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MICROCHANNEL HEAT EXCHANGERS – PRESENT AND PERSPECTIVES

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Lucrarea este o sinteză în domeniul metodelor de realizare şi a particularităţilor funcţionale şi de analiză teoretică ale schimbătoarelor de căldură cu microcanale (MCHEX). Sunt prezentate principalele contribuţii la dezvoltarea unor soluţii constructive şi tehnologii noi în acest domeniu. Sunt evidenţiate diferenţele ce apar la curgerea fluidelor şi transferul de căldură, între rezultatele calculelor teoretice şi cele experimentale pentru MCHEX. Este subliniată influenţa procesului de alunecare a vitezei la perete asupra curgerii şi transferului de căldură din MCHEX. Sunt subliniate avantajele asigurării unor sarcini termice foarte mari obţinute prin folosirea MCHEX. Aceasta justifică interesul în folosirea lor în sistemele termodinamice compacte, în general, şi în special la răcirea Sistemelor Micro-Electro-Mecanice (MEMS).

This paper is a synthesis of the design, operational and theoretical features of microchannel heat exchangers (MCHEX). The main contributions to the development of new design solutions and technologies in this field are presented. Differences by fluid flow and heat transfer between theoretical calculations and experimental results for MCHEX are shown. The influence of the wall velocity slip process on the flow and heat exchange in MCHEX is emphasized. The advantages of a very high thermal load using MCHEX are pointed out. This justifies the interest in their use, generally, in compact thermodynamic systems and, particularly, in cooling Micro-Electro-Mechanical Systems (MEMS).

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1. Introduction

In the last two decades, thermodynamic systems, either motors or thermal converters (cooling systems and heat pumps) have achieved better performances. This can be explained on one hand by the better knowledge of the processes that occur in these systems, thus allowing their design and functional optimization. On the other hand, their improvement comes from using new materials, especially for new types of compact heat exchangers which lead to enhanced heat transfer per unit volume and a higher heat transfer coefficient. In recent years, a very important contribution on increasing the performance of these systems, sometimes to levels once inconceivable, was obtained by using nano-technologies which have allowed the production of a new generation of compact heat exchangers with microchannels. Microchannel Heat Exchangers (MCHEX) have, according to the classification proposed by Kandlikar and Grande [1], a hydraulic diameter D_h=0.2….0.01mm involving the advantage of a large heat exchange surface in a very small volume. Also, at very small sizes, the processes of heat and mass transfer occurring in the dynamic and thermal boundary layers are very effective. These new types of heat exchangers provide high heat transfer coefficients and thus they are up to 45% more compact than the classic ones, at the same thermal performances [2, 3]. Due to high thermal performance, MCHEX are used increasingly in both single-phase (liquid or gas) and two-phase heat exchange (condensation - evaporation); while the disadvantage of higher pumping power is compensated by the lower scale and cost obtained in the case of improved series production based on nano-technologies series production improvement [4]. Using the MCHEX in vapor compression refrigeration systems, microchannel tubes having a lower internal volume will reduce the amount of refrigerant charged in the plant.

The fields of application, the design technologies, the modeling of the process flow and heat transfer in MCHEX are discussed in this paper.

2. Application fields and performances

It is well known that one of the cooling or rapid heating methods of a body with a fluid is the convective at the contact area between the subsystems. The heat flux transferred by convection is given by Newton's law:

$$\dot{Q} = \alpha\, A\, \Delta T_{s,f} \tag{1}$$

where: α is the convective heat transfer coefficient; A is the body surface in thermal contact with the fluid, and $\Delta T_{s,f} = \left(T_s - T_f\right)$ it is the difference between the body surface temperature (T_s) and the fluid temperature (T_f).

The eq. (1) was analyzed with emphasis on simultaneous increase of α and A, but, recently, restrictive conditions regarding the volume of the heat exchanger, have emerged. As a consequence the plate heat exchangers have appeared. These devices, using flow channel dimensions by the order of several millimeters, were originally used for gas applications and after a while for liquids. Design solutions with fins, including those with micro-fins, become prevalent for mono-phase and two-phase flow. The use of serpentine band and tracers in the fluid flow channel are just some of the commonly used methods to increase the local heat transfer, through uniform circulation, interruption of the dynamic boundary layer and increasing the micro-turbulence at the surface. In this context, in recent years the interest of specialists has increased in particular for MCHEX.

Cryogenic industry, due to heat exchangers efficiency reasons, was the first to use compact regenerative heat exchangers with hydraulic diameters less than one millimeter in size. Subsequently, the refrigeration industry obtained significant functional and economical advantages by adopting MCHEX [5, 6].

Today we are witnessing an exceptional development of micro-electronics (integrated circuits cooling – ICC and micro-electro-mechanical systems - MEMS), where the density of information and processor frequency is continuously increasing. In addition, the miniaturization trend leads to a strong decreasing of their performances due the overheating process. Thus, for heat flux densities exceeding 100W/cm^2, further cooling methods and adequate new types of heat exchangers, like the MCHEX, are required [7].

MCHEX have attracted attention because they have a high heat transfer area per operational volume unit, and thus, high heat transfer coefficients (at water flow 10 kW/m^{2}K, for D_h=0.2 mm, up to 100 kW/m^{2}K, for D_h=0.01 mm) and for very small dimension, they provide a very large heat flux transfer (20 kW in a cm^3 of device) [8].

However, due to the very small size of the hydraulic diameter, MCHEX have the disadvantage of high pressure losses (i.e. at water flow of 1 bar/m, for D_h=0.2 mm, up to 1000 bar/m, for D_h=0.01 mm), which requires the use of high pressures (70...110 bar) [9]. Under such circumstances it is clear that thermodynamic systems that use these types of devices must be designed accordingly in order to maintain tightness even at these high pressures. Today, another important disadvantage of using heat exchangers with microchannels is their high price, resulting in a limited use. But this disadvantage will be soon resolved along with the development of modern nano-technologies series production.

3. MCHEX technologies and design

3.1. Generalities on MCHEX technologies

The MCHX improvement was possible in the last two decades because of unprecedented development of materials and constructive solutions based on micro and nano-manufacturing technologies. Today, for the achievement of MCHEX, specialized equipment of great precision is used for casting and etching, laser processing and also pellicle metal deposits, silicon or other materials, such as polymers. Lately, new methods of design and manufacturing have been developed for these miniature exchangers, methods which, in the case of mass production, ensure also a lower price.

Traditional miniaturization technologies are the most accessible approach to produce micro-structures. These miniaturization techniques using traditional machine tools have been adapted to operate under miniature regimes. The domain of restricted use of machine tools becomes smaller as the lithography method is in expansion. Cutting was brought in micro area especially for cutting thin plate-shaped material. Thus, the channel width of $25\mu m$ with an accuracy of order $\pm 4\mu m$ can be obtained with commercially available equipment. For the production of microchannels, based on thin wire like electrodes, devices with micro-electro-discharge were used. Other technologies such as those with ultrasonic cutting and water jet machines were used for fragile materials involving microchannels with very small dimensions. Electro-forming, shaping and stereo-lithography were used at miniature scale by incorporating laser and lithography modeling technology. In this sense, the printing of integrated circuits, for example, is achieved usually with holes having the diameter of $25\mu m$ [9].

Modern technologies are based on the latest scientific advances in technology (laser and photo-lithography) and materials (semiconductors and polymers). Thus, machine tools with point by point processing ensure the reduction of energy consumption and material. Also, laser machines have become tools with very good productivity that can process a wide variety of materials. For example, focused ion beam machines offer many additional benefits and operate in class sizes below $1\mu m$.

3.2. Some details on Danfoss MCHEX

Thanks to the significant advantages obtained by using MCHEX systems, in the last years, more and more companies in the industry of refrigeration and air conditioning switched to the series production of these types of compact heat exchangers. Thus, starting from 2008 the Danfoss-Sanhua company has a series production of microchannel condensers for air conditioning, refrigeration systems, transportation, and others air cooling applications (Fig. 1) [10].

Fig. 1. Design of Danfoss MCHEX [10]

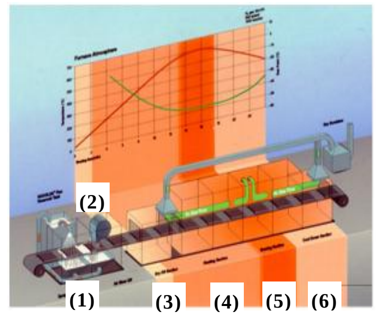

Fig. 2. Technology for Danfoss MCHEX [10]

The integrated technology assembly line of MCHEX is composed of the following sections: (1) spray of fondant flux unit, (2) air blowing section for degassing the fondant flux, (3) drying section, (4) heating section, (5) brazing soldering section and (6) cooling section, as shown in Fig. 2 [10]. In order to ensure a very good quality of the welding process by brazing, in the last three sections of the integrated technology assembly line an inert atmosphere of nitrogen (30 m^3/h of N_2 gas flux for a belt speed of 1 m/min) is used.

The main dimensions and the overall constructive scheme of a Danfoss MCHEX are presented in Fig. 3. A MCHEX is composed of many microchannel tubes that make the connection between the collecting ducts closed by end caps.

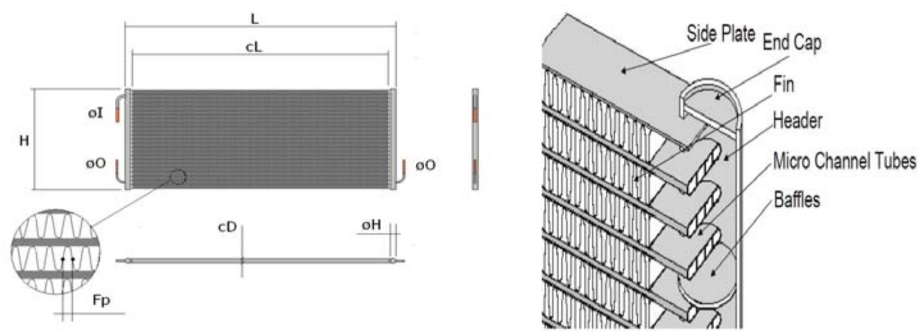

Fig. 3. Overall dimensions and design scheme for Danfoss MCHEX [10]
L - overall length; cL - core length; H - collector length; cD - microchannels tube depth; Fp - coil fin step; ϕH - collector diameter; ϕI - inlet diameter; ϕO - outlet diameter.

The arrangement of microchannel tubes in several passages is done by partition walls mounted inside the collecting ducts. To increase the heat exchange surface, on the side of the secondary air circuit, between the microchannel tubes wavy fin coils are mounted. At the top and bottom of the heat exchanger, additional wavy fin coils are provided using two side plates. The MCHEX components are assembled through a brazing soldering process which takes place at a temperature of 600°C, in conditions of lack of moisture (dew point is about -57°C) [10].

In the next section, according to the manufacturer's data sheet obtained at purchase, the main functional and constructive characteristics of the microchannel condenser type CHX C # A -1.3 X 16 - 2G16 (Danfoss) are presented:

a) *Secondary agent (air)*: volume flow rate $400[m^3/h]$; average speed $2.1[m/s]$; transversal flow area $0.053[m^2]$; pressure drop $21[Pa]$.

b) *Primary agent (R134a)*: mass flow rate $18[kg/h]$; condensing pressure $14.2[bar]$; pressure drop: $0.136[bar]$.

c) *General features of MCHEX*: condenser thermal load $0.907[kW]$; total length $291[mm]$; total height $226[mm]$; diameter of collector $20[mm]$; microtube active length $241[mm]$; microtube width $16[mm]$; microtube height $1.3[mm]$; number of microtubes per exchanger 23; number of microchannels per microtube 16; fin coil step: $1.1 \div 1.4[mm]$; fin coil height: $8.15[mm]$; fin coil thickness: $0.08[mm]$; deployed length per inch of wavy coil: $194.4[mm]$; cross-section flow area per microtube $7.4[mm^2]$.

The flow in this microchannel heat exchanger is optimized in four passes, with the following distribution of the number of tubes with microchannels per pass: 5 tubes/pass 1, 9 tubes/pass 2, 5 tubes/pass 3 and 4 tubes/pass 4 (Fig. 4).

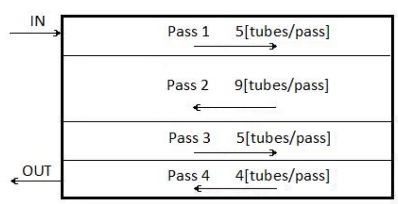

Fig. 4. Number of tubes with microchannels distribution with microchannels per pass for MCHEX type CHX C#A -1.3 X 16 - 2G16 (Danfoss)

Equivalent flow channel diameter on the refrigerant side estimation

The estimation of the flow equivalent diameter and the total internal heat transfer surface on the refrigerant side in the considered MCHEX (CHX C # A - 1.3 X 16 - 2G16) was based on technological data supplied by manufacturer (Danfoss), using the following method.

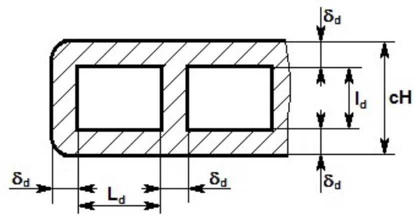

Fig. 5. Cross section through a tube with microchannels

If r_d is the ratio between characteristic dimensions of microchannel (Fig. 5):

$$r_d = L_\mathrm{d} / l_d \tag{2}$$

this is between $r_d = 0.5...2$ [1].

Using this notation, it results that the height of a microchannel is:

$$l_d = \sqrt{A_t / (n_{mc} \cdot r_d)} \tag{3}$$

where: A_t is the cross sectional flow area for all n_{mc} microchannels / microtube.

Consequently, it results that:

- thickness of the partition wall between two consecutive microchannels is:

$$\delta_d = (\mathrm{cD} - n_{mc} \cdot r_d \cdot l_d)/(n_{mc} + 1) \tag{4}$$

where cD is the width of the tube with microchannels.

- upper and lower wall thickness, considered to be equal in condition of equal mechanical strength, will be:

$$\delta_d = (\mathrm{cH} - l_d)/2 \tag{5}$$

where cH is the height of the tube with microchannels.

From equations (4) and (5), ranging ratio $r_d = 0.5...1.5$, and imposing the condition that the partition wall between two microchannels should be equal to the upper and lower wall thickness (tube of equal strength, Fig. 5), the optimum ratio is obtained $r_{d_{opt}} = 0.982$, which according to relations (2) and (3), corresponds to optimum values of microchannel size: $L_{d_{opt}} = 0.674[\mathrm{mm}]$ and $l_{d_{opt}} = 0.686[\mathrm{mm}]$.

Thus, the equivalent diameter of the refrigerant flow through a microchannel results:

$$d_{eq}^{R134a} = 4A_{mc} / P_{wet} = 2L_{d_{opt}} l_{d_{opt}} / \left(L_{d_{opt}} + l_{d_{opt}}\right) = 0.680\,[mm] \tag{6}$$

where: A_{mc} and P_{wet} are the cross section area, and respectively, the wetted perimeter of one microchannel on refrigerant side.

Based on the dimensions of the microchannel, estimated above one can calculate the total inner heat exchange surface of the microchannel tubes with:

$$A_{in}^{mc} = n_{mc} \cdot P_{wet} \cdot cL \cdot n_{tub} = 2n_{mc}(L_{d_{opt}} + l_{d_{opt}}) \cdot cL \cdot n_{tub} = \mathit{2512.78}\ [cm^2] \tag{7}$$

where, n_{tub} is the number of microtubes per heat exchanger.

The inner surface of two collectors of length H and diameter ϕH (Fig. 3) is:

$$A_{in}^{col} = 2\pi \cdot \phi H \cdot H + \pi \cdot \phi H^2 = 252.40 \ [cm^2] \tag{8}$$

Thus, the total inner heat exchange surface will be:

$$A_{in} = A_{in}^{mc} + A_{in}^{col} = 2765.18\,[cm^2] \tag{9}$$

The total inside volume of the heat exchanger, depending on the inner volume of the microtubes (V_{in}^{mc}) and the collector's volume (V_{in}^{col}), will be:

$$V_{in} = V_{in}^{mc} + V_{in}^{col} = A_t \cdot cL \cdot n_{tub} + \pi \cdot \phi H^2 H/2 = 146.132\,[cm^3] \tag{10}$$

Equivalent flow channel diameter on the air side estimation

To estimate the equivalent flow channel diameter and the total external heat transfer surface on the air side, for the considered MCHEX (CHX C # A -1.3 X 16 - 2G16), the calculating scheme of the heat exchanger air flow comb from Fig. 6 and technological data provided by the manufacturer (Danfoss) are used. The following method is proposed:

If the coil fin step is $Fp = 1.1...1.4[mm]$, given by the manufacturer, then the average number of half-loop, considering the extreme values is:

$$n_{hl} = (cL/1.1 + cL/1.4)/2 \approx 196 \tag{11}$$

where cL is the length of the tube with microchannels in the active area.

The value of n_{hl} corresponds to the average step: $Fp_m = 1.232[mm]$.

To calculate the length of one fin, for one step Fp_m (half-loop of coil fin of height *Hp*), the fin loop length is approximated to be equal with the diagonal corrected length of the formed rectangle (Fig. 6).

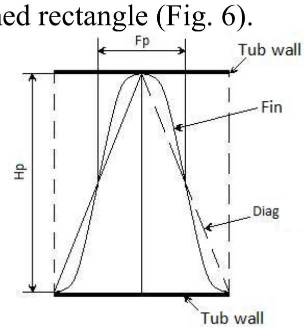

Fig. 6. Calculating scheme for the air flow comb through MCHE

$$Diag = \sqrt{Hp^2 + Fp_m{}^2} \cdot C_{cor} = 9.434[mm] \tag{12}$$

where: $C_{cor} = 1.1445$ is a correction factor that takes into account the fact that, in reality, fin length for a half-loop is greater than the diagonal length. The value chosen for the correction factor is derived from the coil deployed length condition ($194.4mm$), per each inch ($25.4mm$) of real coil, given by the manufacturer.

Thus, it results that the total length of a developed fin is:

$$Fl = Diag \cdot n_{hl} = 1.845[mm] \tag{13}$$

Wetted perimeter of a half-loop coil is calculated with the equation:

$$P_{whl} = 2(Diag + Fp_m) = 21.331[mm] \tag{14}$$

and the total wetted perimeter for a coil will be:

$$P_{wc} = P_{whl} \cdot n_{hl} = 4.173[mm] \tag{15}$$

So, the total wetted perimeter for the entire heat exchanger, considering the upper and lower coils outside the tubes is:

$$P_{wex} = P_{wc}(n_{hl} + 1) = 100.146[mm] \tag{16}$$

Based on the total wetted perimeter estimated through this method with relation (3.15), one can estimate the total heat exchange surface on air side:

$$A_{ex} = P_{wex} \cdot cD = 1.602[m^2] \tag{17}$$

where, cD is the microtube width.

Finally, considering the total wetted perimeter for air flow through the comb of the microchannel heat exchanger, estimated with the relation (17) and the value of cross-section flow area given by the manufacturer, the equivalent diameter of the air flow through the heat exchanger results as:

$$d_{eq}^{aer} = 4A_{ex}/P_{wex} = 2.117\,[mm] \tag{18}$$

4. Functional features and theoretical analysis

4.1. Theoretical vs. experimental results in MCHEX

The main difficulty in the design of MCHEX is the limited knowledge involving engineering methods at this scale. For using MCHEX new methods of calculation for accurate estimation of heat transfer and friction coefficient in single-phase or two-phase flow through channels with hydraulic diameter of $D_h = 10...200\ \mu m$ are needed.

Many of the papers published so far have shown that experimental data for heat transfer and pressure drop during the flow through microchannels differ significantly from the results obtained with the calculation methods used for macro-scale. Thus, Fig. 7 (a) presents heat transfer data obtained in microchannels for single-phase laminar and turbulent gas flow compared with results obtained using classical calculation relations for conventional size channels.

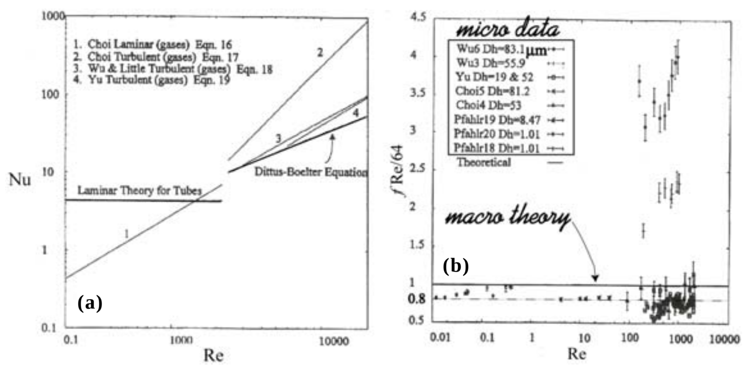

Fig. 7. Single-Phase Microchannel heat transfer and pressure drop data vs. Reynolds numbers from Literature [11]; *(a) Nusselt number; (b) friction coefficient.*

It is noted that for Reynolds numbers lower than 200, experimental values of Nu number and of heat transfer, are 5 times smaller than the solutions obtained for the macro laminar regime. In contrast, for turbulent flow with Reynolds numbers greater than 10000, the Nu number values were up to 4 times higher than the results obtained using the Dittus-Boelter classical relationship. Fig. 7 (b) shows that experimental values of friction factor in laminar regime are 20% lower than those obtained using classical calculation relations. This shows that the experimental values obtained for the friction factor in the turbulent regime are very different from those predicted by macro-theory (either 50% lower, or 300% higher) [11].

The different influences on the energy conservation and momentum equations, arising from fluid flow and heat transfer, which, for heat exchangers with conventional size channels are neglected, become important with the diminution of channel diameter. Thus, in case of MCHEX, the persistent differences between the calculation results based on macro-theory and the micro scale experimental results can be explained and corrected by taking into consideration the influence of the following processes: viscous dissipation, fully developed flow, intermittent flow (compressible or with rarefaction), and the wall conditions (constant or variable temperature and, respectively, constant flow).

Other influences, such as sliding speed, temperature jump, surface roughness, irregular distribution of flow through microchannel networks, the micro-turbulent effect, electric double layer, the use of composite materials and the variation of viscosity with temperature and channel size, play an important role in the processes taking place in miniature systems [12].

From the multitude of processes and boundary conditions that influence the fluid flow and heat transfer in MCHE, we will analyze the influence of wall flow slip, as one of the important processes.

4.2. Wall velocity slip process in MCHEX

In MCHEX the molecular mean free path (λ) becomes comparable with the characteristic dimensions of the fluid flow and collisions between fluid molecules and wall molecules are more numerous than those inside the fluid. In this case, when the fluid molecules strike the solid surface, they can either be reflected normally, preserving the tangential momentum, or can be diffusively reflected. Thus, due to diffuse reflection, unlike macro-flows, where the fluid on the wall has zero velocity ($u_w = 0$), in the case of microchannels flow the fluid will slip from the wall ($u_w > 0$). This phenomenon is shown in Fig 8.

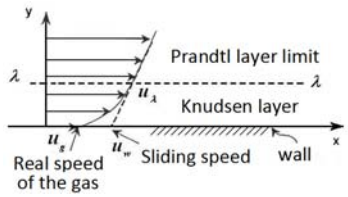

Fig. 8. The occurrence of slip velocity on fluid flow through microchannel

The characteristic size of the channel, relative to the molecular mean free path determines the momentum equation that models the flow as in this case. Calculation errors caused by slip flow occur when the size of channel diameter approaches the size of molecular mean free path. It is defined as a dimensionless criterion called Knudsen number (Kn), the ratio of molecular mean free path (λ) and channel characteristic size (D_h), which is the minimum of the two sizes of a rectangular channel (H), and for circular channel it is the diameter (*D*):

$$Kn = \lambda / D_h \quad [-] \tag{19}$$

For a perfect gas the molecular mean free path can be calculated as a function of pressure (p) and temperature (T) with the formula: $\lambda_{gp} = k_b T / \pi\sqrt{2} d^2 p$, where k_b is the Boltzmann constant ($1.38 \times 10^{-23} J/mol \cdot K$), and d it is the diameter of the molecule. Also for the calculation of molecular mean free path the relationship $\lambda = \mu\sqrt{\pi} / \rho\sqrt{2RT}$, is also recommended which depends on dynamic viscosity μ and gas density ρ [13].

Knudsen criterion (Kn) is used to define boundary conditions for each regime. Continuous flow, where the velocity variation law is continuous, starting from zero, at the wall ($u_w = 0$), is noted for $Kn < 0.001$ [14]. In this flow regime, for gases (He, Ne, Ar, H_2, N_2, O_2, CO_2) and vapors (NH_3, CH_4, air), in tubes with larger diameters than $200 \mu m$, the classic Navier-Stokes equations are valid. In the other flow regimes, where the frequency of interactions between fluid molecules and the wall is reduced, the fluid slip occurs at the wall, and the flow is with rarefaction. In this case, the size of channel hydraulic diameter (D_h) approaches the average molecular path (λ) and the information regarding the existence of the wall is not passed quickly between fluid molecules due to lack of local equilibrium between wall and fluid, resulting in a sliding speed $u_w > 0$ at wall (Fig. 8).

Slip flow regime should be reached at $0.001 < Kn < 0.1$ where the flow can be modeled by Navier-Stokes equations corrected with boundary conditions. Fluid flow in tubes with diameters within the range $0.5...200\ \mu m$ is between continuous and slip flow. For $0.1 < Kn < 3$, the flow is with total slip, case in which the Navier-Stokes equations are no longer valid; in this type of flow, the Boltzmann equation can be used, which are based on statistical distribution of a single particle velocity. Gas flow in tubes with diameters of approximately $1...2\ \mu m$ corresponds to this flow regime. For flow regimes with $Kn > 3$, a free molecular flow is obtained and the tube wall has no influence on the fluid speed distribution.

Knudsen criterion can be expressed in terms of two dimensionless numbers from fluid mechanics, Reynolds (Re) and Mach (Ma) [15] with the relation:

$$Kn = \sqrt{\frac{\pi\gamma}{2}}\frac{Ma}{\mathrm{Re}}\ ; \left(\mathrm{Re} = \frac{wL}{\nu};\quad Ma = \frac{w}{a}\right) \tag{20}$$

where: γ, w, ν and a are the adiabatic exponent, the average flow velocity on the line L, kinematic viscosity and sound speed corresponding to fluid flow conditions.

To calculate the relative sliding speed gradient $(u_w - 0)/\bar{u}$, where $\bar{u}$ is the fluid average velocity in the cross section gas flow through cylindrical micro-tube having the diameter D, Maxwell proposed the following expression, based on the ventilation friction factor (f) and Reynolds number (Re) [15]:

$$\frac{u_w}{\bar{u}} = \lambda\frac{\rho\bar{u}}{\mu}\frac{f}{2} = \lambda\frac{\mathrm{Re}}{D}\frac{f}{2} \tag{21}$$

where μ and ρ are the dynamic viscosity and, respectively, the density of fluid.

Using Maxwell's approximation of the molecular mean free path for rarefied non-polarized gas, such as superheated vapors, and the friction factor expression for laminar ($f = 16/\mathrm{Re}$) and for turbulent ($f = 0.079\,\mathrm{Re}^{-0,25}$) regimes, in macro-flows [14], for relative sliding speed gradient of gas flow through cylindrical micro-tube, eq. (21), on obtain the relations:

- for turbulent flow: $$\frac{u_w}{\bar{u}} = 0.0334\cdot\mu_v\,\mathrm{Re}_v^{0,75}/a\,\rho D \tag{21.a}$$

- for laminar flow: $$\frac{u_w}{\bar{u}} = 12\mu_v/a\,\rho D \tag{21.b}$$

Approximating the molecular mean free path with 1 Ångström, similar expressions for relative speed gradient for liquid flow through micro-cylindrical tube can be obtained [15]. Thus, for turbulent flow it results:

$$\frac{u_w}{\overline{u}} = (0.0395 \times 10^{-10} m) \frac{\mathrm{Re}^{0,75}}{D} \tag{21.c}$$

and for laminar flow it is obtained:

$$\frac{u_w}{\overline{u}} = \frac{8 \times 10^{-10} m}{D} \tag{21.d}$$

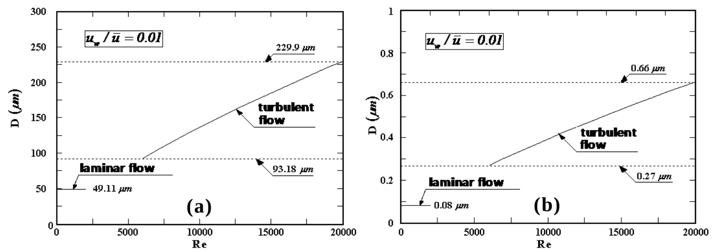

Fig. 9. The diameter of tube where the sliding speed is 1% of the average velocity
(a) for air at 1 bar and 0 °C; (b) for water

Fig. 9, based on relations (4.5.a ÷ 4.5.d), presents the tube diameter values $D_{1\%}$ where the relative sliding speed on wall is $u_w/\overline{u} = 1\%$, for different Reynolds numbers corresponding to laminar and turbulent flow of air and water.

Thus, for laminar flows of air at atmospheric pressure and 0°C, it results that the slip velocity is negligible for tubes with diameters larger than $49.11\,\mu m$. The $D_{1\%}$ diameters for air turbulent flows, are more than double than those obtained for laminar flow and are a function of Reynolds number: $93.2...230\,\mu m$ for $\mathrm{Re} = 6000...20000$. The $D_{1\%}$ diameters for water flow are approximately two orders of magnitude smaller than the diameters for air flow. For example, for water laminar flow $D_{1\%} = 0.08\,\mu m$, while for turbulent flow $D_{1\%} = 0.269...0.664\,\mu m$ for an $\mathrm{Re} = 6000...20000$.

Based on the tangential momentum balance equation at the wall boundary, one can obtain the local expression of fluid flow velocity even in the case of slip flow in microchannels. Thus, for a fully developed laminar flow, in a rectangular tube with height $2H$, the law of velocity variation in cross section, depending on the relative coordinate of calculation position ($\eta = y/H$), can be obtained from the momentum equation as [16]:

$$u = 3/2 \cdot \overline{u} \cdot (1 - \eta^2 + 4Kn) \big/ (1 + 8Kn) \tag{22.a}$$

and respectively, for a circular pipe, of radius R, depending on the relative coordinate position of calculation ($\eta = r/R$), with the expression [16]:

$$u = 2 \cdot \overline{u} \cdot (1 - \eta^2 + 4Kn)\big/(1 + 8Kn) \tag{22.b}$$

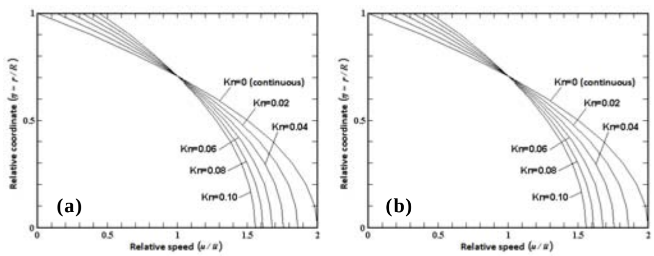

Fig. 10. Laws of relative sliding velocity variation for gaseous flow for different values of Knudsen number *(a) through rectangular micro-channels (equation 22.a)* *(b) through circular micro-channel (equation 22.b)*

Based on the relations (22.a) and (22.b), Fig. 10 (a) and (b), presents the variation laws of relative sliding velocity for gaseous flow through rectangular and circular microchannels for different values. Of Knudsen number from both figures it results that at the wall, where $\eta = 1$ ($y = H$, for rectangular channels, respectively, $r = R$, for circular channels), in case where $Kn = 0$, gas velocity is zero, and thus in this case there is no slip flow. But, as the number Kn increases, the wall slip phenomenon emphasizes and sliding speed increases. Also, the slip phenomenon is stronger in circular microchannels than in rectangular ones.

Thus, for $Kn = 0.1$, the sliding speed is 35% of the average speed for flow through rectangular channels, and 45% of average speed for circular flow channels. The flow velocity in the tube axis ($\eta = 0$), due to the slip flow phenomenon, decreases by 22% with the increase of $Kn = 0...0.1$.

5. Conclusions

In Micro-Channels Heat Exchangers (MCHEX) the use of reduced size channels has led to large heat transfer areas per volume unit and a higher heat transfer coefficient. With these features, one can obtain the transfer of heat fluxes greater than the order of several hundred of W/cm^2.

In terms of production, the miniature systems of MCHEX having hydraulic diameter of D_h=0.2...0.01 mm were considered. In the last two decades, this size range showed a particular interest for research on heat transfer in micro-electronics. In the same period, manufacturing technologies have focused attention on this size range. Today, for the production of MCHEX a wide variety of optimized processes and specialized machinery and tools are available, some derived from traditional processing, others derived from the semiconductor industry and some are ingenious adaptations of some technologies from other fields. In terms of current production possibilities, it appears that there are no dimensional limits for these heat exchangers.

Differences, for heat transfer and pressure loss, between calculations based on classical relations and results of measurements performed in microchannels with hydraulic diameter of $D_h = 20...300 \mu m$ are due to several factors acting simultaneously. Some of these factors may be new microchannel phenomena, such as vortex initiation and transition to turbulent flow for Re numbers lower than those corresponding to the flow through conventional size tubes. Other factors may come from familiar phenomena, which are usually neglected for flows and heat transfer through tubes with conventional dimensions, such as flow slip, viscous dissipation and compressible flow.

Thus, together with viscous dissipation, compressible flow and temperature jump, slip flow (speed jump) is an important factor. Considering the importance of viscous dissipation effect, speed and temperature jump, in the future it is needed to develop some methods of calculation of heat exchangers with micro-channels that take into account these processes.

In general, it can be said that the MCHEX are only at the beginning of their development. We are still in the phase to attempt a better understanding of their characteristics. In the near future with the development of MEMS, of sensors and of miniature automation devices, of exhaust systems for high heat transfer rates and biomedical applications, it is expected that the use of the MCHEX sees an unprecedented development.

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