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Laminar convective heat transfer in chaotic configuration

Vimal Kumar, K.D.P. Nigam *

Department of Chemical Engineering, Indian Institute of Technology, Delhi, Hauz Khas, New Delhi 110 016, India

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Abstract

The convective heat transfer in chaotic configuration of circular cross-section under laminar flow regime at different values of Dean number and Prandtl number is investigated numerically. The chaotic configuration is the combination of 90° bends and coils. The insertion of equidistant 90° bends between the two consecutive coil produces the phenomenon of flow inversion. The hydrodynamics and heat transfer under laminar flow conditions in the chaotic configuration with constant wall flux as a boundary condition is studied. The control-volume finite difference method with second-order accuracy is used. The chaotic configuration shows a 25–36% enhancement in the heat transfer due to chaotic mixing while relative pressure drop is 5–6%. The effect of Prandtl number on fully developed heat transfer coefficient is also reported. It is observed that heat transfer increases with increase in Prandtl number. The stretching and folding phenomenon in chaotic configuration is observed and discussed for heat transfer coefficient and pressure drop in the chaotic configuration. The cyclic oscillation behavior in the heat transfer coefficient with downstream distance in the chaotic configuration and coiled tube is also observed and discussed. It appeared that heat transfer is strongly influenced by flow inversion. The effect of boundary conditions on heat transfer performance in the chaotic configuration as well as in the coiled tube is also carried out. The study is further extended to predict hydrodynamics and heat transfer with temperature-dependent viscosity in the chaotic configuration. A comparative study for heat transfer and friction factor is also carried out for constant and temperature-dependent viscosity in coiled tube and chaotic configuration. It was observed that the heat transfer under heating condition with temperature-dependent viscosity is higher as compared to the constant viscosity result while friction factor shows the reverse phenomenon in the chaotic configuration.

Keywords: Heat transfer; Helical tube; Chaotic configuration; CFD; Uniform wall heat flux

1. Introduction

The modification of the flow in curved tube is due to the centrifugal forces caused by the curvature of the tube, which produce a secondary flow field with a circulatory motion pushing the fluid particles toward the core region of the tube. The laminar flow persists too higher Reynolds number value in helical coils as compared to straight tube, because of the stabilizing effect of this secondary flow. Consequently, the differences in heat and mass transfer performance between helical coil and straight tube is particularly distinct in the laminar flow region. Due to high heat transfer, coiled tubes are used in steam power plants, the water to be heated and boiled into steam is pumped through pipes that are exposed to the hot gases formed in the furnace section of the steam generator. The cooling water circulating through the coiled pipes is heated as the low-pressure steam leaving the turbine is condensed on the outer surface of these pipes. In home heating systems, hot water flows through curved pipes, transferring heat to the leaving space. Besides, this secondary flow also accounts for a considerable decrease in axial dispersion as compared to the straight tube. Extensive reviews on flow fields and heat transfer in curved ducts have been reported by Berger et al. [1], Shah and Joshi [2] and Nandakumar and Masaliyah [3] in literature after the pioneering work of Dean [4,5]. The nondimensional parameter characterizing flow in curved ducts is the Dean number N_{De}, defined

* Corresponding author. Tel./fax: +91 11 26591020.
E-mail address: nigamkdp@gmail.com (K.D.P. Nigam).

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Nomenclature

a radius of the helical pipe, m
A_n coefficient for thermo-physical properties ($n = 1, 2, 3, 4, 5$ and 6)
Cp specific heat, kJ/(kg K)
d diameter of the helical pipe ($= 2a$), m
D coil diameter
f friction factor
k thermal conductivity, W/(m K)
q heat flux, W/m^2
N_{De} Dean number ($= N_{\mathrm{Re}}\sqrt{d/D}$)
N_{Re} Reynolds number ($= \rho U_0 d/\mu$)
N_{Pr} Prandtl number ($= C_p\mu/k$)
N_{Nu} Nusselt number
T temperature, K
U_0 inlet velocity, m/s
u_i axial velocity component, m/s

Greek symbols
ϕ angle, degree
λ curvature ratio (D/d)
Φ wall flux, W/m^2
μ viscosity of fluid, kg/m s
v kinematic viscosity, m^2/s

Subscripts
0 inlet conditions
c coiled tube
m average quantity
s straight tube
w wall condition
θ local quantity

as $N_{\mathrm{De}} = N_{\mathrm{Re}}\sqrt{(d/D)}$, where N_{Re} is the Reynolds number ($U_0 d/v$), U_0 is the average streamwise velocity, d is the width of the channel, v is the kinematic viscosity, and D is the diameter of curved tube. A large variety of thermal boundary conditions can be specified for the energy equations: many such boundary conditions are summarized by Shah and Joshi [2]. The most important and also limiting boundary conditions are axially and peripherally constant wall temperature T_{w} and axially and peripherally constant heat flux at the wall (Φ_{w}). For constant wall temperature conditions Kripklov [6], Roger and Mayhew [7], Kubair and Kuloor [8], Jha and Rao [9], Shuchkin [10] and Janssen and Hoogendroon [11] reported the empirical correlations based on their experimental data, while Akiyama and Cheng [12] and Kalb and Seader [13] reported the heat transfer coefficient numerically. Seban and Mclaughlin [14], Jha and Rao [9], Dravid et al. [15], Singh and Bell [16] and Janssen and Hoogendroon [11] reported empirical correlations for the heat transfer coefficient for fully developed thermal region in case of a uniform peripherally averaged heat flux experimentally. Mori and Nakiyama [17] have obtained an analytical solution by analyzing the velocity and temperature profiles, while Akiyama and Cheng [12,18], Tarbell and Samuell [19], Kalb and Seader [13] and Patankar et al. [20] numerically studies the heat transfer in curved tubes using uniform wall flux. Janssen and Hoogendoorn [11] made a comparison of the overall heat transfer coefficients in case of a constant wall temperature and a constant averaged heat flux for low values of Dean number ($N_{\mathrm{De}} < 17$). They reported that the effect of the boundary condition on heat transfer to be small provided the flow could be considered as isoviscous. They also reported that the non-isoviscous flow effects on the heat transfer are larger than predicted by the Sieder–Tate [21] correction. Manalapaz and Churchill [22] reported the empirical correlations by analyzing the data for both type of boundary conditions (T_{w} and Φ_{w}) for various process conditions. All the researchers have reported that the heat transfer in coiled tube is higher as compared to straight tube because of secondary flows due to centrifugal forces. Beside this the phenomenon of cyclic oscillations in the heat transfer coefficient with increasing the downstream distance in the curved tube [11,14,15] was also reported experimentally and numerically.

Inserting a geometrical perturbation in the laminar flow further enhances the heat transfer in the curved geometry [23–35]. Acharya et al. [23,24], Mokrani et al. [25] experimentally and numerically studied the heat transfer enhancement in a chaotic coil, made of bends, immersed in a constant temperature bath and reported the higher heat transfer as compared to the coiled tube.

Kumar and Nigam [34] studied the heat transfer enhancement in chaotic configuration for the case of thermal boundary condition of constant wall temperature. The geometry considered was proposed by Saxena and Nigam [35,36], bending of helical coils, to cause multiple flow inversion at low flow rates. The heat transfer and hydrodynamics predictions were reported at Dean number ranging from 7 to 400 and Prandtl number 0.74–150. Kumar and Nigam [34] reported that the heat transfer enhancement in the innovative device was higher as compared to the coiled tube. The present study focuses on the development of local and average friction factor and heat transfer coefficient under uniform wall flux as thermal boundary condition. The hydrodynamics and thermal development studies were conducted, using Newtonian fluids, for a range of Dean numbers from 25 to 400 and Prandtl number ranging from 1 to 150. A brief description of the fluid flow in curved channels and its modification arising due to change in the direction of the curvature plane in helical coil and chaotic configuration is presented. All the computations were carried out on a SUN Fire V880 computer in the

Chemical Reaction Engineering Laboratory at Indian Institute of Technology, Delhi.

2. Mathematical formulation

In the present study the geometry considered and the system of coordinates are same as reported in [34]. The bends introduced in between the helical coils are of 90° and each helical tube has same length before and after the bend. The laminar flow and heat transfer develops simultaneously downstream in the helical pipe. The flow is considered to be steady, and constant thermal properties are assumed. The Navier Stokes and energy equations governing the three-dimensional laminar flow in the chaotic configuration were solved in the master Cartesian coordinate system with a control-volume finite difference method (CVFDM) similar to that introduced by Patankar [37]. The detailed numerical methodology used to characterize the heat transfer in chaotic heat exchanger and coiled tube is reported in Kumar and Nigam [34]. The convection term in the governing equations is modeled with the bounded second-order upwind scheme and the diffusion term computed using the multilinear interpolating polynomials nodes. The SIMPLE algorithm [37] is used to resolve the coupling between velocity and pressure.

No-slip boundary condition, $u_i = 0$, and uniform wall heat flux, q_w, are imposed on the wall. At the inlet, the fully developed duct flow velocity profiles and a fixed pressure at the outlet of the chaotic configuration flow inverter were employed. The diffusion flux at the outlet for all variables in the exit direction is set to zero. The numerical computation is considered converged when the residual summed over all the computational nodes at nth iteration were less than or equal to 10^{-5}. The methodology for the numerical simulations and standardization of its accuracy can be found in Kumar and Nigam [34]. To verify the hydrodynamics and

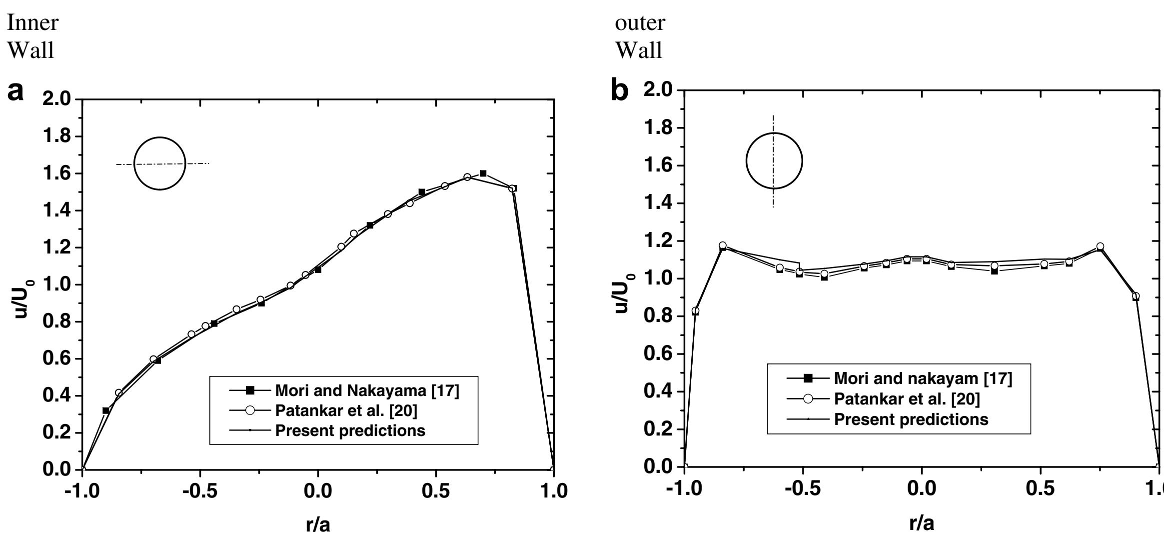


Fig. 1. Comparison of velocity profiles in the present study with Mori and Nakayama [17] and Patankar et al. [20] at (a) horizontal centerline and (b) vertical centerline.

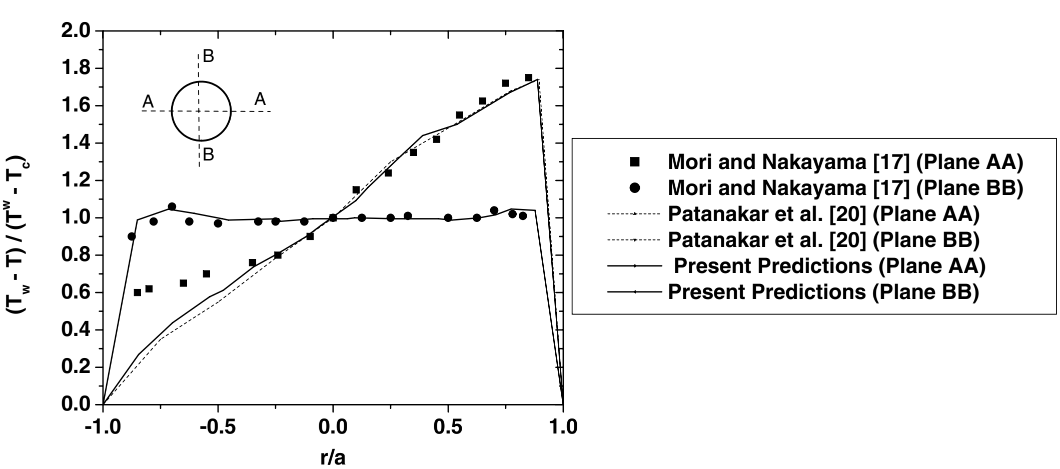


Fig. 2. Comparison of temperature profiles in the present study with Mori and Nakayama [17] and Patankar et al. [20] at (A) horizontal centerline and (B) vertical centerline at $N_{De} = 632.4$, $D/d = 40$ and $N_{Pr} = 0.74$.

heat transfer predictions the fully developed velocity and thermal profiles were first compared with the experimental predictions of Mori and Nakayama [17] and numerical predictions of Patankar et al. [20]. Figs. 1 and 2 show the fully developed velocity and thermal profiles are in well agreement with the literature [17,20].

3. Results and discussion

The numerical computations were carried out for different parameter values ranging for Dean number from 7 to 400 and the Prandtl number from 0.74 to 150 (curvature ratio, $\lambda = 10$) respectively. The physical properties of the fluid during the simulation were kept constant. The study for development of velocity and thermal profiles at various cross-sections and Dean number was also carried out in coiled tube as well as in chaotic configuration. It was observed that for fully developed velocity profiles the maxima moves towards the outer wall of the coiled tube and chaotic configuration. The numerical results of the present study have shown that fully developed velocity fields at any cross-section in chaotic configuration were almost identical as those of coiled tube, which is also in agreement with the predictions reported in [34]. The main difference between the coiled tube and the chaotic configuration can be qualitatively described from their temperature profiles. The Dean roll cells were also locally present after the bend, but after each curvature plane the centrifugal force was reoriented, so the Dean vortices of the previous arm vanished and reappeared in a plane perpendicular to the previous plane. This phenomena result into much weaker temperature gradients in chaotic configuration. The velocity and thermal profiles at various cross-sections and Dean numbers in the present study were similar as reported in [34]. It was observed that the thermal boundary condition does not affect the velocity and thermal profiles in the curved tube.

4. Heat transfer enhancement

4.1. Data comparison

In order to predict the enhancement of heat transfer and friction factor in the proposed innovative chaotic configuration, the present computation technique was checked for

Table 1
Experimental correlations on heat transfer in coiled circular tubes for laminar flow regime under constant wall flux boundary conditions

References	Curvature (λ)	Flow parameter	Heat transfer parameter	Correlation
Berge and Bonilla [19]	30–100	$1 < N_{\mathrm{De}} < 100$	–	$N_{\mathrm{Nu}} = (0.0000229 + 0.00636\lambda)N_{\mathrm{Re}}^{1.29}N_{\mathrm{Pr}}$
Jha and Rao [9]	16.6–37.9	–	–	$\frac{N_{\mathrm{Nu}}}{N_{\mathrm{Nus}}} = [1 + 3.46\lambda]$
Dravid et al. [15]	18.6	$50 < N_{\mathrm{De}} < 2000$	$5.0 < N_{\mathrm{Pr}} < 175$	$N_{\mathrm{Nu}} = (0.76 + 0.65\sqrt{N_{\mathrm{De}}})$
Singh and Bell [16]	20.2 and 41.7	$6.0 < N_{\mathrm{Re}} < 7650$	$241 < N_{\mathrm{Pr}} < 922000$	$N_{\mathrm{Nu}} = (0.224 - 1.369\lambda)\left(N_{\mathrm{Re}}^{[0.521+1.369/\lambda]}\right)$
Janssen and Hoogendoorn [11]	12-100	$20 < N_{\mathrm{De}} < 830$	$20 < N_{\mathrm{Pr}} < 40$	$N_{\mathrm{Nu}} = 0.617(fcN_{\mathrm{Re}}^2)^{0.26}N_{\mathrm{Pr}}^{1/6}$

Table 2
Theoretical correlations on heat transfer in coiled circular tubes for laminar flow regime under constant wall flux boundary conditions

References	Curvature (λ)	Flow parameter	Heat transfer parameter	Correlation
Mori and Nakayama [17] I → 1st approximation II → 2nd approximation	$\lambda \geqslant 1.0$	Laminar	$N_{\mathrm{Pr}} \geqslant 1.0$	$\left(\frac{N_{\mathrm{Nu}}}{N_{\mathrm{Nus}}}\right)_{\mathrm{I}} = 0.1979\sqrt{N_{\mathrm{De}}}/Z$ $\left(\frac{N_{\mathrm{Nu}}}{N_{\mathrm{Nus}}}\right)_{\mathrm{II}} = \left(\frac{N_{\mathrm{Nu}}}{N_{\mathrm{Nus}}}\right)_{\mathrm{I}} 1 + \frac{37.05}{Z}\left\{\left[\frac{1}{40} - \frac{17Z}{120} + \left(\frac{1}{10Z} + \frac{13}{30}\right)\frac{1}{10N_{\mathrm{Pr}}}\right]\frac{1}{\sqrt{N_{\mathrm{De}}}}\right\}^{-1}$ where $Z = \frac{2}{11}\left\{1 + \left[1 + \frac{77}{4N_{\mathrm{Pr}}^2}\right]\right\}$
			$N_{\mathrm{Pr}} < 1.0$	$\overline{\left(\frac{N_{\mathrm{Nu}}}{N_{\mathrm{Nus}}}\right)_{\mathrm{II}}} = \left(\frac{N_{\mathrm{Nu}}}{N_{\mathrm{Nus}}}\right)_{\mathrm{I}} 1 - \frac{37.05}{Z}\left\{\left[\frac{Z^2}{12} + \frac{1}{24} - \frac{1}{120Z} + \left(\frac{4Z}{2} - \frac{1}{3Z} + \frac{1}{15Z^2}\right)\frac{1}{20N_{\mathrm{Pr}}}\right]\frac{1}{\sqrt{N_{\mathrm{De}}}}\right\}^{-1}$ $Z = \frac{1}{5}\left\{2 + \left[\frac{10}{N_{\mathrm{Pr}}^2} - 1\right]^{0.5}\right\}$
Akiyama and Cheng [12]	$\lambda \geqslant 10.0$	Laminar	$(N_{\mathrm{De}}N_{\mathrm{Pr}})^{1/4} \geqslant 3.5$	$\frac{N_{\mathrm{Nu}}}{N_{\mathrm{Nus}}} = 0.181Q(1 - 0.839Q^{-1}) + 35.4Q^{-2} - 207Q^{-3} + 419Q^{-4}$ $Q = (N_{\mathrm{De}}^2N_{\mathrm{Pr}})^{1/4}$ $N_{\mathrm{Nus}} = 4.3636$
Kalb and Seader [13]	10–100	$20 < N_{\mathrm{De}} < 1200$	$0.005 < N_{\mathrm{Pr}} < 0.05$	$N_{\mathrm{Nu}} = 3.31\ N_{\mathrm{De}}^{0.115}\ N_{\mathrm{Pr}}^{0.0108}$
	10–100	$80 < N_{\mathrm{De}} < 1200$	$0.7 < N_{\mathrm{Pr}} < 5$	$N_{\mathrm{Nu}} = 0.913\ N_{\mathrm{De}}^{0.476}\ N_{\mathrm{Pr}}^{0.2}$
Manlapaz and Churchill [22]	$\lambda > 5$	$N_{\mathrm{He}} = 0$–2000 $(H/D = 0$–$1)$	$N_{\mathrm{Pr}} = 0.005$–∞	$N_{\mathrm{Nu}} = \left[\left(\frac{48}{11} + \frac{51/11}{\left(1+\frac{1342}{N_{\mathrm{Pr}}N_{\mathrm{He}}^2}\right)^2}\right)^3 1.816\left(\frac{N_{\mathrm{He}}}{1+\frac{1.15}{N_{\mathrm{Pr}}}}\right)^{3/2}\right]^{1/3}$

its reliability and accuracy by comparing the literature value of fully developed heat transfer coefficient in helical coiled tubes. The experimental and theoretical correlations on heat transfer in coiled circular tubes reported in the literature for laminar flow regime under constant wall flux boundary conditions are given in Tables 1 and 2, respectively. Fig. 3a shows the comparison between the results of present study and the experimental and theoretical data available in the literature for fully developed Nusselt number [13,22] variation with the Dean number. The present CFD predictions agree with the theoretical and experimental data of Kalb and Seader [13] and Manalapaz and Churchill [22] for fully developed Nusselt number. The choice of the literature correlations was based on the process parameters covered in the present study. The comparison was also carried out for the friction factor in coiled tube with the experimental data of Mishra and Gupta [38] and found in good agreement. The maximum deviation between the present predictions and the empirical correlation is less than 3% within the examined parameter range.

Before studying the enhancement in heat transfer in the chaotic configuration a comparative study of heat transfer parameter for the constant wall flux and constant wall temperature boundary condition was made for the simple helical coil and chaotic configuration. Fig. 3b shows the asymptotic values of the peripherally averaged Nusselt number for the fully developed thermal region, both for the boundary condition of $\Phi_w = \text{constant}$ and $T_w = \text{constant}$. The results for coiled tube fairly well agree with the numerical results of Manalapaz and Churchill [22]. However, from the literature it was observed that for $\Phi_w = \text{constant}$, the heat transfer is 8–10% higher as compared to the constant wall temperature condition for coiled tube. The same phenomenon was observed for the chaotic configuration.

4.2. Development of local friction factor and Nusselt number

Fig. 4 shows the development of the local friction factor and Nusselt number at different axial distance ($\phi = 15°$, 30°, 60°, 120° and 180°) in the coiled tube and after the first and second bend in the chaotic configuration at $N_{De} = 100$. The data were plotted from outermost point $\theta = 0°$ to 360° in anti-clockwise direction along the circumference on different cross-sections. For coiled tubes, the region near the inlet, the distribution of f_c (or $N_{Nu,\theta}$) on the circumference is relatively smooth (Fig. 4a). The fully developed velocity profiles from the outlet of coiled tube were fed at the inlet of the first bend. It can be seen from Fig. 4b and c that as φ (curvature angle) increases the phenomenon of stretching and folding takes place due to the chaotic mixing between the fluid elements in the chaotic configuration. Similar phenomenon was also observed after the second bend.

4.3. Development of average friction and Nusselt number

The effect of Dean number on the developments of the circumference average friction factor and Nusselt number in coiled tube and the chaotic configuration are shown in Fig. 5 for $N_{De} = 100$ and 200 respectively. The Nusselt numbers in the coiled tube are reported as a function of axial distance. The variations in heat transfer in the curved tube clearly reflect the oscillating character due to the circulating secondary flow. After every new circulation fluid of a higher temperature flows to the outer tube wall, which leads to a sudden decrease in temperature gradient at the tube wall and therefore a decrease in heat transfer coefficient. Since the thermal boundary layer at the outer wall in the curved tube is thin, the heat transfer is very sensitive to temperature changes of the fluid, which is contrary to the inner wall of the curved tube, where the thermal

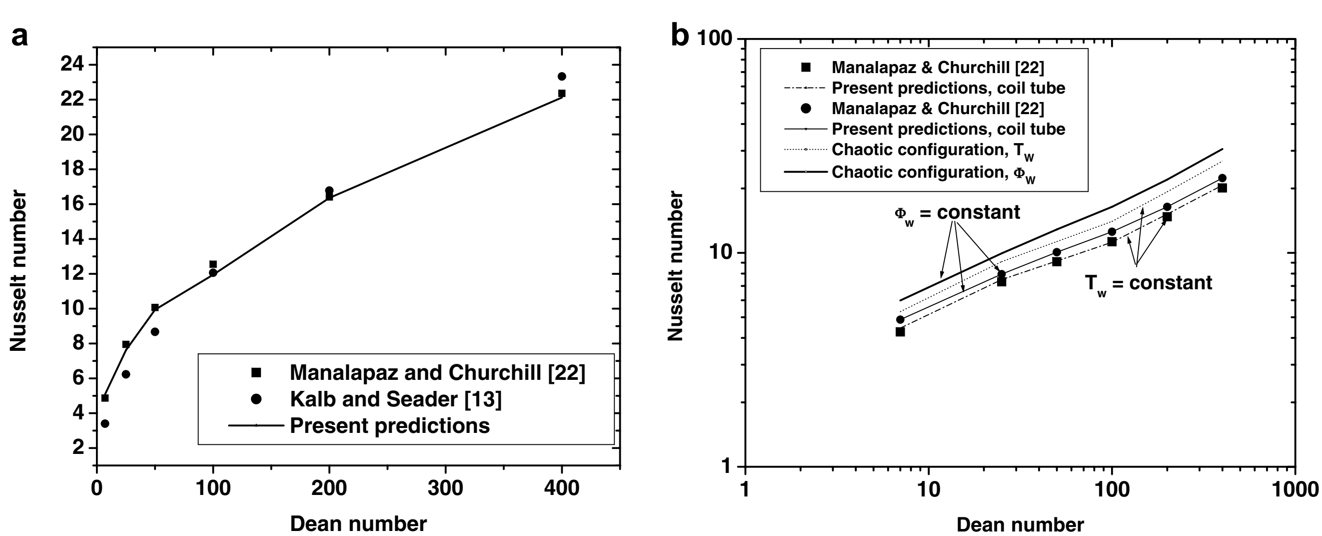
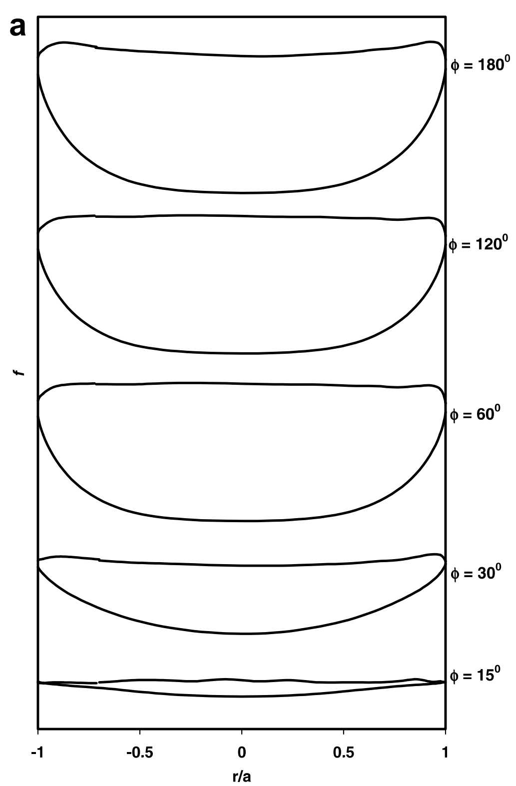
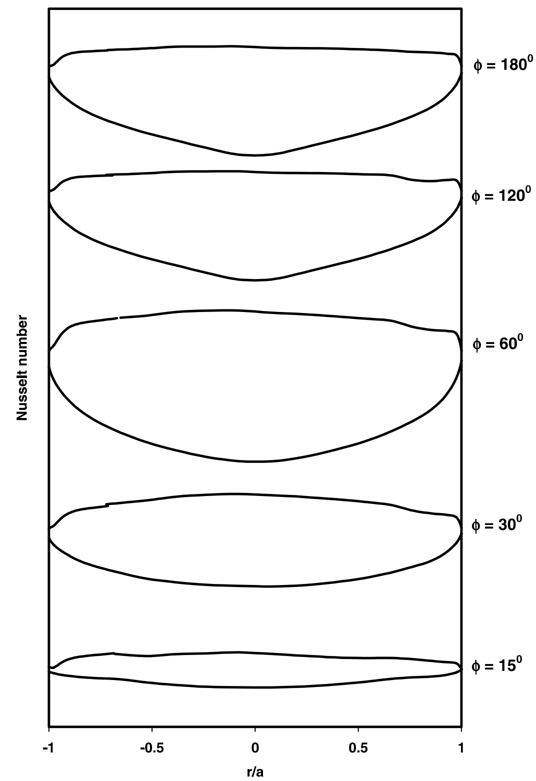


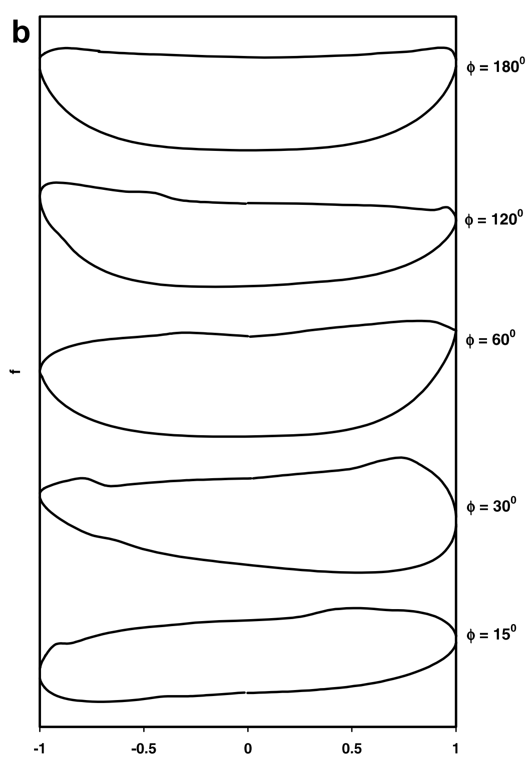
Fig. 3. (a) Nusselt number variation with Dean number in coiled tube and (b) comparison of fully developed Nusselt number for $\Phi_w = \text{constant}$ and $T_w = \text{constant}$ at $N_{Pr} = 7.0$.



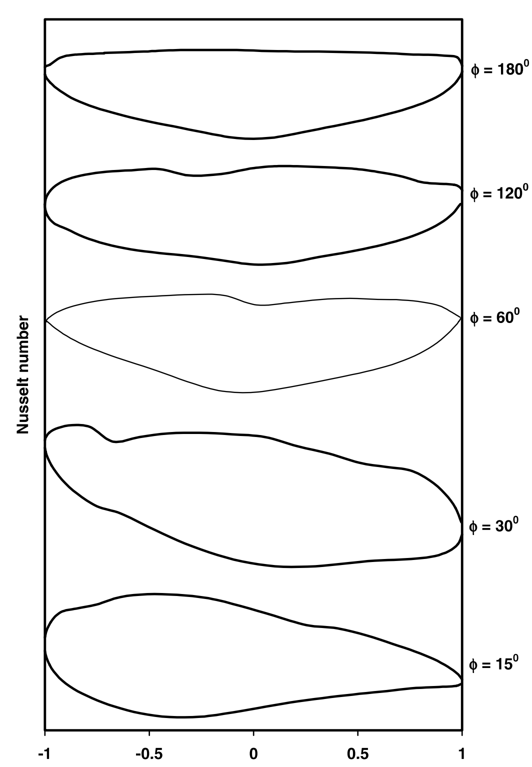
(i) Local friction factor



(ii) Local Nusselt number

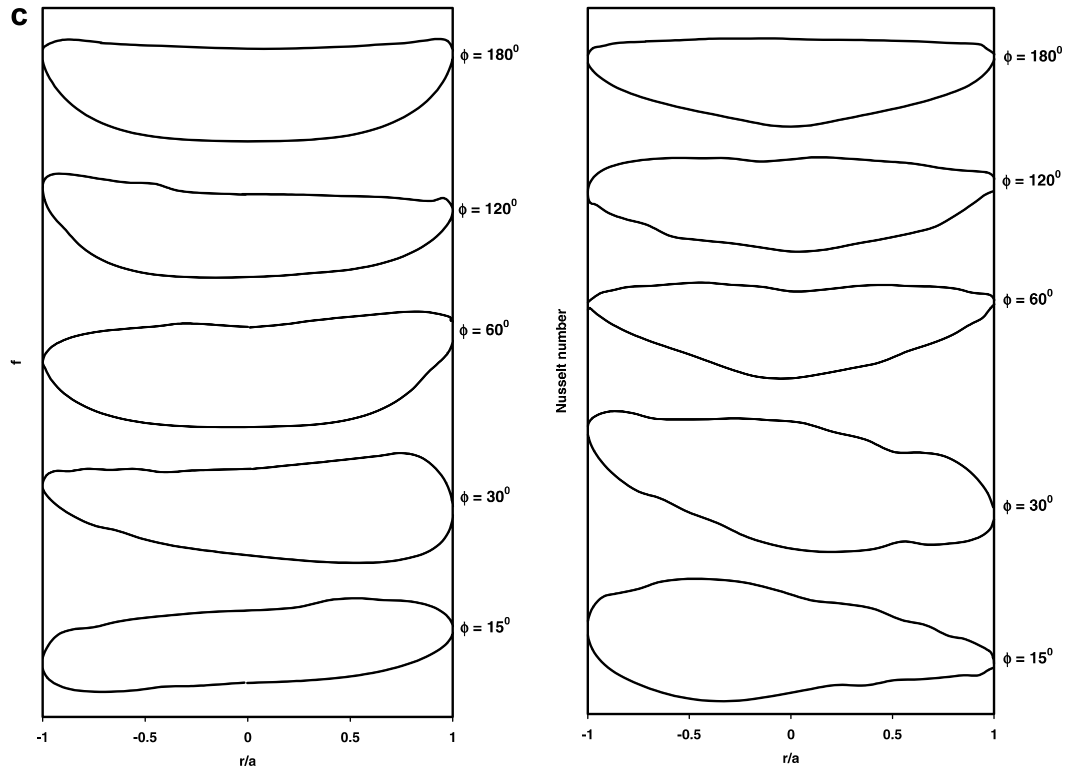


(i) Local friction factor



(ii) Local Nusselt number

Fig. 4. Developments of local friction factor and local Nusselt number on the circumference of (a) coiled tube, (b) after first bend of bent helix and (c) after second bend of bend helix at $N_{De} = 100$ and $N_{Pr} = 7.0$.



(i) Local friction factor

(ii) Local Nusselt number

Fig. 4 (*continued*)

boundary layer is much thicker. Janessen and Hoogendoorn [11], Darvid et al. [15] and Zheng et al. [39] also reported similar oscillations for various Dean numbers in curved tubes.

The whole process of development of f_m (or $N_{Nu,m}$) in chaotic configuration could be divided into three stages characterized by a substantial difference nature. These are summarized below

1. the early developing stage ($\phi < 15^0$), where f_m (or $N_{Nu,m}$) dropped sharply at the increase of ϕ due to the rapid development of the flow (or thermal) boundary layer;
2. the oscillatory developing stage ($15° < \phi < 180°$), where obvious oscillation of f_m (or $N_{Nu,m}$) appeared with the increase of ϕ;
3. the later developing stage ($\phi > 180°$), when f_m (or $N_{Nu,m}$) varied smoothly with the increase of ϕ until the fully developed flow (or heat transfer) was established.

Under the computational conditions used in Fig. 5, the effect of Dean number in chaotic configuration was to reduce the magnitude of f_m and, in contrast, increase the magnitude of $N_{Nu,m}$.

4.4. Flow resistance in coiled tube and chaotic configuration

Fig. 6a shows the variation of friction factor ratio (f/f_s) with Dean number, where friction factor ratio is the ratio of friction factor in coiled tube or chaotic configuration to the friction factor in straight tube. It can be seen from figure that at low values of Dean numbers, there is not much difference between the f/f_s in coiled tube and chaotic configuration. As the Dean number increases the difference in f/f_s between the coiled tube and chaotic configuration increases.

4.5. Heat transfer: effect of Dean number and Prandtl number

The results of the numerical computations for enhancement of heat transfer with Dean number (N_{De}) and Prandtl number (N_{Pr}) in a chaotic configuration having seven 90° bends are shown in Fig. 6b and c respectively. Fig. 6b shows the influence of Dean number on the enhancement of heat transfer in chaotic configuration, as the Dean number increases Nusselt number ratio ($N_{Nu}/(N_{Nu})_s$) increases, where Nusselt number ratio is the ratio of Nusselt number in coiled tube or chaotic configuration to the Nusselt number in straight tube. It can also be observed that there is 22–36% heat transfer enhancement in chaotic configuration comprised of seven bends in terms of the fully developed Nusselt number (N_{Nu}) as compared to the straight coil. The Prandtl number is also seen to have an effect similar to the Dean number (Fig. 6c). The enhanced mixing occurring in the case of the chaotic configuration is a result of convective motion which is strong compared to thermal

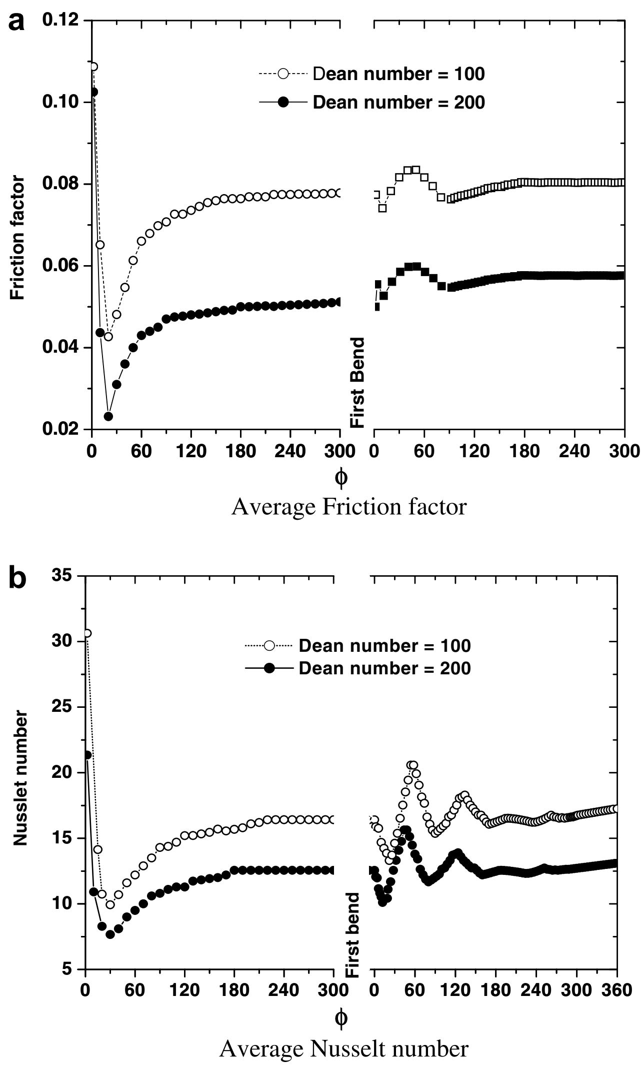


Fig. 5. Effects of Dean number on the development of average (a) friction factor and (b) Nusselt number for chaotic configuration at one bend.

diffusion for the fluids with high Prandtl number. For fluids with low Prandtl number, diffusion tends to smear out the temperature profile and reduces the effective contribution of the convective transfer. It can be seen from Fig. 6c that at low Prandtl number ($N_{\mathrm{Pr}} = 7$), there is 36% enhancement in heat transfer while at higher Prandtl number ($N_{\mathrm{Pr}} = 100$) the enhancement is 70% in the chaotic configuration as compared to the straight helix.

4.6. Percentage increase in heat transfer and pressure drop

Fig. 7 shows the enhancement in heat transfer and relative increase in the pressure drop. The enhancement in heat transfer was estimated by calculating the ratio of heat transfer in chaotic configuration to the straight helix. Similarly in case of pressure drop, the ratio of pressure drop in the chaotic configuration to the straight helix was reported. It can be observed form Fig. 7 that there is an enhancement of 22–36% in heat transfer and the relative pressure drop is 5–6%.

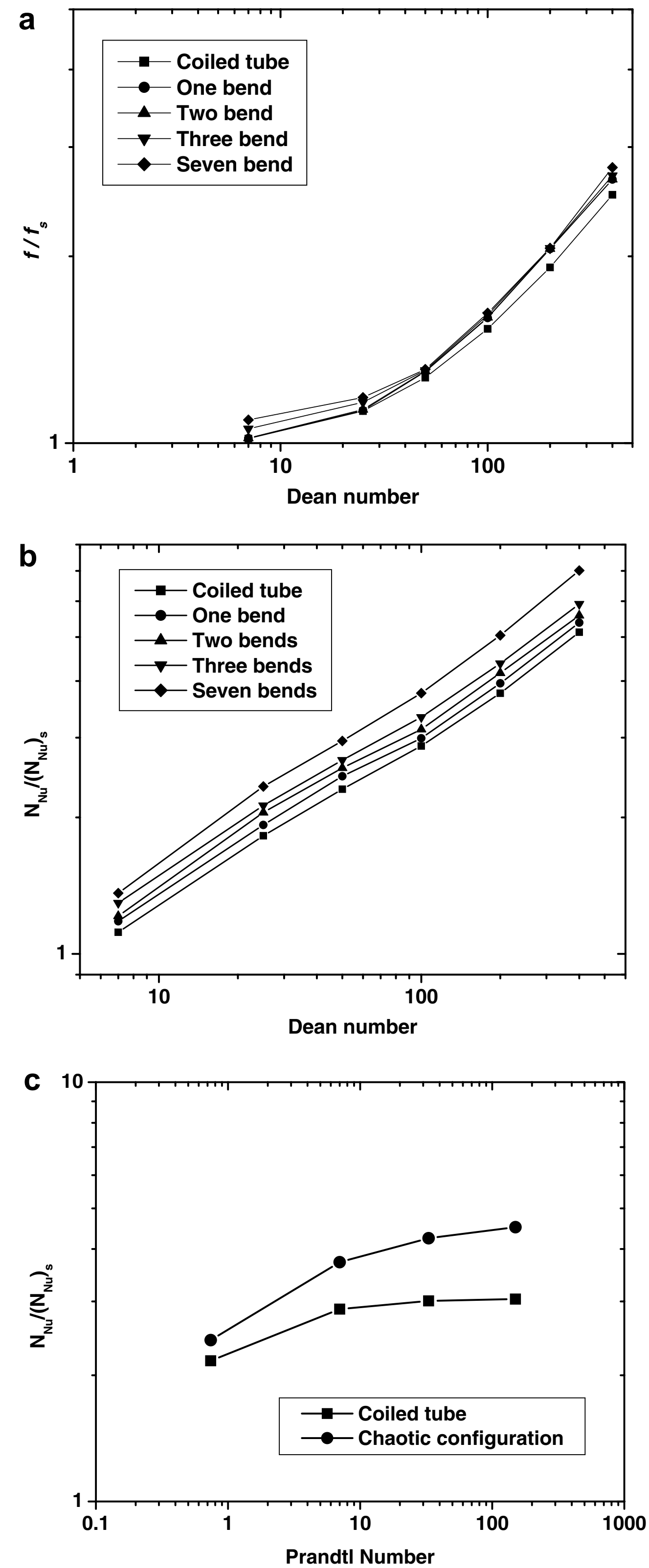


Fig. 6. (a) Friction factor variation with Dean number at $\lambda = 10$; (b) Nusselt number variation with Dean number at $N_{\mathrm{Pr}} = 7.0$ and $\lambda = 10$; and (c) Nusselt number variation with Prandtl number at $N_{\mathrm{Re}} = 316$ and $\lambda = 10$ in coiled tube and chaotic configuration.

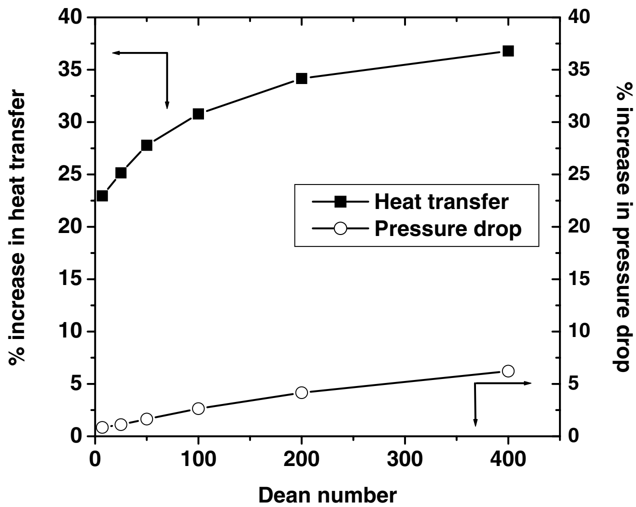


Fig. 7. Relative increase in heat transfer and relative pressure drops in chaotic configuration at $N_{Pr} = 7.0$.

4.7. Effect of temperature-dependent viscosity for highly viscous fluid

The study is further extended to predict the effect of temperature-dependent viscosity on hydrodynamics and heat transfer in the in the coiled tube and chaotic configuration (7-bend geometry). All other fluid properties are considered constant, except the viscosity, during the computations. The temperature-dependent viscosity for the viscous fluid (Diethylene glycol, DEG) is calculated using polynomial function as follows:

$$\mu(T) = A_0 + A_1T + A_2T^2 + A_3T^3 + A_4T^4 + A_5T^5 + A_6T^6 \quad (1)$$

where T is the temperature and A_n ($n = 1, 2, 3, 4, 5$ and 6) are coefficients determined by a polynomial fitting over the temperature-dependent viscosity data of the DEG provided by Incropera and Dewitt [40]. Fig. 8 shows the values of the coefficients of DEG viscosity variation as a function of temperature resulting from Eq. (1). The friction constant (fRe) and heat transfer is calculated for heating and cooling conditions respectively. In the cooling condition, all the thermo-physical properties including viscosity are assumed constant for DEG, and in the heating condition, the DEG viscosity is allowed to vary with temperature. The friction constant (fRe) versus Dean number results for heating and cooling condition is plotted in Fig. 9a. It can be seen from Fig. 9a that the friction constant for temperature-dependent viscosity is less than the constant viscosity results. This is due to the fact that near the wall of the chaotic configuration temperature is higher which results into low viscosity of the fluid in the heating condition and therefore reduces friction constant values. Fig. 9a also shows that as Reynolds number increases, the fRe also increases and tends to approach the result for the cooling condition calculation. This is caused by the temperature impact on the

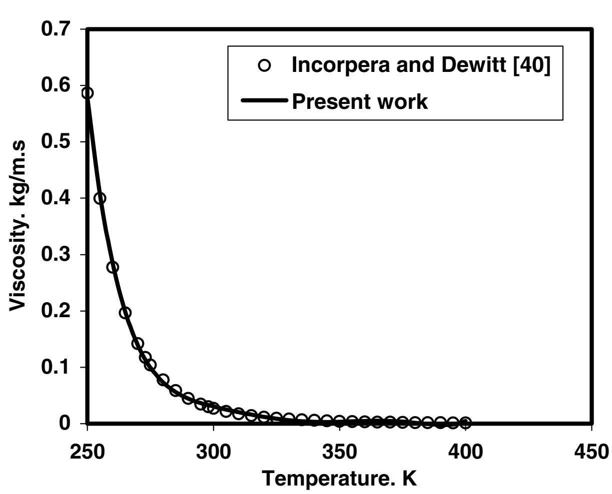


Fig. 8. The temperature-varying viscosity for the diethylene glycol.

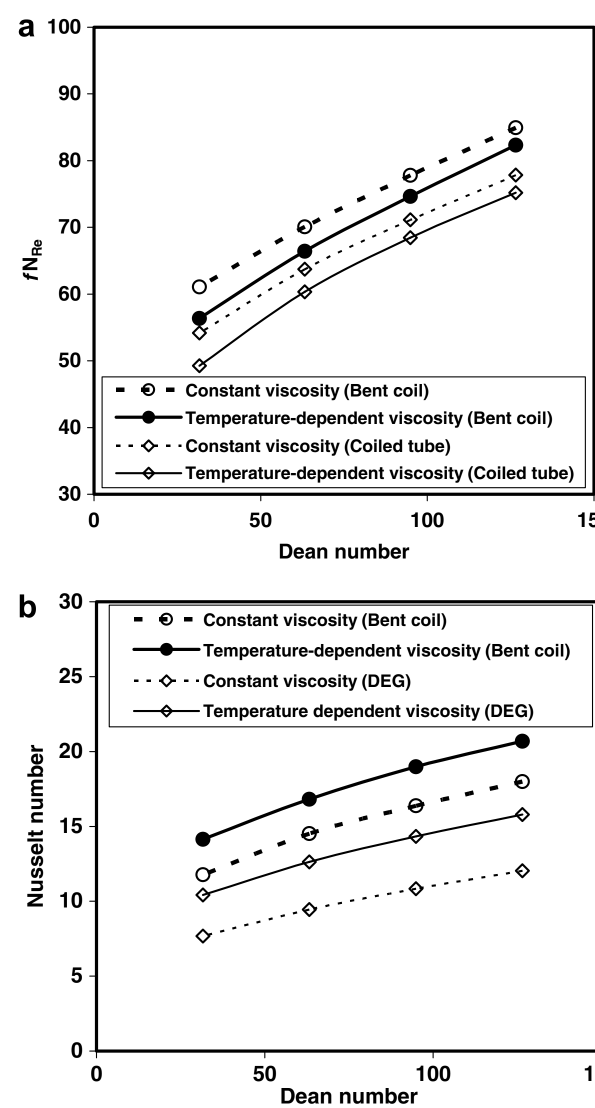


Fig. 9. Nusselt number and friction factor constant variation with temperature-dependent viscosity for diethylene glycol.

viscosity. At low Reynolds number, the fluid will experience higher temperature rise, and therefore, results into lower viscosity. This leads to lower pressure drop and thus the decrease in the values of *fRe*. When Reynolds number increases, the temperature rise of the fluid will drop and the decrease in viscosity will become less appreciative. It can be expected that *fRe* will eventually approach the cooling condition.

Fig. 9b also present the Nusselt number as a function of the Dean number for heating and cooling conditions, respectively. It can be seen from Fig. 9b that the *Nu* values are higher in the heating condition than in the cooling condition (viscosity variations are not taken into account in the cooling condition). It is observed that under heating condition the temperature-dependent viscosity assumption increases the *Nu* results. When the fluid is heated considering temperature-dependent viscosity, the bulk mean temperature is higher than in the constant-viscosity case. This fact cause a decrease of the viscosity values at the inner points of the curved tube section that enhances the secondary flow effect, and consequently the heat transfer rate.

5. Conclusions

Three-dimensional developing flow and heat transfer in helical tubes and a new innovative device of chaotic configuration (bent coil configuration) has been numerically simulated with a control-volume finite difference method. The fluid flow and heat transfer studies were carried out for Dean number ranging from 7 to 400 and Prandtl number ranging from 0.7 to 150 for curvature ratio of 10. The uniform heat flux boundary condition was considered at the wall The flatter temperature distribution in the chaotic configuration is attributed to the radial mixing of fluid elements due to the flow inversion at the 90° bends, which contributes also to the enhancement of the global efficiency of a chaotic configuration as compared to regular coils. Aside from the high thermal efficiency of chaotic configuration, the low mechanical stresses due to the laminar nature of the flow and uniform temperature distribution could be of significant benefit in process industry. A comparative study for the different boundary conditions (Φ_w = constant and T_w = constant) in coiled tube was carried out and compared with the results available in the literature. The heat transfer in chaotic configuration under various thermal boundary conditions was also reported. The development of local and circumferentially averaged friction factor and Nusselt number in the chaotic configuration have been studied at two different values of Dean number (N_{De} = 100 and 200). It was observed that the local friction factor and Nusselt number straches and folded as the axial distance increases. The developments of the circumferential averaged Nusselt number in chaotic configuration is found to be oscillatory before it is fully developed. The chaotic configuration also displays a heat transfer enhancement of 25–36% in terms of the fully developed Nusselt numbers compared to the straight coil over a range of $7 \leqslant N_{De} \leqslant 400$ with little change in the pressure drop. Beside this the hydrodynamics and heat transfer study is further extended for temperature-dependent viscosity for highly viscous fluid (DEG) in the chaotic configuration. It was observed that under heating condition (temperature-dependent viscosity) the heat transfer is higher in case of chaotic configuration as compared to the cooling condition (constant viscosity).

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References

[1] S.A. Berger, L. Talbot, L.S. Yao, Flow in curved pipes, Annu. Rev. Fluid Mech. 15 (1983) 461–512.
[2] R.K. Shah, S.D. Joshi, Convective heat transfer in curved ducts, in: S. Kakac, R.K. Shah, W. Aung (Eds.), Handbook of Single-Phase Convective Heat Transfer, Wiley, New York, 1987 (Chapter 5).
[3] K. Nandakumar, J.H. Masliyah, Swirling flow and heat transfer in coiled and twisted pipes, in: A.S. Mujumdar, R.A. Masliyah (Eds.), Advances in Transport Process, vol. 4, Wiley Eastern, 1986 (Chapter 2).
[4] W.R. Dean, Note on the motion of fluid in a curved pipe, Philos. Mag. 4 (1927) 208–223.
[5] W.R. Dean, The streamline motion of fluid in a curved pipe (second paper), Philos. Mag. S 7 (5) (1928) 673–695.
[6] A.V. Kripklov, Heat transfer in helically coiled pipes, Trudi. Moskov. Inst. Khim. Mashinojtrojenija 12 (1957) 43–56.
[7] G.F.C. Roger, Y.R. Mayhew, Heat transfer and pressure loss in helically tubes with turbulent flow, Int. J. Heat Mass transfer 7 (1964) 1207–1216.
[8] V. Kubair, N.R. Kuloor, Heat transfer to Newtonian fluid in coiled pipes in laminar flow, Int. J. Heat Mass Transfer 9 (1966) 63–75.
[9] R.K. Jha, R. Rao, Heat transfer through coiled tubes in agitated vessels, Int. J. Heat Mass Transfer 10 (1967) 395–397.
[10] V.K. Shuchkin, Correlation for experimental data on heat transfer in curved pipes, Thermal Eng. 16 (1969) 72–76.
[11] L.A.M. Janssen, C.J. Hoogendrawn, Laminar convection heat transfer in helical coiled tubes, Int. J. Heat Mass transfer 21 (1978) 1179–1206.
[12] M. Akiyama, K.C. Cheng, Laminar forced convection heat transfer in curved pipes with uniform wall temperature, Int. J. Heat Mass Transfer 15 (1972) 1426–1431.
[13] C.E. Kalb, J.D. Seader, Fully developed viscous flow. Heat transfer in curved tubes with uniform wall temperatures, AIChE J. 20 (1974) 340–346.
[14] F.A. Seban, E.F. Mclaughlin, Int. J. Heat Mass Transfer 7 (1963) 387–395.
[15] A.N. Dravid, K.A. Smith, E.W. Merrill, P.L.T. Brain, Effect of secondary fluid motion on laminar flow heat transfer in helically coiled pipes, AIChE J. 17 (1971) 1114–1122.
[16] S.P.N. Singh, K.J. Bell, Laminar flow in heat transfer in a helically coiled tube, Proceedings of the 5th International Heat Transfer Conference, vol. II, The Japan Soc. Mech. Engrs., 1974, pp. 193–197.
[17] Y. Mori, W. Nakiyama, Study on forced convective heat transfer in curved pipes, Int. J. Heat Mass Transfer, Part I 8 (1965) 67–82.
[18] M. Akiyama, K.C. Cheng, Graetz problem in curved pipes with uniform wall heat flux, Appl. Sci. Res. 29 (1974) 401–418.
[19] J.M. Tarbell, M.R. Samuels, Momentum and heat transfer in helical coils, Chem. Eng. J. 5 (1973) 117–127.

[20] S.V. Patankar, V.S. Pratap, D.B. Spalding, Prediction of laminar flow and heat transfer in helically coiled pipes, J. Fluid Mech. 62 (1974) 539–551.
[21] E.N. Sieder, G.E. Tate, Ind. Eng. Chem. 28 (1936) 1429–1436.
[22] R.L. Manlapaz, S.W. Churchill, Fully developed laminar convection from a helical coil, Chem. Eng. Commun. 97 (1981) 185–200.
[23] N. Acharya, M. Sen, H.C. Cheng, Heat transfer enhancement in coiled tubes by chaotic mixing, Int. J. Heat Mass Transfer 35 (10) (1992) 2475–2489.
[24] N. Acharya, M. Sen, H.C. Chang, Applications of chaotic heat and mass transfer enhancement, AIChE Symp. Ser. 286 (88) (1992) 44–49.
[25] A. Mokrani, C. Castelain, H. Peerhossaini, The effects of chaotic advection on heat transfer, Int. J. Heat Mass Transfer 40 (13) (1997) 3089–3104.
[26] H. Peerhossaini, C. Castelain, Y. Le Guer, Heat exchanger design based on chaotic advection, Exp. Thermal Fluid Sci. 7 (1993) 333–344.
[27] S.W. Jones, O.M. Thomas, H. Aref, Chaotic advection by laminar flow in twisted pipe, J. Fluid Mech. 209 (1989) 335–357.
[28] Chagny, C. Castelain, H. Peerhossaini, Chaotic heat transfer for heat exchanger design and comparison with a regular regime for a large range of Reynolds numbers, Appl. Thermal Eng. 20 (2000) 1615–1648.
[29] H. Peerhossaini, Y. Le Guer, Effect of curvature plane orientation on vortex distortion in curved channel flow, in: C.D. Andereck, F. Hayot (Eds.), Ordered and Turbulent Patterns in Taylor–Couette Flow, Plenum Press, New York, 1992, pp. 263–272.
[30] H. Peerhossaini, Y. Le Guer, Chaotic motion in the Dean instability flow—a heat exchanger design, Bull. Am. Phys. Soc. 35 (1991) 2229.
[31] C. Castelain, D. Berger, P. Legentilhomme, A. Mokrani, H. Peerhossaini, Experimental and numerical characterization of mixing in a steady spatially chaotic flow by means of residence time distribution measurements, Int. J. Heat Mass Transfer 43 (2000) 3687–3700.
[32] C. Castelain, A. Mokrani, Y. Le. Guer, H. Peerhossaini, Experimental study of chaotic advection regime in a twisted duct flow, Eur. J. Mech. B—Fluids 20 (2001) 205–232.
[33] T. Lemenand, H. Peerhossaini, A thermal model for prediction of the Nusselt number in a pipe with chaotic flow, Appl. Thermal Eng. 22 (2002) 1717–1730.
[34] V. Kumar, K.D.P. Nigam, Numerical simulation of steady flow fields in coiled flow inverter, Int. J. Heat Mass Transfer 48 (2005) 4811–4828.
[35] A.K. Saxena, K.D.P. Nigam, Coiled configuration for flow inversion and its effect on residence time distribution, AIChE J. 30 (1984) 363–368.
[36] K.D.P. Nigam, A.K. Saxena, Residence time distribution in straight and cured tubes, in: N.P. Cheremishinoff (Ed.), Ency. Fluid Mech., vol. 1, Gulf Publishing, USA, 1986, pp. 675–762.
[37] S.V. Patankar, Numerical Heat Transfer and Fluid Flow, Hemisphere, Washington, DC, 1980.
[38] P. Mishra, S.N. Gupta, Momentum transfer in curved pipes: Newtonian fluids, Ind. Eng. Chem. Des. Dev. 18 (1979) 130–137.
[39] B. Zheng, C.X. Lin, M.A. Ebadian, Combined laminar forced convection thermal radiation in a helical pipe, Int. J Heat Mass Transfer 43 (2000) 1067–1078.
[40] F.P. Incropera, D.P. Dewitt, Fundamentals of Heat Transfer, John Wiley & Sons, Inc., New York, 1981.