



Jiyenbeck SUGIROV¹, Bibigul ATSHYBAYEVA², Marzhan SUIMENOVA³, Kulanda SHAIKHIYEVA⁴,
Gulbanu YESBOLAY⁵

Mathematical model of the intensification of convective heat transfer in a bundle of smooth pipes using petal turbulators

ABSTRACT: The relevance of this study is explained by the growing interest in increasing heat transfer by the development of high-performance thermal systems. Increasing the thermal characteristics of heat-exchanger systems is necessary for the efficient use of an energy source. The purpose of this study is to review the existing methods of heat-transfer intensification and examine the mathematical model of such an increase in efficiency when using petal turbulators. This study is based on a high-quality, reliable combination of proven theoretical methods (analysis, synthesis, concreti-

✉ Corresponding Author: Jiyenbeck Sugirov; e-mail: jiy_sugirov@academics.in.net

¹ Department of Construction Engineering, Caspian University of Technology and Engineering named after Sh. Yessenov, Republic of Kazakhstan; ORCID iD: 0000-0002-8109-1658; e-mail: jiy_sugirov@academics.in.net

² Department of Energy and Transport, Caspian University of Technology and Engineering named after Sh. Yessenov, Republic of Kazakhstan; ORCID iD: 0000-0002-2715-8209; e-mail: bi.atshybayeva@scholars.pl

³ Department of Construction Engineering, Caspian University of Technology and Engineering named after Sh. Yessenov, Republic of Kazakhstan; ORCID iD: 0000-0003-3846-2035; e-mail: m.suimenova@scientificcommittee.com

⁴ Department of Construction Engineering, Caspian University of Technology and Engineering named after Sh. Yessenov, Republic of Kazakhstan; ORCID iD: 0000-0002-4320-3639; e-mail: kul.shaiKhiyeva@scientificlaws.com.de

⁵ Department of Construction Engineering, Caspian University of Technology and Engineering named after Sh. Yessenov, Republic of Kazakhstan; ORCID iD: 0000-0002-5026-179X; e-mail: gulb.yesbolay@scientificangle.in.net



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zation, generalization, modelling), and empirical methods. It is the introduction of turbulators into the flow channel that is one of the best methods of increasing passive heat exchange through such advantages as ease of manufacture and operation in combination with low operating and production costs. This study contains both passive and active methods of heat-exchange intensification that have been extensively investigated over the past decade. For this purpose, the newest studies of mainly authors from other countries were used, their detailed analysis was conducted and the results were summed up. In addition, a mathematical model of increasing the thermal efficiency of convective heating surfaces in a bundle of smooth pipes using petal turbulators was investigated, the results of which were tested on an experimental installation. The paper may interest a circle of readers interested in the problem of improving the thermal characteristics of heat exchangers, including researchers, teachers and students of higher educational institutions in the field of heat-power engineering.

KEYWORDS: turbulence, heat boilers, increased heat transfer, flow friction, swirling flow

Introduction

The issues of energy saving and improving heat transfer are becoming increasingly important due to almost depleted energy resources. One of the main reserves for reducing specific fuel costs per unit of heat produced is to increase the efficiency of boilers. This particularly applies to increasing the use of exhaust heat as it is the reason for up to 20% of thermal energy loss. This study covers various types of heating boilers in different regions. In particular, the warm climate of Southern Europe and Central Asia is considered, in which most industrial heating boilers do not need to work at full capacity throughout the year. Exhaust-gas velocities may be underestimated and the processes of convective heat exchange in the tail surfaces may differ from the design ones. Heat exchangers are one of the main components and directly affect the thermal efficiency of systems. It is desirable to conduct an artificial intensification of heat exchange through the installation of various turbulators in air heaters and boiler economizers to save fuel. Turbulators enhance heat exchange, and in some cases, they change the direction of the heat flow impact, creating the effect of “self-cleaning” of loose ash and sediments on convective surfaces (Polukarov et al. 2021; Tanirbergenova et al. 2021). They also substantially improve the heat-transfer characteristics but increase the friction of the liquid on the contacting surfaces and the pressure drop, which depends on the thermophysical characteristics of the liquid. In addition, due to fuel economy and almost complete combustion, the volume of emissions into the environment will decrease (Yu et al. 2022).

Many engineers have drawn attention to the development of the heat transfer rate of compact heat exchangers in the gas-gas and gas-liquid modes. It is well known that the thermal conductivity of a gas is lower than the thermal conductivity of a liquid, and this fact helps to improve the coefficient of convective heat transfer. Research on the tail surfaces is united by the fact that

the temperature difference is quite low, especially at the entrance to the air heater and at the exit from the cold part of the economizer (Batrakov and Zhuk 1994; Kostin et al. 2005). Because of this, the main goal in the development of new designs of tail heating surfaces is to find all ways to intensify the heat exchange in them, to come to the creation of small-sized elements with a low metal cost, while observing minimal ash wear, corrosion and skidding. The artificial turbulence of the free flow in the range of small numbers $Re < 1 \times 10^4$ (the area of mixed flow) is poorly investigated in the literature. The biggest difference in the researchers' studies was the discrepancies in the calculation of heat transfer coefficients and aerodynamic resistances, and especially the values of the proportionality coefficients "C" and "K". Correction coefficients for the number of rows of a bundle of pipes and the geometry of the bundle have similar values for most of the authors.

The effect of artificial turbulence depended on the method of influencing the flow. Thus, Eygens (1940), Prandtl (2000), Gukhman (2010) considered the influence of various flow turbulization methods on a single pipe (the use of threaded grooves in pipes, the use of turbulizing grids, wires, etc.) in some cases led to an increase in the heat transfer of the pipeline up to 40% and a decrease in aerodynamic drag of up to 30%. However, for a tube bundle, according to many researchers, the impact of artificial turbulence is already less substantial than when exposed to an independent pipe. Thus, using various turbulization methods (using well devices, turning the beam at a certain angle, installing partitions in the most convenient places of the beam, installing turbulizing pipes before the main beam, etc.), led to an increase in heat transfer in the tube beam by 15%, and the aerodynamic resistance deviated slightly, depending on the type of turbulence. In addition, a more substantial baffle effect was observed in bundles with a small number of rows of tubes. Explanations of the intensification of transfer processes in channels of various types are often absent or do not agree with many authors; therefore, in this paper, the materials are considered as a basis (Sugirov 2018; Sugirov et al. 2021). In the process of researching and searching for material, it was concluded that the study by these authors is one of the most accurate and up-to-date in this field.

The purpose of this study is to review the existing methods of heat-transfer intensification and examine the mathematical model of such an increase in efficiency when using petal turbulators.

1. Materials and methods

This study is based on a high-quality, reliable combination of proven theoretical methods (analysis, synthesis, concretization, generalization, modelling), and empirical methods (the investigation of experimental studies of researchers and their experience in this or a similar field using similar purpose designs and their analysis by experienced specialists). The theoretical basis of this study mainly consists of case studies consisting of scientific and practical research and review papers, including experimental methods and descriptions of experiments. Analyses,

statistics and other types of work by researchers from other countries, taken from official sources with reliable information, were aimed at investigating a large number of problematic issues that can be encountered in the application of illuminated methods related to the mathematical model of convective heating surfaces when using petal turbulators. The study was conducted in three stages.

In the first stage, the main theoretical basis was prepared, on the analysis of which, the foundation for further conclusions was built. A search and analysis of various reliable sources of information were conducted. A large number of information resources devoted to the intensification of heat exchange, their mathematical models, types of turbulators and their components were considered and filtered. The collected data were investigated and systematized for a simplified, fast, and high-quality understanding of the information. A systematic analysis of the operation and assessment of the condition of the boiler house equipment in operation, existing methods of increasing intensification, and research for new ways of intensifying heat exchange in convective heating surfaces were conducted. The main issues that can be effectively solved through the practical use of modern methods and technologies are outlined.

In the second stage of the study, an analytical comparison and deepening into the materials of the examined studies of other researchers was conducted, comparing them with each other. They mostly included statistics, theoretical information and descriptions of the experience of using various methods to increase heat-transfer coefficients. Experiments, mainly by researchers from other countries, were analyzed in detail on the subjects of turbulator performance, their effect on heat exchange, and variations in their structures and materials. The development of a mathematical model of convective heating surfaces using petal turbulators was investigated. The conclusions were analyzed, verified, and clarified.

In the third stage, material analysis was completed, theoretical and practical conclusions were clarified, and the obtained results were generalized and systematized. The results obtained in the course of this investigation and the conclusions formulated on their basis can be used in future studies of the potential for improving new methods and technologies for increasing the intensification of convective heat transfer in a bundle of smooth pipes when using petal turbulators.

2. Results

In this section, the compilation of a mathematical model in a tube bundle is considered, and the physical foundations of the processes are investigated, based on previous papers (Sugirov 2018; Sugirov et al. 2021). Based on the laws of thermodynamics and transfer phenomena, methods of hydraulic and thermal calculation were created. In physics, such transfer phenomena can be called irreversible processes of a statistical nature, which occur when molecules continuously move and occur mostly in liquids. The phenomenon of transfer is based on several fundamental concepts, such as conservation laws, which in connection with the phenomena of

transfer, should be formulated as continuity equations, and defining equations. Fourier, Fick, and Newton established the laws of mass, heat, and momentum transfer:

$$\begin{aligned}
 \check{\Phi} &= -\lambda \frac{\partial T}{\partial z} = \frac{1}{3} \check{\rho} \check{c}_v \omega_m \sigma \frac{\partial T}{\partial z}; \\
 \check{\mu} &= -\check{\rho} \check{\delta} \frac{\partial \check{x}_j}{\partial z} = \frac{1}{3} \check{\rho} \omega_m \sigma \frac{\partial \check{x}_j}{\partial z}; \\
 \tau &= -\eta \frac{\partial u}{\partial z} = \frac{1}{3} \check{\rho} \check{M} \omega_m \sigma \frac{\partial u}{\partial z}
 \end{aligned} \tag{1}$$

where:

- $\check{\rho}$ – the molar density,
- ω_m – the average velocity of gas molecules,
- σ – the average free path of molecules,
- \check{M} – the mass of one mole,
- $\partial T / \partial z$ – the temperature gradient,
- $\partial \check{x}_j / \partial z$ – the gradient of the molar fraction of component j ,
- $\partial u / \partial z$ – the velocity gradient.

The presented equations show a fundamental connection between the phenomena of transfer and thermodynamics, and it explains why this phenomenon cannot be reversed. Most of the physical phenomena are ultimately associated with systems that seek their lowest energy state using the principle of minimum energy. The closer they get to such a state, the more they try to achieve true thermodynamic equilibrium, during which transport stops and all the forces of motion are absent. All moments of such a balance are directly related to a certain transfer, where heat transfer is the attempts made by the system to get into thermal equilibrium with its environment, according to the same principle as the movement of impulses and mass transfers the system to mechanical and chemical balances.

Turbulence limits the application of equations for flows (1), molecular transfer coefficients λ_t , σ and μ_t . This can be the result of arbitrary movements of turbulent vortices in liquids and gases. The size of these vortices can be from a millimeter (in the heat exchanger tubes), up to meters (in the atmosphere) and is always several percent of the total size of the system. The authors consider the fluid motion in a heat exchanger with a turbulator as an averaged large-scale motion. When the motion changes slightly in time or is stationary, it is more effective for engineering calculations to consider time-averaged equations. When considering the mathematical model of a heat exchanger with a turbulator, it is necessary to classify the flow in this heat exchanger as a flow in straight channels or pipes. Such a model will consist of a section of a rectangular channel with a turbulizing device of length $(l_1 + l_2)$ (Fig. 1) (Sugirov et al. 2021).

The Reynolds equations for averaged turbulent flow in a rectangular Cartesian coordinate system are considered in describing the turbulent flow of fluid in the installation coordinates:

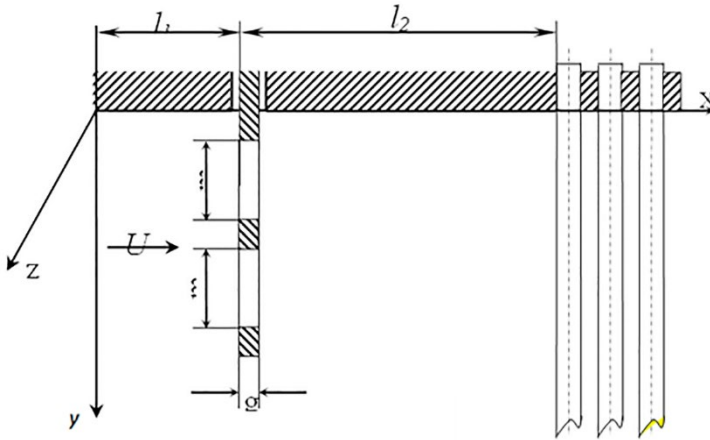


Fig. 1. Scheme of the system with a turbulizing device in the form of partitions with petal cutouts

Rys. 1. Schemat układu z urządzeniem turbulizującym w postaci przegród z wyciętymi płatkami

$$\begin{cases} \rho \left(U \frac{\partial U}{\partial x} + V \frac{\partial U}{\partial y} + W \frac{\partial U}{\partial z} \right) = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x} \tau_{xx} + \frac{\partial}{\partial y} \tau_{xy} + \frac{\partial}{\partial z} \tau_{xz}; \\ \rho \left(U \frac{\partial V}{\partial x} + V \frac{\partial V}{\partial y} + W \frac{\partial V}{\partial z} \right) = -\frac{\partial p}{\partial y} + \frac{\partial}{\partial x} \tau_{yx} + \frac{\partial}{\partial y} \tau_{yy} + \frac{\partial}{\partial z} \tau_{yz}; \\ \rho \left(U \frac{\partial W}{\partial x} + V \frac{\partial W}{\partial y} + W \frac{\partial W}{\partial z} \right) = -\frac{\partial p}{\partial z} + \frac{\partial}{\partial x} \tau_{zx} + \frac{\partial}{\partial y} \tau_{zy} + \frac{\partial}{\partial z} \tau_{zz}; \end{cases} \quad (2)$$

where:

U, V, W – the components of the velocity vector in the direction of the x, y and z axes (Fig. 1),

p – static pressure,

ρ – the density of the liquid (Sugirov et al. 2021).

Differential equations have resulted in a generalized form:

$$\frac{\partial}{\partial x}(UF) + \frac{\partial}{\partial y}(VF) = \frac{\partial}{\partial x} \left(G_F \frac{\partial F}{\partial x} \right) + \frac{\partial}{\partial y} \left(G_F \frac{\partial F}{\partial x} \right) + S_F \quad (3)$$

where:

the variable F denotes alternately U, V, k and ε , and the coefficients G_F and S_F depend on the kind of variable.

The finite difference method, by approximation using difference equations, is used to solve differential equations (3) based on the Patankar method. The investigated system with a turbulating partition was separated by a grid with a step of Δx and Δy , oriented respectively along the x- and y-axis. In the case of a partition, the location of the nodes of the different grids is indicated in Figure 2 (Sugirov et al. 2021). The differential equation (3), will take the form:

$$\begin{aligned} & \frac{(U_j^{i+1} - U_j^i)(F_j^{i+1} - F_j^i)}{\Delta x} + \frac{(V_{j+1}^i - V_j^i)(F_{j+1}^i - F_j^i)}{\Delta y} = \\ & = G_F \frac{F_j^{i+1} - 2F_j^i + F_j^{i-1}}{\Delta x^2} + G_F \frac{F_{j+1}^i - 2F_j^i + F_{j-1}^i}{\Delta y^2} + S_F \end{aligned} \quad (4)$$

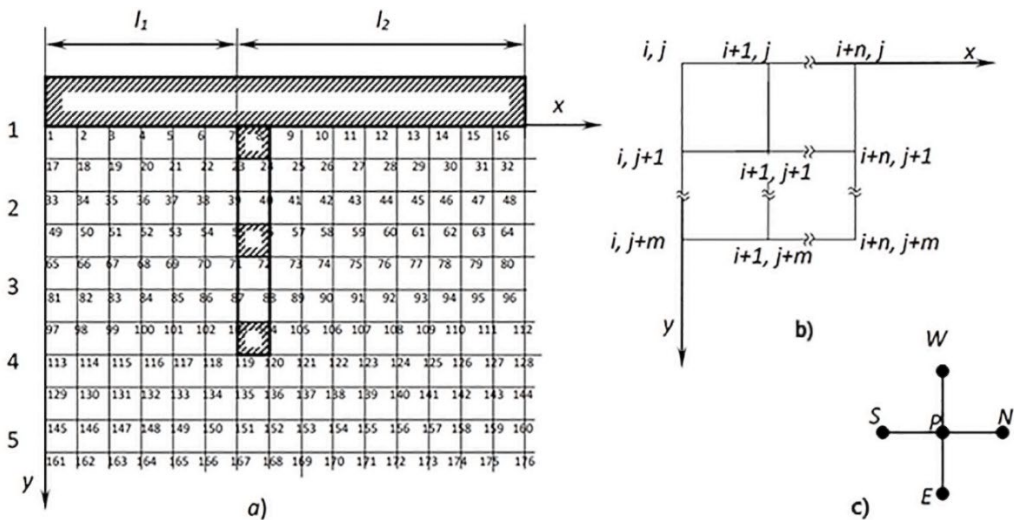


Fig. 2. Location of the difference grid

a – the location of the nodes of the difference scheme; b – the numbering of nodes; c – the designation of adjacent points (Sugirov et al. 2021)

Rys. 2. Lokalizacja siatki różnic

The necessary boundary conditions were established. Since the system consists of a coolant moving in space bounded by the walls of the wind tunnel, the surface of the turbulating device and the section at the inlet and outlet at a distance $(l_1 + l_2)$. For all variables at the input, the Dirichlet condition $F(x, y) = F_o(x_o, y_o)$ is set for a developed turbulent flow. The condition is assumed that on the walls of the aerodynamic tube and the surface of the turbulating device $U = V = 0$, and for k and ϵ , the dependences of the logarithmic law of velocity distribution $di = (i/k)$ (di/dl) are applied. At the boundaries of the mathematical model, the Neumann condition is set for all variables, $dF/dx = dF/du = 0$. Approximating the values of difference equations, differen-

tial equations take an algebraic form, which has the same form for each variable. A system of algebraic equations is obtained:

$$\begin{cases} a_P^U = a_E^U U_E + a_W^U U_W + a_N^U U_N + a_S^U U_S + S_U^U \\ a_P^V = a_E^V V_E + a_W^V V_W + a_N^V V_N + a_S^V V_S + S_U^V \\ a_P^k = a_E^k k_E + a_W^k k_W + a_N^k k_N + a_S^k k_S + S_U^k \\ a_P^\varepsilon = a_E^\varepsilon \varepsilon_E + a_W^\varepsilon \varepsilon_W + a_N^\varepsilon \varepsilon_N + a_S^\varepsilon \varepsilon_S + S_U^\varepsilon \end{cases} \quad (5)$$

where:

- $a^{U, V, k, \varepsilon}$ and $S^{U, V, k, \varepsilon}$ – coefficients that take values depending on the variables U, V, k, ε ,
- P, E, W, N, S – the numbering of nodes located next to each other (Fig. 2c).

In matrix form, the equations that are formed from equation (4) will look like this:

$$H = a\Theta + S \quad (6)$$

where:

$$H = \begin{bmatrix} a_P^U \\ a_P^V \\ a_P^k \\ a_P^\varepsilon \end{bmatrix} a = \begin{bmatrix} a_E^U & a_W^U & a_N^U & a_S^U \\ a_E^V & a_W^V & a_N^V & a_S^V \\ a_E^k & a_W^k & a_N^k & a_S^k \\ a_E^\varepsilon & a_W^\varepsilon & a_N^\varepsilon & a_S^\varepsilon \end{bmatrix} \Theta = \begin{bmatrix} V_n^U \\ V_n^V \\ k_n^k \\ \varepsilon_n^\varepsilon \end{bmatrix} s = \begin{bmatrix} S_U^U \\ S_U^V \\ S_U^k \\ S_U^\varepsilon \end{bmatrix}$$

Systems of difference equations (6) form matrices, of which the matrix of coefficients P , is a matrix of variables. The Gauss-Seidel method was used to reverse the matrices. Velocity distribution profiles were obtained before and after the intensifying partition for variants $\delta = 0.15, 0.3$, and 0.5 when the size of the cutouts m changed from 22 to 26 mm (Fig. 3) (Sugirov et al. 2021), based on previously performed calculations to determine the dimensions of the experimental setup. Considering Figure 3, it was concluded that before the partitions, the profile of the speed remains even, and after the partitions, the speeds change quite sharply. In places where the flow has to collide with the partition, some counteractions form vortices. In places where the flow penetrates through holes in the obstacle, the speeds increase and their profiles remain strictly parabolic. Velocity profiles, which are calculated through specific identical distances ($l_1 + 10n_i$) where $n_i = 2, 4, 6$ for c, d, e , make it clear that the velocity dips that were formed by back pressure are smoothed over time (Sections c-d), and the flow gradually stabilizes.

According to the graph from Figure 3, it can be seen that the change in the size of the notch m slightly affects the speed changes. The largest increase in velocity, and subsequently the increase in flow turbulence, can be observed when $m = 24$ mm. In addition, according to the graph from Figure 3, it can be seen that on A, B and C, the degree of overlap of the flue δ has a substantial

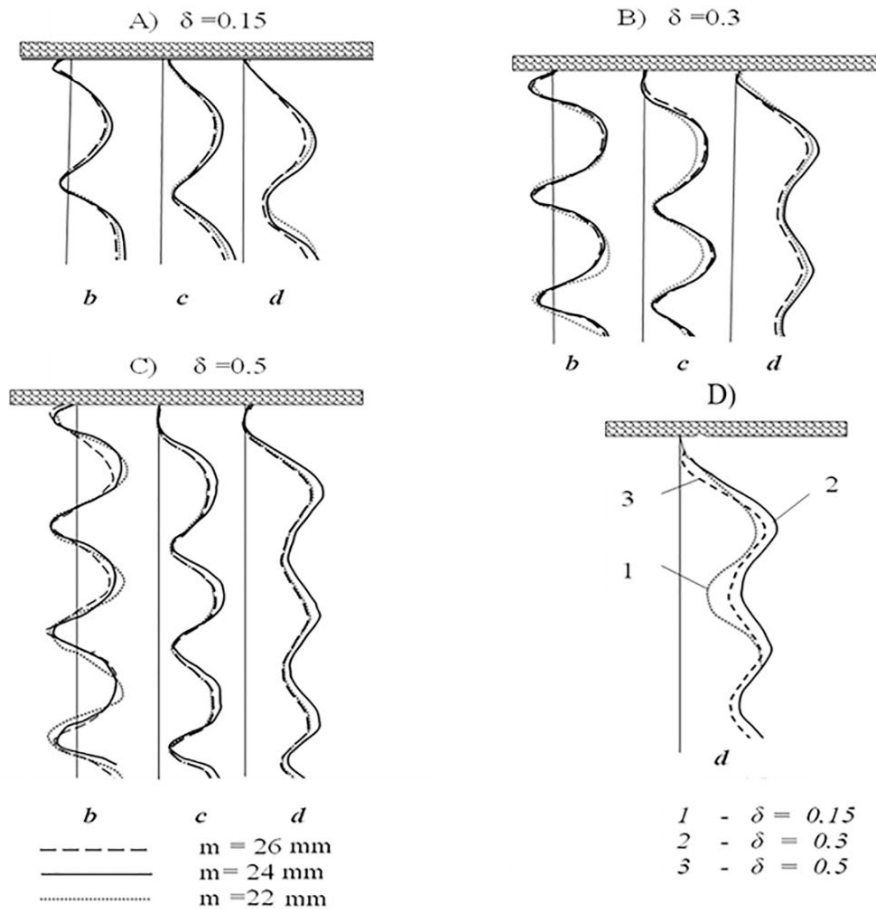


Fig. 3. Comparison of velocity distribution profiles after an intensifying partition at $\delta = 0.15, 0.3$ and 0.5 and a change in the size of the recess *m* from 22 to 26 mm

Rys. 3. Porównanie profili rozkładu prędkości po podziale intensyfikującym przy $\delta = 0,15; 0,3; 0,5$ i zmianę wielkości wgłębienia *m* z 22 na 26 mm

effect on the speed change. Its largest increase can be observed when the partition is extended to a distance of $\delta = 0.3$, the size of the recess *m* is 24 mm and the distance of the turbulator installation to the beam is 40 mm (Section d, Fig. 3D). Therefore, the greatest turbulence of the heat flow will occur here. A similarity coefficient of 1:10 for the double-drum water-tube boiler reconstructed is used and the transverse size of the flue of the experimental installation is assumed to be 210 x 120 mm. As a result, during the manufacturing process of this experimental setup, certain results of mathematical modelling and similarity theory were accepted, such as making the transverse size of the flue 210 x 120 mm, installing a partition at a distance of 40 mm to the beam, and the greatest turbulence will occur when the partition is extended to a distance of 63 mm, with the size of the recess *m* = 24 mm.

The results of mathematical modelling were tested on an experimental setup. The values of velocities were measured with a Prandtl tube at a distance of 20, 40 and 60 mm from the turbulator (Sections c, d, e) and compared with the results of a theoretical study in which the extension of the partition was at a distance $\delta = 0.3$, $l = 63$ mm, the size of the recess was $m = 24$ mm and the installation distance of the turbulator to the beam was $d = 40$ mm. As presented in Figure 4, the experimental data obtained almost coincided with the theoretical data, and this confirmed the correctness of the development of the model (Sugirov et al. 2021).

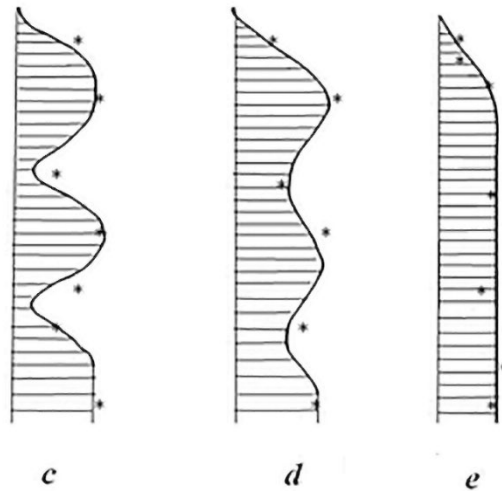


Fig. 4. Comparison of theoretical and experimental values

* – experimental points

Rys. 4. Porównanie wartości teoretycznych i doświadczalnych

3. Discussion

A heat exchanger is a piece of equipment that transfers the heat energy between two liquids with different temperatures when mixed (Erbay et al. 2021). The heat exchanger is widely used in many industries (Promvong and Skullong 2019). The cost of energy, materials and the improvement of heat transfer is a subject of great interest for working on technologies to increase the speed of heat transfer and obtain higher thermal efficiency, but as a result, the pump power increases during the process and its cost becomes higher (Li et al. 2020; Kunik et al. 2016). Heat-transfer improvement technologies play an important role in laminar flow heat transfer, usually laminar flow heat-transfer coefficients of smooth pipes (Salhi et al. 2020; Sarafraz et al. 2019). The heat transfer rate can be improved by disrupting the fluid flow, which can be

achieved by inserting a twisted tape/turbulator into a round tube. The introduction of turbulators into the flow part is one of the best ways to increase heat transfer due to the advantages in the ease of manufacture and application, including low operating costs (Promvongse and Skullong 2020; Sarafraz et al. 2019a). In general, the performance of turbulators strongly depends on their geometry. Increasing convective heat transfer in internal flows with tape inserts in pipes is a well-proven technique used in production practice. Improving heat transfer refers to improving the thermal-hydraulic performance of heat exchangers. The existing methods of improvement can be divided into three different categories: A – passive techniques; B – active methods; C – composite methods.

A – passive techniques. These techniques usually use surface or geometric modifications of the flow channel through the inclusion of inserts or additional devices. They contribute to an increase in the heat-transfer coefficient by disrupting or changing the existing flow behavior (except for the expanded surface), which also leads to an increase in the incident pressure (Sobhan et al. 1994). In the case of extended surfaces for effective heat transfer, the area on the side of the expanded surface increases. Passive techniques have an advantage over active methods since they do not require direct input from external power (Yildiz et al. 1998).

B – active methods. These methods are more difficult to use from a design standpoint since the method requires some external power source to trigger the desired flow of modifications and improvements in the heat-transfer rate. It has limited use because of the need for external power in many practical applications (Durmus 2004; Asulyuk et al. 2017). Compared to passive methods, these methods have not shown much potential as it is difficult to provide external power in many cases.

C – composite methods. Combined augmentation technique, where active and passive methods are used together to further improve the thermal and hydraulic characteristics of the heat exchanger.

Various types of twisted tape: metal strips twisted to the desired shape and size, inserted into the thread. The following are the main categories of twisted tape: twisted tape for the entire length; twisted tape of various lengths; regularly arranged twisted ribbons; tape with attached partitions; tapes with slots; tapes with holes; tapes with various surface modifications. Sobhan et al. (1994) experimentally investigated a shell-and-tube heat exchanger to examine the performance of spiral turbulators. Experiments were conducted with different diameters of winding wire, and the pitch and heat-transfer coefficient were evaluated for a wide range of temperature levels and the flow rate of the inter-tube fluid. The Nusselt number (the Nusselt number is one of the main criteria for the similarity of thermal processes, characterizing the ratio between the intensity of heat exchange through convection and the intensity of heat exchange through thermal conductivity) was used for various winding steps. It is clear that the use of turbulators substantially affects the increase in productivity in the investigated range of the Reynolds numbers (the Reynolds number is a dimensionless quantity characterizing the ratio of inertial forces to viscous friction forces in viscous liquids and gases), which are in a laminar mode. An improvement of up to 70% compared to a bare tube was observed in the external Nusselt number. In the various thread steps used, the best indicators were observed for the case with a winding step of 4 cm,

which indicates its optimality. It has been found that increasing the length of the wire diameter from 1.34 to 1.65 mm improves the total Nusselt number to some extent. The wire of a larger diameter heats up a little better, which increases the heat-transfer coefficient.

Yildiz et al. (1998) placed twisted, narrow, thin, metal strips in the inner tube of a concentric two-tube heat exchanger and investigated their effect on the heat transfer and pressure drop for parallel and counterflow. These turbulators were manufactured by twisting strips at certain angles and are designed to touch the inner wall at every step. In the systems, hot air passed through the inner tube, while cold water flowed through the annular space. Experiments were conducted with an empty inner tube with a Reynolds number of between 3400 and 6900. The effect of turbulators on heat transfer is more pronounced for high Reynolds numbers. Improvements for parallel flow show a parallel direction and are only 10% lower than for counterflow. An increase of about 1.3 times compared to an empty tube with the highest Reynolds number for a pitch of 170 mm.

Kongkaitpaiboon et al. (2010) performed an experimental study of the loss of the convective heat exchange of pressure in a round tube equipped with circular annular turbulators (CAT). They investigated the effect of a circular annular turbulator on heat exchange and the characteristics of liquid friction in the heat-exchanger tube. The experiments were conducted by introducing CAT of various geometries, including three different diameter ratios ($DR = d/D = 0.5, 0.6$ and 0.7) and three different height ratios ($PR = p/D = 6, 8$ and 12). During the tests, air at a temperature of 27°C was passed through a test tube, which was controlled with a homogeneous heat flow barrier. The Reynolds number ranged from 4,000 to 20,000. According to the experimental results, the heat-transfer rate in the tube with CAT increased from about 57 to 195% compared to that in a simple tube, depending on the operating conditions. The influence of the diameter ratio (DR) and the step ratio (PR), the effect on the heat-transfer rate, friction, the behavior of the factor and the thermal coefficient of productivity were investigated under conditions of a uniform thermal barrier flow. CAT with different diameter ratios ($DR = d/D = 0.5, 0.6$ and 0.7) and pitch ratios (6, 8 and 12) was used for the Reynolds number in the range of 4,000 to 20,000. Over the entire range of CAT studies, they offer an improvement in heat transfer from 57 to 195% compared to a simple tube. A maximum thermal efficiency coefficient of 1.07 was found when using a cathode-ray tube with $DR = 0.7$ and $PR = 6$.

Salimpour and Yarmohammadi (2012) conducted an experimental study to determine the effect of twisted tape inserts on the pressure drop during the forced convective condensation of vapors in a horizontal pipe. A set of five tube, with a twist coefficient of 4 has the highest pressure drop. A decrease in the twist coefficient causes a higher disturbance in the liquid film and the vapor core. The condensation pressure drop for a set of five tubes is up to 239% more than for a simple tube and the mass velocity of the refrigerant is $106.8 \text{ kg (M/c}^2\text{)}^{-1}$. A set of two tubes has the lowest pressure drop range, which increases the pressure drop to 151% compared to a smooth tube for a refrigerant mass velocity of $89 \text{ kg (M/c}^2\text{)}^{-1}$. This correlation predicts experimental data with an error range of $\pm 20\%$.

Durmus et al. (2004) investigated the effect of propeller-type turbulators, which were located in a plate tube of a coaxial heat exchanger. The angular blade of the turbulator propeller is be-

tween $10 \leq \theta \leq 40$, the ratio of the propeller diameter to the pipe diameter is between $0.87 \leq Dk$ and $Db \leq 0.94$, (Dk is the propeller diameter, Db is the pipe diameter) and the Reynolds number in the range of 10,000 and 30,000. The turbulator seat with 10 cm increased heat transfer by as much as 28 and 39%, and pressure loss by as much as 17 and 43% compared to turbulators of 20 and 30 cm. A turbulator with a 20° blade angle reduces heat dissipation by 8%. The blade angle is 40° , up to 35% compared to turbulators with a blade angle of 10° . The change in the angle of the blade affected the pressure with losses between 15–40%. In the experiment, it was discovered that the heat exchange is 2–4 times higher, and the pressure loses 8.5 times more than the values of the empty tube of the heat exchanger on the Reynolds number of 10,000–30,000 with different mass flow.

Zamankhan (2010) developed a three-dimensional mathematical model for the examination of augmentation heat transfer in a circular tube with a spiral turbulator. Glycol-water mixtures of various concentrations were used in the inner tube, and pure water was used in the outer tube. Changes in physical properties in a fluid with a temperature change were considered, and k - ε ; k - ω (k is the wave number, ε is the average velocity of turbulence dispersion, ω is the energy dissipation per unit volume) and the modelling of large vortices was performed to simulate disturbances. The simulation results were obtained through a nonlinear change in the Reynolds and Prandtl numbers for a long heat-exchanger model even in the absence of a turbulator. It has been discovered that the presence of a turbulator increases heat transfer, sometimes without turbulence induction, but also increases pressure drop. Comparing their numerical results with those obtained experimentally revealed that the LES model (large eddy simulation) predicts the behavior of real systems. Using a multi-purpose optimization method, such as a genetic algorithm in combination with GPU (graphic processing units)-LES-SPH (smoothed particles hydrodynamics), a set of solutions is obtained that satisfy various levels of compromise. The most suitable solution will be selected from this set. This developed efficiency improvement tool will be particularly suitable for process intensification (i.e. increasing the production capacity per unit volume of equipment) in the chemical industry.

The Promvonge (2008) company presented experimental results on the heat and friction characteristics of the flow in a tube with a uniform heat flow equipped with a 5-mm wire coil with three different coil steps of the ratio $CR = 4, 6$ and 8 and two twisted tapes, the twist coefficients $Y = 4$ and 6 are represented as the Nusselt number and the friction coefficient. The Nusselt numbers are obtained under turbulent flow conditions for twisted tape and spiral wire with three ratios. Notably, the combined wire coil and twisted tape with $CR:Y = 4:4$ provides a higher heat-transfer rate than those with higher $CR:Y$ values for all Reynolds numbers. The increased numerical value of the Nusselt was in the range of 300–685% in the flow of a smooth tube. The coefficient of friction for $CR:Y = 4:4$ turned out to be twice as high as for $CR:Y = 6:4$ or $8:4$, and the coefficient of friction for $CR:Y = 4:6$, was considered as the average value between the values for $CR:Y = 4:4$ and $6:4$. This means that friction losses mainly occur from the use of a wire coil with $CR = 4$, and the average increase in the coefficient of friction is in the range of 30 to 68 times compared to a smooth tube. The heat transfer obtained from both the coil wire and the twisted tape insert was about 130–250% and 180–400%, depending on the Reynolds number.

The increase in the Nusselt number for twisted tapes was about 20–50%, while the coil with wire was about 100–110%. The use of combined twisted tape and wire insert coils also shows a higher heat-transfer rate of about 150–300%. The value of the coefficient of friction for both was about 2–4.5 times higher than for one coil of wire or about 6–12.5 times more than for a twisted tape.

Naga Sarada et al. (2010) presented an experimental study of the potential of a width-reduced insert made of twisted tape to increase the heat transfer rate in a horizontal circular pipe with an internal diameter of 27.5 mm with air as a working fluid. The twisted tapes had three different twist coefficients (3, 4 and 5), each with a different width has 5 types of widths (26–full width, 22, 18, 14 and 10 mm), respectively. The Reynolds' number ranged from 6,000 to 13,500. The percentage of increase in the Nusselt number for tapes of reduced width compared to a conventional tube was about 11–22, 16–31, 24–34 and 39–44 for tapes with widths of 10, 14, 18 and 22 mm, respectively, for the twist coefficient of 3. For full-width tapes, the percentage increase is from 58 to 70% compared to a conventional tube. The percentage increase in Nusselt numbers for a tape of reduced width compared to a conventional tube is about 5–12, 9–22, 13–30 and 23–36% for tapes with widths of 10, 14, 18 and 22 mm, respectively, for a twist coefficient of 4. For full-width tapes, an increase of 36–42% was observed compared to a simple tube. The percentage increase in the Nusselt number for tapes of reduced width compared to a conventional tube was about 2–8, 6–12, 9–19 and 14–27% for tapes with widths of 10, 14, 18 and 22 mm, respectively, for the twist coefficient of 5. For full-width tapes, there was a percentage increase from 22 to 37% compared to a conventional tube. The overall improvement in the ratio of tubes with twisted tape inserts for the entire width was 1.62 for the full width of 26 mm and 1.39 for the reduced width of the 22 mm twisted tape insert.

Eiamsa-ard and Promvongse (2007) conducted an experimental study to examine the behavior of temperature change, transmission losses and friction in a circular wave, in a welded tube with a spiral-tape insert using hot air as a test fluid. In general, the average heat transfer for a wavy tube with spiral tape turned out to be 23–35% better than in a tube with a wavy surface. The corresponding increase in the average Nusselt number for a pipe with a wavy surface with a spiral tape was approximately 330–422% compared to a conventional tube. For a pipe with a wavy surface with a spiral ribbon, the increase in the coefficient of friction was about 50%. The captured average heat-transfer rate for the used tape with a rod turned out to be 8–12% better than without a core rod. Thus, for a tape without a rod, it would be possible to reduce the coefficient of friction by about 50% compared to the tape with a core rod. The results of the existing correlation reasonably agree well within $\pm 10\%$ compared to experimental data for the coefficient of friction, and within $\pm 10\%$ for the Nusselt number. The maximum increase in the heat-transfer rate and the coefficient of friction was about 4.2 and 110 times compared to a conventional tube for the investigated range.

Selvam et al. (2013) conducted experiments on different twisting steps along the width of the twisted tape, and the ratio (y/w is the twist coefficient) for the propeller. A variation of the Nusselt number with the Reynolds number for a pipe equipped with a propeller with three different ratios. The results of the experiment demonstrated that the smaller the y/w ratio, the higher the heat-transfer values. The empirical correlations developed for pitch and Reynolds number

coincided with experimental data within ± 7.28 and $\pm 7.16\%$ for the Nusselt number and the coefficient of friction, respectively.

Conclusions

Increasing heat transfer is a subject of great interest for research that focuses on how to increase the heat-transfer rate and achieve higher efficiency. It is shown that today there is a large informational foundation for the research of various techniques and methods of intensification. Intensification has found wide application in heat exchange and boiler equipment. However, it is necessary to highlight the main problems. Many studies are characterized by the inconsistency of the results obtained. For some industrially promising heat-exchange intensifiers, there is no data on optimal geometric parameters and appropriate modes of their use. It is required to identify and analyse the main possible mechanisms of heat-transfer intensification and the creation and optimization of heat-transfer intensifiers based on fundamental studies of these mechanisms. Increasing convective heat transfer by inserting a twisted petal turbulator in a bundle of smooth pipes is one of the most effective and promising approaches. In this study, different types of turbulators were considered, new promising directions for improving the efficiency of using heat exchangers, the development of a mathematical model in a tube bundle and the determination of geometric parameters of an experimental installation were investigated.

The analysis of existing methods and their categories for improving the performance of heat exchangers was conducted. In addition, a theoretical substantiation and an analysis of experiments by other authors were conducted, which allows experimental confirmation of the possibilities and expediency of increasing the environmental, technical, and economic level of the boiler based on the application of the results of the set of studies. The use of the methods of heat-exchange intensification enables the conclusion to be drawn that tape and petal-type turbulators allow for the deeper cooling of combustion products, leading to substantial fuel savings in the boiler room and reducing the volume of toxic combustion products released into the atmosphere. The findings have been examined the experimental application of which confirms their reliability, efficiency, and the possibility of achieving high technical and economic indicators of boiler houses, and they have been implemented into industrial use.

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Jiyenbeck SUGIROV, Bibigul ATSHYBAYEVA, Marzhan SUIMENOVA, Kulanda SHAIKHIYEVA,
Gulbanu YESBOLAY

Model matematyczny intensyfikacji konwekcyjnej wymiany ciepła w wiązce rur gładkich z wykorzystaniem turbulatorów płatkowych

Streszczenie

Znaczenie tego badania wynika z rosnącego zainteresowania zwiększeniem wymiany ciepła poprzez rozwój wysokowydajnych systemów termicznych. Zwiększenie charakterystyki cieplnej układów wymienników ciepła jest niezbędne do efektywnego wykorzystania źródła energii. Celem niniejszej pracy jest przegląd istniejących metod intensyfikacji wymiany ciepła oraz zbadanie modelu matematycznego takiego wzrostu wydajności przy zastosowaniu turbulatorów płatkowych. Niniejsze opracowanie opiera się na wysokiej jakości, rzetelnym połączeniu sprawdzonych metod teoretycznych (analiza, synteza, konkretyzacja, uogólnienie, modelowanie) oraz metod empirycznych. Ponieważ to właśnie wprowadzenie zawirywacza do kanału przepływowego jest jedną z najlepszych metod zwiększenia biernej wymiany ciepła poprzez takie zalety jak łatwość wykonania i eksploatacji w połączeniu z niskimi kosztami eksploatacji i produkcji. Niniejsze opracowanie obejmuje zarówno pasywne, jak i aktywne metody intensyfikacji wymiany ciepła, które były szeroko badane w ciągu ostatniej dekady. W tym celu wykorzystano najnowsze badania, głównie autorów z innych krajów, dokonano ich szczegółowej analizy i podsumowano wyniki. Ponadto zbadano model matematyczny zwiększania sprawności cieplnej konwekcyjnych powierzchni grzewczych w wiązce rur gładkich za pomocą turbulatorów płatkowych, którego wyniki przetestowano na instalacji doświadczalnej. Artykuł może zainteresować grono czytelników zainteresowanych problematyką poprawy właściwości cieplnych wymienników ciepła, w tym naukowców, nauczycieli i studentów wyższych uczelni z zakresu elektroenergetyki.

SŁOWA KLUCZOWE: turbulencja, kotły ciepłone, zwiększona wymiana ciepła, tarcie przepływu, przepływ wirowy