Experimental research of a two-phase nitrogen natural circulation loop

S Reisz, A Bonelli, B Baudouy
Irfu, CEA, Université Paris-Saclay, F-91191 Gif-sur-Yvette, France
Corresponding author E-mail: bertrand.baudouy@cea.fr

Abstract. Two-phase heat and mass transfer in nitrogen has been experimentally investigated in a natural circulation loop. The experiments were realized in steady-state conditions near atmospheric pressure with a 10 mm inner diameter vertical copper tube, uniformly heated over roughly 1 m in length. The vapor and total mass flow rates and the wall temperature were measured as a function of the tube heat flux density. The investigated ranges-limits are 40 g/s, 25% and 30 kW/m² for the total mass flow rate, the vapor quality and the heat flux density respectively. Transition from single phase natural convection to nucleate boiling flow is evidenced by wall temperature measurements at five different locations along the heated tube. The measurements are compared to the numerical computations of the total mass flow rate that is simulated by a set of conservation equations based on a simple separated flow model. The model catches most of the features of the mass flow rate evolution as a function of the heat flux density. In addition, the wall temperature measurements are used to determine the heat transfer coefficients that are compared with known correlations in both single phase natural convection and boiling flow. A majority of the data falls within ±20% of the predicted values given by the correlations.

1. Introduction
In the past, low temperature circulations loops were included in the refrigeration system of superconducting magnets with, most of the time, helium as a working fluid. Several large magnets were designed, constructed and operated with success with a natural circulation cooling scheme. One can list, for example, the ALEPH magnet for CERN [1], the CLEO II magnet at Cornell University [2], the G0 magnet at Jefferson laboratory [3], the CMS magnet at CERN [4] and more recently the R3B-Glad magnet constructed at CEA Paris-Saclay for GSI [5, 6]. Extensive studies were performed in helium during the design periods of the CMS and R3B-Glad magnets. A 2-m high experimental facility was constructed and served studying the thermo-hydraulic characteristics [7, 8] as well as the parietal heat transfer [9, 10] of such a loop.

Nowadays, with the continued improvement of the high temperature superconducting (HTS) technology, operation of more powerful high field magnets or detectors is possible at liquid nitrogen temperature. Natural circulation loops with nitrogen as a working fluid are good candidates to be included in the cooling scheme of these high-Tc superconductors devices. As opposed to helium as a working fluid, a limited number of studies have been reported in the literature for nitrogen. Umekawa and Ozawa reported the experimental results on heat transfer in two-phase natural circulation loop in the saturated boiling and post-dryouts regimes [11]. They concluded that the heat transfer coefficients in the saturated boiling can be predicted by the
Schröd–Grossman’s correlation developed for non cryogenic fluids under steady flow conditions. Later on, Kim and Chang studied a sub-cooled nitrogen circulation loop and presented heat transfer and mass flow rates results in single phase and two-phase conditions for low mass flow rates up to 1.5 g/s [12]. On a higher loop, an experimental study was presented for heat flux lower than 12 kW/m² in single fluid flow. A mixed convection regime, a combination of natural and forced convection heat transfer, was observed and studied [13].

This paper reports the experimental study performed on our 2-m high circulation loop in two-phase flow nitrogen around atmospheric pressure. The investigated ranges-limits are 40 g/s, 25% and 30 kW/m² for the total mass flow rate, the vapor quality and the heat flux density respectively. A tentative of modeling the thermo-hydraulic behavior of the loop with a separated flow model is proposed and wall heat transfer coefficients are compared with known correlations in both single phase natural convection and boiling flow.

2. The circulation loop facility
2.1. The experimental facility
The experimental facility, already described in [7] and shown in the picture of Figure 1, is built around a fluid reservoir at its top, a feeding downward tubing and a single tube test section where the flow is upward. The test section represents the heat exchanger part of the cryogenic cooling system. As it receives heat load, the overall density of the fluid decreases (either in single or two-phase) and this creates a weight unbalance between the two connected branches. Hence a flow is created. In two-phase flow the vapor exits the reservoir and the liquid level in the reservoir is maintained constant during the experiment.

The test section is a 0.95 m long and 10 mm inner diameter copper tube with a Manganin™ wire heater wrapped around it. The temperature of the tube is measured at five different locations with calibrated 1080 Cernox™ encapsulated sensors. The thermometers are inserted in small copper blocks, brazed on the copper tube, with copper powder grease to improve the thermal contact. The different temperature sensors locations from the bottom of the tubes are 3.1 cm, 24.3 cm, 47.3 cm, 71.3 cm and 92.7 cm. The facility is equipped with two mass flow meters: a gas flow-meter at room temperature measuring the nitrogen vapor mass flow rate, \( \dot{m}_v \), and a Venturi flow-meter measuring the liquid mass flow rate, i.e. the total mass flow rate, \( \dot{m}_t \) inside the downward tube. These two measurements allow to construct the vapor quality, \( x = \dot{m}_v / \dot{m}_t \).

2.2. Instrumentation and experimental errors
The temperature sensors are sold with an accuracy from 12 mK at 77 K to 36 mK at 300 K. The overall precision errors is about the double considering the electronic chain and the overall electromagnetic perturbations. It has been evaluated through direct fluctuation measurements. The thermometer wiring is thermally anchored at 77 K on a copper thermal copper rod directly in contact with the liquid nitrogen through the reservoir. In that way, in the range of the \( \Delta T \) investigated, several Kelvin, heat loss through the temperature sensors wiring is negligible. Therefore the temperature measured represents that of the inner wall. The gas flow meter is capable of measuring up to 50 g/s with an accuracy of 2% of the full scale. The precision on our total mass flow rate measurement varies from 10% at low mass flow rates down to 2% at high mass flow rates. Pressure differences are obtained with a room temperature sensor and their precision errors is around several Pa. The heat flux density is determined through a four-wire measurement technique and adding the error made on the heat transfer area, the overall error is around 1% at 200 W/m² and 0.8% at 30 kW/m².
3. Thermo-hydraulic study

3.1. Model

The model is based on one dimensional conservation equations as it is described in [14, 15] for example. The same set of conservation equations is used when modeling single or two-phase flow with a mass, momentum and energy conservation equation respectively,

\[
\frac{d}{dz}(\rho u) = 0,
\]

(1)

\[
\rho u \frac{du}{dz} = -\frac{dp}{dz} + \rho g \cos \theta - \frac{P}{A} \tau_w,
\]

(2)

\[
\rho u \frac{d}{dz}(h + \frac{u^2}{2} + gz) = \frac{P}{A} q,
\]

(3)

where \( z \) is the curvilinear axis along the length of the loop, \( u, \rho, p \) and \( h \) are respectively the velocity, the density, the pressure and the enthalpy of the fluid. \( P \) and \( A \) are respectively the perimeter and the cross-section of the test section, \( \theta \) is the angle of tubes with respect to the gravity direction, \( \tau_w \) is the average wall shear stress and \( q \) the heat flux density. There are four unknown variables in this equations system that are the velocity \( u \), the pressure \( p \), the enthalpy \( h \) and the mass quality \( x \) that do not appear directly in Eq. (1) to (3) for the moment. But when this system is solved separately in single phase and two-phase flow the number of unknown variables is reduced to three allowing for a complete solution.

In single phase flow, the mass quality is null \( (x=0) \). This stands until boiling appears in the loop, i.e. in the entire downward tubing and the test section until the saturation conditions are reached. The velocity and the density are the ones of the liquid, \( u = u_l \) and \( \rho = \rho_l \). The average
wall shear stressed, $\tau_w$, is expressed by a typical Fanning formulation as,

$$\tau_w = \frac{1}{2} C_{f,l} \rho_l u_l^2,$$

(4)

where $C_f$ is the Darcy friction factor. As the flow is always turbulent ($Re>2000$) in the data presented here, we use the Blasius law $C_f=0.0791.Re^{-0.25}$ for hydraulically smooth tubes [14, 15].

In the two-phase flow region, the variables are modified to account for the two parts of the mixture. For the regimes investigated in nitrogen two-phase flow in this study, a simple separated flow model has been used instead of the homogeneous model as it failed to reproduce our experimental data. In this model, the density and the velocity are replaced by average values as a function of the velocity and density of the two phases,

$$\rho_m = \alpha \rho_v + (1 - \alpha) \rho_l,$$

(5)

$$u_m = \frac{(1 - \alpha) \rho_l u_l + \alpha \rho_v u_v}{\rho_m},$$

(6)

where $\alpha$ is the void fraction and the subscripts $l$ and $v$ stand for the liquid and vapor components respectively. To take account for the friction in the two-phase flow part, the friction factor is multiplied by a coefficient $\phi$, called the two-phase frictional multiplier. So in Eq. (4), $C_{f,l}$ is replaced by $C_{f,l,\phi}$ where $\phi$ is given by the Lockhart-Martinelli correlation [16]. The separated flow model formulation introduces two extra variables, a velocity and the void fraction, and therefore some closure equations must be added for them. We used one of the simplest form of the separated flow model in taking the assumption that the velocity difference between the two phases can be expressed with a slip ratio, $S = u_v/u_l$, given by a empirical expression. For this study, we used the expression given by Huq and Loth [17]. Note that if $S=1$ then the set of equations represents the homogeneous model. The void fraction, $\alpha$, is given by the Lockhart-Martinelli correlation [16]. Finally to close the system, the enthalpy in the boiling region is $h = h_{sat} + L_v x$ since we assume that the flow stays in a saturated boiling regime. The entire system and closure equations have been implemented in the code Comsol Multiphysics® 5.2a in the same manner than in [18, 19] including the regular and the singular pressure drops.

3.2. Massflow rate and vapor quality comparison

The comparison between the experimental data and the model for the total mass flow rate, $\dot{m}_t$, is depicted in Figure 2 as a function of the heat flux density, $q$, dissipated on the tube. The total mass flow rate is computed from the velocity as $\dot{m}_t = \rho u A$ in the single phase region or $\dot{m}_t = \rho_m u_m A$ in the two-phase region. Both numerical and experimental results show clearly two flow regimes namely a gravity dominant regime at low $q$ and a friction dominant regime at higher $q$. In the gravity dominant regime, a small change in vapor quality creates a large change in the void fraction and therefore buoyancy force. As a result, the gravity dominant regime is characterized by an important increase in flow rate with $q$. The increased buoyancy force can be counter-balanced by an increase of the frictional force at higher flow rates due to the increase of the mixture velocity. Thus the friction dominant regime is characterized by a decrease in flow rate as $q$ increases. While both experimental and numerical results show a similar trend, a noticeable difference is found between them. The numerical code under predict the gravity dominant regime, i.e. the maximum mass flow rate is measured to be 42 g/s at 8 kW/m² whereas the code gives a value around 40 g/s at 5 kW/m². In the frictional dominant regime the decrease of $\dot{m}_t$ is measured to be more important than predicted by the code. Since we have not tried different correlations for the void fraction, slip ratio and two-phase friction coefficient, it is difficult to decide on the real reasons of this discrepancy. Nevertheless, one can
Figure 2. Comparison for the total mass flow rate, $\dot{m}$, between the model (---), and the experimental data (○) with respect to the heat flux density, $q$, for the 10 mm copper tube diameter.

suspect that the void fraction given by the code is too high in the gravity dominant regime and therefore shift the entire curve in intensity and $q$ making the $\dot{m}_t$ peak lower at lower $q$ value.

The computations reproduce with a better accuracy the evolution of the vapor quality at the exit, $x$, with respect to the heat flux density. There is a small deviation at heat flux density above 20 kW/m$^2$ as it is shown in Figure 3. This is a consequence of the underestimation of the total mass flow rate by our model, the vapor quality is overestimated after 20 kW/m$^2$.

4. Wall heat transfer

4.1. Boiling curve

A typical boiling curve, $q = f(\Delta T)$, is presented in Figure 4 for the five temperature sensors. The temperature difference is constructed as the difference between the temperature measurement at the wall, $T_w$, and the fluid temperature, $T_f$. The fluid temperature is computed from an energy balance equation in the sub-cooled part of the flow and elsewhere it is taken as the saturation temperature depending only on pressure. Every sensors show a similar temperature evolution typical of any boiling flow. At low heat flux, the curves are linear which is typical of a single phase flow. All the curves are superimposed except for the lowest height ($z=3.1$ cm (●)) which shows a higher slope. The heat transfer is thus higher and so far no acceptable explanation has been found for this temperature measurement even in the light of our previous measurement [13]. Therefore this result will not be considered in the rest of the analysis.

After the single phase region, the wall temperature difference diminishes with increasing heat flux. This transition marks the beginning of the nucleate boiling regimes. The apparition of boiling increases the wall heat transfer coefficient and decreases the temperature difference at the wall. This transition happens at different heat flux density depending on the height in the heated tube since the sub-cooling has a strong influences on it. Boiling appears first where the sub-cooling is lower, i.e. at the top of the tube as the Figure 4 shows. After the apparition of
boiling, the first regime encountered is the partial nucleate boiling where the boiling sites and the bubbles production frequency and size are increasing. Eventually a fully-developed nucleate boiling regime is reached for the highest heat flux. One has to admit that we cannot conclude on this issue since we stopped our experiment due to the limit of our power supply (1 kW). It certainly has been reached for the highest heights (z=71.3 cm (●) and z=92.7 cm (●)) but probably not for the lower heights which show surprising vertical evolution of the $\Delta T$ with $q$ as if the development of the nucleate boiling is delayed with $q$. To answer this question, more investigations must be undertaken to understand the wall heat transfer with respect to the tube heights at higher $q$.

4.2. Comparison with heat transfer correlations
It is instructive to compare the heat transfer coefficient with heat transfer correlations since it can, to some extent, answer the questions concerning the development of the nucleate boiling for this type of flow. The experimental heat transfer coefficient has been constructed as $h_{exp} = q/(T_w - T_f)$ and compared to the Dittus-Boelter’s correlation [20] in the single phase regime and to the Sha’s correlation [21] in the two-phase regime. Figure 5 shows this comparison for the temperature sensors at z=71.3 cm and z=92.7 cm. A vast majority of the data at z=92.7 cm is reproduced by the correlations within $\pm 20\%$, excepted, at the beginning of the the two-phase regime where the nucleate boiling flow is under development. Since the Sha’s correlation has been established for fully developed boiling flow, it is therefore expected that the experimental heat transfer coefficient is lower in this regime than that given by the Sha’s correlation. The experimental heat transfer coefficient obtained at z=71.3 cm is overall lower than that of Sha’s correlations. For the other heights (not shown in fig. 5), most of the data point fall into the $\pm 20\%$ deviation area but they are not following the linear evolution. The correlation is not completely satisfactory and future work should include measurement at higher heat flux and the testing of other void fraction and two-phase flow heat transfer correlations.
Figure 4. Evolution of the wall superheat ($\Delta T = T_w - T_f$) with respect to the heat flux density $q$ for different heights in the tube. $z=3.1$ cm ($\circ$); $z=24.3$ cm ($\varnothing$); $z=47.3$ cm ($\bullet$); $z=71.3$ cm ($\bullet$) and $z=92.7$ cm ($\bullet$).

Figure 5. Comparison between the experimental heat transfer coefficient, $h_{exp}$, and correlations, $h_{correlation}$. The data are compared to the Dittus-Boetler’s correlation [20] for $z=71.3$ cm ($\circ$) and for $z=92.7$ cm ($\bullet$) in the single phase regime and to the Sha’s correlation [21] for $z=71.3$ cm ($\circ$) and for $z=92.7$ cm ($\bullet$) in the two-phase regime. The straight lines (-----) shows the 20% deviation of the correlations.
In the single phase region, the Dittus-Boelter’s correlation give acceptable results for both heights shown in Figure 5. The same results are obtained with other heights, excepted for z=3.1 cm, for which half of the experimental data are two to three higher than predicted by the Dittus-Boelter’s correlation. More analysis needs to be done to investigate this regime at this height and a possible influence of the natural convection as it was found in [13].

5. Conclusions
Heat and mass transfer in a two-phase nitrogen circulation loop have been investigated with the ranges-limits of 40 g/s, 25% and 30 kW/m² for the total mass flow rate, the vapor quality and the heat flux density respectively around the atmospheric pressure. The results have shown that the thermo-hydraulic measurements can be modeled with a simple separated flow model, with a slip ratio, with an acceptable accuracy. Nevertheless, more work must be done to refine the model in finding a better combination of correlations for the slip ratio, void fraction and friction factor. Concerning the parietal heat transfer, in the single phase flow, the data are reproduced with the universal Dittus-Boelter’s correlation with an acceptable accuracy except for the ones taken at the lowest height (z=3.1 cm) that remain unexplained. In the two-phase regime, it has been found that, only at the highest height in the heated tube, the heat transfer regime can be reproduced properly by Sha’s correlation. For lower heights, where the sub-cooling is more important, the boiling curves shows unusual evolutions of the wall temperature difference with the heat flux. The Sha’s correlation is not very successful to reproduce the heat transfer evolution. It would be especially interesting to add, in future studies, experimental data at higher heat flux and try different heat transfer correlations.

References
[1] Desportes H, Bars J and Meuris C 1984 *Journal de Physique* 45 341–345
[2] Monroe C, et al Ross J, Shrimpton G and Smith K 1988 The CLEO II magnet-design, manufacture and tests *Proceedings of the 12th International Cryogenic Engineering Conference, ICEC 12* pp 773–9
[3] Brindza P D, and et al 2006 *IEEE Trans. Applied Supercond.* 16 248–252
[4] Dupont T, Court J C and Perinic G 2010 Commissioning of the CMS cryogenic system after final installation in the underground cavern *Adv. Cryo. Eng., Vol 55 (AIP Conference Proceedings vol 1218)* pp 3–10
[5] Gastineau B and et al 2010 *IEEE Trans. Applied Supercond.* 20 328–331
[6] Gastineau B and et al 2012 *IEEE Trans. Applied Supercond.* 22 900–1004
[7] Baudouy B 2002 *AIP Conference Proceedings* 613 1514–1524
[8] Benkheira L, Souhar M and Baudouy B 2006 *AIP Conference Proceedings* 823 871–878
[9] Baudouy B 2004 *AIP Conference Proceedings* 710 1107–1114
[10] Benkheira L, Baudouy B and Souhar M 2007 *International Journal of Heat and Mass Transfer* 50 3534 – 3544
[11] Umekawa H, Ozawa M and Ishida N 1997 *Heat Transfer - Japanese Research* 26 449–458
[12] Kim M J and Chang H M 2008 *AIP Conference Proceedings* 985 59–66
[13] Baudouy B 2010 *AIP Conference Proceedings* 1218 1546–1553
[14] Wallis G B 1969 *One-dimensional Two-phase flow* (MacGraw-Hill, New York)
[15] Collier J G and Thome J R 1994 *Convective Boiling and Condensation.* (Oxford University Press, Oxford)
[16] Lockhart R W and Martinelli R C 1949 *Chemical Engineering Progress* 45 39–48
[17] Huq R and Loth J L 1992 *J. Therm. Heat Trans.* 6 139–144
[18] Baudouy B 2012 *AIP Conference Proceedings* 1434 717–723
[19] Baudouy B, Bessette A and Four A 2013 *Cryogenics* 53 2 – 6
[20] McAdams W H 1942 *Heat Transmission, 2nd edn.* (McGraw-Hill, New York)
[21] Shah M M 1976 *ASHRAE Trans.* 82 66–86