UDK 539.4.016:621.785:620.195
Original scientific article/Izvirni znanstveni članek

ISSN 1580-2949
MTAEC9, 50(6)939(2016)

THE SIZE EFFECT OF HEAT-TRANSFER SURFACES ON BOILING

VPLIV VELIKOSTI POVRŠIN, KI PRENAŠAJO TOPLOTO NA VRENJE

Petr Kracík, Marek Balas, Martin Lisy, Jiří Pospíšil

Brno University of Technology, Institute of Power Engineering, Faculty of Mechanical Engineering,
Technická 2896/2, 616 69 Brno, Czech Republic
kracik@fme.vutbr.cz

Prejem rokopisa – received: 2015-07-31; sprejem za objavo – accepted for publication: 2015-12-14

doi:10.17222/mit.2015.245

A sprinkled tube bundle is frequently used in technology processes where an increase or decrease of a liquid temperature in a very low-pressure environment is required. Phase transitions of the liquid very often occur at low temperatures at pressures ranging in the thousands of pascals, which enhances the heat transfer. This paper focuses on the issue of a heat-transfer coefficient that is experimentally examined at the surface of a tube bundle. The tube is located in a low-pressure chamber where the vacuum is generated using an exhauster via an ejector. The tube consists of smooth copper tubes of 12 mm diameter placed horizontally one above another. Heating water flows in the bundle from the bottom towards the top at an average input temperature of approximately 40 °C and an average flow rate of approximately 7.2 L min^{-1}. A falling film liquid at an initial temperature of approximately 15 °C at an initial tested pressure of approximately 97 kPa (atmospheric pressure) is sprinkled onto the tubes' surface. Afterwards, the pressure in the chamber is gradually decreased. When reaching the minimum pressure of approximately 3 kPa (abs) the water partially evaporates at the lower part of the bundle. Consequently, the influence of the falling film liquid temperature increase is tested. This gradually leads to the boiling of water in a significant part of the bundle and the residual cooling liquid that drops back to the bottom of the vessel is almost not heated anymore. In this paper we present the influences of the size of the heat-transfer surfaces.

Keywords: sprinkled tube bundle, water, under pressure, heat transfer

Pršenje po snopu cevi se pogosto uporablja v tehnoloških procesih, kjer se zahteva povišanje ali zmanjšanje temperature tekočine v okolju z nizkim tlakom. Fazni prehod tekočine se pogosto pojavi pri nizkih temperaturah in tlakih v območju nekaj tisoč paskalov, kar vpliva na prenos toplote. Članek se nanaša na koeficient prenosa toplote, ki je eksperimentalno določen na površini snopa cevi. Cev je nameščena v nizkotlačni komori, kjer se vakuum ustvarja s pomočjo aspiratorja preko ejektorja. Snop cevi sestavljajo gladke bakrene cevi, premera 12 mm, ki so nameščene horizontalno ena nad drugo. Voda za ogrevanje teče v snop od spodaj proti vrhu s povprečno temperaturo okrog 40 °C in povprečno hitrostjo pretoka okrog 7,2 L min^{-1}. Padajoč vodni film z začetno temperaturo okrog 15 °C in z začetnim tlakom okrog 97 kPa (atmosferski tlak) prši po površini cevi. Tlak se v komori postopno znižuje. Ko doseže minimalni tlak okrog 3 kPa (absolutni tlak) voda delno izhlapi na spodnjem delu snopa. Posledično je preizkušen vpliv naraščanja temperature padajoče vode. To postopno privede do vrenja vode na večjem delu površine snopa in preostala hladilna tekočina, ki kaplja na dno posode, se skoraj ne segreje več. Članek predstavlja vpliv velikosti površine kjer se prenaša toplota.

Ključne besede: pršenje po snopu cevi, podtlak, prenos toplote

1 INTRODUCTION

Due to a decreasing supply of fossil fuels and their increasing price, the minimization of energy consumption becomes an important priority, followed by saving the primary fuel entering energy processes that are supposed to achieve the maximum efficiency possible, and last but not least, using so-called renewable sources of energy. Among these renewable sources, a biomass combustion technology is mainly used to generate thermal energy and electricity in the Czech Republic. Current research aims to reflect these demands well. For instance, research is conducted in the field of optimization of technology for wood mass preparation[1,2] before combustion or further utilization for pellets' production. At the Department of Power Engineering long-term research focuses, apart from other things, on the utilization of waste thermal energy, which is found, for example, in condensers at large energy units, for the possible generation of cool in absorption units.

One of the basic elements of an absorption circuit is an evaporator, inside of which the heat-carrier substance is sprayed onto a tube bundle. Due to a low pressure environment inside the container where the bundle is located the falling film liquid at the tube bundle boils. The heat necessary for boiling is derived from a cooled substance flowing inside the tube bundle.

Under ideal conditions water boils at the whole surface of an exchanger, but in practice it must be considered that in original spots of contact between the water and the exchanger wall the water will not boil at the tubes' surface, but the cooling liquid will merely be heated-up. This paper deals with this very situation – heat transfer behaviour when heating a sprinkled water film that boils in a low-pressure environment for a real tube bundle.

Research in the area of sprinkled exchangers can be divided into two major parts. The first part is research on heat transfer and a determination of the heat-transfer coefficient with respect to sprinkled tube bundles for various liquids, whether boiling or not. For water as the falling film liquid, they were, for example[3–7]. The second part is testing of the sprinkle modes for various tube diameters, tube pitches and tube materials and a determination of individual modes' interface. This area is mainly researched in[8–10].

All the results published so far for water as the falling film liquid apply to one to three tubes, for which the mentioned relations studied are determined in rigid laboratory conditions, defined strictly in advance. The sprinkled tubes were not viewed from the operational perspective where there are more tubes and various modes can occur in different parts with various heat-transfer values.

2 EXPERIMENTAL PART

For the purposes of examination of the heat transfer for sprinkled tube bundles a test apparatus was constructed (**Figure 1**). A tube bundle at which a heat transfer from a heated water flowing inside the tubes into a falling film liquid is studied is placed in a vessel where the low pressure is created by an exhauster through an ejector.

The test apparatus chamber is a cylindrical vessel with a length of 1.2 m with three apertures in which the tube bundle of an examined length of 940.0 mm is inserted. The tube bundle is installed in two fitting metal sheets which define the sprinkled area. The bundle consists of eight copper tubes of diameter 12.0 mm situated horizontally, one above another, with a distribution tube above them, with apertures of the diameter from 1.0 mm to 9.2 mm. The bundle can be operated using only the first four or six tubes too.

Two closed loops are connected to the chamber. A heating one and a sprinkling one. The heating liquid flowing inside the tubes is intended for an overpressure up to 1.0 MPa. The second loop contains flowing falling film liquid. There is a pump, a regulation valve, a flow meter and plate heat exchangers attached to both loops. The plate heat exchanger at the heating loop is connected to a gas boiler, which supplies heat to the heating liquid. The sprinkling loop uses two plate heat exchangers. In the first one the falling film liquid is cooled by cold drinking water from the water mains and the falling film liquid is cooled in the second exchanger by drinking water cooled in a cooler, which regulates the temperature up to 1.0 °C. In order to enable visual control, the heating loop also includes a manometer and a thermometer. The thermal status in individual loops is measured by wrapped unearthed T-type thermocouples on the agents' input and output from the vessel. All the thermocouples were calibrated in the CL1000 Series calibration furnace, which maintains a given temperature with an accuracy of ±0.15 °C. None of the thermocouples exceeded the error ±0.5 °C within the studied range from 28 °C to 75 °C. That is why the total error for the temperature measurement is set uniformly for all the thermocouples along the whole studied range ±0.65 °C.

There are three vacuum gauges measuring the low pressure. The first vacuum gauge is designed for visual control and is a mercury meter, the second one is a

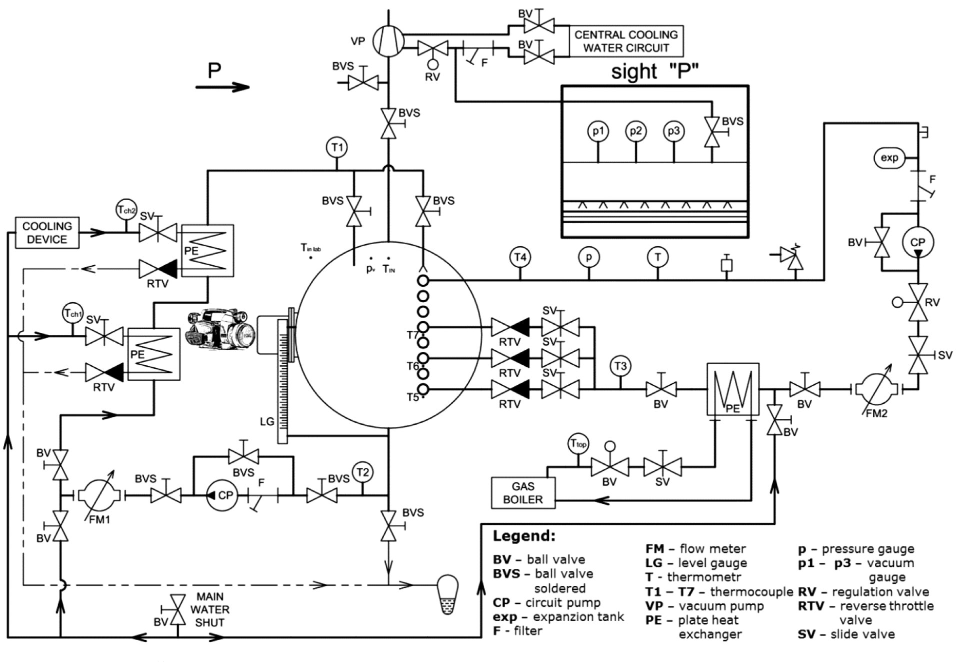

Figure 1: Measurement apparatus diagram
Slika 1: Shema merilnih naprav

digital vacuum gauge Baumer TED6 and enables measurements within the whole desired low-pressure range, but it is less accurate with lower pressure values. To allow a precise measuring of the low spectrum, a third digital vacuum gauge in the range 2.0 kPa to 0 Pa is used. The accuracy of a vacuum gauge, the results of which have been used for the assessment, is 0.5 % from the measured range, i.e., ±0.5 kPa.

Electromagnetic flow meters Flomag 3000 attached to both loops measure the flow rate. The flow meters' range is 0.0078 to 0.9424 L s^{-1}, where the accuracy is 0.5 % from the measured range, i.e. ±0.00467 L s^{-1}. All the examined quantities are either directly (thermocouples) or via transducers scanned by measuring cards DAQ 56 with a frequency of 0.703 Hz.

3 METHODOLOGY OF THE DATA ASSESSMENT

The assessment of the measured data is based on the thermal balance between the operating liquid circulating inside the tubes and a sprinkling loop according to the law of conservation of energy. Heat transfer is realized by convection, conduction and radiation. At lower temperatures the heat transferred by radiation is negligible; therefore, it is excluded from further calculations. The calculation of the studied heat-transfer coefficient is based on Newton's heat-transfer law and Fourier's heat-conduction law that have been used to form the following relation in Equation (1):

$$\alpha_0 = \frac{1}{2\pi r_0\left[\frac{1}{k_s} - \frac{1}{2\pi\alpha_i r_i} - \frac{1}{2\pi\lambda_s}\cdot\ln\left(\frac{r_0}{r_i}\right)\right]} \qquad (1)$$

where α_o [W m^{-2}K^{-1}] is the heat transfer coefficient at the sprinkled tubes' surface

α_i [W m^{-2} K^{-1}] is the heat-transfer coefficient at the inner side of a tube set for a fully developed turbulent flow according to[11],

r_o; r_i [m] are the outer and inner tube radii

λ_S [W m^{-1} K^{-1}] is the thermal conductivity

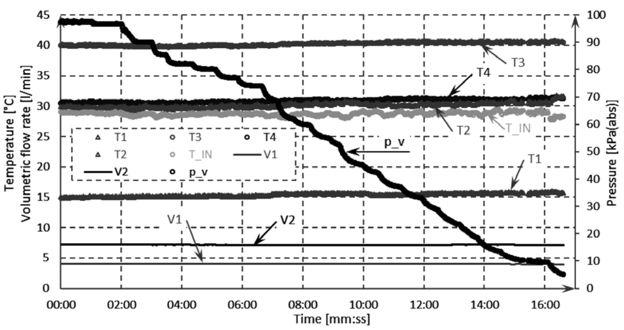

Figure 2: Record of the main measured quantities
Slika 2: Zapis glavnih izmerjenih količin

k_S [W m^{-1} K^{-1}] is the heat admittance based on the above-mentioned laws governing heat transfer, which is calculated from the heat balance of the heating side of the loop, which is why the following Equation (2) must be valid:

$$\dot{Q}_s = k_s L \Delta T_{ln} = \dot{M}_{34} c_p \left(\frac{t_3 + t_4}{2}\right)\cdot(t_3 - t_4) \qquad (2)$$

where M_{34} [kg s^{-1}] is the mass flow of heating water

c_p [J kg^{-1} K^{-1}] is the specific heat capacity of water at constant pressure related to the mean temperature inside the loop

L [m] is the total length of the bundle

ΔT_{ln} [K] is a logarithmic temperature gradient where a counter-current exchanger was considered

The final heat-transfer coefficient for various falling film liquid flow rates is recalculated into the Nusselt number, due to its more common application for a results comparison. For sprinkled exchangers the following form is normally used[3,4] for the recalculation of the heat-transfer coefficient into the Nusselt number:

$$Nu = \alpha_0 \sqrt[3]{\frac{\nu^2}{g\lambda^3}} \quad [-] \qquad (3)$$

where ν [m^2 s^{-1}] is the kinematic viscosity,

g [m s^{-2}] is the acceleration due to gravity and

λ [W m^{-1} K^{-1}] is the liquid film's thermal conductivity.

4 RESULTS

Tube bundles comprising four, six and eight copper tubes were tested and the results are given in this paper. Constant flow rates of the sprinkling liquid and the heating liquid of the required temperature were set first; the pressure in the chamber was then slowly lowered. The initial pressure in the chamber at that moment always equaled the atmospheric pressure.

Three temperature gradients of sprinkling water and heating water 20/40, 15/40 and 20/50 were tested. The first number designates the temperature of the sprinkling water in the distribution tube; the other number designates the temperature of the heating water entering the bundle. The thermal differences were (20, 25 and 30) °C. The flow rate of the heating water in all the experiments was kept at approximately 7.2 L min^{-1}. The flow rates of the sprinkling liquid were carefully selected and remained "constant" throughout the experiments.

Figure 2 shows a recording of the main quantities measured in one of the experiments. Only four tubes were heated in this experiment, which lasted 16 min and 37 s (horizontal axis of the chart). The average temperature of the heating water entering the bottom tube ($T3$) was 40.2 °C ± 0.3 °C, and the average flow rate of the heating water ($V2$) was 7.16 ± 0.03 L min^{-1}. The temperature of the heating water leaving the exchanger is designated as ($T4$) in the chart. The average temperature

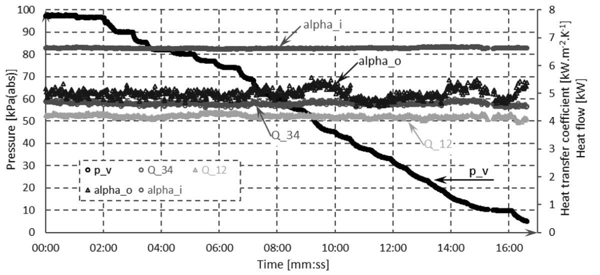

Figure 3: Quantities derived from the measurements
Slika 3: Količine pridobljene iz meritev

of the sprinkling water entering the distribution tube was 15.4 °C ± 0.3 °C, and average flow rate ($V1$) was 4.03 ± 0.02 L min^{-1}. The mass flow of the sprinkling liquid related to the length of the sprinkled area (which is 0.940 mm for all three exchangers) is more commonly used for the comparison. The said mass flow was 0.0713 ± 0.0004 kg/(s m), i.e., the average Reynolds number was 304.4 ± 2.1 [-]. Other values in the chart include the current pressure in the testing chamber (p) (in absolute values), and the temperature of the sprinkling water ($T2$) beyond the tube bundle where the water was heated.

Figure 3 shows the quantities derived from the measurements presented in the previous figure. The quantities in the chart depend on the duration of the experiment. The quantities represented in the chart include the current heat flow extracted from the heating water (Q_34), the current heat flow absorbed by the sprinkling liquid (Q_12), the heat-transfer coefficient on the inside tube wall (alpha_i) and the analyzed heat-transfer coefficient on the surface of the tube (alpha_o). The chart also presents the current pressure in the testing

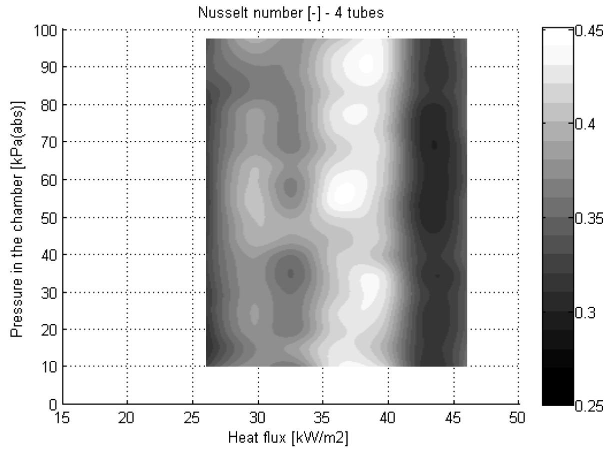

Figure 4: Dependency of the Nusselt number on the pressure in the testing chamber and the heat flow extracted from the heating liquid – four tubes
Slika 4: Odvisnost Nusseltovega števila od tlaka v preizkusni komori in toplotni tok pridobljen z tekočine za ogrevanje – štiri cevi

chamber. The course of the analysed heat-transfer coefficient clearly shows that decrease in pressure in the chamber results in a slightly increased coefficient. The coefficient depends mostly on the temperature of the heating water and the sprinkling water, and not so much on their flow rates.

Altogether 21 experiments on tube bundles with four, six and eight tubes were performed, that is seven experiments for each bundle. **Table 1** presents the measured parameters that helped identify, among others, the mass flow of the sprinkling liquid. The parameters were related to the length of the sprinkled area, the corresponding Reynolds number, and the heat flow extracted from the

Table 1: Core values of the experiments
Tabela 1: Temeljne vrednosti eksperimentov

tubes	T_1 (°C)	T_2 (°C)	T_3 (°C)	T_4 (°C)	V_1 (L/min)	V_2 (L/min)	Γ (kg/(s.m))	Re [-]	q (kW/m^2)
4	19.7 ± 0.1	32.1 ± 0.2	40.1 ± 0.1	32.7 ± 0.1	4 ± 0	7.2 ± 0	0.0711 ± 0.0005	326 ± 2.4	26.03 ± 0.38
	20.2 ± 0.1	29.4 ± 0.3	40.3 ± 0.3	31.7 ± 0.2	6 ± 0	7.2 ± 0	0.1065 ± 0.0004	476.9 ± 2.9	30.26 ± 0.64
	15.4 ± 0.2	30.2 ± 0.4	40.2 ± 0.3	30.9 ± 0.3	4 ± 0	7.2 ± 0	0.0713 ± 0.0004	304.3 ± 1.5	32.66 ± 0.29
	15.7 ± 0.1	28.5 ± 0.3	39.9 ± 0.4	29.9 ± 0.2	5 ± 0	7.2 ± 0	0.0889 ± 0.0003	373.4 ± 1.4	34.97 ± 0.78
	15.5 ± 0.2	27.3 ± 0.2	40.6 ± 0.1	29.8 ± 0.1	6 ± 0	7.1 ± 0	0.1065 ± 0.0006	440.3 ± 3.3	37.57 ± 0.4
	19.7 ± 0.2	38.4 ± 0.4	50.2 ± 0.2	38.8 ± 0.2	4 ± 0	7.2 ± 0	0.0709 ± 0.0005	348.4 ± 2.7	39.76 ± 0.48
	20.1 ± 0.2	33.6 ± 0.2	50.2 ± 0.2	37.1 ± 0.1	6 ± 0	7.2 ± 0	0.1064 ± 0.0005	498.9 ± 2.3	46.08 ± 0.46
6	20.2 ± 0.1	35.6 ± 0.2	40.6 ± 0.1	32.1 ± 0.1	4 ± 0	7.2 ± 0	0.0709 ± 0.0002	339.8 ± 1.6	19.94 ± 0.21
	19.6 ± 0.2	32.0 ± 0.2	39.7 ± 0.2	29.6 ± 0.3	6 ± 0	7.2 ± 0	0.1065 ± 0.0003	487.2 ± 2.9	23.8 ± 0.33
	15.4 ± 0.1	33.2 ± 0.3	40.4 ± 0.3	30.3 ± 0.2	4 ± 0.1	7.1 ± 0	0.0713 ± 0.0017	309.3 ± 7.9	23.71 ± 0.49
	15.5 ± 0.2	32.3 ± 0.2	40.2 ± 0.3	28.2 ± 0.2	5 ± 0	7.2 ± 0	0.0888 ± 0.0002	389.1 ± 1.7	28.2 ± 0.36
	15.4 ± 0.3	31.3 ± 0.5	40.5 ± 0.2	28.0 ± 0.3	6 ± 0	7.2 ± 0	0.1065 ± 0.0003	444.4 ± 3.5	29.03 ± 0.38
	20.2 ± 0.2	40.8 ± 0.8	50.4 ± 0.3	36.0 ± 0.2	5 ± 0	7.2 ± 0	0.0885 ± 0.0005	448.8 ± 5.2	34.03 ± 0.38
	19.9 ± 0.1	38.5 ± 0.5	49.5 ± 0.1	34.2 ± 0.1	6 ± 0	7.2 ± 0	0.1063 ± 0.0003	524.7 ± 3.6	35.64 ± 0.28
8	20.3 ± 0.1	37.2 ± 0.4	40.3 ± 0.2	31.1 ± 0.1	4 ± 0	7.2 ± 0	0.0709 ± 0.0005	346.5 ± 3.2	16.28 ± 0.31
	20.2 ± 0.1	34.7 ± 0.2	40.3 ± 0.1	28.9 ± 0.1	6 ± 0	7.2 ± 0	0.1064 ± 0.0004	505 ± 2.3	19.94 ± 0.13
	15.3 ± 0.2	36.3 ± 0.3	40.2 ± 0.1	28.7 ± 0.2	4 ± 0	7.2 ± 0	0.071 ± 0.0003	324.7 ± 2.5	20.25 ± 0.22
	15.6 ± 0.3	34.1 ± 0.4	39.9 ± 0.4	27.1 ± 0.3	5 ± 0	7.2 ± 0	0.0887 ± 0.0003	397 ± 2.9	22.3 ± 0.53
	15.3 ± 0.5	32.8 ± 0.6	39.7 ± 0.3	26.1 ± 0.5	6 ± 0	7.2 ± 0	0.1063 ± 0.0006	467.4 ± 6.5	23.82 ± 0.51
	20.0 ± 0.2	45.8 ± 1.2	50.4 ± 0.2	36.3 ± 0.4	4 ± 0	7.2 ± 0	0.0709 ± 0.0003	378 ± 3.5	24.93 ± 0.81
	20.2 ± 0.1	41.2 ± 0.7	50.0 ± 0.3	33.0 ± 0.3	6 ± 0	7.2 ± 0	0.1065 ± 0.0003	542.2 ± 4.2	29.73 ± 0.27

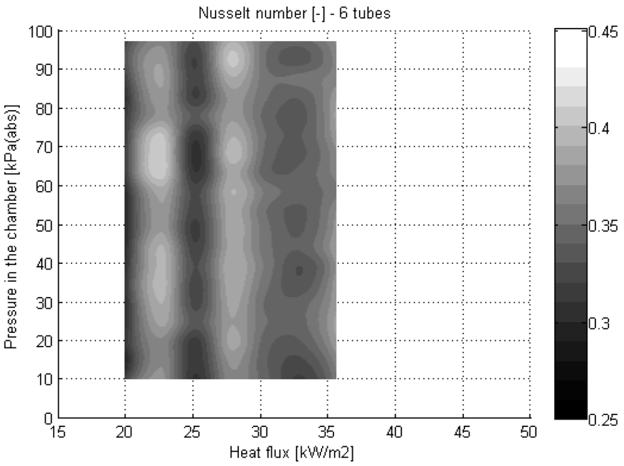

Figure 5: Dependency of the Nusselt number on the pressure in the testing chamber and heat flow extracted from the heating liquid – six tubes

Slika 5: Odvisnost Nusseltovega števila od tlaka v preizkusni komori in toplota pridobljena iz tekočine za ogrevanje – šest cevi

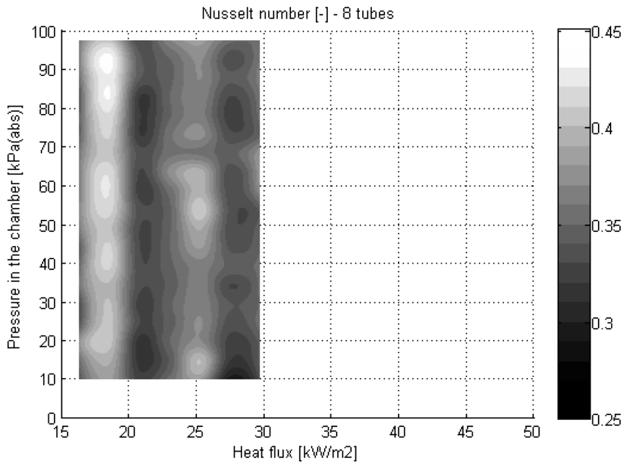

Figure 6: Dependency of the Nusselt number on the pressure in the testing chamber and the heat flow extracted from the heating liquid – eight tubes

Slika 6: Odvisnost Nusseltovega števila od tlaka v preizkusni komori in toplota pridobljena iz tekočine za ogrevanje – osem cevi

heating water, which was related to the size of the heat-exchanging surface.

The heat-transfer coefficient on the surface of the tube bundle was later converted to the Nusselt number. Values acquired at pressures ranging from 5.0 kPa(abs) to 95.0 kPa(abs) in 5.0 kPa increments, and values acquired at atmospheric pressure were selected from the data measured in each experiment. The values were then averaged for each pressure. This helped us to develop matrices of the same sizes for a particular bundle length. The matrices were interpolated using a cubic curve in Matlab software. Contour graphs, shown in **Figures 4** to **6**, were developed using these new matrices. The presented Nusselt number in the figures is dependent on the pressure in the chamber (vertical axis) and the extracted heat flow of the heating water (horizontal axis). The range of the Nusselt number is identical for all three charts, i.e., 0.15 to 0.46 [-], so that individual phases can be compared. However, this procedure was disabled to observe slight increases in the Nusselt number as the pressure decreases in particular measurement sessions.

All three analysed sizes of the tube bundle clearly show three vertical zones that do not connect; these are a result of a different temperature gradient. The temperature gradient is 20/40 in the zone 1, 15/40 in the zone 2, and 20/50 in the zone 3. Temperature gradient 15/40 with the highest Nusselt number seems to be the best option for all three analysed lengths of the bundle. The Nusselt number dropped by almost 0.2 [-] when the temperature of heating water increased. The Nusselt number also drops at very low pressures. The decrease is caused by the fact that heat extracted from the heating liquid is also used for evaporation and not only for heating the heating water. However, the decrease is not very significant and the boiling occurs only at the bottom section of the tube bundle.

Concerning a tube bundle with four tubes, the maximum Nusselt number was attained at a specific heat flow of approximately 35 kW m^{-2} and 38 kW m^{-2}, which corresponds to a temperature gradient of 15/40 °C. The maxima ranged at values around 0.44 [-]. The larger the surface area and the lower the specific loading, the higher the values. In the case of a tube bundle with six tubes, the heat flow is approximatelz 28 kW m^{-2}. In the case of a tube bundle with eight tubes, the maximum reached almost 0.46 [-] at a thermal load of approximately 24 kW m^{-2}.

5 CONCLUSIONS

Sprinkled tube bundles are most commonly used as evaporators since they quickly separate the vapor phase and the liquid phase thanks to a thin liquid film. However, the practice proves that there is no boiling on the first affected tubes, only the liquid is heated. We describe in this paper how the surface area of the tubes affects the heat transfer coefficient on the tube surface. This effect was tested at three thermal differences: (15, 20 and 30) °C. After the flow rate of the heating and sprinkling liquids and their temperatures were set, only the pressure in the testing chamber was changed during the experiment. The initial pressure in the chamber was atmospheric pressure.

The pressure in the chamber and the exhaustion of the air-vapour mixture from the chamber (vapour was formed at the bottom part of the tube bundle at low pressures) had no major impact on the coefficient. It was the exhaustion itself which affected the boundary layers on the tube bundle. A significant change of the coefficient may come only if the boiling on the tube bundle increases.

The impact of the heat-exchanging surface area on the heat-transfer coefficient is evident in zone 3 of particular bundles where the thermal gradient was 20/50 °C. The increase in the surface area results in a decrease in the specific loading of the area but the heat-transfer coefficient increases.

Acknowledgement

Presented results were obtained in frame of the project NETME CENTRE PLUS (LO1202), created with financial support from the Ministry of Education, Youth and Sports of the Czech Republic under the "National Sustainability Programme I".

6 REFERENCES

[1] J. Beniak, J. Ondruška, V. Čačko, Design Process of Energy Effective Shredding Machines for Biomass Treatment, Acta Polytechnica, 52 (**2012**) 5, 1210–2709

[2] J. Beniak, P. Križan, M. Matúš, M. Kováčová, The Operating Load of a Disintegration Machine, Acta Polytechnica, 54 (**2014**) 1, 1210–2709, doi:10.14311/AP.2014.54.0001

[3] L. H. Chien, Ch. H. Cheng, A Predictive Model of Falling Film Evaporation with Bubble Nucleation on Horizontal Tubes, HVAC, 12 (**2006**) 1, 1078–9669, doi:10.1080/10789669.2006.10391168

[4] J. J. Lorenz, D. Yung, A Note on Combined Boiling and Evaporation of Liquid Films on Horizontal Tubes, Journal of Heat Transfer, 101 (**1979**) 1, 0022–1481, doi:10.1115/1.3450914

[5] W. L. Owens, Correlation of Thin Film Evaporation Heat Transfer Coefficients for Horizontal Tubes, Proceedings of the Fifth Ocean Thermal Energy Conversion Conference, Miami Beach, Florida, 1978

[6] W. H. Parken, L. S. Fletcher, V. Sernas, J. C. Han, Heat Transfer Through Falling Film Evaporation and Boiling on Horizontal Tubes, Journal of Heat Transfer, 112 (**1990**) 3, 0022–1481, doi:10.1115/1.2910449

[7] V. Sernas, Heat Transfer Correlation for Subcooled Water Films on Horizontal Tubes, Journal of Heat Transfer, 101 (**1979**) 1, 0022–1481, doi:10.1115/1.3450913

[8] R. Armbruster, J. Mitrovic, Patterns of Falling Film Flow over Horizontal Smooth Tubes, Proceedings of the 10th international heat transfer conference, Brighton, UK, 1994, 3

[9] X. Hu, A. M. Jacobi, The Intertube Falling Film: Part 1 - Flow Characteristics, Mode Transitions, and Hysteresis, Journal of Heat Transfer, 118 (**1996**) 3, 0022–1481, doi:10.1115/1.2822678

[10] J. F. Roques, V. Dupont, J. R. Thome, Falling Film Transitions on Plain and Enhanced Tubes, Journal of Heat Transfer, 124 (**2002**) 3, 0022–1481, doi:10.1115/1.1458017

[11] M. Jícha, Heat and mass transfer, Brno, CERM, 2001, 1, 160