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Research Article

Thermo-hydraulic characteristics of mixed convection of transition flow in a conduit at different tilts

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ABSTRACT

In this study, numerical investigation was carried out on combined free and forced convection for transition flow of water in a conduit that has a mechanical component, which is a three-sided-polygon-perforations-tranverse-axes tape, inserted in its interior. The purpose of the investigation is to find out the thermo-hydraulic characteristics of mixed convection of transition flow in the tube, and compare them with those of forced convection. Fluent software was used to perform the numerical investigation for the transition flow at Reynolds number (Re) in the range 2,150 ≤ Re ≤ 3,700 for the tube bent at sundry tilts (λ) in the range 15° ≤ λ ≤ 90°. The results indicated that the inside temperature and surface temperature of the tube whose fluid flow is forced convection are lower than the case of the tube whose fluid flow is mixed convection, and that the temperatures (tube's inside temperature, dimensionless outlet temperature, and surface temperature) increase as the tube tilt increases. Not only that, as the tube tilt increases, the dimensionless outlet temperature increases, but the outlet velocity decreases. The Nusselt number in the tube whose fluid flow is forced convection is lower than the case of mixed convection. For the tube with flow tilts of 15°, 30°, 60°, and 90°, the Nusselt number are 4.01 to 6.10%, 4.60 to 7.41%, 5.32 to 8.71%, and 6.31 to 10.12%, respectively, greater than the case of the tube whose fluid flow is forced convection, for the Reynolds number considered. The validation of the numerical work with an experimental work confirms the correctness of the numerical work.

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INTRODUCTION

Mixed convection, otherwise known as combined free and forced convection, refers to a situation that occurs in heat transfer in which both free (or natural) and forced convection mechanisms take place at the same time [1, 2]. A change in gravitational body force caused by a change in density of fluid is responsible for the motion of fluid in free convection. In the case of forced convection, a force that is applied externally causes the motion of the fluid [3, 4]. The

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factors that contributes to mixed convection include geometry of flow model, type of working fluid, temperature driving force for heat transfer, boundary conditions, orientation of flow, and regime of flow [5-8].

Different reports on heat transfer applications have been made. These include those of Kilic et al. [9] on cooling of electronic components, Dal and Karaçay [10] on radial air bearing, Kilic [11] on transpiration cooling of hot air and surface of a porous flat plate, and Kilic and Abdulvahitoglu [12] on effect of combination of nanofluids and swirling jets in a radiator of a vehicle.

Utilization of mixed convection heat transfer can be found in various applications, such as oil transportation pipe lines, cooling of electronic equipment, heat exchangers, cooling of rotating parts of gas turbines, boilers, etc. [13-16].

As a result of significance of mixed convection fluid flow in industries, many researches have been done on it with the aim of utilizing its potentials. The effect of inclination on laminar mixed convection of upward flow in a tube was examined by Iqbal and Stachiewicz [17]. The findings showed that as the tube was tilted from its horizontal position to vertical position, the Nusselt number increased. The maximum value of the Nusselt number occurs before the tube reached its vertical location. This location was discovered to be an angle (α) of $20° \leq \alpha \leq 60°$.

Maughan and Incropera [18] experimentally discussed heat transfer in a mixed convection flow of a plate channel. Inclination of between 0^o and 30^o, and Reynolds number of between 125 and 500 were considered for the experiments. It was observed that mixed convection did not dominate on heat transfer in the channel, and that there was a reduction in Nusselt number. The inclined channel yielded a noticeable increase in Nusselt number, as there was an increase of 10% and 15% in Nusselt number for inclination of 20^o and 30^o, respectively, compared with when the channel was horizontal (that is, at 0^o inclination).

Yan [16] numerically looked into heat and mass transfer of slant rectangular ducts under mixed convection of laminar flow. From the results, it was shown that Nusselt number of the flow without buoyancy effect is 0.88% of that with buoyancy effect. This indicates that buoyancy plays a role on characteristics of the flow and transfer of heat in the ducts.

Processes that are associated with heat transfer in mixed convection of air flow in a sloping rectangular duct were investigated by means of experiments by Lin and Lin [19]. The slope of the rectangular duct was up to 26^o and the Reynolds number (Re) was $35 \leq Re \leq 186$. The experimental results revealed that buoyancy and reverse flow in the duct increases the heat transfer. Nusselt number of the mixed convection (that is, combined free and forced convection) is 10% more than the forced convection.

Mixed convection heat transfer in a vertical tube that conveyed air in assisted direction, and later in opposed direction, was discussed experimentally by Mohammed [15]. Laminar flow, for which 400 ≤ Reynolds number ≤ 1600, was considered. It was observed that the surface-temperature along the length of the tube of the flow in opposed direction is 11% higher than that in assisted direction. The Nusselt number of the flow in opposed direction is 12% lower than that in assisted direction.

Patil and Vijay-Babu [7] reported the results of the work that was done experimentally on combination of free and forced convection of laminar flow in a square duct. The observations from the results indicated that at a higher Reynolds number (up to Re = 5,120), heat transfer is influenced by forced convection; at lower Reynolds number (up to Re = 1,100), combination of free and forced convection exercises control on the heat transfer. At the lower Reynolds number, Nusselt number forced convection is 13% less than that of mixed convection.

A study on thermo-hydraulic characteristics of mixed convection in a pipe subjected to laminar flow with Reynolds number between 100 and 2,100 was conducted numerically by Al-asadi et al. [20]. The pipe was inclined at an angle between 30^o and 75^o. The heat transfer for the assisted flow is greater than that of the opposed flow. In addition, the surface temperature of the pipe is directly proportional to its inclination. It was inferred that a gain in the Reynolds number brings about an increment in velocity by up to 19%, but a decrease in surface temperature by up to 16%.

The effects that a vortex promoter has on improvement of heat transfer from electronic components inside a channel that has a rectangular shape were studied by Kilic et al. [9]. Angular position, location, number, and length were the parameters of the vortex promoter that were considered for Reynolds number of 8,000 and vortex promoter angle of between 5^o and 45^o. It was observed that the angle of vortex promoter has an important role on the heat transfer. An increase of 7.1% in Nusselt number was obtained for an angle of 0^o, compared with the case when there was no vortex promoter. A decrease in the Nusselt number was observed for the angle that was either less than or greater than 0^o.

Mahdavi et al. [21] examined the effect of tilt of a pipe under mixed convection flow on migration and deposition of particles of nanofluid inside the pipe. It was found that maximum deposition of between 1% and 5% volume concentration of the nanofluid occurs at inclination of 30^o. At low inclination and concentration of between 0% and 5%, the heat transfer is affected to least extend. For 1% volume concentration, heat transfer coefficient decreases from 436 W/m^2.K at 0^o to 433 W/m^2.K at 30^o, but for 5% volume concentration, the heat transfer coefficient decreases from 360 W/m^2.K) at 0^o to 354 W/m^2.K) at 30^o.

Combined free and forced convection heat transfer and fluid flow in a cylinder whose geometry is a triangle was probed numerically by Altac et al. [22]. A laminar flow assisted in upward direction was assigned to the cylinder. The Nusselt number for forced convection and mixed convection was computed and compared with each other. The results showed that the Nusselt number for mixed convection is 1.25 times that of the forced convection.

Selimefendigil and Öztop [23] examined mixed convection in a hollow area that was occupied with nano-fluid. The vertical segment of the wall of the hollow area was not completely heated, and the segment of the wall that was inclined was kept at constant temperature. Observation from the findings showed that a high value of heat transfer was obtained from a low value of Richardson number (Ri), and that a high value of the fraction of the nano-fluid (β) yielded a high value of heat transfer. At Ri = 0.25 and 1, Nusselt number is 21.9 and 17.6, respectively. At β = 0.01 and 0.04, Nusselt number is 10.0 and 21.9, respectively.

The effects that Reynolds number (500 ≤ Re ≤ 2,000) and Richardson number (0 ≤ Ri ≤ 0) have on heat transfer in forced convection (Ri = 0) and mixed convection (Ri > 0) of laminar flow in a cylinder-shaped annular enclosure was conducted numerically by Turan [24]. It was found that the Nusselt number increases with an increase in Reynolds number. In addition, the Nusselt number of the mixed convection is about 1.14 times of the forced convection.

Shulepova el al. [25] explored the work on combined convection of nanofluid laminar flow in a square enclosure with fin. The influence of fin, Reynolds number, and concentration of the nanofluid on Nusselt number was explored. It was observed that an increase in Reynolds number (Re) and nanofluid concentration (γ) yields an increase in Nusselt number, and that the fin intensifies the Nusselt number. At γ = 0 and Re = 200, the Nussselt number is 7.5. When γ was increased to 0.04 at Re = 200, the Nusselt number increases from 7.5 to 8.3.

Investigation of mixed convection heat transfer on a vertical plate was considered by Akcay et al. [26]. There was application of heat flux on the plate and dimensionless oscillation amplitude between 0.4 and 1.4 was used to perform the investigation. The findings revealed that an increase in the amplitude of oscillation leads to an increase in heat transfer. The highest heat transfer performance of 1.45 is achieved with dimensionless oscillation amplitude of 1.4.

Bhuyan and Giri [27] published the findings on investigation carried out on mixed convection of turbulent flow in air-vapour system. The findings indicated at relative humidity of 0.6, Nusselt number due to condensation and Nusselt number due to sensible heat transfer are 15 and 7.5, respectively; and at relative humidity of 0.8, Nusselt number due to condensation and Nusselt number due to sensible heat transfer are 22 and 7.5, respectively.

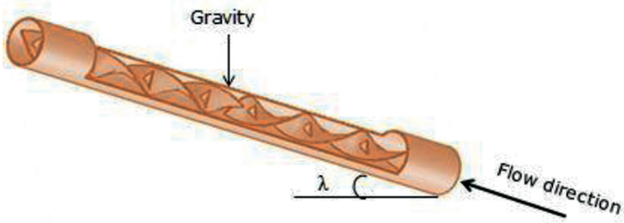

Figure 1. Geometry of flow model.

Mixed convection heat transfer was investigated on a mechanical ball bearing that was filled with different nano-lubricants, namely SiO_2-water and alumina- polyalphaolefin, by Hatami et al. [28]. It was observed through the results of the investigation that for a given quantity of nanoparticle in the lubricants, the nano-lubricant with SiO_2-water has less Nusselt number than the one with alumina- polyalphaolefin, and that the more the nanoparticles volume fraction, the more the Nusselt number. At Reynolds number of 200, the Nusselt number for SiO_2-water and alumina- polyalphaolefin is 3.5 and 7.5, respectively; at Reynolds number of 500, the Nusselt number for SiO_2-water and alumina-polyalphaolefin is 11.2 and 26.9, respectively.

It can be seen that the previous works chronicled above confirm that a report on mixed convection in a slanted tube that has a mechanical component put inside it is not obtainable. Hence, the present work examines numerically the thermo-hydraulic characteristics of mixed convection of transition flow in a tube. A mechanical component, which is a tape with three-sided-polygon-perforations-tranverse-axes, is put in the interior of the tube, and then tilted. The tilt has sundry value.

GEOMETRY OF FLOW MODEL

The model that was used to carry out the present research is shown in Figure 1. The model's geometry comprises a tube and a three-sided-polygon-perforations-tranverse-axes tape inserted in its interior. The tube has an inner diameter of 0.019 m and its length is 1 m. The twisted tape has a width of 0.018 m, a pitch of 0.054 m, and a thickness of 0.001 m. The angle at which the model is bent to the horizontal is represented by λ, and this was varied to obtain different directions of flow. It should be noted that part of the tube in the model (shown in Figure 1) has been removed to enable the tape to be sighted.

MATHEMATICAL MODELLING

The thermo-hydraulic characteristics of mixed convection heat transfer and fluid flow in this work is administered by the equations of Navier-Stokes and energy transport [29, 30]. The working substance is water and the flow is considered to be neither compressible nor unsteady.

The density is given by the relation

$$\rho = \rho_0[1-\beta(T-T_0)] \tag{1}$$

The mass conservation equation is

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \tag{2}$$

The momentum conservation equations are
x-momentum equation:

$$\rho\frac{\partial u}{\partial t}+\frac{\partial(\rho uu)}{\partial x}+\frac{\partial(\rho uv)}{\partial y}+\frac{\partial(\rho uw)}{\partial z}=-\frac{\partial \rho}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right)+gcos\lambda(\rho-\rho_0) \tag{3}$$

y-momentum equation:

$$\rho\frac{\partial v}{\partial t}+\frac{\partial(\rho vu)}{\partial x}+\frac{\partial(\rho vv)}{\partial y}+\frac{\partial(\rho vw)}{\partial z}=-\frac{\partial \rho}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right)+gsin\lambda(\rho-\rho_0) \tag{4}$$

z-momentum equation:

$$\rho\frac{\partial w}{\partial t}+\frac{\partial(\rho wu)}{\partial x}+\frac{\partial(\rho wv)}{\partial y}+\frac{\partial(\rho ww)}{\partial z}=-\frac{\partial p}{\partial z}+\mu\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right) \tag{5}$$

The energy equation is

$$\frac{\partial T}{\partial t}+\frac{\partial(uT)}{\partial x}+\frac{\partial(vT)}{\partial y}+\frac{\partial(wT)}{\partial z}=\frac{\partial}{\partial x}\left(\frac{k}{\rho C_p}\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{k}{\rho C_p}\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(\frac{k}{\rho C_p}\frac{\partial T}{\partial z}\right) \tag{6}$$

where u, v and w are the velocity components; ρ, T, β, p, k, and μ are the fluid's density, temperature, thermal expansion coefficient, pressure, thermal conductivity, and dynamic viscosity, respectively; λ is the tilt; g is the acceleration due to gravity; C_p is the specific heat at constant pressure.

At the pipe inlet, tube diameter (d) = 0.019 m, velocity (u), obtained from u = Re.μ/ρ.d, temperature at inlet (T_i) = 301K were set. The tube wall is subjected to uniform heat flux, expressed as $-k\left(\frac{\partial T}{\partial x}\right)_{d/2'}$ and no-slip condition (i.e. $u_i = u_j = u_k = 0$).

NUMERICAL SIMULATION AND TEST OF GRIDS FOR INDEPENDENCY

A variation of Shear-Stress Transport $\kappa - \omega$ model (that handles transition flow) of Fluent software was applied to the simulation. It was implemented for discretisation of the above-mentioned equations. The discretisation, which was carried out by applying second order upwind scheme, enables the computation of unidentified quantities at the cells' faces. SIMPLE (that is, Semi Implicit Pressure Linked Equations) [30], was employed to connect the velocity's calculation to that of the pressure's calculation.

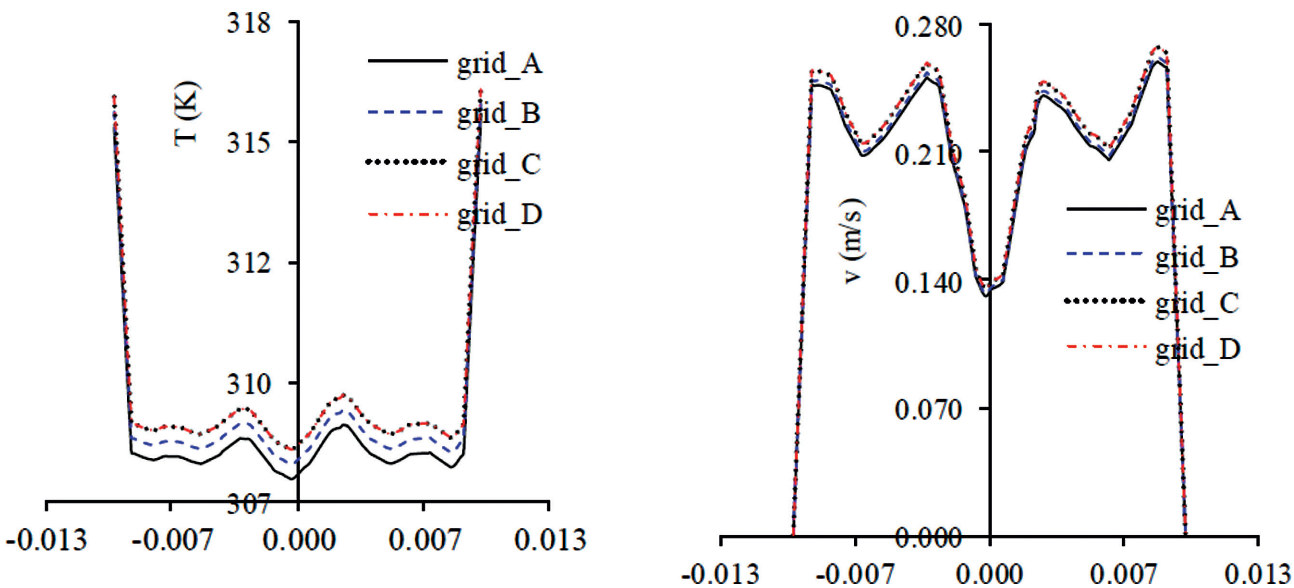

Figure 2. Grid independency test of the tube: (a) temperature, (b) velocity.

In order to be certain that the results of the simulation are accurate, four different grids of the tube were tested for independency. The grids are grid_A, grid_B, grid_C, and grid_D with cells of 1,255,502, 1,658,638, 2,515,756, and 3,018,907, respectively. Data of velocity and temperature for Re = 2,300 at the end of the flow in the tube were obtained. The results are presented in Figure 2. Figure 2 (a) reveals that although the deviation in the value of the temperature in grid_A from that in grid_C is small, it is noticeable. The same occurs in the deviation in the value of the temperature in grid_B from that in grid_C. The disparity between grid_C's temperature and grid_D's temperature is not noticeable, and therefore negligible. This means the results in this case are more satisfactory, compared with that between grid_A and grid_C and that between grid_B and grid_C. Interestingly, similar trend occurs in the values of velocity, as demonstrated in Figure 2 (b). It, therefore, means that grid_C or grid_D can be used for the simulation. Because of consideration of time for convergence of the solution, grid_D was chosen for the simulation.

RESULTS AND DISCUSSIONS

Distribution of Temperature Inside the Tube

The temperature distribution in the induced tube at different tilts is shown in Figure 3. Due to the presence of buoyancy, the tube's inside temperature for combined forced and free convection flow has a gain in its temperature as the slope of the tube increases. The temperature in the tube whose fluid flow is mixed convection at each of the tilts 15°, 30°, 60°, and 90° (Figure 3 (b to (e)) is higher than that in the tube whose fluid flow is forced convection (Figure 3 (a)). The inside temperature at 15°, 30°, 60°, and 90° is 7.42%, 7.61%, 7.84%, and 8.16%, respectively, more than when the flow is forced convection.

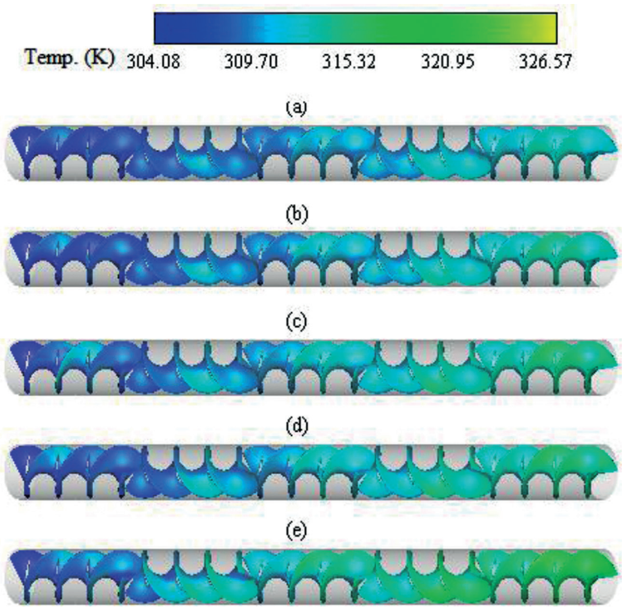

Figure 3. Distribution of temperature inside the tube for tilted flow of mixed convection: (a) forced convection, (b) 15°, (c) 30°, d) 60°, (e) 90°.

Dimensionless Outlet Temperature

Figure 4 shows the implication of the flow tilt on dimensionless outlet temperature. The temperature is the ratio of the difference between the outlet temperature and inlet temperature (T_o–T_i) to the inlet temperature (T_i). It can be inferred from the figure that with the increase in the Reynolds number, there is a decrease in the dimensionless outlet temperature. The temperature does not increase significantly as the flow orientation rises. The increment in temperature is more noticeable at Reynolds number of 3,700 before the tilt of 30°. As the Reynolds number rises from 2,150 to 3,700, the dimensionless outlet temperature drops from 0.3051 to 0.3041 at tilt of 15°, but drops from 0.3052 to 0.30436 as the tilt increases to 90°.

Distribution of Surface Temperature

The distribution of surface temperature is shown in Figure 5. The temperature at the surface increases from the tube's inlet to its outlet. For the Reynolds numbers under consideration, that is, *Re* = 2,150 (Figure 5 (a)), *Re* = 3,000 (Figure 5 (b)), and *Re* =3,700 (Figure 5 (c)), the temperature at the surface increases when the tube tilt increases. Furthermore, the surface temperature for the combined free and forced convection flow is greater than the case of forced convection. For instance, when the Reynolds number is 3,700, (Figure 5 (c)), the surface temperature at 15°, 30°, 60°, and 90° are 6.11%, 6.20%, 6.32%, and 6.71%, in that order, more than when the flow is forced convection.

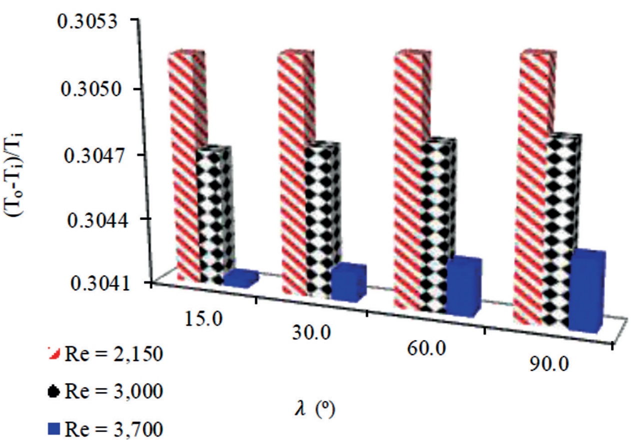

Figure 4. Dimensionless outlet temperature for various Reynolds numbers and flow tilt of mixed convection.

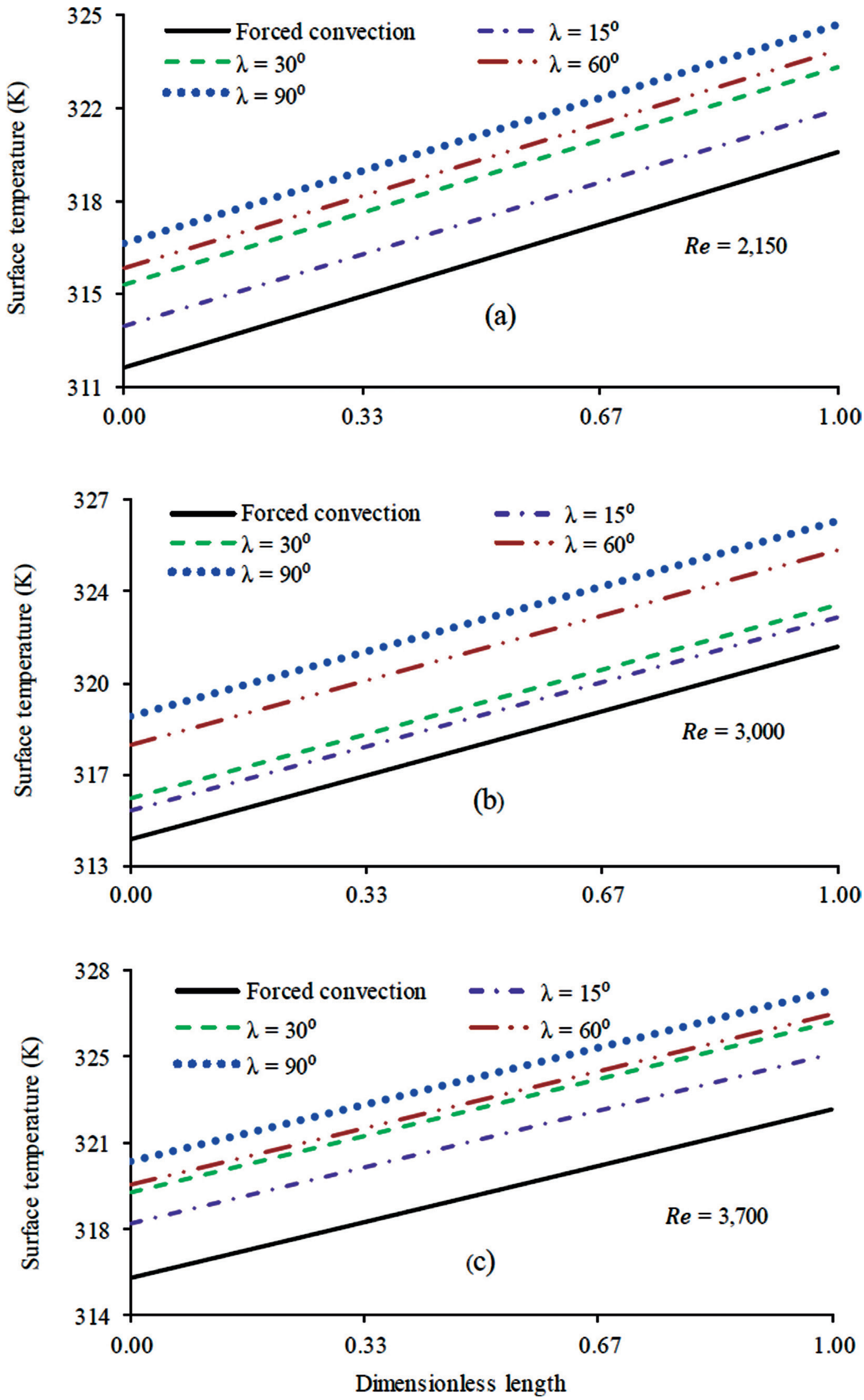

Figure 5. Distribution of surface temperature for various Reynolds numbers and flow tilt of mixed convection.

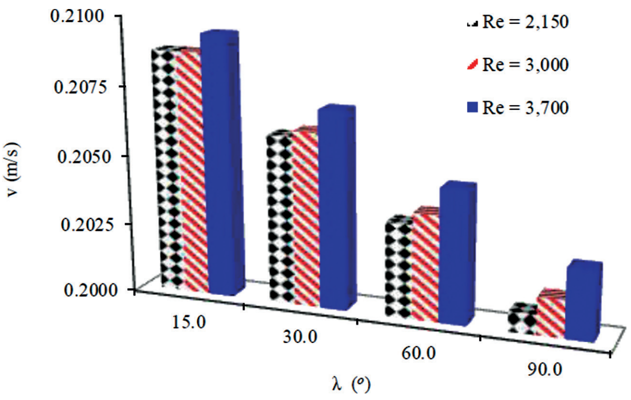

Figure 6. Outlet velocity for various Reynolds numbers and flow tilt of mixed convection.

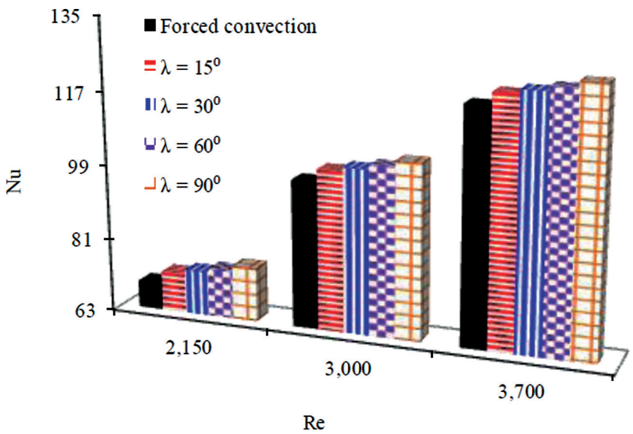

Figure 7. Nusselt number for various Reynolds numbers and flow tilt of mixed convection.

Outlet velocity

The implications of the different values of the Reynolds number and the flow tilt on the outlet velocity of the flow is shown in a graphical form in Figure 6. It is easily understood in the figure that the velocity becomes larger as the Reynolds number increases. Consequent upon a drawback to the movement of the fluid toward the end of the tube, its outlet velocity reduces as the tilt of the flow increases. A sharp reduction in the flow velocity before tilt of 30° is observed. At the tilt of 15°, the velocity is 0.2088m/s, 0.2088m/s, and 0.2094m/s for Reynolds number of 2,150, 3,000, and 3,700, respectively. An increase in the tilt to 90° decreases the velocity to 0.2007m/s, 0.2013m/s, and 0.2025m/s for Reynolds number of 2,150, 3,000, and 3,700, respectively. For Reynolds number of 3,700, the velocity at the tilts of 15°, 30°, 60°, and 90° is 0.2094m/s, 0.2071m/s, 0.2048m/s, and 0.2025m/s, respectively.

Nusselt Number

A dimensionless quantity known as Nusselt number (Nu) is used to express the heat transfer. Its values for Reynolds numbers and flow tilts of the mixed convection are depicted in Figure 7. It is verifiable from the figure that the Nusselt number in the tube whose fluid flow is mixed convection is greater than the case of the tube whose fluid flow is forced convection, and that the maximum augmentation in the heat transfer is obtained when the tube is leaned at 90°. Specifically, the Nusselt number in the tube with flow tilts of 15°, 30°, 60°, and 90° is 4.01 to 6.10%, 4.60 to 7.41%, 5.32 to 8.71%, and 6.31 to 10.12%, in that order, more than the case of the tube whose fluid flow is forced convection, for the Reynolds number considered.

Validation with an Experimental Work

In order to be certain that the results of the numerical work are correct, the numerical results of the surface temperature for the various Reynolds numbers and the different flows were validated with its experimental results, as shown in Figure 8 below.

Table 1. Standard errors from the validation of the numerical results with its experimental results

Flows	Maximum standard errors (%)		
	Re = 2,150	*Re* = 3,000	*Re* = 3,700
Forced convection	3.86	3.24	3.86
λ = 15°	2.78	3.86	3.09
λ = 30°	3.09	4.63	2.62
λ = 60°	3.86	3.09	4.63
λ = 90°	4.63	2.78	3.09

The maximum standard errors of the surface temperature between the simulated results and its experimental results for the different flows for each of the Reynolds number are presented in Table 1. The values of the standard errors confirm that the numerical results and the experimental results satisfactorily agree with one another. This means that the numerical results are correct.

CONCLUSIONS

The present work numerically considers the thermohydraulic characteristics of mixed convection of transition flow in a tube that was modified by inserting a multi-tranverse-axes-three-sided-perforations tape inside it. The modified tube was leaned at different angles of 15°, 30°, 60°, and 90°. The flow regime for the transition flow is 2,150 ≤ Reynolds number ≤ 3,700. The following conclusions can be drawn:

- The inside temperature and the temperature at the surface of the tube whose fluid flow is combined forced and free convection increase as the tilt of the tube increases in value, and the temperatures are greater than those in the tube whose fluid flow is

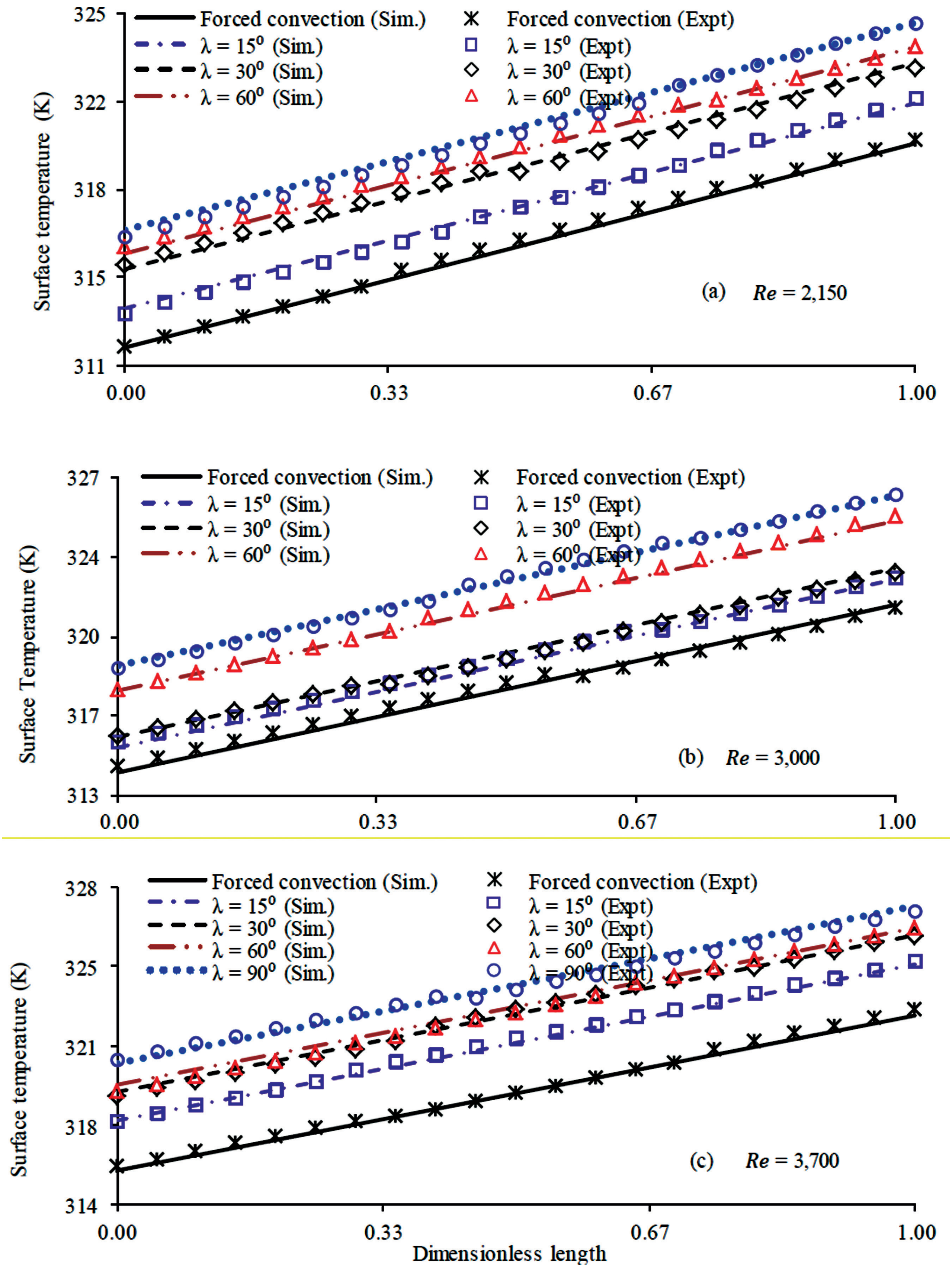

Figure 8. Validation of numerical results of surface temperature with experimental results.

forced convection. The surface temperature at 15°, 30°, 60°, and 90° is 6.11%, 6.20%, 6.32%, and 6.71%, in that order, more than when the flow is forced convection. The inside temperature at 15°, 30°, 60°, and 90° is 7.42%, 7.61%, 7.84%, and 8.16%, respectively, more than when the flow is forced convection.
- The dimensionless outlet temperature drops from 0.3051 to 0.3041 at a tilt of 15° as the Reynolds number rises from 2,150 to 3,700, but drops from 0.3052 to 0.30436 as the tilt increases to 90°.
- The outlet velocity rises as the Reynolds number increases, but reduces as the tilt of the flow increases. At the tilt of 15°, the velocity is 0.2088m/s, 0.2088m/s, and 0.2094m/s for Reynolds number of 2,150, 3,000, and 3,700, respectively. An increase in the tilt to 90° decreases the velocity to 0.2007m/s, 0.2013m/s, and 0.2025m/s for Reynolds number of 2,150, 3,000, and 3,700, respectively.
- The Nusselt number for combined forced and free convection flow (that is, mixed convection flow) is greater than the case of flow that is forced convection. For the tube with flow tilts of 15°, 30°, 60°, and 90° , its Nusselt number is 4.01 to 6.10%, 4.60 to 7.41%, 5.32 to 8.71%, and 6.31 to 10.12%, in that order, greater than the case of flow that is forced convection.
- The outcome of the validation of the numerical results with experimental results reveals the satisfactory agreement among them, and, therefore, the correctness of the numerical results.

This work has, undoubtedly, widened research on heat transfer applications by delving into thermo-hydraulic attributes of mixed convection of transition flow in a slanted tube that has a mechanical component put inside it.

For future research, the areas that can be worked on are other values of Reynolds number, velocity vector to explore flow effect and change of velocity boundary layer and thermal boundary layer, use of different working fluids, and alteration in geometry of flow model.

AUTHORSHIP CONTRIBUTIONS

The work in this article is the contribution of the author.

DATA AVAILABILITY STATEMENT

The author confirmed that the data that supports the findings of this study are available within the article. Raw data that support the findings of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this article.

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