Proceedings of Fifth International Conference on **Enhanced, Compact and Ultra-Compact Heat Exchangers: Science, Engineering and Technology**, Eds. R.K. Shah, M. Ishizuka, T.M. Rudy, and V.V. Wadekar, Engineering Conferences International, Hoboken, NJ, USA, September 2005.

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CONVECTIVE FLOW BOILING OF REFRIGERANT-OIL MIXTURES ON AN ENHANCED TUBE BUNDLE

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ABSTRACT

The effect of oil on convective boiling of R-123 in an enhanced tube bundle is experimentally investigated at 26.7°C saturation temperature. The enhanced tube had 0.23 mm pore diameter, which had been optimized using pure R-123. The effects of oil concentration (0 to 5%), heat flux (10 to 40 kW/m^2), mass velocity (8 to 26 kg/m^2 s) and vapor quality are investigated. The data are compared with the pool boiling counterpart. The oil significantly reduces the bundle boiling heat transfer coefficient. With 1 % oil, the reduction is approximately 35%. Further addition of oil further reduces the heat transfer coefficient. The reduction in the heat transfer coefficient is smaller in a bundle (convective boiling) than in a pool (single-tube pool boiling), with larger difference at a smaller heat flux. The two-phase convection in a bundle may remove the oil-rich layer near the heating surface, and reduce the thermal degradation by oil. Similar to pure R-123, the effects of mass velocity and vapor quality are negligible for the convective boiling of R-123/oil mixture.

INTRODUCTION

In flooded evaporators of large tonnage refrigeration machines, compressor oil is inevitably present. Thus, the evaluation of the effect of oil on convective boiling in a tube bundle is important for proper sizing of the equipment. The oil concentration in the system is approximately 0.5 to 3% (Thome, 1996). Structured enhanced tubes, which are made by reforming the base surface to make fins of standard or special configuration, are widely used in refrigeration equipments. The structured boiling surfaces have been categorized into three groups (Kim and Choi, 2001); those having surfaces pores such as Hitachi Thermoexel-E, those having narrow gaps such as Wieland GEWA-T, and those having pores with connecting gaps such as Wolverine Turbo-B. The literature reveals that most of the studies on oil effect were conducted for a pool boiling using a single tube. Studies on refrigerant-oil mixtures in a tube bundle are very rare.

Many single-tube pool boiling studies on structured enhanced tubes (Wannirachchi et al., 1987; Memory et al., 1993; Webb and McQuade, 1993; Memory et al., 1995a; Kim et al., 2004) reveal that the heat transfer coefficient is significantly reduced (approximately 20 to 40%) by addition of oil. The reduction increases as the oil concentration increases, or the heat flux decreases (Memory et al., 1995a; Kim et al., 2004).

Limited studies on the oil effect in enhanced tube bundles include Gan et al. (1993), Marvillet (1989), Memory et al. (1995b), Tatara and Payvar (2000a, 2000b). With a porous aluminum bundle and R-113 and R-113/R-11 mixtures, Gan et al. (1993) reported that the heat transfer reduction was generally less in a bundle than that in a single tube. The convective effect by the vapor fraction was thought to counter the influence the oil, due to the fact the bundle coefficient outperformed the single tube. Similar observation was made by Marvillet (1989) for a porous aluminum tube bundle containing 84 tubes.

Memory et al. (1995b) examined the R-114/oil convective boiling in structured enhanced (Turbo-B) and porous tube bundles. The bundle heat transfer coefficient decreased as the oil concentration increased, or the heat flux decreased. For the Turbo-B bundle, the reduction was over 30% at 10% oil concentration (at 30 kW/m^2). The single-tube heat transfer reduction was less (10% at 10% oil concentration) (Memory et al, 1995a). This result contradicts with those by Gan et al. (1993) and Marvillet (1989), where the heat transfer reduction was less in a bundle configuration. Memory et al.(1995b) also report that, for the porous tube, the reduction was approximately the same both for the bundle and for the single-tube (approximately 40% at 10% oil concentration).

Tatara and Payvar (2000a, 2000b) investigated the oil effect in a Turbo-BII tube bundle using R-123 and R-134a. The Turbo-BII bundle showed a significant decrease with addition of even a small amount of oil. At 1% oil concentration, the heat transfer coefficient reduced to approximately 30% (for R-123) and 25% (for R-134a) from its no-oil baseline. Unfortunately, no comparison was made with the single tube performance. The bundle heat transfer coefficient decreased as the oil concentration increased, or the heat flux decreased.

The previous studies provide conflicting results on the relative magnitude of heat transfer reduction in a single-tube pool boiling and in a bundle. Gan et al. (1993) and Marvillet (1989) report that the heat transfer reduction is less in a bundle configuration. However, Memory et al. (1995b) report a different trend. In addition, the effects of the mass flux or the vapor quality, which may affect the heat transfer coefficient in a bundle configuration, have not been reported.

In this study, convective boiling experiments were conducted in an enhanced tube bundle using R123-mineral oil mixtures. The enhanced tubes were specially made to have boiling surfaces with pores and connecting gaps. The pore size was optimized to yield the best convective boiling coefficient with pure R-123, and the optimized tube (d_p = 0.23 mm) was subjected to the oil test. The effects of oil concentration, heat flux, mass flux and vapor quality on the heat transfer reduction were investigated.

SAMPLE TUBES

The same tubes as those used for the previous pool and convective boiling tests (Kim and Choi, 2001; Kim et al., 2002; Kim et al., 2004) were used in this study. The surface geometry is shown in Fig. 1. 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 pore size was varied by changing the pressure on the rollers. The pore size is represented by the diameter of a circle inscribed in a triangle.

EXPERIMENTAL APPARATUS

A schematic drawing of the apparatus is shown in Fig. 2, and the detailed drawing of the test section is sh own in Fig. 3. The same apparatus and the test section had been used by Kim et al. (2002) for convective

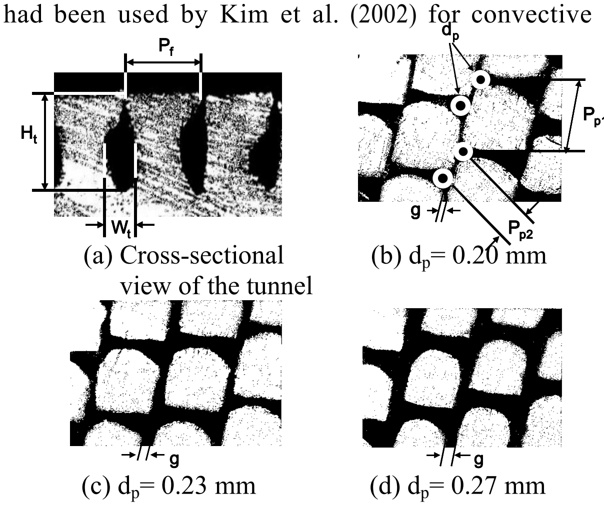


Fig. 1 Enlarged photos showing the present enhanced geometry : (a) cross-sectional view of the tunnel, (b) tube with d_p = 0.20 mm, (c) tube with d_p = 0.23 mm, (d) tube with d_p = 0.27 mm

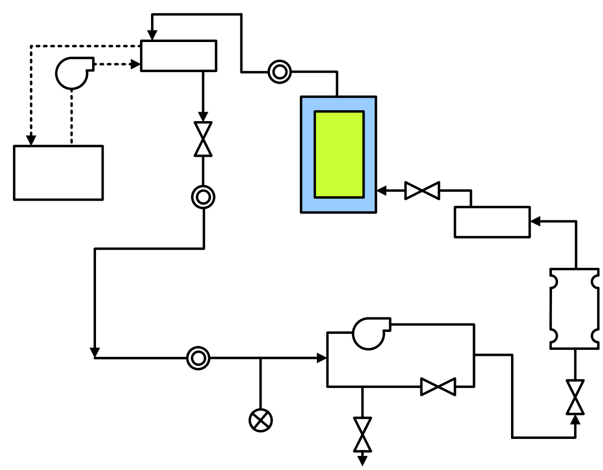


Fig. 2 Schematic drawing of the experimental apparatus

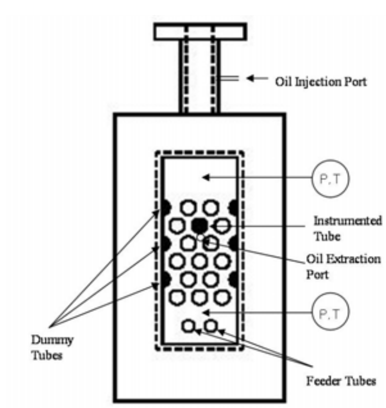


Fig. 3 Detailed drawing of the test section

boiling of pure R-123 and R-134a. Refrigerant enters at the bottom of the test section at a known vapor quality.

Heat is supplied to the tubes in the tube bundle by cartridge heaters. The two-phase mixture leaves the test section and goes to the condensers. Three condensers having 7.5 kW cooling capacity each are connected in parallel. The sub-cooled liquid then pass through a dryer, and goes to the magnetic pump having 1 gpm capacity. The mass flow meter [1.0 gpm, Micromotion DN25S-SS-1] is placed between the magnetic pump and the pre-heater to measure the mass flow rate. Heat is supplied to the pre-heater to obtain a known vapor quality. The two-phase mixture enters the test section through the feeder tubes located at the bottom of the tube bundle. Heat input to the pre-heater determines the inlet vapor quality, which can be controlled independently. Since, the liquid loop is complete by itself, the mass flow rate can be controlled independently. Finally, the heat flux to the test tubes is varied by regulating the line voltage to tubes. Thus, the apparatus was designed to control the vapor quality, mass velocity and heat flux independently.

Figures 3 and 4 show the details of the test section and the instrumented tube respectively. 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. Cartridge heaters of 13.45 mm diameter and 180 mm long were inserted into the test tubes. The heaters were specially manufactured to contain 170 mm long heated section (same length as that of the test tubes) and two 5 mm long unheated end sections. To minimize the heat loss, the unheated sections were covered with Teflon caps and Teflon rings as illustrated in Fig. 4. Before insertion, the heaters were coated with thermal epoxy to enhance the thermal contact with the tubes. The heaters were screwed into the back flange of the test section to form a staggered array of an equilateral triangular pitch of 23.8 mm. For that purpose, the heaters were specially manufactured to contain a male screw at one end. Heat was supplied to all the tubes except for those at the bottom row – its role was to develop flow in the tube bundle.

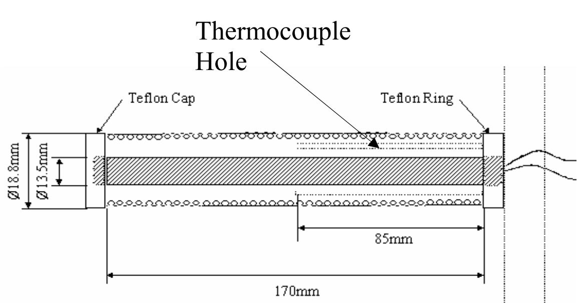


Fig. 4 Detailed sketch of the instrumented tube

The instrumented tube was located at the center of the fifth row from the bottom. Figure 4 shows the cross section of the instrumented tube. The tubes have four thermocouple holes of 1.0 mm 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 steady state heat conduction equation was used to correct for the conduction temperature drop between the thermocouple and the boiling surface.

Saturation temperatures were measured at the top and the bottom of the tube bundle. Calibrated pressure transducers were also used to measure the saturation pressure at the same location. During the experiment, the saturation temperature calculated from the measured pressure was compared with the measured temperature, and they agreed within 0.2^{o}C. The measured temperatures were used for data reduction. For convective boiling, the saturation temperature decreases along the flow passage due to the pressure loss. For the present experiment, the decrease was on the order of 0.5^{o}C. The saturation temperature at the instrumented tube was determined by linear interpolation of the top and bottom temperatures. The oil was injected to the system from the top of the test section. The oil extraction port located near the instrumented tube. A charging cylinder having two ports was used to inject the oil into the system. 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. A 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. At the end of each test, approximately 1 kg of the refrigerant-oil mixture was extracted from the system 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%.

Convective boiling data were taken using R-123 at 26.7^{o}C. The mass velocity was varied from 8 $kg/m^2 s$ to 26 $kg/m^2 s$, heat flux from 10 kW/m^2 to 40 kW/m^2 and vapor quality from 0.1 to 0.9. The oil concentration was varied up to 5%. The vapor quality at the instrumented tube location was determined from Eq. (1).

$$x = \left[\frac{Q_p + NQ_H}{\dot{m}_r} - c_{pr}(T_{p,sat} - T_{p,in}) \right] \qquad (1)$$

Here, Q_p is the heat supplied to the pre-heater, Q_H is the heat supplied to a single heater in the test section, and N is the number of active heaters upstream of the instrumented tube. The test section and the pre-heater were heavily insulated to minimize the heat loss to the environment.

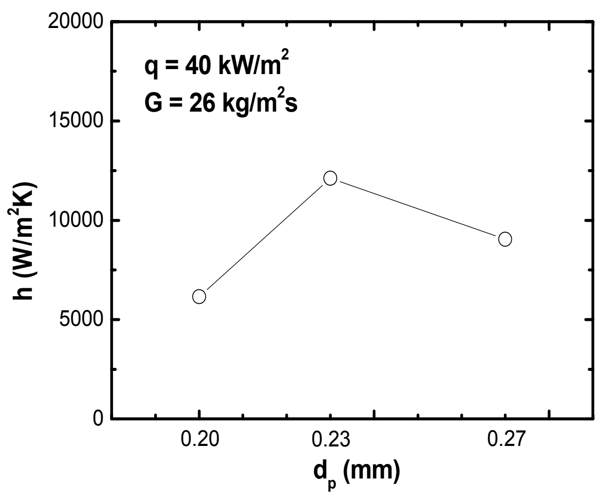
The heat transfer coefficient (h) was determined by the heat flux (q) over wall superheat (T_w – T_{sat}). Calculations of q and h were based on the envelope area, defined by the heated length (170 mm) multiplied by the tube outside perimeter. The input power to the heater was measured by a precision watt-meter [Chitai 2402A] and the thermocouples were connected to the data logger [Fluke 2645A]. The pressure transducers were also connected to the data logger. The thermocouples and the transducers were calibrated and checked for repeatability. The calculated accuracy of the temperature measurement was $\pm 0.1^0$ C, heat flux measurement $\pm$ 0.5%, mass velocity measurement $\pm$ 1% and vapor quality measurement $\pm$ 2 %. An error analysis was conducted following the procedure proposed by Kline and McClintock (1953). The uncertainty in the heat transfer coefficient was estimated to be $\pm$ 2% at the maximum heat flux (40 kW/m^2) and $\pm$ 8% at a low heat flux (10 kW/m^2).

RESULTS AND DISCUSSIONS

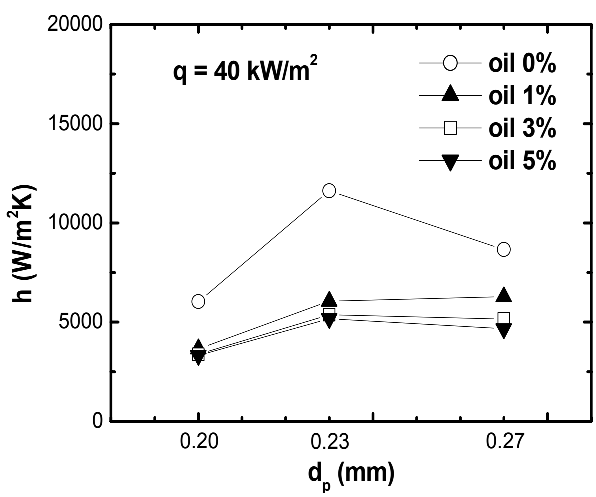
The pool and convective boiling data of pure R-123 at 26.7^{o}C saturation temperature were obtained previously for the three enhanced tubes (Kim and Choi, 2001; Kim et al., 2002), and are reproduced in Fig. 5. The convective boiling data are quality-average values. Also shown in Fig. 5 (b) are the reduced pool boiling heat transfer coefficients by addition of oil (Kim et al., 2004). This figure shows that d_p = 0.23 mm tube yields the highest pool and convective boiling heat transfer coefficient, and maintains the highest pool boiling value even with the oil. Thus, d_p = 0.23 mm tubes were used for the present study to investigate the oil effect on convective boiling heat transfer in an enhanced tube bundle.

The previous study (Kim et al., 2002) revealed that the effects of mass velocity (8 to 26 kg/m^{2}s) and vapor quality on convective boiling of pure refrigerant in an enhanced tube bundle were negligible. The reasons were attributed to the relatively small mass velocity, and the persistence of intermittent two-phase flow in the bundle configuration to a high vapor quality. Similar conclusions are drawn from the present study. Figure 6 (a) shows the heat transfer coefficients at two different mass fluxes (G = 8 kg/m^2 s and 26 kg/m^2 s) at 5% oil concentration. The heat flux was 10 kW/m^2. The effects of mass velocity and vapor quality are negligible. The high heat flux data at 40 kW/m^2 [Fig. 6 (b)] show the same trend.

Figure 7 shows the reduction of heat transfer coeffi cient by addition of oil at three different heat fluxes (q = 10, 20 and 40 kW/m^2). The mass flux is 8 kg/m^2 s. Th ese figures show that the addition of 1% oil significantl y reduces the heat transfer coefficient. Approximately 3 5% reduction is observed with 1% oil. Further addition of oil further reduces the heat transfer (55% reduction a t 5% oil concentration). The effect of



(a) Convective boiling



(b) Pool boiling

Fig. 5 a) Convective boiling data of pure R-123 (Kim et al., 2002), and b) pool boiling data of pure R-123 and R-123/oil (Kim and Choi, 2001; Kim et al., 2004) at T_{sat} = 26.7

vapor quality is negligible. In Figs. 6 and 7, the pure ref
rigerant data are limited to a relatively low quality due t
o limitation of the test apparatus.

The reduction of heat transfer coefficient is re-
plotted in Fig. 8 as a function of oil concentration. The
plotted data are vapor-
quality averaged values. Also shown in the figure are th
e single-tube pool boiling R-
123/oil data obtained from the same tube (Kim et al., 2
004). The reduction in the heat transfer coefficient is s
maller for the convective boiling as compared with the
pool boiling. Figure 8 also shows that the difference bet
ween the pool boiling and the convective boiling is larg
er at a smaller heat flux, where the convective effect is r
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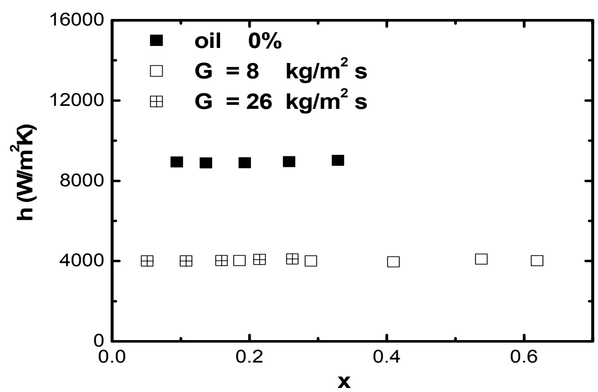
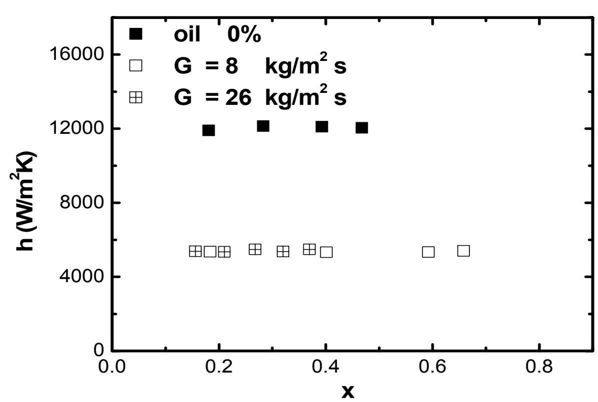
undle configuration.

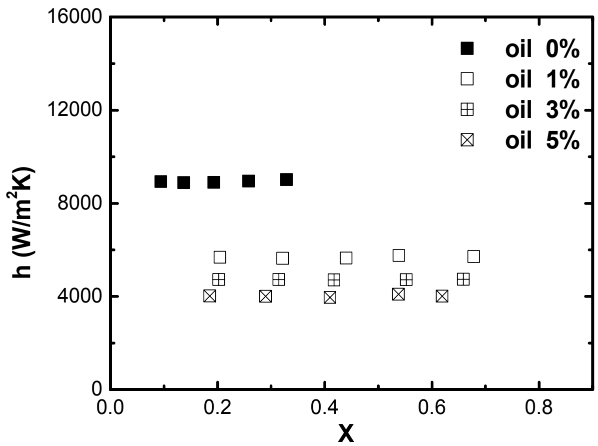


(a) q = 10 kW/m^2

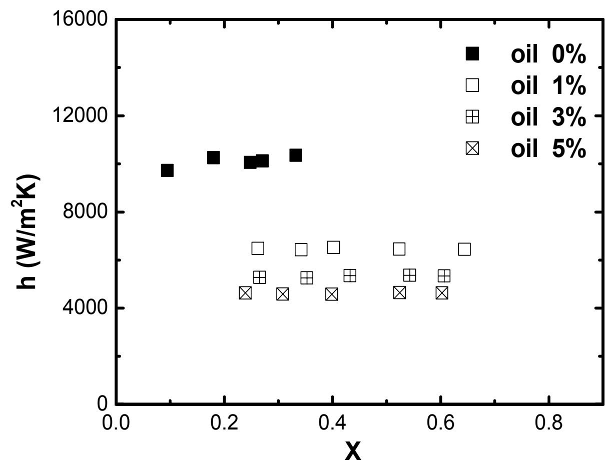


(b) q = 40 kW/ m^2

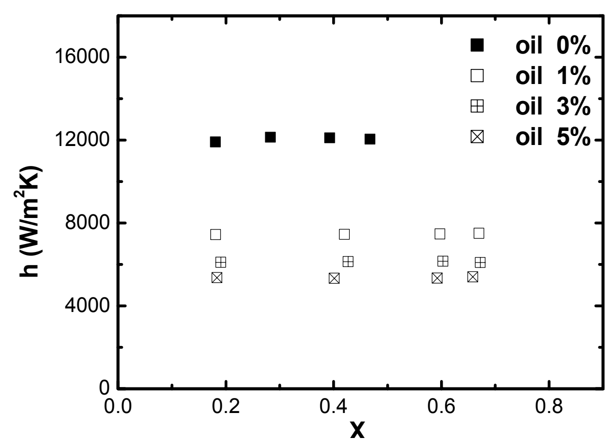
Fig. 6 Effect of mass velocity on the heat transfer coefficient of the enhanced bundle for R-123 at T_{sat} = 26.7 ℃ and 5% oil concentration



(a) q = 10 kW/ m^2



(b) q = 20 kW/ m^2



(c) q = 40 kW/ m^2

Fig. 7 Effect of oil on the convective boiling heat transfer coefficient of the enhanced tube bundle for R-123 at G = 8 kg/m^{2}s and T_{sat} = 26.7 ℃

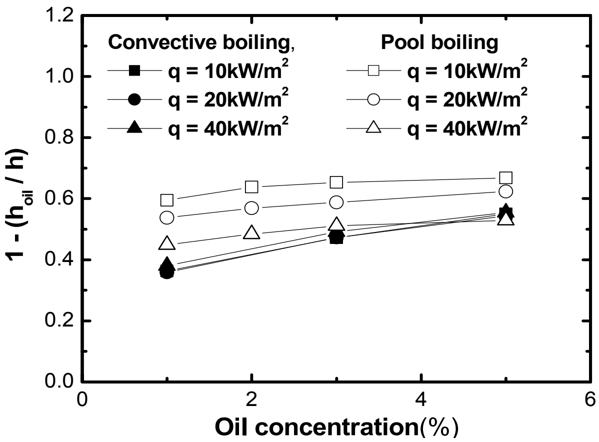


Fig. 8 Effect of oil on the convective boiling heat transfer reduction for R-123 at G = 8 kg/m^2s and T_{sat} = 26.7 □. Data compared with the R-123/oil pool boiling data by Kim et al. (2004)

CONCLUSIONS

In this study, the effect of oil on convective boiling of R-123 in an enhanced tube bundle was investigated at 26.7^{o}C saturation temperature. The enhanced tube had 0.23 mm pore diameter, which had been optimized using pure R-123. The effects of oil concentration (0 to 5%), heat flux (10 to 40 kW/m^2), mass velocity (8 to 26 kg/m^2 s) and vapor quality were investigated. The data are compared with the pool boiling counterpart.

1) The oil significantly reduces the bundle boiling heat transfer coefficient. With 1 % oil, the reduction is a pproximately 35%. Further addition of oil further reduces the heat transfer coefficient (55% reduction at 5% oil concentration).
2) The reduction in the heat transfer coefficient is smaller for the convective boiling than for the pool boiling, with larger difference at a smaller heat flux. The two-phase convection in a bundle may remove the oil-rich layer near the heating surface, and reduce the thermal degradation by oil.
3) Similar to pure R-123, the effects of mass velocity and vapor quality are negligible for the convective boiling of R-123/oil mixture.

ACKNOWLEDGEMENT

This research was conducted by the 2001 Research Grant from R&D Management Center for Energy and Resources of Korea.

NOMENCLATURE

A_{min}	minimum flow area in a bundle	(m^2)
d_p	pore diameter	(m)
c_{pr}	specific heat of the refrigerant	(kJ/kg K)
G	mass flux based on A_{min}	(kg/m^2s)
h	heat transfer coefficient	(W/m^2K)
H_t	tunnel height	(m)
$\dot{m}_r$	mass flow rate of the refrigerant	(kg/s)
N	number of active heaters upstream of the instrumented tube	(dimensionless)
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)
Q_p	heat supplied to the pre-heater	(W)
Q_H	Heat supplied to a single heater in the test section	(W)
$T_{p,in}$	refrigerant temperature at the pre-heater inlet	(K)
$T_{p,sat}$	saturation temperature at the pre-heater	(K)
T_r	pool temperature	(K)
T_{sat}	saturation temperature	(K)
T_w	tube wall temperature	(K)
x	vapor quality	(dimensionless)

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