Intensification of heat exchange in the developed surfaces in flat channels

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Abstract. The paper presents the results of an experimental study of heat transfer during forced water flow from microstructured surfaces in a slotted channel. The investigated micro-structured surfaces were obtained by a resource-saving (waste-free) method of deforming cutting with various constructive shapes and sizes. A description of the experimental setup, methods for conducting and processing experimental data are presented. The maximum increase in heat transfer coefficients in comparison with a smooth surface reaches \( \frac{\text{Nu}}{\text{Nu}_0} = 16.2 \).

Introduction

Modern technologies for the manufacture of radio elements show a tendency to reduce their size and increase specific heat fluxes. Without an effective way to remove the heat generated by radio elements, their temperature will increase, which will lead to equipment failure. Therefore, water cooling systems are important elements of electronic and power equipment used in various sectors of the economy. One of the effective ways to increase the heat transfer coefficient in single-phase forced convection is the use of microstructured surfaces. Webb R.L., Kandlikar S.G., Grande, W.J. [1] M. Reeves, Moreno J. [2], Bessanane N, Si-Ameur M [3], Xu J, Zhang K, Duan J [4], Rebay M., Kakac S. [5], Kravets V.Yu, Konshina V.I., Popova I.A. [6] and many others show the relevance of the topic of intensifying heat transfer from developed surfaces. The search for the optimal form of microfinning of the heat transfer surface continues in terms of thermal and hydraulic efficiency. An important role is played by the technology of manufacturing microstructured surfaces from the point of view of waste-free production. Studies of the influence of the geometric parameters of a microstructured surface on the increase in the heat transfer coefficient, and the subsequent interpretation of the results obtained in real cooling systems, reflect the demands of the real sectors of industry.

The aim of the work is to study the flow structure and the mechanisms of heat transfer enhancement under forced water flow of flat slotted channels with pin and horseshoe structures of various geometries in the range of low Reynolds numbers.

In the course of the work, the following main tasks were performed:

1. To develop and create an experimental stand for studying the structure of the water flow and heat transfer processes of flat slotted channels with pin and horseshoe structures.
2. To develop methods for conducting and processing research results, taking into account the geometric and regime features of the investigated flat slot-left channels with pin and horseshoe-shaped structures.
3. Conduct experiments to study the processes of heat transfer in the forced flow of water from microstructured surfaces in a slotted channel.
4. Analyze the obtained experimental studies and make comparisons with the results of other authors.

1 Object of research

As microstructured surfaces for intensifying heat transfer in the slotted channel, surfaces obtained by the deforming cutting method, which is a combination of cutting and bending the surface layers of the heat exchange surface, were used [1-3]. For non-ferrous alloys, the surface area after processing by the deforming cutting method can be increased up to 2.5 times.

Fig. 1. Fragments of structures: a - pin; b – horseshoe.
In the experiment, we used three plates with pin structures (Fig. 1, a), each of which has a spiral shape (Fig. 1, b) and one plate with horseshoe structures.

Height \( h_{pin} \) and diameter \( d_{pin} \) of pin structures on plates: No. 1 – 4.75 \( \times \) 1 mm, No. 2 – 2.1 \( \times \) 0.5 mm, No. 3 – 0.65 \( \times \) 0.13 mm (Fig. 1, a). The pin structures are arranged in a corridor and staggered order. The height of horseshoe-shaped structures on plate No. 4 \( h_{pin} = 0.2 \) mm (Fig. 1, b).

**2 Description of the experimental stand**

The experimental stand (Fig. 2) for studying the intensification of heat transfer during forced water flow from microstructured surfaces in a slotted channel consists of a system for creating and stabilizing water flow, a system for supplying heat to the microstructured surfaces under study, and a National Instruments data acquisition system. Primary transducers of temperature, pressure, water flow, current strength and voltage drop are connected to 3 remote isothermal terminal blocks NI TBX-1328. Programming of the system, registration and processing of measurement results is carried out in the graphical development environment LabVIEW. Thanks to this, the data from the primary converters are displayed on the monitor in real time.

The hydraulic circuit (Fig. 2) includes a tank 1 with water, from which water is supplied through a filter 2 using a pump 3 through a damper 4 to the working section 6. The damper 4 makes it possible to minimize fluctuations in pressure and water flow caused by work pump 3. Next, the water flow passes through a Coriolis flow meter-regulator 5 of the CORI-Flow brand, the readings of which are recorded by the information collection system. Temperature sensors 7 are installed at the inlet and outlet of the working section 6. The static water pressure drop is measured by a pressure sensor 8. Then the water passes through the cooling radiator 9 and returns back to the reservoir 1.

![Fig 2. Schematic diagram of the bench for studying the processes of heat transfer intensification: 1 - reservoir, 2 - filter, 3 - pump, 4 - damper, 5 - flow meter-regulator, 6 - working section, 7 - thermocouples, 8 - pressure sensor, 9 - heat exchanger, 10 - autotransformer, 11 - electric heating element.](image)

The system for supplying heat to the surfaces under study consists of an electric heating element 11 connected to an autotransformer 8, an ammeter I and a voltmeter V. This allows smooth control of the temperature of the microstructured surface.

The working area (Fig. 3) is a flat slotted channel 2 formed by a plate 3 with pin or horseshoe-shaped structures and a cover 1. The plate 3 is placed on the base of the asbestos-cement box 6. The height of the slotted channel is formed by interchangeable inserts of glass-textolite 5 depending on from the height \( h \) of pin or horseshoe structures. In cover 2, inlet and outlet pipes are mounted. Through the inlet pipe, water enters the flat slotted channel 2, and, accordingly, through the opposite pipe, the water exits it. In the cover 1, in the vicinity of the nozzles, temperature meters 4 (chromel-copel thermocouples) are installed, which allow you to control the temperature of the water at the inlet and outlet of the flat slotted channel. Inside the asbestos-cement box 6 there is an electric heating element 7, which is a winding of nichrome wire. The asbestos-cement box 6 is installed on a fiberglass base 9. The space between the electric heating element 7 and the fiberglass base 9 is filled with basalt wool 8.

The surface temperature of the microstructured surfaces was controlled by eight chromel-copel thermocouples 10, located along the length and width of the plates 3. The thermocouples were pinned on the outer free surface of the plates 3. The temperature of the pin structures and horseshoe-shaped structures was not how thermocouple junction dimensions are comparable to their size.

![Fig 3. Sketch of the working area: 1 – cover, 2 – flat slotted channel, 3 – plate, 4 – thermocouple, 5 – replaceable fiberglass insert, 6 – asbestos-cement box, 7 – electric heating element, 8 – basalt wool, 9 – fiberglass base, 10 – thermocouple.](image)

**3 Methods for conducting experimental research and processing experimental data**

Experimental studies of heat transfer intensification with forced flow of water from microstructured surfaces in a slotted channel were carried out at a steady value of the average flow rate of water flow \( w \) and a steady value of a given heat flux \( Q \). The temperature values of the experimental plates, the inlet and outlet temperatures of water from the working area, mass flow of water, voltage and current on the electric heating element are monitored on the monitor and recorded in the information collection system. Then the next experimental mode is set up.

The average temperature of the wall surface was determined along the length of the plates at 5 points, \( ^\circ C \):
where \( \bar{t}_w = \left( t_{w1} + t_{w2} + \bar{t}_{w3} + t_{w4} + t_{w5} \right) / 5 \)

is the control of the heat flux distribution uniformity at 4 points across the plate width.

According to the measured values of the water temperature at the inlet and outlet, the average temperature of the coolant in the pipe was found, \( ^\circ\text{C} \):

\[
t_f = \frac{(t_{in} + t_{out})}{2},
\]

where \( t_{in} \) and \( t_{out} \) are the water temperatures at the inlet and outlet.

The heat flux transferred to water is determined to control the accuracy of measurements by two methods, \( W \):

a) by the value of electric power:

\[
Q_e = U \cdot I,
\]

where \( U \) and \( I \) are the voltage drop (V) and the current strength (A) on the electric heating element.

b) calorimetric method:

\[
Q = G \cdot c_p (t_{out} - t_{in})
\]

where \( G \) is the mass flow rate of water (kg/s); \( c_p \) is the heat capacity of water at an average temperature in the working area (J/(kg K)); \( (t_{out} - t_{in}) \) is the difference between the temperatures of the water at the exit from the working area and at the entrance to it.

Losses of thermal power to the environment were estimated by the formula for heat transfer from horizontal and vertical plates with free convection in an open volume, \( W \):

\[
Q_i = F \cdot q_i,
\]

for \( \lg(\text{Gr} \cdot \text{Pr}) = 3.3 \ldots 7.3 \) \( \text{Nu} = 0.525(\text{Gr} \cdot \text{Pr})^{0.25} \),

for \( \lg(\text{Gr} \cdot \text{Pr}) = 7.3 \ldots 12 \) \( \text{Nu} = 0.13(\text{Gr} \cdot \text{Pr})^{1/3} \).

The value of \( Q_i \) for forced water flow in the channel, taking into account temperature control on the surface of the working area, low heat transfer coefficients with free convection of air near the working area, wall materials of the working area, made of materials with low thermal conductivity coefficients (thermal insulation) working area, is not more than 1% of the \( Q \) value.

The specific heat flux density was calculated according to the formulas, W/m²:

\[
q_w = \frac{Q_e - Q_i}{F_0} = \frac{Q}{F_0} \quad \text{and} \quad q_w' = \frac{Q_e - Q_i}{F} = \frac{Q}{F}
\]

where \( F_0 \) is the area of the channel surface washed by water, reduced to an initially smooth surface according to the studies of Nunner W. [8], Koch R. [9], and J.F. Tullius [10]. \( F \) is the area of the channel surface washed by water, taking into account the increase in the surface due to the pin structures. The assumption about the use of \( F_0 \) was made as a result of a significant increase in the surface area \( F \) of the plates due to the surface of the pin structures, compared with the initially smooth surface \( F_0 \). The calculated values of the ribbing efficiency coefficients \( \eta_{pin} \approx 0.61 \ldots 0.69 \) for the studied pin structures allow us to make assumptions about the efficiency of heat transfer intensification due to an increase in the surface area \( F > F_0 \).

The heat transfer coefficient was determined by the expressions, W/(m²·C):

\[
\alpha = \frac{q_w}{(t_{w} - t_f)} \quad \text{and} \quad \alpha' = \frac{q_w'}{(t_{w} - t_f)}.
\]

The Reynolds number \( \text{Re}_D \) is determined from the average flow velocity \( w_0 \) and \( w \), equivalent to the diameters \( D_0 \) and \( D \):

\[
\text{Re}_D = \frac{w_0 \cdot D_0}{\mu} \quad \text{and} \quad \text{Re}_D' = \frac{w \cdot D}{\mu}
\]

where \( F_{ch} \) is the cross-sectional area of the channel, taking into account its overlap with pin structures, \( F_{ch} = b \cdot h \) is the cross-sectional area of the initially smooth flat slotted channel No. 0, \( b \) and \( h \) are the width and height of the channel, \( F_{ch} \) is the area of the transverse channel, taking into account its overlap with pin structures structures. This condition is accepted on the basis of the equality of flow rates in \( G_0 = G \) in a flat slotted channel with smooth No. 0 and pin and horseshoe-shaped structures on plates No. 1 – No. 4.

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\]

The Nußelt number \( \text{Nu}_D \) is determined, without taking into account and taking into account pin structures, according to the dependencies:

\[
\text{Nu}_D = \frac{\alpha \cdot D}{\lambda} \quad \text{and} \quad \text{Nu}_D' = \frac{\alpha \cdot D}{\lambda}
\]

where \( \lambda \) is the thermal conductivity of water at an average water temperature.
4 Test experiments

Before the start of the research, test experiments were performed on the heat transfer of a smooth slotted channel No. 0 with forced water flow in the range of \( \text{Re}_D = 30 \,–\, 600 \) numbers. the reliability of the results obtained. It has been established (Fig. 4, a) that the obtained results agree satisfactorily on the average heat transfer of a smooth slotted channel No. 0 with forced water flow in the range of \( \text{Re}_D = 30 \,–\, 600 \) with the known dependence for the laminar regime:

\[
\text{Nu}_D = 0.15 \cdot \text{Re}_D^{0.33} \cdot \text{Pr}_f^{0.43} \cdot \left( \text{Gr}_f \cdot \text{Pr}_f \right)^{0.1} \left( \frac{\text{Pr}_f}{\text{Pr}_w} \right)^{0.25}
\]

The data on the average heat transfer are in satisfactory agreement with the experimental data of Nunner W. [8], Koch R. [9], Dreitser G.A. [11] and Olimpiev V.V. [12]. For surface No. 4 with horseshoe structures, no earlier laminar-turbulent transition was found. This is explained by the low height \( h_{\text{pin}} \) of the horseshoe-shaped structures that sink in the boundary layer, and is in good agreement with the results of Tarasevich S.E [13] on the heat transfer of pipes with metric threads.

The increase in the average heat transfer (without taking into account the increase in the surface due to pin structures) of the slotted channel in the range of numbers \( \text{Re}_D = 30 \,–\, 600 \) for surfaces was: No. 1 \( \text{Nu}/\text{Nu}_0 = 9.5 \,–\, 16.2 \); No. 2 \( \text{Nu}/\text{Nu}_0 = 3.8 \,–\, 7.1 \); No. 3 \( \text{Nu}/\text{Nu}_0 = 1.2 \,–\, 2.2 \) (Fig. 4, a) and significantly depends on the height \( h_{\text{pin}} \) of the pin structures. The high level of growth in the average heat transfer \( \text{Nu}/\text{Nu}_0 \) allows us to attribute the mechanisms of heat transfer enhancement for the considered plates with pin structures to the mechanisms characteristic of the mechanisms of heat transfer enhancement by ordered porous structures [14].

The level of average heat transfer for channel No. 4 with horseshoe structures is comparable to the level of average heat transfer for a smooth channel \( \text{Nu}/\text{Nu}_0 = 1 \). A significant increase in pressure losses in the working section for channel No. 4 reaches \( \Delta \rho/\Delta \rho_0 = 20 \), because the height of the channel with structures is \( h = 0.2 \) mm. Therefore, studies of the average heat transfer of channel No. 4 were carried out only up to the value of \( \text{Re}_D = 300 \). Channel No. 4 with horseshoe structures showed low thermal \( \text{Nu}/\text{Nu}_0 \) and thermohydraulic \( (\text{Nu}/\text{Nu}_0)/(\Delta \rho/\Delta \rho_0) \) efficiency compared to channels with pin structures on plates No. 1 – No. 3.

Experimental studies of heat transfer enhancement under forced water flow from microstructured surfaces in a slotted channel were carried out in the range of \( \text{Re}_D = 30 \,–\, 600 \) for plates No. 1 – No. 4. The channel is formed without a gap above the tops of the pin structures, \( h = h_{\text{pin}} \). The channel for plate No. 2’ is formed with a gap \( h' = 1 \) mm above the tops of the pin structures \( h = h_{\text{pin}} + h' = 3.1 \) mm.

It has been established that in a flat slotted channel with plates No. 1 – No. 3, an earlier laminar-turbulent transition is observed compared to a smooth slotted channel No. 0. An external manifestation of the interaction of pin structures with the flow in the region of transition numbers Re is a decrease in the critical number \( \text{Re}_{\text{cr}} \) at an increase in their height \( h_{\text{pin}} \) (Fig. 4). The experimental data in the considered range, the nature of their change are in satisfactory agreement with the experimental data of Nunner W. [8], Koch R. [9], Dreitser G.A. [11] and Olimpiev V.V. [12]. For surface No. 4 with horseshoe structures, no earlier laminar-turbulent transition was found. This is explained by the low height \( h_{\text{pin}} \) of the horseshoe-shaped structures that sink in the boundary layer, and is in good agreement with the results of Tarasevich S.E [13] on the heat transfer of pipes with metric threads.

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The increase in the average heat transfer for channel No. 2’ with a gap \( h' = 1 \) mm above the tops of the pin structures is \( \text{Nu}/\text{Nu}_0 = 2.54 \,–\, 5.48 \), which is lower than the heat transfer level for channel No. 2. Most of the water flow passes over the tops of the pin structures through the cross section formed by the gap \( h' = 1 \) mm.

The data on the average heat transfer are in satisfactory agreement with the results on the average heat transfer of pipes with metric threads [13], annular and hemispherical protrusions [7,11]. In addition, the use of channel No. 2’ with a gap \( h' = 1 \) mm above the tops of the pin structures allows significantly up to \( \Delta \rho/\Delta \rho_0 = 1.8 \) times to reduce the growth of pressure losses in the working section for channel No. 4 compared to channel No. 2.
It has been established that the increase in pressure losses in the working section for a plate with pin structures No. 1 reaches \( \Delta p/\Delta p_0 = 3.1 - 6.8 \). The low level of hydraulic losses can be explained by the ordered geometric shape of the placement of pin structures on the plate surface (staggered arrangement). Longitudinal pitch between pin structures \( t_1 = 4 \text{ mm} \), transverse pitch \( t_1 = 3 \text{ mm} \). This circumstance makes it possible to determine the advantage of pin structures over highly porous materials and ordered porous materials from [14, 15]. There are no dead-end and longitudinal in the channel with pin structures there are no dead-end and closed pores, in comparison with highly porous materials [14]. In addition, when forming pin structures by the deforming cutting method, there is no loss of material (chips), in comparison with the mechanical processing of ordered porous materials [15].

Satisfactory agreement was obtained between the average heat transfer \( Nu_D \) of the slotted channel with pin structures No. 3 (Fig. 4, a) with the similar results of Tullius J.F. [10] for a slotted channel with pin ribs in the form of a parallelepiped \( (h_{pin} = 0.66 \text{ mm}) \) in the range of \( Re_D = 100 - 600 \).

Comparison of the obtained values for the thermal resistance \( R_{tot,D} = 0.074 \text{ K/W} \) of the slotted channel with the pin structures of surface No. 2 at the flow point of \( 8.44 \times 10^3 \text{ kg/s} \) is in satisfactory agreement with the results of Webb R.L. [16] according to the thermal resistance \( R_{tot,D} = 0.068 \text{ K/W} \) for the "Fin-H" slotted channel with the height of flat microfins \( h_{fin} = 2.1 \text{ mm} \) at the flow point 8.33 \( \times 10^3 \text{ kg/s} \). Some slight difference is due to the height difference and the shape of the pin structures and flat fins.

The graphic dependence of the average heat transfer of channel No. 1 with pin structures on the surface (Fig. 4, b) made it possible to establish that an increase in the heat transfer surface \( F \) due to the presence of pin structures on the surface of the plates and an increase in the average flow rate of water flow \( w' \) are determining mechanisms of heat transfer intensification. The increase in the surface area of No. 1 was \( F/F_0 = 3.5 \text{ times} \) due to the surface area of the pin structures. The increase in the average flow rate \( w/w_0 = 1.3 \) occurred due to a decrease in the cross-sectional area. The calculated value of the fin efficiency factor for surface No. 1 is \( \eta_{pin} \approx 0.61 \). The effect on the intensification of heat transfer due to an increase in the surface area and the average flow rate was \( \approx 62\% \). Additional factors for intensifying the heat transfer of channel No. 1 is the intensive mixing of water in the gaps between the pin structures up to \( \approx 28\% \).

This circumstance is confirmed by the results of visualization of the water flow in the smooth slotted channel No. 0 (Fig. 5, a) and in the channel with plate No. 1 (Fig. 5, b). Visualization was performed by a high-speed Photron Fastcam SA4-500K-C1 camera using dyes. The dye is fed into the channel in portions through a dosing device into the inlet pipe. A video recording of the visualization of the water flow is made. The ink supply to the channel is cut off at the moment of formation of the vortex structure. After the complete development of a portion of ink, a new portion is fed into the channel. In smooth channel No. 0, a white water-based paint was used. Channel 1 used a water-based fluorescent paint that glows in the ultraviolet spectrum. This choice of dye is due to the complexity of estimating the flow structure in channels with pin structures.

It has been established that at low values of the number \( Re_D = 169 \), the water flow at the inlet to the smooth slotted channel No. 0 does not spread uniformly over the width of the channel \( b \). Two vortex structures are formed. The upper vortex structure (Fig. 1, a) rotates clockwise, while the lower vortex structure rotates in the opposite direction. According to the chronology of development, there is an increase in the size of the vortex structures, which become commensurate with the size of the slotted channel. Stagnant zones are formed in the corner areas of the flat slotted channel. This flow pattern is also characteristic with an increase in the number up to \( Re_D = 260 \) with an increase in the speed of rotation of the vortex structures.

With an increase in the Reynolds number to \( Re_D = 414 \) in a smooth slotted channel No. 0, the water flow begins to be more evenly distributed over the width of the channel. Large-scale vortex structures are much smaller. The speed of rotation of the vortex structures and their movement along the length of the channel \( L \) increases. In the corner areas of the flat slotted channel there are stagnant zones.

An excellent picture of the flow of water flow is observed at low values of the number \( Re_D = 169 \) in a flat slotted channel with pin structures No. 1. Starting from the inlet pipe, the water flow is evenly distributed over the width \( b \) and length \( L \) of the channel in a short time interval \( < 0.04 \text{ seconds} \). There are no vortex structures and stagnant areas. This partly explains the increase in the average heat transfer \( Nu/Nu_0 = 3.8 \) with a fairly significant increase in pressure losses in the working section \( \Delta p/\Delta p_0 = 3.1 \). A similar pattern of water flow is observed in the entire range of \( Re_D \) numbers studied.

**Conclusion**

The resource-saving (waste-free) method of deforming cutting made it possible to form ordered geometric structures on the surface of the plates with high values of the ribbing efficiency coefficients \( \eta_{pin} \approx 0.61 - 0.69 \) for the studied pin structures.
Experimental studies of the average heat transfer from surfaces with pin structures No. 1, No. 2, No. 2' and No. 3 with forced water flow in the slot channel showed high thermal \( \text{Nu/Nu}_0 \) and thermohydraulic \( \left( \frac{\text{Nu/Nu}_0}{\Delta p/\Delta p_0} \right) \) efficiency in the range of numbers \( \text{Re}_D = 30 – 600. \) Graphical dependences confirm that the use of pin structures on the surface of the plates can repeatedly (up to \( \text{Nu/Nu}_0 = 16.2 \)) increase the average heat transfer with a comparable increase in pressure losses in the working area reaches (up to \( \Delta p/\Delta p_0 = 0.68 \)). A graphic dependence of the increase in the average heat transfer with an increase in the height of the pin structures has been established.

Experimental studies of average heat transfer for channel No. 4 with horseshoe-shaped structures showed low thermal \( \text{Nu/Nu}_0 \) and thermal-hydraulic \( \left( \frac{\text{Nu/Nu}_0}{\Delta p/\Delta p_0} \right) \) efficiency compared to channels with pin structures on plates No. 1 - No. 3.

Further development of research will be aimed at studying the processes of boiling of various liquids on surfaces with pin and horseshoe structures.

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