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POOL BOILING OF REFRIGERANT-OIL MIXTURES ON ENHANCED TUBES HAVING DIFFERENT PORE SIZES

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ABSTRACT

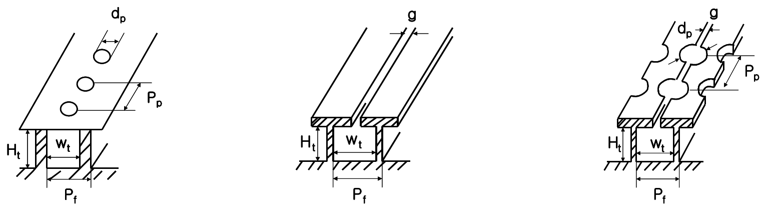
The effect of enhanced geometry (pore diameter, gap width) is investigated on the pool boiling of R-123/oil mixture for the enhanced tubes having pores with connecting gaps. Tubes having different pore diameters (and corresponding gap widths) are specially made. Significant heat transfer degradation by oil is observed for the present enhanced tubes. At 5% oil concentration, the degradation is 26 to 49% for T_{sat} = 4.4°C. The degradation increases 50 to 67% for T_{sat} = 26.7°C. The heat transfer degradation is significant even with small amount of oil (20 to 38% degradation at 1% oil concentration for T_{sat} = 4.4°C), probably due to the accumulation of oil in sub-tunnels. The pore size (or gap width) has a significant effect on the heat transfer degradation. The maximum degradation is observed for d_p = 0.20 mm tube at T_{sat} = 4.4°C, and d_p= 0.23 mm tube at T_{sat} = 26.7°C. The minimum degradation is observed for d_p = 0.27 mm tube for both saturation temperatures. It appears that the oil removal is facilitated for the larger pore diameter (along with larger gap) tube. The highest heat transfer coefficient with oil is obtained for d_p = 0.23 mm tube, which yielded the highest heat transfer coefficient for pure R-123. The heat transfer degradation increases as the heat flux decreases.

1. INTRODUCTION

Vapor compression refrigeration systems use oil-lubricated compressors, and the oil-refrigerant mixture circulates the refrigeration system. The oil concentration can be as high as 5% (Webb and McQuade, 2000). Of particular interest here is the effect of oil on the performance of flooded refrigerant evaporators, for which boiling occurs on the outside of tubes in a bundle. The shell side heat transfer coefficient of the flooded refrigerant evaporator may be obtained by adding the nucleate boiling heat transfer coefficient and the forced convective heat transfer coefficient as suggested by Chen (1963). Thus, knowledge of the nucleate pool boiling heat transfer coefficient of the tube is essential to predict the evaporator performance.

Structured enhanced boiling tubes, which are made by reforming the base surface to make fins of a standard or special configuration, are widely used in flooded refrigeration evaporators. Webb (1994) and Thome (1990) surveyed the techniques used to develop the enhanced boiling surfaces. Kim and Choi (2001) categorized the structured surfaces into three groups; those having surface pores such as Hitachi Thermoexel-E, those having narrow gaps such as Wieland GEWA-T or Trane bent-fin, and those having pores with connecting gaps such as Wolverine Turbo-B. Characteristic dimensions of the surfaces are illustrated in Fig. 1.

The effects of oil concentration on pool boiling of refrigerant-oil mixtures on structured enhanced surfaces have been discussed in several studies. In general, the addition of oil to boiling refrigerants causes a degradation in the heat transfer coefficient. The degradation depends on the saturation temperature and the heat flux. Wanniarachchi et al. (1987) tested R-114/oil mixture on GEWA-T (gap) and Thermoexel-E (pore) tubes. The boiling heat transfer coefficient decreased with the addition of oil. At 3% oil concentration (T_{sat} = 2.2°C, q"= 30 kW/m^2), the degradation in the heat transfer coefficient was 14% for GEWA-T, and 33% for Thermoexel-E. At 6% oil concentration, the degradation was 34% and 54% respectively. Memory et al. (1993) tested R-124/oil mixture on GEWA-K (gap) and



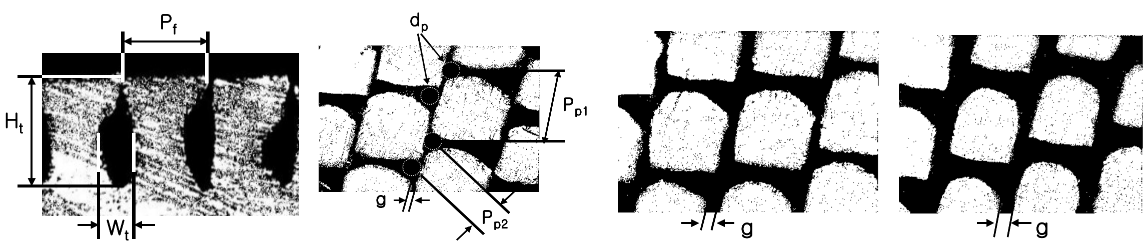
(a) Pores (b) Gaps (c) Pores with connecting gaps

Fig.1 Characteristic dimensions of structured enhanced tubes

Turbo-B (pore/gap) tubes. Interestingly, the boiling coefficient of GEWA-K for low oil concentration (up to 6%) was greater than that for pure R-124. However, the boiling coefficient of Turbo-B decreased with increasing oil concentration. At 6% oil concentration (T_{sat} = 2.2°C, q"= 25 kW/m^2), 24% degradation of heat transfer coefficient was observed. Webb and McQuade (1993) tested R-11/oil and R-123/oil mixtures on GEWA-SE (gap) and Turbo-B (pore/gap) tubes. The boiling coefficient decreased with the addition of oil. At 5% oil concentration in R-11 (T_{sat} = 4.4°C, q"= 30 kW/m^2), the degradation in heat transfer coefficient was 25% for GEWA-SE, and 36% for Turbo-B. The degradation was smaller for R-123. At 5% oil concentration (T_{sat} = 2.2°C, q"= 25 kW/m^2), the degradation was 23% for GEWA-SE and 26% for Turbo-B. Memory et al. (1995) provide R-114/oil data for gapped (GEWA-T, GEWA-K, GEWA-YX), pored (Thermoexel-HE) and pore/gap (Turbo-B) tubes. For gapped tubes at high heat fluxes, small quantities of oil increased the heat transfer coefficient. For pored or pore/gap tubes, however, addition of oil decreased the heat transfer coefficient. These studies show that the heat transfer deterioration by oil is more significant for pore or pore/gap tubes than for gapped tubes. For enhanced tubes, the oil accumulates in the sub-tunnels as a result of boiling of more-volatile refrigerants. It appears that the oil-rich mixture is more easily removed from the sub-tunnels for gapped geometry compared to pored geometry. The boiling coefficients of pure refrigerants are, however, larger for pored tubes than for gapped tubes (Webb and McQuade, 1993).

Most of the studies on the oil effect of enhanced geometries were conducted using commercial enhanced tubes. Because the tunnel and pore shapes of commercial tubes are totally different each other, the effect of geometric parameters cannot be concluded from these data. Only one study systematically varied the enhancement geometry. Zarnescu et al. (2000) investigated the effect of geometric parameters (tunnel and pore size) on the oil degradation of pored tubes. The pored tubes were specially made by soldering a hole-punched copper foil on low finned tubes. The effects of pore diameter (0.18 and 0.23 mm), pore pitch (0.75 and 1.5 mm), and tunnel cross sectional area (0.19 and 0.23 mm^2) were investigated using R-134a/oil mixtures. The oil degradation increased as the tunnel cross section area decreased. The pore diameter and pore pitch were found not to be a major factor.

The literature survey shows no such systematic study on the pore/gap geometry. The representative pore/gap tube is Turbo-B. In this study, tubes having different pore/gap dimensions were specially made, and tested using R-123/oil mixtures. The pure R-123 heat transfer coefficients of the pore/gap tubes have been reported by Kim and Choi (2001).



(a) cross-sectional view of the tunnel (b) d_p= 0.20 mm (c) d_p= 0.23 mm (d) d_p= 0.27 mm

Fig. 2 Enlarged photos showing the present enhanced geometry: (a) cross-sectional view of the tunnel, (b) tube with d_p = 0.20 mm, (c) tube with d_p = 0.23 mm, (d) tube with d_p = 0.27 mm

Table. 1 Geometric dimensions of the sample tubes

No.	d_p (mm)	g (mm)	P_{p1} (mm)	P_{p2} (mm)	P_f (mm)	H_t (mm)	W_t (mm)
1	0.20	0.04	0.71	0.374	0.605	0.54	0.25
2	0.23	0.07	0.71	0.384	0.605	0.54	0.25
3	0.27	0.10	0.71	0.400	0.605	0.54	0.25

2. SAMPLE TUBES

The same tubes tested by Kim and Choi (2001) for pure R-123 were used in this study. The enlarged photos of the tunnels and surfaces of the present pore/gap tubes are shown in Fig. 2. These tubes were made from low integral fin tubes having 1654 fins per meter with 1.3 mm fin height, cutting small notches (0.9 mm depth) on the fins, and then flattening the fins by a rolling process. The resultant tube had triangular pores with connecting gaps and gourd-shaped tunnels. Three tubes with different pore size (and corresponding gap width) were made. The pose size was varied by changing the pressure on the rollers. The geometric dimensions of the surfaces are listed in Table 1. The pore size in the table is represented by the diameter of a circle inscribed in a triangle. The geometric dimensions were measured from enlarged photos taken at twenty different locations. The pore size and the gap width were fairly uniform, and those listed in Table 1 are the averaged values. One notable thing of the present tube is that two pores exist per pore pitch.

3. EXPERIMENTAL APPARATUS AND PROCEDURES

The test apparatus is shown in Fig. 3. The pool boiling test cell consists of a 150 mm inner diameter and 350 mm long copper tube and two flanges. The test tube was mounted on the copper flange at one end of the test cell, and the sight glass was installed on the other end of the test cell. The vapor generated during the test condensed in the external condenser. The brine from the constant temperature bath circulated in the tube-side of the condenser. Tests were performed at two saturation temperatures (4.4°C and 26.7°C). The 4.4°C was chosen because commercial refrigeration chillers operate at the temperature and the 26.7°C was chosen because it is the normal room temperature.

The detailed sketch of the test tube is shown in Fig. 4. The enhanced tubes were specially made from thick-walled copper tubes of 18.8 mm outer diameter and 13.5 mm inner diameter. The length was 170 mm. An electric cartridge heater of 13.45 mm diameter and 180 mm long was inserted into the test tube. Its electric power was controlled by an auto-transformer. The heater was specially manufactured to contain 170 mm long heated section (same length as that of the test tube) and two 5 mm long unheated end sections. To minimize the heat loss, the unheated sections were covered with a teflon cap and a teflon ring as illustrated in Fig. 4. Before insertion, the heater was coated with a thermal epoxy to enhance the thermal contact with the tube.

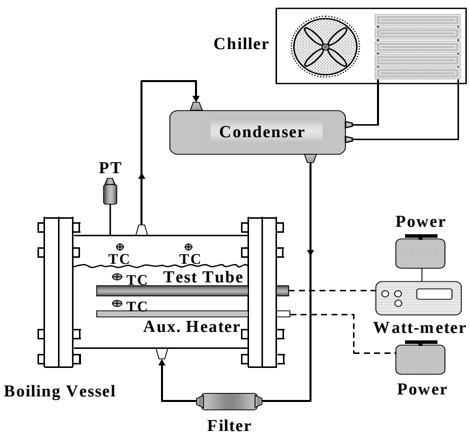


Fig. 3 Schematic drawing of the experimental apparatus.

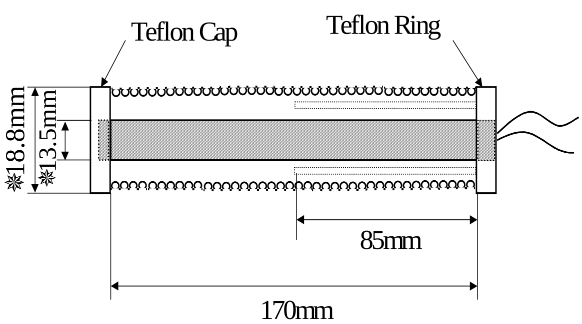


Fig. 4 Detailed sketch of the test tube.

Four thermocouple holes of 1.0 mm diameter were drilled to the center of the tube. Copper-constantan thermocouples of 0.3 mm diameter per wire were inserted into the holes to measure the tube wall temperature. Before insertion, the thermocouples were coated with a thermal epoxy [Chromalox HTRC] to provide good thermal contact with the tube wall. The thermal conductivity of the epoxy is close to that of aluminum. The vapor temperature was measured at two locations on the upper part of the test cell, and the liquid temperature was measured at 20 mm above the test tube and at 20 mm below the test tube.

Oil was injected into the system using a charging cylinder having two ports. The top port of the cylinder was connected to a nitrogen tank. A metered reservoir and an electronic scale supplied precise amount of oil into the cylinder. Up to 10% (mass fraction) mineral oil (alkylbenzene oil with 45.8 m^2/s viscosity at 40^{o}C) was used. The nitrogen gas pushed the oil from the charging cylinder into the test cell through a series of valves. This arrangement allowed initial and additional amounts of oil to obtain desired concentration increments. Tests were conducted for 1%, 2%, 3%, 5%, 7% and 10% oil concentration. For some tests, however, the concentration was limited to 5% because no significant change in heat transfer coefficients was observed for concentrations larger than 5%. When the required oil concentration was attained in the test cell, the tube surface was conditioned employing the surface aging technique "B" of Bergles and Chyu (1982). The method involved degassing the test tube and the pool at the maximum heat flux (approximately 50 kW/m^2) for an hour. The pool was maintained close to the saturation temperature using the constant temperature bath (for 4.4°C saturation temperature) or the auxiliary heater (for 26.7^0 C saturation temperature). The data were taken decreasing the heat flux. Readings were taken 10 minutes after each power change, at which time a steady state condition was attained. Before reading the data, the auxiliary heater was shut off to minimize the convective effects from the heater. Throughout the test, the liquid level was maintained at 5 cm above the test tube. At the end of each test, the mixture was drawn into the pre-weighted cylinder. The sample and the cylinder were then weighed and the mass of the sample was determined. A valve was then cracked, and the refrigerant was carefully flashed off for two days. The mass of oil in the cylinder was then determined by weighing the cylinder. The charged and measured oil concentrations agreed within ± 1%.

The heat transfer coefficient (h) is determined by the heat flux (q") over wall superheat ($T_w - T_r$). The measured pool temperature (T_r) was used to obtain the heat transfer coefficient, following the suggestion by Thome (1996). Calculations of q" and h are based on the envelope area, defined by the heated length (170 mm) multiplied by the tube outside perimeter. The input power to the heater was measured by a precision watt-meter [Chitai 2402A] and the thermocouples were connected to the data logger [Fluke 2645A]. The thermocouples were calibrated and checked for repeatability. The calculated accuracy of the temperature measurement was ± 0.15°C. Tube wall temperature was determined by extrapolating the thermocouple temperatures to the tube wall using Fourier's law.

An error analysis was conducted following the procedure proposed by Kline and McClintock (1953). The uncertainty in the heat transfer coefficient is estimated to be ± 3% at the maximum heat flux (50 kW/m^2) and ± 7% at a low heat flux (10 kW/m^2). The heat flux profile in axial direction may not be uniform because of the heat loss at each ends of the heater. A series of tests were conducted changing the thermocouple location in axial direction. The corresponding heat transfer coefficients agreed each other within ± 3%. All the tests were conducted with thermocouples located at the center of the tube.

4. RESULTS AND DISCUSSION

4.1 Smooth Tube

The boiling heat transfer coefficients of the smooth tube taken at 4.4°C saturation temperature are shown in Fig. 5. The heat transfer coefficient decreases with addition of oil. In Fig. 6, the degradations of the heat transfer coefficient ($1-h/h_{pure}$) are plotted as a function of oil concentration for q"= 8, 20 and 40 kW/m^2. Significant degradation of heat transfer coefficient is observed. At q"= 40 kW/m^2 and T_{sat} = 4.4°C, the degradation is 21% at 2% oil concentration and 37% at 5% oil concentration. Fig. 6 shows that the degradation is more significant at higher heat fluxes. As the heat flux increases, the boiling becomes more vigorous, and the oil is transported to the surface at a faster rate, establishing a thicker oil layer than at low heat fluxes. Jensen and Jackman (1984) comment that the thick oil layer inhibits bubble growth, and leads to the degradation of heat transfer coefficient. Memory et al. (1995) also report larger heat transfer degradation at a higher heat flux. Fig. 6 shows that the effect of saturation temperature is not significant.

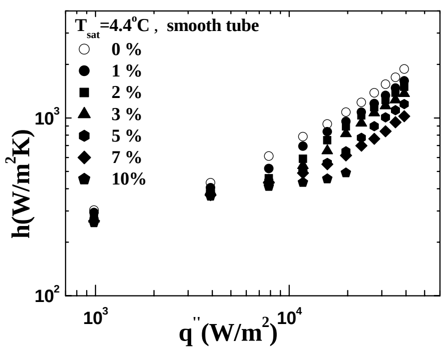


Fig. 5 Effect of oil on boiling heat transfer coefficients of the smooth tube at T_{sat} = 4.4°C

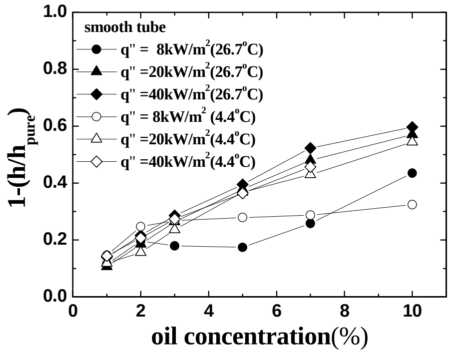


Fig. 6 Effect of oil on the heat transfer degradation for the smooth tube

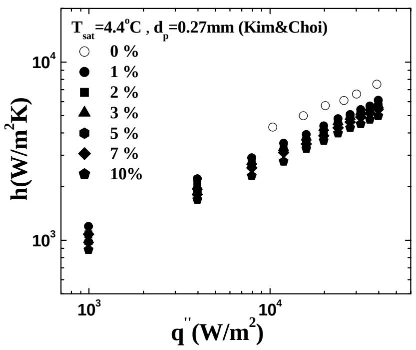


Fig. 7 Effect of oil on heat transfer coefficients of d_p =0.27 mm tube (T_{sat} = 4.4°C)

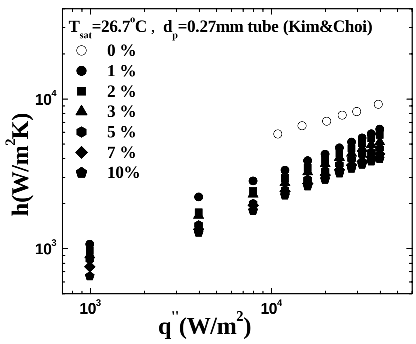


Fig. 8 Effect of oil on heat transfer coefficients of d_p =0.27 mm tube (T_{sat} = 26.7°C)

4.2 Enhanced Tubes

Typical heat transfer coefficient data are shown in Figs. 7 and 8 for d_p = 0.27 mm tube at T_{sat} = 4.4°C and 26.7°C. In the figures, pure R-123 data of Kim and Choi (2001) are shown. For the present tests, we could not obtain the pure R-123 data. Once the test apparatus was contaminated by oil, it was impossible to thoroughly clean the apparatus. The apparatus was cleaned by repeatedly diluting with pure refrigerant. However, a small amount of oil always remained in the apparatus, which induced significant degradation of the heat transfer coefficient. We believe that the pure R-123 data of Kim and Choi (2001) represent the present 0% concentration data, because the same tubes were used, and the same test procedure was taken for both tests.

Figs. 9 to 11 show the degradation of heat transfer coefficients for the three enhanced tubes. Significant degradation (50 to 67% at 5% oil concentration) is observed for T_{sat} = 26.7°C. The degradation is much smaller (26 to 49% at 5% oil concentration) for T_{sat} = 4.4°C. The present smooth tube yielded 29 to 36% degradation at 5% oil concentration for T_{sat} = 4.4°C. Figs. 9 to 11 show that the saturation temperature has significant effect on the heat transfer degradation. The effect of saturation temperature was not significant for the smooth tube. Wang et al. (1999) conducted R-22/oil mixture boiling test on a smooth tube, and reported that the heat transfer degradation by oil decreased as the saturation temperature decreased. The reason was attributed to the decrease of the surface tension and more intense foaming at lower temperatures. For the present test, however, no intense foaming was observed. Figs. 9 to 11 show that significant heat transfer degradation occurs even with a small amount of oil. At 1% oil concentration, 20 to 38% degradation is observed at T_{sat} = 4.4°C. The degradation is more significant (32 to 60%) at T_{sat} = 26.7°C. For the smooth tube, the degradation was less than 15% at 1% oil concentration. It appears that the oil accumulates in the tunnel, resulting in heat transfer degradation. Figs. 9 to 11 show that the degradation increases as the heat flux decreases. This result is in contradiction to the smooth tube result, where the reverse was true. Different

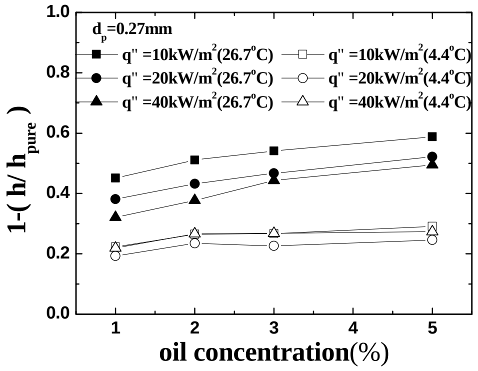


Fig. 9 Effect of oil on the heat transfer degradation for d_p = 0.27mm tube

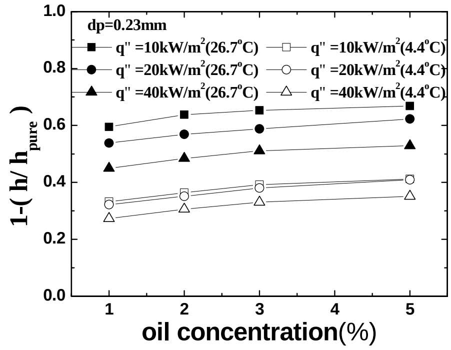


Fig. 10 Effect of oil on the heat transfer degradation for d_p = 0.23mm tube

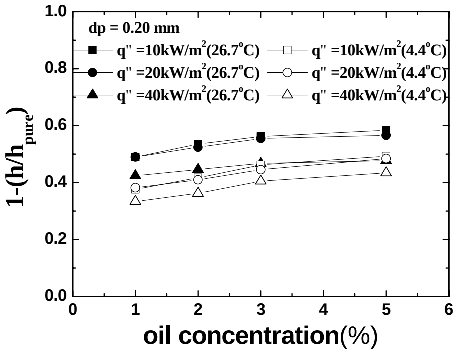


Fig. 11 Effect of oil on the heat transfer coefficients for d_p =0.20 mm tube

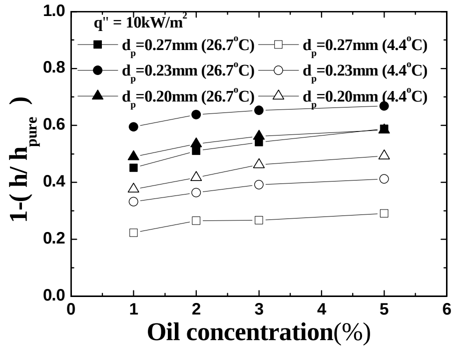


Fig. 12 Effect of oil on heat transfer degradation at q" = 10 kW/ m^2

from the smooth tube, where all the heat transfer occurs at the tube surface, significant portion of the total heat transfer occurs in the tunnel by thin film evaporation for enhanced tubes (Chien and Webb, 1998). It appears that, as the boiling gets more vigorous at a high heat flux, more oil-rich mixture will be removed from the tunnel, resulting in reduced heat transfer degradation.

Fig. 12 shows the heat transfer degradation at q"= 10 kW/m^2. The maximum degradation is observed for d_p = 0.20 mm tube at T_{sat} = 4.4^{o}C, and d_p = 0.23 mm tube at T_{sat} = 26.7°C. The minimum degradation is observed for d_p = 0.27 mm tube for both saturation temperatures. It appears that the oil removal is facilitated for the larger pore diameter (along with larger gap) tube. Although not shown here, the same trend was observed for other heat fluxes.

Fig. 13 shows the heat transfer coefficients at q"= 40 kW/m^2 and T_{sat} = 4.4°C for the three enhanced tubes. The pure R-123 data are also shown. The highest heat transfer coefficient is obtained for d_p = 0.23 mm tube both for pure R-123 and R-123/oil mixtures, although the heat transfer degradation by oil was minimum for d_p = 0.27 mm tube. The present study shows that significant heat transfer degradation is observed for the pore/gap tubes for pool boiling situations. For the bundle, however, forced flow of refrigerant might diffuse the oil-rich mixture in the tunnels. This would result in less performance degradation than is observed for pool boiling. Czikk et al. (1970) performed a study of the effects oil concentrations on R-11 chiller augmented with porous coated tubes, and found that concentrations up to 2% had no effect on the chiller performance. Since the porous coated tube have been shown to have 25% reduction in boiling heat transfer coefficient for 3% oil by Gottzmann et al. (1973), it appears that forced convection might act to alleviate the effect of oil concentration. This issue is currently under investigation by the authors.

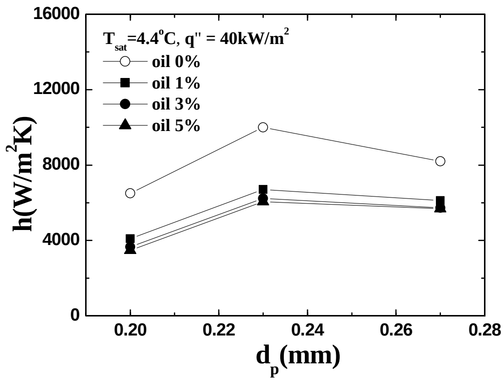


Fig. 13 Effect of pore diameter on the heat transfer coefficients of R-123/oil mixtures at T_{sat} = 4.4C, q"= 40 kW/m^2

5. CONCLUSIONS

In this study, the effect of enhanced geometry (pore diameter, gap width) was investigated on pool boil of R-123/oil mixture for the enhanced tubes having pores with connecting gaps. Listed below are the major findings.

(1) Significant heat transfer degradation by oil is observed for the present enhanced tubes. At 5% oil concentration, the degradation is 26 to 49% for T_{sat} = 4.4°C. The degradation increases 50 to 67% for T_{sat} = 26.7°C.
(2) The heat transfer degradation is significant even with small amount of oil (20 to 38% degradation at 1% oil concentration for T_{sat} = 4.4°C), probably due to the accumulation of oil in sub-tunnels.
(3) The pore size (gap width) has a significant effect on the heat transfer degradation. The maximum degradation is observed for d_p = 0.20 mm tube at T_{sat} = 4.4°C, and d_p= 0.23 mm tube at T_{sat} = 26.7°C. The minimum degradation is observed for d_p = 0.27 mm tube for both saturation temperatures. It appears that the oil removal is facilitated for the larger pore diameter (along with larger gap) tube.
(4) The highest heat transfer coefficient with oil is obtained for d_p = 0.23 mm tube, which yielded the highest heat transfer coefficient for pure R-123.
(5) The heat transfer degradation increases as the heat flux decreases. This result is in contradiction to the smooth tube result, where the reverse is true. It appears that more oil is removed from the tunnel at a higher heat flux by vigorous bubble action.

NOMENCLATURE

d_p	pore diameter	(m)
h	heat transfer coefficient	(W/m^{2}K)
H_t	tunnel height	(m)
P	pressure	(N/m^2)
P_f	fin pitch	(m)
P_p	pore pitch	(m)
P_{p1}	circumferential pore pitch	(m)
P_{p2}	neighboring pore pitch	(m)
P_w	wetted perimeter	(m)
q"	heat flux	(W/m^2)
T_r	pool temperature	(K)
T_{sat}	saturation temperature	(K)
T_w	tube wall temperature	(K)

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