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Fluid Flow and Heat Transfer for Microtechnologies

Abstract – The objective of the present paper is to provide a general overview of the research carried out so far in single-phase heat transfer and fluid flow in capillary (micro) pipes. Laminar flow and laminar-to-turbulent flow transition are analyzed in detail in order to clarify the discrepancies among the results obtained by different researchers.

Experiments performed in the ENEA laboratory indicate that in laminar flow regime the friction factor is in good agreement with the Hagen-Poiseuille theory for Reynolds number below 600-800 for rough pipes (typically stainless steel pipes). For higher values of Reynolds number, experimental data depart from the Hagen-Poiseuille law to the side of higher friction factor values.

In case of smooth pipes (typically fused silica pipes) the agreement can hold up to Reynolds number equal to 2000.

Heat transfer experiments show that heat transfer correlations in laminar and turbulent regimes, developed for conventional (macro) tubes, are not properly adequate for heat transfer rate prediction in microtubes.

Key words: microscale, heat transfer, fluid flow.

INTRODUCTION

Microsystems technology is gaining more and more interest in the scientific and industrial communities. Recently published articles concerning possible future applications of micro technologies predict a big commercial impact on nearly all branches of industry.

The main users of microsystems are industries in the field of microelectronics, chemical, pharmaceutical, and medical technology, included the automotive and aerospace companies. Typical microdevices are cardiac pacemakers, pressure sen-

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sors, accelerometers, inkjet heads, components for telecommunications and components for microelectronics. Generally, these components are characterised by reduced dimensions and increasing power generation. Therefore, the need to transfer high heat fluxes in relatively small surfaces and volumes brings new challenges in heat removal and cooling techniques.

In the present context, a new generation of micro thermal devices have been developed and are gaining importance. These micro thermal devices range from compact heat exchangers for air conditioning and refrigeration systems, to cooling elements for electronic components, portable telephones and computers, and aerospace avionics. Heat removal enhancement is obtained through the increase of heat transfer coefficient and of the heat exchange surface with decreasing channel hydraulic diameter. In particular, the hydraulic diameter in these micro thermal devices ranges from 1 μ m upwards to 2 mm, significantly smaller than macroscale channels that are on the order of 5.0 to 50 mm.

Generally, the classical thermal and fluid dynamic theories developed for "macro systems" are not applicable to fluids in microscale structures. Thermofluid-dynamic phenomena that are generally not noticeable in macro systems, may play a dominant role in microsystems with the consequence that a lot of open questions exist and the few experimental results available in literature reveal that there are still important differences in the conclusions. Therefore, conventional macroscale methods are generally not satisfactory for thermal design of micro thermal devices in single-phase and two-phase flows.

Despite the large industrial interest and demand for thermal-fluid dynamics applications in microgeometries, from a technical point of view we have to face lack of suitable and reliable design methods. There is a dearth of heat transfer and pressure drop correlations which are valid (reliable) for microchannels with an inner diameter smaller than one millimeter. Furthermore, from the scientific point of view, we have to record a scarcity of knowledge of the basic mechanisms of heat transfer at reduced geometric scales. Unfortunately, at least at the present stage of research, the knowledge accumulated on large scale pipes in the past years can hardly be extended to microchannels.

Fluid flow - State of the art

A good review of single-phase fluid flow published results has been recently presented by Obot [2000] both for micropipes and microchannels and by Celata [2004] for micropipes.

Mala and Li [1999] investigated experimentally the flow characteristics of water in stainless steel and fused silica microtubes with diameters ranging from 50 to 254 μ m (L/D = 1200-5000). Especially for smaller pipe diameters experimental results on friction factor show a significant departure from the conventional theory (higher values). Authors found an early transition from laminar to turbulent flow in the Reynolds number range 300-900, while the flow changed to fully developed turbulent flow at Re \geq 1000-1500. Authors explained the unusual phenomenon introducing the roughness viscosity model which takes into account for increase of the momentum transfer in the boundary layer near the wall due to the presence of roughness. Experimental results by Mala and Li [1999] are plotted in Fig. 1, where the pressure gradient along the pipe is plotted against Reynolds number for stainless steel (top graph) and fused silica (bottom graph) micropipes. Average roughness of the pipes is 1.75 µm, although authors do not provide information for each single pipe.

Brutin and Tadrist [2003] performed accurate measurements of friction factors in micropipes ranging from 50 to 530 μ m in diameter fused silica capillary tube in laminar flow, using water. Authors found the friction factor higher or slightly higher than that predicted by the Poiseuille law, the discrepancy increasing as the microtube diameter decreases. Authors claim an accurate measurement of microtube geometry.

Li et al. [2000] studied the frictional characteristic of water flowing in capillary tubes with diameters ranging from 80 μ m to 205 μ m. Three different material (glass, silicon, and stainless steel) were employed in the experiments in order to verify the effect of surface roughness on hydraulic characteristics. The experimental results with smooth tubes made of glass and silicon showed that in laminar flow the friction factor was in good agreement with the classical theory. Figure 2 shows the friction factor versus Reynolds number for these smooth pipes, evidencing the fairly good agreement with the Hagen-Poiseuille theory. The experiments with stainless steel tubes, characterized by higher relative roughness, revealed a product fRe 15% higher than 64 in laminar flow. The transition from laminar to turbulent flow occurred for Re ranging from 1700 to 2000 and no early transition was observed.

Xu et al. [2000] performed experiments of water in microchannels with hydraulic diameter ranging from 30 μ m to 344 μ m and Reynolds number ranging from 20 to 4000. Authors found that friction factor agreed with conventional theories predicted by Navier-Stokes equations.

Judy et al. [2002] performed very accurate experiments on frictional characteristics of fluid flow in microtubes of fused silica and stainless steel. The capillary diameters varied from 15 up to 150 μ m and three different fluids (water, methanol, isopropanol) were tested. The authors concluded that the experimental results were in good agreement with the classical theory. A careful analysis of the experimental uncertainty reveals that error bounds are dominated by measurement of the diameter (in their case measured using a scanning electron microscope, SEM). Authors also found no evidence of transition to turbulent flow in the range Re \leq 2000 as reported by other works.

Yang et al. [2003] using water and R134a in smooth tube (173 μ m) for 350 < Re < 2300 verify that friction factors agree very well with the conventional Poiseuille equation. The laminar-turbulent transition Reynolds number varies from 1200 to 3800 and increases with decreasing tube diameters.



Fig. 1 - Pressure gradient versus Reynolds number for stainless steel (top graph) and fused silica (bottom graph) micropipes with different diameter. Mala and Li [1999].



Fig. 2 - Friction factor vs. Reynolds number for water flow in smooth microtubes. Li et al. [2000].

Yen et al. [2003] using HCFC123 and FC72 in 0.19, 0.3 and 0.51 mm ID tubes claim that the pressure loss characteristics are found to be qualitatively in accordance with the conventional correlation formula.

Lelea et al. [2004] using water in 0.1, 0.3 and 0.5 mm stainless steel tubes for 50 < Re < 800 conclude that experimental results of pressure drop obtained in their experiment confirm that, including the entrance effects, the conventional or classical theories are applicable.

Sharp and Adrian [2004] used water, 1-propanol, and a solution 20%(wt) of glycerol in water, testing glass tubes (50-247 μ m) for 20 < Re < 2900. The pressure drop results confirm the macroscopic Poiseuille flow result for laminar flow resistance for Re < 1800.

Yu et al. [1995] reported the fluid flow characteristics of water in microtubes with diameters of 19, 52, and 102 μ m and 250 < Re < 20000. In laminar flow the friction factor was lower than that predicted by the Navier-Stokes equations (fRe \approx 50), while the laminar-to-turbulent flow transition occurred in the range 2000 < Re < 6000.

Xu et al. [1999] performed experimental and theoretical investigations on water flow in microchannels with hydraulic diameters ranging from 50 μ m to 300 μ m and 50 < Re < 1500. Experimental data revealed that flow transition did not occur in this Reynolds number range. Authors observed that the flow characteris-

tics deviated from traditional theory when the channel dimensions are below 100 μ m. The friction factor was smaller than that predicted by the Hagen–Poiseuille law.

Judy et al. [2000] investigated the frictional characteristics of water, hexane, and isopropanol flowing in fused silica capillaries. The capillary diameters were in the range 20 μ m to 150 μ m. Authors found that for tube diameter lower than 100 μ m friction factor in laminar flow deviated from classical theory, significantly. The deviation is independent of Re depending on the tube diameter. In particular, friction factor was lower than expected and the deviations were lower as the tube diameter decreased, reaching 30% deviation from classical theory for 20 μ m tubes. Figure 3 shows a plot of friction factor versus Reynolds number from laminar to turbulent flow. We can observe lower values of friction factor for laminar flow, while in the turbulent flow region experimental data are within the prediction provided by the Blasius law for smooth pipes.

Other papers from the literature, e.g., Choi et al. [1991], Peng and Wang [1993], Wang and Peng [1994], Peng et al. [1995], and Peng and Peterson [1996], Pfund et al. [1998], and Agostini et al. [2002], among others, investigated single-phase fluid flow in microchannels, finding similar results as those above described for micropipes.

Summarizing the brief description of experimental findings, we can see that some authors found a friction factor in laminar flow significantly higher than that



Fig. 3 - Friction factor versus Reynolds number for water, hexane, and isopropanol flowing in fused silica capillaries. Judy et al. [2000].

predicted by classical theory, with the product fRe higher than 64 for circular microtubes. Some other authors reported friction factors in good agreement with theoretically predicted values, while others reported friction factors lower than the classical theory prediction. This would suggest general disagreement among the researchers on the hydraulics results in terms of friction factor. Generally, though not always true, researchers tend to agree that rough pipes show a friction factor in laminar flow which tends to be higher than classical theory prediction, while hydraulically smooth pipes exhibit friction factors in good agreement with Poiseuille law. On the other hands, the importance of surface roughness on heat transfer and pressure in small pipes has been recently outlined by Kandlikar et al. [2001] for pipe diameter below 0.6 mm. For smaller pipes, D < 100 μ m, there are some contradictory results even for *smooth* pipes. Going down to very small pipe diameter, the experimental uncertainty play an enormous role, being the accuracy and precision of all parameters, and especially the pipe diameter, the most important achievement to thoroughly pursue in the research.

HEAT TRANSFER - STATE OF THE ART

Very few work has been done on single-phase (liquid) heat transfer in micropipes. The most updated state-of-the art review has been prepared by B. Palm [2001], and most of the material reported here is from his courtesy. Other authors proposed interesting reviews on single-phase (liquid) heat transfer in micropipes and/or microchannels, Obot [2000] and Rostami et al. [2000]. The general trend of heat transfer results in microchannels is not so far from that here under described for microtubes.

Before going into details with the review of the current research, it is of interest to quote here, as a reference point to give the baseline values for comparison with the micropipes, the most widely used correlations of heat transfer for laminar and turbulent flow. They are Hausen [1959] correlation for laminar flow, and Dittus-Boelter [1930] and Gnielinski [1976] correlations for turbulent regimes.

The Hausen correlation was developed for long tubes at constant wall temperature:

Nu = 3.66 +
$$\frac{0.19 (\text{Re Pr D/L}_{h})^{0.8}}{1 + 0.117 (\text{Re Pr D/L}_{h})^{0.467}}$$
 (1)

The Dittus-Boelter correlation (valid for Re>10.000) is:

$$Nu = 0.023 \ Re^{0.8} \ Pr^{0.4} \tag{2}$$

The Gnielinski correlation is given by:

$$Nu = \frac{(f/8) (Re - 1000) Pr}{1 + 12.7 (f/8)^{0.5} (Pr^{2/3} - 1)}$$
(3)

where f is the friction factor calculated with the Filonenko [1954] equation:

$$f = 1.82 \text{ Log } (\text{Re} - 1.64)^{-2} \tag{4}$$

Gnielinski correlation may also be used for the transition regime 2300 < Re < 10000.

Yu et al. [1995] tested the heat transfer of water in micropipes with diameters of 19.6, 52.1 and 102 mm. For Reynolds numbers from 6000 to 20000 they suggested the following correlation based on their results:

$$Nu = 0.007 \text{ Re}^{12} \text{ Pr}^{02}$$
(5)

This indicates Nusselt numbers between 8 and 15 times higher than the Gnielinski correlation. Authors presented a theory for turbulent flow and tried to relate the increase in the heat transfer in micropipes to the higher frequency of *bursting* event in the laminar sublayer. In the turbulent flow regime authors also observed a reduction in the friction factor. In laminar flow the heat transfer rate is similar to conventional pipes.

Adams et al. [1998] performed an experimental investigation of heat transfer characteristics of water in microtubes ranging from 0.76 to 1.09 mm. They found that the experimental Nusselt numbers were higher than those predicted by conventional heat transfer correlations. The authors observed that the extent of the heat transfer enhancement (defined as deviation from the conventional theory) increased as the channel diameter decreased and Reynolds number increased. They proposed the following generalized correlation based on their results and those by Yu et al. [1995] (diameter 102 μ m) for the Nusselt number in turbulent flow in circular channels:

$$Nu_{Adams} = Nu_{Gnielinski} \left(1 + F\right) \tag{6}$$

where F is

$$F = 7.6 \ 10^{-5} \ \text{Re} \ (1 - (D/D_0)^2) \tag{7}$$

In Eq. (6) $Nu_{Gnielinski}$ is the Nusselt number calculated with Eq. (3), while D_0 is the tube reference diameter indicated by Adams, whose value is 1.164 mm. For very small diameters, however, the correction factor given by eq. (7) is independent of tube diameter, while for Re \leq 20000, the enhancement factor does not exceed 2.5. It should be noted that the correlation was based on only three tube diameters out of which two were below 1 mm. RESEARCH CARRIED OUT AT THE ENEA INSTITUTE OF THERMAL FLUID DYNAMICS

Experimental Facility

Experiments are carried out using demineralised water, which is degassed by passing through a very small, but continuous quantity of Helium. Being insoluble in water, an atmosphere of only Helium is created above the liquid level, so that all dissolved gases are driven out by their respective partial pressures in the water. A schematic of the test loop is shown in figure 4. After being filtered of solid impurities through a 10 μ m filter, the water passes through a gear pump – for flow rates over 10 ml/min and pressures below 6 bars – or a piston pump with damper – for conquering the large pressure drops through the smallest diameters (up to 120 bars for a 30 μ m tube) at flow rates below 25 ml/min. The water is then, if desired, brought to a certain controlled temperature level with a Peltier cell type pre-heater. This can be convenient to reduce water viscosity and thus head loss.

A 250 μ m K-type thermocouple verifies the actual fluid temperature on entrance into the test section, and pressure transducers on either side of the micro-tube investigated allow the pressure drop over the channel to be established. The mass flow rate is measured with a high precision scale.

The adopted means of measuring the frictional pressure drop per unit of length in the microchannel, therefore without inlet and outlet effects, is schematised in figure 5.

For each type of capillary studied, two lengths are cut from the same tube (so that surface conditions on the inside can be assumed comparable) and mounted in identical fittings. Thus, at equal mass flow rates, the concentrated pressure losses ΔP_{in} and ΔP_{out} must be the same for the two lengths of tube. Therefore, subtracting the total pressure drop along the shorter tube from that measured by the transducers for the longer tube, the difference yields the frictional pressure loss over the extra length of capillary (dp/dl* ΔL).

Then, in the Darcy equation

$$f = 2 \frac{\Delta p}{\Delta L} \frac{D}{\varrho u^2}$$
(8)

the friction factor f for the type of capillary under consideration, can be calculated inserting this difference for Δp , and the difference in length for ΔL . Obviously, the respective lengths of the two tubes are always made to be long enough for fully developed flow to settle. The dimensionless hydrodynamic entrance length is dependent on the Reynolds number for laminar flow: 0.055*Re (Shah and London [1978]). The maximum value is therefore at Re \approx 2200: 120 diameters. The length of the short tube is at least 350-400 diameters, so that in the subtraction of the gross pressure drops the hydrodynamic entrance length effects (which, it is remembered, are considered identical for the two lengths of tube at equal mass



Fig. 4 - Experimental facility used in ENEA experiments.



Fig. 5 - Pressure loss distribution along long and short test sections in ENEA experiments.

flow rates) are cancelled quite comfortably. The difference in length ΔL , typically 100-300 diameters, will then always be representative of frictional head loss only.

The bulk fluid velocity u in eq. (8) is calculated as $u = \Gamma / \rho A$, where A is the cross-sectional area of the microtube, ρ is the fluid density and Γ the mass flow rate.

For some heat transfer tests (Celata et al. [2002], where R114 has been used and Bucci et al. [2003]), where water has been used the capillary tube is heated by the condensation of steam flowing outside the microtube and produced in an external boiler. This specific configuration allows to obtain an indirect measurement of the external microtube wall temperature without placing a thermocouple on the microtube surface. The outer wall temperature is given by the condensation temperature of the vapour and can be obtained by measuring the vapour pressure inside the central tube where the vapour flows. In this way wall temperature is not influenced by the presence of thermocouples on the microtubes which are so small that a contact with any external body may sensibly affect the temperature measurement and the fluid temperature. As a drawback, the information obtained is related to an average heat transfer coefficient along the channel and in some tests this value may be spoiled by the change of the flow regime, i.e., entering in laminar flow and leaving the micropipe in turbulent flow. Besides, indirect heating only allow a coarse control of the thermal power delivered to the fluid (linked to the saturated conditions of the steam). The bulk fluid temperature is measured in the fluid distributors just upstream and downstream microtube using 0.5 mm K-type thermocouples. This makes the bulk exit temperature suitable for an energy balance in the fluid. The knowledge of the inlet and outlet temperatures, together with the measurement of the liquid mass flow rate, allows the computation of the thermal power delivered to the fluid by a heat balance in the coolant.

In the latest configuration of the facility for heat transfer tests the microtube under investigation is mounted inside a stainless steel capsule (see figure 6) which is sucked vacuum by a turbo-molecular vacuum pump (Alcatel ATS-100) to create an environment free of natural convection. To this effect, the level of vacuum must be less than 10⁻³ mbar. The level obtained in our set-up is 2 10⁻⁴ mbar (Edwards Penning Gauge Model 6), so that we can consider the heat loss to the surroundings through convection inexistent.

The heat loss due to radiation is evaluated by considering the formula for two concentric cylindrical surfaces, simplified for the limiting case where the surface of the internal body (OD of the test section is 0.9 mm) is much smaller than the external, concave surface (ID of the vacuum chamber is 100 mm):

$$q'_{rad} L = \sigma A_i \gamma_i \left(T_i^4 - T_e^4 \right) \tag{9}$$

The entity of this loss is of the order of 0.01% in the case of the highest measured temperature difference between the two bodies, and thus considered negligible.



Fig. 6 - Close-up of vacuum chamber and flange with mounted test section (circled), thermocouples and pressure ducts.

The only heat loss term of any importance would then be conduction through connecting wires, leads and tubing. The entity of this loss is difficultly quantifiable, inasmuch as it depends on the mass, conductivity and temperature of the various materials in contact with the test section. It is expected to be of influence, however, only in the case of extremely low fluid flow and high heat input.

A 250 μ m K-type thermocouple measures the fluid temperature on entrance into the test section, and pressure transducers on either side of the microtube investigated allow the pressure drop over the channel to be established (Druck PTX100/IS, 0-35 bar; Transamerica 0-160 bar); also a differential manometer is mounted parallel to the transducers for extra precise differential pressure measurements.

A constant power DC supply is used to heat the test section, which is fitted with Gas Chromatography fittings (Upchurch Scientific) resistant to high temperature and pressure.

At the outlet of the channel, a 50 μ m K-Type thermocouple is made to be inserted inside the (Near-Zero) dead volume of the fitting so that the fluid exit temperature (a critical quantity) is measured as closely as possible. The mass flow rate is measured with a high precision scale.

This latter configuration allows the evaluation of the local heat transfer coefficient, a higher precision and a more refined control of the thermal power delivered to the fluid.

Laminar Fluid Flow

Rough pipes

Although it is common opinion that a not too high relative wall roughness has little effect on laminar flow characteristics, available experimental achievements in micropipes, in spite of the large uncertainty still existing, would warn on the importance of surface roughness in laminar flow for micropipes.

Celata et al. [2002] measured the friction factor for R114 flowing in stainless steel capillary tubes 130 mm in diameter. The Reynolds number varied from 100 to 8000 and the relative channel surface roughness was 2.65%. An enlargement of the cross section of the stainless steel capillary pipe used in the experiments is shown in Fig. 7, where one can realize the large roughness of the pipe. Experimental results plotted in Fig. 8, as the friction factor versus the Reynolds number, show that for laminar flow the friction factor is in good agreement with Hagen-Poiseuille theory for Re less than 600-900. For higher values of Reynolds number, experimental data depart from the Hagen-Poiseuille law to the side of higher f values. It is evident the influence of surface roughness on the friction factor behaviour: a larger surface roughness would seem to produce an early deviation, in terms of Reynolds number, from the laminar flow behaviour. Similar results have been obtained for water flowing in 290 μ m pipe, characterized by a relative roughness of 0.75% (with an average roughness height of 2.166 μ m), Bucci et al. [2003].

These phenomena were observed in normal size tubes by Moody [1944] and Idelchick [1986]. According to Moody, in laminar flow, a value of the friction factor higher than 64/Re is definitely possible. This effect might be caused by the presence of initial turbulences or disturbances at the flow channel inlet and can appear for Reynolds value of 1200. Idelchick describes the same phenomena as connected to the presence of surface roughness: as a consequence, friction factor was affected not only by Reynolds number, but also by the surface roughness.

This has been verified for pipes characterised by a relative roughness larger than 0.7%. Preger and Samoilenko [1966] introduced the Reynolds number corresponding to the onset of the detachment of the friction factor from the Hagen-Poiseuille law. This Reynolds number depends on the tube relative roughness and can be calculated with the following equation:

$$\operatorname{Re}_{0} = 754 \exp\left(\frac{0.0065}{\varepsilon/\mathrm{D}}\right) \tag{10}$$

where ε is the average height of surface asperities.

As an example, eq. (10) provides a value of $\text{Re}_0 = 963$ for the data plotted in Fig. 8, where the experimental deviation is observed for Re = 600-900. The prediction obtained from Preger-Samoilenko equation is not so far from such an experimental evidence.



Fig. 7 - Enlargement of the 130 μm stainless steel microtube cross section, Celata et al. [2002].



Fig. 8 - Friction factor versus Reynolds number in laminar flow region for R114 and a 130 μm stainless steel pipe, Celata et al. [2002].

<u>Smooth pipes</u>

Recent experiments by Celata et al. [2004] using water in fused silica microtubes having inner diameter 31, 50, 101, and 259 μ m and a surface roughness of about 0.05 μ m, have shown how when using a smoother surface the experimental friction factor is quite in a good agreement with the Poiseuille law (fRe = 64) for Re up to 2000. Figure 9 shows the SEM picture of the cross section of the 101 μ m pipe, as a typical example of a smooth pipe, while figure 10 reports the experimental results of the friction factor versus Reynolds number for the tested pipes. The smallest diameter exhibits a larger scatter which can be attributed to the larger experimental uncertainty. For this very smooth pipes equation (10) does not forecast any detachment of the friction factor from the Hagen-Poiseuille law for Re < 2000, as also can be observed experimentally.

Laminar-to-Turbulent Fluid Flow Transition

Rough pipes

According to Schlichting [1979], flow characteristics (laminar, laminar-to-turbulent, turbulent flow regimes) can be recognized in the well known log-log Moody chart. In particular, laminar-to-turbulent flow transition region is characterized by the indefiniteness of behaviour of the f curves versus Re. In this region the lower limit for f is the continuation of the laminar flow line. This line corresponds to a flow in a smooth pipe without any initial disturbances. According to Moody [1944] the presence of initial turbulence in the flow causes the f values to be higher than the laminar flow line, as far as to a Reynolds number of about 1200. According to Schlichting [1979] the presence of roughness on the wetted pipe surface favors the laminar-to-turbulent flow transition. Rough pipes are characterized by a lower Reynolds number than smooth pipes. Roughness give additional disturbances in laminar flow which have to be added to those generated by turbulence already present in the boundary layer.

Preger and Samoilenko [1966] proposed an empirical method for calculating the boundaries of laminar-to-turbulent flow transition for rough pipes. The boundaries of transition region, indicated with Re_1 and Re_2 , can be calculated with the following equations, valid for water flowing in commercial tubes characterised by a relative equivalent roughness $\epsilon/D > 0.7\%$:

$$Re_{1} = 1160 \left(\frac{1}{\epsilon/D}\right)^{0.11}$$
(2);
$$Re_{2} = 2090 \left(\frac{1}{\epsilon/D}\right)^{0.0635}$$
(3)

For the 130 mm stainless steel pipe R114 data (Celata et al. [2002]), plotted in Fig. 11, we have experimental values of $Re_1 = 1850$ and $Re_2 = 2550$, while the Preger and Samoilenko correlations provide 1730 and 2630, respectively. These



Fig. 9 - Cross section of a fused silica capillary pipes, Celata et al [2004].



Fig. 10 - Friction factor, f, versus Reynolds number, Re: smooth tubes, using water as a fluid, Celata et al. [2004].

values are not far from experimental ones, the small discrepancy being in the range 6.5% - 8.7%, can be attributed to the different range of geometrical parameters on which these equations were developed.

<u>Rough pipes</u>

For smooth fused silica tubes ($\epsilon \approx 0.16 \,\mu\text{m}$) transition to turbulent flow seems to occur later. For data plotted in Fig. 10 we have Re₁ around 2000-2200 and Re₂ around 3000-3200. These values correspond acceptably to those discernable in figure 10, although no effect of abruptness is indicated by the equations.

Turbulent Fluid Flow

In the case of fully turbulent flow, friction factor can be found in fairly good agreement with the conventional theory. In other words, the classical equations provided by Blasius, for smooth pipes, and by Colebrook [1939] for rough pipes (see Schlichting [1979]), provides the boundaries inside which the experimental friction factor for micropipes lies. Figure 12 provides a typical example of this behaviour, being the Colebrook theory good enough in predicting the friction factor for the 290 mm stainless steel pipe, Bucci et al. [2003], as long as the turbulent flow is fully developed.

Heat Transfer

<u>Rough pipes</u>

Figure 13 shows the variation of the experimental average Nusselt number with Reynolds number for R114, stainless steel tube, $D = 130 \mu m$, Celata et al. [2002]. Curves of the correlations of Hausen, Dittus Boelter, Gnielinski and Adams are also plotted for comparison. In the graphs the laminar-to-turbulent transition region is indicated by two vertical dashed lines. The graphs show also the variation of the experimental friction factor in order to verify the laminar and turbulent regime. All the parameters are calculated at the mean temperature between inlet and outlet.

As expected, above correlations provide underestimation of experimental data in both flow regimes. Experimental data are underpredicted by the correlations tested.

In heat transfer experiments there are large differences between inlet and outlet liquid temperatures. This results in large differences of Reynolds number values due to the variation of viscosity, between inlet and outlet. In the transition region, the heat transfer regime is laminar in the first part of the flow channel and becomes turbulent near the channel outlet. Therefore, near the boundaries of the transition region the flow regime determination is affected by the strong variation of Reynolds number.



Fig. 11 - Friction factor versus Reynolds number for R114, stainless steel tube, D = 130 $\mu m,$ Celata et al. [2002].



Fig. 12 - Friction factor versus Reynolds number for water, stainless steel tube, D = 290 $\mu m,$ Bucci et al. [2003].



Fig. 13 - Nusselt number and friction factor versus Reynolds number for R114, stainless steel tube, D = 130 $\mu m,$ Celata et al. [2002].



Fig. 14 - Nusselt number versus Reynolds number for water, fused silica tubes, D = 120 $\mu m,$ Celata et al. [2005].

<u>Smooth pipes</u>

Figure 14 shows the variation of the experimental local Nusselt number with Reynolds number for water, fused silica tubes, $D = 120 \mu m$, Celata et al. [2005]. Experiments are carried out in the laminar flow regime, where the appropriate entrance length has to be guaranteed in order to have complete thermal boundary layers developed (Graetz number = RePrD/z < 10). For data plotted in Fig. 14 Gr is always less than 10 in the three wall temperature locations. It is clearly visible that after Re = 2000 the Nusselt number start to deviate from the laminar flow value.

Concluding Remarks

Although many discrepancies exist in the literature about research results on single-phase heat transfer and fluid flow in micropipes, mainly due to the large experimental uncertainty, some conclusions can be somewhat drawn from the experiments carried out so far.

Considering mainly a research carried out at the Institute of Thermal-Fluid Dynamics of ENEA (which results are in agreement with the general trend), experiments indicate that in the laminar flow regime friction factor is in good agreement with the Hagen-Poiseuille theory for Re < 600-800 in the case of rough micropipes and for Re < 2000 for smooth micropipes. The transition from laminar to turbulent regime occurs for Reynolds number in the range 1900-2500, this transition being in good agreement with the flow transition for rough commercial tubes.

Diabatic experiments show that heat transfer correlations in laminar and turbulent regimes developed for conventional tubes are not adequate for calculation of heat transfer coefficient in microtubes.

Further systematic studies are required to generate a sufficient physical knowledge of the mechanisms that are responsible for the variation of the flow structure and heat transfer in microtubes.

NOMENCLATURE

А	cross	section,	m^2
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- c_p specific heat, kJ/kg
- D pipe diameter, m
- F factor defined in Eq. (7)
- f friction factor G mass flux, kg/m²s
- G_z Graetz number, $G_z = \text{RePrD/z}$
- h heat transfer coefficient, kW/m^2K
- k thermal conductivity, kW/m K
- L pipe length, m
- Nu Nusselt number, Nu = hD/k

- p pressure, MPa
- Pr Prandtl number, $Pr = c_p \mu/k$
- Re Reynolds number, $Re = GD/\mu$
- T temperature, °C
- u velocity, m/s
- z axial length, m
- Γ mass flow rate, kg/s
- ε surface roughness, m
- γ body emissivity
- μ dynamic viscosity, kg/s m
- q density, kg/m³
- σ Stefan-Boltzmann constant, 5.669 10-8 W/m²K⁴

REFERENCES

- Adams, T.M., Abdel-Khalik, S.I., Jeter, S.M., and Qureshi, Z.H., 1998. An Experimental Investigation of Single-Phase Forced Convection in Microchannels, *International Journal Heat and Mass Transfer*, Vol. 41, Nos. 6-7, pp. 851-857.
- Agostini, B., Watel, B., Bontemps, A., and Thonon, B., 2002. Friction Factor and Heat Transfer Coefficient of R134a Liquid Flow in Mini-Channels, *Applied Thermal Engineering*, Vol. 22, pp. 1821-1834.
- Brutin, D. and Tadrist, L., 2003. Experimental Friction Factor of a Liquid Flow in Microtubes, *Physics of Fluids*, Vol. 15, n. 3, pp. 653-661.
- Bucci, A., Celata, G.P., Cumo, M., Serra, E., and Zummo, G., 2003. Water Single-Phase Fluid Flow and Heat Transfer in Capillary Tubes, *Thermal Science and Engineering*, Vol. 11, n. 6, pp. 81-89.
- Celata, G.P., Cumo, M., Guglielmi, M., and Zummo, G., 2002. Experimental Investigation of Hydraulic and Single Phase Heat Transfer in 0.130 µm Capillary Tube, *Microscale Thermophysical Engineering*, Vol. 6, pp. 85–97.
- Celata, G.P., 2004. Single-Phase Heat Transfer and Fluid Flow in Micropipes, *Heat Transfer Engineering*, Vol. 25, n. 3, pp. 13-22.
- Celata, G.P., Cumo, M., McPhail, S., and Zummo, G., 2006. Characterization of Fluid Dynamic Behaviour and Channel Wall Effects in Microtube, *International Journal Heat Fluid Flow*, Vol. 27, pp. 134-143.
- Choi, S.B., Barton, R.F., and Warrington, R.O., 1991. Liquid Flow and Heat Transfer in Microtubes, in *Micromechanical Sensors, Actuators and Systems*, ASME DSC-Vol. 32 (edited by Choi, D. et al.) pp. 123-128.
- Colebrook, F., 1939. Turbulent Flow in Pipes with Particular Reference to the Transition Region Between the Smooth and Rough Pipe Laws, *Journal Inst. Civ. Eng.*, No. 4, pp. 14-25
- Dittus, F.W., and Boelter, L.M.K., 1930. Heat Transfer in Automobile Radiators of the Tubular Type, University of California Publications in Engineering, Vol. 2, No.13, pp. 443-461.
- Filonenko, G., 1954. Hydraulic Resistance in Pipes, Teploenergetika, Vol. 1, pp. 40-44.
- Gnielinski, V., 1976. New Equations for Heat Transfer in Turbulent Pipe and Channel Flow, International Chemical Engineering, Vol. 16, pp. 359-368.
- Hausen, H., 1959. Neue Gleichungen f
 ür die W
 ärme
 übertragung bei freier oder erzwungener Strömung, Allg. Waermetech, Vol. 9, pp. 75-79.
- Idelchick, I.E., 1986. Handbook of Hydraulic Resistance, *Hemisphere Publishing Corporation*, Second Edition.

- Judy, J., Maynes, D., and Webb, B.W., 2002. Characterization of Frictional Pressure Drop for Liquid Flows through Microchannels, *International Journal Heat Mass Transfer*, Vol. 45, pp. 3477-3489.
- Judy, J., Maynes, D., and Webb, B.W., 2000. Liquid Flow Pressure Drop in Microtubes, Proc. of the International Conference on Heat Transfer and Transport Phenomena in Microscale, Banff, Canada, October 15-20.
- Kandlikar, S.G., Joshi, S., and Tian, S., 2001. Effect of Channel Roughness on Heat Transfer and Fluid Flow Characteristics at Low Reynolds Numbers in Small Diameter Tubes, Proc. of the 35th National Heat Transfer Conference, NHTC01-12134, Anaheim, CA, June 10-12.
- Lelea, D., Nishio, S., and Takano, K., 2004. The Experimental Research on Microtube Heat Transfer and Fluid Flow of Distilled Water, *International Journal Heat and Mass Transfer*, Vol. 47, pp. 2817-2830.
- Li, Z.X., Du, D.X., and Guo, Z.Y., 2000. Experimental Study on Flow Characteristics of Liquid in Circular Microtubes, *Proc. of the International Conference on Heat Transfer and Transport Phenomena in Microscale*, Banff, Canada, October 15-20.
- Mala, Gh.M., and Li, D.Q., 1999. Flow Characteristics in Microtubes, *International Journal of Heat and Fluid Flow*, Vol. 20, pp. 142-148.
- Moody, L.F., 1944. Friction Factors for Pipe Flow, Transactions of the ASME, pp. 671-684, Nov. 1994.
- Morini, G.L., 2004. Single-Phase Convective Heat Transfer in Microchannels: A Review of Experimental Results, *International Journal of Thermal Sciencs*, Vol. 43, pp. 631-651.
- Obot, N.T., 2000. Toward a Better Understanding of Friction and Heat/Mass Transfer in Microchannels A Literature Review, *Proc. of the International Conference on Heat Transfer and Transport Phenomena in Microscale*, Banff, Canada, October 15-20.
- Palm, B., and Peng, X.F., 2004. Single-Phase Convective Heat Transfer, in *Heat Transfer and Fluid Flow in Microchannels*, G.P. Celata, ed., Begell House, New York.
- Palm, B., 2001. Heat Transfer in Microchannels, Microscale Thermophysical Engineering, Vol. 5, pp. 155-175.
- Peng, X.F., and Peterson, G.P., 1996. Convective Heat Transfer and Flow Friction for Water Flow in Microchannels Structures, *International Journal Heat and Mass Transfer*, Vol. 39, No. 12, pp. 2599-2608.
- Peng, X.F., and Wang, B.X., 1993. Forced Convection and Fluid Flow Boiling Heat Transfer for Liquid Flowing Through Microchannels, *International Journal Heat and Mass Transfer*, Vol. 36, No. 14, pp. 3421-3427.
- Peng, X.F., Wang, B.X., Peterson, G.P., and Ma, H.B., 1995. Experimental Investigation of Heat Transfer in Flat Plates with Rectangular Microchannels, *International Journal Heat and Mass Transfer*, Vol. 38, No. 1, pp. 127-137.
- Pfund, D., Rector, D., Shekarriz, A., Popescu, A., and Welty, J.R., 1998. Pressure Drop Measurements in a Microchannel, in *Micro-Electro-Mechanical Systems (MEMS) – 1998*, DSC-Vol. 66, pp. 193-198, ASME.
- Preger, E.A., and Samoilenko, L.A., 1996. Investigation of Hydraulic Resistance of Pipelines in the Transient Mode of Flow of Liquids and Gases, *Issled, Vodosnabzhen, Kanalizatsii (Trudy LISI)*, no. 50, pp. 27-39, Leningrad.
- Rostami, A.A., Saniei, N., and Mujumdar, A.S., 2000. Liquid Flow and Heat Transfer in Microchannels: A Review, *International Journal of Heat and Technology*, Vol. 18, No. 2, pp. 59-68.
- Shah, R.K., and London, A.L., 1978, Laminar Flow Forced Convection in Ducts, *Advances in Heat Transfer*, Academic Press, New York, p. 91, p. 248.

Schlichting, H., 1979. Boundary Layer Theory, McGraw-Hill series in mechanical engineering.

Sharp, K.V., and Adrian, R.J., 2004. Transition from Laminar to Turbulent Flow in Liquid Filled Microtubes, *Experiments in Fluids*, Vol. 36, pp. 741-747.

- Wang, B.X., and Peng, X.F., 1994. Experimental Investigation on Liquid Forced Convection Heat Transfer Through Microchannels, *International Journal Heat and Mass Transfer*, Vol. 37, No. 1, pp. 73-82.
- Xu, B., Ooi, K.T., Wong, N.T., and Choi, W.K., 2000. Experimental Investigation of Flow Friction for Liquid Flow in Microchannels, *International Communications in Heat Mass Transfer*, Vol. 27, No. 8, pp. 1165-1176.
- Xu, B., Ooi, K.T., Wong, N.T., Liu, C.Y., and Choi, W.K., 1999. Liquid Flow in Microchannels, Proc. of the 5th ASME/JSME Joint Thermal Engineering Conference, San Diego, California, March 15-19.
- Yang, C.Y., Wu, J.C., Chien, H.T., and Lu S.R., 2003. Friction Characteristics of Water, R-134a, and Air in Small Tubes, *Microscale Thermophysical Engineering*, Vol. 7, pp. 335-348.
- Yen, T.H., Kasagi, N., and Suzuki, Y., 2003. Forced Convective Boiling Heat Transfer in Microtubes at Low Mass and Heat Fluxes, *International Journal Multiphase Flow*, Vol. 29, pp. 1771-1792.
- Yu, D., Warrington, R., Barron, R., and Ameel, T., 1995. An Experimental and Theoretical Investigation of Fluid Flow and Heat Transfer in Microtubes, ASME/JSME Thermal Engineering Conference, Vol. 1, ASME.