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Arthur Gibadullin
Geetha Devi
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Preface

The organizing team is proud to publish the selected papers of the IV International Symposium, (MST2024), held from 23 to 25 December 2024 in the Springer Proceedings in Physics Series. The MST2024 Symposium intends to be an international platform for the potential researchers and materials engineers to present and showcase the recent developments and techniques in the field of physics, materials science, biological, chemical and mechanical materials. All the submitted papers were undergone three extensive peer reviews by qualified experts as independent reviewers in the field selected by the symposium committee.

As a result of the work of all reviewers, out of 32 received papers, the top 12 papers were accepted for the publication. The peer review focused on the quality of the content, the relevance of the research, novelty and scientific significance and the originality of the work. Reviewers, members of the program committee and members of the organizing committee were kept away from the peer review process to avoid any conflicts of interest.

The symposium was intended to provide international experience of the participants in the field of physics and materials science. In particular, the following issues were considered:

– Formation of scientific and practical proposals for the development of materials science, chemistry, new polymers and composites;
– Development of fundamental and applied research in the field of chemical physics and physical chemistry and related sciences;
– Formation of recommendations aimed at improving the quality of materials and developing the scientific and technical base of new materials and composites.

In the symposium, the authors of the articles presented and displayed their novel research findings for broad discussion, searching for options for interaction and gather new approaches for the further development of their research.

A distinctive feature of the symposium is that authors and speakers were from the different nationals like People's Republic of China, the Sultanate of Oman, the Republic of Azerbaijan, the Russian Federation and the Republic of Uzbekistan spoke at it. Researchers from different countries presented their research outcome and achievements in the field of physics, materials science and new materials.

Thus, the symposium offered knowledge exchange and gain advanced knowledge for all participants, and the results of the symposium were beneficial for state and regional authorities for the development of industry and new areas of activity, international and supranational organizations, scientific and professional community.

The organizing committee of the symposium expresses their gratitude to the Springer staff that supported the publication of this collection. In addition, the organizing committee would like to thank the symposium participants, reviewers and those who helped to successfully organize this symposium.

Moscow, Russia — Arthur Gibadullin
Muscat, Oman — Geetha Devi
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Chapter 11
Research of the Physical Basis of Heat Transfer in Heat Exchanger Devices

Saydulla Khuzhakulov, Zakir Pardaev, Ismoildjon Muradov, and Timur Shodiev

Abstract The purpose of this article is to determine the dependence of heat transfer processes on balance and criterion equations, heat transfer in external and internal heat exchange, criterion equations of heat transfer in heat exchange devices, the dependence of heat on physical parameters, and mathematical modeling of the process. The heat transfer coefficient was calculated based on criterion equations related to the Nusselt, Prandtl (from the table), Grasshoff, Pecle and Reynolds numbers. When a pipe with d_0 = 32 mm was studied, according to the calculations based on criteria, it was determined that the heat transfer coefficient is 14.5×10^3 W/m^2 K and the heat flux is 29.14×10^3 W when the Nusselt number is 730.47. When the heat exchange is calculated when the flow crosses a large number of pipe bundles, the Reynolds number for the flow at a speed of 4 m/s is equal to 12.7×10^4, that is, the flow is turbulent. According to the criterion equations, the Nusselt number reaches 749.7, and the value of the heat transfer coefficient reaches 14,021 W/m^2 K. Based on the results of mathematical modeling based on the selected boundary conditions, it was determined that the coefficient of heat transfer is greater, it depends on the flow regime of heat carriers, Reynolds, Nusselt criteria and Prandtl number, and graphs of dependence were obtained.

11.1 Introduction

Heat exchangers are an integral part of industrial processes, especially heating, ventilation, and cooling processes in thermal power plants, due to their high efficiency in transferring heat between working bodies [1–4]. In terms of their structural design, concentric tube heat exchangers are distinguished by their simplicity, economy, and widespread use [5, 6]. The performance of heat exchangers is significantly affected by their geometric parameters and flow conditions [7, 8]. Factors such as the inner diameter of the pipe, the ratio of diameters, the length, and the Reynolds number

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of the hot and cold coolants can significantly affect the heat transfer coefficient and pressure differences in the heat exchanger.

Furthermore, the correct choice of the working medium has a significant impact on the thermal characteristics [9, 10], as the selection of different heat carriers (e.g., air–air, air–water, water–air, and water–water) can change the thermal and hydraulic characteristics of the system and thus affect the overall efficiency of the system.

11.1.1 Statement of the Problem

In contrast to the concept of heat conduction, when heat energy is transferred from a stationary or moving medium through its molecular structure, the movement of the medium causes heat transfer. A moving medium is understood to be a liquid or heat carrier [11].

Thus, the convective transfer of heat between layers of a heat carrier flow with different temperatures is called convective heat transfer. In practice, heat exchange between a heat carrier flow and a wall surface, i.e. convective heat transfer, is often observed. In this process, heat is transferred in one direction, i.e. either from the heat carrier to the wall surface, or vice versa, from the wall surface to the heat carrier flow. The purpose of this article is to determine the dependence of heat transfer processes on balance and criterion equations, heat transfer in external and internal heat exchange, criterion equations of heat transfer in heat exchange devices, determination of the dependence of heat on physical parameters, and mathematical modeling of the process.

11.2 Materials and Methods

The basic equation representing heat transfer is Newton's equation:

$$Q = \alpha \cdot \left(T_j - T_w\right) \cdot F_w, \quad q = \alpha \cdot \left(T_j - T_w\right), \tag{11.1}$$

here Q—heat flow in heat supply, W; T_j—average temperature of the heat carrier over the longitudinal heat exchange surface F_w; T_w—average temperature of the heat exchange surface; $T_j - T_w = \Delta T_w$—temperature difference in heating; $q = Q/F_w$—heat flow density on the heat exchange surface, W/m^2; α—heat transfer coefficient, $W/(m^2\ K)$.

Heat transfer problems are solved by calculating the given heat flux Q at given values of the heat exchange surface F_w, or, conversely, by determining the required surface F_w at which heat is transferred at a given value of the heat flux Q.

In general, the heat transfer coefficient is calculated based on the following equations:

$$Nu = f(Re, Gr, Pr, \ldots) \tag{11.2}$$

Derivation of criteria of constituent similarity in this expression is presented in educational and scientific literature [11–13].

From the expression (11.2), the Nusselt number is the quantity determined by calculation, which contains the required coefficient α:

$$Nu = \frac{\alpha \cdot l_0}{\lambda_j}, \tag{11.3}$$

here l_0—descriptive dimension of the process, $\lambda_j(T_j)$—is the thermal conductivity of the heat carrier and is determined as a physical parameter from references [16] and depending on its type and temperature from [10].

The next constituent, the numerical value of which is the Reynolds number, describes the flow regime of a heat carrier moving with a velocity w_j relative to the heat exchange surface:

$$Re = \frac{\rho_j \cdot w_j \cdot l_0}{\mu_j}, \tag{11.4}$$

The criterion equation describing the intensity of heat exchange in free convection conditions of the heat carrier is the Grasshof criterion:

$$Gr = \frac{\beta_j \cdot g_0 \cdot \Delta T_w \cdot l_0^3}{\nu_j^2}; \tag{11.5}$$

this expression takes into account the temperature difference between the coolant and the surface of the object ΔT_w and the acceleration of the environment; $\nu_j(T_j) = \frac{\mu_j}{\rho_j}$—kinematic viscosity of heat carriers; $\beta_j(T_j)$—volume expansion coefficient, for gases $\beta_\Gamma = \frac{1}{(T_j)}$.

This value is also taken from the reference books for liquids or calculated from the density chosen for a given temperature interval:

$$\beta_j(\Delta T_w) = \frac{(\rho_w - \rho_j)}{\rho_j}. \tag{11.6}$$

Along with the coefficients λ_j (T_j) and μ_j (T_j) of liquid heat carriers, depending on the nature and temperature of the heat carrier, the physical parameter Prandtl criterion Pr_j (T_j) is determined from the references (most often [13, 15, 16] is used for liquids and gases).

To perform heat calculations, the heat physical properties of heat carriers can be determined from analytical relationships over a specified temperature range [15].

Often, complex similarity numbers are used in criterion equations, such as the Peclet number $Pe = Re \cdot Pr$, the Reley number $Ra = Gr \cdot Pr$.

The shape factor of the body Φ—takes into account all the features of the heat carrier flow around it, depending on the size and shape of the surface of the body.

When developing and calculating a mathematical model of the heat transfer process, the recommended governing equations of heat transfer help to generalize the experimental results by a maximum of 10–20%. When solving a specific problem, when selecting governing equations from reference books, it is necessary to pay attention to the following:

- Depending on the external factors that cause the movement of the heat carrier, it is divided into free and forced convection. However, there may also be a mixed convection case;
- Forced convection can be performed for cases of unrestricted flow of the heat carrier and movement of the heat carrier in pipes and channels. Calculation of unrestricted flow is an external problem, calculation of the heat carrier moving through the pipe is an internal problem [17];
- Depending on the intensity of movement, forced convection can occur in laminar or turbulent modes of the heat carrier.

The governing equations of heat transfer, depending on the given and a number of other features, differ fundamentally from each other. Accordingly, the choice of a governing equation for a specific case of heat transfer is of great importance in solving the problem posed. The similarity criteria included in Eq. (11.2) are the physical parameters of heat transfer of liquid and gaseous heat carriers. The method of accurately estimating the parameters in the process is very complicated. In technical calculations, the method of presenting experimental data and calculation results in tabular form is widespread. It should be taken into account that the data given in different reference books for the same type of working body may differ in value. To avoid this situation, it is advisable to refer to the reference materials used in the calculation practice.

When selecting the physical parameters of heat carriers from reference tables, it is necessary to distinguish between the given values from a certain range of values. In this case, for accurate calculations, it is necessary to use the interpolation method. For example, to determine the parameter A based on the given parameter B, the following formula is used:

$$A = A_q + \left(A_{up} - A_l\right) \cdot \frac{B - B_l}{B_{up} - B_l}, \tag{11.7}$$

where A is the parameter $A_{up} < A < A_l$ value to be searched between the upper and lower (larger and smaller) values in the table; B is a given parameter of the function A(B) and its value is found in the interval $B_{up} < B < B_l$.

In external heat transfer problems, the criterion equation in the form of Eq. (11.2) for calculating heat transfer by free convection over a body in a uniform unconfined isobaric environment does not take into account the criterion number Re for the case where there is no forced motion of the heat carrier, and the dynamics of the motion can be fully described by the magnitude of the criterion Gr. It is customary to call the product of the criteria $Gr \cdot Pr = Ra$ the Reley number, in which case expression (11.2) is written as follows [11, 13]:

$$Nu_p = C \cdot (Ra_p)^n \cdot \left(\frac{Pr_p}{Pr_w}\right)^{0,25}, \tag{11.8}$$

where C is a constant and the power index n is a function of the number Ra and is determined from the numerical values and depending on the shape of the heat exchange surface as follows:

- For vertical walls $Ra_p = 1 \times 10^3$ to 1×10^9, $C = 0.75$–0.8; $n = 0.25$;
- For vertical walls $Ra_p > 6 \times 10^9$, $C = 0.15$; $n = 0.333$;
- For horizontal pipes $Ra_p = 1 \times 10^3$ to 1×10^9, $C = 0.50$; $n = 0.25$.

The index "p" in the expression denotes the physical properties of the heat carrier (β, λ, ν, Pr) related to the similarity criterion. These values are chosen as the average value based on the calculated temperature of the moving heat carrier $t_p = ((t_w + t_j))/2$.

The following expression is used to calculate the average heat transfer coefficient of the vertical surface under the conditions of $q_w = \text{const}$:

$$Nu_j = 0.75 \cdot Ra_j^{0,25} \cdot \left(\frac{Pr_j}{Pr_w}\right)^{0,25}, \tag{11.9}$$

in this case, the temperature of the heat carrier at the boundary of the moving layer is taken. When determining the heat transfer over a horizontal cylindrical pipe, the constant value in the expression is taken to be equal to 0.5.

Equation (11.8) includes a correction for the number Pr during temperature changes and, at the same time, for the physical parameters of the thickness of the moving layer formed on the heat exchange surface [11, 12].

Heat transfer under forced convection conditions. The problems of heat transfer in this case are also worked out on the basis of the criterion Eq. (11.2). The structure and constants of this equation are determined from the flow regime of the heat carrier relative to the heat exchange surface. Also, based on the boundary conditions of the process, the problem of heat transfer is divided into external and internal problems [11, 13].

Internal heat transfer problems are characterized by the flow of the heat carrier inside pipes and channels, its interaction with the wall surface and filling the horizontal surface of the channel. In this case, the flow regime is divided into laminar $Re \leq 2300$, turbulent $Re > 10^4$ and transitional $2300 \leq Re < 10^4$ regimes.

Also, the laminar flow regime is divided into viscous ($Re \leq 2300$, $Ra \leq 8 \times 10^5$), in which natural convection does not significantly affect heat transfer, and viscous-gravitational ($Re \leq 2300$, $Ra > 10^6$), in which free convection affects the heat carrier layers. In the process of analyzing scientific works and studying scientific and technical literature, work was carried out to apply Eq. (11.2) in real situations and calculate it.

In deaerators, pyrolysis plants, device for recycling municipal solid waste [14] and condensers, the process of heat transfer of the coolant in pipes and channels under forced convection conditions is considered.

The most important heat transfer process in all thermodynamic processes is the heat transfer of the working bodies moving in the pipes.

The calculation of the heat transfer of the coolant flow in filled channels or pipes is an internal issue of the heat transfer process, and in such cases the following characteristic parameters of the process are selected [12, 13]:

- Equivalent or hydraulic diameter of the channel d_e:

$$d_e = \frac{4F_0}{P}, \tag{11.10}$$

where F_0 is the cross-sectional area of the channel (or the cross-sectional area through which the heat carrier flows); P is the perimeter of the cross-sectional area through which the heat carrier flows;

- The flow velocity in the channel w_m (or mass velocity), which is found from the flow rate of the heat carrier:

$$G = \rho \cdot w_m \cdot F_0, \quad \text{where } w_m = \frac{G}{(\rho \cdot F_0)}, \frac{\text{m}}{\text{s}}. \tag{11.11}$$

The universal formula for calculating the Reynolds number for internal flows, taking into account expression (11.11), is written as follows:

$$Re = \frac{\rho \cdot w_m \cdot d_e}{\mu} = \frac{4G}{\mu \cdot P}, \tag{11.12}$$

where μ is the dynamic viscosity of the heat carrier, its value is selected from the reference according to the temperature value T [15, 16].

The choice of the heat transfer criterion equation of the form (11.2) and its appearance as a function of the number Re is shown in the calculation of complex heat processes.

11.2.1 Recommendations

Recommendation 1. When calculating the heat transfer of heat-carrying fluids, special attention should be paid to the Prandtl number $Pr(T_j)$. Its value, unlike gases, is equal to 100 and sometimes 1000, and varies depending on the temperature of the heat-carrying fluids. The change in the Pr number is taken into account by the correction $(Pr_j/Pr_w)^{0.25}$.

Recommendation 2. To determine the heat transfer in internal heat exchange problems, a change is made to the hydrodynamic stability section ε_l. This value takes into account the influence of the heat carrier entering the pipe on the intensive heat exchange. The correction value depends on the flow regime of the coolant and the relative size of the pipe (l/d_0) and is given in the form of a table and formula in the reference books [15].

Recommendation 3. The choice of the criterion equation for heat transfer depends on the exact calculation conditions.

11.3 Results and Discussions

When water is pumped at a speed of $w_j = 4$ m/s in a 1-m-long HE pipe channel, the effect of the shape of the channel's cross-sectional surface on the heat transfer coefficient and the magnitude of the heat flux transferred from the wall to the water flow is studied and modeled for

- A pipe with a diameter of $d_0 = 32$ mm;
- A rectangular channel with an aspect ratio of 1:25.

The average water temperature along the length of the channel is $t_m = 40$ °C and the channel wall temperature $t_w = 80$ °C, and the channels are assumed to have the same cross-sectional area F_0 and water flow velocity w_j (Fig. 11.1).

Based on the average water temperature $t_m = 40$ °C, we obtain the physical parameters of heat from [4]:

$$v = 0.658 \times 10^{-6}\,\frac{\mathrm{m}^2}{\mathrm{s}};\quad \lambda = 0.635\,\frac{\mathrm{W}}{\mathrm{m\,K}};\quad Pr_j = 4.31;$$

$$t_w = 60\,^{\circ}\mathrm{C} - Pr_w = 2.99.$$

For case 1: we determine the equivalent diameter of the channel:

$$d_э = \frac{4 \cdot F}{P} = \frac{4}{\pi} \cdot \frac{\pi}{4} \cdot \frac{d_0^2}{d_0} = d_0 = 32\,\mathrm{mm},$$

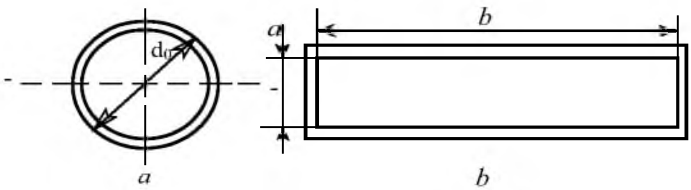


Fig. 11.1 1-m-long pipe and rectangular channel heat exchanger

- The equivalent diameter of a circular pipe is equal to its internal diameter.

To determine the flow regime of water in the pipe, we calculate the criterion equation Re:

- The equivalent diameter of a circular pipe is equal to its inner diameter.

To determine the flow regime of water in the pipe, we calculate the Re criterion equation:

$$Re = \frac{w_j \cdot d_0}{\nu_j} = \frac{4 \cdot 0.032}{0.658} \cdot 10^6 = 19.45 \cdot 10^4 > 10^4$$

i.e. the flow regime is turbulent.

Reference [15] the criterion equation for the heat transfer coefficient is selected from the literature and calculated. For the case where Pr > 0.7 in the turbulent movement of heat carriers along the pipe, it is recommended to use the following criterion equation:

$$Nu_j = 0.021 \cdot Re_j^{0.8} \cdot Pr_j^{0.43} \cdot \psi(Pr) \cdot \varepsilon_l, \tag{11.13}$$

where $\psi(Pr) = \left(\frac{Pr_j}{Pr_w}\right)^{0.25} = \left(\frac{4.31}{2.99}\right)^{0.25} = 1.09$ is the correction factor that determines the change in the temperature of the heat carrier on the cross-sectional surface of the pipe, since the Pr number is highly dependent on temperature for all flowing fluids, and in addition, the temperature range of heat carriers is also large.

The correction factor for the initial flow point of the heat carrier in the pipe also has the value $\varepsilon_l < 1$, subject to the condition $\frac{l}{d_0} = \frac{1.0}{0.032} = 31.25 < 50$ [14].

$$Nu_j = 0.021 \cdot \left(19.45 \cdot 10^4\right)^{0.8} \cdot 4.31^{0.43} \cdot 1.09 = 730.47;$$
$$\alpha = \frac{Nu_j \cdot \lambda}{d_0} = \frac{730.47 \cdot 0.635}{0.032} = 14.5 \times 10^3 \frac{\mathrm{W}}{\left(\mathrm{m}^2\,\mathrm{K}\right)}.$$

We determine the heat flow during heat transfer:

$$Q_0 = \alpha \cdot F_w \cdot (t_w - t_m) = 14.5 \cdot 10^3 \cdot 3.14 \cdot 0.032 \cdot 1.0 \cdot (60 - 40)$$
$$= 29.14 \cdot 10^3\ \mathrm{W}.$$

For case 2: since the dimensions of the channel passage surface are not given, further studies are carried out using the relative method. For channels with the same cross-sectional area, the following can be written:

$$F_0 = a \cdot b = \frac{\pi \cdot d_0^2}{4} = 25 \cdot a^2, \tag{11.14}$$

because according to the initial conditions a:b = 1:25. Finally $a = \left(\frac{\pi}{100}\right)^{0.5} \cdot d_0 = \left(\frac{1}{100 \cdot \pi}\right)^{0.5} \cdot \pi d_0$.

To calculate the perimeter P of the channel, we find the following relationship:

$$P_0 = \pi \cdot d_0; \quad P_k = 2 \cdot (a + b) = 2 \cdot a \cdot \left(1 + \frac{b}{a}\right) = 52a.$$

$\frac{P_k}{P_0} = 52 \cdot \left(\frac{1}{100 \cdot \pi}\right)^{0.5} = 2.93$, i.e., in a channel with the same cross-sectional area, the perimeter of the channel with an aspect ratio (a:b = 1:25) is almost three times larger than the perimeter of the pipe section.

To compare the intensity of heat transfer, the ratio of the Re numbers in the criterion equation is found. The answer to the question of whether the water flow regime in the channels changes or not is found.

We determine the Re number through the coolant flow:

$$Re = \frac{4 \cdot G}{\mu \cdot \Pi}, \tag{11.15}$$

where $G = \rho \cdot w_j \cdot F_0 = const$, because according to the initial conditions, $w_j = const$, $F_0 = const$, and also if $\rho = const$, the temperature indicators of the water are also considered unchanged.

According to this condition, Re $\approx$ P^{-1}—that is, the Re criterion is inversely proportional to the channel perimeter:

$$\frac{Re_k}{Re_0} \sim \frac{P_0}{P_k} = \frac{1}{2.93}.$$

$$Re_k = \frac{Re_0}{2.93} = \frac{19.45 \cdot 10^4}{2.93} = 6.64 \cdot 10^4 > 10^4.$$

When moving from a pipe to a rectangular channel, the flow regime of water does not change, therefore, the criterion equation remains unchanged.

We will study the effect of the equivalent diameter of the channel on the heat transfer intensity. From the criterion Eq. (11.2) it is known that: $\alpha \sim d_э^{-1} \cdot Re^{0.8} \sim P \cdot P^{-0.8} = P^{0.2}$, that is, when $F_0 = const$, as the perimeter surface of the channel increases, its equivalent diameter decreases, and the heat transfer intensity increases. Then, the heat transfer coefficient in the channel.

$$\frac{\alpha_k}{\alpha_0} \sim \left(\frac{P_k}{P_0}\right)^{0.2} = 2.93^{0.2} = 1.24$$

increases by 24%.

The heat flux change is calculated based on the above:

$$\frac{Q_k}{Q_0} \sim \frac{\alpha_k}{\alpha_0} \cdot \frac{F_k}{F_0} \sim \frac{\alpha_k}{\alpha_0} \cdot \frac{P_k}{P_0} \sim \left(\frac{P_k}{P_0}\right)^{1.2} = 2.93^{1.2} = 3.633.$$

From these derived solutions, it can be concluded that the heat transfer coefficient of the flow at the same cross-sectional area is 3.63 times greater for the heat transfer from a rectangular channel than for the heat transfer from a pipe. This is explained by the increase in the intensity of heat transfer in the channel and the increase in the perimeter of the cross-sectional area there.

If the flow of the heat carrier heated in the external environment is accompanied by a high-temperature heat transfer flow inside the pipe, and the flow is directed perpendicular to the pipe surface, the model for calculating the heat transfer process using standard equations is as follows.

11.3.1 Express the Calculation Results

The algorithm for setting and solving the problem of heat transfer during forced flushing of pipes and pipe bundles in the horizontal direction is carried out in a similar way to the general calculation method, with the form and composition of the heat transfer criterion Eq. (11.2) changing. The composition and form of these equations are selected from the reference literature for specific cases encountered in practice [15].

In our case, a boiling heat carrier flow moving in a straight pipe with an outer diameter $d = 32$ mm is cooled by a water flow in the horizontal direction (Fig. 11.2). In this case, if the water flow velocity $w_j = 4.0$ m/s, the average water temperature $t_j = 20$ °C, and the temperature on the pipe wall surface $t_w = 60$ °C, we determine the coefficient of heat transfer from the pipe wall to the cooling water.

To calculate the heat transfer of a cylindrical channel with a fluid flow transversely, the following criterion equations should be used [15]:

(1) For the case where $8 < Re_j < 1 \cdot 10^3$

$$Nu_j = 0.5 \cdot Re_j^{0.5} \cdot Pr_j^{0.38} \cdot \psi(Pr), \tag{11.16}$$

(2) For the case where $1 \cdot 10^3 < Re_j < 2 \cdot 10^5$

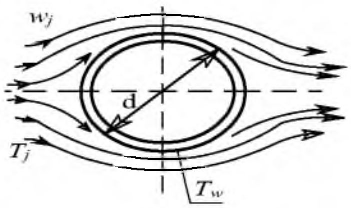


Fig. 11.2 Providing heat during transverse flushing of the pipe surface

$$Nu_j = 0.25 \cdot Re_j^{0.6} \cdot Pr_j^{0.38} \cdot \psi(Pr), \tag{11.17}$$

In Eq. (11.16), $\psi(Pr) = \left(\frac{Pr_j}{Pr_w}\right)^{0.25}$ is the correction for the change in the parameters of the heat carrier near the heat transfer surface; the diameter of the cylindrical body is taken as the defining parameter, the indices "j" and "w" denote the physical properties of the heat carrier and are selected based on the average temperature t_j of the moving flow and the average temperature t_w of the heat carrier on the heat exchange surface [17].

In this case, the flow regime when the pipe surface is washed by a water flow is determined. For the case where the water temperature $t_j = 20$ °C, the physical parameters of the heat carrier are selected from the reference [15]:

$$\nu = 1.003 \cdot 10^{-6} \frac{\mathrm{m}^2}{\mathrm{s}}; \quad \lambda = 59.85 \cdot 10^{-2} \frac{\mathrm{W}}{(\mathrm{m\,K})}; \quad Pr = 7.01.$$

In $t_w = 60\,°\mathrm{C}$, $Pr_w = 2.99$.

The value of the number Re according to the selected parameters:

$$Re = \frac{w_j \cdot d}{\nu} = \frac{4 \cdot 0.032}{1.003 \cdot 10^{-6}} = 12.7 \cdot 10^4 > 10^4$$

indicates that the water flowing through the pipe is moving turbulently.

Now the heat transfer criterion equation can be solved. Because in the case under review

$$1 \cdot 10^3 < Re_j < 2 \cdot 10^5$$

and this indicates that expression (11.17) should be used.

Substituting exponents to solve the criterion equation gives the following result:

$$\begin{aligned} Nu_j &= 0.25 \cdot Re_j^{0.6} \cdot Pr_j^{0.38} \cdot \psi(Pr) \\ &= 0.25 \cdot \left(12.7 \cdot 10^4\right)^{0.6} \cdot 7.01^{0.38} \cdot \left(\frac{7.01}{2.99}\right)^{0.25} = 749.7. \end{aligned}$$

The value of the coefficient of heat transfer to water through the pipe wall:

$$\alpha = \frac{Nu_j \cdot \lambda}{d_1} = \frac{326.7 \cdot 0.5985}{0.032} = 14.021 \cdot 10^3 \frac{\mathrm{W}}{(\mathrm{m}^2\,\mathrm{K})}.$$

11.4 Conclusion

Heat transfer is a physical process of heat (cold) transfer between the surface of solid bodies and the working environment (heat carriers) that wash them. Heat carriers consist of gases, liquids and solutions. Heat transfer is also a process that occurs as a result of convection and radiation heat exchange, as well as heat transfer.

The process of heat exchange through the surface area of heat exchange devices is the basis for the creation of all types of heat energy devices.

Regardless of whether the wall that separates the heat and the different temperature environments has a circular, flat, angular or flat shape, their materials will have different heat transfer properties. This parameter, in turn, depends on the temperature of heat carriers, wall thickness and a number of other thermal physical parameters.

In our case, the heat transfer process of heat carriers with high velocities at small temperature differences was modeled. According to the modeling results, it was found that the heat transfer coefficient depends mainly on the flow regime of the heat carriers, the Reynolds and Nusselt criteria, and the Prandtl number (Figs. 11.3, 11.4, 11.5, 11.6, 11.7 and 11.8).

Figure 11.3 shows a graph of the dependence of the heat transfer coefficient on the Nusselt criterion when the flow regime is laminar and turbulent, Fig. 11.4 the graph of the dependence of the heat transfer coefficient on the Nusselt criterion in laminar mode is depicted, Fig. 11.5 the graph of the dependence of the laminar flow regime on the Nusselt criterion is depicted when the heat carrier consumption is around 1 t/s, Fig. 11.6 the graph of the dependence of velocity on the Nusselt criterion in turbulent flow regime is depicted, Fig. 11.7 the graph of the dependence

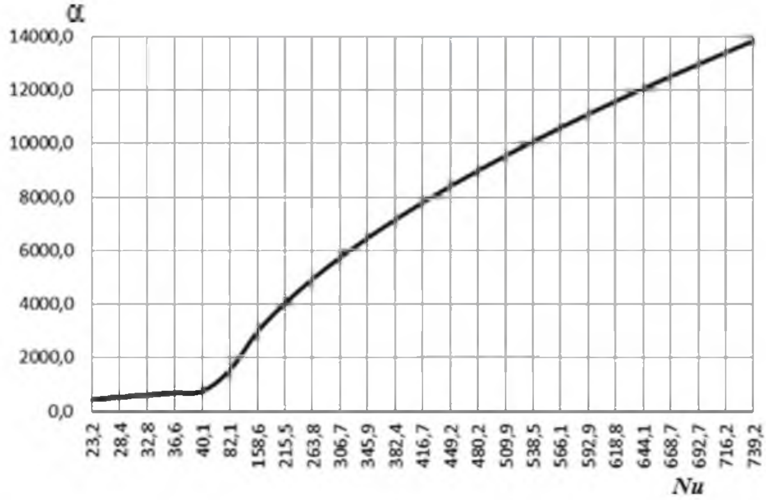


Fig. 11.3 The graph of the dependence of the heat transfer coefficient on the Nusselt criterion when the flow regime is laminar and turbulent

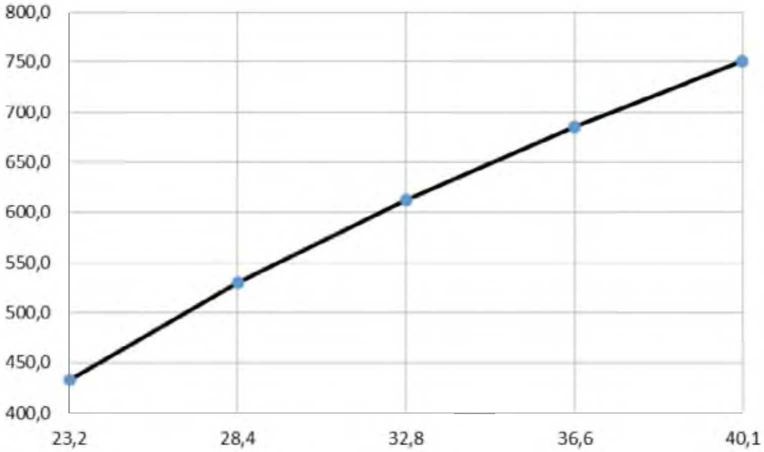
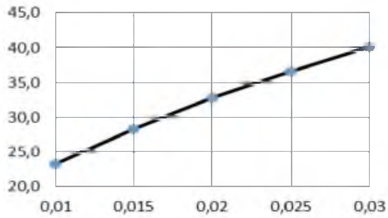


Fig. 11.4 Graph of the dependence of the heat transfer coefficient on the Nusselt criterion in laminar mode

Fig. 11.5 Graph of dependence of the laminar flow order on the Nusselt criterion when the heat carrier consumption is around 1 t/s



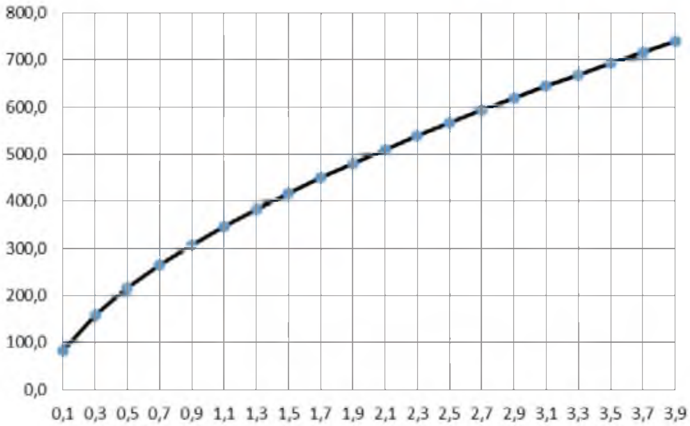


Fig. 11.6 The graph of the dependence of the speed on the Nusselt criterion in the turbulent flow regime

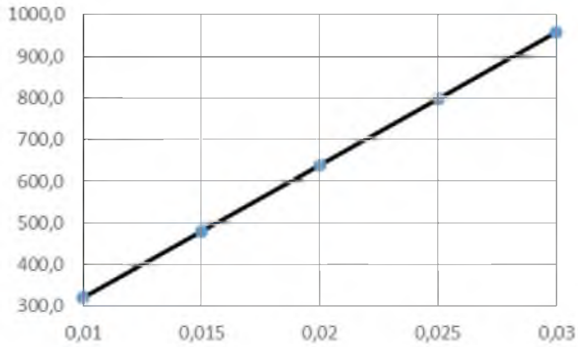


Fig. 11.7 Graph of the dependence of the velocity on the Reynolds number in the laminar flow regime

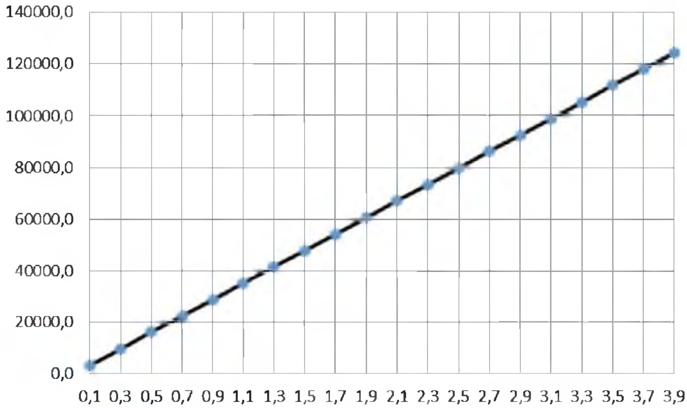


Fig. 11.8 Criterion (Reynolds) graph of the dependence of the speed of the heat carrier on the flow pattern

of velocity on Reynolds number in laminar flow regime is depicted and Fig. 11.8 Criterion (Reynolds) graph of the dependence of the speed of the heat carrier on the flow pattern is depicted.

The heat transfer coefficient of the heat carrier is used to describe the intensity of the heat exchange process. The numerical value of the heat transfer coefficient is equal to the heat flow generated per unit surface under the influence of temperature pressure (α, W/(m^2 K)), that is, it indicates the amount of heat transferred from the calculated surface unit in a unit of time. The heat transfer coefficient depends on the flow rate of the heat carrier, the type of flow, the geometry of the surface, etc.

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