Forced convection heat transfer of subcooled liquid nitrogen in a vertical tube

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Abstract. Experimental research on forced convection heat transfer of subcooled liquid nitrogen ranging from the pressures of 0.3 MPa to its supercritical pressure is carried out for wide ranges of inlet temperature and flow velocity. A stainless steel tube heater with the inner diameter of 5.4 mm and the length of 100 mm is mounted vertically. The heat transfer coefficients in non-boiling region and the DNB (departure from nucleate boiling) heat fluxes are higher for higher flow velocity and higher subcooling. The trend of the heat transfer coefficients in the non-boiling region agree with those by the Dittus-Boelter correlation, although they are unaffected by the flow velocity for \(Re<8000\). The lowest limits agree with those obtained for no forced flow. The correlation of DNB heat flux that can describe the experimental data is presented. On the other hand, forced convection heat transfer of supercritical nitrogen deteriorates when the heated surface temperature, \(T_w\), exceeds its pseudo-critical temperature, \(T_c'\), although they are similar to those under subcritical pressure for \(T_w<T_c'\). The heat transfer characteristics of supercritical nitrogen can be predicted by authors’ correlation.

1. Introduction

It is important for the design of HTS superconducting cable to understand forced convection heat transfer of subcooled liquid nitrogen. Until now, heat transfers in a pool of saturated and subcooled liquid nitrogen have been mainly investigated [1-6]. Coeling et al. [1] reported that natural convection heat transfer form a flat disk plate to saturated liquid nitrogen and hydrogen can be predicted by conventional correlations. Shiotsu et al. [2-5] investigated steady-state and transient heat transfer from a horizontal wire in subcooled liquid nitrogen. However, there are few investigations for forced convection heat transfer of liquid nitrogen. Tatsumoto et al. [7] developed the experimental apparatus that produced a stable forced flow up to Reynolds number, \(Re\), of \(10^5\) without a pump. They measured forced convection heat transfer of subcooled liquid nitrogen and supercritical nitrogen in a horizontal tube [7, 8]. They reported that the heat transfer coefficients in non-boiling region agreed with the Dittus-Boelter correlation [9]. Departure from nucleate boiling (DNB) heat fluxes in forced flow could not be predicted by a correlation for water [10]. The heat transfer characteristics under supercritical pressure deteriorated when the wall temperature exceeded its pseudo critical temperature.

In this study, forced convection heat transfer from a vertical tube heater to subcooled liquid nitrogen and supercritical nitrogen in a vertical tube were measured for wide ranges of pressure, inlet temperature and flow velocity.
2. Experimental apparatus and method

An experimental apparatus that easily produced a forced flow of liquid nitrogen was developed as shown in figure 1. The details are presented in other paper [7]. The main cryostat and a compressed nitrogen gas cylinder are put on a scale that can measure up to 150 kg within 0.01 kg resolution. The main cryostat is connected to a receiver tank through a flexible transfer line with a manual valve. Test tube heater is installed to one end of the transfer line in the main cryostat.

![Figure 1](image1.png)

**Figure 1.** Experimental apparatus for a forced flow of liquid nitrogen.

Figure 2 shows the test tube heater that is vertically immersed in the main cryostat. The test heater made of stainless steel is 5.4 mm in inner diameter, \(d\), 100.0 mm in length, \(L\), and 0.3 mm in thickness, \(t\). The outside of the test heater is insulated by Fiber-Reinforced Plastic (FRP) block, which is 112.0 mm in length and 66.0 mm in outer diameter. The length of the FRP block is slightly longer than the heater in order to electrically insulate it from the transfer line. The test heater has an entrance length of 200 mm, which corresponds to \(37d\). This is because the forced flow through the tube heater, which has

![Figure 2](image2.png)

**Figure 2.** Schematic of test heater.
Reynolds number, $Re$, higher than 3000 in this study, can be regarded as turbulent flow. Liquid nitrogen flows through the heater in the opposite direction of gravity. The heat leakage to the liquid nitrogen through the FRP block is estimated to be less than 2 %.

Forced flow through the heater is produced by adjusting the valve opening and the pressure difference between the tanks. The average mass flow rate is measured from the weight change of the main cryostat and the nitrogen gas cylinder. It was confirmed that the weight change was unaffected by the pipes connected to the main tank because a flexible tube was partially adopted. The flow measurement error is estimated to be within 0.25 g/s. The liquid temperature is measured by a platinum resistance temperature sensor.

The heating current to the test heater is supplied by a power amplifier. The power amplifier can supply a direct current of up to 400 A at a power level of 4.8 kW. The input signal of the power amplifier is controlled by an analog computer so that the heat generation rate in the test heater agrees with a desired value. The electric resistance of the test heater is measured using a double-bridge circuit including the test heater as a branch of the bridge, and then the average temperature is estimated by the aid of the resistance temperature characteristic that has been previously measured at saturated temperatures ranging from atmospheric pressure to critical pressure. The double bridge circuit is first balanced at bath temperature. The output voltages of the bridge circuit, together with the voltage drops across the potential taps of the heater and across a standard resistance, are amplified and are simultaneously sampled at the interval of 30 ms. The heat generation in the heater is calculated from the measured voltage drop across the heater and the standard resistance. The surface heat flux is the difference between the heat generation rate and the time rate of change of energy storage in the heater.

The average surface temperature of the heater, $T_w$, and heat flux, $q$, are calculated from the average temperature and the heat generation by solving the conduction equation in the radius direction of the heater.

The double bridge circuit to measure the heater resistance has the accuracy of $1 \times 10^{-4}$. The gradient of the resistance-temperature relation is lowest at the boiling point, where the gradient is about $30 \mu\Omega/K$ at the resistance of 15 mΩ. As a result, the temperature deviation of about 0.05 K can be measured by the bridge. However, as the dynamic range of heating current is about 30, temperature deviation of about 0.15 K can be measured at the start of heating. Experimental error is estimated to be within the heater surface temperature of 0.2 K and the heat flux of 2 %.

The forced convection heat transfers from inner side of the vertical heater are measured at the pressures ranging from 0.3 MPa to 3.5 MPa by quasi-steadily increasing heat inputs ($Q=Q_0e^{\tau}$, $\tau = 2.0$ s) The inlet temperatures, $T_{in}$, are varied from 78 K to around its saturated temperature, $T_{sat}$.

3. Results and discussion

3.1. Steady state heat transfer characteristics

Figure 3 shows forced convection heat transfer curves at a pressure, $P$, of 0.5 MPa and the inlet subcooling, $\Delta T_{sub}$, of (a) 10 K and (b) 16.2 K on the graph of heat flux, $q$, versus the temperature difference, $\Delta T_L$, between the surface temperature, $T_w$, and the inlet temperate, $T_{in}$. With an increase in the heat input, the heat flux gradually increases, and then the nucleate boiling occurs at the heat flux of $q_i$, where $T_w$ is higher than saturated temperature, $T_{sat}$. The heat flux rapidly increases up to a departure from nucleate boiling (DNB) heat flux, $q_{DNB}$ with relatively little increase in wall superheat. The heat transfer mechanism shifts to film boiling after the DNB heat flux. For a low flow velocity such as 0.22 m/s, the heat flux rapidly decreases after the DNB heat flux and the surface temperature, $T_w$, drastically increases. It seems that the whole heated surface would be instantaneously covered with a vapor film at the DNB heat flux.

With an increase in the flow velocity, the heat transfer coefficient in the non-boiling region, $h_n$, the heat flux at the onset of nucleate boiling, $q_i$, and DNB heat flux become higher. The heat transfer curves in the nucleate boiling regime exist on the identical extrapolation curve. For the flow velocity of 1.6 m/s, the heat flux continues to increase after the DNB heat flux, although the gradient of the
heat transfer is lower than that in nucleate boiling. For higher flow velocity, the thermal boundary
layer would become thinner and the temperature rise of the mainstream from the inlet temperature, \( T_{in} \),
would be small. The heat transfer would be dominated by nucleate boiling except for around the outlet
of the heater where vapor film would be locally formed at the DNB heat flux. With a further increase
in the heat input, the vapor film becomes progressively larger toward the heater inlet. It is assumed
that the heat transfer would be dominated by intermixture of nucleate boiling and film boiling. All
heated surfaces would be eventually covered with a vapor film at the heat flux of \( q_f \) shown in Fig.3 (b),
and then the heat transfer characteristic shifts to film boiling regime. With a further increase in flow
velocity, the reduction of the heat flux becomes smaller after \( q_f \) and the surface temperature, \( T_w \),
continuously increases.

With an increase in subcooling, the values of the heat transfer coefficient in the non-boiling region,
\( h_n \), the heat flux at the onset of nuclear boiling, \( q_i \), and the DNB heat flux, \( q_{DNB} \), are higher. However,
it seems that the difference between \( q_i \) and \( q_{DNB} \), where heat transfer characteristic is dominated by
nucleate boiling regime, becomes smaller.

\[
\Delta T_L (T_w - T_{in}) (K)
\]

(a) Inlet subcooling, \( \Delta T_{sub} \), of 10 K. (inlet temperature, \( T_{in} \), of 84 K)

(b) Inlet subcooling, \( \Delta T_{sub} \), of 16.2 K. (inlet temperature, \( T_{in} \), of 77.84 K)

**Figure 3.** Forced convection heat transfer curves for the pressure of 0.5 MPa.
Forced convection heat transfer processes for higher pressures and large subcooling of 25.7 K; (a) $P = 1.0$ MPa and (b) $P = 2.0$ MPa are shown in figure 4. As the flow velocity increases, the heat transfer coefficients in non-boiling region become larger and the difference between $q_i$ and $q_{DNB}$ becomes smaller. And the obvious DNB heat flux and inception heat flux of nucleate boiling also disappear above a certain flow velocity.

![Figure 4](image)

**Figure 4.** Forced convection heat transfer curves at subcooling of 25.7 K for high pressure.

Forced convection heat transfer processes at a supercritical pressure of 3.5 MPa for the inlet temperature, $T_{in}$, of 78 K are shown in figure 5. For supercritical nitrogen at a pressure of 3.5 MPa, pseudo-critical temperature, $T_c^*$, where the maximum specific heat of fluid appears, is 126.8 K. For $T_w < T_c^*$, the heat flux gradually increases with an increase in the heat input. The heat transfer coefficient becomes higher as the flow velocity increases as well as the case of subcritical pressure. For $T_w > T_c^*$, the heat transfer deteriorates progressively. It seems that the improvement of the heat transfer is observed in high $\Delta T_L$ region. For flow velocity of 0.24 m/s, the improvement appears at $\Delta T_L = 80$ K. It seems that the surface temperatures at the onset of the improvement are higher for higher flow velocities.

![Figure 5](image)

**Figure 5** Forced convection heat transfer at supercritical pressure of 3.5 MPa for the inlet temperature, $T_{in}$, of 78 K.
3.2. Heat transfer coefficients below saturated temperature and pseudo-critical temperature

Figure 6 shows the effects of Reynolds number, $Re$ on Nusselt number, $Nu$, and Prandtl number, $Pr$, below $T_{sat}$ and $T_{c'}$. The values of $Nu$ are obtained by using the measured heat transfer coefficients, $h_m$. The data of $Nu/Pr^{0.4}$ for no forced flow ($v = 0$ m/s) and $\Delta T_w = 10$ K are also shown in terms of a broken line. For $Re$ higher than 8000, the trend of the measured values of $Nu/Pr^{0.4}$ agree with those predicted by the Dittus-Boelter correlation [9], which is well known as a correlation for single phase turbulent forced convection heat transfer in non-boiling region.

$$Nu = 0.023Re^{0.8}Pr^{0.4}$$ (1)

On the other hand, for $Re < 8000$, the values of $Nu/Pr^{0.4}$ deviate from the curve with $Re^{0.8}$ and approach a constant value that corresponds to that for no forced flow. It is found that the lower limit of heat transfer of forced flow is determined by that of natural convention. For comparison, the experimental data for the same structure tube mounted horizontally [7, 8] are also shown in the figure. It is found that the heat transfer coefficient in non boiling region is not affected by the support orientation.

![Figure 6. Heat transfer characteristics of subcooled nitrogen below saturated temperature, $T_{sat}$, and supercritical nitrogen below pseudo-critical temperature, $T_{c'}$.](image)

3.3. DNB heat flux

Figure 7 shows the influence of the flow velocity on the DNB heat flux at the pressure of 0.5 MPa and 2.0 MPa for various inlet subcoolings, $\Delta T_{sub}$, by way of example. The DNB heat fluxes are higher for higher flow velocity and subcooling. Hata et al. [10] measured the DNB heat fluxes of subcooled water in vertical tubes for the pressures ranging from 0.16 MPa to 1.0 MPa in order to establish the database for designing the divertor for a helical type fusion experimental device. They derived the following correlation for DNB heat flux that could describe their experimental data within 15% difference [10].

$$Bo = 0.082 \left[ \frac{1}{d} \left( \frac{\sigma}{g(\rho_l - \rho_g)} \right) We^{-0.3} \left( \frac{L}{d} \right)^{-0.1} Sc_{in}^{0.7} \exp\left( - \frac{L/d}{0.53Re^{0.4}} \right) \right]$$ (2)

$$Bo = \frac{q_{DNB}}{Gh_{fg}}$$ (3)

$$We = \frac{G^2d}{\rho_l \sigma}$$ (4)
where \( Bo \) is the boiling number, \( We \) is the Weber number, \( Sc_{in} \) is a non-dimensional inlet subcooling, \( G \) is the mass flux, \( h_{fg} \) is the latent heat of vaporization, \( \sigma \) is the surface tension, \( g \) is the acceleration of gravity, \( \rho \) is the density, \( c_{p} \) is the specific heat at constant pressure, \( \Delta T_{sub} \) is the inlet liquid subcooling, \( L \) is the tube length, and \( d \) is the tube diameter.

\[
Sc_{in} = \frac{C_p \Delta T_{sub}}{h_{fg}}
\]  

(5)

The DNB heat fluxes predicted by equation (2) are also shown in figure 7 for comparison. The predicted values for the pressure of 0.5 MPa and the inlet subcooling, \( \Delta T_{sub} \) of 4.9 K are almost in agreement with the experimental data, although they are a little bit lower. For \( \Delta T_{sub}=15.9 \) K, the deviation becomes larger with increase in flow velocity. For example, the predicted value at \( v = 3.0 \) m/s is about 35% higher than the experimental data. For higher pressure of 2.0 MPa, the tendency is similar to that for 0.5 MPa. Accordingly, it seems that the effect of subcooling on the DNB heat flux of liquid nitrogen would be smaller than that of water.

The effect of the subcooling on the DNB heat flux are considered as shown in figure 8, and a correlation of DNB heat flux based on the experimental data is derived. All the data of DNB heat

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\text{Figure 7. Effect of flow velocity on DNB heat flux.}
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\text{Figure 8. Relationship between BoSc}_{in}^{-0.55} \text{ and We for subcooled liquid nitrogen in forced flow.}
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fluxes obtained in this work are plotted on log ($BoSc_{in}^{-0.55}$) vs. log ($We$) graph. All the values of $Bo Sc_{in}^{-0.55}$ exist on the straight line with $We^{-0.3}$. The following correlation can describe the experimental data of subcooled liquid nitrogen flowing through the vertical tube within ±15% difference.

$$Bo = 0.014 We^{-0.3} Sc_{in}^{0.55}$$  \hspace{2cm} (6)

The DNB heat fluxes for a horizontal tube [7] are also shown in the figure for comparison. Although the data of $Bo Sc_{in}^{-0.55}$ for the horizontal tube also agree with those for vertical tube for $We > 20$, they exist lower than the straight line with $We^{-0.3}$ for $We < 20$. For low flow velocity, it would be easy to collect bubbles at the upper side of the tube heater due to the influence of buoyancy. Therefore, it is considered that the effect of the support orientation of the tube heater on the DNB heat flux appears.

3.4. Heat transfer correlation for supercritical nitrogen

As shown in figure 6, the heat transfer coefficients for $T_w < T_c$ agree with those predicted by the Dittus-Boelter correlation. Unlike the case of subcritical pressure, the heat transfer deterioration appears for $T_w < T_c$ as shown in Figure 5. Authors [11] have presented a new generalized correlation of heat transfer to supercritical fluid flow by modifying the Dittus-Boelter correlation.

$$Nu = 0.023Re^{0.8} Pr^{-0.4} F_c$$  \hspace{2cm} (7)

$$F_c = \left[1 + 108.7 \left(\frac{d}{L}\right)^{2} \right]^{0.25} \left[1 + 0.002 \left(\frac{\Delta T}{T_{cr}}\right) \left(\frac{\rho_w}{\rho_B}\right)^{0.34} \left(\frac{\mu_B}{\mu_w}\right)^{0.17}\right]$$  \hspace{2cm} (8)

Prandtl number, $Pr$, is defined using the average heat capacity of $\overline{Cp}$ as follows.

$$\overline{Pr} = \frac{\overline{Cp}}{\lambda_B}$$  \hspace{2cm} (9)

They have reported that the correlation can predict the experimental data of supercritical helium measured by Shiotsu et al. [12] and those for non cryogenic fluids such as water and carbon dioxide at supercritical pressure [13-15]. Figure 9 shows the effects of $Re$ on $Nu$, $Pr$ and $F_c$ for various temperature differences, $\Delta T$, between $T_w$ and $T_c$. Here, thermophysical properties are taken at the inlet temperature except for $C_p$. With an increase in $Re$, the measured values of $Nu \overline{Pr}^{-0.4} F_c^{-1}$ under supercritical pressure, which is independent of $T_{in}$ and $\Delta T$, increase along the straight line with the

![Figure 9](image-url)

**Figure 9.** Comparison of heat transfer coefficient under supercritical pressure with authors’ correlation. ($\Delta T$: temperature differences between surface temperature, $T_w$, and pseudo-critical temperature, $T_c$; $T_{in}$: inlet temperature)
gradient of $0.023Re^{0.8}$. It is confirmed that the correlation can also predict the experimental data for supercritical nitrogen.

4. Conclusions
The steady-state heat transfers from inner side of the vertical heater to forced flow of subcooled liquid nitrogen were measured for various inlet temperatures and flow velocities under pressures from 0.3 to 3.5 MPa. Experimental results led to the following conclusions.

The heat transfer coefficients of subcooled liquid nitrogen in the non-boiling region, $h_n$, inception of nuclear boiling, $q_i$, and the DNB heat fluxes, $q_{DNB}$, become higher with an increase in flow velocity. With a further increase in flow velocity, an obvious DNB heat flux disappears.

The heat transfer characteristics in non boiling region agreed with those by the Dittus-Boelter correlation. However, for $Re < 8000$, they approached a constant value that corresponds to that for no forced flow.

Non-dimensional equation of DNB heat flux for forced flow of subcooled water by Hata et al. [7] was not express the experimental data in liquid nitrogen, especially for large subcooling. A correlation of DNB heat flux in forced flow of subcooled liquid nitrogen was presented by considering the effect of subcooling.

For $T_w < T_c'$, the heat transfer characteristic under supercritical pressure was similar to that under subcritical pressure. When the heated surface temperature exceeded $T_c'$, the heat transfer deterioration appeared. It was confirmed that the heat transfer characteristics can be predicted by authors’ correlation previously presented for various fluids.

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