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# Effects of pipe turns on vertical helically coiled tube heat exchangers for water heating in a household refrigerator

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## Abstract

A new coupled numerical model of the overall vapor-compression cycle and water heater was developed in order to analyze the influence of condenser coil designs on the heat transfer process of a helical coil heat exchanger immersed in a storage tank. The model was developed using the CFD code ANSYS Fluent. In the numerical simulation, the helical condenser coil was treated as an internal heat source and the initial heat flux was obtained from experimental results and delivered to each coil element in the CFD model as boundary condition.

The coupled model was validated with experimental results and showed that the numerical results of the proposed model are in good agreement with the experimental data with the deviation of  $\pm 1.6\%$ . Then, the impact of the main representative geometrical parameter on the overall heat transfer coefficient and water temperature distribution was investigated. The results indicated that the temperature distribution and overall heat transfer coefficient increase with the increase of the pipe turns of a helically coiled tube heat exchanger.

**Keywords:** Helically coiled tube heat exchangers, Structure optimization, Computational Fluid Dynamics, Heat transfer, Water heater, Natural convection

## 1 Introduction

Compactness, ease of maintenance, higher thermal efficiency, high operating pressure, and strong temperature gradients are all advantages of helically coiled tube heat exchangers for household hot water generation [1–4]. Because of its practical importance, several studies have been conducted over the last several decades on the evolution of the heat transport in helical coils. Ye et al. [5] investigated numerically the performance of a heat pump water heater (HPWH) with a wrap-around condenser coil. When compared to constant-pitch coils, the heat transfer coefficient and average COP of variable-pitch coils rose by 21.91% and 10.75%, respectively. Dai et al. [6]

conducted a numerical examination of heat pump water heater performance analysis (HPWH). When compared to a standard helical coil, the heat transfer coefficient of a variable-diameter coil was 19.06% higher and the average COP was 3.97% higher, according to the coupling simulation data. The heat pump water heater with submerged helically coiled tubes was numerically analyzed by Missaoui et al. [7]. The findings revealed that when the size of the water tank reduces, the overall heat transfer area expands, potentially increasing water velocity throughout the heating cycle. Zhang et al. [8] confirmed that modifying the coil sectional structure increases the heat transfer performance of refrigerant in condenser coils marginally. To examine transient thermal behavior, Kim et al. [9] created a dynamic model of a water heater system driven by a heat pump. The smaller the water tank, the greater the transient performance deterioration. Therefore, tank size should be adjusted in the design phase to reduce both heat loss and performance degradation. Pieiropontevetra

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et al. [10] modeled the behavior of a helical coil used as a condenser in a heat pump utilized for household hot water heating in a tank. Heat transfer in refrigeration and water storage were also addressed in the paper. A variable-diameter condenser coil positioned at the bottom of a water tank was proposed by Chuan-Chao et al. [11]. The temperature stratification was lowered as a result of his research, which gave additional coil design reference. Li et al. [12] introduced a linked model to simulate the transient performance of a heat pump water heater (HPWH) with a water tank during the heating-up process. A quasi-steady model to predict the behavior of the vapor-compression system and a CFD model to represent the transient flow and temperature fields in the water tank made up this linked model. Zhou et al. [13] investigated the geometric parameters of a wrap-around condenser for a water tank in an experimental and numerical study. The findings reveal that a small diameter copper tube (5 mm) has a greater heat transfer effect and offers cost savings. CFD analysis of single-phase flows inside helically coiled tubes was carried out by Jayakumar et al. [14]. The findings reveal that the diameter of the pitch circle, tube pitch, and pipe diameter all have a direct impact on heat transfer in a helically coiled tube. Norihiro et al. [15] conducted an experimental study on the heat transfer and pressure drop in coiled smooth tubes and coiled internally helical-grooved copper tubes with different coil diameters. The results indicated that the pressure drop in a coiled grooved tube was two to three times larger than that of a coiled smooth tube. Chingulpitak et al. [16] performed experiments as well as numerical investigations for the refrigerant R134a flowing in the helical capillary tubes. They concluded that the tube diameter presents a significant effect on mass flow rate. Dabas et al. [17] performed design optimization of a cylindrical shell and helical coil. They concluded that the heat transfer rate increases with the increase in refrigerant mass. Kaewon et al. [18] performed experiments to investigate the R134a flowing inside the helically coiled capillary tubes. They found that the capillary diameter presents a significant effect on the mass flow rate than the other variables. In the water heating process, Lau et al. [19] proposed a multi-functional heat pump system. The findings reveal that the heat source combinations have an impact on the system's heating capacity and coefficient of performance (COP). Kumar et al. [20] conducted an experimental investigation on the heat transfer coefficient of helically coiled tube heat exchanger with  $\text{Al}_2\text{O}_3$ /water nanofluid under laminar flow condition. The results indicated that the heat transfer coefficients increase with increasing inner Dean number and particle volume concentration. Shen et al. [21] developed and validated a quasi-steady-state heat pump water heater model with a stratified

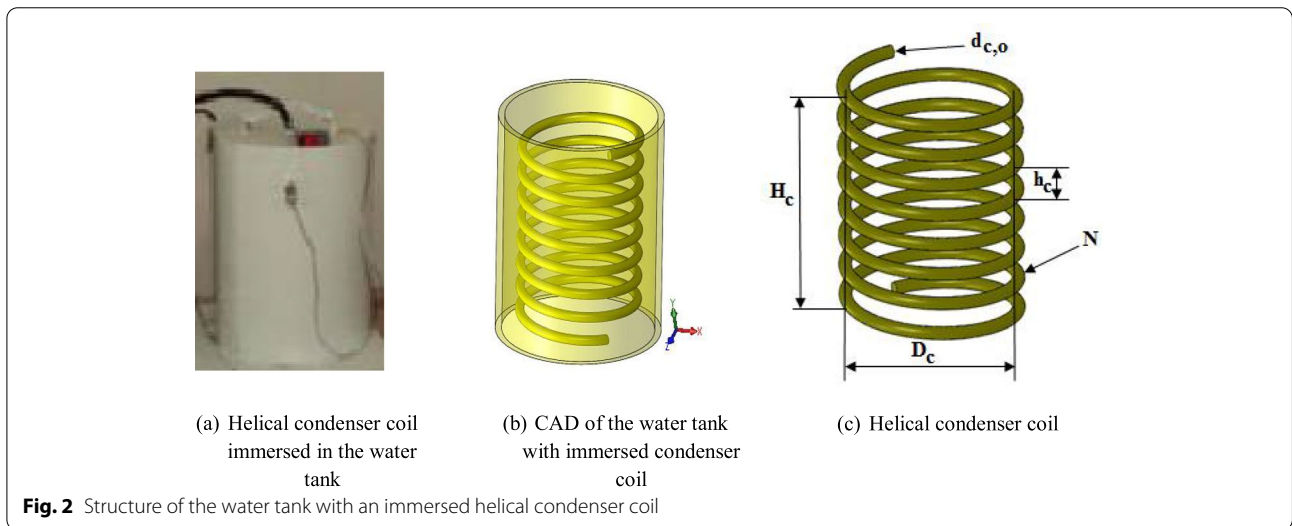
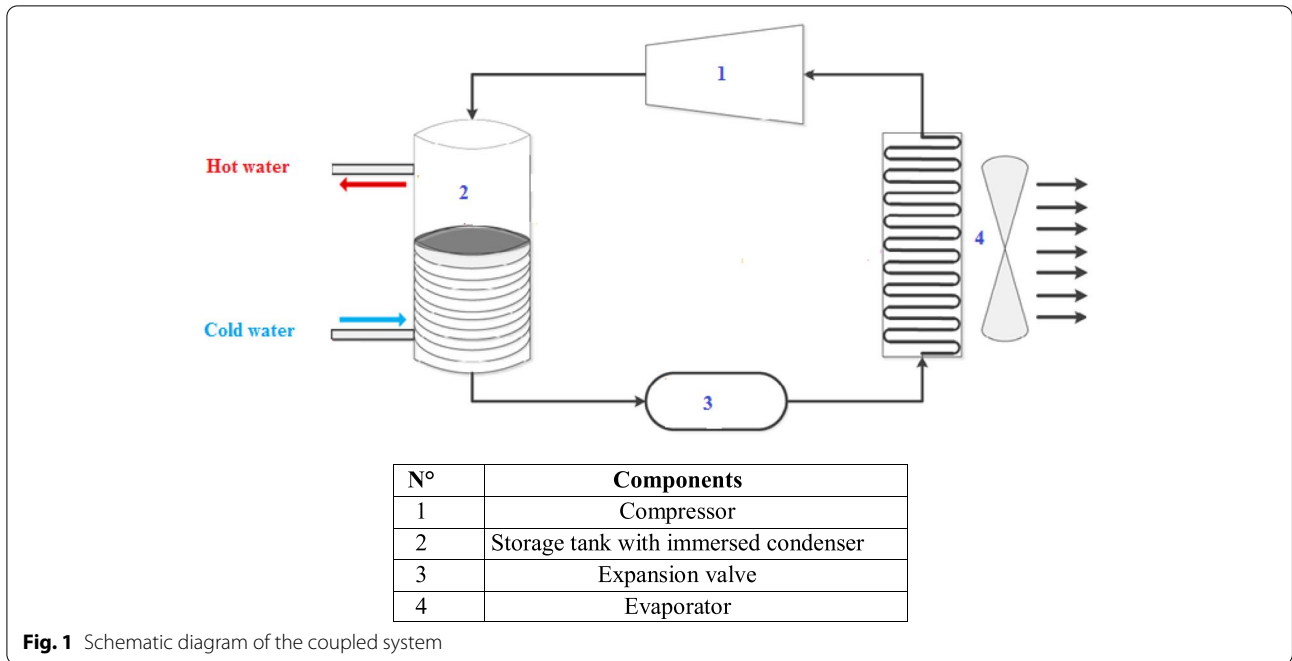
water tank and wrapped-tank condenser. The model was used to run parametric simulations to study the impact of different design aspects. A numerical and experimental study on an air source heat pump water heater was conducted by Wang et al. [22]. The findings revealed that the COP of the system may be enhanced. From these anterior studies, there are no entirely relevant citations in a literature search on the subject. Although significant research has been done on the heat transfer characteristics and COP in the water heating process, less has been done on the impact of alternative condenser coil structure designs on the refrigeration machine's operating performance for water heating. In the present manuscript, a household refrigerator with a water heating unit is designed for simultaneous domestic use, which can supply hot water during the whole year. This new design of the coupled system could reduce the thermal stratification, increase the water temperature, and reduce the total time required for domestic hot water production. Therefore, the present paper is an attempt to enhance the heat transfer coefficient of an immersed helical coil heat exchanger by increasing the pipe turns. The goal of this investigation is to study the temperature distribution and the heat transfer coefficient of the new coupled system and to compare the influence of the pipe turns of the helical condenser coil on the thermal characteristics of the domestic refrigerator for hot water production.

### 1.1 Experimental set-up

The vapor-compression cycle and water heating system employed in this study are primarily made up of an evaporator, compressor, and water tank with immersed condenser and thermostatic expansion valve. A domestic refrigerator with a water heating process uses an electrically powered vapor-compression cycle to transfer energy from the surrounding air to water in a tank, boosting the water's temperature. In order to study the effects of condenser coil geometry on the operating characteristics of the domestic refrigerator for water heating, a prototype of the coupling system was built based on the schematic diagram, as shown in Fig. 1.

The water tank with an immersed helical condenser coil, shown in Fig. 2, is the subject of this work. In the experimental study, the condenser coil was directly immersed in the storage tank. The configuration details of the helical coil are illustrated in Table 1. Moreover, the volume of the plastic water tank is equal to 50 L. The tank has a stainless steel wall with a 50-mm-thick polyurethane thermal insulation layer.

In the experiment, the storage tank is well insulated with glass wool and the heat losses to the surrounding ambient are neglected. During the whole heating time, the total amount of water in the storage tank is



**Table 1** Main structural parameters of the helical condenser coil

Structural parameter	Symbol	Value
Coil diameter (m)	$D_c$	0.320
Inner diameter of the tube (m)	$d_{c,i}$	0.004
Outer diameter of the tube (m)	$d_{c,o}$	0.006
Coil height (m)	$H_c$	0.38
Coil pitch (m)	$h_c$	0.03
Length (m)	$L_c$	13.074
Number of turns (-)	$N$	13

conserved. Furthermore, the refrigerant R134a is used as the working fluid. The constant room temperature is kept at 20 °C and a 50 L of cold water with an initial temperature of 20 °C is linked directly to the tap water pipe at the bottom of the tank, while the hot water with 51 °C is dispensed from the top.

### 1.2 System model development

In order to predict the operation performance of a domestic refrigerator coupled with a water heater and find the influence of geometric parameters on its operation, a numerical model has been formulated in this study.

### 1.2.1 Compressor model

Since the compression of refrigerant vapor is assumed to be a polytropic process, the mass flow rate of refrigerant  $m_r$  is obtained from:

$$m_r = \eta_v \cdot \frac{V_h}{3600 \cdot v_s} \quad (1)$$

where  $\eta_v$  is volumetric efficiency,  $V_h$  is the theoretical displacement volume, and  $v_s$  is the suction-specific volume.

The volumetric efficiency  $\eta_v$  is obtained from:

$$\eta_v = \frac{V_r}{V_{th}} \quad (2)$$

where  $V_r$  is the real measured aspired volume and  $V_{th}$  is the theoretical volume of the compressor chamber.

The electric power consumed by the compressor can be defined as:

$$W = \frac{m_r(h_{co,r,o} - h_{co,r,i})}{\eta_{co}} \quad (3)$$

where  $h_{co,r,o}$  is the enthalpy of the compressor outlet,  $h_{co,r,i}$  is the enthalpy of the compressor inlet, and  $\eta_{co}$  is the efficiency.

### 1.2.2 Thermostatic expansion valve model

The expansion is assumed isenthalpic. Therefore, the inlet enthalpy  $h_{val,in}$  and outlet enthalpy  $h_{val,out}$  value are equal and can be written as follows:

$$h_{val,in} = h_{val,out} \quad (4)$$

### 1.2.3 Evaporator model

In the evaporator model of the coupled system, the heat equation can be presented as follows:

$$Q_{ev} = m_r \cdot (h_{e,r,o} - h_{e,r,i}) \quad (5)$$

where  $m_r$  is the refrigerant mass flow rate and  $h_{e,r,i}$  and  $h_{e,r,o}$  are inlet and outlet refrigerant enthalpy in the evaporator, respectively.

The heat transferred from the evaporator to the air can be written as:

$$Q_{air} = m_a \cdot (h_{e,a,o} - h_{e,a,i}) \quad (6)$$

where  $m_a$  is the mass flow rate of the air,  $h_{e,a,o}$  is the enthalpy of the air at the outlet of the evaporator, and  $h_{e,a,i}$  is the enthalpy of the air at the inlet of the evaporator.

The total heat transfer can be expressed as:

$$Q_e = Q_a = U_e A_e \Delta T_e \quad (7)$$

where  $Q_a$  is the energy in the air side,  $U_e$  is the overall heat transfer coefficient,  $A_e$  is the heat transfer area, and  $\Delta T_e$  is the temperature difference.

### 1.2.4 Condenser model

The helical condenser coil is immersed in the water tank and the storage tank is well insulated with a 50-mm-thick polyurethane thermal insulation layer. Therefore, the heat loss from the water tank to ambient is negligible. The heat transfer in the helical condenser coil is calculated by:

$$Q_c = m_r \cdot (h_{c,r,i} - h_{c,r,o}) \quad (8)$$

where  $m_r$  is the mass flow rate of the refrigerant,  $h_{c,r,i}$  is the enthalpy of the refrigerant at the inlet, and  $h_{c,r,o}$  is the enthalpy of the refrigerant at the outlet of the compressor.

The heat transferred from the helical condenser coil to the water is given by the following equation:

$$Q_w = C_{p,w} \cdot m_w \cdot (T_{w,o} - T_{w,i}) \quad (9)$$

where  $C_{p,w}$  is the specific heat of the water,  $m_w$  is the mass flow rate of the water,  $T_{w,i}$  is the inlet of the water temperature, and  $T_{w,o}$  is the outlet of the water temperature.

The amount of heat transferred from the helical condenser coil to the water is determined by the following relation:

$$Q_c = Q_w = U_c \cdot A_c \cdot (T_{c,r} - T_w) \quad (10)$$

where  $Q_w$  is the energy in the water side,  $U_c$  is the overall heat transfer coefficient,  $A_c$  is the heat transfer area of the condenser,  $T_w$  is the average water temperature, and  $T_{c,r}$  is the refrigerant temperature.

The overall heat transfer coefficient can be determined from the following equation [23]:

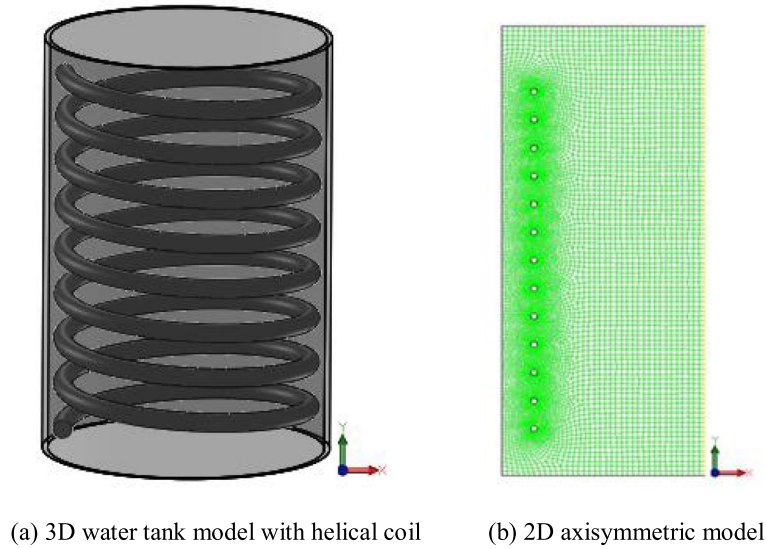
$$U_c = \frac{1}{\left[ \frac{1}{h_r} + d_i \cdot \frac{\ln(d_o/d_i)}{2 \cdot \lambda_c} + \frac{1}{h_w} \right]} \quad (11)$$

where  $h_r$  is the heat transfer coefficient of the refrigerant side,  $h_w$  is the heat transfer coefficient of the water side,  $\lambda_c$  is the thermal conductivity of the copper coil, and  $d_i$  and  $d_o$  are the inner and outer diameter of the tube.

The heat flux between the water and the immersed helical condenser coil is given as follows:

$$q = \frac{Q_w}{\pi \cdot d \cdot L} \quad (12)$$

where  $Q_w$  is the heat transferred to the water,  $d$  is the outer diameter of the tube, and  $L$  is the total length of the condenser coil.



**Fig. 3** Simplified model of a water tank with a helically coiled tube heat exchanger

**1.2.5 Water tank model**

In the numerical simulation of the coupled model, we have considered that:

- The 3D water tank was simplified as a 2-D axisymmetric model.
- During the heating process, the amount of the hot water is conserved as no water consumption during heating mode [24].
- The water was defined as laminar [25].
- The density was treated with the Boussinesq approximation [26].
- The tank bottom, tank top, and tank wall were assumed adiabatic [27, 28].

**1.2.6 Performance index**

The COP of the new coupled system is calculated from the thermal energy rejected by the helical condenser over the electrical energy consumed by the compressor and is expressed by the following equation:

$$COP = \frac{Q_c}{W_c} \tag{13}$$

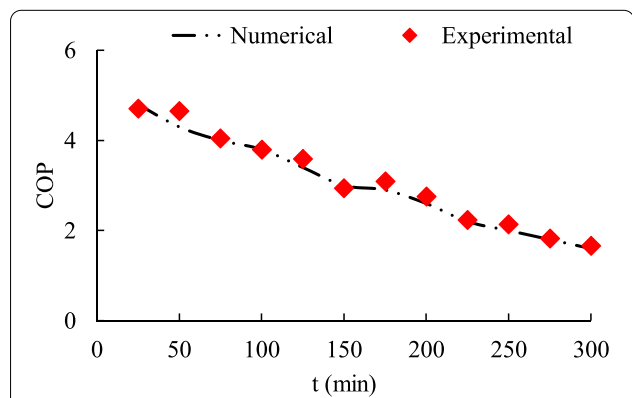
**1.3 Numerical simulation method**

The numerical model is developed by considering the software “ANSYS Fluent” to solve the Navier-Stokes equations using a finite volume discretization method. The commercial software Computational Fluid Dynamics (CFD) code ANSYS FLUENT 16.2 is utilized for the present study.

**1.3.1 Physical model**

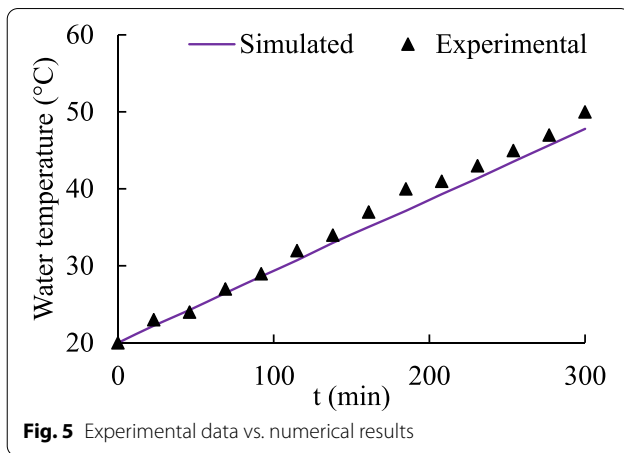
In ANSYS Fluent, a CFD model is created to simulate the heating process of the water in the storage tank from this new coupled technology. The water tank is roughly shaped like a cylinder, having a height of 0.48 m and a diameter of 0.38 m. This CFD model is planned to be a two-dimensional (2-D) axisymmetric domain, as shown in Fig. 3b, due to the symmetry in the geometry and physical boundary conditions. The cylindrical water tank centerline corresponds to the axisymmetric boundary. The condenser coil was assumed to be made up of several tiny circles in the symmetry plane in this CFD model.

The buoyancy-driven flow is considered to be laminar, and the SIMPLE pressure-velocity coupling scheme is chosen. The density difference caused by the temperature difference is the driving force for water flow in the tank. The Boussinesq approximation is a frequent and

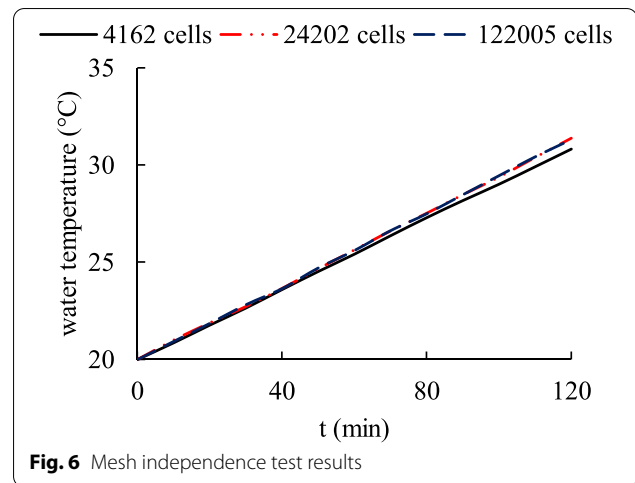


**Fig. 4** Numerical results for COP against experimental data





**Fig. 5** Experimental data vs. numerical results



**Fig. 6** Mesh independence test results

straightforward approach to think about water density change. Except for the buoyancy element in the momentum equation, this approximation treats density as a constant value in all solved equations. Furthermore, it is assumed that the density changes linearly with temperature.

$$(\rho - \rho_0)g \cong -\rho_0\beta(T - T_0)g \tag{14}$$

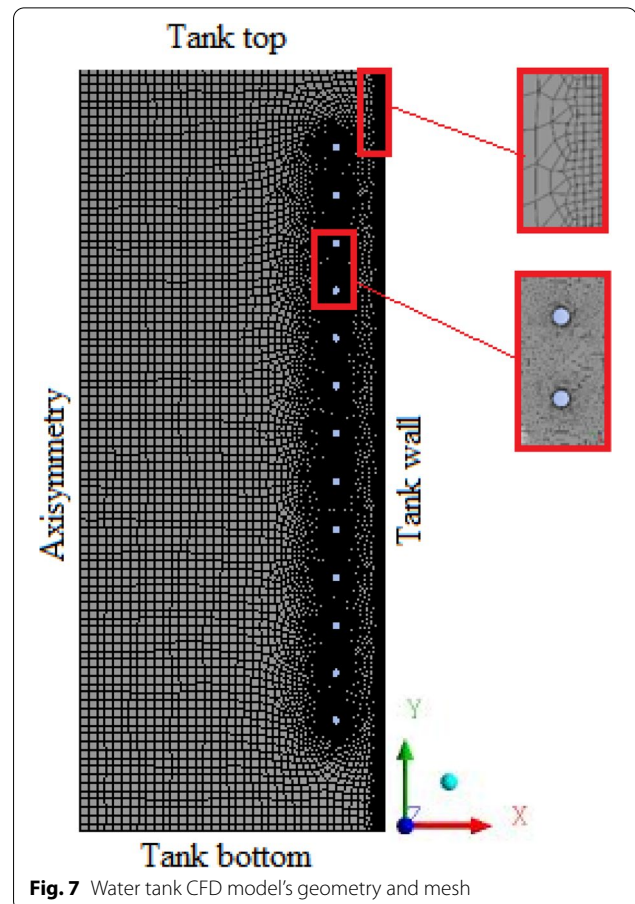
where  $\rho_0$  is the constant density of flow,  $T_0$  is the operating temperature, and  $\beta$  is the thermal expansion coefficient. The above equation is obtained by using the Boussinesq approximation  $\rho = \rho_0(1 - \beta\Delta T)$  to eliminate  $\rho$  from the buoyancy term. This approximation is accurate as long as changes in actual density are small.

### 1.3.2 Model validation

To validate the accuracy of the new coupled model, a household refrigerator with a water heating system including a thermostatic expansion valve, evaporator, compressor, and water tank with immersed condenser coil was used. The simulation findings were in good agreement with the experimental data, with a maximum variance of 1.6%, as shown in Fig. 4.

In order to guarantee the accuracy of numerical results, the experimental results of water temperature during

the whole heating time were compared to the numerical data obtained from the new coupled model containing both vapor-compression cycle and water heating unit, as shown in Fig. 5. According to the obtained results, it has been noted that the theoretical results were mostly lower than the experimental results. This is mainly due to the simplified two-dimensional axisymmetric model of



**Fig. 7** Water tank CFD model's geometry and mesh

**Table 2** Thermophysical properties

Water property	Symbol	Values	Units	Method
Density	$\rho$	994	kg/m <sup>3</sup>	Boussinesq
Specific heat	$C_p$	4183	J/(kg K)	Constant
Thermal conductivity	$\lambda$	0.6107	W/(m K)	Constant
Dynamic viscosity	$\mu$	0.0007196	kg/(m s)	Constant
Thermal expansion coefficient	$\beta$	0.000347	1/K	Constant

the water tank with an immersed condenser coil. In reality, in the proposed model, the helical condenser coil was assumed as many small circles in the vertical direction. However, the maximum difference between the experimental and the numerical results for the water temperature is within 7.67%.

**1.3.3 Governing equations**

This section presents the equations used to simulate the laminar flow of the water inside the cylindrical water tank and heat transfer from the heat pipe’s surface via convection.

The continuity equation is written as follows:

$$\frac{\partial u_x}{\partial x} + \frac{\partial u_y}{\partial y} = 0 \tag{15}$$

The momentum equations are written as follows:

$$\frac{\partial u_x}{\partial t} + u_x \frac{\partial u_x}{\partial x} + u_y \frac{\partial u_x}{\partial y} = -\frac{1}{\rho_{w0}} \frac{\partial P}{\partial x} + v_w \left[ \frac{\partial^2 u_x}{\partial x^2} + \frac{\partial^2 u_x}{\partial y^2} \right] \tag{16}$$

$$\frac{\partial u_y}{\partial t} + u_x \frac{\partial u_y}{\partial x} + u_y \frac{\partial u_y}{\partial y} = -\frac{1}{\rho_{w0}} \frac{\partial P}{\partial y} + v_w \left[ \frac{\partial^2 u_y}{\partial x^2} + \frac{\partial^2 u_y}{\partial y^2} \right] - g\beta(T_w - T_{w0}) \tag{17}$$

The energy equation is defined by:

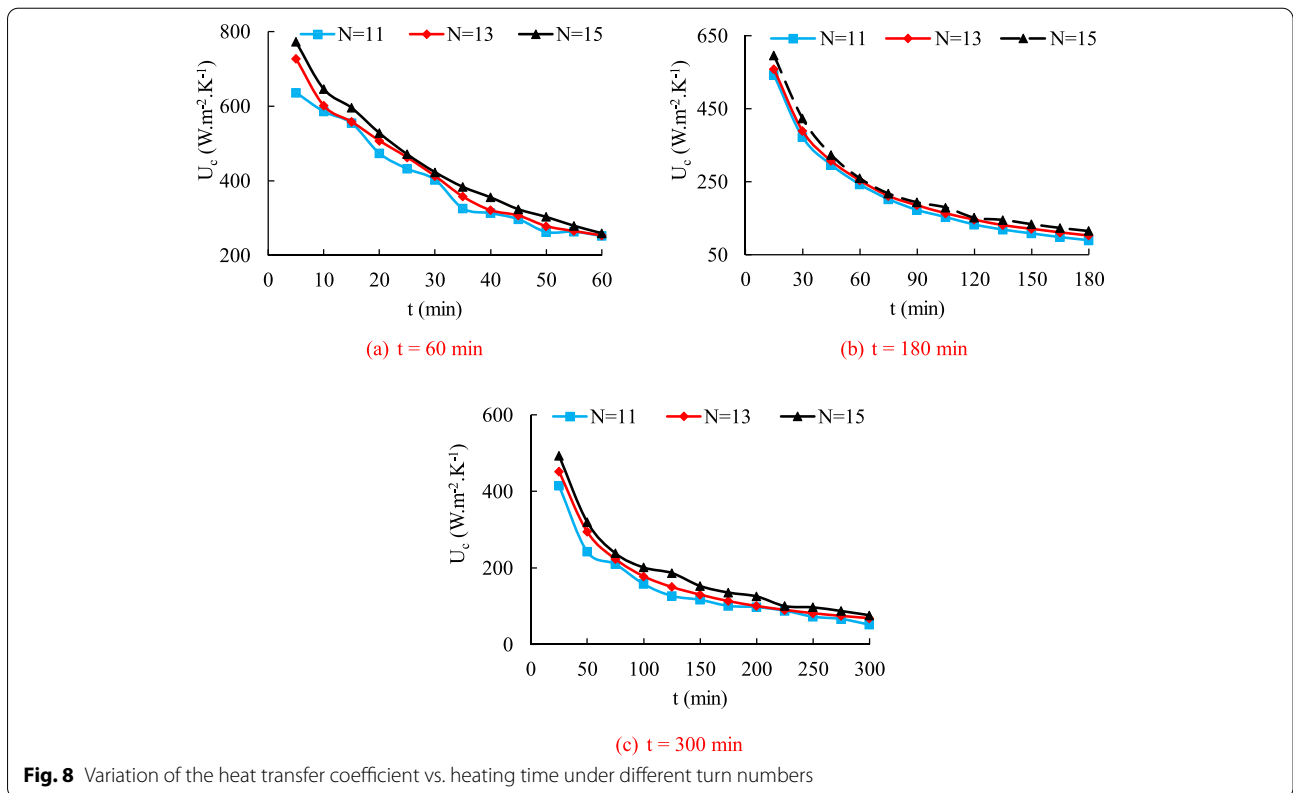
$$\frac{\partial T}{\partial t} + u_x \frac{\partial T}{\partial x} + u_y \frac{\partial T}{\partial y} = \lambda_w \left[ \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right] \tag{18}$$

**1.3.4 Boundary conditions**

All solid surfaces of the water tank are required to have no-slip and adiabatic walls. Experimental results are used to establish the heat flux of the condenser coil, which is then used as the initial boundary condition in the CFD model. For the present numerical study, the initial water temperature is set to be 20 °C and the time step is selected to be equal to 1 s. For pressure-velocity coupling, the SIMPLE method is utilized. The momentum and energy equations are treated using the second-order upwind method. The PRESTO technique is used to solve the pressure equation in laminar fluid flow. Based on the relationship between temperature and pressure obtained from the experiment, the thermophysical properties of the water are computed using Engineering Equation Solver (EES) software, as shown in Table 2.

**1.3.5 Mesh**

In our anterior published paper [29], we have developed a numerical model to study the water heating process in

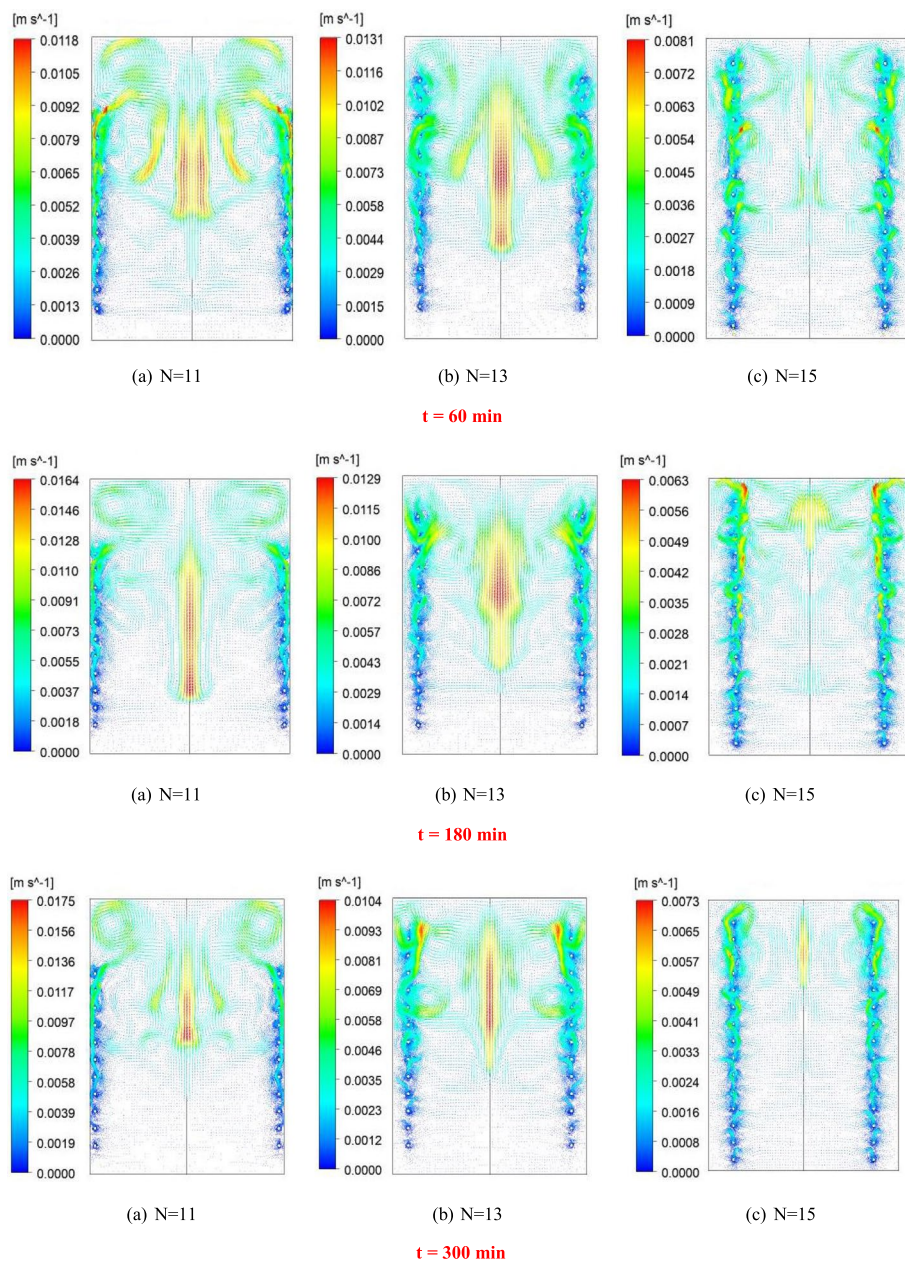


**Fig. 8** Variation of the heat transfer coefficient vs. heating time under different turn numbers

a cylindrical water storage tank. Particularly, we are interested on the mesh resolution effect on the flow simulation results. In fact, the size of the mesh has been changed and then the obtained results have been compared. Particularly, different cases corresponding to coarse and refined meshing have been studied. According to this study, the best result regarding precision and time is found for the mesh containing 24,202 cells, as shown in Fig. 6. This choice leads to a better result with regard to the precision

and the resolution time. In this new paper, we focus on the geometrical parameter effect and we have to change the pipe turns and keep constants all the other parameters. Based on our anterior study [30], we have chosen the time step size with a value equal to  $\Delta t = 1$  s.

As shown in Fig. 7, the grids are refined locally near the condenser coil for the calculation accuracy. The grids became increasingly looser as they moved away from the coil.



**Fig. 9** Velocity fields

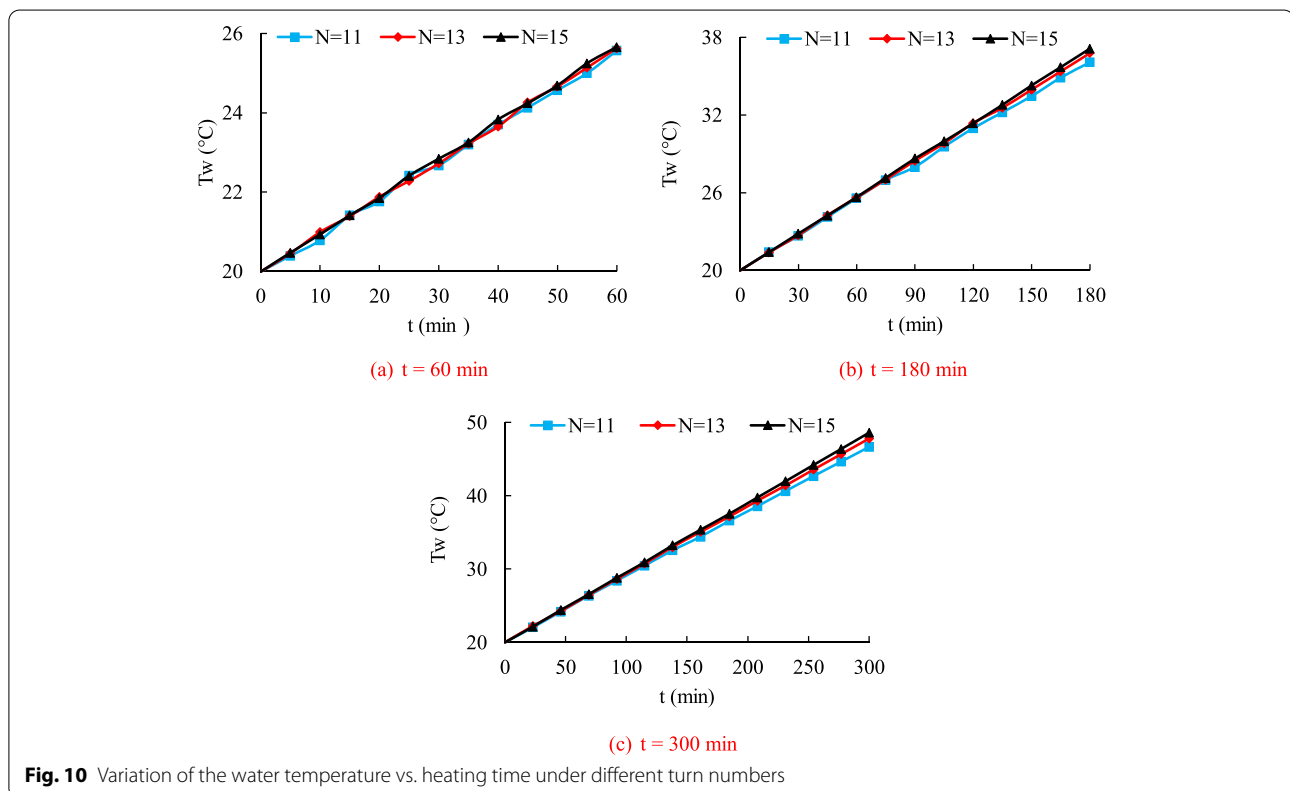


## 2 Results and discussion

The pipe turns are one of the elements of relevance in this study since they have a direct impact on the amount of heat transfer area accessible in the helical condenser coil, as well as the heat transfer coefficient. The effect of the pipe turns' number on the heat transfer coefficient in the helical condenser coil has been shown in Fig. 8. As it can be seen, the heat transfer coefficient decreases with the increase of the heating time. At  $t = 60$  min, the average heat transfer coefficient under the three structures is equal to  $399.93 \text{ W m}^{-2} \text{ K}^{-1}$ ,  $421.08 \text{ W m}^{-2} \text{ K}^{-1}$ , and  $444.98 \text{ W m}^{-2} \text{ K}^{-1}$  for  $N = 11$ ,  $N = 13$ , and  $N = 15$ , respectively. While examining these results, it has been noted that the heat transfer coefficient  $U_c$  of  $N = 15$  was the largest, which is 11.26% higher than that of  $N = 11$ . In fact, with the increase of the turn number, the heat transfer area increases which can increase the heat flux and improve the thermal performance of the helical condenser coil. At  $t = 180$  min, the average heat transfer coefficient  $U_c$  is equal to  $210.79 \text{ W m}^{-2} \text{ K}^{-1}$ ,  $223.32 \text{ W m}^{-2} \text{ K}^{-1}$ , and  $238.43 \text{ W m}^{-2} \text{ K}^{-1}$  for  $N = 11$ ,  $N = 13$ , and  $N = 15$ , respectively. Compared to the copper pipe turn with  $N = 11$  and  $N = 13$ , the heat transfer coefficient  $U_c$  is 13.11% and 6.76% higher. At the same time of heating, the heat transfer coefficient is improved with increasing the pipe turns, resulting in reducing the condensing temperature and the electrical power consumption of

the compressor. According to these results, it has been noted that the heat transfer coefficient  $U_c$  in the water tank is improved with the increase of the pipe turn number, which contributes to accelerating the water heating process.

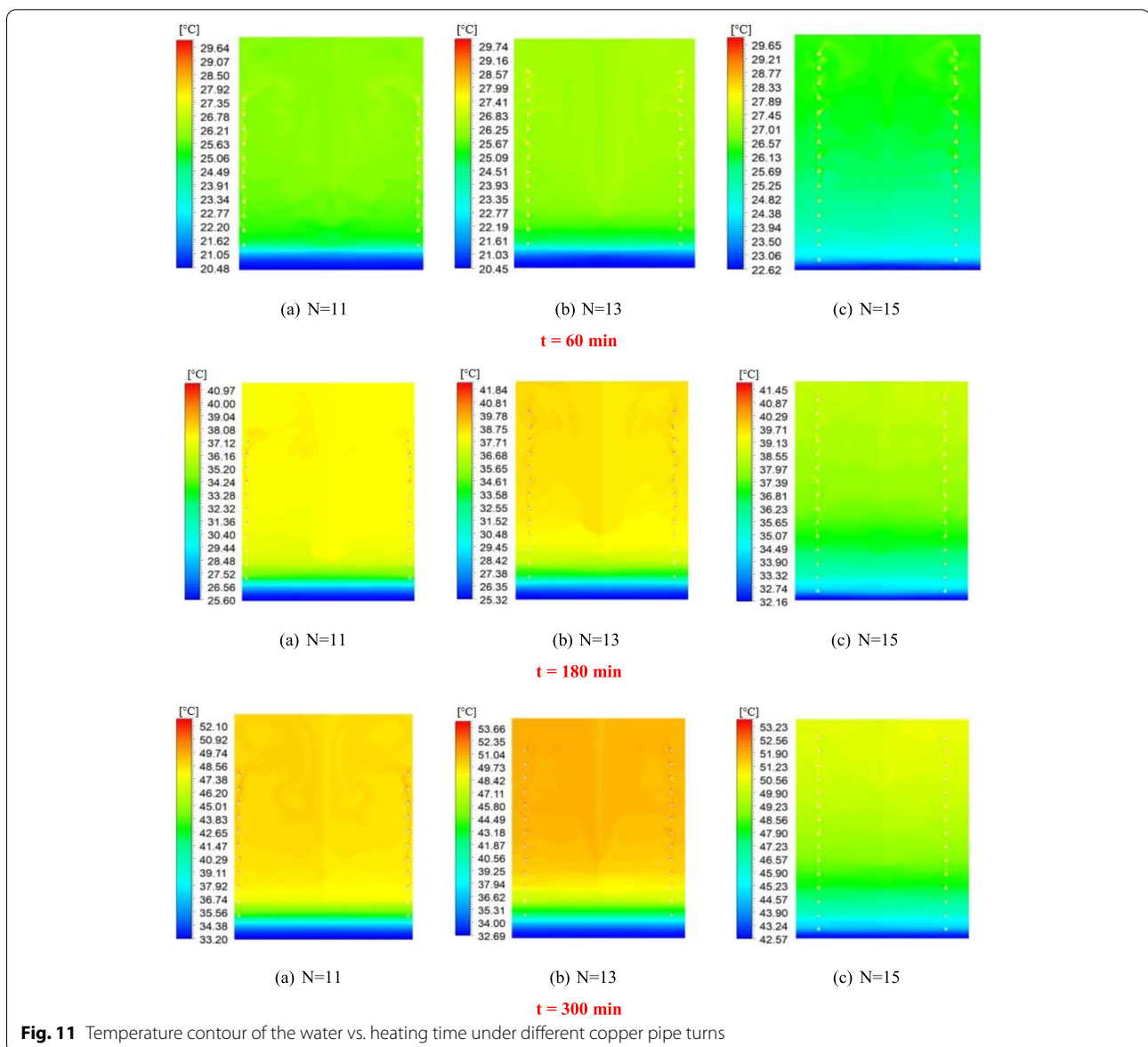
Figure 9 shows the velocity vector of the water in the storage tank at  $t = 60$  min,  $t = 180$  min, and  $t = 300$  min for three different turn numbers defined by  $N = 11$ ,  $N = 13$ , and  $N = 15$ . While examining these results, it can easily be noted that the helical pipe turn has a direct effect on the water velocity inside the tank. For  $t = 60$  min, the average water velocity is equal to  $V = 0.0060$  m/s,  $V = 0.0059$  m/s, and  $V = 0.0027$  m/s for the helical coil with  $N = 11$ ,  $N = 13$ , and  $N = 15$ , respectively. Compared to  $N = 13$  and  $N = 15$ , the water velocity of  $N = 11$  was 0.016% and 1.22% higher. Clearly, the water velocity gradient is getting smaller as the pipe turns increase. Indeed, the velocity field in the storage tank is affected by the turn number of the helical coil heat exchanger. In fact, with smaller pipe turns, the refrigerant flow is changed which leads to stronger water circulation in the lower part of the storage tank. At  $t = 180$  min, the average water velocity is equal to  $V = 0.0062$  m/s,  $V = 0.0049$  m/s, and  $V = 0.0022$  m/s for  $N = 11$ ,  $N = 13$ , and  $N = 15$ , respectively. Obviously, with the increase of the heating time from  $t = 60$  min to  $t = 180$  min, the effect of the turn number on the water velocity becomes stable. During the whole



heating process,  $t = 300$  min, the average water velocity is equal to  $V = 0.0048$  m/s,  $V = 0.0053$  m/s, and  $V = 0.0022$  m/s for  $N = 11$ ,  $N = 13$ , and  $N = 15$ , respectively. According to the obtained results, the pipe turns have no strong regularity on the water velocity vector.

In order to investigate the influence of different turn numbers on the thermal performance of helical condenser coil for water heating, the water temperature distribution is studied. The total lengths of the condenser coil are equal, while the initial pipe turn  $N$  is 11, 13, and 15, respectively. In order to analyze the obtained numerical results, Fig. 10 shows the water temperature variation in the storage tank under different pipe turn defined respectively by  $N = 11$ ,  $N = 13$ , and  $N = 15$  at  $t = 60$  min,  $t = 180$  min, and  $t = 300$  min. At  $t = 60$  min, the

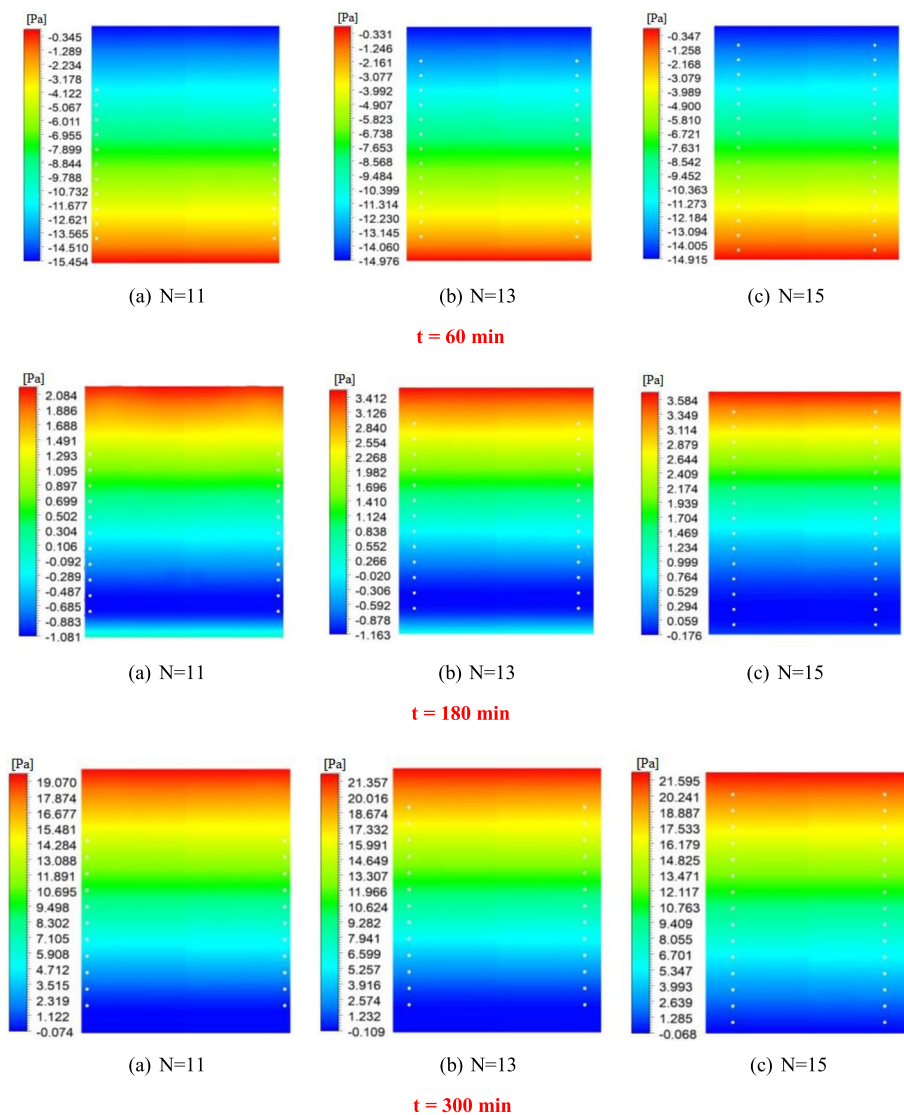
average water temperature under the three structures is equal to  $33.42$  °C,  $33.93$  °C, and  $34.26$  °C, respectively. While examining these results, it can easily be noted that the average water temperature of  $N = 15$  is the largest, which is 2.51% higher than that of  $N = 11$ . It is obvious that the increase of pipe turns can significantly reduce the water heating time. At  $t = 180$  min, the effects of the spiral pipe turn on the water temperature distribution are obvious. In fact, with increasing the pipe turn, the vertical length of the helical condenser coil increases the heat transfer area and improves the thermal performance of the helical condenser coil. Furthermore, the increase of the pipe turns can increase the heat flux of the helical coil heat exchanger, resulting in the quickly mixing process between different temperature layers in the storage



tank. With increasing the heating process to  $t = 300$  min, it is clear that the turn number has a direct effect on the water temperature distribution in the cylindrical tank. Indeed, it has been noted that the water temperature in the tank can be improved with increasing the pipe turn number.

Figure 11 shows the distribution of the water temperature versus heating time under three different pipe turn numbers defined respectively by  $N = 11$ ,  $N = 13$ , and  $N = 15$ . In order to compare the turns of copper pipe on the heating effect of the water in the storage tank, the total length of the condenser and the pipe spacing are kept constant. We have changed only the condenser coil diameter. Thus, with increasing the pipe turns, the condenser coil diameter decreases in order to maintain the same

length of the condenser. According to these results, it has been noted that the water temperature is on its maximum when the pipe turns with  $N = 15$  was used. With a heating time equal to  $t = 60$  min, the maximum temperature difference in the tank altitude direction is  $9.16\text{ }^\circ\text{C}$ ,  $9.29\text{ }^\circ\text{C}$ , and  $7.03\text{ }^\circ\text{C}$  for  $N = 11$ ,  $N = 13$ , and  $N = 15$ , respectively. In fact, when the pipe turns increase, the contact area between the copper pipe and the water increases leading to an increase of the heat flux and the water temperature. Furthermore, with the increase of the pipe turns, the heat transfer performance of the heat exchanger is increased and the natural convection between the water and the helical pipe became more severe. At  $t = 180$  min, the maximum temperature difference in the vertical direction of the tank under the three copper pipe turn is equal



**Fig. 12** Distribution of the static pressure vs. heating time under different copper pipe turns

to 15.37 °C, 16.52 °C, and 9.29 °C, respectively. Obviously, with increasing the heating time, the water temperature difference in the storage tank is decreased with increasing the pipe turns. In addition, with increasing the pipe turns, the water around the helical coil is quickly heated which leads to an increase of the buoyancy-driven flow. During the whole heating process,  $t = 300$  min, the maximum temperature difference in the vertical direction of the tank under the three copper pipe turn is equal to 18.9 °C, 20.97 °C, and 10.66 °C for  $N = 11$ ,  $N = 13$ , and  $N = 15$ , respectively. According to these results, it is clear that the temperature stratification decreases with increasing the spiral pipe turns of the condenser coil. The comparison between these results confirms that the condenser coil design has a direct effect on the water temperature distribution.

Figure 12 shows the distribution of the static pressure in the cylindrical water tank at  $t = 60$  min,  $t = 180$  min, and  $t = 300$  min under different pipe turns defined respectively by  $N = 11$ ,  $N = 13$ , and  $N = 15$ . At  $t = 60$  min, the distribution of the static pressure of the water for three different coil geometry defined respectively by  $N = 11$ ,  $N = 13$ , and  $N = 15$  show that the depression zone is located in the upper part of the storage tank. With increasing the heating time from  $t = 60$  min to  $t = 180$  min, the static pressure starts to increase gradually from the bottom tank to the top tank. This is mainly due to the water density variation. Indeed, the static pressure is found to be weak under condenser coil with  $N = 11$  and increases gradually with the increase of spiral pipe turns. This fact is due to the convective heat transfer coefficient increase with the increase of the spiral pipe turn from  $N = 11$  to  $N = 15$ . The comparison between these results confirms that the helical coil heat exchanger design has a direct effect on the static pressure distribution during utilizing the waste heat for domestic hot water production.

### 3 Conclusion

In this paper, numerical investigations were carried out to study the effect of the copper pipe turns on the thermal performance of a helical condenser coil for water heating. This result compares different designs of a helical condenser coil by the change of the pipe turn number defined by  $N = 11$ ,  $N = 13$ , and  $N = 15$  when keeping the other structural parameters constant. From the above analysis, it can be concluded that:

- The numerical results of the new coupled model were subsequently validated by experimental data. In addition, the water temperature distribution and the coefficient of performance show good agreement with a maximum deviation of about 7.67% and 1.6%, respectively. The proposed model can be used to predict the

heat transfer, water velocity, and water temperature from the similarly dimensioned heat exchanger and can be applied to design and improve an existing heat exchanger for the refrigeration system.

- With appropriately increasing the pipe turns, the faster the water in the storage tank will be heated. When the pipe turns of the helical coil heat exchanger are increased, the heat transfer performance is shown as the best with excellent temperature uniformity of the water.
- The helical condenser coil design has a direct effect on the operating characteristics of the domestic refrigerator for hot water production.

These research results will be considered in the development of the future domestic refrigerator water heaters with internal condensers.

#### Nomenclature

$C_p$ : Specific heat (J/kg K);  $t$ : Time (s);  $T$ : Temperature (°C);  $q$ : Heat flux (W/m<sup>2</sup>);  $m$ : Mass flow rate (kg/s);  $h_c$ : Coil pitch (mm);  $h$ : Refrigerant enthalpy (J/kg);  $\eta_v$ : Volumetric efficiency(%);  $A$ : Area (m<sup>2</sup>);  $\Delta T$ : Temperature difference (°C);  $d$ : Diameter (mm);  $h_i$ : Enthalpy of the refrigerant at the inlet of the expansion device (J/kg);  $h_o$ : Enthalpy of the refrigerant at the outlet of the expansion device (J/kg);  $V_t$ : Theoretical displacement volume (m<sup>3</sup>);  $v_s$ : Suction-specific volume (m<sup>3</sup>/kg);  $W$ : Power of the compressor (W);  $\eta_{co}$ : Total efficiency of the compressor;  $Q$ : Heat energy (W);  $U$ : Overall heat transfer coefficient (W/m<sup>2</sup>K);  $A$ : Heat transfer area (m<sup>2</sup>);  $\Delta T$ : Temperature difference between air and refrigerant (K).

#### Greek symbols

$\lambda$ : Thermal conductivity (W/m K);  $\rho_w, \rho$ : Constant density of water (kg/m<sup>3</sup>);  $\rho$ : Variable density of water (kg/m<sup>3</sup>);  $\mu$ : Viscosity (kg/m s);  $\beta$ : Thermal expansion coefficient (1/K);  $\nu$ : Kinematic viscosity (m<sup>2</sup>/s);  $\alpha$ : Heat transfer coefficient (W/m<sup>2</sup> K).

#### Subscripts

w: Water; c: Condenser; e: Evaporator; co: Compressor; v: Expansion valve; i: Inlet; o: Outlet; r: Refrigerant; a: Air.

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#### Authors' contributions

Study conception and design: S. Missaoui, Z. Driss; data collection: S. Missaoui; conceived and designed the analysis: R. Ben Slama; analysis and interpretation of results: S. Missaoui; draft manuscript preparation: S. Missaoui, Z. Driss; other contribution: B. Chaouachi. All authors reviewed the results and approved the final version of the manuscript.

#### Declarations

#### Competing interests

The authors declare that they have no competing interests.

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