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# Numerical Modeling of Water Flow in an Annular Helicoidal Pipe: Hydrodynamics of Laminar and Turbulent flow

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**ABSTRACT**

In the present study, isothermal fluid flow in an annular helically-coiled tube has been modeled for different fluid flow regimes. The three-dimensional governing equations for mass and momentum have been solved, and k-ε turbulence model was used to model the turbulent flow regime. The fluid is water at Dean numbers ranging from 125 to 5600. It was found that centrifugal forces create a high velocity region at the outer side of the annular helicoidal pipe walls. The acceleration forces acting on the fluid flow create high pressure region near the outer pipe wall. The friction factor values for both laminar and turbulent regimes were compared with the experimental data and correlations from literature. Friction factor decreases as the tendency for turbulence increases. The best consistency between correlation and CFD model was observed for turbulent regime.

**KEYWORDS:** Computational fluid dynamics, Annular hlicoidal pipe, Friction factor.

## 1. INTRODUCTION

Helical coiled tubes are of special interest because they have many practical advantages, such as compactness, easy manufacture and high efficiency in heat and mass transfer. They are extensively used in compact heat-exchangers, heat-exchanger network, heating or cooling coils in the piping systems, intake in air-crafts, fluid amplifiers, coil steam generators, refrigerators, nuclear reactors, thermosyphones, and other heat transfer equipment involving phase change, chemical plants as well as in the food and drug industries [1].

The pipe curvature causes centrifugal forces to act on the flowing fluid, resulting in a secondary flow pattern perpendicular to the main axial flow. This secondary flow pattern generally consists of two vortices, which move fluid from the inner wall of the tube across the center of the tube to the outer wall. Upon reaching the outer wall it travels back to the inner wall following the wall [2].

Mishra and Gupta [3] conducted an experimental investigation to study momentum transfer in curved pipes. The effect of pitch on the friction factor at the same diameter was observed, and as the pitch increased, the centrifugal forces decreased for the same Reynolds number, thus weakening the secondary flow field, and ultimately straight tube behavior was approached. Naphon and Wongwises [4], in particular, recently reviewed more than 100 studies addressing both single and two-phase flow through curved tubes. Two-phase flow pressure drop in helical coils, in particular, received considerable attention, both in adiabatic and diabatic flow conditions.

Murai et al. [5] studied structure of air–water two-phase flow in helically coiled tubes. The obtained results show that owing to the curvature of the tube, which provides centrifugal acceleration to the two-phase flow, the flow transition from bubbly to plug flow is considerably quickened compared to that in the flow in a straight tube. Also, in comparison with an upward inclined straight tube, small bubbles vanish away from the liquid slug in the case of a strong curvature owing to the centrifugal acceleration. M. Moawed. [6] investigated the forced convection from helical coiled tubes with different parameters**.** Their results showed that, for the same $P/d_o$ (Pitch of coil tube/Outer diameter of coil tube), the higher values of Nusselt number can be obtained with a high value of $D/d_o$ (Coil diameter/ Outer diameter of coil tube) while the small value of Nusselt number can be obtained with a small value of $D/d_o$. Austen and Soliman [7] studied laminar flow and heat transfer in helically coiled tubes with substantial pitch. The influence of pitch on the pressure drop and heat transfer characteristics of helical coils was explored for the condition of uniform input heat flux. Water was used as the test fluid. Significant pitch effects were noted in the friction factor and Nusselt number results at low Reynolds number.

Garimella et al. [8] have investigated the heat transfer in coiled annular ducts to estimate the heat transfer coefficient inside the inner duct to map the shell side head transfer coefficient. They reported that coiling augments the heat transfer coefficient above the values for a straight annulus in the laminar region. However, this augmentation was less than that for a coiled circular tube and decreases at the transition region. Xin et al. [9] studied the effects of coil geometries and the flow rates of air and water on pressure drop in both annular vertical and

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horizontal helicoidal pipes. Experiments were performed for the superficial water Reynolds number from 210 to 23,000 and superficial air Reynolds number from 30 to 30,000. Their results showed that the transition from laminar to turbulent flow covers a wide Reynolds number range. Choi et al.[10] numerically studied the steady laminar flows in coiled annular ducts and observed the evolution of secondary flow and the effect of radius ratio on the flow development. It was concluded that the flow in a curved annular duct is not necessarily fully developed earlier when the radius ratio was larger owing to the complicated interaction between the viscous and centrifugal forces. Annular helicoidal pipes could not only provide more compactness in volume, but also controlled main flow and secondary flow due to the inner-wall boundary effects, making the friction factor and heat transfer characteristics different from straight circular tubes [11]. For annular helicoidal pipe, the experimental results of Xin et al.[12] indicated that the single-phase and two-phase flow pressure drops in annular helicoidal pipe differ from that in helical pipe with a circular cross-section. The main aim of this study is to examine the CFD of single-phase flow of water in an annular helicoidal pipe, picturing the velocity and pressure distributions and to make a comparison between the calculated friction factor from CFD model, experimental data and well-known correlations.

**2. Experiments**

Fig. 1 shows the geometry of test section. The setup consisted of an annular helicoidal pipe with 9 turns, inner coil of 9.53 mm diameter (I.D.), and outer coil of 6.22 mm. The coil has a length of 2.286 m when stretched. The diameter and the pitch of the outer coil are 196.85 mm and 25.4 mm, respectively [12].

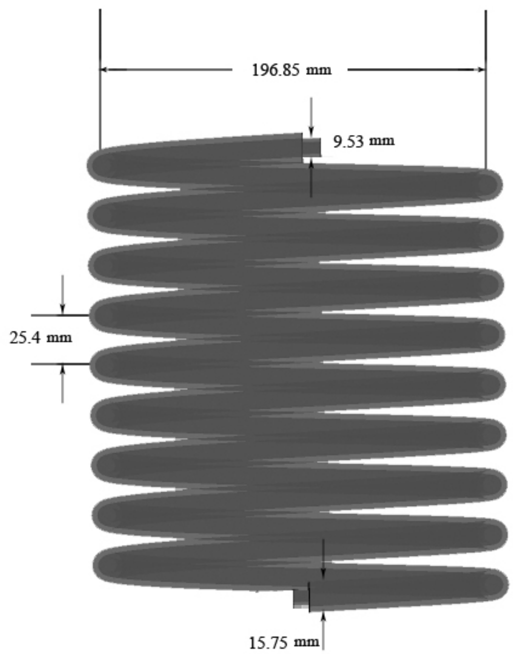


Fig.1. The schematic of annular helicoidal pipe

**3. Governing equations**

**3-1 Continuity equations**

$$\nabla . \left( \rho \mathbf{V} \right) = 0 \tag{1}$$

**3-2 Momentum conservation**

$$\nabla.(\rho\ \mathbf{VV}) = -\nabla P + \nabla.(\mu_{eff} (\nabla \mathbf{V} + (\nabla \mathbf{V})^{T})) + B \tag{2}$$

**B** describes the body force acting on the fluid.

The $k-\varepsilon$ turbulence model is used for turbulency modeling in transient and turbulent flows.

$$\nabla.(\rho V k) - \nabla.((\mu + \frac{\mu_t}{\sigma_k})\nabla k) = \mu_{eff} \nabla V.\nabla(\nabla V + \nabla(\nabla V)^T) - \frac{2}{3}\nabla.V(\mu_{eff}\nabla.V + \rho k) - \rho\varepsilon \quad (3)$$

$$\nabla.(\rho V \varepsilon) - \nabla.((\mu + \frac{\mu_t}{\sigma_\varepsilon})\nabla \varepsilon) = C_1 \frac{\varepsilon}{k}(\mu_{eff} \nabla V.\nabla(\nabla V + \nabla(\nabla V)^T)) - \frac{2}{3}\nabla.V(\mu_{eff}\nabla.V + \rho k) - C_2\rho\varepsilon \quad (4)$$

Where $\mu_{eff} = \mu + \mu_t$ and $\mu_t = C_\mu \rho \frac{k^2}{\varepsilon}$

Where $C_1 = 1.44$, $C_2 = 1.92$, $C_\mu = 0.09$, $\sigma_k = 1.0$, $\sigma_\varepsilon = 1.3$ are the standard $k - \varepsilon$ model constants.

**4. Boundary conditions**

At the inlet, for different Dean numbers of various flow regimes, corresponding normal speeds were used. No turbulence option was chosen for analysis of laminar flow. For transition and for turbulence cases, k-ε model with 5% and 10% inlet turbulent intensity were applied. Totally, 13 runs were made according to different Dean numbers for laminar, transient, and turbulent flows. In this study, at the outlet, pressure boundary condition was applied. No slip boundary condition was used at walls. In this analysis, a superficial velocity corresponding to Dean numbers 125, 140, 200, 300, 380, 600, 1100, 1550, 1900, 2300, 3200, 4200 and 5600 at the inlet are specified, respectively.

**5. Solution method**

The simulations were carried out as 3-D single-phase of water flow in an annular helicoidal pipe. Convergence criterion of 1.0E-5 for RMS of all of the equations were used. Hybrid scheme was used for discretization of all equations and SIMPLEC algorithm was used for pressure-velocity coupling. The effect of gravitational force is applied in this analysis.

## 6. RESULTS AND DISCUSSION

figures 2 and 3 show the contour plots of velocity at four positions; 1*180$^\circ$, 6*180$^\circ$ , 9*180$^\circ$ and outlet for different flow regimes. Thirteen Dean numbers of 125, 140, 200, 300, 380, 600, 1100, 1550, 1900, 2300, 3200, 4200 and 5600 for laminar, transient and turbulent flow were applied. As depicted in Fig. 2, more uniform high fluid speed between two tubes can be seen in the regions near the outer wall, as the Dean number increases. This is due to the effect of centrifugal force. But the slowest fluid speed was observed in the regions near to the inner tube walls. The location of high velocity region shifted slightly down in successive turns of pipe, and the band of very high water velocity (dark red band) is slightly wider, too. However, in the case of flow in a straight pipe, high velocity region is observed at the central regions of the pipe.

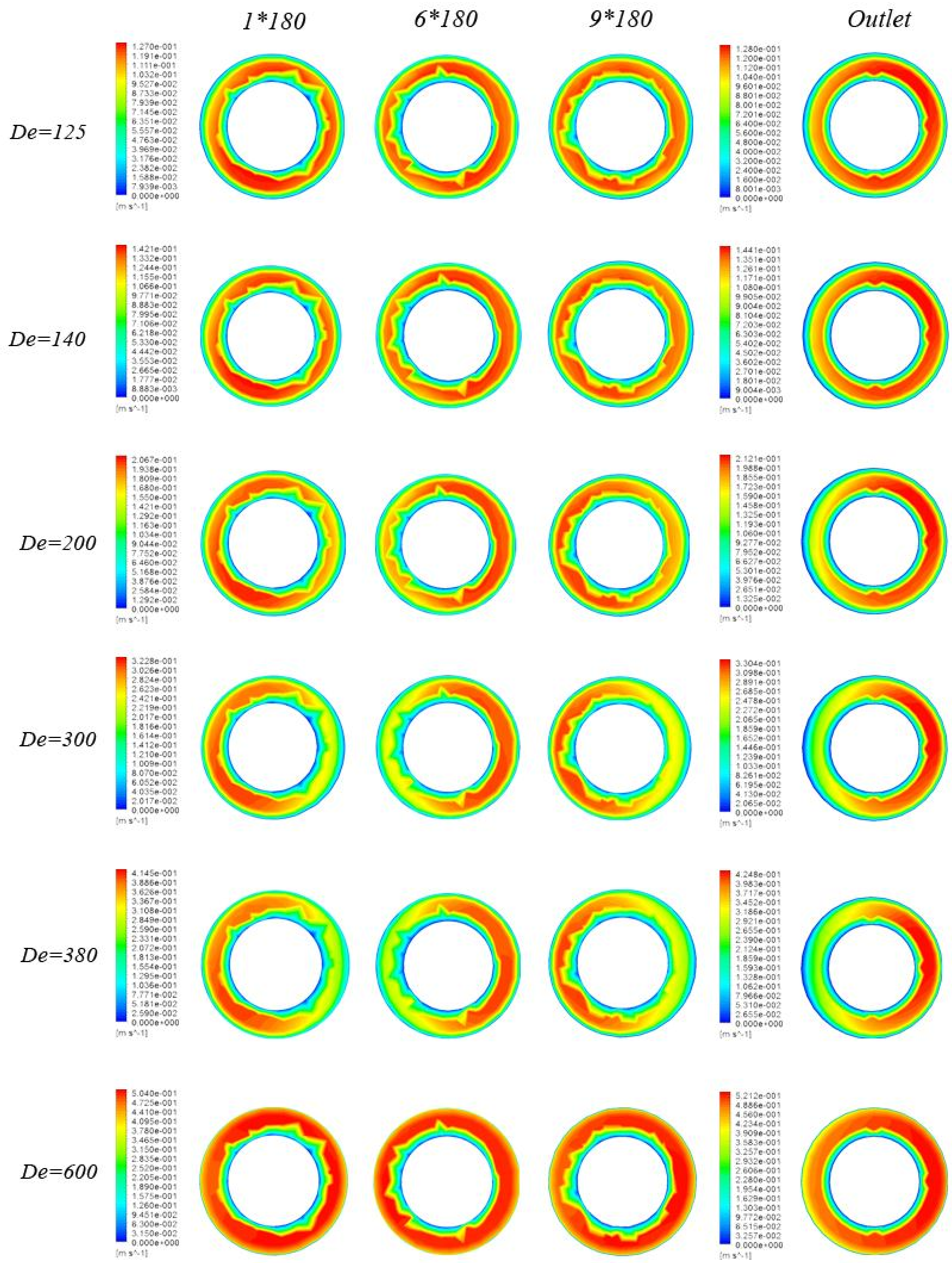


Fig.2. the contour plots of velocity at different positions (Laminar Regime)

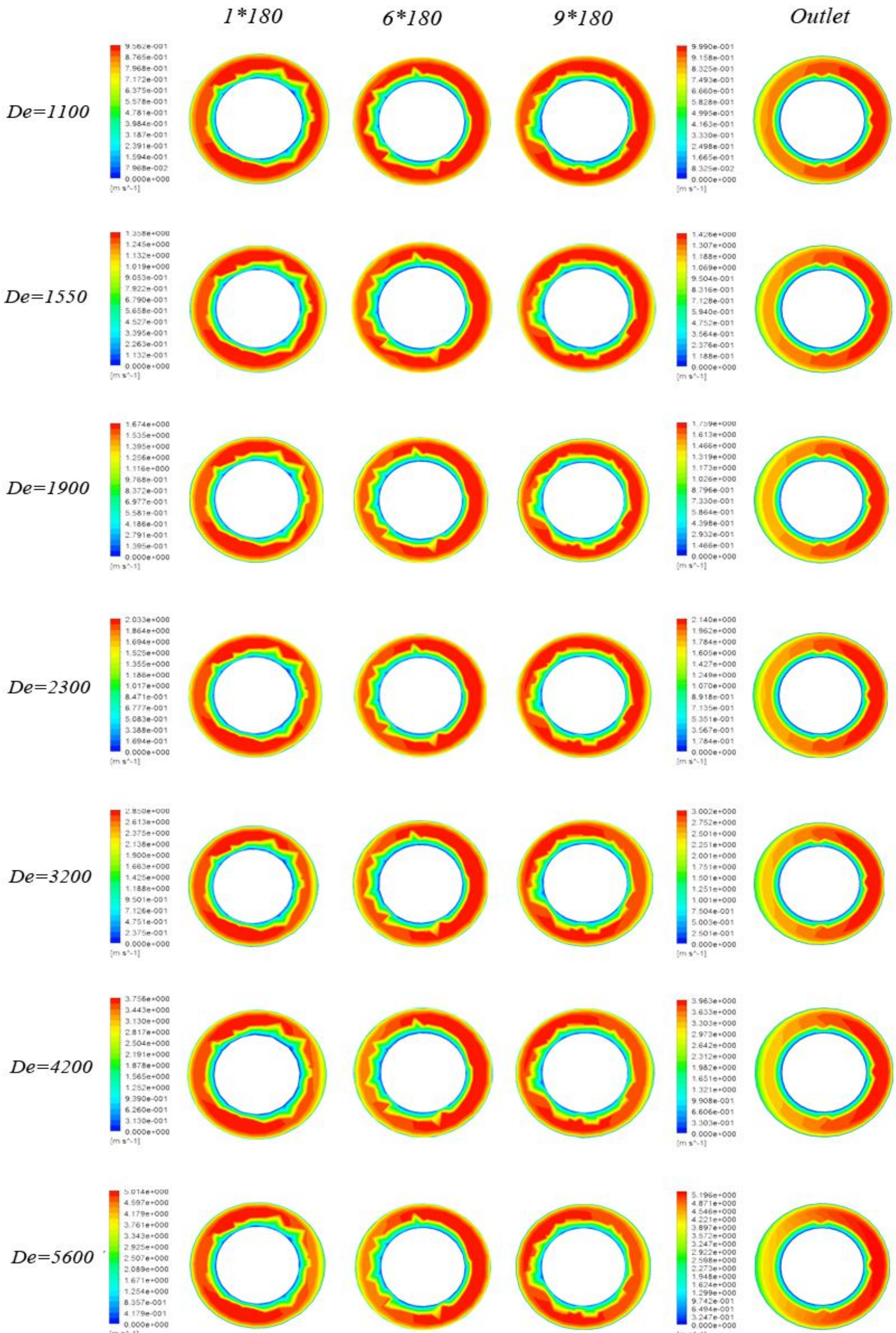


Fig.3. the contour plots of velocity at different positions (Turbulent Regime)

Figures 4 and 5 show the contour plots of pressure gradient at four positions; 1*180°, 6*180°, 9*180° and outlet of annular helicoidal pipe for water flow with 125, 140, 200, 300, 380, 600, 1100, 1550, 1900, 2300, 3200, 4200 and 5600 Dean numbers for laminar, transient and turbulent flow. The Reynolds and Dean numbers are defined as:

$$Re = \frac{\rho u(d_o - d_i)}{\mu} \tag{3}$$

$$De = Re\left(\frac{d_o - d_i}{D}\right)^{1/2} \tag{4}$$

The friction factor for single phase flow is determined using:

$$f = \frac{(\Delta p/L)(d_o - d_i)}{\frac{1}{2}\rho u^2} \tag{5}$$

The critical Dean numbers predicted by Srinivasan's equation [1] is 1139 for this test section.

The curvature of the pipe causes centrifugal force to act on the fluid. The acceleration forces acting on the fluid flow in the pipe create high pressure region nearby the outer wall. The CFD model results of pressure drop, were used to calculate the friction factor using Eq (5). CFD results of friction factor were compared with experimental data. Well-known correlations of Manalapaz and Churchill [13] for laminar regime and Ito [14] for turbulent regime were also used to calculate friction factor for each regime as shown in Fig. 6. Comparison of applied methods, based on the Mean Average Error (MAE) is presented in table 1.

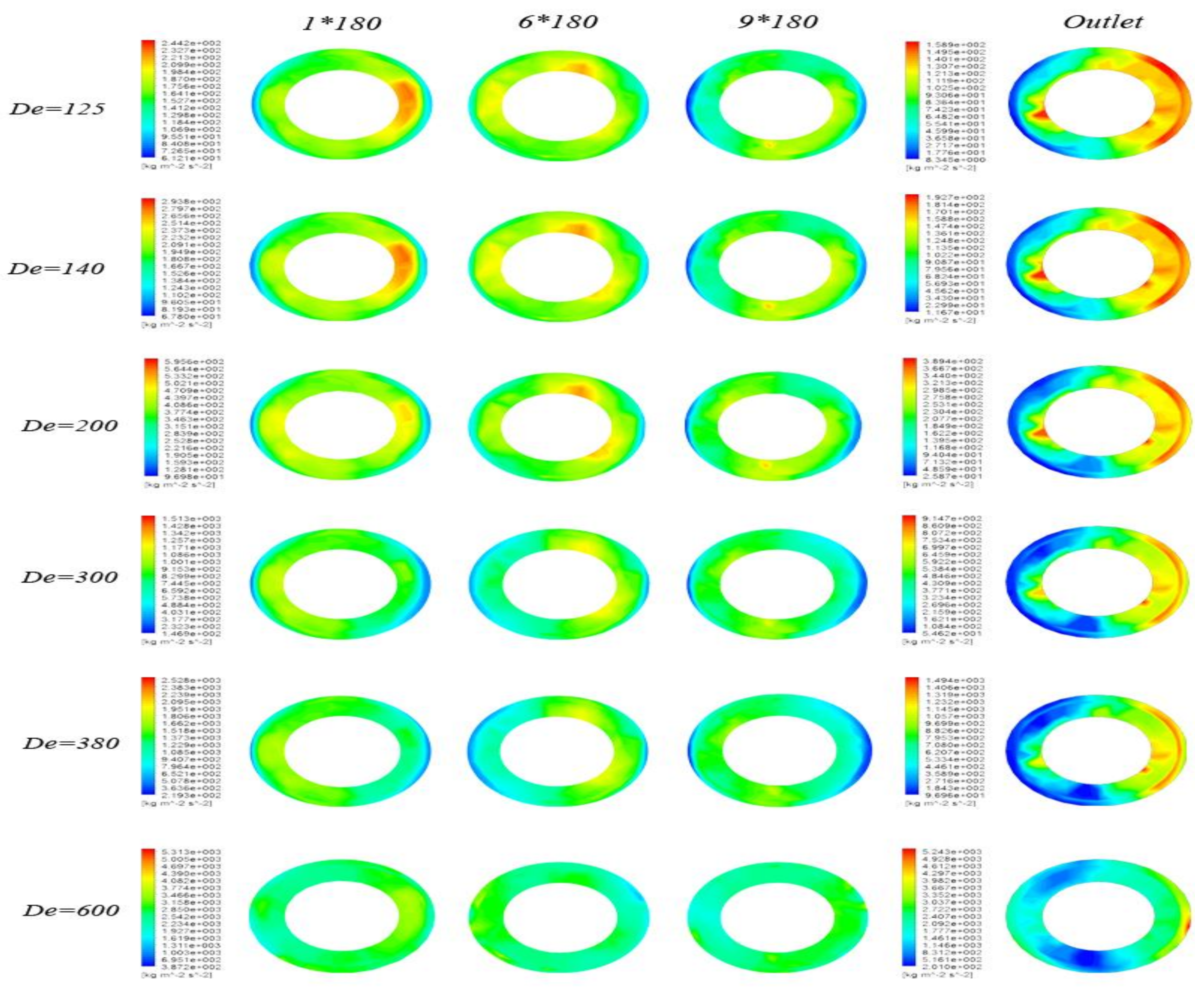


Fig.4. the contour plots of pressure gradient at different positions (Laminar Regime)

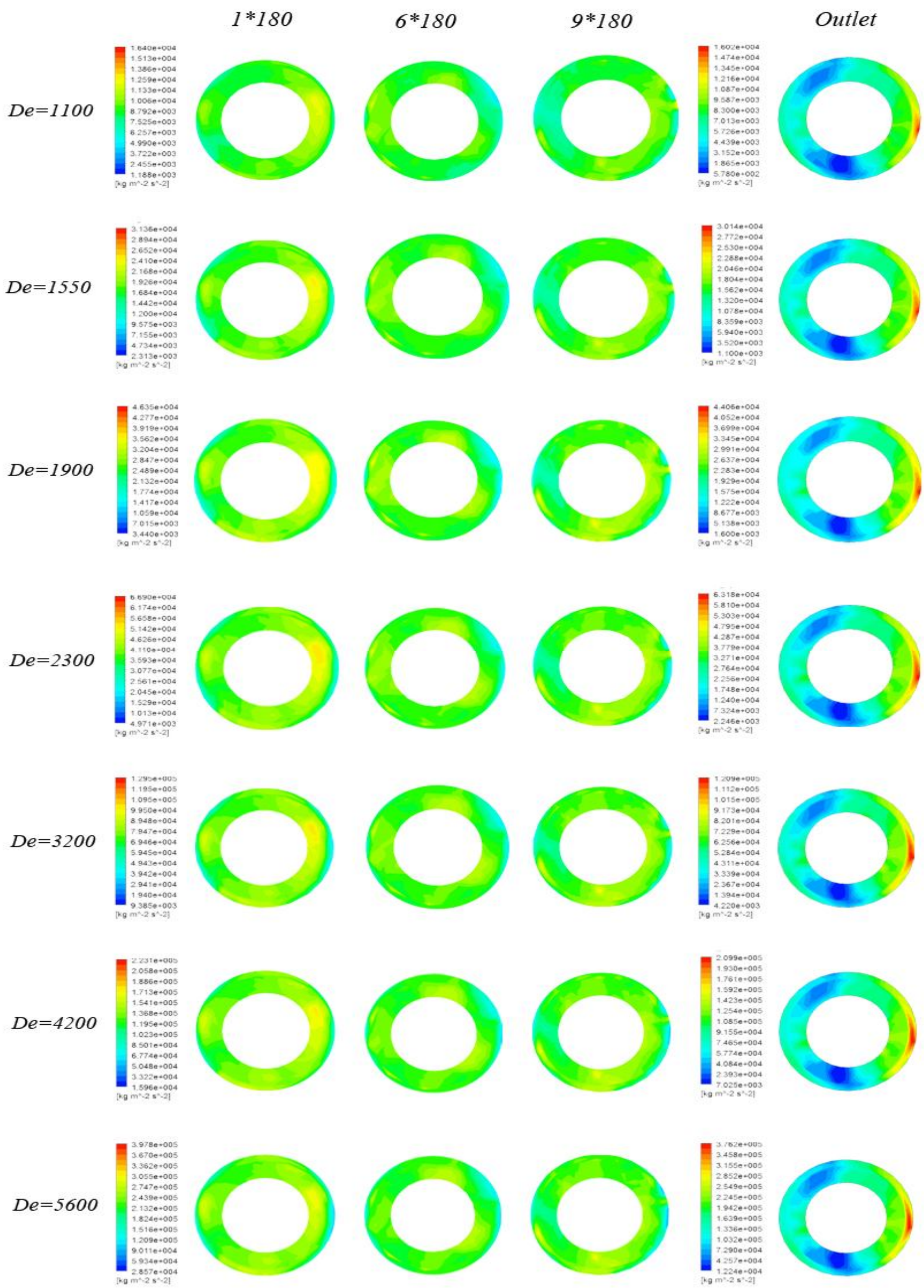


Fig.5. the contour plots of pressure gradient at different positions (Turbulent Regime)

Figure 6 shows the friction factors versus Dean Numbers. As can be seen in this figure, the friction factor decreases when increasing Dean number or on the other hand with enhancing velocity and, consequently, increasing turbulency. Also it can be seen that CFD model gives better results at turbulent regime with Dean numbers of higher than 1000. Mean average error of 0.23 and 0.057 were calculated for friction factor of experimental data compared with CFD model for laminar and turbulent regimes, respectively. Values of 0.072 and 0.12 were calculated for mean average error of friction factor for Manalapaz and churchill correlation relative to experimental data and Ito correlation compared with experimental data, respectively, as well. Values of 0.16, 0.058 of mean average error of friction factor were found for Manalapaz and churchill correlation compared with CFD model and Ito correlation compared with CFD model for laminar and turbulent regimes, respectively, too. Based on the results of Fig. 6 and Table 1, the CFD model can predict the friction factor with acceptable accuracy, when compared with experimental data. Similar to the Manalapaz and Churchill (1980) correlation- in comparison to experimental data- MAE for laminar flow is higher than turbulent flow. Although, the MAE of correlations relative to CFD model in both regimes are acceptable, but results are too closer for turbulent regime. The Manalapaz and Churchill's correlation for laminar regime shows more consistency with experimental data, as compared to the CFD model, while CFD model results are closer to experimental data in comparison with Ito's correlation for turbulent regime.

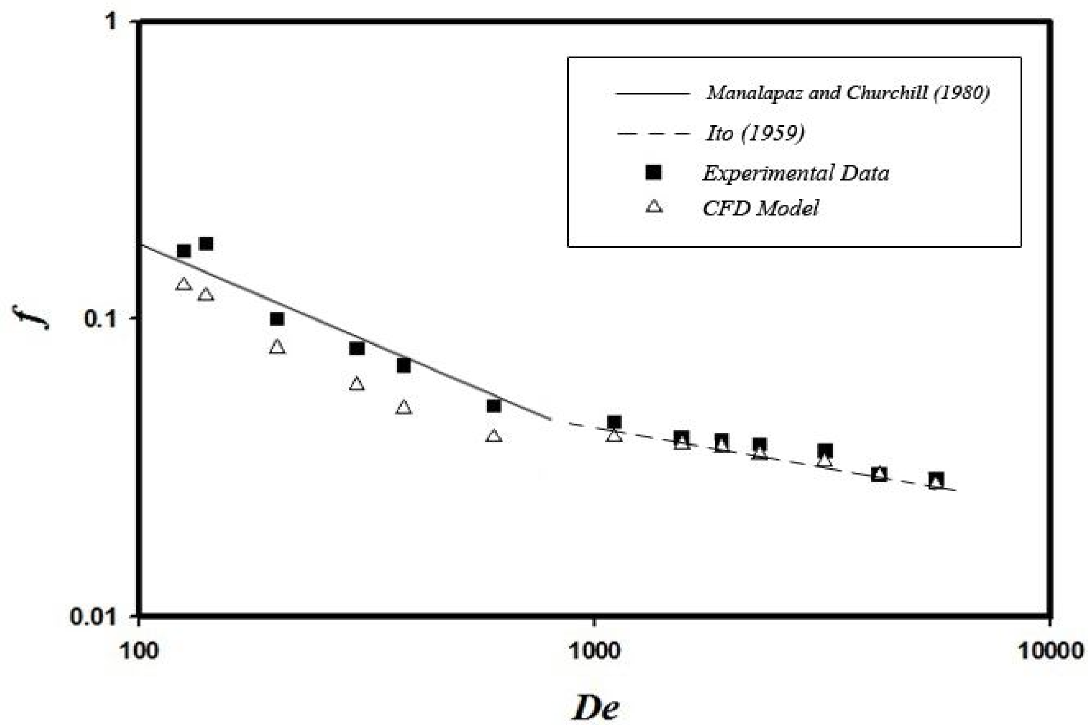


Fig.6. Friction factor for single liquid phase flow in annular helicoidal pipe

Table.1. Comparison of the friction factors of different methods for laminar and turbulent regimes

| **a) Laminar regime** | **MAE** |
|---|---|
| CFD Model-Experimental Data | 0.23 |
| Manalapaz Correlation-Experimental Data | 0.072 |
| Manalapaz Correlation-CFD Model | 0.16 |
| **b) Turbulent Regime** | |
| CFD Model-Experimental Data | 0.057 |
| Ito Correlation-Experimental Data | 0.12 |
| Ito Correlation-CFD Model | 0.058 |

## 7. Conclusion

Laminar and turbulent single-phase of water flow through an annular helicoidal pipe is simulated using computational fluid dynamics (CFD). It was found that centrifugal forces create a high velocity region near the outer wall of the annular helicoidal pipe. The acceleration forces acting in the fluid flow in the pipe create high pressure region nearby the outer pipe wall. Friction factor decreases as the turbulency increases. Although, the MAE of correlations- CFD model in both regimes are acceptable, but results are too closer for turbulent regime. The Manalapaz and Churchill's correlation for laminar regime shows more consistency with experimental data, as compared to the CFD model, while CFD model results are closer to experimental data in comparison with Ito's correlation for turbulent regime.

**Nomenclature:**

| | |
|---|---|
| $d_i$ | inner diameter (m) |
| $d_o$ | outer diameter (m) |
| $D$ | coil diameter (m) |
| $p$ | pressure (N/m$^2$) |
| $u$ | superficial velocity (m/s) |
| $x$ | spatial position (m) |
| $f$ | friction factor |
| $L$ | pipe length (m) |
| ***Greek letters*** | |
| $\rho$ | density of fluid (kg/m$^3$) |
| $\mu$ | dynamic viscosity (kg/ms) |
| $\Delta p$ | pressure difference (Pa) |
| ***Subscripts*** | |
| $i$ | inside/inner |
| $o$ | outside/outer |
| ***Dimensionless numbers*** | |
| $N_{Re}$ | Reynolds number |
| $N_{De}$ | Dean number |

## REFERENCES


[1] S. Ali, Pressure drop correlations for flow through regular helical coil tubes, Fluid Dynam. Res. 28 (2001) 295–310.

[2] S.A. Berger, L. Talbot, L.S. Yao, Flow in curved pipes, Annual Review of Fluid Mechanics 15 (1983) 461–512.

[3] P. Mishra, S. N. Gupta, Momentum Transfer in Curved Pipes, 1. Newtonian fluids, *Ind. Eng. Chem* 18 (1979) 130–137.

[4] P. Naphon, S. Wongwises, A review of flow and heat transfer characteristics in curved tubes, Renewable and Sustainable Energy Reviews 10 (2006) 463–490.

[5] Y. Murai, S. Yoshikawa, S. Toda , M. Ishikawa and F. Yamamoto "Structure of air–water two-phase flow in helically coiled tubes" Nuclear Engineering and Design, 236 (2006) 94–106.

[6] M. Moawed "Experimental study of forced convection from helical coiled tubes with different parameters", Energy Conversion and Management, 2 (2011), 1150-1156.

[7] D.S. Austen, H.M. Soliman. Laminar flow and heat transfer in helically coiled tubes with substantial pitch Journal of Experimental Thermal and Fluid Science 1(1988)183–194.

[8] S. Garimella, D.E. Richards, R.N. Christensen, Experimental investigation of heat transfer in coiled annular ducts, J. Heat Mass Transfer 37 (1998) 329–336.

[9] R.C. Xin, M.A. Ebadian, The effects of Prandtl numbers on local and average convective heat transfer characteristic in helical pipes, J. Heat Transfer 119 (1997) 467–473.

[10] H.K. Choi, S.O. Park, Laminar entrance flow in curved annular ducts, Int. J. Heat Fluid Flow 13 (1992) 41–49.

[11] G.T. Karahalios, Mixed-convection flow in a heated curved pipe with core, Phys. Fluids A 2 (1990) 2164–2175.

[12] R.C. Xin, A. Awwad, Z.F. Dong, M.A. Ebadian, An experimental study of single-phase and two-phase flow pressure drop in annular helicoidal pipes, Int. J. Heat Fluid Flow 5 (1997) 482–488.

[13] R. L. Manalapaz, and S. W. Churchill, Fully developed laminar flow in a helically coiled tube of finite pitch. Chem. Eng. Comm., 7 (1980) 57-78.

[14] H. Ito, Friction factor for turbulent flow in curved pipes. J. Basic Eng., 81(1959) 123-134.