Heat transfer and structure of flow at boiling of refrigerant R134a in channels with inserts in the form of finned twisted tape

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Abstract. The paper presents the results of experimental study of heat transfer for the refrigerant R134a flow in the channels with finned twisted tape inserts at bubble boiling. The flow regimes implemented under the given conditions are shown. The stable cord-like flows appear at boiling in channels with twisted tape inserts and high vapor content when the liquid phase moves in the form of a stream (cord) along the central part of the tape, which is not an active heat exchange surface. At boiling this can lead to an increase in the length required for complete liquid evaporation. Existing geometric modifications of twisted tapes are used in the heat-exchange equipment at forced convection of the coolant and do not solve the problem of cord-like flows elimination. The present work discusses the experimental study of heat transfer at boiling of refrigerant R134a in the channels with twisted tape inserts that have fins on its surface.

1. Heat transfer and structure of the flow at boiling of refrigerant R134a in the channels with inserts in the form of finned twisted tapes

Heat transfer at boiling of refrigerant R134a in the channels with inserted finned tapes having a relative pitch of twist \(y=s/d=3; 4; 6\), pitch of fin installation \(t=60; 40; 20\) mm, at fixed values of the fin height \(h=1\) mm and installation angle \(\varphi=45^\circ\) (the fins are installed in the opposite direction to the tape twisting) was experimentally studied in the range of the regime parameters: Reynolds number depending on fluid circulation velocity \(Re_0=3\cdot10^4\div6.5\cdot10^4\), heat flow density \(q=99\div326\) kW/m\(^2\), boiling parameter \(Bo=0.00052\div0.0045\), and the estimated mass flow vapor content according to the supplied heat flow at the outlet of the working section reached \(X=0.07\div0.74\). The diameter of the working section \(d=0.01\) m, and the relative tube length \(L/d=50\) [1, 2].

The evaluation of the regime parameters influence on the heat transfer at boiling of the refrigerant R134a in the channel with finned twisted tape inserts has shown that the influence of Reynolds number \(Re_0\), the heat flow density \(q\) and relative pitch of twist \(y=s/d\) in channels with twisted tape inserts is similar to that detected in channels with smooth twisted tapes [1].

With a decrease of the dimensionless pitch the heat transfer increases insignificantly in the area from \(t/s=1\) to \(t/s>1\) and decreases with further reduction of the dimensionless pitch at \(t/s<1\). The decrease in heat transfer at the tightest pitch of the fin installation \(t=20\) mm is related to the change in the flow structure, namely, to the formation of stagnant zones on the channel surface beyond the
contact point of the fin butt end with the channel wall. With a decrease of the Reynolds number below 4.5\times10^4 the formation of a stable dry spot in this area was visually recorded (Figure 1). With further increase in the heat flux density above \( q > 222.9 \text{ kW/m}^2 \) a heat exchange crisis occurred along with the sharp increase of the channel wall temperature followed by the channel destruction.

Figure 1 shows a visually recorded flow regime in the channel with finned twisted tape \( y=4 \), with pitch of the fin installation \( t=20 \text{ mm} \) at the boundary of the heat transfer crisis beginning. Local values of the heat transfer along the working section length at the crisis boundary are shown in Figure 2. The heat exchange crisis is in section No. 5, as indicated by the decrease of the heat transfer coefficient even before the crisis began, with the crisis coming the wall temperature rises sharply by more than 200 °C in this section with subsequent growth in the remaining sections.

The experimental data on heat transfer at boiling of the refrigerant R134a in the channel with finned twisted tape inserts are shown in Figure 3. As it can be seen from the graph, the influence of the pitch of fin installation on the twisted tape surface on the heat transfer at various relative pitches does not qualitatively differ from the conclusions drawn earlier, that in the area from \( t/s=1 \) to \( t/s>1 \) the heat transfer increases and at pitch \( t/s<1 \) the heat transfer decreases to the level of the smooth twisted tapes.

The presence of fins affects the heat transfer at boiling insignificantly but fin installation changes the flow structure. The fin installation on the surface leads to an increase of the vapor amount inside the channel due to the formation of vortex structures behind the fin, in which active vaporization takes place, and also because of the decrease of fluid saturation temperature as a result of the increase of hydraulic resistance. The increase of vapor content with other regime parameters being equal in the

**Figure 1.** Photograph of flow regime in channel with twisted tape \( y=4 \), with pitch of fin installation \( t=20 \text{ mm} \) at: \( Re_0=31459.8; \ q=222.9 \text{ kW/m}^2; \ p=627.9 \text{ kPa}; \ t_{\text{inlet}}=23.6°C; \ X=0.488 \)

**Figure 2.** Local values of heat transfer coefficient at boiling of refrigerant R134a in channel with twisted tape \( y=4 \), with pitch of fin installation \( t=20 \text{ mm} \) at: \( Re_0=31459.8; \ q=222.9 \text{ kW/m}^2; \ p=627.9 \text{ kPa}; \ t_{\text{inlet}}=23.6°C; \ X=0.488 \)

**Figure 3.** Heat transfer at boiling of refrigerant R134a in channel with finned twisted tapes \( y=3; 4; 6 \) with pitches of fin installation \( t=60; 40; 20 \text{ mm} \)

The presence of fins affects the heat transfer at boiling insignificantly but fin installation changes the flow structure. The fin installation on the surface leads to an increase of the vapor amount inside the channel due to the formation of vortex structures behind the fin, in which active vaporization takes place, and also because of the decrease of fluid saturation temperature as a result of the increase of hydraulic resistance. The increase of vapor content with other regime parameters being equal in the
channels with finned twisted tapes leads to an earlier coming of the heat exchange crisis, as compared to channels with smooth twisted tape inserts, due to the appearance of dry spots on the heat-release surface.

As noted above at boiling of refrigerant in the channels with twisted tape insert \( y=4 \) with pitch of fin installation \( t=20 \) mm, the formation of stagnant zones with the decrease of the liquid film thickness and the formation of a stable dry spot behind the contact point of the fin butt end with the channel wall at a decrease of the Reynolds number \( Re_0<4.5\cdot10^4 \) was recorded. In the channel with twisted tape insert \( y=6 \) with pitch of fin installation \( t=20 \) mm, a similar picture of flow regime with the formation of a stable dry spot in this area was observed. With an increase of Reynolds number \( Re_0>4.5\cdot10^4 \) the liquid inleakage with a short-term wetting of the surface by a thin liquid film and the formation of ruptures of the liquid film in the contact zone of the fin butt end and the channel wall occurred. The flow of liquid film on the wall surface in the channel with finned twisted tape \( y=6 \) and \( t=20 \) mm was less uniform than in the channel with the tape \( y=4 \) and \( t=20 \) mm.

The boundary of the beginning of heat transfer crisis in the channel with finned tape with the relative pitch of twist \( y=6 \) and pitch of fin installation \( t=20 \) mm is lower than for \( y=4 \) and \( t=20 \) mm and corresponds to the heat flow density \( q\geq161.8 \) kW/m\(^2\). Local values of heat transfer along the working section length at the heat transfer crisis boundary in the channel with twisted tape \( y=6 \) and pitch of fin installation \( t=20 \) mm are shown in figure 4.

We failed to achieve the critical thermal loads in the channel with finned twisted tape \( y=3 \) with pitch of fin installation \( t=20 \) mm even at the maximum possible power of the transformer.

It was observed that due to the fins installed on the surface the increase of vaporization occurs inside the channel because of the formation of vortex structures behind the fin in which active vaporization takes place. It is noted that with a decrease of the relative pitch of tape twisting a more uniform distribution of the liquid on the heat exchange surfaces occurs due to the increase of mass forces as a result of which the critical heat flows increase.

It is important that in this channel there are neither dry spots in the entire studied range, nor discontinuities of the liquid film on the heat-release surface and in the contact zone of the fin butt end with the channel wall. The liquid distribution on the heat-release surface in the channel with finned tape \( y=3 \) and \( t=20 \) mm is more uniform than with the tape \( y=4 \) and \( t=20 \) mm as indicated by higher heat transfer coefficients. The local values of heat transfer along the working section length at the maximum possible heat flow density \( q=326.5 \) kW/m\(^2\) in the channel with twisted tape \( y=6 \) and pitch of fin installation \( t=20 \) mm are shown in figure 5.

**Figure 4.** Local values of heat transfer coefficient at boiling of refrigerant R134a in channel with twisted tape \( y=6 \), with pitch of fin installation \( t=20 \) mm at: \( Re_0=32462.5 \); \( q=161.8 \) kW/m\(^2\); \( p=618.4 \) kPa; \( t_{inlet}=23.0^\circ \)C; \( X=0.330 \)

**Figure 5.** Local values of heat transfer coefficient at boiling of refrigerant R134a in channel with twisted tape \( y=3 \), with pitch of fin installation \( t=20 \) mm at: \( Re_0=31433.1 \); \( q=326.5 \) kW/m\(^2\); \( p=661.4 \) kPa; \( t_{inlet}=24.3^\circ \)C; \( X=0.740 \)
On the basis of the presented data on the flow regimes in the channels with finned twisted tapes at boiling of refrigerant R134a we can conclude that fin installation affects the heat transfer at boiling insignificantly. The fin installation with the smallest pitch $t=20\text{ mm}$ at $Re_{0}<4.5\cdot10^4$ in channels with relative pitch of twist $y>4$ leads to the formation of stagnant zones that promote the formation of dry spots resulting in an earlier heat transfer crisis than in the channels with smooth twisted tapes. This again confirms the data of the V.I. Subbotin [3] that the maximum effect of twisting is observed when $s/d$ is less than 4.

The increase of the heat transfer coefficient in the channels with finned twisted tape inserts in the studied range of geometric parameters compared to the straight channel at boiling is on average from 1.52 to 1.75 times.

The experimental data generalization allowed obtaining the dependence for calculation of the heat transfer coefficient with finned twisted tape inserts in the form (1):

$$Nu = 10.33 \, Re^{0.7} \, K^{0.2} \, (s/d)^{-0.15} \, Pr^{0.43} \tag{1}$$

Dependence (1) with an error of $\pm22\%$ at a confidence probability of 0.95 generalizes the experimental data, and it is valid for calculation of the heat transfer at boiling of refrigerant R134a in the channels with finned twisted tape inserts with relative pitches of twist $y=s/d=3; 4; 6$, pitch of fin installation $t=60; 40; 20\text{ mm}$, at fixed values of the fin height $h=1\text{ mm}$ and installation angle $\phi=45^\circ$ (the fins are installed in the opposite direction to the tape twisting) in the range of the regime parameters: Reynolds number depending on the fluid circulation velocity $Re_{0}=3\cdot10^4÷6.5\cdot10^4$, heat flux density $q=99÷326\text{ kW/m}^2$, boiling parameter $Bo=0.00052÷0.0045$ and the estimated mass flow vapor content according to the supplied heat flow at the outlet of the working section reached $X=0.07÷0.74$.

The diameter of the working section channel $d=0.01\text{ m}$, and the relative tube length $L/d=50$.

2. Conclusion

It is found that the influence of regime parameters on the heat transfer at boiling of refrigerant R134a in the channel with finned twisted tape inserts is the same as in the channels with smooth twisted tapes. Because of the fins installed on the surface the increase of vaporization occurs inside the channel as a result of the formation of vortex structures behind the fin in which active vaporization takes place. The fin installation affects the heat transfer at boiling insignificantly, however, the fin installation changes the flow structure, increases the amount of vapor by additional generation of vapor inside the channel at the decrease of the liquid saturation temperature as a result of the increases of hydraulic resistance. The increase of vapor content with other regime parameters being equal in the channels with finned twisted tapes leads to an earlier coming of the heat exchange crisis than in channels with smooth twisted tape inserts due to the appearance of dry spots on the heat-release surface. The increase of the heat transfer coefficient in the channels with finned twisted tape inserts in the studied range of geometric parameters compared to the straight channel at boiling is on average from 1.52 to 1.75 times.

It is noted that with a decrease of the relative pitch of tape twisting a more uniform distribution of the liquid on the heat exchange surfaces occurs under the action of mass forces, as a result of which the critical heat flows increase. The maximum effect of twisting is observed when $s/d$ is less than 4.

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