ANALYSIS OF TUBE BUNDLE HEAT TRANSFER TO VERTICAL FOAM FLOW Paper CIT04-0105

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Abstract. Phenomena of foam flow and heat transfer to it are rather complex. Foam is two–phase flow, which structure changes while it passes an obstacle: bubble divides into the smaller bubbles and liquid drains down from flow. All these peculiarities make extremely complicated an application of analytic methods for their study. Thus experimental method of investigation was selected in our work. The investigations were performed on the experimental laboratory stand consisting of foam generator, vertical channel and staggered bank of the horizontal tubes. Cross section of the channel had square profile with side dimension 140 mm. Tubes of the bank were located in three vertical rows with five tubes in each of them. Outside diameter of tubes was 20 mm. Experiments were performed within Reynolds number diapason for gas 190÷450, foam volumetric void fraction – 0.996÷0.998. Direction of foam motion in vertical channels also influences on heat transfer intensity. Investigations of heat transfer process to upward and downward moving statically stable foam flow from horizontal tube bank was performed.

Keywords. *foam flow, void fraction, heat transfer, tube bundle, upward and downward flow.*

1. Introduction

Heat is one of the most widely used kinds of energy. Thermal processes are spread within different industry branches. Rapidly developing modern technologies enable us to implement and control the most complicated technological processes. The tasks of reducing energy costs, and at the same time product costs, are becoming increasingly urgent. Heat and mass exchange frequently accompany technological processes. Depending upon the character of a process, heat can be consumed or transferred. In heat exchangers most frequently used heat agents are liquid or gas. But most probably, the use of alternative heat agents in some cases would enable us to achieve better results and allow the reduction of energy input.

Foam is distinguished by especially large inter–phasic contact surface and can be applied for the conduction of different purpose heat and mass exchanges. The efficiency of mentioned processes depends on the capacity "to control" the foam. Presently, the most widely researched processes include hydrodynamic, heat and mass exchangers taking place in the flows of dynamic and fire-fighting foam (Sadoc and Rivier, 1997). However, these processes are investigated insufficiently especially in the regime of statically stable foam, which regime can be defined according Gylys (1998). Statically stable foam is formed up only of detergent solutions having reduced surface tension. Even small concentration of detergents may be the reason of intensive generation of statically stable foam due to bubbling of gas. Bubble in statically stable foam keeps initial dimensions within broad limits of time interval from several seconds to days even after termination of gas supply. Gylys (1998) showed that 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.

The application of statically stable foam flow in heat exchangers is perspective in both economic and engineering aspects. Studies of Gylys (1998) showed that heat exchangers with statically stable foam as carrier agent for heat transfer have many advantages: small quantities of consumed liquid, relatively large heat transfer rate, low energy consumption required for foam delivery to heat exchange space and etc.

2. Experimental Equipment

The experimental set–up (Gylys et all, 2002) consists of the following main parts: foam generation channel, 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).

The experimental channel had a riddle at the bottom of experimental part. For the experiments with downward moving foam flow the tube bundle was located in the output part of channel. The whole experimental channel was made of glass in order to observe visually foam flow structure and the size of foam bubbles. The cross section of the channel had dimensions 0.14 x 0.14 m. The height of experimental channel was 1.8 m. Foam flow was generated on the riddle. The water solution of detergents was used in experiments. Concentration of detergents was kept constant and it was equal 0.5 %. Liquid was delivered from reservoir to the riddle from upper side; gas was supplied to the riddle from below. Foam flow was produced during gas and liquid contact. Liquid in experiment was used only once and was not returned back to the reservoir.

Schematic view of experimental section of the channel with tube bundle is presented in Fig.2. The bundle of tubes consisted of three vertical rows with five tubes in each. Spacing among the centres of the tubes $s_1=s_2=0.035$m. All tubes had an external diameter of 0.02 m. The heated tube was made of copper and had an external diameter of 0.02 m also. The endings of the tube were sealed and insulated. The tube was heated electrically. An electric current value was measured by ammeter and voltage by voltmeter. The temperature of foam flow was measured by two calibrated thermocouples: one in front of the bundle and one behind. The temperature of heated tube surface was measured by eight calibrated thermocouples. Six of them were placed around central part of heated tube and two of them were placed in both sides of the tube at 50 mm distance from the central part.

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

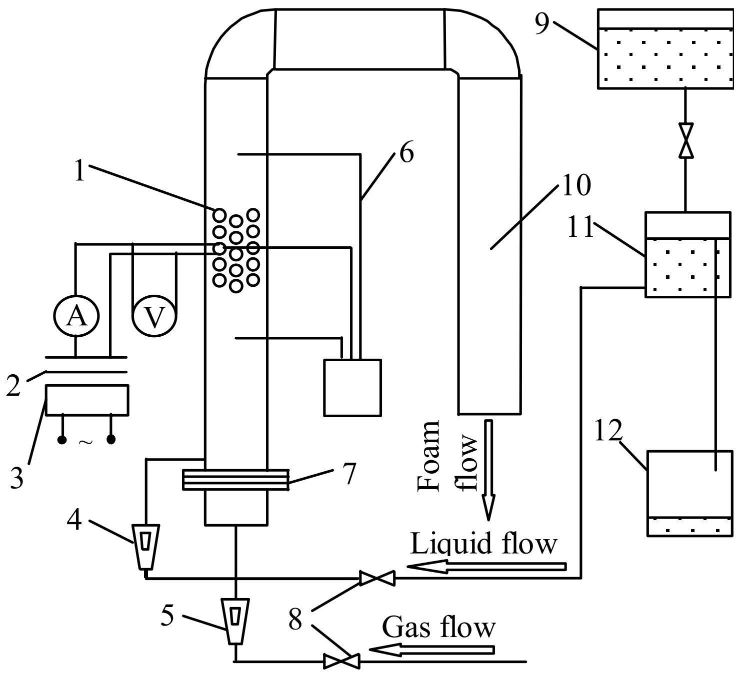

Figure 1. Scheme of experimental equipment: 1–foam channel; 2–transformer; 3–stabilizer; 4–liquid flow meter; 5–gas flow meter; 6–thermocouples; 7–foam generation riddle; 8–gas and liquid control valves; 9–liquid reservoir; 10–output channel; 11–liquid level control reservoir; 12–liquid receiver

The experiments were provided for different values of mean volumetric void fractions β=0.996, 0.997 and 0.998. Volumetric void fraction was calculated by the equation

$$\beta = \frac{G_g}{G_g + G_l} \tag{1}$$

where G_g – gas volumetric flow rate, m^3/s; G_l – liquid volumetric flow rate, m^3/s.
The foam flow rate can be written as

$$G_f = G_g + G_l \tag{2}$$

where G_f – foam volumetric flow rate, m^3/s.

Temperature of tube surface and foam flow, electric current and voltage were measured and recorded during the experiments. Our preliminary investigations showed that hydraulic and thermal regime stabilizes completely within 90 s after the change of experiment conditions. Therefore measurements were started not earlier than 90 s after adjustment of foam flow parameters. Heat flux density on the tube surface q_w was calculated after registration of electric current and voltage.

Difference of temperature ΔT (between the mean temperatures of the foam flow $\overline{T}_f$ and tube surface T_w) was calculated after record of heated tube surface and foam flow temperatures.

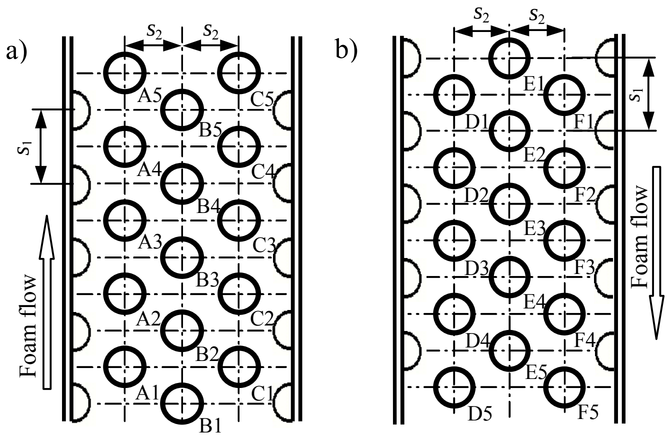

Figure 2. Tube bundle in upward (a) and downward (b) foam flow

The average heat transfer coefficient was calculated as

$$\overline{\alpha} = \frac{q_w}{\Delta T} \tag{3}$$

The Nusselt number was computed by formula

$$\overline{Nu}_f = \frac{\overline{\alpha} d}{\lambda_f} \tag{4}$$

where λ_f is the averaged thermal conductivity of the statically stable foams flow, obtained from the correlation

$$\lambda_f = \beta\lambda_g + (1-\beta)\lambda_l \tag{5}$$

where λ_g – gas thermal conductivity, W/(m·K); λ_l – liquid thermal conductivity, W/(m·K).
The gas Reynolds number was computed by formula

$$\overline{Re}_g = \frac{G_g d}{F \nu_g} \tag{6}$$

where d – external diameter of tube, m; F – cross section area of experimental channel, m^2; ν_g – gas kinematic viscosity, m^2/s.

All experiments and measurements were repeated in order to avoid measurement errors and to increase reliability of investigation results.

The experimental uncertainties according Schenck (1972) in the range of test data variation: β=1.7±2.2 %, $\overline{\alpha}$=1.9±8.0%, $\overline{Nu}_f$ =2.0±8.1 %, $\overline{Re}_g$ =1.9±2.2 %.

3. Results

The experimental results show great dependencies of heat transfer intensity on mean gas velocity $\overline{w}_g$ and volumetric void fraction β. With an increase of $\overline{w}_g$ and decrease of β, heat transfer intensity increases. Data of heat transfer intensity as function of $\overline{Re}_g$ for the first tube of the middle line of the tube bundle in upward moving statically stable foam flow are shown in Fig. 3. When flow Reynolds number $\overline{Re}_g$ changes within the limits 190–470, heat transfer intensity of the first tube B1 in the middle line tube bank increases (when foam volumetric void fraction β=0.996), and $\overline{Nu}_f$ changes from 260 to 1090 (Fig.3).

The last tube in the middle line of the bundle is an exception from others. It was determined that the decrease of volumetric void fraction from 0.997 to 0.996 intensifies heat transfer rate of the last tube in the bundle greatly. This can be explained by the following reasons. The foam is wetter and larger quantity of liquid is transferred within them. A layer of smaller foam forms up above the bundle. This layer occurs when larger foam bubbles are divided into smaller ones. In this layer more intensive drainage of foam takes place, and at the same time, heat exchange is intensified. The comparison of heat transfer intensity of the bundle's middle line tubes in the flow of statically stable foam at void volumetric fraction β=0.997 is shown in the Fig.4.

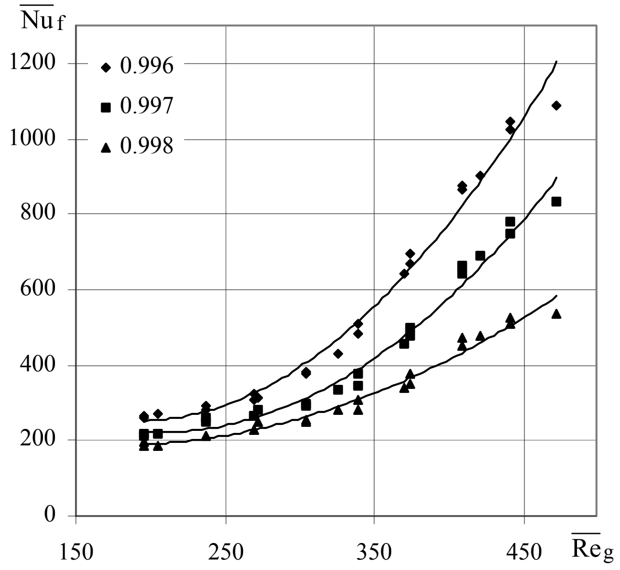

Fig. 3 Heat transfer of the first tube (B1) in the middle line of the bundle in upward moving foam flow; *β*=0.996, 0.997, 0.998

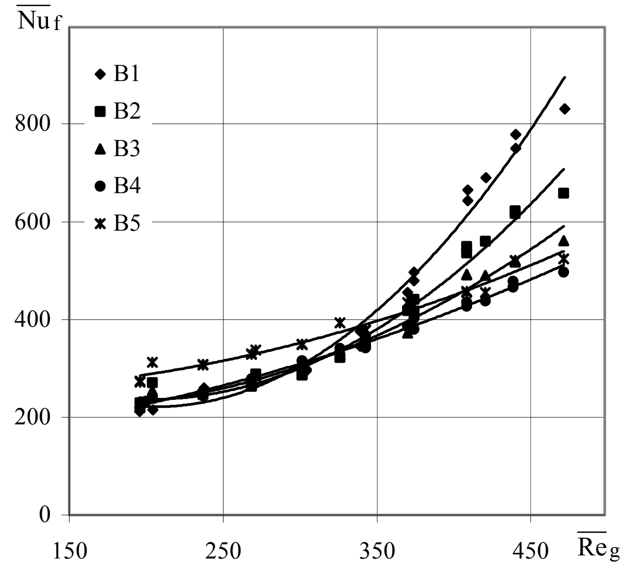

Fig. 4 Heat transfer intensity of the tubes in the middle line of the bundle in upward moving foam flow, *β*= 0.997

When foam flow Reynolds number $\overline{Re}_g$ changes within the limits 190÷300, heat transfer intensity slightly depends on tube's position in the middle line of the bundle. This conclusion is applicable only for the first four tubes of the middle line. We can assert that the heat transfer intensity of the first four tubes in the middle line of the bundle is similar. Only the fifth tube is distinguished by the best heat transfer rate to statically stable foam flow within above indicated conditions. This probably due to the fact that when gas velocity is small, foam flow consists of large size bubbles (d_b= 8÷14 mm). Large bubbles of foam passing through the bundle of tubes are divided into smaller bubbles (d_b=2÷5 mm). It means that a layer of small bubbles forms up over the bundle and drainage process between heated surface and statically stable foam becomes more intensive.

When Reynolds number of flow $\overline{Re}_g$ exceeds 300 the heat transfer rates of tube B1 and B2 increase greatly (Fig.4). Heat transfer of tube B3 and B4 is smaller, which indicates the change of flow character near different tubes of the row. It should be noted that heat transfer intensity of the fifth, i.e. the last tube in the line B5, is larger than that of the fourth B4. The main reason of this change of heat transfer intensity was already mentioned. When mean gas velocity increases, the foam of smaller bubbles starts forming up in the foam generating channel. Smaller foam bubbles, greater number and more dense distribution of them make the foam flow more homogenous and better washing heated surfaces.

Visual observations and comparison of upward and downward moving foam flow (Fig. 5) allows noticing obvious differences. Downward moving foam flow has better condition liquid drainage. Separated from foam liquid forms layer in corners of the channel and on the walls of it and moves down under the action of gravity forces. Presence of the liquid layer increases channel hydraulic resistance for upward moving foam flow and reduces it for downward one.

When flow Reynolds number $\overline{Re}_g$ changes within the limits 190÷325 (when β=0.996), foam flow consists of large size bubbles (d_b= 8÷14mm). Passing through the bundle of tubes large bubbles of foam divide into smaller ones. The liquid drainage process becomes more intensive. As a result the heat transfer intensity increases. Therefore the fifth E5 tube has the best heat transfer rate to downward statically stable foam flow. Heat transfer intensity of the fourth E4, the third E3, the second E2 and the first E1 tubes grows slower in the same order (Fig. 6a).

When $\overline{Re}_g$ exceeds the value 325 at β=0.996, heat transfer of the first tube E1 increases more intensively in comparison with other tubes. It can be explained by the fact that the foam of smaller bubbles starts forming up in the foam generating channel and the structure of the foam flow varies not much passing the bundle of tubes. When flow Reynolds number $\overline{Re}_g$ is near 430 the heat transfer intensity is the best of the first tube, less of the second and so on, according to increasing number of them.

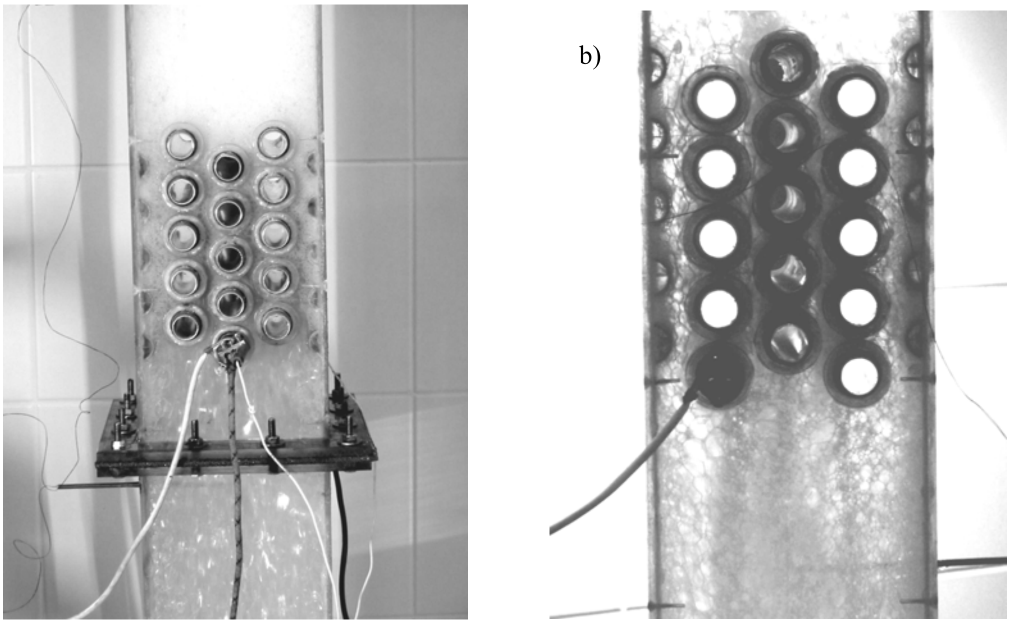

Figure 5. Statically stable upward (a) and downward (b) directed foam flow and tube bundle in the experimental chann.

The heat transfer intensity is nearly independent from tube position in the line when downward directed statically stable foam flow at volumetric void fraction of β=0.998 passes the tube bundle and Reynolds number $\overline{Re}_g$ changes within the limits 190–235 (Fig. 6b).

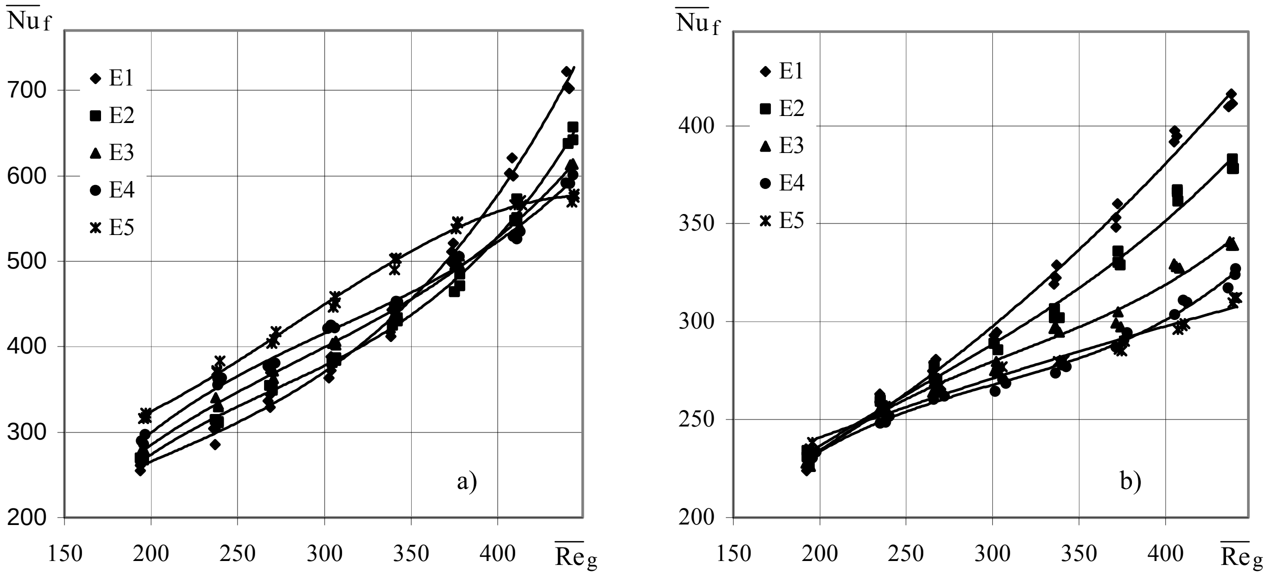

Figure 6. Heat transfer intensity of the tubes in the middle line of the bundle in downward foam flow: a) β= 0.996, b) β= 0.998

Heat transfer rate of side-line tubes in the bundle in cross-upward statically stable foam flow was investigated in similar way. The experimental results of the dependence of mean heat transfer coefficient on mean gas velocity were generalized by using the dependence of Nusselt and Reynolds similarity criteria. This dependence within interval $190 < \overline{Re}_g < 300$ of upward moving foam flow at volumetric void fraction β=0.996; 0.997; 0.998 can be expressed by eqn (7):

$$\overline{Nu}_f = c\overline{Re}_g^n \tag{7}$$

where c, n – coefficients; n = -60(β - 1.0072); c = a(β - 0.99).

On average, for entire middle line in the bundle a=1735, for entire side–line in the bundle a=1880, and for entire tube bundle a=1820.

To determine heat transfer rate depending on Reynolds and Nusselt similarity criteria within the interval $300 < \overline{Re}_g < 450$ and at foam volumetric void fraction β=0.996; 0.997; 0.998 the following equation may be used (Gylys, 2002)

$$\overline{Nu}_f = c\beta^u \overline{Re}_g^m \tag{8}$$

where c, u, m – coefficients.

For entire middle line of tubes in the bundle: u=890, m= -200(β-1.004), c=2; for entire side–line of tubes in the bundle: u=1000, m= -200(β-1.0007), c=127; for entire tube bundle allocated in staggered order: u=950, m= -200(β-1.004), c=2.3.

The experimental results of downward foam flow were generalized by using the dependence of Nusselt and Reynolds number similarity criteria. This dependence within interval $190 < \overline{Re}_g < 430$ and at foam volumetric void fraction β=0.996; 0.997; 0.998 can be expressed by eqn (8) where for middle line of the bundle: m=206(1-β); u=993; c=113. For side–line of the bundle: m=226(1-β); u=1014; c=176. For tube bundle allocated in staggered order: m=217(1-β); u=1003; c=144.

4. Conclusions

1. The heat transfer intensity of tube bundle to vertical foam flow depends not only on foam flow velocity and volumetric void fraction but on foam flow direction as well.
2. When analyzing the dependence of heat transfer upon the direction of vertical foam flow, it was observed that where the foam flow velocity ($\overline{Re}_g \in [190; 300]$) is lower, heat exchange is more intensive in downward foam flow. With an increase of foam flow velocity, ($\overline{Re}_g \in [300; 440]$), tubes are more intensely cooled in the upward foam flow.
3. Experimental results of tube bundle's heat transfer to vertical cross foam flow were summarized by criterion equations, which enable to calculate heat transfer rate of the entire bundle or separate tube of the bundle at different values of void volumetric fractions and regime parameters of statically stable foam flow. The obtained criteria equations can be used for the computation and designing of foam heat exchangers.

5. References

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