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HEAT TRANSFER AND HYDRODYNAMICS IN HEAT EXCHANGERS OF INDEPENDENT ELECTRIC MICROSOURCES (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 (**d_h <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 (η_e<(6-8%)). η_e 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.

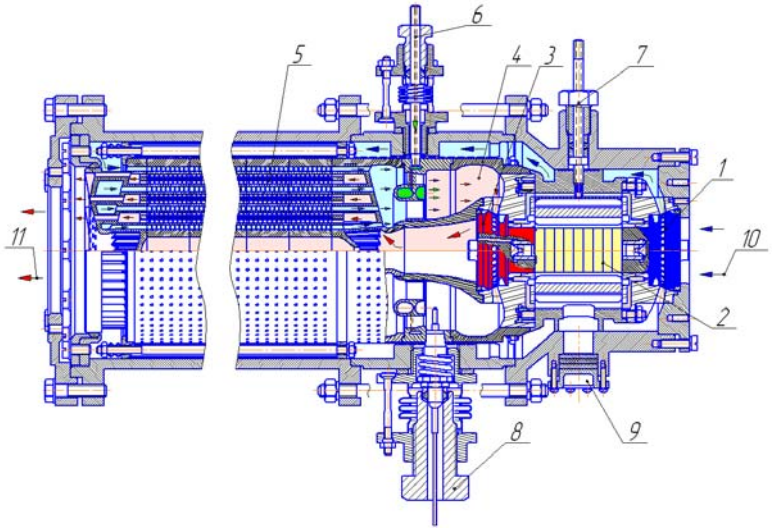


Figure 1 Miniature regenerative type gas turbine engine with structural ceramics components.
Designations: 1 – compressor, 2 – electric generator, 3 – turbine, 4 – combustion chamber, 5 – air heater, 6 – fuel supply, 7 – startup compressed air, 8 – igniter , 9 – electric power removal to customer, 10 – air at compressor inlet, 11 – exhaust gases to be removed to air heater.

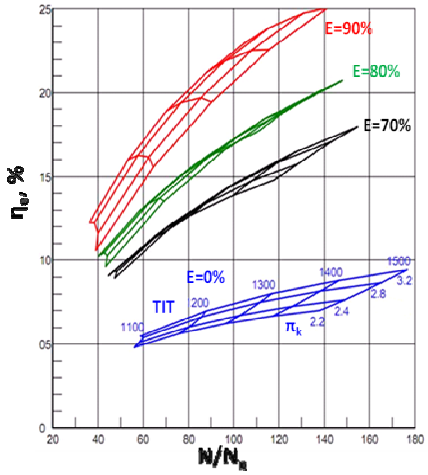


Figure 2 Effect of regeneration ratio E on η_e of regenerative miniature gas turbine engine at the rated power N_n=3kW at loads N/N_a=40-110% [**4**].

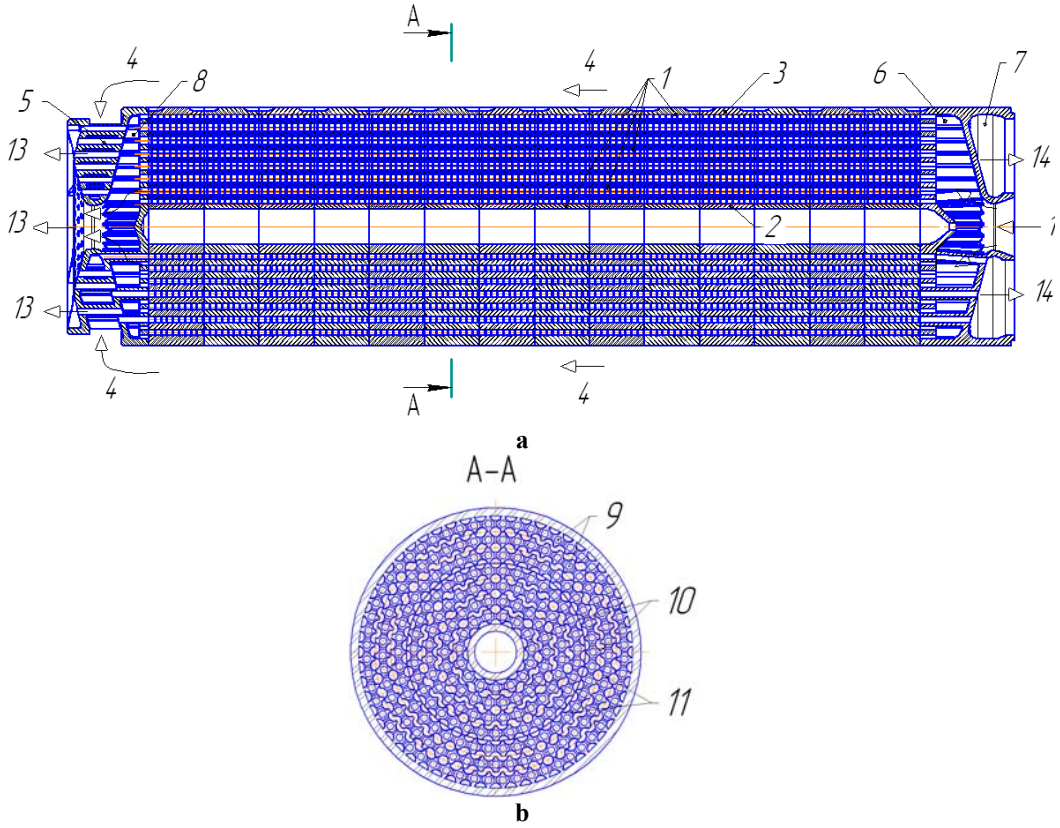


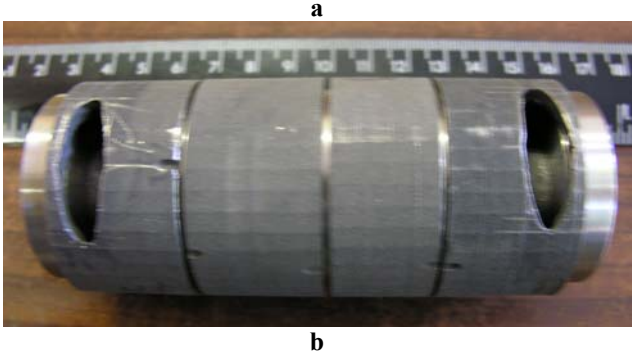
Figure 3 Longitudinal (a) and cross (b) sections of μAH matrix for regenerative micro gas turbine engine [**6**].

μHP made of structural ceramics is used for heating the compressed air (π_k=2.5) to a temperature of t_a"=1,005ºC (E=0.86) and is intended for use in the "hot "atmosphere at a temperature t_g'≈1,150ºC. 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- 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.

a



b

Figure 4 Photos of standard sections of matrix (a), general view of metal model of μAH with coolant collection/distribution chambers (b).

The tests of the metal model were carried out at the maximum gas temperature not exceeding t_g, M<300ºC. The results of the model testing are presented in §4.

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 $\mathbf{d_h}$ 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-isothermity, 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>>δ_b (where δ_b 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 1

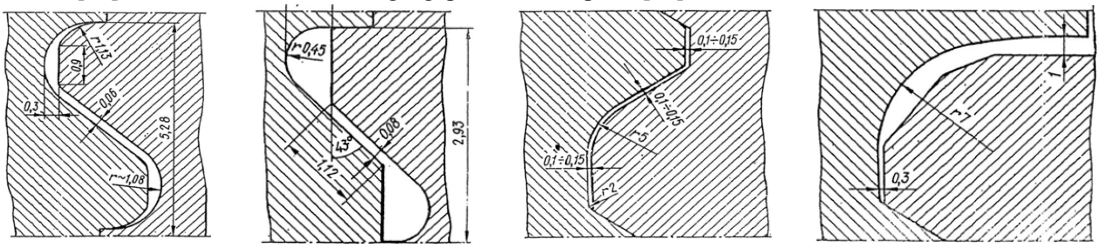
Test findings of studies over hydrodynamics in micro-channels [**7-12**]

№	Tubes, μm	Tube material. Roughness of micro-channel walls	Coolant	ξ_f; operation range Re	Transition range Re	Source
1	50-254	Rough channel	Water	$\xi_f \neq \xi_{f.0}$ at low $\mathbf{d_h}$; $\xi_f \approx \xi_{f.0}$ at high $\mathbf{d_h}$	300-900; >1,000-1,500	**[8]**
2	50-300	-	Water	$\xi_f < \xi_{f.0}$	>1,500	**[8]**
3	300-750	Metal, machined	Water, methanol		700-1,100	**[8]**
4	19;52;102	-	Nitrogen; water	$\xi_f < \xi_{f.0}$ at laminar duty; 250-20,000	2,000-6,000	**[8]**
5	30-344	-	Water	$\xi_f = \xi_{f.0}$; 20-4000	-	**[8]**
6	320	-	Water	$\xi_f > \xi_{f.0}$; 60-600	-	**[8]**
7	80-166	Glass, silicon, stainless steel (rough)	Water	$\xi_f = \xi_{f.0}$ smooth tube; $\xi_f \approx 1,15\xi_{f.0}$; rough tube.	1,700-2,000	**[8]**
8	20-150	Melted silica	Water, hexane, isopropanol	$\xi_f < \xi_{f.0}$ at $\mathbf{d_h}$<100 $\xi_{f.min} = 0,7\xi_{f.0}$	-	**[8]**
9	130	-	R114	$\xi_f = \xi_{f.0}$; at Re<600; $\xi_f > \xi_{f.0}$; Re>600	1,900-2,500	**[8]**
10	760-1090	-	water		-	**[8]**
11	50-254	Stainless steel; melted silica	water	$\xi_f \neq \xi_{f.0}$ at low $\mathbf{d_h}$	-	**[9]**
12	133-343	Slot	water	$\xi_f \neq \xi_{f.0}$	-	**[9]**
13	51-169	Trapezoidal silicon micro channels			-	**[9]**
14	50-1120	Trapezoidal micro channels	Distilled water	Re>600 $\xi_f \approx \xi_{f.0}$	-	**[9]**
15	15-150	Stainless steel; melted silica	Water, methanol, isopropanol	Re=8-2,300 $\xi_f \approx \xi_{f.0}$	-	**[9]**
16	26-291	Trapezoidal silicon micro channels	Deionized water	$\xi_f \approx \xi_{f.0}$; laminar duty	-	**[9]**
17	173-4010	Tubes	Air, water, coolant R-134a	$\xi_f \approx \xi_{f.0}$	1,200-3,800	**[9]**
18	50 and more	-		$\xi_f \approx \xi_{f.0}$	-	**[9]**
19	30 and more; Non-ferrous tubes: 848;1237;1699; length 310 mm	Circular tubes, round tubes, all smooth; straight channels (Ra<0.1μm); Copper and stainless steel: roughness Ra<1 μm	Deionized water; coolant R-134a	$\xi_f \approx \xi_{f.0}$	-	**[9]**
20	310-2,000	Stainless steel	Air; M_{out}<0,15	Re=10^3-45*10^3 $\xi_f \approx \xi_{f.0}$	Re<2,300 **[12]**	**[10]**
21	Slotted mounting cooling channels	Stainless steel. Smooth slotted channels	Air	Re=10^3-16*10^3; $\xi_f \approx \xi_{f.0}$; $\xi_f \neq f(M)$	Re>3,500	**[11]**

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**].



a **b** **c** **d**

Figure 5 Mounting gaps and cooling channels for conventional shank joints [**12**].
Designations: **a, b** – mounting gaps of herringbone GTE shanks; **c** - the same as before for GT-6-750 by the Turbomotor Works (TMZ); **d –** cooling channels of shanks for TMZ GT-6-750.

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

For air, $\xi_f \neq f(M)$ 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 ξ_f.

At Tu> 5-7%, the flow duty is crisis-free.

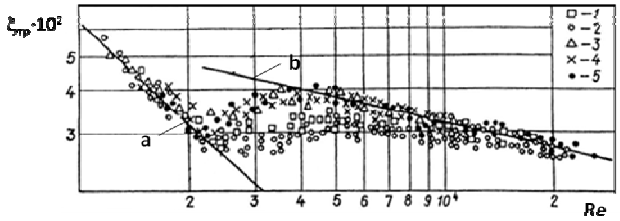


Figure 6 Relationship ξ_f=f (Re) for mounting gaps of rotor blade shanks in (1) and b (2), and, also, cylindrical capillary channels of 390 (3); 510 μm (4) and 2mm (5) at the adiabatic air flow.

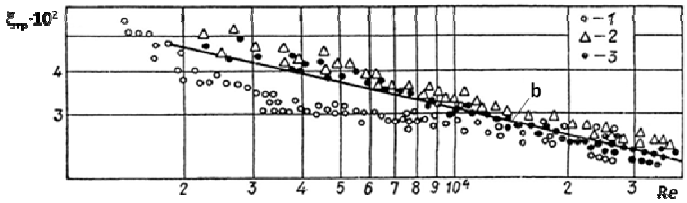


Figure 7 Relationship ξ_f=f (Re) for mounting gaps b (1) and cylindrical capillary of 0.51mm diameter (2) and 2mm (3) at the heat supply to air.

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).

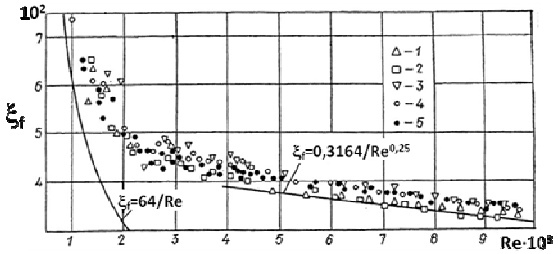


Figure 8 Relationship ξ_f=f (Re) for cylindrical channel at various turbulence: 1;2- Tu=1-1.5%; 3;4;5- Tu=5-18%.

Tu increase reduces the critical Reynolds number of the transition start from 2,300 to (1.7-1.8)10^3 and does not change ξ_f at the turbulent flow, starting at Re> 3,500. At 1,900> Re> 1,000, ξ_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 ξ_f, are high (Fig. 9) [**13**].

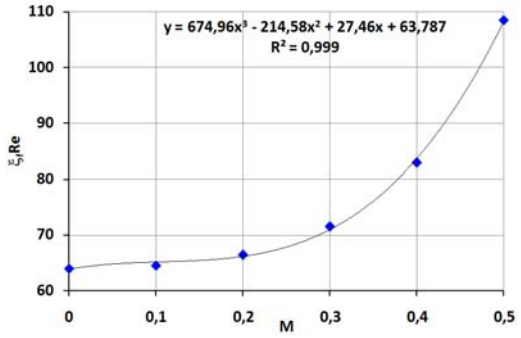


Figure 9 Effect of Mach number M on the product (ξ_fRe) at the laminar gas flow in μC [**13**].

It is seen that at the gas flow in the microtubes you should not neglect the influence of compressibility. The pressure variations in the gas flow in micro channel can be quite large, while the local Mach number increases with increasing the distance from the entrance, and can become very large even at a low Mach number at the entrance. This phenomenon is critical for the micro systems designing as the **micro channel choking** (M=1 at the channel exit) can occur with the gas flowing even when the input Mach number is much smaller than unity (M<<1.0). In micro devices, the gas flow velocity is associated with the diameter of the channel by the dependence, varying from cubic to linear, as the flow in the micro channel can vary from the subsonic to the sound value.

$$\xi_f = \xi_{f.0}(1+M^2/(1.5-0.66M-1.44M^2)), \quad (1)$$

where M is the local Mach number;

$$\xi_{f.0}=64/\mathrm{Re}. \quad (2)$$

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

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 $d_h >> \delta_b$.

However, for micro channels with a non-uniform roughness, as well as for "conventional" tubes, the calculation of the boundary values $Re_{c.1}$ and $Re_{c.2}$ can be performed using the formulas given in the handbook by Idelchik I.E. [**16**], according to which: $Re_{c.1}=1{,}160\ (\delta_s/d_h)^{0{,}11}$ with $(\delta_s/d_h)>0.007$; $Re_{c.2}=2090(\delta_s/d_h)^{0.0636}$ (for tubes with any roughness). In the transition zones of the Reynolds numbers, Celata [**8**] recommends the following relationship to calculate ξ_f in μC:

$$Re_0<Re<Re_{c1}\ \xi_f=4.4\ Re^{-0.595}\exp(-0.00275/(\delta_s/d_h)), \quad (3)$$

where $Re_0=1{,}900$;

$$Re_{c1}<Re<Re_{c2}\ \xi_f=(\xi_{f2}-\xi_{f1})\exp(-(Re_{c2}-Re))^2+\xi_{f1}. \quad (4)$$

Here:

at $(\delta_s/d_h)\le 0.007$

$$\xi_{f,1}=0.032;\ \xi_{f2}=7.244\ Re^{-0.643}; \quad (5)$$

at $(\delta_s/d_h)>0.007$

$$\xi_{f,1}=0.0758-0.0109/(\delta_s/d_h)^{0.286}; \quad (6)$$

$$\xi_{f,2}=0.145//(\delta_s/d_h)^{-0.244}. \quad (7)$$

At the self-similar turbulent condition, ξ_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, ξ_f depends on the Re number and the relative roughness; whereas at the transition to the turbulent condition ξ_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 ξ_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**].

2. CONVECTIVE HEAT TRANSFER in MICRO CHANNELS

Use of very small diameter channels with their relative length $(l/d_h)> 100$ leads to the appearance of the laminar flow condition, at which, given relatively low flow rates and hence small values of «D d_h/l» (Pe d_h/l<10, where Pe=RePr is the Peclet number; l is the length of the channel), the Nusselt number Nu ceases to depend on this complex and tends to a constant value Nu_{min} (Fig.10), that is, the convective heat transfer coefficient is independent of the inertial forces (flow rate of the working media) [**17-19**].

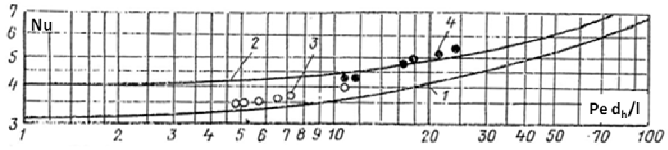


Figure10 Convective heat transfer in the micro channels under condition Pe d_g/l<10 for rectangular cross-section channels with various aspect ratio ψ=h/b (where h, b are thickness and width of the micro channel [**17**]).

Designations: 1,2 - ψ=1; 3 calculation; 3,4 - ψ=1;2,62 – experience [**18**].

The number Nu_{min} and the hydraulic resistance coefficient of friction ξ_f in this case have a significant influence on the cross-sectional shape of the channel. For the capillary tubes of circular cross-section, $Nu_{min,t}=3.66$, and the rectangular $Nu_{min,t}=7.5\text{-}8.24$ as function of the aspect ratio ψ=h/b. Here,

$Nu_{min,t}$ is the minimum Nusselt number for the constant wall temperature t_w. With a decrease in ψ, $Nu_{min,t}$ increases. Reduction of the complex "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 ξ_f , $Nu_{min.t}$ you can use the following approximation formulas:

$$\xi_f Re=57.736\ \psi^2-95.072\ \psi+ 94.792; \quad (8)$$

$$Nu_{min,t} =8.7699\ \psi^2-13.524\ \psi+ 7.8092. \quad (9)$$

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**].

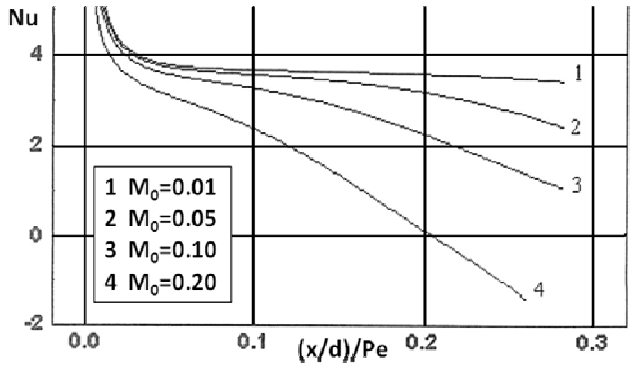


Figure 11. Effect of Mach number on the convective heat transfer at gas flow in the micro channels [**13**].

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. $T_{a.w}>T$ (curve 4, M=0.2).

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

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:

$$r = Rr^{0.33}-\Delta r_x. \quad (11)$$

Here:

$$\Delta r_x=7.16\cdot10^5 Re_x^{0.4}\ f(x/d_h); \quad (12)$$

x is the characteristic size in the Reynolds number;
$f(x/d_h)=1$ at x/d_h=0-15 (initial segment);

$$f(x/d_h)=1+0.0413(x/d_h-15) \text{ at } x/d_h=15\text{-}27\ [\mathbf{15,19}]. \quad (13)$$

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.

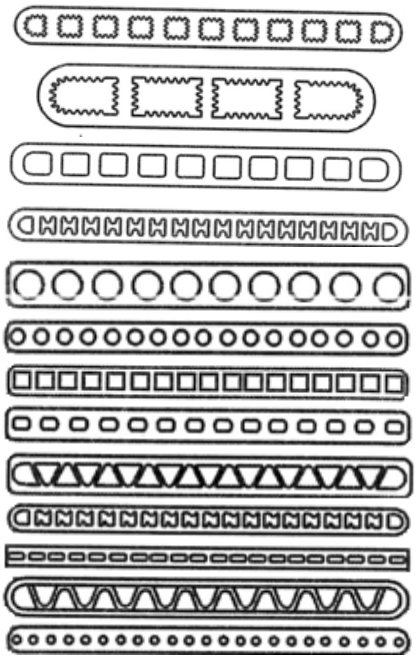


Figure 12 Shapes and sizes of the cross section of the micro channels in flat multi-channel tubes applied to manufacture miniature heat exchangers [**7**].

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=133-367μm. 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:

$$\mathbf{Nu=C_1\ Re^{0.62}Pr^{0.33}} \quad \mathbf{(14)}$$

or for an array of parallel rectangular **channels**

$$\mathbf{Nu=0.1165Re^{0.62}Pr^{0.33}\ (d_h/\ b)^{0.81}\ (h/b)^{-0.79}}, \quad \mathbf{(15)}$$

which is applicable at 900≥Re≥80 and 367≥d_h≥133μm.

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

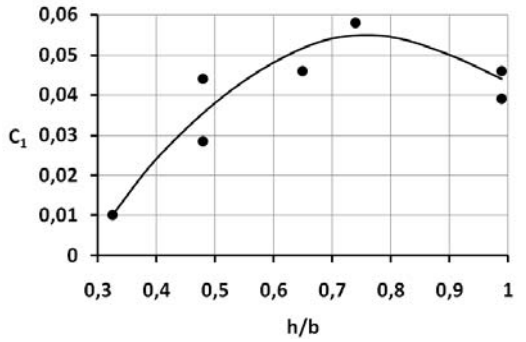


Figure 13 Effect of the relative height (h/b) and d_g on the convective heat transfer in the micro channels [**7**].

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.

Table 2

Geometries of studied micro channels and empiric coefficient in the criteria equation by Peng **[8]**

№	b, mm	h, mm	L, mm	d_h, mm	h/b	C_1
1	0.4	0.3	50	0.343	0.75	0.058
2	0.3	0.3	50	0.3	1.0	0.0384
3	0.4	0.2	50	0.267	0.50	0.0426
4	0.3	0.2	50	0.24	0.667	0.0472
5	0.2	0.2	50	0.20	0.1,0	0.0468
6	0.3	0.1	50	0.15	0.333	0.0104
7	0.2	0.1	50	0.133	0.50	0.0285

For conditions where Re≥1,000, Peng suggested a correlation $Nu=C_1\ Re^{0.8}Pr^{0.33}$ 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 mircochannels (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 (δ less than 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 ξ_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 δ=0.004 mm, while the relative roughness ratio is Δ=0.143...0.0615. With such Δ, greatly exceeding the limit value Δ_{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_{sur}=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 δ_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 ξ_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, ξ_f is 20...40% higher that calculated by the Hagen-Poiseuille's law ξ_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 ξ_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, $De=Re\sqrt{d_h/D_{ben}}=26\text{-}7{,}000$ and $(d_h/D_{ben})=6.2\text{-}62.5$, where D_{ben} is the bending diameter, the laminar flow with macro vortexes occurs, at which:

$$Nu=0.0575\ Re^{0.8}Pr^{0.43}\ (d_h/D_{ben})^{0.21}(Pr_f/Pr_w)^{0.25} \qquad (16)$$

For calculation of ξ_f, the Tschukin's formula can be used:

$$\xi_f=0.0385(D_{ben}/d_h)^{,.5}+0.312Re^{\mathbf{0.25}}\ \textbf{[22].} \qquad \textbf{(17)}$$

For the enhancement of the heat transfer, longitudinally profiled tubes with the "convergent-divergent" type configuration were applied. According to the Mighay's data [26], their thermal efficiency at $Re>10^4$ is significantly higher that achieved by other methods of improving the convective heat transfer in tubes. Systematic experimental studies of the convergent-divergent type models (CDC) and the divergent-convergent channels (DCC), carried out by E.F. Kuznetsov and B.P. Ivanov [25] in a wide range of the Reynolds numbers $5.0<Re<2\cdot10^5$, showed that for some CDC and DCC models within the zone of transition from the laminar to the turbulent flow, a 2.5-3 times growth of the convective heat transfer was obtained with a lagging resistance increase.

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.

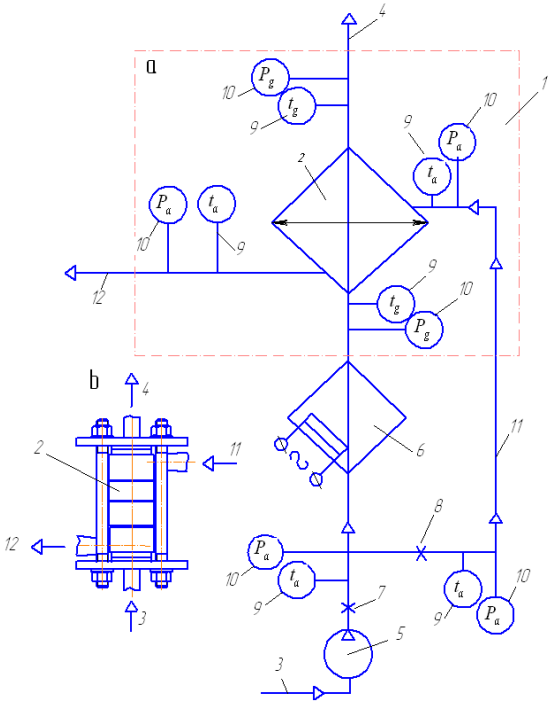


Figure 14 Schematic diagram (a) of the experimental setup for thermal-engineering test of μAH model (b)**.**

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 Q_g; Q_a; Δp_g; Δp_a in Figure 15 are compared with the calculated $Q=k\Delta TF$, 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_g; Δp_a in the matrix model paths are compared with calculated values determined using recommendations by I.E. Idelchik [**16**] for the "conventional" tubes.

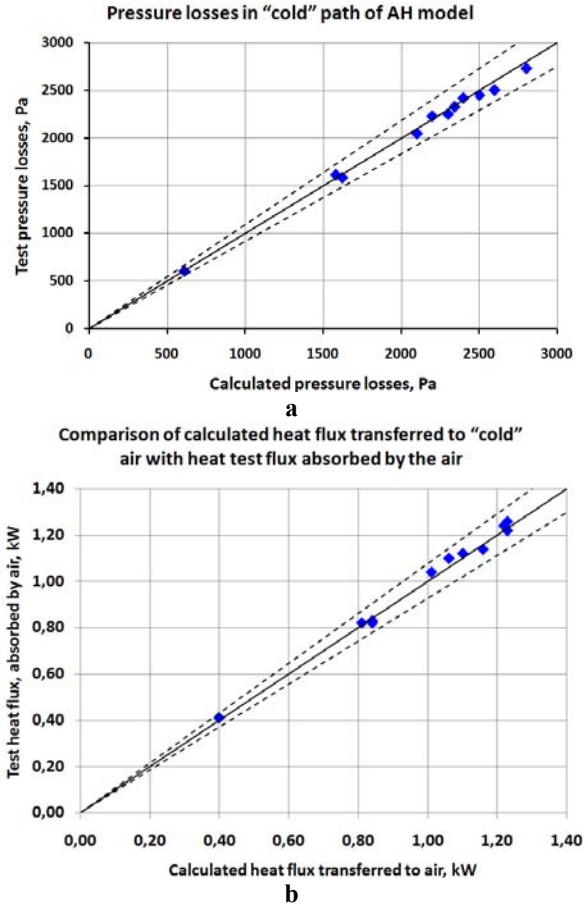


Figure 15 Comparison of test values Δp_g; Δp_a; Q_g; Q_a with calculated $\Delta p_{g,c}$; $\Delta p_{a,c}$ (a); Q (b) by recommendations [**1;16;19**].

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 $d_h >> \delta_b$
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 μC leads to a significant heat transfer enhancement.
9. Increasing the roughness of the μC 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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