Determination of thermal parameters of a shell and tube heat exchanger with increased turbulization of the working fluid

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Abstract. Shell and tube heat exchangers are still widely used. This equipment is widely used in heat supply systems, food, chemical and petroleum industries. Therefore, conducting research to improve their performance is an important area. By changing the geometry of the heat exchange surface, the flow of the working fluid is further turbulated and the heat exchange area increases. The heat exchanger with a developed heat exchange surface is proposed, which differs from the serial device (with smooth tubes). Thus, a plate equipped with a cylindrical edge is installed on the heat exchange tube. The working fluid in the inter-tube space, flowing around the edges, is turbulated. This reduces the thickness of the laminar sublayer on the plate and edge. Accordingly, the heat transfer from the heat exchange surface to the streamlined fluid increases. When new elements appear in the design of the heat exchange surface, it is necessary to develop the calculation of temperature parameters for these surfaces. Formulas for calculating the average surface temperature of the edge and plate under conditions of turbulent flow around these surfaces are obtained analytically. The temperatures of the plate and edge will determine the Prandtl criterion Pr. In the future, this is necessary to find the main parameter of the heat exchanger – the heat transfer coefficient.

1. Introduction
Improvement of heat supply systems is an urgent task at the present time. Therefore, the design of boilers, heat networks, heat points, etc. is made using modern high-performance equipment and materials [1]. Such equipment also includes heat exchangers used for heating, municipal and technological hot water supply.

Shell and tube heat exchangers are still popular equipment in heat supply (figure 1). These devices are widely used in various industries. This spread of shell and tube heat exchangers is explained by their resistance to pressure drops, the ability to work in environments with high pressure values (up to 14 MPa) and temperatures (up to 600 °C), a slight decrease in the heat transfer coefficient with moderate overgrowth of the heat exchange surface [2]. Improving the technical and operational characteristics of these devices requires additional research and justification.
Figure 1. Shell and tube heat exchanger:
1 – cooling liquid outlet pipe (pipe space); 2 - pipe for the supply of heated fluid; 3 – body; 4 – connecting roll; 5 - pipes connecting the inter-pipe space; 6 - pipe for hot (heating) fluid supply; 7 – outlet pipe of the heated fluid.

2. Materials and methods
In Russia and abroad, to improve the thermal characteristics, work is being actively carried out to improve the designs of heat exchangers [3]. According to the method of modernization of the structure for the intensification of heat exchange, we proposed the following classification:

1) constructive solutions:
   - arrangement of movement of the heat carrier in the body;
   - changing the geometry of the heat exchange surface;
2) the use of external physical fields;
3) adding chemicals to the heat carrier.

It should be noted that of all the ways to intensify heat transfer, “changing the geometry of the heat transfer surface” is the most common. This can be represented as a change in the geometry of the surface of the heat exchange tube: the use of spherical holes on the inner surface [4], biconcave spherical recesses [5]; semicircular hollows [6]. It is also possible to use tubes of spiral surface [7, 8], one smooth tube in the form of a spiral [9, 10]; in the form of a conical spiral [11].

We proposed an original design of a shell and tube heat exchanger [12]. The main difference from the serial [13] is that the plate (mounted on the tube) is equipped with a cylindrical edge (figure 2).

This allows increasing the heat exchange surface. Also, the working fluid is turbulated when it flows around the edge. This creates additional turbulization of the heated liquid on the plate and edges (in the inter-tube space). In turn, turbulization will contribute to the destruction of the wall-mounted laminar layer on the plate and edges, which means that heat transfer from the heat exchange surface to the heated fluid will increase [14]. In the end, this will lead to the increase in the performance of the heat exchanger for the heat carrier: hot or superheated water.

It is important to note that when new elements of the heat exchange surface – edges and plates – occur, it is necessary to develop a method for calculating the average temperature of these surfaces.
Figure 2. Heat exchange surface of the shell and tube heat exchanger: 1 – tube; 2 – plate; 3 – edges of cylindrical shape; \( U_m \) – velocity of the fluid in the inter-tube space, m/s; \( R_0 \) – edge radius, \( d_p \) – edge diameter, \( l_n \) – length of the plate between the edges, \( h \) – height of the plate.

To find the temperature change inside a cylindrical edge with radius \( R_0 \), m, and length \( h \), m, with a forced heat source \( q_0 \), W/m³, represented as a thin plate located along the axis of the cylinder (figure 3), we use the stationary Fourier thermal conductivity equation in cylindrical coordinates [15]:

\[
\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} + \frac{q_0}{\lambda} = 0,
\]

where \( \lambda \) – coefficient of thermal conductivity of a cylindrical body, W/(m·K).

Figure 3. Scheme of the cylindrical edge for determining the temperature inside the cylindrical body with the heat source in the form of a plate: 1 – twirl, 2 – plate, 3 – edge.

It should be noted that due to the axial symmetry, the temperature distribution function inside the cylinder does not depend on the coordinate \( \varphi \).

In the state of thermodynamic equilibrium, the amount of heat allocated per unit of time in a unit of volume of a cylindrical body is determined by the ratio:

\[
q_0 = \frac{q_{sz}}{V},
\]

where \( V \) – edge volume, m³, \( q_{sz} \) – the amount of heat generated by the plate, W.

It is important to note that based on [15], the amount of heat generated by the plate is determined by the formula:

\[
q_{sz} = -\lambda_0 S \frac{\partial T}{\partial z},
\]

where \( \lambda_0 \) – coefficient of thermal conductivity of the plate, W/(m·K).
where \( \lambda_0 \) - coefficient of thermal conductivity of the plate, W/(m K), \( S \) – surface area of the plate, m\(^2\), \( \frac{\partial T}{\partial z} \) - temperature gradient along the \( z \) axis, K/m.

The volume of a cylindrical edge is determined by:

\[
V = \pi R_0^2 h .
\]  

(4)

The surface area of the plate is equal to:

\[
S = 2R_0 h .
\]  

(5)

Then equation (1) taking into account formulas (2) – (5) takes the form:

\[
\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} - \frac{2\beta}{\pi R_0} \frac{\partial T}{\partial z} = 0 .
\]  

(6)

where the following mark is introduced:

\[
\beta = \frac{\lambda_0}{\lambda} .
\]  

(7)

Using the Fourier method [16], we look for the solution of equation (6) in the form:

\[
T(r, z) = T_1(r)T_2(z) .
\]  

(8)

After the transformations, we present the final equation of temperature change \( T(R_0, z) \), K, over the surface of a cylindrical edge [17]:

\[
T(R_0, z) = T_0 I_0 \left[ \beta \left( \frac{R_0 \ln \left( \frac{T_0}{T_1} \right)}{\beta h} \right)^{-1} \left( \frac{T_0}{T_1} \right)^{\frac{z}{R_0}} \right] .
\]  

(9)

where \( T_0 \) – temperature at the point of contact of the plate and tube with the coordinate \( z=0 \), \( T_1 \) – temperature at the end of the plate with the coordinate \( z=h \).

The obtained dependence (9) allows determining the average temperature on the edge surface using the formula:

\[
T_{av} = I_0 \beta \left[ \frac{R_0 \ln \left( \frac{T_0}{T_1} \right)}{\beta h} \right]^{-1} \ln \left( \frac{T_0}{T_1} \right) .
\]  

(10)

where \( I_0 \) – the Bessel function of the zero order of the first row.

To determine the average surface temperature of the plate with height \( h \) and thickness \( 2\delta, \) m, we place the plate as shown in figure 4.
The temperature distribution over the height of the plate, which is flowed by the turbulent flow of fluid, is determined by the following dependence [18]:

\[ T_n = T_0 \sum_{n=0}^{\infty} a_n \left( e^{-\frac{v_n y}{\delta}} + e^{\frac{v_n y}{\delta}} \right) \cos \left( \frac{v_n y}{\delta} \right), \]  

where value \( a_n \) is equal to:

\[ a_n = \frac{2 \sin c(v_n)}{(1+\sin c(2v_n)) - (1+\varepsilon_n)}. \]  

Parameter \( v_n \) is a dimensionless value and is found as:

\[ v_n = \pi n + \alpha_n. \]  

Value \( \alpha_n \) is determined from the tables offered in [18], and depend on dimensionless value \( \Lambda \):

\[ \Lambda = \frac{v_n}{\delta / \lambda}. \]  

where \( \lambda \) - coefficient of thermal conductivity of the fluid, W/(m K); \( \delta \) – thickness of the boundary layer, m.

Value \( \varepsilon_n \) is determined as:

\[ \varepsilon_n = \frac{\nu_n - \Lambda e^{-\nu_n h / \delta}}{\nu_n + \Lambda}. \]  

Then the average value of the plate surface temperature \( T_{cp}^n \), K, can be calculated as follows:

\[ T_{cp}^n = \frac{T_0}{h} \frac{1}{2\delta} \int_0^h \int_0^\delta \sum_{n=0}^{\infty} a_n \left( e^{-\frac{v_n y}{\delta}} + e^{\frac{v_n y}{\delta}} \right) \cos \left( \frac{v_n y}{\delta} \right) dy \right) dx. \]  

As a result of converting expression (16), we obtain a formula for determining the average temperature of the surface of the plate that is flowed by a turbulent flow:

\[ T_{cp}^n = \frac{T_0}{h} \sum_{n=0}^{\infty} a_n \sin \nu_n \frac{\nu_n}{\nu_n^2} \left( e^{-\nu_n} + e^{\nu_n} \frac{\nu_n}{\delta} - 1 - \varepsilon_n \right). \]
3. Results
Thus, important thermal parameters of the new elements of the heat exchange surface of the shell and tube apparatus with increased turbulence of the working fluid in the inter-tube space were determined: the average surface temperatures of the cylindrical edge \( T_p \) (formula 10) and the plate \( T_{\text{cp}} \) (formula 17). The values of these temperatures are necessary to find the Prandtl numbers \( \text{Pr} \) for the edge and plate. Knowing The \( \text{Pr} \) value, it is possible to determine the Nusselt number \( \text{Nu} \) for new elements of the heat exchange surface of a shell and tube apparatus with increased turbulence of the working fluid.

It is known that the coefficient of heat transfer from the heat exchange surface is calculated based on the geometric dimensions (for the plate – length, m, for the edge – diameter, m), the coefficient of thermal conductivity of the fluid \( \text{W}/(\text{m K}) \) and the number \( \text{Nu} \) [15]. Hence, knowing the coefficient of heat transfer from the edge \( \alpha_p \), \text{W}/(\text{m}^2 \text{K}) and a plate of \( \alpha_{\text{np}} \), and the coefficient of heat transfer from the outer surface of the tube \( \alpha_n \) [19], it is possible to calculate the average heat transfer coefficient (according to M.A. Mikheev’s research) from the outer, heating surfaces of heat transfer \( \alpha_2 \) with the changed geometry:

\[
\alpha_2 = \frac{\alpha_n F_n + \alpha_{\text{np}} F_{\text{np}} + \alpha_p F_p}{F_n + F_{\text{np}} + F_p}, \quad (18)
\]

where \( F_n, F_{\text{np}}, F_p \) - the area of the outer surface, pipe, plate, edge respectively, \text{m}^2.

It is important to note that by defining the parameter \( \alpha_2 \), it is possible to calculate the heat transfer coefficient of a shell and tube heat exchanger with increased turbulence using the well-known formula [19].

4. Summary
Thus, the proposed procedure for calculating the temperature parameters of the heat exchange surface is necessary for further development of the method for selecting heat exchangers with increased turbulence of the working fluid. The development of a software package for selecting the standard size and calculating the design and technological parameters of such heat exchangers is also of interest.

5. References
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