Study of In–line Tube Bundle Heat Transfer to Downward Foam Flow

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Abstract: An experimental investigation of heat transfer from the in–line tube bundle to the two–phase foam flow was performed. Statically stable gas–liquid foam flow was used as a coolant. Investigation was performed on the experimental laboratory set–up consisting of the foam generator, an experimental channel and tube bundle. Regularities of the heat transfer of the tube bundle to the downward foam flow under the 180˚ degree turn were analysed in the work. Heat transfer character of frontal and further tubes to downward foam flow is different in comparison with the one–phase coolant flow. After the turn, local void fraction of the foam is less on the inner side of the foam flow. Therefore heat transfer intensity of the inner side–line tubes is higher than for other tubes of the bundle. Results of the investigation were generalized by criterion equation, which can be used for calculation and design of the statically stable gas–liquid foam heat exchangers with the in–line tube bundles.

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

1 Introduction

Smaller coolant mass flow rate, relatively large heat transfer rate, low energy consumption required for coolant delivery to heat transfer place may be achieved by usage of two–phase gas–liquid foam flow as a coolant [1, 2]. Characteristics of one gas–liquid foam type – statically stable foam – showed its perfect availability for this purpose [1].

Tube bundles of different types and geometry are used in heat exchangers and single–phase coolants such as water or air usually are used for heat and mass transfer process in industrial heat exchangers. Therefore heat transfer of different tube bundles to one–phase fluids was investigated enough [3, 4], but there is no sufficient data concerning heated surfaces heat transfer to the statically stable foam flow yet. Therefore problems arise when foam systems and heat exchangers are designed. In our previous works heat transfer of alone circular surface – tube, and such tubes line to upward statically stable foam flow was investigated [1]. Next experimental series with staggered (spacing between centres of the tubes across the bundle was s_1=0.07 m and spacing along the bundle was s_2=0.0175 m) and in–line (s_1= s_2=0.03 m) tube bundles in upward and downward foam flow were followed [5, 6, 7].

Presently the heat transfer of the in–line tube bundle (s_1=0.03 m, s_2=0.06 m) to vertical downward after 180° degree turning foam flow was investigated experimentally. It was determined a dependency of tubes' heat transfer intensity on flow velocity and foam volumetric void fraction. Apart of that, influence of tube position in the bundle on heat transfer intensity was investigated also. The structure of foam is one of the factors which influences on the heat transfer intensity of the tubes to the foam flow. When foam flow is passing through the bundle of tubes foam bubbles are intermixed, some bubbles collapsed or divided into smaller bubbles. In order to observe visually the vertical downward foam flow crossing the tube bundles, the walls of the experimental channel were made from transparent material.

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

2 Experimental Set–up

The in–line tube bundle consisted of five vertical rows with three tubes in each. Spacing between centres of the tubes across the experimental channel was s_1=0.03 m and spacing along the channel was s_2=0.06 m. A schematic view of the experimental channel with tube bundles is shown in the Fig. 1. External diameter of all the tubes was equal to 0.02 m. An electrically heated tube – calorimeter had an external diameter equal to 0.02 m also. During the experiments calorimeter was placed instead of one tube of the bundle. An electric current value of heated tube was measured by an ammeter and voltage by a voltmeter. Temperature of the calorimeter surface was measured by eight calibrated thermocouples: six of them were placed around the central part of the tube and two of them were placed in both sides of the tube at a distance of 50 mm from the central part. Temperature of the foam flow was measured by two calibrated thermocouples: one in front of the bundle and one behind it.

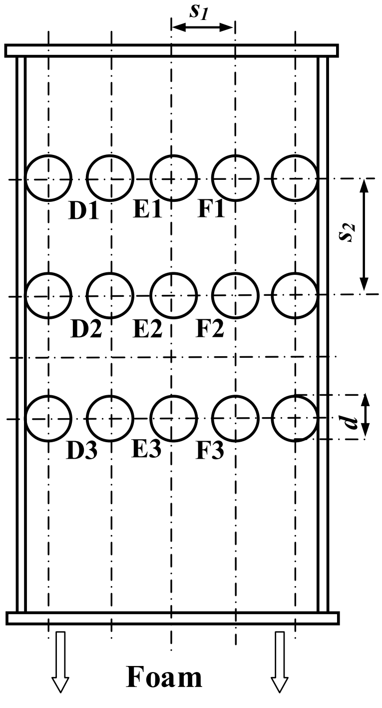

Fig. 1. In–line tube bundle in foam flow

The experimental set–up consisted of the following main parts: experimental 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 [5, 6, 7].

Cross section of the experimental channel had dimensions 0.14 x 0.14 m; height of it was 1.8 m. Radius of the channel turning (R) was equal to 0.17 m.

Statically stable foam flow was used for an experimental investigation. This type of gas–liquid foam was generated from water solution of detergents. Concentration of detergents was kept constant and was equal to 0.5%. Foam flow was produced during gas and liquid contact on the riddle, which was installed at the bottom of the experimental channel. Liquid was delivered from the reservoir to the riddle from the upper side; gas was supplied to the riddle from below.

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 investigation a relationship was obtained between Nusselt number (heat transfer intensity) Nu_f from one side and foam flow volumetric void fraction β and gas flow Reynolds number Re_g from the other side:

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

Nusselt number was computed by formula

$$Nu_f = \frac{hd}{\lambda_f}, \tag{2}$$

where λ_f is the thermal conductivity of the statically stable foam flow, W/(mK), computed by the equation

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

An average heat transfer coefficient was calculated as

$$h = \frac{q_w}{\Delta T}. \tag{4}$$

Gas Reynolds number of foam flow was computed by formula

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

Foam flow volumetric void fraction can be expressed by the equation

$$\beta = \frac{G_g}{G_g + G_l}. \qquad (6)$$

It is known [5] that there are four main regimes of the statically stable foam flow in the vertical channel of rectangular cross section:
– 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.

Experiments were performed within Reynolds number diapason for gas (Re_g): 190÷440 (laminar flow regime) and foam volumetric void fraction (β): 0.996÷0.998. Foam flow gas velocity was changed from 0.14 to 0.32 m/s. Heat transfer coefficient (h) varied from 200 to 2000 W/(m^{2}K).

4 Results

Statically stably foam initially moved vertically upward, then made 180° degree and R=0.17 m radius turning and moved downward crossing the in–line tube bundle.

The main three parameters of foam flow influence on heat transfer intensity of different tubes of the bundle: foam structure, distribution of local flow velocity and distribution of local foam void fraction across and along the experimental channel.

Foam structure can be characterized by diameter of the foam bubble (d_b). This parameter depends not only on the foam volumetric void fraction (β), but on the foam flow generation conditions as well. Larger size bubbles foam flow is generated if the feeding gas rate G_g and accordingly the Re_g is low.

Diameter of the foam bubbles is d_b=15±2 mm for the volumetric void fraction of foam β=0.998 and Re_g=190. Diameter of foam bubbles for drier (β=0.997) foam flow is equal to 10±1.5 mm and for the driest (β=0.996) foam flow d_b=5±1 mm at the same conditions (Re_g=190). Increase of G_g influences on generation of foam flow with smaller bubbles (size of the bubble is about 1.5÷2 times lower), therefore foam flow becomes more homogenous and heat transfer process intensifies.

Liquid drainage process influences on the distribution of the foam local void fraction and accordingly on heat transfer intensity of the tubes. Liquid drainage from foam phenomena depends on gravity and capillary [8, 9]. 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 electrostatic and molecular forces on drainage is insignificant [8]. Gravity forces act along the upward and downward foam flow. While foam flow makes a turn the gravity forces act across and along the foam flow. Liquid drains down from the upper channel wall and local void fraction increases (foam becomes drier) here as well. After the turn, local void fraction of foam is less (foam is wetter) on the inner – left side of the cross–section (tubes D, Fig. 1). The flow velocity distribution in cross section of the channel transforms after turn too.

Heat transfer intensity of the first tubes of the in–line bundle to downward wettest (β=0.996) foam flow is shown in Fig. 2. Increasing foam flow gas Reynolds number (Re_g) from 190 to 440, heat transfer intensity (Nu_f) of the tube D1 increases twice (from 750 to 1486), that of the tube E1 increases by 2.6 times (from 406 to 1058), and that of the tube F1 increases by 3 times (from 256 to 757) for foam volumetric void fraction β=0.996.

When Re_g=440, the heat transfer intensity of the tube D1 is by 1.4 times more than that of the tube E1 and twice more than that of the tube F1.

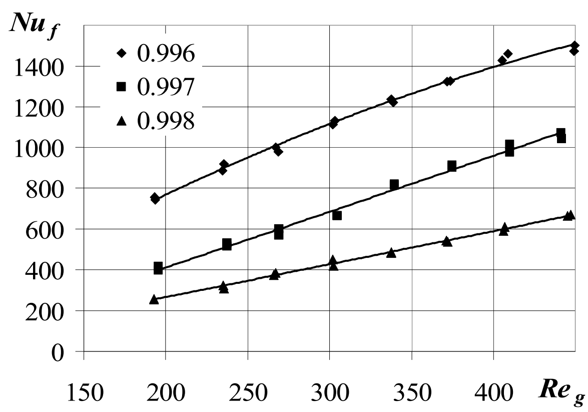

Fig. 2. Heat transfer of the first tubes D1, E1 and F1 to downward foam flow, β=0.996

Heat transfer intensity of the first tubes of the in–line bundle to downward driest (β=0.998) foam flow is shown in Fig. 3. Heat transfer intensity (Nu_f) of the tube D1 increases by 1.8 times (from 431 to 768), that of the tube E1 increases twice (from 310 to 614), and that of the tube F1 increases by 1.9 times (from 253 to 475) for Re_g=190÷440 and β=0.998.

Heat transfer intensity of the tube D1 is twice better to the wettest (β=0.996) foam flow in comparison with the heat transfer of the same tube to the driest (β=0.998) foam flow. The same for the tube F1 is by 1.4 times better to the wettest (β=0.996) foam flow in comparison with the heat transfer of the same tube to the driest (β=0.998) foam flow.

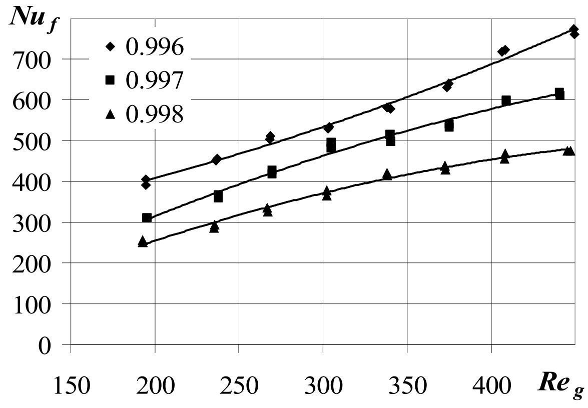

Fig. 3. Heat transfer of the first tubes D1, E1 and F1 to downward foam flow, β=0.998

Increasing foam flow gas Reynolds number (Re_g) from 190 to 440, heat transfer intensity (Nu_f) of the tube D3 increases twice (from 778 to 1566), that of the tube E3 increases by 2.3 times (from 468 to 1061), and that of the tube F3 increases by 2.7 times (from 252 to 687) for foam volumetric void fraction β=0.996 (Fig. 4).

In one–phase flow case heat transfer intensity of the frontal tubes is equal to about 60% of the third tubes heat transfer intensity [3]. It is different in two–phase foam flow case. Heat transfer intensity of the tube D3 is only by 3% better than that of the tube D1 for Re_g=190÷440 and β=0.996. Heat transfer intensity of the tube E3 is by 3% better than that of the tube E1 and heat transfer intensity of the tube F3 is by 1% better than that of the tube F1 for the same conditions.

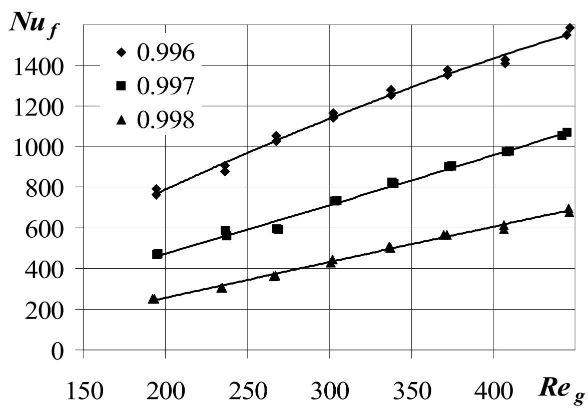

Fig. 4. Heat transfer of the third tubes D3, E3 and F3 to downward foam flow, β=0.996

Heat transfer intensity of the third tubes of the in–line bundle to downward driest (β=0.998) foam flow is shown in Fig. 5. Heat transfer intensity of the tube D3 increases by 1.6 times (from 415 to 668), that of the tube E3 increases by 1.8 times (from 320 to 584), and that of the tube F3 increases by 1.8 times (from 252 to 462) for Re_g=190÷440 and β=0.998.

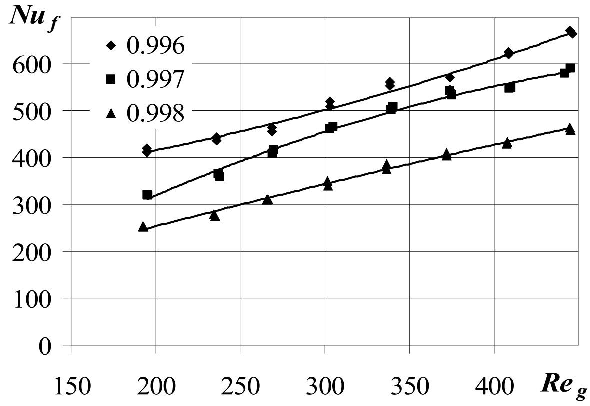

Fig. 5. Heat transfer of the third tubes D3, E3 and F3 to downward foam flow, β=0.998

Heat transfer intensity of the first tubes is better than that of the third tubes to the driest (β=0.998) foam flow. Heat transfer intensity of the tube D1 is by 8% better than that of the tube D3 for Re_g=190÷440 and β=0.998. Heat transfer intensity of the tube E1 is by 3% better than that of the tube E3 and heat transfer intensity of the tube F1 is by 6% better than that of the tube F3 for the same conditions.

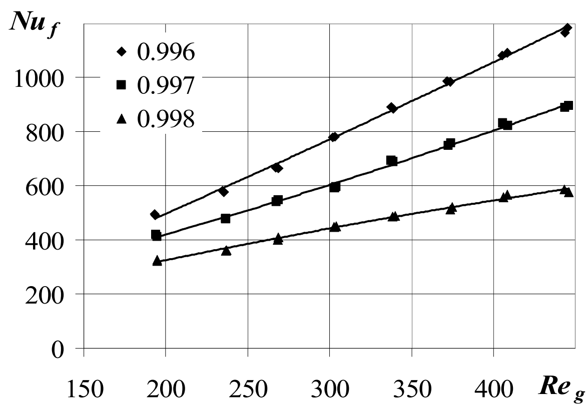

Fig. 6. Average heat transfer of the tubes of the in line bundle to downward vertical foam flow: β=0.996, 0.997 and 0.998

An average heat transfer rate was calculated in order to analyze the experimental results of in–line tube bundle. An average heat transfer intensity of the tubes of the in–line bundle to downward vertical foam flow is shown in Fig. 6. Changing Re_g from 190 to 440, an average heat transfer intensity of the tubes increases by 2.4 times for β=0.996; by 2.1 times for β=0.997, and by 1.8 times for β=0.998.

Experimental results of investigation of heat transfer of the in–line tube bundle to downward after 180° turning statically stable foam flow were generalized by criterion equation using dependence between Nusselt number Nu_f and gas Reynolds Re_g number. This dependence within the interval 190<Re_g<440 for the in–line tube bundle (s_1=0.03 and s_2=0.06 m) in downward foam flow with the volumetric void fraction β=0.996, 0.997, and 0.998 can be expressed as follows:

$$Nu_f = c\beta^n Re_g^m \, . \tag{7}$$

On average, for entire left (D) side–line of the in–line tube bundle in the downward foam flow: c=19.8, n=347, m=130(1.003–β). On average, for entire middle (E) line of the in–line tube bundle in the downward foam flow: c=50.8, n=1230, m=249.5(1.001–β). On average, for entire right (F) side–line of the in–line tube bundle in the downward foam flow: c=47, n=1422, m=280.2(1.0001–β) and on average, for the whole in–line tube bundle (s_1=0.03 and s_2=0.06 m) in the downward foam flow c=22.4, n=675, m=167.8(1.002–β).

5 Conclusions

Heat transfer from in–line tube bundle to laminar downward after 180° degree turning foam flow was investigated experimentally.

Three main parameters of foam flow influence on heat transfer intensity of different tubes of the tube bundles: foam structure, distribution of local flow velocity and distribution of local foam void fraction across and along the experimental channel.

Heat transfer intensity of the left (D) side–line tubes is higher than that of the middle (E) and right (F) side–line tubes.

Heat transfer intensity of the third tubes is better than that of the first tubes to the wettest (β=0.996) foam flow. It is different with driest (β=0.998) foam flow. Heat transfer intensity of the first tubes is better than that of the third tubes in that case.

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

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;
Re	Reynolds number;
T	temperature, K;
h	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;
w	wall of heated tube.

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