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Influence of the Foam Flow Turn on the Staggered Tube Bundle Heat Transfer Intensity

J. GYLYS and T. ZDANKUS
Energy Technology Institute
Kaunas University of Technology
Kaunas, K. Donelaicio 20, LT–44239
LITHUANIA

S. SINKUNAS
Department of Thermal and Nuclear Energy
Kaunas University of Technology
Kaunas, K. Donelaicio 20, LT–44239
LITHUANIA

Abstract: Foam is two–phase system with number of specific peculiarities, which extremely complicates an application of analytic methods for the study of heat transfer in foam. The heat transfer of the tubes located in staggered tube bundle to foam flow after the 180 degree turn was investigated experimentally. Investigation was performed on the experimental laboratory set–up consisting of the foam generator, foam channel and the staggered bundle of the horizontal tubes. The statically stable foam flow was used for the experimental investigation. Volumetric void fraction of the foam was 0.996÷0.998; gas velocity for foam flow was changed from 0.14 to 0.32 m/s; heat transfer coefficient varied from 160 to 1270 W/(m^2·K). The results of the investigation are presented in this paper.

Key-Words: heat transfer, void fraction, foam flow, flow turn, staggered tube bundle, experimental channel.

1 Introduction

Foams are suitable for different purposes. Foam is distinguished by especially large inter–phase contact surface and can be used as coolant in heat exchangers or in foam apparatus. One significant problem appears in this case – foam must keep its initial structure and bubbles' dimensions within broad limits of a long time intervals. Characteristics of one type of foam – statically stable foam – demonstrated its perfect availability for this purpose [1]. Statically stable 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 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, 2]. However, the concentration of detergents in solution predetermines the gas content of generated foam. Larger detergents' concentration allows generating foam of smaller liquid content [3]. For experimental foam production the concentration of detergents must ensure required stability of foam and satisfy defined requirements to volumetric void fraction [1].

Heat transfer of different tube bundles to one–phase fluids was investigated enough, but practically there is insufficiency of the data of the tube bundles heat transfer to foam flow. Heat transfer of alone cylindrical tube and of tube line to upward statically stable foam flow was investigated in our previous works [1]. The experimental series with staggered tube bundle in upward foam flow [4, 5] followed as well.

Presently experimental investigation of heat transfer process from the staggered tube bundle to the downward (after the turn) moving statically stable foam flow was performed. It was determined dependence of heat transfer intensity on flow parameters: flow velocity, 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. Results of investigations were generalized using relationships between Nusselt number and Reynolds number and volumetric void fraction of foam. Those investigations showed sufficient influence of the turn on the void fraction distribution across the foam channel, and on the heat transfer intensity of the different tubes at the same cross section of the channel as well.

There was performed investigation in order to examine the influence of the 180 degree turn (from up to down) on the heat transfer intensity across the staggered tube bundle.

2 Experimental Set–up

The experimental set–up consisted 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).

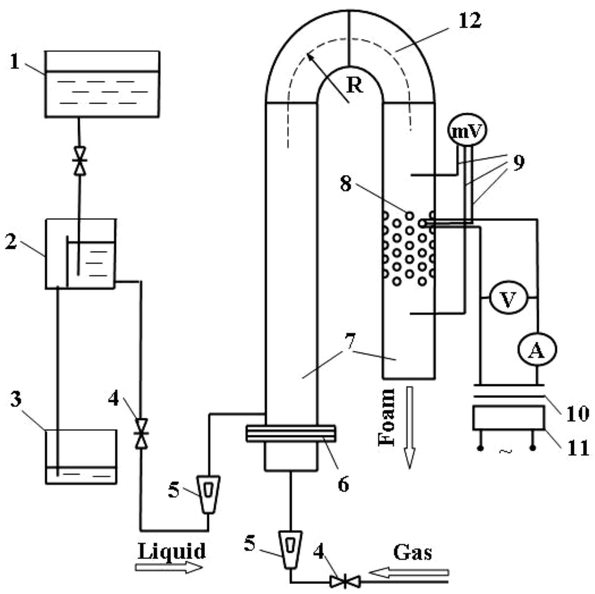

Fig. 1. Experimental set–up scheme: 1–liquid reservoir; 2–liquid level control reservoir; 3–liquid receiver; 4–gas and liquid control valves; 5–flow meter; 6–foam generation riddle; 7–experimental channel; 8–tube bundle; 9–thermocouples; 10–transformer; 11–stabiliser; 12–foam flow turn

Experimental channel had a riddle at the bottom and experimental part. The cross section of the channel had dimensions 0.14 x 0.14 m^2. The height of experimental channel was 1.8 m. Foam flow was generated on the riddle. Water solution with detergents was delivered from reservoir to the riddle from sides; gas was supplied to the riddle from below. When gas and liquid contacted, foam flow was produced. Liquid in experiment was used only once and was not returned back to reservoir.

A riddle of the foam generator was made of stainless steel plate with thickness of 2 mm. The diameter of the holes was 1 mm; spacing among centers of the holes was 5 mm. The holes were located in staggered order.

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

The bundle of tubes consisted of three vertical rows with five tubes in each. Spacing among the centers of the tubes $s_1 = s_2 = 0.035$ m. All tubes had an external diameter of 0.02 m. Heated tube was made of copper and had an external diameter of 0.02 m also. Endings of tube were sealed and insulated to prevent heat losses through them. Tube was heated electrically. An electric current value was measured by ammeter and voltage by voltmeter.

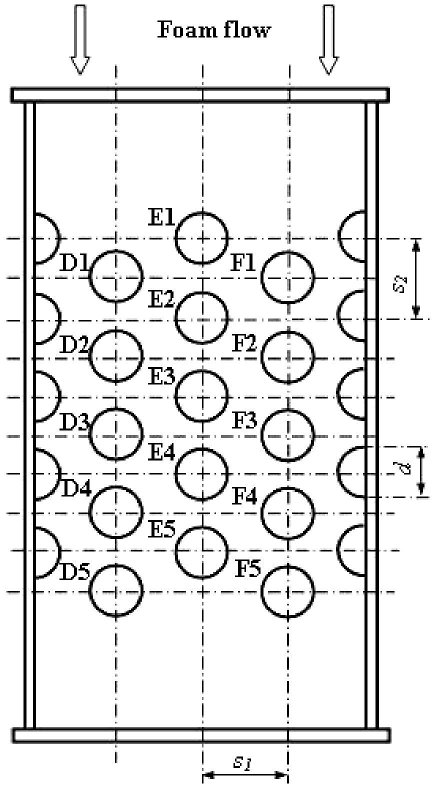

Fig. 2. Tube bundle in foam flow

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 heated tube and two of them were placed in both sides of the tube at 50 mm distance from the central part.

Water solution was used in experiments. Concentration of detergents was kept constant and it was equal 0.5%.

The radius R of the turn 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 a relationship was obtained 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:

$$\overline{Nu_f} = f\left(\beta, \overline{Re_g}\right). \tag{1}$$

The Nusselt number was computed by the formula

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

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

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

The 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

$$\overline{Re_g} = \frac{G_g d}{A \nu_g} . \tag{5}$$

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

Fig. 3. Statically stable foam flow and tube bundle in the experimental channel

Experiments were performed within Reynolds number diapason for gas 190÷440 and foam volumetric void fraction – 0.996÷0.998. The velocity of the gas for foam flow was changed from 0.14 to 0.32 m/s. The heat transfer coefficient varied from 160 to 1270 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.

The walls of experimental channel, including 180° turn, were made from transparent material and during experiments foam flow was observed visually (Fig. 3).

The experimental uncertainties [6] in the range of test data variation: $\overline{\alpha}$=1.9÷8.0%, $\overline{Nu}_f$=2.0÷8.1% and $\overline{Re}_g$ = 1.9÷2.2%.

4 Results

The experimental results show great dependencies of tube bundle heat transfer intensity on gas Reynolds number of foam flow, volumetric void fraction and tube position in the bundle.

Generated on the foam generation riddle statically stable foam was directed vertically upward, then made 180° degree turning and moved downward and crossed the staggered tube bundle.

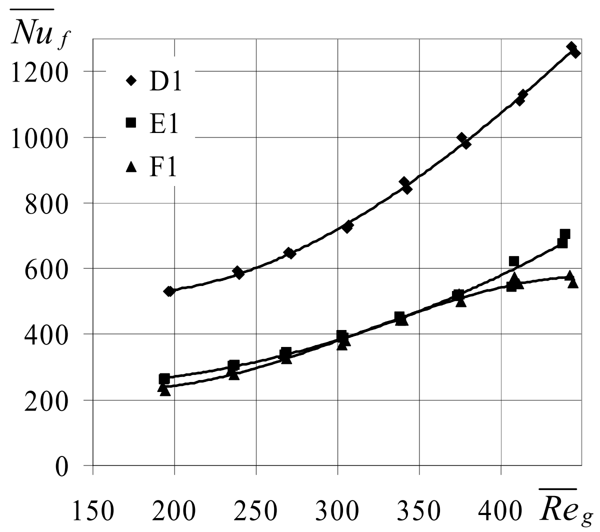

Fig. 4. Heat transfer of the tubes D1, E1 and F1 in downward foam flow, β=0.996

The most important process – liquid drainage from foam – must be taken into account during analysis of the tube bundle heat transfer to statically stable foam flow. Gravity and capillary forces influence liquid drainage from foam flow by Plateau channels system. 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 influence on liquid flow from the foam bubble walls to Plateau channels is insignificant and mainly is predetermined by forces of surface tension. During drainage process geometric characteristics of foam bubbles are changing: walls of bubbles and Plateau channels are thinning; volumetric gas fraction is increasing. The gravity forces act along the upward and downward foam flow. While foam flow makes turn the gravity forces act across the flow and liquid drains down from the upper channel wall and the real void fraction increases here as well. After the turn, the real void fraction of foam is less (foam is wetter) on the left side of the cross–section (D tubes on Fig. 2). The foam 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.

The comparison of the D1, E1 and F1 tubes' heat transfer intensity to downward foam flow at the volumetric void fraction β=0.996 after turning is shown in Fig. 4.

Changing gas Reynolds number for foam flow from 190 to 440, heat transfer intensity of the tube D1 increases by 2.4 times, that of the tube E1 – 2.6 times, and that of the tube F1 – 2.4 times, for β=0.996 (Fig. 4). The heat transfer intensity of the tubes E1 and F1 is almost the same until $\overline{Re}_g$=375. The heat transfer of the side tube D1 is in average twice more than of that of the tubes E1 and F1 for the whole interval of $\overline{Re}_g$, for β=0.996.

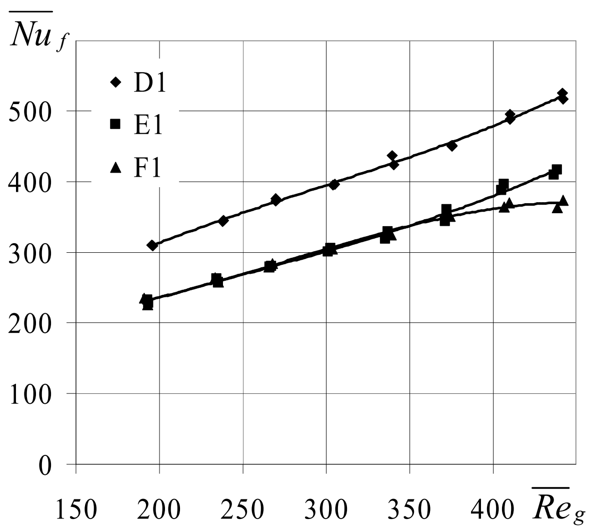

Fig. 5. Heat transfer of the tubes D1, E1 and F1 in downward foam flow, β=0.998

The comparison of the D1, E1 and F1 tubes' heat transfer intensity to downward foam flow at the volumetric void fraction β=0.998 is shown in Fig. 5. When $\overline{Re}_g$ increases from 190 to 440, the heat transfer of the tube D1 increases by 1.7 times, the heat transfer of tube E1 increases by 1.8 times and that of the tube F1 – 1.6 times, for β=0.998. The heat transfer of the tube D1 is better in whole interval of $\overline{Re}_g$ and is in average 1.3 times more than of that of the tubes E1 and F1, for β=0.998.

When $\overline{Re}_g$ =440, the heat transfer intensity of the tube D1 to the wettest foam flow (β=0.996) is by 2.4 times more in comparison with the heat transfer of the same tube to the driest foam flow (β=0.998). In the case of the tube F1, the heat transfer to the wettest foam flow (β=0.996) is by 1.5 times more than that to the driest foam flow (β=0.996).

The comparison of the D3, E3 and F3 tubes' heat transfer intensity to downward foam flow at the volumetric void fraction β=0.996 after turning is shown in Fig. 6. If the heat transfer intensity ($\overline{Nu}_f$) of the first tube D1 to foam flow (β=0.996) varies from 532 to 1270 (2.4 times), for $\overline{Re}_g$ =190÷440; the heat transfer of tube D3 increases less – $\overline{Nu}_f$ varies from 310 to 521 (about 1.7 times).

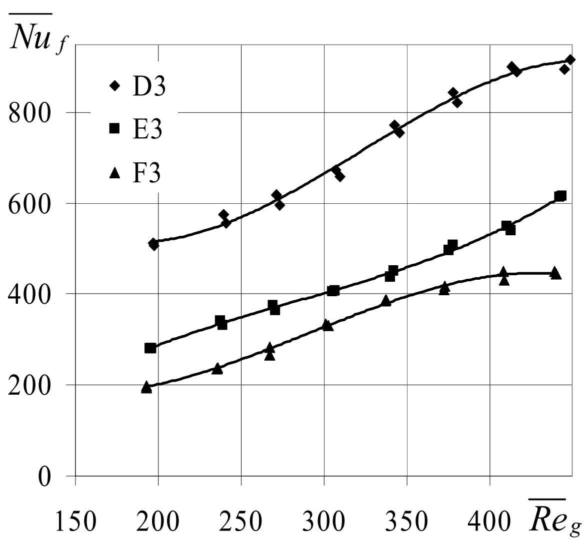

Fig. 6. Heat transfer of the tubes D3, E3 and F3 in downward foam flow, β=0.996

The heat transfer of the tube D3 is better in average in 66% than the heat transfer intensity of the tube E3 and the heat transfer of the tube E3 is better in average in 30% than the heat transfer intensity of the tube F3, for β=0.996 (Fig. 6). It is different in comparison with the case of the first tubes (Fig. 4). While foam flow passing the obstacle – tube bundle, the tubes change the moving direction of the foam bubbles and distribution of the foam real void fraction in the experimental channel cross–section become more gradual. It is more obvious in the case with driest (β=0.998) foam flow (Fig. 7).

In the case of the one–phase flow the heat transfer intensity of the frontal tubes are equal to about 60% of the third tubes heat transfer intensity [7] and the velocity distribution in cross–section is the main factor which influences the different heat transfer intensity of the middle and side tubes. So, it is different with foam flow, because the liquid drainage, changing foam structure, and the collapse of foam bubbles are acting together on the tube's heat transfer process.

Experimental results of heat transfer of the staggered tube bundle to directed statically stable foam flow after turning 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 staggered tube bundle in downward 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 entire middle line in the bundle k=142, n=1091, m=224.31–224.25 β.

On average, for whole staggered tube bundle c=134, n=1025, m=223.25–223.2 β.

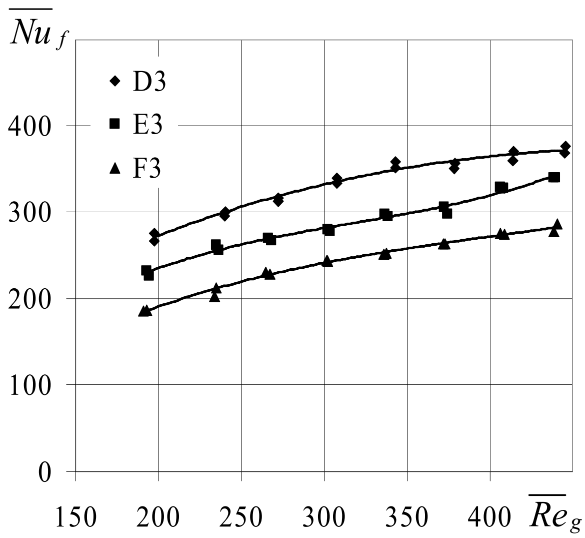

Fig. 7. Heat transfer of the tubes D3, E3 and F3 in downward foam flow, β=0.998

5 Conclusions

Heat transfer of the staggered tube bundle to downward statically stable foam flow after turning was investigated experimentally.

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

The experimental investigation showed that the heat transfer of the frontal tubes to downward foam flow is better than of that of the third tubes. It is different in comparison with one–phase fluid flow case also.

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

The experimental investigations of different configuration tube bundles heat transfer to vertical flow of the statically stable foam are in prospect.

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.

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