Heat transfer of coolants in channels with swirling and continuous wall roughness

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Abstract. The work presents the results of experimental study of heat transfer in rough tubes, including tubes with twisted tape insert, at water and R134A one-phase flow. The roughness of tube inner surface is obtained by thread cutting. The combined influence of turbulizing factors on the heat transfer intensity is analyzed. Generalizing dependences are obtained for heat transfer calculation.

1. Introduction
The research of combined influence of two types of intensifiers is important to enhance processes in the tubes and channels of various heat exchange equipment. The use of swirling in the channels with rough surfaces for the boiling flows allows significantly reducing overall dimensions of the evaporator since the emerging mass forces contribute to the fluid motion to the surface with the developed evaporation centers. The extensive review of heat transfer and hydraulic resistance in channels with rough surfaces is presented in work [1]. It is noted that at present, the number of papers considering hydrodynamics and heat transfer of one and two-phase flows at the combined influence of swirling and artificial roughness of the channel surface is limited.

2. Experimental research of heat transfer
2.1. Heat transfer in straight rough channels at one-phase flow
The research of heat transfer in rough tubes was performed using a water test bench and a test bench with refrigerant R134A. The detailed description of these test benches is presented in [2, 3]. Heat was supplied by electrocontact (ohmic) method, i.e. by passing electric current directly through the tube wall. Distribution of wall temperatures on the tube surface was defined by means of 28 thermocouples arranged on the outer surface of the tube. The roughness of the inner surface of the tube was obtained by thread cutting with pitches of the thread \( t = 0.3 \ldots 1.0 \) mm and average height of projection \( \Delta = 0.07 \ldots 0.65 \) mm (Table 1). The inner tube diameter \( d \) (about 10 mm) was defined on the roughness cavity.

Studies of heat transfer for distilled water flow were performed at Reynolds number \( \text{Re}=6000 \ldots 160000 \) and heat flux \( q=30 \ldots 190 \) kW/m\(^2\), and for refrigerant R134A flow – at \( \text{Re}=26000 \ldots 100000, q=8 \ldots 20 \) kW/m\(^2\).
Table 1. Parameters of rough surfaces.

| Profile 1 | Profile 2 | Profile 3 | Profile 4 |
|-----------|-----------|-----------|-----------|
| $t=0.3$ mm | $t=0.4$ mm | $t=0.5$ mm | $t=0.6$ mm |
| $\Delta=0.07$ mm | $\Delta=0.07$ mm | $\Delta=0.14$ mm | $\Delta=0.14$ mm |
| $\Delta/d =0.007$ | $\Delta/d =0.007$ | $\Delta/d =0.014$ | $\Delta/d =0.014$ |

| Profile 5 | Profile 6 | Profile 7 | Profile 8 |
|-----------|-----------|-----------|-----------|
| $t=0.5$ mm | $t=0.75$ mm | $t=1.00$ mm | $t=1.25$ mm |
| $\Delta=0.25$ mm | $\Delta=0.35$ mm | $\Delta=0.45$ mm | $\Delta=0.65$ mm |
| $\Delta/d =0.025$ | $\Delta/d =0.035$ | $\Delta/d =0.045$ | $\Delta/d =0.065$ |

Typical curves of changes of heat transfer coefficient $\alpha$ for one-phase flows of water and refrigerant R134A depending on relative length $x/d$ are shown in Figure 1 a, b.

Figure 2 shows the experimental data on heat transfer in straight rough channels with different depth of cutting for the distilled water flow. As can be seen an increase of roughness height $\Delta$ and increment of Reynolds number Re leads to an increase of heat transfer. The values of the Nusselt number $Nu$ for rough tubes with profiles 1 and 2 practically coincide with the Nusselt number for a smooth tube since small roughness peaks ($\Delta=0.07$ mm) disturb the flow only slightly.

Figure 1. Local values of heat transfer coefficient $\alpha$ in a straight rough tube with $\Delta=0.45$ mm and $t=1.0$ mm for distilled water (a) and refrigerant R134a (b)
A significant increase of heat transfer (by 38…49%) occurs even in channels with Δ=0.14 mm. The coincidence of the heat transfer level for tubes with profile 3 and profile 4, which have the same cutting depth but different pitch, indicates that the frequency density of the protrusions location has no influence on heat transfer.

Figure 3 presents the comparison of Nusselt numbers for the straight rough tube with Δ=0.35 mm and d=10 mm with Nusselt numbers, calculated by the formulas for heat transfer in rough tubes, proposed by Hausen [5], Yaglom, Kader [6, 7] and Galin [8]. As it can be seen from the graph the heat transfer values calculated by the formula of Hausen [5], Yaglom and Kader [6, 7] as well as the experimental data obtained by Isachenko, Agobabov, Galin [4] are slightly overestimated. However, there is a coincidence of the heat transfer level with the Galin’s data [8] at Reynolds numbers Re=10000…40000.

Calculations of heat transfer by other dependences presented in [1] give significant divergences. This once again confirms that there is no universal approach for taking the roughness into account.

To calculate the heat transfer of the tubes with continuous artificial roughness the generalizing dependence of the following form was obtained

\[
\frac{Nu}{Nu_0} = 1 + 2.8 \cdot (\Delta/d)^{0.3},
\]

where for straight smooth tube

\[
Nu_0 = 0.021 \cdot Re^{0.8} Pr_f^{0.43} (Pr_f/Pr_w)^{0.25}.
\]

Pr_f and Pr_w are the Prandtl numbers at flow temperature and wall temperature.
2.2. Heat transfer in rough channels with flow swirling

Flow swirling in rough tubes was organized by twisted tape inserts [9, 10] with relative pitch of twisting \( s/d = 2.5 \ldots 6 \) at a turn by 180°. The thickness of tapes \( \delta \) was 0.8 mm.

Heat transfer for the distilled water flow in the tubes with twisted tape and continuous artificial roughness in the form of a metric thread with a complete cutting profile was studied at Reynolds numbers calculated on the basis of the roughness cavity diameter from 5000 to 80000 and heat flux \( q = 50 \ldots 250 \text{ kW/m}^2 \). The Reynolds numbers for a metric thread with a complete cutting profile for the refrigerant R134A flow were in the range from 10000 to 50000, and the heat flux \( q = 9 \ldots 21 \text{ kW/m}^2 \).

Analysis of the experimental data has shown that the flow swirling leads to heat transfer enhancement, and its intensity increases with an increasing degree of the tape twisting \( s/d \) (s – pitch of twist). The heat transfer enhancement in the rough tube with flow swirling for \( \Delta = 0.25 \text{ mm} \) is 133\ldots173\%, for \( \Delta = 0.35 \text{ mm} \) it is 142\ldots183\%, for \( \Delta = 0.45 \text{ mm} \) it is 152\ldots192\%, and for \( \Delta = 0.65 \text{ mm} \) it is 163\ldots202\% relative to the smooth straight tube.

Figure 4 shows experimental dependences of heat transfer in rough channels with flow swirling for a fixed degree of twisting and various cutting depth of the metric thread.

Heat transfer enhancement in the rough tube with flow swirling relative to heat transfer in the smooth tube with flow swirling calculated according to Manglik and Bergles [9] formula for \( \Delta = 0.25 \text{ mm} \) is about 60\%, for \( \Delta = 0.35 \text{ mm} \) it is 70\%, for \( \Delta = 0.45 \text{ mm} \) – 80\%, and for \( \Delta = 0.65 \text{ mm} \) – 90\%.

The experimental study has shown that the heat transfer depends most strongly on the relative height \( \Delta/d \) of roughness peaks and the degree of flow swirling \( (s/d) \) for both coolants.

Generalization of experimental data for rough tubes with flow swirling was carried out for roughness with a complete cutting profile in the form of a triangular thread. The dependence has the form

\[
\frac{Nu}{Nu_s} = 1 + 2 \cdot (\Delta/d)^{0.3},
\]

where Nusselt number for smooth tube with twisted tape insert is according to [9]
3. Conclusions
The study of the distilled water and coolant flows in the tubes with rough walls in the form of the metric thread with complete cutting profile have shown a significant influence of the profile height on the heat transfer intensity while the influence of the thread pitch has not been found.

The twisted tape installation in the channel with rough walls gives an additional increment in heat transfer, and this occurs with an increase of the twisting degree. The generalizing dependences for the tubes with and without the tape are obtained. There the roughness influence in the form of complex ($\Delta/d$) was included in equal degree – 0.3, which proves the absence of the wall-adjacent turbulence suppression by the flow swirling.

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