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NUMERICAL ANALYSIS OF AIR FLOW IN HELICALLY RIB-ROUGHENED TUBES

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Abstract. *This work presents numerical information for a single-phase forced convection in a circular tube containing a helically rib roughness. The analysis reports the heat transfer and friction characteristics for air flow with three helix angles (30, 49 and 70º). Several mesh options were evaluated, with 750.000, 1.000.000, 1.300.000, 1.500.000, 3.200.000 nodes. The numerical model results are compared with experimental results provided experimentally by Webb et al. (1980) and are not well correlated due to coarse mesh imposed by computational limitations.*

Keywords: *enhanced surfaces, rib-roughened tubes, forced convection, numerical analysis*

1. INTRODUCTION

Increase the capacity of a heat exchanger without implying a higher cost, increase its surface area, working fluid flow rate, or even use higher thermal conductivity materials, may be a challenge. Commonly used in heat exchangers, passive methods of performance does have like main characteristics the absence of external work to generate performance gain, such as slotted tubes. As the name implies, these tubes have internal grooves on its inner surface or micro fins that induce a secondary flow of fluid forming a vortex that mixes the low temperature fluid flowing in the central region of the tube with fluid which is in contact with the tube wall in a higher temperature, reducing mean temperature difference for heat exchange, increasing the heat transfer capacity of the heat exchanger. The increased heat transfer fluid generated by micro fins has a cost: higher pressure drop generated by the flow. Increasing pressure drop the working fluid flow rate is reduced aiming to maintain the same flow rate value a higher pumping power is required. This work aims to develop a computational model for rib-roughened tubes studied experimentally by Webb et al. (1980) using CFD tolls and use Webb's results to validate numerical models.

2. LITERATURE REVIEW

According to Bergles et al. (2000) performance increasing techniques could be classified as active or passive techniques. Active techniques use external energy or work to induce performance enhancement, such as mechanical or electrical aid, acoustics excitation or vibration source. However, passive methods consists in special geometries surfaces that provides disturbances to the flow, increasing heat exchanger performance without additional energy or any kind of external work , for example: rib-roughened inner or outer surfaces.

Many studies have performed with passive techniques of heat exchanger performance improvement, specifically rib-roughened surfaces, Bergles and Webb (1978) registered in their research more than 282 papers about forced convection over enhanced surfaces. Jakob and Fritz (1931) showed the influence of surface finish has on the convection heat transfer coefficient during ebullition processes. Nikuradse (1933) conducted experiments with a series of tubes, each one internally coated with sand grains classified as grain size. Nikuradse's data in the transitionally rough regime show the friction factor gradually increasing above the smooth curve, reaching a minimum value, before rising and leveling off to the fully rough value. Its intuitive that flow over a rough surface will experience a larger drag than a smooth surface, it is not completely clear when and how a rough surface begins affect the mean velocity profile, much less the higher order moments and turbulence structure. As the Reynolds number increases, a point is reached where the size of the smallest eddies in the flow is comparable to the size of the roughness elements k, and the viscous effects are no longer sufficient to damp the effects of the perturbation. Beyond this transitional regime, a point is reached at higher Reynolds numbers where the flow becomes fully rough, and further increases in Reynolds number no longer have an effect on the friction factor. Also, Nikuradse (1933) created a dimensionless friction factor correlation for flow over these kinds of tubes, as shown on equation (1), (2) and (3) and almost all the subsequent rough-wall studies compare the value of the friction factor in the fully rough regime with Nikuradse's values and hence assign an equivalent sand grain roughness to the surface under consideration.

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$$B(e^{+}) = \sqrt{\frac{2}{f}} + 2{,}5 * \ln\left(\frac{2e}{D}\right) + 3{,}75 \quad (1),$$

$$e^{+} = \frac{e * u^{*}}{v} = (\frac{e}{D})Re\sqrt{f/2} \quad (2),$$

$$f = \frac{\Delta P * D}{2 * L} * \frac{\rho}{G^{2}} \quad (3),$$

where $B(e^{+})$ is the friction factor parameter developed by Nikuradse (1933), e^{+} is roughness Reynolds number, f is the friction factor, $e\ (m)$ is the rib height, $D\ (m)$ specifies inside diameter of pipe (at base of ribs) , $\Delta P\ (Pa)$ is the pressure drop, $L\ (m)$ represents flow length, $\rho\ \left(\frac{Kg}{m^{3}}\right)$ is the fluid density e $G\ \left(\frac{Kg}{s\ m^{2}}\right)$ specifies mass velocity (mass flow per unit area).

Webb et al. (1971) developed an experimental analysis for pressure drop and heat transfer for turbulent flow in tubes having repeated-rib roughness, as shown on figure 1, using air as working fluid. Also, developed correlations based on law of the wall similarity for friction, same method employed by Nikuradse (1933) for sand grain roughness and heat-momentum transfer analogy for heat transfer correlation, which was first used by Dipprey and Sabersky (1963). The correlations proposed by Webb et al. (1971) are verified with experimental data taken with $0{,}01 < e/D < 0{,}04$ and $10 < p/e < 40$ and covering the range $0{,}71 < Pr < 37{,}6$.

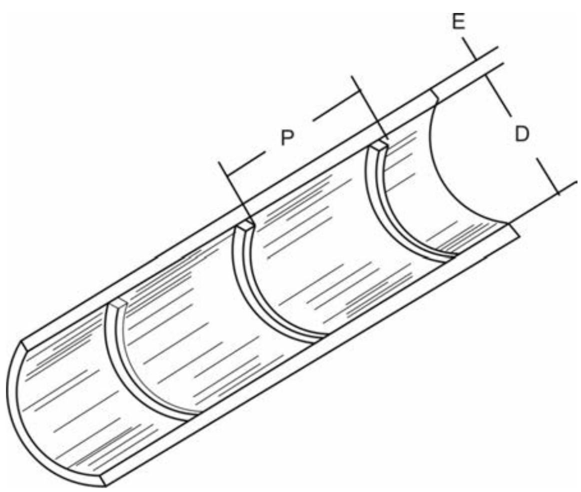

Figure 1: Geometrie studied by Webb et al. (1971).

The studies performed by Webb et al. (1971) concluded that repeated-rib friction data are well correlated using law of the wall similarity with a logarithmic velocity distribution, as shown on equation 4, 5 and 6, also the heat momentum transfer analogy, based on law of the wall similarity, adequately correlates the repeated-rib heat transfer data over a wide range of *e/D, p/e* and *Pr.*

$$u^{+} = A\ ln(y^{+}) + B \quad (4),$$

$$y^{+} = \frac{yu_{*}}{v} \quad (5),$$

$$u_{*} = \sqrt{\left(\frac{\tau_{0}}{\rho}\right)} \quad (6),$$

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Continuing with previous studies, Webb et al. (1980) returned to study rib-roughened tubes, in this case helically ribs, figure 2 shows the pipes configuration used on experiment:

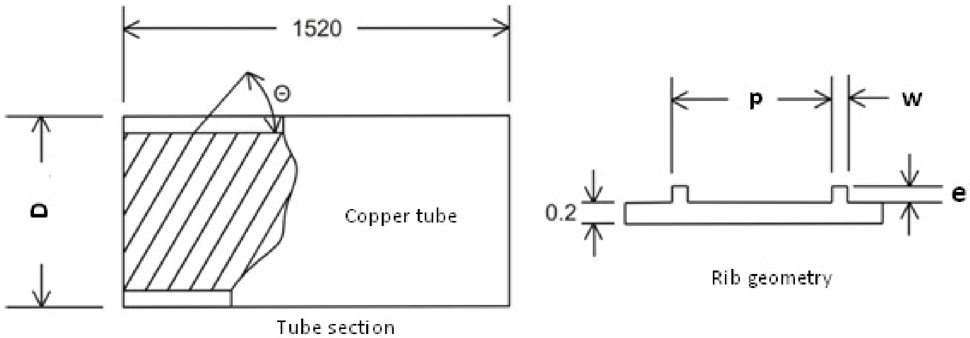

Figure 2: Helically rib-roughened tube used by Webb et al. (1980).

where $D\ (mm)$ is the pipe inner diameter, $p\ (mm)$ is the distance between ribs, $w\ (mm)$ represents ribs width equals to 0,3 mm, $e\ (mm)$ ribs height equals to 0,3 mm and Θ is ribs helix angle.

Webb et al. (1980) used in their analysis the fallow dimensionless parameter: *e/D, w/e, p/e and* Θ, representing an additional variable in their work comparing with Webb et al. (1971). In their experiment, Webb et al. (1980), analyzed three finned tubes configuration and one smooth tube sample, as shown on table 1, for experiment validation based on literature.

Tube	θ	p
1	70°	3,81 mm
2	49°	3,81mm
3	30°	3,81mm

Table 1: Finned tubes configuration used by Webb et al. (1980).

In their study, Webb et al. (1980) used air as working fluid, experimenting in Reynolds number between 6000 < Re < 65000 and as well as their work in 1971, measure pressure drop (ΔP) in 1,52 m length test section, between air inlet and outlet For flow development, another rib roughened pipe segment, with 1,52 m length was positioned before test section. In test section, eight thermocouple equally spaced along pipe section were used to measure temperature gain during flow, externally another thermocouple was installed to measure tube wall temperature. In order to minimize measure error temperature gain was limited from 8 to 12 K, also air flow was considered developed when temperatures values do not vary for at least three minutes as well as air outlet temperature. Before initiating experiments with rib roughened tubes a smooth tube was tested and determined friction factor value and heat transfer coefficient and compared measured values with Prandtl-Karman calculated friction factor value and Petukhov-Popov equation for Stanton number, aiming to calibrate and validate the experiment bench. For their analysis for friction factor on finned tubes, correlations developed by Nikuradse (1933) were used, described on equations 1, 2 and 3 especially modified for helically ribs, as shown on equation 7.

$$B(e^+) = \left[\sqrt{\frac{2}{f}} + 2{,}5 * \ln\left(\frac{2e}{D}\right) + 3{,}75\right] * \left(\frac{\alpha}{50}\right)^{0{,}16} \quad (7),$$

where $\alpha\ (°)$ represents rib helix angle. For heat transfer analysis, a correlation developed by Dipprey and Sabersky (1963) , model based on heat-momentum transfer analogy applied to a two region flow model, assuming a roughness influenced viscous wall region and a turbulent outer region that is insensitive to roughness, showed on equation 8.

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$$g(e^+, Pr, \alpha) = [\frac{f/2St-1}{\sqrt{f/2}} + B(e^-, \alpha)](\frac{\alpha}{50})^j \quad (8),$$

where g represents heat transfer correlating parameter for helical rib roughness, e^+ is roughness Reynolds number, Pr Prandtl number, α (°) helix number, f its friction factor, St is Stanton number and $B(e^+, \alpha)$ as given on equation (7) represents friction factor correlating parameter for helical rib roughness. The rough tube Stanton number data are shown in figure 3, figure 4 shows friction factor vs. Reynolds number.

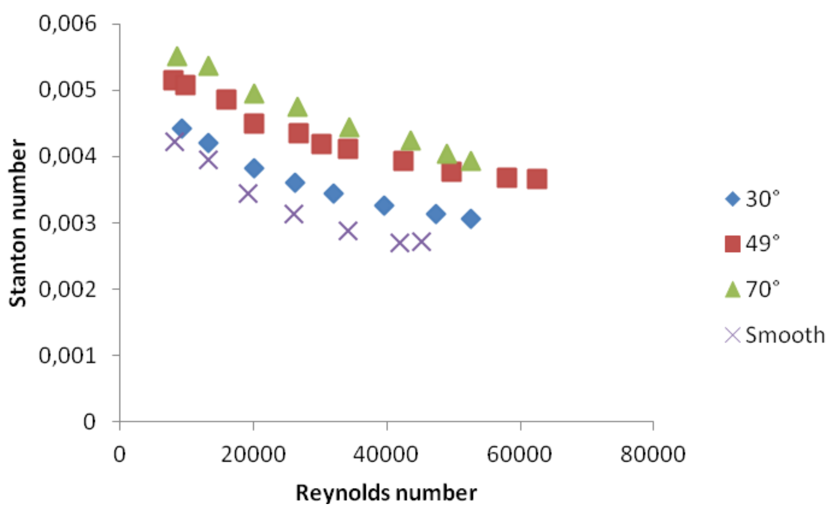

Figure 3: Stanton number vs. Reynolds number for tested ribs helix angles studied by Webb et al. (1980)

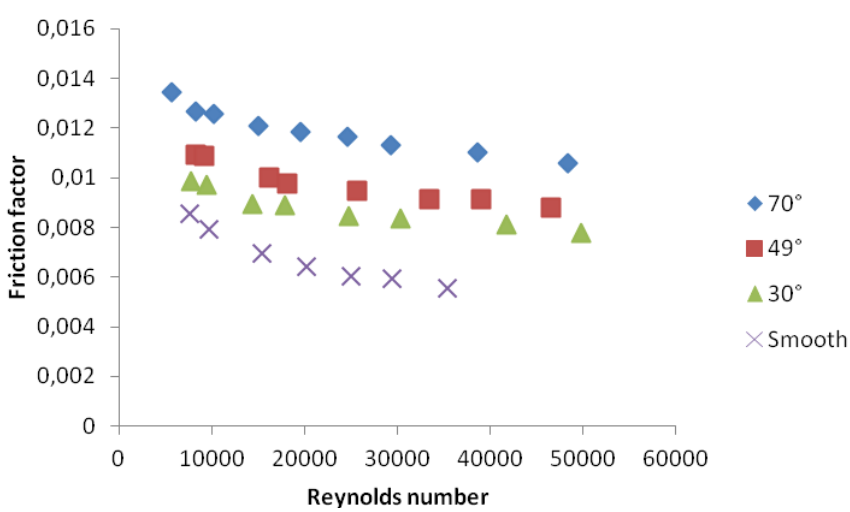

Figure 4: Friction factor vs. Reynolds number for tested ribs helix angles studied by Webb et al. (1980)

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These figures show the Stanton number decreases as helix angle is decreased. However, further examination on figure 5 shows the friction factor decreases at a greater rate than Stanton number. This suggests that the friction contribution due to form drag a large helix angles is greater than the heat transfer due to the flow separation and reattachments. It also suggests the definition of an optimum helix angle, which will give maximum thermal performance per unit friction power. Webb et al. (1980) concluded that the analysis clearly establishes that helical rib roughness yields greater heat transfer per unit friction than transverse rib roughness, studied by Webb et al. (1971) and the preferred helix angle is approximately 49º, also analysis of the correlated results establishes a preferred operating condition of $e^+ \cong 20$, providing high heat transfer performance with minimum friction penalty, as shown on figure 6 and 7, also experimental data are well correlated with Nikuradse (1933) model, Dipprey and Sabersky (1963).

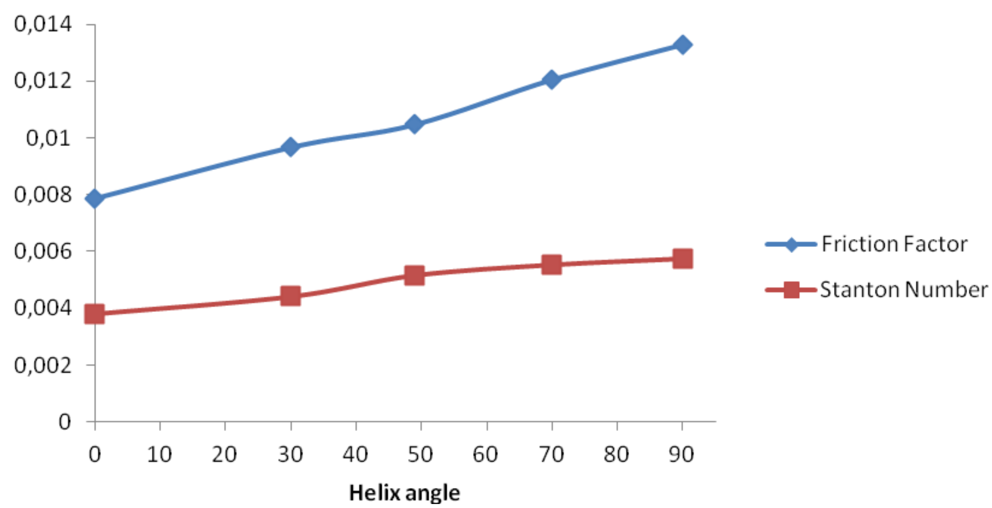

Figure 5: Friction factor and Stanton number vs. helix angle for Re=10000.

Several experimental studies about tubes with enhanced inner surface proved its advantages compared with smooth tubes. An alternative to experimental analysis is use Computational Fluid Dynamics (CFD) tool to solve flow problems and through numerical methods perform calculation required to simulate an interaction of liquids and gases with volume control defined by boundary conditions.

Promvonge et al. (2011) performed a numerical work to examine turbulent periodic flow and heat transfer characteristics in a three dimensional square-duct with inline 60º V-shaped ribs placed on two opposite heated walls , as shown on figure 6, with isothermal-flux condition. The computations are based on the finite volume method with the SIMPLE algorithm for pressure-velocity coupling, governed by the Reynolds averaged Navier-Stokes (RANS) equations and the energy equation. For data validation simulations with smooth square duct was first performed and results compared with literature correlations and results is found to be in excellent agreement, for both Nusselt number and friction factor, within $\pm$ 5,3 and $\pm$ 4,8% deviation. Air is used as working fluid with the flow rate in terms of Reynolds numbers ranging from 10000 to 25000 and introduced at a uniform inlet velocity while a pressure outlet condition was applied at the exit.

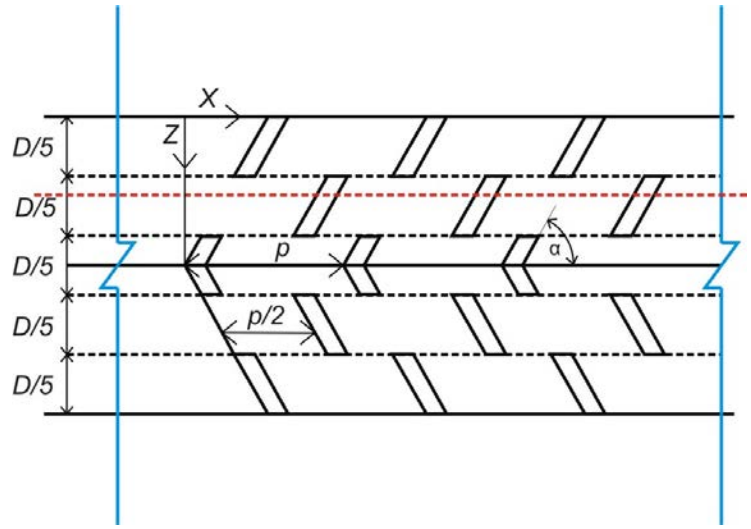

Figure 6: Periodic V-shaped ribs studied by Promvonge et al. (2011).

Promvonge et al. (2011) concluded that vortex formation between ribs helps to induce impingement flows on duct wall leading to higher heat transfer, from 200 – 370% higher than the smooth duct. However, the heat transfer augmentation is related to the increased friction loss ranging 8 – 10,5 times when compared with smooth duct.

Due to the benefits brought by Computational Fluid Dynamics numerical analysis, such as a faster development, savings time and costs decreasing number of experiments and based on experimental data developed by Webb et al. (1980), this study aims to develop and validate a numerical model for turbulent air flow on helically rib roughened tubes using commercial CFD code CFX 14.0 to analyze model flow characteristics.

3. PROBLEM MODELLING

3.1 Flow configuration

The flow system of interest is a horizontal helically rib roughened tube analyzed, described on figure 3, in three helix angle configuration, 30º, 49º and 70º, shown on table 1. As well as experimental study developed by Webb et al. (1980) the test section modeled for numerical simulation has 1,52 m length with eight variable measures points equally spaced along pipe simulation section. Assuming entering flow are developed with defined inlet velocity along pipe axis, the air enter the pipe at an inlet temperature, $T_{in} = 305\ K$, also uniform and constant heat flux on pipe wall, even in the ribs, in maintained at 1 kW/m².

3.2 Mathematical modeling

The numerical model for fluid flow and heat transfer in the helically rib roughened tube is developed under the following assumptions:

- Steady three-dimensional flow and heat transfer.
- The flow is periodic, fully developed and incompressible.
- Constant fluid properties.
- Body forces, viscous dissipation and radiation heat transfer are ignored.

Based on the above assumptions, air flow through the finned tube is governed by the Shear Stress Transport (SST) equations and the energy equation. The SST turbulence of Menter (1994) model is an eddy-viscosity model which includes two main peculiarities: It is a combination of k - ω model, used inside boundary layer and k - ε model on the boundary layer outer region; a limitation of the shear stress in adverse pressure regions is introduced. The k - ε model has two main weaknesses: it over predicts the shear stress in adverse pressure gradient flows and it requires near-wall

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modification, such as low Reynolds number damping functions. The k - ω turbulence model is better at predicting adverse pressure gradient flow and the standard model developed by Wilcox (1988) does not use any damping functions. However, the disadvantage of k - ω model is the dependence on the free-stream value of ω. In order to improve both the k - ε and k - ω model, Menter (1994) suggested a combination of those two models, described on equations 9 and 10.

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_i}\left(\Gamma_k \frac{\partial k}{\partial x_j}\right) + \widetilde{G_k} - Y_k + S_k \quad (9),$$

$$\frac{\partial}{\partial t}(\rho \omega) + \frac{\partial}{\partial x_i}(\rho \omega u_i) = \frac{\partial}{\partial x_i}\left(\Gamma_\omega \frac{\partial \omega}{\partial x_j}\right) + \widetilde{G_\omega} - Y_\omega + D_\omega + S_\omega \quad (10),$$

where, k $\bar{G}_k^{'}$ represents turbulence kinetic energy generation due to mean velocity gradient, $\bar{G}_\omega^{'}$ represents ω generation, Γ_k and Γ_ω represents the effective diffusivity of k and ω, respectively. Y_k and Y_ω are k and ω dissipation due to turbulence, D_ω represents cross diffusion term, S_ω and S_k are user defined terms.

All governing equations were discretized by the High resolution advection scheme, using a finite volume approach. The solutions were converged when residual values were less than 10^{-4} for all variables. There are five parameters of interest in the present work, Reynolds number, defined by equation 11, Stanton number, defined by equation 12, friction factor, represented by equation 3, pressure difference between pipe inlet and outlet and heat transfer convection coefficient, defined by equation 13.

$$Re = \frac{\rho * v * D}{\mu} \quad (11),$$

$$St = \frac{h}{c_p \rho v} \quad (12),$$

$$h = \frac{Q}{A\, \Delta T} \quad (13),$$

where $\rho \left(\frac{kg}{m^3}\right)$ is the fluid density, $v \left(\frac{m}{s}\right)$ represents mean fluid velocity, $D\ (m)$ is the pipe inner diameter, $\mu\ (Pa * s)$ is the fluid dynamic viscosity, $c_p \left(\frac{J}{Kg * K}\right)$ is the fluid specific heat, $Q\ (W)$ is the heat flow, $A\ (m^2)$ heat transfer surface area, $\Delta T\ (K)$ is the temperature difference between pipe inner wall and fluid mean temperature.

3.3 Grid independence study

A grid independence procedure was implemented, testing five different grid arrangements on helically rib roughened tube geometries characterized on table 1 using air as working fluid. The mesh options have 750.000, 1.000.000, 1.300.000, 1.500.000, 3.200.000 nodes and all of them have a denser elements concentration on next to wall regions, variable results for each mesh option are shown on figure 7. Greater amount of nodes couldn't be tested due to computer limitations, not allowing further convergence analysis. Taking into account variables values convergence in function of number of nodes and limitations imposed by computational processing, the fifth option , with 3.200.000 nodes, was chosen, showed on figure 8. Also, the y^+ value was monitored ensuring accurate resolution on near wall region. The y^+ is a non-dimensional distance from the wall to the first mesh node, to use a wall function approach for a particular turbulence model with confidence, its necessary to ensure y^+ values are within a certain range. The SST turbulence model requires a small values of y^+, commonly smaller than 1. Larger y^+ values CFX uses logarithmic wall function to calculate the shear stress at the wall losing precision.

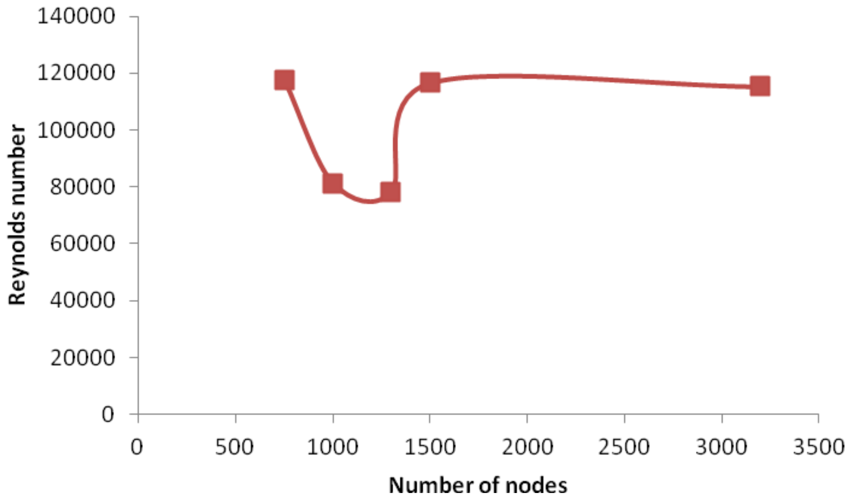

Figure 7: Mesh tests performed.

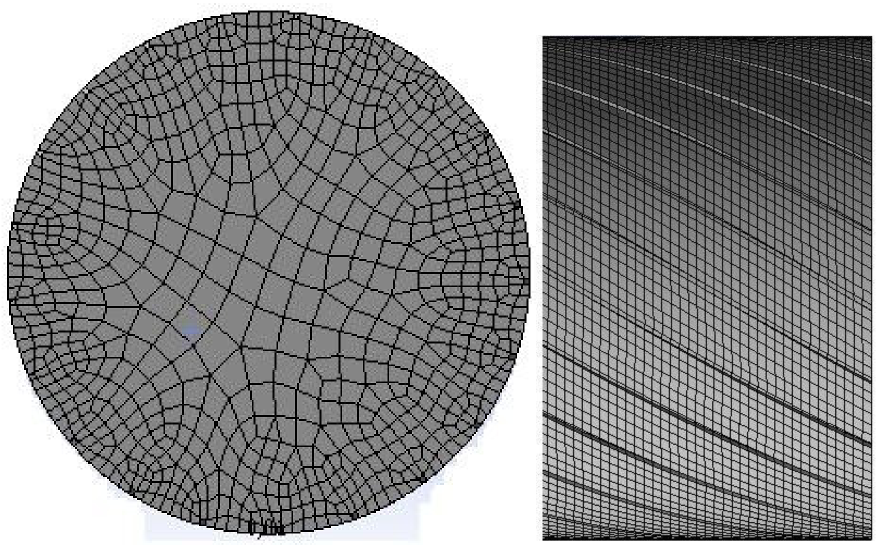

Figure 8: Mesh adopted on numerical simulations.

4. RESULTS AND DISCUSSION

Numerical simulation study was carried out using three different tubes in different inlet velocities shown at constant temperature (305 K) and heat flux (1 kW/m²) provided through the wall. The simulations performed with 3.200.000 nodes performed for three helix angles configurations: 30, 49 and 70º for all inlet velocities presented an averaged y^{+} value of 13, showing that on near wall region calculations were made through logarithmic wall functions, less accurate than SST model, especially for low Reynolds number values. Numerical simulations results on heat transfer, showed on figure 9, and friction factor on figure 10, demonstrate that helically rib roughened tube with 70° helix angle, produces largest Stanton number but also has largest friction factor. With 30° helix angle heat transfer and friction factor are lower than 49 and 70° as expected.

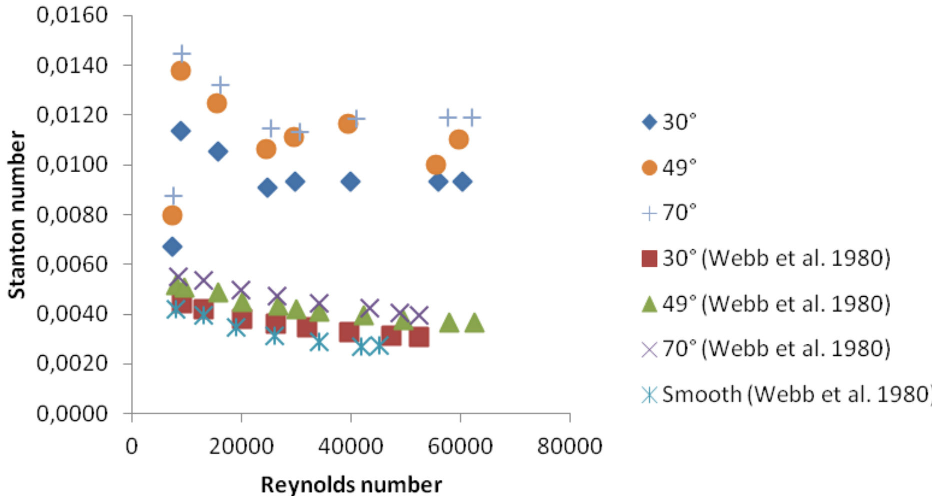

Figure 9: Stanton number vs. Reynolds number numerical results.

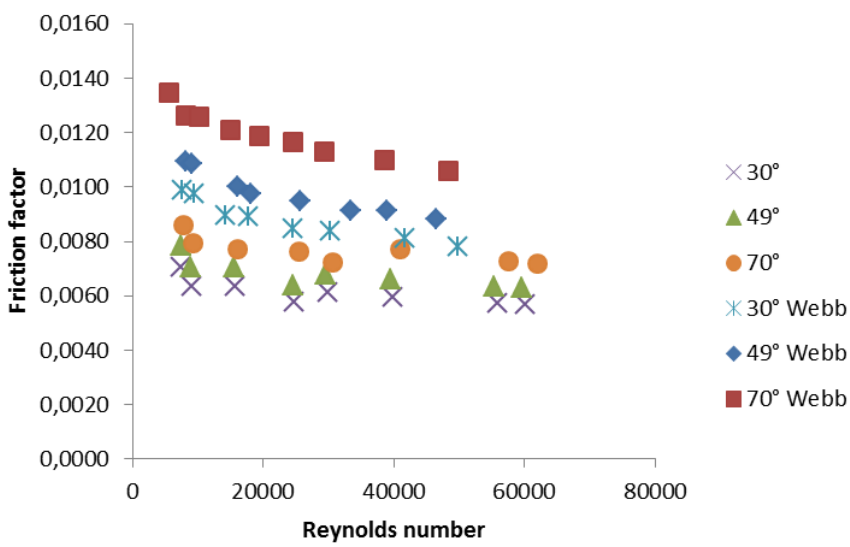

Figure 10: Friction factor vs. Reynolds number numerical results.

Stanton number and friction factor numerical values presented discrepancy when compared with experimental data developed by Webb et al. (1980). Also in free turbulence, such as smooth tubes, at sufficiently large Reynolds number the effect of viscosity on the over-all behavior and gross structure of the flow can be neglected, in grooved tubes, at any Reynolds number, there is a region close to the wall where the behavior of flow is determined by the fluid viscosity. At a distance from the wall, the direct effect of fluid viscosity on the turbulence gross structure may became small and negligible. Those distinct regions couldn’t be determined on simulations evaluated due to course mesh on near wall region. Figure 11 show near wall region flow on axis direction in function of turbulence kinetic energy, at this region a detachment boundary layer zone should be seen fallowed by reattachment, vortex or secondary flow developed.

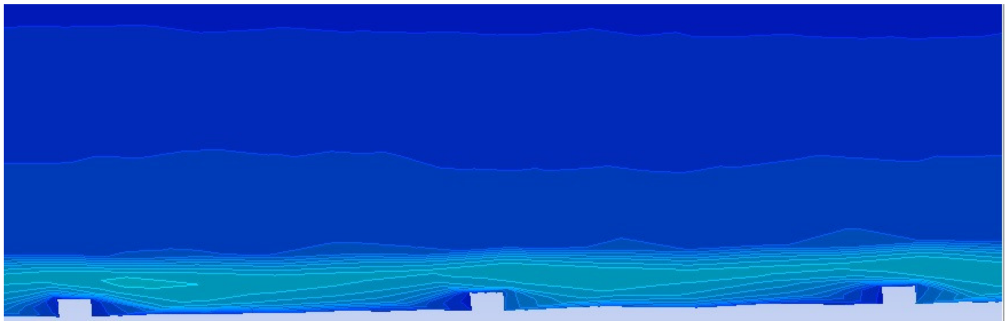

Figure 11: Turbulence kinetic energy on near wall region.

5. CONCLUSION

Numerical simulations for a single-phase forced convection in a circular tube containing a helically rib roughness were developed based on experimental data provides by Webb et al. (1980). Results simulations showed a high y^+ values, demonstrating that simulations were performed with a coarse mesh, further mesh refining weren't made due to computational limitations. Limitation consequences are explicit comparing numerical vs. experimental results. Stanton number and friction factor numerical results showed same behavior that experimental data: Stanton number and friction factor increases increasing ribs roughened helix angle. Further analysis should be done in lower values of y^+ using more capable computer resources and reaching numerical values closer to experimental data.

6. ACKNOLEDGMENTS

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7. REFERENCES

Jakob,M. , Fritz, W., 1931. "Versuche über den verdampfungsvorgang". *Forschung auf dem Gebiete des Ingenieurwesens 2*, 435-447.

Nikuradse, J, 1933. "Laws of flow in rough pipes", *Translation of "Strömungsgesetze in rauhen rohren."-VDI-Forschungsheft 361*.

Webb, R. L., Eckert, E. R. G. and Goldstein, R. J.,1971. "Heat transfer and friction in tubes with repeated-rib roughness". *Int J. Heat Mass Transfer 14*, 601 – 617.

Bergles, A. E. and Webb, R. L., 1978. "Bibliography on augmentation of convective heat and mass transfer". *Previews Heat Mass Transfer 4*, 89-106.

Gee, D. L., and Webb, R. L., 1980. "Forced convection heat transfer in helically rib-rouchened tubes". *Int. J. Heat Mass Transfer 23, 1127-1136.*

Wilcox, D. C., 1988. "Reassessment of the Scale-Determining Equation for Advanced Turbulence Models". *AIAA Journal 26. No. 11, 1299-1310.*

Menter, F. R., 1994. "Two-equation eddy-viscosity turbulence models for engineering applications". *AIAA Journal 32, No. 8, 269-289.*

Promvonge, P., Changcharoen, W., Kwankaomeng, S., Thianpong, C., 2011. "Numerical heat transfer study of turbulent square-duct flow through inline V-shaped discrete ribs". *International Communications in heat and mass transfer 38, 1392-1399.*