Influence of Artificial Roughness on Convective and Boiling Heat Transfer in the Rotating Flow

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Abstract: - The results of an experimental study of the process of convective and boiling heat transfer in the vessel with stirrer for smooth and rough ring-shaped pipes are presented. Experimentally was found three regimes of the influence of artificial roughness on the convective heat transfer: 1. regime, where roughness has no influence on heat transfer; 2. regime of partial influence of roughness on heat transfer; 3. fully developed roughness effect regime. It was established that creation of two-dimensional artificial roughness on the heated surface causes the essential (~90%) intensification of convective heat transfer in case of fully developed roughness effect regime. The similitude equation for calculating convective heat transfer coefficient, which generalizes well experimental data both for the smooth and the rough surfaces is proposed. In case of boiling the influence of roughness appears on the initial stage of boiling and in case of fully developed nucleate boiling there was no intensification of heat transfer.

Key-Words: - Heat, Transfer, Boiling, Roughness, Mixing.

1 Introduction

In the heat exchangers generally, and particularly in stirred tanks, which are widely used in chemical and food industry, heat transfer processes frequently accompany the chemical reactions. As a result, it is obvious that heat exchange intensification in such kind of apparatus has great practical value. Among of numerous methods of intensification of stirring and heat transfer currently the mostly effective and well studied is the method of using reflective spacers [1]. Together with this, the effectiveness of the method of artificial roughness appeared quite high in case of turbulent heat exchange in channels [2-6] has not been investigated in case of rotating flow. First works dedicated to this issue, as far as we know, were published by authors [7-9]. Boiling process in such conditions as far as we know was not studied as well.

2 Test Unit

The test unit **(**Fig.1) - stainless steel cylindrical vessel (1), with inner diameter D=200mm and height 300 mm was manufactured in order to investigate the influence of artificial roughness on heat transfer intensity in the rotating flow.

The heating low voltage AC electric power was supplied to experimental ring shaped stainless steel pipe (5) via copper conductors (7) installed through the bottom of the vessel. The outer diameter of pipe was 10 mm, average diameter of pipe ring -140 mm. The heated pipe ring was located at 50 mm from the bottom of the vessel coaxially with mixer's shaft. The admitted heat was disposed with liquid (distilled water) in the vessel. The wall of the vessel was cooled from outside by water (16) using cooling jacket (13). In case of saturated boiling vessel was insulated and vapour was condensed on the water (15) cooled coil (6). On the cover of the vessel was installed electric motor (4) connected to the shaft of the mixer (2). The paddles (3) with various dimensions were mounted on the mixer's shaft on the specific level from the bottom of the vessel. Thermocouple (10) with case (9) was immersed in the vessel with liquid for measuring liquid temperature.

Experiments were carried out in case when the paddles of the mixer were located in the different levels from the heated experimental pipe. Impeller mixers with vertical paddles were used. Mixers with various diameters (d), with different quantity (Z) and width (b) of the paddles, were used: d=35 mm, 50 mm, 65 mm, 100 mm, 120 mm, 180 mm; Z=2, 4, 6; b=10 mm, 20 mm, 30 mm. The angles between paddles were equal.

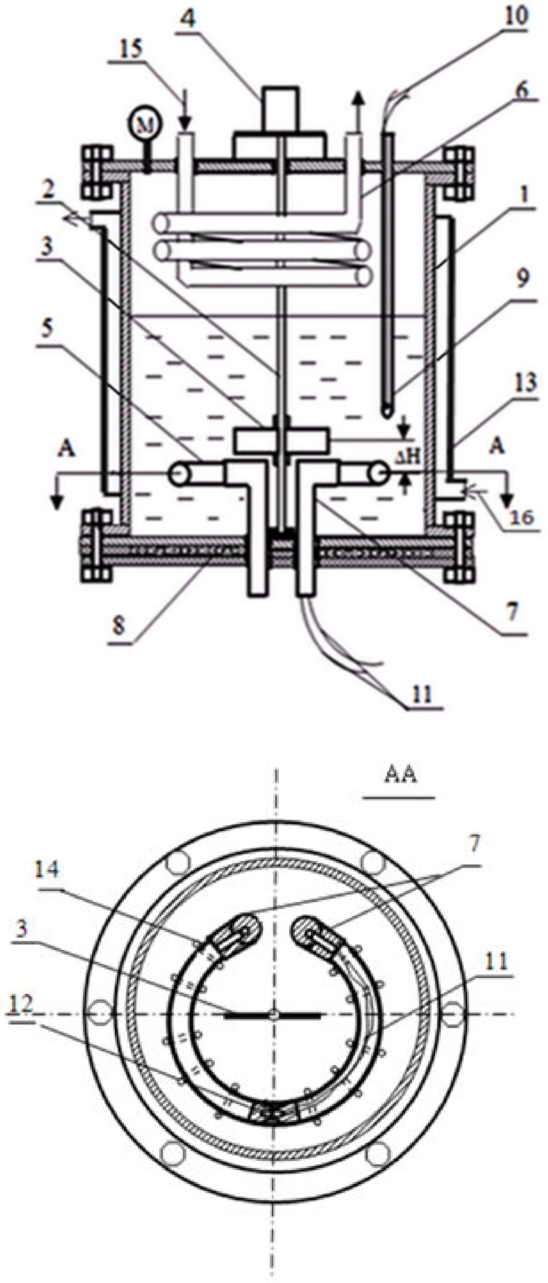

Fig.1 Test unit
1-Vessel; **2**-Shaft; **3**- Paddle;**4**-El. motor; **5**-Ring-shaped pipe; **6**-Cooling coil; **7**-Conductors; **8**-Auxiliary heater; **9**-Case; **10,11**–Thermocouples; **12**–Chamber; **13**–Cooling jacket; **14**–roughness elements; **15,16**-Cooling water.

Roughness elements- wire rings or washers were attached on the heated experimental pipe. The height (h) of the element of the roughness and the average pitch (s) between the elements varied: h=0.25 mm, 0.5 mm, 1.15 mm, 1.4 mm; s/h=3.5, 7.1, 7.5, 8, 10, 20, 40.

The inner surface temperature of the experimental pipe was measured by three chromel-alumel thermocouples, placed in Teflon chamber (12). The temperature of the outer surface of experimental pipe was calculated using well-known formula. The temperature of the distilled water in the vessel was measured also using chromel-alumel thermocouple, which was placed in the case (10) filled with oil. The voltage on the ends of thermocouples was measured by digital multimeter.

Heat transfer coefficient was determined by formula:

$$\alpha = \frac{q}{t_w - t_f} \tag{1}$$

The results of the experiments are performed using modified Re number in the range: $10^4 \leq Re \leq 350 \cdot 10^3$; Pr number in the range: $2 \leq Pr \leq 6.5$.

3 Convective Heat Transfer

Experiments were carried out both for smooth and rough surfaces (pipes). Part of experimental data is shown in Fig.2-4.

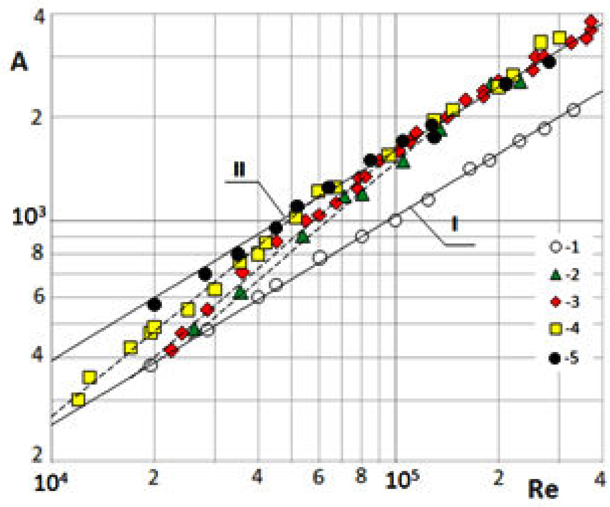

Fig.2. Relation of heat transfer intensity from Reynolds number, d=65 mm, b=10 mm; ΔH=0.
1. Smooth surface;
Rough surfaces:
2. h=0.25mm, s/h=10; **3**. h=0.5mm, s/h=10;
4. h=1.15 mm, s/h=10; **5.** h=1.4mm, s/h=7.1.
I – According to formula (3) (Smooth);
II – According to formula (3) (Rough).

In the Fig. 2 experimental results for smooth and rough surfaces are represented as a relation A=f (Re), where

$$A = \frac{Nu}{\Pr^{m}(D/d)^{k}(D/H)^{0.25}} \qquad (2)$$

For smooth surfaces m=0.33; k=0.35. For rough surfaces m=0.38; k=0.2. These results were obtained for $\Delta H=0$. In the Fig. 2 three different regimes can be seen: **1**. Roughness has no influence on heat transfer. **2**. Regime of partial appearance of roughness effect. **3**. Regime, where roughness effect is fully developed. In the regime 2, increasing roughness element's height heat transfer intensity increases accordingly and in the regime 3 heat transfer intensity is not related with roughness element's height.

In the Fig. 3 experimental data are presented for smooth and rough (h=0.25mm, s/h=10) surfaces when ΔH=30mm. Here up mentioned three regimes are seen more clearly.

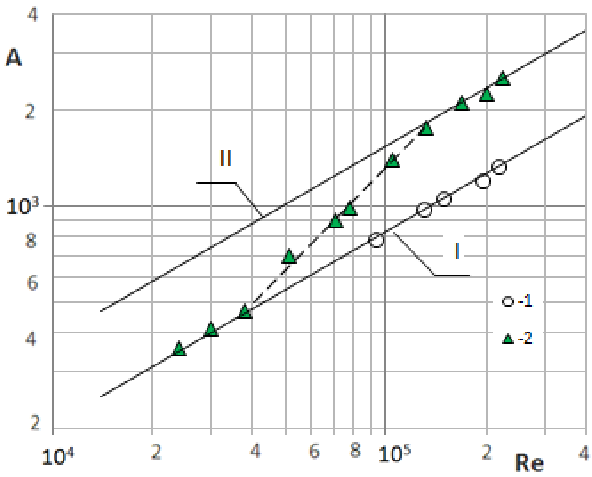

Fig.3 Relation of heat transfer intensity from Reynolds number, b=10 mm; ΔH=30 mm.
1. Smooth surface; Rough surfaces: **2.** h=0.25mm, s/h=10;
I – According to formula (3) (Smooth);
II – According to formula (3) (Rough).

According to Fig. 2 and Fig. 3 the effect of roughness was higher when ΔH=30mm.

Experimental data, shown in Fig. 4 represent relation of heat transfer intensification on the geometrical parameter (s/h).

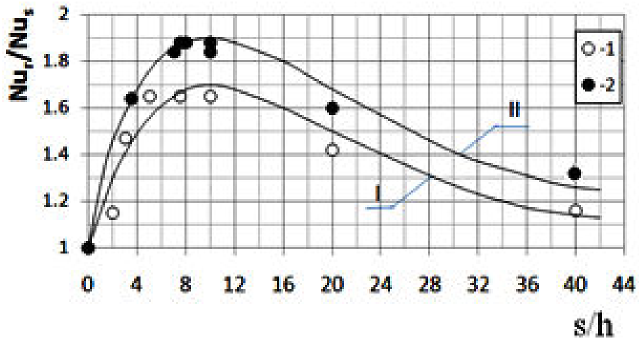

Fig.4 Relation of heat transfer intensity from geometrical parameter s/h. Re=2·10^5, Pr=3, d=120 mm, b=10mm. **1.** Experimental data, ΔH=0;
2. Experimental data, ΔH=30mm.
I – According to formula (3), ΔH=0;
II – According to formula (3), ΔH=30 mm.

As it is clear from the Fig.4, the character of relation of $Nu_r/Nu_s=f(s/h)$ is similar despite of change of ΔH. It was found that the optimal range of artificial roughness geometrical parameter pitch-to height ratio is $7 \le s/h \le 10$. In the range $7 \le s/h \le 10$ the meaning of Nu_r/Nu_s is higher for 20% in case of ΔH=30mm, than in case of ΔH=0. The ratio Nu_r/Nu_s is the highest (1.9) in case of ΔH=30mm and $7 \le s/h \le 10$.

Based on experimental study the following similitude equation was obtained:

$$Nu = 0.82\,\mathrm{Re}^{0.62}\,\Pr^{0.33}\left(\frac{D}{d}\right)^{0.35}\left(\frac{D}{H}\right)^{0.25}\left(\frac{25b}{H}\right)^{0.35} \times \qquad (3)$$
$$\cdot\left(1+\frac{|\Delta H|}{b}\right)^{-0.12}\left(\frac{Z}{2}\right)^{0.35}(\mu/\mu_w)^{0.14}\varepsilon_r,$$

Where in case of smooth surface $\varepsilon_r=1$, and for rough surfaces

$$\varepsilon_r = \Pr^{0.05}\left(\frac{D}{d}\right)^{-0.15}\left(1+\frac{|\Delta H|}{b}\right)^{0.1}\left(\frac{Z}{2}\right)^{-0.1} \times$$
$$(1+0.2(s/h)\exp(-0.1(s/h))).$$

In the Fig.5 is represented comparison between formula (3) and experimental data as a relation A^*=f(Re), where

$$A^* = \frac{Nu}{f(\Pr,G)} \qquad (4)$$

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$$f(\mathrm{Pr}, G) = \mathrm{Pr}^{\,0.33}\left(\frac{D}{d}\right)^{0.35}\left(\frac{D}{H}\right)^{0.25}\left(\frac{25b}{H}\right)^{0.35} \times$$
$$\cdot\left(1+\frac{|\Delta H|}{\mathrm{b}}\right)^{-0.12}\left(\frac{Z}{2}\right)^{0.35}(\mu/\mu_{\mathrm{w}})^{0.14}\,\varepsilon_{\mathrm{r}} \qquad (5)$$

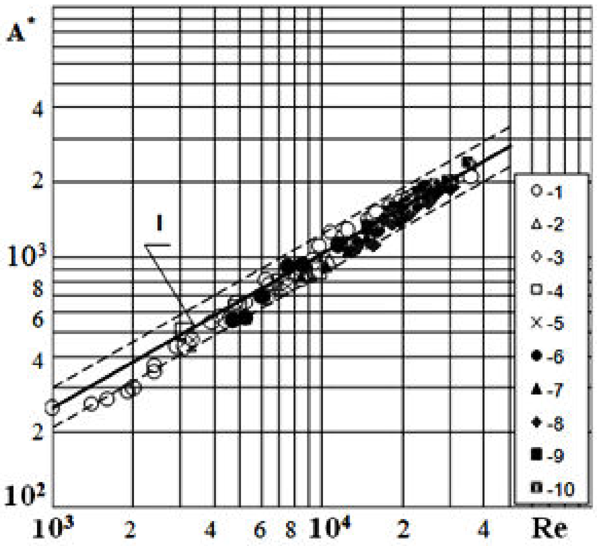

Fig.5 Relation of heat transfer intensity from Re number**. I** – According to formula (3); Experimental data.Smooth surface: b=10 mm, **1**. Z=2; **2**. Z=4; **3**.Z=6;**4**. b =20 mm; **5**. b=30 mm; Rough surfaces: b=10 mm, Z=2: **6**. s/h=5; **7**. s/h=8; **8**. s/h=10; **9**. s/h=20; **10**. s/h=40.

According to the obtained results heat transfer intensification due to artificial roughness in apparatus with mixer is less, than in rough canals. This difference can be explained based on ideas below.

Let us consider that in mixed liquid movement consists of three components – tangential, radial and axial. If we consider that one part of pipe is overflowed with tangential component and another with normal (radial and axial) for smooth surface can be written:

$$Nu_s = c_1 Nu_{s\tau} + c_2 Nu_{sn} \qquad (6)$$

where c_1 and c_2 can be determined experimentally. Since in our case roughness elements are displaced transversal with tangential component of the flow, obviously heat transfer intensification can be caused only due to this component. Taking into consideration above mentioned,

$$Nu_r = c_1 \varepsilon_{\mathrm{r}} Nu_{s\tau} + c_2 Nu_{sn} \qquad (7)$$

According to data, obtained in the case of canals when s/h=10-14, ε_{r}=2.4 by V. Gomelauri [3].

It can be seen from formula (7) that in our case the growth of heat transfer is less than in canals.

4 Boiling

Researches done in [10] show that in case of nucleate boiling of subcooled liquid (distilled water) flow in annulus artificial roughness increased heat transfer intensity in the regime of initial boiling and did not affect in case of developed nucleate boiling.

To find if the similar mechanism works in case of rotating liquid flow experiments were done in apparatus with stirrer both for smooth and rough pipes. Flat two paddle stirrer was used, d=65mm. Roughness was created attaching wire rings to ring-shaped stainless steel pipe. Artificial roughness geometrical parameters were: element height - h=1mm, pitch-to-height ratio- s/h=7. Distilled water was used as coolant.

In the Fig. 6 experimental data is presented both for smooth and rough pipes. Data can be divided by three zones: convective zone, transitional nucleate boiling zone and developed nucleate boiling zone. In the convective and transitional nucleate boiling zones influence of artificial roughness on the heat transfer is significant. Increasing heat flux mentioned influence becomes less and in the regime of developed nucleate boiling is equal to zero.

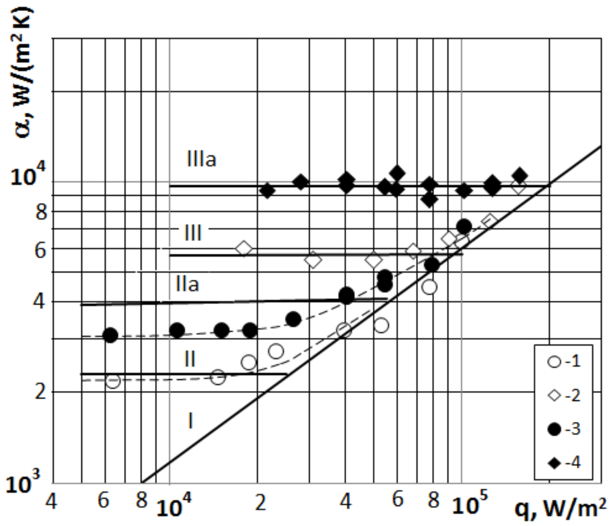

Fig. 6 Relation between heat transfer coefficient and heat flux Δt_{sc}= t_{st} - t_{tf}=7^{0}C,d=65mm, b=10mm.
1- smooth surface, n=2.33 RPS;
2- smooth surface, n=10 RPS;
3-rough surface, h=1mm, s/h=7, n=2.33RPS;
4-rough surface, h=1mm, s/h=7, n=10 RPS.
I- developed nucleate boiling, n=0;
II, IIa- acording to formula (3) for smooth and rough surfaces correspondingly, n=2.33RPS;
III, IIIa- acording to formula (3) for smooth and rough surfaces coreresponingly, n=10 RPS.

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Fig. 6 shows that in the convective zone coincidence of experimental data to the formula (3) when n=10 RPS is good, and when n=2.33RPS is not. This should be caused because at high RPS and accordingly at high Re numbers roughness effect is fully developed. At low RPS roughness effect develops partially.

5 Conclusions

In case of convective heat transfer creating two-dimensional artificial roughness on the surface of ring shaped heated pipe immersed in the apparatus with mixer significantly increases heat transfer intensity; roughness effect is more significant when mixer and heated pipe are placed at different levels.

Optimal range of two-dimensional artificial roughness geometrical parameter was found 7.5< s/h <10, where maximal intensification of heat transfer intensity (90%) was reached.

Similitude equation for calculating heat transfer coefficient generalizing well experimental data has been obtained.

In case of nucleate boiling two-dimensional artificial roughness increased heat transfer intensity in the regime of initial boiling, and did not in case of developed nucleate boiling.

6 Nomenclature

A	$[m^2/c]$	thermal diffusivity
α	$[w/m^2K]$	heat transfer coefficient
B	[m]	width of the paddles
c	[-]	constant
D	[m]	diameter of the vessel
d	[m]	diameter of the mixer
ΔH	[m]	level difference between mixer and heated pipe
H	[m]	level of water in the vessel
h	[m]	height of roughness elements
λ	[W/mK]	heat conductivity coefficient
ν	$[m^2/c]$	cinematic viscosity
n	[1/c]	mixer's RPS
$Nu=\alpha D/\lambda$	[-]	Nusselt number
$Pr=\nu/a$	[-]	Prandtl number
q	$[w/m^2]$	heat flow density
$Re=nd^2/\nu$	[-]	Modified Reynolds number
s	[m]	pitch between roughness elements
t	[K]	temperature
z	[-]	quantity of the paddles

Subscripts

f	Fluid
r	rough
s	Smooth
sc	Sub cooled
w	wall

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