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Conceptual design and analysis of a cryogenic system for a new test facility for high temperature superconductor current leads (HTS CLs)

Abstract*. In view of the R&D program on HTS CL at EPFL-CRPP PSI Villigen it is foreseen to extend the capabilities of an existing facility (JORDI) in order to be able to perform tests of HTS CL in different conditions. The present work is focused on the conceptual design of the cryogenics needed for the facility upgrade considering the requirement of an as extended as possible range of operation. Two cooling options for the copper part of a CL are considered. The use of heat exchangers for the optimization of the cryogenic system is discussed and their size is assessed.*

Streszczenie. *W związku z trwającymi w EPFL-CRPP PSI Villigen pracami B&R nad nadprzewodnikowymi krioprzepustami prądowymi (HTS CLs) przewiduje się uruchomienie tam stanowiska eksperymentalnego do prowadzenia testów HTS CLs w różnych warunkach. W pracy zaprezentowano projekt koncepcyjny układu kriogenicznego dostarczającego chłodziwo do tego stanowiska. Rozważono możliwość dwóch opcji chłodzenia części miedzianej HTS CLs. Przedyskutowano konieczność zastosowania wymienników ciepła w układzie kriogenicznym i oszacowano ich rozmiar.(**Projekt koncepcyjny i analiza termodynamiczna układu kriogenicznego dla nowego stanowiska eksperymentalnego do testowania nadprzewodnikowych krioprzepustów prądowych**)*

Keywords: HTS current leads, thermal-hydraulic analysis, cryogenic system, heat exchanger.
Słowa kluczowe: nadprzewodnikowe krioprzepusty prądowe, analiza cieplno-przepływowa, układ kriogeniczny, wymiennik ciepła.

Introduction

Current leads (CLs) transfer the operating current from a room temperature power supply to a superconducting cable at cryogenic temperature. Conventional CLs are made of metals, whereas advanced high-current CLs consist of a metallic heat exchanger(HEX) part connected with either a low-Tc superconductor or high-Tc superconductor (HTS) material. It has been shown that the use of HTS CLs leads to a reduction of the refrigeration power consumption typically by a factor 3 with respect to the optimally designed conventional metallic CLs [1]. Further improvement of this promising technology would be advantageous. In particular, the development of new cooling concepts for HTS CLs is needed, aimed at reduction of the required amount of HTS material and further reduction of the refrigeration power consumption.

Experimental investigations of HTS CLs are foreseen at the EPFL-CRPP (Villigen PSI), which requires the extension of the capabilities of the existing JORDI facility in order to perform tests of HTS CLs in different cooling conditions. The present work is focused on the conceptual design and thermal-hydraulic analysis of the cryogenics needed for the facility upgrade considering the requirement of an as extended as possible range of operation. The designed cryogenic circuit should be capable to provide enough coolant for the most demanding planned test conditions. In our analysis we consider: (i) a variable helium mass flow rate at low temperature (LT) to obtain the required mass flow rate at a particular intermediate temperature (IT) from mixing the existing LT and room temperature (RT) lines, (ii) two cooling options for the HEX part of a CL, (iii) the use and sizing of heat exchangers in the cryogenic circuit.

Basic assumptions and description of the designed cryogenic circuit

The basic assumptions resulting from the limitations of the existing refrigerator and from the functional requirements are as follows:

1. The existing refrigerator is able to provide up to 4 g/s of LT high pressure (HP) helium (at 4.5 K and 10 bar).
2. The maximum current of the HTS CL is planned to be 18 kA, which results in the maximum heat load of about $\dot{Q}_{ce} = 5.2$ W at the cold end of a HTS CL.
3. During operation the cold end of the HTS CL should be kept at maximum temperature of 5 K.
4. Two cooling options of HEX part of a CL, as specified in Fig. 1, should be possible.
5. The facility should be able to provide up to 2.5 g/s (Option 1) or 6 g/s (Option 2) of IT helium at 50 K to 80K for cooling the HEX part of a CL.

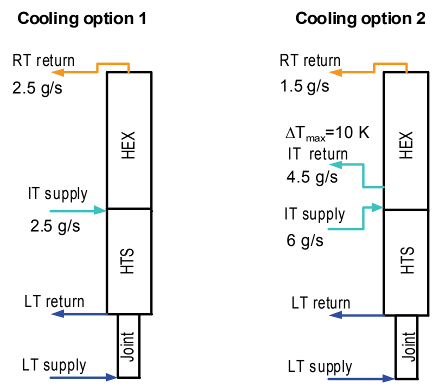

Fig. 1. Cooling options for HTS CL

The basic elements of the cooling system concept are presented schematically in Fig. 2. The vacuum-insulated transfer line (stainless steel pipe, D_{in}/D_{out} = 16/21.3 mm, about 30 meters in length, wrapped with 25 layers of superinsulation) carries the LT HP helium from the remote refrigerator coldbox (point 0) to the test vacuum vessel. We propose to use a bath heat exchanger HX1 to pre-cool helium warmed-up in the transfer line prior to flowing to the cold end of a HTS CL. At the exit of the cold end of a HTS CL (point 3) helium is divided into two parts. One flow is expanded through the Joule-Thomson valve CV15 to provide the liquid helium to the bath in HX1. The LT low pressure helium gas leaving HX1 is fed back to the refrigerator. The second part is mixed with the warm helium at point M, through a mixing chamber to equalize pressures of the two branches of the circuit, to provide the IT helium for cooling the HEX part of a CL and the thermal shield. It results from the preliminary analysis that it is impossible to obtain the maximum mass flow rate of the IT helium

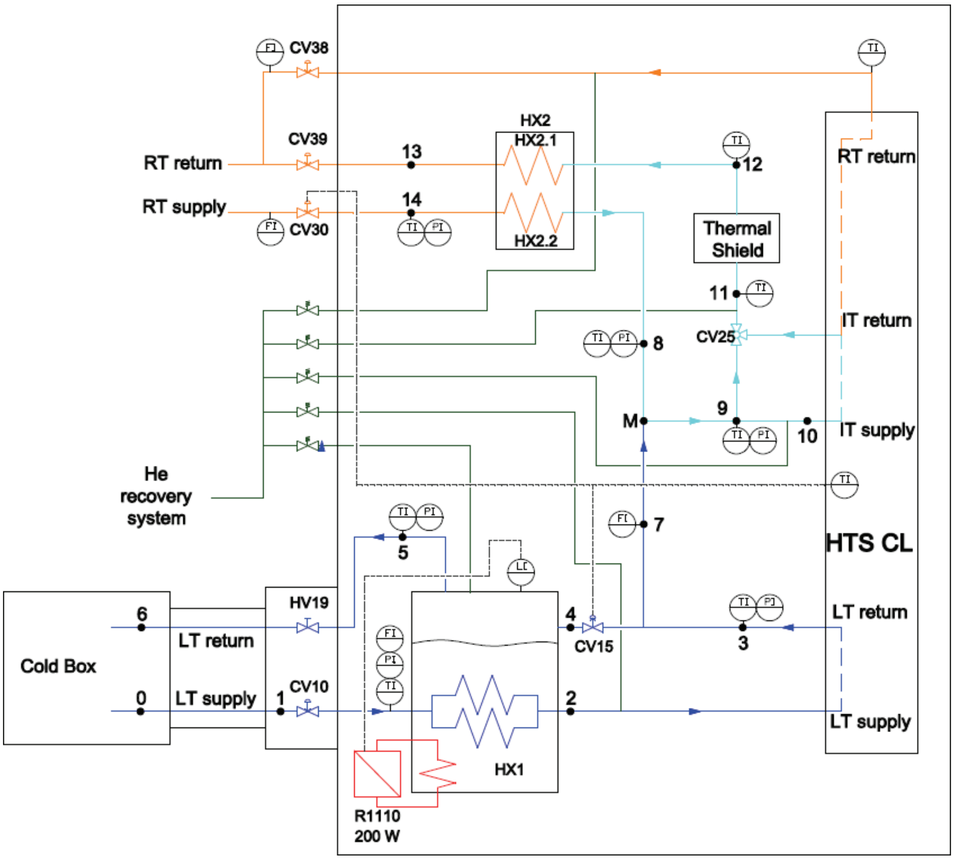

Fig. 2. Scheme of the cryogenic circuit for the HTS CL test facility

required in the cooling option 2 by mixing the available LT helium with the RT helium, thus the use of a recuperator HX2 to pre-cool the RT helium is indispensable. The appropriate cooling option is chosen using the tri-flow on-off valve CV25. The RT helium exiting the warm end of the cooper part of a CL is sent directly to the compressor. The IT helium (at point 11) is used to cool the thermal shield and afterwards to pre-cool the room temperature helium stream in the heat exchanger HX2, prior to flowing to the compressor.

Thermal – hydraulic analysis of the cryogenic circuit

The analysis is performed for the total mass flow rate of the LT HP helium $\dot{m}_0 = 3$ to 4 g/s and for the most demanding functional requirements, i.e. assuming the maximum expected heat loads in the circuit as well as the minimum temperature (50 K) and the maximum mass flow rate of the IT helium supply to the HEX part of CL (at point 8), i.e. $\dot{m}_8 = 2.5$ g/s and 6 g/s for the cooling option 1 and 2, respectively. At each point of the cryogenic circuit the helium mass flow rate as well as the state parameters (p and T) are determined using the mass and energy balance equations and the correlations mentioned below. The calculations are performed with the aid of Mathcad.

The convective heat transfer coefficient between the bulk of helium flowing in a pipe and the pipe surface is calculated using the standard Dittus-Boelter correlation for a forced turbulent flow [2]:

(1) $$\mathrm{Nu}_{DB} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{n}, \quad \mathrm{Re} > 10^4, \quad 0.7 \le \mathrm{Pr} \le 160,$$

where n = 0.4 for heating of the fluid, and n = 0.3 for cooling of the fluid. For the heat transfer between the helium in the bath and the pipe surface we use the Churchill-Chu correlation for the free convection from a horizontal isothermal cylinder with the outer diameter D [3]:

(2) $$Nu_{CC} = \left\{0.6 + \frac{0.387\,\mathrm{Ra}_D^{1/6}}{\left[1 + (0.56/\mathrm{Pr})^{9/16}\right]^{8/27}}\right\}^2, \quad \mathrm{Ra}_D < 10^{12}.$$

To assess the Darcy friction factor for flows in pipes we use the standard Blasius correlation:

(3) $$f_B(\mathrm{Re}) = 0.3164\,\mathrm{Re}^{-0.25}, \quad 3\cdot10^3 < \mathrm{Re} < 10^5,$$

which holds for a turbulent flow in a smooth circular tube [2]. For the flow in the outer pipe of the counterflow pipe-in-tube heat exchanger HX2 we use the correlation for a turbulent flow in a concentric annular duct of radii r_{in} and r_{out} [2]:

(4) $$f_A(\mathrm{Re}) = \left[1 + 0.0925\left(r_{in}/r_{out}\right)\right] f_B(\mathrm{Re}).$$

In Eqs. (1) - (4) $\mathrm{Nu} = HD_h\mu/k$ is the Nusselt number, $\mathrm{Re} = \rho v D_h/\mu$ is the Reynolds number, $\mathrm{Pr} = C_p\mu/k$ is the Prandtl number, $\mathrm{Ra}_D = g\beta\rho C_p D^3(T_w - T_b)/(\mu k)$ is the Rayleigh number, H is the heat transfer coefficient, D_h is the hydraulic diameter of a duct, μ, k, C_p, ρ and β are the helium dynamic viscosity, thermal conductivity, specific heat, density and thermal expansivity, respectively, v is the flow velocity, g is the gravitational acceleration, T_w and T_b are temperatures of a wall and a bath, respectively. The helium thermophysical properties are calculated by a 2D cubic spline interpolation of the data taken from [4].

The helium temperature at the inlet of the heat exchanger HX1 (point 1) is calculated by solving the energy balance equation:

(5a) $$\dot{m}_0 h(p_0,T_0)+\dot{Q}_{TL}=\dot{m}_0 h(p_1,T_1)\,,$$

(5b) $$p_1=p_0-\Delta p\,,\quad \Delta p=f_{DB}L\dot{m}^2/(2D_{in}\rho A^2)\,,$$

(5c) $$\dot{Q}_{TL}=\dot{q}_{TL}\pi D_{out}L\,,$$

where h is the helium specific enthalpy, Δp is the pressure drop in the transfer line calculated assuming incompressible flow, L is the pipe length, A is the flow area, $\dot{Q}_{TL}$ is the total heat load in the transfer line and $\dot{q}_{TL}=6$ W/m^2 is the maximum heat flow rate per unit area specified by the transfer line manufacturer.

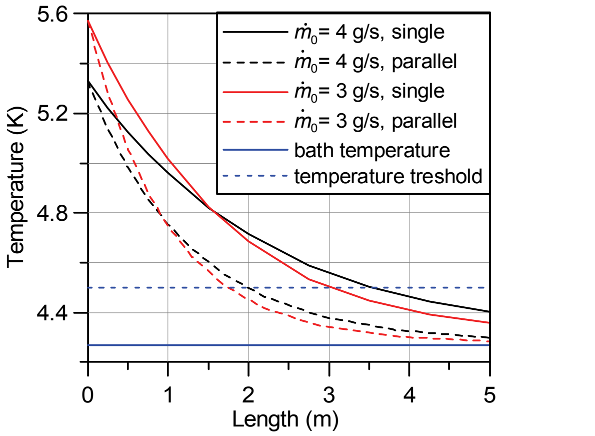

Fig. 3. Temperature profile in the heat exchanger HX1 at different LT helium mass flow rates.

We assume that the heat exchanger HX1 consist of a spiral copper pipe (D_{in}/D_{out} = 6/8 mm) immerged in a helium bath at temperature T_b = T_4 = 4.27 K, which is the temperature of boiling helium at pressure $p_b = p_4$ =105 kPa. Two possible options for HX1 are analyzed: (i) a single pipe and (ii) two pipes connected in parallel. The length of HX1 is assessed assuming that the helium temperature at its outlet (point 2) should not exceed 4.5 K. The steady state heat transfer from helium at temperature T_{He} flowing in the pipe to helium in the bath is described by the equations:

(6a) $$\dot{q}_l=\pi D_{in}H_{DB}(p_{He},T_{He})\cdot(T_{He}-T_{win})\,,$$

(6b) $$\dot{q}_l=2\pi k_{Cu}(T_{win})\frac{T_{win}-T_{wout}}{\ln(D_{out}/D_{in})}\,,$$

(6c) $$\dot{q}_l=\pi D_{out}H_{CC}(p_b,T_b,T_{wout})\cdot(T_{wout}-T_b)\,,$$

where $\dot{q}_l$ is the heat flow rate to the bath per unit length of pipe, $T_{w\,in}$ and $T_{w\,out}$ are the temperatures of the inner and outer surface of the pipe, respectively. To ensure almost isothermal flow conditions the total length of a pipe in HX1 is considered as a sum of ten pieces of different lengths dL_i. The system of Eqs. (6) is solved for the unknown $\dot{q}_l$, T_{win} and $T_{w\,out}$ in the successive pieces, assuming that T_{He} and p_{He} are equal to their values at the inlet of each piece. The temperature and pressure at the outlet of each piece are calculated in an analogous way as for the transfer line (see Eqs. (5a)-(5c)). The total heat flow rate to the bath is:

(7) $$\dot{Q}_{HX1}=\sum_{i=1}^{10}\dot{q}_{l_i}dL_i\,.$$

The results of our calculations are presented in Fig. 3. It is seen that to obtain the required helium outlet temperature either a single pipe of length about 4 m, or two pipes of length about 2.5 m connected in parallel may be used in HX1.

Since the calculated values of pressure drop in relatively long transfer line and HX1 are very small (less than 1.2kPa) we assume that the pressure drop in short junctions is negligible small, thus

(8) $$p_2=p_3=p_7=p_8=p_9=p_{10}\,.$$

The helium temperature at point 3 is calculated by solving the energy balance equation

(9) $$\dot{m}_0 h(p_2,T_2)+\dot{Q}_{ce}=\dot{m}_0 h(p_3,T_3)\,.$$

The vapor fraction of helium at point 4 ($frac$) is determined assuming the isenthalpic flow in the Joule-Thomson valve:

(10) $$h(p_3,T_3)=frac\cdot h_g(p_4,T_4)+(1-frac)h_l(p_4,T_4)\,,$$

where h_g and h_l are the specific enthalpy of gaseous and liquid helium, respectively. The helium mass flow rate $\dot{m}_4$ needed in the bath in HX1 is calculated from the energy balance equation

(11) $$\dot{Q}_{HX1}=\dot{m}_4(1-frac)\Delta h_{vap}\,,$$

where Δh_{vap} = $h_g(p_4,T_4)$ - $h_l(p_4,T_4)$ is the helium specific enthalpy of vaporization.

The cold and warm helium mass flow rates to be mixed in the mixing chamber (point M) are calculated from mass balances

(12) $$\dot{m}_7=\dot{m}_0-\dot{m}_4\,,\qquad \dot{m}_8=\dot{m}_9-\dot{m}_7$$

and the temperature of the warm helium, T_8, is obtained from the energy balance equation

(13) $$\dot{m}_9 h(p_9,T_9)=\dot{m}_7 h(p_7,T_7)+\dot{m}_8 h(p_8,T_8)\,.$$

The helium mass flow rate and temperature at the inlet of the thermal shield (point 11) are calculated as

(14a) $$\dot{m}_{11}=\dot{m}_9-2.5\text{ g/s},\quad T_{11}=T_9\quad \text{for the Option 1,}$$

(14b) $$\dot{m}_{11}=4.5\text{ g/s},\ T_{11}=T_9+10\text{ K}\quad \text{for the Option 2.}$$

The existing thermal shield is a copper cylinder (diameter D_{TS} = 0.8 m and height L_{TS} = 1.5 m). The copper spiral cooling channel (assumed: inner diameter D_c = 8 mm, length L_c = 7 m) is brazed at the outer surface of the cylinder. The cylinder is wrapped with multilayer superinsulation (assumed thickness: t_{MLI} = 11 mm). The total heat load in the thermal shield, $\dot{Q}_{TS}$, is calculated as:

(15a) $$\dot{Q}_{TS}=\sigma E A_{TS\,total}\left(T_{w3}^4-T_{w2}^4\right),$$

(15b) $$\dot{Q}_{TS}=A_{TS\,tb}k_{MLI}(T_{w2}-T_{w1})/t_{MLI}+\\+2\pi k_{MLI}L_{TS}(T_{w2}-T_{w1})/\ln\left(1+t_{MLI}/D_{TS}\right),$$

where T_{w1}, T_{w2}, T_{w3} are the temperatures at the outer surface of the copper cylinder, outer surface of superinsulation and at the inner wall of the cryostat vessel, respectively, $A_{TS\,tb}$ is the area of the top and bottom of the cylinder, $A_{TS\,total}$ is its total area, σ is the Stefan-Boltzmann constant and $E = \varepsilon_2\varepsilon_3/(\varepsilon_2+\varepsilon_3-\varepsilon_2\varepsilon_3)$ is a factor which involves the emissivities of the cryostat wall (ε_3) and superinsulation (ε_2) [5]. The system of Eqs. (15) is solved for the unknown $\dot{Q}_{TS}$ and T_{w2} assuming: T_{w3} = 300 K, $T_{w1} = T_{11}$, ε_3 = 0.5, ε_2=0.05 and k_{MLI} = 1.5e-4 W/(mK) [5]. We have chosen the values of these parameters to assess the highest possible heat load in the thermal shield (the most pessimistic scenario). The helium parameters at the outlet of the thermal shield (p_{12} and T_{12}) are calculated by solving the system of equations:

(16a) $$\dot{m}_{11}h(p_{11},T_{11})+\dot{Q}_{TS}=\dot{m}_{11}h(p_{12},T_{12}),$$

(16b) $$\dot{m}_{11}=\frac{A_c(p_{11}-p_{12})}{(v_{12}-v_{11})+\frac{L_c}{4D_c}\left[f_B(\mathrm{Re}_{11})v_{11}+f_B(\mathrm{Re}_{12})v_{12}\right]}.$$

Eq. (16b) is an approximate relation for the compressible flow [6].

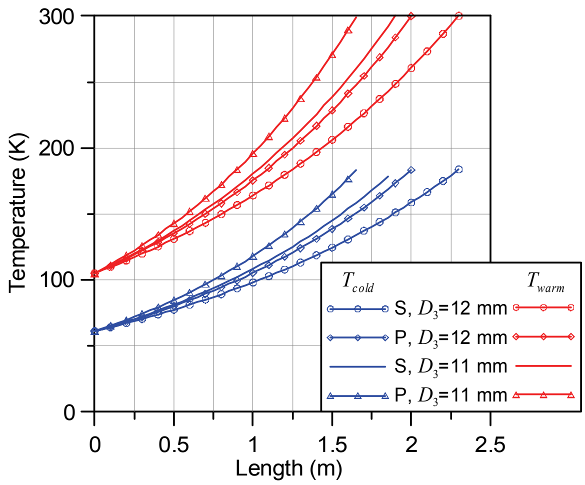

Fig. 4. Temperature profiles in the heat exchanger HX2 for the cooling Option 2 at $\dot{m}_0$ = 4 g/s, D_1 = 6 mm, D_2 = 8 mm and two HX2 configurations (S-single, P-two in parallel) and diameters of the outer pipe.

Two possible simple realizations of the counterflow recuperator HX2 are studied (i) a single pipe-in-tube and (ii) two pipe-in-tubes connected in parallel. We perform the calculations for the inner pipe made of copper with inner and outer diameters: D_1=6 mm and D_2=8 mm, respectively, and for the outer tube with the inner diameter D_3=11 mm or 12 mm. The governing 1D steady state energy transport equations for flow in HX2 may be written in the form

(17a) $$\begin{cases}\dot{m}_{11}C_p(p_{cold},T_{cold})\dfrac{dT_{cold}}{dx}=H_l\pi(T_{warm}-T_{cold})\\ \dot{m}_8C_p(p_{warm},T_{warm})\dfrac{dT_{warm}}{dx}=-H_l\pi(T_{warm}-T_{cold})\end{cases},$$

where H_l is the overall linear heat transfer coefficient:

(17b) $$H_l=\begin{Bmatrix}\left[D_1h_{DB}(p_{cold},T_{cold})\right]^{-1}+\\ +\ln(D_2/D_1)/\left[2k_{Cu}(T_m)\right]+\\ +\left[D_2h_{DB}(p_{warm},T_{warm})\right]^{-1}\end{Bmatrix}^{-1},$$

and T_m = $(T_{cold} + T_{warm})/2$ is the mean temperature. It is assumed that the cold helium flows in the central inner pipe. We have chosen this option, since in the alternative case the value of the overall heat transfer coefficient is lower.

The temperature profiles in HX1 are computed by solving numerically, using the finite difference method, Eqs. (17) with the boundary conditions:

(18) $$T_{cold}(0)=T_{12}, \quad T_{warm}(0)=T_8.$$

The length of HX2 is assessed from the number of integration steps needed to obtain $T_{warm} \geq 300$ K. An example of the calculated temperature profiles in HX2 is presented in Fig. 4. The calculated lengths of the pipes in HX2 at various mass flow rates and HX2 configurations are summarized in Table 1. It is seen in Table 1 that applying of the parallel configuration does not reduce significantly the required length of pipes in HX2. A more effective way to intensify the heat transfer in HX2 seems to be increasing the Reynolds number by reducing the outer pipe diameter or extending the surface of heat exchange and the friction factor, e.g. by using a corrugated inner pipe.

Table 1. Minimum lengths of HX2 for D_1=6 mm and D_2=8 mm and different cooling options, HX2 configurations (S-single, P-two in parallel) and diameters of the outer pipe. For the Option 1 at $\dot{m}_0$ =3.5 g/s and 4 g/s pre-cooling of the RT helium in HX2 is not necessary.

$\dot{m}_0$ (g/s)	Cooling Option	Minimum length of HX2 (m)			
		D_3=12 mm S	D_3=11mm S	D_3=12 mm P	D_3=11 mm P
3.0	1 2	2.85 8.00	2.50 6.70	2.50 6.95	2.20 5.85
3.5	1 2	- 4.05	- 3.35	- 3.50	- 2.95
4.0	1 2	- 2.30	- 1.90	- 2.05	- 1.70

Summary and conclusions

A cryogenic circuit for the new HTS CLs test facility has been designed and the thermal-hydraulic analysis of the proposed circuit has been performed. It results from our analysis that the proposed cryogenic circuit is capable to provide sufficient amount of coolant needed to perform tests of HTS CL's in the most demanding planned conditions.

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