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Development boiling to sprinkled tube bundle

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Abstract. This paper presents results of a studied heat transfer coefficient at the surface of a sprinkled tube bundle where boiling occurs. Research in the area of sprinkled exchangers can be divided into two major parts. The first part is research on heat transfer and determination of the heat transfer coefficient at sprinkled tube bundles for various liquids, whether boiling or not. The second part is testing of sprinkle modes for various tube diameters, tube pitches and tube materials and determination of individual modes' interface. All 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 may occur in different parts with various heat transfer values. The article focuses on these processes. The tube is located in a low-pressure chamber where vacuum is generated using an exhauster via ejector. The tube consists of smooth copper tubes of 12 mm diameter placed horizontally one above another.

1 Mathematical model of heat transfer of boiling liquid

Lorenz and Yung created a mathematical model published at **[1]** in 1979 for prediction of a heat transfer coefficient at the surface of a single sprinkled horizontal tube using a simplified geometry. The results of for instance **[2, 3, 4]** were compared with this model. The mathematical calculation considers a single vertical half of a horizontal tube that has been unrolled.

The mathematical model further takes into account that the temperature of the liquid falling on a heated tube equals the temperature of a saturated liquid. On these grounds two basic regions are formed at the tube during consequent heating. The first one is a transition convective region where boiling develops and the second one represents a fully developed boiling where evaporation occurs. The total heat transfer coefficient at the surface of a sprinkled horizontal tube equals a sum of partial heat transfer coefficients [at nucleate boiling effect (α_{ob}), at thermal developing region (α_{od}) and at fully developed region (α_{oc})] which are defined below and corrected by length for which their mean value has been set against the total studied length (of the unrolled area)

$$\bar{\alpha}_o = \alpha_{ob} + \alpha_{od} \cdot \frac{L_d}{L} + \alpha_{oc} \cdot \left(1 - \frac{L_d}{L}\right) \qquad (1)$$

The heat transfer coefficient at nucleate boiling effect (also referred to as boiling nucleus) is based on a sufficient heat transfer at which the liquid starts to boil. According to **[1]** higher heat transfers with boiling temperature are achieved in thin liquid films than in steam environment and a conservative approach is the application of a relation defined by Rohsenow **[5]** who uses in his equation besides other things latent heat [enthalpy of vaporization l_v (J·kg^{-1})]

$$\alpha_{ob} = \frac{\mu \cdot l_v}{C_{sf}^3 \cdot \sqrt{\frac{g_0 \cdot \sigma}{g \cdot \rho}}} \cdot \left(\frac{c_p}{l_v \cdot Pr}\right)^3 \cdot \Delta T^2 \qquad (2)$$

and parameter C_{sf} is a function of the surface tension of the liquid and of the surface which the liquid flows down from. Parameter C_{sf} for water and various surfaces is for example in **[6, tab. 9.1]**.

Cooper (1984) **[7]** created relatively simple relation for experimental data of heat transfer at nucleate boiling effect that is only a function of reduced pressure, molar mass and roughness of the surface of a sprinkled tube bundle. Cooper formed several identical functions. One of them applies to common logarithm

$$\alpha_{ob} = \left\{90 \cdot \frac{\Delta T_{ws}^{0.67}}{\sqrt{M}} \cdot \frac{p_r^m}{[-\log_{10}(p_r)]^{0.55}}\right\}^{3.0303} \qquad (3)$$

where p_r (–) is a reduced pressure, i.e. pressure in the tube bundle's environment divided by critical pressure of the liquid that flows around the tube bundle. ΔT_{ws} (K) stands for the overheating temperature, i.e. the temperature of the tube wall is deducted by the temperature of saturation of the liquid flowing around it at a given environment pressure. The capital "M (kg·kmol^{-1})" represents molar mass and the exponent m (–) is dependent on the roughness of the tube surface

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$$m = 0.12 - 0.2 \cdot \log_{10}\left(\frac{Ra}{0.4}\right) \qquad (4)$$

Ra (μm) stands for an average (mean) arithmetical deviation of the studied profile of the given surface calculated from the roughness profile. More in **[8]**

The thermal developing region where the heat is used for liquid overheating and where boiling does not occur yet, but evaporation does, is limited by the length L_d (m) **[1]**

$$L_d = \frac{c_p \cdot \sqrt[3]{\Gamma^4}}{4 \cdot \pi \cdot \rho \cdot \lambda} \cdot \sqrt{\frac{3 \cdot \mu}{g \cdot \rho^2}} \qquad (5)$$

this equation is derived for laminar to slightly turbulent flow and the mean value of a heat transfer coefficient at the tubes' surface in this region is **[1]**

$$\alpha_{od} = \frac{3}{8} \cdot c_p \cdot \frac{\Gamma}{L_d} \qquad (6)$$

which is based on energy balance.

For a fully developed region at a vertical smooth tube the following two equations are recommended by **[1]** for a calculation of a heat transfer coefficient at laminar (and also pseudo laminar) and at turbulent flow according to Chun and Seban **[9]**

$$\alpha_{oc,lam} = 0.821 \cdot \sqrt[3]{\frac{\lambda^3 \cdot g}{\nu^2}} \cdot \sqrt[0.22]{\frac{\mu}{4 \cdot \Gamma}} \qquad (7)$$

$$\alpha_{oc,tur} = 0.003\,8 \cdot \sqrt[3]{\frac{\lambda^3 \cdot g}{\nu^2}} \cdot \sqrt[0.4]{\frac{4 \cdot \Gamma}{\mu}} \cdot \sqrt[0.65]{\frac{\nu}{a}} \qquad (8)$$

The laminar mode considers the influence of ripple and waves as well that have an effect on the increase of heat transfer and the decrease of film's thickness. Both equations are a function of the Reynolds and Prandtl number and they should be applied for a condition at a constant heat transfer or at constant thermal conditions at the tube wall. Correlation between the equations (7) and (8) creates a transition boundary defined as a functional dependence of the Reynolds number on the Prandtl number

$$Re_{f,tr} > \frac{5\,800}{Pr^{1.06}} \qquad (9)$$

This equation does not define a real boundary between pseudo laminar and turbulent mode, but the transition between the equations (7) and (8) only.

2 Mathematical model for two-phase flow

Chien and Cheng (2006) **[3]** state that the model of Chun and Seban (1971) **[9]** does not comply well with practically measured results for various coolants, i.e. shows a significant error. For these reasons they suggest applying this model, which draws on the fact that the calculation of heat transfer coefficient at a tube surface should also consider occurrence of bubbles, to a horizontal tube. The model is based on the superposition principle, i.e. the final heat transfer coefficient consists of multiple elements which add up to create its resultant. This model has been developed by Chen (1966) **[10]** and it is given by the basic equation

$$\alpha_o = S \cdot \alpha_{ob} + \alpha_{cv} = S \cdot \alpha_{ob} + E \cdot h_l \qquad (10)$$

which is identical also for flow inside a channel, where S (–) represents the suppression factor of heat transfer coefficient at nucleate boiling effect (α_{ob}). The factor is determined by the equation (3). A modified form of the mentioned equation which is defined by Cooper **[7]** can be used which uses natural logarithm and the overheating temperature is replaced by the heat transfer density.

$$\alpha_{ob} = 93.5 \cdot \frac{\dot{q}^{0.67}}{\sqrt{M}} \cdot \frac{p_r^{\,0.12-0.434\,3\cdot\ln\left(\frac{Ra}{0.4}\right)}}{[-0.434\,3 \cdot \ln(p_r)]^{0.55}} \qquad (11)$$

For the equation (11) Chien and Cheng [3] state the deviation ±10% at the largest height of protrusions Ra=0.1 μm.

The quantity α_{cv} in the equation (10) is a convective heat transfer coefficient that equals the product of a two-phase convection enhancement factor E (–) and the heat transfer coefficient for a single-phase liquid flow. Chien and Cheng (2006) **[3]** suggest to use the following functions to calculate the suppression factor and the two-phase convection enhancement factor

$$S = \frac{1}{1 + 2.56 \cdot 10^{-6} \cdot \left(Re_f \cdot E^{1.25}\right)^{1.17}} \qquad (12)$$

$$\left.\begin{aligned} E &= 1\,[-] && for\ \frac{1}{X_{tt}} < 1\,[-] \\ E &= 2.35 \cdot \left(0.213 + \frac{1}{X_{tt}}\right)^{0.736} && for\ \frac{1}{X_{tt}} \geq 1\,[-] \end{aligned}\right\} \qquad (13)$$

where X_tt [–] is the Martinelli parameter

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.875} \cdot \left(\frac{\mu_f}{\mu_g}\right)^{0.125} \cdot \left(\frac{\rho_g}{\rho_f}\right)^{0.5} \qquad (14)$$

3 Measurement apparatus

For the purposes of examination of the heat transfer at sprinkled tube bundles a test apparatus has been constructed; see the diagram at Fig. 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 low pressure is created by an exhauster through an ejector.

The test apparatus chamber is a cylindrical vessel of the length 1.2 m with three apertures in which the tube bundle of the examined length 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 the 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 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.

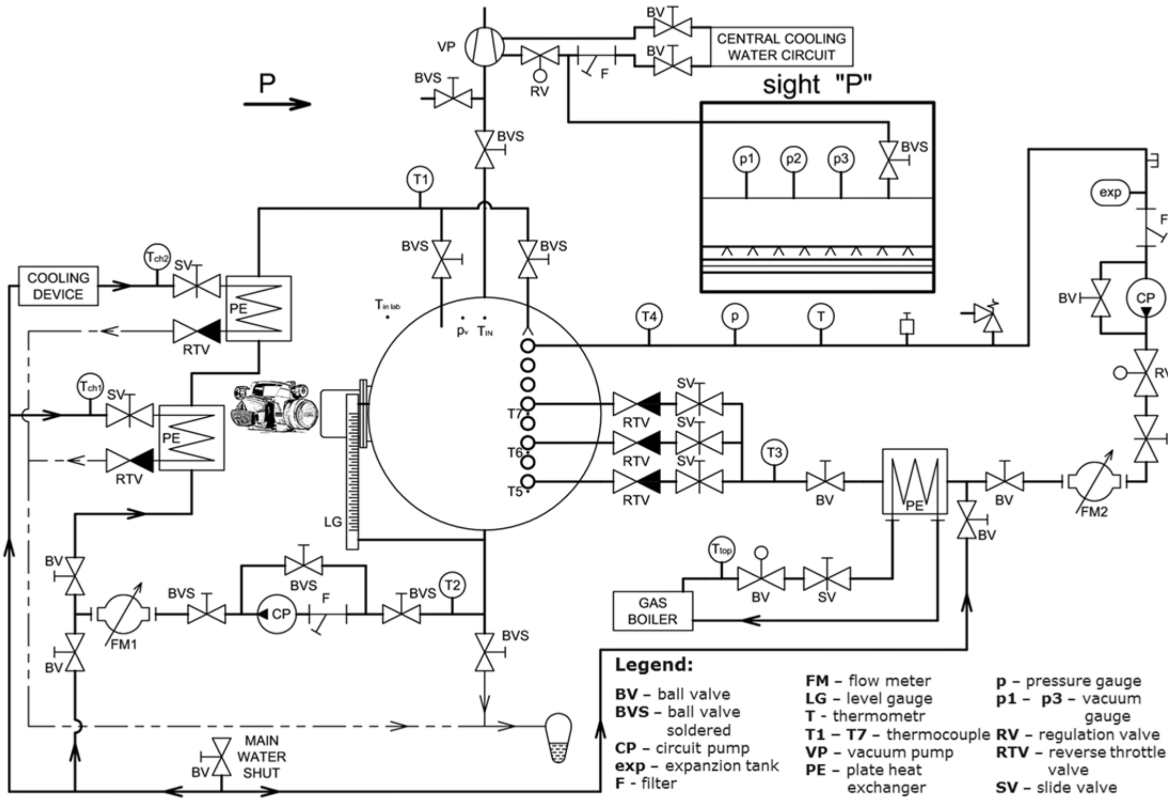

Figure 1. Measurement Apparatus Diagram

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 a cold drinkable water from water mains and the falling film liquid is cooled in the second exchanger by a drinkable 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 thermocouples have been calibrated in the CL1000 Series calibration furnace which maintains a given temperature with the accuracy ±0.15 °C. None of the thermocouples has exceeded the error ±0.5 °C within the studied range from 28 °C to 75 °C. That is why the total error at temperature measurement is set uniformly for all 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 it is a mercury meter, the second one is a digital vacuum gauge Baumer TED6 and enables measuring 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 of the range 2.0 kPa – 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 0.007 8 ÷ 0.942 4 $l \cdot s^{-1}$, where the accuracy is 0.5% from the measured range, i.e. ± 0.004 67 $l \cdot s^{-1}$. All examined quantities are either directly (thermocouples) or via transducers scanned by measuring cards DAQ 56.

4 Methodology of data assessment

The assessment of the measured data is based on the thermal balance between the operation 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. In 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 the Newton's heat transfer law and Fourier's heat conduction law that have been used to form the following relation

$$\alpha_o = \frac{1}{2 \cdot \pi \cdot r_o \cdot \left[\frac{1}{k_S} - \frac{1}{2 \cdot \pi \cdot \alpha_i \cdot r_i} - \frac{\ln\left(\frac{r_o}{r_i}\right)}{2 \cdot \pi \cdot \lambda_S} \right]} \qquad (15)$$

where α_o ($W \cdot m^{-2} \cdot K^{-1}$) is the heat transfer coefficient at the sprinkled tubes' surface, α_i ($W \cdot m^{-2} \cdot K^{-1}$) is the heat transfer coefficient at the inner side of a tube set for a fully developed turbulent flow according to [11, page 59 and 138], r_o; r_i (m) are outer and inner tube radii, λ_S ($W \cdot m^{-1} \cdot K^{-1}$) is thermal conductivity and k_S ($W \cdot m^{-1} \cdot K^{-1}$) is heat admittance based on the above mentioned laws governing heat transfer which is calculated from heat balance of the heating side of the loop, that is why the following must be valid:

$$\dot{Q}_{34} = k_S \cdot L \cdot \Delta T_{ln} = \dot{M}_{34} \cdot c_p \left(\frac{t_3 + t_4}{2}\right) \cdot (t_3 - t_4) \qquad (16)$$

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 and ΔT_{ln} (K) is a logarithmic temperature gradient where a counter-current exchanger was considered

The above mentioned calculation is valid only for heat transfer from water flowing inside the tubes into a falling film liquid which does not boil at the surface or where insignificant boiling may occasionally occur at the last tube. In case boiling occurs at a major part of the tube bundle, i.e. the tube bundle is sprinkled by a saturated water or higher at a corresponding pressure in a chamber p_V (Pa), two-phase flow should be considered. The amount of developed steam can be calculated according to the law of conservation of mass. I.e. the sum of the mass flow rate of steam M_v (kg·s^{-1}) and the mass flow rate of the water flowing down after the exchanger M_f (kg·s^{-1}) must equal the mass flow rate flowing from a distribution tube M_1 (kg·s^{-1}). However, there are two unknown variables in this equation. That is why it is necessary to apply the law of conservation of energy as well. I.e. the sum of the energy necessary to develop steam Q_v (W) and the energy transferred to a falling film water Q_f (W) must equal the energy supplied by the heating liquid Q_{34} (W). Then the following is valid for the calculation of the amount of developed steam:

$$\dot{M}_v = \frac{\dot{Q}_{34} - \dot{M}_1 \cdot [i_2(t_2; x=0) - i_1(t_1; x=0)]}{i_{IN}(t_{IN}; p_v) - i_2(t_2; x=0)} \qquad (17)$$

under the condition that the temperature of the falling film water t_1 (°C) in a distribution tube is max. at the saturation boundary or slightly below it. In the mentioned equation, i_{IN} (J·kg^{-1}) stands for the enthalpy of a developed steam cushion at the temperature t_{in} (°C), and i_2 (J·kg^{-1}) stands for the enthalpy of a heated falling film water at the temperature t_2 (°C), after the last tube the test tube bundle.

The studied heat transfer coefficient can be determined on the basis of the Newton's law of heat transfer where the wall temperature is set as a mean temperature of the heating water and the temperature of the liquid flowing around it as a mean value of the temperature of a falling film water flowing out of the distribution tube and the temperature of a steam cushion.

$$\left.\begin{aligned} \dot{Q}_v &= \alpha_{o,v} \cdot S \cdot (T_w - T_\infty) \\ \Rightarrow \alpha_{o,v} &= \frac{\dot{M}_v \cdot [i_{IN}(t_{IN}; p_v) - i_1(t_1; x=0)]}{S \cdot \left(\frac{t_3 + t_4}{2} - \frac{t_{in} + t_1}{2}\right)} \end{aligned}\right\} \qquad (18)$$

Where S (m^2) stands for a heat-transfer region, i.e. the surface of the tube bundle.

5 Experiments results

Experiments published in this paper have focused on the influence of a heat-exchanging surface size on boiling development at a sprinkled tube bundle at two thermal levels. A tube bundle has been assembled consisting of eight tubes positioned horizontally one above another where only the first four, six or all eight tubes were heated. The input heating liquid temperature (t_3) was maintained at the level of approx. 40°C or 50°C with one degree centigrade deviation and at the flow rate (V_1) of 7.2 litres per minute with a deviation up to 0.1 litre per minute.

The falling film liquid flow rate was adjusted and maximum low pressure was created in the vessel by an exhauster for each experiment. The difference in pressures at individual thermal levels was given by the fact that with more vapour generation of a higher thermal level the exhauster's absorption capacity decreased. The heat source itself was regulated by the temperature of the falling film water at the distribution tube's output. Summary of basic data gained by measurement is provided in Table 1. Apart from the quantities mentioned above the table also contains falling film liquid mass flow rate related to the sprinkled zone length (Γ_1), Reynolds number of the liquid phase (Re_f), Prandtl number of the liquid phase (Pr_f), pressure in a chamber (p_v) during the whole experiment, specific heat transfer derived from the heating liquid for boiling (q“) that was determined on the basis of the law of conservation of energy and mass, and a total heat transfer derived from the heating liquid (q).

Due to the fact that the waveform of a falling film liquid heating on the bundle at tested flow rates is approximately linear, it was possible to apply linear approximation to find the borderline of saturated liquid at the bundle. Figure 2 shows four sets of points in relation to the Prandtl number of the liquid phase and on the falling film liquid temperature at the distribution tube's output, deducted by the temperature of saturation corresponding to the current pressure in a chamber. Points "0%" mark a state when the falling film water temperature flowing out of the exchanger reaches a saturation borderline. Points "50%" mean that the saturation state occurs between the 4th and 5th tube in case of an 8-tube bundle etc. The results for the Prandtl number up to approx. 4.7 come from experiments where the heating water at the bundle's input was approx. 50°C. Higher values of the Prandtl number correspond to the heating water of approx. 40°C.

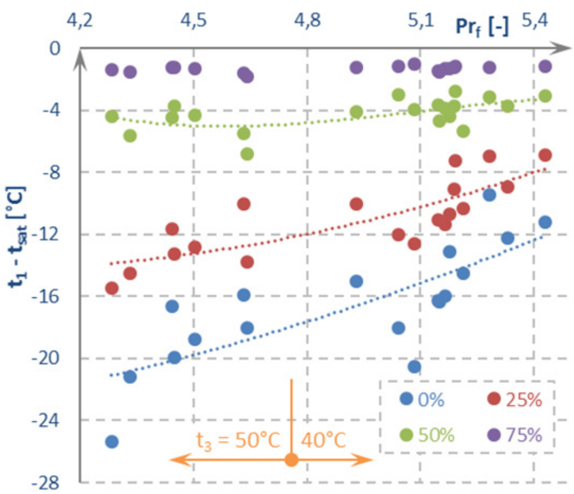

Figure 2. Boiling Development at Sprinkled Tube Bundle

Table 1. Basic information about the experiments

	t_3 (°C)	V_1 (l/min)	Γ_1 (kg/(s.m))	Re_f (-)	Pr_f (-)	p_v (kPa)	q" (kW/m²)	q (kW/m²)
4 tubes	40.4 ± 0.3	4.04 ± 0.03	0.0713 ± 0.0005	361.8	5.3	4.4 ± 0.1	3.8 ± 0.9	12.1 ± 2.5
	40.4 ± 0.3	6.03 ± 0.01	0.1063 ± 0.0003	544.5	5.3	4.5 ± 0.2	3.4 ± 1.0	12.4 ± 3.5
	40.3 ± 0.4	4.03 ± 0.00	0.0710 ± 0.0001	369.0	5.2	4.7 ± 0.3	4.4 ± 0.9	10.4 ± 2.0
	40.4 ± 0.2	6.03 ± 0.01	0.1063 ± 0.0002	531.0	5.4	4.2 ± 0.2	4.8 ± 1.2	14.5 ± 3.8
	50.3 ± 0.4	4.03 ± 0.01	0.0708 ± 0.0003	405.5	4.6	6.2 ± 0.4	8.0 ± 1.6	18.3 ± 3.9
	50.1 ± 0.3	6.03 ± 0.02	0.1062 ± 0.0005	610.5	4.6	6.2 ± 0.5	7.3 ± 1.4	19.8 ± 4.5
6 tubes	40.6 ± 0.2	4.03 ± 0.02	0.0710 ± 0.0003	367.3	5.2	4.7 ± 0.1	3.4 ± 0.6	7.6 ± 1.1
	40.5 ± 0.2	6.02 ± 0.02	0.1062 ± 0.0003	552.1	5.2	4.7 ± 0.2	3.5 ± 1.1	9.4 ± 2.7
	40.3 ± 0.3	4.06 ± 0.01	0.0715 ± 0.0001	374.4	5.1	4.8 ± 0.2	2.7 ± 0.5	6.8 ± 1.9
	40.5 ± 0.3	5.02 ± 0.01	0.0885 ± 0.0003	462.0	5.2	4.8 ± 0.2	3.2 ± 0.8	8.3 ± 2.5
	50.2 ± 0.4	4.03 ± 0.03	0.0710 ± 0.0006	416.7	4.5	6.7 ± 0.5	6.8 ± 1.6	12.6 ± 2.8
	50.1 ± 0.5	5.01 ± 0.02	0.0881 ± 0.0003	524.2	4.5	6.9 ± 0.3	6.5 ± 1.4	13.5 ± 2.7
	50.4 ± 0.2	6.05 ± 0.01	0.1064 ± 0.0002	633.9	4.4	7.0 ± 0.6	7.3 ± 1.6	15.2 ± 3.4
8 tubes	40.5 ± 0.2	4.02 ± 0.01	0.0708 ± 0.0001	370.2	5.2	4.8 ± 0.2	2.5 ± 0.7	6.6 ± 1.7
	39.9 ± 0.7	4.02 ± 0.04	0.0708 ± 0.0007	375.2	5.1	5.0 ± 0.2	1.9 ± 1.0	5.5 ± 2.1
	40.2 ± 0.5	5.04 ± 0.01	0.0887 ± 0.0001	473.8	5.0	5.1 ± 0.2	1.8 ± 0.8	5.3 ± 1.9
	40.4 ± 0.2	6.03 ± 0.02	0.1063 ± 0.0003	554.4	5.2	4.7 ± 0.2	2.1 ± 0.8	6.8 ± 2.4
	40.4 ± 0.5	8.08 ± 0.00	0.1423 ± 0.0001	776.5	4.9	5.4 ± 0.2	1.4 ± 0.6	5.4 ± 1.8
	50.2 ± 0.3	4.04 ± 0.04	0.0710 ± 0.0007	436.4	4.3	7.6 ± 0.3	4.4 ± 0.8	9.7 ± 1.6
	50.3 ± 0.3	6.02 ± 0.02	0.1058 ± 0.0003	645.9	4.3	7.4 ± 0.5	4.2 ± 0.8	10.7 ± 2.3

Figure 2 clearly shows that at the temperature difference of -3 and more, the boiling occurs at more than 60% of the tube bundle's height (considered from the last tube upwards). Simultaneously, the heat transfer coefficient was being stabilized at this value. That is why an average value and a standard deviation have been calculated starting at this borderline up for the purpose of further comparison. Figure 3 shows heat transfer coefficients of the vapour phase that have been determined according to the equation (18) in relation to the Reynolds number. Red tones represent an 8-tube bundle, green tones represent a 6-tube bundle and blue tones represent a 4-tube bundle. A lighter tone displays t_3 = 50°C and a darker tone displays t_3 = 40°C. This legend is applied to the following diagrams too, with the exception of the last one.

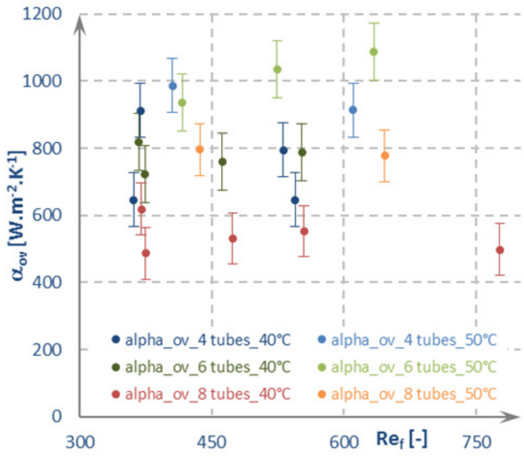

Figure 3. Heat Transfer Coefficient of Vapour Phase

A deviation is highlighted at individual points that is uniform at the given number of tubes. It was determined according to the average value of individual deviations increased by a standard deviation of this average value. Deviations ranged between 7 and 25%.

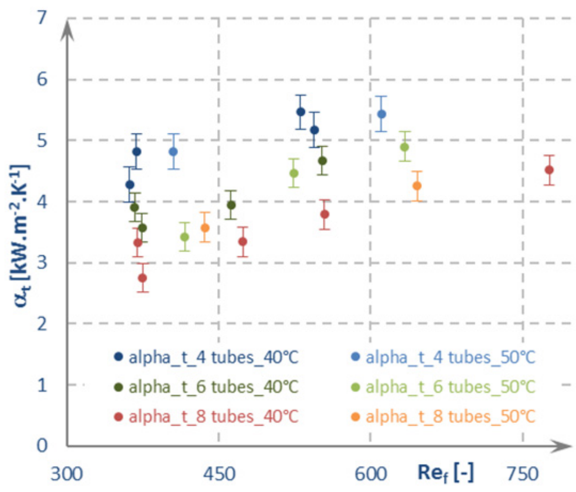

Figure 4. Total Heat Transfer Coefficient

The liquid phase coefficient given by the equation (15) as well as the vapour phase coefficient is derived from total energies at the input and output of the bundle. For this reason the coefficients are valid for the whole bundle and they represent its mean value. On the basis of the superposition principle it is therefore possible to total up both coefficients, thus creating a total mean heat transfer coefficient at the tube's surface which is displayed in Figure 4 also in relation to the Reynolds number of the liquid phase. Figure 5 shows the values of the generated vapour amount in relation to the Reynolds number.

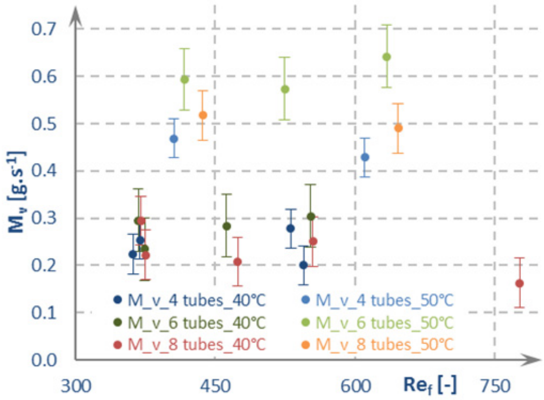

Figure 5. Generated Vapour Amount

The last Figure 6 displays all values of the total mean coefficient in single colour (green) and compares them with two mathematical models given in the theoretical part. Equations of these models were completed by the state values gained by our measurement. Blue values have been calculated according to the Lorenz and Yung model, i.e. by the basic equation (1), and red values have been calculated according to the Chien and Cheng model (2006), i.e. by the basic equation (10).

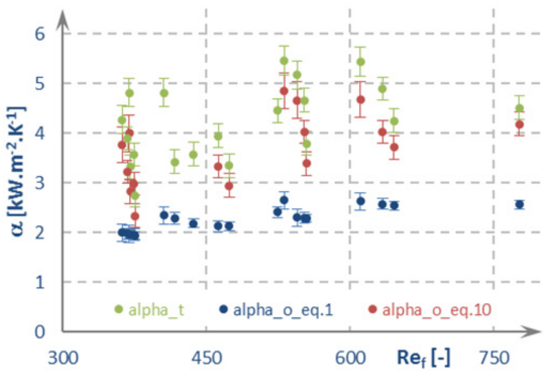

Figure 6. Results Comparison with Other Authors

Conclusion

The objective of this paper was to describe the influence of heat-exchanging surface size on boiling development at a sprinkled tube bundle at two thermal levels of a heating liquid that flows inside the tubes. Boiling development, i.e. its consequent testing on a bundle, was realized using the temperature of a falling film water that has been sprinkled on a tube bundle. It has been proved that the lowest values of the heat transfer coefficient of the vapour phase occurred at the 8-tube bundle and fluctuated around 550 W/(m^2.K). On the contrary, the highest values (almost double) have been reached at the 6-tube bundle. The vapour phase coefficient as well as the generated vapour amount showed to be functionally independent on the value of the total falling film liquid flow rate.

While the convenient number of tubes cannot be clearly determined at the thermal level of 40°C because the vapour amount fluctuated around the mean value of 0.25 g/s, the higher thermal level provides more clear-cut results. The highest values have been reached with six tubes where the vapour amount fluctuated around approx. 0.6 g/s. 8 tubes follow where the amount was approx. 0.5 g/s and the mean value at four tubes oscillated around 0.45 g/s. Higher amounts of generated vapour at six tubes can be justified by a better distribution of the temperature field at the bundle which is visible at the boiling development. Whereas the gradient increases between six and eight tubes, it falls sharply due to further reduction of the heat-exchanging surface (best visible at 50 and 75%). The total heat transfer coefficient at the tube surface consists mostly of the liquid phase coefficient. That is why it is also functionally dependent on the value of the falling film liquid flow rate. The highest values have been reached at a 4-tube bundle. Values of a 6-tube bundle were approx. 20% lower and the difference at an 8-tube bundle was up to approx. 40%.

Coefficients suggested by us have been compared with two other models. State values gained by our measurement have been put into these models. The first model by Chien and Cheng did not correspond very well, i.e. the calculated values were lower in the range of approx. 30 to 56%. Contrary to this, the second model showed a relatively good concordance. Values were lower only in the range of approx. 7 to 17%.

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