Приборы и оборудование

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Novel Trends of Development and Perfection the Modern Heat Exchangers

Some actual aspects of advancement the problem of improvement the heat exchange equipment are considered in the paper. First of all the actual items related to middle and high temperature recuperators are discussed with proper up-to — date approaches.

The classification of flue gases heat recovery appliances has been proposed along with the statement and analysis of the main characteristics of the recovery plants and option the ways of optimization the mentioned characteristics.

The problem of Reynolds analogy (similarity of relative change the heat transfer phenomenon and variation the hydraulic resistance) within the channels of different purpose and of various cross-section supplied with and without the obstacles has been analyzed in application to separate cases of flow along the surfaces equipped with the cavities (dimples) or the convex elements.

Thermal Performance Factor (TPF) of the heat exchange process is qualitatively like to Reynolds analogy factor and is highly depended upon rate of heat transfer and of friction factor in conditions of the scheme under consideration for flow over the surface or flow within the channel. The various media has been compared used as a working body in the heat exchanger's channels: gaseous, liquid and the nanofluids, the last appeared in practice since 2000.

Analysis has been carried out on effect of using the secondary energy emitters (SEE) arranged inside the tube channels, on resulting heat flux by heat exchange between outward flow of combustion products and the inner air flow. *Bibl. 23, Fig. 6.*

Key words: convection, heat exchanger, heat recovery, heat transfer media, nanofluid, Nusselt number, radiation, Reynolds analogy, thermal-hydraulic efficiency.

Introduction

The Heat Exchanger apparatuses are related to the most wide spread facilities in power and heat engineering, in heat supply systems and by air conditioning, by completion the thermal engines to be a part of equipment for chemical, food and other technological processes. Significant portion of heat exchange plants is related to heat recovery units particularly within the middle and high temperature range. This type of equipment is used in the furnaces and the boilers to enhance the power efficiency and to reduce the fuel flow rate. Another purpose and opportunity of the recuperators — ensuring of especially enhanced operation temperatures in case of the high temperature technologies, for example — by the glass melting processes.

Different principles of effect upon heat transfer media and various mechanisms of enhancement the heat transfer have been developed by studying and realization the exchange processes.

Heat transfer enhancement (intensification) and attendant increase of power efficiency of the plant makes the basic problem by studying and development the heat exchangers. This problem is of especial significance for the case when the gaseous media make the working bodies (agents) by both sides of heat exchange surface while the resulting heat fluxes are characterized by low values [1].

As a rule the heat transfer enhancement is accompanied by increase of the energy for transportation the working agents. Two directions of enhancement the heat transfer are under consideration. First direction is related to the approach when heat flux increase is provided without an account of additional energy losses.

The second direction is composed of the statement on heat flux increase while given value of energy is spent for pumping over the heat transfer media [1].

Heat recovery of flue gases from the furnace and boiler plants is usually provided by preliminary preheating of initial combustion components: air-oxidant (mainly) — or/and the low-calorific fuel gas — at the expense of heat content (total enthalpy) of combustion products (CP). Mechanism of heat recovery — using of *sensible heat* of CP — is realized in traditional recuperative (regenerative) heat exchangers. The processes and the plants grounded upon thermochemical recuperation (TChR) have been spread and been taken for application at the last time. Recently the furnaces with TChR have been implemented in industry [2, 3]. Under this approach the *latent heat* of endothermic chemical reactions is absorbed in conditions of natural gas conversion with oxidizing components, first of all with the combustion products, H_2O, CO_2 [2, 3].

a. Background and classification of advanced heat recovery facilities for different energy constituents

1. The heat recovery units for the furnace and boiler plants represent the particular group of heat exchange facilities. The distinctive peculiarity of this plants makes the temperature area of operation the recuperative heat exchangers and significant input of radiative heat transfer in high and middle temperature processes and plants under conditions of increased values of resulting heat fluxes.

Along with the general regularities being typical for any heat exchange plants, for the recuperative heat exchangers the great portion of resulting heat flux is provided due direct and indirect radiation constituents by both sides of heat exchange surface within the course of heat irradiation from the combustion products (*primary — hot — heat transfer medium*) through the intermediate body(ies) to the working substance (*secondary — cold — heat transfer medium*).

Level of working temperatures for heat recovery plants is predetermined by final, exit temperature of working substance preheat. As a rule the combustion air is used as a working substance in heat recovery plants of the furnaces and boilers. Sometimes the low-calorific fuel gas could serve as a working agent.

In the latter case the corresponding gas recuperative devices can complement the ordinary air recuperators or be used as the only apparatuses of the heat recovery system. In frame of the smart energy saving and environmentally benign technologies the traditional preheating of the initial combustion component(s) could be accompanied by *humidification of an oxidant* (air flow) at a separate stage of recovering the flue gases heat [4–7].

In the modern advanced heat utilizers, presented by the most common centralized furnace systems as well as the individual recuperators of heating and heat treatment furnaces of metallurgy and of mechanical engineering, the combustion air preheating temperature reaches 600 °C and more [8].

2. At up-to-date stage of development the systems of heat recovery, the combined processes of air preheating and humidification were popularized and for example been realized by using the contact water vapor pump (WVP) systems and the special means been proposed in accordance with the Maisotsenko Cycle (M-Cycle) [4–7].

The WVP could serve as a first stage of air heating system while the M-Cycle makes the evaporative cooling technology that cools any fluid (air or liquids) ***below the wet bulb temperature*** and approaches the dew point temperature without adding humidity in a single stage heat and mass exchanger.

The M-Cycle is a thermodynamic conception which captures an energy from the air by utilizing the psychrometric renewable energy available from the latent heat of water evaporating into the air [6]. Present state of art provides an application of M-Cycle and heat recovery in various heating, ventilation, and air-conditioning (HVAC) systems; cooling systems and gas turbine power cycles. It has been found that the hybrid, ejector, and desiccant based MAC (M-Cycle Air conditioning) systems enable a huge energy saving potential to achieve the sensible and latent load of AC in humid regions. Similarly, the overall system performance is significantly improved when the M-Cycle is utilized in cooling towers and evaporative condensers. Furthermore, the M-Cycle conception in gas turbine cycles has been realized recently in which the M-Cycle recuperator provides not only hot and humidified air for combustion but also recovers the heat from the turbine exhaust gases. The M-Cycle nature helps to provide the cooled air for turbine inlet air cooling and to control the pollution by reducing NO_x formation during combustion. Comparison of three distinguished Maisotsenko gas turbine power cycles with the conventional cycles shows the M-Cycle's power advantages [5]. Maisotsenko disclosed a configuration wherein a main stream of air is passed along a dry duct, simultaneously passing an auxiliary air stream counter currently along a moist duct

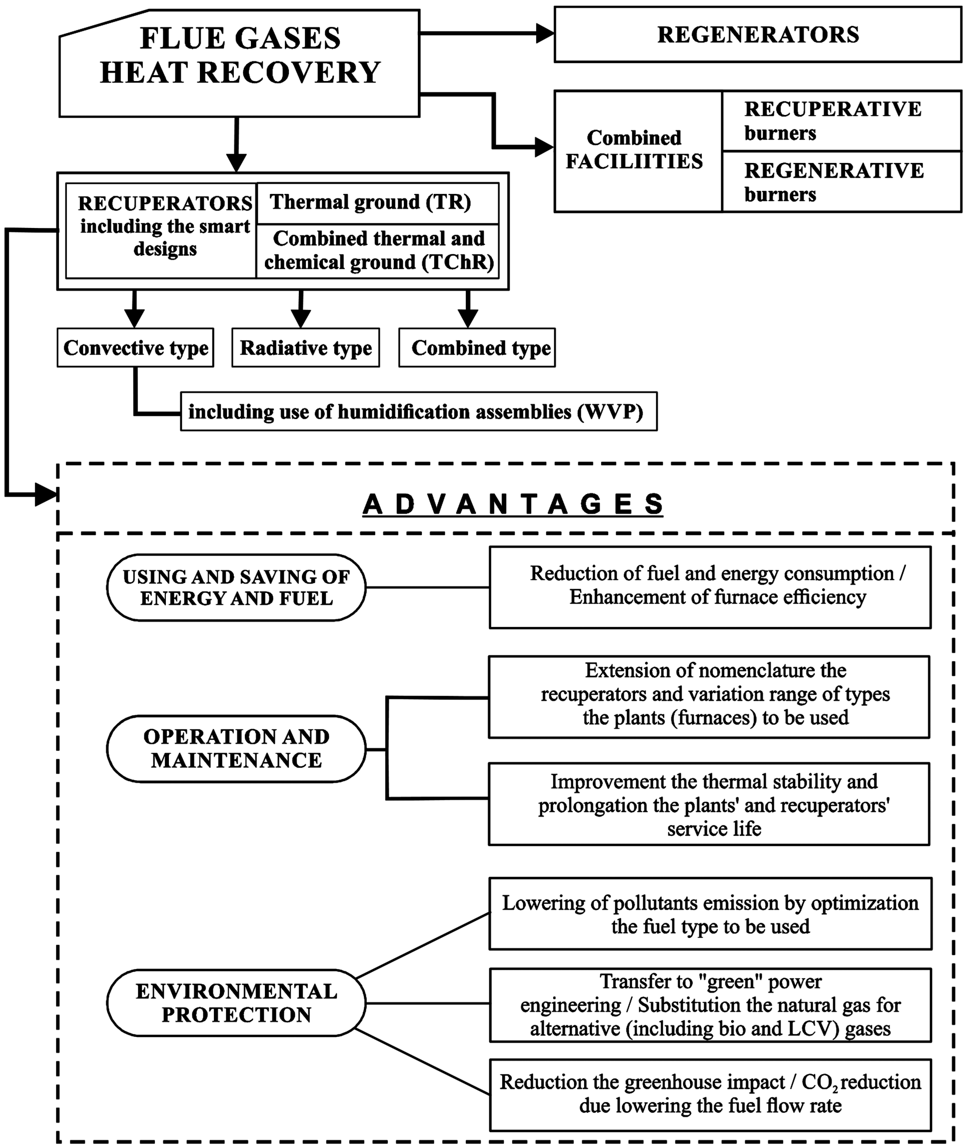

Fig. 1. Diagram on interconnection between the actual and perspective types of heat recovery plants (HRP) in frame of proposed classification the advantages by main features of HRP (Heat Exchangers).

which is in heat-exchange relation with the dry duct. The auxiliary stream is obtained by subdividing the total stream into main and auxiliary streams. It gives opportunity to get the cold air *below the wet bulb and approaching the dew point temperature of outside air.* The M-Cycle's cooling capacity increases (instead of decreases) when the temperature of air being cooled increases. This distinctive feature makes any M-Cycle air coolers ideally suited for an air flow preparation by inlet to turbine combustion chamber.

The M-Cycle can significantly improve the operating efficiency of the Brayton Cycle, the thermodynamic cycle upon which all gas turbines operating. The M-Cycle can be used cost effectively to change the density and mass flow of the air supplying to the compressor.

By adding the M-Cycle inlet cooling system directly to the turbine system, the user can also take advantage of exhaust air stream which is cooler the ambient air and more humid than obtainable from any humidifiers.

The performance of three highly efficient advanced indirect evaporative air coolers: the «classical» cross-flow heat and mass exchangers related to Maisotsenko Cycle and two novel combined M-Cycle air coolers proposed by authors [7]. The novel heat and mass exchangers are based on a combination of parallel-current and counter-flow or cross and counter-flow schemes. The main conclusion is that proposed solutions are characterized by higher cooling efficiency than the cross-flow M-Cycle unit. Combined cross-regenerative counter flow heat and mass exchanger has obtained the highest overall performance [7].

3. In Fig. 1 is presented the diagram generalizing the classification of the existent types of heat recovery facilities: *recuperators, regenerators, combined devices*: recuperative and regenerative burners joining the thermal energy generation by fuel combustion and flue gases heat recovery in the single device.

The eligible systems include the recuperator with the burner arranged in the single body — *recuperative burner* — or pair of regenerators, each of them been fitted with the burner — *regenerative burners*.

Simultaneously in Fig. 1 are shown the advantages of application the modern heat recovery systems: reduction of fuel flow rate and energy consumption for fuel plants (boilers, furnaces); increase of thermal stability and prolonged service life of heat exchangers; reduction of pollutants formation and of greenhouse gases impact.

Mentioned consideration reflects our approach to creation the universal classification for the heat exchangers of different purpose and of various energy background including the analysis of heat recovery facilities [9].

Usually any technique of heat transfer enhancement, particularly — within the heat exchangers (HE) is considered to be related to one of three methods realized by means of proper facilities: active, passive or compound ones [10]. Active method is realized due application the external energy source or physical fields (electrostatic, magnetic).

To the examples of similar kind are related the cases of using the pulsations, fluid vibration, surface vibration, generation of sound waves etc. Passive method is realized due sudden change in profile of the channel or any geometrical alteration in passage conditions like forming the sequences (series) of arrangement the inserts inside the tubes. The mentioned technique belongs to the category of the most popular passive methods of heat transfer enhancement.

b. Similarity of heat transfer and hydraulic resistance phenomena

Similarity of heat transfer and hydraulic resistance as phenomenon to be studied is considered for the case of channels and surfaces with regular relief pattern by air flow when the only friction constituent of hydraulic resistance is included into consideration (without vortices formation).

Principal characteristics of the flow inside the channel with the insert in accordance with the most of publications are followed of heat transfer and hydraulic resistance regularities and include the set of parameters:

1) Rate of heat transfer (total coefficient of heat transfer α_{Σ}, indirect dimensionless index — Nusselt number);

2) Friction Factor f;

3) Thermal Performance Factor (TPF) $\eta = (Nu_T/Nu_0)/(f_T/f_0)^{1/3}$ under conservation the invariable (constant) pumping power. Subscript «T» denotes the channel with the twisted tape insert while «0» denotes the plain tube channel.

Parameter mentioned above and called as TPF is similar to that serving as Reynolds analogy factor *k* [11]. The latter value makes ratio of two dimensionless characteristics each of them presented the relative value (own ratio) of corresponding parameters. In numerator is considered the relative heat transfer characteristic, in denominator — the relative hydraulic resistance indices. Both indicators: of heat transfer and of hy-

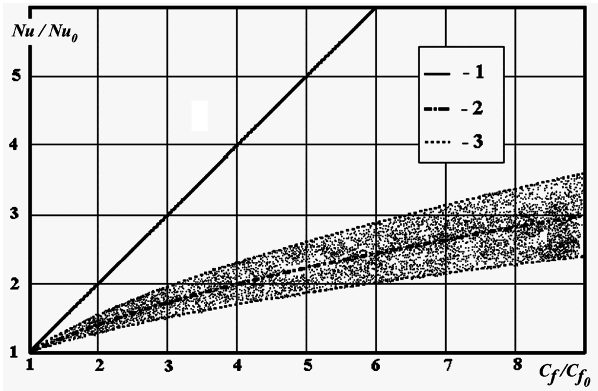

Fig. 2. Dependence of heat transfer index Nu/Nu_0 compared with smooth surface on ratio C_f / C_{f0} of the friction coefficients (intensification ways) [11]: 1 — Reynolds analogy $Nu/Nu_0 = C_f/C_{f0}$; 2 — Nunner curve $Nu/Nu_0 = (C_f/C_{f0})^{0.5}$; 3 — area of known experimental results (except for vortex intensification methods).

draulic resistance — are comparing the specific values for the channel under consideration, particularly connected with arrangement the inserts inside the tubes, and for the basic channels (plain tubes).

The heat transfer efficiency by using any special appliances, surface profiles and roughness structures or by change the operation mode could be presented by ratio of Nusselt numbers while the index of hydraulic resistance for the cases of friction constituent should be given by ratio of

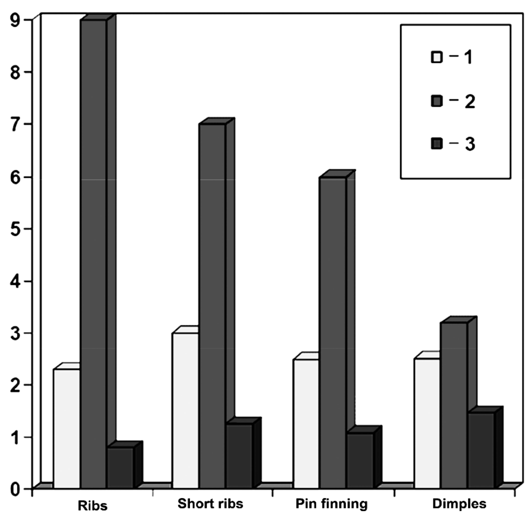

Fig. 3. Thermal-hydraulic efficiency of surfaces with various heat transfer intensifiers [11]: 1 — heat transfer enhancement as compared with smooth surface; 2 — hydraulic resistance growth as compared with smooth surface; 3 — thermal-hydraulic efficiency parameter k.

friction coefficients. As a result the similarity equation known as Reynolds analogy

$$k_1 = (Nu/Nu_0)/(C_f/C_{f0}) = 1, \qquad (1)$$

has been found to be valid mainly for an air flow [11] when Prandtl number Pr >1.0.

Generally the analized correlation could be presented in the next form (Fig. 2):

$$k = (Nu/Nu_0)/(C_f/C_{f0})^n. \qquad (2)$$

In equations (1), (2) the parameters Nu, C_f — Nusselt number and friction coefficient for the case of the channel or surface with the heat transfer intensifiers; Nu_0, C_{f0} — Nusselt number and friction coefficient for the basic case of the heat exchange (HE) channel or surface without the intensifiers (smooth channel or surface).

Origin of vortex structures on the heat exchange surface for example in case of supplying the surfaces with the dimples (Fig. 3) leads to disturbance the boundary layer and to heat transfer enhancement.

In Fig. 2 are shown the correlations between ratios Nu/Nu_0 and C_f/C_{f0} accordingly Reynolds analogy (equation (1) and (2) when n =1) and by Nunner (equation (2) when n = 0.5). Usually by using of value n = 0.5 in equation (2) somebody would be succeeded to meet the actual heat transfer characteristics to predicted data in case of arrangement the channels or surfaces with HE intensifiers [11].

In Fig. 3 are shown the thermal and hydraulic characteristics of the surfaces supplied with various types of the HE surface intensifiers [11]. The numerical values of characteristics are significantly differed; as the most preferable of the turbolizers are considered those providing the maximum thermal-hydraulic efficiency and minimum hydraulic resistance. To set a goad example of the surface disturbers compared in Fig. 3 the dimples could be recognized as the most effective surface structures.

c. Rate of heat transfer and technique for its enhancement

1. The heat transfer enhancement from potential core to and within the boundary layer is mainly stipulated by vortices generation [11]. The following three types of mechanisms of vortices (eddies) formation at the obstacle’s surface are arisen:

— ***the first type*** — Teylor/Gertler vortices (eddies) generation by reason of boundary layer instability;

— ***the second type*** — is connected with vortices formation in swirling flow of liquid (gas) independently on viscosity of medium (including the ideal gas). This case is related to most common way of heat transfer rate enhancement in HE using the twisted tapes (TT);

— ***the third type*** mechanism could be detected by working body flow through the profile or along the surface relief with consequence of the non-uniformities as a result of alternation of cavities and projections (lugs).

Particularly this case is referred to the process of flowing over the surfaces with relief formed by hemispherical cavities (the dimples) [11].

2. Like the HE intensifiers been demonstrated in Fig. 3 for the case of flowing over the open surfaces, it could be suitable to consider the turbulizing inserts firstly in form of various twisted tapes (TT) as the principal elements for the channels.

The manufacturers inform that turbolators in form of TT are custom made and available in a variety of configurations to fit most tube sites and many types of heat exchange equipment providing an increase of heat transfer efficiency and extending the service life of equipment by means of eliminating the hot and cool spots that cause the thermal stress [12].

The mentioned manufactures [12] with 75 years experience of designing and fabrication have provided since 1973 the custom turbulators for wide range of heat exchange equipment including shell and tube heat exchangers and fire tube boilers.

Dozen types of TT used at the channels of different cross-sections are known and have been experimentally investigated [13, 14].

An investigation on heat transfer and pressure drop characteristics in turbulent flow regime through the circular tube fitted with typical twisted tapes has been carried out for the case of varying the length and alternation of clockwise and counter-clockwise twisted tapes (ACCT tapes). The results indicated that ACCT tapes have a good heat transfer performance as compared with typical twisted tapes of the same length. The heat transfer enhancement is growing with increasing the length of twisted tapes [13].

The effect of clearance ratio and twist ratio tape inserts on heat transfer coeficient and on pressure drop characteristics of an air flowing within the tube under uniform heat flux have experimentally examined for the Reynolds number range nearly from $5 \cdot 10^3$ to $25 \cdot 10^3$. Twisted tapes which were used by study have five values of twisted ratio (TR = 2.0; 2.5; 3.0; 3.5, and 4.0) and three different clearance ratio (CR = 0; 0.0178; 0.0357). Experimental results have showed that the Nusselt number was decreased with increase of the clearance ratio and with twist ratio. On the other hand, friction factor decreases with increase of the clearance ratio but it increases with decrease of the twist ratio as expected. Also the Nusselt number increases and friction factor decreases with increase of Reynolds number.

Helical and screw type twisted tape inserts, the TT inserts with circular rings demonstrate an advanced version of twisted tape inserts. Great investigations were performed to examine the thermo-hydraulic performances of these tapes. A square duct fitted with helical screw tape with variable twist ratios TR was experimentally evaluated. The values of Nu to have a direct relations with Re and inverse relation with TR. The values of Nu and f were stated to be 3.64–18 times and 2.31–6.56 times higher — respectively — in comparison with a plain square channel [15]. Last time R&D have provided an opportunity to widen the different techniques for heat transfer enhancement without great rise of hydraulic resistance the channel due combination of various alterations within the channel profile (for example — combined circular ring — twisted tape set-up as well as TT inserts equipped with wire nails inducing the vortex flow).

3. Another way to enhance heat transfer makes combination of hydrodynamic disturbances due arrangement the different protrusion's within the channel or using the twisted tapes of any type with nanofluids as a working media [16–18]. Effects of nanofluid flow and protrusion transverse rib roughness, on the thermal and hydrodynamic performance of square channel were analyzed. The nanofluid was prepared as solid nanoparticles of Al_2O_3 suspended indistilled water. Thermal hydrodynamic performance was determined for different sets of protrusion transverse rib roughness and flow parameters. The highest value of thermal hydrodynamic has been observed at nanoparticle concentration of 4.0 % [19].

A great experience has been accumulated since year 2000 regarding an application of combination the obstacles within the channels (ribs of different form) with the nanofluids as heat exchange medium. Mentioned combination enhances both heat transfer rate and hydraulic resistance of the modernized channels.

d. Heat transfer media

The heat transfer medium — gaseous or liquid one — exerts a great influence upon the TPF characteristic. In case of the gas been used as the working medium, the losses of mechanical energy to overcome the friction forces are great enough to be compared with the heat energy flux transferred within the heat exchangers HE. It must be taken into account that the cost of mechanical energy makes much higher value than that for the equivalent heat energy flux (usually 3–5 times) for the most of power plants.

The power consumption for transportation the HE medium by the length of HE channels and to overcome the friction and resistance forces (it means the energy capacity for pumping the HE medium) is much lower by value in comparison with the transferred heat flux. That's why the liquid heat transfer medium from mentioned standpoint has a power preferences comparing the gaseous medium [11].

Last time period in accordance with given comparison of the gaseous and liquid media is characterized by great interest to the researches been carried out with application of turbo-lator-twisted tapes arranged within the HE channels while using the nanofluids serving as heat transfer media by this approach.

Beginning of year 2000 the trend of using the nanofluids with nanoparticles composed of the metals or oxides likeAl_2O_3, SiO_2,TiO_2,CuO, Fe_3O_4 suspended in the distilled water to serve as heat transfer medium — has received a great popularity due opportunity to enhance the Nu number up to 40 % and more [16–19].

Monotonous increase of Nu with value of concentration φ of Al_2O_3 nanoparticles of 30–45 nm in range of φ = 1–4 % is connected with growth of Prandtl number and heat conductivity of nanofluid.

It has been shown that the thermal and hydraulic (hydrodynamic) performance η of heat exchanger channels is mostly affected by application the special nanofluids. Application of spherical nanoparticles in liquid heat transfer medium provides the best results in comparison with a pure H_2O in case of particles concentration $\varphi \approx 4$ %. But the growth of final impact depends on some flow characteristics including the operative, structural ones, on geometry (relative sizes) of the system to be studied, on material of the particles (nanoparticles) and on thermophysical properties of the moving (transporting) flow. The temperature of the flow was studied in rane of 20–60 °C being considered as the parameter influencing on transferred heat.

By consideration the nanofluids as heat transfer medium the most noteworthy factors of nanofluids make the type and substance of nanoparticles, their form, sizes and concentration φ. For example it was experimentally shown an increase of Nu number by 44.64 % and 41.82 % for SiO_2 and Al_2O_3 nanofluids, respectively [20]. Rise of Nu value in case of increase the particles concentration in nanofluid has been generally stated. Meanwhile was found that Nu enhancement makes 12 % (in case of φ = 1 wt. %) and 32 % (when φ = 0.5 wt. %) respectively by alteration ц value for tested nanofluids [21].

When gas or liquid medium passes the dimple type cavities the large-scale dynamic vortex structures are arisen which are observed both in laminar and turbulent flow modes. Similar structures are observed as well at low subsonic velocities (at Mach number M < 0.3 when the compressibility of the gas media is negligible), at high subsonic and at supersonic velocities (when the compressibility of the medium cannot be neglected) [11].

e. Analysis of impact the inner secondary emitters on heat transfer process and intensity

Due modern approaches and opportunities of Heat&Mass Transfer Theory [22] and calculations background [23] it has been succeeded to solve the applied tasks on advancement the effective designs of middle and high temperature heat exchangers.

As has been mentioned earlier, the installation of secondary emitters SEE inside the tubes allows the using of inserts serving as additional heating surfaces for enhancement the resulting heat flux transferred by convection both from the inner tube surface and from the SEE surfaces to the radiatively transparent medium — air coolant inside the tube channel (Fig. 4):

$$\begin{aligned} Q_a &= Q_{conv,t} + Q_{conv,in} = \\ &= \int_{F_t} q_{conv,t}\, dF_t + \int_{F_{in}} q_{conv,in}\, dF_{in} = \\ &= \int_{F_t} \alpha_{conv,t} (T_t - T_a)\, dF_t + \\ &+ \int_{F_{in}} \alpha_{conv,in} (T_{in} - T_a)\, dF_{in} \end{aligned} \tag{3}$$

where Q_a — the total resulting heat flux transmitted in the air channel and absorbed by the air flow; $Q_{conv,t}$, $Q_{conv,in}$ — the resulting heat fluxes transferred by convection to the air flow from the inner surface of the channel (tube) and from the surface of the inner (in-tube) inserts, respectively; $q_{conv,t}$, $q_{conv,in}$ — the local heat fluxes, trans-

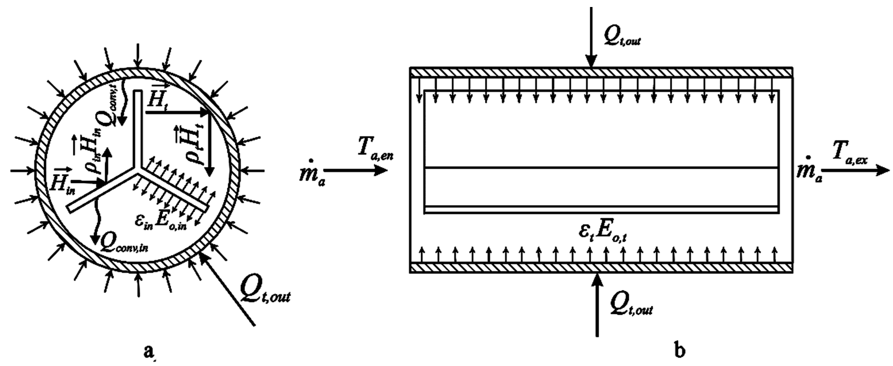

Fig. 4. Calculation layout of incident and resulting fluxes within the tube section of recuperative heat exchanger. The vectors of total heat fluxes are shown conditionally both by direction and by module.

ferred by convection to the air flow from the inner tube surface of the channel and from the surface of the inner inserts, respectively; $\alpha_{conv,t}$, $\alpha_{conv,in}$ — the local coefficients of convective heat transfer to the air flow from the inner tube surface and from the surface of the inserts, respectively; T_a, T_t, T_{in} — local temperatures of the air flow, of the inner tube wall (temperature is assumed to be of fixed constant value by the wall thickness) for the tube channel under consideration; F_t, F_{in} — heat exchange surfaces of the tube channel and the insert — respectively.

From another standpoint the heat transfer process could be considered regarding conditional heat exchange surface (tube inner surface) dividing the space between primary and secondary heat transfer media — combustion products and an air flow — respectively. The calculations have been carried out for three — loops plant ($i \equiv 1, 2, 3$) of the tube recuperative section made of tube of inner diameter D = const.

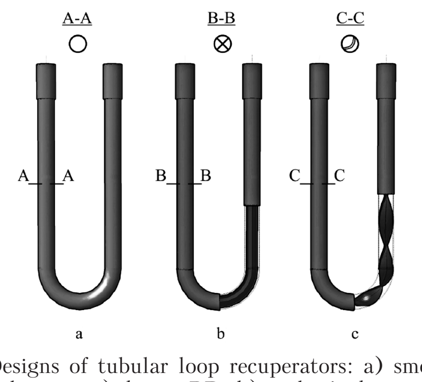

Fig. 5. Designs of tubular loop recuperators: a) smooth-tube (without the inserts) design BD; b) author's design MD with flexible cruciform inserts – secondary emitters mounted along the whole length of heat-exchange tubes; c) designs SP1 and SP2 with spiral inserts. The cross sections of the channels are shown above the loops.

Each of the loops is of U-like form and has the similar design and nearly equal sizes. The procedures connected with thermal process within the tube walls and the impact of wall thickness are assumed as a negligible factors.

By this approach the conditional average value of heat transfer coefficient from the tube walls to an air flow within each of the i-th loops generally could be presented by equation:

$$\bar{\alpha}_i = \frac{c_{pm} \cdot \dot{m}_a \left(t_{a,i,ex} - t_{a,i,en}\right)}{\int\limits_{F_{l,i}} t_w \, dF_{l,i} - \pi D \int\limits_{L_i} t_a \, dL_i}, \qquad (4)$$

where the numerator gives the heat flux $Q_{a,i}$ absorbed by air flow within the i-th loop ($i \equiv 1, 2, 3$) of heat exchange surface $F_{l,i}$.

In this case the average heat transfer coefficient within the recuperative section RS (subscript «rs») makes the conditional sum of heat transfer by convection and radiation from the inner surface of RS:

$$\bar{\alpha}_{rs} = \bar{\alpha}_{rs,conv} + \bar{\alpha}_{rs,rad} = \sum_{i=1}^{3} \bar{\alpha}_i F_{l,i} F_{rs}^{-1}. \qquad (5)$$

It means that for the case of transparent medium like an air flow within the tube channels equipped with the inner SEE, the radiative flux (presented by component $\bar{\alpha}_{rs,rad}$) is transferred from the inner tube surface to the insert's surface, afterwards is adiabatically absorbed and removed from the channel wall by flow convection.

The Fig. 6 demonstrates the results of comparison the values of average heat transfer coefficients $\bar{\alpha}_{rs}$ in dependence on flue gases (combustion products) temperature $T_{fl} = T_{CP}$ by application of different types of recuperative sections (designs of tube recuperator: MD, SP1, SP2) with the basic design BD.

Within the range of moderate temperatures (lower than 800–850 K) the turbulizing inserts provide higher values of $\bar{\alpha}_{rs}$ for the cases of using the sections of types SP1, SP2 with the spiral inserts. If T_{CP} > 500 °C (773 K) the radial rib inserts SEE exceed the opportunities of enhancement the convective heat transfer compo-

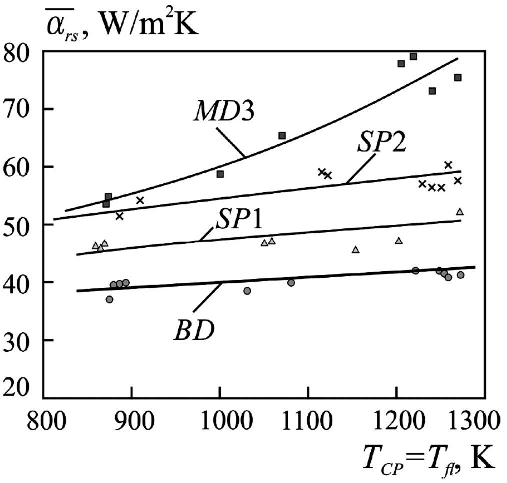

Fig. 6. Dependence of average value of inner heat transfer cofficient (HTC) $\overline{\alpha}_{rs}$ for tube U-like recuperative section conisting of three loops (Fig.5) of different design on temperature within the combustion chamber T_{fl} of firing rig. The $\overline{\alpha}_{rs}$ values are conditionally related to inner surface of the tubes within the section. Mass (volume) air flow rate $\dot{m}_a$ = 0.053 kg/s ($\dot{V}_a$= 150 nm³/h = idem).

nent. As a result the recuperative section MD3 becomes the most effective HE in range of high temperatures ($T_{CP} \geq 500$ °C) [9].

As could be seen by observation the Fig. 6, the dependences of coefficients $\overline{\alpha}_{rs}$ on T_{CP} have the similar character for the recuperative sections of types SP1, SP2 like that for basic design BD. The curves $\overline{\alpha}_{rs}$ (T_{CP}) have the slightly increased values with T_{CP}. That's peculiar to convective heat exchange with very low radiative component.

Unlike mentioned dependences in case of using the recuperative section with the cross-shaped insert MD3 beginning of T_{CP} ~ 800 K the $\overline{\alpha}_{rs}$ value is changed suddenly by growth the flue gases temperature. It means an increase of radiative heat transfer role within the range T_{CP} = = 500–1000 °C (773–1273 K) to be analyzed (Fig. 6).

Conclusion

1. The most important trends and approaches for advancement the designs and calculation procedures of heat exchange equipment are considered in the paper, first of all — from the standpoint of heat transfer enhancement. Main attention has been paid to the middle — and high temperature heat exchangers (HE) using as the heat recovery facilities of the furnace and boiler plants.

By classification of HE the centralized recuperators and regenerators, the thermal and thermochemical recuperators, recuperative and regenerative burners are picked out and considered.

By this approach the main characteristics of furnaces: using and saving the fuel and energy, operation and maintenance, environmental protection — related to application the heat recovery plants are evaluated for further optimization.

2. Similarity of processes the heat transfer and hydraulic resistance is considered for different schemes of the HE with evaluation the Reynolds analogy factor k. It has been shown that maintanance of analogy factor k > 1.0 or equation $Nu/Nu_0 = C_f / C_{f,0}$ make the criteria confirming the same change of thermal and hydraulic characteristics by divergence of plant's conceptual design from the basic version.

3. Main techniques of heat transfer enhancement are realized and compared by means of arrangement the turbulizing inserts within the channels of HE or by attaching the visible roughness for heat exchange surfaces or by arrangement of various obstacles at the surface under consideration. At the moment the most spread type of the inserts make the twisted tapes (TT) within the channels of different cross-section. From the standpoint of providing the minimum hydraulic resistance the most popular solution belongs to arrangement the dimples by heat exchange surface of different design and profile: plain and axially symmetric (cylinder), of different curvature — curvilinear geometry.

The examples of the turbulizing elements been computed, tested and analyzed in literature are presented by the dimples of symmetric form (hemispherical cylindrical), V-shape dimples, asymmetric holes, by combination of elements: the ribs-dimples, protrusions-dimples. Combination of different obstacles this SEE flow could be used for enhancement the thermal performance of heat exchange surface. installed

4. It has been ascertained by the results of own author's experimental researches that more simple than TT designs of the secondary energy emitters (SEE) in form of radial ribs are preferable for using to enhance heat transfer by cross-flow heat exchange of in-tube air flow with combustion products moving in intertube space provide the better opportunities. By this the Reynolds analogy serves as unambiguous criterion for goodness of fit the transfer processes for local and great systems.

References

1. Zhukauskas A.A. [Convective transfer in heat exchangers], Moscow : Nauka, 1982. 472 p. (Rus.)

2. Khinkis M., Kozlov A., Kalensky D., High Efficiency Combustion System for Stirling Engine. *Proceedings of the 2nd Thermal and Fluid Engineering Conference, TFEC2017 4th International Workshop on Heat Transfer, IWHT2017,* April 2–5, 2017, Las Vegas, NV, USA–TFEC–IWHT–2017–18111, pp.1–6.
3. OPTIMELT system at Leerdam. *Praxair*/www.lifeoptimelt.com/life8.htm, 2018.10.19., pp. 1–4.
4. Guillet, R. The Humid Combustion to Protect Environment and to Save the Fuel: the Water Vapor Pump and Maisotsenko Cycles Examples. *International Journal of Energy for a Clean Environment*, 2011, No. 12 (2–4), pp. 259–271.
5. Bogdanova V., Moreau J., Guillet R., The water vapor pump technologies back in France. *CIEC,* June, 2014, pp. 1–4. — https://www.ciec.fr/wp-content/themes/ciec/assets/fichier/english-pave.pdf.
6. Mahmood M.H., Sultan M., Miyazaki T., Maisotsenko V. Overview of the Maisotsenko cycle — A way towards dew point evaporative cooling, *Renewable and Sustainable Energy Reviews,* 2016, Vol. 66, pp. 537–555, ISSN 1364-0321. — https://doi.org/10.1016/j.rser.2016.08.022.
7. Pandelidis D., Anisimov S., Rajski K., Brychcy E., Sidorczyk M. Performance comparison of the advanced indirect evaporative air coolers. *Energy,* 2017. Vol. 135. pp. 138–152.
8. Fantuzzi M. Revamping of reheating furnaces. *Heat Processing.* 2015. No. 4. pp. 51–57.
9. Soroka B., Vorobyov M., Zgurskyy V., Karabchievskaya R. Heat transfer and Aerodynamic Resistance of Modern High-Temperature Tube Heat Exchanger Equipped with the in-Tube Inserts. *Proceedings of XV Minsk Intern. Heat and Mass Transfer Forum*. Scientific Edition, Minsk, 23–26 May, 2016. Vol. 3, pp. 206–210. (Rus.)
10. Omidi M., Farhadi M., Jafari M., A comprehensive review on double pipe heat exchangers. *Appl. Therm. Eng.* 2017. Vol. 110. pp. 1075–1090.
11. Burtsev S., Vasil'ev V., Vinogradov Yu., Kiselev N., Titov A. Experimental study of parameters of surfaces coated with regular relief. *Scientific Periodical of the Bauman MSTU,* El #FS77-482111, # 04212000025, January, 2013. pp.263–290. — https://technomag. bmstu.ru/doc/ 532996.html #01 / (Rus.)
12. Maximize Thermal Efficiency. Minimize Footprint. Media Library Funding Incentives. *Fuel Efficiency LLC. Clyde*, New York, 2018, pp.1–6.
13. Man C., Lv X., Hu J., Sun P., Tang Y. Experimental study on effect of heat transfer enhancement for single-phase forced convective flow with twisted tape inserts. *International Journal of Heat and Mass Transfer*, 2017, Vol. 106, pp. 877– 883.
14. Man Z. C., Yao J., Wang C. The Experimental study on the Heat Transfer and Friction Factor Characteristics in tube with a new kind of twisted tape insert. *Int. Commun. Heat Mass Transf.* 2016. Vol. 75. pp. 124–129.
15. Bas H., Ozceyhan V., Heat transfer enhancement in a tube with twisted tape inserts placed separately from the tube wall. *Experimental Thermal and Fluid Science.* 2012. Vol. 4l. pp. 51–58.
16. Yang L., Du K. A comprehensive review on heat transfer characteristics of TiO_2 nanofluids. *Int. J. Heat Mass Transfer*. 2017. Vol. 108. pp.11–31.
17. Wadekar V. Ionic liquids as heat transfer fluids — an assessment using industrial exchanger geometries. *Appl. Term. Eng.* 2017. Vol. 111. pp. 1581–1587.
18. Sundar L.S., Sharma K.V., Singh M.K., Sousa A.C.M. Hybrid nanofluids preparation, thermal properties, heat transfer and friction factor, A review. *Renewable and Sustainable Energy Reviews.* 2017. Vol. 68. pp. 185–198.
19. Kumar S., Kothiyal A., Bisht M., Kumar A. Effect of Nanofluid Flow and Protrusion Ribs on Performance in Square Channels: an Experimental Investigation. *Journal of Enhanced Heat Transfer.*, 2019. Vol. 26. Iss. 1. pp. 75–100.
20. Ahmed H., Yusoff M., Hawlader M.. Ahmed M.I., Salman B.H., Kerbeet A.S. Turbulent Heat Transfer and Nanofluid Flow in a triangular duct with vortex generators. *Int. Journal of Enhanced Heat Transfe.* 2017. Vol. 26. Iss. 1. pp. 105:495–504.
21. Agromayor R., Cabaleiro D., Pardinas A.. Vallejo J., Fernandez-Seara J., Lugo L. Heat Transfer Performance of Funktionalized Graphene Nanoplatelet Aqueous Nanofluid. *Materials (Basel).* 2016. Vol. 9/455. pp. 1–18. doi:10.3390/ma 9060455.
22. Specht E. Heat and Mass Transfer for high temperature processes in Thermoprocessing: *Fundamentals. Calculations. Processes. Essen, Vulkan Verlag*, 2017, 596 p.
23. Soroka B.S. Intensification of Thermal Processes in Fuel Furnaces. Kiev : Naukova Dumka, 1993. 413 p. (Rus.)

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Новые тенденции разработки и совершенствования современных теплообменников

Рассмотрены некоторые актуальные аспекты совершенствования теплообменного оборудования. Обсуждаются с использованием соответствующих современных подходов к процессам переноса актуальные вопросы интенсификации теплообмена в средне- и высокотемпературных рекуператорах.

Предложена классификация теплообменного, в том числе теплоутилизационного оборудования. Представлены и проанализированы основные характеристики рекуперативных установок.

Проблема аналогии Рейнольдса (подобия относительного изменения интенсивности теплообмена и изменения гидравлического сопротивления при варьировании конструктивных геометрических и режимных факторов) рассмотрена применительно к отдельным случаям потока вдоль однородной поверхности, а также каналов, снабженных лунками или выпуклыми элементами.

Коэффициент термического совершенства (TPF) процесса теплообмена качественно сходен с фактором аналогии Рейнольдса и сильно зависит от интенсивности теплообмена и коэффициента трения в условиях рассматриваемой схемы: при обтекании поверхности или течения внутри канала. В качестве рабочего тела в каналах теплообменника сравнивались различные среды: газообразные, жидкие и наножидкостные (последние практически используются с 2000 г.).

Проведен анализ влияния вторичных излучателей энергии (SEE), расположенных внутри трубных каналов, на результирующий тепловой поток в условиях теплопередачи между потоком продуктов сгорания и внутритрубным воздушным потоком. *Библ. 23, рис. 6*

Ключевые слова: аналогия Рейнольдса, излучение, конвекция, наножидкость, рекуперация теплоты, теплогидравлическая эффективность, теплоносители, теплообменник, утилизация теплоты, число Нуссельта.

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Новітні тренди розробки та удосконалення сучасних теплообмінників

Розглянуто деякі актуальні аспекти удосконалення теплообмінного устаткування. Обговорюються з використанням відповідних сучасних підходів до процесів переносу актуальні питання інтенсифікації теплообміну у середньо- та високотемпературних рекуператорах.

Запропоновано класифікацію теплообмінного, у тому числі теплоутилізаційного устаткування. Наведено та проаналізовано основні характеристики рекуперативного устаткування.

Проблема аналогії Рейнольдса (подібності відносної зміни інтенсивності теплообміну та зміни гідравлічного опору при варіюванні конструктивних геометричних та режимних факторів) розглянута стосовно до окремих випадків потоку вздовж однорідної поверхні, а також каналів, обладнаних лунками чи випуклими елементами.

Коефіцієнт термічної досконалості (TPF) процесу теплообміну якісно подібний до фактора аналогії Рейнольдса та у значній мірі залежить від інтенсивності теплообміну та коефіцієнта тертя в умовах схеми, що розглядається: при обтіканні поверхні або течії всередині каналу. Як робоче тіло у каналах теплообмінника порівнювалися різноманітні середовища: газоподібні, рідинні та нанорідинні (останні практично ви-

користовуються з 2000 р.).
Проведено аналіз впливу використання вторинних випромінювачів енергії (SEE), розташованих всередині трубних каналів, на результуючий тепловий потік в умовах теплопередачі між потоком продуктів згоряння та повітряним потоком всередині труб. *Бібл. 23, рис. 6.*

Ключові слова: аналогія Рейнольдса, випромінювання, конвекція, нанорідина, рекуперація теплоти, теплогідравлічна ефективність, теплоносії, теплообмінник, утилізація теплоти, число Нуссельта.

Список литературы

1. Жукаускас А.А. Конвективный перенос в теплообменниках. М. : Наука, 1982. 472 с.
2. Khinkis M., Kozlov A., Kalensky D. High Efficiency Combustion System for Stirling Engine. *Proceedings of the 2nd Thermal and Fluid Engineering Conference, TFEC2017 4th International Workshop on Heat Transfer, IWHT2017,* April 2–5, 2017, Las Vegas, NY, USA–TFEC– IWHT–2017–18111, P. 1–6.
3. OPTIMELT system at Leerdam. *Praxair* / www.life-optimelt.com/life8.htm, 2018.10.19, P. 1–4.
4. Guillet R. The Humid Combustion to Protect Environment and to Save the Fuel: the Water Vapor Pump and Maisotsenko Cycles Examples. *International Journal of Energy for a Clean Environment*, 2011, № 12 (2–4), P. 259–271.
5. Bogdanova V., Moreau J., Guillet R. The water vapor pump technologies back in France.*CIEC,* June, 2014, P. 1–4. — Режим доступа: https://www. ciec.fr/wp-content/themes/ciec/assets/fichier/english-pave pdf.
6. Mahmood M.H., Sultan M., Miyazaki T., Maisotsenko V. Overview of the Maisotsenko cycle — A way towards dew point evaporative cooling. *Renewable and Sustainable Energy Reviews.* 2016. Vol. 66. P. 537–555. — Режим доступа: https://doi.org/10.1016/j.rser.2016.08.022.
7. Pandelidis D., Anisimov S., Rajski K. et al. Performance comparison of the advanced indirect evaporative air coolers. *Energy.* 2017. Vol. 135. P. 138–152.
8. Fantuzzi M. Revamping of reheating furnaces. *Heat Processing*. 2015. № 4. P. 51–57.
9. Сорока Б.С., Воробйов М.В., Кудрявцев В.С., Згурский В.А., Карабчиевская Р.С. Теплообмен и аэродинамическое сопротивление современных высокотемпературных рекуператоров с внутритрубными вставками. *Материалы XV Минского международного форума по тепло- и массобмену : Тезисы докладов и сообщений.* Минск, 23–26 мая 2016 г. Т. 3. С. 206–210. — 20 стр. (CD).
10. Omidi M., Farhadi M., Jafari M. A comprehensive review on double pipe heat exchangers. *Appl. Therm. Eng.* 2017. Vol. 110. P. 1075–1090.
11. Бурцев С.А., Васильев В.К., Виноградов Ю.А. и др. Экспериментальное исследование характеристик поверхностей, покрытых регулярным рельефом. *Научная периодика им. Баумана МВТУ*, El #FS77-482111, # 04212000025, январь, 2013. С. 263–290.— Режим доступа: https://technomag. bmstu.ru/doc/532996.html #01.
12. Maximize Thermal Efficiency. Minimize Footprint. Media Library Funding Incentives, *Fuel Efficiency LLC. Clyde.* New York. 2018. P. 1–6.
13. Man C., Lv X., Hu J., Sun P., Tang Y. Experimental study on effect of heat transfer enhancement for single-phase forced convective flow with twisted tape inserts *International Journal of Heat and Mass Transfer*. 2017. Vol. 106. P. 877–883.
14. Man Z. C., Yao J. and Wang C. The Experimental study on the Heat Transfer and Friction Factor Characteristics in tube with a new kind of twisted tape insert, *Int. Commun. Heat Mass Transf.*, 2016, Vol.75, P. 124–129.
15. Bas H., Ozceyhan V. Heat transfer enhancement in a tube with twisted tape inserts placed separately from the tube wall. *Experimental Thermal and Fluid Science.* 2012. Vol. 4l. P. 51–58.
16. Yang L., Du K. A comprehensive review on heat transfer characteristics of TiO_2 nanofluids. *Int. J. Heat Mass Transfer*. 2017. Vol. 108. P. 11–31.
17. Wadekar V. Ionic liquids as heat transfer fluids — an assessment using industrial exchanger geometries. *Appl. Term. Eng.* 2017. Vol. 111. P. 1581–1587.
18. Sundar L.S., Sharma K.V., Singh M.K., Sousa A.C.M. Hybrid nanofluids preparation, thermal properties, heat transfer and friction factor, A review. *Renewable and Sustainable Energy Reviews.* 2017. Vol. 68. P. 185–198.
19. Kumar S., Kothiyal A., Bisht M., Kumar A. Effect of Nanofluid Flow and Protrusion Ribs on Performance in Square Channels: an Experimental Investigation. *Journal of Enhanced Heat Transfer.* 2019. Vol. 26. Iss. 1. P. 75–100.
20. Ahmed H., Yusoff M., Hawlader M. et al. Turbulent Heat Transfer and Nanofluid Flow in a triangular duct with vortex generators. *Int. Journal of Enhanced Heat Transfer.* 2017. Vol. 26. Iss. 1. P. 105:495–504.
21. Agromayor R., Cabaleiro D., Pardinas A. et al. Heat Transfer Performance of Funktionalized Graphene Nanoplatelet Aqueous Nanofluid. *Materials (Basel).* 2016. Vol. 9/455. P. 1–18. doi:10.3390/ma 9060455.
22. Specht E. Heat and Mass Transfer for high temperature processes in Thermoprocessing: *Fundamentals. Calculations. Processes. Essen, Vulkan Verlag*, 2017, 596 p.
23. Сорока Б.С. Интенсификация тепловых процессов в топливных печах. Киев : Наук. думка, 1993. 413 с.

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