Показано перевагу використання пластинчастих теплообмінних апаратів з різною висотою гофрування каналів в разі суттєвої відмінності витрат теплоносіїв. Це дозволяє зменшити площу поверхні теплопередачі, повністю реалізувати допустимі втрати тиску, збільшити швидкості в каналах, що підвищує опір забруднення пластин. Розглянуто конкретні приклади з проектування теплообмінників підготовки гарячої води в системі централізованого теплопостачання

Ключові слова: пластинчасті теплообмінники, висота гофрування, аналіз теплопередачі, опір забруднення, гаряче водопостачання

Показано преимущество использования пластинчатых теплообменных аппаратов с различной высотой гофрировки каналов в случае существенного отличия расходов теплоносителей. Это позволяет уменьшить площадь поверхности теплопередачи, полностью реализовать допустимые потери давления, увеличить скорости в каналах, что повышает сопротивление загрязнению пластин. Рассмотрены конкретные приложения по проектированию теплообменников подготовки горячей воды в системе централизованного теплоснабжения

Ключевые слова: пластинчатые теплообменники, высота гофрировки, анализ теплопередачи, сопротивление загрязнению, горячее водоснабжение

UDC 621.565.954.4
DOI: 10.15587/1729-4061.2018.120546

ADVANTAGES OF USING CHANNELS WITH DIFFERENT CORRUGATION HEIGHT IN THE PLATE HEAT EXCHANGERS

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1. Introduction

Traditionally, plate heat exchangers (PHE) have the same corrugation on the heating (hot) and heated (cold) sides of heat carriers. This is determined by their designs, special features of plates manufacturing, cost of creation and implementation in industry of plates with new geometric parameters and developed mathematical models for their design. The plates with the same corrugation form channels that are equal in area and their characteristics and behavior during operation have been sufficiently explored.

Currently, the exception is wide-channel heat exchangers that make it possible to combine channels with different flow sections on the hot and cold sides. Almost all manufacturers of heat exchange equipment produce heat exchangers of this type. These devices were designed as an alternative to spiral and welded cross-flow heat exchangers and are intended primarily to operate with contaminated media with various impurities such as fibers, particles etc. in the flow. Such apparatuses are produced with a unilateral and bilateral wide channel in the stack of plates. In the case of a unilateral plates' stack, a wide channel is used only by one of the heat carriers. For the bilateral design, a wide channel is provided from both carriers. In most applications, the use of this type of apparatuses requires implementation of a multi-pass composition to provide a rational velocity of heat carriers in channels.

A significant difference in consumption on the sides of cold and hot heat carriers when using heat exchangers of the same height and corrugation leads to operation with low velocity of a heat carrier on the side with low consumption. This fact triggers emergence and intensive growth of deposits on the heat transfer surface. Contamination, caused in this way, leads to disruption in operation mode of an apparatus and a forced halt for clean-up. In the case of using channels with different corrugation height (cross-section area), velocities in the channels are aligned and intensity of emergence and growth of contaminants falls sharply. The relevance and practical value lies in the fact that the proposed approach allows extension of operation term of a heat exchanger before it is stopped for clean-up. This makes it possible to provide continuity of the technological process and reduce operating costs.

There are a number of industrially produced plate heat exchangers that use an asymmetric structure of channels, for example, "Off-Set" design, manufactured by "Funke" company (Germany) [1]. According to "Funke", due to varying configuration of the channels' profile, the cross-section area on the sides of heat carriers may differ by one-third. This makes it possible to reduce the overall area of the heat transfer surface by up to 17 % in comparison with the standard symmetric configuration.

In the industry there are applications, particularly associated with recuperative usage of plate heat exchangers, when according to energy recovery conditions, there is a significant difference between consumption of the hot and cold heat carriers. In this case, when providing for heat transfer requirements, temperature mode of operation and satisfying conditions for loss of pressure in the apparatus, velocity in the channels of a heat exchanger drops sharply on the side of a heat carrier with lower consumption. This circumstance substantially affects a decrease in magnitude of heat transfer coefficient of this heat carrier and total heat transfer coefficient of the apparatus. In addition, a decrease in velocity in the channels leads to a decrease in magnitude of tangential stress on the wall and, as a consequence, to intensification of contamination of the heat exchange surface during operation. Thus, despite attractiveness of installation of the PHE apparatus in terms of energy recuperation (energy saving), its operation conditions are extremely unprofitable. A large heat exchange surface, determined by the use of a heat carrier with high consumption, leads to a decrease in velocity of the heat carrier on the side with low consumption.

A similar situation can occur when using a heat carrier with a very low coefficient of thermal conductivity on one side. This leads to a significant decrease in heat transfer coefficient and an increase in the plate stack (heat exchange surface) for provision of parameters for functioning of the apparatus. Approximately the same picture is observed in the case when one of the heat carriers has significantly higher viscosity than another, which requires designing a large number of channels to satisfy conditions for loss of pressure in the apparatus. This leads to a significant decrease in velocity of heat carrier on the other side of the heat exchanger with all the ensuing consequences.

2. Literature review and problem statement

Design and simulation of PHE with corrugated plates is described in detail in monographs [2, 3]. The algorithms and programs for a variety of plates of various corrugation sizes and types were developed and implemented [4]. This makes it possible to take into account different temperature profiles in all channels, high thermal efficiency, distribution of overall heat transfer coefficient and pressure drop [5, 6]. Generalized mathematical models were presented without taking into account particularities of specific applications in various industries.

Experimental and numerical study of heat transfer and pressure losses was presented in paper [7], for plates with corrugation angle of 30°, experiments were performed for different values of channel height at a fixed corrugation height on one of the heat carriers. Water was used as a heat carrier. Numerical analysis was conducted with the use of CFD (ANSIS) for conditions of performing the experiment. Based on the obtained results, the authors concluded that intensity of heat transfer and pressure drop decrease when the width of channels increases, and increase when corrugation angle increases. In paper [8], based on the experimental data, correlation ratios for determining local and mean value of Nusselt number for plates with corrugation angle of 15–45° were obtained, which enables the authors to refine the form of correlation ratios depending on corrugation slope angle and, as a consequence, to increase accuracy of calculations.

Article [9] presented experiments on determining of heat transfer characteristics for fully extended flow of heated (cold) and heating (hot) water flowing in alternating corrugated channels with corrugation angle of 30°. A systematic comparative analysis of heat exchangers with corrugated plates of the sinusoidal and cylindrical shapes, forming channels of different cross-section, was given. The impact of corrugation slope angle and pitch on the local and integrated heat and mass transfer was explored. It was shown that a change in nature of the flow, caused by different geometry of channels, has a significant impact on uniformity and amount of local heat and mass transfer, as well as magnitude of pressure losses. Three different types of flow, which differ from each other depending on geometry and flow parameters, were identified. In addition to detailed analysis of the flow, comparison of local heat and mass transfer and pressure losses for these geometries was made. Measurements were performed at the same velocity of flow, height and width of the channel of cross-corrugated structure.

The plate structure with channels of various flow sections is implemented in air chillers-heaters [10, 11]. Here, welded cross- and parallel-flow heat exchangers with corrugated channels of triangular form are used. Three-dimensional finite element simulation, using k-ε model for fully extended turbulent flow, was presented. The computed flow fields provided inaccessible information about the structures of the flow and mechanisms of heat transfer improvement. The distribution of local heat transfer coefficient was received.

Given the above, there arises the problem of creation of PHE designs with different corrugation height in channels (of different cross-section area of channels), which makes it possible to overcome disadvantages of using traditional plate apparatuses in the applications described above. The main thing is to answer the question of whether it is appropriate to do so in terms of costs of development and manufacturing of these devices, creation of mathematical calculation models and determining of the most advantageous positions for their installation.

3. The aim and objectives of the study

The aim of present study is to study the feasibility and desirability of creation and application of plate heat exchangers with different corrugation height of channels (various cross-sections of channels). This design will allow decreasing the heat transfer surface area and provide a more effective design of heat exchangers in municipal heat and hot water supply points.

To accomplish the set goal, the following tasks must be solved:

– to develop a mathematical model for calculation and design of a plate heat exchanger with different corrugation height of channels of heat carriers;

– to provide prediction and selection of corrugation height prior to design according to data of the problem;

– in order to conduct multivariate calculations of plate heat exchangers with channels of different cross-section, to develop an algorithm for calculation and to execute its practical implementation;

– to carry out practical designing of plate heat exchangers in the communal heat supply system for comparison of the proposed approach with the traditional calculation.

4. Mathematical model and algorithm for the calculation of plate heat exchanger with different corrugation height of channels

Let us consider a plate heat exchanger, in which channels on the hot and cold sides have different cross-section area (different corrugation height), but the same width and length of the plate. Loss of pressure on the sides of heat carriers Δp_h, Pa on the hot side and Δp_c, Pa the cold side can be written down as

$$\Delta p_h = B_h \cdot \mathrm{Re}_h^{-m_h} \cdot \frac{l_{prh}}{d_{ekvh}} \cdot \frac{\rho_h w_h^2}{2},$$

$$\Delta p_c = B_c \cdot \mathrm{Re}_c^{-m_c} \cdot \frac{l_{prc}}{d_{ekvc}} \cdot \frac{\rho_c w_c^2}{2}. \qquad (1)$$

In ratio (1), pressure losses in the ports and attachments are not taken into account, the heat exchanger is considered to be single-pass. In these equalities, indices h and c refer to the hot and cold side, respectively; ρ_h, ρ_c are the density, kg/m^3; w_h, w_c are the velocities of heat carriers in the channels, m/s; l_{pr} is the reduced length, m; d_{ekv} is the equivalent diameter, m; Re in the Reynolds number; B and m are the constants. .

These ratios can be rewritten in the form of

$$\Delta p_h = B_h \cdot \left(\frac{w_h \cdot 2\delta_h}{\nu_h}\right)^{-m_h} \cdot \frac{b \cdot \Phi_h}{2\delta_h} \cdot \frac{\rho_h w_h^2}{2},$$

$$\Delta p_c = B_c \cdot \left(\frac{w_c \cdot 2\delta_c}{\nu_c}\right)^{-m_c} \cdot \frac{b \cdot \Phi_c}{2\delta_c} \cdot \frac{\rho_c w_c^2}{2}, \qquad (2)$$

where δ_h, δ_c are the corrugation height in the channels, m; ν_h, ν_c is the kinematic viscosity, m^2/s; b is the width of the plate, m; Φ is the coefficient of area increase due to corrugation. Velocity of the heat carriers in the channels of the plate heat exchanger can be represented as

$$w_h = \frac{v_h}{n_{chh} \cdot b \cdot \delta_h} \quad \text{and} \quad w_c = \frac{v_c}{n_{chc} \cdot b \cdot \delta_c}, \qquad (3)$$

where v_h, v_c are the volumetric consumption of the hot and cold heat carrier, respectively, m^3/s; n_{chh}, n_{chc} are the number of channels on the hot and cold side. Without violation of generality of reasoning we will consider that the number of channels is equal, that is, $n_{chh}=n_{chc}=n_{ch}$.

We will write down the ration of pressure losses in PHE channels, using (2)

$$\frac{\Delta p_h}{\Delta p_c} = \frac{B_h}{B_c} \cdot \frac{\Phi_h}{\Phi_c} \cdot \left(\frac{\delta_c}{\delta_h}\right)^3 \cdot \frac{\rho_h}{\rho_c} \cdot \left(\frac{v_h}{v_c}\right)^2 \times$$
$$\times \left(\frac{v_h}{\nu_h}\right)^{-m_h} \cdot \left(\frac{v_c}{\nu_c}\right)^{m_c} \cdot \left(\frac{2}{n_{ch} \cdot b}\right)^{-m_h+m_c}.$$

From this ratio on condition of full satisfaction of pressure losses for the hot and cold sides, that is, $\Delta p_h=[\Delta p_h]$ and $\Delta p_c=[\Delta p_c]$, having assigned the value of corrugation height in the channel of one of the heat carriers, it is possible to determine the value of corrugation height in the channels for the other heat carrier.

$$\left(\frac{\delta_c}{\delta_h}\right)^3 = \frac{[\Delta p_h]}{[\Delta p_c]} \cdot \frac{B_c}{B_h} \cdot \frac{\Phi_c}{\Phi_h} \cdot \frac{\rho_c}{\rho_h} \cdot \left(\frac{v_c}{v_h}\right)^2 \times$$
$$\times \left(\frac{v_h}{\nu_h}\right)^{m_h} \cdot \left(\frac{v_c}{\nu_c}\right)^{-m_c} \cdot \left(\frac{2}{n_{ch} \cdot b}\right)^{m_h-m_c}. \qquad (4).$$

It is possible to simplify expression (4) and obtain an approximated but simpler analogue.

Value Φ, for example, for sinusoidal corrugation can be approximately calculated from formula

$$\Phi = \frac{1}{6}\left(1 + \sqrt{1+\eta^2} + 4\sqrt{1+\eta^2/2}\right),$$

where $\eta = 2 \cdot \pi \cdot \delta / \Lambda$ or $\eta = 2 \cdot \pi / 4 \cdot tg\gamma$, Λ is the corrugation pitch, the distance between two adjacent points of a change in curvature, which can be approximately considered equal to the distance between the peaks of the sinusoidal wave, m; δ is the corrugation height, m, Fig. 1. For characteristics of corrugation, we used magnitude $tg\gamma = 2\delta/(\Lambda/2)$, which reflects the degree of corrugation "curvature".

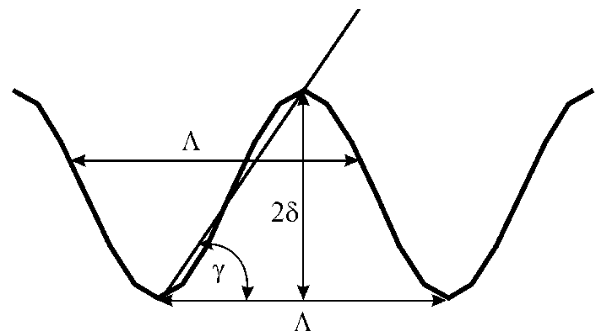

Fig. 1. Basic parameters of sinusoidal corrugation

Value of magnitude Φ for industrially produced plate heat exchangers typically ranges from 1.6–1.7. That is why the ratio of magnitudes of Φ_h/Φ_c, or vice versa, can be accepted equal to unity. For media with close density values, it is possible to assume that $\rho_h \approx \rho_c$. Value of coefficient m for industrially produced plates at developed turbulent motion of heat carriers varies considerably. For plates with high resistance (angle of corrugation slope to the axis of the plate is 60°), it is close to zero value, for plates with low resistance (angle of corrugation slope to the axis of the plate is 30°), it is about 0.15. Given this, expression (4) can be rewritten in the form

$$\left(\frac{\delta_c}{\delta_h}\right)^3 = \frac{[\Delta p_h]}{[\Delta p_c]} \cdot \frac{B_c}{B_h} \cdot \left(\frac{v_c}{v_h}\right)^2.$$

It is possible to determine the value of corrugation height, corresponding to minimum heat exchange area, from the condition of equality of pressure losses of heat carriers in the hot and cold channels. This equality can be represented as follows

$$\left(\frac{\delta_c}{\delta_h}\right)^3 = \frac{B_c}{B_h} \cdot \left(\frac{v_c}{v_h}\right)^2 \cdot R, \qquad (5)$$

where

$$R = \frac{\Phi_c}{\Phi_h} \cdot \frac{\rho_c}{\rho_h} \cdot \left(\frac{v_h}{\nu_h}\right)^{m_h} \cdot \left(\frac{v_c}{\nu_c}\right)^{-m_c} \cdot \left(\frac{2}{n_{ch} \cdot b}\right)^{m_h-m_c}.$$

Thus, having corrugation height on the side of the heat-carrier with high consumption, it is possible to determine corrugation height on the side with lower consumption of the heat carrier from expression (4). After recalculation of corrugation height, the value of velocities in the channels of the heat exchanger is determined using expression (3).

On the other hand, from determining of magnitude of heat carrier velocity in the channel of the plate heat exchanger (3) and the equation of thermal balance, for example on the hot side, it is possible to obtain the correlation of the form

$$\frac{w_h}{w_c}=\frac{t_{22}-t_{21}}{t_{11}-t_{12}}\cdot\frac{cp_c}{cp_h}\cdot\frac{\rho_c}{\rho_h}\cdot\frac{\delta_c}{\delta_h},$$

where t_{11}, t_{12} are the input and output temperatures of the heating heat carrier; t_{21}, t_{22} are the input and output temperatures of the heated heat carrier; cp_h, cp_c are the specific thermal capacity of the hot and cold heat carrier. From this expression, using the ratio of corrugation heights, it is also possible to find values of velocities from (4), having previously calculated one of them using one of the ratios (3).

Having determined values of velocities in channels, it is necessary to verify if conditions on heat load of heat exchanger Q, *W*, are met, that is, if it serves its primary purpose. To do this, we will use the basic equation of heat transfer of the exchanger in the form of $Q=K\cdot F\cdot\Delta t_{\ln}$, K is the heat transfer coefficient, W/(m^2·K); F is the heat transfer surface area of the heat exchanger, m^2; $\Delta t_{\ln}$ is the mean logarithmic temperature difference, K [2]. The designed heat exchanger must satisfy the condition

$$\frac{n_{ch}\cdot\Delta t_{\ln}\cdot(f_{plh}+f_{plc})}{\dfrac{1}{\alpha_h}+\dfrac{1}{\alpha_c}+\dfrac{\delta_{wall}}{\lambda_{wall}}}-Q\geq 0, \qquad (6)$$

where f_{plh}, f_{plc} are the area of the plate on the hot and cold channels, respectively, m^2; the area of the plate $f_{pl}=b\cdot l_0\cdot\Phi$, where l_0 is the linear length of the corrugated surface of the plate; δ_{wall} is the width of the corrugated plate, m; λ_{wall} is the coefficient of thermal conductivity of material, from which the plate is made, W/(m·K); formation of deposits and existence of contaminants on the heat transfer surface is not taken into account so far. The value of heat transfer factors on hot α_h and cold α_c sides can be calculated using criterial ratio for the Nusselt number

$$Nu=A\cdot\mathrm{Re}^n\cdot\mathrm{Pr}^{n_1}\cdot(\mu/\mu_w)^{n_2},$$

where μ is the dynamic viscosity of the heat carrier, Pa·s; μ_w is the dynamic viscosity of the heat carrier, calculated at mean temperature of the wall of the heat exchanger; constant n_1 according to different authors takes the values ranging from 0.3–0.43; constant n$_2$ is most often accepted equal to 0.14 [2, 12]. Values of constants A and n are individual and depend on characteristics of the plate, corrugation type and parameters. Subsequently, we will accept n_1=0.43 and n_2=0.14. Then for heat transfer factors for the hot and cold sides considering (3), it can be written down as

$$\alpha_h=\frac{\lambda_h}{2\delta_h}\cdot A\cdot\left(\frac{2\cdot v_h}{n_{ch}\cdot b\cdot\nu_h\cdot f_{plh}}\right)^n\cdot\mathrm{Pr}_h^{0{,}43}\cdot\left(\mu_h/\mu_{wall\,h}\right)^{0{,}14}$$

and

$$\alpha_c=\frac{\lambda_c}{2\delta_c}\cdot A\cdot\left(\frac{2\cdot v_c}{n_{ch}\cdot b\cdot\nu_c\cdot f_{plc}}\right)^n\cdot\mathrm{Pr}_c^{0{,}43}\cdot\left(\mu_c/\mu_{wall\,c}\right)^{0{,}14}.$$

Substituting n_{ch} in inequality (6), which is non-linear relatively to unknown magnitude, we verify if this inequality is satisfied. In case if inequality (6) for found n_{ch} is not satisfied, it is necessary to increase this value until inequality (6) is satisfied. Resulting value n_{ch} will be the minimal number of channels in the device, satisfying conditions on heat transfer and hydraulic losses in the apparatus. Considering new value n_{ch}, velocities in channels (3) and pressure losses (2) are recalculated.

5. Practical design of plate heat exchangers with different corrugation height for municipal heat and hot water supply facilities

5. 1. Validation of mathematical model at calculation of heat exchanger for heating hot water

As an example of the proposed approach to using plates with different corrugation height, we will consider calculation of the plate heat exchanger for the position of hot water heating. The following design parameters were accepted. Thermal load of the heat exchanger Q=620 kW. Input temperature of the hot heat carrier (network water) t_{inh}=70 °C; output temperature of the hot heat carrier (water) t_{outh}=30 °C. Inlet temperature of the cold heat carrier (tap water) t_{inc}=5 °C; output temperature of the cold heat carrier (tap water) t_{outc}=60 °C. Permissible pressure losses on the hot and cold sides are [Δp_h]=[Δp_c]=25 KkPa.

These are traditional parameters for calculation of the heat exchanger for water heating of hot water supply (HWS) by the parallel scheme of attachment of the heat exchanger at the heat point [12] t_{outc}=55 °C is sometimes accepted as estimated output temperature of tap water. However, this has no special effect on calculation of the heat exchanger. Volumetric consumption of heat carriers on the hot side v_h=13.7 m^3/h and v_c=9.7 m^3/h and, consequently, their ratio is equal to v_h/v_c=1.4. Logarithmic temperature difference is equal to $\Delta t_{\ln}$=16.4. Thermophysical characteristics of the heat carriers are calculated at mean calorimetric temperature of flows, respectively, on the hot side of 54.5 °C and 38.7 °C and mean temperature of the wall was calculated as their arithmetical mean value. Thickness of plates δ_{wall}=0.5 mm, thermal conductivity factor λ_{wall}=16 W/(m·K) (stainless steel AISI 316).

As geometrical parameters of the channels of the heat exchanger, we will accept parameters of plate M6M, industrially manufactured by company "Alfa Laval" (Sweden) on the hot side, and M6 on the cold side, Table 1. For both plates, it was accepted that width of the plates is b=213 mm and length is l= 412 mm.

Table 1

Geometrical parameters of corrugated plates

Brand	δ, mm	Λ, mm	Φ	$f_{ch}\cdot10^3$ m^2	f_{pl}, m^2	l_{pr}, m
M6M	3	10.15	1.618	0.630	0.14	0.666
M6	2	6.35	1.683	0.432	0.15	0.694

Values of coefficients in ratios (1) and (8) were accepted as follows: for M6M – A=0.27; B=4.55·10^{-3}; for M6 – A=0.25; B=2.40·10^{-3}, coefficients m=0 and n=0.7 [13]. Such parameters of the plates correspond to plate corrugation angle of 60° or according to terminology of "Alpha Laval", to HN-plate. These coefficients are true for the values of the Reynolds number, exceeding 400.

Calculation of corrugation height on the hot side δ_h at the assigned value of corrugation height on the cold side δ_c=2 mm from formulas (4) and (5) gives respectively 3.026 mm and 3.074 mm, which actually corresponds to corrugation of M6M plate equal to 3 mm.

For this position, single-pass heat exchangers M6 and M6M with different corrugation height were calculated. Calculation parameters are shown in Table 2.

Table 2

Calculation parameters of heat exchangers

Brand	Number of plates	Heat exchange area, m^2	Heat transfer factor, W/(m^2·K)	Pressure losses, KPa	
				Hot side	Cold side
M6	53	7.65	4956	23.50	12.48
M6M	91	12.46	3031	4.33	2.40
M6M/M6	51	7.25	5210	14.17	13.05

An analysis of Table 2 indicates that the use of channels with different corrugation height M6M/M6 leads to a slight decrease in heat transfer surface in comparison with heat exchanger M6 and a significant difference compared to M6M. In addition, there is an increase in heat transfer coefficient and alignment of pressure losses on the sides of the heat carrier.

5. 2. Effectiveness of using an approach with different corrugation for calculation of heat exchangers of two-stage hybrid scheme for hot water heating

As another example of the proposed approach to using plates with different corrugation height, we will consider calculation of the plate heat exchangers on the position of hot water heating hot water by the two-stage hybrid scheme [12].

This scheme of attachment or heat exchangers is characterized by the fact that during the heating season, reverse water from the heating system of a consumer (buildings, houses or groups of houses) is used for preheating tap water in the heat exchanger of stage 1. This water is mixed with network water, coming out from the heat exchanger of stage 2 and is fed to the heat exchanger of stage 1 as the heating heat carrier. A specific feature here is that a large flow of the heating low-temperature heat carrier is supplied to the heat exchanger of stage 1 (approximately by 2.0–3.0 times more than consumption of heated tap water). This creates a number of problems with hydraulic calculation of the heat exchanger and, in particular, requires verification of its serviceability under other operation modes.

The following calculation parameters were accepted. Heat load of the heat exchanger Q=367 kW. Input temperature of the hot heat carrier (mixed output network water from the heat exchanger of stage 2 and reverse network water for the heat exchanger of heating system) t_{inh}=46 °C; output temperature of the hot heat carrier (network water) t_{outh}=30 °C. Input temperature of cold heat carrier (tap water) t_{inc}=5 °C; output temperature of the cold heat carrier (tap water) t_{outc}=40 °C. Permissible pressure losses on the hot and cold sides are $[\Delta p_h]=[\Delta p_c]$=50 kPa.

These are traditional parameters for calculation of the heat exchanger for heating water for hot water supply (HWS) by two-stage hybrid scheme of attachment of the heat exchanger at the heating point [12].

Volumetric consumption of heat carriers on the hot side v_h=20 m^3/h and v_c=9 m^3/h and, respectively, their ratio is equal to v_h/v_c=2.2. Logarithmic difference of temperatures is $\Delta t_{\ln}$=13.4. Thermophysical characteristics of the heat carriers are calculated at mean calorimetric temperature of flows, respectively, on the hot side of 40.7 °C and 20.5 °C, and mean temperature of the wall was calculated as their mean arithmetic value. Thickness of plate δ_{wall}=0.5 mm, thermal conductivity coefficient λ_{wal}=16 W/(m·K) (stainless steel AISI 316). As a plate on the side of the heating heat carrier, we accepted a model plate with corrugation angles of 60° (N-plate) with parameters: δ=4 mm; Λ=9 mm; Φ=2.13; f_{ch}=0.852 m^2; f_{pl}=0.187 m^2; l_{pr}=0.880 m. As constant values in criterial equations, the following magnitudes were accepted [13]: A=0.246; B=4.064·10^{-3}, coefficients m=0 and n=0.7.

At the assigned value of corrugation height on the cold side δ_c=2 mm from formulas (4) and (5) for corrugation height on the hot side δ_h we will obtain 3.97 mm and 4.074 m^3 respectively. This corresponds to corrugation of plate M10M, in which corrugation height is equal to 4 mm.

Table 3

Calculation parameters of heat exchangers

Brand	Number of plates	Heat exchange area, m^2	Heat transfer factor, W/(m^2·K)	Pressure losses, KPa	
				Hot side	Cold side
M6	56	8.10	3393 (Margin 43 %)	44.80	10.14
M6M	43	5.74	4826	42.98	8.94
M10M/M6	29	4.72	6378 (Margin 9.2 %)	47.72	35.90

Analysis of Table 3 shows that the use of channels with different corrugation height M10M/M6 leads to a significant decrease in heat transfer surface in comparison with heat exchangers M6 and M6M. In addition, there is an increase in heat transfer factor and alignment of pressure losses on the sides of the heat carrier.

6. Discussion of results of research into simulation of plate heat exchangers with different corrugation height in channels

Analysis of data in Tables 1, 2 shows that the use of PHE with different corrugation height of the channels leads primarily to a decrease in heat transfer surface area of the apparatus, that is, to an actual decrease in capital costs. Conditions for permissible losses of pressure from both heat carriers are almost completely satisfied, which favorably affects operation dynamics of these apparatuses. In addition, by aligning velocities of heat carriers in the channels of the apparatuses, magnitude of tangential stress on the walls of the plates increases. The latter circumstance makes it possible to slow down the processes of emergence of sediment and contamination on the heat transfer surface during the operation process and extend the term of continuous operation between cleanup stops.

Presented results have direct practical value when using plate heat exchangers at thermal points of heating and hot water supply systems. In addition, the proposed approach can be used for any applications, where there is a substantial difference in consumption of heat carriers, for example, in recuperation heat exchangers. Application of apparatuses with different corrugation height is promising in the petrochemical industry, especially for separation of preheating of raw materials.

The presented research is a continuation of earlier studies to improve designs of plate heat exchangers, analysis of thermophysical processes in channels, simulation of contaminants in the process of operation.

The main obstacle for application of plate heat exchangers with different corrugation height is complexity of structural implementation of the proposed approach. By analogy with wide-channel apparatuses, this can be achieved, for example, through structural inserts. Normal alternation of plates with different corrugation leads to emerging of channels with averaged properties. Any of the structural implementations of the proposed approach presupposes the use of seals of various types in one apparatus, which requires refinement when determining dimensions of a plate stack at its assembly.

6. Conclusions

1. A mathematical model of calculation and design of the plate heat exchanger with different corrugation height in the channels was developed. The use of the model makes it possible to decrease the calculated heat exchange surface, to increase heat transfer coefficient, to align pressure losses in channels. Configuration of heat exchangers due to increased velocity leads to a decrease in intensity of contamination of the heat exchange surface.

2. The ratio, enabling us to calculate in advance the values of corrugation height at the design stage in every specific application, was obtained. This relationship links permissible magnitudes of pressure losses in channels, geometric parameters of the plate and corrugation, thermophysical parameters of heat carriers and the number of channels in the apparatus.

3. To conduct multivariate calculations of plate heat exchangers with channels of different cross-section, the calculation algorithm was developed and implemented in practice.

4. Reliability and advantages of the proposed approach were demonstrated on the example of calculation of the plate heat exchangers of hot water preparation by the parallel and two-stage hybrid schemes. The obtained results showed that the heat exchangers with different corrugation height, selected for calculation positions, have a smaller heat transfer surface area (smaller plate stack) than traditional devices. This makes it possible to decrease capital costs when designing a heating point, and due to an increase in velocity of heat carriers in channels, to reduce intensity of emerging of deposits on the walls of plates.

Acknowledgement

The study was performed with the financial support from the Ministry of Education of Ukraine.

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