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# INFLUENCE OF TUBE BUNDLE GEOMETRY ON HEAT TRANSFER TO FOAM FLOW

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## ABSTRACT

Usage of two–phase gas–liquid foam flow as a coolant allows achieving relatively large heat transfer intensity with smaller coolant mass flow rate. However number of foam peculiarities complicates an application of the analytical methods for heat transfer investigation. Presently an experimental method of investigation was selected as the most suitable. Due to the fact that tube bundles of different geometry my be used in foam apparatus, the experimental investigation of heat transfer of the in–line tube bundle with different spacing between tubes to vertical foam flow was performed. Spacing among the centres of the tubes across the in–line tube bundle was 0.03 m and spacing along the bundle was 0.03 m. In other case the spacing along the bundle was 0.06 m. Results of investigation showed that an effect of "shadow" is slight and heat transfer is higher for the tubes of the in–line tube bundle with more spacing between the tubes' centres along the bundle.

## INTRODUCTION

Typical heat exchangers usually consist of several vertical parts in which coolant changes its direction from vertical upward to vertical downward and vice versa. The turning of foam flow from the upward to downward flow direction influences on the distribution of the foam volumetric void fraction and on the flow velocity in the cross–section of the foam channel as well. Consequently the investigation of foam flow turning influence on heat transfer peculiarities must be performed for its later application in heat exchangers.

Foam is two–phase flow and structure of it changes while it passes obstacle: bubbles are changing their size and liquid drainage is going on. Continuous drainage of liquid from foam [1, 2], diffusive gas transfer [1] and disintegration of inter–bubble films [3, 4] destruct the foam flow at the same time. Thus investigation of heat transfer process in foam flow is rather complex and presently an experimental method of investigation was selected in our work as the most suitable.

Our previous works were devoted to the investigation of heat transfer of alone cylindrical tubes to upward statically stable foam flow [5]. Next experimental series were performed for the tube line placed in upward foam flow [5].

Tube bundles of different types and geometry are used in heat exchangers. Therefore we performed an investigation of staggered [6, 7] and in–line [8] tube bundles heat transfer to vertical upward and downward foam flow. It was determined the dependence of heat transfer intensity on the flow parameters: flow velocity, volumetric void fraction of foam and liquid drainage from foam. Apart of this, influence of tube position in the bundle to heat transfer intensity was investigated also.

## NOMENCLATURE

| | | |
|---|---|---|
| $A$ | [m$^2$] | Cross–section area of the experimental channel |
| $c$ | [–] | Coefficient |
| $d$ | [m] | External diameter of tube |
| $d_b$ | [m] | Diameter of foam bubble |
| $G$ | [m$^3$/s] | Volumetric flow rate |
| $h$ | [W/m$^2$K] | Average heat transfer coefficient |
| $I$ | [A] | Electric current value |
| $m$ | [–] | Coefficient |
| $n$ | [–] | Coefficient |
| $Nu$ | [–] | Nusselt number |
| $q$ | [W/m$^2$] | Heat flux density |
| $R$ | [m] | Radius of the channel turn |
| $Re$ | [–] | Reynolds number |
| $s$ | [m] | Spacing between the centres of the tubes |
| $T$ | [K] | Temperature |
| $U$ | [V] | Voltage drop |

Special characters

| | | |
|---|---|---|
| $\beta$ | [-] | Volumetric void fraction |
| $\lambda$ | [W/mK] | Thermal conductivity |
| $\nu$ | [m$^2$/s] | Kinematic viscosity |

Subscripts

| | |
|---|---|
| $f$ | Foam |
| $g$ | Gas |
| $l$ | Liquid |
| $w$ | Wall of heated tube |

Main task of this work was to investigate experimentally and compare heat transfer intensity of two in–line tube bundles with different geometry. The results of our experimental investigation are presented and discussed in this paper.

## EXPERIMENTAL SET–UP AND METHODOLOGY

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 [6, 7, 8]. The whole experimental channel was made of glass in order to observe visually foam flow structure and size of foam bubbles. 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%.

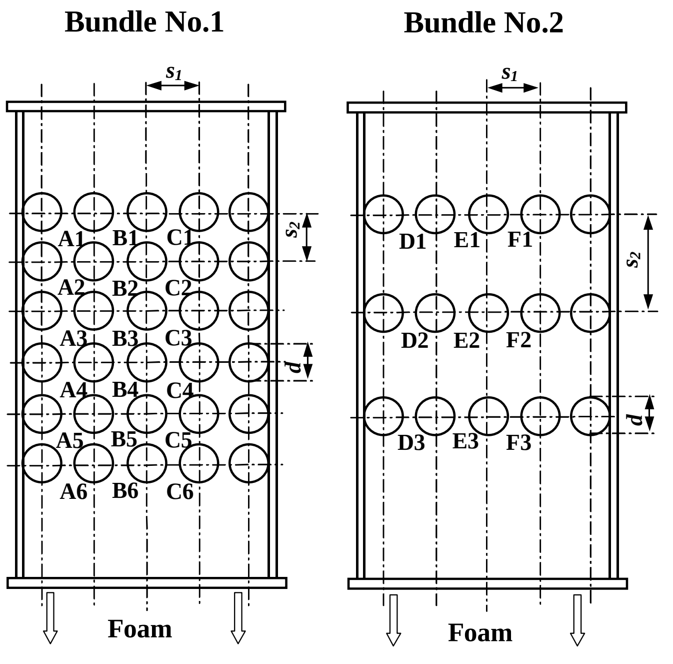


**Figure 1** In–line tube bundles in foam flow

Two in–line tube bundles were used during experimental investigation. A schematic view of the experimental channel with tube bundles is shown in the Figure 1. The in–line tube bundle No. 1 consisted of five vertical rows with six tubes in each. Spacing between centres of the tubes was $s_1=s_2=0.03$ m. The in–line tube bundle No. 2 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. 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.

During the experimental investigation a relationship was obtained between an average heat transfer coefficient $h$ from one side and foam flow volumetric void fraction $\beta$ and gas flow Reynolds number $Re_g$ from the other side:

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

Nusselt number was computed by formula

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

Where $\lambda_f$ is the thermal conductivity of the statically stable foam flow, W/(mK), computed by the equation

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

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

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

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 ($\beta$): 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 2200 W/(m$^2$K).

## RESULTS

The object of experimental investigation was an influence of tube bundle geometry on heat transfer intensity to foam flow. Initially an experiments with in–line tube bundle No. 1 were performed; then the tube bundle No. 2 was placed instead of the previous bundle and experiments followed.

Heat transfer intensity of the bundles' third, fourth and further tubes to one–phase flow is almost the same [9, 10]. In two–phase foam flow case heat transfer of the third and further tubes of the bundles varies slightly and differences are insignificant [6, 7, 8]. Therefore heat transfer intensity of the third tubes to foam flow was compared. Heat transfer intensity of the A3, B3, C3 and D3, E3, F3 tubes of the in–line bundles to downward after turning foam flow at the volumetric void fraction $\beta$=0.996 is shown in Figure 2.

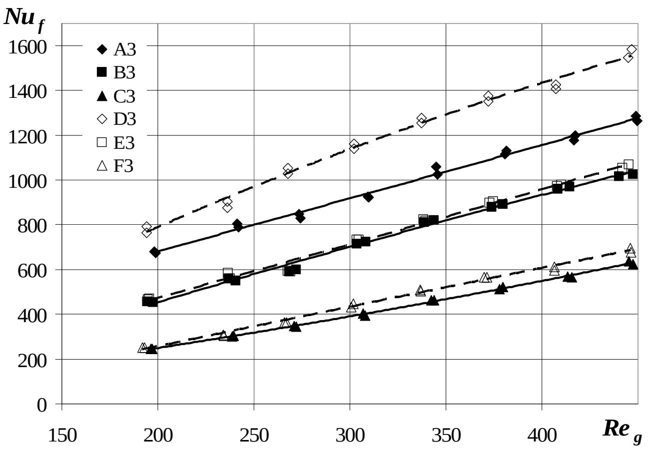


**Figure 2** Heat transfer of the tubes A3, B3, C3 and D3, E3, F3 to downward foam flow, $\beta$=0.996

Three main parameters of foam flow influence on heat transfer intensity of different tubes of the bundles: foam structure, distribution of local flow velocity and distribution of local foam void fraction across and along the experimental channel. 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 [1, 2]. 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 [1]. 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 A and D, Figure 1). The flow velocity distribution in cross section of the channel transforms after turn too.

Foam structure can be characterized by diameter of the foam bubble, but this parameter depends not only on the foam volumetric void fraction, 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 $\beta$=0.998 and $Re_g$=190. Diameter of foam bubbles for drier ($\beta$=0.997) foam flow is equal to 10±1.5 mm and for the driest ($\beta$=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.

Heat transfer intensity of the third tubes of the in–line bundle No. 2 is higher than that of the bundle No. 1 (Figure 2). Increasing foam flow gas Reynolds number ($Re_g$) from 190 to 440, heat transfer intensity ($Nu_f$) of the tube A3 increases by 1.9 times (from 677 to 1275), by 2.2 times (from 454 to 1020) of the tube B3, and by 2.6 times (from 246 to 631) of the tube C3 for foam volumetric void fraction $\beta$=0.996. Heat transfer intensity of the tubes D3 for the same $Re_g$ increases twice (from 778 to 1566), by 2.3 times (from 468 to 1061) of the tube E3, and by 2.7 (from 252 to 687) of the tube F3 for $\beta$=0.996. When $Re_g$=440 heat transfer intensity of the A3 tube is twice higher than that of the tube C3 and heat transfer intensity of the D3 tube is higher than that of the tube F3 by 2.3 times.

Heat transfer intensity of the tube D3 is higher than that of the tube A3 on average by 20%, the heat transfer of the tube E3 is higher than that of the tube B3 on average by 4%, and the heat transfer of the tube F3 is higher than that of the tube C3 on average by 7% for $\beta$=0.996 and $Re_g$=190÷440.

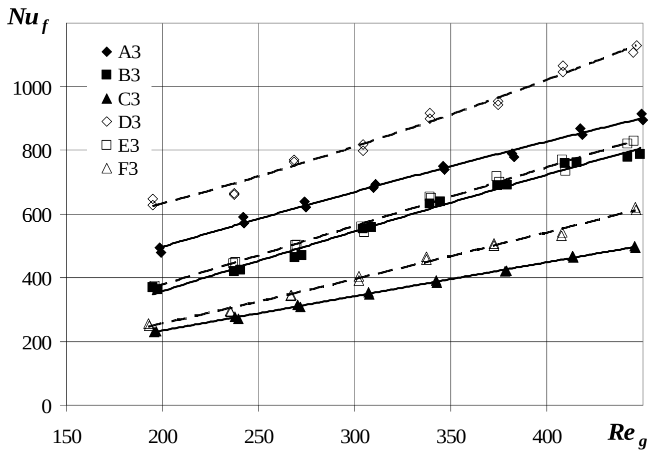


**Figure 3** Heat transfer of the tubes A3, B3, C3 and D3, E3, F3 to downward foam flow, $\beta$=0.997

Heat transfer intensity of the A3, B3, C3 and D3, E3, F3 tubes of the in–line bundles to downward after turning foam flow at the volumetric void fraction $\beta$=0.997 is shown in Figure 3. With increase of $Re_g$ from 190 to 440, heat transfer intensity of the tube A3 increases by 1.9 times (from 488 to 904), by 2.1 times (from 366 to 783) of the tube B3, by 2.1 times (from 231 to 496) of the tube C3, by 1.8 times (from 637 to 1118) of the tube D3, by 2.2 times (from 373 to 825) of the tube E3, and by 2.4 (from 252 to 617) of the tube F3 for $\beta$=0.997. When $Re_g$=440 heat transfer intensity of the A3 tube is higher than that of the tube C3 and heat transfer intensity of the D3 tube is higher than that of the tube F3 by 1.8 times.

Heat transfer intensity of the tube D3 is higher than that of the tube A3 on average by 26%, the heat transfer of the tube E3 is higher than that of the tube B3 on average by 4%, and the heat transfer of the tube F3 is higher than that of the tube C3 on average by 20% for $\beta$=0.997 and $Re_g$=190÷440.

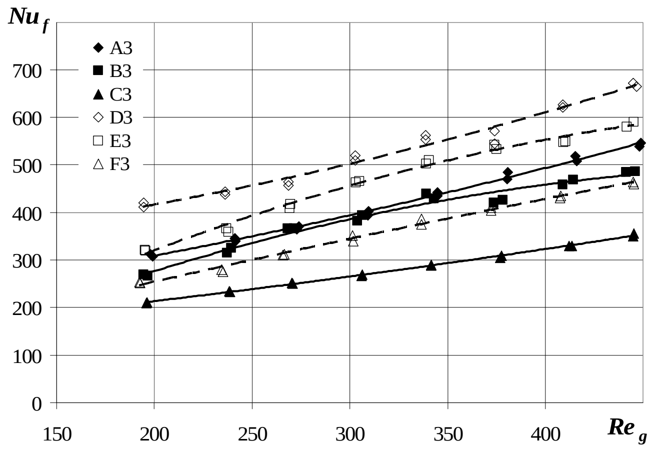


**Figure 4** Heat transfer of the tubes A3, B3, C3 and D3, E3, F3 to downward foam flow, $\beta$=0.998

Heat transfer intensity of the A3, B3, C3 and D3, E3, F3 tubes of the in–line bundles to downward after turning foam flow at the volumetric void fraction $\beta$=0.998 is shown in Figure 4. With increasing of $Re_g$ from 190 to 440, heat transfer intensity of the tube A3 increases by 1.7 times (from 310 to 542), by 1.8 times (from 268 to 486) of the tube B3, by 1.7 times (from 210 to 353) of the tube C3, by 1.6 times (from 415 to 668) of the tube D3, by 1.8 times (from 320 to 585) of the tube E3, and by 1.8 (from 252 to 462) of the tube F3 for $\beta$=0.998. When $Re_g$=440 heat transfer intensity of the A3 tube is higher than that of the tube C3 and heat transfer intensity of the D3 tube is higher than that of the tube F3 by 1.5 times.

Heat transfer intensity of the tube D3 is higher than that of the tube A3 on average by 27%, the heat transfer of the tube E3 is higher than that of the tube B3 on average by 20%, and the heat transfer of the tube F3 is higher than that of the tube C3 on average by 27% for $\beta$=0.998 and $Re_g$=190÷440.

An average heat transfer rate was calculated in order to analyze and compare the experimental results of different in–line tube bundles. An average heat transfer intensity of the tubes of the in–line bundle No. 1 and No. 2 to downward after turning foam flow is shown in Figure 5.

The effect of "shadow" takes place in the case of the in–line bundle No. 1 therefore an average heat transfer intensity of the tubes of the in–line bundle No. 2 is higher than that of the tubes of the in–line bundle No. 1 for the whole interval of the $Re_g$ ($Re_g$=190÷440). Changing $Re_g$ from 190 to 440, an average heat transfer intensity of the tubes of the in–line bundle No. 1 to downward foam flow increases by 2.1 times for $\beta$=0.996; twice for $\beta$=0.997, and by 1.7 times for $\beta$=0.998; and that for the tubes of the in–line bundle No. 2 is by 2.4 times for $\beta$=0.996; by 2.1 times for $\beta$=0.997, and by 1.8 times for $\beta$=0.998.

An average heat transfer intensity of the tubes of in–line bundle No. 2 is higher than that of the tubes of the in–line bundle No. 1 on average by 21% for $\beta$=0.996, by 23% for $\beta$ =0.997 and by 27% for $\beta$=0.998 to downward foam flow for whole interval of $Re_g$ ($Re_g$=190÷440).

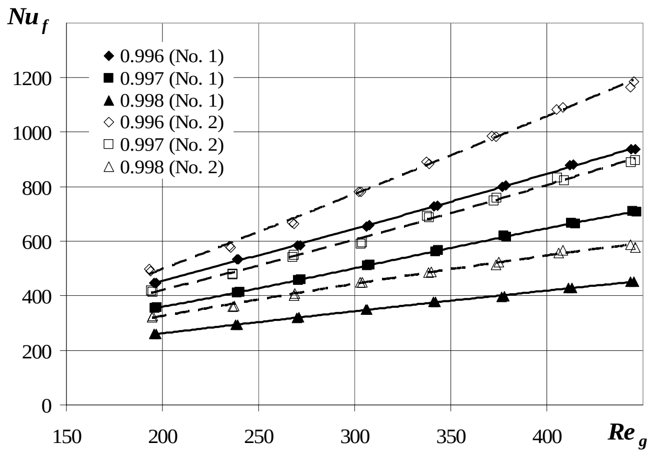


**Figure 5** Average heat transfer of the tubes of the in line bundle No. 1 and No. 2 to downward foam flow: $\beta$=0.996, 0.997 and 0.998

Experimental results of investigation of heat transfer of the in–line tube bundles 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 in downward foam flow with the volumetric void fraction $\beta$=0.996, 0.997, and 0.998 can be expressed as follows:

$$Nu_f = c\beta^n \mathrm{Re}_g^m \qquad (7)$$

On average, for the whole in–line tube bundle No. 1 ($s_1$=$s_2$=0.03 m) in the downward foam flow $c$=12.7, $n$=334, $m$=114.6(1.004–$\beta$).

On average, for the whole in–line tube bundle No. 2 ($s_1$=0.03 and $s_2$=0.06 m) in the downward foam flow $c$=22.4, $n$=675, $m$=167.8(1.002–$\beta$).

## CONCLUSIONS

Heat transfer of two in–line tube bundles with different geometry to vertical 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.

Liquid drainage process significantly transforms the "cross–sectional" distribution of local void fraction of the downward foam flow and acts on the heat transfer intensity of the tubes. Therefore heat transfer intensity of the left (A and D) side–line tubes is higher than that of the middle (B and E) and right (C and F) side–line tubes.

The effect of "shadow" is slight and heat transfer is higher for the tubes of the in–line tube bundle with more spacing between the tube centres along the bundle.

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.

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