Experimental investigation of hydrodynamics and heat exchange in the ring channel with heat exchangers in the modes of single-phase convection and bubble boiling

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Abstract. The effectiveness of the heat exchange intensifier “rib-twisted wire” is considered in this paper. The main goal is to study the influence of the wire coiling step \( t \) on heat transfer and hydraulic resistance for different values \( \hat{H} \) of the dimensionless height of the edge \( H \), as well as some results on heat exchange during bubbly boiling in an annular channel. Show:

• a brief description and an image of the heat exchange intensifier “rib-twisted wire”;
• generalized results of studies of heat exchange and hydraulic resistance in the annular channel in the single-phase convection with different geometric characteristics of the intensifier;
• empirical correlations of the generalized experimental results that allow to calculating the coefficient of hydraulic resistance and heat transfer in the range of regime parameters in the single-phase convection that is being studied.
• some results of experiments in bubbly boiling regimes and near-critical thermal loads.

1.Introduction

Today various methods for heat exchange intensification have been proposed and are used effectively [1, 2]. However, the application of these methods to a convex heated surface does not lead to a noticeable increase in heat transfer, and sometimes a negative result is observed. The work carried out earlier by the team of authors [3, 4] showed the effectiveness of using the method of heat exchange intensification on a convex heated surface founded the interaction of swirling and axial (transit) flows [5]. The point of the method is the organization of two interacting streams with the help of the “rib-twisted wire” intensifier in the annular channel. Analysis of extensive experimental data [3, 4] showed that the use of intensifiers of this type provides a significant increase in the heat transfer coefficient in comparison with a smooth annular channel with an acceptable increase in hydraulic resistance. The maximum value of the heat transfer coefficient was obtained with the value of the dimensionless height of the edge \( \hat{H} = 0.35 \), and the maximum of the hydraulic resistance coefficient was obtained with \( \hat{H} = 0.23 \). Since all the experiments were carried out at the same step of the intensifier wire coiling \( t = 50 \) mm, it seemed appropriate to expand the studies in which the value of the coiling step would change for different values \( H \) of the dimensionless height of the edge \( H \).
2. Description of the intensifier “rib-twisted wire”

The research cycle was carried out at the "TVS MEI" test-bed, operating in the range of technological parameters corresponding to the VVER-1000 reactor. Its description and main technical characteristics are presented in [6] and were discussed at conferences [7].

Fig. 1 shows the image of the used intensifier. Longitudinal ribs (2) were attached to the outer surface of the heating element (1), on which a twisting device (wire coiling) was mounted (3). All the elements of the intensifier were made of steel grade 08X18H10T, and their fastening was carried out with laser welding. The intensifiers were installed to provide the clearance of 0.2 mm between the unheated concave surface and the wire coiling, conditioned by the structural features of the installation and the convenience of mounting.

![Fig. 1. Elements of the intensifier and their location on the working area: 1 – heating element, 2 – longitudinal rib, 3 – wire coiling.](image)

The detailed description of the working area design and the intensifier is presented in [4].

The main geometric characteristics of the intensifier are the height of the longitudinal edge $h$, the diameter of the twisted wire $d$ and the coiling step $t$. For the convenience of generalization of the experimental data, the dimensionless height of the edge $H$ and the twist coefficient $k$ related to the step $t$ are introduced:

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

where $d_1 = 12.9$ mm is the internal diameter of the annular channel (outer diameter of the heating element), $d_2 = 16.3$ mm is the outer diameter of the annular channel and $d$ is the diameter of the median line of the swirling flow, determined by the ratio:

$$\tilde{d} = \frac{d_2 + d_1}{2} + h.$$

Table 1 shows the values of the geometric characteristics of the intensifiers used. The “+” sign in the table indicates that this step was used for the represented value of $H$.

| Parameter | Value |
|-----------|-------|
| $h$, mm   | 1.00  0.90  0.75  0.60  0.50  0.40 |
| $d$, mm   | 0.50  0.60  0.75  0.90  1.00  1.10 |
| $H$       | 0.59  0.53  0.44  0.35  0.29  0.23 |
3. Effect of an intensifier on heat exchange and hydrodynamics in the single-phase convection

To calculate the heat transfer coefficient in a smooth annular channel, the modified Isachenko-Galin formula is used:

\[ Nu = 0.017 \, Re^{0.8} \, Pr^{0.4} \left( \frac{Pr_{\infty}}{Pr_{cm}} \right)^{0.25} \left( \frac{d_2}{d_1} \right)^{0.18} . \]

The measurement results are presented as \( Nu/A = f(Re) \), where \( A = Pr_{\infty}^{0.4} \left( \frac{Pr_{\infty}}{Pr_{cm}} \right)^{0.25} \left( \frac{d_2}{d_1} \right)^{0.18} \).

Parameter \( A \) considers the properties change (kinematic viscosity and coefficient of thermal diffusivity) at various temperatures of the coolant in the working area, as well as the geometric dimensions of the annular channel.

The influence of the geometric parameters of the intensifier on the heat transfer in the form of the dependences \( Nu/A = f(Re) \) and \( Nu/A = f(H) \) is shown on Fig. 2.

![Fig. 2. The dependence of \( Nu/A \) on the parameter \( H \) (Re = 45000):](image)

- \( 1 - t = 40 \, \text{mm} \).
- \( 2 - t = 50 \, \text{mm} \).
- \( 3 - t = 60 \, \text{mm} \).
- \( 4 - t = 100 \, \text{mm} \).
- \( 5 - \) smooth annular channel (experiment).

As follows from the analysis of Fig. 2 the dependence \( Nu/A = f(H) \) is not monotonic and has a maximum value at \( H = 0.35 \) for all values of the coiling step \( t \). The same value was obtained in [4], in which the research was carried out only for a coiling step \( t = 50 \, \text{mm} \).

One of the conditions for the efficiency of any heat exchange intensifier is exceedance the heat transfer coefficient against the hydraulic resistance coefficient. For this purpose, studies of hydrodynamic resistance were carried out in the entire range of \( H \) and \( t \). During the experiment, measurements of pressure losses, mass flow and temperatures at the inlet and outlet of the working area were made.

Fig. 3 shows the dependence of the coefficient of hydraulic resistance on the dimensionless parameter \( \hat{H} \) for all steps of wire coiling. This dependence is not monotonic, and the maximum \( \varphi/\varphi_0 \) is observed at \( \hat{H} = 0.23 \) for all values of \( t \).
4. Calculated ratio

The obtained array of experimental data allows getting equations for hydraulic resistance and coefficient with the given type of intensifier.

The structure of the equations has the following form:

\[
\frac{\xi}{\xi_0} = 1 + C_1 k^{m_1} \left( \frac{1}{H^k} \right)^{n_1}, \quad \frac{Nu}{Nu_0} = 1 + C_2 k^{m_2} \left( \frac{1}{H^k} \right)^{n_2}
\]

As a result of the experimental data processing, the following equations were obtained:

\[
\frac{Nu}{Nu_0} = 1 + 0.37 k^{0.18} \left( \frac{1}{H^k} \right)^{0.43} \tag{1}
\]

\[
\frac{\xi}{\xi_0} = 1 + 0.54 k^{0.55} \left( \frac{1}{H^k} \right)^{0.31} \tag{2}
\]

Here \(Nu_0\) and \(\xi_0\) are the Nusselt number and the hydraulic resistance coefficient of the smooth annular channel, respectively. The values of the exponents \(m_1, m_2\) and the constants \(C_1, C_2\) in the equations are obtained by averaging the entire array of experimental data, and the deviations of the experimental data from the results of calculations on the obtained correlations do not exceed the experimental data error estimated by the authors at 12%.

5. Results of bubbly boiling experiments

To review the effectiveness of the studied intensifier in the bubbly boiling, experiments were carried out using the technique, described below:

1. A preselected working pressure was set in the circuit. The circulation pump was switched on, and a certain consumption rate of the coolant was selected.
2. At the selected pressure, using the heaters located in the circuit, the value of the water inlet temperature in the working area was set equal to the saturation temperature.
3. In the absence of a load on the working area heater, the indications of the wall thermocouples were recorded in order to test them and to measure the thermal losses in the working area.
4. The load was applied to the main heater of the working area, and the bubbly boiling regime was reached by a gradual increase in the power of the heater. The indications of thermocouples located on the internal adiabatic surface of the heater tube, the temperature of the water at the inlet and outlet of the working area, the pressure in the circuit, the pressure losses in the working area, the current strength and the voltage on the heater were recorded. In the process of entering the new mode, all parameters were recorded by ASNII. The start of bubbly boiling was characterized by a sudden change in the consumption rate of the coolant and pressure losses in the working area, which happened due to the formation of the vapor phase and additional pressure losses to change the velocity of the flow.
Fig. 4 shows the distribution of the temperature field along the length of the working area at a pressure $p = 3.0$ MPa for the intensifier with parameters $\dot{H} = 0.35$ and $t = 50$ mm. Fig. 5 shows the distribution of the liquid temperature in the annular channel at a constant power of the main heater for different consumption rates.

![Fig. 4: Distribution of temperatures of the wall (1) and the liquid (2, 3, 4) along the length of the working area ($t = 50$ mm, $\dot{H} = 0.35$ and $G = 0.25$ kg/s): 1 – $N_{el} = 55$ kW; 2 – $N_{el} = 55$ kW; 3 – $N_{el} = 50$ kW; 4 – $N_{el} = 40$ kW.]

![Fig. 5: Distribution of coolant temperature along the length of the working area ($N_{el} = 27$ kW): 1 – $G = 0.189$ kg/s; 2 – $G = 0.235$ kg/s.]

The processing of the experimental results enabled calculating the heat transfer coefficient at various regime parameters. The experimental values of the heat transfer coefficient under bubbly boiling conditions were $\alpha_{exp} = (110\div180)$ kW / (m$^2$·K). The boiling component $\alpha_{boil}$ was isolated using the interpolation formula of S.S. Kutateladze:

$$\alpha_{exp} = \alpha_{con} \sqrt{1 + \left(\frac{\alpha_{boil}}{\alpha_{con}}\right)^2} = \alpha_{con}^2 + \alpha_{boil}^2.$$  

Values $\alpha_{con}$ were selected from the experimental data obtained earlier.

Comparison of experimental values of $\alpha_{boil}$ with known correlations of Borishanskiy V.M., Yagova V.V. and Rassokhina N.G. has showed a noticeable increase in $\alpha_{boil}$ due to the introduction of an intensifier.

The near-critical and critical regimes that determine the boiling crisis appeared at input capacities of more than 50 kW and mass expenditures ($0.20 \div 0.35$) kg/s. The values of the critical thermal loads obtained in 3 working areas are experimentally established. The heat exchange crisis occurred approximately in the same zone along the channel path so quickly that the protection system did not have time to prevent its destruction. A typical example of the destroyed working area is shown in Fig. 6.
At the same time, the values of critical heat flux density turn out to be equal to $q_{cr1} = 1.86 \text{ MW/m}^2$ at $\rho w = 3079 \text{ kg/(m}^2\text{s})$; $q_{cr2} = 1.97 \text{ MW/m}^2$ at $\rho w = 3720 \text{ kg/(m}^2\text{s})$; and $q_{cr3} = 2.23 \text{ MW/m}^2$ at $\rho w = 4490 \text{ kg/(m}^2\text{s})$. The geometry of the annular channel with intensifier is such that, for a coolant flow with an inlet temperature close to saturation temperature and significant thermal loads, all heat exchange regimes apparently take place: convective, local and developed bubbly boiling, transient boiling, and heat exchange crisis. Calculations carried out using the known correlations [9] have showed that the experimental values obtained by the authors turn out to be lower than the calculated ones. Work in this direction continues.

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