Study of In–line Tube Bundle Cooling in Vertical Foam Flow

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Abstract: Smaller coolant mass flow rate and higher heat transfer intensity may be reached using two–phase foam flow as a coolant, therefore heat exchangers with smaller size and mass can be manufactured. Heat transfer of the in–line tube bundle to the vertical foam flow was investigated experimentally ad hoc. Experimental investigation was provided within the laminar regime of the statically stable foam flow. Dependency of the tube bundle heat transfer on the foam flow velocity, direction and volumetric void fraction was analyzed. In addition to this, influence of the tube position in the bundle on heat transfer intensity was investigated also. Experimental results show, that the heat transfer to vertical foam flow is different in comparison with one–phase flow. Peculiarities of the foam play significant role in that case. The results of the investigations were generalized by criterion equation, which can be used for the calculation and design of the statically stable foam heat exchangers with the in–line tube bundles.

Key-Words: heat transfer, foam flow, flow turn, in–line tube bundle, experimental channel.

1 Introduction

Single phase coolant (liquid or gas) is widely used for heat removal in heat exchangers. Nevertheless usage of two–phase system as a coolant (foam – particular) in some cases allows to achieve better results and significantly reduces energy consumption [1].

Foam is distinguished by especially large inter–phase contact surface and can be applied for different heat and mass transfer purposes. Efficiency of these processes depends on the capacity "to control" the foam. 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. Thus, this type of foam is suitable for heat transfer process in different foam apparatus.

An application of the statically stable foam as a coolant and the design of the heat exchangers with foam coolant are impossible without deep knowledge of processes which are going on during the contact of foam flow and surface. Number of specific foam's peculiarities: foam structure changing [2, 3]; drainage of liquid from foam [4, 5]; diffusive gas transfer [3] and destruction of inter–bubble films [3, 6] – complicate an analytical investigation of heat transfer process. Today an experimental method of investigation can be treated as the most suitable.

Heat transfer of the different heated surfaces and tube bundles to the one–phase coolants was investigated quite enough [7, 8], but practically there are not enough data concerning of tube bundle heat transfer to the foam flow. Our previous works [1] were devoted to the investigation of heat transfer of a single tube and tube line to the upward statically stable foam flow. Next experimental series with the staggered tube bundle in the upward and downward foam flow followed as well [2, 9, 10]. It was determined the dependence of heat transfer intensity on the foam flow velocity, volumetric void fraction and liquid drainage from the foam. Apart of this, influence of the tube position in the bundle on heat transfer intensity was investigated also. Results of investigations were generalized using relationship between Nusselt number and Reynolds number and the volumetric void fraction of the foam.

Presently an experimental investigation of the heat transfer process from the in–line tube bundle to the vertical statically stable foam flow was performed. Main task of the investigation was to determinate an influence of the foam flow parameters on the tube bundle heat transfer intensity.

2 Experimental Set–up

The experimental set–up consisted of the following main parts: foam generation channel, tube bundle, gas and liquid control valves, gas and liquid flow meters, liquid storage reservoir, liquid level control reservoir, air fan, electric current transformer and stabilizer (Fig.1).

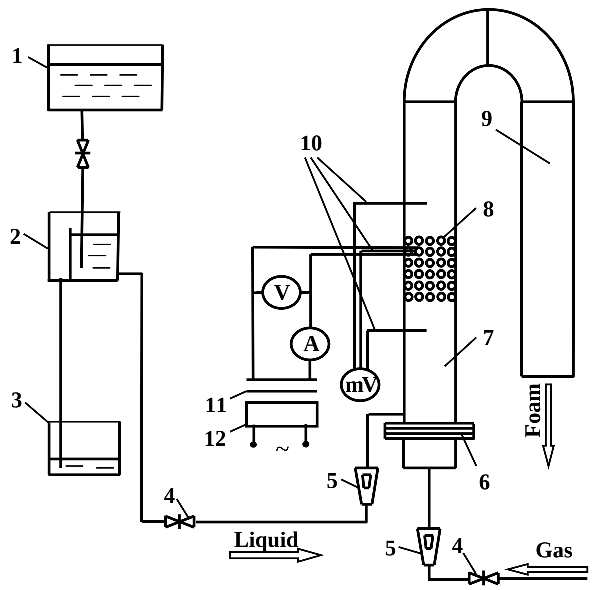

Fig. 1. Experimental set–up scheme: 1–liquid reservoir; 2–liquid level control reservoir; 3–liquid receiver; 4–gas and liquid control valves; 5–flow meter; 6–foam generation riddle; 7–experimental channel; 8–tube bundle; 9–output channel; 10–thermocouples; 11–transformer; 12–stabiliser

Statically stable foam flow was used for the experimental investigation. The foam–able liquid for the statically stable foam generation was produced from water solution with the detergents. Concentration of the detergents was kept constant at 0.5 % in all experiments. Foam–able liquid was supplied from the reservoir onto the special riddle for the foam generation from all channel sides; gas was supplied to the plate from the bottom. Air from the compressor via receiver of was used as “gas” in all investigations. Foam flow was produced during gas and liquid contact on the riddle.

Foam generator riddle was made from stainless steel plate with thickness of 2 mm. Diameter of holes was 1 mm; spacing among centers of the holes was 5 mm. Holes were located in staggered order.

Cross section of the channel had dimensions 0.14 x 0.14 m^2; height of experimental channel was 1.8 m.

Schematic view of experimental section of the channel with the in–line tube bundle can be seen in Fig. 2.

Tube bundle consisted of five vertical rows with six tubes in each. Spacing among the centers of the tubes was $s_1 = s_2 = 0.03$ m. All tubes had an external diameter of 0.02 m. One tube was heated electrically. This tube was made of copper and had an external diameter of 0.02 m also. Endings of the heated tube were sealed and insulated in order to prevent heat loss through them. During the experiments heated tube was placed instead of one tube of the bundle. An electric current value was measured by ammeter and voltage by voltmeter.

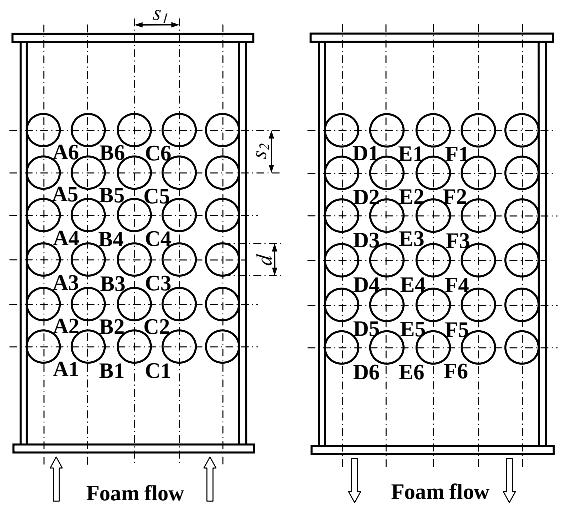

Fig. 2. Tube bundle in foam flow

Temperature of foam flow was measured by two calibrated thermocouples: one in front of the bundle and one behind. Temperature of heated tube surface was measured by eight calibrated thermocouples. Six of them were placed around central part of theheated tube and two of them were placed in both sides of the tube at 50 mm distance from the central part.

Radius of the channel turning was equal to 0.17 m.

Measurement accuracies for flows, temperatures and heat fluxes were of range correspondingly 1.5%, 0.15÷0.20% and 0.6÷6.0%.

3 Methodology

During the experimental investigations was obtained a relationship between the average heat transfer coefficient $\overline{\alpha}$ from one side and foam flow volumetric void fraction β and gas flow Reynolds number $\overline{Re}_g$ from the other side:

$$\overline{Nu}_f = f\left(\beta, \overline{Re}_g\right). \tag{1}$$

Nusselt number was computed by the formula

$$\overline{Nu}_f = \frac{\overline{\alpha} d}{\lambda_f}\ , \tag{2}$$

where λ_f is a thermal conductivity of the statically stable foam flow, W/(m·K), obtained from the equation

$$\lambda_f = \beta\lambda_g + (1-\beta)\lambda_l\,. \tag{3}$$

An average heat transfer coefficient was calculated as

$$\overline{\alpha} = \frac{q_w}{\overline{\Delta T}}\,. \tag{4}$$

Gas Reynolds number of foam flow was computed by the formula

$$\overline{Re}_g = \frac{G_g d}{A v_g}\,. \tag{5}$$

Foam flow volumetric void fraction can be expressed by the equation

$$\beta = \frac{G_g}{G_g + G_l}\,. \tag{6}$$

The following parameters were measured and recorded during the experiments: temperatures of the heated tube surface and foam flow, electric current and voltage. Investigations showed that hydraulic and thermal regime stabilises completely within five minutes after the change of experimental conditions. Therefore measurements were started not earlier than five minutes after adjustment of new foam flow parameters. Heat flux density on the tube surface q_w was calculated using the following formula:

$$q_w = \frac{UI}{\pi d l}\,. \tag{7}$$

After record of heated tube surface and foam flow temperatures, the temperature difference $\overline{\Delta T}$ (between mean temperatures of foam flow $\overline{T}_f$ and tube surface $\overline{T}_w$) was calculated.

Experiments were performed within Reynolds number diapason for gas 190÷440 and foam volumetric void fraction – 0.996÷0.998. Gas velocity for foam flow was changed from 0.14 to 0.32 m/s. Heat transfer coefficient varied from 200 to 1440 W/(m^2·K).

All experiments and measurements were repeated in order to avoid measurement errors and to increase the reliability of the investigation results. The statistical analysis of the data showed that all results are reliable, precise and reproducible.

4 Results

Initially the experimental investigation of in–line tube bundle heat transfer to upward statically stable foam flow was performed. Next, the tube bundle was installed in the output channel and the experiments with downward foam flow followed.

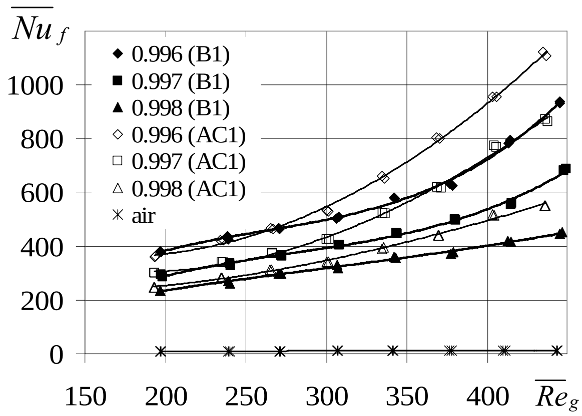

Fig. 3. Heat transfer of the first tube in the middle–line (B1) and in the side–line (AC1) to upward foam flow; β=0.996, 0.997 and 0.998 and the heat transfer of the tube B1 to airflow

Data of heat transfer intensity as a function of upward foam flow velocity for the tubes A1, B1 and C1 of the bundle and for comparison heat transfer intensity of the tube B1 to the one–phase airflow is shown in Fig. 3. The experimental results proved that the heat transfer intensity of in–line tube bundle to the foam flow is much higher than that for the one–phase airflow under the same conditions.

Side–line tubes A and C were located at the same distances from the experimental channel centre (and from the middle–line tube). Foam flow local void fraction and local velocity were the same near those tubes, and heat transfer intensity of A and C tubes were identical as well. Therefore an average heat transfer intensity of the accordingly tubes of the bundle side–lines (A and C) was calculated for the further experimental results analyzis.

Changing gas Reynolds number $\overline{Re}_g$ from 190 to 440, heat transfer intensity of the tube B1 increases about 2.4 times, for β=0.996; 2.2 times, for β=0.997, and 1.9 times, for β=0.998 (Fig. 3). An average heat transfer intensity of the first side–line tubes (AC1) for the same $\overline{Re}_g$ values increases 3.1 times, for β=0.996; 2.9 times, for β=0.997, and 2.2 times, for β=0.998 (Fig. 3). It is evident that heat transfer intensity of the side–line tubes is better than of the middle–line tube B1.

Comparison of an average heat transfer intensity of the third side–line tubes (AC3) and the heat transfer intensity of the third middle–line tube (B3) in upward foam flow is shown in Fig. 4. Heat transfer of the side–line tubes AC3 is better than that of the B3 on average 9% for β=0.996, 13% for β=0.997 and 14% for β=0.998.

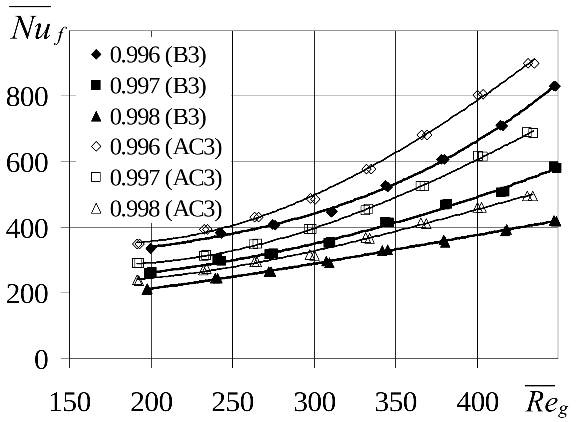

Fig. 4. Heat transfer of the third tube in the middle–line (B3) and in the side–line (AC3) to upward foam flow; β=0.996, 0.997 and 0.998

Comparison of the heat transfer intensity for the middle–line tubes in the upward foam flow at the volumetric void fraction β=0.997 is shown in the Fig. 5. Heat transfer of the first tube is better than of the other tubes for the whole interval of $\overline{Re}_g$. Heat transfer of the second tube is better than that of the third tube; heat transfer of the third tube is better than that of the fourth tube; heat transfer intensity of the fifth tube is better than that of the fourth tube and less than that of the third and the sixth. Heat transfer intensity of the sixth – the last tube is higher than that of the third tube when <330 and less when increases from 330 to 440. Analogical situation is in the case with side–line tubes.

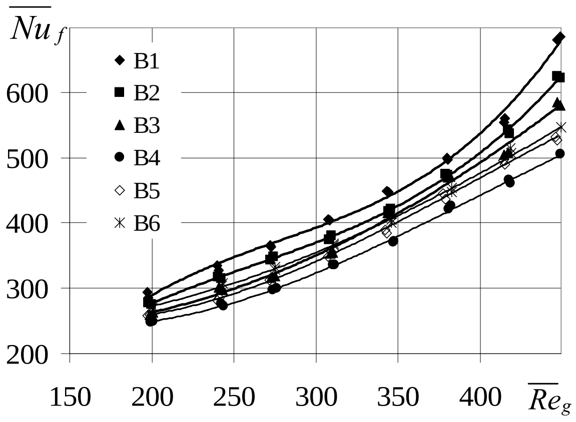

Fig. 5. Heat transfer intensity of the tubes in the middle–line (B) to upward foam flow, β= 0.997

In one–phase flow case heat transfer intensity of the frontal tubes is equal to about 60% of the third tubes heat transfer intensity; heat transfer of the second tubes are equal to about 90% of the third tubes heat transfer intensity; and heat transfer intensity of the fourth and furthered tubes in the in–line tube bundles is like of the third tubes [7]. An experimental investigation with two–phase foam flow shows that the best heat transfer is of the first–frontal tubes of the in–line bundle, less of the second, and so on. The heat transfer of the last and the next–to–last tubes is the exceptional case. Foam structure variation and liquid drainage intensity differences along the channel take place in that case.

The experimental investigation with vertically downward moving after the 180° degree turning foam flow followed. Distribution of the foam local void fraction across the channel transforms during the flow turning. This transformation mainly depends on liquid drainage process from the foam and must be analyzed.

Liquid drainage from foam phenomena is influenced by gravity and capillary. In a vertical direction these forces are acting together. In a horizontal direction influence of gravity forces is negligible and influence of capillary forces is dominating. Influence of the electrostatic and molecular forces on drainage is insignificant. Gravity forces act along the upward and downward foam flow. While foam flow makes turn gravity forces act across foam flow and liquid drains down from the upper channel wall, therefore local void fraction increases here as well. After the turn, local void fraction of foam is less (foam is wetter) on the left side of the cross–section (D tubes on Fig. 2). Flow velocity distribution in cross section of the channel transforms after turn too. All mentioned factors influence on the tube bundle heat transfer intensity to statically stable foam flow.

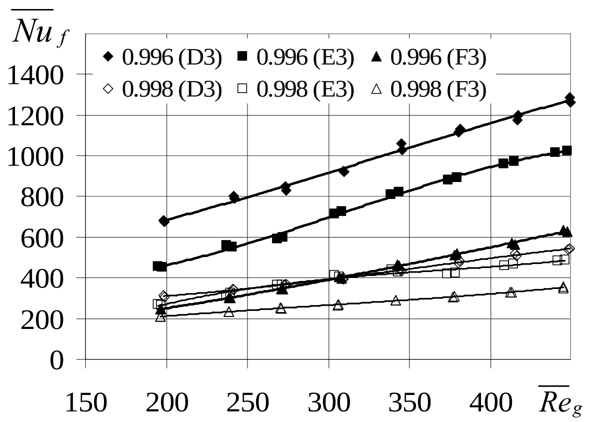

Fig. 6. Heat transfer intensity of the tubes D3, E3, F3 to downward foam flow, β= 0.996 and 0.998

Comparison of heat transfer intensity of the tubes D3, E3 and F3 to the downward foam flow at the volumetric void fraction β=0.996 and β=0.998 after turning is shown in Fig. 6. Heat transfer of the tube D3 is better on average 31% than heat transfer intensity of the tube E3 and heat transfer of the tube E3 is better on average 73% than heat transfer intensity of the tube F3, for β=0.996 (Fig. 6). Heat transfer of the tube D3 is better on average 7% than heat transfer intensity of the tube E3, and heat transfer of the tube E3 is better on average 41% than heat transfer intensity of the tube F3, for β=0.998. Heat transfer of the tube D3 to the wettest foam flow (β=0.996) is by 2.3 times better than that to the driest foam flow (β=0.998).

Comparison of heat transfer intensity of the middle line tubes to downward foam flow at the volumetric void fraction β=0.997 is shown in Fig. 7. Heat transfer of the first tube (E1) increases more intensive in comparison with other tubes and is better than that of the other tubes for whole interval of gas Reynolds number of the foam flow ($\overline{Re}_g$=190÷440). Heat transfer of the second tube (E2) is less than that of the first tube (E1) and so on. Heat transfer of the fourth (E4), fifth (E5) and the last (E6) tubes are nearly the same in whole interval of the $\overline{Re}_g$. A phenomenon of "shadow" effect takes place in downward foam flow; therefore heat transfer intensity for the following tubes is less than for the frontal tubes.

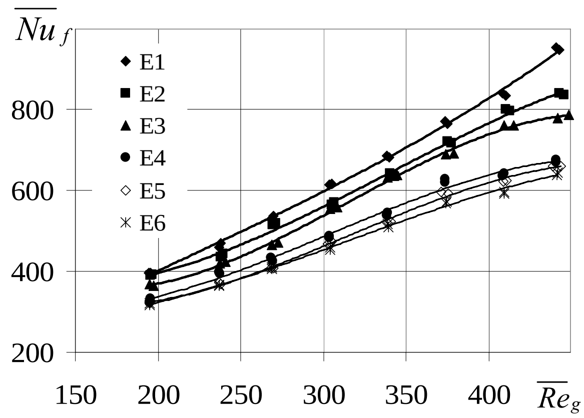

Fig. 7. Heat transfer intensity of the tubes in the middle–line (E) to downward foam flow, β= 0.997

Experimental results of heat transfer of the in–line tube bundle to upward and downward after turning statically stable foam flow were summarised by criterion equations using dependence between Nusselt number $\overline{Nu}_f$ and gas Reynolds number $\overline{Re}_g$ for the foam flow. This dependence within the interval 190 < $\overline{Re}_g$ < 440 for the in–line tube bundle in upward and downward after 180 turning foam flow with the volumetric void fraction β =0.996, 0.997, and 0.998 can be expressed as follows:

$$\overline{Nu}_f = c\beta^n \overline{Re}_g^m . \quad (8)$$

On average, for the entire middle–line (B) in the upward foam flow: c=5.7, n=340, m=102.1(1.006–β); on average, for the entire side–line (A and C) in the upward foam flow c=6.26, n=613, m=147.8(1.004– β) and on average, for the whole in–line tube bundle in the upward foam flow c=5.9, n=479, m=125.3(1.005– β).

On average, for entire left (D) side–line in the downward foam flow c=4, n= –286.5, m=22.5(1.03–β). On average, for entire middle (E) line in the downward foam flow c=16.1, n=518, m=140.7(1.003–β). On average, for entire right (F) side–line in the downward foam flow c=106.6, n=1345, m=270.5(1.0004– β) and on average, for whole in–line tube bundle in the downward foam flow c=12.7, n=334, m=114.6(1.004– β).

5 Conclusions

Heat transfer of the in–line tube bundle to upward and downward after 180º turning statically stable foam flow was investigated experimentally.

Heat transfer intensity of the in–line tube bundle to the foam flow is much higher (from 25 to 100 times) than that to the one–phase airflow under the same conditions.

Distribution of the foam's local void fraction and flow velocity in cross–section of the channel were the main factors which influenced on heat transfer intensity of the different tubes.

Experimental investigation showed that the heat transfer intensity of the frontal and further tubes to downward foam flow is different in comparison with one–phase fluid flow.

Experimental results were generalized by criterion equations, which can be used for the calculation and design of the statically stable foam heat exchangers.

Nomenclature:

A	cross–section area of exper. channel, m^2;
c, n, m	coefficients;
d	external diameter of tube, m;
G	volumetric flow rate, m^3/s;
Nu	Nusselt number;
q	heat flux density, W/m^2;
R	radius of the turn, m;
Re	Reynolds number;
T	temperature, K;
α	average heat transfer coefficient, W/(m^2·K);
β	volumetric void fraction;
λ	thermal conductivity, W/(m·K);
ν	kinematic viscosity (m^2/s).

Indexes:

f	foam;
g	gas;
l	liquid;
w	wall of heated tube.

References:

[1] J. Gylys, *Hydrodynamics and Heat Transfer under the Cellular Foam Systems*, Technologija, Kaunas, 1998.

[2] J. Gylys, S. Sinkunas, T. Zdankus, Analysis of tube bundle heat transfer to vertical foam flow, *Engenharia Termica*, Vol.4, No. 2, 2005, pp. 91–95.

[3] V. Tichomirov, *Foams. Theory and Practice of Foam Generation and Destruction*, Chimija, Moscow, 1983.

[4] B. Fournel, H. Lemonnier, J. Pouvreau, Foam Drainage Characterization by Using Impedance Methods, *3rd Int. Symp. on Two–Phase Flow Modelling and Experimentation*, Pisa, Italy, 2004, p. [1–7].

[5] Anh V. Nguyen, Liquid Drainage in Single Plateau Borders of Foam, *Journal of Colloid and Interface Science*, Vol.249, No.1, 2002, pp. 194–199.

[6] N. G. Vilkova and P. M. Kruglyakov, Investigation of foam and emulsion destruction under the great pressure gradients, *Advances in Colloid and Interface Science*, Vol.108–109, 2004, pp. 159–165.

[7] A. Zukauskas, *Convectional Heat Transfer in Heat Exchangers*, Nauka, Moscow, 1982.

[8] G. F. Hewitt, *Heat exchanger design handbook 2002*, York, Begell House, 2002.

[9] J. Gylys, M. Jakubcionis, S. Sinkunas, T. Zdankus, Analysis of staggered tube bank heat transfer in cross foam flow, *1st International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics*, South Africa, vol. 1, part 2, 2002, pp. 1115–1120.

[10] J. Gylys, S. Sinkunas and T. Zdankus, Experimental Study of Staggered Tube Bundle Heat Transfer in Foam Flow, *5th International Symposium on Multiphase Flow, Heat Mass Transfer and Energy Conversion*, Xi'an, China, 2005, p. [1–6].