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Performance of extended surface from a cryocooler for subcooling liquid nitrogen by natural convection

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Abstract

Natural convection of subcooled liquid nitrogen under a horizontal flat plate is measured by experiment. This study is motivated mainly by our recent development of cryocooling systems for HTS power devices without any forced circulation of liquid nitrogen. Since the cold surface of a GM cryocooler is very limited, the cooling plate immersed in subcooled liquid nitrogen is thermally anchored to the cryocooler located at the top in order to serve as an extended surface. A vertical plate generating uniform heat flux is placed at a given distance under the cooling plate so that subcooled liquid may generate cellular flow by natural convection. The temperature distributions on the plates and liquid are measured during the cool-down and in steady state, from which the heat transfer coefficients are calculated and compared with the existing correlations for a horizontal surface with uniform temperature. A fair agreement is observed between two data sets, when the heat flux is small or the plate temperatures are relatively uniform in horizontal direction. Some discrepancy at higher heat flux is explained by the cellular flow pattern and the fin efficiency of the extended surface, resulting in the non-uniformity of the horizontal plate.
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1. Introduction

Liquid nitrogen cooling systems have been developed for the HTS (high temperature superconductor) power devices over the past several years since liquid nitrogen is a cheap and excellent cooling medium and also a good electrical insulator. Most of commercial HTS systems cooled by liquid nitrogen employ a cryocooler in order to acquire the continuous operation and at the same time to eliminate the boil-off and maintain the liquid at the subcooled state. In cooling for HTS power applications, subcooled liquid nitrogen has several advantages over saturate liquid [1,2]. First of all, the temperature of subcooled liquid is lower than that of saturated liquid for a given pressure, which results in an increased critical current of HTS. Another advantage of subcooling is the augmentation of electric insulation characteristics since the probability of bubble generation on the heated surface decreases when the liquid is cooled well below its saturation temperature.

Recently, the compact and efficient cooling systems by natural convection of subcooled liquid nitrogen have been reported [3–5]. Chang et al. [3] have proposed a new cryogenic concept for HTS transformer, operating in the range of 63–66 K by natural convection of subcooled liquid nitrogen between the HTS magnets and the vertical copper plate extended from a GM cryocooler. In order to confirm the feasibility of the natural convection cooling for HTS transformers using subcooled liquid nitrogen, an experiment was designed, constructed and performed [4]. In this system, as the cryocooler is located at the top, the temperature of solid bodies including the vertical copper plate and HTS magnets decrease upwards, which result in vertically

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Nomenclature

A	surface area (m^2)
C	specific heat (kJ/kg K)
g	gravitational acceleration (m/s^2)
h	convective heat transfer coefficient (W/m^2 K)
h_{eff}	effective heat transfer coefficient (W/m^2 K)
k	thermal conductivity (W/m K)
L	characteristic length (m)
m	mass (kg)
Nu	Nusselt number, $h \cdot L/k$
P	perimeter which encompass the area (m)
Q	heat transfer rate (W)
Ra	Rayleigh number, $g\beta \cdot \Delta T \cdot L^3/(\nu \cdot \alpha)$
r	radius (m)
T	temperature (K)
t	time (s)
z	distance from bottom of vertical plate (m)
Greek letters	
β	thermal expansion coefficient (1/K)
η	efficiency
ρ	density (kg/m^3)
ν	kinematic viscosity (m^2/s)
Subscripts	
1	coldhead
2	horizontal cooling plate
C	cooling
f	fin
H	heating
l	liquid
s	surface

segregated cellular flows and an augmentation of heat transfer between the copper plate and HTS magnets. Kang et al. [5] have developed the 6.6 kV/200 A inductive SFCL (superconducting fault current limiter) cooled by the natural convection of subcooled liquid nitrogen which is generated by the horizontal copper plate extended from a GM cryocooler.

Since the cold surface area of a GM cryocooler is very limited, the cooling plate is thermally anchored to the cryocooler in order to extend the heat transfer area in the cooling systems for HTS fault current limiters or HTS transformers mentioned above. The size of the cooling plate, however, is restricted by the shape of HTS magnets and the electrical insulation, especially for higher voltage applications. Therefore, it would be a wise design of cooling with only a horizontal cooling plate if the cooling performance satisfies the design criterion. From thermal point of view, the heat transfer rate is obviously greater when the surface area is larger. However, the fin efficiency of the extended surface decreases because the extended surface itself represents a conduction resistance to heat transfer from the original surface [6].

In this paper, the performances of the horizontal cooling plate or extended surface for subcooling liquid nitrogen by natural convection are investigated. For various sizes of the cooling plates extended from a GM cryocooler, the heat transfer coefficients are measured during the cool-down and in steady state and are compared with the existing correlations developed for the horizontal surface with a uniform temperature. In addition, the cooling performances of the horizontal plate are quantitatively discussed in terms of the fin efficiency of extended surface. This experimental data should contribute to the understanding the natural convection of subcooled liquid nitrogen under a horizontal cooling plate as well as the designing the cooling system for HTS power devices, such as HTS fault current limiters or HTS transformers.

2. Experimental apparatus and procedures

A schematic overview of natural convection experiment is shown in Fig. 1. The experiment is composed of a cryocooler, a horizontal cooling plate and a vertical heating plate immersed in liquid nitrogen. The cryocooler is mounted on the top plate of the cryostat whose inner diameter is 305 mm and height is 1040 mm. A rectangular shape of heating plate with 255 mm wide and 500 mm high is vertically placed at the center of the cryostat. A circular shape of cooling plate is horizontally located above the heating plate and thermally anchored to the coldhead of cryocooler by oxygen-free copper braids. The thickness of the cooling plate is 10 mm, and the diameter is a main variable in the range between 136 mm and 273 mm as described later.

A single-stage GM cryocooler (Cryomech Model AL63) provides the cooling to the experiment. Shown in Fig. 2 is the refrigeration capacity provided by the manufacturer [7] and measured by our own test [2]. The flexible tinned copper braids are used for a thermal connection between the coldhead and the horizontal cooling plate as well as a protection of the coldhead from thermal contraction. A flat Thermofoil™ heater is sandwiched between two identical stainless steel plates and cryogenic epoxy is applied to ensure good contact between the heater and the plates. The heater covers the entire surface and supplies a constant heat flux per unit area to the plates. The heating power is regulated with a DC power supply (HP-6253A, 0–30 V/0–3 A).

Temperature is measured with silicon diodes (Lakeshore DT-470-SD) at the coldhead of the GM cryocooler, the center of the cooling plate, and the lower end of heating

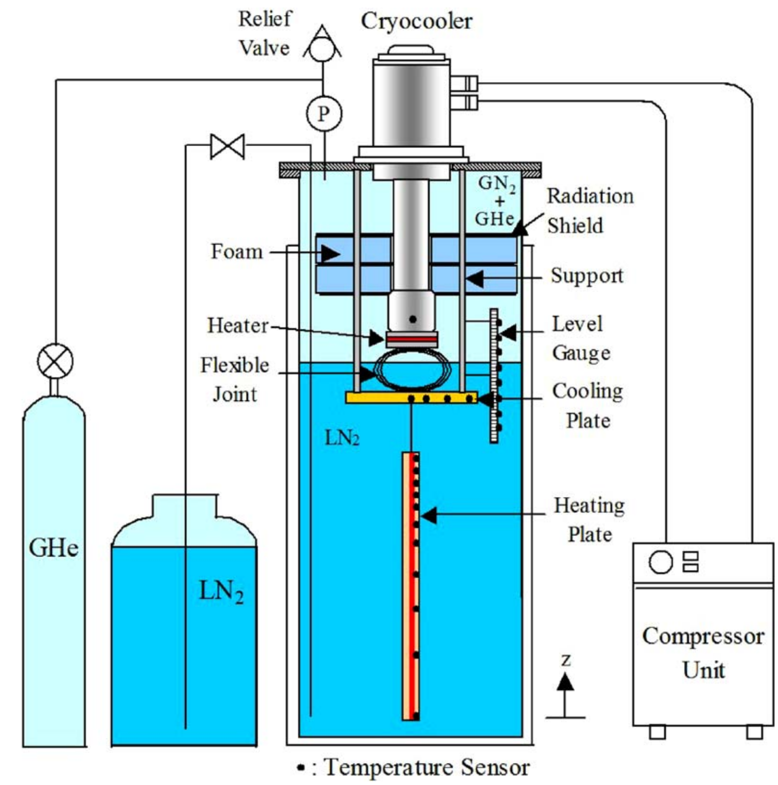

Fig. 1. Schematic of experimental apparatus.

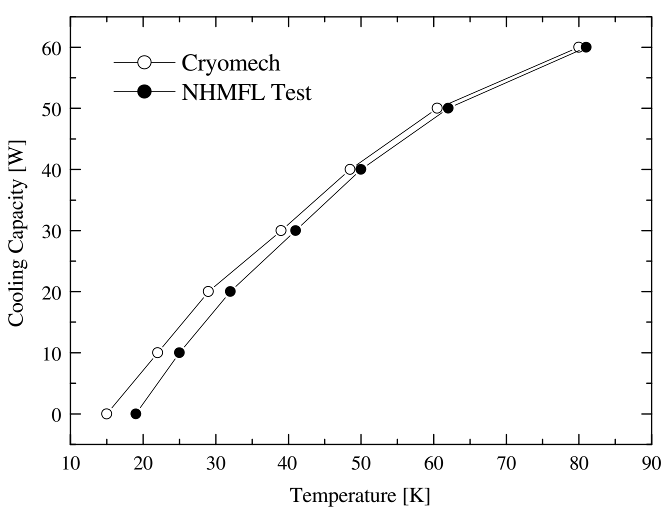

Fig. 2. Listed and measured cooling capacity of Cryomech AL 63.

plate. Additional surface temperatures are measured with E-type (chromel-constantan) thermocouples at many locations of the cooling and heating plates, while platinum resistance thermometers (Lakeshore PT103) are used for determining the absolute temperature of liquid and the top of the heating plate. The liquid temperature is averaged from the individual data of three thermometers, located approximately 300 mm below an edge of the cooling plate and equally spaced in horizontal direction, and five thermocouples located just below the free surface for level gauging as described below. The liquid level and the temperature distribution near the free surface are also measured with E-type thermocouples whose accuracy is ±30 mK for temperatures in the range of 63–77 K [2]. These thermocouples are used in many temperature measurements due to their advantages of short time-constant and no self-heating due to bias electric current [8]. A relief valve at 105 kPa is installed to maintain the inside pressure constant.

Another Thermofoil™ heater is installed at the coldhead of the GM cryocooler in order to avoid the freezing of liquid. The heating plate is attached to the coldhead with the brass screws and Cryocon™ grease is applied between coldhead and plate as a thermal contact medium, ensuring the maximum thermal conductance. The heating power is regulated with a DC power supply. The horizontal cooling plate is suspended at the top plate of cryostat with gravitational supports made of threaded GFRP rod.

A number of thermocouples are used to determine the spatial temperature distribution of the plates. The locations of all thermocouples on the plates are indicated in Fig. 1. The lead wires of the temperature sensors are connected through the holes from the opposites sides of the test section to minimize the disturbance of liquid flow. The holes housing for the thermocouple beads are located on the centerline on the plates. The sensing beads of thermocouple are dipped in cryogenic thermal grease in order to prevent reaction with the fluid.

At the initial phase of the experiment, the cryostat is filled with liquid nitrogen at 77 K and then cooled down to near its freezing temperature (63 K) by the extended heat exchange surface connected to the cryocooler. Once the cryostat is cooled down, a uniform heat flux is supplied to the heater and heating plate so that liquid nitrogen under the cooling plate experiences natural convection. The temperature distributions on the cooling and heating plate and liquid are measured in steady state, from which the heat transfer coefficients are calculated. Variables in this experiment are the size of cooling plate (whose surface area is 20%, 40%, 60% and 80% of total free surface area of liquid) and the magnitude of heat flux.

The time to reach steady state is determined by monitoring the plate and liquid temperature until the time rate of change is less than 0.06 K/h. Temperature is recorded every 2 min with a data acquisition system operated through LabView™ software. The heat flux supplied to the plates is measured by the electrical current and supplied voltage to the heater, which are monitored and checked from the multimeter (Keithley Model 2700) having an accuracy of 6.5 digits. The noise level of supplied heat flux in this measurement is on the order of 0.01 W. Once the system reaches a steady state, the temperatures on the cooling and heating plates are collected through the 40 channels scanner card (Differential Multiplexer Module 7708).

The fluid properties, thermal expansion coefficient (β), thermal conductivity (k), kinematic viscosity (υ), and density (ρ), are evaluated at the average temperature between the cooling and the heating plates. The reason why we have taken this average temperature can be explained in the following two ways. During the initial cool-down, the cooling plate is the only driving source of the natural convection because no power is supplied to the heating plate. As a result, the flow pattern is basically similar to the convection under a cooling surface in an open liquid, where the properties should be evaluated at the mean film temperature (that is, the average of the cooling plate temperature and the open liquid temperature) [18]. In this case, the liquid temperature is basically same as the heating plate. When the power is supplied to the heating plate, however, both the heating and cooling plates simultaneously drive the natural convection, and the flow pattern is rather similar to the convection in an enclosed space. For the convection in enclosures, the fluid properties can be more accurately evaluated at the average temperature of the heating and cooling plates [18].

3. Results and discussion

3.1. Size of cooling plate

Fig. 3 shows the temperature history of the coldhead, the cooling plate and liquid after turning on the cryocooler when the surface area of the cooling plate is 60% of free surface area of liquid. During the initial cool-down, the temperature decreased almost at a constant rate, and it took approximately 10 h for the coldhead to reach 63 K, the freezing temperature of nitrogen. At this point, the

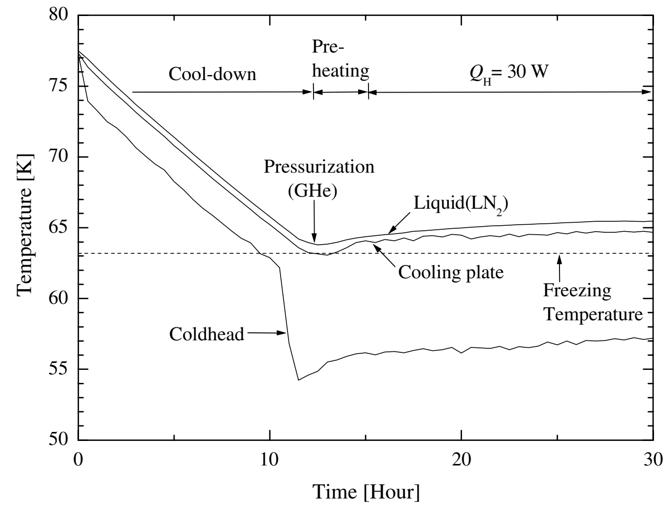

Fig. 3. Temperature history of coldhead, cooling plate and liquid when the area ratio of cooling plate is 60%.

coldhead temperature dropped quickly to 54 K, while the cooling plate in liquid nitrogen remained nearly constant at 63 K, as the liquid begins to freeze on the plate surface. Helium gas was then supplied to maintain atmospheric pressure so that the liquid is in the subcooled state. When the electric heater at the coldhead of cryocooler was on, the coldhead temperature increased gradually and the temperature difference between the cooling plate and liquid became measurable. The heating power was varied up to 30 W, and than set at 30 W until the cooling system achieved a steady state. The same procedure was repeated with the increased heating powers, 32 and 34 W.

During the cool-down period, the temperature decreases very slowly, at approximately 0.016 K/min, which could make the reasonable assumption of the quasi-steady state for the system. The effective heat transfer coefficient, h_{eff}, can be estimated from the net heat transfer rate during the cool-down.

$$Q_{\text{ref}} = mC\left(-\frac{\mathrm{d}T}{\mathrm{d}t}\right) \approx A \cdot h_{\text{eff}}(T_1 - T_{\text{s}}) \tag{1}$$

where m and C are the mass and the average specific heat of liquid nitrogen and the vessel, respectively. The effective heat transfer coefficient during cool-down is 308.1 W/m^2 K when the surface area ratio of the cooling plate is 60%. At the steady state, the effective heat transfer coefficients are estimated from Eq. (2).

$$Q_{\text{ref}} = Q_{\text{C}} - Q_{\text{H}} = A \cdot h_{\text{eff}}(T_1 - T_{\text{s}}) \tag{2}$$

In the experiment, the total power (Q_{ref}) at the horizontal plate is determined by subtracting the electric heating power to the coldhead (Q_{H}) from the cooling power of the cryocooler (Q_{C}) that is determined from the cooling capacity shown in Fig. 2. The h_{eff}'s are 298.4, 299.1 and 299.8 W/m^2 K when the heating powers to the coldhead are 30, 32 and 34 W, respectively. Comparing to the h_{eff}

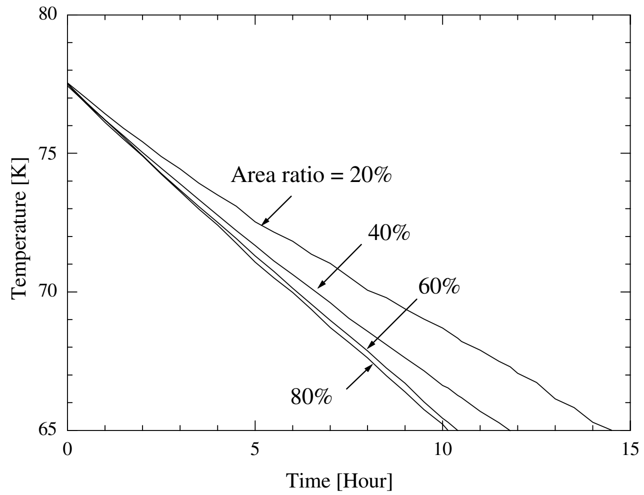

Fig. 4. Temperatures of liquid during cool-down for 20%, 40%, 60% and 80% area ratio of cooling plates.

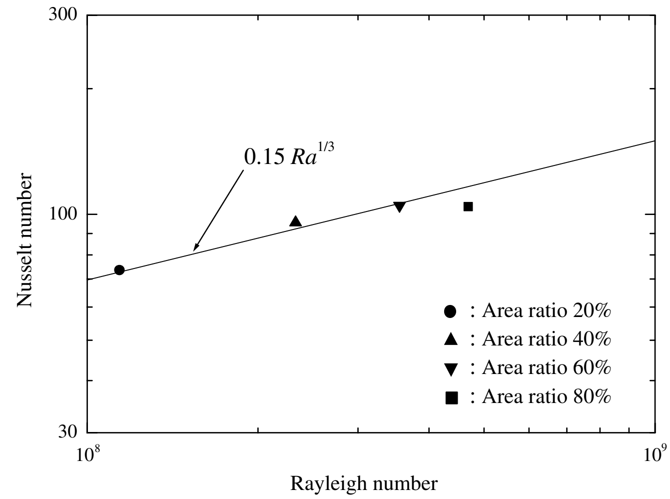

Fig. 5. Average Nusselt number versus Rayleigh number for 20%, 40%, 60% and 80% area ratio of cooling plates.

during cool-down, we note that the variation is within 3% and the effective heat transfer coefficient does not depend on the heating power to the coldhead.

The liquid temperature during cool-down for various sizes of the horizontal cooling plate is plotted in Fig. 4. The elapsed time of the temperature from 77 K to 65 K increases as the size of the cooling plate decreases. In other words, the larger cooling plate has a stiffer slope than the smaller one. This slope is related to the surface size and effective heat transfer coefficient, $A \cdot h_{\mathrm{eff}}$. In Eq. (1), the time rate of temperature change is proportional to the $A \cdot h_{\mathrm{eff}}$, if the amount of liquid and temperature difference between liquid and surface are same. The time rates of temperature change are 0.865, 1.044, 1.188 and 1.219 K/h when the area ratios of cooling plates are 20%, 40%, 60% and 80%, respectively. Even though the size of cooling plate increases 1.5 times from 40% to 60%, the elapsed time does not decreases 1.5 times since the efficiency of the extended surface decreases with increasing size, and the details will be discussed later.

The average Nusselt number (Nu) is estimated from Eq. (3).

$$Nu = \frac{h_{\mathrm{eff}} \cdot L}{k} = \frac{L}{k \cdot A} \cdot \frac{Q_{\mathrm{C}} - Q_{\mathrm{H}}}{T_{\mathrm{l}} - T_{\mathrm{s}}} \tag{3}$$

In Eq. (3), L, the characteristic length [6], is determined from the ratio of the plate surface area and perimeter which encompasses the area, $L = A_{\mathrm{s}}/P$. Fig. 5 shows the average Nusselt number as a function of average Rayleigh number (Ra) for various sizes of the cooling plate and compares it with the existing correlations which are developed for the natural convection under a horizontal cooling surface with a uniform temperature [9–16]. As the size of the cooling plate increases, Nu as well as Ra increases because Nu and Ra are defined based on the characteristic length. A fair agreement is observed between two data sets, when the size of the cooling plate is small. As the size of the cooling plate increases, the measured heat transfer becomes smaller than the existing correlations. This discrepancy can be explained by the fin efficiency of the extended surface. As the cold surface area of a GM cryocooler is very limited, a horizontal cooling plate is thermally anchored to the coldhead of a cryocooler in order to enhance the heat transfer rate between the coldhead and the adjoining liquid. The horizontal cooling plate, therefore, could be considered as a circular fin [17,18] and the effective heat transfer coefficient is expressed as

$$h_{\mathrm{eff}} = h\left(1 - (1 - \eta_{\mathrm{f}})\frac{A_{\mathrm{f}}}{A}\right) \tag{4}$$

where A and A_{f} denote the total surface area and the surface area of the fin, respectively. As the size of the extended surface increases, the effective heat transfer coefficient decreases because the fin efficiency is less than unity, $\eta_{\mathrm{f}} < 1$, and decreases with increasing the size of surface area. The measured effective heat transfer coefficients are 380.6, 345.6, 299.3 and 242.4 W/m^2 K when the area ratios of cooling plates are 20%, 40%, 60% and 80%, respectively.

The fin efficiency of the horizontal cooling plate for various plate sizes is plotted in Fig. 6. The solid lines in Fig. 6 are the fin efficiency of circular fins with constant thickness drawn after Ref. [6,17], which are based upon the corrected fin geometry with tip convection. The fin efficiency of the horizontal cooling plate is determined from the solution of an extended surface [6] utilizing the geometry of the cooling plate and the effective heat transfer coefficients measured in steady state. The fin efficiency of the cooling plate decreases with increasing the size of the cooling plate as we expected earlier. The fin efficiency of the cooling plate is 0.75, 0.50, 0.39 and 0.33 when the area ratios of cooling plates are 20%, 40%, 60% and 80%, respectively. Good agreement is observed between two data sets with the less than 1% variation.

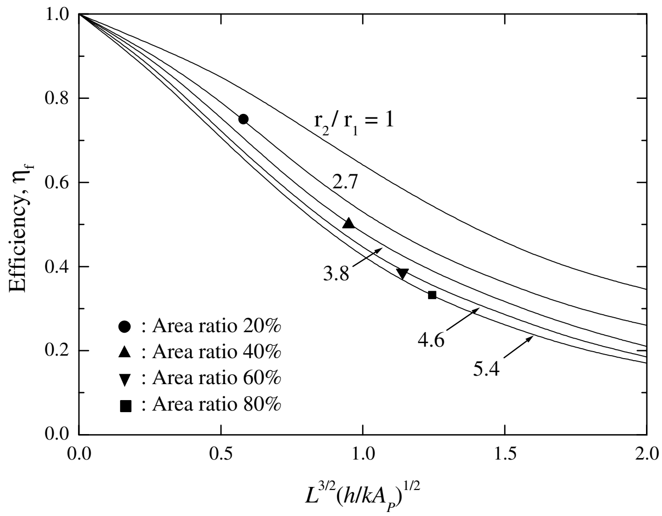

Fig. 6. Fin efficiency of the extended surface for 20%, 40%, 60% and 80% area ratio of cooling plates.

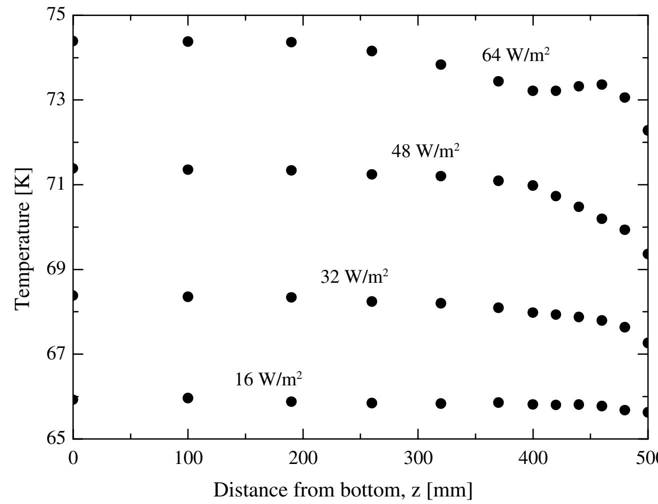

Fig. 8. Vertical temperature distributions of heating plate when the area ratio of cooling plate is 40% with a constant heating power to the coldhead.

3.2. Effect of heating power

Fig. 7 shows the measured average Nusselt numbers against Rayleigh numbers for various values of heat flux at the vertical heating plate and compares them with the existing engineering correlation for the natural convection under a horizontal cooling surface with a uniform temperature [6,15,16,18]. The area ratio of cooling plate is 40% and the electric heating power to the coldhead is 30 W. The Nu and Ra increase with heat flux because the temperature difference between the cooling plate and liquid increases. When the heat flux is smaller than 56 W/m^2 or corresponding Ra is less than 3.63×10^8, a good agreement is observed between the experiment and the correlation because the temperatures of cooling plate are relatively uniform in horizontal direction. As the heat flux increases over 56 W/m^2 or Ra exceeds 3.63×10^8, however, the measured heat transfer becomes considerably greater than the existing engineering correlation. This discrepancy can be explained by the thermal boundary conditions in the experiment where the surface temperatures of vertical plate decrease upward. Such a boundary condition can cause vertically segregated cellular flows and enhanced heat transfer under the horizontal cooling plate. Even though it is hard to tell the exact point of multi-cellular flow occurring, two cellular flow occurs when the heat flux is over 64 W/m^2 or Ra exceeds 3.76×10^8.

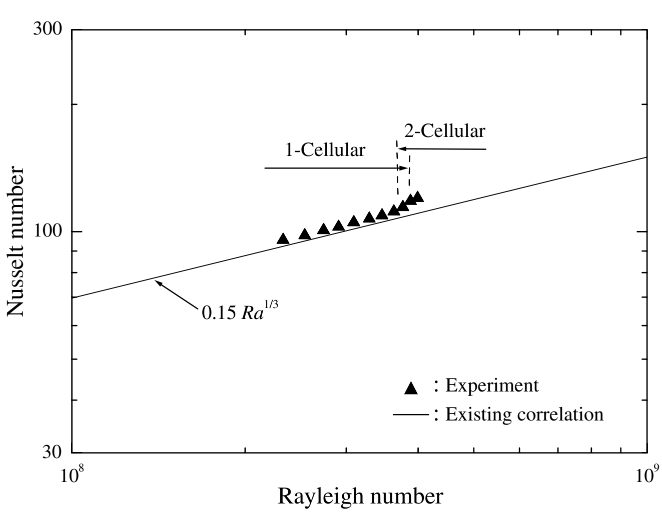

Fig. 7. Average Nusselt number versus Rayleigh number when the area ratio of cooling plate is 40% with a constant heating power to the coldhead.

Shown in Fig. 8 is the measured temperature distributions on the vertical heating plate when the heat flux is varied from 16–64 W/m^2. When the heat flux is small, the temperatures on the vertical heating plate are relatively uniform. As the heat flux increases, the temperature becomes higher all over the heating plate, and the temperature gradient becomes steeper near the top of the plate. When the heat flux is 64 W/m^2, the wavy temperature distribution that is derived from multi-cellular flow [2,19] is observed. The vertical heating plate is used to simulate the AC loss or heat dissipation in the power applications. As the AC loss or heat flux increases, the heat transfer under a horizontal cooling plate increases and the multi-cellular flow occurs at a certain value of heat flux, which results in an enhanced heat transfer and the effective heat transfer coefficient is much greater than the existing correlation as shown in Fig. 7.

4. Conclusions

Experiments were successfully performed to investigate the performance of a horizontal cooling plate for subcooling liquid nitrogen by natural convection. The effective heat transfer coefficient is in a good agreement with the existing engineering correlation when the size of cooling plate is small. At larger sizes of cooling plate, the effective heat transfer coefficient is clearly smaller than the existing correlation because of the decreased fin efficiency. Natural convection under a horizontal flat plate is accelerated with increasing the heat flux at vertical heating plate. When the heat flux is over $56\,\mathrm{W/m^2}$ or Ra exceeds 3.63×10^8, the multi-cellular flow starts to occur which result in the enhanced heat transfer under a horizontal flat plate with approximately 6–10% greater heat transfer coefficient than the existing engineering correlation.

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