Forced convective heat transfer to supercritical nitrogen in a vertical tube

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The results of investigation on forced convective heat transfer to supercritical nitrogen are presented. The experiments are conducted with a specially constructed setup. The test section, made of a 20–22-mm-diameter stainless steel tube is heated along 1471 mm of its length by passing direct current. The flow is turbulent and oriented upward. Working parameters during experiments were $p=4$ MPa; $\dot{m}=12$–26 kg/(m^2·s); $T_{in}=101$–125 K; $q_{in}=9$–21 kW/m^2. The calculated values of the criteria were $Re_b=3.9\times10^3$–4.2×10^4 and $\overline{Gr}_b=4.6\times10^7$–$1.5\times10^{10}$. Experimental curves of the heat transfer coefficient show two peaks when the difference between wall and bulk temperatures spans the region of severe variations of the physical properties of the fluid.

Keywords: forced convection; heat transfer coefficient; supercritical nitrogen; upward flow

Introduction

The design and optimization of cryogenic devices cooled by supercritical fluid flows require generalized recommendations on heat transfer prediction. However, the severe variations of physical properties upon temperatures in the vicinity of the critical point lead to a complex interference between fluid dynamics and heat transfer. So a separate description of hydrodynamics and heat transfer by adequate dimensionless terms is usually impossible. Our knowledge of heat transfer characteristics is by no means satisfactory for the identifying operating regions in which conventional heat transfer correlations can be aptly applied or the region where either augmentation or deterioration of heat transfer coefficient might occur. The magnitude of the augmentation and deterioration cannot be estimated properly either. So much work must be done before an exact and complete method for heat transfer description in a supercritical region can be established. This paper reports experimental data on heat transfer studies to a supercritical nitrogen flowing upward in a circular tube.

Experimental apparatus and procedures

The complete scheme of the experimental installation is given in Ref. 1. The nitrogen flowing through a closed loop is compressed up to the working pressure. Being cooled in a heat exchanger and in a liquid nitrogen pan, the fluid enters the test section. On the section outlet pressure is reduced to 0.1 MPa, and the flow rate of the stream is measured. Figure 1 shows the scheme of the experimental chamber. The test section (1) is a circular stainless steel tube, 20–22 mm in diameter and 1658 mm long, heated along 1471 mm of its length by passing direct current. On both ends it is joined to the rest of the loop by electrically insulated flanges (2) to prevent current leakage. The bulk enthalpy of flow is measured by inlet and outlet mixing chambers (3), joined to the entrance and to the exit of the test section, respectively. Between the mixing chamber and the test section two outlets (4) are made for the working pressure to be measured. For reducing possible heat losses, the test section is insulated by a copper shield (5) and put into a Dewar flask (6), the inside of which is evacuated to less than 100 Pa. The heat losses can be estimated from the temperature distribution along the copper shield (measured by four thermocouples), the vacuum in the Dewar, and the dimensions of the system. The wall temperature outside is measured by differential copper-constantan thermocouples (7) uniformly distributed at 15 cross sections along the tube. The apparatus allows a variation of the angle between the fluid flow and the gravitational field from 0 to 90, as well as reversing the direction of the flow.

Before the experiments were started in the supercritical region, some data were taken at 0.5, 1, and 4 MPa and at the inlet temperature of 100 K in order to check the accuracy of the thermocouples. The difference in the obtained values was lower than 0.5°. All experiments were conducted at a test section pressure of 4 MPa to reduce potential errors during data processing. However, we cannot express our results in dimensionless terms because of the lack of data concerning the physical properties (λ, in particular) for supercritical nitrogen at such a pressure. The experimental results were processed using data for the thermophysical properties of nitrogen.[2,3]

Results and discussion

The distribution of the inner wall surface temperature versus the bulk stream enthalpy, calculated by the heat balances, are given in Figures 2 and 3, where the fluid temperature corresponding to the bulk enthalpy is shown (thick solid line) as well, together with the experimental points related to each particular flow regime. To determine the inner wall surface temperature, we considered the test section as a circular wall

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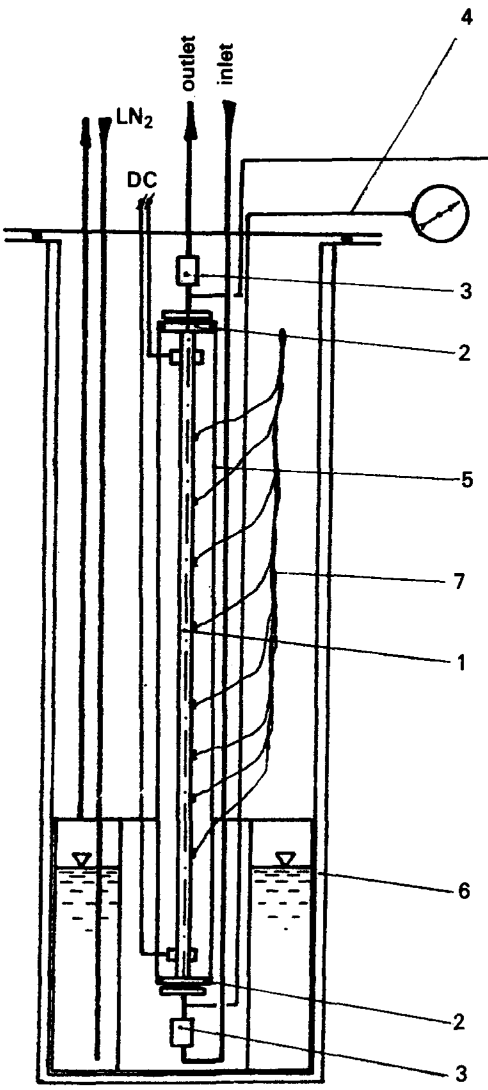

Figure 1 Experimental setup

with internally distributed heat sources where heat exchange occurs through the inside tube surface only. The temperature drop through the wall, calculated from

$$\delta t = q_{in} r_2^2 r_1 \frac{2\ln(r_2/r_1) + (r_1/r_2)^2 - 1}{2k(r_2^2 - r_1^2)} \tag{1}$$

is subtracted from the measured wall temperature. An error

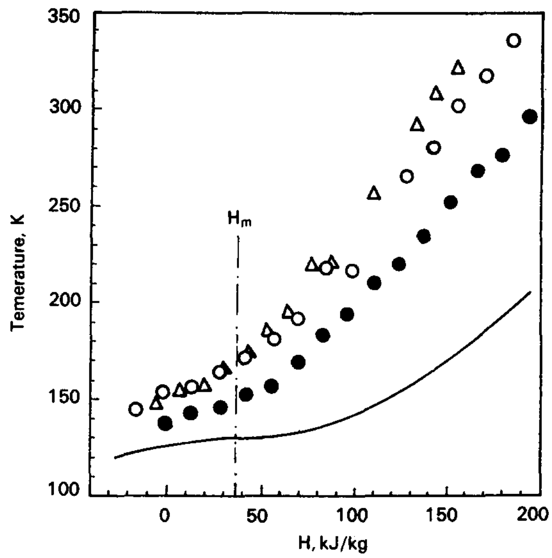

Figure 2 See Figure 3 caption

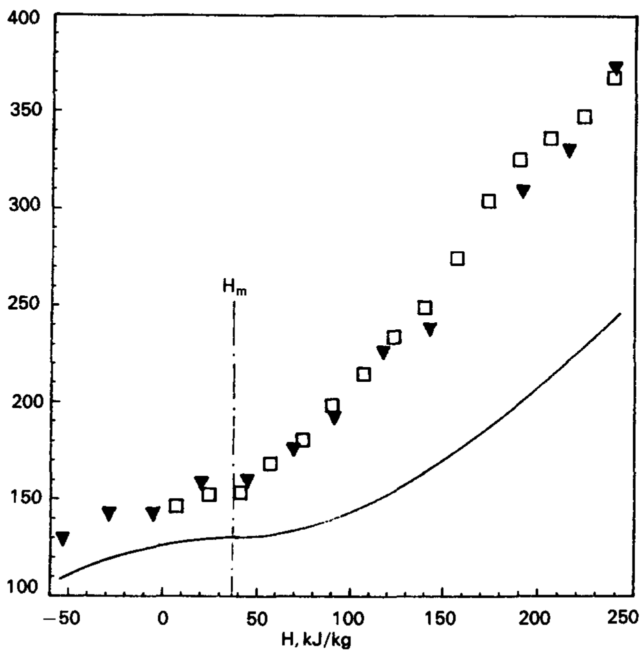

Figure 3 Temperature distribution versus bulk enthalpy at pressure $p = 4$ MPa. ▼: $\dot{m} = 11.91$ kg/(m²·s); $Q = 1738$ W; ●: $\dot{m} = 14.61$ kg/(m²·s); $Q = 1079$ W; □: $\dot{m} = 16.53$ kg/(m²·3); $Q = 1464$ W; ○: $\dot{m} = 23.29$ kg/(m²·s); $Q = 1822$ W; △: $\dot{m} = 25.46$ kg/(m²·s); $Q = 1826$ W

Notation

R	Electrical resistance, Ω·m
g_n	Gravity, m/s²
H	Enthalpy, kJ/kg
I	Dc magnitude, A
k	Thermal conductivity of stainless steel, W/(m·K)
$\dot{m}$	Mass flow rate, kg/(m²·s)
p	Pressure, Pa
q_{in}	Average heat flux, related to inner wall surface, W/m²
T_{in}	Inlet bulk temperature, K
α	Heat transfer coefficient, W/(m²·K)
η	Viscosity, kg/(m·s)
λ	Thermal conductivity coefficient of nitrogen, W/(m·K)
ρ	Density, kg/m³
$\bar{\rho}$	$\int_{T_b}^{T_w} \rho \, dT/(T_w - T_b)$
$\overline{Gr}_b$	$8\rho_b(\rho_b - \bar{\rho}) r_1^3 g_n/\eta^2$
Re_b	$2\dot{m} r_1/\eta$
r_1, r_2	Inner, outer pipe radius, m

Subscripts

w	Wall
b	Bulk

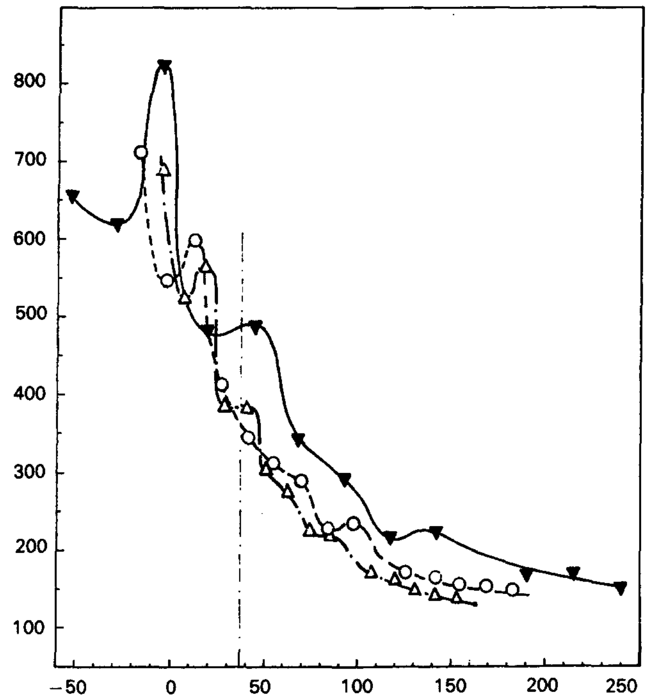

Figure 4 See Figure 5 caption

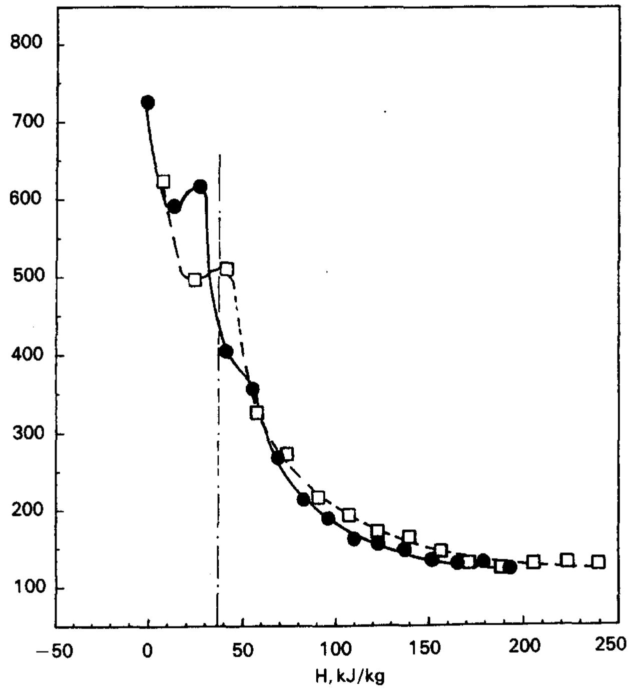

Figure 5 Heat transfer coefficient versus bulk enthalpy at pressure $p=4$ MPa. ▼: $\dot{m}=11.91$ kg/(m²·s); $Q=1738$ W; ●: $\dot{m}=14.61$ kg/(m²·s); $Q=1079$ W; □: $\dot{m}=16.53$ kg/(m²·3); $Q=1464$ W; ○: $\dot{m}=23.29$ kg/(m²·s); $Q=1822$ W; △: $\dot{m}=25.46$ kg/(m²·s); $Q=1826$ W

can occur if the heat power leakage to the insulating vacuum is neglected. The calculation of that power leakage was carried out by using both measured temperatures at the shield, residual pressure in the Dewar, and the enthalpy balance for the test section. Power leakage was estimated to be less than 5%. We computed average heat flux in a particular cross section related to the inner diameter of the tube by the equation

$$q_{in}=\frac{I^2R}{2r_1\pi^2(r_2^2-r_1^2)} \tag{2}$$

For our experiments $q_{in}=9$–$21\ \mathrm{kW/m^2}$. The curves in Figures 2 and 3 exhibit a similar behavior to that previously reported.[4,5] The lower magnitude of the wall temperature peaks obtained in our experiments is probably due to the relatively lower values of q_{in}.

The pecularities of heat transfer in the vicinity of supercritical temperatures are better illustrated in Figures 4 and 5, where the calculated values of heat transfer coefficient versus bulk flow enthalpy are plotted. One can see that a sharp maximum of α appears when bulk enthalpy is lower than H_m (enthalpy value corresponding to pseudocritical temperature), and a weaker peak appears in the vicinity of H_m. With further increase of bulk enthalpy for three of the present regimes, a slight minimum in α occurs. Such behavior of the heat transfer coefficient probably results from the joint influence of forced fluid convection with variable properties and buoyancy effects.

In order to assess how far buoyancy effects influence heat transfer W. B. Hall and J. D. Jackson's criterion was applied:[6]

$$\frac{\overline{\mathrm{Gr}}_b}{\mathrm{Re}_b^{2.7}}<10^{-5} \tag{3}$$

The calculated values of the ratio on the left side of Equation 3 were between 3.88×10^{-4} and 0.48 for our results. That means a violent influence of buoyancy effects for the present data should be expected. Hall[7] mentioned that Equation 3 must be regarded with caution when the heat flux is too low to produce a temperature difference incapable of spanning the region of rapid property variations. This is not the case for the data presented. The difference between the wall and the bulk temperatures is sufficient to cover the entire region of density drop. Nevertheless we could not say that Figures 2 or 3 show a significant deterioration in heat transfer.

References

1 Dimitrov, D., Georgiev, J., Zahariev, A., and Kovachev, V. An experimental equipment for heat transfer studies of supercritical cryogenic liquid forced flow in tubes. *Univ. Ann. Techn. Phys., Sofia*, 1985, **22**, 127

2 Wargaftik, N. *Handbook of thermophysical properties of gases and liquids*, Nauka, Moscow, 1972

3 Strobridge, T. *The Thermodynamic Properties of Nitrogen from 64 to 300 K between 0.1 and 200 Atmospheres*. National Bureau of Standards, Washington, D.C., 1962

4 Shitsman, M. Impairment of the heat transmission at supercritical pressures. *Teplofiz. Vis. Temp.*, 1963, **1**(2)

5 Jackson, J. and Evans-Lutterodt, K. Impairment of turbulent forced convection heat transfer to supercritical presure CO_2 caused by buoyancy forces. Rept. N.E. 2, Simon Engg. Labs., Univ. of Manchester, 1968

6 Hall, W. B. and Jackson, J. D. Heat transfer near the critical point. *Sixth Int. Heat Transfer Conf.*, Toronto, Canada, Aug. 7–11, 1978, vol. 6, pp. 377–392

7 Hall, W. B. Heat transfer near the critical point. *Advances in Heat Transfer*, 1971, 7, 2–86