Experimental investigation of heat transfer efficiency on the convex surface of the annular channel of varying geometry intensifiers

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Abstract. The work is a continuation of the experimental studies on the enhancement of heat transfer in the fuel assembly on the experimental stand in National Research University "Moscow Power Engineering Institute". The description of the experimental setup, construction and main geometrical parameters of intensifier are presented. The new experimental data on the pressure loss and heat transfer coefficient using an edge enhancer - twisted wire single-phase convection mode are presented. In the research, the range mode parameters and geometric characteristics of the intensifier were extended. The relation of the coefficients of hydraulic resistance and the Nusselt number of steps twist twisted wire was found, the effect of the ribs on the heat transfer coefficient was shown. It is found that for any twist pitch ranging from 20 to 100 mm corresponds to a maximum heat transfer rib height \( \hat{H} = 0.35 \). An increase in the heat transfer coefficient in the convex heating surface was experimentally obtained.

1. Introduction

Experimental studies conducted by a team of authors in [1] described above have shown that the use of intensifiers heat in which a positive effect is achieved through the interaction of swirling and transit (axial) flows in the annular channel is justified and effective. The intensifier, the main element of which was the system, produces formation of swirling and transit flows in the circle gap: edge-twisting wire. The interaction of flows with each other, with the ribs and the walls of the swirl ring wire channel throughout the length of the heated surface occurs within a plurality of additional and powerful vortex flow turbulence.

In [1] presents the results of experiments carried out previously in a wide range of regime parameters at twist wire step \( t = 50 \times 10^{-3} \) m.

It was found that

- a significant increase in heat transfer (80%);
- depending on the coefficients of hydraulic resistance and heat transfer from the relative height of the ribs are nonmonotonic, are clearly marked peaks;
• the maximum value of the heat transfer coefficient obtained at the value \( H = 0.35 \), and the coefficient of hydraulic resistance - at \( H = 0.23 \).

To complete the work in the field of single-phase convection authors believed it was appropriate to conduct new studies, sufficiently detailed thermal-hydraulic characteristics, varying the step intensifier spin at different values of the dimensionless ribs height \( H \).

2. The description of experimental installation and test section
The Experimental studies were carried out on the installation whose main systems, equipment, work area design and methodological principles are described in detail in [2].

To determine the temperature field along the length of the heating element 11 thermocouples were installed on the inner surface of test section. Using compensation wires via the switching device thermocouples connected to the Centre for Automation of Thermophysical Research (CATR). To measure the voltage directly at the working area on the inner surface of the heated pipe potentiometric bends installed. The working section is placed in a high-pressure vessel (HPV), the inner wall which with the cylindrical surface of the heating element forms a circular channel. The size of the annular channel is \((d_2 - d_1)/2 = 1.7 \times 10^3\) meters, less than the size of the capillary constant. Therefore, this annular channel can be attributed to the type of gap or mini channels. HPV placed vertically in the hydraulic circuit.

Main characteristics of the annular channel (Figure 1):
- the inner diameter of the annular channel (the outer diameter of the heating element) – \( d_1 = 12.9 \times 10^{-3} \) m;
- the outer diameter of the annular channel (the inner diameter of HPV) – \( d_2 = 16.3 \times 10^{-3} \) m;
- the inner diameter of the heating element – \( d_3 = 10.0 \times 10^{-3} \) m;
- area of the flow cross section of the annular channel – \( S = 77.9 \times 10^{-6} \) m
  
- hydraulic diameter of the smooth annular channel – \( d_g = d_2 - d_1 = 3.4 \times 10^{-3} \) m;
- the length of the heating element corresponds to the distance between the selection pressure – \( l = 0.7 \) m;
- length of the upper current lead – \( l_1 = 0.55 \) m;
- length of the lower current lead – \( l_2 = 0.25 \) m.

Figure 1. Scheme of annular channel: 1 – annular channel; 2 – heating element; 3 – the inner surface of HPV; 4, 5 – lower and upper current lead, respectively; 6, 7 – potentiometric bend

3. Technical parameters and construction of intensifier
Intensifier is designed to create in an annular channel two interacting flows, the presence of which increases the efficiency of heat transfer. The description of the intensifier presented in [1].

General view of the intensifier mounted on a convex surface of the test section shown in Figure 2.
Experiments on heat transfer coefficients and hydraulic resistance were carried out at the steps twist intensifier $t = 20 \cdot 10^{-3}$ m, $30 \cdot 10^{-3}$ m, $40 \cdot 10^{-3}$ m, $50 \cdot 10^{-3}$ m, $60 \cdot 10^{-3}$ m, $100 \cdot 10^{-3}$ m. The series of several experiments to determine heat transfer and hydraulic resistance at various rib heights (obtained over 300 experimental points, including twisting during winding wire directly on a heated work area) was carried out. A comparison of the array of experimental data for the $t = 50 \cdot 10^{-3}$ m [3] showed good reproducibility of measurements.

In [4], the main geometric characteristics are the intensifier rib height $h$ and the diameter of the wire twisting $d$. In processing the experimental data used dimensionless parameter $\hat{H}$ (relative height of the ribs) and the coefficient of swirl flow $k$, associated with a twist pitch $t$:

$$\hat{H} = \frac{h}{(d_2 - d_1)/2}, \quad k = \pi \bar{d}/t,$$

where $\bar{d}$ - the diameter of the middle line of the swirling flow, and $\hat{H}$ inherently determines the ratio of the swirling flow characteristics and transit (axial) flows.

4. Pressure drops and hydraulic resistance

Use of intensifiers, of course, leads to a substantial increase in pressure loss in the work area. Therefore, one object of this study was to determine the optimum field spin steps in which the increase of the hydraulic resistance, at least not more than the heat transfer coefficient values increase. In the experiments, direct measurements were determined by the pressure loss and mass flow rate of the coolant for various $\hat{H}$. Figure 3 shows the dependence of the mass flow of coolant pressure loss for different twist wire steps with $\hat{H} = 0.44$.

![Figure 3. Pressure drops - mass flow rate of the coolant relationship (p = 3.0 \cdot 10^6 Pa, T_m \approx 100^\circ C, \hat{H} = 0.44): 1 – smooth annular channel; 2 – annular channel with ribs; 3 - t = 100 \cdot 10^{-3} \text{ m}; 4 - t = 60 \cdot 10^{-3} \text{ m}; 5 - t = 40 \cdot 10^{-3} \text{ m}]

According to the data presented in Figure 3 can draw the following conclusions:

- presence of the ribs without wire winding has practically no effect on the hydraulic channel resistance;
- a decrease in step spin (spin rate) pressure loss increases sharply (Figure 4).

![Figure 4. Pressure drops - the intensifier swirl pitch t relationship (p = 3.0 \cdot 10^6 \, \text{Pa}, T_m \approx 100^\circ \text{C}, G=0.30 \, \text{kg/c}): 1 - \dot{H}= 0.23, 2 - \dot{H}= 0.35, 3 - \dot{H}= 0.44]

From analysis of Figure 4 shows that there is a bundle of data on the hydraulic losses at the height of the ribs.

On Figure 5 shows the hydraulic drag coefficient on the Reynolds number for different steps twist when \( \dot{H} = 0.23 \). Hydraulic resistance coefficient \( \zeta \) is calculated as follows:

\[
\zeta = \frac{2 \Delta p d_g S^2 \rho}{G^2 l},
\]

where \( \Delta p \) - pressure loss in the work area; \( d_g = 4S/P \) - hydraulic diameter of the annular channel with an intensifier; \( P \) - wetted perimeter; \( S \) - area of the flow cross section of the annular channel; \( \rho \) - density of the coolant; \( G \) - mass flow rate of the coolant; \( l \) - the length of the working area.

The Reynolds number is calculated by the relation:

\[
\text{Re} = \frac{G d_g}{\mu S},
\]

where \( \mu \) - dynamic viscosity coefficient of the coolant.
Figure 5. The pressure drop coefficient - Reynolds number relationship ($\dot{H} = 0.23$): 1 – $t = 40 \cdot 10^{-3}$ m; 2 – $t = 50 \cdot 10^{-3}$ m; 3 – $t = 60 \cdot 10^{-3}$ m; 4 – $t = 100 \cdot 10^{-3}$ m; 5 – smooth annular channel (calculation).

For comparison, the graph shows the calculation of $\xi$ (Re) using a modified Filonenko equation for a smooth annular channel:

$$\xi = \frac{1}{(1,82 \lg Re - 1,64 - 0,19 (d_1 / d_2)^{0.25})^2}.$$

Presented graph suggests a very tangible effect of geometrical parameters intensifier by a factor of hydraulic resistance. This is confirmed by data on the relation of the coefficient of hydraulic resistance in the parameter $\dot{H}$ obtained earlier [1].

The influence of parameters intensifier ($\dot{H}$ and $t$) on the coefficient of hydraulic resistance is particularly evident in the direct dependence of $\xi$ ($\dot{H}$) for different steps twist $t$ and $\xi$ ($t$) for different dimensionless ribs heights $\dot{H}$. The dependence of the coefficient of hydraulic resistance by twisting step is shown in Figure 6 (for $Re = 60000$) at different $\dot{H}$.

Figure 6. The pressure drop coefficient - the intensifier swirl pitch $t$ relationship (Re = 60000): 1 - $\dot{H} = 0.23$, 2 - $\dot{H} = 0.35$, 3 - $\dot{H} = 0.44$

From Figure 6 that bundle dependence $\xi$ ($t$) for $\dot{H}$ ranges - (8 - 10)%. 
Figure 7 shows the dependence of $\xi / \xi_0 = f (H)$ at various steps twisting wire, $\xi_0$ - hydraulic resistance coefficient of fluid flow in a smooth annular channel, is calculated using the equation (1).

The experimental data presented in Figure 7 allow the following conclusions:

- the value of the hydraulic resistance coefficient significantly depends on the pitch of twist. With increasing $t$ hydraulic resistance coefficient decreases;
- dependence of the hydraulic drag coefficient on the dimensionless parameter $H$ for all steps twisted wire is nonmonotonic and is close to $H = 0.23$ there is a maximum hydraulic losses. Such dependence $\xi (H)$ is retained over the entire range of Reynolds numbers.
- maximum $\xi / \xi_0$ value for different steps spins at a Reynolds number $Re = 60000$ in the range from 1.3 to 2.2.

5. Heat transfer coefficient

The aims of the carried out research systematic study are:

- the impact of the step twist $t$ (twist coefficient $k$) in the heat transfer coefficient at various flow characteristic of the interacting flows determined by the dimensionless parameter $H$;
- determining the value $H$, corresponding to the maximum heat transfer for different steps $t$.

Methods of measurement and processing of the experimental data for the study of heat transfer in detail presented in [2].

The modified Isachenko-Galin equation for a smooth annular channel was using for processing the experimental data.

$$Nu = 0.017 Re^{0.8} Pr_l^{0.4} \left( \frac{Pr_l}{Pr_w} \right)^{0.25} \left( \frac{d_2}{d_1} \right)^{0.18}.$$ 

The measurement results are presented in the form of $Nu/A = f(Re)$, where

$$A = Pr_l^{0.4} \left( \frac{Pr_l}{Pr_w} \right)^{0.25} \left( \frac{d_2}{d_1} \right)^{0.18}.$$

The parameter A takes into account the change in the coolant properties at different temperatures, as well as the geometric dimensions of the annular channel.

The relationship $Nu/A = f(Re)$ for different swirling steps of intensificator $t$ is illustrated in Figure 8.
As it follows from Figure 8 steps to a decrease spin (spin rate increase) significantly increases heat transfer.

Influence of geometrical parameters on heat transfer intensifier in the form of dependences \( \frac{Nu}{A} = f(\dot{H}) \) is shown in Figure 9.

As appears from analysis Figure 9 dependence \( \frac{Nu}{A} = f(\dot{H}) \) is nonmonotonic and has a maximum value at \( \dot{H} = 0.35 \). The same value was obtained in the previous study, in which the studies were conducted only for the spin step \( t = 50 \cdot 10^{-3} \) m.

6. Conclusions

The studies of heat transfer and hydrodynamics in the annular channel with heat transfer intensifiers. Operating parameters: mass velocity \( \rho_w = 1000 \div 4000 \text{ kg/(m}^2\text{ c)} \), a pressure \( p = (3.0 \div 7.0) \cdot 10^6 \text{ Pa} \), the coolant temperature at the entrance of the operating portion \( T = 100^\circ \text{C} \), the heat flux \( q = (0.23 \div 0.70) \cdot 10^6 \text{ W/m}^2 \).

A study of the effectiveness of the method of interacting streams in order to intensify the heat exchange on the convex heating surface. It was found that the coefficient of heat transfer and hydraulic resistance depends essentially on the geometric characteristics of the intensifier: \( \dot{H} \) parameter that determines the intensity of the axial and swirling flows and step spin intensifier \( t \). It was
established experimentally that for different steps twist intensifier for maximum hydraulic resistance is observed at $\dot{H} = 0.23$, and on heat transfer - with $\dot{H} = 0.35$.

This work was supported by grant 16-19-10457 RNF.

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