

Heat convection enhancement of heat exchanger with cylindrical spring inserts: A numerical analysis

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ABSTRACT

In this research, the characteristics of thermo-hydraulic for water flow into a double-pipe heat exchanger (DPHE) with cylindrical spring insert inside a tube side was numerically investigated. Three-dimensional numerical simulation of turbulence model utilizing ANSYS-Fluent version 2021 R1 commercial software was performed. Different numbers of cylindrical spring inside a tube side of heat-exchanger, namely four, six and eight besides the smooth (without springs) tube case were simulated. The water as working fluid in both annular and tube sides was used. Various Reynolds numbers in tube side ranging from 4000 to 10,000 and it fixed at 7000 in annular side. The aim of present research is to simulate and analyze numerically the influence of cylindrical spring inserts number, Reynolds number and surface temperatures on the thermo-hydraulic characteristics of water flow into a double pipe heat-exchanger under forced heat convective conditions. The numerical results seem that the coefficient of overall heat transfer are greater about 14%, 18.7% and 21.4% for cases of 4, 6 and 8-springs respectively compared with smooth tube case. The large pressure drop occurs in 8-springs case about 2.1 times than that smooth tube. Besides, the 6-springs case has a bigger values of thermal performance factor about 4% and 9.5% than that 8-springs and 4-springs respectively.

KEYWORDS

Numerical simulation, Double pipe heat exchanger, Cylindrical spring, Water flow, Forced convective.

INTRODUCTION

The heat dissipation processes for two or more fluid flows with various temperatures have industrial and domestic applications. Generally, the heat-exchangers are utilized with this aim. Many different types of a tabulator devices namely, wire springs and twisted tapes utilized for improvement thermal performance of heat-exchangers. The coil spring inserts are one of the passive heat technique to improve heat transfer rate interdependent in different industrial applications such as cooling and air conditioning systems, preheaters system and heat exchangers. The coil spring inserts are considered an easy manufacturing and installing, besides cost is less. The wire springs provide an acceptable heat transfer improvement with minimum pumping power required as comparison than other tabulator devices [1].

A literature review that will display illustrate many researches of insertion configuration in the tube side of double pipe heat-exchanger. San et al. [2] evaluated experimentally the pressure drop and rate of heat dissipation for a circular-tube utilizing coiled wire inserts with various pitch length and diameter of spring. They noted that the increment of spring diameter and decreasing spring pitches, the values of Nusselt number increases. Zimparov and Penchev [3] evaluated and compared heat and hydraulic properties of coiled-spring and twisted-tape insertions into a heat exchanger under turbulence-flow condition. They noted that the coiled-spring insertions give more efficiency and performance compared with twisted-tape insertion. Kumar et al. [4] developed numerically a new configuration of inserts in tube side of double pipe heat-exchanger (DPHE). They modeled the triangular perforated twisted-tape configuration with V cuts in both laminar and turbulent flow zones. They showed that the thermal performance factor (TPF) of a new shape (triangular perforated twisted-tape) about 1.49 and it's greater than that the classical twist-tape insert.

Nakhchi et al. [5] investigated numerically the heat performance of turbulence flow into tubes of heat exchanger. They used a perforated louvered-strip insertions and various slant-angles. They noted that the re-

circulation of flow during a circular holes of the perforated louvered-strips enhances rate of heat transfer compared with the louvered-strips without holes. They showed that the peak of heat performance factor is (1.84) for double-perforated louvered-strips at slant angle of 25° . Al-Obidi and Chaer [6] presented a numerical study of heat and hydraulic behaviors for fluid flow into a circular tube of heat-exchanger with and without twisted-tapes inserts at a wide range of Reynolds number. They utilized various twisted-tap widths. They noted that the wider width twisted-tap gives largest heat performance about 29%. Nakhchi and Esfahani [7] numerically computed and analyzed the structure of flow and heat performance through a heat-exchanger tubes using perforated hollow cylinders under turbulence flow condition. Different models in an aspect ratio (tube diameter to diameter of hollow cylinder) was studied and simulated. They noted that the maximum heat performance about 1.45 for aspect ratio of 0.74 at low values of Reynolds number.

Khashayar et al. [8] conduct numerical study of spring wire fitted inside tube in a double-pipe heat exchanger (DPHE) under laminar flow regions. They conclude that taking the important of proper wire coils fitted inside tube which the Nusselt values improved to 1.77 times as compared with smooth case. Jolaghani and Lavasan [9] investigated experimentally the effect of spiral springs on heat transfer and pressure drop in DPHE. The spring have a square cross section area which inserted at the tube side. Different cross section area and spring pitches are tested, which inserted at the tube side. They found that the case of inserting the coils into the internal tube, the coefficient of heat transfer increases about (45%) and pressure drop increases about (77%) than that without-coils case conditions. Sharafeldein et al. [10] conduct experimentally the effect spiral spring on thermal hydraulic performance through a pipe under high turbulence flow. They used a spiral-springs with various diameters and pitches inside the pipe. They found the performance criteria within (1.01 to 1.28) which evidenced that the spiral-springs insert gives a larger heat-transfer rate. Generally, more of researches in this field (using springs or coil wires insert) are conducted with experimental work while a few researchers are employed the numerical simulations like, [8] and [11]. The present research simulates and analysis numerically the influence of cylindrical springs number interior a tube side, Reynolds number and surface temperatures on the properties of thermo-hydraulic for turbulent water-flow into a double-pipe heat exchanger.

Computational details

The study exploit finite-volume technique based on commercial software of ANSYS Fluent version 2021 R1. The numerical model are conducted to analyze three dimensional (3D) incompressible Navier-Stokes fluid flow and heat transfer through double-pipe heat exchanger. The explanations of computational flow domain, grid mesh and the boundary conditions as follows:

Physical model and meshing

The physical model (double pipe heat-exchanger) illustrated in Figure 1, consists of a tube and shell with internal diameters of ($d_i = 28.5$ mm) and ($D_i = 50.8$ mm) respectively and tubes length of ($L = 1000$ mm). The shell has inlet and outlet ports which have the same internal diameters of tube. A tube is extended in length at entrance side with 500 mm to obtain a fully developed fluid into heat-exchanger. Different numbers of springs wire have modeled in the cases namely, 4, 6 and 8 springs as shown in Figure 2a. The smooth case represents (double pipe heat-exchanger) without springs. The springs are open free ended as shown in Figure 2b. The spring geometry and dimensions are the same for all cases, the technical specifications clarified in the table1. A cartesian cut cell mesh method type is adopted that generated by cutting the solid bodies out of a background cartesian mesh [12]. The mesh treatment is done with taking a consideration the spring curvatures and contact regions between the spring surface and the tube in order to capture the details of boundary layer flow. The enlarge view of mesh generation in parts of DPHE are shown in Figure 3. For the best accuracy results with less computational time, the mesh independence test has been conducted. The grid independence of (DPHE) has been tested. The results of simulation for 8-springs insert case at tube and annular sides are shown in Figure 4. They are show that the overall heat transfer (U) is tending to be stable after mesh cell number about 2,230,000.

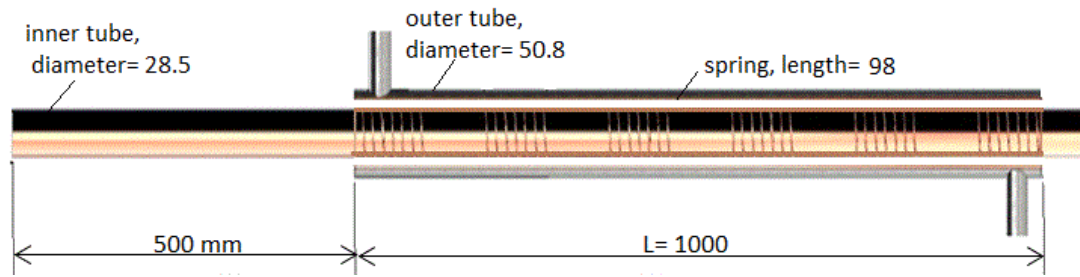
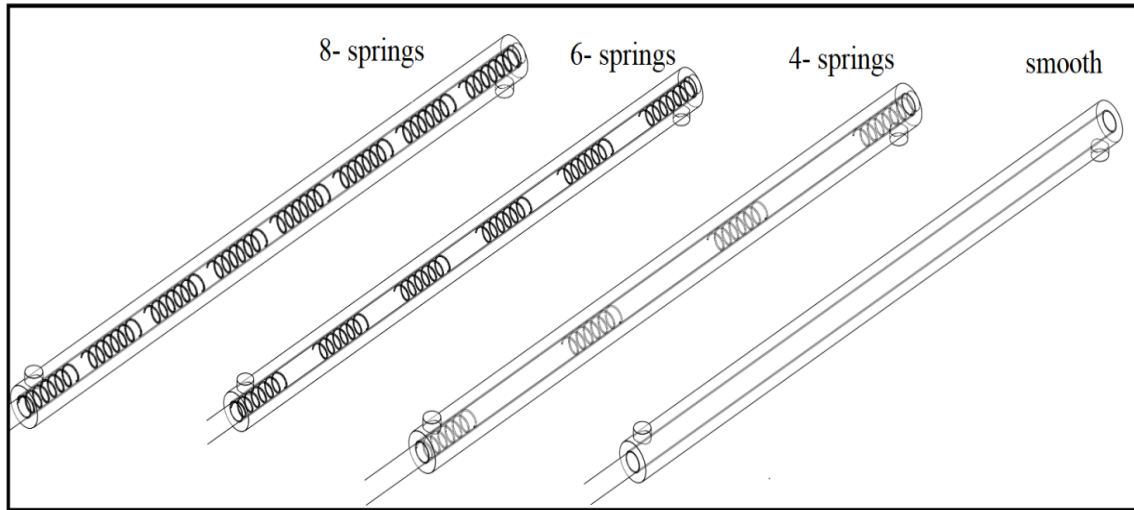
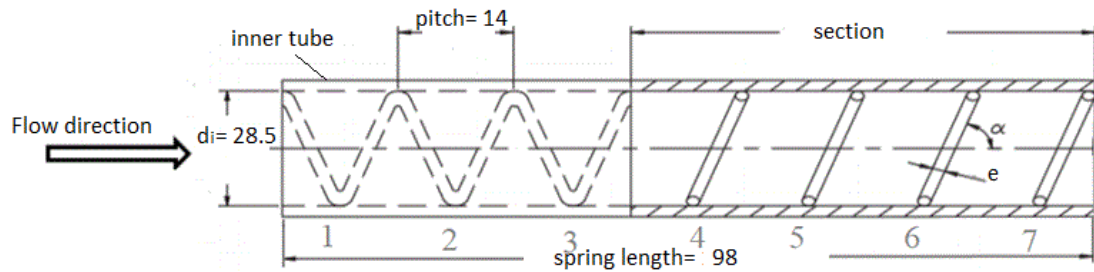


Figure 1. Physical model of double pipe heat-exchanger (all dimensions in mm).



(a)



(b)

Figure 2. Heat exchanger models:(a) models simulated of (DPHE) (b) section of inner tube with cylindrical spring geometry (all dimensions in mm).

Table 1. Geometrical dimensions of cylindrical spring.

Dimensions		Values
Pitch length (p)	14 (mm)	
Spring diameter =internal tube diameter (di)	28.5 (mm)	
Spring free length (L_s)	98 (mm)	
Wire diameter (e)	2.5 (mm)	
Helix angle (α)	80°	
Number of turns (N_t)	7	

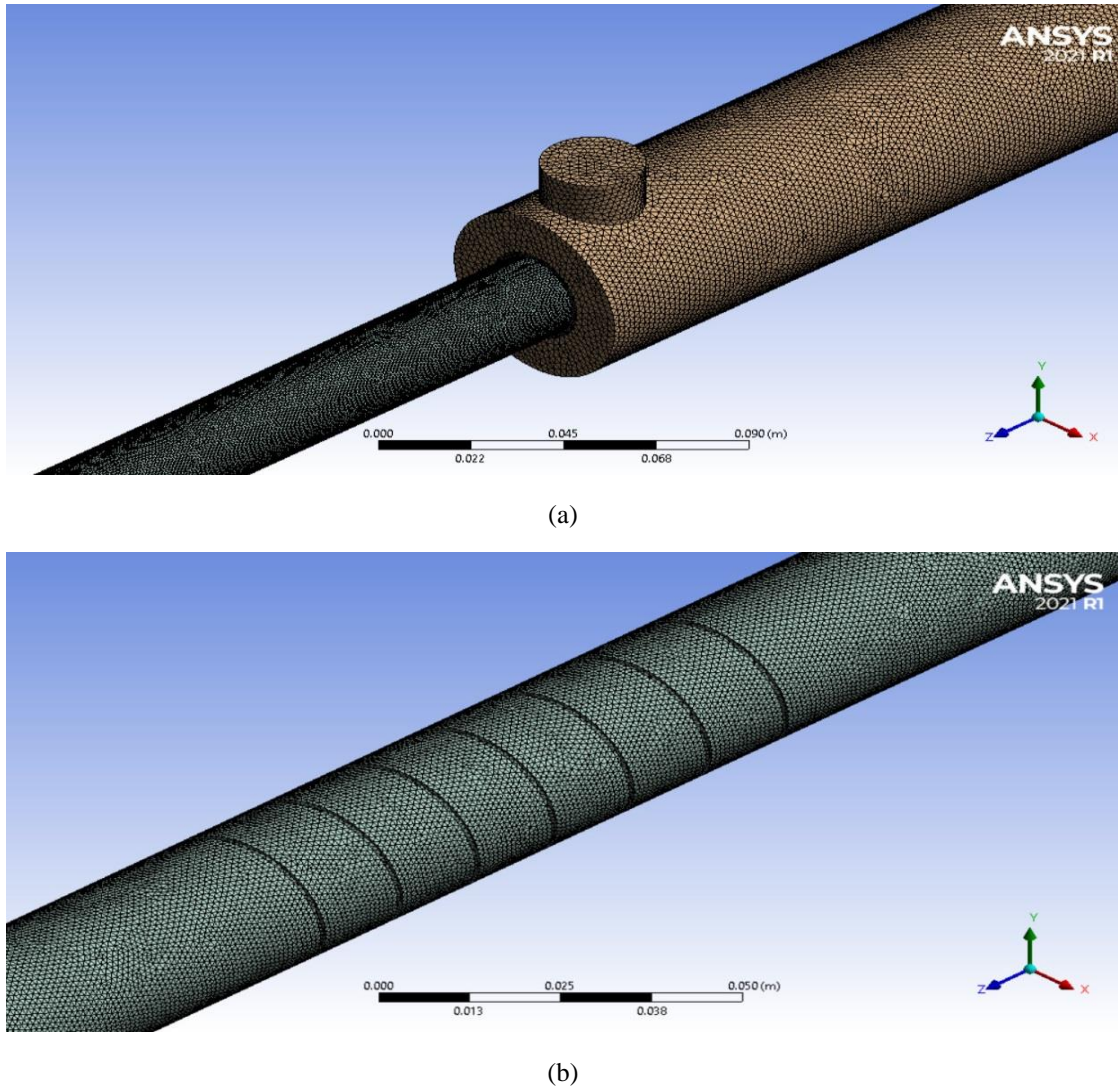


Figure 3. Mesh grids generation (a) cut of view part side of heat exchanger (b) one spring inside tube along z axis position.

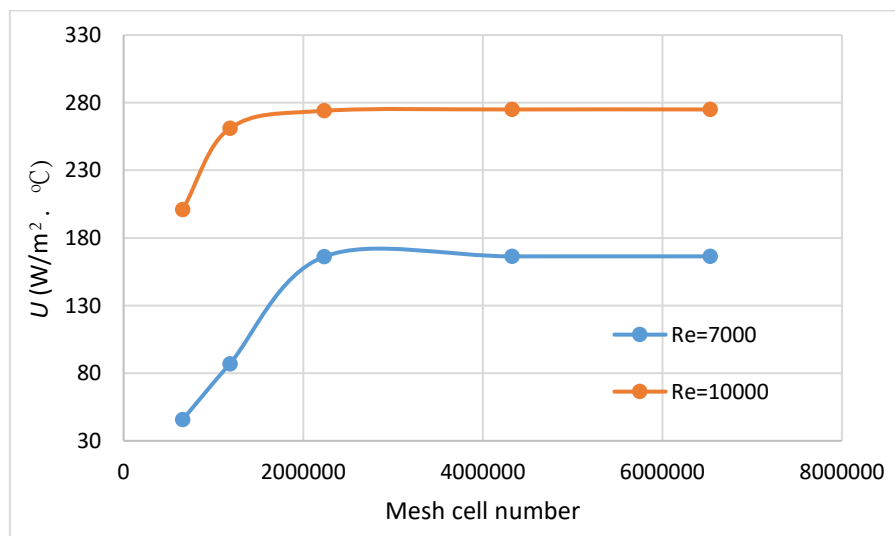


Figure 4. Behavior of overall heat-transfer coefficient (U) versus number of mesh cells for 8-springs inserts (DPHE) case.

Governing equations and mathematical model

The (3D) numerical model of water flow and heat transfer in a double-pipe heat exchanger (DPHE) was developed under the following assumptions:

- A three dimension, incompressible and steady-state for water flow and heat transfer.
- both fluids (water) are turbulence.
- The radiation effects were neglected.
- No slip condition and uniform inlets water velocity.
- Both fluids are constant properties.

The governing equations (in Cartesian coordinates) are [13, 14]:

- The continuity equation:

$$\left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z}\right) = 0$$

- The momentum equations:

x- momentum equation

$$\left(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z}\right) \rho = -\frac{\partial P}{\partial x} + \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right) \mu$$

y- momentum equation

$$\left(u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z}\right) \rho = -\frac{\partial P}{\partial y} + \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2}\right) \mu$$

z- momentum (in axial direction) equation

$$\left(u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z}\right) \rho = -\frac{\partial P}{\partial z} + \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2}\right) \mu$$

where, ρ and μ are the density and dynamic viscosity of water in respectively.

P is the local water pressure.

The energy equation:

$$\rho C_p \frac{\partial T}{\partial t} + \rho C_p \left(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z}\right) = k \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}\right)$$

where, C_p and k are water specific heat and the thermal conductivity respectively.

The standard (k- ϵ) model is selected in this study which is mostly used for heat exchangers simulation. For the (k - ϵ) model, turbulence modeling is depended on the transport equation of turbulence kinetic energy (k) and its rate of dissipation (ϵ), which are writing as the following in respectively.

$$\rho u_j \frac{\partial k}{\partial x_j} = \frac{\partial}{\partial x_j} \left[\mu_t \frac{\partial k}{\partial x_j} \right] + G_k - \rho \epsilon$$

$$\rho u_j \frac{\partial \epsilon}{\partial x_j} = \frac{\partial}{\partial x_j} \left[\mu_t \frac{\partial \epsilon}{\partial x_j} \right] + C_{1\epsilon} \frac{\epsilon}{k} G_k + C_{2\epsilon} \rho \frac{\epsilon^2}{k}$$

where,

$$G_k = \mu_t \left[\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right] \frac{\partial u_i}{\partial x_j}$$

and,

$$\mu_t = C_\mu \rho \frac{k^2}{\epsilon}$$

The factors constant values are in the following values:

$$\sigma k = 1.0, \sigma \varepsilon = 1.3, C_\mu = 0.09, C_{1\varepsilon} = 1.44, C_{2\varepsilon} = 1.92$$

Boundary conditions

For boundary conditions setting, inner conditions of the tube and annular sides of DPHE are set as the rate of mass flow inner condition with a constant temperature. For current study, the hot fluid enters into the internal tube at 50°C with different flow rate ranged between 0.08 to 0.2 m/s while the cold fluid enters into the external tube at 20°C and flow rate of 0.26 m/s. Both flows are turbulent and the turbulence intensity is ($I_{in} = 5\%$). The outer conditions are arranged as the pressure outlet condition with constant backflow turbulence intensity. The annular side surface is arranged as adiabatic surface and the tube side surfaces are coupled surfaces under (no-slip) boundary condition.

Mathematical model

The numerical average heat transfer rates (Q_{avg}) is computed as follows:

$$Q_{avg} = \frac{Q_t + Q_a}{2}$$

where, (Q_t) and (Q_a) are the heat transfer rates on tube and the annulus sides respectively, and calculated as [13]:

$$Q_t = m_t C_{pt} (T_{h,i} - T_{h,o})$$

$$Q_a = m_a C_{pa} (T_{c,o} - T_{c,i})$$

Define the overall heat transfer coefficient (U) based on inside surface tube area as:

$$U = \frac{Q_{avg}}{A_i LMTD}$$

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln \left(\frac{\Delta T_1}{\Delta T_2} \right)}$$

where, A_i is the surface area of internal tube and $LMTD$ is the logarithmic-mean temperature difference.

$$\Delta T_1 = T_{h,i} - T_{c,o}$$

$$\Delta T_2 = T_{h,o} - T_{c,i}$$

The total pressure drop (P_t) of double-pipe heat exchanger is evaluated as:

$$P_t = \Delta P_t + \Delta P_a$$

where, ΔP_t is the pressure difference between input and output points at ends of tube sides, (ΔP_a) is the pressure difference between input and output ports at shell side. Define, the thermal performance factor (TPF) as the ratio of overall convection heat transfer coefficients according to change the total pressure drop of both tube and annular sides. It is a vital parameter that indicates the potential of spring wire in comparison with the smooth case [13, 14].

$$TPF = \frac{(U/Us)}{\left(\frac{(\Delta P_t + \Delta P_a)}{(\Delta P_{ts} + \Delta P_{as})} \right)^{0.333}}$$

where, ΔP_{ts} and ΔP_{as} are the pressure difference of tube and annular side for the smooth model respectively. If ($TPF > 1$) indicates that the rate of heat transfer is bigger than those losses of pump power.

Define, the Reynolds (Re) number as [14]:

$$Re = \frac{\rho V D}{\mu}$$

where, V is water inlet temperature, D is the hydraulic diameter of both tube and shell inlet port side, and ρ and μ are water density and water viscosity respectively. Both Reynolds numbers (Re) of tube and annular sides are calculated based on tube and ports inlet diameter respectively.

Validation of numerical model

To validity the numerical prediction, the Nusselt number and friction factor of smooth tube case was validated against the experimental results of Pourahmad and Pesteei work [15] as shown in Figure 5. The maximum deviation between two works are 8.3% and 13% for Nusselt number and friction factor respectively.

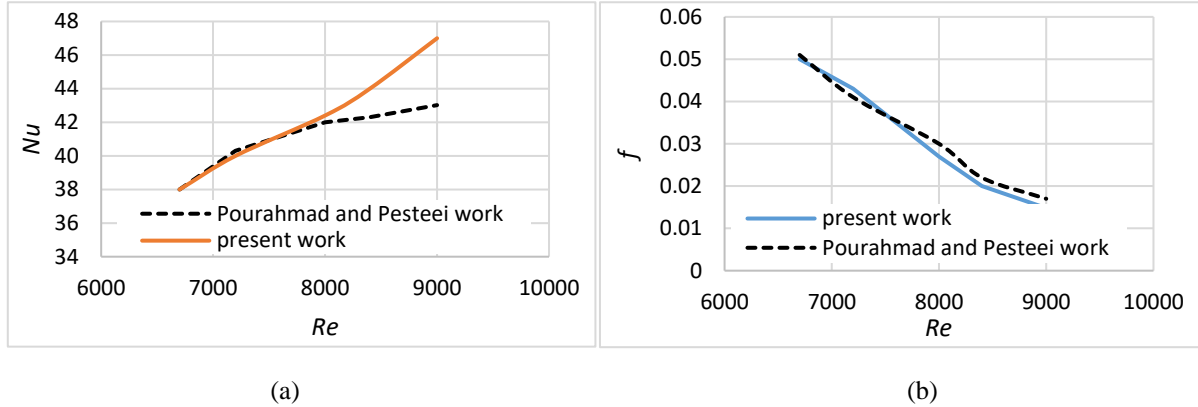


Figure 5. Validation of present work with Pourahmad and Pesteei work [15] (a) Nusselt number (b) friction factor.

RESULTS AND DISCUSSION

To investigate the flow field, Figure 6 displays the velocity contours in the middle section of DPHE cases at Reynolds number of (8500). As clear seen the velocity contours is high along the cylindrical springs and reduced in the spaces between them. This explained the effect of wire spring for generated more turbulence. When the water moves through the springs, the flow is disturbed induced continual vortex flow whole length for each springs. Then, the vortex flow is to decay along the free space in tube and then returned back to generate at another cylindrical spring length. The lesser number of springs inside tube provided poorer mixing in fluid. Moreover, pressure drop is viewed in the Figure 7. Generally, the pressure drop at annular side is constant with a small value of (41 Pa) for all models because the cold-water flow rate is fixed. All models with spring insertions are bigger performance compared with smooth tube model because the dissipation of dynamic pressure for water due to the flow resistance caused by the vortex flow.

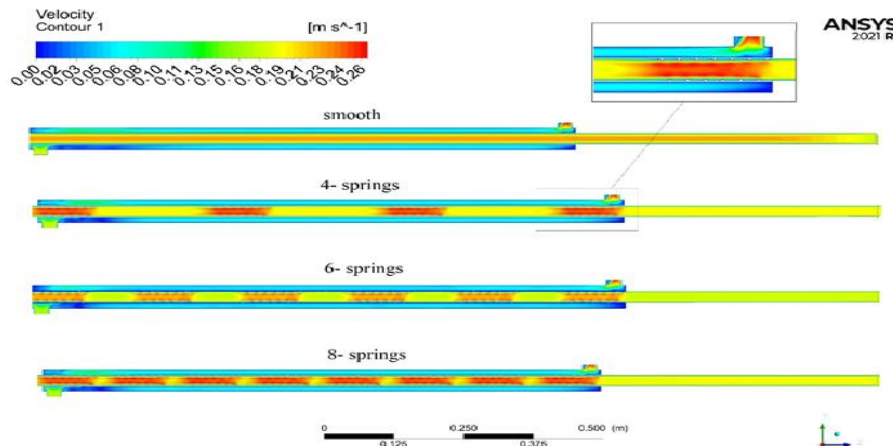


Figure 6. The velocity counters along (DPHE) for smooth tube, 4, 6 and 8- springs models.

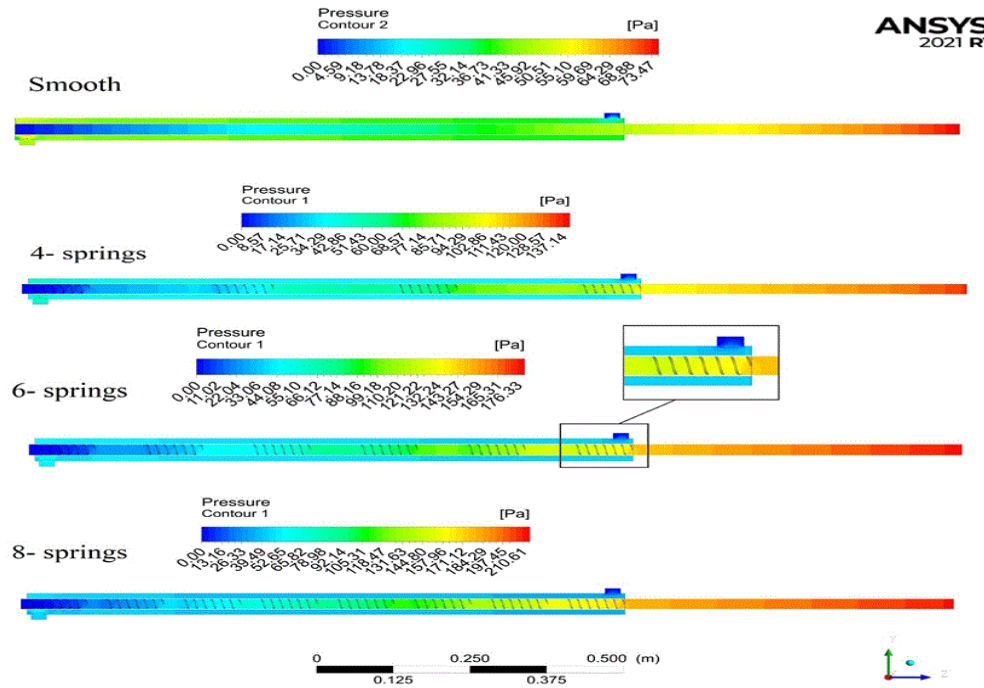


Figure 7. The pressure counters along (DPHE) for smooth tubes, 4, 6 and 8- springs models.

To understand the thermal field in different models, Figure 8 shows the temperature profiles in different models in the middle section of DPHE. Based on analyzing of results for all models, the temperatures of hot water decreases towards the tube exit while the cold water is to be hotter towards the exit port at the annular side. This explains the heat exchanged between hot and cold water for all studied models. The water temperature at exit tube section in smooth model is hottest than other model. This attribute due to spring effect for enhance the rate of heat transfer by minimizing the thermal boundary layer thickness and disturbing velocity boundary layer and this permitting the core fluid to mix with the boundary fluid. Temperatures of core fluid for 8-springs model are coldest flowed than the 6-springs-model. This declare the effect of increase the number of cylindrical springs to allow more mixing along the tube length. The results of heat transfer enhancement in terms of overall heat transfer coefficient (U) at different Reynolds number (Re) are shown in Figure 9.

For all cases, the overall heat transfer increases with increasing Reynolds number because the increasing of vortex flow velocity as flow increase. Based on Reynolds number ranged and number of springs, overall heat transfers of flow in tube with 4-springs model improved by 14% as compared than smooth model while in 8-springs model, the improved is jumped up to 21.4% than smooth model. Figure 10. shows the behavior of pressure drop against Reynolds number for each models. It is found the values of pressure drop increased with increasing Reynolds number in all cases. The pressure drop of flows in the 8-springs model increased up to 2.1 times than the smooth model. Figure 11 shows the variation in the (TPF) with different Reynolds numbers of tube side (Re). For all models, the TPF is high in magnitude at low Reynolds number, this illustrates the heat transfer is high with less pressure drop. The 6-springs model gives higher (TPF) for all Reynolds number values ranged from (1.028 to 1.257) which indicates the highest heat transfer rate and the lowest pressure drop. The (TPF) for some values of (Re) of 4-springs and 8-springs models are ($TPF < 1$), that explain the pressure drop is a dominate.

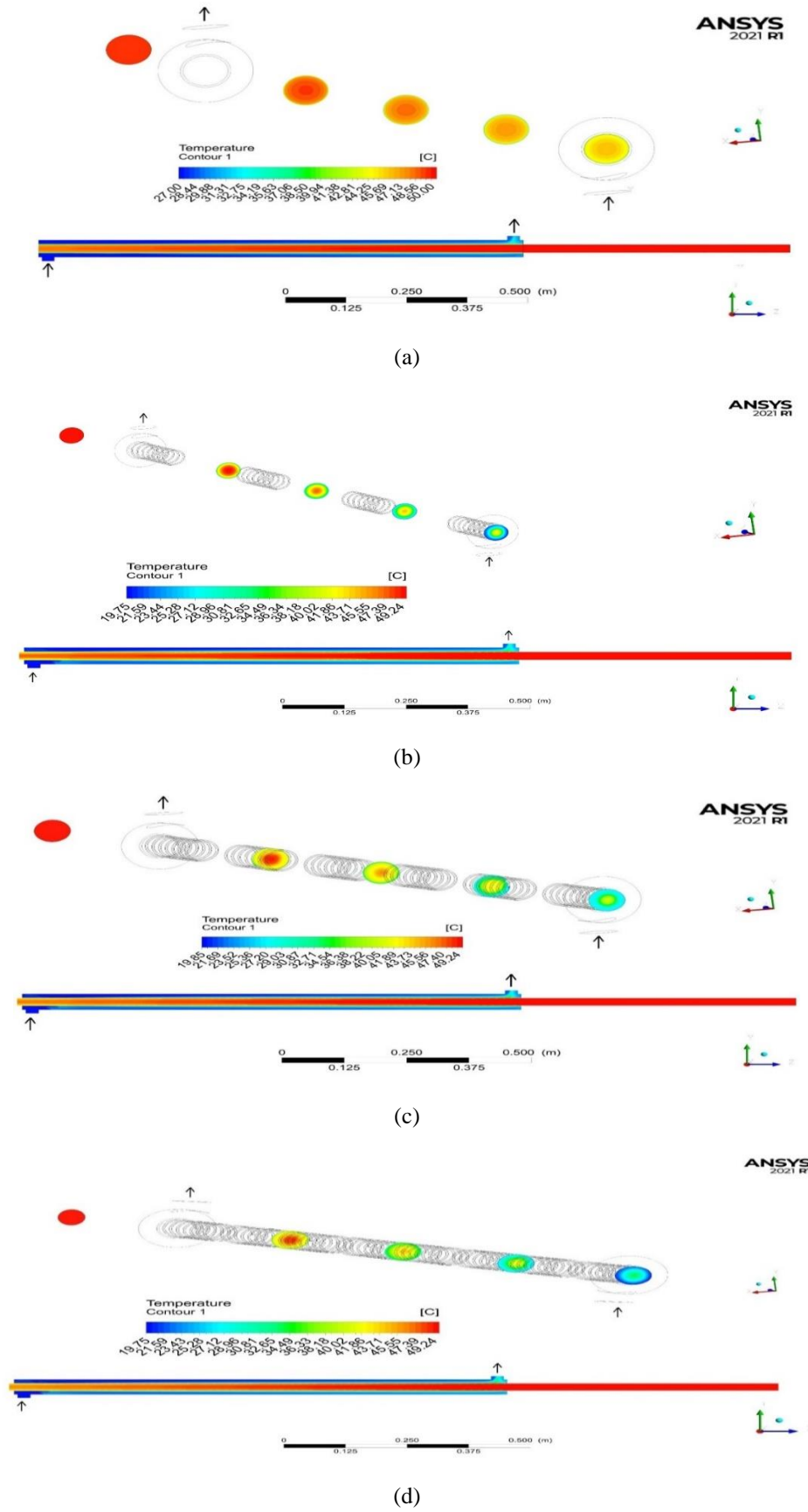


Figure 8. The temperature counters along (DPHE) for model: (a) smooth tube (b) 4-springs (c) 6-springs (d) 8-springs.

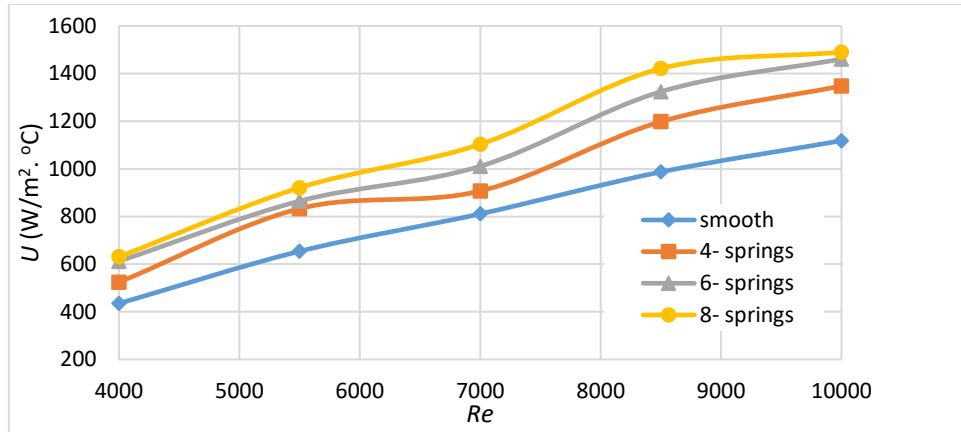


Figure 9. Behavior of overall heat transfer coefficient (U) with Reynolds number at tube side for different models.

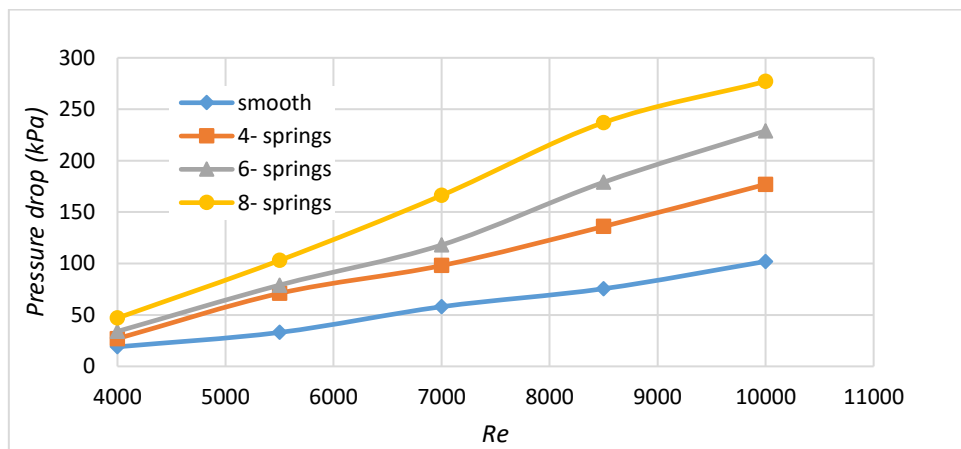


Figure 10. Behavior of pressure drop with Reynolds number (Re) at tube side of heat-exchanger for different models.

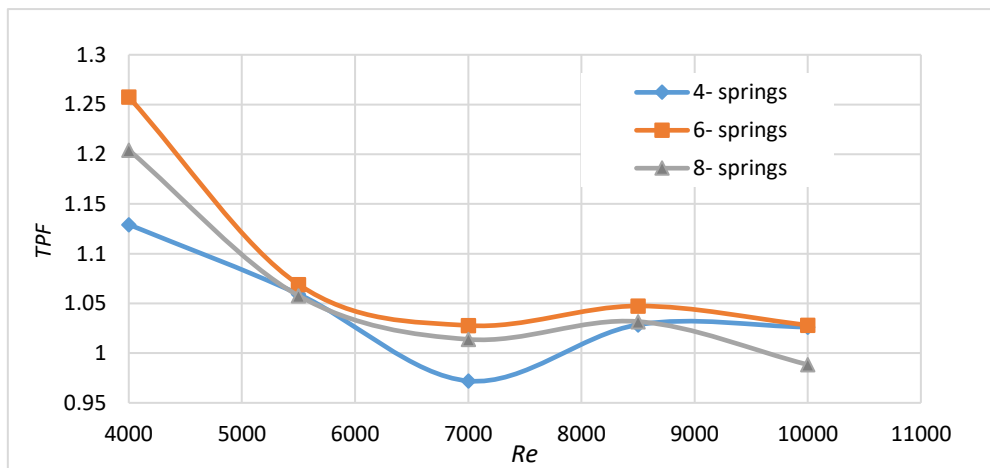


Figure 11. Thermal performance factor (TPF) versus Reynolds number (Re) at tube side for model with different numbers of spring.

CONCLUSIONS

Four models of double pipe heat-exchanger (DPHE) like, smooth tube model and 4, 6 and 8-springs models have been tested numerically. Improvement of heat transfer, friction factor and thermal performance factor

(TPF) have been studied and analyzed. Also, the influence of springs number, surface temperatures and Reynolds number on thermal and hydraulic properties were studied. The main conclusions can be listed as:

1. Rate of heat transfer increased as number of springs increase, the overall heat transfer coefficient is bigger about 14%, 18.7% and 21.4% for models of 4, 6 and 8-springs respectively compared than smooth tube model.
2. The large pressure drop occurs in 8-springs model about 2.1 times than that smooth tube.
3. The 6-springs model has a maximum values of thermal performance factor (TPF) about 4% and 9.5% than that 8-springs and 4-springs respectively.
4. Maximum deviations between the present work and available previous work about 8.3% and 13% for Nusselt number and friction factor respectively.

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