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NUMERICAL INVESTIGATION OF TURBULENCE MODELS FOR MINICHANNELS

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ABSTRACT

Due to having considerably small diameters compared to the macro channels; validation of conventional models and correlations and, examination of heat transfer and flow characteristics for mini/micro channels have been an attractive subject for last decades. In this study, classical turbulence models are compared and applicability of the conventional correlations is investigated for the flow through minichannels having diameter range between 1.2 and 0.25 mm. For the flow considered, fluid (R134a) enters the horizontal channel with a prescribed temperature and velocity, absorbs heat from the surrounding and then leaves the channel. Reynolds number is chosen in a range between 5000 and 20000 in order to cover the turbulent regime. For the first step of the study, in order to investigate the use of conventional turbulence models, Standard k-ε, RNG k-ε, Realizable k-ε, Standard k-w and Reynolds Stress models are employed to estimate friction factor and Nusselt number values for 0.5 mm diameter channel. These numerical results are compared with those calculated by conventional correlations existing in the literature. According to the comparison, none of the models create a dramatically deviation and Standard k-ε is determined as the model giving the closest results to the conventional values. As second step of the study, Standard k-ε model is applied for the flow through the minichannels having diameter of 1.2, 1, 0.8, 0.5 and 0.25 mm, respectively. Friction factor and Nusselt number values estimated numerically via Standard k-ε model are compared with those calculated by conventional correlations and existing relevant experimental data. According to the study, it is concluded that the numerical friction factor values are found to be close to the conventional values. The most discrepancy exists when diameter is less than and equal to 0.5 mm. Furthermore, numerical Nusselt number values are found to be close to conventional values estimated with the correlation proposed by Gnielinski (1976) while they are lower estimated for channels having diameter of 1.2, 1 and 0.8 mm and over estimated for 0.5 and 0.25 mm diameter channels. As a result, conventional correlations and turbulence models are found to be applicable for the diameter range and the flow investigated.

INTRODUCTION

There have been many studies performed in order to investigate its flow and heat transfer characteristics since mini/microchannels have become a research area. Some of the studies can be summarized as follows. Yun Heming et al. (2006) investigated the flow and heat transfer by CFD modeling of the rectangular minichannels with hydraulic diameter between 0.2 and 1.4 mm. As a result they found that the friction factor increases with increasing aspect ratio and decreasing Reynolds number. Transitional Reynolds number decreases with decreasing hydraulic diameter. Celata et al. (2006) experimentally studied the evaluation of frictional pressure drop by taking the channel diameter, shape and aspect ratio, inclination, working fluid and heat input into consideration in different channels having hydraulic diameter between 0.259 and 1.699 mm. As a result, they determined that Poiseuille and Blasius equations are valid regardless of the fluid, cross-section and inclination. Steinke and Kandlikar (2006) reviewed the studies investigating the single-phase friction factor in microchannels and compared this literature with their experimental results. They concluded that conventional friction factor correlations are valid in laminar regime for the channel diameters considered. Adams et al. (1997) investigated the forced convectional heat transfer of single-phase turbulent flow through circular microchannels having 0.76 and 1.09 mm diameter. They compared the results with a previous experimental data and concluded that the experimental and predicted Nusselt numbers fit each other within ± % 18.6. Owhaib et al. (2004) experimentally investigated the heat transfer characteristics of single-phase forced convection of R134a through circular microchannels having 1.7, 1.2 and 0.8 mm inner diameters. They compared their experimental results with classical correlations and suggested correlations for microchannels. They found that the experimental results show good agreement with the classical correlations in turbulent region. Also they determined that none of the suggested correlations for microchannels agreed with the experimental data. Lelea et al. (2004), numerically and experimentally studied the microchannel heat transfer and fluid flow for distilled water flowing with Re-number range up to 800 through 0.1, 0.3 and 0.5 diameter microtubes. They determined that the conventional theories including the entrance effects are applicable for the case considered. Hetsroni et al. (2005) considered the heat transfer in the frame of continuum model corresponding to small Knudsen number. They analyzed circular, triangular, rectangular, and trapezoidal micro-channels with hydraulic diameters ranging from 60 μm to 2000 μm, and investigated the effects of geometry, axial heat flux due to thermal conduction through the working fluid and channel walls, and energy dissipation on heat transfer. As a result they concluded that the effect of energy dissipation on heat transfer in microchannels is negligible under typical flow conditions, axial conduction in the fluid and wall affect significantly the heat transfer in micro-channels and the thermal entry length should be considered by comparison between experimental and numerical results. Caney et al. (2007) investigated frictional pressure drop and heat transfer during single-phase flow in vertical minichannel with the aim of determining the validity of classical correlations available for conventional size channels. The Reynolds number is chosen in order to cover the laminar regime as well as the beginning of the turbulent regime. According to the experimental frictional pressure drop measurements they concluded that the classical correlations can be accurately applied. With a correction on temperature measurements they found that the heat transfer measurements are in fair agreement with the classical literature results. Yang and Lin (2007) performed an experimental investigation on forced convective heat transfer performance of water flowing through microtubes with inner diameters ranging from 123 to 962 μm. According to the test results they stated that the conventional heat transfer correlations for laminar and turbulent flow can be well applied for predicting the fully developed heat transfer performance in microtubes. They also concluded that the discrepancy between the test results and the predicted value increases with decreasing tube size. Agostini et al. (2004) presented their friction factor and heat transfer coefficient experimental results obtained with a liquid flow of R134a in rectangular minichannels. They investigated two test sections with different mass flux and heat flux ranges. As a result they concluded that the literature correlations for large tubes were found to predict their results reasonably well.

In this study, different conventional turbulence models are employed numerically in order to examine their applicability and to estimate friction factor and Nusselt number values for R134a flow through minichannels having diameter range between 1.2 and 0.25 mm. Reynolds number is chosen in a range between 5000 and 20000 in order to cover the turbulent regime. As a first step, according to the comparison between conventional and numerical values of friction factor and Nusselt number given by different turbulence models for 0.5 mm diameter channel, most appropriate turbulence model for the flow investigated is determined. Then, in the second step, turbulent characteristics of the flow and the heat transfer though the minichannels are investigated via this model for different Reynolds numbers and results are compared with those estimated by conventional correlations and existing experimental studies.

SIMULATION DETAILS

Drawing of the channel investigated in the study is shown in Figure 1. R134a flow, having an inlet and outlet temperature of 20 ºC and 60 ºC relatively, at 300 kPa with a Reynolds number ranging from 5000 to 20000 in a circular channel has been assumed. Fluid enters the channel with a prescribed temperature and velocity, absorbs heat from the surrounding and leaves the channel. Heat flux from the channel wall is estimated for each Reynolds number in order to obtain constant temperature at the outlet. Geometry and numerical grid are constituted by using GAMBIT. 2D, axi-symmetric geometry is used in numerical analysis. Aspect ratio of grids is determined to be less than five due to convergence criteria and thus between 10^6 and 2.18×10^5 nodes are obtained for the computational region, based on the channel diameter. Grid sizes with respect to the channel diameters, D, are given in Table 1.

Table 1: Grid size vs. channel diameter

D (mm)	Cell	Faces	Nodes
1.2	998400	2007296	1008897
1	832000	1674480	842481
0.8	665600	1341664	676065
0.5	416000	842440	426441
0.25	208000	426420	218421

Numerical simulation is performed thanks to the commercial program, FLUENT, that is using finite volume method. Second order upwind scheme for spatial discritization are utilized for the solution. Gradients are evaluated via Green-Gauss Cell Based method and SIMPLE algorithm is employed for pressure-velocity coupling. Calculations based on the channel diameters and Reynolds numbers considered, and on the thermophysical properties of the fluid at mean temperature show that Gr/Re^2 values for all cases investigated are found between 1.38×10^{-3} and 7.85×10^{-6}. Besides, density variation along the channel is inconsiderable, i.e. 14 %. Hence, natural convection through the channel is neglected and the fluid is assumed to be incompressible. Enhanced Wall Treatment method is applied for all models in order to investigate the near wall region, properly. Boundary conditions given for channel wall, inlet and outlet are heat flux, uniform velocity inlet and pressure outlet, respectively. Uniform velocity at the inlet is provided in order to observe the hydrodynamic entrance length.

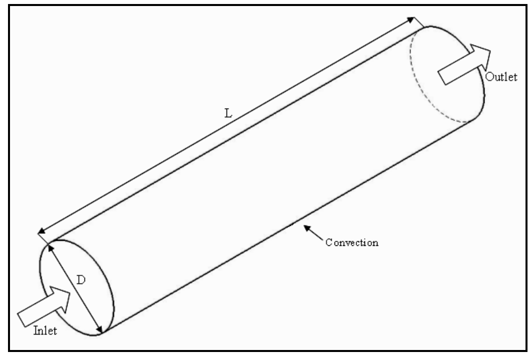

Figure 1: Schematic of the channel considered

Fluid is chosen to be superheated vapor to avoid two-phase flow along the channel. Numerical Nusselt numbers for each turbulence model are estimated via Newton's Law of Cooling with temperature values obtained from the numerical analysis. Mass-weighted average of the fluid temperature in radial direction for 26 points taken along the channel and, area-weighted wall temperature at these points are used to calculate the local heat transfer coefficient. By using local heat transfer coefficient, local and averaged Nusselt number values are estimated. Estimation procedure is illustrated with Figure 2.

$$q''_x = h_x \left(T_w - T_{ave}\right) \tag{1}$$

$$Nu_x = \frac{h_x d}{k} \quad , \quad Nu = \frac{1}{L}\int_0^L h_x dx \tag{2}$$

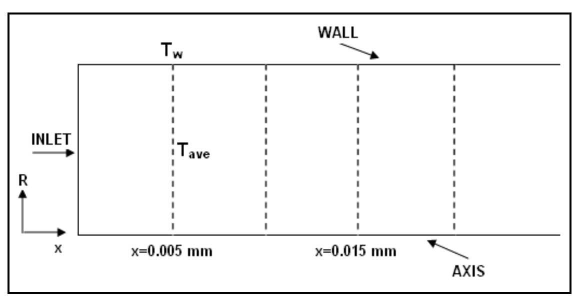

Figure 2: Sketch for estimation procedure of heat transfer variables

In order to validate the mesh structure, numerical frictional factor values for Reynolds number of 600 and 1000 are compared with those obtained by Yang et al. (2003). As a result, numerical friction factors for

Reynolds numbers equal to 600 and 1000 are obtained to be 0.10768 and 0.06568 while their experimental and theoretical values ($f = 64/\mathrm{Re}$) are 0.116 and 0.072 and, 0.1066 and 0.064, respectively. Hydrodynamic entrance length for turbulent flow is estimated with Equ. 3, and found to be between 1/16 th and 1/113th of the channel length according to the Reynolds numbers and diameters investigated.

$$L_{hy} = 1.359\,\mathrm{Re}^{0.25} \cdot d_{hy} \qquad (3)$$

Besides, numerical results of entrance length for 0.5 and 0.25 mm diameter channels at are illustrated in the figure, below.

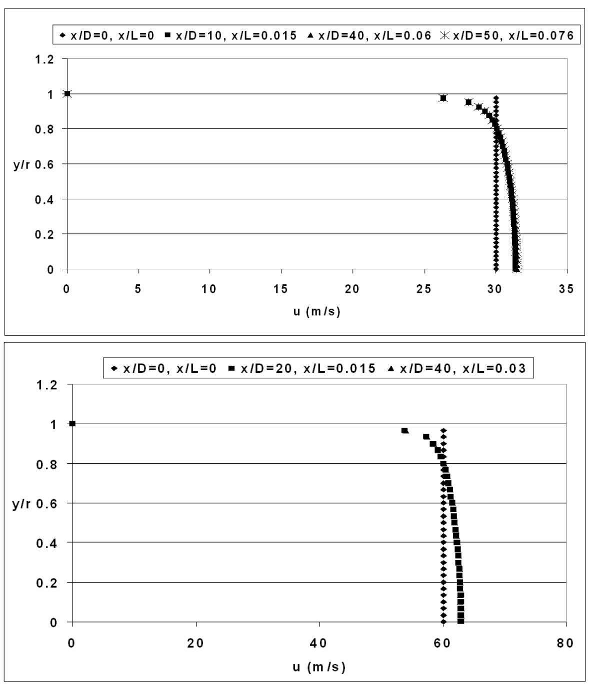

Figure 3: Entrance length of 0.5 mm (upper) and 0.25 mm (lower) diameter channels for Re=15000

Thanks to the results calculated via Equ. 3 and to the numerical values exemplified in Figure 3, entrance length of all cases investigated correspond to max. 1/13 th of the channel length, L. Thus the turbulent flow in the channel is assumed to be fully developed. Relative error for heat and mass flux and, y^+ values at the channel wall are taken as the convergence criteria of the numerical analyses. Relative error (%) formulation for any value is given in Equ. 4. Results for 1.2 mm and 0.25 mm diameter channels are written in table, below.

$$\Delta v = \left(v_{inlet} - v_{outlet}\right) / v_{inlet} \qquad (4)$$

Table 2: Relative error for heat and mass flux and, y^+ values at the channel wall

Reynolds Number		5000	10000	15000	20000
1.2 mm	$\Delta\dot{m}$	0	0	0	0
	$\Delta\dot{Q}$	0.46	0.46	0.3	0.35
	y^+	0.96	1.7	2.41	3.11
0.25 mm	$\Delta\dot{m}$	0	0	0	0
	$\Delta\dot{Q}$	0.2	0.1	0.7	0.2
	y^+	1.63	3.49	5.21	6.76

Friction factors for turbulent flow are calculated with the correlation proposed by Petukhov (1970),

$$f = \left(0.7904 \ln \mathrm{Re} - 1.64\right)^{-2}, 3000 \le \mathrm{Re} \le 5\times10^{6} \qquad (4)$$

Conventional correlations used to evaluate turbulent Nusselt number are listed in Table 3.

Table 3: Conventional correlations to evaluate turbulent Nusselt number

Dittus-Boelter: $Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4}$, $\mathrm{Re} \ge 10000$
Gnielinski: $Nu = \dfrac{(f/8)(\mathrm{Re}-1000)\mathrm{Pr}}{1+12.7(f/8)^{1/2}\left(\mathrm{Pr}^{2/3}-1\right)}$ $3000 \le \mathrm{Re} \le 5\times10^{6}$, $f = \left(1.82\cdot\log\mathrm{Re}-1.64\right)^{-2}$ $0.5 \le \mathrm{Pr} \le 2000$
Petukhov et al.: $Nu = \dfrac{(f/8)\mathrm{Re}\,\mathrm{Pr}}{K+12.7(f/8)^{1/2}\left(\mathrm{Pr}^{2/3}-1\right)}$ $10^{4} < \mathrm{Re} < 5\times10^{6}$ $K = 1.07 + \dfrac{900}{\mathrm{Re}} - \dfrac{0.63}{(1+10\cdot\mathrm{Pr})}$
Wu and Little: $Nu = 0.00222\,\mathrm{Re}^{1.09}\,\mathrm{Pr}^{0.4}$ $\mathrm{Re} > 3000$
Choi et al: $Nu = 3.82\cdot10^{-6}\,\mathrm{Re}^{1.96}\,\mathrm{Pr}^{1/3}$ $2500 < \mathrm{Re} < 20000$
Yu: $Nu = 0.0007\,\mathrm{Re}^{1.2}\,\mathrm{Pr}^{0.2}$, $6000 < \mathrm{Re} < 20000$

RESULTS AND DISCUSSION

In this paper conventional turbulence models and classical correlations for friction factor and Nusselt number are examined for minichannels. Firstly, five different conventional turbulence models are compared for a minichannel with 0.5 mm diameter, numerically. Enhanced wall treatment is employed to all models to investigate near wall, properly. Numerical friction factor and Nusselt number values estimated via different turbulence models for Reynolds number between 5000 and 20000 are compared with the results calculated by conventional correlations. In order to determine the most suitable conventional correlation for Nusselt number, numerical values for 0.5 mm diameter channel are compared with those calculated by the correlations given in Table 1. According to the results, correlation proposed by Gnielinski (1976) is found to be the one giving the closest values to those of numerical and, is concluded as the conventional correlation for Nusselt number.

Conventional and numerical results for friction factor and Nusselt number values with respect to (wrt) turbulence models are shown in Fig. 4 and Fig. 5, respectively. In these figures, correlation proposed by Petukhov (1970) is used as classical correlation for friction factor.

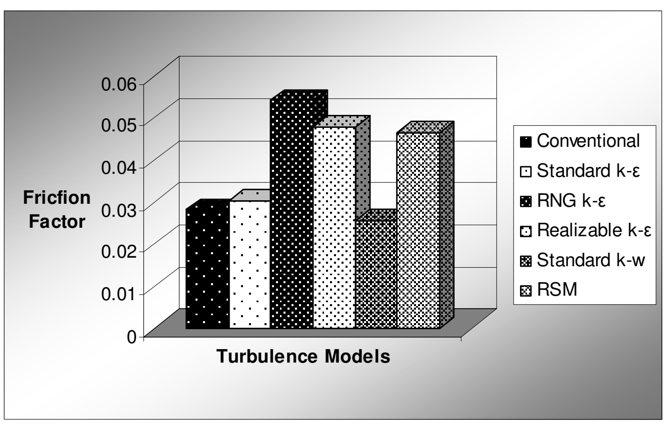

Figure 4: Conventional and numerical values of friction factor wrt turbulence models

As expected, different results are obtained due to varying solution methodology of the turbulence models. According to the results shown in Fig. 4 and Fig. 5, Standard k-ε model gives the most reasonable results for both friction factor and Nusselt number and therefore, is chosen to be the most appropriate turbulence model among the others, for the present study.

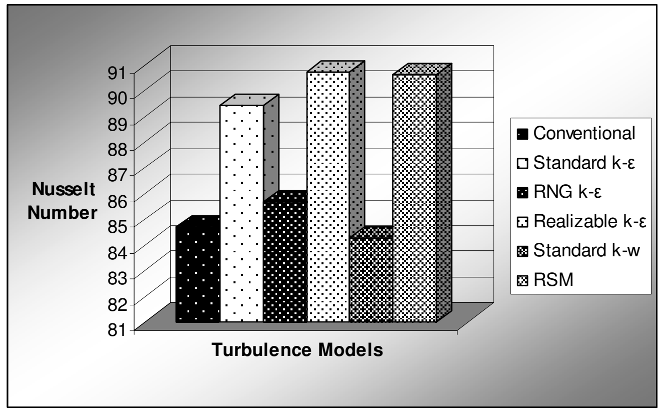

Figure 5: Conventional and numerical values of Nusselt number wrt turbulence models

In the second part of the paper, friction factor and Nusselt number values are estimated via Standard k- ε model for minichannels having diameter between 1.2 and 0.25 mm. Results obtained numerically are compared with the existing experimental values in the literature and with the conventional values to investigate the applicability of classical correlations for minichannels. Conventional and numerical values of friction factor versus Reynolds number for different diameter channels are shown in Figure 6, where the equation proposed by Petukhov (1970) is taken to be the conventional correlation.

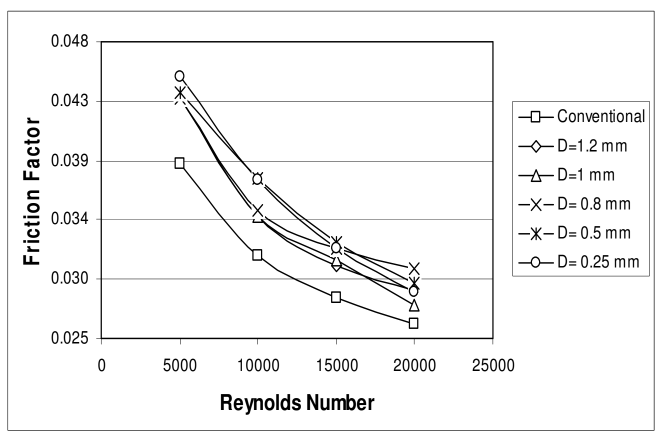

Figure 6: Friction factor values wrt Reynolds number for different diameter channels

From the results shown in Fig. 6, it can be seen that the numerical and conventional values for friction factor are close to each other while difference between the values increases with decreasing diameter. For minichannels considered, friction factor increases with decreasing diameter and decreases with increasing Reynolds number, thus the characteristics are similar to that of macro channels. The most discrepancy occurs while passing from transition region to turbulent region for the channels with diameter less than 0.8 mm. Thus this interval is determined to be a transition from the point of view of channel diameter.

Steinke and Kandlikar (2006) reviewed the experimental studies which investigated the friction factor in microchannels with various cross-sections, Reynolds number ranges and hydraulic diameters. They gathered the data of all experimental studies considered and plotted the non-dimensional Poiseuille number, C*, versus Reynolds number. Here C* is defined with the equation, below.

$$C^* = \frac{(f\,\mathrm{Re})_{\mathrm{exp.}}}{(f\,\mathrm{Re})_{theory}} \tag{5}$$

In Figure 7 and 8, numerical and experimental C^* values are shown. In order to calculate numerical C^*, $(f\,\mathrm{Re})_{numerical}$ is substituted instead of $(f\,\mathrm{Re})_{\mathrm{exp.}}$ in Equ.5.

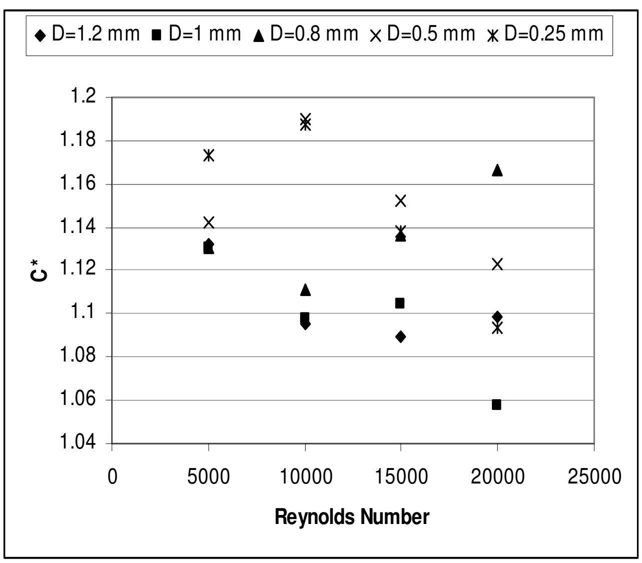

Figure 7: Numerical C^* values (Present study)

It can be seen from the figures that, the numerical and experimental values for C^* are close to each other for the Reynolds range considered.

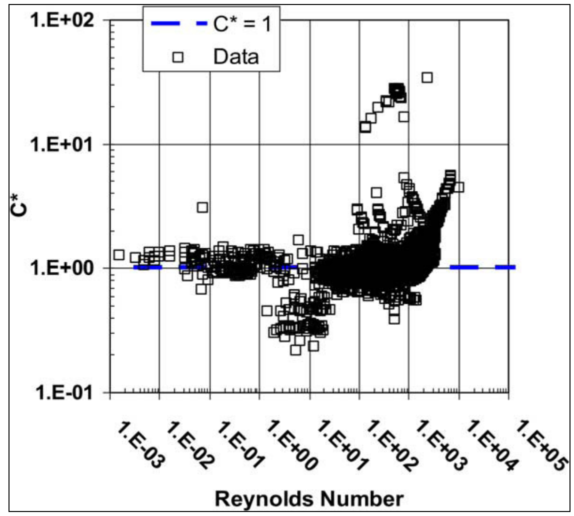

Figure 8: Experimental C* values [Steinke and Kandlikar (2006)]

As a result it can be concluded that the conventional friction factor correlation for turbulent flow is applicable for the diameter interval considered.

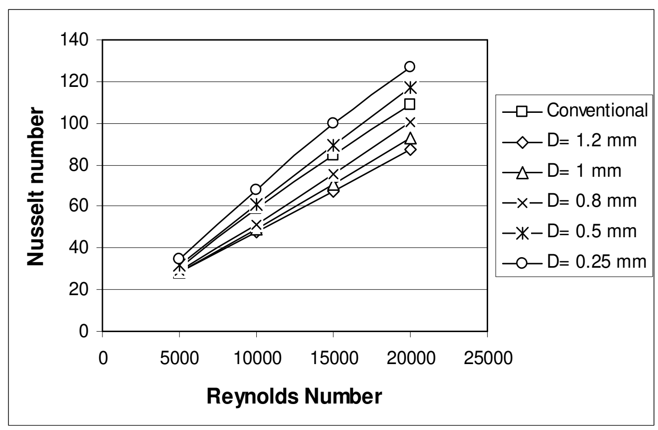

Figure 9: Nusselt number values wrt Reynolds number for different diameter channels

Conventional and numerical values of Nusselt number versus Reynolds number for minichannels investigated are shown in Figure 9, where the equation proposed by Gnielinski (1976) is taken to be the conventional correlation. In the figure it can be seen that numerical results are close to conventional values and Nusselt number increases with decreasing diameter and increasing Reynolds number. Numerical Nusselt number values are lower estimated for 1.2, 1 and 0.8 mm diameter channels while over estimated for those having 0.5 and 0.25 mm diameter. Thus 0.5 mm and below can be determined as the diameter value changing the quantity of Nusselt number.

In Figure 10, numerical Nusselt number values for 1.2 mm and 0.8 mm diameter channels are compared with their experimental values given by Owhaib et al. (2004).

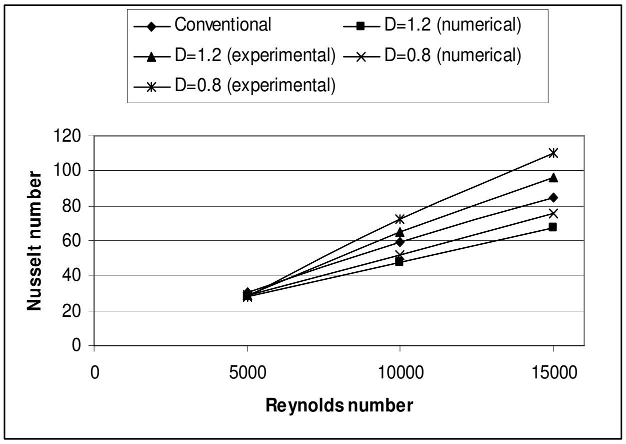

Figure 10: Conventional, numerical and experimental [Owhaib et al. (2004)] values of turbulent Nusselt number

As seen from Figure 9 and 10, applicability of the conventional Nusselt number correlations for minichannels considered can be concluded; although the numerical results under-predict the values once compared to those of experimental.

CONCLUSIONS

In this paper conventional turbulent models are examined for minichannels having diameter between 1.2 and 0.5 mm. Then, friction factor and Nusselt number values are estimated via the turbulence model that is determined to be the most appropriate among other models. Numerical results are compared to the values calculated with conventional correlations in the literature. Outcomes of the study can be listed as follows:

- Since radial velocity gradient from the wall increases significantly with decreasing diameter, grid size in this region should be well adjusted and the numerical simulation should focus on the near wall treatment, in order to avoid the divergence. Therefore, it is suggested to use enhanced wall treatment rather than wall functions when turbulent flow in mini/micro channels is studied.
- According to the comparison between the numerical and conventional values of friction factor and Nusselt numbers for 0.5 mm diameter minichannel, Standard k-ε model is determined to be the most appropriate turbulence model among the others.
- Friction factor values estimated with Standard k-ε model for 1.2-0.25 mm diameter channels are found to be close to the conventional values. Therefore it is concluded that conventional correlation for friction factor is applicable for the diameter interval considered.
- Most discrepancy in numerical friction factor values exists when diameter is less than and equal to 0.5 mm and thus it is determined to be a deviation point from the point of view of diameter.
- Nusselt number values calculated with Standard k-ε are found to be close to conventional values estimated with the correlation proposed by Gnielinski (1976).
- Numerical Nusselt number values are lower estimated for channels having diameter of 1.2, 1 and 0.8 mm while it is over estimated for 0.5 and 0.25 mm diameter channels. Hence as in friction factor results, there is a deviation for diameters below and equal to 0.5 mm.
- Furthermore, in order to validate the mesh structure, numerical friction factor values for laminar flow are compared with conventional and experimental results obtained by Yang et al (2003). In conclusion, conventional correlation for laminar friction factor is also applicable for minichannels considered.

NOMENCLATURE

q''_x	Local heat flux (W/m^2)
h_x	Local heat transfer coefficient (W/m^{2}K)
T_w	Wall temperature (K)
T_{ave}	Mass-weighted averaged fluid temperature (K)
Nu_x	Local Nusselt number
d, d_{hy}	Hydraulic diameter (m)
$\Delta\dot{m}$	Relative error for mass flux (%)
$\Delta\dot{Q}$	Relative error for heat flux (%)
k	Heat conduction coefficient (W/mK)
L, L_{hy}	Pipe length, hydrodynamic entrance length (m)
Gr	Grashof number
Re	Reynolds number
f	Friction factor
Pr	Prandtl number
C^*	Non-dimensional Poiseuille number

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