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Analysis of staggered tube bundle heat transfer to vertical foam flow

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Abstract

In general heat transfer intensity between solid surface and coolant (fluid) depends on three main parameters: heat transfer coefficient, size of heat exchange surface and temperature difference between surface and fluid. Sometimes the last two parameters (surface size and temperature difference) are strictly limited due to the process or technological requirements, and only increase of heat transfer coefficient is allowed. Simplest way offering sufficient increase in heat transfer rate (heat transfer coefficient as well) is to go from the laminar fluid flow regime to the turbulent one by increasing flow velocity. In many cases it helps despite such disadvantages like more complicated fluid supply system, rise of fluid flow mass rate and growth of energy usage for pumping. But in some cases, for example, in space application, in nuclear engineering, etc. there is not allowed to use high flow velocity of coolant – gas (due to vibration danger) or to apply high mass rate of coolant – liquid (due to limitation concerning weight or mass). One of the possible solutions of that problem could be the usage of two-phase flow as a coolant. An idea to use such two-phase coolant for heat removal from the solid surface is not new. Boiling liquid (water especially), gas flow with liquid droplets and other two-phase systems are widely used for heat and mass transfer purposes in various industries like food, chemical, oil, etc. An application of such two-phase coolants has lot advantages; high value of heat transfer coefficient is one of the most important. Unfortunately nothing is ideal on the Earth. Restrictions on vibration, on coolant weight (or mass rate); necessity to generate two-phase flow separately from the heat removal place; requirements on very low coolant velocities and other constraints do not allow using such type of two-phase coolant for purposes which were mentioned above (space application especially). As a possible way out can be usage of the statically stable foam flow produced from gas (air) and surfactant solutions in liquid (water). Our previous investigations [J. Gylys, Hydrodynamics and Heat Transfer under the Cellular Foam Systems, Technologija, Kaunas, 1998] showed the solid advantages of that type of two-phase coolant, including high values of heat transfer coefficient (up to 1000 W/m^2 K and more), low flow velocities (less than 1.0 m/s), small coolant density (less than 4 kg/m^3), possibility to generate foam flow apart from the heat removal place, etc.

This article is devoted to the experimental investigation of the staggered tube bundle heat transfer to the vertical upward and downward statically stable foam flow. The investigations were provided within the laminar regime of foam flow. The dependency of the tube bundle heat transfer on the foam flow velocity, flow direction and volumetric void fraction were analyzed. In addition to this, the influence of tube position in the bundle was investigated also. Investigation shows that the regularities of the tube bundle heat transfer to the vertical foam flow differ from the one-phase (gas or liquid) flow heat transfer peculiarities. It was showed that the heat transfer intensity of the staggered tube bundle to the foam flow is much higher (from 25 to 100 times) than that for the one-phase airflow under the same conditions (flow velocity). The results of the investigations were generalized using criterion equations, which can be applied for the calculation and design of the statically stable foam heat exchangers with the staggered tube bundles.

Keywords: Gas–liquid foam; Statically stable foam; Foam flow; Heat transfer; Flow turn; Staggered tube bundle

1. Introduction

The gas–liquid foam has especially large inter-phase contact surface which allows using it as a coolant in the heat exchangers or in foam apparatus. The usage of foam

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Nomenclature

A_{ch}	cross-section area of the experimental channel, m^2
A_w	surface area of the heated tube, m^2
a, c, k	coefficients, dimensionless
d	external diameter of the tube, m
d_b	diameter of the foam bubble, mm
G	volumetric flow rate, m^3/s
h	average heat transfer coefficient, $W/(m^2\ K)$
I	amperage, A
n, m, u	coefficients, dimensionless
Nu	Nusselt number, dimensionless
q	heat flux density, W/m^2
Re	Reynolds number, dimensionless
s	spacing between the centres of the tubes, m
T	temperature, K
U	voltage, V
W	velocity, m/s

Greek symbols

β	volumetric void fraction, dimensionless
λ	thermal conductivity, W/(m K)
ν	kinematic viscosity, m^2/s

Subscripts

f	foam flow
g	gas
l	liquid
w	wall

for this purpose is restricted by the foam flow capability to keep its initial structure for a long time intervals. Characteristics of one foam type – statically stable foam – showed its perfect availability for this purpose [1]. Statically stable foam is such type of foams, which keeps its initial dimensions of bubbles within broad limits of time intervals, from several seconds to days, even after termination of the foam generation [1,2]. The generation of statically stable foam is predetermined by the content of detergents in a solution. On adsorptive layers of foam bubble, strong detergents form up viscous and firm spatial structures, protecting foam films from thinning and cracking. Even small concentration of detergents may be the reason of intensive generation of statically stable foam due to the bubbling of gas. There exists minimum concentration of detergents for different kinds of detergents and different liquids, at the presence of which a certain liquid volume can be transformed into a flow of statically stable foam [1]. For the experimental foam generation the concentration of detergents must ensure required stability of foam and satisfy defined requirements to the volumetric void fraction [2].

The two-phase gas–liquid foam heat exchangers compared with liquid one has a number of advantages: small coolant mass flow rate is required for heat transfer, heat transfer rate is high, mass of all system is much smaller, and energy consumption for foam delivery into heat transfer zone is lower. But there is no yet sufficient data concerning heated surfaces heat transfer to the statically stable foam flow. Therefore problems arise when foam systems and heat exchangers are designed. Main aim of our investigation there was to estimate the peculiarities of heat transfer from tube bundle to foam flow and to develop the method for design of foam heat exchangers.

The up-to-date scientific research field of gas–liquid foam is oriented in preference to foam generation process and its specific peculiarities [3,4], to foam physics and foam flow rheology [5,6], etc. The heat transfer from the heated surfaces to the foam flow is not investigated enough at present time. The application of gas–liquid foam as a coolant and the design of modern heat exchangers with foam coolant are impossible without knowledge of regularities which take place during heat transfer from heated surfaces to the foam flow. Gas–liquid foam is the two-phase system with number of different attendant processes: drainage of liquid from the foam [7–9], diffusive gas transfer [2] and destruction of inter-bubble films [10]. Structure of foam flow especially changes while it passes obstacle: bubbles size is changing and liquid drainage is going on. All these peculiarities make very difficult an application of analytical methods of investigation. Thus experimental method was selected for the investigation of tube bundle heat transfer to the statically stable foam flow.

Typical heat exchangers usually consist of several vertical parts in which coolant changes its direction from vertical upward to vertical downward and vice versa. The turning of foam flow from the upward to downward flow direction influences on the distribution of the foam volumetric void fraction and on the flow velocity in the cross-section of the foam channel as well. Consequently the investigations of foam flow turning influence on heat transfer peculiarities must be performed for its later application in heat exchangers.

Our previous works were devoted to the investigation of heat transfer of alone cylindrical tubes to upward statically stable foam flow. Next experimental series were performed for the tube line placed in upward foam flow [1].

Presently an experimental investigations of staggered tube bundle heat transfer process to the vertical upward and downward statically stable foam flow were performed [11,12]. It was determined the dependence of heat transfer intensity on the flow parameters: flow velocity, volumetric void fraction of foam and liquid drainage from foam. Apart of this, influence of tube position in the bundle to the heat transfer intensity was investigated also. Results of investigation were generalized using relationship between Nusselt number and Reynolds number and the volumetric void fraction of the foam.

2. Experimental set-up

The investigations were performed on the experimental laboratory set-up consisting of foam generator, experimental channel, staggered tube bundle, measurement instrumentation and auxiliary equipment (Fig. 1).

Statically stable foam flow, which was used for the experimental investigation, was generated from the detergents water solution. Concentration of the detergents was kept constant at 0.5% in all experiments. Foamable liquid was supplied from the reservoir onto the special perforated plate (riddle); gas (air) was delivered through the plate from the bottom gas chamber. Foam flow was produced during gas and liquid contact. Foam flow parameters control was fulfilled using gas and liquid valves.

Foam generation riddle was installed at the bottom of the experimental channel and was made from stainless steel plate with a thickness of 2 mm; orifices were located in a staggered order; their diameter was 1 mm; spacing among the centres of the holes was 5 mm.

The experimental channel was composed of three main parts: the vertical channel for the upward foam flow; the channel turn and the vertical channel for downward foam flow. The radius R of the channel turn was equal to 0.17 m. Cross-section of the channel had dimensions $0.14 \times 0.14\ m^2$; height was 1.8 m; walls were made from the transparent material in order to observe foam flow visually.

Staggered bundle of the tubes consisted of three vertical rows with five tubes in each. Spacing between centres of the tubes was $s_1 = 0.07$ m and $s_2 = 0.0175$ m. All tubes had an external diameter of 0.02 m. Schematic view of the experimental section with tube bundle is presented in Fig. 2. One bundle's tube (calorimeter) was heated electrically. This tube was made of copper and had an external diameter of 0.02 m also. The ends of the heated tube was sealed and insulated to prevent heat loss through them. During the experiments calorimeter was placed instead of one of the bundle's tube.

Temperature of the foam flow was measured by two calibrated thermocouples: one in the front of the bundle and one behind. Temperature of the heated tube surface was measured by eight calibrated thermocouples: six of them were placed around central part of the heated tube and

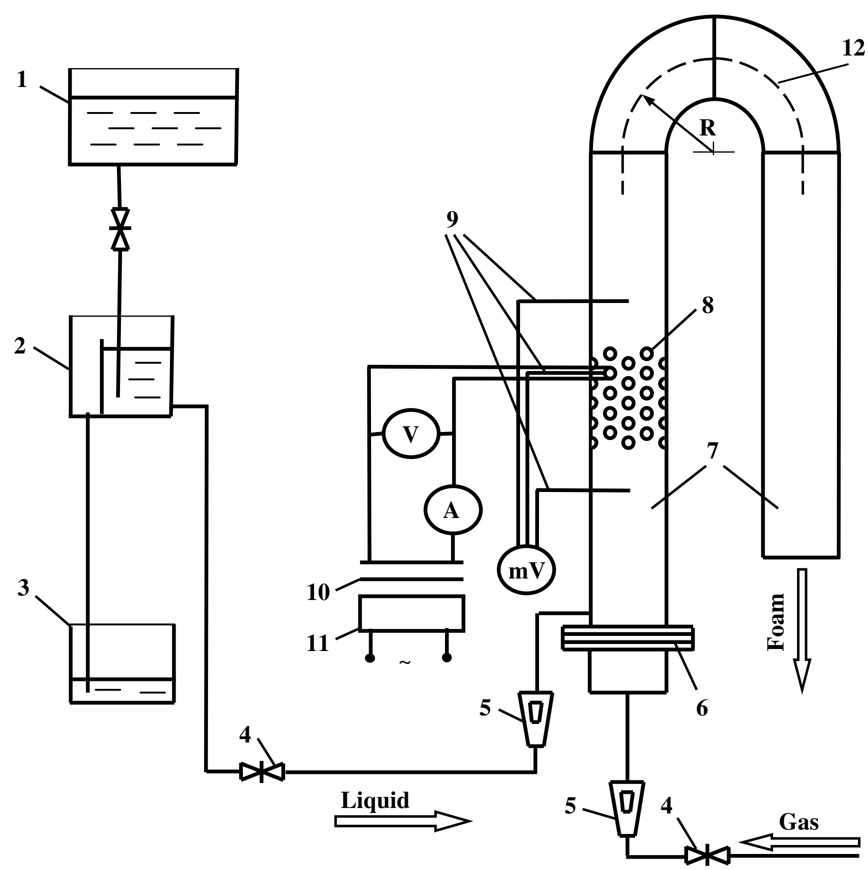

Fig. 1. Experimental set-up scheme: 1 – liquid reservoir; 2 – liquid level control reservoir; 3 – liquid receiver; 4 – gas and liquid control valves; 5 – flow meter; 6 – plate for the foam generation; 7 – experimental channel; 8 – tube bundle; 9 – thermocouples; 10 – transformer; 11 – stabilizer and 12 – channel turn.

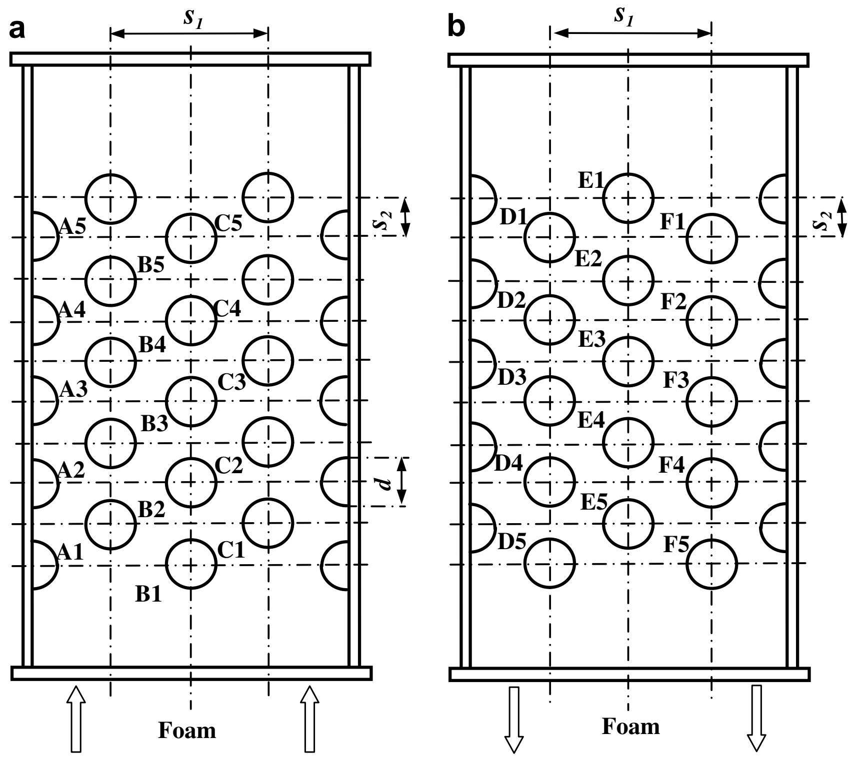

Fig. 2. The staggered tube bundle in the experimental channel for the upward (a) and downward (b) foam flow.

two were placed in both sides of the tube at 50 mm distance from the central part. Accuracy of all thermocouples was ±0.5 K across its operating temperature range of 273.15–373.15 K (0–100 °C). Voltage (in mV) of thermocouples was measured using precision multimeter.

Gas and liquid flow rates were measured by flow meters. Accuracy of flow meter was $\pm 0.1 \times 10^{-3}$ m^3/s for gas (air) across all operating range, which varied from 0 to 10×10^{-3} m^3/s; and it was $\pm 0.25 \times 10^{-6}$ m^3/s for liquid (detergent solution) across all operating range, which varied from 0 to 40×10^{-6} m^3/s.

Calorimeters power supply system's voltage was stabilized by stabilizer and was reduced by transformer; electric current magnitude was measured by ammeter and voltage – by voltmeter. Accuracy of the ammeter measurements were ±0.1 A across all its operating range, which was from 0 to 10 A; accuracy of the voltmeter measurements were ±0.05 V across all its operating range, which was from 0 to 25 V.

3. Methodology

Statically stable foam flow generated on the riddle was directed vertically upward and after 180° turning moved vertically downward. The same staggered tube bundle was used for heat transfer investigation under the upward and downward foam flows. The influence of foam flow volumetric void fraction β and flow velocity w_f on an average heat transfer coefficient h was obtained:

$$h = f(\beta, w_f). \tag{1}$$

The experiments were provided for three different values of the foam volumetric void fractions $\beta = 0.996$, 0.997 and 0.998. Volumetric void fraction was computed according to the following equation:

$$\beta = \frac{G_g}{G_g + G_l}. \tag{2}$$

The foam flow velocity was computed using such formula:

$$w_f = \frac{G_g + G_l}{A_{ch}}. \tag{3}$$

Our previous investigations showed that hydraulic and thermal regimes stabilize completely within five minutes after changing of the experimental conditions. Therefore measurements were started not earlier than five minutes after the adjustment of the foam flow parameters. Heat flux density on the heated tube surface q_w was calculated after registration of electric current and voltage:

$$q_w = \frac{UI}{A_w}. \tag{4}$$

An average temperature difference $\Delta T = T_w - T_f$ between foam flow T_f and tube surface T_w was calculated using thermocouple measurement results and was further applied for determination of the heat transfer coefficient:

$$h = \frac{q_w}{\Delta T}. \tag{5}$$

Gas Reynolds number of foam flow was computed using the following formula:

$$Re_g = \frac{G_g d}{A_{ch} \nu_g}. \quad (6)$$

Nusselt number was computed by the equation:

$$Nu_f = \frac{hd}{\lambda_f}, \quad (7)$$

where λ_f – thermal conductivity of the statically stable foam, obtained from the equation:

$$\lambda_f = \beta\lambda_g + (1-\beta)\lambda_l. \quad (8)$$

Heat transfer intensity of the staggered tube bundle tube to airflow was computed by the formula [13]:

$$Nu = 0.4Re_g^{0.6} Pr_g^{0.36} \left(\frac{Pr_g}{Pr_w}\right)^{0.25}, \quad (9)$$

where (9) formula is valid for the range $40 < Re_g < 1000$.

Previous our investigations [1] showed that four main regimes of the statically stable foam flow in the vertical channel of rectangular cross-section can be achieved:

Laminar flow regime $Re_g = 0$–600.
Transition flow regime $Re_g = 600$–1500.
Turbulent flow regime $Re_g = 1500$–1900.
Emulsion flow regime $Re_g > 1900$.

Experimental results presented here were received for the laminar flow regime of the statically stable foam flow.

All experiments and measurements were repeated several times in order to reduce the measurement errors and to increase the reliability of the results. The statistical analysis of the data showed that all experimental results are reliable, precision and reproducible.

During the experimental investigations the parameters varied within the following limits: foam volumetric void fraction β varied from 0.996 to 0.998; foam flow velocity w_f varied from 0.14 to 0.32 m/s; average heat transfer coefficient h varied from 235 to 1581 W/(m^2 K).

4. Results

The experimental investigation proved the preliminary estimation that heat transfer intensity of the staggered tube bundle to the foam flow is much higher than to the one-phase airflow under the same conditions (flow velocity). Data of the tube bundle tubes A1, B1 and C1 heat transfer intensity as a function of the upward foam flow velocity and for comparison heat transfer intensity of the tube B1 as a function of one-phase airflow velocity are shown in Fig. 3.

Foam local volumetric void fraction and flow local velocity had correspondingly the same values near the tubes A and C and therefore heat transfer intensity of those tubes was identical. For that reason the data of heat transfer intensity of the tubes A and C was grouped and was presented as data of the side-line tubes AC (Figs. 3 and 4).

Foam flow velocity w_f increase from 0.14 to 0.32 m/s influences growth of heat transfer coefficient (h) of the middle-line tube B1 by 4 times (from 335 to 1350 W/(m^2 K)) for foam volumetric void fraction $\beta = 0.996$ and by 3.5 times for $\beta = 0.997$ (from 288 to 1002 W/(m^2 K)), and by 2.7 times for $\beta = 0.998$ (from 246 to 658 W/(m^2 K)). It was noticed that influence of foam flow volumetric void fraction on the tube B1 heat transfer intensity is more significant for the faster moving foam flow ($w_f = 0.32$ m/s). Besides that heat transfer intensity of the same tube B1 is twice better to the wettest foam flow ($\beta = 0.996$) in comparison to the driest foam flow ($\beta = 0.998$).

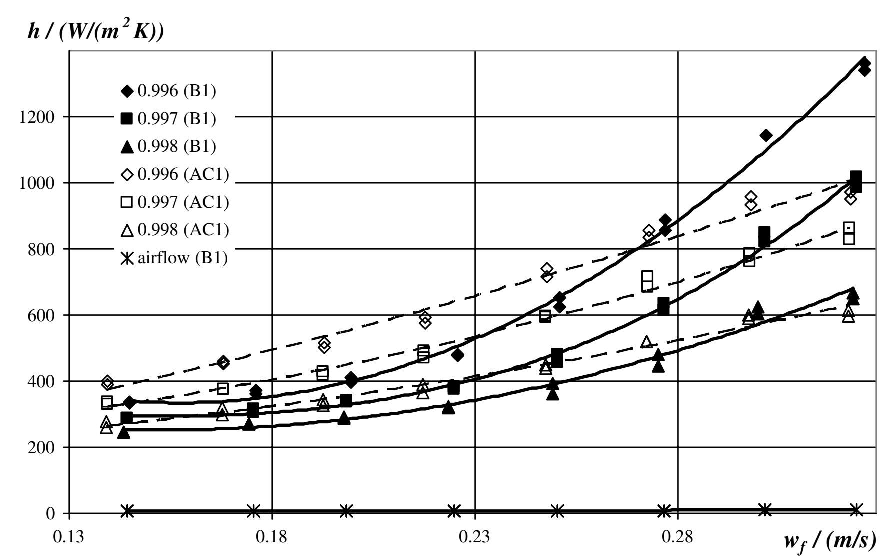

Fig. 3. The heat transfer of the tubes A1, B1 and C1 to the upward foam flow: $\beta = 0.996$, 0.997, 0.998 and to airflow.

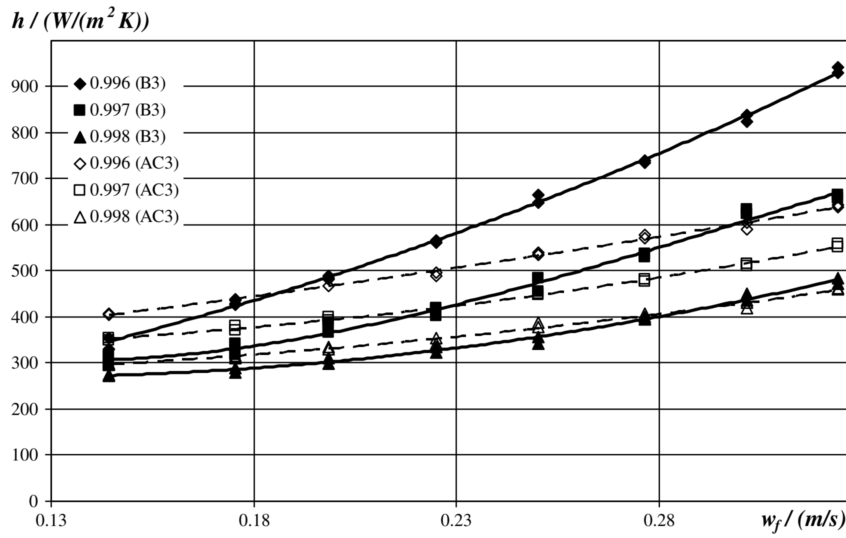

Fig. 4. The heat transfer of the tubes A3, B3 and C3 to the upward foam flow: $\beta = 0.996$, 0.997 and 0.998.

The side-line tubes A1 and C1 heat transfer intensity depends on foam flow velocity less than that of the middle-line tube B1. Foam flow velocity w_f increase from 0.14 to 0.32 m/s affects heat transfer intensity of the side-line tube A1 and C1 growth by 2.5 times for the foam with $\beta = 0.996$ and $\beta = 0.997$, and by 2.2 times for $\beta = 0.998$.

Cross channel distribution of the foam flow local velocity and local volumetric void fraction influences on the tubes heat transfer intensity also. Maximum value of the foam flow local velocity usually is registered at the central part of the channel. Local volumetric void fraction distribution is different from that. Usually foam is dryer at the centre of the channel and is wetter near the channel's walls. The third factor which influences on the tubes heat transfer intensity is the foam structure. Diameter of the foam bubbles characterizes foam structure and depends not only on the foam volumetric void fraction, but on the foam flow generation conditions as well. Larger size bubbles (diameter is more than 8 mm) foam flow is generated if the feeding gas rate G_g and accordingly the foam flow velocity w_f is low. Increase of G_g (or w_f) influences on generation of foam flow with smaller bubbles, therefore foam flow becomes more homogenous. As a result, the foam local volumetric void fraction distribution across the channel becomes smoother and an influence of the foam flow velocity distribution on the tubes heat transfer intensity increases therefore heat transfer of the middle-line tube B1 is better than that of the side-line tubes. The experimental investigation shows that heat transfer intensity of the side-line tubes AC1 in comparison with the middle-line tube B1 is better till $w_f = 0.27$ m/s for $\beta = 0.996$, till $w_f = 0.294$ m/s for $\beta = 0.997$ and till $w_f = 0.299$ m/s for $\beta = 0.998$ (Fig. 3).

Foam bubbles are intermixed, some bubbles collapsed or divided into the smaller bubbles during foam flow pass through the tube bundle. Distribution of the foam local volumetric void fraction across the channel becomes more evenly. Heat transfer intensity of the tubes A3, B3 and C3 to the upward foam flow is shown in Fig. 4. Increase of foam flow velocity w_f from 0.14 to 0.32 m/s raises heat transfer intensity of the middle-line tube B3 by 2.7 times for the foam with $\beta = 0.996$ and by 2.2 times for $\beta = 0.997$, and by 1.75 times for $\beta = 0.998$. Heat transfer intensity of the side-line tubes AC3 increases by 1.6 times for the foam with $\beta = 0.996$, $\beta = 0.997$ and $\beta = 0.998$ in the same interval of w_f.

Investigation showed that heat transfer intensity of the tube B3 is twice better for the wettest foam flow ($\beta = 0.996$) in comparison with the driest foam flow ($\beta = 0.998$) at the fastest foam flow ($w_f = 0.32$ m/s) conditions, and by 1.3 times only at the slowest foam flow ($w_f = 0.14$ m/s) conditions. Heat transfer intensity of the side-line tubes AC3 is about 1.4 times better for the wettest foam flow in comparison with the driest foam flow for the whole investigated interval of the foam flow velocity w_f.

Investigation of the staggered tube bundle heat transfer in one-phase flow [13] showed that heat transfer intensity of the frontal (first) tubes is equal to about 60% of the third tubes heat transfer intensity, heat transfer intensity of the second tubes is equal to about 70% of the third tubes heat transfer intensity, and the heat transfer intensity of the fourth and further tubes is the same like of the third tubes. Flow velocity distribution in the cross-section of the channel (pipe) is the main factor which makes different heat transfer intensity of the middle and side tubes. Heat transfer of the tubes of the staggered bundle to airflow was calculated by Eq. (9) and is presented in Fig. 5. The regularities of the different tubes heat transfer to the vertical foam flow are different in comparison with the one-phase airflow. The peculiarities of the foam plays significant role in that case.

Comparison of heat transfer intensity of the middle-line tubes to upward foam flow at a volumetric void fraction $\beta = 0.997$ is shown in Fig. 6. It can be seen that heat transfer intensity of the tubes from the first (B1) to fourth (B4)

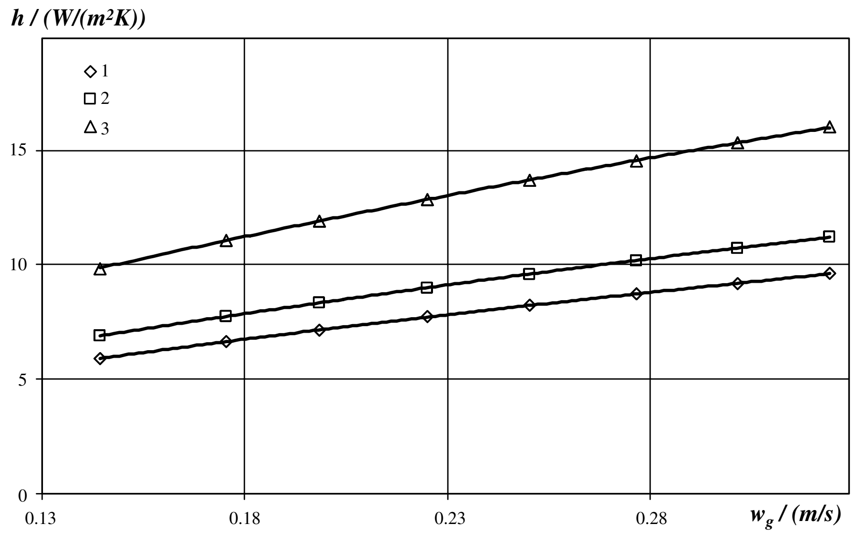

Fig. 5. The heat transfer intensity of the tubes of the staggered tube bundle to the airflow: 1 – first tube, 2 – second tube, 3 – third, fourth and fifth tubes.

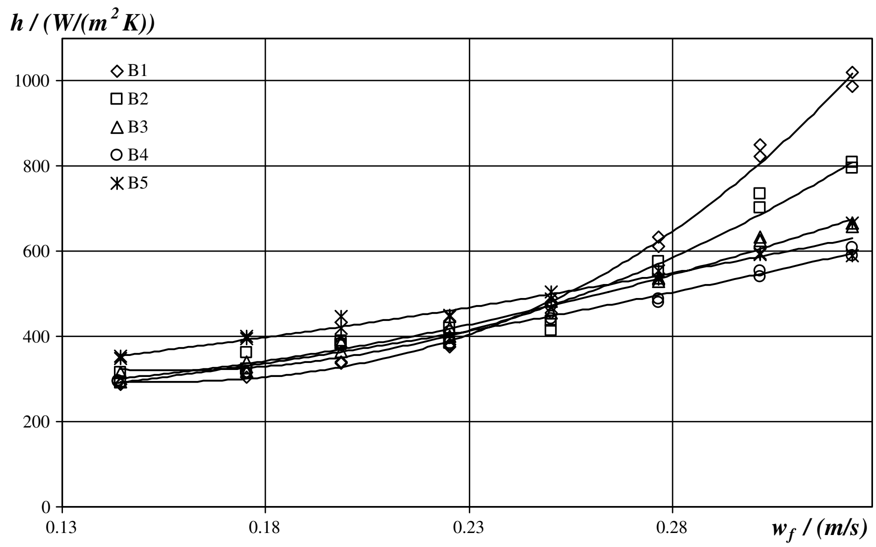

Fig. 6. The heat transfer intensity of the tubes in the middle-line (B) of the bundle to the upward foam flow; $\beta = 0.997$.

slightly depends on the position of the tube for the interval of the foam flow velocity w_f from 0.14 to 0.235 m/s. Heat transfer intensity of the last tube (B5) of the middle-line of the bundle differs (is better) from the others. It can be explained by the fact that low foam flow velocity corresponds to generation of the foam flow consisting of quite large bubbles ($d_b = 11 \pm 3$ mm). Large bubbles passing through the bundle of the tubes are divided into smaller bubbles ($d_b = 3.5 \pm 1.5$ mm). Consequently drainage process near the surfaces of the last tubes becomes more intensive, and increase of heat transfer rate can be noticed as well. Rise of foam flow velocity w_f from 0.235 to 0.32 m/s influences more intensive generation of smaller foam bubbles, which make foam flow more homogenous, with better wetting conditions of the heated surfaces. Therefore heat transfer rate of the tubes B1 and B2 increase much significantly. Otherwise heat transfer rate of the tubes B3 and B4 is lower due to change of the flow character near the different tubes of the line.

Experimental investigation with vertically downward moving after 180° turn foam flow showed additional peculiarities. Foam flow local volumetric void fraction distribution across the channel transforms during flow turning. This transformation depends mainly on liquid drainage from the foam; therefore drainage must be taken into account during analysis.

Liquid drainage from the foam mainly is influenced by the gravity and capillary forces; influence of the electrostatic and molecular forces is negligible [2,14,15]. Gravity and capillary forces are acting together in vertical direction, however influence of gravity forces is negligible in horizontal direction consequently influence of capillary forces is dominating. Gravity forces act along the upward and downward moving foam flow, but on the foam flow turn those forces act across foam flow. As a result, liquid drains down from the foam near upper channel wall and a local volumetric void fraction increases here as well. Therefore volumetric void fraction of the foam is less (foam is wetter) on the left side of the channel (D tubes, Fig. 2b). Flow velocity distribution across channel transforms after turning also. All these mentioned factors influence on tube bundle heat transfer intensity. Comparison of heat transfer intensity of the tubes D1, E1 and F1 to the downward foam flow at the volumetric void fraction $\beta = 0.996$ and $\beta = 0.998$ after the turning is shown in Fig. 7.

Foam flow velocity (w_f) changes from 0.14 to 0.32 m/s increases heat transfer intensity of the tube D1 by 2.4 times, tube E1 – by 2.6 times, and tube F1 – by 2.4 times, for the same value of the volumetric void fraction $\beta = 0.996$ (Fig. 7). Heat transfer intensity of the tubes E1 and F1 is almost identical up to $w_f = 0.28$ m/s, but further increase of the foam velocity makes heat transfer intensity of tube E1 more than that of the tube F1. Heat transfer of side tube D1 is approximately twice more than that of the tubes E1 and F1 for the whole interval of the flow velocity w_f, for $\beta = 0.996$. Analogous situation can be observed for the driest foam flow ($\beta = 0.998$), however heat transfer coefficient (h) of the tubes D1, E1 and F1 is significantly less than for the $\beta = 0.996$. Heat transfer intensity of the tube D1 to the wettest foam flow ($\beta = 0.996$) is in average twice higher than that for the driest foam flow ($\beta = 0.998$); and for the tubes E1 and F1 foam flow with $\beta = 0.996$ allows to reach about 36% higher heat transfer rate than for the foam flow with $\beta = 0.998$. Tube D1 heat transfer intensity to the driest foam flow ($\beta = 0.998$) is near 31% higher than that of the tubes E1 and F1. This influence is slightly different for the same tubes under the wettest ($\beta = 0.996$) foam flow.

Tubes D3, E3 and F3 heat transfer intensity to downward foam flow after 180° turning at the volumetric void fraction $\beta = 0.996$ and $\beta = 0.998$ is shown in Fig. 8. It can be noticed that heat transfer rate of tube D3 approximately by 64% is higher than heat transfer intensity of tube E3, and heat transfer of tube E3 almost by 29% is higher in comparison with heat transfer intensity of tube F3, for the same value of $\beta = 0.996$. Heat transfer of tube D3 to the wettest foam flow ($\beta = 0.996$) is more than twice higher than that for tube F3.

Foam's flow local volumetric void fraction distribution across the experimental channel becomes more gradual due to influence of the obstacle – tube bundle. It is more obvious in the case with the driest ($\beta = 0.998$) foam flow (Fig. 8). Tube D3 heat transfer rate to the driest foam flow ($\beta = 0.998$) is higher in average by 16% than that of tube E3, and in average by 38% than that of tube F3.

Heat transfer intensity of the middle-line tubes to the downward foam flow at the volumetric void fraction $\beta = 0.997$ is shown in Fig. 9. Figure shows that heat transfer intensity of the middle-line tubes is almost independent of the tube position (except the fifth tube) for the downward foam flow passing tube bundle at the volumetric void fraction $\beta = 0.997$ and foam flow velocity w_f changing from 0.14 to 0.23 m/s. During foam flow velocity w_f changes within the limits 0.23–0.32 m/s heat transfer intensity is best of the first tube, less of the second, more less of the third and so on except the last-fifth tube. The heat transfer intensity of the fifth tube is less than that of the other tubes only after w_f achieves value equal to 0.31 m/s.

Today it is difficult to compare heat transfer intensity of the tube bundle separate tubes placed in upward and downward foam flows. Main reason of that – separate tubes heat transfer intensity differences across the channel. Therefore an average heat transfer coefficient (h) of the entire tube bundle to the upward and downward foam flow was calculated. It was observed that heat transfer between the stag-

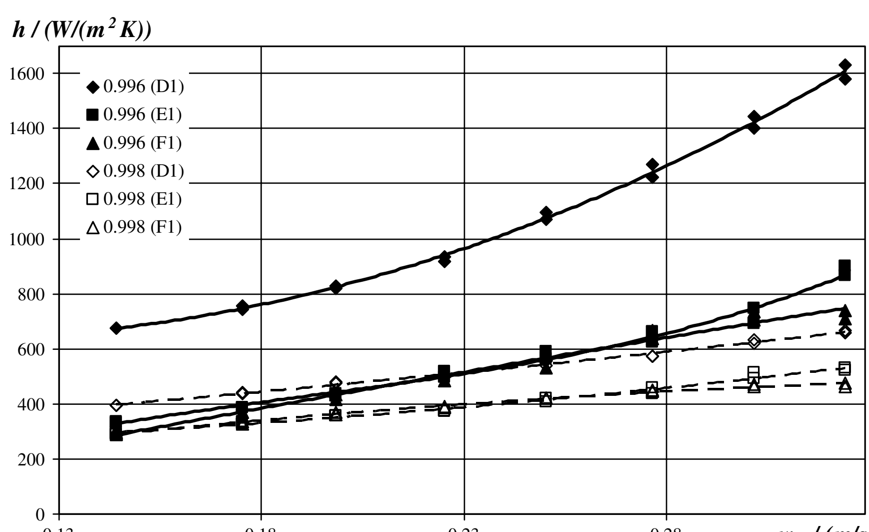

Fig. 7. The heat transfer of the tubes D1, E1 and F1 to the downward foam flow: $\beta = 0.996$, 0.997 and 0.998.

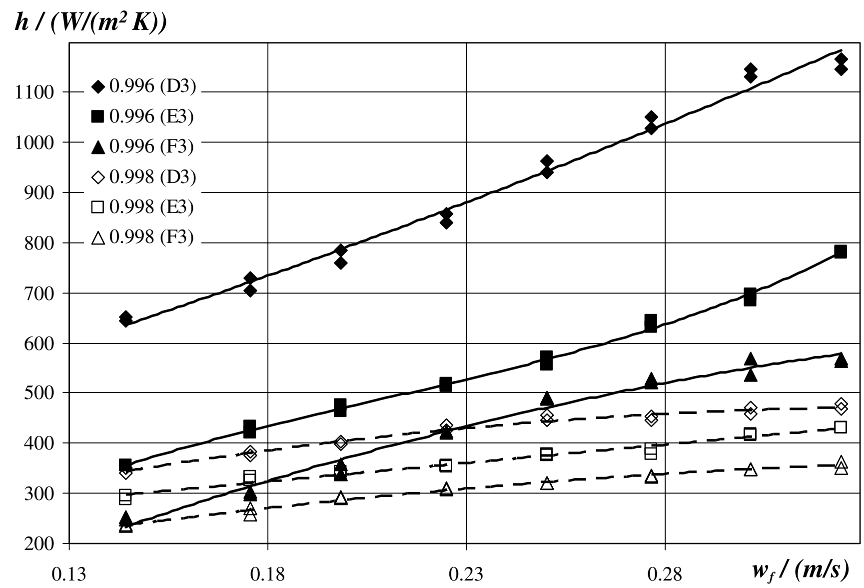

Fig. 8. The heat transfer of the tubes D3, E3 and F3 to the downward foam flow: $\beta = 0.996$, 0.997 and 0.998.

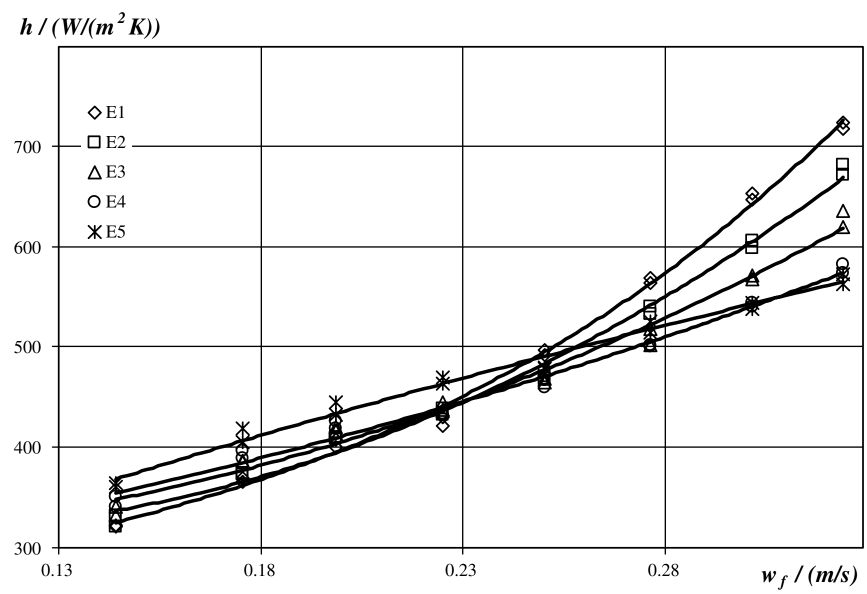

Fig. 9. The heat transfer intensity of the tubes in the middle-line (E) of the bundle to the downward foam flow; $\beta = 0.997$.

gered tube bundle and downward foam flow is more intensive for the foam flow velocity interval $w_f = 0.14$–0.23 m/s. Otherwise, at higher foam flow velocity ($w_f = 0.14$–0.23 m/s) staggered tube bundle was more intensely cooled in upward foam flow.

Experimental results of the staggered tube bundle heat transfer to upward and downward (after 180° turn) foam flow were summarized by the criterion equations, which are suitable for calculation and design of the foam-type apparatus and heat exchangers. Generalization of the experimental results was performed using dependence between Nusselt number Nu_f and foam flow gas Reynolds number Re_g.

This dependence for upward foam flow with the volumetric void fraction $\beta = 0.996$; 0.997; 0.998 within the interval $190 < Re_g < 300$ (0.14 m/s $< w_f <$ 0.23 m/s) can be expressed by the following equation:

$$Nu_f = cRe_g^m, \tag{10}$$

where $m = 60(1.0072 - \beta)$; $c = a(\beta - 0.99)$.

An average magnitude of the coefficient a determined as follows: for entire middle-line (B) in the bundle: $a = 1735$;

for entire side-line (A or C) in the bundle: $a = 1880$; and for the entire staggered tube bundle: $a = 1820$.

In order to determine heat transfer rate for the foam volumetric void fraction $\beta = 0.996$; 0.997; 0.998 and foam flow velocity $0.23\ \text{m/s} < w_f < 0.32\ \text{m/s}$ $(300 < Re_g < 440)$ the following equation can be used:

$$Nu_f = c\beta^n Re_g^m. \qquad (11)$$

An average magnitude of the coefficients n, m, c determined as follow: for entire middle-line (B) of the tubes in the bundle: $n = 890$, $m = 200(1.004 - \beta)$, $c = 2$; for entire side-line (A or C) of the tubes in the bundle: $n = 1000$, $m = 200(1.0007 - \beta)$, $c = 127$ and for entire tube bundle allocated in the staggered order: $n = 950$, $m = 200(1.004 - \beta)$, $c = 2.3$.

The same dependence for the downward foam flow at the volumetric void fraction $\beta = 0.996$; 0.997; 0.998 within the interval $190 < Re_g < 440$ ($0.14\ \text{m/s} < w_f < 0.32\ \text{m/s}$) can be expressed also by the Eq. (11).

An average magnitude of the coefficients n, m, c determined as follow: for entire side-line D in the bundle: $c = 153$, $n = 881$, $m = 217.66–217.65\beta$; for entire middle-line E in the bundle $c = 142$, $n = 1091$, $m = 224.31–224.25\beta$; for entire side-line F in the bundle: $c = 318$, $n = 1560$, $m = 282.06–282.1\beta$ and for the whole staggered tube bundle $c = 134$, $n = 1025$, $m = 223.25–223.2\beta$.

5. Conclusions

Staggered tube bundle heat transfer to the upward and downward foam flow after 180° turn was investigated experimentally.

Dependency of tube bundle heat transfer on foam flow velocity, direction, volumetric void fraction and tube position in the bundle was analyzed.

Heat transfer rate using laminar (velocity $0.14 < w_f < 0.32\ \text{m/s}$) foam flow varied from 235 to 1581 $\text{W/(m}^2\,\text{K)}$ and was much higher (from 25 to 100 times) than using single gas flow.

Differently from single-phase flow heat transfer rate of the first bundle tubes to foam flow was higher than of the following tubes.

Foam flow turn influenced on foam flow local velocity and local volumetric void fraction distribution across the experimental channel.

Heat transfer to downward foam flow was more intensive for foam flow velocity range from 0.14 to 0.23 m/s. Upward foam flow was more preferable for higher foam flow velocity (from 0.23 to 0.32 m/s).

Results of investigation were generalized by criterion equations, which can be used for calculation and design of the statically stable foam heat exchangers with the staggered tube bundles.

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