Factors influencing on the thermal flow with the cross-section of the corridor tube bundle in low-frequency non-symmetric pulsations

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Abstract. Factors influencing on the thermal flow of cylinder of corridor tube bundle in pulsating flow of fluid was analyzed numerically. The range of Reynolds numbers was Re <1000 pulsation frequency f = 0.5, Hz, relative amplitude A / D = 1.25. It is shown that the increase in the thermal flow in tube bundle when pulsations are applied to the flow of fluid is mainly due to the convective component. The change in the temperature pressure and turbulence contributes less to the increase in the thermal flow.

Nowadays the study of heat transfer in pulsating flows in the elements of heat exchange equipment and in other industrial equipment is of great practical importance [1-3]. In past studies [4-6], an experimental and numerical method was used to study the heat transfer in the case of low-frequency, nonsymmetric pulsations of the fluid flow in a corridor tube bundle. It is established that when external pulsations are applied to the fluid flow, an increase in the external heat transfer of the corridor tube bundle takes place.

In this paper, by means of numerical simulation, it is planned to analyze the factors influencing the change in the thermal flow in the low-frequency asymmetric pulsations of the fluid flow in the corridor tube bundle for the Reynolds number range Re <1000 of the pulsation frequency f = 0.5, Hz, relative amplitude A/D = 1, 25.

Mathematical model

A numerical method was used to describe the heat transfer in a tube bundle during pulsating flows. The computational region of the model (Figure 1) consisted of a two-dimensional bundle of tubes arranged in a corridor order. The diameter of the tubes was D = 0.01 m, the pitch of the tubes was horizontal and vertical $S_{1,2}$ = 0.013 m. The number of tubes in the direction of the liquid was z = 10.

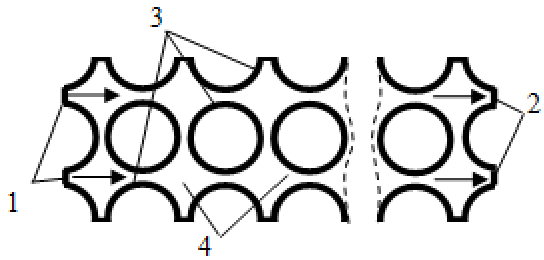

Fig. 1. Calculated region of the model: 1 – fluid inlet; 2 - fluid outlet; 3 - wall; 4 - annular space

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The incompressible fluid flow was described by the system of Navier-Stokes equations (Reynolds averaged Navier-Stokes RANS)

$$\frac{\partial \bar{u}_j}{\partial x_j} = 0 \qquad (1),$$

$$\frac{\partial \rho \bar{u}_i}{\partial t} + \frac{\partial \rho \bar{u}_j \bar{u}_i}{\partial x_j} = -\frac{\partial \bar{p}}{\partial x_i} + \frac{\partial}{\partial x_i}\left((\mu + \mu_t)\left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) \right), \quad (2)$$

where $\bar{u}_i$, $\bar{u}_j$ – averaged speed components; ρ – fluid density; μ – dynamic viscosity; $\bar{p}$ – pressure; μ_t – turbulent viscosity; (i = 1, 2 j, = 1, 2).

Heat transfer was described by the equation (Fourier-Kirchhoff)

$$c_p\left(\frac{\partial}{\partial t}\rho \bar{T} + \frac{\partial}{\partial x_j}(\rho u_i \bar{T}) \right) = \frac{\partial}{\partial x_j}\left[(\lambda + \lambda_t)\frac{\partial \bar{T}}{\partial x_j} \right], \qquad (3)$$

where c_p – heat capacity of fluid; $\lambda_t = c_p \mu_t / \Pr_t$ – turbulent thermal conductivity; $\Pr_t = 0{,}85$ – turbulent Prandtl number; $\bar{T}$ – temperature.

Turbulent viscosity was calculated using the Spalart-Allmaras turbulence model (SARC) [7]

For stationary flow at the inlet in the calculated region (Fig. 1), the velocity u_n depending on the desired Reynolds numbers $\mathrm{Re} = Du_n / \nu$ (4), where ν – kinematic viscosity of fluid, m^2/sec; where u_n – the velocity of the fluid averaged over the narrowest cross-section of the tube shell space, m/sec; D – outer diameter of the tube in the bundle, m. To create a nonstationary flow in the tube at the input, a velocity profile was defined as a function of time $u_n(t)$ With the necessary frequency f and relative amplitude A/D pulsations, where A – displacement of fluid particle backward, m in the narrowest section of the annular space of the bundle. The type of the velocity profile was set in a manner similar to the profile obtained experimentally in the work. [4] In Fig. 2, as an example, the velocity profile for the conditions f = 0.5 Hz, A/D = 1.25, Re = 100 is given. The averaged velocity $\langle u_n \rangle$ over the pulsation period T_p corresponds to the stationary flow velocity, i.e. $\langle u_n \rangle = u_n$.

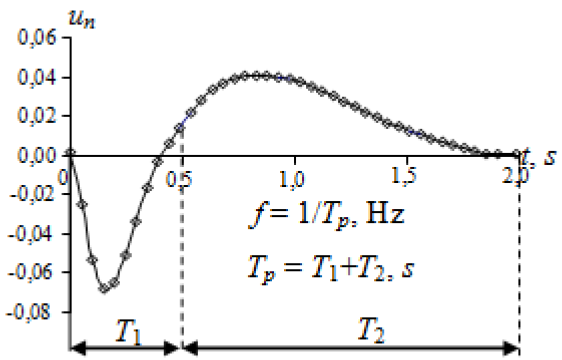

Fig. 2. The velocity profile for a nonstationary flow for f = 0,5 Hz A/D=1,25; Re = 100

On the walls of the tubes the following boundary conditions were set: fluid temperature T_{wall} = 315 K; Speed u = 0, m / sec; pressure $\partial P / \partial n = 0$ and absolute roughness Δ = 0, m.

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On the area between the tube halves, from above and below, the constancy of speed $\partial u / \partial n = 0$, pressure $\partial P / \partial n = 0$, and temperature $\partial T / \partial n = 0$ was determined. The pressure at the exit from the bundle was constant P = 101325 Pa.

ANSYS Fluent 14.0 was used to solve the system of equations (1-3). The system of equations was solved by the finite volume method (FVM) for which the calculation area was divided into 64856 reference volumes. The average volume size was 2×1^{-4} m from the condition for the dimensionless coordinate in the boundary layer $y^+ = 1$. [7] The time step was 0.01 sec.

The thermal flow q was averaged over the wall surface of the fifth cylinder in the bundle and in time (for r = D / 2, depending on the angular mark - φ (Figure 3))

$$q_{r,\varphi} = \frac{\int_{\varphi_1}^{\varphi_2} q d\varphi}{\pi D/16} \tag{4}$$

The mean $q_{r,\varphi}$ on time was carried out for one period of pulsations T_p

$$q_{t,r,\varphi} = \frac{\int_0^{T_\Pi} q_{r,\varphi} dt}{T_\Pi}. \tag{5}$$

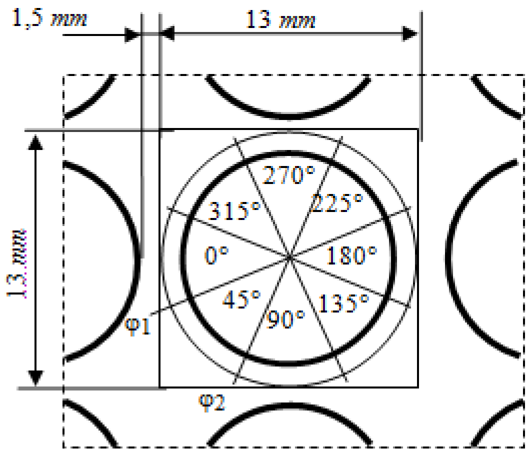

Fig. 3. The layout of the control cylinder and the angle mark φ

Table 1. The angle mark

	0°	4 5°	90 °	13 5°	18 0°	22 5°	27 0°	31 5°
2	3 37,5	2 2,5°	67, 5°	11 2,5°	15 7,5°	20 2,5°	24 7,5°	29 2,5
1	2 2,5°	6 7,5°	11 2,5°	15 7,5°	20 2,5°	24 7,5°	29 2,5°	33 7,5°

The temperature of the fluid was averaged over the region around the cylinder for $x = [0, 2L + D]$, $y = [0, 2L + D]$ (Fig. 3)

$$T = \frac{\int_{y_1}^{y_2}\int_{x_1}^{x_2} T dx dy}{(2L+D)^2 - \pi D^2/4}. \tag{6}$$

To averaging over time T, we used the following expression

$$T_t = \frac{\int_0^{T_\Pi} T dt}{T_\Pi}. \tag{7}$$

The fluid velocity module $\vec{u}$, $u_i, i = 1, 2$ was as follows,

$$U = |\vec{u}| = \sqrt{\sum_{i=1}^{2} u_i^2}. \tag{8}$$

The velocity module averaged over time U_t, corresponded to a period of time t = [0, T_p].

The velocity module was also averaged over the region around the fifth cylinder in the beam for r = [D/2, D/2+L] and was according to the formula

$$U_{r,\varphi} = \frac{\int_{r_1}^{r_2}\int_{\varphi_1}^{\varphi_2} U d\varphi dr}{S(\varphi, r)}. \tag{9}$$

Here $S(\varphi, r)$ area of averaging, φ – «angle mark».

To averaging over time $U_{r,\varphi}$, we used the following expression

$$U_{t,r,\varphi} = \frac{\int_0^{T_p} U_{r,\varphi} dt}{T_p} \tag{10}$$

Turbelent heat transfer λ_t averaged over the region around the cylinder at r = [D/2, D/2+L]

$$\lambda_{t_{r,\varphi}} = \frac{\int_{r_1}^{r_2}\int_{\varphi_1}^{\varphi_2} \lambda_t d\varphi dr}{S(\varphi, r)}. \tag{11}$$

Here $S(\varphi, r)$ area of averaging, φ – « angle mark ».

To averaging over time $\lambda_{t_{r,\varphi}}$, we used the following expression

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$$\lambda_{t_{t,r,\varphi}} = \frac{\int_0^{T_p} \lambda_{t_{r,\varphi}} dt}{T_p} . \tag{12}$$

In order to better understand the processes taking place in the pulsating current, and also to identify the causes that lead to an increase or decrease in the thermal flow, in a nonstationary flow, in comparison with the stationary one, the change in a number of forces (quantities) that contribute to the change in the thermal flow has been studied. This is the contribution of the convective component, turbulent, the change in the thermal flow due to the thermal conductivity and the average driving force of the process - the temperature pressure.

The convective component was estimated by the velocity of the flow taken modulo $U_{r,\varphi}$.

The turbulent component and the change in the thermal flow due to thermal conductivity were estimated by the effective thermal conductivity

$$\lambda_{e_{r,\varphi}} = \lambda + \lambda_{t_{r,\varphi}} , \tag{13}$$

$$\delta\lambda_{e_{r,\varphi}} = \delta\lambda + \delta\lambda_{t_{r,\varphi}} . \tag{14}$$

Since the fluctuations in thermal conductivity $\delta\lambda$ were of the order of 0.01%, then $\delta\lambda_{e_{r,\varphi}} \approx \delta\lambda_{t_{r,\varphi}}$, in fact, the turbulent component $\delta\lambda_{e_t}$ was actually estimated as a gain. The temperature pressure ΔT was estimated as follows

$$\Delta T = T_{wall} - T \tag{15}$$

The dimensions of the area for determining $U_{r,\varphi}$, $\lambda_{e_{r,\varphi}}$,, as well as the location of their local zones along the perimeter of the cylinder at different angle marks φ are shown in Fig. 3. The values $U_{r,\varphi}$, $\lambda_{e_{r,\varphi}}$ were averaged over the entire area around the cylinder (from 0° to 360°, and also by one eighth inner region of the ring, depending on the angle mark φ, which took values from 0° to 315° (Table 1). φ= 0°, the frontal part of the cylinder is assumed to be perpendicular to the flow of liquid circulating through the bundle. The increase $\delta\Theta$ in the sought value in the pulsating flow Θ_p compared with the stationary flow Θ_{st} was estimated as follows

$$\delta\Theta = \left(\Theta_p / \Theta_{st} - 1\right) \cdot 100 \text{ \%,}. \tag{16}$$

Further, for the sake of convenience, we omit the subscript *r* in (5), (6), (10) - (15), omit the subscript φ for the averaged quantities over the entire region around the cylinder in the same formulas.

When observing the change in the above-listed quantities λ_e, ΔT, U, together with the thermal flow density q in the nonstationary flow as compared to the stationary one, one can indirectly answer the question at the expense of which value the increase or decrease in the thermal flow in the liquid flow occurs when pulsations are applied, and also trace how pulsations affect λ_e, ΔT, U, while it remains unknown what contribution each of these quantities makes to the change in the thermal flow.

The contribution of each of the above values to the change in the thermal flow is estimated on the basis of the Newton-Richman law and the Peakle Pe criterion.

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From the Newton-Richmann law it is known that $q = \alpha\Delta T$ (17). Here heat transfer α can be represented as the product of heat transfer due to convection α_c and the sum of heat transfer due to turbulent thermal diffusivity $a_t = \lambda_t /(c_p\rho)$ (18) and the usual temperature conductivity *a*

$$\alpha = \alpha_c \cdot (a_t + a). \quad (19)$$

Where sum $a_t + a = a_e = \lambda_e /(c_p\rho)$ (20) – effective thermal conductivity.

The contribution ratio α_c and a_e can be expressed in terms of the effective Peakle number Pe_e

$$\alpha_c / a_e \approx \mathrm{Pe}_e = U \cdot L / a_e, \quad (21)$$

where *L* =1,5 – ring width, mm around the cylinder (Fig.3). Let's find a_e from (22)

$$a_e = \alpha_c / \mathrm{Pe}_e, \quad (22)$$

taking this into account, we rewrite equation (20) in the following form

$$\alpha = (\alpha_c \cdot \alpha_c)/\mathrm{Pe}_e = 2\alpha_c / \mathrm{Pe}_e, \quad (23)$$

Find $\alpha_c = \sqrt{\alpha \cdot \mathrm{Pe}_e}$. (24)

The contribution of each quantity α_c, a_e, and ΔT in the change in the heat flux was estimated among themselves as a percentage, where the sum of these three values was 100%, for this purpose the rule of logarithms

$$\ln(q) = \ln(\alpha_c \cdot a_e \cdot \Delta T) = \ln(\alpha_c) + \ln(a_e) + \ln(\Delta T); \quad (25)$$

$$\ln(\alpha_c)/\ln(q)\cdot 100 + \ln(a_e)/\ln(q)\cdot 100 + \ln(\Delta T)/\ln(q)\cdot 100 = 100, \%. \quad (26)$$

Results and its discussion

Below are the results of numerical simulation for *f* = 0,5 Hz, *A*/*D* = 1,25 and Re < 1000 (Fig. 4).

From Fig. 4 it can be judged how changes in the averaged temperature averaged over the region around the cylinder and in time $\delta\Delta T_t$ convective δU_t and turbulent component $\delta\lambda_{e_t}$ can affect the change in the thermal flow δq_t at different numbers Re. On Fig.4 it is shown that δq_t depends more on δU_t and $\delta\Delta T_t$. Increase δq_t grows from the minimum value 50 % to maximum 125 %, in the range of numbers Re from 100 to 300, while an increase of the growth of temperature pressure is observed $\delta\Delta T_t$. Further in the range of numbers Re from 300 to 900 occurs the decline δq_t together with $\delta\Delta T_t$ and δU_t. The value of the increase in effective thermal conductivity $\delta\lambda_{e_t}$ varies the opposite of the speed increase δU_t. If $\delta\lambda_{e_t}$ grows with increase of number Re then δU_t declines.

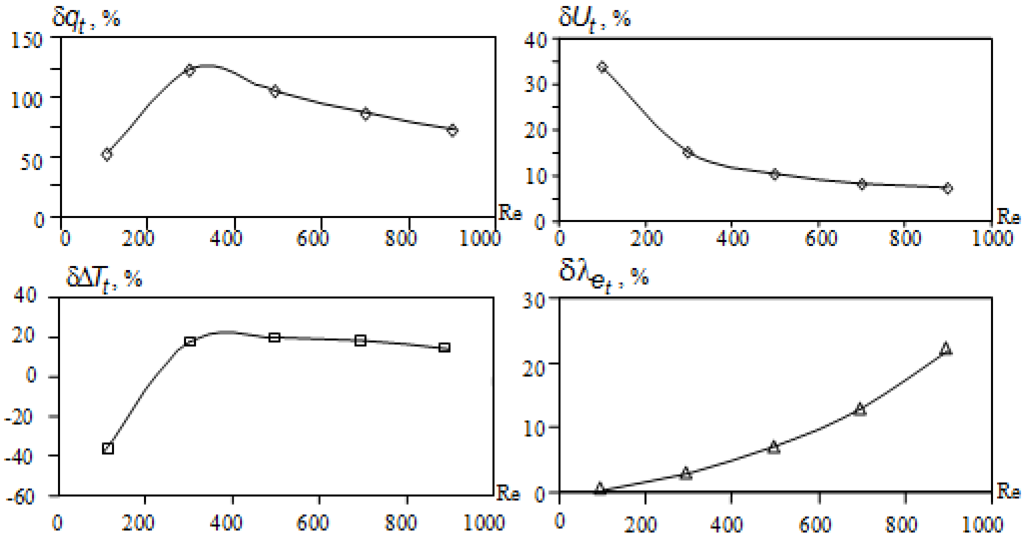

Fig. 4. Relations δq_t, δU_t, $\delta\Delta T_t$, $\delta\lambda_{e_t}$ from Re for f = 0,5 Hz, A/D = 1,25

Below are the factors contributing to the change in thermal flow as a percentage, with a total of 100% (see formula 26). According to Fig. 5 that in both steady-state and pulsating flows, a significant contribution to q is exerted by the convective component α_c. The part of α_c among the factors , influencing the change in the thermal flow lies in the range α_c = [59÷69] %, part of ΔT = [16÷23] %, a_e = [11÷20] %. In stationary flow, with increasing of numbers Re the part of α_c increases, but a_e decreases, with an almost unchanged part of ΔT. In the pulsating flow with increasing numbers Re to 500, the fractions a_e and α_c decrease, but ΔT increases, with an increase to 900, the proportion of shares remains the same.

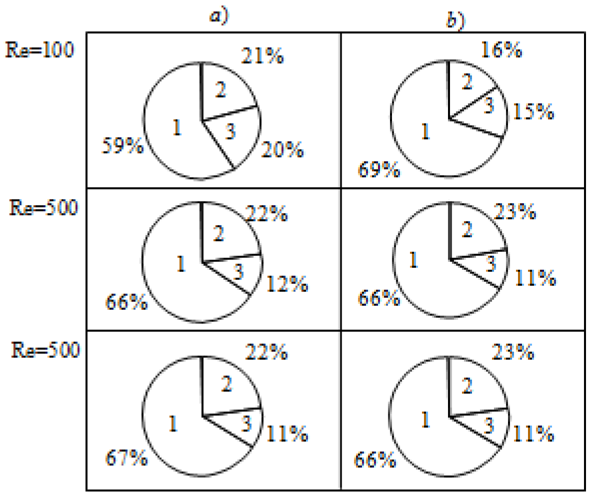

Fig. 5. Factors affecting the thermal flow a in percentage: a) stationary flow; b) pulsating current,1 – α_c ;2 – ΔT; 3 – a_e

Consider how the values of local increments vary $\delta q_{t,\varphi}$, $\delta U_{t,\varphi}$, $\delta\lambda_{e_{t,\varphi}}$ (Fig. 6) on the perimeter of the cylinder, depending on the numbers Re. Maximum $\delta q_{t,\varphi}$ and $\delta U_{t,\varphi}$ in the entire range of Re observed in the frontal and aft part of the cylinder, at φ = 0°, and 180°, which is quite obvious, since it is in these zones in the steady flow that stagnant zones with poor circulation of the fluid arise. Minimum values $\delta q_{t,\varphi}$ and $\delta U_{t,\varphi}$ fixed at $\varphi = 90°$. Minimal value of increase $\delta\lambda_{e_{t,\varphi}}$ occurs at $\varphi = 0°$ and 180°, in the range of numbers Re from 100 to 500 growth $\delta\lambda_{e_{t,\varphi}}$ was negative. Maximum value $\delta\lambda_{e_{t,\varphi}}$ fixed at $\varphi = 45°$. The change of numbers Re differently influence on the growth of $\delta q_{t,\varphi}$, $\delta U_{t,\varphi}$ and $\delta\lambda_{e_{t,\varphi}}$. The increase of values $\delta q_{t,\varphi}$ is observed throughout the perimeter of the cylinder with increasing Re from 100 to 300, then there is a decline up to Re ≈ 900. The growth of $\delta U_{t,\varphi}$ decreases with increasing Re by all perimeter of cylinder. The growth of $\delta\lambda_{e_{t,\varphi}}$ increases in all range of numbers Re and all φ.

By Fig. 6 it could be done the conclusion that when numbers Re from 100 to 900 and frequency $f = 0{,}5$ Hz, $A/D = 1{,}25$ large growth of $\delta q_{t,\varphi}$ when $\varphi = 0°$ and 180° connected with large growth of $\delta U_{t,\varphi}$ at these ranges.

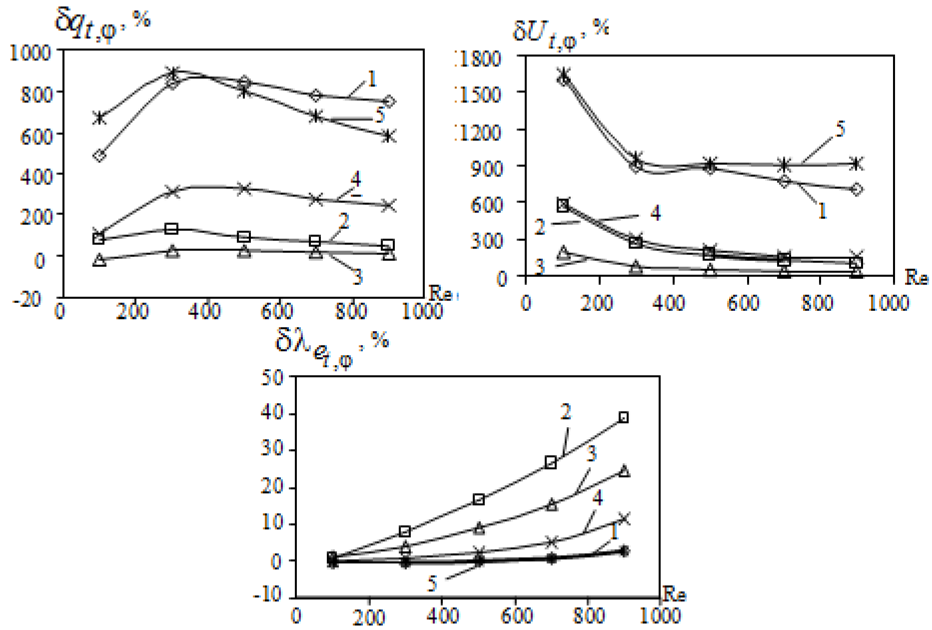

Fig. 6. The growth of $\delta q_{t,\varphi}$, $\delta\lambda_{e_{t,\varphi}}$ and $\delta U_{t,\varphi}$ by perimeter of cylinder, $f = 0{,}5$ Hz,

$A/D = 1{,}25$: 1 – φ = 0°; 2 – φ = 45°; 3 – φ = 90°; 4 – φ = 135°; 5 – φ = 180°

In Fig. 7, 8 there are shown the dependences of the increments δq, δU, $\delta\Delta T$, $\delta\lambda_e$ from Re for one period of pulsations T_p. Growth δU is observed at the initial moment of time from 0 to 0,1sec, then for some time the values δU fluctuate up to the time $t = 1$ sec, after which a decrease to $t = 2$ sec occurs. The maximum increase for the entire period of time Tp is fixed for Re≈100. The maximum increase in the increase in the effective thermal conductivity $\delta\lambda_e$ is observed in the time t from 0.9 to 1.2 sec. At Re numbers of about 100 and 500, the increase in the temperature pressure $\delta\Delta T$ sags in a period of time from t = 0.1 sec to 0.9 sec, at Re about 900 from t = 0.1 sec to 0.7 sec, then the values $\delta\Delta T$ for all Re rise slightly higher than the corresponding instant t = 0 sec, after which the curves decrease to 2 sec. At Re numbers of about 500 and 900, there are small periods of time of not more than 0.1 sec, at which $\delta\Delta T$ have negative values. When Re ≈ 100 the gain $\delta\Delta T$

was negative for the whole period of pulsation T_p. δq begin to grow from t = 0.1 sec to 0.3 sec, at Re numbers of about 500 and 900, then it starts to decrease down to t =2 sec.

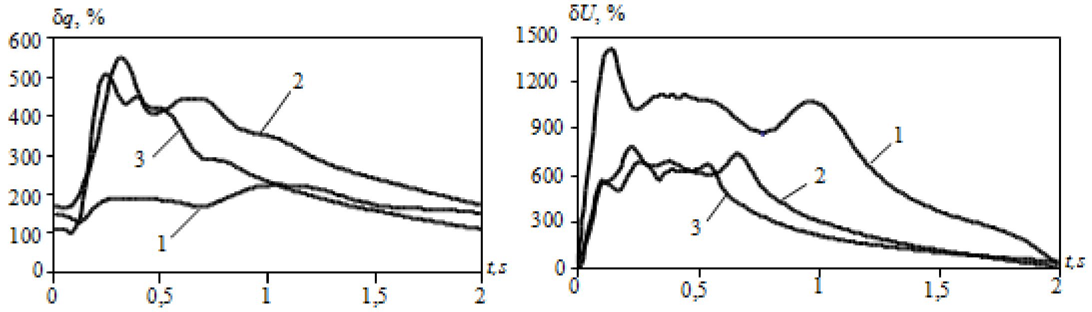

Fig. 7. Dependencies δq and δU from t at f = 0,5 Hz, A/D = 1,25

1 – Re = 100; 2 – Re = 500; 3 – Re = 900

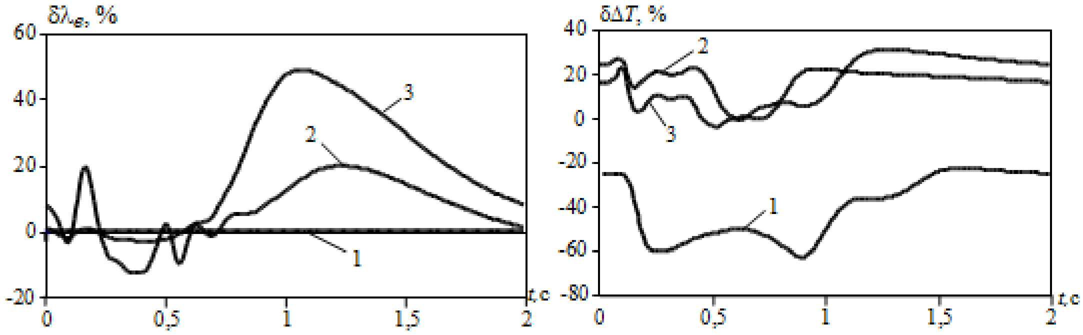

Fig. 8. Dependencies $\delta\lambda_e$ and $\delta\Delta T$ from t at f = 0,5 Hz, A/D = 1,25:

1 – Re = 100; 2 – Re = 500; 3 – Re = 900

Conclusion

The results of a numerical experiment show that the change in the thermal flow in tube bundle during pulsating flows in tube bundles is mainly due to the convective component U. The temperature pressure ΔT and the turbulent component λ_e as a whole contribute less to the change in the thermal flow. In this case, there are pulsation regimes for which a decrease in the temperature pressure ΔT and the turbulent component λ_e are observed in comparison with the stationary flow. An increase in the Reynolds number Re leads to a decrease in the convective component U, in comparison with the stationary flow. This can be attributed to the fact that as the Re numbers increase, the ratio of the pulsation velocity to the stationary one decreases, which is expressed by the ratio fA/u_n, where u_n is the velocity in the stationary flow, which increases with the increase in the numbers Re. λ_e with increasing Reynolds numbers Re increases, while the temperature pressure ΔT can both decrease and increase with the Reynolds number Re.

An increase in the local values of the convective component U_φ along the perimeter of the cylinder is mainly observed in the frontal and aft parts of the tube. This is due to the fact that in these areas with stationary flow there are stagnant zones with poor circulation of the fluid.

An increase in the local turbulent component λ_{e_φ} along the perimeter of the cylinder during pulsating currents is observed in the places where the jet collides with the surface of the cylinder, i.e. in the lower part of the cylinder and the area between the front and bottom. Values $\lambda_{e,\varphi}$ are significantly lower in the frontal and aft part of the cylinder. Such a distribution $\lambda_{e,\varphi}$ along the perimeter of the cylinder in the nonstationary flow is probably due to the fact that for stationary flow in these regions the flow has a lower turbulence, with smaller separation zones than in the aft part of the cylinder, where a pronounced vortex zone is formed.

Considering the change in the convective component in time U, we can distinguish two lengths of time at which growth U occurs. The first is associated with the acceleration of the flow, with the reverse flow of the fluid flow in the bundle of tubes, the second is associated with the acceleration of the fluid when it moves in the forward direction. For instantaneous values ΔT of the temperature pressure in the time interval between the segments at which growth U occurs, a decrease ΔT is observed. The periods of time at which the growth of the instantaneous values λ_e of the turbulent component arise, can vary significantly, depending on Re. In this case, the increase in the thermal flow during the pulsation period is mainly observed, for the same periods of time at which the convective component U increases.

When considering the factors influencing the thermal flow in percentage ratio as a result of mathematical modeling, it was found that a greater contribution to the change in the thermal flow for tubes bundle is about 59-69% for both a steady flow and a non-steady flow of fluid in tube bundle has a convective component, the temperature pressure dT in 16-23% and about 11-20% is the contribution of the effective thermal conductivity a_e.

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