HEAT TRANSFER AND HYDRODYNAMICS IN HEAT EXCHANGERS OF INDEPENDENT ELECTRIC MICRO SOURCES (IEM)

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ABSTRACT
IPMs are increasingly used as power plants (PP) for decentralized energy supply, robotics, transportation, gas pipelines, micro vehicle, etc. applications. IPM’s efficiency is ensured by the increased working media initial temperature TIT and the use of efficient miniature air heaters (μAH), which compactness and low material consumption are achieved through the use of some structural materials that do not require cooling at temperatures up to 1350ºC, and application of some novel technologies of implementation of micro channels in its matrix with the hydraulic diameter of less than 1.5 mm (dh <1.5 mm).

The reliability and efficiency of the μAH and gas turbine engine are generally provided at the design stage. This is accomplished using reliable computational relationships repeatedly proven by compact heat exchanger development practice.

Reducing the hydraulic diameter of the micro channel causes a significant reduction in the Reynolds numbers Re that determine the hydrodynamic flow condition of gas/air in the μAH paths. Condition changing affects their hydraulic resistance and heat exchange intensity. As a rule, micro channels have an arbitrary cross-sectional shape, while the absolute roughness of the walls of any structure is commensurable with the absolute size of their cross section.

In connection with this, for development of a miniature air heater for IPM, it is first necessary to identify the factors that have a significant impact on the mass and heat transfer in their micro channels and to establish the applicability of the "classical" equations of thermal and hydrodynamic similarity for the thermal hydraulic calculations of μAH.

In this paper, a survey of experimental and theoretical studies on the hydrodynamics and heat transfer in μCs for μAH and results of the thermal-engineering tests of the metal model of μAH is presented, this μAH having a matrix containing the microchannels of the same dimensions and configuration of the cross-section as are in the matrix of a full-scale μAH.

Analysis of the published material from a number of studies and experimental data obtained on the metal models has shown that "classic" calculation relationships of an acceptable accuracy [1], used for "normal" channels, can also be used for the thermal-hydraulic calculations of μAH.

INTRODUCTION
It is known that ~30% of the electricity produced in developed countries is spent on the energy supply to houses, public buildings and other structures that do not require mandatory centralized power supply. Each year, the volume of this market segment is growing and increasingly occupied by the decentralized energy supply sources, which serve as a grid support. The decentralization process is driven by frequent natural and man-made disasters, inevitably causing long interruptions in the supply of electricity and heat to thousands of customers.

Usually, power plants are used as stand-alone micro-power and/or mini-sources (IPMs) with internal combustion engines (ICE) or gas turbine engines (GTE) [2] serving as heat engines; use of GTE being thereat preferred due to its lower exhaust emissions. A major shortcoming of the current simple cycle μGTEs is that their efficiency is low due to the scale factor and limited capabilities of heat-resistant metal alloys that do not allow increasing the TIT≤900ºC.

A significant increase in the TIT and, thus, the efficiency, reliability and service life can be achieved using non-shrink structural ceramic materials with high creep rupture strength at temperatures of 1300-1400ºC [3]. Even at a high temperature (TIT=1350ºC), the miniature gas turbine engine has a very low efficiency (ηe=(6-8%)). ηe increasing is achieved through the regeneration of heat, which is realized by incorporating a mini recuperative air heater into the power generation makeup [4] (Fig. 1).

This solution provides a high engine compactness, low mass (3-4 times less that of metal products), and low-cost mass production [2]. We see that the engine compactness is largely determined by the geometrical dimensions of the applied μAH.
manufactured using non-shrinkage non-shrinking ceramics [5] and the laser prototyping technology [6]. The μGTE efficiency increase due to regeneration application of various thermal effectiveness (E=70, 80, 90%) is reflected by the plots, borrowed from the paper by Visser et al [4].

The μAH design concept applicable for the regenerative 2.0 kW engine is shown in Fig. 3.

**Figure 1** Miniature regenerative type gas turbine engine with structural ceramics components.

**Designations:** 1 – compressor, 2 – electric generator, 3 – turbine, 4 – combustion chamber, 5 – air heater, 6 – fuel supply, 7 – startup compressed air, 8 – igniter, 9 – electric power removal to customer, 10 – air at compressor inlet, 11 – exhaust gases to be removed to air heater.

**Figure 2** Effect of regeneration ratio E on ηε of regenerative miniature gas turbine engine at the rated power Nn=3kW at loads N/Nn=40-110% [4].

**Figure 3** Longitudinal (a) and cross (b) sections of μAH matrix for regenerative micro gas turbine engine [6].
µHP made of structural ceramics is used for heating the compressed air ($\pi_c=2.5$) to a temperature of $t_g'=1,005^\circ$C (E=0.86) and is intended for use in the "hot" atmosphere at a temperature $t_g'=1,150^\circ$C. It consists of a matrix (1), located between the inner (2) and outer (3) shells, the latter (3) of which is separated from the cylindrical body (not shown) of heat exchanger by the annular gap, which serves to move the compressed "cold" air (4) and to provide a counter flow of the working media of microchannels, distribution (5), (6) and collection (7), (8) chambers of these media.

The matrix (1) includes concentric cylindrical shells of uniform thickness and different diameters, where the air (9) micro channels of circular cross section are placed at an equal pitch in the middle of the circle with a cross section, while the gas (10) micro channels of the semicircular cross-section and interconnected by the radial (11) micro channels, leveling the distribution of gas over the peripheral gas micro channels in each shell, are placed on the periphery. The direction of the working media flow is indicated by arrows: air (4), (14), gas (12), (13).

To carry out detailed thermal-engineering studies, a metal model (Fig. 4) of µHP is made; its matrix having micro channels of the same cross-sectional configuration and geometrical dimensions as in the full-scale heat exchanger. The difference between the model and full-scale µHP is a number of cylindrical shells that form the matrix. Thus, the model has three shells, while the full-scale heat exchanger has seven shells.

The careful analysis of experimental and theoretical studies dedicated to identification of the factors that affect the mass and heat transfer in regenerative µAHs was made by Celata G.P. [7]. Authors of the articles quoted in this review note an impact on the hydrodynamics and heat transfer during the motion of gas/air along channels of a small hydraulic diameter ($d_h<1.5$) imposed by such factors as:

- the absolute hydraulic diameter $d_h$ of the channel;
- the configuration of the cross section and its relative dimensions as applied to the channel of the rectangular section, with width b and height h of the channel;
- the relative roughness (RR) of their walls;
- the turbulence ratio at the channel inlet, and, also, the non-stationarity, non-isothermality, flow duty, coolant discontinuity, etc.

Ambiguous and contradictory nature of the mass and heat transfer processes in micro channels, obtained by the studies, is largely due to the difference in processing techniques and the generalization of the experimental data, a different structure of the working areas of the experimental facilities, differing methods of conducting experiments and precision measurements performed.

The results of the experiments show that the empirical correlations developed for the calculation of the coolant flow and heat transfer during the motion of coolants in conventional tubes [1], for which $d_h>>\delta_l$ (where $\delta_l$ is the boundary layer thickness) are not fully applicable to micro channels. It is necessary to carry out a more systematic research to clarify their relation to micro channels.

Celata G.P. [7] notes that some additional mechanisms affecting the formation of boundary layers influence the coolant flow; it is the lack of adhesion of coolant, wall electro-viscosity effects, the compressibility of the gas flow, etc.

1. HYDRAULIC RESISTANCE AT GAS FLOW ACROSS MICRO CHANNELS

A number of experimental studies reviewed in [8] and dedicated to the microscale thermal hydraulics indicate the presence of significant differences with coolants flowing in micro-and macrochannels, in particular, during the transition from the laminar to the turbulent condition (Table 1).

A detailed review of the hydraulic resistance during the single-phase liquid flow in smooth micro channels with varying cross-sectional configuration and orientation in space, with no heat supply to the liquid, and when it is heated in the laminar, transitional and turbulent duty flows, for media of different viscosities is presented in a comprehensive experimental study by Celata G.P. et al [9]. The authors found that:

- the classical equations of similarity (Hagen-Poiseuille, Blasius) allow to calculate the coefficients of friction with respect to micro channels of any cross-section with the accuracy of +/- (10-30)% at various orientations in space, with the hydraulic diameter ranging $d_h=259-1,699$ microns;
- given the heat supply/heat sink, it is necessary to consider the change in viscosity of the working media.
<table>
<thead>
<tr>
<th>№</th>
<th>Tubes, μm</th>
<th>Tube material. Roughness of micro-channel walls</th>
<th>Coolant</th>
<th>$\xi_f$; operation range Re</th>
<th>Transition range Re</th>
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<tr>
<td>1</td>
<td>50-254</td>
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<td>Water</td>
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<td>[8]</td>
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<td>-</td>
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<td>700-1,100</td>
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<td>19;52;102</td>
<td>-</td>
<td>Nitrogen; water</td>
<td>$\xi_f &lt; \xi_{f,0}$ smooth tube; $\xi_f = 1.15\xi_{f,0}$ rough tube</td>
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<td>[8]</td>
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<tr>
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<td>-</td>
<td>Water</td>
<td>$\xi_f = \xi_{f,0}$</td>
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<td>-</td>
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<tr>
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<td>Glass, silicon, stainless steel (rough)</td>
<td>Water</td>
<td>$\xi_f &lt; \xi_{f,0}$ at laminar duty; $\xi_f = 1.5\xi_{f,0}$ at high $d_h$</td>
<td>1,700-2,000</td>
<td>[8]</td>
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<td>20-150</td>
<td>Melted silica</td>
<td>Water, hexane, isopropanol</td>
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<td>R114</td>
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<td>12</td>
<td>133-343</td>
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<td>water</td>
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<td>-</td>
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<td>Distilled water</td>
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<tr>
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<td>Water, methanol, isopropanol</td>
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<td>-</td>
<td>[9]</td>
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<tr>
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<td>Deionized water</td>
<td>$\xi_f &lt; \xi_{f,0}$ laminar duty</td>
<td>-</td>
<td>[9]</td>
</tr>
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<td>173-4010</td>
<td>Tubes</td>
<td>Air, water, coolant R-134a</td>
<td>$\xi_f &lt; \xi_{f,0}$</td>
<td>1,200-3,800</td>
<td>[9]</td>
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<tr>
<td>18</td>
<td>50 and more; Non-ferrous tubes: 848;1237;1699; length 310 mm</td>
<td>Circular tubes, round tubes, all smooth; straight channels (Ra=0.1μm); Copper and stainless steel; roughness Ra&lt;1 μm</td>
<td>Deionized water R-134a</td>
<td>$\xi_f &lt; \xi_{f,0}$</td>
<td>-</td>
<td>[9]</td>
</tr>
<tr>
<td>19</td>
<td>30-2,000</td>
<td>Stainless steel</td>
<td>Air; $M_{in}&lt;0.15$</td>
<td>$\xi_f &lt; \xi_{f,0}$</td>
<td>Re&lt;2,300</td>
<td>[12]</td>
</tr>
<tr>
<td>20</td>
<td>Slotted mounting cooling channels</td>
<td>Stainless steel. Smooth slotted channels</td>
<td>Air</td>
<td>$\xi_f &lt; \xi_{f,0}$</td>
<td>Re&gt;3,500</td>
<td>[11]</td>
</tr>
</tbody>
</table>

Notes.
$\xi_{f,0}$ – friction factor, determined by the Hagen-Poiseuille’s or Blasius’ laws.
Results of the extensive research by Celata et al [9], carried out in 2006, serve as confirmation of the previously (50 years ago, 1956) obtained experimental data for the air flow in capillary channels [10] and in the mounting gaps of herringbone rotor blade shanks (Fig. 5) [11]. These materials have been reported in the well-known monograph by Shvets I.T. and Dyban E.B. dedicated to the air cooling of gas turbines parts [12].

In papers [10-12], it was stated that the air flow in the slotted smooth channels of herringbone shanks is $\xi_f = \xi_{f0}$. The deviation from the Blasius relation with $Re = (3-4) \times 10^3$ does not exceed 15%, while at $Re = (1-1.9) \times 10^3$, $\xi_f > \xi_{f0}$ the deviation is 20-40% (Fig. 6-8).

For air, $\xi_f = f(M)$ is up to values of the Mach number $M = 0.9-0.95$.

The heated air increases the turbulence ratio at the entrance, which leads to an increase in $\xi_f$.

At $T_u > 5-7\%$, the flow duty is crisis-free.

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At the heat supply to the flow, a smooth (crisis-free) transition from the Poiseuille’s relationship to the Blasius’ relationship takes place. A similar effect on the patterns of transition is imposed by increasing the level of the initial turbulence $Tu$ (Fig. 8).

Tu increase reduces the critical Reynolds number of the transition start from 2,300 to (1.7-1.8) $\times 10^3$ and does not change $\xi_f$ at the turbulent flow, starting at $Re > 3,500$. At $1,900 < Re < 1,000$, $\xi_f$ is 20-40% higher the value obtained using the Poiseuille relationship.

At the gas flow in micro channels, a number of investigators found out an influence of the compressibility of the gas stream. Guo et al [8] revealed that the change in the gas density in the flow direction can be quite large if the high pressure losses from friction in the flow direction, leading to a change in the velocity profile and friction coefficient $\xi_f$ are high (Fig. 9) [13].
of the boundary values Re<sub>c.1</sub> and Re<sub>c.2</sub> can be performed using roughness, as well as for “conventional” tubes, the calculation of transition zones of the Reynolds numbers, Celata \[12\] according to which: Re<sub>c.1</sub>=1,160 (for tubes with any roughness). In the formulas given in the handbook by Idelchik I.E. \[13\], and Deych \[14\], we can conclude that there is no influence of the absolute size of the micro channel in the transition from the laminar to the turbulent flow, which allows to calculate the hydrodynamics in the capillary channels of the criteria relations given in the references \[1,16\].

At the self-similar turbulent condition, \(\xi_f\) in the micro channels, as in conventional tubes, does not depend on the Reynolds number and is determined solely by the relative roughness value of the channel walls.

For the micro channels in the transition zone, \(\xi_f\) depends on the Re number and the relative roughness; whereas at the transition to the turbulent condition \(\xi_f\) gradually decreases with increasing the Re number, reaching the lowest values of the quadratic (self-similar) mode.

For the laminar flow in the channel with an equivalent relative roughness over 0.7%, the deviation from the Hagen-Poiseuille’s law in the direction of increasing \(\xi_f\) is likely to occur. The greater is the relative roughness value, the lower is the Re number, at which the deviation occurs.

Based on the analysis of results of the experiments carried out \[9-12\], we can conclude that there is no influence of the absolute size of the micro channel in the transition from the laminar to the turbulent flow, which allows to calculate the hydrodynamics in the capillary channels of the criteria relations given in the references \[1,16\].

The surface roughness of channels, providing a significant effect on the pressure drop and heat transfer in the micro channels, differs in its structure and the absolute size of the roughness of “conventional” tubes, for which \(d_h>>\delta_r\).

However, for micro channels with a non-uniform roughness, as well as for “conventional” tubes, the calculation of the boundary values Re<sub>c.1</sub> and Re<sub>c.2</sub> can be performed using the formulas given in the handbook by Idelchik I.E. \[15\], according to which: Re<sub>c.1</sub>=1,160 (\(\delta_r/d_h\) \(0.11\) with \(\delta_r/d_h\)>0.007; Re<sub>c.2</sub>=2090(\(\delta_r/d_h\) \(0.0636\) (for tubes with any roughness). In the transition zone of the Reynolds numbers, Celata \[8\] recommends the following relationship to calculate \(\xi_f\) in \(\mu\)C:

\[
\xi_f = \xi_{\text{fo}}(1+M^2(1.5-0.66M-1.44M^2)),
\]

where \(M\) is the local Mach number;

\[
\xi_{\text{fo}} = \frac{64}{\text{Re}}.
\]

This conclusion contradicts the experimental data of Shvets I.T., Dyban E.P.\[12\], and Deych \[15\], who state that \(\xi_f\) is essentially independent of the Mach number (up to \(M=0.9-0.95\)).

The number Nu<sub>min</sub> and the hydraulic resistance coefficient of friction \(\xi_f\) in this case have a significant influence on the cross-sectional shape of the channel. For the capillary tubes of circular cross-section, Nu<sub>min</sub>=3.66, and the rectangular Nu<sub>min</sub>=7.5-8.24 as function of the aspect ratio \(\psi=h/b\). Here,
\( \text{Nu}_{\text{min},t} \) is the minimum Nusselt number for the constant wall temperature \( t_w \). With a decrease in \( \psi \), \( \text{Nu}_{\text{min},t} \) increases.

Reduction of the complex \( \frac{Re d_h}{l} \) promotes the Prandtl number \( Pr \), which is less than 1.0 for gas coolants (e.g., air) \[17, 19\]. With respect to the slotted micro channels, for calculation of \( \xi_f \), \( \text{Nu}_{\text{min},t} \) you can use the following approximation formulas:

\[
\xi_f = 57.736 \psi^2 - 95.072 \psi + 94.792; \tag{8}
\]

\[
\text{Nu}_{\text{min},t} = 8.7699 \psi^2 - 13.524 \psi + 7.8092. \tag{9}
\]

Under the laminar gas flow in a microchannel (as opposed to the conventional channels \[1,12\]), the convective heat transfer is significantly affected by the Mach number (Fig. 11) \[13\].

![Figure 11](image) Effect of Mach number on the convective heat transfer at gas flow in the micro channels \[13\].

It is evident that the effect of the Mach number \( M \) on the convective heat transfer in the micro channels is already apparent at \( M=0.05 \) (curve 2), and a situation is likely to occur that the heat flow is directed from the wall to gas, i.e. \( T_{a.w} > T_g \) (curve 4, \( M=0.2 \)).

\[
T_{a.w} = T_g (1 + (r(\kappa-1)/2) M^2), \tag{10}
\]

where \( T_{a.w} \) is the adiabatic wall temperature, \( T_g \) is the temperature of the gas flow; \( r \) is the temperature recovery coefficient at the gas flow in the channel, equal to:

\[
r = Rf^{0.33} \Delta x; \tag{11}
\]

Here:

\[
\Delta x = 7.16 - 10^3 Re_{x}^{0.4} f(x/d_h); \tag{12}
\]

\( f(x/d_h) = 1 \) at \( x/d_h = 0-15 \) (initial segment);

\( f(x/d_h) = 1+0.0413(x/d_h-15) \) at \( x/d_h = 15-27 \) \[15,19\].

The cross-sectional shape of the micro channels is very diverse (Fig. 12). It has a significant effect on the rate of heat transfer in a channel, especially under the laminar condition.

![Figure 12](image) Shapes and sizes of the cross section of the micro channels in flat multi-channel tubes applied to manufacture miniature heat exchangers \[7\].

The difference in the convective heat transfer is due to the secondary currents generated in such channels. They lead to significant differences in the heat transfer coefficients in comparison with a circular channel, having the same hydraulic diameter. It is shown in publications devoted to the study of heat transfer in the micro channels (\( d_h < 1.5 \) mm) that the technology of manufacture of the test micro channels of a noncircular section causes a number of effects that lead to significant differences in the convective heat transfer, compared with round section micro tubes. The available information on the convective heat transfer in the circular and noncircular micro channels does not provide trustworthy recommendations for the thermo-hydraulic calculations of a compact heat exchangers with micro channels. More systematic researches are needed to consider each of the parameters that affect the transport processes at the gas flow in them. However, it is unacceptable to neglect the experimental information already available, summarized in the paper by Palm and Peng \[8\] and applying to the heat transfer in noncircular micro channels with \( d_h = 133-367 \mu \text{m} \). They found a significant difference in the intensity between the heat exchange in micro- and "conventional" channels.

At the laminar flow in the micro channels, the test data is generalized by the equations:

\[
\text{Nu} = C_1 Re^{0.62} Pr^{0.33} \tag{14}
\]

or for an array of parallel rectangular channels

\[
\text{Nu} = 0.1165 Re^{0.62} Pr^{0.33} (d_h/b)^{0.81} (h/b)^{-0.79}, \tag{15}
\]
which is applicable at $900 \geq Re \geq 80$ and $367 \geq dh \geq 133 \mu m$.

Here, $C_1$ is the empirical coefficient, which depends both on the hydraulic diameter $dh$, and the ratio of the channel thickness $h$ to its width $b$ (Fig. 13).

It is evident that there is an optimal height $(h/b)$, which minimizes the thermal resistance in a rectangular channel. In accordance with Peng’s experiments, this height is $h/b = 0.75$.

![Figure 13 Effect of the relative height ($h/b$) and $d_h$ on the convective heat transfer in the micro channels [7].](image)

Table 2

<table>
<thead>
<tr>
<th>№</th>
<th>$b$, mm</th>
<th>$h$, mm</th>
<th>$L$, mm</th>
<th>$d_h$, mm</th>
<th>$h/b$</th>
<th>$C_1$</th>
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<td>0.4</td>
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<td>50</td>
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<tr>
<td>4</td>
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<td>0.667</td>
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</tbody>
</table>

For conditions where $Re \geq 1,000$, Peng suggested a correlation $Nu = C_1 Re^{0.19}Pr^{0.33}$ for the turbulent flow, which, unfortunately, has a considerable scatter of experimental data.

On the basis of the analysis of experimental data on the convective heat transfer in the micro channels under the turbulent flow condition, it was established that for channels with hydraulic diameter $d_h = 1.2$ mm is critical for $d_h$, at which you can use "classical" equations of the thermal similarity for calculation of the convective heat transfer [1,16,19].

In Zotov’s study [14] over the air flow in slotted microchannels $(h = 0.048...0.18$ mm), a stratification of the experimental data, depending on the relative roughness of the channel walls, was revealed despite a relatively small height of the roughness $(\delta < 1.2$ mm). In some experiments, the air stream speed at the outlet of the channels reached the sound level. The results of this work are consistent with the materials reported by Watts [20], which studied the hydraulic resistance of friction $\xi_f$ at the flow of nitrogen, argon and hydrogen in planar channels of glass and silicone of the height $h = 0.028...0.065$ mm, the average height of roughness bumps on the surface is less than $\delta = 0.004$ mm, while the relative roughness ratio is $\Delta = 0.143...0.0615$. With such $\Delta$, greatly exceeding the limit value $\Delta_{lim} = 15 / Re = 0.03...0.015$ for any commercially available tubes, stratification should take place, as is noted in the handbook by Idelchik I.E. [16].

Effect of the isotropic roughness on the heat transfer intensity with air $(R_u = 0.01...0.7$ MPa) flowing in slotted micro channels is examined in the experimental study [12]. The slotted channels of stainless steel and glass are employed. The absolute roughness of the walls of the channels was $\delta_s = 0.1; 1; 2; 3; 4; 7; 10$ µm, and the thickness of pads, forming the channel, is $h = 0.05; 0.06; 0.09$ mm. The pad thickness was measured with the accuracy up to $1$ µm. The channel width is $b = 3.1$ mm, the length is $L = 13.5; 22.6$ mm. To calculate a slotted length averaged channel $\xi_f$, relationship that applies to an object that has no heat exchange with the environment was used, while the resistance coefficient along the length of the channel is constant.

On the basis of the analysis of results of the experiment, we can conclude that there is no influence of the absolute size of the micro channels on the transition from the laminar flow to the turbulent flow, which allows for calculation of the heat exchange in the capillary channels using conventional criteria relations [1,16,19].

All the capillary tubes, where the air flow was investigated [10-12], are hydraulically smooth, so no stratification of the experimental data at the turbulent duty was revealed. Increasing the level of the initial turbulence reduces $Re_{c1}$ to $1,700...1,800$, and practically does not change the friction resistance coefficient in the turbulent air flow.

At a higher turbulence $(Tu > 5...7\%)$, the crisis-free transition from the laminar to the turbulent flow is implemented (Fig. 8), while at $Re = 1,000...1,900$, $\xi_f$ is 20...40% higher that calculated by the Hagen-Poiseuille’s law $\xi_f = 64 / Re$ for the circular cross-section tubes.

The experimental materials [9-12] confirm the applicability of the classical equations of similarity for calculating the convective heat transfer with gases flowing in the micro channels, and the validity of using the Hagen-Poiseuille’s and Blasius’ formulas to determine $\xi_f$.

3. HEAT TRANSFER ENHANCEMENT

To intensify the heat transfer inside the micro channels, some known methods can be used [21], namely the coolant flow twisting [22,23] and the longitudinal channel configuration variation, contributing to the emergence of pressure fluctuations in a moving stream [24-26].

With the gas/air flowing across the curvilinear channel, an uneven distribution of velocities and pressures is established in its sections, transverse pressure gradients emerge, causing a secondary vortex flow that is superimposed on the main stream. The secondary flow lines are locked in the secondary flow channel cross section.

The secondary flow consists of two streams, which are directed to the convex surface near the walls, and in the center of the channel they are directed to the concave surface and have the symmetrical – helical character.
The structure of the secondary flow and the loss of energy, caused by it, depends on the geometrical shape of the channel and the flow condition, defined by the Reynolds numbers and Mach numbers. The secondary flows enhance the heat exchange at any flow mode [22].

At the Dean number, De=Re√dh/D ben=26-7,000 and (dh/D ben)=6.2-62.5, where D ben is the bending diameter, the laminar flow with macro vortexes occurs, at which:

\[
\text{Nu}=0.0575 \text{Re}^{0.8} \text{Pr}^{0.43} (\text{dh}/\text{D ben})^{0.21} (\text{Pr}_f/\text{Pr}_w)^{0.25} (16)
\]

For calculation of ξf, the Tschukin’s formula can be used:

\[
\xi_f=0.0385(\text{D ben}/\text{dh})^{0.5} +0.312 \text{Re}^{0.25} (22). (17)
\]

At the supply (11)/outlet (12) "cold" compressed air has a similar equipment (except for the heater) and instrumentation (8), (9), (10), allowing to determine the heat flux absorbed by the compressed air, and the pressure losses associated with pushing it along the micro channels of the μHP model air paths.

Results of the experimental values of Qg; Qc; Δp; Δpa in Figure 15 are compared with calculated Q=kΔTF, where Q is the heat flux transferred through the heat exchange surface F of the model matrix, k is the coefficient of heat transfer, which calculation is performed using the "classical" equation of similarity [1,19], delta T is the mean log temperature head.

The experimental values of the pressure loss Δp; Δpa in the matrix model paths are compared with calculated values determined using recommendations by I.E. Idelchik [16] for the "conventional" tubes.

4. THERMAL-ENGINEERING TESTS OF METAL MODEL OF μAH.

The experimental study of the hydraulic resistance and heat transfer model was made on the experimental setup (Fig. 14), consisting of the working section (1) with a metal model (2) of μAH, supply (3)/outlet (4) "hot" air parts placed on it.

The compressor (5), serving both model paths (2), is placed within the area (3). It is equipped with an electric heater (6), a flow meter (7), pressure gauge (8) and thermal sensors (9), designed to measure the parameters of the gas at the inlet and outlet of the model (within the heat removal area (4)).

The presence of this equipment and instrumentation enables to determine empirically the heat flux released by the "hot" air (gas) in the gas path of the μAH model and to identify pressure losses in it.

The supply (11)/outlet (12) "cold" compressed air has a similar equipment (except for the heater) and instrumentation (8), (9), (10), allowing to determine the heat flux absorbed by the compressed air, and the pressure losses associated with pushing it along the micro channels of the μHP model air paths.

Results of the experimental values of Qg, Qc, Δp, Δpa in Figure 15 are compared with the calculated Q=kΔTF, where Q is the heat flux transferred through the heat exchange surface F of the model matrix, k is the coefficient of heat transfer, which calculation is performed using the "classical" equation of similarity [1,19], delta T is the mean log temperature head.

The experimental values of the pressure loss Δp, Δpa in the matrix model paths are compared with calculated values determined using recommendations by I.E. Idelchik [16] for the "conventional" tubes.

Figure 14 Schematic diagram (a) of the experimental setup for thermal-engineering test of μAH model (b).

Figure 15 Comparison of test values Δp, Δpa, Qg, Qc with calculated Δp, Δpa (a); Q (b) by recommendations [1;16;19].
SUMMARY

1. Experimental data on the mass and heat transfer at the coolants flowing in micro channels ($d_h < 1.5$) are ambiguous and often contradictory; they do not provide specific guidance on the thermal hydraulic calculation of miniature heat exchangers, in which matrix similar micro channels are applied.

2. For the thermal - hydraulic design of miniature heat exchangers, it is recommended to use "classical" empirical equations of the thermal and hydrodynamic similarity obtained for "conventional" tubes and channels, for which $d_h \gg \delta_h$.

3. When using "classical" equations of similarity with respect to the micro channels $d_h < 1.5$ mm, the trustworthiness of the calculated data will be lower than it is the case with "conventional" tubes.

4. Calculation findings obtained using "classical" equations of similarity should be compared with the recommendations of individual researchers, whose empirical equations are placed in this paper.

5. We need to continue research on:
   - a definition of the pressure losses variations in the micro channels,
   - a variation in the density and velocity of the gas/air flows,
   - the Mach number effect on the pressure drop and heat transfer

6. To improve the heat transfer coefficients in both paths of the matrix, channels with slotted or flat-oval cross-sectional configurations should be applied.

7. Reduction of the hydraulic resistance is facilitated using the convergent-divergent type channels.

8. The coolant flow twisting in the $\mu C$ leads to a significant heat transfer enhancement.

9. Increasing the roughness of the $\mu C$ walls enhances their hydraulic resistance and will have an insignificant effect on the convective heat transfer due to a total immersion of the roughness bulges in the laminar boundary layer.

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