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Heat transfer between the non-standard tube bundle and statically stable foam flow

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Nomenclature

A - cross section area of experimental channel, m^2; *d* - outside diameter of tube, m; *G* - volumetric flow rate, m^3/s; *h* - average coefficient of heat transfer, W/(m^{2}K); *k*, *m*, *n* - coefficients; *Nu* - Nusselt number; *q* - heat flux density, W/m^2; *Re* - Reynolds number; *T* - average temperature, K; β - volumetric void fraction; λ - thermal conductivity, W/(mK); ν - kinematic viscosity, m^2/s.
Indexes
f - foam; *g* - gas; *l* - liquid; *w* - wall of heated tube.

1. Introduction

Foam is a dispersion system, consisting of a number of gases (air, steam) bubbles – foam lattice, separated by liquid films of considered thickness. Gas in this case is treated as a dispersed phase and fluid – as a viscosity adjuster. The liquid films separating gas bubbles form specific "skeletons", which are the bases of the foam structure. Foam dispersion medium may be solid material, but this type of foam is not a subject of this investigation.

The usage of the statically stable aqueous foam as a coolant was a main task of our investigation, therefore, the characteristics and structure of the foam varies in time [1, 2 and 3]. Such processes like: drainage of the liquid from the foam [4, 5], diffusive gas transfer between the foam bubbles [2], division, junction and destruction of the foam bubbles [2, 6] complicate an application of the analytic methods for the heat transfer study under the foam flow. Therefore an experimental method was selected for our investigation.

In energy and technology industries, in different technological processes [7, 8, and 9], heat and mass transfer is performed using various types of the heat exchangers [10, 11 and 12]. Currently, the most prevalent type is recuperator type heat exchangers (heating and heated heat carriers are separated by a wall). For such heat transfer process usually tube bundles are used due to easy production, simple mounting, grouping, and good compactness properties. At the same time tube bundles have good thermal characteristics and durability.

Usually two ways are used for layout tubes in the bundles. These are staggered and in-line tube bundles. The main geometric parameters characterizing the bundle are the outer diameter of the tubes *d*, and the steps between the pipes: across s_1 (the distance between the tube axis across the flow) and lengthwise s_2 (the distance between the pipes put one after another in accordance with the direction of flow, axis). It is important to specify the number of tube rows in the bundle and the number of tubes in each row.

Heat carrier can flow through the tubes and heat or cool the coolant inside the tube or vice versa. Direction of the heat carrier, hydro-flowing around the tube bundle, depends on the particular situation, and can be very diverse: along, across or at an angle against the tube bundle. The heat exchange, when the tube bundles are flow–rounded by a single-phase stream, has been examined in details [10, 12]. The investigation was carried out by changing the physical properties and flow regimes of the stream flowing-rounded the tubes. The heat exchange for the case when the cross and longitudinal steps of the stream flowing-around tube bundle were changed was investigated as well. The heat exchange of the tube bundle with both, smooth and faceted tube surface has been analyzed also [10, 12].

Application of the two-phase coolants, such as aqueous foam, in practice could significantly reduce material and energy demands, simultaneously sustaining the proper heat transfer intensity on the heated surfaces [13]. Such advantages of aqueous foam give a chance to create a compact, light, safe and economic heat exchanger [1, 13].

The major objectives of our present researches are to determine and estimate the influence of tube bundles type and geometry on the intensity of tube bundles heat transfer to the foam flow. It had been provided experiments with staggered [14] and in-line tube bundles [13, 15]. The dependence of the non-standard tube bundle heat transfer intensity on foam flow velocity and volumetric void fraction are determined and discussed in this work. One of the main objectives for this investigation was to determine the optimal type of tube bundle therefore it was important to investigate the effectiveness of our non-standard tube bundle for heat exchange process and compare the results of investigation with such results of the typical staggered and in-line tube bundles.

2. Experimental set-up

Experimental set-up, used during experiments, consisted of foam flow generating equipment, experimental channel, non-standard tube bundle, measuring instruments and auxiliary equipment (Fig. 1).

Statically stable foam flow was generated from the detergents solution in water (concentration of detergents: 0.5%) during gas and liquid contact on the perforated plate.

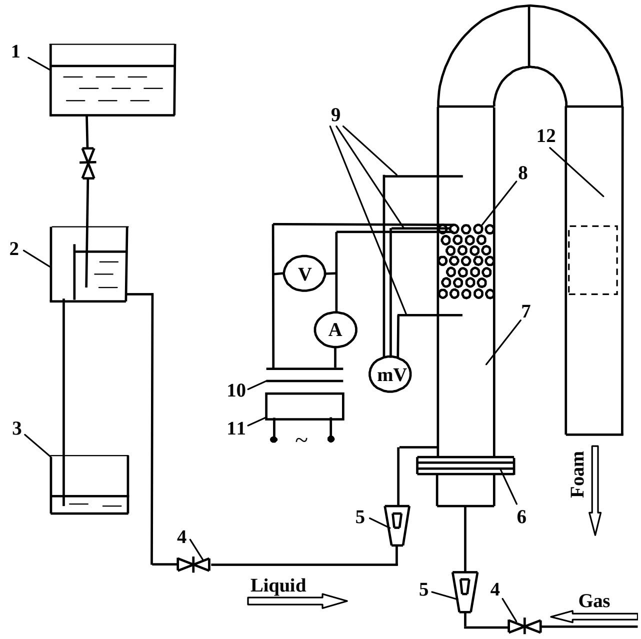


Fig. 1 Scheme of the experimental set-up: *1* - solution tank; *2* - container to maintain constant level; *3* - solution reservoir for overflows; *4* - control valve; *5* - flow meter; *6* - perforated plate; *7* - experimental channel; *8* - tube bundle; *9* - termocouples; *10* - transformer; *11* - stabilizer; *12* - output channel

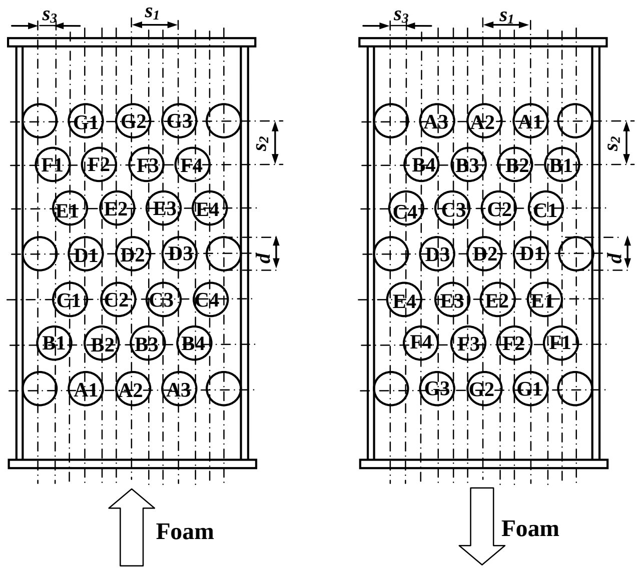
The experimental channel had cross section which dimensions were 0.14 x 0.14 m^2; height was 1.8 m.

Non-standard tube bundle (Fig. 2) consisted of seven rows of tubes (diameter d = 0.02 m, amount of tubes in a row: 1st = 5, 2nd = 4, 3rd = 4, 4th = 5, 5th = 4, 6th = 4, 7th = 5). The spaces between the centers of the tubes in a row across the tube bundle were $s_1 = 1.5d$ m and the distance between axis piloted out through tubes centres in horizontal rows was $s_2 = 1.5d$ m. The tubes in a second row were moved into the right side considering the first row with a distance $s_3 = 0.5d$ m. The third row tubes were moved with the same distance $s_3 = 0.5d$ m to the right side regarding the second row tubes. Fourth row tubes were aligned horizontally the same way as the first. The fifth, sixth and seventh rows were stated complexly like a mirror – image of the first three rows. Due to this kind of complicated array of tubes, the tube bundle was named "non-standard".

Experimental investigation of the heat transfer between the tubes of the non-standard tube bundle and upward statically stable foam flow was performed initially (Fig. 1). Then the tube bundle was reinstalled to the output part of the experimental channel and the investigation with the downward after 180° turning foam flow was provided.

Experiments were performed according to the methodology which was used during our previous works [13-15].

Accuracy of the temperature measurements was ±0.5 K for its operating temperature range of 273.15 to 373.15 K (0-100°C). Accuracy of flow measurements was ±0.1 x 10^{-3} m^3/s for gas (air) across all operating range, which varied from 0 to 10 x 10^{-3} m^3/s; and it was ±0.25 x 10^{-6} m^3/s for liquid (detergent solution) across all operating range, which varied from 0 to 40 x 10^{-6} m^3/s. Accuracy of the ammeter measurements were ±0.1 A across all its operating range, which was from 0 to 10 A; accuracy of the voltmeter measurements were ±0.05 V across all its operating range, which was from 0 to 25V.



a b

Fig. 2 Non-standard tube bundle in upward (a) and downward (b) foam flow

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

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

where foam flow volumetric void fraction

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

Nusselt number

$$Nu_f = \frac{hd}{\lambda_f} \tag{3}$$

Thermal conductivity of the statically stable foam flow λ_f, W/(m·K)

$$\lambda_f = \beta\lambda_g + (1-\beta)\lambda_l \tag{4}$$

An average heat transfer coefficient

$$h = \frac{q_w}{\Delta T} \tag{5}$$

An average temperature difference (ΔT) between the average temperatures of foam (T_f) and tube surface (T_w)

$$(\Delta T = T_w - T_f) \tag{6}$$

Gas Reynolds number of the foam flow

$$Re_g = \frac{G_g d}{A \nu_g} \tag{7}$$

Experiments were performed within limits of Reynolds number diapason for gas (Re_g): 190~440 (laminar flow regime) and foam volumetric void fraction (β): 0.996~0.998. Gas velocity for foam flow was changed from 0.14 to 0.32 m/s.

3. Results

The heat transfer process between the tubes of the bundle and vertical upward foam flow was investigated initially.

The comparison of heat transfer intensity (Nu_f) of the tubes A1, A2 and A3 of the first horizontal line to the upward foam flow is shown in the Fig. 3. The tubes A1, A2 and A3 were the first obstacle for the foam flow from its generation place to the bundle. The tubes A1 and A3 were located at the same distance from the vertical axis of the experimental channel, therefore local void fraction of foam and foam flow local velocity had correspondingly the same values near mentioned tubes and the heat transfer intensity of those tubes was identical. The data of Nu_f-Re_g relationship of the tubes A1 and A3 are presented in the Fig. 3 as A1&A3.

An influence of two main parameters of foam flow such as the cross-sectional distribution of the flow local velocity and the cross-sectional distribution of the local void fraction of the foam compensates each other within the interval of Re_g from 190 to 400. Therefore the difference between heat transfer intensity of the middle tube A2 and side tubes A1 and A3 to upward foam flow is negligible and reaches only 2% for $\beta = 0.996$ and 0.997 and less than 1% for $\beta = 0.998$ within the mentioned interval of Re_g. The structure of foam becomes more homogenous when Re_g is more than 400 and the velocity becomes the main factor of the influence on the tubes' heat transfer intensity. Therefore, the heat transfer between tube A2 and foam flow is more intensive than that of the A1 and A3.

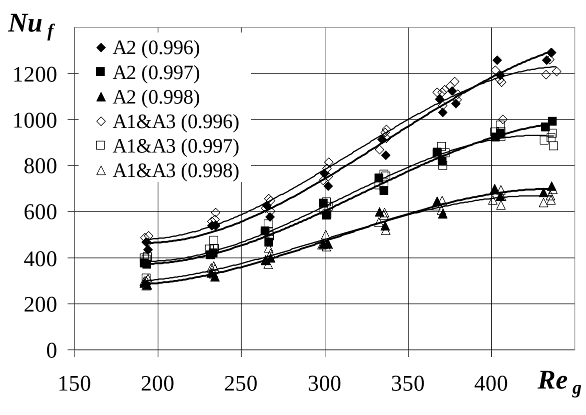


Fig. 3 Heat transfer intensity of the tubes A1, A2 and A3 to upward foam flow, $\beta = 0.996$, 0.997 and 0.998

Foam flow gas Reynolds number (Re_g) increases from 190 to 440, heat transfer intensity (Nu_f) of the tube A2 increases by 2.8 times (from $Nu_f = 450$ to 1273) for foam with volumetric void fraction $\beta = 0.996$; by 2.6 times (from $Nu_f = 372$ to 977) for $\beta = 0.997$, and by 2.4 times (from $Nu_f = 285$ to 697) for $\beta = 0.998$. The heat transfer intensity of the tubes A1 and A3 increases by 2.6 times (from $Nu_f = 470$ to 1238) for $\beta = 0.996$; by 2.4 times (from $Nu_f = 374$ to 911) for $\beta = 0.997$, and by 2.2 times (from $Nu_f = 97$ to 664) for $\beta = 0.998$ and $Re_g = 190$~440.

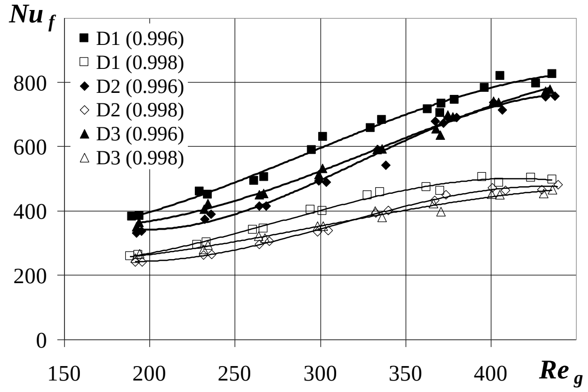


Fig. 4 Heat transfer intensity of the tubes D1, D2 and D3 to upward foam flow, $\beta = 0.996$ and 0.998

The comparison of heat transfer intensity (Nu_f) of the tubes D1, D2 and D3 of the fourth horizontal line to the upward foam flow is shown in the Fig. 4. The foam flow passes obstacles: the first, second and the third horizontal lines of tubes by reaching the tubes D1, D2 and D3. After the fourth line some bubbles strike against the tubes of the fifth horizontal line and change their moving direction. The cross-sectional distribution of foam flow velocity and void fraction is transformed near the tubes of the fourth tube line. Therefore the heat transfer intensity between the tubes of the fourth horizontal line and foam flow is different in comparison with the case of the first horizontal line of the tubes.

With increasing of Re_g from 190 to 440 the heat transfer intensity (Nu_f) of the tube D1 to the upward foam flow increases by 2.1 times (from $Nu_f = 384$ to 811), the heat transfer intensity of the tube D2 increases by 2.3 times (from $Nu_f = 335$ to 764) and that of the tube D3 increases by 2.2 times (from $Nu_f = 359$ to 775) for foam volumetric void fraction $\beta = 0.996$. The Nu_f of the tube D1 increases by 1.9 times (from $Nu_f = 262$ to 501), the Nu_f of the tube D2 increases twice (from $Nu_f = 241$ to 487) and the Nu_f of the tube D3 increases by 1.8 times (from $Nu_f = 260$ to 460) for $\beta = 0.998$ and $Re_g = 190 \div 440$.

The intensity of the heat transfer between the tube D1 and upward foam flow is by 11.2% higher than that of the tube D2 and by 7.9% higher than that of the tube D3 for $\beta = 0.996$, and the Nu_f of the tube D1 is by 10% higher than that of the tubes D2 and D3 for $\beta = 0.998$ within limits of Re_g from 190 to 440. The difference between the Nu_f of the tubes D2 and D3 is 3.1% for $\beta = 0.996$ and only 0.2% for $\beta = 0.998$.

The heat transfer intensity of the tube A2 (middle tube of the first horizontal line) to the upward foam flow is on average by 1.6 times higher than that of the tube D2 (middle tube of the fourth horizontal line) for $\beta = 0.996$, by 1.5 times higher than that of the tube D2 for $\beta = 0.997$ and by 1.4 times higher than that of the tube D2 for $\beta = 0.998$, ($Re_g = 190\sim440$).

It is difficult to compare the heat transfer intensity of the other tubes (A1, A3 and D1, D3) of the first and fourth (and other) lines, because the positions of the tubes in the cross-section of the channel were different. Therefore average heat transfer intensity (Nu_{f_av}) of the tubes of each horizontal line to upward foam flow was calculated for the better experimental results analysis. An average heat transfer of the tubes of the first line means the average heat transfer intensity of the tubes A1, A2 and A3 to foam flow and so on with other lines of tubes.

The comparison of average heat transfer intensity of tubes of horizontal lines to upward foam flow for $\beta = 0.997$ is shown in the Fig. 5. The heat transfer process between the tubes of the first line ant foam flow is the most intensive. The heat transfer intensity of the tubes of the second, third, fourth, fifth and sixth horizontal rows to foam flow differs from each to other less than 12%. The tubes of the last (seventh) line are under different conditions – the washing of its backsides are better than that of the other tubes. Therefore the heat transfer process between the tubes of the last horizontal line and foam flow is more intensive than that of the tubes of the second, third, fourth, fifth and sixth lines.

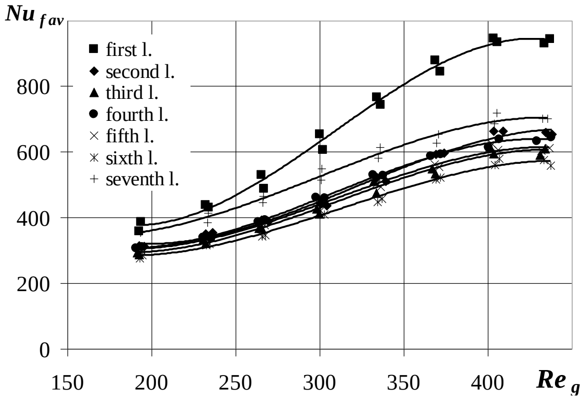


Fig. 5 An average heat transfer intensity of the tubes of horizontal lines to upward foam flow, $\beta = 0.997$

After the experiments with upward foam flow the experimental set-up was reinstalled and the experiments with downward foam flow followed. The foam flow was generated at the bottom of the experimental channel, and then foam moved upward, made the 180° degree turning and moved downward crossing the tube bundle. Local velocity and void fraction distribution for that case wasn't symmetrical at the cross-section before reaching the tube bundle. Foam was wetter on the right side of the cross-section (in the direction of flow) near tube A3, and foam was drier on the left side of the cross-section near tube A1. Foam local velocity cross-sectional distribution was transformed after the turn also.

Comparison of heat transfer intensity (Nu_f) of the tubes A1, A2 and A3 of the first horizontal line to the downward foam flow is shown in the Fig. 6.

With increasing of Re_g from 190 to 440 the Nu_f of the tube A1 to the downward foam flow increases by 2.3 times (from $Nu_f = 336$ to 762), the heat transfer intensity of the tube A2 increases by 2.1 times (from $Nu_f = 473$ to 1005) and that of the tube A3 increases by 1.8 times (from $Nu_f = 659$ to 1181) for foam with $\beta = 0.996$. The Nu_f of the tube A1 increases by 1.6 times (from $Nu_f = 266$ to 427), the Nu_f of the tube A2 increases by 1.7 times (from $Nu_f = 289$ to 483) and the Nu_f of the tube A3 increases by 1.9 times (from $Nu_f = 293$ to 549) for $\beta = 0.998$ and $Re_g = 190\div440$.

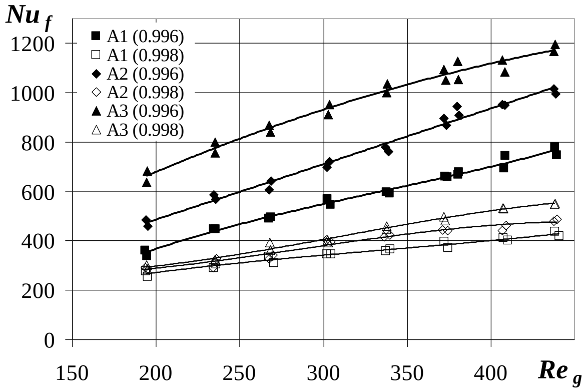


Fig. 6 Heat transfer intensity of the tubes A1, A2 and A3 to downward foam flow, $\beta = 0.996$ and 0.998

The Nu_f of the tube A3 is on average by 25.7% higher than that of the tube A2 and by 65.6% higher than that of the tube A1 for $\beta = 0.996$, and the Nu_f of the tube A3 is by 9.1% higher than that of the tube A2 and by 21.6% higher than that of the tube A1 for $\beta = 0.998$ within limits of Re_g from 190 to 440.

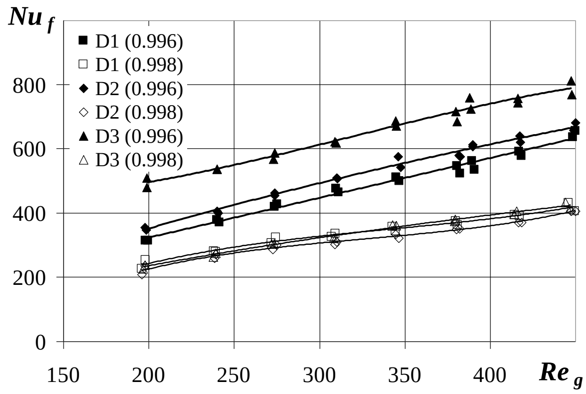


Fig. 7 Heat transfer intensity of the tubes D1, D2 and D3 to downward foam flow, $\beta = 0.996$ and 0.998

The comparison of heat transfer intensity of the tubes D1, D2 and D3 of the fourth horizontal line to the downward foam flow is shown in the Fig. 7. By increasing of Re_g from 190 to 440 the Nu_f of the tube D1 increases by 2.1 times (from $Nu_f = 314$ to 647), the Nu_f of the tube D2 increases by 1.9 times (from $Nu_f = 351$ to 681) and that of the tube D3 increases by 1.6 times (from $Nu_f = 495$ to 790) for foam with $\beta = 0.996$. The Nu_f of the tube D1 increases by 1.7 times (from $Nu_f = 240$ to 418), the Nu_f of the tube D2 increases by 1.8 times (from $Nu_f = 222$ to 406) and the Nu_f of the tube D3 increases by 1.8 times (from $Nu_f = 233$ to 423) for $\beta = 0.998$ and $Re_g = 190\div440$.

The Nu_f of the tube D3 by 23.6% higher than that of the tube D2 and by 33.9% higher than that of the tube D1 for β=0.996 and Re_g=190÷440. The difference of heat transfer intensity of tubes D1, D2 and D3 to downward foam flow is negligible and is no more than 6% for β=0.998 and Re_g=190÷440.

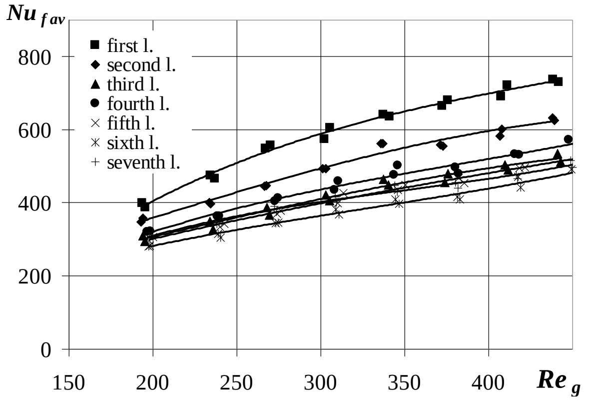


Fig. 8 An average heat transfer intensity of the tubes of horizontal lines to downward foam flow, $\beta = 0.997$

Average heat transfer intensity (Nu_{f_av}) of the tubes of each horizontal line to downward foam flow was calculated like in the case of upward foam flow. The comparison of average heat transfer intensity of tubes of horizontal lines to downward foam flow for $\beta = 0.997$ is shown in the Fig. 8.

The heat transfer process between tubes of the first line ant downward foam flow is the most intensive. The heat transfer intensity of the tubes of the second row to foam flow is less than that of the first tube. The heat transfer intensity of the tubes of the third, fourth, fifth, sixth and seventh horizontal rows to foam flow differs from each to other within interval from zero to 18%.

An average heat transfer intensity of the tubes of entire non-standard tube bundle to foam flow was calculated in order to compare the efficiency of the investigated non-standard tube bundle with that of the in-line 1.5×1.5 tube bundle (Fig. 9).

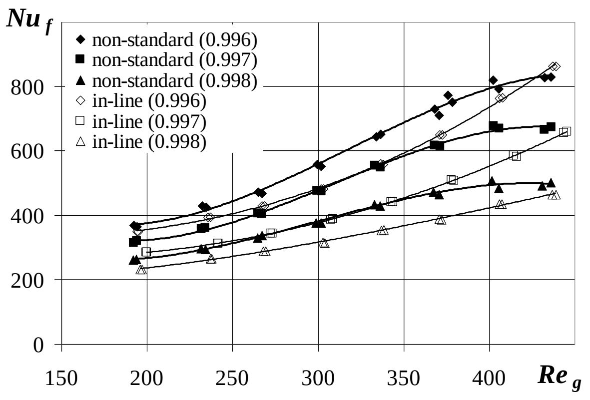


Fig. 9 Comparison of average heat transfer intensity of the tubes of the non-standard and in-line bundles to upward foam flow, β = 0.996, 0.997 and 0.998

Average heat transfer intensity of the tubes of the non-standard bundle to upward foam flow is higher than that of the tubes of the in-line bundle on average by 10.5% for β=0.996, by 15.5% for β =0.997 and by 15.2% for β=0.998 and for the interval of Re_g from 190 to 440 (except the one point when Re_g=440 and β=0.996).

The situation is different in the case of downward foam flow. The in-line arrangement of the tubes of the bundle influences more intensive heat transfer between tubes of the bundle and foam flow (Fig. 10). Average heat transfer intensity of the tubes of the in-line bundle to downward foam flow is higher than that of the tubes of the non-standard bundle on average by 22.0% for $\beta = 0.996$, by 17.4% for $\beta = 0.997$ and by 9.5% for $\beta = 0.998$ and for the interval of $Re_g = 190 \div 440$.

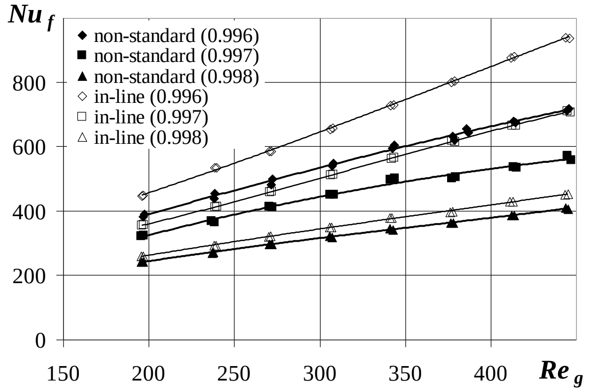


Fig. 10 Comparison of average heat transfer intensity of the tubes of the non-standard and in-line bundles to downward foam flow, β = 0.996, 0.997 and 0.998

The experimental results were generalized by using the dependence of Nusselt and gas Reynolds similarity criteria. This dependence within interval $190 < Re_g < 440$ of upward and downward foam flow at volumetric void fraction β = 0.996; 0.997; 0.998 can be expressed by the equation

$$Nu_f = k\left(\frac{\beta}{1-\beta}\right)^m Re_g^{\ n}, \tag{8}$$

Computation of average heat transfer intensity of the tubes of non–standard tube bundle to upward foam flow: k=210, m= –0.79 and n=0.95.

Computation of average heat transfer intensity of the tubes of non–standard tube bundle to downward foam flow: k=705, m= –0.79 and n=0.72.

4. Conclusions

1. Heat transfer process between the tubes of the non-standard bundle and vertical flow of statically stable foam was investigated experimentally. It was determined influence of the volumetric void fraction, foam flow velocity and flow direction on the heat transfer intensity.

2. Heat transfer process between the tubes of the first line ant upward foam flow is most intensive. Heat transfer intensity of second, third, fourth, fifth and sixth horizontal rows of the tubes to foam flow differs each from other less than 12%. Heat transfer process between the tubes of the last horizontal line and foam flow is more intensive than that of the tubes of the second, third, fourth, fifth and sixth lines.

3. Heat transfer process between the tubes of the first line ant downward foam flow is most intensive like in the case of the upward foam flow. Heat transfer intensity of the tubes of the second row to foam flow is less than that of the first tube. The heat transfer intensity of the tubes of the third, fourth, fifth, sixth and seventh horizontal rows to foam flow differs each from other within interval from zero to 18%.

4. An average heat transfer intensity of the tubes of the non-standard bundle to upward foam flow is higher than that of the tubes of the in-line 1.5×1.5 tube bundle. Case of downward foam flow is different. An average heat transfer intensity of the in-line 1.5×1.5 tube bundle to downward foam flow is higher than that of the tubes of the non-standard bundle.

5. Criterion Eq. (8) may be applied for calculation and design of the statically stable foam heat exchangers with non-standard tube bundles.

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ŠILUMOS MAINAI TARP NESTANDARTINIO VAMZDŽIŲ PLUOŠTO IR STATIŠKAI STABILIŲ PUTŲ SRAUTO

R e z i u m ė

Praktikoje naudojant dvifazius šilumnešius, tokius kaip vandeninės putos, galima žymiai sumažinti medžiagines ir energetines sąnaudas, tuo pat metu užtikrinant reikiamą šilumos mainų intensyvumą. Be to, dvifazis putų šilumnešis pasižymi papildoma savybe, leidžiančia valdyti šilumos mainų intensyvumą keičiant putų tūrinį debitinį dujingumą. Tokie vandeninių putų pranašumai suteikia galimybę sukurti kompaktišką, lengvą, saugų ir ekonomišką šilumokaitį. Šiame straipsnyje pateikti ir aptarti šilumos mainų tarp „nestandartinio“ vamzdžių pluošto ir vertikaliai kylančio bei po 180 laipsnių posūkio besileidžiančio putų srauto eksperimentinio tyrimo rezultatai.

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HEAT TRANSFER BETWEEN THE NON-STANDARD TUBE BUNDLE AND STATICALLY STABLE FOAM FLOW

S u m m a r y

Applying the two-phase coolants, such as aqueous foam, in practice could significantly reduce material and energy demands, simultaneously sustaining the proper heat transfer intensity. Moreover, the two-phase foam coolant has additional possibility for control of the intensity of heat transfer by changing volumetric void fraction of the foam. Such advantages of aqueous foam give a chance to design a compact, light, safe and economic heat exchanger. The results of an experimental investigation of heat transfer between “non-standard” tube bundle and vertically upward and downward after 180 degree turning foam flow are presented and discussed in this paper.

Keywords: experimental investigation, heat transfer, two-phase flow, aqueous foam, tube bundle, heat exchanger.

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