Tube bundle's cooling by aqueous foam

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Abstract

Single-phase coolants, such as water, oil or air, are mostly used in industrial apparatus. But usage of two-phase coolants, such as aqueous foam, can significantly reduce material and energy expenditures, simultaneously sustaining proper heat transfer intensity on heated surfaces. Comparing with single-phase coolant, the two-phase foam coolant has additional possibility to change the intensity of heat transfer by changing volumetric void fraction of foam. This enables wider range of regulation of heat transfer intensity. An experimental investigation of heat transfer between in-line tube bundle and aqueous foam flow was performed. One type of aqueous foam – statically stable foam – was used as a coolant. Vertically downward moving foam flow crossed the in-line tube bundle. Spacing between the centres of the tubes across the in-line tube bundle was 0.03 m and spacing along the bundle was 0.03 m. During an experimental investigation it was determined dependence of heat transfer intensity on flow parameters: flow velocity, volumetric void fraction and liquid drainage from foam. Apart of this, influence of tube position in the bundle on heat transfer was investigated. Experimental results were summarized by criterion equations, which are suitable for the design and calculation of foam apparatus. Mentioned experiments are the continuation of our previous investigation with foam flow moving downward after 180-degree turning.

Keywords: heat transfer, aqueous foam flow, in-line tube bundle, void fraction of foam.

1 Introduction

Heat transfer between different heated surfaces and aqueous foam flow is the area of our researches. It was noticed that usage of two-phase coolants, such as

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aqueous foam, could significantly reduce material and energy expenditures, simultaneously sustaining proper heat transfer intensity on heated surfaces. Comparing with single-phase coolant, the two-phase foam coolant has additional possibility to change the intensity of heat transfer by changing volumetric void fraction of foam. Small density and mentioned properties of foam type coolant enables to create compact, light and economic heat exchanger with simple and safe operation using two-phase foam flow.

Characteristics of one type of aqueous foam, namely statically stable foam, demonstrate its perfect availability for heat transfer process [1]. It appears that investigated by us statically stable foam keeps its initial structure and bubbles' dimensions within broad limits of time intervals, from several minutes to days, even after termination of the foam generation. Thus, this type of coolant was used as heat transfer working fluid in our investigation.

The statically stable foam has all specific peculiarities of aqueous solution foams: drainage of liquid from foam [2, 3], diffusive transfer of gas between bubbles [4], division and collapse of foam bubbles [1, 4]. During foam flow contact with heated surfaces some foam bubbles are destroyed and additional liquid flow appears. All those mentioned above phenomena are closely linked with each other and make extremely complicated an application of analytic methods for their study. Thus experimental method of investigation was selected in our work.

In our previous researches heat transfer of alone cylindrical surface–tube and then of tube line to upward statically stable foam flow was investigated [1]. Next experimental series with staggered tube bundle [5, 6] and in-line tube bundles of different geometry [7–9] in upward and downward after 180-degree turning foam flow followed.

Presently the experimental set-up design was changed and foam flow moved downward without any turning. During an experimental investigation it was determined dependence of heat transfer intensity on flow parameters: flow velocity, volumetric void fraction and liquid drainage from foam. Apart of this, influence of tube position in the bundle on heat transfer was investigated. The results of investigation were compared with the results of our previous researches and influence of foam flow turning on heat transfer intensity of different tubes of the bundle was determinate.

Results of investigation 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 calculating of heat transfer intensity of the in-line tube bundle.

2 Experimental set-up

Experimental set-up (fig. 1) consisted of the following main parts: vertical experimental channel, in-line 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. Cross section of the experimental channel had dimensions 0.14 x 0.14 m^2; height of it was 1.8 m. Walls of the

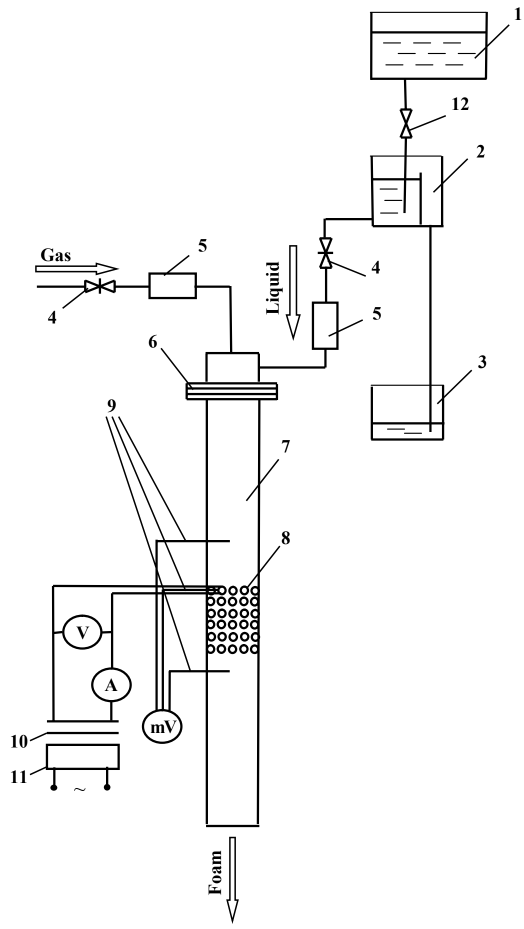

Figure 1: Experimental set-up scheme: 1 – liquid reservoir; 2 – liquid level control reservoir; 3 – liquid receiver; 4 – gas and liquid control valve; 5 – flow meter; 6 – foam generation plate; 7 – experimental channel; 8 – tube bundle; 9 – thermocouples; 10 – transformer; 11 – stabilizer; 12 – valve.

channel were made from the transparent material in order to observe foam flow visually.

Statically stable foam – one type of aqueous foam – was used as coolant for our experiments. Statically stable foam flow was generated from the detergents water solution. 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. Foam flow was generated during gas and liquid contact. Foam flow parameters control was fulfilled using gas and liquid valves.

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Perforated plate for foam generation was installed at the upside of the experimental channel and was made from stainless steel plate with a thickness of 2 mm; orifices were located in a staggered order; their diameter equal 1 mm; spacing between the centres of the holes equal 5 mm.

In-line tube bundle was used during experimental investigation. This bundle of the tubes consisted of six rows with five tubes in each. Spacing between centres of the tubes across and along the tube bundle was equal to 0.03 m (s_1=s_2=0.03 m). All tubes had an external diameter of 0.02 m. Schematic view of the experimental section with tube bundle is presented in fig. 2. One tube-calorimeter was heated electrically. This tube was made of copper and had an external diameter of 0.02 m also. The ends of the heated tube was sealed and insulated to prevent heat loss through them. During the experiments calorimeter was placed instead of one of the bundle's tube. 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.

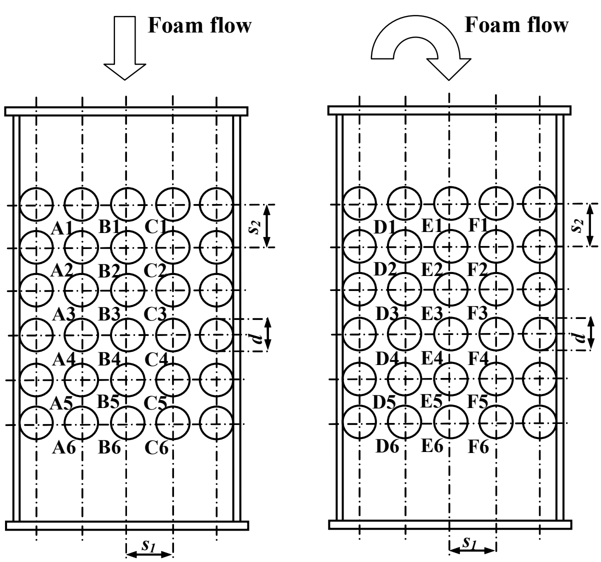

Figure 2: In-line tube bundle in downward and downward after turning foam flow.

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

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:

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

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

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

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

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

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

The process of heat transfer between tubes of in-line tube bundle and vertical downward laminar foam flow was the object of our experimental investigation.

Comparison of heat transfer intensity (Nu_f) of the third line tubes A3, B3 and C3 to the downward foam flow is shown in the fig. 3. The heat transfer intensity of tubes located at different places across and along the experimental channel is under the influence of distribution of local flow velocity and local foam void fraction across and along the channel. The maximum value of the foam flow local velocity is in the centre of the channel cross-section and decreases following to the walls of the channel. It is different with local void fraction of the foam. The foam is dryer in the centre of the channel cross-section and it is wetter near the channels walls. The distribution of the foam local void fraction decreases by foam flow passing the tube bundle. Side tubes A and C were located at the same distance from the vertical axis of the experimental channel, therefore foam local void fraction and foam flow local velocity had correspondingly the same values near the side tubes A and C and the heat transfer intensity of those tubes was identical. Some points of results data of the tubes A and C covered each other. Therefore an average heat transfer intensity of side tubes (AC) was calculated for the better experimental results analysis.

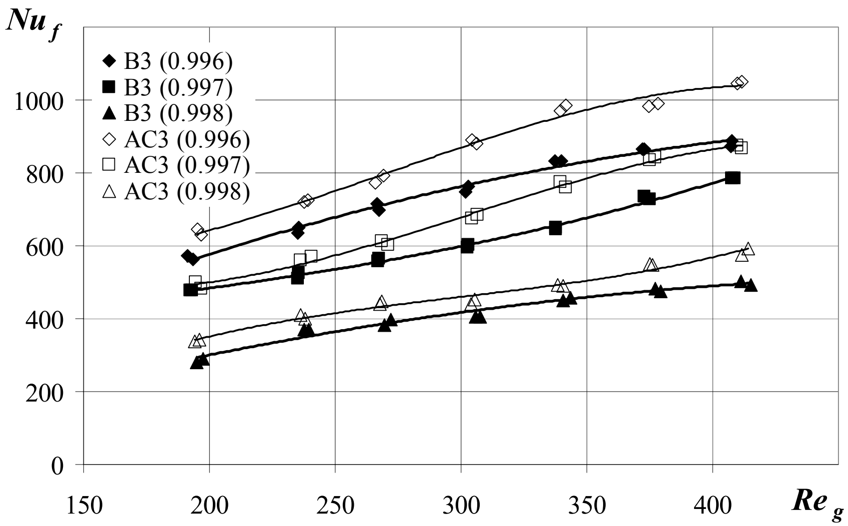

Figure 3: Heat transfer intensity of the tubes A3, B3 and C3 to downward foam flow, β=0.996, 0.997 and 0.998.

By increasing of foam flow gas Reynolds number (Re_g) from 190 to 410, heat transfer intensity (Nu_f) of the tube B3 from middle-column to downward foam flow increases by 1.6 times (from 567 to 880) for foam with volumetric void fraction β=0.996; by 1.6 times (from 476 to 785) for β=0.997, and by 1.7 times (from 285 to 498) for β=0.998. The heat transfer intensity of the tube B3 is on average by 1.9 times higher to the wettest foam flow (β=0.996) in comparison with the driest foam flow (β=0.998). The Nu_f of the side-column third tubes A3 and C3 grows by 1.6 times (from 637 to 1047) for β=0.996; by 1.8 times (from 491 to 871) for β=0.997, and by 1.7 times (from 340 to 584) for β=0.998 (Re_g =190–410). The heat transfer intensity of the side-tubes (A3 and C3) is on average by 1.8 times greater to the wettest foam flow in comparison with the driest foam flow. An average heat transfer intensity of the third side-tubes A3 and C3 (AC3) is higher than that of the third middle-tube (B3) by 15% for β=0.996; by 12% for β=0.997, and by 13% for β=0.998 (Re_g =190–410).

Situation is different when foam flow initially moved vertically upward then after the 180-degree turning moved vertically downward crossing the tube bundle [8]. Distribution of foam flow local void fraction across the channel changes during flow turning. This transformation depends mainly on liquid drainage from the foam; therefore drainage must be taken into account during analysis. Gravity forces act along the upward and downward foam flow, but in the foam flow turning those forces act across foam flow also. Therefore liquid drains down from the foam near upper channel wall and local void fraction increases (foam becomes drier) here as well. After the turn, local void fraction of foam is lower (foam is wetter) on the internal left side of the channel cross-section (near tube column D), and local void fraction of foam is higher (foam is

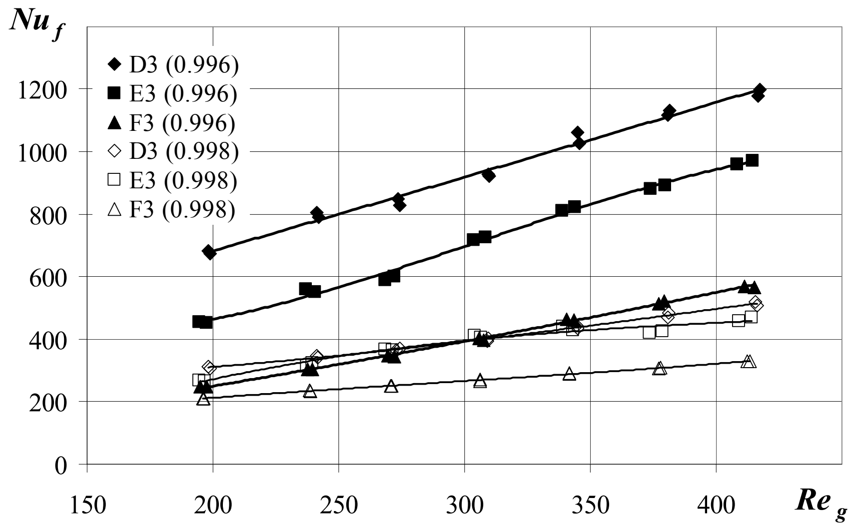

Figure 4: Heat transfer intensity of the tubes D3, E3 and F3 to downward after 180-degree turning foam flow, β=0.996, 0.997 and 0.998.

drier) on the external right side of the cross-section (near tube column F). Flow velocity distribution in cross section of the channel transforms after turn also. Comparison of heat transfer intensity of the third line tubes D3, E3 and F3 to the downward after 180-degree turning foam flow for the volumetric void fraction β=0.996 and β=0.998 is shown in the fig. 4.

Within the interval of foam flow gas Reynolds number (Re_g) from 190 to 410, heat transfer intensity (Nu_f) of the tube D3 is higher on average 31% than heat transfer intensity of the tube E3 and heat transfer of the tube E3 is higher on average 73% than that of the tube F3, for β=0.996. Heat transfer of the tube D3 is higher on average 7% than heat transfer intensity of the tube E3, and heat transfer of the tube E3 is higher on average 41% than that of the tube F3, for β=0.998. Heat transfer of the tube D3 to the wettest foam flow (β=0.996) is by 2.3 times higher than that to the driest foam flow (β=0.998).

An average heat transfer rate of middle-column tubes (B and E) was calculated in order to analyse the influence of foam flow turning on the heat transfer intensity of the tubes in the middle-column of the bundle. An average heat transfer intensity of the middle-column tubes (B) to downward foam flow and an average heat transfer intensity of the middle-column tubes (E) to downward after 180-degree and R=0.17 m radius foam flow turning is presented in the fig. 5.

Foam is wetter and local velocity of foam is higher in the centre of the channel cross-section in the case of foam flow without the turning. Therefore an average heat transfer intensity of the middle-column tubes B is higher than that of the middle-column tubes E (downward after 180-degree turning foam flow)

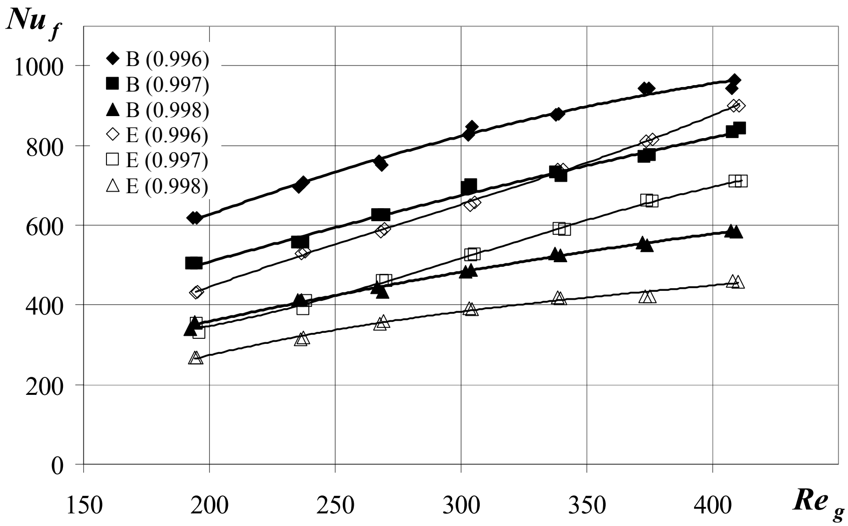

Figure 5: An average heat transfer intensity of the tubes B to downward foam flow and an average heat transfer intensity of the tubes E to downward after 180-degree turning foam flow, β=0.996, 0.997 and 0.998.

on average by 22% for β=0.996; by 28% for β=0.997, and by 27% for β=0.998 (Re_g=190–410).

Experimental results of investigation of heat transfer from the in-line tube bundle to downward and downward after 180-degree turning 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 downward and downward after 180-degree turning foam flow with the volumetric void fraction β=0.996, 0.997, and 0.998 can be expressed as follows:

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

For the entire middle-column (B) in the downward foam flow on average c=3.4, n=–500, m=41.4(β–1.018).

For the entire middle-column (E) in the downward after 180-degree turning foam flow on average c=16.1, n=518, m=140.7(1.003–β).

4 Conclusions

Heat transfer of in-line tube bundle to vertical downward laminar foam flow was investigated experimentally.

Heat transfer intensity of the third side-tubes (A3 and C3) is on average by 13–15% higher than that of the third middle-tube (B3) for β=0.996–0.998 and Re_g =190–410.

Foam is wetter and local velocity of foam is higher in the centre of the cross-section of the channel in the case of downward foam flow without turning. Therefore heat transfer intensity of the middle-column tubes to downward foam flow is on average by 22–28% higher than that of the middle-column tubes to downward after 180-degree turning foam flow.

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; h – average coefficient of heat transfer, W/(m^2·K); Nu– Nusselt number; q – heat flux density, W/m^2; Re – Reynolds number; $\overline{T}$ – average temperature, 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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