ABSTRACT

of the dissertation for the academic degree of Doctor of Philosophy / PhD on specialty 6D072400 – Technological machines and equipment

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Development and design of mixing and surface heat exchangers with turbulators of heat carriers

Actuality of the dissertation research. For industrial enterprises, one of the main tasks is to improve the efficiency of heat exchangers. To solve this problem is possible either by applying already tested methods, or by searching for new opportunities, facilitating the intensification of heat exchange in heat exchange equipment.

It is known that the intention to intensify the ongoing processes is related to the increase in energy costs and, from an economic point of view, it is not always feasible to use maximum characteristics of high-speed operating modes. In relation to heat exchangers, the criteria for creating effective structures also include the technological effectiveness of their assembly, the reduction of material consumption and the increase (preservation) of strength characteristics.

As shown by numerous data, of all known methods of heat transfer intensification in tubes, the greatest attention is paid to the effective and technologically feasible method of artificial flow turbulization by ring diaphragms. The essence of the proposed method is as follows. On the outer surface of the tube the periodically arranged annular grooves are applied by knurling. In this case, on the inner side of the tube there are formed annular diaphragms of smooth configuration. The annular diaphragms and grooves are turbulizing the flow in the wall layer and ensuring the intensification of heat transfer outside and inside the tubes. In this case the outer diameter of tubes is not increased, which makes it possible to use these tubes in tight bundles and not to change the existing heat exchanger assembly technology.

The knurled tubes study results showed the increased efficiency of heat exchange process at low and medium Reynolds numbers. There are no results of studies with using the flow core turbulators.

For this reason, the studies of convective heat exchange in channels and tubes with regularly located intensifiers of boundary layers and a flow core at medium and above medium Reynolds numbers are actual and have both scientific and wide practical interest.

Subject of studies: hydrodynamic laws and heat exchange processes.

Purpose of work: development of scientific bases for heat exchange processes in mixing and surface heat exchangers, creation of scientifically grounded methods of their design and engineering, testing of the obtained results in pilot-industrial conditions with their further implementation in the industry.

In accordance with the set goal, the following **problems** were being solved:

- ascertainment of hydrodynamic laws and heat exchange characteristics, when the heat carrier is flowing in a circular tube with transversely annular protrusions and flow core turbulators;
- simulation of the process of vortex streamlining of annular protrusions in tubes and flow core turbulators;
- obtaining of the calculated dependences of resistance and heat transfer coefficients in tubes with transversely annular protrusions and flow core turbulators;
- experimental study of hydrodynamic characteristics and heat exchange parameters in the internal and external streamlining of tubular elements of the mixing heat exchanger with changing the operating and design parameters, and obtaining of calculated dependences;
- experimental study of hydrodynamic characteristics and heat exchange parameters in the external streamlining of tubular elements of the surface heat exchanger with changing the operating parameters, and obtaining of calculated dependences;
- development of the scientifically grounded engineering design procedure for mixing and surface type heat exchangers with tubular packing and the recommendations for design and operation;
- industrial testing of study results.

Scientific novelty of the study:

- there is proposed a physical model of vortex interaction in tubes during streamlining of peripheral annular protrusions and spherical flow core turbulators at medium Reynolds numbers with determination of the coefficients, taking into account the degree of vortex interaction behind the ring diaphragms, spherical flow core turbulators and their total interaction;
- there are derived the equations for calculation of heat transfer intensity and heat carrier hydraulic resistance coefficients in tubes with ring diaphragms and spherical flow core turbulators at medium and above medium Reynolds numbers;
- there are derived the equations for calculation of resistance coefficients, hydraulic resistance and the retained liquid quantity, which are taking into account the mechanisms of vortex interaction in the volume of a regular tubular packing of the mixing heat exchanger;
- based on the use of a dissipative approach, there is derived an equation for determination of mass transfer coefficients in the gas phase and, based on the analogy of heat and mass transfer processes, there is derived an equation for calculation of heat transfer coefficients of the mixing heat exchanger;
- there are derived the equations for calculation of hydraulic resistance and heat transfer coefficients, which are taking into account the mechanisms of vortex interaction in the volume of a tube bundle of the surface heat exchanger.

The theoretical importance of the study is that on the basis of theoretical and experimental studies of vortex flow interaction mechanisms there is scientifically proven the design procedure for mixing and surface heat exchangers.

Practical value. There is developed the structure of heat exchange tubes with knurling and flow core turbulators, protected by the innovation patent RK №

28151, as well as the structure of a heat and mass transfer apparatus, protected by the innovatiion patent RK №30217.

There are developed the design procedures, recommendations for designing and operation of mixing and surface heat exchangers.

Publications on the research topic. On the subject of dissertation there were published 25 articles, including 15 articles in the materials of international conferences, 2 articles in the edition, included in the international database of scientific journals Scopus, 6 articles in journals, recommended by the Committee for Control in Education and Science of the Ministry of Education and Science of the Republic of Kazakhstan, and there was received 2 innovation patents of the Republic of Kazakhstan. The content of the articles covers the main content of the dissertation.

Introduction gives the assessment of current state of the scientific problem to be solved, the basis and the initial data for topic development, the substantiation of the necessity to carry out the research work, the information about the planned scientific and technical level of development and the methodological support of the dissertation, shows the actuality and novelty of the topic, the relationship of this work with other research works, the purpose, object, subject and research objectives, the methodological basis, the provisions, submitted for defense of the dissertation, the practical value and approbation of practical results.

Section 1 gives an up-to-date presentation of the issue on heat exchange in mixing and surface heat exchangers. Along with this, the analysis of work and methods of calculation of the main hydrodynamic and heat-exchange characteristics of the equipment is carried out, the research objectives are set.

Section 2 describes the experimental unit for study of the convective heat exchange and the hydrodynamic characteristics of a tube-in-tube heat exchanger with knurled tubes and a flow core turbulator. Tubes with knurled annular channels have D= 20×2,5 mm. The relative height of tube protrusions: d/D=0, 96 and 0,875 (d – the diameter of protrusions). The pitch of protrusions: t=3 and 7mm. The ball-shaped flow core turbulator $d_{ш}$=0,6 and 2mm, the pitch $t_{яд}=(4\text{-}5)d_{ш}$.

Parameters in the heater: the wall temperature $t_с$=29,1...50,8°C; the inlet water temperature $t_в'$=11,4...17,1°C and the outlet water temperature $t_в''$=13,8...26,7°C.

With increasing Re, the effect of intensification $\alpha/\alpha_{гл}$ increases. Moreover, the growing intensification effect is apparent with the diaphragm height growth and the presence of flow core turbulators. At high Reynolds numbers of the dropping liquid flow it is advisable to use turbulators of small height.

For all knurled tubes with and without flow core turbulators, the resistance coefficient decreases with the growth of Re, which favorably affects the intensification.

The conducted studies of the heat transfer coefficient and the dependence of the relative hydraulic resistance coefficient from the pitch t/h showed that the studied characteristics have a maximum, falling on t/h≈10, where h is a scale.

A physical model is proposed to explain this. Considering the mechanism of formation and interaction of toroidal vortices in tubes with transversely annular protrusions and flow core turbulators, it was found that at certain pitches of turbulators arrangement it is possible to achieve an in-phase mode of vortex interaction, when there is provided the coincidence of vortex formation time and time of vortex flight from one source to another.

To calculate the degree of interaction of vortices behind the ring projections, the equation is suggested:

$$\theta_h = 0,85 + 0,15Sin\left[\frac{\pi}{2}\left(4\frac{t_h}{h \cdot m_h} + 1\right)\right] \tag{2.1}$$

To calculate the degree of interaction of vortices behind spherical thickenings, the equation is suggested:

$$\theta_{яд} = 0,85 + 0,15Sin\left[\frac{\pi}{2}\left(4\frac{t_{яд} \cdot Sl_{ш}}{d_{ш} \cdot m_{ш}} + 1\right)\right] \tag{2.9}$$

Provided that $t_{яд}/d_{ш}=0.47 \cdot t_h/h$, the combined factor, taking into account the interaction mechanisms behind the annular protrusions and the flow core turbulators, can be determined by the formula:

$$\theta_{\Sigma} = \theta_h \cdot \theta_{яд} \tag{2.12}$$

On the basis of the obtained experimental data, there are derived the equations for calculation of the relative coefficients of heat transfer and resistance in tubes with annular protrusions and flow core turbulators:

$$A = \frac{\alpha}{\alpha_{гл}} = 1,6 \cdot \theta_{\Sigma} \cdot \left[\left(\frac{d_{ш}}{d}\right)^2 + \left(\frac{d}{D}\right)^{-0,5} + 0,35 \cdot \left(\frac{D}{d}\right) \cdot \mathrm{Re} \cdot 10^{-3}\right] \tag{2.15}$$

$$B = \frac{\xi}{\xi_{гл}} = 1,4 \cdot \theta_{\Sigma} \cdot \left(\frac{d}{D}\right)^{-3,94} \exp\left[\left(1,2 \cdot \left(\frac{d_{ш}}{d}\right)^2 + 0,3 \cdot \frac{d}{D}\right) \cdot 10^{-4} \cdot \mathrm{Re}\right] \tag{2.18}$$

Section 3 presents the results of studies of the packed mixing device with heat carrier turbulators depending on the operating and design parameters.

The range of operating parameters in the studies: gas velocity $w_г$= 1÷5 m/s; irrigation density L =10÷75 $m^3/m^2 \cdot hr$; air temperature $t_{возд.}$= 20÷100 ^0C; heat carrier temperature in the tubular bundle $t_ж$=16÷100 ^0C; the range of design parameters: the pitch between packing elements vertically $t_в/b$ - 1÷5; horizontally $t_р/b$ - 1,5÷4; the size of tubular elements: d = 0,025 m; ℓ = 0,34 m; the size of turbulators: the relative height of annular protrusions: d/D=0,96 and 0,875; the pitch between annular protrusions t/h=10; spherical thickenings $d_ш/d$=0,04 и 0,145; the pitch between spherical thickenings (4-5) $t_{яд}/d_ш$.

Studies of the hydraulic resistance of tubes in the tube bundle showed that the size of knurling and the size of flow core axial turbulators have a significant impact on the hydraulic resistance, especially in the mode of developed turbulence. Friction losses increase significantly with the implementation of the mechanism of simultaneous vortex formation behind the annular protrusions and ball flow core turbulators. The values of heat transfer coefficients and Nusselt numbers in the entire range of Reynolds numbers increase, while with increasing temperature, the values of heat transfer coefficients and Nusselt numbers decrease.

As a result of studies of hydrodynamic laws of fluid motion in tubes, depending on the Reynolds numbers, the equation is derived:

$$\Delta p = \Delta p_1 + (z-2)\cdot\Delta p_2 + z\cdot(\Delta p_3 + \Delta p_4 + \Delta p_4 + \Delta p_{\text{тр}} + \Delta p_5) + (z-1)\cdot (\Delta p_6 + \Delta p_7 + \Delta p_8 + \Delta p_9) + \Delta p_{10}, \tag{3.9}$$

Here $\Delta p_1 - \Delta p_{10}$ are pressure losses in local resistances, Pa; z – number of motions in the tube bundle.

In the equation for calculation of friction losses $\Delta p_{\text{тр}}$ for tubes with knurling and flow core turbulators, the friction factor, with taking into account (3.20), has the form:

$$\xi = B\cdot\lambda_{\text{тр}} \tag{3.14}$$

To calculate the average heat transfer coefficient, the equation is given:

$$\propto = \frac{\lambda^{0,57}\cdot w_{\text{тр}}^{0,8}\cdot c^{0,43}\cdot \rho_{\text{ж}}^{0,43}}{d_{\text{тр}}^{0,2}\cdot \nu_{\text{ж}}^{0,37}} \tag{3.15}$$

The studies of hydraulic resistance ΔP_L and retained liquid quantity h_0 of the mixing apparatus packed zone with tube arrangement pitches $t_в$=2d и$t_р$=2d have shown that that with increasing Reynolds numbers for gas $Re_г$ and liquid $Re_ж$, the parameters under study increase. With increasing Reynolds numbers $Re_г$, the values of heat transfer coefficients α and also Nusselt numbers Nu grow.

To calculate the hydraulic resistance of the irrigated apparatus, a well-known equation for devices with a regular movable packing was used, in which a formula was proposed to determine the resistance coefficients:

$$\xi_L = 0,25\cdot\theta_в\cdot\theta_р\cdot \mathrm{Re}_{жс}^{0,1} \tag{3.18}$$

The amount of liquid retained is calculated by the equation:

$$h_o = 0,54\cdot\xi_L\cdot\frac{H}{t_в}\cdot\frac{\rho_г W_г^2}{2\cdot\rho_{жс}\cdot g\cdot\varepsilon_0^2}, \tag{3.26}$$

The equation for calculating the heat transfer coefficient is as follows:

$$\alpha = B_\alpha \cdot c_p^{/} \left[\xi_L \frac{D_г^2 \cdot U_г^3}{t_в \cdot \varphi_{яч} \cdot \nu_г} \right]^{1/4}, \quad (3.42)$$

where $B_\alpha = 7,28 \cdot \left(\frac{\varphi}{1-\varphi} \right)^{1/4}$ - the experimental coefficient, taking into account the contact heat exchange surface.

With taking into account

$$\lambda = 1{,}9 \cdot C_P^{/} \cdot \rho_Г \cdot \nu_Г, \quad (3.43)$$

equation in the criterion form is written as follows:

$$Nu_г^* = A_\alpha \cdot \frac{\mathrm{Re}_г^{*3/4} \cdot \mathrm{Pr}_г}{Sc^{1/2}}, \quad (3.44)$$

where $A_\alpha = 0,1 \cdot A_г$ - the dimensionless parameter, characterizing the interaction of vortices in an irrigated tubular bundle; $Nu^* = \frac{\alpha \cdot t_в}{\lambda}$ and $\mathrm{Re}^* = \frac{U_г \cdot t_в}{\nu_г}$ - the modified Nusselt and Reynolds numbers related to cell height; $Pr = \frac{c \cdot \rho_г \cdot \nu_г}{\lambda}$ - the Prandtl number.

Section 4 presents the results of studies of hydrodynamics and heat transfer in surface tubular heat exchangers.

The equation for determination of hydraulic resistance at the single-pass scheme of the heat carrier movement in tubes in the shell-and-tube heat exchanger with motionless tube grids has the form:

$$\Delta p = \Delta p_1 + \Delta p_2 + \Delta p_{тр} + \Delta p_3 + \Delta p_4 \quad (4.1)$$

The total resistance of the inter-tube space of shell-and-tube heat exchangers with transverse partitions (figure 4.1) is determined by the equation:

$$\Delta p = \Delta p_5 + \frac{l}{l_п} \Delta p_{мт} + \left(\frac{l}{l_п} - 1 \right) \Delta p_6 + \Delta p_7 \quad (4.3)$$

To calculate the pressure loss on friction in the inter-tube space of the heat exchanger, we propose an equation:

$$\Delta p_{мт} = \lambda_{тр} \cdot \frac{D}{t_в} \cdot \frac{\rho_ж \cdot W_{мт}^2}{2} \quad (4.4)$$

Here D is the internal diameter of the apparatus, m; $t_{\text{в}}$ – the tube arrangement pitch along the way of the moving stream.

The friction coefficient of friction $\lambda_{\text{тр}}$ is calculated by the formula:

$$\lambda_{\text{тр}} = 2{,}275 \cdot \theta_{\text{в}} \cdot \theta_{\text{р}} \cdot Re^{-0{,}2} \tag{4.5}$$

To calculate the heat transfer coefficient α, the formula is proposed:

$$\alpha = \theta_{\text{в}} \cdot \theta_{\text{р}} \cdot \frac{w^{0{,}65} \cdot c^{0{,}36} \cdot \rho^{0{,}36} \cdot \lambda^{0{,}64}}{\nu^{0{,}29} \cdot d^{0{,}35}} \tag{4.8}$$

Section 5 gives recommendations for designing, design procedure and implementation of mixing and surface heat exchangers with heat carrier turbulators.

The recommendations for designing contain the information on operating and design parameters, while the recommendations for design procedure - the basic hydrodynamic and heat exchange characteristics of mixing and surface heat exchangers with heat carrier turbulators.

According to the study results, an industrial mixing apparatus with a tubular packing was designed, which was implemented at JSC "Aktobe plant of chrome compounds" in the production of chromium oxide with an environmental effect on environmental protection measures of 30,09 million tenges / year. The surface heat exchanger with knurled tubes and flow core turbulators was implemented at the ZHF LLP "Kazphosphate" (NDFZ) in the production of thermal phosphoric acid with an estimated economic effect of 4.2 million tenges per year.

Conclusion contains brief conclusions on the results of dissertation studies, evaluation of the completeness of set tasks’ solutions, recommendations and initial data on the specific use of the results, an assessment of the technical and economic efficiency of implementation and the scientific level of work performed in comparison with the best achievements in this field.

Conventional signs: D – diffusion coefficient, m^2/sec.; internal diameter of the pipe, m; h - height, m; θ – coefficient characterizing the degree of vortex interaction; *Sl* – Strouhal number; $m_{\text{ш}}$ – mass of the ball, kg; A – coefficient; B – coefficient; α – heat transfer coefficient, $W/m^2 \cdot K$; ζ – resistance coefficient; d – nozzle diameter, drops, holes, particles, m; *Re* – Reynolds number; ℓ – length, m; ρ – density, kg/m^3; ξ – hydraulic resistance coefficient; λ - coefficient of heat conductivity, W /(m·K); coefficient; H – height, m; w – velocity, m/ sec.; W – irrigation density of the apparatus, kg / (m^2·sec.); flow rate, m / sec.; *Nu* – Nusselt number; *Pr* – Prandtl number;