FR9501247

To be presented at International Cryogenic Engineering Conference 15, June 7-10, 1994, Genova, Italy

CEA-CONF--11916

Construction and Preliminary Testing of Perforated-Plate Heat Exchangers For Use In Helium II Refrigerators

François Viargues, Gérard Claudet, Peter Seyfert

CEA/ Département de Recherche Fondamentale sur la Matière Condensée, Service des Basses Températures, Centre d'Etudes Nucléaires de Grenoble, 38054 Grenoble Cedex 9, France

Heat exchangers in cryogenic refrigerators must meet conflicting requirements for high thermal efficiency and low frictional pressure drops. This is particularly true in helium II refrigerators where the gas stream leaves the cold source at 15 mbar. Perforated-plate heat exchangers are likely to offer better overall performances for that application than the widely used plate-fin heat exchangers. Through their design they combine a high thermal conductance in transverse direction with a low thermal conductance in axial direction. The present paper reports on the initial stage of development of a fabrication method using photochemical milling of copper plates and stainless steel spacers, and silver brazing for bonding. Comparison of preliminary test measurements and calculations in the 80 to 300 K temperature range with flow rates between 1 and 5 g/s has confirmed that predictable results can be obtained for the thermal efficiency and the pressure drop with an accuracy of a few percent.

INTRODUCTION

Cooling by helium II of large scale superconducting devices (magnets or accelerating cavities) has become a key technology for a number of currently studied research projects. The most advanced among all , the Large Hadron Collider of C.E.R.N. shall use 16 refrigeration units with a capacity of 1000 W at 1.8 K each [1]. In a more distant future, linear accelerators like TESLA [2] are likely to require several 1.8 K cold sources delivering up to 5000 W. Other applications of similar size in the field of superconducting magnet energy storage or controlled thermonuclear fusion may be envisaged.

Mechanical helium II refrigerators work by adding a further stage to ordinary 4.5 K helium refrigerators which are presently available for capacities of up to 10 or 20 kW. The essential duty of this stage is to maintain a constant vapour pressure in the range of 10 to 15 mbar over a liquid helium bath by pumping away the cold gas at the required speed.

Considering the huge mass flow rates to be handled, severe design criteria must be adopted for all components of this stage, especially as regards the investment costs, the long term reliability, and the thermodynamic efficiency. The latter decisively determines the energy consumption, hence the running cost. From the experience gathered so far, it appears that the solution is to be sought in an optimized combination of cold compressors and high efficiency heat exchangers.

Heat exchangers for use in helium II refrigerators which the present paper will concentrate on must combine pressure losses in the cold gas stream as low as a few millibars with thermal efficiencies close to 95 % in some cases. The only way to achieve this goal is to provide large flow cross-sections and short flow paths. In conventional aluminium plate-fin exchangers, this design concept produces large heat losses by axial conduction through the body of the exchanger made of high conductivity material. As a result the thermal efficiency drops dramatically. Coiled tube exchangers have the same disadvantage and, in addition, are difficult to fabricate by industrial methods.

Perforated-plate heat exchangers seem to offer better prospects. By virtue of their particular configuration they allow for high thermal conductance in transverse direction while restraining axial heat flow. An excellent literature review on perforated-plate heat exchangers has recently been presented by Venkatarathnam and Sarangi [3]. We will report here on our attempts to fabricate this type of heat exchangers and on first results of how they performed in the 80 to 300 K temperature range.

MANUFACTURE OF THE HEAT EXCHANGERS

The exchangers we have attempted to develop essentially consist of a stack of copper perforated plates which are interleaved with stainless steel spacers. The purpose of the holes in the copper plates is to increase the active surface area (usually by a factor of 2 to 3). The spacers inserted between the copper plates ensure uniform flow distribution on a short range (between neighbouring holes) and create local turbulence which favours wetting of the front and back faces of the plates so enhancing heat transfer. In addition, being of low conductivity material they are crucial for reducing axial heat flow. Figure 1 shows two models of plates which use rectangular shaped multiple passages for the two fluid streams.

Fabrication of the Perforated Plates and Spacers

The copper perforated plates are characterized by a great number of small holes, generally more than 10000 perforations per plate, with less than 1 mm in diameter. Such plates would be impossible to machine in conventional ways such as drilling or embossing. We have used photochemical milling for fabrication of both copper perforated plates and stainless steel spacers.The thickness of both types of plates was 0.5 mm which seemed to be an upper limit if deviations from the cylindrical shape of the holes ('sausaging') were to be kept in reasonable limits. A certain scatter ($\sim \pm 10$ %) in the mean diameter of the holes (here 0.55 mm) also appeared to be inherent to the method.

Brazing of a heat exchanger

The final assembly of a heat exchanger consists in stacking a set of perforated plates and spacers with one yoke at each end, inside a special fixture, and then having all these components brazed together in a high-vacuum furnace.

The method chosen for brazing uses the copper-silver eutectic which forms when the silver-plated separators are in contact with the copper plates in proper conditions of temperature and contact pressure. The brazing width, given by the geometry of spacers, has been set at 2.4 mm and a weight load has been applied to the stack to ensure a correct contact of all the surfaces to be brazed. The brazing fixture is covered with graphite plates in order to avoid sticking to the exchanger after the heat treatment.

The contact area between the copper plates and the stainless steel spacers is a critical place for the separation of the two fluid streams from each other and of both fluid streams from outside. Therefore, an absolute tightness is required for the brazing, especially considering that the heat exchanger has to be operated in vacuum. The brazing must also resist the tensile stress which will be induced by the high pressure hot stream.

The method has proven to be very reliable. Regularly carried out tests on all fabricated models have shown no detectable leaks down to a sensitivity level of 10^{-10} mbar l s^{-1}.

CALCULATIONS

For the purpose of prediction of exchanger performances we have attempted to calculate the effectiveness and the pressure drop for situations where the inlet flow rates and temperatures as well as all parameters of a given heat exchanger are known.

Effectiveness has been calculated by using the well-known NTU-concept (see for example [4]). The term UA, overall heat transfer coefficient times heat transfer surface area, which usually occurs in the definition of N_{TU}, has been rewritten as nU in the present case, where n is the number of plates used in the exchanger and U has the meaning of an overall heat transfer coefficient <u>per plate</u>. U is related to the parameters of a plate by :

$$U = \left(\frac{1}{(A\eta h)_h} + \frac{b}{k_s A_s} + \frac{1}{(A\eta h)_c} \right)^{-1}$$

The first and third term in this expression represent the heat transfer between the hot (subscript h) or cold (subscript c) stream and the corresponding set of channels. A is the total heat exchange surface area provided for the stream considered (including wall surface of holes), h the corresponding convective heat exchange coefficient and η the so-called fin efficiency. η accounts for the fact that heat transfer from or to the channels is composed of a convection component and a plate conduction component [5]. An empirical correlation by Mikulin et al [6] applying globally to convective heat transfer with perforated plates has been used for h.The middle term in the above expression for U describes the conductive heat transport across the partition separating two channels. b is the width of the partition, A_s its cross-sectional area (plate thickness times channel length) and k_s is the thermal conductivity of the plate material.

The effectiveness calculated along these lines has eventually been corrected for the detrimental effect of axial heat conduction across the solid structure of the heat exchanger [7].

The pressure drop across a stack of perforated plates is believed to arise mainly from the singularities at entry and exit of the holes in the plates. In our calculations, we have used a formula and an experimentally determined friction factor by T. Guo [8].

$$\Delta p = \frac{v^2}{2\rho}\frac{L}{d_h}\left(73.61\mathrm{Re}^{-0.768} + 0.757\right) \quad \text{where } d_h = \frac{4 \text{ x swept volume}}{\text{total wetted surface area}}$$

Here v is the fluid velocity in the holes, ρ the fluid density, L the length of the stack and Re is the Reynolds number based on the above defined d_h as characteristic length.

TESTING

We have built a first series of heat exchangers for a nominal flow rate value of 5 g s^{-1}. Since testing facilities providing flow rates of that size at helium II temperatures are not easily available we have started our development by adopting design specifications for the first stage of a standard helium refrigerator. Performance measurements on these exchangers have been carried out by means of a relatively simple test-rig using liquid nitrogen and a 100 m^3 h^{-1} / 15 bar screw compressor. All exchangers were tested in counterflow configuration with fixed inlet temperatures: 77 K for the cold stream and 300 K for the hot stream.

The tests of two exchanger models will be reported here. While both models are made up of 250 pairs of plates and spacers, they differ in the geometry given to flow passages:

A model no.1 plate is shown in Figure 1a. The low-pressure cold stream (total area of flow cross-section 45.15 cm^2) is split into three channels, one at the centre and one next to each of the two outer edges of the plate. Two channels carrying high-pressure gas are arranged so as to have a low-pressure channel on either side. In addition, all channels of a stream communicate between each other through lateral extensions of the perforated zones. This permits continuous equalization of pressure and is supposed to ensure uniformity of flow distribution.

Figure 1b shows a model no.2 plate. Here the low-pressure cold stream is split into two channels (total area of flow cross-section 22.27 cm^2), one on each side of the single high-pressure channel at the centre. No communication was provided between the two separated low-pressure channels.

Experimental results of the effectiveness as a function of mass flow rate (per unit flow cross-section of cold stream) for the two exchanger models are shown in Figure 2. Cold stream pressure has been kept constant at 1 bar in all measurements, while warm stream pressure was varied in the 5 to 15 bar range with no effect at all on the efficiency.

Also shown in Figure 2 is the result of the above mentioned calculations. They appear to be in fairly good agreement with the results obtained on model no.2. The substantially lower values recorded for model. no.1 are interpreted as a consequence of insufficient heat exchange in the perforated zones connecting the different channels. This geometry is therefore not recommended.

Cold stream pressure drop along the perforated plates was also checked. Figure 3 shows experimental and calculated results. Obviously, there is not much difference between model no.1 and 2, but agreement with calculation is seen to deteriorate as the flow rate rises. This is presently not understood and needs more investigation.

OUTLOOK

The results obtained so far with our perforated-plate heat exchangers have encouraged us to carry on with our development program. A new prototype for a 6 g. s^{-1} / 1.8 K Joule-Thomson stage has recently been delivered to the LHC project group at CERN and test results are expected to be available in autumn 1994.

REFERENCES

1 Lebrun, P Superfluid Helium Cryogenics For the Large Hadron Collider Project at CERN, to be presented at Int. Cryogenic Eng. Conf. - 15, June 7 - 10, 1994, Genova (Italy)

2 Trines, D., Dwersteg, B., Kaiser, H., Matheisen, A., Meyer, G., Proch, D., Wolff, S., Lengeler, H., Nicol, T., Peterson, T., Alessandria, F., Castagna, F., Tazzari, S., Cavedon, J.M., Gastebois, J. Conceptual Design of the Cryostat and the Cryogenic Supply For a Superconducting Linear Collider In: High Energy Accelerator Conference 1992

3 Venkatarathnam, G., Sarangi, S. Matrix Heat Exchangers and Their Application In Cryogenic Systems Cryogenics (1990) 30 907-918

4 Kays, W. and London, A.L. In: Compact Heat Exchangers McGraw-Hill (1964)

5 Fleming, R.B. A Compact Perforated-Plate Heat Exhanger In: Adv. Cryogenic Eng. (1969) 14 197-204

6 Mikulin E.I, Shevich, Yu.A., Potapov, V.N., Solntsev, M.Ya., Yusova, G.M. Study of Matrix-Type Heat Exchangers Made of Perforated Plates Chemical and Petrol Engineering (1980) 514-519

7 Kroeger, P.G. Performance Deterioration in High Effectiveness Heat Exchangers Due To Axial Heat Conduction Effects In: Adv. Cryogenic Eng. (1967) 12 363-372

8 Guo, T. private communication

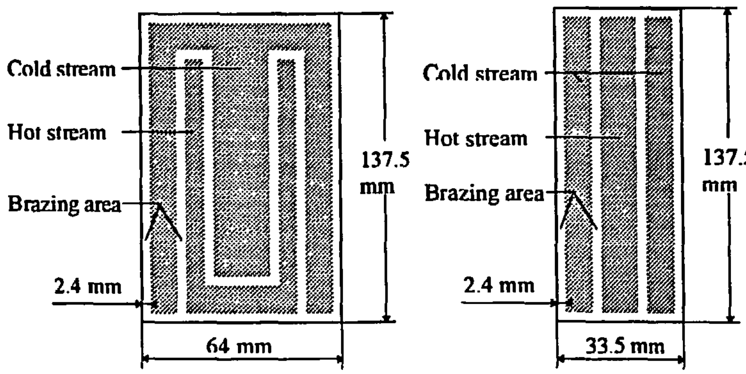

Figure 1 Layout of perforated plates (at left: model no.1, at right: model no.2)

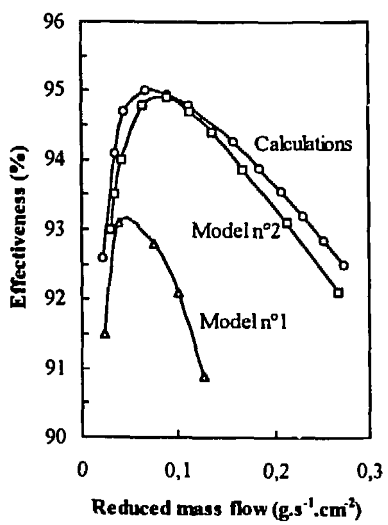

Figure 2 Measured and calculated effectiveness

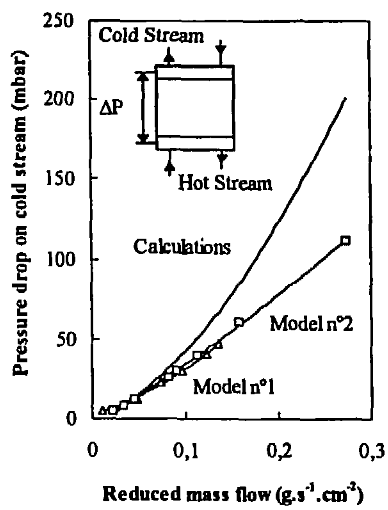

Figure 3 Measured and calculated pressure drop