressurized He II is contained inside of the tubes and the connection pipe between the heat exchanger and the cold mass, only those spaces are subject to the quench pressure of up to 20 bar. The pumping line pipe and heat exchanger shell side volume are subject only to the pressures of the LHC pumping system.

INTERNAL HEAT EXCHANGER

A second way to remove heat from the magnet cold masses is the use of a heat exchanger internal to the cold mass. The use of an internal heat exchanger will certainly require multiple heat exchanger tubes to distribute the heat flow within the cold mass and keep the cold mass cooling channel sizes reasonable. An advantage of using an internal heat exchanger is that the thermal path from the pressurized He II to the saturated He II is shortened and will make available more temperature margin.

The internal heat exchanger studies assume a cold mass cross-section of a large aperture Nb_3Sn quadrupole [2]. These cold masses have a 400 mm outer diameter with a 90 mm beam pipe aperture. There are also eight large longitudinal cooling channels, each with a cross-sectional area of 50 cm^2. Each cooling channel could accommodate a number of heat exchanger tubes, such as one 45 mm (1.75 in) diameter tube or seven 19 mm (0.75 in) diameter tubes.

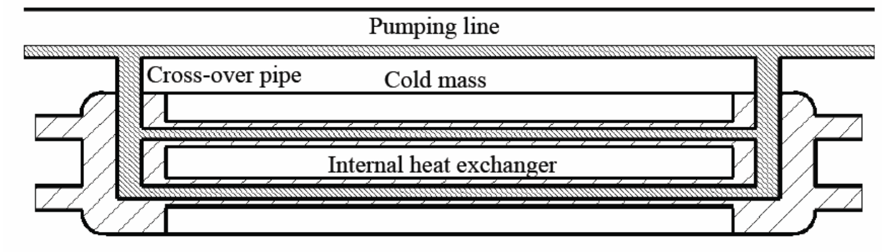


FIGURE 4. Schematic representation of an internal heat exchanger for the upgraded inner triplet.

One possible operating mode for an internal heat exchanger is to completely fill all the heat exchanger tubes with He II, filling just up into the pumping line which is assumed to be above the cold masses as shown in Figure 4. Two effects must then be considered: longitudinal conduction, and the height of the liquid column.

The use of multiple smaller tubes is preferable in this case. The longitudinal temperature drop is reduced due to the larger He II cross-sectional area. The radial temperature drop through the heat exchanger tube wall is also reduced due to the larger He II-tube interface surface area.

The height of the liquid column from the heat exchanger tubes to the He II liquid surface sets another limitation on the allowable temperature drop. A large temperature drop from the magnet center to the liquid surface will result in the local saturation temperature being exceeded. Boiling will then occur, drastically changing the system heat transfer characteristics. The shaded area of Figure 5 estimates the range of column heights for the cold mass and pumping line placement considered here. For a 250 W distributed heat load within a quadrupole, the calculated temperature drop from the magnet center to the magnet end is at least 150 mK, depending on the quantity and size of heat exchanger tubes used. From Figure 5, this would result in boiling within the magnet and is therefore an unacceptable cooling scheme.

The use of an internal heat exchanger would therefore require multiple heat exchanger tubes partially filled with He II, essentially multiple bayonet heat exchangers. Additional study would be required to determine the feasibility of this option.

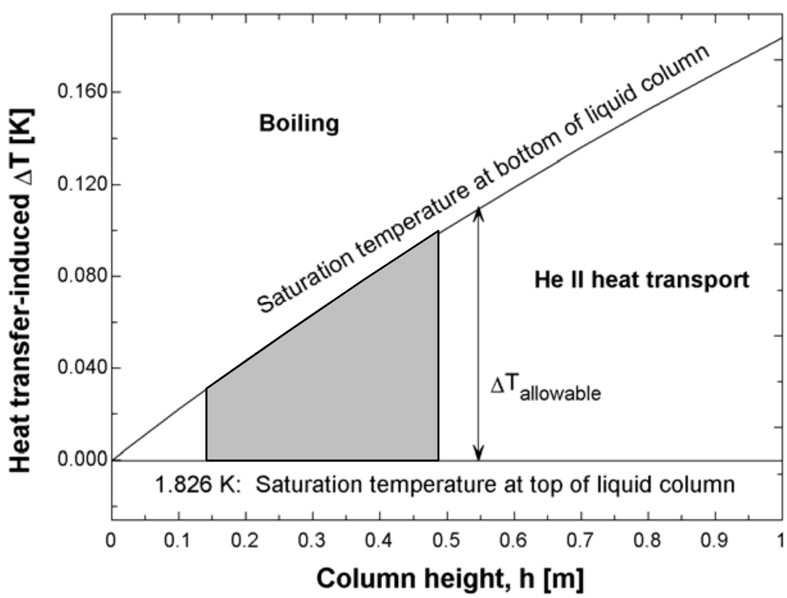


FIGURE 5. Heat transfer-induced temperature drop ΔT vs. column height for boiling and He II heat transport.

CONCLUSIONS

Design studies have been conducted to understand the requirements of a He II heat exchanger for removing the beam-induced heat load of an upgraded inner triplet.

The external bayonet heat exchanger currently in use is not capable of handling the increased heat loads. Nearly the entire surface would need to be wetted with He II, and significant heat transfer enhancement also would be required.

The design and analysis of a shell and tube heat exchanger, located alongside the magnet cold masses, has been presented. This is a feasible, relatively straightforward design.

Use of an internal heat exchanger would require multiple bayonet-style heat exchangers in order to prevent localized boiling within the saturated He II system.

The design study results presented here are a first step towards understanding the requirements of a heat exchanger for an upgraded inner triplet. All design options must be evaluated in terms of both thermal and mechanical considerations as well as integration with the LHC cryogenics system.

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