Experimental efficiency evaluation of the of various geometry twisted bands for heat transfer intensification in the heat-exchange channels of a nuclear power unit equipment

A E Sobornov, S M Dmitriev, R R Ryazapov, A V Mamaev and A V Kotin
NNSTU n.a. R.E. Alekseev, Minin Street, 24, Nizhny Novgorod, 603155, Russia

e-mail: wisestjedi@mail.ru

Abstract. The paper is devoted to experimental study of heat exchange and pressure drop of channel with various geometry twisted bands. The studies were carried out in the range of operating modes parameters of the nuclear power units’ heat exchange equipment. The three different designs of intensifiers are presented in the paper. The values of heat transfer coefficient and pressure drop are obtained. The dependences of the Nusselt number (Nu) on the Reynolds number (Re) were calculated. The comparative analysis of intensifiers is made. The efficiency factor was also calculated on experimental data. The most optimal geometry form of intensifier was selected.

1. Introduction
One of the topical tasks of modern nuclear power engineering is to increase the energy efficiency of NPU equipment. The most rational way to solve this problem is the heat and mass transfer processes intensification because it allowing to avoid a significant increase in the mass and size characteristics of heat exchange equipment. [1]. In terms of capital costs, passive intensification methods are preferred. The main action of such methods is aimed at turbulence increasing or destruction of the boundary layer, as well as at the modernization of the heat exchange surface, which causes a significant increase of the local heat transfer coefficient.

Among of all passive heat transfer intensifying methods in the channels of a nuclear power units, one should single - using of inner twisted metallic bands. The main advantages of this method are the lower cost of intensifiers manufacture and installation and a lower channel pressure drop. It is important to note that the efficiency of this type of intensifiers using are largely depends of temperature gradient between the moving coolants.

At nominal flow rates, the temperature gradient between coolants remains constant along the length of the channel, which provides maximum thermal load. The non-optimal flow rates ratio leads to decreasing of the temperature difference between coolants. The temperature fields of the coolants at different operation modes of the countercurrent heat exchanger are presented on figure 1. Line without points for \( t_1(z) \), and line with points for \( t_2(z) \). Temperature distributions were calculated using the formulas from [2].

In modes with a non-optimal ratio flows relation, the calculated heat flow from the coolant side is reduced by 45-50%. The using of constant twist step intensifiers along the entire length of the channel
under these conditions is irrational, because it leads to a significant increase in the heat transfer coefficient only in areas with a significant temperature gradient. Authors consider that using of the intensifiers with a variable twist step at the inlet (outlet) section are allows equalizing the temperature gradient and reducing the dependence of the heat transfer intensification efficiency on temperature field.

Figure 1. The temperature fields of the coolants at different operation modes of the countercurrent heat exchanger

A rapid change in the boundary layer structure in these areas also leads to an increase of channel pressure drop [3]. Thus, the application of this method of heat transfer intensification is impossible without a comprehensive analysis of the hydrodynamic and heat transfer processes in the coolant flow.

Within the framework of this work to solve the complex problem of choosing the twisted band optimal step combination experimental studies of thermal hydraulics processes in a viscous fluid flow in channels were carried out. The studies were carried out in the range of operating modes parameters of the nuclear power units’ heat exchange equipment. The main objectives of this work were:

- the selection of the optimal twisted band step combination of the heat transfer intensifier on the base of the effectiveness assessment results;
- creation of an experimental data for the validation and verification of numerical methods designed to calculate the flow characteristics during fluid movement in curvilinear channels.

2. Experimental branch and technique

Experimental studies of the heat transfer intensification were carried out on the FT-80 testing branch, which created to study the hydrodynamic and heat transfer processes in steam generating systems, as well as overturning circulation processes. The test branch includes three hydraulic circuits. The first and second circuits are hydraulically closed and using under excess pressure, in the range from 10 to 15 MPa, the third circuit is hydraulically open and serves to cool the main equipment.

The general view of the experimental model is shown in figure 2a, 2b. The model is a heat exchanger channel made according to the "pipe in pipe" principle. The case of the model consists of two parts, which are after welding the longitudinal welds form a circular cross-section channel with an inner diameter of 17 mm. A heat exchange tube with an outer diameter of 13 mm and a wall thickness of 1.5 mm, made of PT7M titanium alloy, is placed along the channel axis (Figure 2c).

In the model, the countercurrent flow of coolants is organized. The secondary circuit coolant moves upwards along the inner tube and it is heated by the primary circuit coolant that moves top down in the annular gap. This coolants movement scheme is the most optimal because becomes possible to achieve...
the maximum value of the temperature gradient with minimum impact of thermal driving head forces on the heat transfer process and movement of coolants.

Figure 2. The experimental equipment
a) experimental model;
b) section of model;
c) twisted bands with different step combination
The heat exchange surface active part length is 2440 mm. The channel total length, in which thermocouples is installed, is 2280 mm. The temperature of the coolants and the heat exchange surface is measured in 11 sections along the height of the model. Each section contains one thermocouple for primary coolant flow, and two thermocouples at different depths (0.7 and 1.3 mm) of heat exchange surface to determine the heat flux. In 7 out of 11 sections, thermocouples are installed to measure the temperature of the secondary coolant. All thermocouples are made of Chromel/Alumel alloy (Type K). The diameter of thermocouples sensitive part is 0.5 mm, the measurement error is 0.2 °C.

The intensifier is a strip made of 12Cr18Ni10Ti stainless steel 1 mm thick and 9.8 mm wide, twisted around the central axis (Figure 2b). The heat exchange intensifier length is equal to the length of the experimental model active part. There is study of several intensifiers within of this work:

- with decreasing step s/d = 8-6-4;
- with alternating step s/d = 4-6-8-6-4 and s/d = 8-6-4-6-8.

The length of constant swirl step fragments was determined on the basis of experimental and computational data, which obtained as a result of heat transfer studies in channels with constant step twisted intensifiers [4].

Experimental studies were carried out with the following parameters of coolants:

- a pressure of primary coolant - 11 MPa, secondary coolant - 5 MPa;
- the primary coolant highest temperature is in the range 250 ÷ 260 °C;
- the secondary coolant temperature at inlet is in the range 40 ÷ 50 °C;
- the mass flux of the primary coolant is in the range of 130 ÷ 540 kg/(m²·s);
- the mass flux of the secondary coolant is in the range of 150 ÷ 460 kg/(m²·s).

The secondary coolant is moved like laminar flow with macro eddies.

The experimental technique assumed the following actions:

- creation a predetermined ratio of coolant flow rates by using the control and monitoring system;
- waiting some time for transient processes termination and thermal parameters stabilization;
- recording temperature field, pressure fields, coolant flow rates to data files.

Registration and processing of sensor signals were carried out using a self-developed software, based on the Agilent 34980a unit.

An important part of the experimental study was to calculate heat loss. The maximum value of heat losses does not exceed 1%. Such a small heat losses value allows excluding them from further analysis.

3. Study results

The determination the dependency of the heat transfer intensity on the secondary coolant flow rate is based on experimental data. First, the using a various twist step intensifier in the secondary coolant channel made it possible to increase the heat load of the heat exchanger. A dependence of the heat load on the coolant flow rates ratio is shown in Figure 3. For comparison, the same graph shows the dependence for a constant twist intensifier with 40 mm step [4].
The selected ratios of the constant step sections lengths made it possible to increase the heat flux in the entire range of flow rates. It has been found that at low values of the temperature gradient, the intensifier $s/d = 8-6-4-6-8$ is the most effective in heat transfer intensification.

The heat transfer efficiency evaluation is carried out according to the averaged values of the ratio of convective to conductive heat transfer. The dependences of the Nusselt number ($Nu$) on the Reynolds number ($Re$) is calculated for secondary coolant for various intensifiers (Figure 4).

It is possible to draw a conclusion about the greatest heat transfer intensification by a $s/d = 8-6-4-6-8$ intensifier in the entire range of coolant flow rates. The high values of the heat transfer coefficient are explained, firstly, by the presence of a large part with $s/d = 4$, and secondly, by a secondary flows high turbulence caused by the high velocity field gradient at the sections where twisting step changed.

To do a more correct comparison of the intensifiers effectiveness it should be also compared channel pressure drop, due to the complication of the geometric shape. To solve this problem, an experimental
study of channel pressure drop with constant thermal properties was carried out. The study results are shown in Figure 5. The pressure drop study was carried out at atmospheric pressure in the same range of the coolant velocity. In this case, it became possible to use thermocouple fittings for pressure sensors connecting. As a result, significantly increased the accuracy of the obtained values of the loss coefficient. This also explains the Reynolds numbers interval difference on the figures 4 and 5.

![Figure 5. The dependence of loss coefficient on Reynolds number](image)

The greatest pressure drop value is possessed by a channel with an intensifier with s/d = 8-6-4-6-8. Thus, the using of an intensifier with s/d = 8-6-4-6-8 in tube channel, on the one hand, are provides the highest value of the heat transfer coefficient, and, on the other hand, are characterized by the greatest value of pressure drop. This ambiguity makes it difficult to choose the most optimal intensifier geometry and compels to do a comparative analysis of intensifiers efficiency.

4. The effectiveness evaluation

Comparative analysis of intensifiers is made with technique from [1]. The method is based on a comparison of heat transfer intensity with and without an intensifier at equal power need for coolant moving. In this case, the efficiency factor consists of three components:

\[ k_{eff} = k_F \cdot k_N \cdot k_Q \] (1)

where \( k_F \) - ratio of heat exchange surfaces areas with and without an intensifier, \( k_N \) - ratio of hydraulic power, \( k_Q \) - ratio of thermal load. The calculated values of efficiency ratio at the optimal ratio of coolant flow rates and the ratio equal to 4 are shown on figure 6a and 6b. The analysis showed that the using of various twisted intensifiers makes it possible to increase the efficiency of heat exchange equipment in the entire range of flow rates. The channel with an intensifier s / d = 8-6-4-6-8 is characterized by the highest efficiency factor.
5. Conclusion
Analysis of the results of experimental studies of hydrodynamics and heat transfer processes in a viscous fluid flow in channels with various twisted intensifiers allowed us to draw the following conclusions:

- the using of various twisted intensifiers makes it possible to increase the efficiency of heat exchange equipment in the entire range of flow rates;
- the efficiency factor, when using of intensifiers with a various twist step, is higher than with a constant twist step. This effect is most pronounced in operating modes with a non-optimal coolant’s flows ratio;
- under the operating conditions of the existing heat exchange equipment, the intensifier with s/d = 8-6-4-6-8 has the most optimal geometry form.

Acknowledgments
Study is made with the financial support from the Ministry of Education and Science of the Russian Federation within the national project "Science and Universities" initiated for the youth research labs creation (scientific topic: "Hydrodynamic and heat and mass transfer processes in nuclear power plants equipment")

References
[1] Budov V M, Dmitriev S M 1989 High heat load exchangers for nuclear power plants (Moscow: Energoatomizdat) p 174
[2] Kern D, Kraus A 1977 Heat transfer of complex surfaces (Moscow: Energiya) p 464
[3] Shchukin V K 1980 Heat transfer and hydrodynamics of internal flows in the mass forces fields (Moscow: Mechanical engineering) p 240
[4] Sobornov A E 2012 The 9th International Topical Meeting on Nuclear Thermal-Hydraulics, Operation and Safety Kaohsiung (Taiwan: NUTHOS-9) p 13