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Superfluid Helium cooling and compact Heat Exchanger for HL-LHC D2 Recombination Dipoles

B Rousset[1], F Bancel[1], S Claudet[2], F Millet[1], P Nivelon[1], A Perin[2], R Van Weelderen[2]

[1] Univ. Grenoble Alpes, CEA INAC-SBT, F-38000 Grenoble, France
[2] CERN, CH-1211 Geneva 23, Switzerland

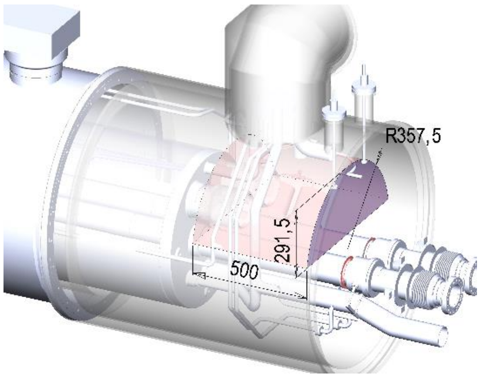
Corresponding author : bernard.rousset@cea.fr

Abstract. New D2 recombination dipoles with a larger aperture than in the LHC dipoles are required for the future **H**igh **L**uminosity LHC at CERN. These 13.5 m-long D2 magnets are proposed to be conduction cooled in a static bath of pressurized He II. Their cooling is provided via pressurized He II channels located in the D2 iron yoke and thermally connected to a saturated bath installed at one end of each D2 dipole. The heat transfer between the pressurized He II static bath and the bath pumped down to 16.4 mbar (1.8 K) is performed in a heat exchanger under study at CEA. Various design solutions were studied and evaluated to define the more suitable solution fulfilling on the one hand D2 cooling requirements (up to 70 W) and on the other hand D2 cryostat integration constraints. The paper will report on the D2 cooling needs and constraints, present the studied options and detail the main design features of the selected solution for a compact heat exchanger for D2 dipoles.

1 Introduction

The High Luminosity Upgrade of the LHC (HL-LHC) at CERN will provide instantaneous luminosities up to five times larger than the LHC nominal value. To do so, most existing LHC superconducting magnets will be replaced, including the D2 recombination dipole magnet. This 13.5 m-long magnet is in a “standalone” configuration and is characterized by a beam induced heat load of 70 W that is mostly localized at one end of the magnet as shown in figure 1. An exploratory study and an evaluation of different cooling architectures and technology options for such 13.5 m-long stand-alone magnets operating in superfluid helium have been presented in a recent paper [1]. The selected baseline solution consists in immersing the magnet in a static bath of pressurized He II at 0.13 MPa and to cool it by heat transport through He II; a Tore Supra-like solution [2]. The magnet cooling is provided via pressurized He II channels in the D2 iron yoke thermally connected to a saturated He II bath installed at one end of the dipole. The heat transfer between the pressurized He II volume and the saturated He II bath is performed with a heat exchanger HX-D2 as shown in Figure 1. Several design solutions were studied and evaluated to define the more suitable one fulfilling both the D2 cooling requirements and the D2 cryostat integration constraints.

The heat exchanger HX-D2 shall remove the heat loads in the nominal conditions, i.e. to extract 70 W from the D2 magnet with a saturated bath at 1.8 K at the liquid-gas interface and a maximum temperature at the magnet interface of 2.044 K. The HX-D2 must also maintain this cooling capacity in the two following extreme situations: when the saturated bath is at 2 K or when the pressure of the pressurized He II bath is at 0.4 MPa as detailed in [1]. The D2 magnet will be installed in the LHC tunnel where there are strict limitations on both the longitudinal and the transversal spaces. It is therefore of primary importance for the heat exchanger to be compact to ease its integration into the magnet cryostat.



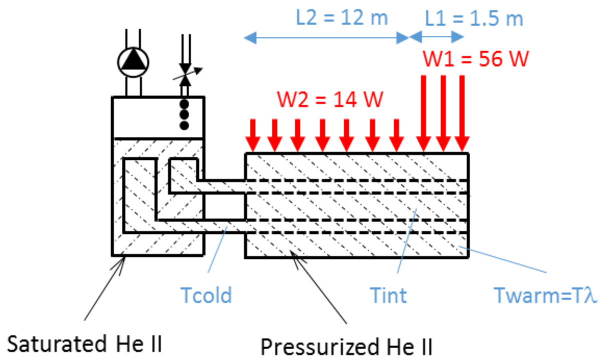


Figure 1. HL-LHC D2 integration constraints (left) and cooling scheme with distributed heat loads (right) [1]

2 Design assumptions and parameters

The following assumptions and technical choices are made to fulfill the cooling requirements and to perform the present study:

- The heat exchanger core is to be made of identical copper sectors (e.g. pipes) working in parallel.
- The He II directly connected to the liquid He II bath is called hereafter the "saturated" He II even if it is slightly subcooled by the hydrostatic pressure of the liquid bath level.
- The ratio of the cross sections allocated to the pressurized He II and the saturated He II is inversely proportional to the ratio of their He II thermal conductivity function (TCF) at the power 1/3.4 as demonstrated in the following equations.
- The He II thermal conductivity function (TCF) is assumed to be constant in the pressurized and saturated baths (i.e. independent of the temperature for small temperature differences (~0.02 K)).
- A copper thermal conductivity of 6.T (W/m/K) with T in K is assumed. This value corresponds to a Cu-C1 copper with a RRR of about 5 (laboratory measurements).
- Due to small temperature differences along the heat exchanger, the Kapitza conductance between copper and He II is considered constant and fixed at $600.T^3$ (W/m²/K) with T in K [3].

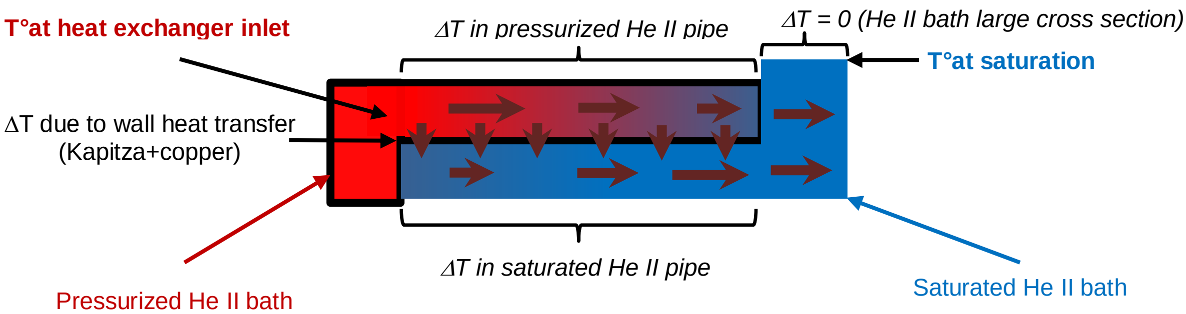


Figure 2. Heat transfer and temperature profiles in one heat exchanger sector

It should be noted that the copper wall does not represent the major thermal barrier compared to Kapitza resistances. Furthermore, the "errors" introduced with the above assumptions are negligible compared to the uncertainties due to the material and to the manufacturing choices; for an identical material, the Kapitza resistance between copper and He II may vary by a factor larger than 2 between dirty and clean surfaces (ranging from $400.T^3$ to $900.T^3$). It should also be outlined that the assumption for He II cross-section is in fact a design choice to keep the same temperature difference along the pressurized and the saturated He II sectors (figure 2). That also imposes that the pressurized He II sector (from the magnet towards the cold source) gradually releases the heat flux to the saturated He II sector.

To fulfill this criterion, we apply the "classical" He II heat transport equation in a 1D-pipe of length L and cross section area S with a heat flux q and a temperature difference. Due to the targeted small ΔTs in both sectors, the classical equation becomes for the pressurized and the saturated He II sectors:

$$T_{pres}(0) - T_{pres}(L) = \frac{L}{4.4\, g(\langle T\rangle,\mathrm{pres})} {q_{pres}}^{3.4}(0) \text{ and } T_{sat}(0) - T_{sat}(L) = \frac{L}{4.4\, g(\langle T\rangle,sat)} {q_{sat}}^{3.4}(L) \quad (1)$$

where $g(T,P)$ is the He II thermal conductivity function (TCF) at given pressure and temperature (The *sat* and *pres* indexes refer respectively to the saturated He II and the pressurized He II).

And consequently the ratio of He II cross sections is:

$$\frac{S_{sat}}{S_{pres}} = \left(\frac{g(\langle T\rangle,pres)}{g(\langle T\rangle,sat)}\right)^{\frac{1}{3.4}} \quad (2)$$

The longitudinal temperature difference ΔT_{long} along one of the HX-D2 sectors can then be easily calculated once the cooling power to be extracted, the cross sections and the length of the sectors are defined. The transverse temperature difference ΔT_{trans} through the pipe wall and the two Kapitza resistances are added to the longitudinal one applying the equation (3):

$$\Delta T_{trans} = \frac{W}{\pi.nb.d.L}\left(\frac{e}{k(\frac{T_{sat}+T_{pres}}{2})} + \frac{1}{h_{kap(T_{sat})}} + \frac{1}{h_{kap(T_{pres})}}\right) \quad (3)$$

with d pipe diameter, L pipe length, e wall thickness, k pipe wall conductivity, h_{kap} Kapitza resistance at liquid-solid interface, T_{sat} He II temperature inside the pipe (liquid He II close to saturation), T_{pres} He temperature at the outer pipe wall (pressurized He II) and *nb* the number of pipes r.

The total temperature difference ΔT_{total} between the temperature at the liquid-vapor interface and the pressurized helium at the magnet interface is the sum of ΔT_{long} , ΔT_{trans} and the temperature difference ΔT_{bath} across the He II bath liquid above the heat exchanger (ΔT_{bath} could be negligible in the case of a bath with a large cross section and limited liquid height above the heat exchanger). Finally, to prevent film boiling, it is necessary to check that the saturation line is not reached at any point inside the pipes filled with the saturated He II liquid. If not, the liquid level must be increased to fulfill the inequality (4).

$$P_{sat}(T_{sat\_max}) < P_{sat}(T_{sat\_bath}) + \rho.g.h \quad (4)$$

with T_{sat_max} the maximum temperature of saturated liquid, T_{sat_bath} the temperature of the liquid at the bath liquid-vapour interface, ρ the liquid density, g the gravity constant and h the difference in altitude between the maximum saturated temperature and the liquid-vapour interface locations.

3 Optimized design study

The combination of the previous assumptions and equations are applied to compare various possible heat exchanger designs. As the smallest cross section has to be allocated to the saturated He II (higher TCF), the logical choice is to consider pipes filled with saturated He II surrounded by pressurized He II. The problem to be solved consists then to define the number of pipes, their length and their orientation (horizontal versus vertical) and then the associated pressurized He II parameters. The equations above show that for a given total cross section and a given length, the longitudinal ΔT_{long} (due to He II heat transports) does not depend on the number of pipes, whereas increasing the heat transfer area (decreasing ΔT_{trans}) can be achieved by increasing the number of pipes. Practical considerations like the wall thickness and the minimum gap between pipes for manufacturing issues shall also to be considered together with design constraints due to the differential pressures that can reach 2 MPa in case of a magnet quench. Taking into account these inputs, Cu-C1 copper pipes, 10 mm inner diameter pipes with 1 mm thickness are selected for the heat exchanger. The cold source temperature being defined at the He II bath liquid-vapor interface, increasing the pipe length (while keeping other parameters constant) results in two opposite effects: on one hand, it increases the heat exchange surface and decreases the ΔT_{trans} but on the other hand, it increases the distance between the maximum temperature and the cold source which increases the ΔT_{long}. In addition, in the case of horizontal pipes, increasing the ΔT_{long} also results in an increasing demand for saturated He II subcooling, i.e. an increase in liquid level above the heat exchanger to comply with equation (4), which in turn can result in a slight increase in the overall ΔT and also an increase of the size of the heat exchanger system. So, even if it is not obvious, increasing the length of the heat exchanger while keeping its cross section constant can result in a higher total temperature difference. The general way to solve this kind of problem, i.e. to find the minimum of the function expressed in equation (5), is to select $L = \sqrt{\frac{A}{B}}$ and the corresponding ΔT is then defined by equation (6).

$$\Delta T = \Delta T_{total} = \Delta T_{trans\cdot} + \Delta T_{long\cdot} = \frac{A}{L} + B.L \tag{5}$$

$$\Delta T = \sqrt{AB} + \sqrt{BA} = 2\sqrt{AB} \tag{6}$$

In the present case, one obtains $A = \frac{\left(\frac{W}{nb.\pi\frac{d^2}{4}}\right)^{3.4}}{4.4\ g(\langle T\rangle,sat)}$ and $B = \frac{W}{nb.\pi d}\left(\frac{e}{k(\frac{T_{sat}+T_p}{2})} + \frac{1}{h_{kap(T_{sat})}} + \frac{1}{h_{kap(T_p)}}\right)$.

The values A and B depend on the number of pipes as well as on the total power resulting in an optimum length. Some space constraints may also impose a maximum length L_{max}. and a maximum cross section S_{max} for the vessel in which the heat exchanger is installed. In this case, the best design can be determined with an iterative method, taking into account these constraints, which should have a ΔT_{total} lower than ΔT_{max}. The maximum cross section is used to calculate the cross section allocated for pipes filled with saturated He II using equation (2), which gives the number of pipes. Then both the corresponding ΔT given by (1) and (3) and L can be calculated. If ΔT_{total} is higher than ΔT_{max}, some of the geometrical parameters (length, maximum cross section or pipe diameter) are adjusted and iterative calculations are performed to define the optimum parameters.

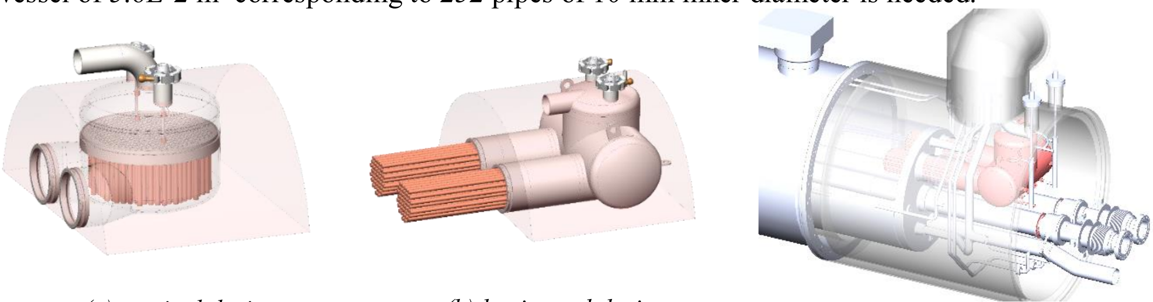
4 HX-D2 selected design

The HX-D2 design selection is performed assuming a heat load of 70 W to be extracted at $T_{pres\_max} = 2.044\ K$ and P=0.4 MPa with a cold source at $T_{sat} = 2\ K$, which correspond to the very worst cases. The interface holes in the D2 cold mass vessel are two tubes of 0.136 m inner diameter (i.e. $1.45\ 10^{-2}$ m² cross section area). Using Sato fits [4], the TCF is equal to $4.89\ 10^{14}$ at saturation pressure and 2.02 K and $1.59\ 10^{14}$ at 0.4 MPa and 2.04 K. The theoretical number of pipes can be expressed by the following equation:

$$nb = \frac{S_{tot} * f_1}{\frac{\pi d_{int}^2}{4} + f_1.\left(\frac{\pi d_{ext}^2}{4} - \frac{\pi d_{int}^2}{4}\right)} \quad \text{with } f_1 = \left(\frac{g(\langle T\rangle,P)}{g(\langle T\rangle,sat)}\right)^{\frac{1}{3.4}} / \left(1 + \left(\frac{g(\langle T\rangle,P)}{g(\langle T\rangle,sat)}\right)^{\frac{1}{3.4}}\right) \tag{7}$$

A numerical application gives 65 pipes, an optimal length of 1.2 m and a total ΔT of 13 mK. In reality, the pipes cannot be so long as the maximum available length L_{max} is limited to 0.1 m for vertical designs and to 0.5 m for horizontal designs. Additionally, pipes cannot be evenly spaced as they have boundaries (walls) and the real pipe length and number must be decreased accordingly, which can be done as the calculated ΔT is below the required value $\Delta T_{max} = 44\ mK$. Another possibility to provide more geometrical freedom is to install the heat exchanger away from the magnet and to select a larger cross section area than those allowed in the magnet interface holes. Figure 3a shows such an architecture with a vertical heat exchanger installed in an external saturated bath as validated in existing magnet He II cooling such as in Tore Supra [2].

The vertical pipes filled with saturated He II have the liquid–vapor interface just above their tops (minimal need of hydrostatic pressure subcooling). As the height of the saturated bath (liquid and vapor parts) must be small to fit inside the existing vacuum vessel, the length/height of heat exchanger pipes shall be small (0.1 m) and compensated by a larger number of pipes (far from the optimum defined previously). ΔT_{long} will be consequently small (<< 1 mK) and the ΔT_{total} is essentially defined by ΔT_{trans}. To fulfill the specifications (i.e. $\Delta T_{total} \leq 44$ mK), a minimum internal cross section area for the vessel of 5.6E-2 m² corresponding to 252 pipes of 10 mm inner diameter is needed.



(a) vertical design *(b) horizontal design* *(c) D2 integration of the final HX-D2*

Figure 3. Examples of studied HX-D2 designs and proposed HX-D2 integration.

A more compact solution consists to introduce a portion of the heat exchanger inside the end of the D2 cold mass vessel where some free volume filled with pressurized He II is present (figure 3c). Pipes filled with saturated He II are horizontal in this configuration (figure 3b). Such design allows longer pipes and a reduced pipe number. Following design iterations between geometrical constraints and thermal performance, we selected a horizontal design that uses the maximal allocated space and that offers thermal performance margins. The selected HX-D2 design has 52 pipes of a 0.5 m length per D2 hole and gives a ΔT_{total} < 25 mK (20 mK for ΔT_{trans} and 5 mK for ΔT_{long}). This horizontal design with penetrating tubes has been fully detailed and integrated in the magnet cryostat. A prototype of the HX-D2 is now under construction and will be tested at CEA/SBT in a multi test cryostat connected to a 1.8 K refrigerator [5] in 2019.

5 Conclusions and perspectives

The present paper describes a method to design He II-He II heat exchangers and applies it for the heat exchanger HX-D2 for the future high luminosity LHC recombination dipoles D2. Horizontal pipes filled with saturated He II and penetrating inside the extremity of the D2 cold mass vessel constitute the more compact and efficient solution. Manufacturing of a prototype is now in progress and thermal performance tests are scheduled at CEA/SBT 1.8 K test facility in 2019.

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