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ANALYSIS OF THE IN–LINE TUBE BUNDLE HEAT TRANSFER TO THE FOAM FLOW

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ABSTRACT

Foam is distinguished by especially large inter–phase contact surface and can be used as a coolant in heat exchangers or in foam apparatus. But it should be noted that foam is a two–phase system with a number of peculiarities, which are closely linked with each other and make extremely complicated an application of analytic methods for the study of heat transfer in foam. Experimental investigation of heat transfer process from the in–line tube bundle to the laminar upward and downward directed foam flow was performed. Dependency of the heat transfer intensity on the following parameters was determined: flow velocity, direction of flow, volumetric void fraction of foam and liquid drainage from foam. Apart of this, influence of tube position of the bundle to heat transfer was investigated. The results of the study of in–line tube bundle heat transfer to foam flow are presented in this paper.

Keywords: heat transfer, void fraction, vertical foam flow, in–line tube bundle.

NOMENCLATURE

A cross section area of experimental channel, m^2
c coefficient
d outside diameter of tube, m
G volumetric flow rate, m^3/s
I amperage, A
l tube length, m
m coefficient
n coefficient
Nu Nusselt number
q heat flux density, W/m^2
Re Reynolds number
T temperature, K
U voltage, V

$\bar{\alpha}$ 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

INTRODUCTION

Our previous studies (Gylys, 1998) showed that characteristics of one type of foam – statically stable foam – allow it to be used as a coolant. Statically stable foam is such type of foams, which keeps its initial structure and bubbles' dimensions within broad limits of time intervals, from several seconds to days, even after termination of the foam generation (Gylys, 1998). This type of foam can be generated from the solutions, which have less then pure liquid surface tension. Even small concentration of detergents may be the reason of intensive generation of statically stable foam due to the bubbling of gas. There exists a 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 (Tichomirov, 1983).

Phenomena of foam flow and heat transfer in it are rather complex. Foam is two–phase flow and its structure changes while it passes obstacle: bubbles are changing their sizes and liquid drainage is going on. The continuous drainage of liquid from foam (Fournel, Lemonnier and Pouvreau, 2004), diffusive gas transfer and disintegration of inter–bubble films (Sadoc and Rivier, 1997) destruct the foam flow at the same time.

Heat transfer of different tube bundles to one–phase fluids was investigated enough (Zukauskas, 1982), but there practically aren't data of tube bundles heat transfer to foam flow. In our previous works heat transfer of a single cylindrical surface – tube and of tube line to upward statically stable foam flow was investigated (Gylys, 1998). Next experimental series with staggered tube bundle in upward and downward foam flow followed (Gylys et al, 2005). Main task of these investigations was to determine the influence of the foam flow parameters, such as flow velocity, direction of flow, volumetric void fraction of foam and liquid drainage from foam, on the tube bundle heat transfer intensity. Influence of tube position in the bundle on heat transfer intensity was investigated also.

Presently experimental investigation of heat transfer process from the in–line tube bundle to the vertical upward and downward statically stable foam flow was performed.

Results of investigations were generalized using relationships between Nusselt number and Reynolds number and volumetric void fraction of foam. The obtained generalized equation can be used for the designing of foam heat exchangers and calculating of heat transfer intensity of the in–line tube bundle.

EXPERIMENTAL SET–UP

The experimental set–up consisted of the following main parts: experimental channel, in–line 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). Cross section of the experimental channel had dimensions 0.14 x 0.14 m; height of it was 1.8 m.

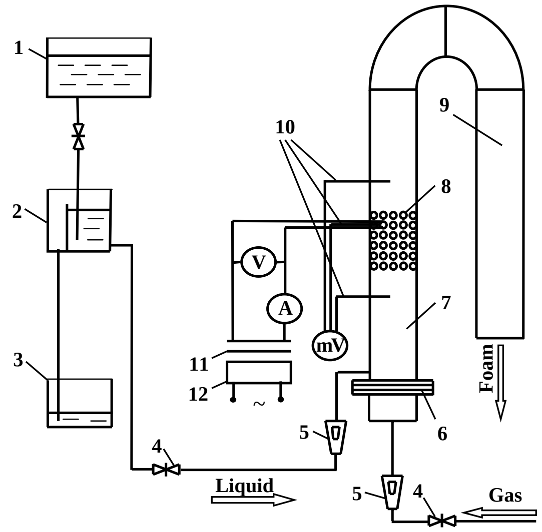

Figure 1: Experimental set–up scheme: 1–liquid reservoir; 2–liquid level control reservoir; 3–liquid receiver; 4–gas and liquid control valves; 5–flow meter; 6–foam generation riddle; 7–experimental channel; 8–tube bundle; 9–output channel; 10–thermocouples; 11–transformer; 12–stabilizer

The tube bundle consisted of five vertical rows with six tubes in each (Fig. 2). Spacing among centers of the tubes across and along the channel was s_1=s_2=0.03 m. An external diameter of the tubes was equal to d=0.02 m. Electrically heated tube – calorimeter was made from copper and had an external diameter equal to 0.02 m also. An electric current value was measured by ammeter and voltage by voltmeter. Temperature of foam flow was measured by two calibrated thermocouples: one in front of the bundle and one behind it. Temperature of heated tube surface was measured by eight calibrated thermocouples: six of them were placed around central part of the tube and two of them were placed in both sides of the tube at 50 mm distance from the central part.

Water solution of detergents was used in experiments. Concentration of detergents was kept constant and it was equal to 0.5 % by mass. Foam flow was produced during gas and liquid contact on the riddle, which was mounted at the bottom of the experimental channel. Liquid was delivered from reservoir to the riddle from upper side; gas was supplied to the riddle from below. Detergent solution was used in experiment only once and it was not returned back to the reservoir.

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

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

Foam flow volumetric void fraction can be expressed by the equation

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

Gas Reynolds number of foam flow was computed by the formula

$$\overline{Re}_g = \frac{G_g d}{A v_g}\,. \tag{3}$$

The Nusselt number was computed by the formula

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

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

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

The average heat transfer coefficient was calculated as

$$\overline{\alpha} = \frac{q_w}{\overline{\Delta T}}\,. \tag{6}$$

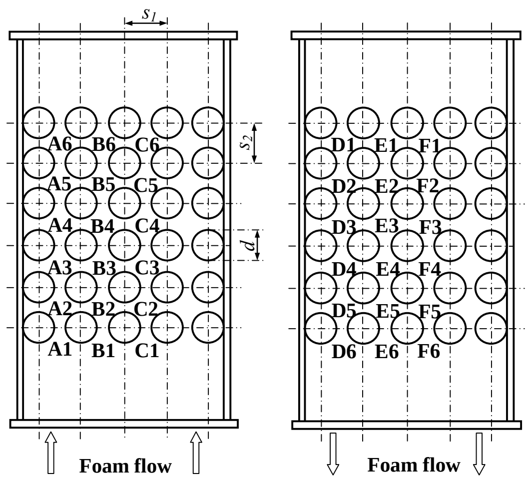

Figure 2: The in–line tube bundle in upward and downward foam flow

Investigations showed that hydraulic and thermal regime stabilises completely within five minutes after the change of experimental conditions. Therefore data recording of 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:

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

The difference of temperature between the mean temperatures of the foam flow $\overline{T}_f$ and tube surface $\overline{T}_w$ was calculated by the equation

$$\overline{\Delta T} = \overline{T}_w - \overline{T}_f . \qquad (8)$$

The investigation of tube heat transfer from the bundle consisted of three series of experiments, which were provided for three different values of mean volumetric void fractions β=0.996, 0.997 and 0.998 within laminar regime of foam flow.

All experiments and measurements were repeated in order to reduce 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.

The walls of experimental channel were made from transparent material and during experiments foam flow was observed visually (Fig. 3).

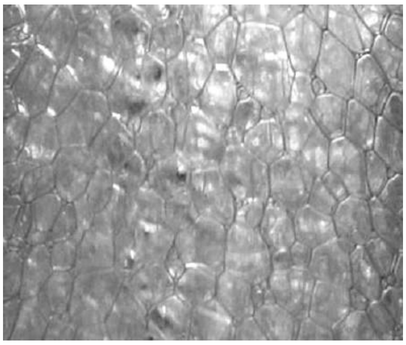

Figure 3: Statically stable foam flow in the experimental channel

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

RESULTS

The investigations of heat transfer between the tube bundle and foam flow were provided in upward and downward statically stable foam flow. For the experiments with downward moving foam flow the tube bundle was located in output part of channel. The experimental results show great dependencies of heat transfer intensity on supplied gas rate and volumetric void fraction β. 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 and downward statically stable foam flow is shown in Fig. 4.

As the gas Reynolds number for the foam flow $\overline{Re}_g$ increases from 190 to 440, heat transfer intensity ($\overline{Nu}_f$) of the first tube (B1) in upward foam flow increases by 2.5 times for the foam with volumetric void fraction β=0.996 and by 2.3 times for β=0.997, and by 1.9 times for β=0.998. The heat transfer intensity of the first tube (E1) in downward foam flow increases by 2.5 times for β=0.996 and by 2.4 times for β=0.997, and by 1.8 times for β=0.998.

The heat transfer intensity of the first tubes in downward foam flow is better than in upward flow (Fig. 4), because downward moving foam flow has a better condition of liquid drainage. Separated from the foam, liquid forms a layer in the corners of the channel and on its walls, moving downward under the action of gravity forces. The presence of this liquid layer increases channel hydraulic resistance for upward moving foam flow and reduces it for downward flow.

The heat transfer intensity of the tube B1 for the foam flow is much higher than that for the one–phase airflow under the same conditions (Fig. 4).

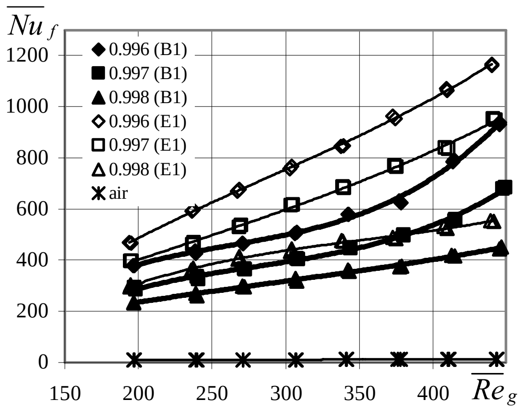

Figure 4: Heat transfer of the first tube for upward (B1) and downward (E1) foam flow; β=0.996, 0.997 and 0.998; and the heat transfer of the tube B1 in airflow

The comparison of the heat transfer intensity of the third middle–line tube in upward (B1) and downward (E1) foam flow is shown in Fig. 5.

The heat transfer of the third tube in downward foam is better on average 41 % for β=0.996, 46 % for β=0.997 and 26 % for β=0.998 than that of the third tube in upward foam flow.

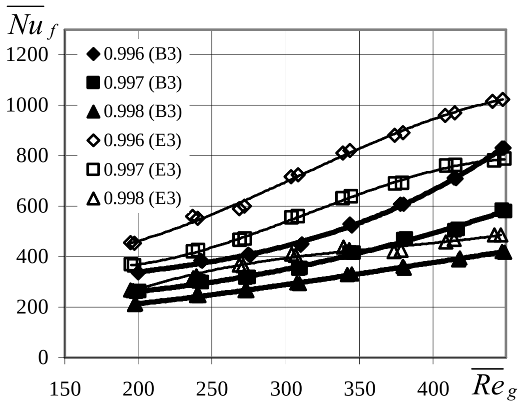

Figure 5: Heat transfer of the third tube for upward (B3) and downward (E3) foam flow; β=0.996, 0.997 and 0.998

In comparison with the first middle–line tube the heat transfer intensity of the third tube in upward and downward foam flow cases are worse for whole interval of $\overline{Re}_g$ ($\overline{Re}_g$=190÷440).

The comparison of 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. 6. The heat transfer of the first tube is better than of the other tubes for the whole interval of $\overline{Re}_g$. The heat transfer of the second tube is better than that of the third tube. The heat transfer of the third tube is better than that of the fourth tube. The 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. The heat transfer intensity of the sixth – the last tube is higher than that of the third tube when $\overline{Re}_g$ <330 and less when $\overline{Re}_g$ increases from 330 to 440.

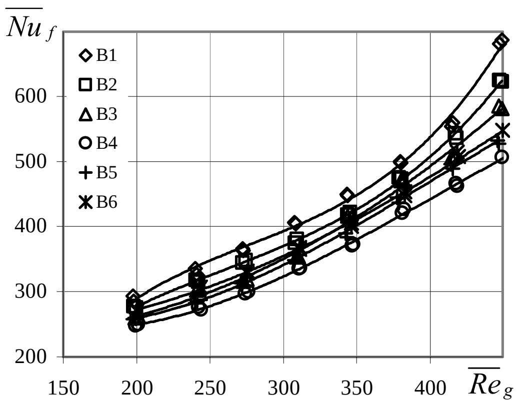

Figure 6: Heat transfer intensity of the tubes in the middle–line in upward foam flow, β= 0.997

The 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.

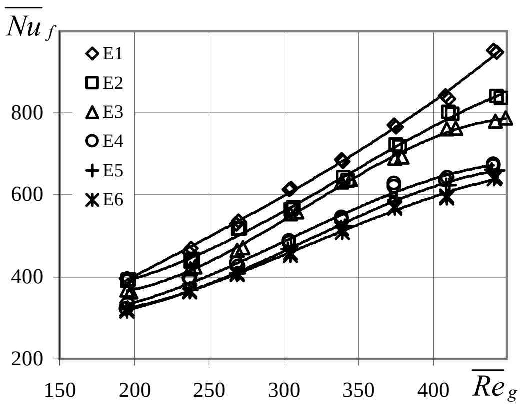

Figure 7: Heat transfer intensity of the tubes in the middle–line in downward foam flow, β= 0.997

The heat transfer of the first tube (E1) increases more in intensity in comparison with other tubes and is better than that of the other tubes for whole interval of gas Reynolds number of foam flow ($\overline{Re}_g$ =190÷440). The heat transfer of the second tube (E2) is less than that of the first tube (E1) and so on. The heat transfer of the fourth (E4), fifth (E5) and the last (E6) tubes are nearly the same in whole interval of the $\overline{Re}_g$.

In one–phase flow case the heat transfer intensity of frontal tubes are 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 the heat transfer intensity of the fourth and furthered tubes in the in–line tube bundles are similar to the third tubes. 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 in upward foam flow are the exceptional case. The variation of foam structure and different intensity of liquid drainage along the channel take place in that case.

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

$$\overline{Nu}_f = c\beta^n \overline{Re}_g^m . \qquad (9)$$

On average, for the whole in–line tube bundle in upward foam flow: c=5.9, n=479, m= 125.3(1.005–β) and on average, for the whole in–line tube bundle in downward foam flow: c=12.7, n=334, m= 114.6(1.004–β).

CONCLUSION

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

The heat transfer between in–line tube bundle and foam flow is more intensive in downward foam flow for whole gas Reynolds number of foam flow $\overline{Re}_g$ =190÷440.

The experimental investigation showed that the heat transfer of the frontal tubes to foam flow is the best. It is different in comparison with one–phase fluid flow case. The experimental results were generalized by criterion equations, which can be used for the calculation and design of the statically stable foam heat exchangers.

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