KAUNAS UNIVERSITY OF TECHNOLOGY
LITHUANIAN ENERGY INSTITUTE

Tadas Ždankus

ANALYSIS OF STAGGERED TUBE BANK HEAT TRANSFER IN DOWNWARD TWO PHASE FLOW

Summary of Doctoral Dissertation

Technological Sciences, Power and Thermal Engineering (06T)

Kaunas, 2004

This scientific work was carried out in 2000–2004 at Kaunas University of Technology.

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The official defense of the dissertation will be held at 14:00 on December 20, 2004 at the public session at Dissertation Defense Hall (K. Donelaičio g. 73–403) at Kaunas University of Technology.

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The sending–out of the summary of the Dissertation is on November 20, 2004.

The Dissertation is available at the libraries of Kaunas University of Technology (K. Donelaičio g. 20, Kaunas) and Lithuanian Energy Institute (Breslaujos g. 3, Kaunas).

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ŠILUMOS MAINŲ TYRIMAS BESILEIDŽIANČIAM DVIFAZIUI SRAUTUI APTEKANT ŠACHMATINĮ VAMZDŽIŲ PLUOŠTĄ

Daktaro disertacijos santrauka

Technologijos mokslai, energetika ir termoinžinerija (06 T)

Kaunas, 2004

Disertacija rengta 2000 – 2004 metais Kauno technologijos universitete.

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Disertacija ginama 2004 m. gruodžio 20 d. 14:00 val. viešame energetikos ir termoinžinerijos mokslo krypties tarybos posėdyje, kuris įvyks Kauno technologijos universitete, disertacijų gynimo salėje (K. Donelaičio g. 73 – 403).

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Disertacijos santrauka išsiųsta 2004-11-20.

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Increment of heat transfer intensity and decrement of heat carrier mass at the same time is the main task of scientific investigations of heat exchangers design. Such task is difficult to solve with one-phase heat carriers. Two–phase carriers, especially statically stable foam, appear more promising.

Two–phase foam flow has number of peculiarities, which make application of analytic methods for investigation of heat transfer in the foam flow extremely complicated. Thus experimental method of investigation was selected in this work. Direction of foam motion in vertical channels also influences on heat transfer intensity. Heat transfer process in vertical foam flow moving upward was investigated deep enough in earlier research works of other scientists. The heat transfer process in downward foam flow is not investigate yet. Design of modern heat exchangers with statically stable foam heat carrier is impossible without knowledge about tube bank heat transfer in both upward and downward statically stable foam flow. The scientific investigation of that problem is aim of this work.

The aim of the work

The aim of the work is to investigate heat transfer of the staggered tube bank to downward statically stable foam flow.

The tasks of the work

The following main tasks were raised for this work

- To investigate experimentally heat transfer of the staggered tube bank to downward statically stable foam flow.
- To determinate influence of liquid drainage from foam flow to heat transfer of tube bank in downward foam flow.
- To investigate influence of tube position in the bank, foam flow velocity and foam volumetric void fraction to heat transfer in downward statically stable foam flow.
- To investigate analytically distribution of downward fluid flow velocity in channel of the rectangular cross-section.
- To summarize obtained experimental results by criterion equations.

Object of investigations

The heat transfer of the staggered tube bank to downward statically stable foam flow is the object of investigations.

Novelty of the work

- For the first time heat transfer of the staggered tube bank to downward statically stable foam flow was investigated experimentally.
- Influence of downward foam flow parameters to heat transfer of tube bank has been determinated.

- Influence of tube location in the bank to the heat transfer with foam flow has been determinated.
- New analytical relationship of the fluid flow local velocity distribution in cross section of rectangular shape was developed.
- The experimental results of tube bank's heat transfer to vertical cross foam flow were summarized by criterion equations, which enable to calculate heat transfer rate of the entire bank or separate tube of the bank.

Actuality of the work

- The experimental results of tube bank's heat transfer to vertically foam flow were summarized by criterion equations, which enable to calculate heat transfer rate of the entire bank or separate tube of the bank at different values of void volumetric fractions and regime parameters of statically stable foam flow.
- Estimation of the influence of liquid drainage from foam flow to heat transfer of tube bank in downward foam flow allows to increase accuracy of design works.
- Results of own experimental investigations were generalized by criterion equations what makes them reliable and suitable for design of modern staggered ordered type heat exchangers with statically stable foam heat carrier.

Presentation of the work

Results of this study were presented at ten international and republican scientific conferences.

Approbation of this work

Seven articles were published on the subject of dissertation in censored international publications, two in prestigious Lithuanian research journals and three in not censored publications.

Scope of the work

The dissertation consists of introduction, 5 chapters and conclusions represented on 112 pages of text with 79 figures, 7 tables and 101 references.

Velocity Distribution in Rectangular Cross Section of Fluid Flow

Phenomenon of velocity distribution in cross section of steady uniform fluid flow remains under constant interest of scientists. It influences dynamic characteristics of a flow, also parameters of heat mass exchange phenomenon.

Hagen and Poiseuille were the first who managed to construct equation for laminar fluid flow local velocity in pipe of round cross section

$$u = \frac{\rho g h_f}{4\mu l}\left(r_0^2 - r^2\right). \tag{1}$$

Here u is local velocity; ρ is fluid density; g is acceleration of gravity; h_f is friction loss; μ is dynamic viscosity; l is distance between cross sections under consideration; r_o is pipe radius; r distance from the pipe axis to the point under consideration. The equation vas derived analyzing forces of pressure, friction and gravity acting cylindrical element of the flow in the center of the pipe. It is based on the assumption of zero velocity adjacent to a solid boundary, also linear relationship of shear stress τ to velocity gradient du/dy i.e.

$$\tau = \mu \frac{du}{dy}. \tag{2}$$

Equation (2) is valid for laminar regime only; therefore equation (1) should be applied for laminar regime only.

Fluid flow in narrow split is another phenomenon often meet in engineering practice and described in any fluid mechanics manual. Known formula of velocity distribution for that case

$$u = \frac{\rho g h_f}{2\mu l}\left(y_0^2 - y^2\right). \tag{3}$$

Here y_0 is a half of split width; y is distance from center of split to the rated point. This equation was constructed on the ground of the same assumptions like (1). Fluid flow in narrow split as a role is laminar therefore expansion validity of the equation has no sense.

Velocity distribution equation in rectangular cross section of laminar fluid flow may be constructed analyzing interaction between forces, acting symmetric parallelepiped shape element of the flow in the center of it. All six sides of the parallelepiped are acted by pressure, sides – by shear stress, all volume – by gravity forces. To simplify analysis let us substitute distributed loads of pressure, shear and gravity by the mean forces of pressure F_1, F_2, F_3, F_4, F_5 and F_6, shear F_7 and gravity F_8 (see Fig. 1). Projecting all these forces to the axis of the flow the following equation may be received

$$F_1 - F_2 - F_7 + F_8 = 0. \tag{4}$$

The mean forces of pressure acting bottom and top of the parallelepiped may be expressed as

$$F_1 = 4xyp_1 \text{ and } F_2 = 4xyp_2. \tag{5}$$

The mean force of shear stress may be expressed as a product of shear stress (see Eq.(2)) and area of corresponding sides ly or lx of the parallelepiped taking into account decrement of velocity u with increment of distance from the center of the flow x or y.

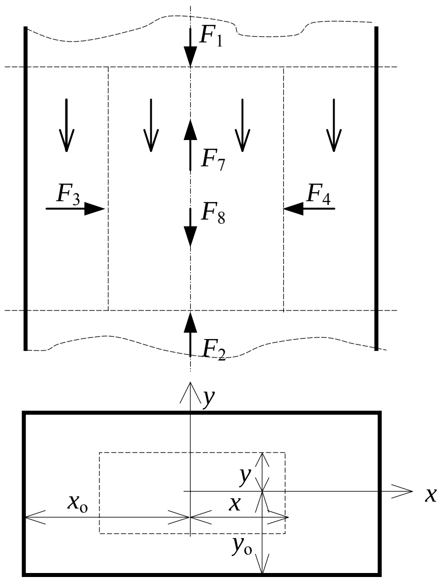

Fig.1. Scheme of flow, dimensions and force vectors

The mean shear force acting all four sides

$$F_7 = -4l\mu\left(y\frac{\partial u}{\partial x} + x\frac{\partial u}{\partial y}\right). \qquad (6)$$

The mean gravity force depends on mass of fluid contained in parallelepiped $4xyl\rho$ and acceleration of gravity, i.e.

$$F_8 = 4xyl\rho g\,. \qquad (7)$$

Here ρ is density of the fluid.
Applying expressions (5), (6) and (7) Eq. (4) may be recorded in such shape

$$4xyp_1 - 4xyp_2 - \left(-4l\mu\left(y\frac{\partial u}{\partial x} + x\frac{\partial u}{\partial y}\right)\right) + 4xyl\rho g = 0\,. \qquad (8)$$

Dividing the equation by $4xy\rho g$, adding and subtracting velocity head $\frac{\alpha w^2}{2g}$, where α is Coriolli coefficient and w is mean velocity of the flow (w=w_1=w_2), Eq. (8) may be rewritten in such shape

$$\left(\frac{p_1}{\rho g}+\frac{\alpha w_1^2}{2g}+l\right)-\left(\frac{p_2}{\rho g}+\frac{\alpha w_2^2}{2g}\right)=-\frac{l\mu}{\rho g}\left(\frac{\partial u}{x\partial x}+\frac{\partial u}{y\partial y}\right). \tag{9}$$

The left side of this equation consists of hydrodynamic heads of cross sections 1 and 2 $H_1=\frac{p_1}{\rho g}+\frac{\alpha w_1^2}{2g}+l$ and $H_2=\frac{p_2}{\rho g}+\frac{\alpha w_2^2}{2g}$ difference of which expresses friction loss of energy h_f. Thus from Eq. (9) follows

$$h_f=-\frac{l\mu}{\rho g}\left(\frac{\partial u}{x\partial x}+\frac{\partial u}{y\partial y}\right). \tag{10}$$

Taking into account that ratio of dynamic viscosity μ and density ρ equals to kinematical viscosity ν and denoting

$$\frac{h_f g}{l\nu}=k. \tag{11}$$

After calculations such formula of velocity distribution in cross section was solved

$$u=k\frac{x_0 y_0}{4}\sqrt{\left(1-\left(\frac{x}{x_0}\right)^2\right)\cdot\left(1-\left(\frac{y}{y_0}\right)^2\right)}. \tag{12}$$

When $x=x_o$ or (and) $y=y_o$ then $u=0$, what confirms rightness of Eq. (12). When $x=y=0$ then

$$u=u_{max}=k\frac{x_0 y_0}{4}. \tag{13}$$

It follows from parabolic character of velocity change in both (x and y) directions of cross section, that maximal velocity

$$u_{max}=2w. \tag{14}$$

This relationship allows to express local velocity u through mean velocity $w=Q/A$, where $A=4x_o y_o$ is area of cross section and to get more convenient expression of the formula

$$u=2w\sqrt{\left(1-\left(\frac{x}{x_0}\right)^2\right)\cdot\left(1-\left(\frac{y}{y_0}\right)^2\right)}. \tag{15}$$

It follows from (14) that mean velocity w equals to half of maximal local velocity u_{max}. Expressing it by (13), k – by (11) and solving received equation with respect to h_f friction loss we arrive to broadly known Darsy–Weisbach formula

$$h_f=\lambda\frac{l}{y_0}\frac{v^2}{2g}. \tag{16}$$

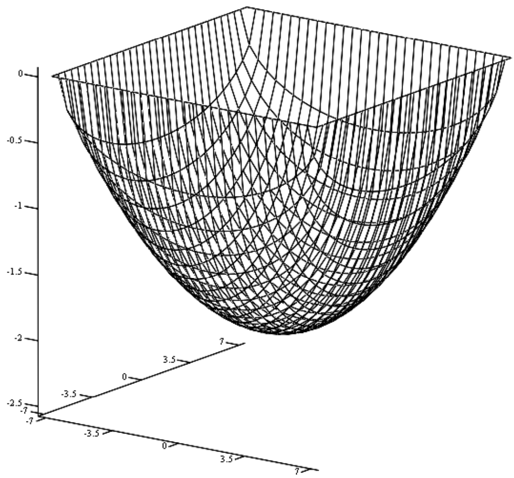

Fig. 2. Model of velocity distribution in cross section

Friction factor in it is being expressed by similar like for pipe of round cross section formula

$$\lambda = \frac{64}{Re} . \tag{17}$$

Reinols number is expressed here through dimension of cross section x_0 , i.e.

$$Re = \frac{w x_0}{\nu} . \tag{18}$$

Model of one–phase fluid (rate G=0,004 m^3/s w=0,204 m/s) comparative velocity $u_w = \frac{u}{w}$ distribution in rectangular (0.14x0.14 m^2) cross section shown in Fig. 2.

The equation (15) may be applied for solution of defined engineering problems, for example heat–mass transfer tasks. An application of this equation is complicated when the foam flow is used. Phenomena of foam flow and heat transfer to it are rather complex. Foam is two–phase flow which structure changes while it passes an obstacle, bubble divides into the parts and liquid drains down from flow. All these peculiarities make extremely complicated an application of analytic methods for their study. Thus experimental method of investigation was selected in this work.

Experimental Equipment

The experimental set–up consists of the following main parts: foam generation channel, gas and liquid control valves, gas and liquid flow meters, liquid storage reservoir, liquid level control reservoir, air fan, electric current transformer and stabilizer (Fig.3).

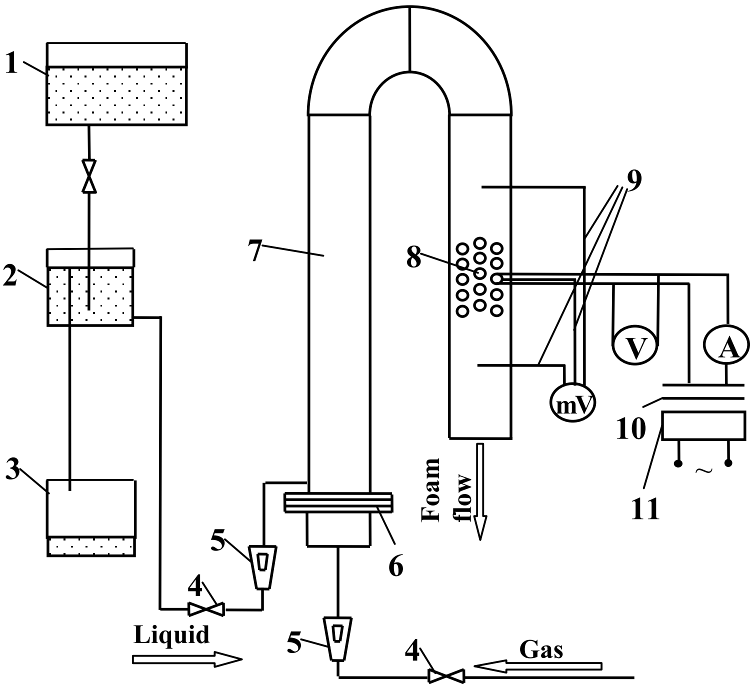

Fig. 3. Experimental set–up scheme: 1–liquid reservoir; 2– liquid level control reservoir; 3–liquid receiver; 4–gas and liquid control valves; 5–flow meter; 6–foam generation riddle; 7–channel; 8–tube bank; 9–thermocouples; 10–transformer; 11–stabilizer

The experimental channel had a riddle at the bottom of experimental part. For the experiments with downward foam flow the tube bank was located in the output part of channel. The whole experimental channel was made of glass in order to observe visually foam flow structure and the size of foam bubbles. The cross section of the channel had dimensions 0.14 x 0.14 m. The height of experimental channel was 1.8 m. Foam flow was generated on the riddle. The water solution of detergents was used in experiments. Concentration of detergents was kept constant and it was equal 0.5 %. Liquid was delivered from reservoir to the riddle from upper side; gas was supplied to the riddle from below. Foam flow was produced during gas and liquid contact. Liquid in experiment was used only once and it was not returned back to the reservoir.

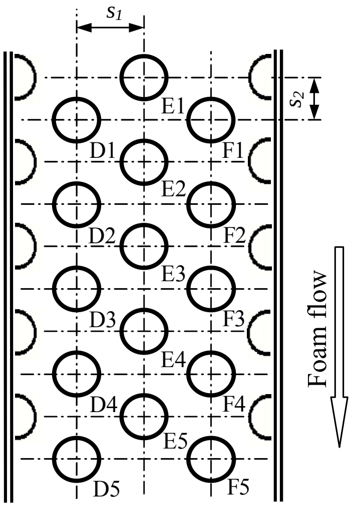

Fig. 4. Tube bank in foam flow

Schematic view of experimental section of the channel with tube bank is shown in Fig. 4. The bank of tubes consisted of three vertical rows with five tubes in each. Spacing among the centers of the tubes s_1=0.07 m, s_2=0.0175 m. All tubes had an external diameter of 0.02 m. The heated tube was made of copper and had an external diameter of 0.02 m also. The endings of the tube were sealed and insulated. The tube was heated electrically. An electric current value was measured by ammeter and voltage by voltmeter. The temperature of foam flow was measured by two calibrated thermocouples: one in front of the bank and one behind. The temperature of heated tube surface was measured by eight calibrated thermocouples. Six of them were placed around central part of heated tube and two of them were placed in both sides of the tube at 50 mm distance from the central part.

Methodology

The foam flow volumetric void fraction can be expressed by the equation

$$\beta = \frac{G_g}{G_g + G_l}, \tag{19}$$

were: G_g –gas volumetric flow rate, m^3/s; G_l – liquid volumetric flow rate, m^3/s.

The foam flow rate can be written as

$$G_f = G_g + G_l. \tag{20}$$

The temperature of the heated tube surface and the foam flow, electric current and voltage were measured and recorded during the experiments. The preliminary investigations showed that hydraulic and thermal regime stabilizes completely within 5 minutes after the change of experiment conditions. Therefore measurements were started not earlier than 5 minutes after adjustment of foam flow parameters. After registration of electric current and voltage the heat flux density on the tube surface q_w was calculated. After record of heated tube surface and foam flow temperatures, the difference of temperature $\overline{\Delta T}$ (between the mean temperatures of the foam flow $\overline{T}_f$ and tube surface $\overline{T}_w$) was calculated. The average heat transfer coefficient was calculated as

$$\overline{\alpha} = \frac{q_w}{\overline{\Delta T}}. \tag{21}$$

The Nusselt number was computed by formula

$$\overline{Nu}_f = \frac{\overline{\alpha} d}{\lambda_f}, \tag{22}$$

where λ_f is the thermal conductivity of the statically stable foams flow, W/(m·K), computed from the equation

$$\lambda_f = \beta \lambda_g + (1 - \beta)\lambda_l, \tag{23}$$

where: λ_g – gas thermal conductivity, W/(m·K); λ_l – liquid thermal conductivity, W/(m·K).

The gas Reynolds number of foam flow was computed by formula

$$\overline{Re}_g = \frac{G_g d}{A \nu_g}, \tag{24}$$

where: d – external diameter of tube, m; F – cross section area of experimental channel, m^2; ν_g – gas cinematic viscosity, m^2/s.

The liquid Reynolds number of foam flow was calculated as

$$\overline{Re}_l = \frac{G_l d}{F \nu_l}. \tag{25}$$

All experiments and measurements were repeated in order to avoid measurement errors and to increase reliability of investigation results.

During experimental investigations the parameters varied within limits: $\overline{T}_f \in [285;\ 291]$ K; $\overline{T}_w \in [296;\ 331]$ K; electric current of heated tube $I \in [8,4;\ 9,6]$ A; voltage of heated tube $U \in [11,3;\ 13,2]$ V; $q_w \in [10,74;\ 12,78]$ kW/m^2. Values of Nusselt and Reynolds numbers were: $\overline{Nu}_f \in [210;\ 850]$; $\overline{Re}_g \in [190;\ 440]$; $\overline{Re}_l \in [9,8;\ 22,3]$, when β=0,996; $\overline{Re}_l \in [7,5;\ 16,8]$, when β=0,997; $\overline{Re}_l \in [4,9;\ 10,9]$, when β=0,998.

The statistical analysis of data showed that all results are reliable, precision and reproducible.

Results

Investigation of tube heat transfer in the bank consisted of three series of experiments. The experiments were provided for different values of mean volumetric void fractions β=0.996, 0.997 and 0.998. Changes in foam volumetric void fraction influence not only on the magnitude of mean heat transfer intensity but also on the character of relationship. If foam is wetter (β=0.996), the intensity of heat transfer reacts more sensitively to the change in foam flow velocity and varies more intensively if compared with dryer foam (β=0.998). Velocity of generated foam flow was changed by changing flow rate of supplied air. When the air flow rate, corresponding to a particular velocity of flow and desired volumetric void fraction (β=0.996, 0.997 or 0.998) of foam are known, the required amount of solution can be calculated.

The experimental results show great dependencies of heat transfer intensity on foam flow gas and liquid velocities, volumetric void fraction β and tube position in the bank. Data of heat exchange intensity as function of $\overline{Re}_g$ for the first tube of the middle line of the tube bank in downward statically stable foam flow shown in Fig. 5.

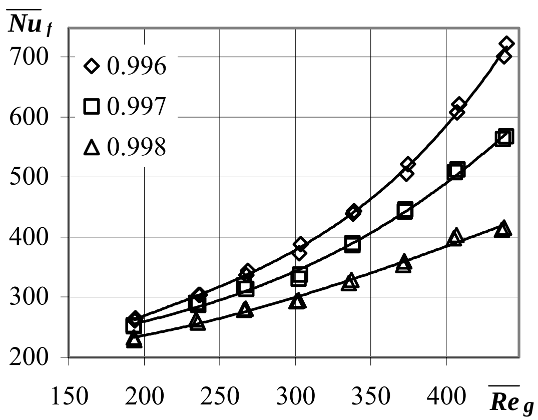

Fig. 5. Heat transfer intensity of the first tube (E1) in the middle line of the bank in downward foam flow; β=0.996, 0.997, 0.998

Heat transfer intensity of the first tube (E1) in the middle line of the bank increases with an increase of $\overline{Re}_g$ and decrease of β. When flow Reynolds number $\overline{Re}_g$ changes within the limits 190–440, heat exchange intensity ($\overline{Nu}_f$) of the first tube (E1) increases from 261 to 722, when foam volumetric void fraction β=0.996; from 253 to 568, when β=0.997 and from 233 to 416, when β=0.998 (Fig. 5).

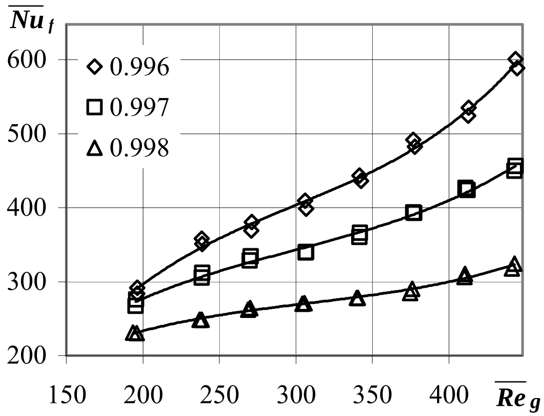

Fig. 6. Heat transfer intensity of the fourth tube (E4) in the middle line of the bank in downward foam flow; β=0.996, 0.997, 0.998

Heat transfer intensity of the fourth tube (E4) is worse than the first tube in the middle line of the tube bank (Fig. 6).

When flow Reynolds number $\overline{Re}_g$ changes within the limits 190–300 (when β=0.996), foam flow consists of large size bubbles (d_b= 8–14mm). Passing through the bank of tubes large bubbles of foam divide into smaller ones. The liquid drainage process becomes more intensive. As a result the heat transfer intensity increases. Therefore the fifth E5 tube has the best heat transfer rate to downward statically stable foam flow. Heat transfer intensity of the fourth E4, the third E3, the second E2 and the first E1 tubes grows worse in the same order (Fig. 7).

When flow Reynolds number $\overline{Re}_g$ exceeds 300 at β=0.996, heat transfer of the first tube E1 increases more intensive in comparison with other tubes. It can be explained by the fact that the foam of smaller bubbles starts forming up in the foam-generating channel and the structure of the foam flow varies not much passing the bank of tubes. When flow Reynolds number $\overline{Re}_g$ is near 430 the heat transfer intensity is best of the first tube, less of the second and so on according increasing number of them.

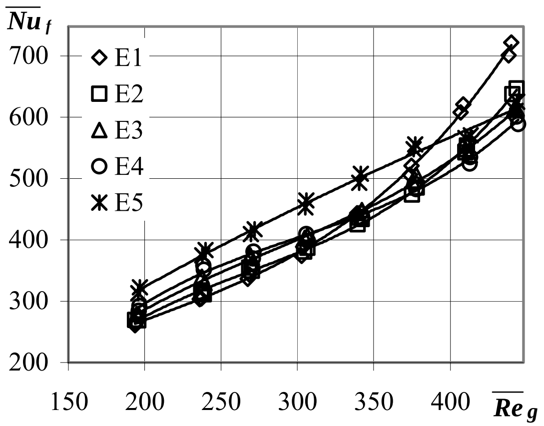

Fig. 7. Heat transfer intensity of the tubes in the middle line of the bank in downward foam flow, β=0.996

The heat transfers with more lines of tubes are mostly useful in energetic. It is difficult to define witch vertical line of tubes is the middle line and witch is the side–line. The position of tube in point of fluid flow is more important parameter. In the investigates the tubes of bank were sorted to groups in point of foam flow. There were five groups of tubes: first tubes, second tubes and so on. We calculated the average heat transfer intensity of each group of tubes. Such calculations allowed us to escape distribution of foam flow real void fraction in cross–section. Data of heat exchange intensity as function on $\overline{Re}_g$ for the fourth tubes of the tube bank at the point of foam flow in downward statically stable foam flow shown in Fig. 8. The heat transfer changing character is like the same of the fourth tube in the middle line while $\overline{Re}_g$<350. The increment of $\overline{Nu}_f$ subject to $\overline{Re}_g$ become worse when $\overline{Re}_g$ >350 in comparison with the fourth tube of the middle line (Fig. 8).

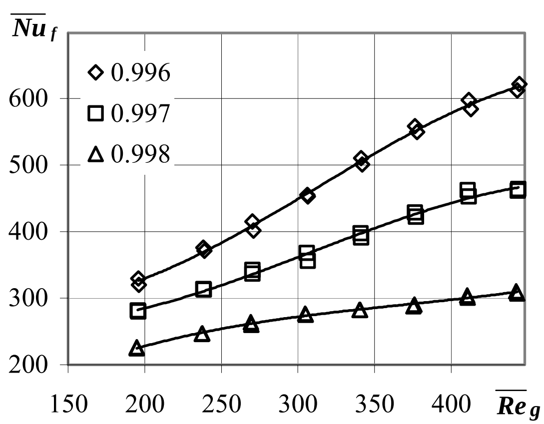

Fig. 8. Average heat transfer intensity of the fourth group of tubes in downward foam flow, β=0.996

The comparison of average heat transfer intensity of each group of tubes when β=0.996 shown in Fig. 9. When flow Reynolds number $\overline{Re}_g$ changes within the limits 190–430 the average heat transfer intensity is best of the first group of tubes, less of the second, the second less of the third and so on. The average heat transfer intensity of second, third and fourth groups of tubes are nearly independent on tube position in the line when downward statically stable foam flow at volumetric void fraction of β=0.996 passes the tube bank and Reynolds number $\overline{Re}_g$ changes within the limits 190–235. The average heat transfer intensity of mention groups of tubes more differentiate when $\overline{Re}_g$>300.

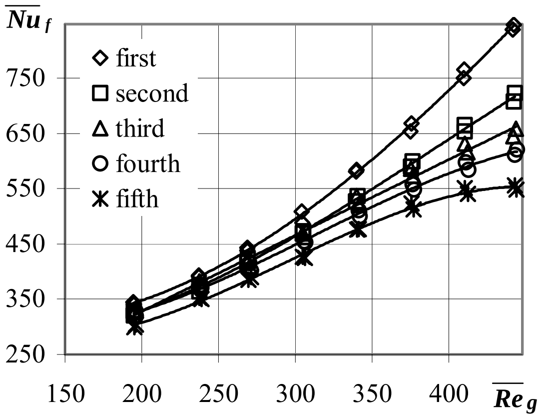

Fig. 9. Average heat transfer intensity of the groups of tubes in downward foam flow, β=0.996

During experiments, liquid Reynolds number $\overline{Re}_l$ of the driest foam (β=0.998) was changed within the following limits: $\overline{Re}_l \in [4.9; 10.9]$. Liquid Reynolds number $\overline{Re}_l$ of foam flow with volumetric void fraction β=0.997 was changed within such limits: $\overline{Re}_l \in [7.5; 16.8]$. For the wettest foam ($\beta$=0.996) $\overline{Re}_l$ was changed within limits $\overline{Re}_l \in [9.8; 22.3]$.

Data of heat exchange intensity as a function of $\overline{Re}_l$ for the first tube (E1) of the middle line of tube bank in downward statically stable foam flow is shown in Fig. 10.

Results of the investigations allow to make interesting observations. Value of $\overline{Nu}_f$, describing fixed intensity of heat exchange, can be reached by using foam flow with different volumetric void fraction and by selecting a certain velocity of the foam flow. As an example, let's assume that the intensity of heat exchange of the first bank tube (E1) must be equal $\overline{Nu}_f$=400. That intensity of heat exchange is achieved using driest foam (β=0.998) flow, where the foam flow velocity is such that the $\overline{Re_l}$ must be $\overline{Re_l}$=10.4. The same heat transfer intensity can be achieved using wetter foam (β=0.997) when $\overline{Re_l}$=13.1. For the wettest foam (β=0.996) flow such intensity of heat exchange is achieved when $\overline{Re_l}$=15.6.

Where foam liquid $\overline{Re_l}$ varies within the limits of $\overline{Re_l} \in$[9.8; 10.9], the different intensity of heat exchange can be ensured by using foam with different volumetric void fractions (Fig. 10).

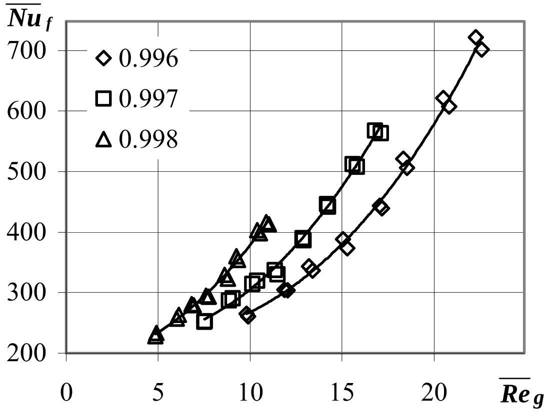

Fig. 10. Heat transfer intensity of the first tube (E1) in the middle line of the bank in downward foam flow; β=0.996, 0.997, 0.998

Data of heat exchange intensity as function of $\overline{Re_l}$ for the fourth tube of the middle line of tube bank in downward statically stable foam flow is shown in Fig. 11.

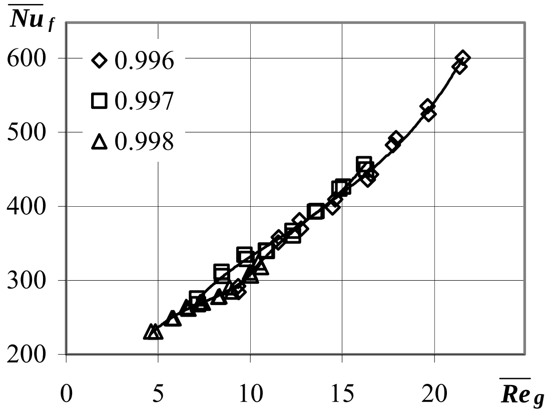

Fig. 11. Heat transfer intensity of the fourth tube (E4) in the middle line of the bank in downward foam flow; β=0.996, 0.997, 0.998

There is no significant difference of heat transfer intensity when foam flow of different volumetric void fraction (β=0.996, 0.997 and 0.998) is used. This notice is applicable to the dependencies of average heat transfer intensity of fourth group of tubes on $\overline{Re}_l$ (Fig. 12) too.

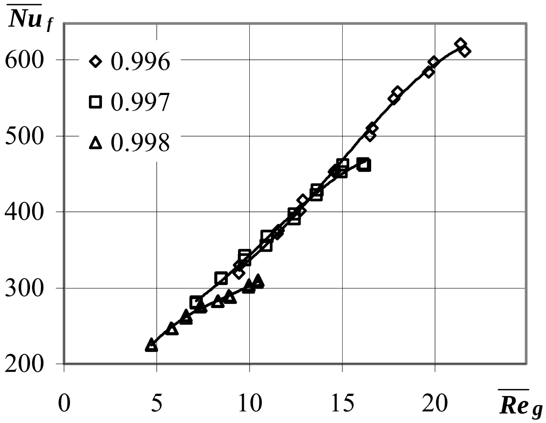

Fig. 12. Average heat transfer intensity of the fourth group of tubes in downward foam flow; β=0.996, 0.997, 0.998

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

$$\overline{Nu}_f = k\beta^n \overline{Re}_g^m \tag{26}$$

where $k=31.4e^{0.53\cdot P}$; $m=(219.65-219.3\cdot\beta)\cdot P^{121.55-122.3\cdot\beta}$; n=1250 for E1, n=1052 for E2, n=1042 for E3, n=1055 for E4, n=1150 for E5.

On average, for entire middle line in the bank k=142, n=1091, m=224.31-224.25β.

Equation (26) for tube groups of bank: $k=54.2e^{0,34\cdot P}$; $m=(233.48-233.25\cdot\beta)\cdot P^{82.84-83.4\cdot\beta}$; n=1183 for E1, n=1046 for E2, n=986 for E3, n=991 for E4, n=976 and k=233 for E5.

On average, for whole staggered tube bank k=134, n=1025, m=223.25-223.2β.

The experimental results were generalized by using the dependence of Nusselt criteria and liquid Reynolds number. This dependence within interval $\overline{Re}_l \in[9.8; 22.3]$, of downward foam flow at volumetric void fraction β=0.996; $\overline{Re}_l \in[7.5; 16.8]$, at β=0.997 and $\overline{Re}_l \in[4.9; 10.9]$, at β=0.998 can be expressed by equation (27)

$$\overline{Nu}_f = k_l \beta^{n_l} \overline{Re}_g^{m_l} \tag{27}$$

where m_l=(218.77-218.4·β)·$P^{118.64-119.4\cdot\beta}$; k_l=300 and n_l=751 for E1, k_l=297 and n_l=618 for E2, k_l=357 and n_l=582 for E3, k_l=356 and n_l=500 for E4, k_l=395 and n_l=473 for E5.

On average, for entire middle line in the bank k_l=330, n_l=574, m_l=219.82-219.75β.

Equation (27) for tube groups of bank: m_l=(234.04-233.8·β)·$P^{80-80.55\cdot\beta}$; k_l=303 and n_l=652 for E1, k_l=295 and n_l=555 for E2, k_l=291 and n_l=478 for E3, k_l=303 and n_l=460 for E4, k_l=288 and n_l=428 for E5.

On average, for whole staggered tube bank k_l=298, n_l=513, m_l=220.31-220.25β.

Conclusions

1. For the first time the heat transfer of the staggered tube bank to downward statically stable foam flow was investigated experimentally.
2. The phenomenon of "shadow" effect takes place in process of heat transfer of the investigated staggered tube bank to downward statically stable foam flow. In fact of that the heat transfer intensity of following tubes is less than of frontal tube.
3. The influence of heat transfer of second, third and fourth tubes to downward foam flow depend mainly on liquid velocity ($\overline{Re}_l$) of foam flow, the foam volumetric void fraction influences the process less.
4. The intensity of staggered tube bank heat transfer to statically stable foam flow depends on direction of flow: the process in drier foam (β=0,998) is more intensive in the first and less in the further rows of tubes in downward flow compared with that in upward foam flow.
5. The heat transfer between tube bank and foam flow is more intensive in downward foam flow when $\overline{Re}_g \in$[190; 300] and in upward foam flow when $\overline{Re}_g \in$[300; 440].
6. An analytical equation (Eq. 15) was obtained for computation of local velocity in downward laminar flow of one–phase fluid in channel of rectangular cross section. Statically stable foam flow is two–phase system with some specific peculiarities, therefore an application of mentioned equation is in perspective.
7. Experimental heat transfer results of the staggered tube bank to downward statically stable foam flow were summarized by criterion equations within interval $\overline{Re}_g \in$[190; 440] and β=0.996; 0.997; 0.998 or $\overline{Re}_l \in$[9.8; 22.3], at β=0.996; $\overline{Re}_l \in$[7.5; 16.8], at β=0.997 and $\overline{Re}_l \in$[4.9; 10.9], at β=0.998. The obtained generalized equations can be used for the designing of foam heat exchangers and calculating of heat transfer intensity of the separate tube of the bundle.

Nomenclature

$\overline{\alpha}$ – average coefficient of heat transfer, W/(m^2·K); β – volumetric void fraction; λ – thermal conductivity, W/(m·K); ν – cinematic viscosity, m^2/s; ρ – density, kg/m^3; A – cross section area of experimental channel, m^2; a, c, k, n, m – coefficients; d – external diameter of tube, m; d_b – diameter of foam bubble, m; G – volumetric flow rate, m^3/s; g – acceleration of gravity, m/s^2; l – distance between cross sections under consideration, m; P – tube position in the bank; q – heat flux density, W/m^2; r_o – pipe radius, m; r – distance from the pipe axis to the point under consideration, m; T – temperature, K; u, w – velocity m/s; y_0 – a half of split width, m; y – distance from center of split to researching point, m. ***Subscripts*** f – foam; g – gas; l – liquid.

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Tadas Ždankus was born on 1971 in Kaunas. In 1989 he graduated Kaunas 29th secondary school, studied in Kaunas University of Technology and obtained Bachelor degree on Electrical Engineering at Electrical Engineering and Control Systems Faculty. In 1995 Tadas Ždankus obtained Master of Science degree on Technologies Control Systems. Since 1995 he works in Kaunas Power Plant. In 2000 Tadas Ždankus started Doctoral studies on Power and Thermal Engineering at Kaunas University of Technology, Department of Thermal and Nuclear Energy.

Reziumė

Kasmet didėjant šiluminės energijos sąnaudoms vienu iš svarbiausių termoinžinerijos ir energetikos mokslo uždavinių tampa naujų šilumnešių, įgalinančių intensyvinti šilumos mainus, tuo pačiu sumažinant energetines bei medžiagines proceso sąnaudas, paieška. Šiuo požiūriu ypač perspektyviu šilumnešiu laikytinos dvifazės sistemos, tame tarpe, statiškai stabilios putos. Atlikti tyrimai parodė, kad putų srautui aušinant įkaitusius paviršius, pasiekiamos palyginti didelės šilumos atidavimo koeficiento reikšmės.

Pastaruoju metu jau yra tirtas pavienio vamzdžio, vamzdžių eilės ir šachmatinio vamzdžių pluošto šilumos atidavimas kylančiam statiškai stabilių putų srautui. Kadangi putos pasižymi tik joms būdingomis specifinėmis savybėmis, todėl nėra žinoma kaip keisis šilumos mainų intensyvumas pakeitus putų srauto kryptį. Kuriant modernius daugiaėjus ar besileidžiančio putų srauto šilumokaičius, žinoti vamzdžių pluošto šilumos mainus yra būtina ir kylančiame, ir besileidžiančiame putų sraute.

Pagrindiniai putų srauto parametrai lemiantys vamzdžių pluošto šilumos mainų intensyvumą yra putų srauto greitis ir dujingumas. Eksperimentiniai tyrimai atlikti vertikaliu kanalu besileidžiančiam statiškai stabilių putų srautui skersai aptekant šachmatinį vamzdžių pluoštą. Vamzdžių pluoštas buvo sudarytas iš trijų vertikalių eilių vamzdžių po penkis kiekvienoje. Statiškai stabilių putų srautas leidosi uždaru stačiakampio skerspjūvio kanalu. Analizuojant tėkmės greičio pasiskirstymo skerspjūvyje įtaką skirtingose skerspjūvio vietose patalpintų vamzdžių šilumos mainams, iškilo problema. Tėkmės greičio pasiskirstymo skerspjūvyje lygtys skirtos tik apvalaus skerspjūvio vamzdžiams arba plyšiams. Analitiniu būdu buvo gauta lygtis, aprašanti besileidžiančio vertikalaus laminarinio vienfazio srauto greičių pasiskirstymą uždaro stačiakampio kanalo skerspjūvyje. Kadangi statiškai stabilių putų srautas yra dvifazė sistema, kurioje vyksta visa eilė specifinių procesų, šios lygties pritaikymas komplikuotas.

Analizuojant gautus eksperimentinius duomenis, nustatyta, kad besileidžiančiam statiškai stabilių putų srautui aptekant tirtą šachmatinį vamzdžių pluoštą, pasireiškia, taip vadinamasis "šešėlio" efektas, dėl kurio srauto kryptimi pluošto tolesniųjų eilių vamzdžiai aušinami prasčiau už priekinius. Svarbią įtaką šilumos mainų intensyvumui daro ir skysčio drenažas iš putų. Šachmatinio vamzdžių pluošto šilumos atidavimo besileidžiančiam statiškai stabilių putų srautui skaičiavimui siūlomos kriterinės lygtys, esant putų dujingumui β=0,996, 0,997 ir 0,998, o putų srauto dujų Reinoldso skaičius $\overline{Re}_g \in [190;\ 440]$, arba pagal putų srauto skysčio $\overline{Re}_l \in [9,8;\ 22,3]$, kai β=0,996; $\overline{Re}_l \in [7,5;\ 16,8]$, kai β=0,997 ir $\overline{Re}_l \in [4,9;\ 10,9]$, kai β=0,998. Šios lygtys gali būti taikomos projektuojant daugiaėjus putų šilumokaičius, o taip pat apskaičiuojant bet kurio šachmatinio pluošto vamzdžio šilumos mainų intensyvumą.

UDK 536.24 (043)
SL 344. 2004-11-17. 1 leidyb. apsk.l. Tiražas 70 egz. Užsakymas 428.
Išleido leidykla „Technologija“, K. Donelaičio g. 73, 44029 Kaunas
Spausdino leidyklos „Technologija“ spaustuvė, Studentų g. 54, 51424 Kaunas