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Two-Phase Electronic Cooling Using Small Diameter Tubes

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Abstract

Boiling heat transfer characteristics of Freon R-113 is experimentally investigated in small diameter vertical tubes, D =1.45, 2.8 mm and L =100 mm under a wide pressure range of 19 to 269 kPa and a natural circulation condition. Traces of wall temperatures at different elevations indicate that at all heat flux range the fluctuation of wall temperature is negligibly small on every system pressure within the present experimental range. Boiling curves indicate qualitatively similar to that of normal pool boiling. Except the entrance region of the test section the flow is annular in view of measured vapor flux. Effect of diameter on boiling heat transfer is almost consistent with the findings of Klimenko and Ishibashi-Nishikawa at system pressures below atmospheric but at pressures above atmospheric, the effect is negligible.

Introduction

Recently compact heat exchanger in various configurations have received attention for boiling heat transfer applications since heat transfer coefficient and rate are generally much larger than those characteristic of convection heat transfer without phase change. Confined device cooling incorporates several small and short coolant passages similar to those described in this paper.

Refrigerants may be the preferred coolant since fluorocarbons possess high dielectric strength, boil at moderate pressures with low saturation temperatures, and compatible with materials commonly used in electrical and electronic devices. Flow boiling of fluorocarbon refrigerants has been investigated in small channels by many researchers. Wambsganss *et al.*[10] investigated the flow boiling of R-113 in a small diameter horizontal tube of 2.92 mm, which diameter corresponds to the diameter to Laplace constant ratio of 2.95 to 3.02, at a pressure range of 124 to 160 kPa with a mass flux of 50 to 300 kg/m^{2}s. They reported that the majority of their boiling data has the plug and slug pattern flow regimes, the heat transfer trends show a dominance of the nucleation mechanism, and the pool boiling correlation of Stephan-Abdelsalam[8] predicts well the measured results. Bowers-Mudawar[1] investigated flow boiling of R-113 in small diameter tubes of 0.51 and 2.54 mm, which diameters correspond to the diameter to Laplace constant ratio of 0.52 and 2.6, at a pressure of 138 kPa with a mass flux of 105 to 458 kg/m^{2}s per one channel of 2.54 and 0.51 mm, respectively. They concluded that the heat transfer regime in their experimental range was the same as the ordinary pool boiling and the tube diameter effect on boiling heat transfer was negligible. Yao *et al.*[11] conducted boiling experiment using R-113 with three different gaps 0.32, 0.80, and 2.58 mm at atmospheric pressure in an annular channel. They reported that in the thermal hydraulic diameter less than 4.0 times of the Laplace constant, the isolated deformed bubble regime occurs at low heat flux and the coalesced deformed bubble regime is observed at high heat flux. They also concluded that the effect of space confinement to boiling heat transfer increases with decreasing gap size. Ishibashi-Nishikawa[2] presented the effect of fluid space on the boiling heat transfer using various annular working fluid spaces, ΔR of 0.57 to 83.5 mm for atmospheric and above atmospheric conditions and reported that the boiling heat transfer depends on the boiling space, ΔR as $h \;\alpha\; \Delta R^{-2/3}$. Since they used an annular boiling space in which only inner tube was heated and the outer tube was adiabatic, the heat transfer characteristics and flow behavior of the working fluid in the space surrounded by the heated and unheated surfaces might be different from that in a single tube. Klimenko *et al.*[3] investigated the boiling heat transfer for a two-phase flow of nitrogen at the mass flux of 60 to 800 kg/m^{2}s in stainless steel tubes with diameters of 4.0 to 18.9 mm and at above atmospheric pressure in the range of 150 to 800 kPa. They reported that in the channel diameter less than 1.5 times of the Laplace constant, the deformation of vapor bubble results in an increase of boiling heat transfer. Klimenko[4] also concluded that in small diameter tubes the boiling heat transfer depends as $h \propto (D/B)^{-2/3}$. However, it is not clear that this finding can be applied to ordinary fluids or fluorocarbon refrigerants since the properties of ordinary fluids and refrigerants are much different from those of the cryogenics.

The findings mentioned above indicate that for the enhancement of boiling heat transfer the flow regime and behavior in a boiling passage should be taken into account besides the geometry parameter such as the ratio of tube diameter to Laplace constant. When R-113 is used as a working fluid in a thermosyphon or saturated cooling for electrical and electronic devices under a natural circulation condition, for very compact devices the heat transfer space should be as small as possible. Actually so far no comprehensive investigation has been performed on boiling heat transfer in a small diameter tube under a natural circulation condition. The purpose of our experimental investigation is to examine the boiling heat transfer regime in a small diameter tube below and above atmospheric pressures under natural circulation condition and to clarify the enhancement phenomenon by comparing the obtained results with established correlations.

Experimental apparatus and test procedure

A schematic diagram of the experimental apparatus is shown in Fig. 1 and dimensions of the section and the locations of thermocouples are presented in Fig. 2. The test section(1) is made of stainless steel tubes with inner diameter of 1.45 and 2.8 mm and a length of 100 mm. The condenser section is made by a

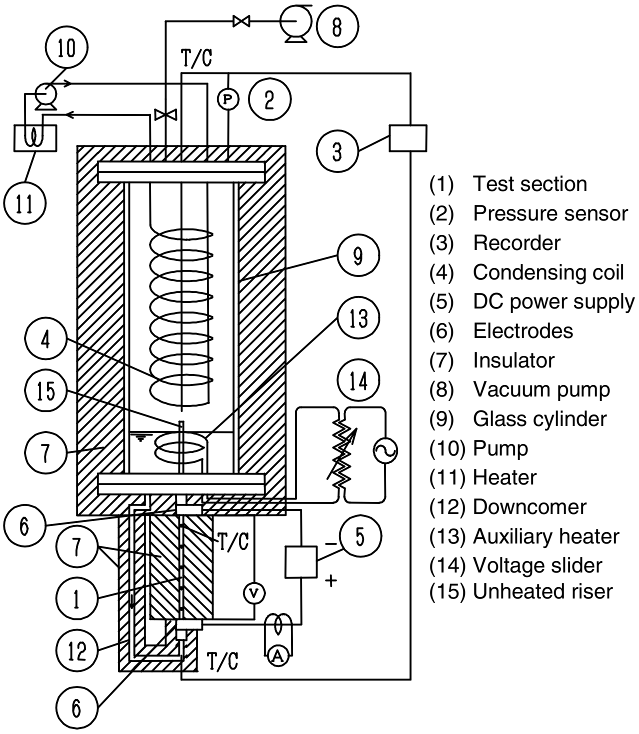


Fig. 1 Schematic diagram of experimental apparatus

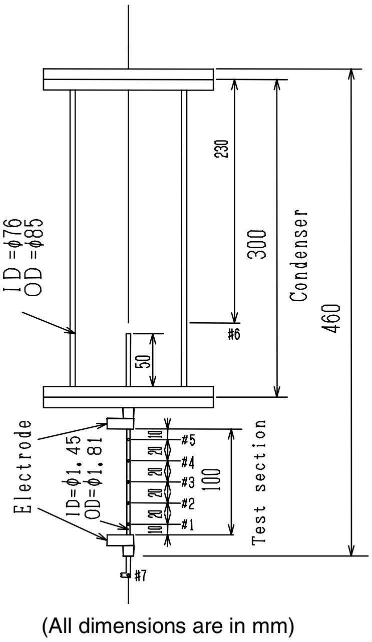


Fig. 2 Dimensions of the test section and locations of the thermocouples

glass cylinder(9) and a cooling coil(4) in which a water coolant with a constant temperature is circulated. The wall temperature of heating section is measured at five locations with 0.1 mm diameter chromel-alumel thermocouples placed on the outer surface of the tube. Two chromel-alumel sheathed thermocouples are inserted into the condenser section and the inlet of test section to measure the vapor temperature and the inlet liquid temperature, respectively. The system pressure is calculated from the saturated condition corresponding to the vapor temperature measured in the condensing section. An auxiliary heater(13) is installed at the bottom of condenser section to enable experimental condition to be saturated since the sub-cooling of inlet liquid gives a large hydraulic fluctuation like a geysering in the natural circulation condition. The condenser and the heated sections are connected with an unheated riser(15) of 50 mm long tube to prevent a back flow from the condenser directly to the test section. The downcomer (12) is a low resistance flow path made of a 8 mm inner diameter tube. The test section is heated directly with a manually controlled DC power supply(5). Four hundred data per channel are scanned through a data acquisition system with a scan speed of 0.4 second per one datum. The experiments are conducted under natural circulation condition, and the experimental parameters and their ranges are indicated in Table 1. The experimental heat flux range covers up to near critical heat flux condition. Uncertainties in the measurement of system pressure and heat fluxes are determined as 1 and 2 percent, respectively.

D (mm)	q (kW/m^2)	P_s (kPa)	B (mm)	D/B
1.45	8.34~28.05	19	1.12	1.29
	13.58~41.0	43	1.07	1.35
	16.91~43.89	68~70	1.04	1.39
	26.67~50.08	93~94	1.01	1.45
	22.62~53.03	142~144	0.97	1.49
	31.66~59.70	204~206	0.94	1.54
	34.36~69.39	268~269	0.90	1.61
2.8	18.89~46.70	25~27	1.12	2.50
	11.51~51.0	41~42	1.07	2.61
	19.0~60.90	64	1.04	2.69
	19.09~76.93	92~93	1.01	2.77
	25.23~89.61	145~146	0.97	2.89
	40.12~103.0	205~207	0.94	2.98
	58.26~117.15	262~267	0.91	3.07

Table 1 Experimental parameters and their ranges

Results and discussion

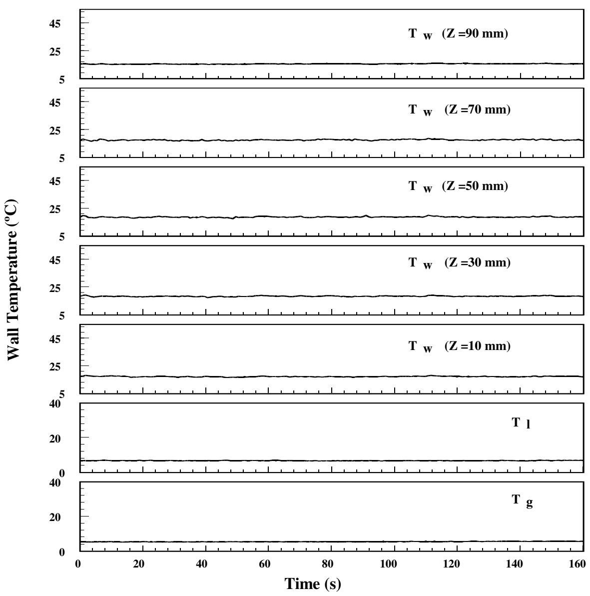
Temperature traces and boiling curve

Figure 3 shows time traces of measured temperatures at five elevations of the heated wall, inlet liquid and vapor for Freon R-113 in the 1.45 mm tube at a pressure of 19 kPa and low heat flux condition. They show steady state traces without any fluctuations. At other conditions, temperature traces are almost similar to those in this figure.

Boiling curves in both tubes are shown in Fig. 4. The represented wall temperature is calculated by simple averaging the measured ones with time since the fluctuations are negligible. Curves are qualitatively similar to that of the normal pool boiling, except the data at higher heat flux region in the 1.45 mm tube, which are very close to be critical.

Flow regime

The heated section in the present study is long enough compared with diameter and the flow pattern must change along the heated length. The flow pattern must be evaluated since the pattern



D =1.45 mm, q =8.34 kW/m^2, P_s =19 kPa

Fig. 3 Traces of heated wall, inlet liquid, and vapor temperatures for Feron R-113

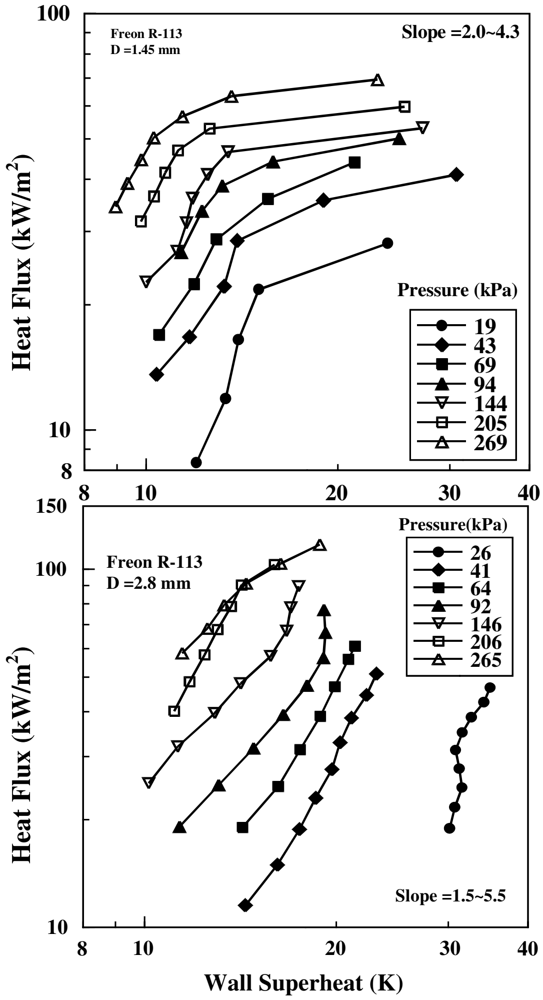


Fig. 4 Boiling Curves

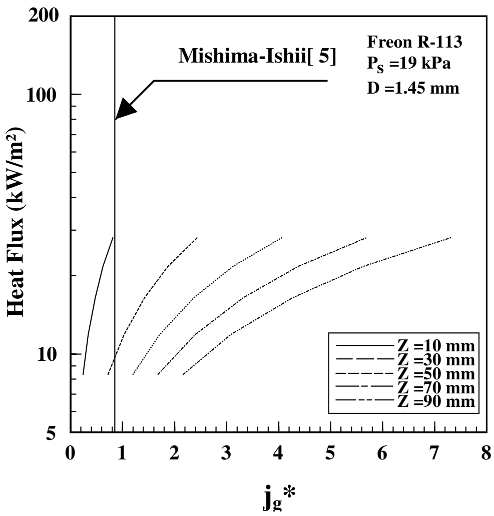


Fig. 5 Relationship between heat flux and dimensionless vapor flux

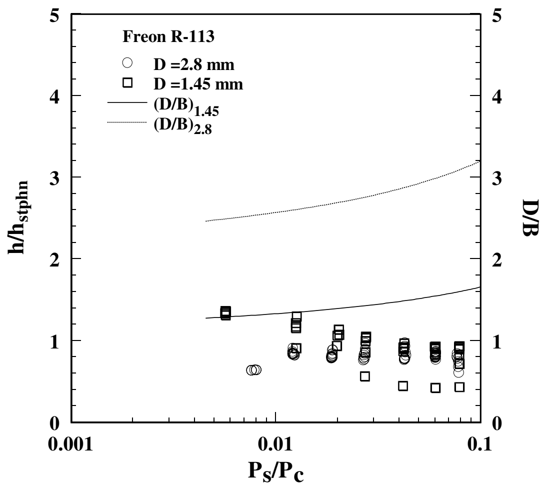


Fig. 6 Enhancement of boiling heat transfer in small diameter tube

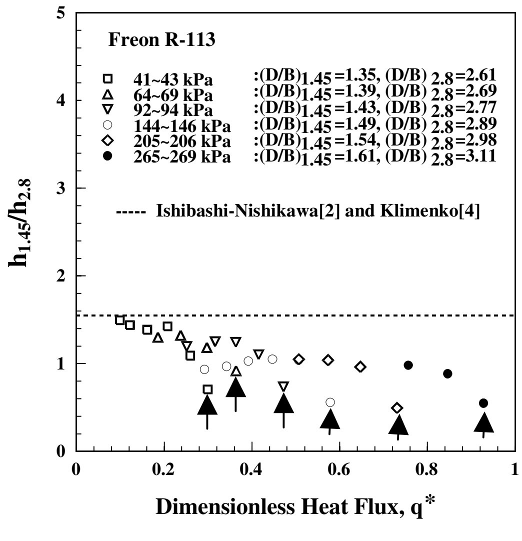


Fig. 7 Effect of diameter on boiling heat transfer

might affect the heat transfer. Figure 5 shows the relationship between heat flux and a dimensionless vapor flux at five different elevations in the 1.45 mm tube at a pressure of 19 kPa. Wallis[9] proposed that for the annular flow regime the dimensionless vapor flux is required to be greater than 0.9. Mishima-Ishii[5] also developed the following dimensionless vapor flux for the transition between churn-turbulent flow and annular flow regimes:

$$j_g^* = \sqrt{\alpha - 0.11} \tag{1}$$

Within our experimental range the void fraction, α is predicted to be 0.72 using the following correlation proposed by Mishima-Hibika[6] developed in a small diameter tubes(D =1.0 to 4.9 mm) for an air-water system:

$$\alpha = 1/\left[1.2 + 0.51\exp(-691D)\right] \tag{2}$$

Finally, the value of the dimensionless flux for the transition is calculated to be 0.78 by Eq. (1) as shown by a vertical line in Fig. 5. This value is very close to be 0.9 proposed by Wallis mentioned above. Comparing the experimental vapor flux with the predicted one, the flow regime might be annular within our experimental range except the entrance region below Z =10 mm of the test section. At other conditions in both tubes similar results are obtained. Therefore, nucleate boiling in a liquid film or film evaporation without any boiling might be a predominant

Enhancement of heat transfer

The rate of enhancement of boiling heat transfer is presented in Fig. 6 through comparison with pool boiling correlation of Stephan-Abdelsalam[8] which predicts well the measured boiling heat transfer coefficients of R-113 in 2.92 mm diameter horizontal tube of Wambsganss *et al.*[10]. The figure indicates that a little enhancement of boiling heat transfer is obtained in the 1.45 mm tube up to system pressure below atmospheric. According to Klimenko *et al.*[3] the enhancement is expected if D/B is less than 1.5. Even within the range of the ratio no enhancement is observed.

Effect of diameter on boiling heat transfer

The tube diameter effect on boiling heat transfer is presented in Fig. 7. The ordinate shows the ratio of the measured heat transfer coefficients in 1.45 to 2.8 mm diameter tubes. Dimensionless heat flux is considered as the abscissa including Laplace constant, which was considered by Rohsenow[7] as a bubble Reynolds number. The previous findings about the effect of diameter on boiling heat transfer of Ishibashi-Nishikawa[2] and Klimenko[4] become similar if the boiling space in an annular tube is considered to be equal to the diameter in a circular tube. Since they proposed that heat transfer coefficient is proposed that heat transfer coefficient is proportional to $D^{-2/3}$, then the ratio of heat transfer coefficient in 1.45 to 2.8 mm tubes is a constant value of 1.55, independent of pressure and working fluids. The figure indicates that effect of diameter on boiling heat transfer is almost consistent with the findings of Klimenko and Ishibashi-Nishikawa at system pressures below atmospheric but at pressures above atmospheric, the effect is negligible. However, in the 1.45 mm tube, the measured heat transfer coefficients at a higher heat flux condition shown by the arrow indicate a little lower than those in 2.8 mm tube due to lower critical heat flux than that for the 2.8 mm tube.

Conclusions

Experimental study for boiling heat transfer of Freon R-113 in small diameter tubes under below and above atmospheric pressures and natural circulation conditions give the following results:

[1] Boiling curves in both tubes are qualitatively similar to that of the normal pool boiling.
[2] Considering vapor flux within the present experimental range, the flow regime is annular above Z =10 mm of the test section.
[3] No enhancement of boiling heat transfer is obtained which contradicts the finding of Klimenko *et al.*[3].
[4] Effect of tube diameter on boiling heat transfer is consistent with the findings of Ishibashi-Nishikawa[2] and Klimenko[4] except near critical heat flux data.

Nomenclature

B : Laplace constant, $\sqrt{\sigma / g(\rho_l - \rho_g)}$, m
D : inside diameter of tube, m
g : gravitational acceleration, m/s^2
h : heat transfer coefficient, W/m^{2}K
h_{fg} : latent heat of vaporization, J/kg
j_g : superficial vapor velocity, $4qZ/(Dh_{fg}\rho_g)$, m/s
j_g^* : dimensionless vapor flux, $j_g\sqrt{\rho_g} / \sqrt{gD(\rho_l - \rho_g)}$
L : heated length, m
P : pressure, kPa
q : heat flux, W/m^2
q^* : dimensionless heat flux, $qB/(h_{fg}\mu_l)$
T : Temperature, K
Z : axial distance from bottom of the heated tube, m
α : void fraction
ρ : density, kg/m^3
μ : dynamic viscosity of liquid, Pa•s
σ : surface tension, N/m

Subscripts

c : critical condition
g : vapor of working fluid
l : liquid
s : saturation condition
$stphn$: stephan
w : wall

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