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Pool boiling heat transfer of propane, isobutane and their mixtures on enhanced tubes with reentrant channels

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Abstract

Pool boiling heat transfer experiments were carried out on a conventional smooth tube and four enhanced tubes with reentrant surfaces using propane, isobutane and their mixtures as working fluids for six saturation temperatures. The heat transfer performance is very different for different surface–fluid combinations. Compared to the smooth tube, the mixture boiling heat transfer degradation is more significant for the enhanced tubes. The current data are compared with available literature data for the same fluids and also with data for R12 and R134a. Experimental results of boiling hysteresis and for twin-tube bundles are also provided. Further explanations for the different heat transfer performances is provided by means of visualization in an accompanying paper [Y. Chen, M. Groll, R. Mertz, R. Kulenovic, Visualization and mechanisms of pool boiling of propane, isobutane and their mixtures on enhanced tubes with reentrant channels, submitted to Int. J. Heat Mass Transfer (H/S 04016)].

Keywords: Pool boiling; Heat transfer; Mixtures; Propane/isobutane; Enhanced tube; Boiling hysteresis

1. Introduction

The role of chlorofluorocarbons (CFCs) in the process of ozone depletion and their contributions to global warming is now widely accepted. Hydrocarbons offer the possibility of a cheap, readily available and environmentally acceptable alternative to CFCs and CFC substances [2]. Propane and isobutane are widely used as a refrigerants [3]. Mixtures of propane and isobutane can be composed to obtain a saturation temperature/pressure curve as well as a cooling capacity very similar to those of R12, and have therefore been investigated as servicing or retrofitting refrigerants [2]. Investigations show a better COP for hydrocarbons than for R12 in refrigeration systems e.g. [4]. Though there are some published boiling heat transfer data for propane, the data for isobutane and propane/isobutane mixtures are scarce (refer Section 3.6), and it seems that no such data are available for enhanced surfaces.

The commercial enhanced boiling tubes may be categorized into three groups: *low-finned tubes* (e.g. Gewa-K), *modified finned tubes* (e.g. Gewa-T, Thermoexcel-E and Turbo-B) and *porous layer coated tubes* (e.g. High Flux) [5]. Generally, for pure fluids, the heat transfer performances of the High Flux, Thermoexcel-E and Turbo-B surfaces are better than those of the Gewa-T and Gewa-K surfaces, at least at low and moderate heat fluxes [5–9]. Modified finned tubes can be further divided into three groups: *tubes with narrow gaps* (e.g.

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Nomenclature

F_E	improvement factor [–]
F_{Mix}	degradation factor [–]
h	heat transfer coefficient [$W\,m^{-2}\,K^{-1}$]
h_1	heat transfer coefficient of the light component [$W\,m^{-2}\,K^{-1}$]
h_2	heat transfer coefficient of the heavy component [$W\,m^{-2}\,K^{-1}$]
h_{id}	ideal heat transfer coefficient [$W\,m^{-2}\,K^{-1}$]
q	heat flux [$W\,m^{-2}$]
T_m	measured averaged temperature of the thermometers [K]
T_S	saturation temperature [K]
T_W	wall temperature [K]
ΔT	wall superheat [K]
$\Delta T_{B,loc}$	local wall superheat of tube bundle [K]
ΔT_{id}	ideal wall superheat [K]
$\Delta T_{Sg,loc}$	local wall superheat of single tube [K]
x	liquid mass composition of light component [%]
Greek symbols	
ϕ	outer diameter of tube [m]
ϕ_{in}	inner diameter of tube [m]
λ	thermal conductivity of wall [$W\,m^{-1}\,K^{-1}$]

Gewa-T), *tubes with surface pores* (e.g. Thermoexcel-E) and *tubes with surface pores connected by gaps* (e.g. Turbo-B and the PB-tubes used in this work (Fig. 2)) [10].

For mixtures, the heat transfer performance can differ substantially depending on the kind of enhanced surface and on the mixture used. For the High Flux porous tubes, the boiling heat transfer coefficient of ethanol-water mixtures can be greater than the ideal value [11,12]. The reduced degradation suffered by the High Flux tube compared with the smooth tube, according to Thome [11], is the result of a liquid Prandtl number augmentation of the liquid-phase convection process inside the porous matrix, which partially counterbalances the negative mass transfer effects. For acetone/water mixtures, the High Flux tube has a sharp drop in performance with the addition of small concentrations of acetone, this drop is much greater than that for the smooth tube [5]. At higher concentrations, the performance of the High Flux tube increases substantially, while the smooth tube values remain low [5]. However, for R113–R11 mixtures, the High Flux tube suffers a much severer degradation than the smooth tube [13]. For finned tubes (e.g. Gewa-K), the reported data show a similar mixture effect as that for the smooth tubes [14]. The plain tube mixture boiling correlation predicted the finned tube mixture data fairly well [5].

Very limited data has been reported for boiling of mixtures on reentrant tubes. For boiling of R113–R11 mixtures, the Turbo-B surface suffers a much higher degree of heat transfer degradation than the smooth surface [13]. This is also true for boiling of R114–oil mixtures on the Turbo-B, Thermoexcel-HE and High Flux surfaces [8]. A surmise for the large degradation of mixture boiling heat transfer on the reentrant tubes is that there is a large buildup of the less volatile component next to the heated surface due to the inability of the free stream to affect the heat transfer process in the subsurface channels [13].

The main objective of this paper is to provide a comprehensive pool boiling database for propane, isobutane and their mixtures on smooth and reentrant enhanced tubes. In the accompanying paper [1], the boiling mechanisms, especially the mixture effects on the enhanced boiling, will be studied by means of visualization and by evaluating the bubble dynamics data, which provides partial explanations for the different heat transfer performances as will be shown in the present paper.

2. Introduction to experiments

The experimental setup is mainly composed of test vessel, test section, power supply unit, PC based data acquisition system and visualization equipment (Fig. 1). The test section consists of an outer tube with the test surface and a copper inner tube (16.02 mm in diameter) shrunk into the outer tube. The surface is heated by a heater cartridge. In the copper inner tube wall, four platinum resistance thermometers (PT100) are guided in four axial grooves, 90° apart, to different axial locations.

Pure propane N35, pure isobutane N35 and three propane and isobutane mixtures are used as working fluids. The mixtures have mass fractions of *propane* of **5**%, **50**% and **95**% which are called **5**–**95**% *mixture*, **50**–**50**% *mixture* and **95**–**5**% *mixture*, respectively. Table 1 shows the experimental saturation pressure corresponding to different saturation temperatures for the various fluids. These values are very close to those calculated by using a software developed by NIST [15].

Five kinds of carbon steel (ST35.8) tubes with outer diameter (ϕ) of about 19 mm and length of about 115 mm are used, one conventional smooth tube ($R_a = 1.1\ \mu m$) and four structured reentrant tubes with sub-surface channels and surface openings named PB1 to PB4 (Fig. 2), respectively. The mean values of the channel geometries are listed in Table 2.

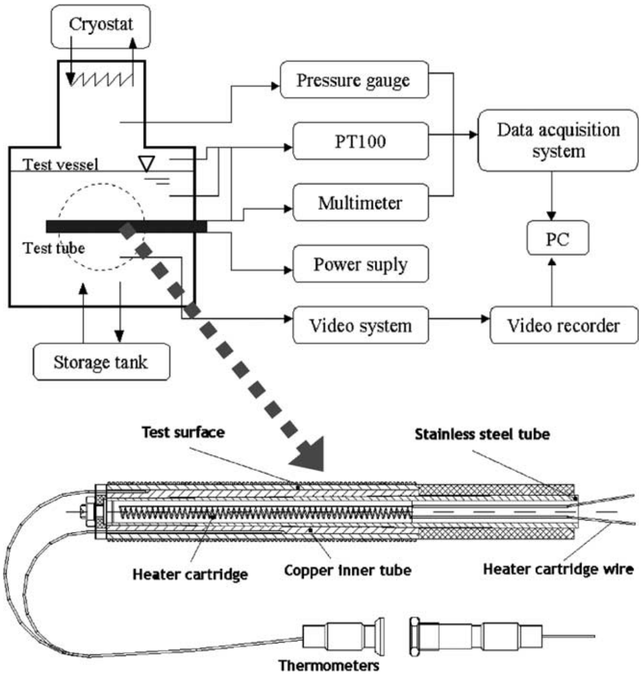


Fig. 1. Experimental setup and test section.

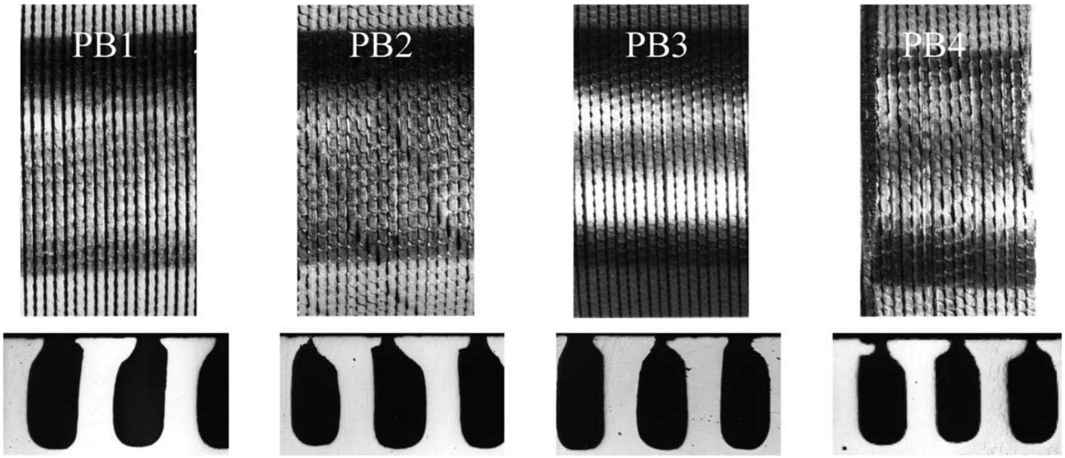


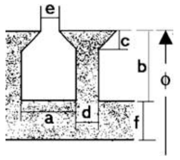
Fig. 2. Photos of enhanced tubes (upper: structured surface; lower: sub-surface channels).

Table 1
Saturation pressure (bar) for various fluids at given saturation temperature

T_S (K)	Isobutane	5–95% Mix.	50–50% Mix.	95–5% Mix.	Propane
243	0.47	0.50	1.10	1.60	1.67
253	0.72	0.79	1.65	2.35	2.44
263	1.08	1.18	2.36	3.33	3.45
273	1.55	1.70	3.31	4.63	4.74
283	2.17	2.44	4.55	6.25	6.37
293	3.00	3.30	5.97	8.17	8.36

Table 2
Geometric data of enhanced structures (mm, refer to the schematic cut-view on the r.h.s.)

	ϕ	a	b	c	d	e	f	Deviation of "e" (%)
PB1	19.22	0.279	0.694	0.169	0.220	0.164	0.812	12.8
PB2	19.14	0.305	0.686	0.205	0.218	0.081	0.794	32.3
PB3	19.16	0.290	0.679	0.168	0.228	0.076	0.811	47.5
PB4	19.17	0.299	0.711	0.184	0.209	0.062	0.862	38.4



Single tube as well as twin-tube experiments are carried out under saturation condition. For the twin-tube experiments, the two tubes are located horizontally with one above the other with the center to center distance of 24.8 mm. The chosen saturation temperatures for the experiments range from 243 to 293 K with an interval of 10 K. The heat flux ranges from about 1 to 100 kW/m^2. For each saturation temperature, the heat flux is first set to go up then go down, step by step, with a time interval of about 10 min. To ensure that the experimental procedure is correct, for each liquid-surface combination, experiments are repeated at least for one saturation temperature.

The mean heat transfer coefficient h is calculated by

$$h = q/(T_W - T_S) \tag{1}$$

where, the heat flux q is calculated from the total power input, T_S is the saturation temperature and T_W is the average surface temperature of the tube wall which can be calculated from the measured average temperature of the thermometers, T_m, by using Fourier's law, as

$$T_W = T_m - \frac{q\phi \ln(\phi/\phi_{in})}{2\lambda_L} \tag{2}$$

with $T_m = \frac{1}{4}\sum_{i=1}^{4} T_{m,i}$.

Here, ϕ_{in} is the inner diameter of the tube (16 mm), λ is the thermal conductivity of the wall. For the enhanced tubes, heat flux and wall temperature are measured at the base of the sub-surface channels. That means, ϕ in Eq. (2) refers to the diameter measured at the channel base. Due to the thermal contact resistance between the test tube and the inner tube, the measured temperature T_m is somewhat higher than the actual value at the inner surface of the test tube. However, it is estimated that the difference is less than 2% of the wall superheat.

The uncertainty of temperature measurement is ±0.1 K. The relative uncertainty of heat flux is 4.3% for $q = 2$ kW/m^2 and 1.9% for $q = 100$ kW/m^2. The uncertainty of heat transfer coefficient is calculated based on the proposed formula by Kline and McClintock [16]. For the smooth tube, the wall superheat is relatively big, the relative uncertainty of the heat transfer coefficient is ranging from about 1.3% to 10%. However, for enhanced tubes, such as PB3 (with isobutane as working fluid), the wall superheat is small, the relative uncertainty of the heat transfer coefficient can be up to 100% at low heat fluxes, and can be over 10% even at relatively high heat fluxes.

3. Results and discussion

3.1. Boiling hysteresis

Fig. 3 show the boiling curves obtained by first increasing and then decreasing the heat flux with the start point indicated by an arrow. Boiling hysteresis was found for all surfaces and it is much more pronounced during boiling of the 50–50% mixture than the pure fluids for each surface. For the various tubes used, the smooth tube has the highest temperature overshoot (short: T-overshoot). Almost no T-overshoot is seen for boiling of isobutane on PB3 and also for boiling of propane on PB4. For the 50–50% mixture, the

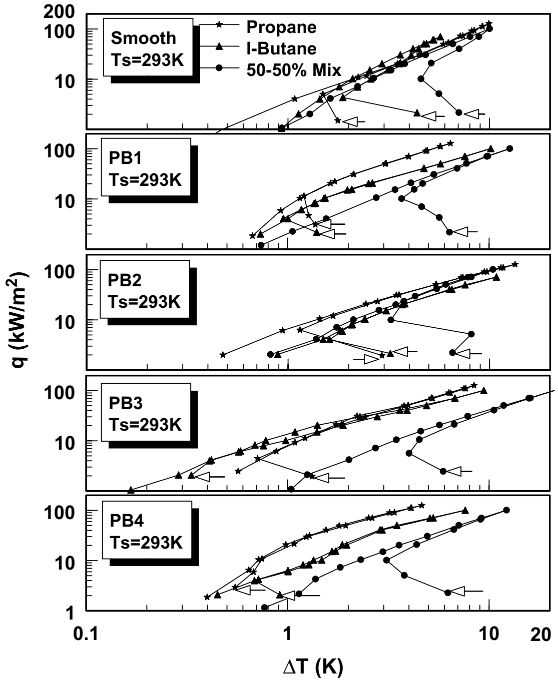


Fig. 3. Boiling curves for ascending and descending heat fluxes.

T-overshoot of PB2 disappears at $q > 20$ kW/m^2, while for other tubes small T-overshoots exist at a much higher heat flux. It is interesting to see that, in general, PB2, PB3 and PB4 are the best performing tube (descending heat flux) for the 50–50% mixture, isobutane and propane, respectively (see next section). Thus, it seems that, the lower the T-overshoot, the higher the heat transfer performance. This is also supported by some other data [6,17].

Big T-overshoot (from 4 to 15 K) was found by Marto and Lepere [6] for boiling of R113 and FC-72 on the High Flux, Thermoexcel-E and Gewa-T tubes. The best performing tube (High Flux) has the lowest T-overshoot and vice versa. By fitting a shroud over the Gewa-T tube with openings at the top and bottom, the heat transfer performance is greatly improved and the T-overshoot is greatly reduced [17]. Trewin et al. [13] showed a T-overshoot as high as 10 K for the small-porosity High Flux surface in R113, while almost no T-overshoot was found for the medium-porosity High Flux and the Turbo-B surfaces. In this case, though the Turbo-B surface shows a higher heat transfer performance than the small-porosity High Flux surface at low heat fluxes, the latter performs better than the medium-porosity High Flux. The smooth surface shows a moderate T-overshoot at $q < 8$ kW/m^2. The lack of a big T-overshoot for the Turbo-B surface was attributed to the interconnected spiraling channel which facilitates vapor to travel circumferentially around the tube [13]. Marto and Lepere [6] attributed the large initiation superheat for the Gewa-T surface to the big openings between the fins which facilitates the flooding of the channels.

The above explanations are incomplete, since the results from Trewin et al. [13] and Marto and Lepere [6] are in conflict with those from Kedzierski [9]. For R123, Kedzierski [9] found a noticeable T-overshoot for the Turbo-BII-LP surface for $q < 60$ kW/m^2 by fluid heating, while almost no T-overshoot was found for the Gewa-K surface. For electrical heating, a small T-overshoot was found for the High Flux surface, and it was also not found for the Gewa-K surface. Apparently, for enhanced surfaces, boiling hysteresis is also strongly influenced by different surface–fluid combinations.

The much higher initiation superheat required by mixtures as shown in Fig. 3 was also found by Thome and his coworkers [12,18] during boiling of ethanol–benzene mixtures on a smooth and a High Flux tube. Shakir and Thome [18] attributed the positive deviation in the initiation superheat to the effects of supersaturation and mass diffusion on trapped vapor nuclei. Since, when the heat to the surface is shut off, the vapor nucleus remaining in a cavity is supersaturated with the less volatile component (with respect to its equilibrium composition), and therefore this component partially condenses which results in a smaller nucleus than it would be for a pure fluid.

3.2. Heat transfer coefficient

Experimental results of five tubes are presented in Figs. 4–8 as h vs. q for different fluids at six saturation temperatures. In general, h increases with q and T_S. For the smooth tube, the linear curves are seen in the logarithmic plots (Fig. 4). For the enhanced tubes, the

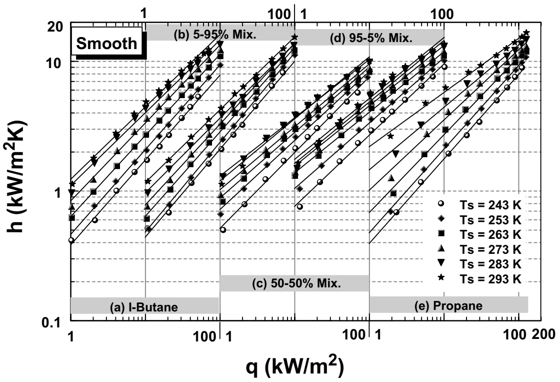


Fig. 4. Heat transfer coefficient for the smooth tube.

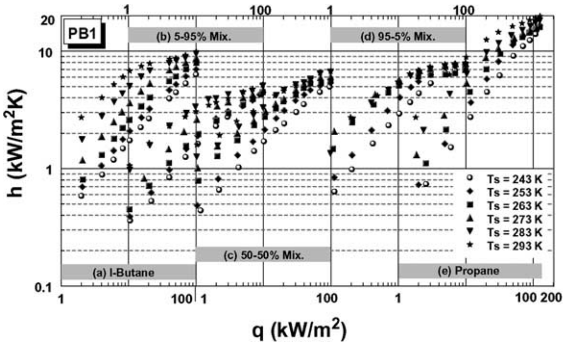


Fig. 5. Heat transfer coefficient for the enhanced tube PB1.

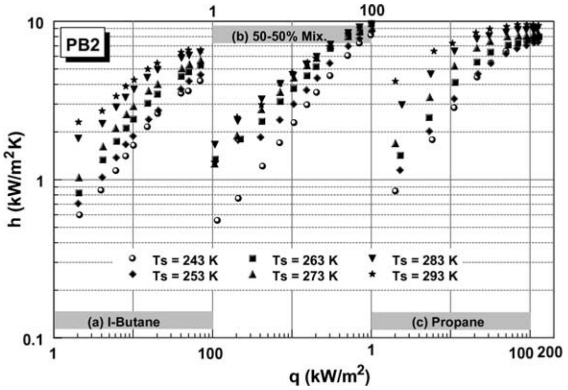


Fig. 6. Heat transfer coefficient for the enhanced tube PB2.

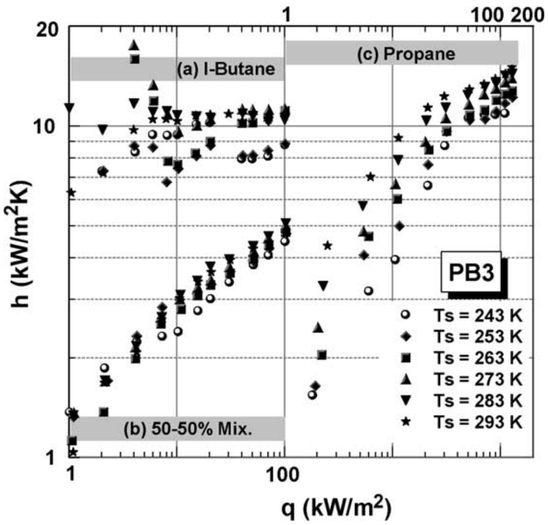


Fig. 7. Heat transfer coefficient for the enhanced tube PB3.

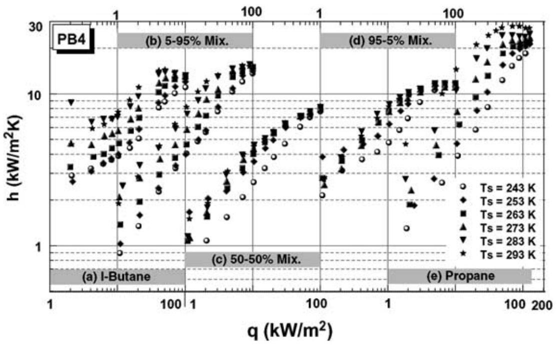


Fig. 8. Heat transfer coefficient for the enhanced tube PB4.

curves are generally not linear. At a heat flux of about 20 to 50 kW/m^2, depending on different surface–fluid combinations, the slope of the curves starts to become smaller, for the tube PB4 in case of pure fluids, especially at elevated temperature, even to decrease (Fig. 8). This phenomenon was also found for other reentrant tubes e.g. [19].

The heat transfer performances are very different for different surface–fluid combinations. While PB1 and PB2 show a modest performance (Figs. 5 and 6), PB4 has rather high heat transfer coefficients, especially for the pure components and the 5–95% mixture (Fig. 8). PB3 shows a surprising performance for boiling of isobutane (Fig. 7), as h remains essentially constant (at around 7–11 kW/m^2, which is extremely high for low heat fluxes) with increasing q. Two additional repeated experiments have confirmed this behavior. This cannot be attributed to the large measurement errors mentioned in Section 2.4 for low heat fluxes. It is related to the physical properties of isobutane and the surface geometry of PB3 (refer [1]). However, PB3 performs rather poor for the 50–50% mixture ($h < 5$ kW/m^{2}K), especially at high heat fluxes.

The heat transfer coefficient tends to increase as the pressure increases. This is caused by the increase in site density, which is due to a decrease in the equilibrium superheat required for boiling site activation with increasing pressure. For the mixtures, the heat transfer coefficients are generally lower than those of the pure components and depend less on pressure. Especially at high heat fluxes ($q > 40$ kW/m^2) and for the enhanced tubes, the curves for different saturation temperatures come closer together. This is due to the fact that, the degradation in heat transfer for mixtures is not independent on pressure, rather, it becomes more pronounced as the pressure increases. Therefore, it is not possible for a mixture to have a band width between two constant pressure curves (shown in the h–q chart) equal or bigger than that for the pure components. This effect of pressure has three possible explanations: (1) as the pressure increases, the extent of turbulent motion of liquid near the heating surface decreases due to reduction in bubble size and density difference between liquid and vapor phases, which results in a decrease in mass transfer rate [20]; (2) the increase of site density resulting from increasing pressure leads to a smaller area for the diffusion of the light component from the bulk to the heating surface [21]; (3) since the overall wall superheat decreases rapidly with increasing pressure, if assuming a constant local rise in the boiling point ($\Delta\theta$), then $h/h_{id} = \Delta T_{id}/\Delta T = \Delta T_{id}/(\Delta T_{id} + \Delta\theta)$, still decreases [22].

In fact, the effect of pressure can be so significant that the heat transfer coefficient even decreases with increasing pressure. For example, Jungnickel et al. [23] showed that, for the R13/R12 mixture with 75% mole R12 at $q = 60$ kW/m^2, h is 4.6 kW/m^{2}K for $p_S = 2$ bar, which is much higher than 2 kW/m^{2}K for $p_S = 20$ bar. However, for $q < 13$ kW/m^2, h still increases with pressure. This abnormal behavior can also be seen in Figs. 4–8, e.g. in many cases, the mixture heat transfer coefficients for $T_S = 283$ K are slightly higher than those for $T_S = 293$ K.

3.3. Degradation of mixture boiling heat transfer

The degradation factor, F_{Mix}, is defined here as

$$F_{\mathrm{Mix}} = \frac{h}{h_{\mathrm{id}}} = \frac{\Delta T_{\mathrm{id}}}{\Delta T} = \frac{h[xh_2 + (1-x)h_1]}{h_1 h_2} \qquad (3)$$

where x is the mass fraction of light component, h is the mixture heat transfer coefficient; subscript id refers "ideal"; 1 and 2 refer the light and heavy component, respectively; ΔT_{id} is the ideal wall superheat which is defined as the linear interpolation of the wall superheats of two pure components with respect to the mass fraction [24].

Fig. 9a–c show the measured relationships between F_{Mix} and q at $T_{\mathrm{S}} = 243$ and 293 K for the three mixtures. In general, F_{Mix} is lower than unity, viz. $h < h_{\mathrm{id}}$. However, for certain circumstances, especially in case of low saturation temperature and for the smooth tube, F_{Mix} can be greater than unity. This behavior was also found for boiling of *n*-propanol/water [25], ethanol/water mixtures [26] and for the same mixtures as used in this study [27], while no explanation was given. F_{Mix} is generally much higher for $T_{\mathrm{S}} = 243$ K than for $T_{\mathrm{S}} = 293$ K. This effect of saturation pressure has been already explained in the former section. The heat flux has a complicated influence on F_{Mix}. However, for the 95–5% mixture (Fig. 9c), F_{Mix} decreases with increasing heat flux (not for the smooth tube at $T_{\mathrm{S}} = 293$ K).

From the surfaces tested, the performances of the smooth tube and tube PB2 for mixtures are relatively good compared with the pure components. For PB1, 3 and 4, F_{Mix} is generally much lower than that for the smooth tube. The extent of degradation for a specific mixture is quite different for different enhanced surfaces. Here, examples are given for $T_{\mathrm{S}} = 293$ K (solid symbols in Fig. 9). For the 5–95% mixture (Fig. 9a), essentially no degradation is found for PB4 except at low heat fluxes, while for PB1 the degradation factor is around 0.5. For the 95–5% mixture (Fig. 9c), both PB1 and PB4 have a similar degradation factor ranging from about 0.8 at $q = 2$ kW/m^2 to 0.5 at $q = 100$ kW/m^2. For the 50–50% mixture (Fig. 9b), PB2 suffers the least heat transfer degradation with F_{Mix} of about 0.85 for $q < 20$ kW/m^2 and up to 1.2 at $q = 100$ kW/m^2. PB1, 3 and 4 have a F_{Mix} ranging from about 0.3 to 0.5 for most of the heat flux range.

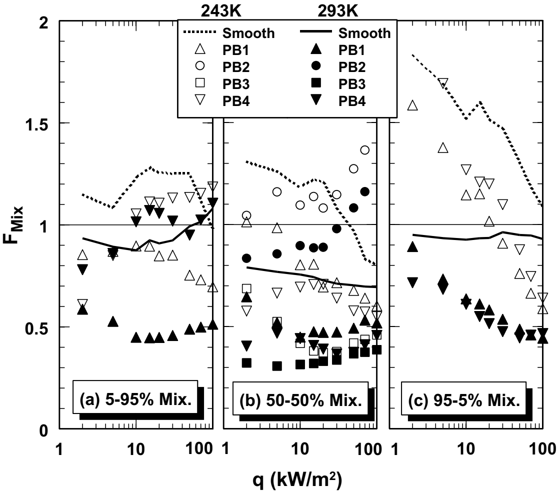


Fig. 9. Degradation factors of three mixtures for different surfaces.

3.4. Improvement factors for the enhanced tubes

Fig. 10a–d show the improvement factors F_{E}, defined as the ratio of heat transfer coefficients of enhanced and smooth tubes, for four enhanced tubes at two saturation temperatures (243 and 293 K). F_{E} is generally significantly higher for the pure fluids (symbols connected by lines) than for the mixtures (symbols). In case of high heat fluxes, and especially for the mixtures, the heat transfer coefficients of the enhanced tubes can be even lower than those of the smooth tube, viz. $F_{\mathrm{E}} < 1$. This is related to the higher degree of heat transfer degradation for the enhanced tubes than for the smooth tube.

For a specific surface, F_{E} strongly depends on the tested fluids and also working conditions. For **PB1** (Fig. 10a), F_{E} up to 1.8 is found for propane. For isobutane improvements can be seen for $T_{\mathrm{S}} = 293$ K and $q < 30$ kW/m^2; a further increase of q leads to F_{E} lower than unity (down to about 0.7 at $q = 100$ kW/m^2). Deterioration is shown for all mixtures with F_{E} down to about 0.4 for the 5–95% mixture. For **PB2** (Fig. 10b), F_{E} of about 1.5 is found for the 50–50% mixture and propane at small heat fluxes ($q < 2$ kW/m^2). However, for both pure components, especially isobutane, F_{E} can be down to about 0.5 as the heat flux approaches 100 kW/m^2. For **PB3** (Fig. 10c), a very high improvement factor (up to 12) is achieved for isobutane at low heat fluxes; it decreases rapidly with increasing heat flux, F_{E} becomes slightly less than unity at $q \approx 100$ kW/m^2. A

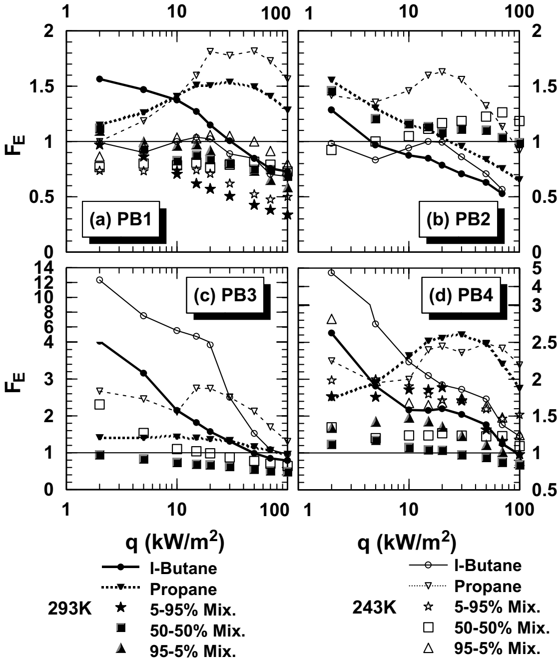


Fig. 10. Improvement factor for four enhanced tubes.

moderate improvement is found for propane. The 50–50% mixture has F_E less than unity for $q > 10$ kW/m^2, and it goes down to about 0.5 at high heat fluxes. Best performance is obtained for **PB4** (Fig. 10d) with F_E ranging from about 1.7–2.6 for propane, 1–5 for isobutane, 1–2 for the 5–95% and 95–5% mixtures. The only deterioration for PB4 is for the 50–50% mixture starting at $q = 30$ kW/m^2, where F_E is about unity and goes down to 0.85 at $q = 100$ kW/m^2.

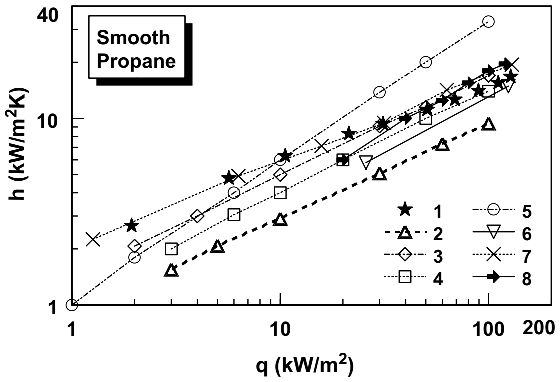
3.5. Comparison with literature data

Fig. 11 shows the comparison of heat transfer performance for boiling of propane on the smooth tube with literature data given in [27–30,32]. These data come close to each other with two exceptions. One is the data for a stainless steel tube (curve no. 2 [28]) which show the poorest heat transfer performance due to its low thermal conductivity (14.3 W/mK). The other is shown as curve no. 5 [29] for boiling on a small-diameter copper tube which has the highest heat transfer coefficient for $q > 10$ kW/m^2. The present data (curve no. 1) show a relatively high h among the data shown, especially at low heat fluxes, which might be due to the high surface roughness.

The published data on boiling of isobutane are very limited. Fig. 12 shows the comparison of heat transfer performance for boiling of isobutane on smooth surfaces. The present data (curve no. 1) is much higher than those from [31] (curve no. 3) for boiling on a copper tube which has a very low surface roughness, while it is close to the data from [27] (curve no. 4) for a flat surface (note that the saturation temperature for the latter case is 347.6 K). Curve no. 2 shows the measurements made about 10 months earlier than curve no. 1 on the same tube, but with isobutane N25 (instead of N35) and using type K thermocouples (instead of PT100). The agreement of the two experiments is relatively good.

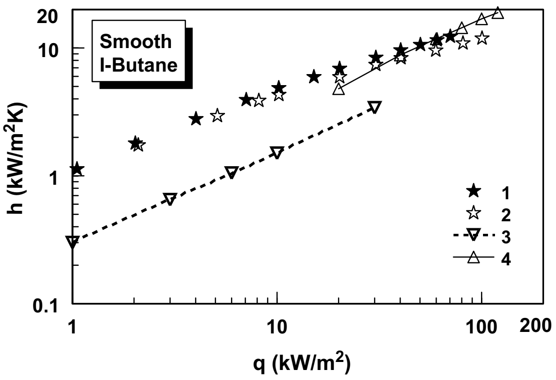
Fig. 13 shows the comparison of the degradation factor for boiling of propane/isobutane mixtures on the smooth surface with the data given in [27]. Note that the present data (solid symbols) are calculated based on the same saturation temperature (293 K), while the data from [27] (hollow symbols) are based on the same saturation pressure (12 bar). The degradation factor from [27] is lower than the present one, which is due to the higher system pressure for the former. The two sets of data show that, for low composition of propane, F_{Mix} is close to unity. With increasing mass fraction of propane (x), F_{Mix} decreases, until at a certain value (around 0.5), F_{Mix} reaches its lowest level and then increases with further increasing of mass fraction of propane.

Fig. 14 shows the comparison of heat transfer performance of the enhanced tubes PB1-4 with that of the Gewa-T [28], Gewa-K [28], Gewa-TX [29] and finned tube [32] for boiling of propane. The thermal design curve given by O'Neill et al. [33] for the High Flux



No.	Reference	μ(mm)	Material	Ra (ϕm)	T_S (K)
1	this work	19	carbon steel	1.1	293
2	Mertz [28]	20	stainless steel	0.48	293
3	Mertz [28]	19	CuNi	0.35	293
4	Mertz [28]	19	Cu	0.45	293
5	Gorenflo et al. [29]	8	Cu	0.55	293
6	Myers & Katz [32]	19	Cu		274.3
7	Clements & Colver [30]	flat surface	gold-plate		310.4
8	Cioulachtjian & Lallemand [27]	flat surface	Cu	0.35	307.8

Fig. 11. Comparison of heat transfer performance with literature data for boiling of propane on smooth surfaces.



No.	Reference	ϕ (mm)	Material	Ra (μm)	T_S (K)
1	this work	19	carbon steel	1.1	293
2	this work	19	carbon steel	1.1	293
3	Hsieh & Yang [31]	20	Cu	0.05	291
4	Cioulachtjian & Lallemand [27]	flat surface	Cu	0.35	347.6

Fig. 12. Comparison of heat transfer performance with literature data for boiling of isobutane on smooth surfaces.

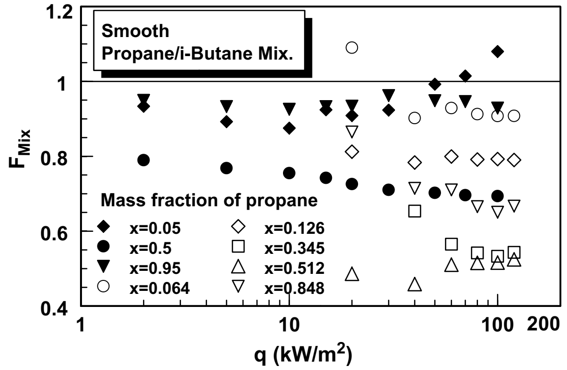


Fig. 13. Comparison of degradation factor with literature data (hollow symbols) for boiling of propane/isobutane mixtures on smooth surfaces.

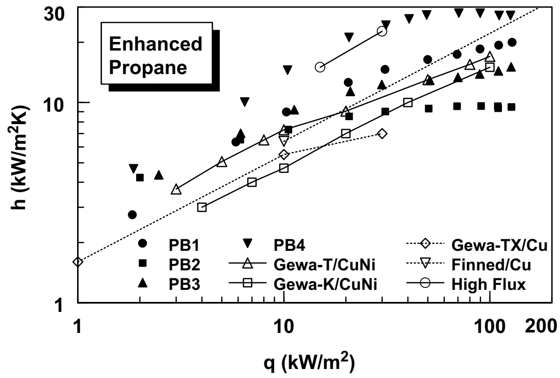


Fig. 14. Comparison of heat transfer performance with literature data for boiling of propane on enhanced surfaces.

surface is also shown. Overall, the PB-tubes outperform the Gewa- and finned tubes at relatively low heat fluxes (while the Gewa- and finned tubes perform consistently better with increasing heat fluxes). PB4 has a heat transfer performance similar to High Flux, which is significantly better than other enhanced surfaces.

The performances of the smooth and enhanced tube PB4 in propane, isobutane and the 50–50% mixture are compared to the data of R12 and R134a given by Webb and Pais [7] for copper tubes (17.5–19.1 mm in diameter) at $T_S = 299.85$ K, as shown in Fig. 15a and b. For the smooth surfaces (Fig. 15a), the performance of the hydrocarbons is generally better than that of R12 and R134a especially for the pure components. The performance of the 50–50% mixture begins to be lower than R12 and R134a for q higher than about 20 kW/m^2. For the enhanced tubes (Fig. 15b), while the performance of PB4 greatly depends on different hydrocarbons, the performance of Turbo-B as well as Gewa-TX is similar for R12 and R134a. In general, Turbo-B performs better than PB4/isobutane, but worse than PB4/propane. For pure fluids, Gewa-TX shows the lowest performance which, however, is still much better than that for PB4 in the 50–50% mixture.

3.6. Twin-tube performance

Experimental results of twin-tube bundles are available only for three mixtures on the smooth tube and tube PB4, and isobutane on PB4. Fig. 16 shows the comparison of the bundle factor F_B, which is defined as the ratio of the upper tube heat transfer coefficient to the heat transfer coefficient of the same tube without

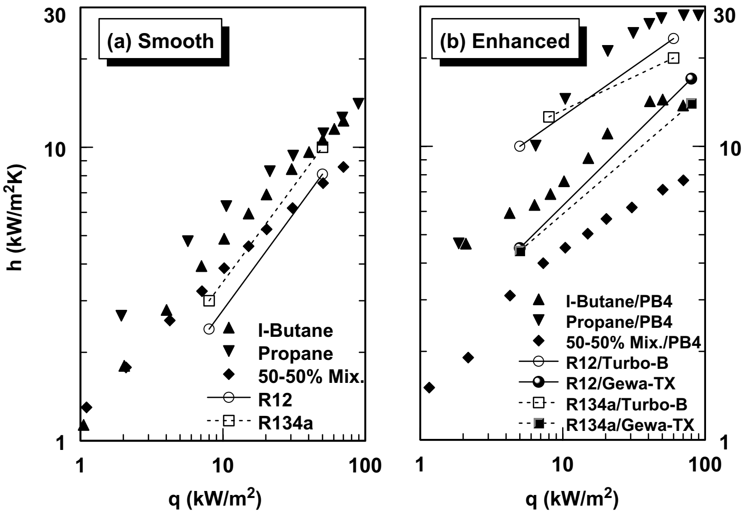


Fig. 15. Comparison of heat transfer performance of the smooth and enhanced tube PB4 in propane, isobutane and the 50–50% mixture with the data of R12 and R134a given in [7].

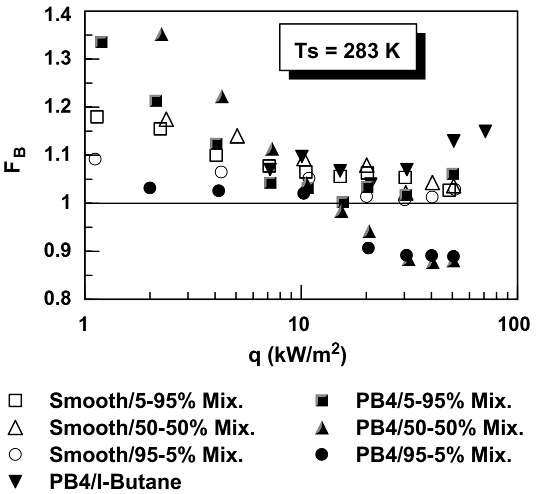


Fig. 16. Comparison of bundle factors for isobutane and three mixtures on smooth tube and enhanced tube PB4 at $T_{\mathrm{S}} = 283$ K.

onflow.[1] For the smooth tube, F_{B} ranges from 1 to 1.2 and it decreases with increasing heat flux. The 5–95% mixture has a bundle factor slightly higher than the 95–5% mixture, but slightly lower than the 50–50% mixture. For PB4, F_{B} is higher than unity for isobutane and the 5–95% mixture with a minimum at $q \approx 20\ \mathrm{kW/m^2}$. For the 50–50% and 95–5% mixtures, F_{B} decreases continuously with increasing heat flux until about $q = 30\ \mathrm{kW/m^2}$, where F_{B} is about 0.9, and becomes constant. For both smooth and enhanced tubes, F_{B} decreases with increasing system pressure (not shown).

To understand the twin-tube performance, comparison is made between the local wall superheats of a tube with ($\Delta T_{\mathrm{B,loc}}$) and without ($\Delta T_{\mathrm{Sg,loc}}$) heating the other tube. An example is given in Fig. 17 as the ratio of $\Delta T_{\mathrm{B,\ loc}}$ and $\Delta T_{\mathrm{Sg,loc}}$ against q for the upper and lower tubes at four positions (bottom, left, top and right) during boiling of the 50–50% mixture on PB4. In general, with increasing heat flux, the ratio $\Delta T_{\mathrm{B,loc}}/\Delta T_{\mathrm{Sg,loc}}$ increases from a value less than unity to one greater than unity. This means that the local heat transfer coefficient of both upper and lower tubes of the bundle is improved at low heat fluxes compared with the single tube performance, while it is deteriorated at high heat fluxes.

For a given heat flux, $\Delta T_{\mathrm{B,loc}}/\Delta T_{\mathrm{Sg,loc}}$ is generally lower for the upper tube (hollow symbols) than for the lower tube (solid symbols), which means that the improvement of heat transfer for the upper tube by heating the lower tube is more efficient than that for the lower tube by heating the upper tube. This is obviously due to the fact that bubbly flow moves upwards driven by buoyancy force. Thus the wall superheat on the bottom surface of the upper tube (with the lower tube heated) is the lowest due to the direct impingement of the onflow from the lower tube, while the top surface has the highest wall superheat and the wall superheats on the two side walls lie in between.

Fig. 17 also shows that heating the upper tube has a positive effect on the heat transfer of the lower tube at relatively low heat fluxes. The deterioration of heat transfer of a tube ($\Delta T_{\mathrm{B,loc}}/\Delta T_{\mathrm{Sg,loc}} > 1$) with heating of the other

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[1] This definition is slightly different from that defined in [34], where the bundle factor is defined as the ratio of average heat transfer coefficient for the whole bundle to that of a single isolated tube of similar surface.

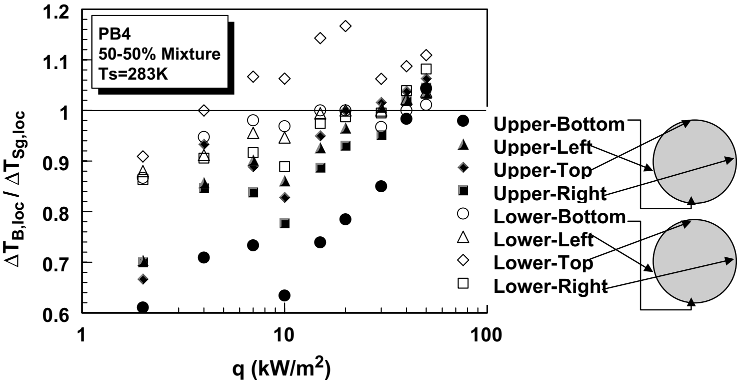

Fig. 17. Ratio of local wall superheat with the other tube heated ($\Delta T_{B,loc}$) and without the other tube heated ($\Delta T_{Sg,loc}$).

one at relatively high heat fluxes is probably due to mixture effects, because the liquid around the tubes will be richer with less volatile component for the case that both tubes are heated than for the case that only a single tube is heated, especially in the region between the two tubes where, bubbles are so dense that diffusion of the light component is greatly retarded. This is indicated by a much higher ratio $\Delta T_{B,loc}/\Delta T_{Sg,loc}$ on the top surface of the lower tube than on the other three positions.

The present results are in agreement with data from the literature [32,34,35]. Memory et al. [34] conducted experiments for boiling of R114 and R114/oil mixtures on multi-tube bundles for both smooth and enhanced surfaces ($\phi = 15.9$ mm, gap: 3.2 mm). For pure R114 at $q = 5$ kW/m^2, F_B is 1.52, 1.79 and 1.29 for the smooth, finned and structured tube bundle, respectively; F_B decreases to unity at $q = 50$ kW/m^2 for all bundles. The higher heat transfer performance for the upper tubes was attributed to convection effects as well as bubbles from lower tubes impinging on, and sliding around the upper tubes, and secondary nucleation which occurs where sliding bubbles are present [34]. Convection effects were thought to be dominant for the multi-tube bundle [34]. With increasing heat fluxes, boiling from the surface itself begins to dominate, diminishing other mechanisms. This happens for structured reentrant surfaces at lower heat fluxes. For a porous tube bundle, a bundle factor between 1.06 and 0.7 was found, which was attributed to the high vapor quality around the top tubes [34]. F_B less than unity was also reported in [35] for a finned tube bundle at heat fluxes greater than 30 kW/m^2 with R11 as working fluid.

4. Conclusions

(1) Boiling hysteresis is found for all surfaces. The temperature overshoot for the smooth tube is generally higher than that for the enhanced tubes. The mixtures show much stronger hysteresis than the pure components. The data generally show that, the lower the temperature overshoot, the higher the heat transfer performance.

(2) Compared with the smooth tube, the enhanced tubes generally show different heat transfer improvements depending on the specific enhanced surface, working fluid and experimental conditions. However, at high heat fluxes, and especially for the mixtures, the heat transfer coefficients of the enhanced tubes can be even lower than those of the smooth tube.

(3) In boiling of mixtures, the enhanced tubes suffer much more pronounced heat transfer degradation than the smooth tube.

(4) Among the four enhanced tubes, the heat transfer performance of PB1 is relatively good for propane, but poor for isobutane and mixtures; PB2 is the best for the 50–50% mixture, but the worst for propane and isobutane; PB3 shows very high heat transfer performance at $q < 20$ kW/m^2, but the worst for the 50–50% mixture and a poor one for propane; PB4 is the best for propane, the 5–95% and 95–5% mixtures and also for isobutane at $q > 20$ kW/m^2.

(5) For the twin-tube experiments, a bundle factor up to 1.3 is found. However, for the enhanced tubes at relatively high heat fluxes, the bundle factor can be lower than unity during boiling of mixtures which is mainly attributed to mixture effects.

(6) Boiling of hydrocarbons on the smooth surface generally has a higher heat transfer coefficient than for R12 and R134a, especially for the pure fluids. For the mixtures, the performance of the enhanced tubes can be much poorer than for boiling of R12 and R134a on Gewa-TX and Trubo-B surfaces.

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