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Low cryogen inventory, forced flow Ne cooling system with room temperature compression stage and heat recuperation

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ABSTRACT

We present design and commissioning results of a forced flow cooling system utilizing neon at 30 K. The cryogen is pumped through the system by a room-temperature compression stage. To decouple the cold zone from the compression stage a recuperating counterflow tube-in-tube heat exchanger is used. Commissioning demonstrated successful condensation of neon and transfer of up to 30 W cooling power to the load at 30 K using only 30 g of the cryogen circulating in the system at pressures below 170 kPa.

1. Introduction

The temperature region of 20–40 K attracts increasing attention in the recent years. Helium becomes scarce, expensive and unreliable commodity especially for applications looking forward to the future. Progress of the HTS technology and new materials create demand for cooling systems operating between LN and LHe [1]. At such temperatures one can take advantage of increased critical fields of HTS or use less expensive, better workable MgB_2. These temperatures are also energy efficient compared to LHe and can be achieved with a single stage cryocooler. Cooling below 50 K is not feasible with sub-cooled nitrogen/oxygen mixtures. The lowest temperature of 50.1 K can be achieved with sub-cooled 22/78 eutectic N_2/O_2 [1]. Below 50 K all N_2/O_2 mixtures freeze. For applications where a cryogenic medium is requested at temperatures 20–50 K, the only options are either hydrogen or neon. The former has a well developed cryogenic technology, mainly in large scale facilities such as missile propellant storage tanks and liquefiers or, in the past, in bubble chambers at accelerator facilities [2,3]. However, mainly due to safety concerns, laboratory use of liquid hydrogen is now limited. Neon on the other hand is laboratory friendly, safe and gains increasing attention in the recent time after being first evaluated for cryogenics back in the 1960s [4,5].

Numerous systems using neon were recently reported. Systems without circulation have their main application in material research. Such systems include laboratory liquefiers [6,7] and bath boiling systems with re-condensation [8]. Systems with natural convection such as thermosyphons [9] have a variety of applications in machinery cooling including HTS MRI magnets [10], generators [11] and motors [12]. They are robust, low-maintenance and cost efficient. Natural convection however puts strong limitations on the geometry of the cryogenic parts and heat flows, and makes the system rigid in the operation settings once built [9,13].

In this paper we report on the design and test operation of a Ne-based forced circulation cryogenic cooling system. Controlled flow can be advantageous compared to natural convection in certain configurations with geometrical constraints and distributed loads. Compared to He gas cooling such a system offers higher temperature stability at lower flow rate and cryogen pressure. The reported system features forced circulation provided by a compressor stage operated on air at ambient temperature outside of the cryostat. To save the cooling power of the cryocooler the warm gas coming from the compressors is precooled in a counterflow heat exchanger by the return flow of the cryogen from the cold zone.

2. Description of the system

The system was designed to deliver about 20 W of cooling power at 30 K to a load located in a separated cryostat about 1 m below the cooling module. See the load description in Fig. 6 Due to the design constrains of the load the maximum design pressure of the cryogen was set to a few hundreds kPa with pressure relief valves opening at 420 kPa absolute pressure. Therefore this system hereafter is referred to as Low Pressure Cooling System (LPCS). The schematic of the LPCS is given in Fig. 1. The LPCS can be broken

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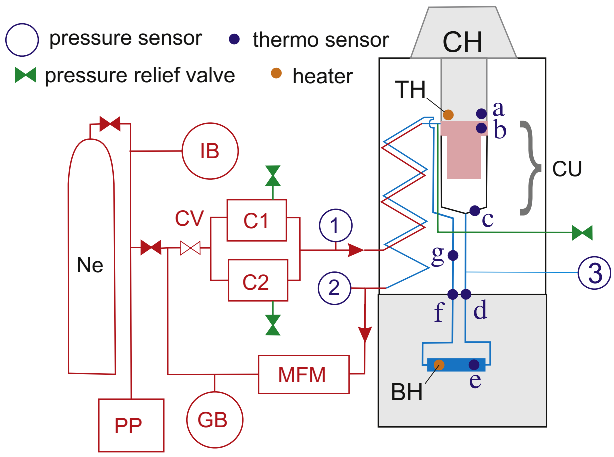

Fig. 1. Simplified scheme of LPCS. See description in Section 2.

down into 3 regions – room temperature setup, LPCS cryostat and load cryostat.

In the room temperature region two membrane compressors Vakuubrand ME 4R NT (C1,C2 in Fig. 1) push neon into the LPCS cryostat. The flow rate is measured by a thermal mass flow meter (MFM) and is regulated by a manual control valve (CV) in front of the compressors. Three pressure sensors monitor the Ne pressure on the entrance, exit and in the cold zone of the circuit (see Fig. 1). The working pressure is defined by the filling pressure, the amount (if any) of gas injected in the system after cooling down and by the settings of the control valve. Prior to filling with neon the system is evacuated by a membrane purging pump (PP). The initial filling is performed from a high pressure gas cylinder with a pressure reducing valve. Additional gas can be added into the system using small injection buffer (IB) volume. In order to improve the gas flow stability a gas buffer (GB) volume is introduced.

In the LPCS cryostat warm neon enters the counterflow tube-in-tube heat exchanger. The direct flow ascending from the bottom to the top gets precooled in contact with the cold return flow streaming from the top to the bottom. Precooled neon enters the condenser unit (CU). Inside the CU, neon gets in contact with a cold copper body (see Fig. 4) conductively cooled by a single stage cryocooler Leybold 140 T fed by Coolpack 6000 compressor. After the CU the cryogen goes to the load cryostat via thermally decoupled vacuum to vacuum cryogen feedthroughs (see Fig. 9). Shortly before the transition to the load cryostat a 1 mm steel capillary tube picks up the pressure in the cryogenic region and guides it to the pressure sensor number 3 outside of the cryostat. The temperature distribution is controlled at several points of LPCS by one DT-670 (Lakeshore) silicon diode (a) on the cold head and six Pt1000 resistive sensors (b–g) controlling the temperature at the top of the CU (b), the bottom of the CU (c), direct and return feedthroughs (d and f), the load (e) and the return line 20 cm above the feedthrough (g).

The load was represented by a cryogen-filled test HTS magnet. The load was equipped with a temperature sensor (e) and a heating element (bottom heater – BH) to measure the cooling power of the LPCS. The load was suspended in the load cryostat on 4 stainless steel wires and isolated with 30 layers of multilayer isolation (MLI). The test magnet was equipped with high-current 25 mm^2 copper current leads to energize the coil. The current leads for 60 A from 290 K to 30 K introduce conductive heat transfer into the load of about 4.4/4.8 W (idle/energized) if optimized using the standard recipe [14]. To balance the cooling power of the LPCS and its thermal load, a heating element (top heater – TH) was mounted on the cold head to sink the extra cooling power.

3. Design of the system

The most critical component of the system is the heat exchanger. The purpose of the exchanger is to remove sensible heat from the direct flow so that less of the cryocooler power (W) is wasted to precool the gas, and more is used for phase transition of neon. The thermal capacity c_p of neon changes from 1.062 kJ/kg K at 40 K to 1.031 kJ/kg K at 300 K [15] (independent of pressure in the 100–1000 kPa range). With the enthalpy of phase transition equal to $\Delta H = 85.8$ kJ/kg [15], the phase transition of Ne is energetically equivalent to a change of its temperature by some 80 K. To use the cooling power efficiently, the difference between the temperature of gas entering the CU and the condensation temperature should be significantly lower than 80 K. In our design we limit it to 10 K, i.e. $T_1^C = 40$ K. Notations $T_{1,2}^{C,H}$ are explained in Fig. 2. We want to use only the phase transition for cooling of the load, therefore we assume the return flow at the cold end to have $T_2^C = 30$ K. On the room temperature side we expect the flows to have $T_1^H = 300$ K and $T_2^H = 290$ K.

To design the heat exchanger fulfilling our needs we first define the operational flow regime using Reynolds number. The Reynolds number is given by $Re = GD_h/\eta$, where η is the viscosity, D_h the effective hydrodynamic diameter, and $G = 4\dot{m}/A$ is the mass flow over cross-section area A. The effective diameter is $D_h = 4 \times A/l$ where l is the perimeter exposed to the stream. The mass flow rate can be estimated as $\dot{m} = W/\Delta H = 0.56$–$0.64$ g/s. The neon viscosity changes from $\eta = 31.6\mu$ Pa s at 300 K and 100 kPa to 6.8μ Pa s at 40 K and 100 kPa (independent of pressure in the range 100–1000 kPa above 40 K [15]). Taking the practical dimension of the inner tube as 10 $\times$ 1 mm and the flow rate of 0.64 g/s, one gets Reynolds numbers of 3200–15,000 depending on the local temperature, which is above the transition to the turbulent regime at about 2000 for annular flows. Thus, the theory of turbulent heat exchangers can be applied [16]. In a closed cycle system one can estimate the length of the exchanger as follows:

$$L = \frac{\alpha}{\beta}\left(T_1^H - T_1^C\right) \quad \text{where} \tag{1}$$

$$\alpha \approx \dot{m}c\left(\frac{1}{h_1 S_1} + \frac{1}{h_2 S_2}\right), \quad \beta = T_1^C - T_2^C. \tag{2}$$

$S_{1,2}$ are the perimeters of the respective flows. The coefficients $h_{1,2}$ are the heat transfer coefficients, defined such that the rate of heat transfer from the gas to the tube per unit area is $\dot{Q} = h\Delta T$. The coefficient h depends on η, c_p, the thermal conductivity κ and the effective tube diameter D_e. Introducing the Nusselt's number $Nu = hD_e/\kappa$ one can use the Dittus–Boetler correlation [17]

$$Nu = 0.023 Re^{0.8} Pr^{0.4}, \tag{3}$$

where Pr is Prandtl's number $Pr = \eta c_p/\kappa$. h is expressed in W/m^2 K, κ in W/m K, c_p in J/kg K, D_e in meters, G in kg/m^2 s and η is in Pa s. Substituting the definitions of Nu, Re and Pr in Eq. (3) one gets

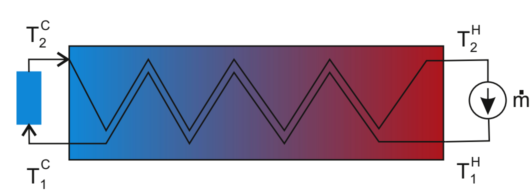

Fig. 2. Definition of the flows in the heat exchanger. Direct (1) and return (2) flow temperatures on cold (C) and hot (H) ends of the circuit.