Optimization of a condensed-neon cooling system for a HTS synchronous motor with Gd-bulk HTS field-pole magnets

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Abstract. The axial-gap synchronous machine developed in our laboratory is based on Gd-bulk HTS field-pole magnets, able to trap a part of the magnetic flux they are submitted to when cooled down below $T_c$. At the liquid nitrogen temperature, by the Pulsed-Field Magnetization (PFM), 1.04 T was trapped in 60 mm-diameter and 20 mm-thickness magnets, leading to an output power of the motor of 10 kW at 720 rpm. To enhance this performance, we have to increase the total amount of trapped flux in the bulk, the shortest way being to decrease the temperature of the bulk HTS. Thus, we focused on the improvement of the condensed-neon cooling system, a closed-cycle thermosyphon, so that it provided enough cooling power to lead the rotor plate enclosing the magnets to a low temperature. The present study implied coming out with a new fin-oriented design of the condensation chamber; hence, the numeric calculations and FEM software (ANSYS) heat transfer simulations were conducted for various shapes and positions of the fins. The trapezoidal design offering the best efficiency was then manufactured for testing in a heat-load test configuration, leading to cooling times divided by three and a maximum heat load endured of 55 W.

1. Introduction

The use of High-temperature superconductors (HTS) has been spreading in the recent years, thanks to the improvement of their performances coming with a significant decrease of their production price. The application to rotating machines, motors and generators, is particularly interesting as high torque and compactness are both required to propose a design surpassing conventional magnets. This field has thus attracted a good number of laboratories and companies ([1], [2]), as well as ours which has been developing, among others, a compact axial-type bulk-HTS synchronous rotating machine for ship propulsion since 2001 [3].

The ability for bulks to trap a magnetic flux, flux pinning, is highly dependent on the temperature of the bulk. At 29 K, it has been shown than a field of more than 17 T [4] can be trapped in a bulk which would have only kept 1.2 T at the liquid nitrogen temperature [5]. Increasing the trapped flux also means increasing the output power of the rotating machine, hence the need to enhance the cooling system to reach a low temperature and keep it low even in case of heat invasion or generation.

Thermosyphons are a very efficient way to transport heat from the part to cool down (i.e. the rotor plate containing the bulk HTS in our case) to the cold head of the cryocooler [6]. We have already been using a condensed-neon closed-cycle thermosyphon for some time, but it was lacking efficiency.
The addition of optimized fin arrays is something which has long been studied in heat transfer science ([7], [8]), hence the idea to come over an all-new design of a fin-oriented condensation chamber.

The calculations were thus conducted for the best efficiency while keeping in head the manufacture of such a shape of an array of fins. First comparisons and evaluations were made by the use of the FEM software ANSYS. The modified condenser was then manufactured and added to the thermosyphon for load tests with a heater attached to the evaporator. Some problems recurrent with the use of thermosyphons are also reported, when not solved.

2. Optimization of the design of the condensation chamber

2.1. Thermal calculations

The main phenomenon driving the condensation heat transfer in the presence of vertical longitudinal fins is the convection allied with the buoyancy coming from the increase of density of the gas along with the decrease of its temperature. The efficiency of this convection highly relies on the heat transfer coefficient, commonly symbolized as \( h \). This value differs for every experimental configuration, meaning an experimental measurement is necessary for an accurate use, but it is possible to get an estimation by following calculations [9] based on the properties of neon (essentially heat of vaporization, density, specific heat, dynamic viscosity [10] and thermal conductivity [11] at low temperatures). In our case, these calculations led to an estimated heat transfer coefficient of 6,160 W m\(^{-2}\) K\(^{-1}\).

The material chosen for those fins was naturally C1020 oxygen-free copper (OFHC), as its very high thermal conductivity is a very important advantage, at the condition it is annealed under vacuum. The OFHC copper we used for the fins as well as for the heat exchange plate separating the cold head of the cryocooler and the condenser was pure at 99.99%, leading to a theoretical thermal conductivity of 6,000 W m\(^{-1}\) K\(^{-1}\) at 25 K. Based on these properties, optimization calculations ([7], [8]) and manufacture considerations, we opted for a 100-mm wide eleven-fin array, based on a trapezoidal design, each fin’s thickness being 1 mm at the top and 0.5 mm at the tip, all separated by a 8.25 mm pitch. The calculations are following efficiency criteria, beginning with four different fins’ shapes: rectangular, triangular, parabolic and hyperbolic. The most efficient thickness and pitch were then obtained for each geometry and the already existing condensation chamber decided the limits in size. For cost considerations and manufacturing feasibility, we adopted a trapezoidal geometry (a thickness of less than 0.5 mm at the tip being difficult to reach). The radial symmetry was not adopted considering that a constant pitch is also a factor of efficiency.

2.2. FEM simulations

The previous condenser and heat exchange plate were made of stainless steel (SUS 304), a particularly unsuitable material considering its poor thermal conductivity of about 3 W m\(^{-1}\) K\(^{-1}\) at 25 K. A brief simulation in condensation conditions can give us a quick comparison of the expected change in efficiency, first by the replacement of SUS 304 by OFHC and then by the addition of fins (see table 1).

| Conditions of the calculations | Heat transfer coefficient [W m\(^{-2}\) K\(^{-1}\)] | Cold head temperature [K] | Neon temperature (condensation at atmospheric pressure) [K] |
|-------------------------------|----------------------------------|-----------------|----------------------------------|
| SUS 304                      | 3.74                             | 25              | 27.07                            |
| Theoretical max heat transfer [W] | OFHC w/o fins | 175              |                                  |
|                               | OFHC with fins                  | 754              |                                  |

Table 1. Conditions and results of the thermal simulation conducted with ANSYS.
These values are based at the same time on an infinite quantity of neon as well as an infinite cooling capacity of the cold head, hence the high values for the OFHC. We can yet notice that the SUS condenser was anyway limited at a heat transfer below 4 W. The cooling capacity of our Cryomech AL330 cryocooler running at 50 Hz is certified for 57.25 W at 25 K. It would seem that a simple change of the material would have been enough to make full use of this cooling power. Still, the addition of optimized trapezoidal fins makes the condenser even more efficient, supposedly resulting in shorter cooling times and faster reactivity against a possible sudden increase of the bulks’ temperatures.

The fin array was then manufactured by Wire Electrical Discharge Machining with the design previously described and annealed at 300 °C during 48 hours. An inspection test conducted by the supplier, Hitachi Cable, Ltd., confirmed the copper used was pure at 99.99 %. The manufactured fin array is displayed in figure 1.

![Figure 1. The manufactured optimized fin array.](image)

3. Experiments

3.1. Mounting of the test facility.

The assembly was made on a vertical axis for both old- and new-type condensers. The cold head of the cryocooler is attached to the condensation chamber through a heat exchange plate in which, for the new type, a Watlow cartridge heater has been inserted to control the temperature of the condensation with a Lakeshore 331S temperature controller. The plate is attached to the upper part of the condensation chamber, which is actually the fin array in the case of the new type (see figure 2a for the upper part of the assembly). A gas input pipe enables the direct flow of neon from the outside to this chamber. The bottom of the condenser is linked to the OFHC evaporator by a 40-cm long and 12.7-mm thick SUS 304 tube (figure 2b). The load test is conducted thanks to a Watlow cartridge heater located in a copper block attached under the evaporator. The whole is as much as possible wrapped with superinsulation and placed in a vacuum chamber to prevent at the same time radiation and convection from the outside.
3.2. Measurements.
The temperature of the cold head is measured thanks to a silicon-diode sensor (DT-670-SD-1.4L/Lakeshore, calibrated by the supplier from 1.4 to 325 K) placed in the heat exchange plate. It is directly linked to the temperature controller previously described. Two Au-Ag thermocouples are located on the fins in the case of the new-type condenser, and another one is free inside the chamber to have an idea of the neon-gas temperature. These three thermocouples’ data are recorded with a Keithley Model 2001 multimeter. The last temperature measurement is a Cernox (CX, CX-1030-SD) sensor located under the evaporator. Like the SD sensor, it is linked to the Lakeshore 331S. This is the main indicator of the advancement of the condensation. The wires of the thermocouples and various sensors are all taped to the evaporator playing the role of thermal anchor, so that no heat from the outside comes and interfere with the values of temperature. A schematic summary of all of this is displayed in figure 3. All the data are recorded by a home-made five-second step Labview acquisition program with a National Instruments GPIB-USB-HS controller.

Figure 2. (left, a) 1 – cold head, 2 – heat exchange plate, 3 – cold head control heater, 4 – neon input pipe, 5 – condensation chamber and 6 – adiabatic tube; (right, b) 7 – evaporator and 8 – load heater.

Figure 3. The schematic of the complete new-type thermosyphon. In the case of the old type, fin tip thermocouples are naturally absent. SD stands for silicon diode, TC for thermocouple and CX for Cernox.
3.3. Experimental conditions.
The main experiment conducted here consists, after the neon is fully condensed, in gradually increasing the output power of the heater located under the evaporator, simulating a load. The load was increased every time the temperature was saturating or after fifteen minutes anyway, with a 5-W step driven by a variac. That way, it is possible to determine the maximum load-value the cooling system can cope with. It is an important value to know for the design of the real motor. We could thus compare efficiencies of both old-and new-type condensers, for now without rationing the quantity of neon and using a 12.7 mm outer-diameter tube.

Then, we forgot about the old-type condenser to focus on the new one. The optimum quantity of neon is another parameter we have been trying to determine during these experiments. Even if a closed-cycle thermosyphon allows using the same gas for a very long time, the minimum quantity has yet been investigated. After having cooled down the whole and then taken out the initial quantity of neon, successive load tests have been conducted, gradually increasing the quantity of neon of a volume $V_0$ until we arrive to the saturation of the load limit.

4. Results and discussion

4.1. Comparison of the condensation time
The first thing we could notice by replacing the cooling system was the drastically shorter time of the condensation process. The evolution of temperatures in function of time of both old- and new-type cooling systems can be seen in figure 4.

![Figure 4. Comparison of the full-cooling time of the evaporator.](image)

The time for the evaporator to be at the temperature of liquid neon (approximately 30 K here) changed from 525 to 165 minutes by the replacement of the material and the addition of fins. While the decrease of the temperature of the cold head is more or less the same during the first hour, we can see the condensation of the neon gas is not performed at the same temperature in both cases. While occurring at 22.4 K for the old type, corresponding to a cooling power of 45 W from the cryocooler (determined following the cooling capacity curve of this model), it occurs at 33.6 K for the new type, corresponding to 91 W of heat extracted. This additional quantity of transferred heat is one of the explanations of such a change in the duration of condensation. Contrarily to the new-type case, the cold head has never been set to 25 K for the old type, as there was no interest to do so, considering its already poor efficiency. Indeed, no freezing (noticeable with the new condenser, when the setting is under 25 K, by a sudden increase in temperature of the evaporator) has ever been observed in the SUS condenser, even at very low temperatures.
4.2. Load tests
The reaction of both types of condensers is shown on the figure 5, in terms of temperatures of cold head and evaporator, as well as in terms of neon pressure inside the condenser.

![Figure 5. Temperature and pressure variations during the load test.](image)

The pressure allows an increase of the efficiency of the condensation but will not be allowed to go over 0.45 MPa in the motor because of some parts of the cryo-rotary joint allowing the flow of cryogen from a static to a rotating part. The old SUS type’s gas pressure increases from the first heat load of 5 W. It is also accompanied with an increase of the temperature, showing that even 5 W is over the maximum load it can safely handle. At 15 W, the temperature is still stable at 40 K but the high value of the pressure (more than 0.7 MPa) makes it unusable inside the motor. The limit is thus considered at 10 W.

As for the OFHC fin-oriented condenser, there is no effect of the load until 45 W. Then, the pressure and temperature increase gradually but the three values are still in an acceptable range at 55 W. At 60 W, the temperature of the evaporator increases continually, making it impossible to show it on the graph.

4.3. Influence of the quantity of neon
Now we focus only on the new-type condenser to determine its characteristics. After a complete cooling of the thermosyphon, we purged all the neon inside with a vacuum pump. From that point, an arbitrary quantity of neon, corresponding to opening the input valve during five seconds at a 0.5 MPa pressure, is reintroduced. The condenser’s total volume has been determined at 0.92 L thanks to design drawings, meaning we introduce approximately 4.6 L of neon gas in normal conditions (if we neglect the condensation already certainly taking place during these five seconds). Having introduced this volume we called V₀, we performed a load test in the same conditions as before, until we reach a high pressure or an unstoppable rise of the evaporator’s temperature. Another volume V₀ of neon is then added to the one already inside the thermosyphon for another load test starting from 0 W, until we reach the 55-W limit previously encountered. The successive load tests are reported in figure 6.
The difference of temperature between the cold head and the evaporator is reported in function of the heat load applied and of the quantity of neon gas introduced. Starting from a 40-W limit for a V0 volume, 50 W for 2 and 3 V0, it becomes 55 W for 4 and 5 V0. Yet, the 4-V0 volume proposes a steep increase of the temperature which was not present in the first load test, hence the choice of 5 V0 as the optimal value. In normal conditions of temperature and pressure, 5 V0 corresponds to 23 L of neon gas or 0.017 L of liquid neon after full condensation, which is quite little. The decrease of the difference of temperature at 55 W was not present in the previous load test. It may come from the fact that there is an optimum quantity of neon after which inputting more only results in obstructing the flow inside the tube and decreasing the efficiency of the heat transfer.

5. Conclusion
We have conducted in this study the design optimization and manufacture of a new condensation chamber for thermosyphon, for use in HTS rotating machines as a way to cool down the rotor plate containing the HTS field-pole magnets. Both the heat exchange plate and the fin array included in the condensation chamber have been manufactured in 99.99 % pure oxygen-free copper, following a design achieved thanks to thermal calculations and simulations. The fin array was finally decided as a trapezoid-shaped eleven-fin array, each of them proposing a thickness of 1 mm at the top and 0.5 mm at the tip. Kept at 25 K thanks to a cartridge heater located in the heat exchange plate fixed to the cold head, this important modification allows to use the cryocooler near its full extent, i.e. until a 55-W load limit. Condensation times have also been drastically shortened. The next step should be the design of concentric tubes in order to avoid counter-flow problems.

6. References
[1] Kummeth P, Frank M, Nick W, Nerowski G and Neumueller H W 2005 Proc. of the 17th Int. Symp. on Superconductivity (ISS 2004) – Advances in Superconductivity XVII (Physica C: Superconductivity vol 426-431 part 2) (Amsterdam: Elsevier) pp 1358-64
[2] Kwon Y K, Kim H M, Baik S K, Lee E Y, Lee J D, Kim Y C, Lee S H, Hong J P, Jo H S and Ryu K S 2008 Proc. of the 20th Int. Symp. on Superconductivity (ISS 2007) (Physica C: Superconductivity vol 468 issues 15-20) (Amsterdam: Elsevier) pp 2081-6
[3] Sano T, Kimura Y, Yamaguchi K, Izumi M, Ida T, Sugimoto H and Miki M 2008 Proc. of the 6th Int. Workshop on the Processing and Applications of Bulk Superconductors (PASREG) (Cambridge, UK, 13-13 September 2007) (Materials Science and Engineering: B vol 151 issue 1) (Amsterdam: Elsevier) pp 111-6
[4] Tomita M and Murakami M 2003 Nature 421 517-20
[5] Nariki S, Hinai H, Sakai N, Murakami M and Otsuka M 2002 Physica C: Superconductivity vol
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