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Heat Transfer During Pool Boiling of Water, Methanol and R141b on Porous Coated Horizontal Tube Bundles

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ABSTRACT

This paper presents the results of experimental investigation of water, methanol and R141b refrigerant boiling on horizontal porous coated tube bundles supposed to represent a flooded-type evaporator. Experiments were carried out for a bundle of 19 tubes in triangular layout for two pitch-to-diameter ratio values – 1.7 and 2.0 – in atmospheric and sub-atmospheric pressure conditions. Average heat transfer coefficients – both local, row-specific, and for the whole bundle – were determined. The boiling process on a bundle was visualized with a CCD camera and laser sheet technique. A Nusselt type correlation equation allowing the calculation of the average heat transfer coefficient for boiling on porous coated tube bundles has been proposed.

Key words: treated surface, pool boiling, tube bundle**,** passive technique

INTRODUCTION

Literature data indicate that during nucleate pool boiling on single tube with porous coating heat transfer coefficient can be an order of magnitude higher than for smooth tube at that same wall superheat. Comprehensive surveys of the studies related to the boiling – among others, on porous coated surfaces are given in (Thome, 1990, Webb, 1994). More recent review on pool boiling enhancement was discussed by Bergles (1997). New surveys related to enhanced boiling are presented in (Ribatski and Thome, 2006, Cieśliński, 2011, Yang and Liu, 2013, Bergles and Manglik, 2013). Heat transfer mechanisms of enhanced surfaces are presented extensively in a web book (Poniewski and Thome, 2008). Based on the above mentioned reviews that are recommended as reference studies, below a literature survey dealing only with boiling on porous coated tube bundles is presented.

From the practical point of view a question arises about heat transfer enhancement during pool boiling on tube bundle made of porous coated tubes and secondly, is it possible to use data obtained for single porous coated tube in design of flooded-type evaporator. Two-phase interactions that occur in tube bundles during boiling are very complex and can vary with heat flux, operating pressure, fluid properties, tube surface, pool height and bundle layout. 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.

Data presented in the open literature regarding pool boiling on porous coated tube bundles are very scarce and fragmentary and suggest slight heat transfer augmentation in comparison with smooth tube bundles for the same thermal conditions. Czikk et al. (1970), realized one of the first published investigations of boiling on porous coated tube bundle. Two prototypes of liquid coolers (70 kW capacity) were tested. In the first cooler the test bundle was made of twenty copper tubes with 0.3 mm copper porous layer, porosity of 60% with 19.9 mm of OD and of 1.52 m length. The tube bundle was built as a four row triangular arrangement with pitch-to-diameter-ratio of 1.2. As a working fluid was used refrigerant R11, which was boiling in temperature 1 °C. Heat flux range was from 31.4 kW/m^2 to 45.6 kW/m^2. The tube bundle average heat transfer coefficient of the 5x4 tube bundle attained 27.4 kW/m^{2}K to 41.5 kW/m^{2}K and has been a little larger compared with heat transfer coefficient for horizontal round plate coated with identical porous layer during pool boiling of refrigerant R11. Czikk et al. (1970), established, that 2% oil in mixture didn't cause visible change of heat transfer coefficient, and flood below the top row of tubs in a bundle did not influence on efficiency of cooler. Second cooler was made of 18 copper tubes with 0.25 mm copper porous layer, porosity of 60% with 23.7 mm of OD and of 0.7 m in length. The tube bundle was built as a 7 row triangular layout with pitch-to-diameter-ratio of 1.195. As a working fluid was used refrigerant R11, too. The tube bundle average heat transfer coefficient was as that received on horizontal round plate coated with identical porous layer during pool boiling of refrigerant R11. Starner and Cromis (1977), showed that for the same efficiency of water coolers, the total length of porous coated tubes with 0.18 mm porous layer and porosity of 60% can be even about half shorter than the length of standard low-finned tubes (1026 fin/m) with the same diameter of 19 mm if the boiling liquid will be refrigerant R22. Xiulin et al. (1989), investigated nitrogen boiling under atmospheric pressure on tubes with 0.4 - 0.5 mm copper porous layer, porosity of 40 – 50% and average pore radius of 120 μm. It was established that heat transfer on tube bundle with two vertical tubes is more intensive than in case of individual tube and it depends on distance between them. For investigated range scale between tubes (3 - 12 mm) heat transfer for the smallest scale was the best. For larger range scale heat transfer rate decreased. Danilowa et al. (1992), studied boiling of R22 and R717 on tube bundle model made of smooth and porous coated steel tubes. The tube bundle was built as a five row triangular arrangement with pitch-to-diameter-ratio of 1.35, with aluminium porous layer of 0.26 - 1.0 mm thick, porosity of 25 - 44%, and average pore radius of 24 - 40 μm, and 20 mm of OD. Heat transfer coefficient on a porous coated tube bundle for upper and bottom rows was the same, for the heat flux range from 1 kW/m^2 to 10 kW/m^2. Intensity of heat transfer on porous coated tube bundle decreased with number of tube rows in tube bundle increase. Jensen et al. (1992), investigated boiling of R113 on bundles made of 68 smooth or porous coated copper tubes. The tube bundle was built as a 15 row triangular layout with two pitch-to-diameter-ratios (1.17 and 1.5). The 101.6 mm long, 19.1 mm of OD tubes were constructed of copper. Pressure range was from 0.2 to 0.6 MPa, heat flux from 5 kW/m^2 to 80 kW/m^2, mass velocity from 50 to 500 kg/m^{2}s and quality (inlet to vessel) from 0 to 80%. Jensen et al., 1992 established that heat transfer coefficient of porous coated tube bundle is considerably higher than for smooth tube bundle. The value of average heat transfer coefficient for pool boiling on a porous tube bundle is almost the same as average heat transfer coefficient for a single tube with identical porous layer, and secondly the influence of change of mass velocity and inlet quality is very small on porous coated tube bundle heat transfer coefficient. Memory et al. (1995), investigated boiling of refrigerant R114 on smooth and porous coated tube bundles with 15 High Flux tubes in a staggered triangular-pitch layout with pitch-to-diameter-ratio of 1.2. The porous tube bundle data contradict results for the other bundles, showing a bundle effect near 1 for the whole range of heat flux covered. Memory et al. (1995) claimed, that nucleation from the porous coated surface is the dominant mechanism at low as well as high heat fluxes (2 – 60 kW/m^2), aside from the tube position in a column. The results for pure R-114 confirm that, in general, a bundle factor should be incorporated into the design of flooded evaporators and the use of single tube data will be conservative. The one exception to this is the porous coated tube bundle, where single tube data may even overestimate the heat transfer especially at higher heat fluxes. Furthermore, bundle enhancements agree fairly well with single tube enhancements. McNeil et al. (2002), investigated boiling of pentane at atmospheric pressure in a slice kettle reboiler (241 electrically heated tubes, 17 rows and 17 columns in line). The tubes were 19 mm in diameter and 56 mm long with 0.08 mm porous layer. The shell was 738 mm in diameter and 56 mm long. Heat flux range was from 10 to 50 kW/m^2. The pool boiling results show that the High Flux tubes produce heat transfer coefficients that are up to five times larger than their plain tube counterparts. In flow boiling the enhancement is 3 – 6 times. Hsieh et al. (2003), studied nucleate pool boiling from plasma coated tube bundles immersed in saturated R-134a. The bundles were composed of 15 tubes (of which the number of heated/ instrumented tubes was varied) arranged in four different configurations with a pitch-to-diameter ratio of 1.5. A rectangular vessel, 370 mm deep, 650 mm high and 370 mm wide, with an electrically heated multi-tube bundle was used to simulate a portion of a refrigerant-flooded evaporator. The tube bundle were fabricated from commercially available, 20 mm diameter smooth cooper tubes with 0,1 mm porous copper layer, porosity of 5.7%, and average pore radius of 4 μm. Heat flux range was from 0.1 to 30 kW/m^2. The saturation temperature was kept near 18 ºC. The liquid level was maintained approximately 100 mm above the test tube. The average bundle heat transfer coefficient was found to be three times greater than for smooth tube bundle and five times greater than an isolated single smooth tube. Hsieh et al. found that bundle factor decreases to unity at high heat fluxes. Secondly, remarkable enhancement of heat transfer for top rows in vertical-in-line and staggered configurations were observed upon the heat fluxes imposed. A configuration factor was defined. Its value is less than 1 for several cases in some geometric arrangements. Both bundle factor and configuration factor showed a consistent trend for the associated tube bundle configurations, and an optimum geometry arrangement could be selected based on the complex effect of the types of tube bundle configurations and heat fluxes. Schäfer et al. (2005) studied boiling of refrigerant R-134a on a porous, plasma coated tube bundle. The tube bundle consisted of up to 4 tubes with a pitch-to-diameter-ratio of 1.33 in a rectangular arrangement. The tube bundle were fabricated from 16 mm ID tubes. Thickness of the porous copper layer was 170 μm, porosity of about 20%, and average pore radius of 41 μm. Heat flux was varied in the range from 2 to 100 kW/m^2. Length of the oil heated tubes was 350 mm. Schäfer et al. established boiling heat transfer improvement particularly for low heat fluxes over a wide range of saturation pressures (1-25 bar). Furthermore, bundle factor decreased to 1 for higher heat fluxes. Recently, Kim et al. (2011) investigated boiling of R-123/oil mixtures on the bundle composed of 14 tubes with only one instrumented. The enhanced tubes were made from thick-walled copper tubes of 18.8 mm outer diameter and 13.5 inner diameter. The length of the tubes was 170 mm. The pore diameter ranged from 0.2 to 0.27 mm. It was established, that the tube with pore diameter of 0.23 mm yielded the highest heat transfer coefficient with pure refrigerant as well as with refrigerant/oil mixture, although the difference decreases with addition of oil. Furthermore, the tube with the largest pore diameter showed the smallest degradation.

The purpose of the present paper is to provide a comprehensive database for water, methanol and refrigerant R141b boiling on a small bundle of porous coated tubes that represents 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.

EXPERIMENTAL APPARATUS AND PROCEDURE

Figure 1 shows a schematic diagram of the experimental apparatus. Essentially, it is consisted of a cylindrical test vessel, a horizontal tube bundle, a condenser, the measuring system, visualization system and electric power supply system. The bundle consists of 19 electrically-heated smooth or porous coated 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 in-bundle visualization. Commercially available stainless steel tubes having 10 mm OD and 0.6 mm wall thickness were used to fabricate tube bundles. Aluminium porous coatings of 0.15 mm thick with porosity of about 40% and mean pore radius of 2.8 μm were produced by plasma spraying. 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. Details of the heating section are shown in Figure 2. The liquid level was maintained at about 15 mm above top row of tubes in the bundle. Tube bundle was mounted inside a cylindrical test vessel made of stainless steel having a diameter and length of 0.3 m. The vessel is equipped with three inspection windows for direct observation and visualization of the boiling process. Four K-type thermocouples installed in the grooves of a copper sleeve placed inside the tube were used to measure inside temperature of a tube. Experiments were performed for atmospheric and sub-atmospheric pressure within heat flux range from 15 to 50 kW/m^2.

DATA REDUCTION AND UNCERTAINTY ESTIMATION

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

$$q = \frac{UI}{\pi D_o L} = \frac{P}{\pi D_o L} \tag{1}$$

the experimental uncertainty of heat flux was estimated as follows:

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

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 extended from ± 1.3% to ± 1.2% for maximum and minimum heat flux, respectively.

Single tube wall superheat was calculated as

$$\Delta T = t_{out} - t_l \tag{3}$$

where the outside temperature was estimated from the formula (Chiou et al., 1997)

$$t_{out} = t_{in} - UI\frac{\ln(D_{out}/D_{in})}{2\pi\lambda_t L} \tag{4}$$

Inside temperature t_{in} of a tube was calculated as the arithmetic mean of four wall temperatures indicated by thermocouples placed inside a tube and measuring circumferential temperature distribution – Figure 2. Liquid temperature t_l was measured directly by a thermocouple placed in a pool below a tube bundle.

Average tube bundle wall superheat was calculated as arithmetic mean of 19 tube wall superheats

$$\Delta\overline{T}_b = \frac{\sum_{i=1}^{i=19}\Delta T}{19} \tag{5}$$

Row average wall superheat was calculated as

$$\Delta\overline{T}_{row} = \frac{\sum_{i=1}^{i=n}\Delta T}{n} \tag{6}$$

where n is a number of tubes in a row.

Absolute measurement error of the average tube bundle wall superheat, estimated from the systematic error analysis, equals ± 0.26 K.

Row average heat transfer coefficient $\overline{\alpha}_{row}$ and average tube bundle heat transfer coefficient $\overline{\alpha}_b$ were calculated as

$$\overline{\alpha}_{row} = \frac{q}{\Delta\overline{T}_{row}} \text{ or } \overline{\alpha}_b = \frac{q}{\Delta\overline{T}}, \tag{7}$$

respectively.

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

$$\Delta\overline{\alpha} = \sqrt{\left(\frac{\partial\overline{\alpha}}{\partial q}\Delta q\right)^2 + \left(\frac{\partial\overline{\alpha}}{\partial\Delta T}\delta T\right)^2} \tag{8}$$

The maximum error for average heat transfer coefficient was estimated to ± 2.3%.

RESULTS AND DISCUSSION

In order to validate the apparatus as well as experimental procedure, the present data were compared to the data reported by other researchers. A comparison between the present experimental results and data obtained by Cieśliński (2002) and Lakhera et al. (2009) for water boiling on horizontal tubes with porous coatings of about the same thickness ~0.15 mm, fabricated by thermal spraying is shown in Figure 3. Despite the differences in porosity of porous coatings satisfactory agreement has been obtained with literature data.

Independent of tested liquid, pitch-to-diameter ratio and operating pressure, higher heat transfer coefficients were obtained for porous coated tube bundle than for smooth tube bundle. As an example, Figure 4 shows boiling curves for R141b while boiling on smooth and porous coated tube bundles with pitch-to-diameter ratio of 1.7 at atmospheric pressure.

Experiments revealed that independent of pitch and operating pressure the highest bundle average heat transfer coefficients were obtained for boiling water, which results from its excellent thermophysical properties, above all the high heat of evaporation. As an example Figure 5 shows experimental results for three tested boiling liquids and bundle pitch-to-diameter ratio of 2.0 recorded at atmospheric pressure.

For all tested liquids, both atmospheric and sub-atmospheric pressure, higher heat transfer coefficients were obtained for greater pitch-to-diameter ratio examined, i.e. 2.0. The higher was heat flux the bigger were wall superheat differences between tube bundle with pitch-to-diameter ratio of 1.7 and 2.0, respectively. It seems that present results confirm the primary guideline in setting enhanced tube pitches suggested by Thome (1990, p. 288), i.e.: "the larger the tube pitch, the smaller the tendency for vapor blanketing to occur at the top of the bundle". Exemplarily, Figure 6 illustrates influence of pitch-to-diameter ratio for water boiling at atmospheric pressure.

Independent of pitch-to-diameter ratio and kind of liquid tested higher heat transfer coefficients were obtained for atmospheric pressure, what is compatible with few literature data. The higher was heat flux the lower were average wall superheat differences between data for atmospheric and sub-atmospheric pressure. As an example Figure 7 displays boiling curves for methanol boiling at atmospheric and sub-atmospheric pressure and bundle pitch-to-diameter ratio of 1.7.

Figure 8 and Figure 9 illustrate average heat transfer coefficients for rows of tubes and selected heat fluxes in the case of water boiling at atmospheric pressure on porous coated tube and smooth tube bundles, respectively. For smooth tube bundle (Figure 8) heat transfer coefficient increases with increase from one row to another in the upward direction, and for a given row the value of heat transfer coefficient increases with heat flux increase. This observation agrees with published data (Slesarenko et al, 1982, Rebrov et al., 1989). Contrary to smooth tube bundle, row average heat transfer coefficient for porous coated tube bundle decreases from bottom to top row of tubes (Figure 9). This phenomenon can be explained by enormous nucleation ability of porous coating. During boiling on porous coating the number of generated vapour bubbles is so big that the upper rows of tubes become inundated with vapour what leads to decrease in heat transfer performance. Similar behaviour, i.e. disappearance of heat transfer augmentation for the upper porous coated tube was observed by Fujita et al. (1986) during experiments with boiling of refrigerant R113 for the two-tube configuration. As an example, Figure 10 illustrates boiling process on porous coated tube bundle.

As it is seen in Figure 11 bundle effect decreases monotonically with heat flux increase, while bundle factor decreases as well but up to the heat flux about 25 kW/m^2. For higher heat flux, i.e. fully developed boiling regime, bundle factor is almost constant and reaches value about 0.95. It means that average bundle heat transfer coefficient is approximately 5% lower than that for a single porous coated tube.

A multidimensional regression analysis based on the least squares method was used to establish correlation equation for prediction of porous coated tube bundle average heat transfer coefficient – Eq. (9).

$$Nu = 837.3 Bo^{0.185} \left(\left(\ln \frac{p}{p_{cr}} \right)^2 \right)^{-1.312} \left(\frac{s}{D} \right)^{-0.023} \Pr^{0.76} \qquad (9)$$

where:

$Nu = \frac{\overline{\alpha}_b D_t}{\lambda_l}$ - Nusselt number,

$Bo = \frac{q La \rho_l}{\rho_v r \mu_l}$ - boiling number,

$La = \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}}$ - characteristic length.

Proposed correlation includes all tested variables in dimensionless form, i.e.: heat flux, thermophysical properties of boiling liquids, pitch-to-diameter ratio and reduced pressure. A comparison of predicted data against the experimentally obtained under the present investigation is displayed in Figure 12. For about 96% of experimental points the discrepancy between experimental data and values calculated from proposed correlation is lower than ±20%. Versatility of present correlation was checked by comparison predictions made by use of proposed correlation (Eq. 9) with experimental data obtained by Hsieh et al. (2003). Figure 13 shows average heat transfer coefficient obtained by Hsieh et al. (2003) for saturated refrigerant R134a boiling on a porous coated tube bundle with 6 heated tubes and 5 dummy tubes (above) in a triangular layout with pitch-to-diameter ratio of 1.5. This arrangement best corresponds to that used in the present study. For this case study Hsieh et al. (2003) have proposed very simple correlation for average heat transfer coefficient calculation

$$\overline{\alpha}_b = 22.21q^{0.54} \tag{10}$$

It is worth noting that in paper (Hsieh et al., 2003, Tab. 3) incorrect form of the Eq. 10 is presented, i.e. $q = 22.21\overline{\alpha}_b^{0.54}$. As results from Figure 13 present correlation (Eq. 9) underpredicts experimental data obtained by Hsieh et al. (2003) for lower heat fluxes and overpredicts for higher heat fluxes. Having in mind quite different liquid-porous surface combination tested by Hsieh et al. (2003), present correlation reproduces their experimental data with reasonable agreement. Besides, present correlation was successfully applied in calculations of heat transfer performance of two-phase thermosyphon heat exchanger (Cieśliński and Fiuk, 2013).

Nomenclature
D - diameter [m],
g - acceleration due to gravity [m/s^2],
I - current [A],
L - tube active 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],
ΔT – wall superheat [K]
U – voltage drop [V]

Greek symbols
$\overline{\alpha}$ - average heat transfer coefficient [W/m^{2}K],
λ - thermal conductivity [W/mK],
ρ - density [kg/m^3],
μ - viscosity [Ns/m^2],
σ - surface tension [N/m].

Subscripts
atm – atmospheric
b – bundle
cr - critical,
in – inner,
l – liquid,
n – number of tubes in a row
out – outer,
row – row,
t – tube,
v - vapour

CONCLUSIONS
The following conclusions can be made from the present investigation on boiling heat transfer on porous coated tube bundles under atmospheric and sub-atmospheric 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 sub-atmospheric 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 sub-atmospheric pressure.
4. Row average heat transfer coefficient decreases from bottom to top row of tubes.
5. The value of bundle factor for heat flux above 25 kW/m^2, was nearly equal to 1, it means that average heat transfer coefficient for boiling on a porous tube bundle is almost the same as average heat transfer coefficient for a single tube with identical porous layer.
6. A Nusselt-type relation has been proposed to predict 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.

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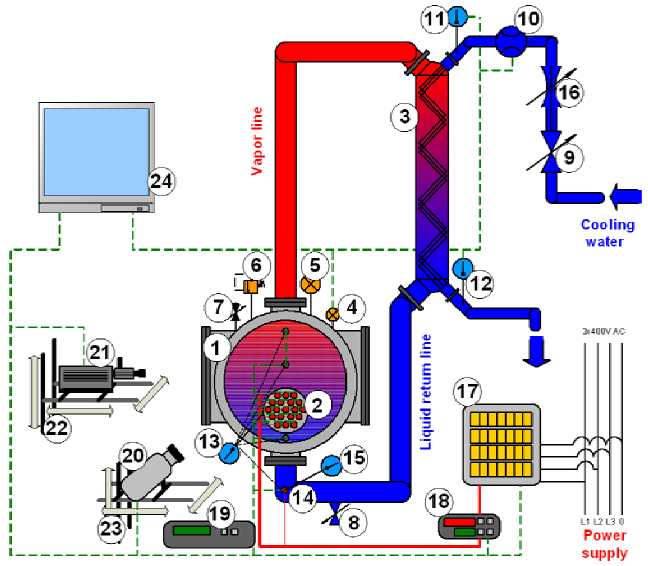

Fig. 1. Schematic diagram of the experimental apparatus; 1 – test vessel, 2 – tube bundle, 3 – condenser, 4 – pressure transducer, 5 – pressure gauge, 6 – safety-valve, 7 – valve to setting of pressure in test vessel, 8 – drain valve of test liquid, 9 – drain valve of cooling water, 10 – flowmeter, 11, 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

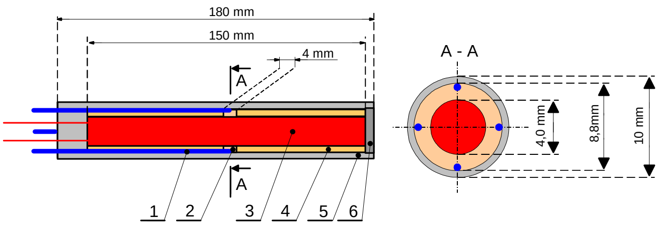

Fig. 2. Details of the heating section; 1 – thermocouple, 2 – ebonite ring, 3 – cartridge heater, 4 – copper sleeve, 5 – tested tube, 6 – cap

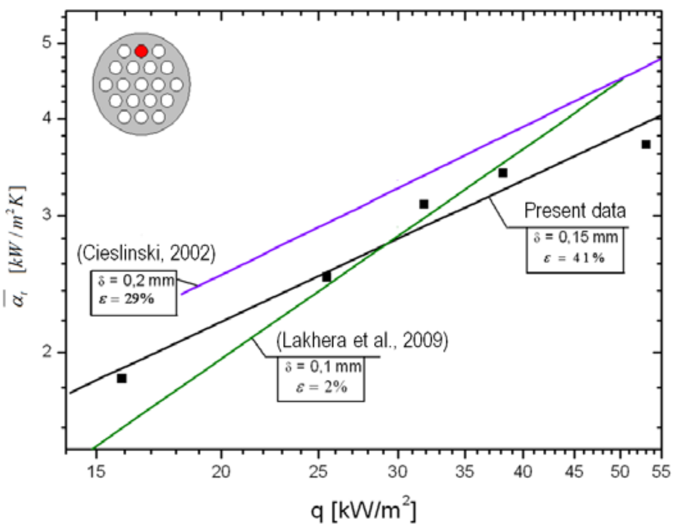

Fig. 3. Validation of present experimental results

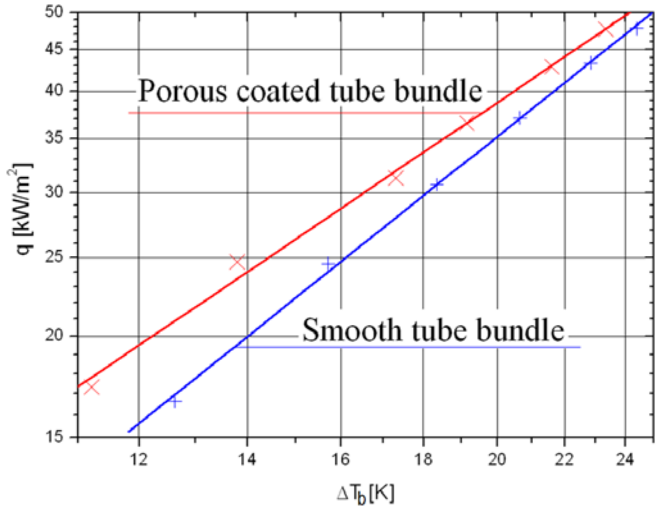

Fig. 4. Boiling curves of R141b on smooth and porous coated tube bundles and pitch-to-diameter ratio of 1.7 at atmospheric pressure

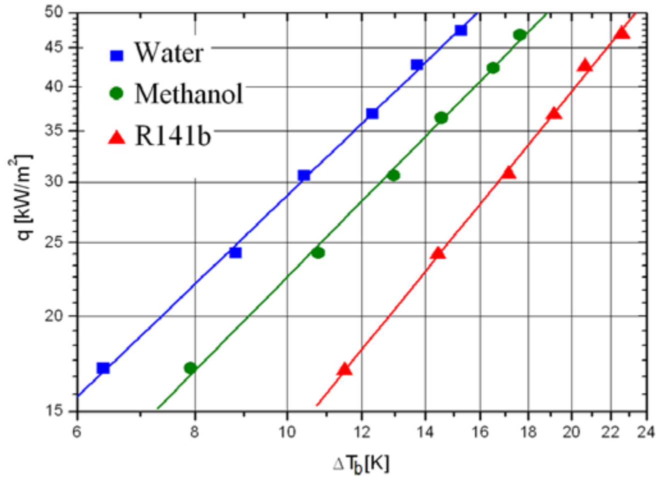

Fig. 5. Boiling curves for three tested liquids at atmospheric pressure and bundle pitch-to-diameter ratio of 2.0

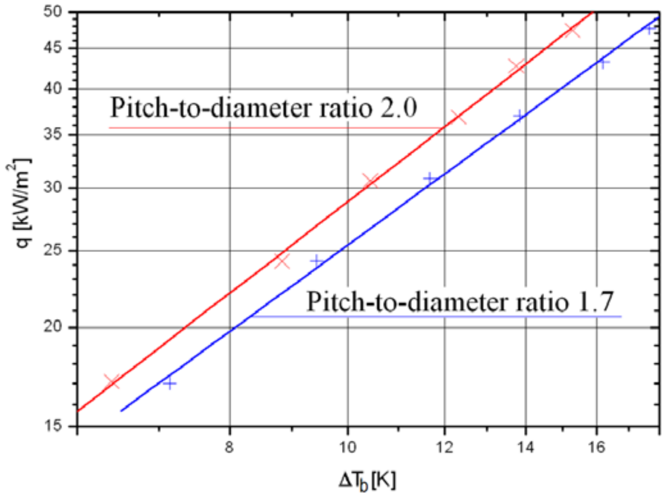

Fig. 6. Influence of pitch-to-diameter ratio for water boiling at atmospheric pressure; pitch-to-diameter ratio: + 1.7, × - 2.0

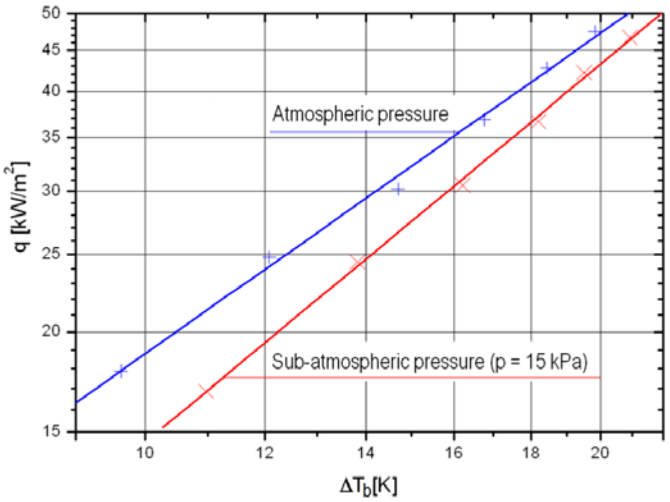

Fig. 7. Influence of operating pressure for methanol boiling
on porous coated tube bundle with pitch-to-diameter ratio of 1.7

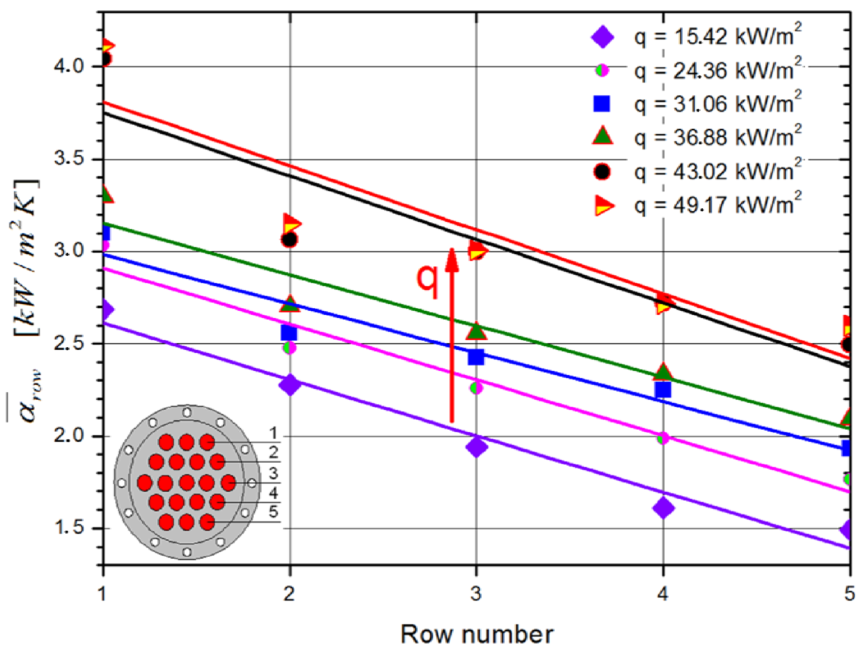

Fig. 8. Row average heat transfer coefficient for water boiling
on a smooth tube bundle of pitch-to-diameter ratio of 1.7 at atmospheric pressure

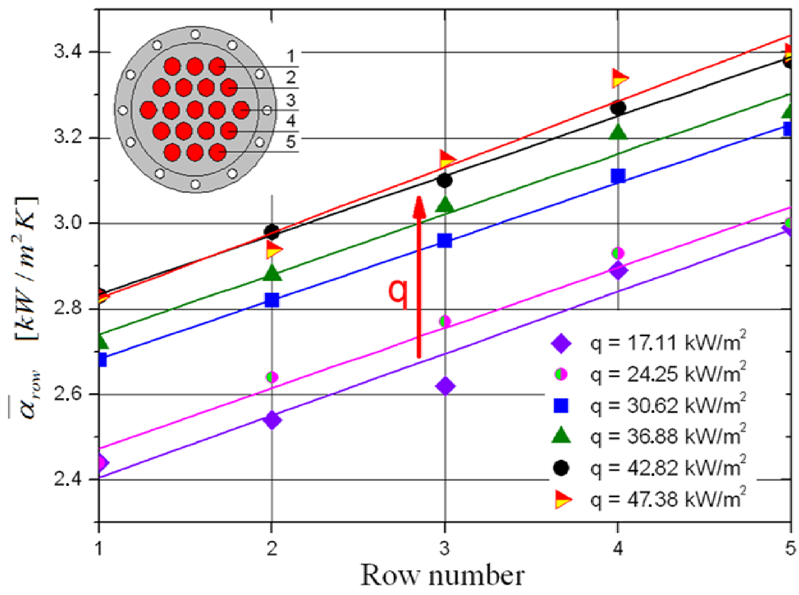

Fig. 9. Row average heat transfer coefficient for water boiling
on porous coated tube bundle of pitch-to-diameter ratio of 2.0 at atmospheric pressure

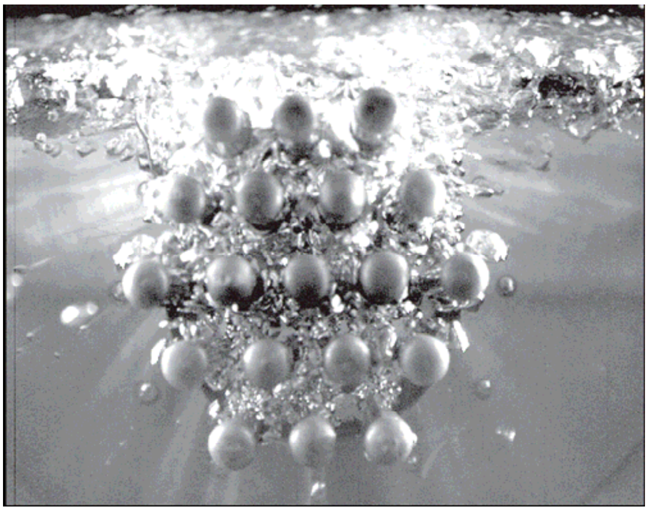

Fig. 10. Visualization of intensifying steam bubble effect for water boiling process on a porous coated tube bundle; *s/D = 1.7*, at atmospheric pressure *(p = 100.5 kPa)* and heat flux *q = 30.25 kW/m²*

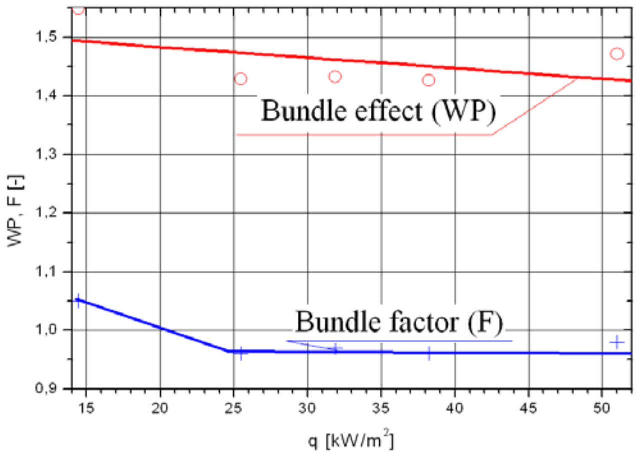

Fig. 11. Bundle factor (F) and bundle effect (WP) for methanol
boiling at atmospheric pressure on smooth bundle with pitch-to-diameter ratio of 1.7

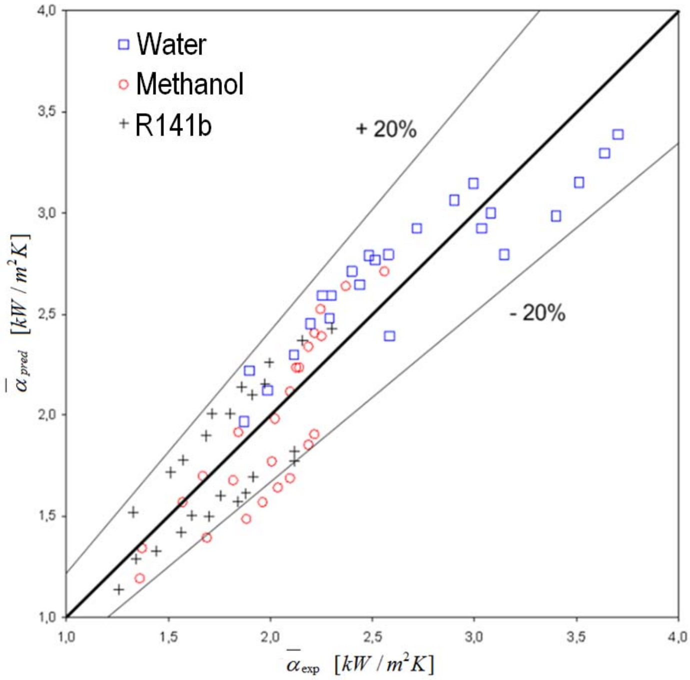

Fig. 12. Predicted vs. experimental average heat transfer coefficients
on porous coated tube bundle

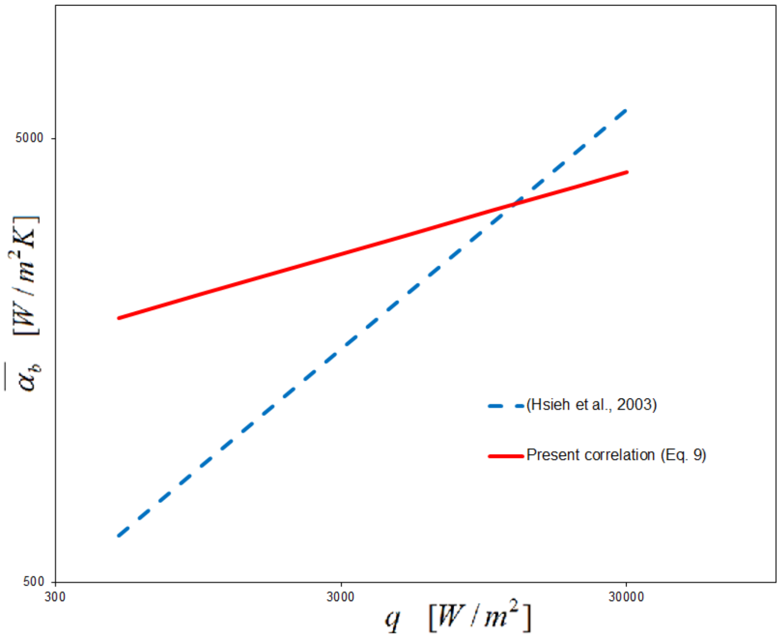

Fig. 13. Comparison of predicted and experimental data obtained by Hsieh et al. (2003)