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Numerical Analysis of Roughness Effect on Mini and Microchannels: Hydrodynamics and Heat Transfer

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Abstract**—**A three-dimensional numerical simulation of flow through mini and microchannels with designed roughness is conducted here. The effect of the roughness height (surface roughness), geometry, Reynolds number on the friction factor, and Nusselt number is investigated. The study is carried out by employing CFD software, CFX. Our work focuses on a water flow inside a circular mini-channel of 1 mm and microchannels of 500 and 100 μm in diameter. The speed entry varies from 0.1 m/s to 20 m/s. The general trend can be observed that bigger sizes of roughness element lead to higher flow resistance. It is found that the friction factor increases in a nonlinear fashion with the increase in obstruction height. Particularly, the effect of roughness can no longer be ignored at relative roughness height higher than 3%. A significant increase in Poiseuille number is detected for all configurations considered. The same observation can be done for Nusselt number. The transition zone between laminar and turbulent flow depends on the channel diameter.

Keywords**—**Heat transfer, hydrodynamics, micro-channel, roughness.

I. Introduction

INTEREST in microdevices has been constantly growing over the past decade. In particular, for devices such as micro heat exchangers and micro machines, a considerable amount of research activity has been devoted to the understanding of small scale fluid phenomena. However, although a large pool of experimental data for both pressure drop and Nusselt number is available, the full comprehension of the microscale flow behavior is still an open problem.

Several studies exhibit contradictory results for both mechanical and thermal characteristics of the flow [1]-[3]. This is generally due to difference in the many parameters that characterize these studies such as geometry, usually made of complex multichannels, the hydraulic diameter, the shape and surface roughness of the channels, the fluid nature, the boundary conditions, the flow regime and the measurements and calculating techniques itself. For a fundamental insight into microfluidics, it may then be useful to reduce as much as possible the number of parameters.

While most literature references [3], [4] are on the role of the surface roughness in the microscale laminar regime agree in describing that an increase in Poiseuille number $f \cdot Re$, with respect to the conventional theory, a much higher uncertainty arises when the effects of surface roughness on heat transfer are considered.

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Work in the area of roughness effects on friction factors in internal flows was pioneered by [5] and [6]. Their work was, however, limited to relative roughness values of less than 5%, a value that may be exceeded in microfluidics application where smaller hydraulic diameters are encountered. Many previous works have been performed through the 1990s with inconclusive and often contradictory results.

Moody [7] presented these results in a convenient graphical form. The first area of confusion is the effect of roughness structures in laminar flow. In the initial work, [6] concluded that the laminar flow friction factors are independent of relative roughness ε/D for surfaces with $\varepsilon/D < 0.05$. This has been accepted into modern engineering textbooks. The effect of pitch on friction factor is another important area.

While most literature references on the role of the surface roughness in the microscale laminar regime agree in ascribing to it an increase in Poiseuille number $C = f \cdot Re$, with respect to the conventional theory, a much higher uncertainty arises when the effects of surface roughness on heat transfer are considered. According to [8], a high relative roughness of the walls increases the convective heat transfer because of the regeneration of the thermal boundary layer. On the other hand, [9], comparing their experimental results with the numerical ones obtained by solving a conjugate heat transfer problem, justify the measured lower Nusselt with the surface roughness effects.

Reynaud et al. [10] explain Nusselt numbers lower than those predicted by conventional theory by considering the non-uniformity of heat flux at the walls.

Xu et al. [11] performed experimental investigations on water flow in microchannels with hydraulic diameters ranging from 50 to 300 μm. They observed that the flow characteristics deviated from conventional theory for channel dimensions below 100 μm. The friction factor was smaller than that predicted by the Hagen–Poiseuille law. They also present the result for liquid flow in 30–344 μm diameter channels at Re numbers between 20 and 4000. The characteristics of flow in micro-channels agree with the conventional behavior predicted by the Navier–Stokes equations. Furthermore, they concluded that if any non-Navier–Stokes flow phenomena existed, their influence was masked by experimental uncertainty.

Bavière et al. [12] carried out numerical modelling of laminar flows in rough-wall micro-channels using rectangular prism rough elements in periodical arrays. The numerical results confirmed that the flow is independent of the Reynolds number in the range 1–200.

Friction factors in mini-scale and micro-scale rough channels were systematically investigated by [3]. The effects of the roughness element height, pitch, and relative roughness on friction factor were studied experimentally. A model was proposed to predict the early transition from laminar to turbulent flow due to roughness effects.

The present work consists in characterizing the dynamic and thermal field of a flow in circular mini and micro-channel equipped with structured roughness using a commercial code CFX 5.15.

Here, we will take advantage of the flexibility of CFD tools in order to isolate and estimate the magnitude of the roughness effect. In particular, we will verify if it is possible to detect simple relationships between roughness, pressure losses and heat transfer, or if these effects are sensitive to the geometrical details and, thus, not easy to generalize.

II. Flow Configuration and Boundary Conditions

The configuration of the flow is a circular, mini and micro channel with a diameter D (Fig. 1). The channel length, L, is chosen so that the fully developed flow is reached at inlet. The following assumptions were made for the numerical method: (i) The fluid is water, incompressible, continuum and Newtonian, (ii) All the fluid properties are constant, (iii) Steady flow character, (iv) The effect of gravity and other forms of body forces are negligible.

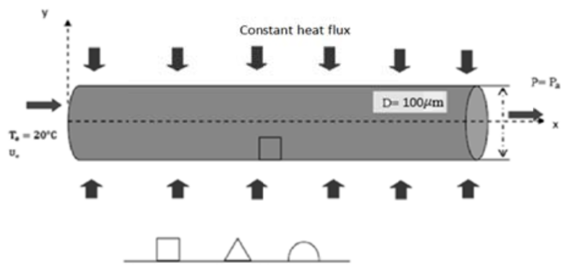

Fig. 1 Circular micro-channel with roughness geometry

The Reynolds number Re = U.D/ν, is based on the incoming velocity U and the channel diameter D. The selected flow conditions are taken as follow: the flow velocity U at the entrance of the channel "inlet" is specified, U = 0.1 to 25m/s. The diameter of the channel varies from mini to micro scale with values: D = 1mm; 0.5mm; and 0.1mm. The boundary conditions are; mass conservation at outlet and no-slip at the wall surface. A heat flux is applied to the wall with constant value = 100000w/m2K. The following properties are used: ρ = 988.05 kg/m3 and ν = 1,004.10-6 m2/s.

III. Numerical Technique

The 3-D Navier-stokes equations are used to describe the flow in the mini and micro-channel. The resolution of these differential equations requires the choice of a suitable numerical method to solve the problem. For our case, we choose the finite-volume method. The turbulence model used is the SST model (Shear stress turbulence model).

To simplify the study and reduce the elements number of the mesh volumes and to speed up the computation time, we opted for an axisymmetric configuration.

We specially pay attention in selected grid. The mesh employed is of tetrahedral elements with prismatic refining at the wall, illustrated in Figs. 2 and 3.

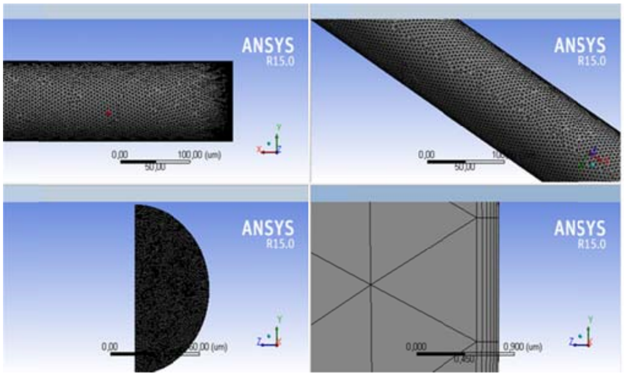

Fig. 2 Wall mesh

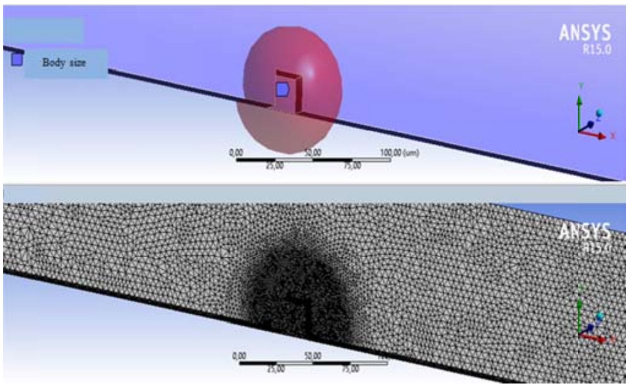

Fig. 3 Grid around roughness

Non uniform grid was generated and grid refinement close to the wall and sudden contraction zone was applied. Several successive grid refinements have been carried out in very case to get negligible effect of the mesh in the solutions. We considered about 1.2 million nodes and 6 million tetrahedral elements. The CFD used is a combination of two complementary softwares: The CFX mesh processor, which makes it possible to prepare the geometry and to generate grid and the CFX solver, who solves the equations modeling the phenomenon. The CFX solver will terminate the run when the equation residuals calculated using the method specified is below the target residual value. A convergence criterion of 10^{-4} was used to ensure negligibly small iteration errors.

IV. Results and Discussion

A. Smooth Mini-Channel

This part is the study of a smooth mini-channel of diameter D = 1 mm and length L = 10 cm. The inlet speed is varying between 0.1 and 25 m /s, to scan the different flow regimes; laminar and turbulent. The purpose of this study was the comparison of numerical simulation with experimental results and theoretical ones. The results are represented in Fig. 4 for Poiseuille number and Fig. 5 for Nusselt number.

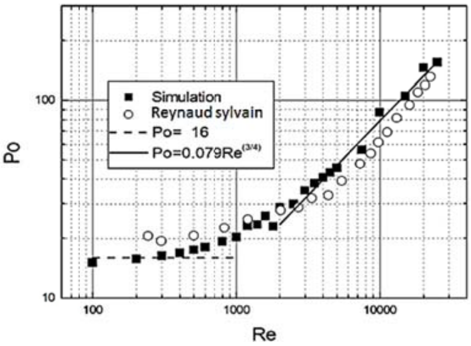

Fig. 4 Comparison between experimental and theoretical results of the Poiseuille number with numerical simulations

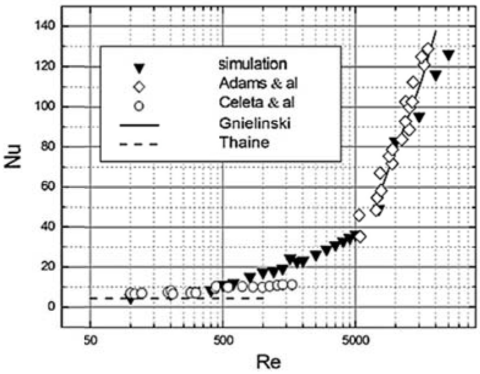

Fig. 5 Comparison between experimental and theoretical results of the Nusselt number with numerical simulations

The first observation concern the laminar flow Re <1000. The numerical results seem closer to the experimental ones. For Reynolds numbers greater than 5000, the numerical Poiseuille number is of the same order of magnitude as the experimental results of [10]. However, evolution does not show exactly the same slope. We further observe that the evolution of the number of Poiseuille undergoes a change in slope for a Reynolds number between 1000 and 5000. This change can be interpreted as the transition from laminar to turbulent regime. In this case, it would be in the range of Reynolds numbers admitted for conventional channels. Note, Fig. 5, that the numerical results of Nusselt number do not differ from those derived from the theoretical prediction, particularly for the turbulent regime. To validate our numerical results we compared them with the experimental study of [9] for the turbulent flow and with those of [13] for the laminar flow. The numerical results evolve similarly with the experimental results with a slight gap.

The transition zone is obtained for Re between (1000 and 5000). The results confirm the early transition to transition, already observed by further authors. It's also found that the value of the average Nusselt number increases gradually as the Re increases, according to the law of [14] corrected by [9], for turbulent regime. By cons, for laminar flow, the value of the Nusselt number is slightly higher than those obtained by [15].

B. Effect of Channel Diameter

Figs. 6 and 7, represent, respectively, the average values of the Poiseuille number and Nusselt number vs Reynolds number, for three different diameters (D=1mm; D=0.5mm and D=0.1mm).

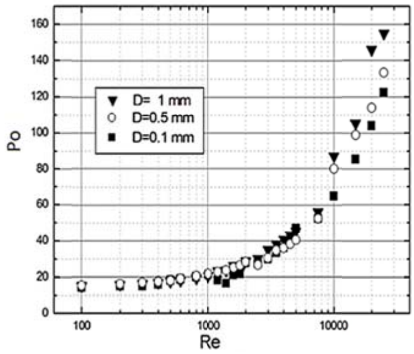

Fig. 6 Evolution of Poiseuille number with Reynolds number

For Re number below 1000, the values of Po for the three diameters is almost the same (Fig. 6). This demonstrates that the change of the diameter, down to values considered, has not a big influence on the values of the Poiseuille number. However, for higher values of Re, a slight gap starts to become visible. The average number of Poiseuille is slightly greater for diameter (D = 1 mm).

Fig. 7 shows that the heat transfer, quantified by the Nusselt number is not much influenced by the reduction in the diameter of the channel. This could be explained by the fact that the limit of mini-micro channels is not reached yet, or, the fact that the fluid flow in micro-channel can be studied through the results from the classical theory, with certain precautions.

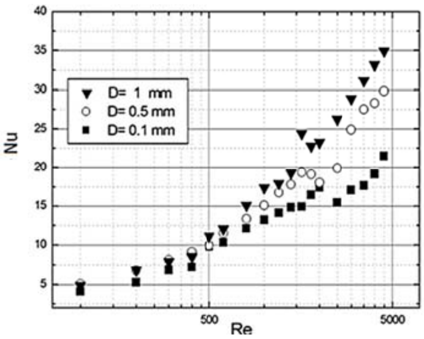

Fig. 7 Evolution of Nusselt number with Reynolds number

C. Micro-Channel with Roughness

In this section we analyze the dynamic and thermal effect for a micro-channel of diameter D = 0.1 mm and a length L = 1cm equipped with roughness of different forms. We set the height of the relative roughness ε/D to 25% and the flow is considered laminar (Re = 500). This study will involve three configurations (cubic roughness, pyramid, and a half-spherical roughness).

Fig. 8 represents the velocity and temperature distribution within the micro-duct for the different form studied.

Fig. 9 presents changes in values of the Nusselt number for different forms of roughness. Note that the value of the average Nusselt number is higher for a cubic roughness. Chengbin Zhang [16] also made similar findings and interpreted this increase to the form of the cubic roughness, which generates considerable stagnation and recirculation in lover and downstream of the roughness. This promotes heat transfer. Unlike spherical and pyramidal shape have a smaller angle of attack. These shapes allow the flow to be profiled and therefore generate very little vortices compared to the cubic roughness.

To study the effect of the roughness height, we opted for a cubic roughness, placed far from inlet section at x / L = 40%. Reynolds number considered is Re = 500. The height of roughness ranges from ε / D = 5% and 45%.

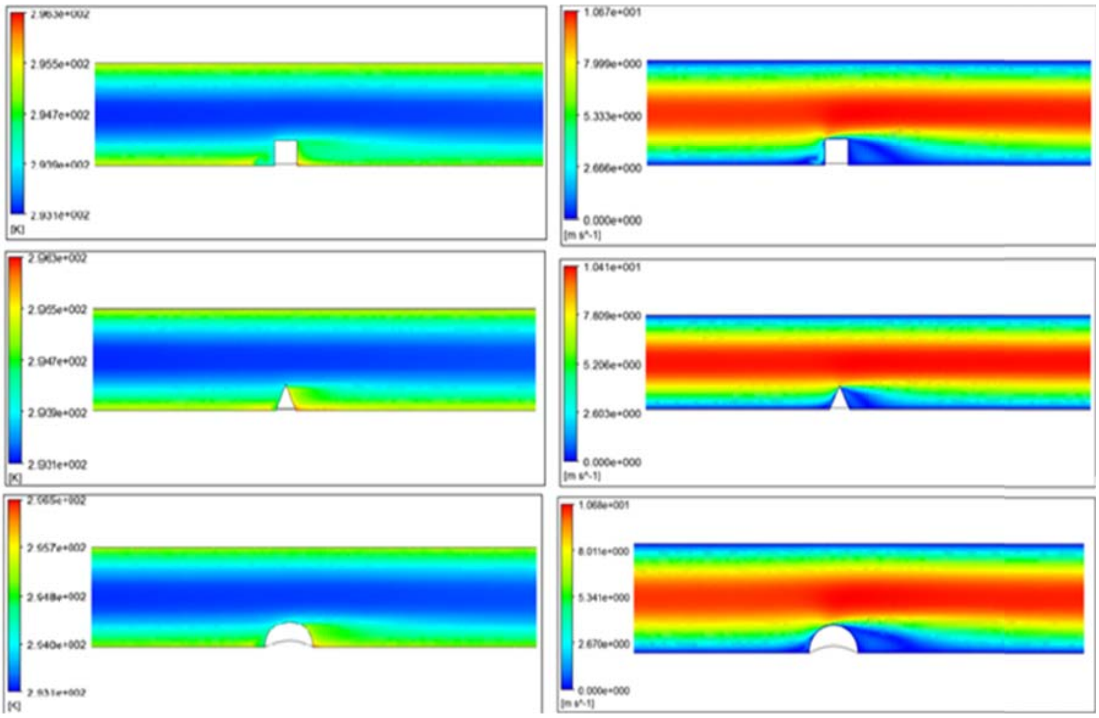

Fig. 8 Velocity and temperature distribution for different roughness form (Re = 300: ε/D=35%))

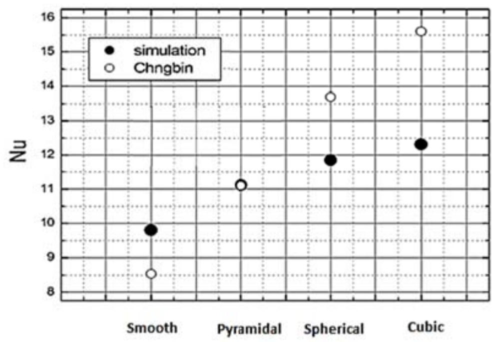

Fig. 9 Evolution of Nusselt number with roughness form (Re = 500; ε/D=25%)

The effect of the presence of the roughness can be felt by the increasing in the value of the average Nusselt number. From Fig. 10, we find that for values of ε /D < 17%, the value of Nu increases slightly compared to a smooth mini-channel. For roughness value of ε/D > 17%, Nusselt number increase considerably. This fact was also observed by [16] and interpreted by non-attachment of the boundary layer due to the surface roughness after the higher roughness. The dynamic and thermal boundary layers, causes an obstruction to heat transfer. We also studied the effect of the variation of the roughness bases. Fig. 11 represents a comparison between the values of the average Nusselt number for two types of roughness. Cubic roughness (variable base) and rectangular roughness (constant base ε/D = 12%).

The results show that for values of ε / D less than 17%, no difference was observed between the two configurations. However, for ε/D> 17%, even if the evolution of the Nusselt number follows the same pace, a difference of values is noted. For a given height, roughness cubic causes a significant increase in heat transfer. This is due to the greater distance between the fluid and the wall, preventing the reformation of the boundary layer and thus improving the transfer.

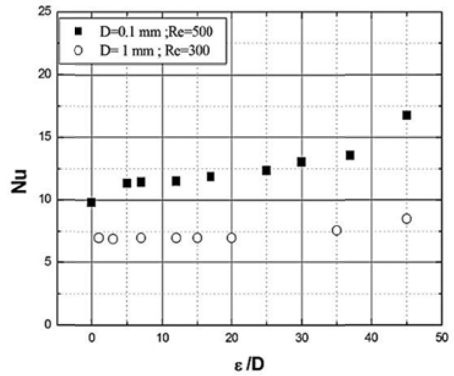

Fig. 10 Evolution of Nusselt number with roughness height (Re = 500)

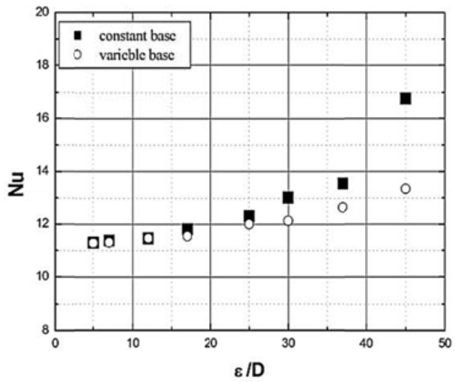

Fig. 11 Evolution of Nusselt number with roughness height (Re = 500)

Fig. 12 shows the numerical results obtained for a mini and micro-channel with two roughness placed differently from each other. The relative roughness considered is 25%. Note for a laminar flow, the average Nusselt number increases with the distance until a maximum value S / ε = 4. The value of Nusselt then decreases gradually. This was also found by [16] and [4]. This can be interpreted by the fact that at a spacing S, the boundary layer has time to link between the two rough, causing a decrease in the value of the Nusselt number.

The value of the average Nusselt number increases with the increase of roughness, as we can see in Fig. 13. This increase was predictable. Indeed, the non-attachment of the dynamic and thermal boundary layers increases heat transfer, and in the case of several rough this occurs repeatedly, so the phenomenon is superimposed to generate a greater transfer.

V. Conclusion

The objective of this paper was to identify the impact of roughness on the thermal behavior of a fluid in a circular micro-channel, but also to be aware of possible behavioral difference between mini and micro channels. The study was performed using the numerical code CFX 15.0 and digital resolution finite volume method.

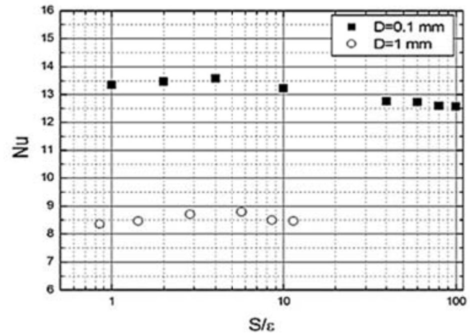

Fig. 12 Nusselt number vs distance between two roughness (ε / D = 25%) (Re = 500)

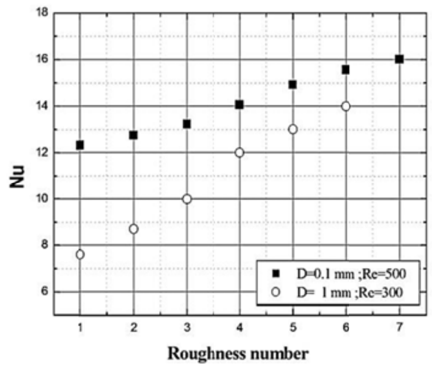

Fig. 13 Evolution of the Nusselt number with roughness number

The average values of the Nusselt number, in laminar flow approximate theoretical predictions and the experimental results of [15] and [13], but the trend seems slightly different when the Reynolds number increases up then 1000 for turbulent regime. Particularly the numerical results are consistent with experimental results of [9] and the predictions of [14]. We also investigated the effect of spacing between two successive roughness. The important effect is a significant increase in the value of the average Nusselt number. This increase can be interpreted by the perturbation of dynamic and thermal boundary layers, which causes better heat transfer. The same observations were made by [16] and [4]. They explain this phenomenon by the fact that from a given distance, the boundary layer has time to recreate between two successive roughness.

The effect of roughness height was also investigated. We denote an increase in the average Nusselt number with the relative intensity of the roughness. This increase is significant from a relative intensity of ε /D> 17%. Chengbin Zhang [16] made the same observation in his study with different value of ε /D.

The effect of the roughness shape was also considered, where it is found that the shape of the roughness has a major impact on heat transfer. Numerical results show that cubic roughness has a significant effect on heat transfer. Chengbin Zhang [16] also found this fact, interpreted as being due to the shape of the cube which generates interference upstream and downstream of the roughness. This is not the case of the sphere and the pyramid shape which allow the flow to be shaped, and thus improve the very little heat transfer.

The results show that until diameter of 0.1 mm, the same behavior of the flow is denoted for dynamic and thermal aspect. This could be explained by the fact that the limit of mini-micro channels is not reached yet, or, the fact that the fluid flow in micro-channel can be studied through the results from the classical theory, with certain precautions.

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