

# Forced Convective Boiling of Refrigerant-Oil Mixtures in a Bundle of Enhanced Tubes Having Pores and Connecting Gaps

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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 pores (0.23 mm diameter) and connecting gaps (0.07 mm width), which had been optimized using pure R-123. The effects of oil concentration (0 to 5%), heat flux (10 to 40 kW/m<sup>2</sup>), mass velocity (8 to 26 kg/m<sup>2</sup>s) and vapor quality are investigated. 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 data are also compared with the pool boiling counterpart. 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. Similar to pure R-123 case, the effects of mass velocity and vapor quality are negligible for the convective boiling of R-123/oil mixture.

*Key words:* Enhanced tube, Bundle, Refrigerant-oil mixture, Convective boiling, Pore and connecting gaps

## Nomenclature

$c_{pr}$  : specific heat of refrigerant [kJ/kgK]  
 $d_p$  : pore diameter [mm]  
 $d_o$  : tube outer diameter [m]  
 $g$  : gap width [mm]  
 $G$  : mass flux [kg/m<sup>2</sup>s]  
 $h$  : heat transfer coefficient [W/m<sup>2</sup>K]  
 $H_t$  : tunnel height [mm]  
 $L$  : tube length [m]  
 $\dot{m}_r$  : mass flow rate of refrigerant [kg/s]  
 $N$  : number of tubes  
 $P_f$  : fin pitch [mm]  
 $P_{p1}$  : circumferential pore pitch [mm]  
 $P_{p2}$  : neighbouring pore pitch [mm]  
 $q$  : heat flux [W/m<sup>2</sup>]  
 $Q_H$  : heat supplied to a single tube [W]  
 $Q_p$  : heat supplied to pre-heater [W]  
 $T_{in}$  : inlet refrigerant temperature [K]  
 $T_w$  : tube wall temperature [K]  
 $T_{sat}$  : saturation temperature [K]

$W_t$  : tunnel width [mm]

$x$  : quality

## 1. 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 varies from 0.5% to 3%<sup>[1]</sup>. 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<sup>[2]</sup>; those having surface 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.

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Many single-tube pool boiling studies on structured enhanced tubes<sup>[3-7]</sup> 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<sup>[6, 7]</sup>. Limited studies on the oil effect in enhanced tube bundles include Gan et al.<sup>[8]</sup>, Marvillet<sup>[9]</sup>, Memory et al.<sup>[10]</sup>, Tataru and Payvar<sup>[11, 12]</sup>. Using a porous aluminum bundle with R-113 or R-113/R-11 mixtures, Gan et al.<sup>[8]</sup> reported that the heat transfer reduction was generally less in a bundle than that in a single tube. The convective effect in a bundle was thought to counter the influence the oil. Similar observation was made by Marvillet<sup>[9]</sup> for a porous aluminum tube bundle containing 84 tubes. Memory et al.<sup>[10]</sup> examined the R-114/oil convective boiling in structured (Turbo-B) or 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. The single-tube heat transfer reduction was less (10% at 10% oil concentration)<sup>[6]</sup>. This result contradicts with those by Gan et al.<sup>[8]</sup> and Marvillet<sup>[9]</sup>, where heat transfer reduction was less in a bundle configuration. Memory et al.<sup>[10]</sup> also reported 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). Tataru and Payvar<sup>[11, 12]</sup> 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 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). 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 reveal conflicting results on the relative magnitude of heat transfer degradation in a single-tube pool boiling with that in a bundle. Gan et al.<sup>[8]</sup> and Marvillet<sup>[9]</sup> report that the heat transfer degradation is less in a bundle configuration. However, Memory et al.<sup>[10]</sup> 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 had been optimized to yield the best convective boiling coefficient with pure R-123<sup>[13]</sup>, 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.

## 2. Sample tube

The same tube for the previous pool<sup>[2, 7]</sup> and convective boiling<sup>[13]</sup> test was used in this study. The surface geometry is shown in Fig. 1. The tube was 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. The pore size is represented by the diameter of a circle inscribed in a triangle pore. Detailed geometric dimensions of the tube is listed in Table 1.

## 3. Experimental apparatus

A schematic drawing of the apparatus is shown in Fig. 2, and the detailed drawing of the test section is shown in Fig. 3. The same apparatus and the test section had been used by Kim et al.<sup>[13]</sup> for convective

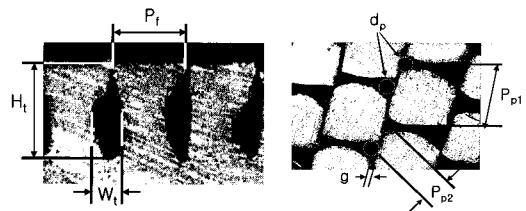


Fig. 1. Enlarged photos showing the present enhanced geometry ( $d_p = 0.23$ mm) (a) cross-sectional view (b) planar view.

Table 1. Geometric dimensions of the sample tubes.

geometry	dimension
$d_p$ (mm)	0.23
$g$ (mm)	0.07
$P_{p1}$ (mm)	0.71
$P_{p2}$ (mm)	0.384
$P_f$ (mm)	0.605
$H_t$ (mm)	0.54
$W_t$ (mm)	0.25

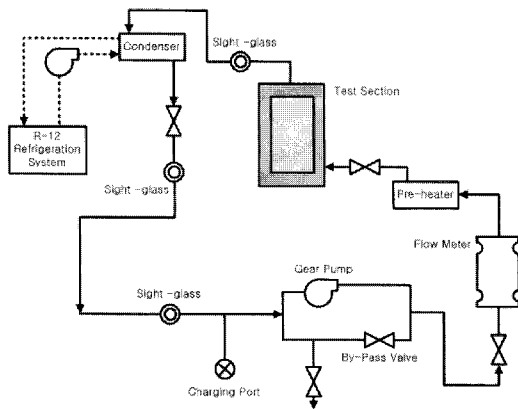


Fig. 2. Schematic drawing of the experimental apparatus.

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 from condensers 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 also 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.

Figs. 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.

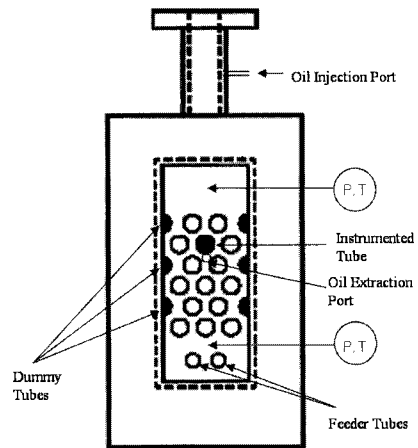


Fig. 3. Detailed drawing of the test section.

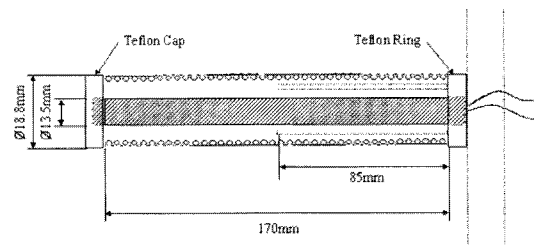


Fig. 4. Detailed drawing of the instrumented tube.

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.

The instrumented tube was located at the center of the fifth row from the bottom. Fig. 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°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°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<sup>2</sup>/s viscosity at 40°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°C.

The mass velocity was varied from 8 kg/m<sup>2</sup>s to 26 kg/m<sup>2</sup>s, heat flux from 10 kW/m<sup>2</sup> to 40 kW/m<sup>2</sup> and vapor quality from 0.1 to 0.9. The mass flux was determined based on minimum flow area across the bundle. The oil concentration was varied up to 5%. The vapor quality at the instrumented tube location was determined from Eq. (1).

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

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,

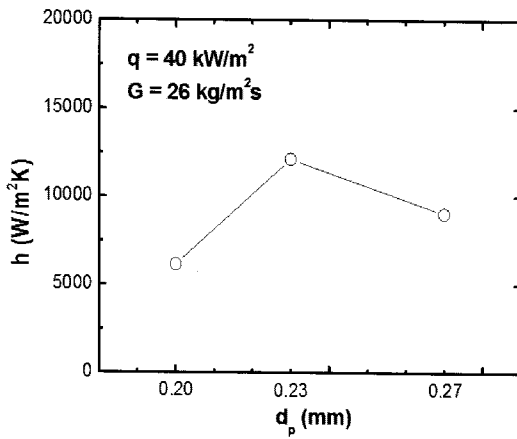
and  $N$  is the number of active heaters upstream of the instrumented tube.  $T_{sat}$  is the saturation temperature at the test section and  $T_{in}$  is the temperature at inlet of the pre-heater. The test section and the pre-heater were heavily insulated to minimize the heat loss to the environment.

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 ±0.1°C, heat flux measurement ±0.5%, mass velocity measurement ±1% and vapor quality measurement ±2%. An error analysis was conducted following Kline and McClintock<sup>[14]</sup>. The uncertainty in the heat transfer coefficient was estimated to be ±2% at the maximum heat flux (40 kW/m<sup>2</sup>) and ±8% at a low heat flux (10 kW/m<sup>2</sup>).

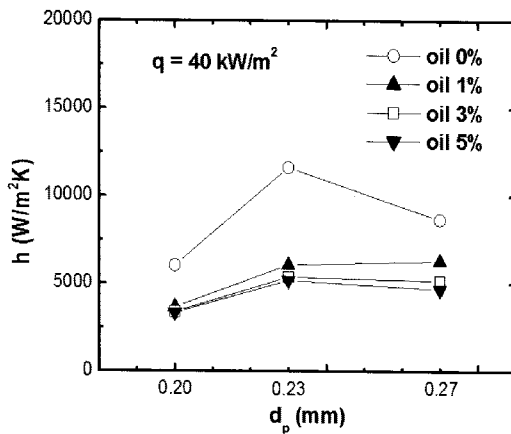
### 3. Results and discussions

The effect of pore diameter on convective boiling of pure R-123 at 26.7°C saturation temperature were investigated previously<sup>[13]</sup>, and are reproduced in Fig. 5. The convective boiling data are quality-average values. Also shown in Fig. 5(b) are the pool boiling heat transfer coefficients by addition of oil<sup>[7]</sup>. 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<sup>[13]</sup> revealed that the effects of mass velocity 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. Fig. 6(a) shows the heat transfer coefficients at two different mass fluxes



(a) Convective boiling



(b) Pool boiling

 Fig. 5. (a) Convective boiling data of pure R-123<sup>[13]</sup> (b) pool boiling data of R-123/oil<sup>[7]</sup> at  $T_{sat} = 26.7^\circ C$ .

( $G = 8 \text{ kg/m}^2\text{s}$  and  $26 \text{ kg/m}^2\text{s}$ ) at 5% oil concentration. The heat flux was  $10 \text{ kW/m}^2$ . The effects of mass velocity and vapor quality are negligible. The high heat flux data at  $40 \text{ kW/m}^2$  [Fig. 6(b)] show the same trend.

Fig. 7 shows the reduction of heat transfer coefficient by addition of oil at three different heat fluxes ( $q = 10, 20$  and  $40 \text{ kW/m}^2$ ). The mass flux is  $8 \text{ kg/m}^2\text{s}$ . These figures show that the addition of 1% oil significantly reduces the heat transfer coefficient. Approximately 35% reduction is observed with 1% oil. Further addition of oil further reduces the heat transfer (55% reduction at 5% oil concentration). The effect of vapor quality is negligible. In Figs. 6 and 7, the pure refrigerant data are limited to a relatively low quality due to limitation of the test apparatus. One

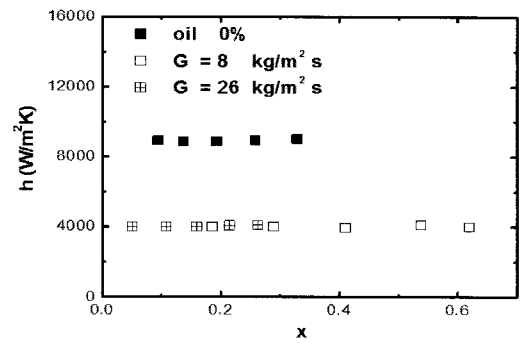
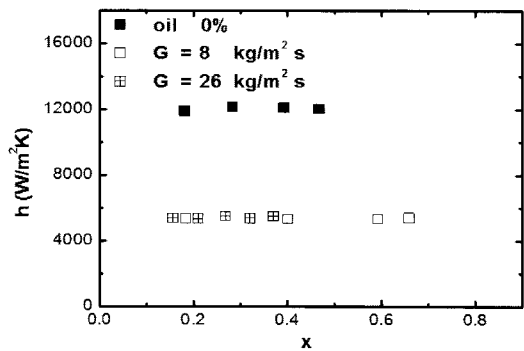
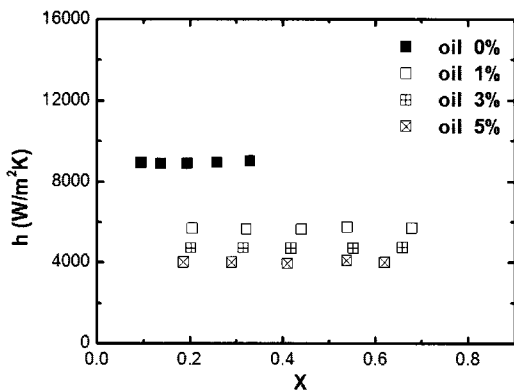
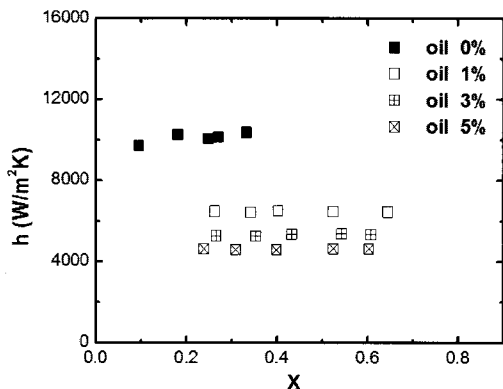

 (a)  $q = 10 \text{ kW/m}^2$ 

 (b)  $q = 40 \text{ kW/m}^2$ 

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

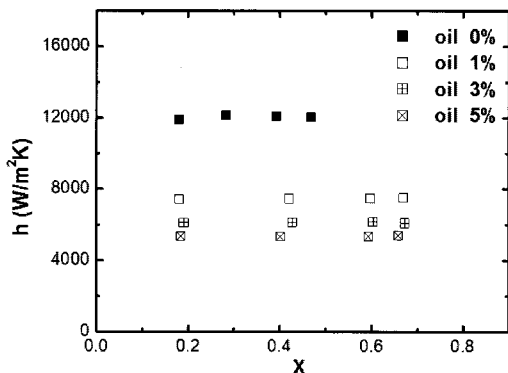
thing to be noted is that the pure R-123 data shown in Figs. 6 and 7 are those obtained by Kim et al.<sup>[13]</sup>. Once the apparatus was fouled with oil, it was almost impossible to thoroughly clean the apparatus to obtain pure refrigerant data. The reduction of heat transfer coefficient is re-plotted in Fig. 8 as a function of oil concentration. The plotted data are vapor-quality averaged values. Also shown in the figure are the single-tube pool boiling R-123/oil data obtained from the same tube<sup>[7]</sup>. The reduction in the heat transfer coefficient is smaller for the convective boiling as compared with the pool boiling. Fig. 8 also shows that the difference between the pool boiling and the convective boiling is larger at a smaller heat flux, where the convective effect is relatively large compared with the boiling effect. The two-phase convection in a bundle may remove the oil-rich layer near the heating surface, and reduce the thermal degradation by oil. Gan et al.<sup>[8]</sup> and Marvillet<sup>[9]</sup> also reported reduced thermal degradation in a bundle configuration.



(a)  $q=10 \text{ kW/m}^2$



(b)  $q=10 \text{ kW/m}^2$



(c)  $q=40 \text{ kW/m}^2$

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

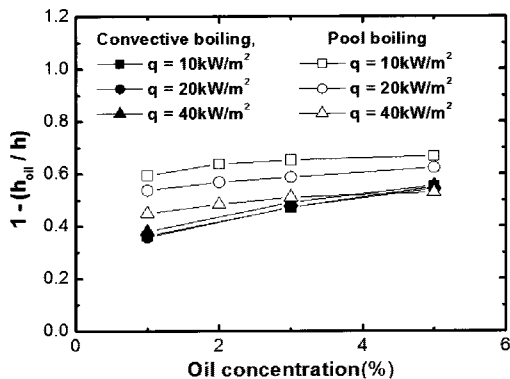


Fig. 8. Effect of oil on the convective boiling heat transfer reduction for R-123 at  $G = 8 \text{ kg/m}^2\text{s}$  and  $T_{\text{sat}}=26.7^\circ\text{C}$  Data compared with the R-123/oil pool boiling data by Kim et al.<sup>[7]</sup>.

**4. Conclusions**

In this study, the effect of oil on convective boiling of R-123 in an enhanced tube bundle was investigated at  $26.7^\circ\text{C}$  saturation temperature. The enhanced tube had  $0.23 \text{ mm}$  pore diameter, which had been optimized using pure R-123. The effects of oil concentration (0 to 5%), heat flux (10 to  $40 \text{ kW/m}^2$ ), mass velocity ( $8$  to  $26 \text{ kg/m}^2\text{s}$ ) 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 approximately 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**

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