TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

No. 123, 2011, 85–98

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Nucleate pool boiling heat transfer from small horizontal smooth tube bundles

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Abstract

Measurements of boiling heat transfer coefficients in water, methanol and refrigerant R141b are reported for the bundles of smooth tubes that represent a portion of a flooded-type evaporator. Each bundle contained 19 instrumented, electrically heated tubes in a staggered triangular-pitch layout. The effect of heat flux, tube pitch and operating pressure is studied in the paper. Bundle factor and bundle effect are discussed as well. A correlation for prediction of a bundle average heat transfer coefficient is proposed.

Keywords: Boiling heat transfer; Smooth tube bundle

Nomenclature

D	–	diameter, m
g	–	acceleration due to gravity, m/s^2
I	–	current, A
L	–	active tube length, m
p	–	pressure, N/m^2
P	–	electrical power, W
q	–	heat flux, W/m^2
r	–	latent heat of vaporization, J/kg
s	–	pitch, m
t	–	temperature, °C
U	–	voltage drop, V
w	–	velocity, m/s

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Greek symbols

$\overline{\alpha}$ – average heat transfer coefficient, W/m^{2}K
λ – thermal conductivity, W/m K
ρ – density, kg/m^3
μ – viscosity, Ns/m^2
σ – surface tension, N/m

Subscripts

cr – critical
i – inner
l – liquid
o – outer
v – vapour

1 Background

In order to improve the design of shell-and-tube evaporators, they can be divided into three groups: reboilers, commonly used in food or chemical industry – Fig. 1a, flooded evaporators dominating in refrigerating systems – Fig. 1b, and two-phase themosyphon heat exchangers applied in heat recovery systems – Fig. 1c, numerous studies have been performed to understand bundle boiling heat transfer. In particular, the effect of tube position within a bundle, operating conditions, bundle layouts and tube spacing was examined [1–4]. Several correlations for the prediction of heat transfer coefficient of individual tube as well as tube bundle were proposed [5–8]. As it is pointed out in [9] the boiling mechanism in a flooded refrigerant evaporator is different from that which occurs in a kettle reboiler used in the process industry. As a result it is difficult to use information from one type of bundle and fluid combination, and apply it to another situation.

It is a known fact that heat transfer coefficients for a tube bundle are usually larger than those for nucleate pool boiling on a single tube under the same conditions. This is referred to as bundle factor – defined as a ratio of average heat transfer coefficient for the whole bundle to that of a single tube with the same surface (Fig. 2a) or bundle effect – defined as a ratio of heat transfer coefficient for an upper tube in a bundle with lower tubes heated to that for the same tube heated alone in the bundle (Fig. 2b). Bundle factor is typically slightly lower than the bundle effect and is of more use to a designer who can use it with a single tube data to estimate average bundle coefficients, by application for instance Palen [10] formula

$$\bar{\alpha}_b = \alpha_{nb} F_b + \alpha_{nc} , \tag{1}$$

where: α_{nb} – nucleate pool boiling heat transfer coefficient for single tube, α_{nc} – single phase free convection heat transfer coefficient for tube bundle, and F_b – bundle factor.

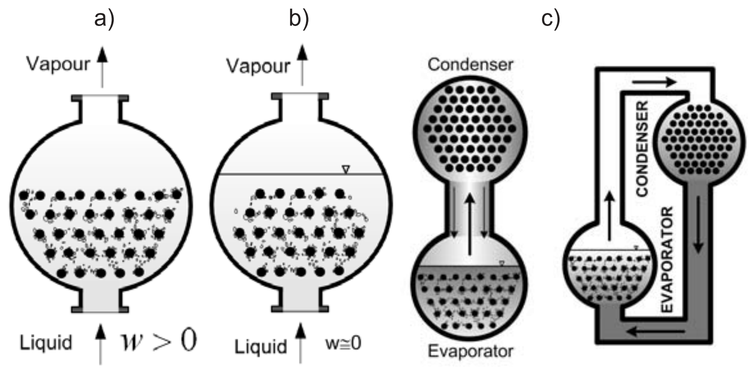

Figure 1. Shell-and-tube evaporators: a) reboiler, b) flooded-type evaporator, c) two-phase themosyphon heat exchanger.

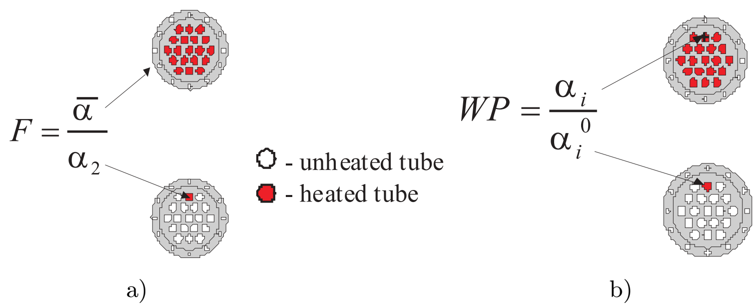

Figure 2. Illustration to the definitions of bundle factor a), and bundle effect b).

Very important factor that has influence on bundle boiling heat transfer is a gap between the outer tube of a bundle and the vessel wall (shell). Two solutions can be differentiated in the industrial practice: small tube bundle, where the tube bundle occupies only a small part of a cross-section of the vessel – Fig. 3a, and large tube bundle, where the tube bundle fulfils almost the entire cross section of the vessel – Fig. 3b. According to Gupta *et al.* [6] for small gap between the outer tube of the bundle and the vessel wall and under cross-flow velocity conditions, the uniform distribution of vapour bubbles over the entire cross-section of the channel can be assumed and, as a result, the heat transfer relations can be expressed in terms of the void fraction or the Martinelli parameter. This approach is not suitable for small tube bundle placed in a large channel because in such system an accurate estimation of the local void fraction is not possible.

Under low cross-flow velocity conditions (kettle reboilers, flooded evaporators) a gap between the outer tube of the bundle and the vessel wall strongly influences internal recirculation of liquid and therefore effects heat transfer performance of the heat exchanger.

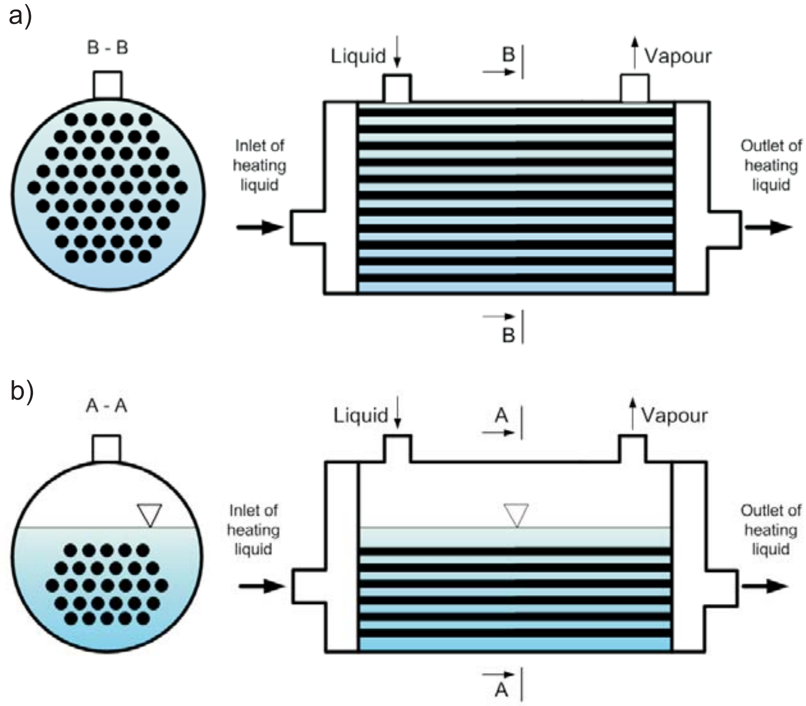

Figure 3. Tube bundle: a) large, b) small.

Very limited number of papers is available in the literature particularly concerning the bundle effect and bundle factor. Marto and Anderson [2] made measurements of boiling heat transfer coefficients in R-113 for a bundle of 15 electrically heated, smooth copper tubes arranged in an equilateral triangular pitch with s/D=2. It has been reported that lower tubes within a bundle can significantly increase the nucleate boiling heat transfer from the upper tubes at low heat fluxes. Qiu and Liu [4] investigated experimentally the effects of tube spacing, positions of tubes and test pressures on the boiling heat transfer of water in restricted spaces of the compact staggered bundles consisting of smooth horizontal tubes. A compact staggered bundle with a tube spacing of 0.3 mm displayed a maximum heat transfer enhancement in the range of the low and moderate heat fluxes. For boiling in compact bundles, the position of the heated tubes within the bundle has some effect on heat transfer for the tube spacings of 1.0 and 0.5 mm. However, the position of heated tubes has hardly any effect on the heat transfer for a tube spacing of 0.3 mm. For the case where the tube spacing is equal to or larger than 1.0 mm, the effect of pressure on the heat transfer enhancement was insignificant. Gupta *et al.* [6] conducted experiments with water boiling at atmospheric pressure on a smooth stainless steel tube arrangements: two or three tubes placed one above the other at different pitch distances (s/d=1.5–6.0) of commercial finish having a 19.05 mm outside diameter heated directly by means of a high alternating current. Heat transfer characteristics of the lowermost tubes in a tube bundle have been found to be independent of the presence of the bundle. Besides, the maximum enhancement of the order of 100% was observed for the top tube of a 1 x 3 tube bundle under low heat flux conditions. Kumar *et al.* [7] studied boiling of water at subatmospheric pressure on twin tube arrangement. Two copper, electrically heated, tubes having 32 mm outer diameter were placed one over another. Negligible influence of the tube materials on heat transfer coefficients was reported. Leong and Cornwell [11] performed experimental study with large, 241-tube arrangement that simulated slice of the reboiler. Experiments have been conducted with refrigerant R113 at atmospheric pressure. The local heat transfer coefficients for bottom row tubes in the tube bundle were almost the same as for a single tube, whilst for the top row tubes were considerably higher. Gupta [12] investigated nucleate boiling heat transfer in the electrically heated 5 x 3 in line horizontal tube bundle under pool and low cross-flow conditions of saturated water near atmospheric pressure. For configuration tested as well as for single column tube bundles heat transfer coefficients were nearly the same as those on a single tube. Maximum heat transfer coefficient on the top tube of central column was about seven times higher than that on the bottom tube at the same heat flux of 23 kW/m^2 under pool boiling conditions.

The purpose of the present paper is to provide a comprehensive nucleate boiling database for water, methanol and refrigerant R141b from a small bundle of smooth tubes that represents a portion of a flooded-type evaporator. The effect of heat flux, tube pitch and operating pressure is studied in the paper. Bundle factor and bundle effect are discussed as well. A correlation for prediction of a bundle average heat transfer coefficient is proposed.

2 Experimental apparatus and procedure

Figure 4 shows a schematic diagram of the experimental apparatus. Essentially, it consists of a cylindrical test vessel made of stainless steel having a diameter and length of 0.3 m, horizontal smooth tube bundle, condenser, measuring system, visualization system and an electric power supply system. The vessel is equipped with three inspection windows for direct observation and visualization of the boiling process. Vapour from the test vessel flows through a stainless steel tube having an inside diameter 50 mm and enters the condenser, installed above the vessel. The flow rate of cooling water through the condenser was regulated by the manual valve and measured by a flowmeter. Test liquid entering the vessel is at the saturated state and its temperature is controlled by a heating element placed in the preheater.

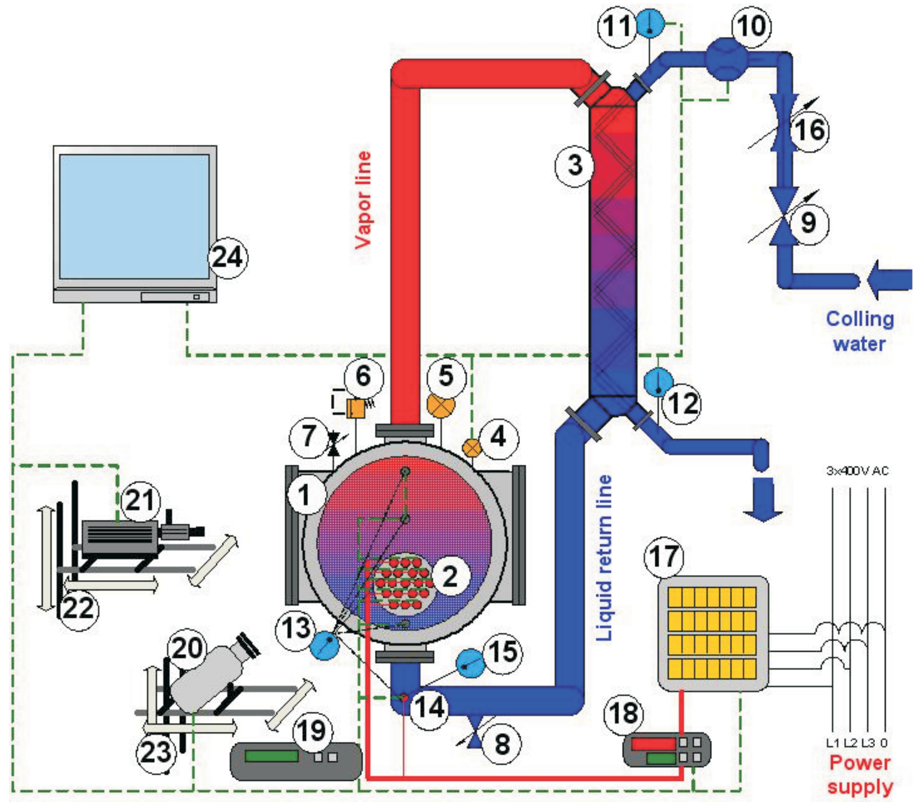

Figure 4. Schematic diagram of the experimental apparatus: 1 – test vessel, 2 – tube bundle, 3 – condenser, 4 – pressure transducer, 5 – pressure gauge, 6 – safety-valve, 7 – valve to setting of pressure in test vessel, 8 – drain valve of test liquid, 9 – drain valve of cooling water, 10 – flowmeter, 11, 12, 13, 15 – thermocouples, 14 – preheater, 16 – manual valve of flow control, 17 – wattmeter, 18 – regulators, 19 – multiplexer, 20 – high speed camera, 21 – CCD camera, 22 – mobile support (3d) of CCD camera, 23 – mobile support (3d) of high speed camera, 24 – computer aided data acquisition system.

The evaporator was designed to simulate a slice of a flooded-type evaporator. The bundle consists of 19 electrically-heated smooth tubes which are arranged in a staggered triangular-pitch layout with a pitch-to-diameter ratio of 1.7 and 2.0. The bundles were cantilever-mounted from the back wall of the evaporator to permit the in-bundle visualization. A stainless steel tube of $R_a = 0.40$ μm having 10 mm OD and 0.6 mm wall thickness formed a single test heater. Electrical energy supplied to heating elements is controlled by electronic regulators. Each cartridge heater is equipped with a separate regulator. Each tube was 180 mm long and effective length was 155 mm. The liquid level was maintained at ca. 15 mm above top row of tubes in the bundle.

As pointed out in [13,14] great care must be exercised with the cartridge heater and temperature measuring instrumentation to ensure good accuracy of the measurement of the inside temperature of the heating cylinder. Each tube was equipped with four thermocouples evenly spaced at 90° on the inner wall and midway between the heated length of a tube. The wall temperature t_w was calculated from the formula [15]

$$t_w = t_i - UI\frac{\ln{(D_o/D_i)}}{2\pi\lambda L} \;, \tag{2}$$

where t_i was calculated as the arithmetic mean of four measured inside wall temperatures. More details about experimental stand and procedure are given in [16].

2.1 Uncertainty estimation

The uncertainties of the measured and calculated parameters are estimated by the mean-square method. Heat flux was calculated from the formula

$$q = \frac{UI}{\pi D_o L} = \frac{P}{\pi D_o\, L} \;. \tag{3}$$

The experimental uncertainty of heat flux was estimated as follows

$$\Delta q = \sqrt{\left(\frac{\partial q}{\partial P}\Delta P\right)^2 + \left(\frac{\partial q}{\partial D_o}\Delta D_o\right)^2 + \left(\frac{\partial q}{\partial L}\Delta L\right)^2} \;, \tag{4}$$

where the absolute measurement errors of the electrical power ΔP, outside tube diameter ΔD_o and active length of a tube ΔL are 10 W, 0.02 mm, and 0.2 mm, respectively. So, the maximum overall experimental limits of error for heat flux ranged from ±1.3% for maximum heat flux to ±1.2% for minimum heat flux.

The experimental uncertainty for the average heat transfer coefficient is calculated as

$$\Delta\bar{\alpha} = \sqrt{\left(\frac{\partial\bar{\alpha}}{\partial q}\Delta q\right)^2 + \left(\frac{\partial\bar{\alpha}}{\partial\Delta T}\delta T\right)^2} \;, \tag{5}$$

where the absolute measurement error of the wall superheat, δT, estimated from the systematic error analysis, equals to ±0.2 K. The maximum error for average heat transfer coefficient was estimated to ±2.3%.

3 Results and discussion

In order to validate the apparatus as well as experimental procedure, the present data for refrigerant R141b at atmospheric pressure were compared with those predicted by Palen correlation – Eq. (1), and experimentally obtained by Chang *et al*. [17]. Figure 5 shows comparison of present experimental data with Palen correlation prediction where nucleate pool boiling α_{nb} and single phase free convection α_{nc} heat transfer coefficients were calculated using Cooper [18] and Michejev [19] correlations, respectively. The bundle factor was assumed to be equal to 1.5. Palen correlation underpredicts present data for heat flux $q < 35$ kW/m^2 and overpredicts for $q > 35$ kW/m^2 and it is not surprising having in mind that the bundle factor F_b depends primarily on the heat flux, bundle configuration, and bundle diameter. It seems that Palen correlation may serve as a first step design correlation. The experimental data for boiling of R141b are found to be in reasonable agreement with those obtained experimentally by Chang *et al*.

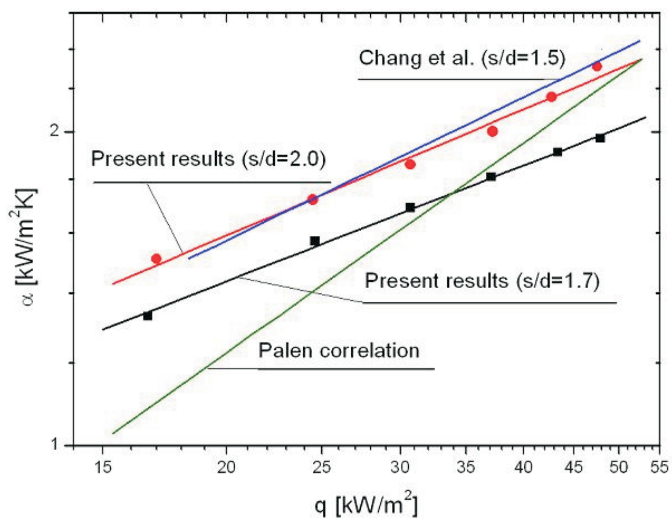

Figure 5. Comparison of present results with Chang *et al.* experiment [17] and Palen correlation for R141b at atmospheric pressure.

Independent of pitch and operating pressure the highest bundle average heat transfer coefficients were obtained for boiling water. As an example Fig. 6 shows experimental results for three tested boiling liquids and bundle pitch-to-diameter ratio of 1.7 recorded at atmospheric pressure. For all tested liquids, both atmospheric and subatmospheric pressure, higher heat transfer coefficients were obtained for greater pitch-to-diameter ratio examined, i.e. 2.0. Exemplarily, Fig. 7 illustrates influence of pitch-to-diameter ratio for refrigerant R141b boiling at atmospheric pressure.

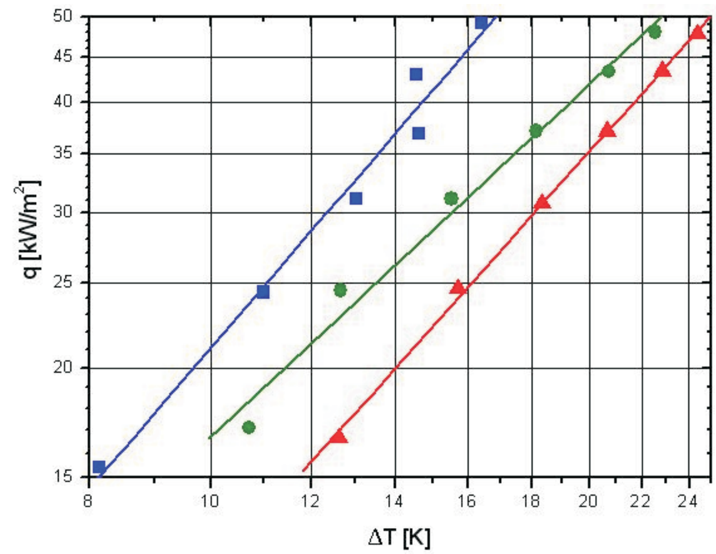

Figure 6. Boiling curves for three tested liquids at atmospheric pressure and bundle pitch-to-diameter ratio of 1.7: – water, • – methanol, – R141b.

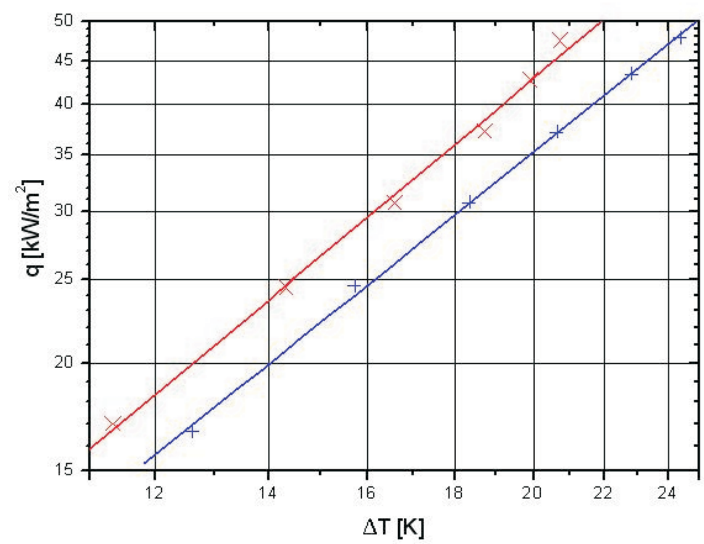

Figure 7. Influence of pitch-to-diameter ratio for R141b boiling at atmospheric pressure; pitch-to-diameter ratio: + – 1.7, × – 2.0.

Independent of pitch and kind of liquid tested higher heat transfer coefficients were obtained for atmospheric pressure. As an example Fig. 8 displays boiling curves for methanol boiling at atmospheric and subatmospheric pressure and bundle pitch-to-diameter ratio of 2.0.

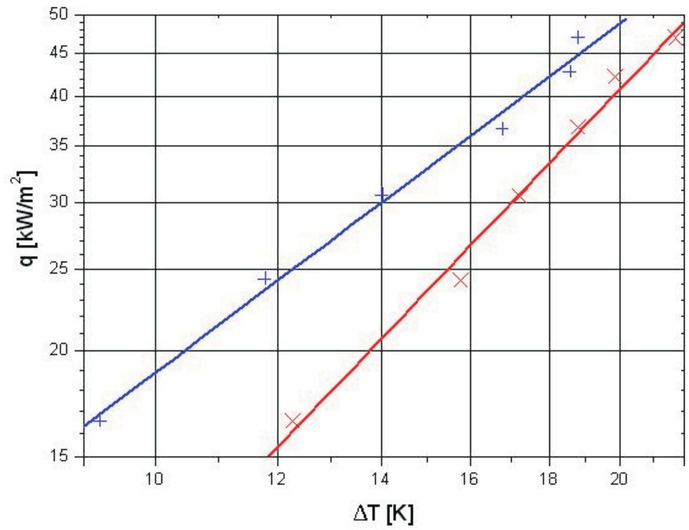

Figure 8. Influence of pressure for methanol boiling on smooth tube bundle with pitch-to-diameter ratio 2.0: $\times$ – sub-atmospheric pressure (14 kPa), $+$ – atmospheric pressure.

Figure 9 illustrates average heat transfer coefficients for the row of tubes and selected heat fluxes in the case of water boiling at atmospheric pressure on the tube bundle with pitch-to-diameter ratio 1.7. The higher was the heat flux the higher was heat transfer coefficient for all rows of tubes and simultaneously average heat transfer coefficient increases from bottom to the top row of tubes. Bundle factor and bundle effect (Fig. 10) decrease with heat flux increase. Likewise literature data [13] bundle effect is slightly higher than the bundle factor.

A multidimensional regression analysis using the least squares method was used to establish the correlation equation for prediction of bundle average heat transfer coefficient

$$\mathrm{Nu} = 521.7\mathrm{Bo}^{0.305}\left[\left(\ln\frac{p}{p_{cr}}\right)^2\right]^{-1.48}\left(\frac{s}{D}\right)^{0.74}\mathrm{Pr}^{0.67}\,, \tag{6}$$

where:
$\mathrm{Nu} = \dfrac{\overline{\alpha} D_o}{\lambda_l}$ – Nusselt number,
$\mathrm{Bo} = \dfrac{qLa\rho_l}{\rho_v r \mu_l}$ – boiling number,

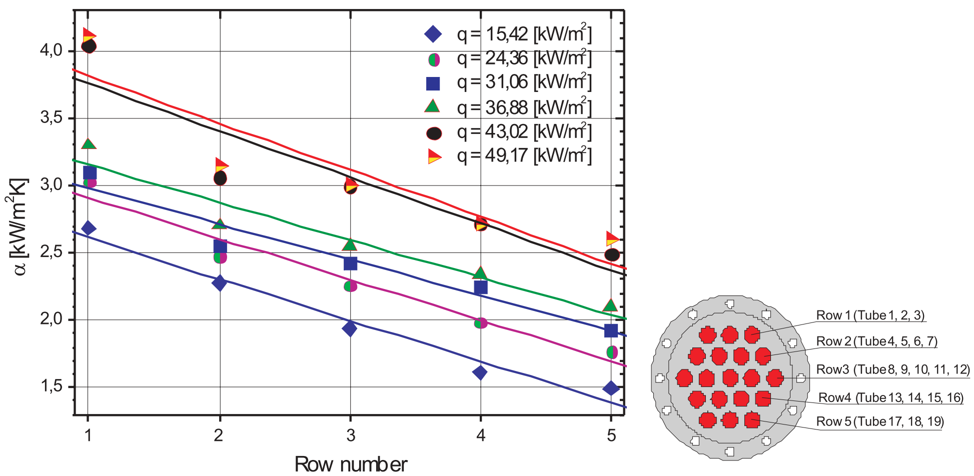

Figure 9. Row average heat transfer coefficient against row position in a tube bundle.

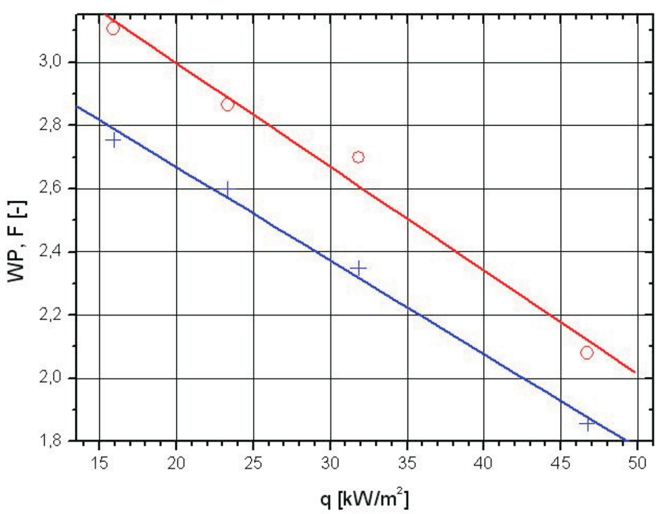

Figure 10. Bundle factor (F – +) and bundle effect (WP – ∘) for methanol boiling at atmospheric pressure on smooth bundle with pitch-to-diameter ratio of 2.0.

$La = \sqrt{\dfrac{\sigma}{g\left(\rho_l - \rho_v\right)}}$ – characteristic length,

$\Pr = \dfrac{\nu}{a}$ – Prandtl number.

A comparison of predicted data against the experimentally ones obtained within the present investigation is displayed in Fig. 11. For about 96% of experimental points the discrepancy between experimental data and values calculated from proposed correlation is lower than ±40%.

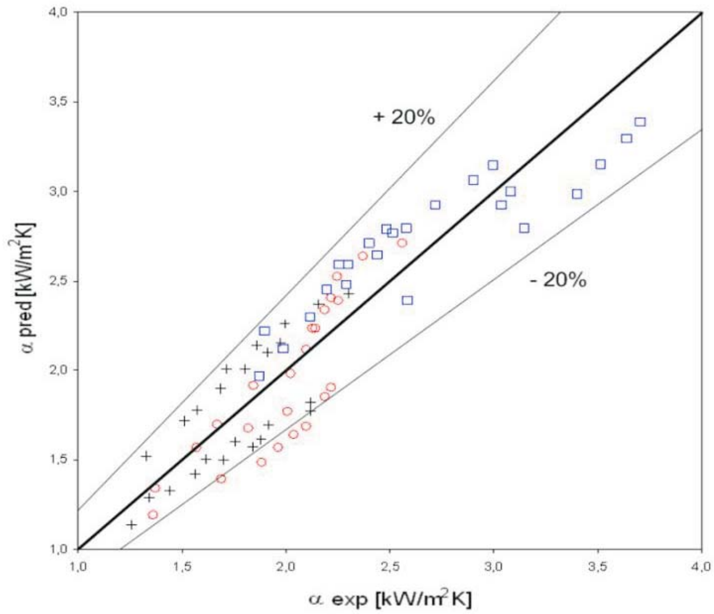

Figure 11. Predicted vs. experimental average heat transfer coefficients in smooth tube bundle: × – water, ∘ – methanol, + – R141b.

4 Conclusions

The following conclusions can be drawn from the present investigation on boiling heat transfer in smooth tube bundles under atmospheric and subatmospheric pressures:

1. Distilled water has an evident superiority over methanol and refrigerant R141b for each tested tube bundle.
2. For atmospheric pressure higher heat transfer coefficients were recorded than for subatmospheric pressure.
3. Increase in pitch-to-diameter ratio results in average heat transfer coefficient increase for all three liquid tested as well as atmospheric and subatmospheric pressure.
4. Row average heat transfer coefficient increases from the bottom to top row of tubes.

5. A Nusselt-type relation has been proposed to predict the heat transfer coefficient and the predicted values correlate satisfactory with the experimental data related to water, methanol and refrigerant R141b over some range of pressure.

Received 30 August 2011

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