Peculiarities of heat transfer from in-line tube bundles to upward aqueous foam flow

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Abstract

Four in-line tube bundles with different geometry were investigated for establishing their performance in terms of heat transfer enhancement. Two-phase aqueous foam was used as a coolant. Such coolant was considered, because our previous research showed that large heat transfer intensity may be reached even at small mass flow rate of the foam. Spacing among the centres of the tubes across the first in-line tube bundle was 0.03 m and spacing along the bundle was 0.03 m. In the second case spacing among the centres of the tubes across the bundle was 0.03 m; spacing along the bundle was 0.06 m. In the third case spacing was accordingly 0.06 and 0.03 and in the last case spacing was accordingly 0.06 m and 0.06 m. During an experimental investigation it was determined a dependence of heat transfer intensity on flow parameters. The investigation of heat transfer from the bundle to upward vertical foam flow was provided for three different values of foam volumetric void fractions β=0.996÷0.998. The velocity of the foam flow was changed from 0.14 to 0.30 m/s. The heat transfer coefficient varied from 200 to 2000 W/(m^{2}K) for the above mentioned foam flow parameters.

Keywords: in-line tube bundle, aqueous foam, upward flow, heat transfer.

1 Introduction

Heat exchangers with different type and geometry of tube bundles are usually used in industry. Two-phase foam coolant has a number of advantages in comparison with single-phase coolant. For example, it requires smaller coolant flow rates, and lower energy consumption for delivering the coolant to the region where heat transfer process takes place. Two-phase foam flow can signally reduce material expenditures. A totally new, modern and economic heat exchanger with simple and safe exploitation using two-phase foam flow in heat transfer could be created. It would be compact, light and with greater intensity of heat transfer.

Aqueous foam flow has number of peculiarities which always exists during heat transfer process: drainage of liquid from foam [1, 2], diffusive transfer of gas between bubbles [3], division and collapse of foam bubbles [4]. Those phenomena are closely related with each other and make an application of analytical study of the heat transfer processes in the foam flow extremely complicated. So, an experimental approach has been selected as the most suitable for our present investigation.

Heat transfer of different tube bundles to one-phase fluid was investigated enough [5, 6], but practically there are no data about tube bundles heat transfer to foam flow. This work is an extension of our previous works [7–9]. Heat transfer from four in-line tube bundles with different types of geometries to the vertical upward statically stable foam flow was investigated experimentally. The influence of in-line tube bundles geometry was analyzed for average heat transfer intensity by increasing the distance between tubes centres of tube. Dependence of heat transfer intensity on flow parameters was determined. Objectives of this investigation are to determine an optimal geometry of in-line tube bundle which guarantees a maximal heat transfer to two-phase foam flow.

The results of investigation were treated analyzing relationships between Nusselt and Reynolds numbers and volumetric void fraction.

2 Experimental set-up

Investigation was performed on the experimental set-up consisting of experimental channel, foam generator, in-line tube bundle, gas and liquid flow meters, liquid storage reservoir, compressor, electric current transformer and stabilizer [7–9]. Cross section of the channel had dimensions (0.14 x 0.14) m^2; height of experimental channel was 1.8 m.

Statically stable foam (one type of aqueous foam) was used for experiments [7–9]. Foam-able liquid for the statically stable foam generation was produced from water solution with the detergents. Concentration of the detergents was kept constant at 0.5% in all experiments. Foam-able liquid was supplied from the reservoir onto the special perforated plate for the foam generation from all channel sides; gas was supplied to the perforated plate from the bottom. Air from the compressor via receiver was used as “gas” in all investigations. Foam flow was produced during gas and liquid contact on the perforated plate.

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The perforated plate was made from stainless steel plate with thickness of 2 mm. Holes were located in staggered order. Diameter of the holes was 1 mm; spacing among centres – 5 mm.

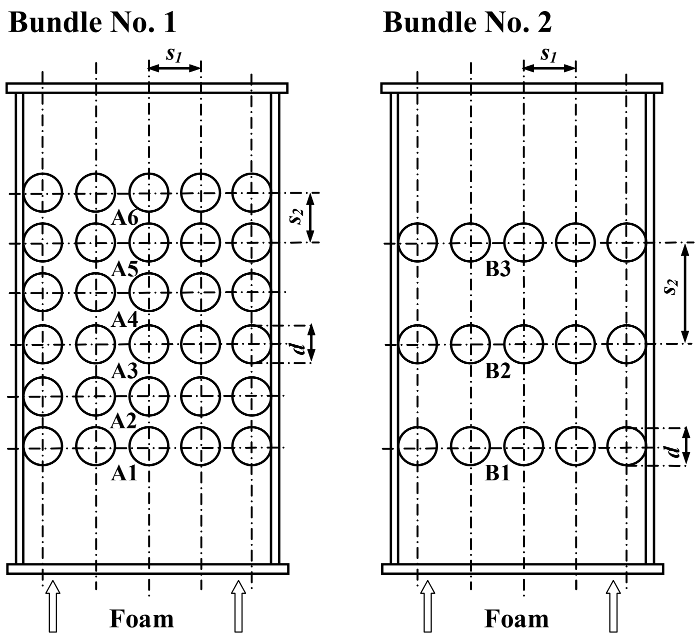

Figure 1: In-line tube bundles No. 1 and No. 2 in upward foam flow.

Four in-line tube bundles were used during experimental investigation. Schematic view of the experimental channel with tube bundles No. 1 and No. 2 is shown in the Fig. 1. The in-line tube bundle No. 1 consisted of five columns 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 (Fig. 1) consisted of five columns 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.

The in-line tube bundle No. 3 (Fig. 2) consisted of three columns with six tubes in each. Spacing between centres of the tubes across the experimental channel was s_1=0.06 m and spacing along the channel was s_2=0.03 m. The in-line tube bundle No. 4 (Fig. 2) consisted of three columns with three tubes in each. Spacing between centres of the tubes was s_1=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 calorimeter and two of them were placed in both sides of the calorimeter 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.

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

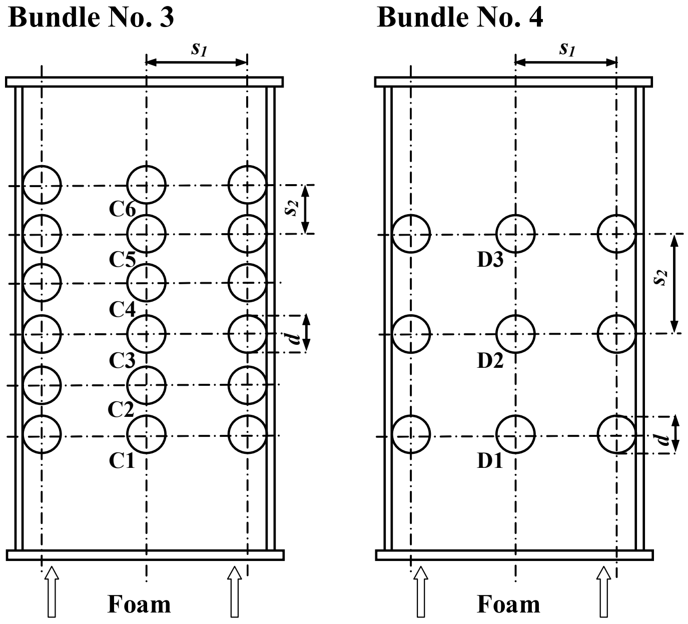

Figure 2: In-line tube bundles No. 3 and No. 4 in upward foam flow.

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

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

Nusselt number was computed by formula

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

Here λ_f is the thermal conductivity of the statically stable foam flow, W/(m·K), computed by the equation

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

An average heat transfer coefficient we calculated as

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

Gas Reynolds number of foam flow we computed by formula

$$\mathrm{Re}_g = \frac{G_g d}{A \nu_g}. \tag{5}$$

Foam flow volumetric void fraction we expressed by the equation

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

Experiments we performed within limits of Reynolds number diapason for gas (Re_g): 190–410 (laminar flow regime) and foam volumetric void fraction (β): 0.996–0.998. Gas velocity for foam flow was changed from 0.14 to 0.30 m/s.

3 Results

Heat transfer from the tube bundle No. 1 to the vertical upward foam flow was investigated initially [8, 9]. Then tube bundle No. 2 was installed in an experimental channel instead of the bundle No. 1 and so on.

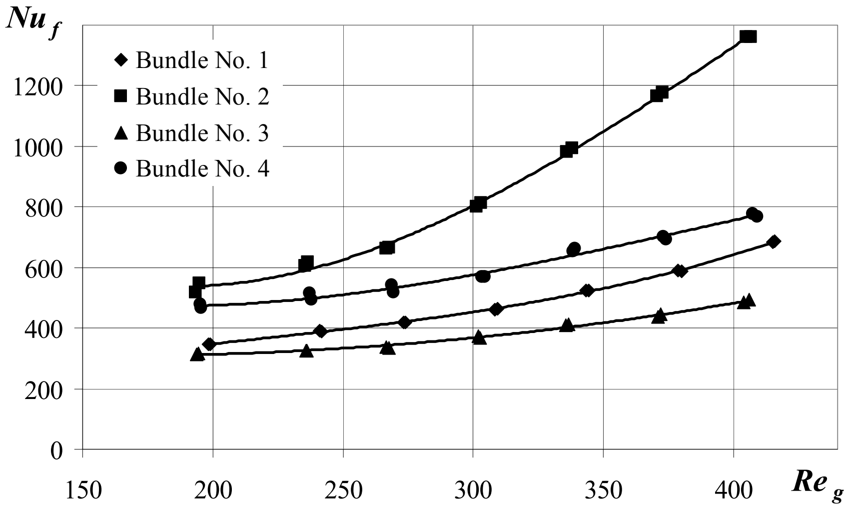

Figure 3: An average heat transfer intensity of the tubes from middle column of the in-line bundle No. 1, No. 2, No. 3 and No. 4 to upward foam flow, β=0.996.

Heat transfer intensity of the tubes located at the different places across and along the in-line tube bundle to vertical upward foam flow was investigated experimentally. In the foam flow case heat transfer intensity of the different tubes is under the influence of distribution of local flow velocity and local foam void fraction across and along the channel [8, 9]. The third factor which influences the heat transfer intensity of tubes is the foam structure.

An average heat transfer rate of tubes from middle column was calculated in order to compare the experimental results of in-line tube bundles with different geometries. An average heat transfer intensity of the tubes from middle column (A, B, C and D) of the in-line bundles No. 1, No. 2, No. 3 and No. 4 to upward foam flow at β=0.996 is shown in Fig. 3.

The effect of "shadow" takes place in the case of reduced spacing between centres of the tubes in the columns (tube bundles No. 1 and No. 3). Therefore an average heat transfer intensity of the tubes from middle column of the bundles No. 2 and No. 4 is higher than that of the bundles No. 1 and No. 3.

Local foam flow velocity increases while foam passes the rows of tubes. Local velocity is higher for the tube rows with more tubes in each (tube bundles No. 1 and No. 2). Therefore an average heat transfer intensity of the tubes from middle column of the bundles No. 1 and No. 2 is accordingly higher than that of the bundles No. 3 and No. 4.

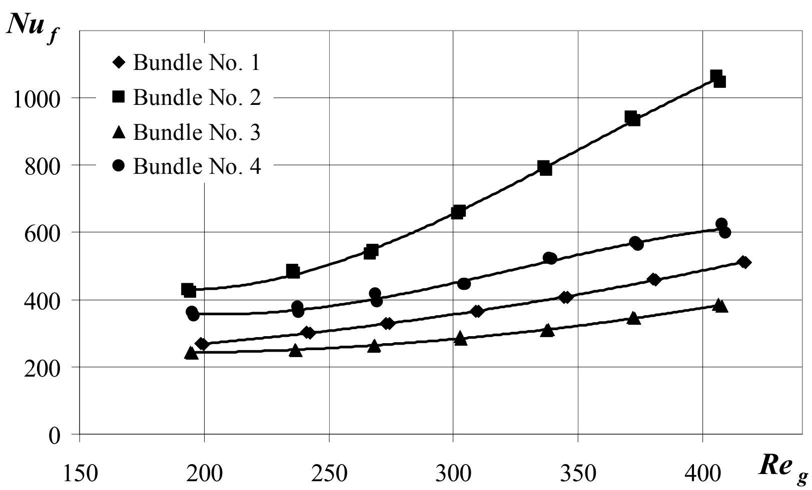

Figure 4: An average heat transfer intensity of the tubes from middle column of the in-line bundle No. 1, No. 2, No. 3 and No. 4 to upward foam flow, β=0.997.

By increasing of foam flow gas Reynolds number (Re_g) from 190 to 410, heat transfer intensity (Nu_f) of the tubes from middle column to upward foam flow increases twice of the in-line bundle No. 1, by 2.5 times of the bundle No. 2 and by 1.6 times of the bundles No. 3 and No. 4 for β=0.996. An average heat transfer intensity of the tubes from middle column of the bundle No. 2 is on average by 1.8 times higher than that of the bundle No. 1, by 2.3 times higher than that of the bundle No. 3 and by 1.5 times higher than that of the bundle No. 4 for β=0.996 and Re_g=190÷410.

An average heat transfer intensity of the tubes from middle column (A, B, C and D) of the in-line tube bundles No. 1, No. 2, No. 3 and No. 4 to upward foam flow at β=0.997 is shown in Fig. 4. In this case foam is drier and heat transfer intensity of all investigated bundles is less in comparison with foam volumetric void fraction (β) equal to 0.996. Changing Re_g from 190 to 410, an average heat transfer intensity of the tubes from middle column to upward foam flow increases by 1.9 times of in-line bundle No. 1, by 2.5 times of the bundle No. 2, by 1.6 times of the bundle No. 3 and by 1.7 times of the bundle No. 4 for β=0.997. An average heat transfer intensity of the tubes from middle column of the bundle No. 2 is on average by 1.9 times higher than that of the bundle No. 1, by 2.3 times higher than that of the bundle No. 3 and by 1.5 times higher than that of the bundle No. 4 for β=0.997 and Re_g=190–410.

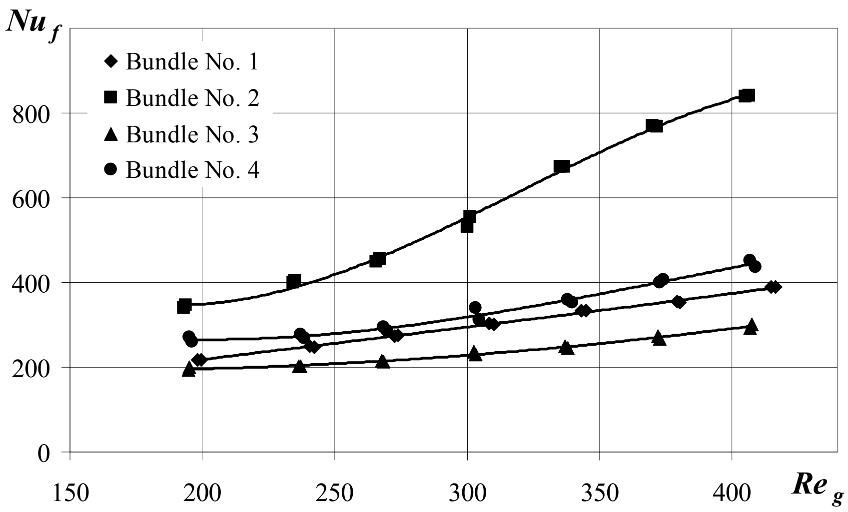

Figure 5: An average heat transfer intensity of the tubes from middle column of the in-line bundle No. 1, No. 2, No. 3 and No. 4 to upward foam flow, β=0.998.

An average heat transfer rate of the tubes from middle column of the in-line tube bundles No. 1, No. 2, No. 3 and No. 4 to upward foam flow at β=0.998 (the driest foam of our investigation) is shown in Fig. 5. With increase of Re_g from 190 to 410, an average heat transfer intensity of the tubes from middle column to upward foam flow increases by 1.8 times of in-line bundle No. 1, by 2.4 times of the bundle No. 2, by 1.5 times of the bundle No. 3 and by 1.7 times of the bundle No. 4 for β=0.998. An average heat transfer intensity of the tubes from middle column of the bundle No. 2 is on average by 1.9 times higher than that of the bundle No. 1, by 2.4 times higher than that of the bundle No. 3 and by 1.7 times higher than that of the bundle No. 4 for β=0.998 and Re_g=190–410.

Experimental results of investigation of heat transfer from the in-line tube bundles to upward laminar 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 $190 < Re_g < 410$ 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:

$$Nu_f = c\beta^n \operatorname{Re}_g^m . \tag{7}$$

On average, for whole middle column (A) of the in-line tube bundle No. 1 (s_1=s_2=0.03 m) to the upward foam flow c=5.7, n=340, m=102.1(1.006–β).

On average, for whole middle column (B) of the bundle No. 2 (s_1=0.03 and s_2=0.06 m) to the upward foam flow c=0.29, n= –125, m=14.3(1.089–β).

On average, for whole middle column (C) of the bundle No. 3 (s_1=0.06 and s_2=0.03 m) to the upward foam flow c=7.4, n= –111, m=22.8(1.023–β).

On average, for whole middle column (D) of the bundle No. 4 (s_1=0.06 and s_2=0.06 m) to the upward foam flow c=25.4, n=363, m=77.5(1.006–β).

4 Conclusions

Heat transfer from four in-line tube bundles with different geometry to vertical laminar upward statically stable foam flow was investigated experimentally.

Heat transfer intensity of in-line tube bundles to upward foam flow is higher for bundles with increased spacing between tubes centres along the bundle (bundles No. 2 and No. 4).

Local foam flow velocity increases while foam passes the rows of tubes and influences the heat transfer intensity of tubes. Therefore heat transfer intensity is higher for the tube bundles with reduced spacing between tubes centres across the bundle (bundles No. 1 and No. 2).

Criterion equation (7) may be applied for calculation and design of the statically stable foam heat exchangers with in-line tube bundles.

Nomenclature

A – cross section area of experimental channel, m^2; c, m, n – coefficients; d – outside diameter of tube, m; G – volumetric flow rate, m^3/s; Nu– Nusselt number; q – heat flux density, W/m^2; Re – Reynolds number; $\overline{T}$ – average temperature, K; h – average coefficient of heat transfer, 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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References

[1] Gylys, J., Hydrodynamics and Heat Transfer Under the Cellular Foam Systems, Technologija: Kaunas, 1998.
[2] Fournel B., Lemonnier H., Pouvreau J., Foam Drainage Characterization by Using Impedance Methods, 3rd Int. Symp. on Two-Phase Flow Modelling and Experimentation, Pisa, Italy, p. [1–7], 2004.
[3] Tichomirov V., Foams. Theory and Practice of Foam Generation and Destruction, Chimija: Moscow, 1983.
[4] Garrett P. R, Recent developments in the understanding of foam generation and stability. Chemical Engineering Science, **48(2)**, pp. 367–392, 1993.
[5] Zukauskas A., Convectional Heat Transfer in Heat Exchangers, Nauka: Moscow, p. 472, 1982.
[6] Hewitt, G. F., Heat exchanger design handbook 2002, York, Begell House, 2002.
[7] Gylys J., Zdankus T., Sinkunas S., Giedraitis V., Study of In-line Tube Bundle Cooling in Vertical Foam Flow. WSEAS Transactions on Heat and Mass Transfer, Iss. 6, **1**, pp. 632–637, 2006.
[8] Gylys, J.; Sinkunas, S.; Zdankus, T.; Giedraitis, V., Different type tube bundle heat transfer to vertical foam flow, Proceedings of the 5th International Conference on Nanochannels, Microchannels and Minichannels, ICNMM 2007, Pueble, Mexico, pp. 449–455, 2007.
[9] Gylys, J.; Sinkunas, S.; Zdankus, T.; Giedraitis, V., Balcius, A., Foam flow turn influence on the in-line tube bundle heat transfer intensity, Computational Methods and Experimental Measurements XIII, Prague, Czech Republic, pp. 457–464, 2007.