TEST Engineering & Management

*March-April 2020*
*ISSN: 0193-4120 Page No. 10985 - 10997*

# Study to the Thermal Performance of Serpentine-Tube Solar Collector at Two Different Tilt Angles

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***Article Info***
***Volume 83***
***Page Number: 10985 - 10997***
***Publication Issue:***
***March - April 2020***

***Article History***
***Article Received**: 24 July 2019*
***Revised**: 12 September 2019*
***Accepted**: 15 February 2020*
***Publication**: 13 April 2020*

***Abstract:***
The global demand for electricity is increasing, and the resources depletion of fossil fuel, have necessitated an epitome shift in the development of efficient technologies and sustainable energy. Air-conditioning represents the main reason in the increasing of demand for electricity particularly in summer. The use of renewable energy is a viable solution to mitigate this problem. This work studies the thermal performance of a serpentine-tube solar collector for two different design at two different tilt angles to employ the proposed design in the improvement of air conditioning system. In the current study, the 3-D model design has been used with two different lengths. The first has a tube of 1.65m length with two turns. The second has a tube of 2.35m length with four turns. ANSYS Fluent 16.1 is used in the modeling and simulating of the two design. The working fluid in the suggested model is R22, where the refrigerant is directly exposed to the solar radiation. Computational fluid dynamics (CFD) simulation employs to predict the refrigerant temperature at the outlet of serpentine-tube (ST). The results show the increment in temperature due to the intensity of solar radiation, at angle 13o to the first proposed design, is about 15.42% and for the second design is 24.99%, while at angle 30o it was boosted by 19.34% for the second design and 10.86% for the first model.
***Keywords:*** *thermal performance, CFD simulation,solar collector, serpentine-tube, tilt angle*.

Nomenclature subscripts
A cross sectionarea ($m^2$) $ga$ gain
$C_p$ specific heat (j/kg.K) i inside
Do outside diameter (m) in tube inlet
Di inside diameter (m) o outside
I solar irradiance ($W/m^2$) out tube outlet
k thermal conductivity ($W/m^2$. K) T total
L tube length (m) p constant pressure
$\dot{m}$ mass flow rate (kg/s) t turbulent
N number of turns
$P$ pressure (pa) Abbreviations
Q heat gain (J/kg) CFD computational fluid dynamics
T temperature (K) HRF heat removal factor
u, v, w velocity (m/s) FPSC flat-plate solar collector
Greek symbols SC solar collector

TEST Engineering & Management
March-April 2020
ISSN: 0193-4120 Page No. 10985 - 10997

| | | | |
|---|---|---|---|
| ρ | density (kg/m$^3$) | SFPSC | serpentine tube flat plate solar collector |
| ε | turbulent dissipation rate (m$^2$/s$^3$) | SSC | serpentine tube solar collector |
| μ | viscosity (kg/m. s) | ST | serpentine tube |
| $\mu_t$ | turbulent viscosity (kg/m.s) | | |
| $C_{\varepsilon 1}, C_{\varepsilon 2}, C_{\mu}$ | constant | | |
| η | serpentine-tube solar collector efficiency | | |
| $\sigma_k$,$\sigma_\varepsilon$ | constant | | |

## 1. INTRODUCTION

In recent decades, the high demand for fossil fuels represents one of the main challenges in power generation. The expected increase in the average of world energy consumption in 2035 is approximately 49% from 2007. So, all countries should be paying attention to the use of energy sources, and the use of fossil fuel have to reduce as possible and move to resources of renewable energy like solar energy, wind energy, hydropower energy, etc.

The solar collector (SC) is the device used for converting the received power from the sun radiation into thermal energy via a transport medium, hence the thermal effects increases. The solar radiation absorbs and collects by (SC) then transforms into thermal energy in the form of an increase in temperature which heats the flowing fluid through (SC) [1]. The mostly common type of (SC)s isflat-plate solar collector (FPSC) that use for heating of residential water and space as well as in the industrial application. The parallel tubes type of (FPSC)s is the most available currently, it also known as traditional (FPSC) [2]. These types associated with a high flow rate that need high cost of operating. Moreover,the traditional FPSC was in working for a long time with no considerable changes in operational principles, shape, and design [3]. The arrangement of solar collector represents an important factor in the determining of thermal performance.In the systems of low-flow, due to the early starting of turbulent flow that encourages the application of heat transfer, the performance of the serpentine solar collector (SSC) can be better than, the conventional, parallel tube solar collector[4]. For the solar collector of the same: surface area, the spacing between tubes and the tube diameter, the serpentine collector is better than conventional collector[5].Theperformance difference between the FPSC and SFPSC is mainly owing to the coefficient of internal heat transfer. However, SFPSC was ignored previously due to the large pumping required at the high flow rates.Due to the design geometry of solar collector (SC), the heat removal factor (HRF) and the collector efficiency factor (CEF) are difficultly expressed. If a heat break takes place in the middle between the serpentine tubes, the collector can be studied as Thermal performance of a serpentine absorber plate a conventional collector. If the break does not provide, the decrease in performance is expected and an increase of complex analysis is required [6]. The determination of (HRF) for (SSC) is mostly complicated than that for traditional (FPSC) [7]. The analysis of the conventional collector is unlike (SSC), the heat transfer between the tubes exists in (SSC). Many published works have analytical solutions that performed to address the governing differential equations of heat transfer in absorber serpentine tubes.The (HRF) analytical solution of serpentine-tube attached to a two parts plate was discovered. Where the two segments analytical solution was feasible for any number of sectors with a minor error [8]. Then discovered that this result leads to more errors than predicted via obtained analytical solutions to a given set of parameters for N = 3 and 4[9].The solution assumes, the plate heat

March-April 2020
ISSN: 0193-4120 Page No. 10985 - 10997

transfer, one-dimensional and ignores heat transfer through a U-bend portion. The number of turns variation to (SSC) was also analyzed. when the number of turns increases, the value of (HFR) approximate to the values at the turn numbers N = 1 [10]. At the situation of analyzing the longest straight collector with no inflection, the model will be the same as (FPC) model, except the coefficient of internal heat transfer will be different[5]. There are a few publications that reportedan experimental resultto the flow of serpentine (SC)s. An experimental study conducted for two (SSC)s that have the same geometrical shape.In the first SSC, the serpentine tube welded to the absorption plate to the whole collector, but in the second SSC, the bent tube was not connected thermally to the absorber plate. The two collectors put underneath the sun radiation and the performance had been measured under meteorological matching conditions. In the experiment, the efficiency of the collector that welded to the absorption plate is about 2 to 2.5% or upper[11]. Decreasing the distance between the tubes was the most efficient method to enhance the uniformity of temperature distribution, and the use of absorbents with better thermal properties, that can partially ameliorate temperature distribution. Nevertheless, the increment in fluid speed at tube inlet has a small effect on the creation of a uniform temperature field, although it can minimize the temperature of the plate. The temperature affected via the row configuration of the tube, and no uniform temperature range can be obtained with further bends[12].[13]examined numerically the effect of the parameters on the temperature, which is providing the service of the hot-water via immersing a heat exchanger tube, inside a flat plate combined collector solar water heater by CFD analysis. Hot water is drawn to the service indirectly, via a submerged heat exchanger. To intensify the process of heat transfer, the storage water is moved by recycling it via a pump, which continues only when the service water flows inside the heat exchanger. Three main factors, which affect performance, are improved: the location of the heat exchanger relative to the walls of the tank, the length of the heat exchanger, and the diameter of the pipe. All these factors are explored to increase the temperature of the service water outlet. The optimal configuration adjustment time is also calculated. The optimal location was found to lay the heat exchanger in contact with the front and back walls of the tank, with an optimum inner tube diameter of 16 mm, while an acceptable heat exchanger length was found to be about 21.5 m. [14]conducted a CFD and a finite element method (FEM) analysis on a FPSC for designing and fabricating a low-cost solar water heater to rustic areas of Oman. The upgraded prototype provided 100 liters of hot water with an average temperature of 50 ° C at 6 p.m., which are accepted conditions for smaller families in Oman. also, an experimental study was carried, where many experiments are conducted at different conditions for validating the suggested design and achieved satisfying results. the outlet temperature of the water is 83 ºC, which is gained in the afternoon period, and this temperature would be ideal for the home applications in Oman.

All the previous researches used the water as working fluid in this type of the solar collector, while in this study the thermal performance of the serpentine tub exposes directly to solar irradiation, via using R22 as a working fluidis investigated.Also the effect of the tilt angle one the amount of the solar irradiation, which is received by the serpentine-tube. The effect of a number of turns and length of the tube is also investigated. ANSYS 16.1 employs to design and simulate both design at two tilting angle. The solar irradiation is measured under Hillah city environment conditions. The proposed designs are suggested to employin solar cooling applications.

March-April 2020
ISSN: 0193-4120 Page No. 10985 - 10997

## 2. MATERIALS AND METHODS

The software simulationANSYS FLUENT version 16.1 utilizesin geometrical modelingand simulation for both proposed designs, then the design has been transformed to mesh, and the design parameters of geometry parts are selected to the meshing section. In order to get better analysis for thermal and flow, the fine mesh size was chosen to further evaluation for the flow of many elements and updated for differentiating the input.The working fluid that pass through the tube is Freon-22 (R22). The 3D structure of geometrical model for the proposed two designs in this work are shown in Figure1. (a)&(b)all dimensions are illustrated in these figures. Both designs are modeled via ANSYS workbench 16.1.The hexahedral mesh is used for both designs as shown in Figure (2).To implement mesh independency via monitoring convergent history of predicted temperature at the outlet, for various refinement levels of mesh at heat flux (780W/m$^2$) as depicted in Figure (3.). It shows the outlet temperature difference is less than 0.12% at the employment of elements higher than 2917668 for the tube.

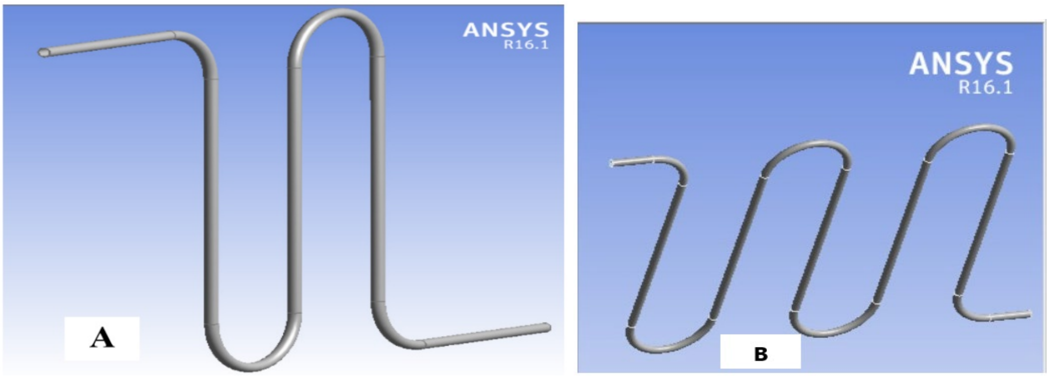


Figure 1. geometry of the serpentine tube (a)the first design. (b) the second model.

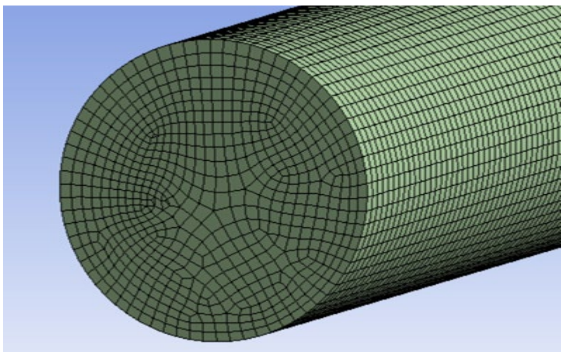

Figure 2. Computational grid of the system

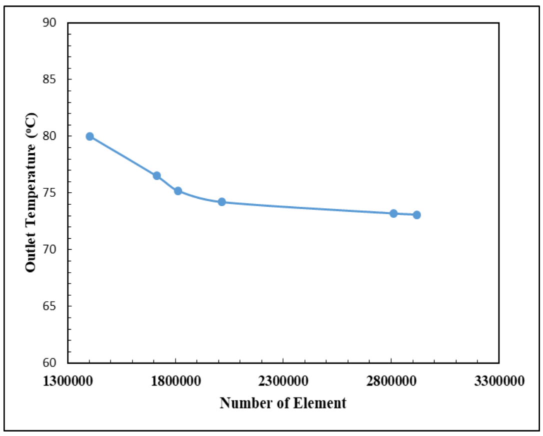


Figure 3. accuracy of Prediction Outlet Temperature with Number of Element

### 2.1 governing equations & boundary condition

The governing equations are solved according to the following assumptions: Steady state, single phase and turbulent flow, three dimensional, incompressible flow, the gravity effect is in considerable along the vertical axis by determining the value of the negative acceleration, and heat generation is neglected.

Continuity equation**[15]**:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \qquad (1)$$

March-April 2020
ISSN: 0193-4120 Page No. 10985 - 10997

Momentum equation:

The momentum equation components in Cartesian coordinates are **[15]**:

in $x$-direction:

$$\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=-\frac{1}{\rho}\frac{\partial P}{\partial x}+\nu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right) \quad (2)$$

Energy equation:

The energy equation in Cartesian coordinates can be written as **[15]**:

$$\rho\, C_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=\left(u\frac{\partial P}{\partial x}+v\frac{\partial P}{\partial y}+w\frac{\partial P}{\partial z}\right)+k_s\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \quad (3)$$

Turbulence Model (The Standard $k$-$\varepsilon$ Model):

This model is the most commonly used in CFD for simulating the characteristics of mean flow for the conditions of turbulent flow. It is a model of two equations that presents a comprehensive description of turbulence by partial differential equations of two transported variables. The first variable employs in determining the turbulence energy, which is termed turbulent kinetic energy (k). The second variable is the turbulent dissipation (ε), which utilizes in determining the turbulent rate of dissipation for the turbulent kinetic energy. The transport equations in **[15]**are used for determining (k & ε) the turbulence kinetic energy and rate of dissipation respectively.The turbulent viscosity $(\mu_t)$ is calculated via combining $(\varepsilon\&k)$ [16]**.**

$$\mu_t=\rho C_\mu\frac{k^2}{\varepsilon} \quad (4)$$

where $(C_\mu)$ is a constant, which is selected as (0.09) in ANSYS.Thetemperature and pressure,design values of refrigerant,are experimentally measured at the compressor outlet for an air-condition of capacity 3.5 KW. Table (2) presents the material properties of (ST) the values of copper properties considers according to [16]. Physical properties like thermal conductivity k, specific heat $C_p$, and refrigerant R22 viscosity μ are assumed as a polynomial function of temperature are shown in table (1).

Table (1) refrigerant (R22) correlation of the physical properties

| Property | Correlation | Ref. |
|---|---|---|
| Viscosity | $\mu(T) = -0.7369597 + 4.9394676E\text{-}02T^1 - 1.2112332E\text{-}05T^2$ | [17] |
| Conductivity | $k(T) = -4.0615383E\text{-}04 + 1.80841025E\text{-}05T^1 + 6.18803419E\text{-}08T^2$ | [17] |
| Specific heat | $C_p(T) = 335.591265 + 0.654343T^1 + 4.97567E\text{-}04T^2 - 2.93811E\text{-}06T^3 + 3.65937E\text{-}09T^4 - 1.95317E\text{-}12T^5 + 3.902384E\text{-}016T^6$ | [17] |

March-April 2020
ISSN: 0193-4120 Page No. 10985 - 10997

Table (2) copper properties

| Parameter | Symbol | Value |
|---|---|---|
| Density (kg/m3) | ρ | 8978 |
| Specific heat (kj/kg-k) | Cp | 381 |
| Thermal conductivity (w/m-k) | k | 387.6 |

The mass flow inlet boundary condition is used for the inlet of the serpentine-tube (ST) for all the two designs. The values of inlet refrigerant (R22) mass flow is considered constant and measured experimentally. The boundary conditions parameters at tube inlet, at tube wall and also the two design dimensions are shown in Table (2).The wall of the tube is subjected to the constant heat flux of solar radiation. The value of heat flux represents the average of collected solar radiation [18] in May 2019. Heat flux used, for both experimental and CFD testing, as solar insolation for simulating the distribution of temperature by the flow of R22 via tube, the heat flux is represented in the following equation[16].

Table (3)boundary condition at inlet and the wall of the tube

| Parameter | The first model | | The second model | |
|---|---|---|---|---|
| Temperature inlet ($T_{inlet}$) | 355(K) | 355(K) | 355(K) | 355(K) |
| Solar irradiance (I) | 856.6(W/m$^2$) | 576.6(W/m$^2$) | 856.6(W/m$^2$) | 576.6(W/m$^2$) |
| Mass flowrate ($\dot{m}$) | 0.0034(kg/s) | 0.0034(kg/s) | 0.0034(kg/s) | 0.0034(kg/s) |
| Tube length (L ) | 1.65 (m) | 1.65(m) | 2.35(m) | 2.35(m) |
| Number of turns (N) | 2 | 2 | 4 | 4 |
| Outside diameter (Do) | 9.525(mm) | 9.525(mm) | 9.525(mm) | 9.525(mm) |
| Inside diameter (Di) | 8(mm) | 8(mm) | 8(mm) | 8(mm) |
| Tilt angle | 13$^o$ | 30$^o$ | 13$^o$ | 30$^o$ |

**2.2 Method of Solution**

A computational code has been developed, according to the finite volume method, using ANSYS CFD code. The steady pressure-based solver used for solving the governing equations. The second-order upwind scheme employs for discretizing the equations of momentum and energy. Semi Implicit Pressure Linked Equation (SIMPLE), the double-precision algorithm utilizes to combine pressure with velocity. The numerical computation considered converged when the total residual massand sum of residual energy is lesser than $10^{-4}$ as shown in table(4).

Table (4) residual error for the tested case

| Equations | Continuity | X-Velocity | Y-Velocity | Z-Velocity | Energy | K | E |
|---|---|---|---|---|---|---|---|
| **Residual Error** | $10^{-5}$ | $10^{-4}$ | $10^{-4}$ | $10^{-4}$ | $10^{-6}$ | $10^{-4}$ | $10^{-4}$ |

### 2.3 Parameters Calculations

The following equation used in the calculation of (SFPSC) coefficient of performance [18]

$$\eta = \frac{Q_{ga}}{A_C I_T} \dots (8)$$

$$Q_{ga} = \dot{m} \times Cp \times (T_{out} - T_{in}) \dots (9)$$

$Q_{ga}$ : heat gained from the serpentine tube solar collector

$\eta$ : serpentine-tube solar collector efficiency

$T_{out}$: temperature at ST outlet

$T_{in}$: temperature at ST inlet

## 3. RESULTS DISCUSSION

The solution is found under the case of computational fluid dynamics, and the required parameters presented in post-processing. The temperature distribution of flow inside the tube is predicted thereby using CFD simulation to estimate the efficiency factor of STC in addition to the other parameters that clarify the performance of collector.Eventually, the resultsacquire by simulation analysis of computational fluid dynamics for both designs is compared according to two different angles and two different lengths in addition to the number of turns.

### 3.1. The Contours of the Simulation

Figure (4) shows the temperature contours of refrigerant-22 for the first design at angle 13$^o$, the diagram illustrates that the refrigerant temperature atthe inlet ofserpentine tube (ST) is($T_{in}$) 355K,which is recorded at the air condition compressor (the air condition is a split unit with capacity of one ton).when the flow passed through the pipe, the temperature of the fluid at the outlet of the tube increase to become 364.352K. while the temperature difference between inlet and outlet of the serpentine tube at angle 30$^o$ for the same designis 4.432$^o$C as shown in figure (6). Figure (7) shows the outlet temperature for second design, it is increased with 15.151$^o$C at tilt angle 13$^o$ while at angle 30$^o$ the refrigerant entered with 355K and exit with 362.894k as shown in Figure (8).

The results presents thatthe thermal performance is directly proportional: to the|serpentine-tube length and number of turnsunder the same boundary condition, and tothe value of the absorbed solar irradiation by serpentine-tube.Thus the performance of the serpentine-tube collector increases when the temperature difference between inlet and outlet increase. The proposed system has a better heat transfer, andthe obtained numerical results is obviously demonstrating that the suggested design provides a considerable enhancement in performance.

Figure (8) and (9) display contours density of ST, from the diagram, the gradually decreases in density along the tube can be seen, which belongs

March-April 2020
ISSN: 0193-4120 Page No. 10985 - 10997

to the charactristic of R-22 that fall under the properties of compressible fluid, whereas the temperature and the pressure increase the density of the fluid decrease.

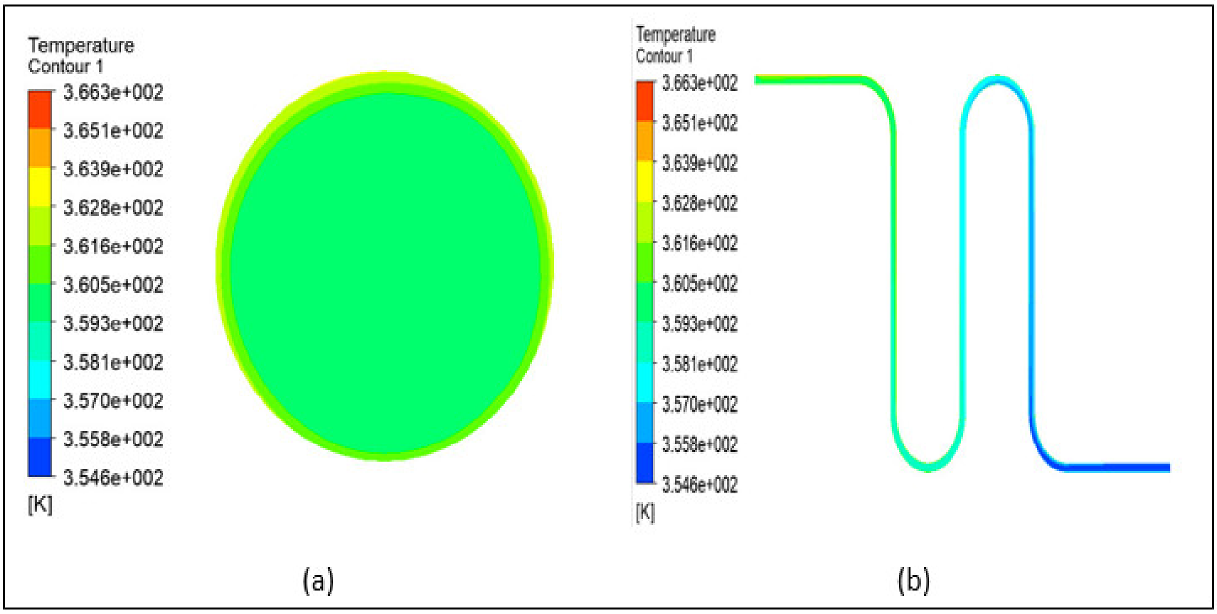


Figure 4. Serpentine tube temperature contour |of the first model at tilt angle 13°(a) outlet temperature (b) temperature distribution

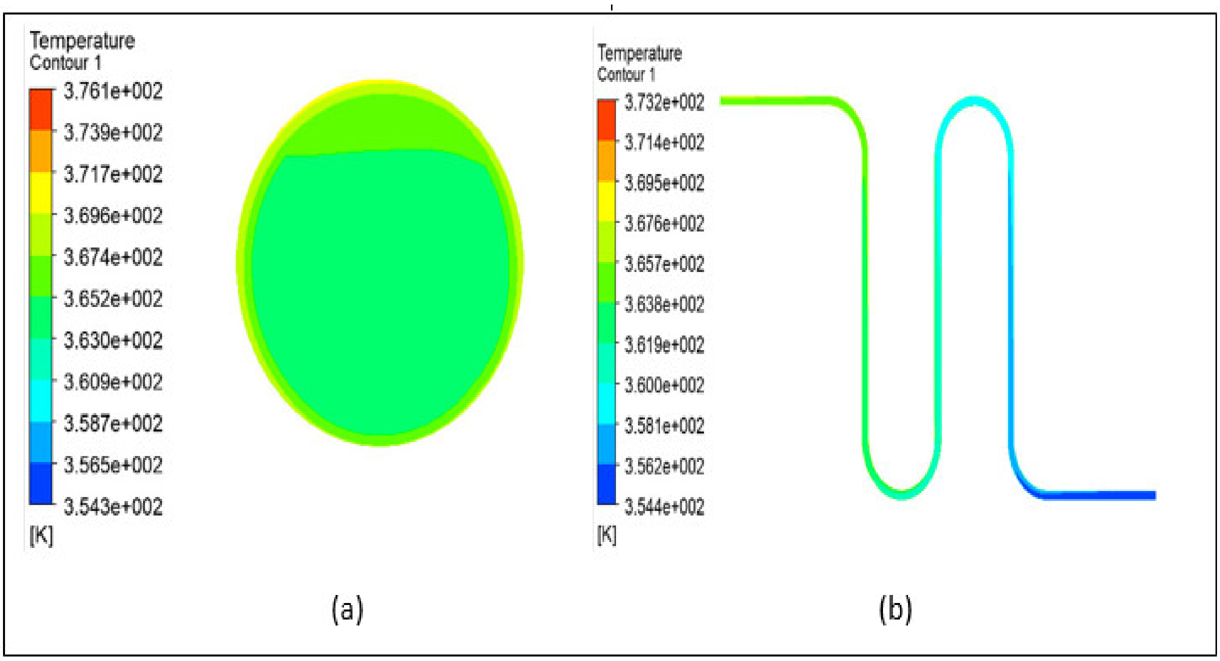


Figure 5. Serpentine tube temperature contour of the first design at tilt angle 30° (a) outlet temperature (b) temperature distribution

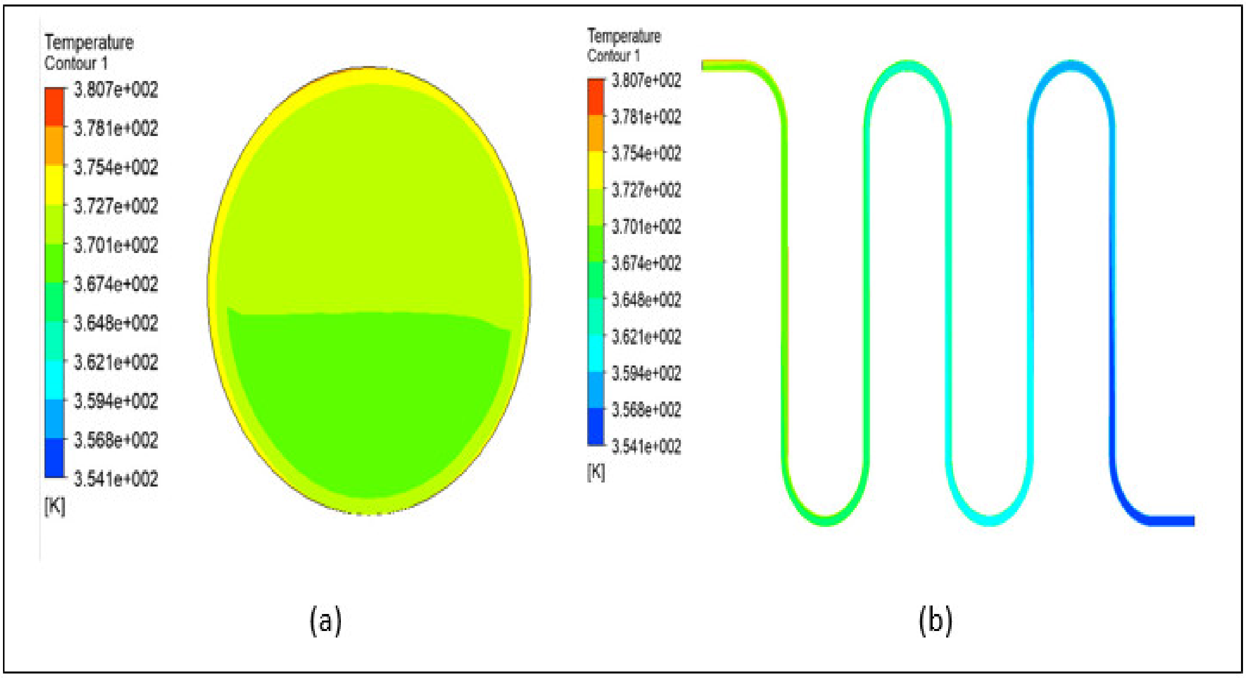


Figure 6. Serpentine tube temperature contour of the second design at tilt angle 13°(a) outlet temperature (b) temperature distribution

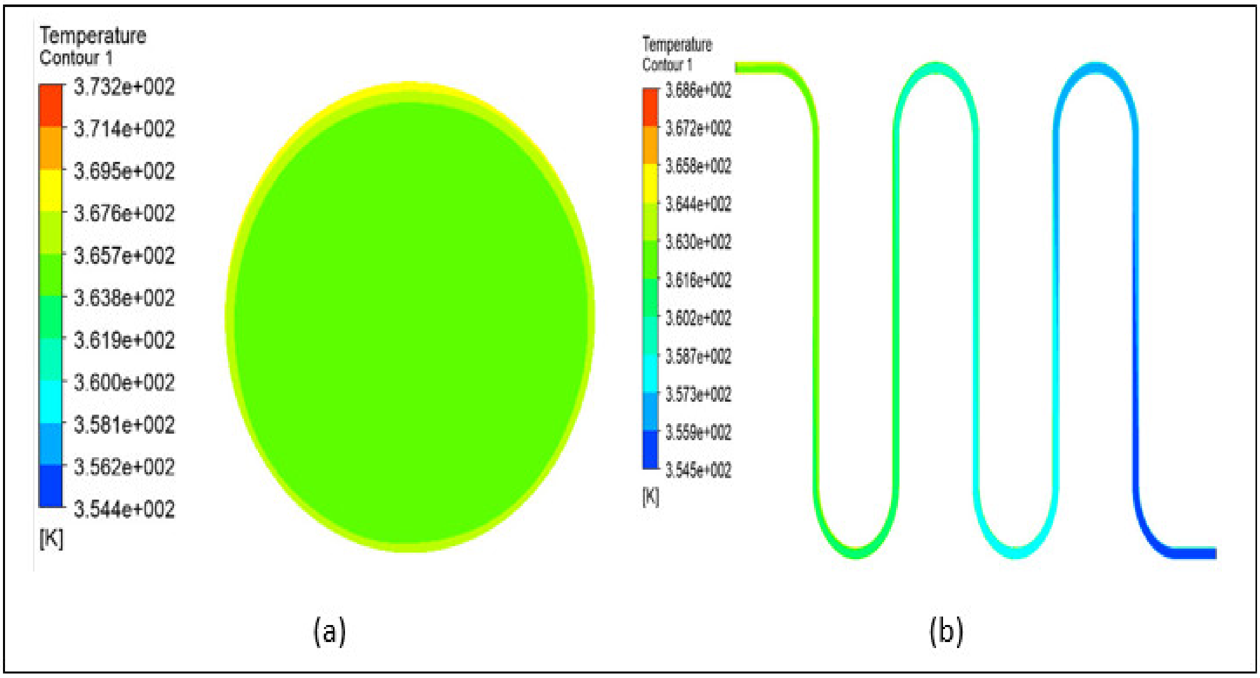


Figure 7. Serpentine tube temperature contour of the second design at tilt angle 30°(a) outlet temperature (b) temperature distribution

March-April 2020
ISSN: 0193-4120 Page No. 10985 - 10997

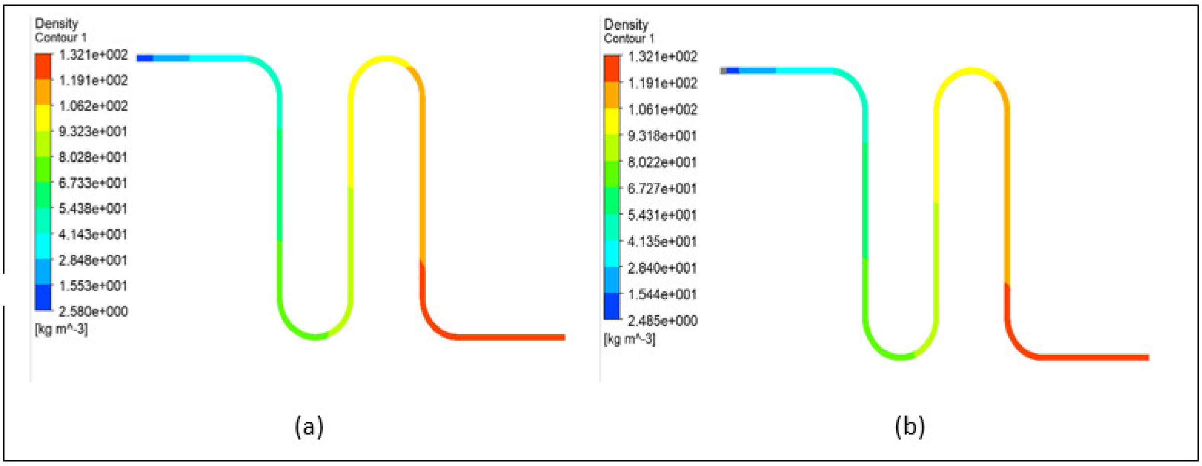


Figure 8. Serpentine-tube density contour of the first design (a) at tilt angle 13° (b) at tilt angle 30°

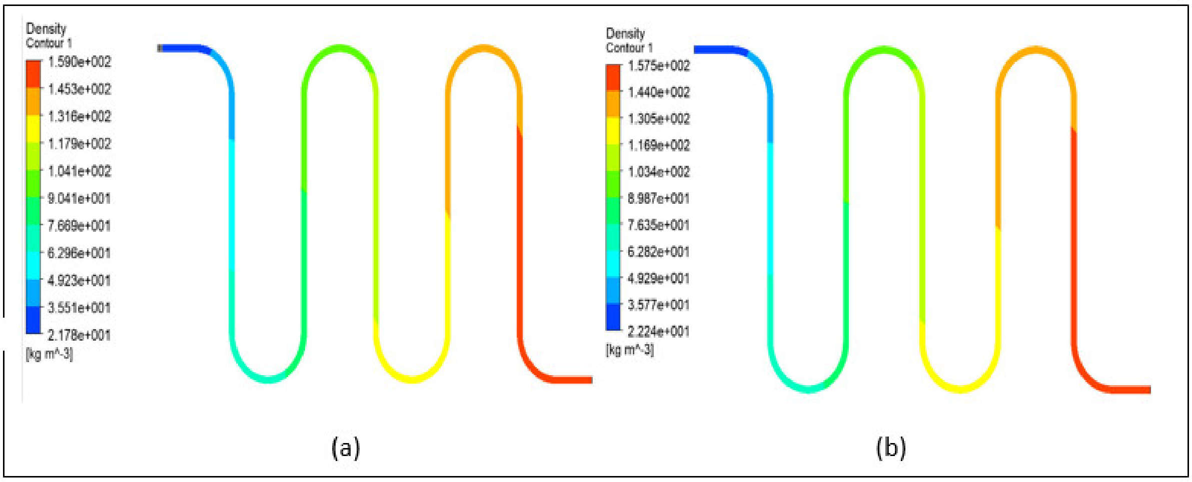


Figure 9. Serpentine-tube density contour of the second design (a) at tilt angle 13° (b) at tilt angle 30°

### 3.2. The Temperature Distribution Along the ST

Fig. (10) shows the simulation results of temperature distribution at two different tilt angles, the first tilt angle is 13°, it represents the average tilt angle of month (May)in Babylon governorate [20],and the other tilt angle is 30°which represents the average tilt angle along the year for Babylon governorate[21]. The intensity of solar radiation is measured by TES module (132) solar power meter data logging device which ranges from 0 to 2000 W/m$^2$, where this device is set up to the south (azimuth angle = 0°) and the device contains a lens which is placed parallel to the surface of the solar.

The readings are recorded in 6th May 2019 under ambient temperature 38°C. The time of recorded reading was in the same day for both angles, where the reading at tilt angle 13° recorded at 11:00 a.m, while at angle 30° recorded at 11:15 a.m. The value of SIR that recorded at angle 13° higher than the value at angle 30°, the tilt angle effects on the solar irradiation that received by the absorber surface. The results show that the difference between the temperature at tube inlet and the first position on 10cm distance after exposes to the solar irradiation is about 0.693°C at 13°and 0.375 °C at angle 30°for the first design.On the other hand, in the second designthe temperature distribution was obtained and compared for 9 positions the comparison was between the tilt angle 13° and 30°.

Figure (11) illustrates the difference between inlet and outlet of the first position at angle 13° is about 1.622°C, while at angle 30° was 0.844°C This related to the value of the solar irradiation that the tube exposes to it. The temperature distribution between the position inside the tube for the first design is in range (0.6°C to 2 °C)at angle 13° and (0.3°C to 1°C) at angle 30°. In the second design the temperature difference inside the serpentine-tube increased gradually from 1.6°C to 3°Cat angle 13° and from 0.6°C to 1.2°C at angle 30°.

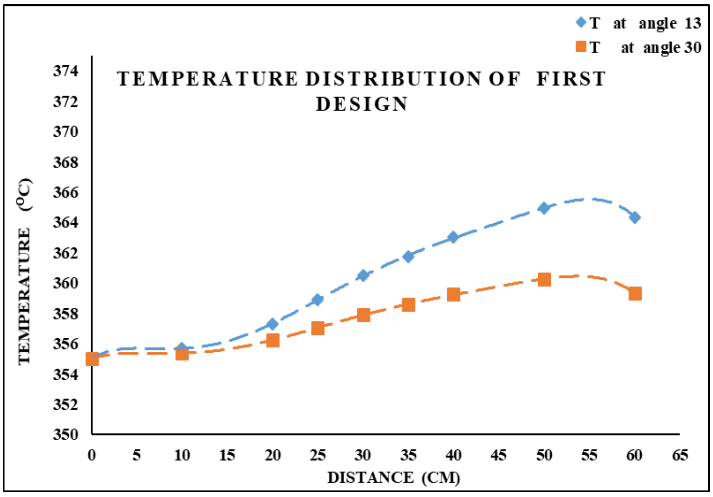


Figure10. temperature distribution along ST at tilt angles 13° and 30° for the first design

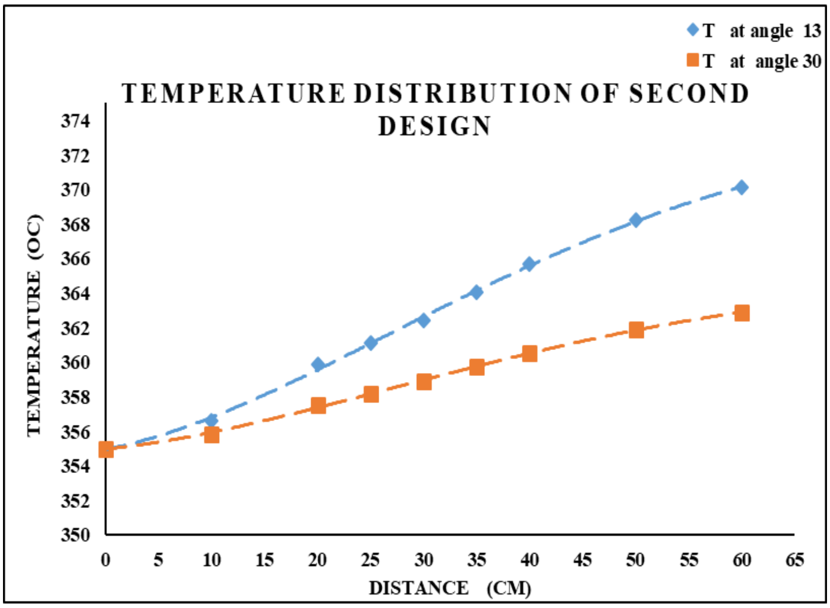


Figure 11. Temperature distribution along ST at tilt angles 13° and 30° for the second design

### 3.3. The Thermal Performance of the ST

Figure (12) shows the collector thermal efficiency with tilt angle for two designs of serpentine. The results show that the length of serpentine-tube and the number of turns have a significantly effect on the collector efficiency, when the length of tube and number of turns increase the thermal performance increase. Also the tilt angleeffects on the thermal performance of the solar collector, where the tilt angle limited the absorbed solar irradiation by the collector. From the result of study, it observes that a higher thermal performance is achievedfor the second design at angle 13°, where the efficiency is about25% at angle 13° and 38.6% at angle 30° for the same design, while in the first design the efficiency was 21.8% and 17.66% at angle 13°

and 30° respectively.

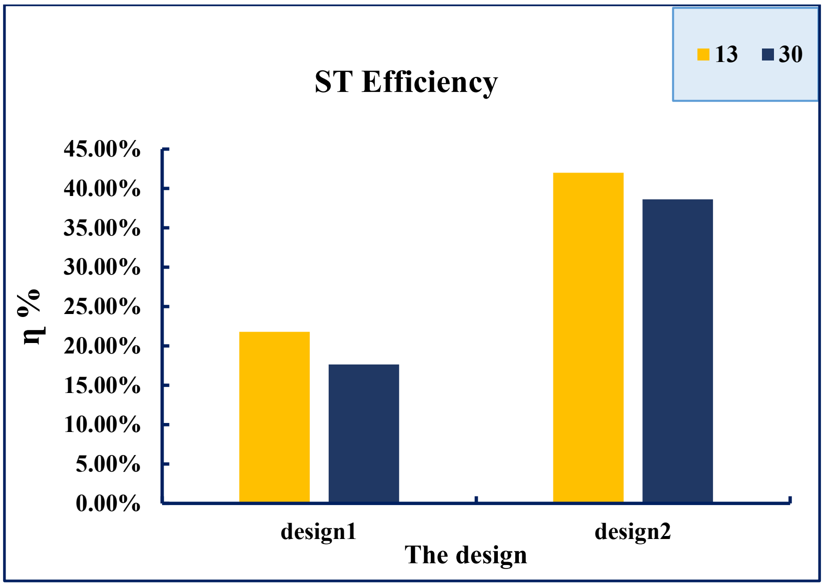


Figure 12. The effect of tilt angle on the efficiency of serpentine tube for both design

## 4. CONCLUSION

This paper represents a CFD simulation to the serpentine-tube collector (STC) to predict the enhancement in the temperature of the working fluid at the serpentine-tube outlet. From the mentioned results can be concluded the following

1. The system performance generally affected by solar collector shape, size in addition to the flow rate of working fluid.
2. The value of the received solar irradiation by the data logging device of a solar power meter is affected by the tilt angle. The tube length and number of serpentine turns have a significant effect on the thermal performance, whereas they are increased the thermal performance increases.
3. The obtained results show that the temperature of the refrigerant STC outlet at angle 13°for the first model ($T_{out}$) became and 364.4K and for the second model temperature was 370.2K, while at angle 30° the temperature at STC outlet was 359.3K for the first model and 362.9K for the second model.
4. The thermal performance enhancement of the second design was 9.08% and 7.13% at 13° and 30° respectively, while for the first design the improvement was 5.37% at angle 13° and 3.88% at angle 30°.

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