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ABSTRACT

This study experimentally investigated the flow boiling heat transfer and pressure drop characteristics of refrigerant R32 in a horizontal small-diameter microfin tube that had an outer diameter of 3.0 mm and an equivalent diameter of 2.6 mm. The test microfin tube had a helix angle of 10°, a fin height of 0.1 mm, and 25 fins. The flow boiling heat transfer coefficient and the pressure drop were measured against a mass velocities range of 100–400 kg/(m^{2}s), a heat flux range of 5–20 kW/m^2, and a saturation temperature of 15°C. The heat transfer coefficient was found to increase with an increase in the heat flux and the vapor quality in the pre-dryout region. The effect of tube diameter on heat transfer in small-diameter microfin tubes – tubes with equivalent diameters of 2.6 mm and 3.7 mm – that had the same number and height of fins, is significant in the forced convection and thin liquid film evaporation heat transfer dominant regions compared to the nucleate boiling dominant region.

1. INTRODUCTION

Compact, high-performance heat exchangers used in air-conditioning and electric cooling systems show improved performance and need smaller amounts of refrigerant charge when they are developed using small diameter tubes with outer diameters less than 5 mm. Several studies have sought to improve the performance of heat exchangers by investigating heat transfer and pressure drop during boiling and condensation flow in microfin tubes. Microfin tubes of smaller diameter are found to improve the performance of heat exchangers as well as reduce their size. The boiling heat transfer and flow characteristics of conventional large diameter tubes are different from those of small-diameter microfin tubes, because the effect of surface tension and shear stress on those parameters increases with decreasing tube size. Although refrigerant R32 has been commercially available in recent years for use in residential and industrial air-conditioning systems, only limited research is available on the boiling heat transfer and pressure drop characteristics of R32 in microfin tubes. Research into microfin tubes with outer diameters smaller than 4mm is especially meager when compared to the data available on conventional large-diameter microfin tubes.

In the recent past, Dang *et al.* (2010) investigated the flow boiling heat transfer of CO_2 inside a small-diameter microfin tube with a mean inner diameter of 2.0 mm, using a mass velocity range of 360–720 kg/(m^{2}s). Baba *et al.* (2012) experimentally investigated the flow boiling characteristics of R32 and other refrigerants in a horizontal microfin tube with a mean inner diameter of 5.2 mm. Diani *et al.* (2014) performed experiments with refrigerant R1234ze(E) using small-diameter microfin tubes that had a 3.4 mm fin-tip diameter and a 4 mm outer diameter, adjusting mass velocity to be in the range 190–940kg/(m^{2}s), at a saturation temperature of 30°C. They have proposed two prediction correlations for heat transfer and pressure drop in microfin tubes. Diani *et al.* (2015) also experimented with R1234yf inside a microfin tube that had a 3.4 mm fin-tip diameter and reported their findings.

This paper is concerned with the flow boiling heat transfer and pressure drop characteristics of refrigerant R32 in a horizontal, small-diameter microfin tube, with an outer diameter of 3.0 mm and an equivalent diameter of 2.6 mm. The experiments were carried out with mass velocity values in the range 100–400 kg/(m^2s), heat flux in the range 5–20 kW/m^2, and a saturation temperature of 15°C. The effects of vapor quality, mass velocity and heat flux are clarified. The measured inside a 2.6 mm microfin tube were compared with those inside a 3.7 mm microfin tube of the same fin height and fin number.

2. EXPERIMENTAL APPARATUS AND METHOD

2.1 Experimental Apparatus

Figure 1 shows a schematic diagram of the experimental apparatus used in this study. The test apparatus consists of a magnetic gear pump, a double-tube heat exchanger, an electric pre-heater, a test section, a condenser, a liquid receiver, and a sub-cooler. Liquid refrigerant discharged from the gear pump flows into the water heat exchanger and electric heater. The electric heater heats the test refrigerant to obtain the desired vapor quality at the inlet to the test section. The liquid refrigerant returns to the gear pump through the condenser, the liquid receiver, and the sub-cooler. The refrigerant mass flow rate is measured by the Coriolis mass flow meter with an accuracy of ±0.5%. The refrigerant flow rate is controlled by the flow regulating valves in the main and bypass loops.

Figure 2 shows a detail of the test section. AC current heats the test tube. The tube wall temperature is measured by T-type thermocouples with a measuring accuracy of ±0.05 K. These thermocouples are buried in the test tube wall separated by 50 mm intervals. The absolute pressure transducer measures the refrigerant pressure at the inlet of the measuring section with an accuracy of ±1.4 kPa. The pressure drop between the inlet and the outlet of the measuring section is measured using a differential pressure transducer with an accuracy of ±0.2 kPa. The measuring lengths of the heat transfer and the pressure drop are 400 and 550 mm respectively. The test section is encased with insulation and placed inside a box in which the air temperature is controlled to match the evaporation temperature of the test refrigerant in the test section to reduce the heat gained by the test section from the surrounding air.

The test tube is a small-diameter microfin tube with an outer diameter of 3.0 mm. The parameters of the microfin test tube are as follows: number of fins - 25, helix angle - 10°, fin height - 0.1 mm. The equivalent inner diameter of the microfin tube is 2.6 mm, where equivalent inner diameter means the inner diameter of a smooth tube with the same internal free flow area as this tube. The enlargement of the heat transfer area, which is the ratio of the actual heat transfer area to the surface area of a smooth tube with the same equivalent inner diameter, is 1.4.

The experiments were performed using R32 as the test refrigerant at a saturation temperature of 15 °C at the inlet to the test section, a mass velocity in the range 100–400 kg/(m^2s) and a heat flux in the range 5–20 kW/m^2.

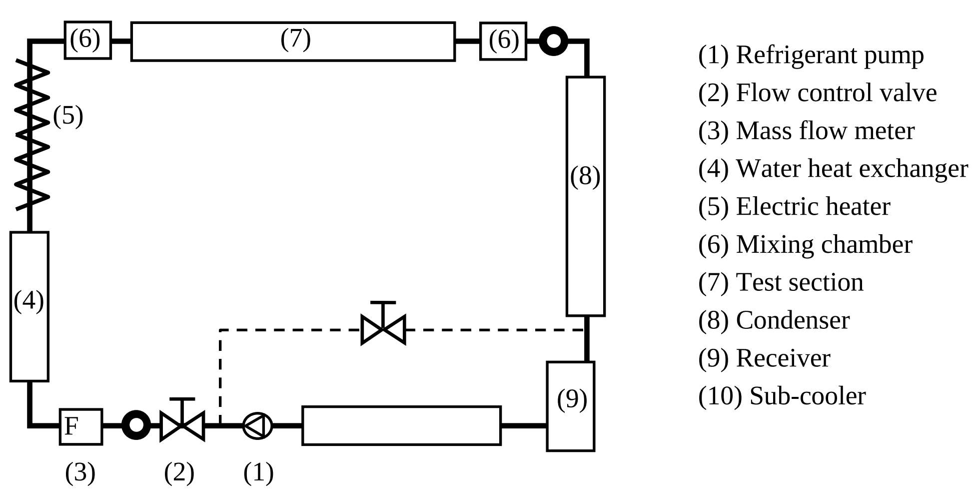


Figure 1: Schematic diagram of the experimental apparatus

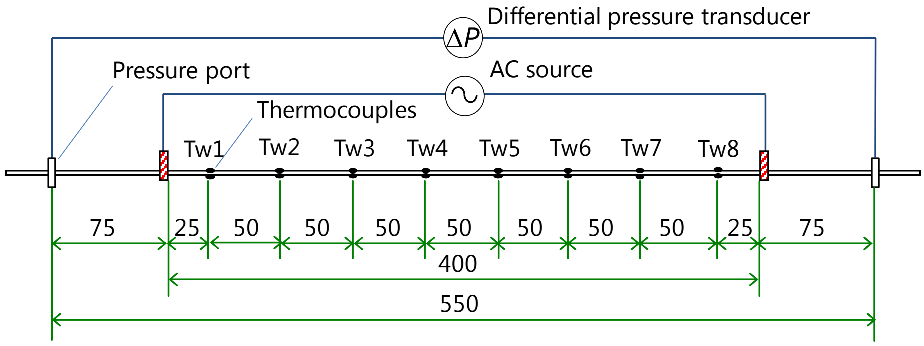


Figure 2: Detail of the test section

Furthermore, the experiment includes results on heat transfer coefficient and pressure drop for refrigerant R32 in microfin tubes with equivalent diameters of 2.6 mm and 3.7 mm, and the same fin height and fin numbers.

2.2 Data Reduction Method

The specific enthalpy at the inlet of the test section was calculated using the bulk specific enthalpy at the inlet of the electric pre-heater, the heat transfer rate, and the heat loss in the pre-heater. The bulk specific enthalpy at the inlet of the test section was calculated using the following heat balance equation:

$$h_{\mathrm{TS,in}} = h_{\mathrm{E,in}} + Q_{\mathrm{E}} / m \tag{1}$$

where $h_{\mathrm{TS,in}}$ and $h_{\mathrm{E,in}}$ are respectively the bulk specific enthalpies at the inlets of the test section and the electric-preheater, Q_{E} is the heat transfer rate in the electric pre-heater, and m is the flow rate of the test refrigerant. The distribution of the vapor quality in the test section was determined using the bulk specific enthalpy calculated from the input heat transfer rate in the heating section. The heat flux q was calculated by the following equation:

$$q = Q_{\mathrm{TS}} / A_{\mathrm{H}} = Q_{\mathrm{TS}} / (\eta \pi d_{\mathrm{eq}} L) \tag{2}$$

where Q_{TS} is the heat transfer rate calculated based on the input AC power in the heating section, while A_{H}, d_{eq}, and L are the actual heat transfer area, equivalent inner diameter, and effective heating length of the test microfin tube. η is the surface enlargement ratio calculated from the actual and nominal heat transfer areas. The heat transfer coefficient α is given by the following equation:

$$\alpha = q / (T_{\mathrm{w}} - T_{\mathrm{R}}) \tag{3}$$

where T_{R} is the saturation temperature of the refrigerant and T_{w} is the temperature of the inner tube wall. The tube wall temperature is the average temperature of the top and bottom sections of the test tube. The ratio of the heat transfer rate between the test tube and its surroundings to the rate of heat input was found to be less than 8% and 2% with the heat flux being 5 kW/m^2 and 20 kW/m^2, respectively. The inner tube temperature was considered equal to the outer tube wall temperature. The uncertainty of the boiling heat transfer coefficient was estimated to be within ±10% for most of the data, and the maximum error was evaluated to be 30% under these conditions – a heat flux of 5 kW/m^2, a mass velocity of 200 kg/(m^{2}s), and the presence of a high vapor quality region.

The frictional pressure drop ΔP_{F} was calculated by the following equation:

$$\Delta P_{\mathrm{F}} = \Delta P_{\mathrm{mes}} - \Delta P_{\mathrm{A}} \tag{4}$$

where ΔP_{mes} is the measured pressure drop between the inlet and the outlet of the test tube, and ΔP_{A} is the acceleration pressure drop associated with the quality change in the measuring section. The acceleration pressure drop is calculated by the following equation:

$$\Delta P_{\text{A}} = \Delta\left[\frac{G^2 x^2}{\xi \rho_{\text{V}}} + \frac{G^2 (1-x)^2}{(1-\xi)\rho_{\text{L}}}\right] \tag{5}$$

where x is vapor quality, and ξ is the void fraction. The void fraction was estimated using the equation from Kondou *et al.* (2008). The properties of the test refrigerant were calculated using NIST Refprop (Lemmon *et al.*, 2010).

3. EXPERIMENTAL RESULTS AND DISCUSSION

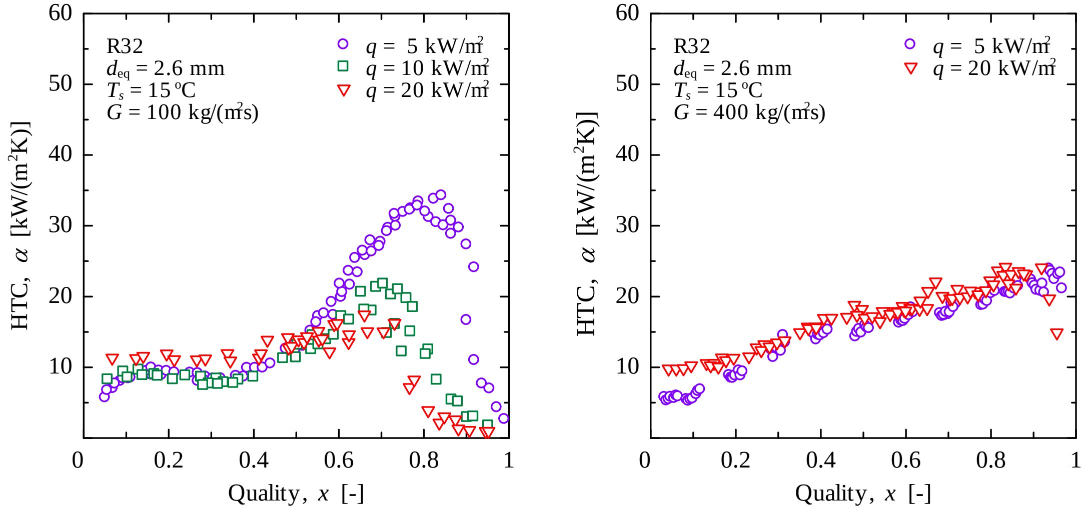
3.1 Flow Boiling Heat Transfer

Figures 3 (a) and 3 (b) show the measured heat transfer coefficient in the microfin tube with an equivalent diameter of 2.6 mm for mass velocity values of 100 and 400 kg/(m^{2}s) at a saturation temperature of 15°C. The horizontal and vertical axes, respectively, show the vapor quality x and the heat transfer coefficient α . In case of a mass velocity of 100 kg/(m^{2}s), the heat transfer coefficient increases with increasing heat flux due to an increase in nucleate boiling when vapor quality x<0.5. The heat transfer coefficient dramatically increases with an increase in vapor quality, so that at a vapor quality of 0.4, the heat flux of 5 kW/m^2. The enhanced heat transfer occurs through an extremely thin liquid film that forms at the fins. This liquid film meniscus is formed at the top of the tube due to the capillary force and shear stress. In contrast to the previous condition, the heat transfer coefficient decreases with increasing heat flux, for a vapor quality greater than 0.5. The liquid thin film formed at the fin-tips dries under conditions of high heat flux and high vapor quality. Furthermore, with an increase in vapor quality the heat transfer coefficient rapidly decreases due to the dryout, as it happens in smooth tubes. It was noted that the quality of the dryout decreases with increase in heat flux at G = 100 kg/(m^{2}s).

In the case of a mass velocity of 400 kg/(m^{2}s), the heat transfer is enhanced by increasing heat flux as well as that of a mass velocity of 100 kg/(m^{2}s) under a vapor quality of 0.3 due to the enhancement of nucleate boiling heat transfer. Nucleate boiling heat transfer is dominant for high heat fluxes and for regions of low vapor quality. However, the difference in heat flux is small when compared to a mass velocity of 100 kg/(m^{2}s) for vapor quality in the range x>0.3. Forced convection heat transfer is dominant in a region with high mass velocity and high vapor quality. Therefore, the liquid film that is formed covering the inner circumference of the tube becomes uniformly thick due to the effect of vapor shear stress. No dryout is observed at a heat flux of 5 kW/m^2 when the mass velocity is 400 kg/(m^{2}s).

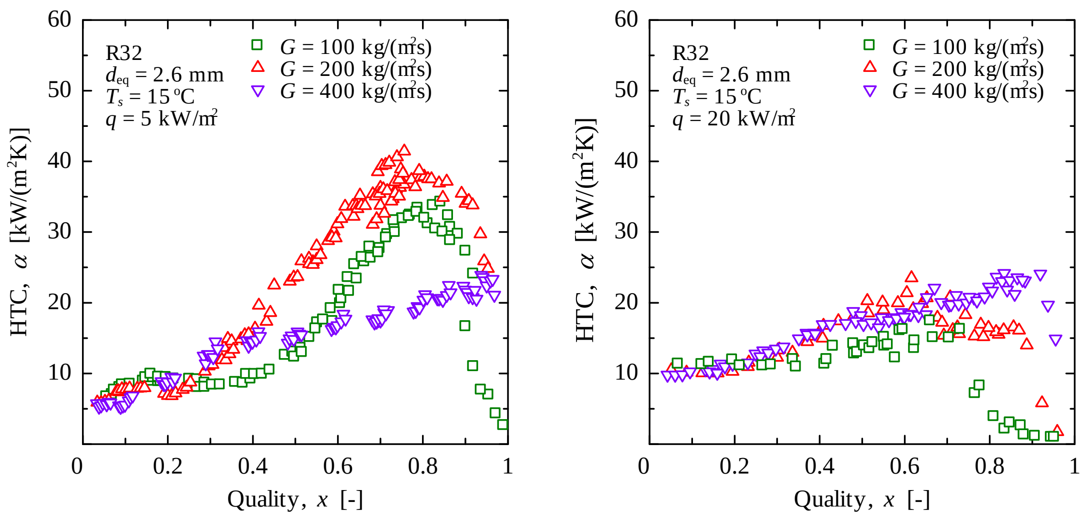
Figures 4 (a) and 4 (b) show the measured heat transfer coefficients in the microfin tube for heat flux values of 5 kW/m^2 and 20 kW/m^2, respectively, at the saturation temperature of 15 °C. When q = 5 kW/m^2, the effect of the mass velocity on heat transfer is small for a vapor quality that is in the range x<0.3. The heat transfer coefficient increases with increasing vapor quality in the pre-dryout region, especially for mass velocity values of 100 kg/(m^{2}s) and 200 kg/(m^{2}s). The flow pattern changes to separated flow with the thin liquid film at the top of the tube to enhance heat transfer by liquid surface tension and vapor shear stress. For further increase in vapor quality, the heat transfer coefficient rapidly decreases at over dryout vapor quality. The heat transfer coefficient decreases with a further increase in mass velocity, because the circumferential liquid film thickness becomes uniform. Finally, the heat transfer coefficient achieves its highest value at G = 200 kg/(m^{2}s) in the dominant region of the evaporation heat transfer through the thin liquid film formed at the fins.

The effect of mass velocity on heat transfer is smaller for a heat flux of 20 kW/m^2 when compared to 5 kW/m^2. The heat transfer coefficient for mass velocity G = 100 kg/(m^{2}s) is lower than the heat transfer coefficient of the mass velocity range G = 200 kg/(m^{2}s)-400 kg/(m^{2}s), because the flow pattern is different in the two cases. Figures 5 (a) and 5 (b) show the measured heat transfer coefficients at the top and bottom of the tube for mass velocity values 100 kg/(m^{2}s) and 400 kg/(m^{2}s). For mass velocity G = 400 kg/(m^{2}s), there is no difference in the heat transfer coefficients for the top and bottom of the tube. It can be noted that the flow pattern for mass velocity G = 400 kg/(m^{2}s) is annular flow and the film has uniform thickness. On the other hand, the heat transfer coefficient at the



(a) G = 100 kg/(m^{2}s) (b) G = 400 kg/(m^{2}s)

Figure 3: Effect of heat flux on boiling heat transfer coefficient.



(a) q = 5 kW/m^2 (b) q = 20 kW/m^2

Figure 4: Effect of mass velocity on boiling heat transfer coefficient.

top of the tube is higher than it is at the bottom, for mass velocity G = 100 kg/(m^{2}s). In a low vapor quality region, the heat transfer coefficient at the top of the tube is as high as 2.5 times its value at the bottom. This distribution of the heat transfer coefficient suggests that the flow pattern is separated flow with the thin liquid film at the top of the tube for mass velocity G = 100 kg/(m^{2}s) and a vapor quality in the range x<0.8. However, the heat transfer coefficient rapidly decreases at over dryout vapor quality as about x = 0.8.

Figure 6 shows a comparison between the measured heat transfer coefficient in this study and the data for a microfin tube with an outer diameter of 4.0 mm and equivalent diameter of 3.7 mm, as reported by Inoue *et al.* (2016). The microfin tube in the Inoue *et al.* (2016) study has the same number of fins and fin height as in this study. The heat transfer in a 2.6 mm microfin tube is greater than it is in a 3.7 mm microfin tube. For mass velocity G = 100 kg/(m^{2}s), the effect of difference in which tube diameter on heat transfer for the vapor quality range x>0.4 is greater than it is for x<0.4. Further, the heat transfer coefficient for mass velocity G = 400 kg/(m^{2}s) increases with a decreasing tube diameter. It can be observed that the enhancement ratios of the heat transfer coefficients are 1.3–1.6

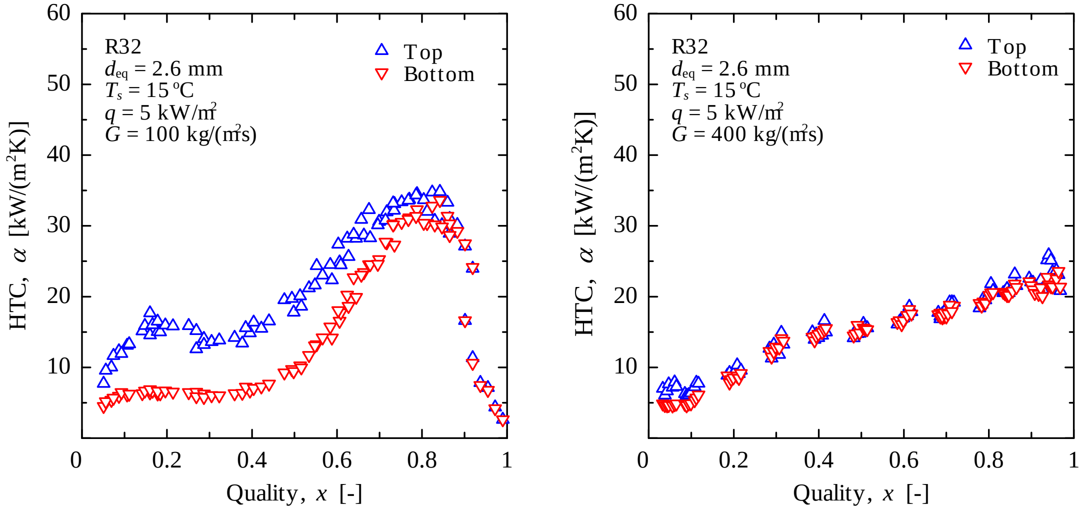


Figure 5: Heat transfer coefficient at top and bottom of tube.

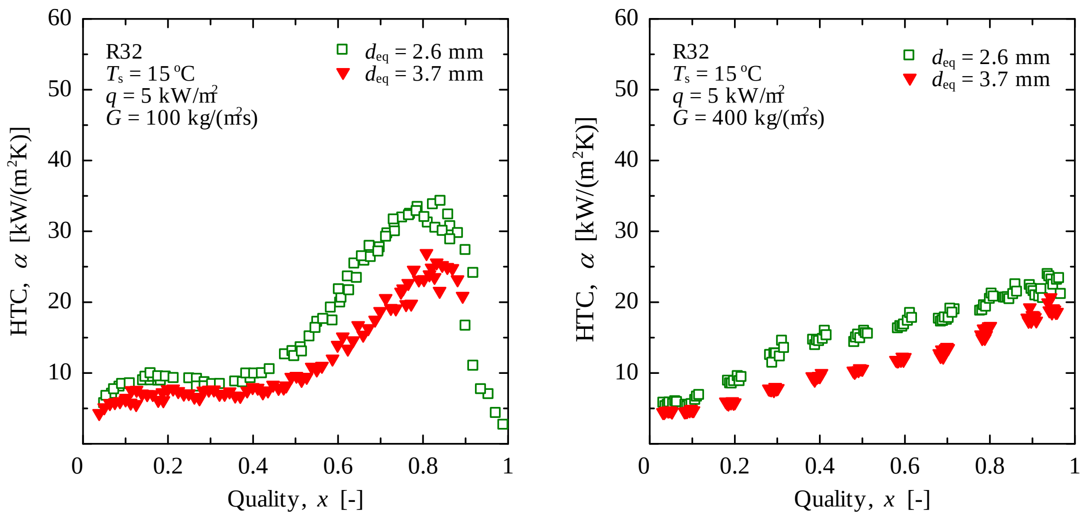


Figure 6: Effect of tube diameter on the boiling heat transfer for microfin tubes.

times the corresponding ratios for a 3.7 mm microfin tube. Therefore, the effect of tube diameter on heat transfer in small-diameter microfin tubes is significant in the regions of forced convection and thin liquid film evaporation heat transfer dominant compared to the nucleate boiling dominant region. Dryout vapor quality occurs at a slightly smaller vapor quality with decreasing tube diameter when mass velocity G = 100 kg/(m^{2}s); however dryout is not observed at G = 400 kg/(m^{2}s) and q = 5kW/m^2.

3.2 Frictional Pressure Drop

Figure 7 shows the frictional pressure drop in the test microfin tube for flow boiling a heat flux of 5 kW/m^2 and a saturation temperature of 15 ºC. The vapor quality change in the measuring section is 0.04–0.07. The frictional pressure drop increases with increasing vapor quality in the range x < 0.8. The frictional pressure drop for a mass velocity of 400 kg/(m^{2}s) is 4 times compared to pressure drop for a mass velocity of 200 kg/(m^{2}s) at the same vapor quality, because of the increase in vapor shear stress. The experimental frictional pressure drop is compared to the prediction correlations of Kubota *et al.* (2001) and Diani *et al.* (2014) as shown in the figure. The mean deviations of correlation are -8.2 % and 27.4 % for Kubota *et al.* (2001) and Diani *et al.* (2014) respectively. The correlation for

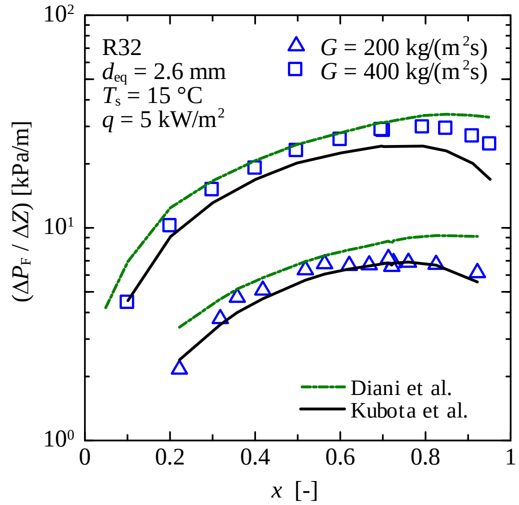


Figure 7: Frictional pressure drop during flow boiling at q = 5 kW/m^2.

Kubota *et al.* (2001) was based on data from microfin tubes with an inside fin root diameter in the range 6.3–6.8 mm and using refrigerants R410A and R407C; however, it agrees well with our experimental data for small-diameter microfin tubes.

4. CONCLUSIONS

This study experimentally investigated the heat transfer and pressure drop characteristics of flow boiling R32 in a horizontal small-diameter microfin tube with an equivalent diameter of 2.6 mm. The main conclusions are summarized as follows:
(1) The heat transfer coefficient increases with increasing heat flux and vapor quality in the pre-dryout region. However, heat transfer coefficient decreases with increasing heat flux, for mass velocity G = 100 kg/(m^{2}s) and vapor quality x>0.5.
(2) The heat transfer coefficient increases with increasing vapor quality in the pre-dryout region, especially for mass velocity values of 100 and 200 kg/(m^{2}s) owing to the flow pattern changing to separated flow with the thin liquid film evaporation at the top of the tube to enhance heat transfer by liquid surface tension and vapor quality. For further increase in mass velocity, the heat transfer coefficient decreases owing to the circumferential liquid film thickness becoming uniform.
(3) The dryout phenomenon is observed at low mass velocity. Dryout vapor quality decreases with increasing heat flux and decreasing tube diameter.
(4) The effect of tube diameter on boiling heat transfer in a forced convection and thin liquid film evaporation heat transfer dominant region is larger than it is in the nucleate boiling dominant region.
(5) The frictional pressure drop correlation of Kubota *et al.* (2001) matches our flow boiling experimental data. The correlation can be applied to such small-diameter microfin tubes.

NOMENCLATURE

A_H	heat transfer area	(m^2)
d	diameter	(m)
G	mass velocity	(kg/(m^{2}s))
h	specific enthalpy	(J/kg)
L	heat transfer length	(m)
Q	heat transfer rate	(W)
q	heat flux	(W/m^2)
T	temperature	(K)
T_s	saturation temperature	(K)
x	vapor quality	(-)

α	heat transfer coefficient	$(W/(m^2K))$
$\Delta P / \Delta Z$	pressure drop gradient	(Pa/m)
η	surface enlargement ratio	(-)

Subscripts

E	pre-heater
eq	equivalent
f	friction
in	inlet
R	refrigerant
TS	test section
w	tube wall

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