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Optimization of the Heat Exchanger in a Flat Plate Indirect Heating Integrated Collector Storage Solar Water Heating System

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Abstract

Due to the environmental impact of energy usage, consumers need to be encouraged to use renewable energy sources such as solar energy. The indirect heating flat plate integrated collector storage solar water heating system is one of the compact systems for domestic water heating. It incorporates the collection of a solar energy component and a hot water storage component in one unit. The objectives of this study were to investigate the effect of different parameters on the thermal performance of this system with the aim of reducing both the initial and the running costs. The outlet service water temperature was used as a measure of performance, because it is an indicator of the energy acquired from the solar radiation. The continuity, momentum and energy equations of the fluids involved in the system were numerically solved in a steady state condition, using FLUENT software. Three-D CFD models were developed and validated using previous experimental results. A standard k-ω turbulent model was used in the optimization of the heat exchanger because it produced good agreement with the experimental results. The surface-to-surface radiation model was included. The effect of single and double row heat exchangers with different lengths was investigated. Circular and elliptic cross-section pipes were also examined. Mass flow rates of 500 and 650 L/h were chosen. The effect of the angel of collector was also examined. The results showed that the single row HX of 10.8 m length for both the elliptical and type B tube gave high service water outlet temperature (acceptable for heat exchanger design) and with

low pumping power. This resulted in an increase in the thermal efficiency and a significant reduction in both the initial and the operating costs of the system.

Nomenclature

f_i: Body force
t: Time
p: Fluid pressure
U_i: Fluid mean velocity component (u, v, w)
$\overline{u_i' u_i'}$: Reynolds stress tensor
x_i: Cartesian coordinates (x, y, z)
$\overline{u_i' T'}$: Turbulent heat flux tensor
Pr: Turbulent Prandtl number
υ: Fluid kinematic viscosity
T: Fluid mean temperature
ρ: Density

1. Introduction

As a consequence of the increase in the world's population, human development, the increase in individual income and the aspiration for more comfortable life styles, power consumption has increased significantly over the last three decades resulting in an increase in the carbon emissions. The global rate of CO_2 emissions from fossil fuel burning and industrial process has increased by 1.1% per year between 1990 and 2000. This increase has been accelerated to more than 3% per year for the period between 2000 and 2004 [1]. Moreover, the world's population is expected to increase to 8.1 billion by 2030 [2]. Achieving the Millennium Development Goals for 2030 will require that the average carbon emissions for each individual are reduced to 3.7 tCO2/year [2]. Carbon emissions must be reduced and this can be achieved by increasing the percentage of energy generated from clean resources like solar, wind, geothermal and hydraulic energy.

Research involving clean sources of energy such as solar energy, has increased significantly over the last four decades, particularly after the World Oil Crisis in 1973 [3]. Solar energy can be used in the industrial, commercial and domestic sectors. In the domestic applications, households consume energy in air conditioning, heating, water heating, lighting and other applications. An environmentally and economically important and costly use occurs in the production of domestic hot water, which accounts for approximately 14% of the domestic energy consumption in the United States [4]. An economic and efficient system is required to encourage households to use solar water heating.

The solar collectors are devices which capture the solar energy and transfer it into thermal energy that increases the internal energy in the fluids, and hence increases their temperature. There are several types of solar collectors, including the flat plate collector, evacuated tube, parabolic trough, central receiver and dish concentrator [5]. The temperature that the different types produce is a key indicator of their relevance to a particular application. For example, the evacuated tube type collector can produce 90-200^{o} C and the parabolic trough can produce outlet fluid temperatures between 260-400^{o} C, while the central receiver can produce 500-800^{o} C. According to Lovegrove and Luzzi [5], the highest outlet fluid temperature (500-1200^{o} C) is produced by the dish concentrator type, which is appropriate for power generation. The flat plate collector, however, is used for applications that require a temperature lower than 100^{o} C such as domestic hot water.

There are many types of flat plate collector, including coupled units, conventional indirect circuit solar thermosiphons and flat plate integrated collector. The flat plate integrated collectors are characterized by incorporating the collection of the solar energy with the storage of hot water in one unit [6]. This reduces the cost of the system as there are no connecting pipes and only a small area is required for installation [7, 8]. There are two types of the flat plate integrated collectors; the direct heating system in which the service water flows into the storage tank and is directly heated through the collector (Figure 1) and the indirect heating system in which the service water passes through a serpentine tube that is immersed in the stored fluid (Figure 2), [9]. Construction of the storage tank in the indirect heating system is cheap by comparison with the direct heating system and other flat plate collector like coupled units and conventional indirect circuit solar thermosiphons [8]. This is due to the fact that the storage tank is not connected directly to municipal water pressure (the storage water is not being refreshed) and the pressure inside the tank is low [8]. However, the storage tank in the direct heating type needs to be manufactured from a high corrosion resistant material that is able to withstand high pressures. This increases tank construction costs to more than 50 percent of the total system price [9]. The objective of this work is to investigate ways to enhance the efficiency of the indirect heating system and make it more economically attractive to consumers by minimising both the initial and operating costs. To make these systems more efficient, two issues need to be addressed; reducing the heat loss from the system and enhancing the heat gained from the sun.

Kumar and Rosen [10] investigated five strategies for reducing heat loss in a flat plate integrated collector system. They tested: (1) a single glass cover without night insulation, (2) a single glass cover with night insulation cover; (3) a double glass cover without night insulation cover; (4) transparent insulation with a single glass cover and (5) an insulating baffle plate with a single glass cover. They found that case 3 provided the greatest thermal performance while case 5 had the lowest thermal efficiency. AL-Khaffajy and Mossad [11] investigated the optimum size for the upper and lower air gap spacing in a flat plate integrated collector system with double glass covers. They found that the combination of 15 mm and 35 mm for the top and lower air gap respectively, gives the least heat loss.

The energy gained by the solar collectors can be enhanced by identifying the optimum solar collector angle and orientation to capture the maximum solar radiation and this depends on the collector location in the globe [12, 13]. Moreover, enhancing the heat transfer to the heated water in the solar collectors has been widely investigated. Kumar and Rosen [6] used a corrugated absorber surface to enhance the heat transfer between the absorber and the storage water in a direct heating integrated flat plate collector. They found that the system performance improved when using a corrugated absorber. Chen *et al.* [14] studied an indirect heating integrated flat plate collector operated with paraffin instead of water and their objective was to enhance the heat transfer. They added aluminium foam to increase the thermal conductivity of the paraffin since the aluminium has high thermal conductivity (202.4 W/m K). They found that adding aluminium foam enhanced further the heat transfer between the storage fluid (paraffin wax) and the service water.

Gertzos and Caouris [9] investigated another heat transfer enahncement approach which is using a pump to circulate the storage water in the indirect heating flat plate collector (Figure 3). This resulted in an increase in the outlet service water temperature. Gertzos and Caouris [15] optimized four parameters that have an effect on the mean tank water velocity, and hence on the outlet service water temperature. These parameters were the inlet and outlet position of the circulating pump, the diameter of the inlet and outlet of the circulating tube, and the arrangement of the interconnecting fins. The optimum inlet diameter was 8 mm, while the optimum inlet position was 100 mm from the top right side and the outlet was 337.5 mm from the lower left side (Figure 3). For the interconnecting fins, they found that the case using five fins of 10 cm length was the best option. Gertzos, Caouris and Panidis [8] investigated the effect of the heat exchanger tube (HX) position relative to the storage tank wall, tube length and diameter for the system with circulating pump. The position where the tube was placed inside the tank touching the top and lower wall of the tank was found to be the best. The optimum diameter was found to be 16 mm and they found that there is no need for a further increase in the tube length of 21.68 m in their system. Mossad and AL-Khaffajy [16] investigated two configurations of the heat exchanger in an indirect heating flat plate collector without circulating pump. They concluded that both the initial and running cost of the system could be minimized by using a single row heat exchanger touching the top wall of the tank.

Previous studies in the indirect heating system [8, 9, 15, 17] which investigated the heat transfer between the storage and service water assuming that the temperature of the storage water was fixed at 60^{o} C for the steady state investigation or the initial temperature of the storage water was 60^{o} C or 80^{o} C for the transient investigation. These studies used a pump to circulate the storage water. The present study investigated the whole collector consisting of the double glass covers and the heat exchanger. This study used the assumption that the temperature of the absorber surface is constant at 60^{o} C. This assumption was considered to be more realistic than the previous assumptions for two reasons. First, based on energy balance, the absorber temperature can reach 60^{o} C when solar incident radiation is 650 W/m^2, but there is no evidence that the storage water reaches this temperature evenly. Second, the absorber is the heat source in the system rather than the storage water. Therefore, assuming the absorber temperature is constant; including the effect of the double glass covers with no circulating pump is a more realistic model and was expected to use less energy. The service water flow rate is chosen as 500 and 650 L/h, as have been used in the investigation by other researchers [8, 15, 17].

2. MATHEMATICAL MODEL

To predict the temperature and velocity of the service water as well as the air in the gap, the equations governing the flow must be solved. According to Nakayama [18], the continuity, momentum and energy equation for transient, incompressible and turbulent flow can be written in abbreviated form as follows:

Continuity equation

$$\frac{\partial U_j}{\partial x_j} = 0 \qquad (2.1)$$

Momentum equation

$$\frac{\partial U_i}{\partial t} + \frac{\partial U_j U_i}{\partial x_j} = -\frac{1}{\rho}\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\upsilon\left[\frac{\partial U_i}{\partial x_j} + \frac{\partial U_i}{\partial x_j}\right] - \overline{u_i' u_i'}\right) + f_i \qquad (2.2)$$

Energy equation

$$\frac{\partial \mathbf{T}}{\partial t} + \frac{\partial (U_j \mathbf{T})}{\partial x_j} = \frac{\partial}{\partial x_j}\left(\frac{\upsilon}{Pr}\frac{\partial \mathbf{T}}{\partial x_j} - \overline{u_i' T'}\right) \qquad (2.3)$$

The above equations are non-linear partial differential equations and the analytical solution is impossible except for very simple cases, but these equations can be solved numerically. The present study used FLUENT software which uses a finite volume approach. The pressure-based type solver was used. The effect of gravity was included by enabling the full buoyancy effect. The variation of the properties of air with temperature has been included by using the incompressible ideal gas equation for estimating the density and kinetic theory equation for specific heat, thermal conductivity and viscosity. The variation of properties of water with temperature was also included using equations (2.4a-d), which were recommended by Gertzos, Pnevmatikakis and Caouris [17].

$$\rho = -1.3187 * 10^{-7} T^4 + 1.8447 * 10^{-4} T^3 - 9.9428 * 10^{-2} T^2 + 23.28T - 1113.5 \quad (2.4)$$

$$\mu = 3.533 * 10^{-11} T^4 - 4.8141 * 10^{-8} T^3 + 2.4637 * 10^{-5} T^2 - 0.0056188T + 0.48281 \quad (2.5)$$

$$C_p = 3.321729 * 10^{-6} T^4 - 4.459811 * 10^{-3} T^3 + 2.248733T^2 - 5.041488 * 10^2 T + 4.654524 * 10^4 \quad (2.6)$$

$$k = 6.2068 * 10^{-10} T^4 - 8.0897 * 10^{-7} T^3 + 3.8437 * 10^{-4} T^2 - 7.7569 * 10^{-2} T + 6.1019 \quad (2.7)$$

The velocity-pressure coupling was treated by using the SIMPLE algorithm. A first order upwind scheme was used for Momentum, Turbulent Kinetic Energy and Turbulence Dissipation. For the residual, 10^{-4} was used as the convergence criteria for all terms: x-velocity, y-velocity, z-velocity, kinetic energy, epsilon and continuity, except for the energy, which was 10^{-8}. The standard k-ω and realizable k-є turbulence models were used in the CFD validation to identify the best model.

3. Validation of the CFD model

In order to get confidence of the chosen turbulence model used in the CFD, the experimental work by Gertzos, Pnevmatikakis and Caouris [17] was chosen to validate the CFD model. The system they used consisted of a storage water tank and a serpentine tube immersed in the storage water (Figure 4). This case was chosen because it shared the most important features with the model investigated in this study. The volume of the storage tank and the absorber area were the same for both cases and both have serpentine heat exchanger. The Reynolds number of the service water was 2.6×10^4 for the experimental case, while it was varied in this study between 1.5×10^4 and 2.6×10^4. This indicate the similarity of the flow and hence the appropriateness of the turbulence model.

3.1 Experimental procedure

Gertzos, Pnevmatikakis and Caouris [17] examined the heat transfer between the service and storage water with and without a circulating pump. Their experimental results of the case without a circulating pump were used in this study. The storage tank was made of iron (k= 63.9 W m^{-1} k^{-1} and 1.7mm thick) and its inner dimensions were 81 x 135 x 10 cm containing about 109 L of water. It was insulated using glass wool (k= 0.041W m^{-1} k^{-1} and 5 Cm thickness). The tube was made of iron (k = 63.9 W m^{-1} k^{-1}, 2 mm thickness and 10mm inside diameter). The total length of the tube inside the tank was 16.2 m. The outer part of the tube was insulated by conventional foam with 1.9 cm thickness and 0.037W m^{-1} k^{-1} thermal conductivity.

The experimental procedure took place indoors. The angle of the tank was 45^o from the horizontal plane, the inlet service water temperature was in the range of 16.5-17.8^o C and the flow rate for the case that was chosen in this validation was 500 L/h. The procedure was as follows: The storage water was heated, using successive passages through the external heater. When the temperature of the storage water reached 80^o C, the heating process was stopped and the service water flow was started through the tube. The inlet, outlet service water and the storage temperatures at four different positions (T_1, T_2, T_3 and T_4 as shown in Figure 4) were measured at one second time intervals. They were averaged and recorded every 30 seconds, for an energy withdrawal period of one hour.

3.2 CFD simulation

A computational grid was developed for the same setup as Gertzos, Pnevmatikakis and Caouris [17] using ANSYS 13.0 software. The number of elements was 1,130,000 and most of them were hexahedral (Figure 5). The continuity, momentum and energy equations were solved in transient conditions for a one hour period. The realizable k-ϵ and standard k-ω turbulence models were tested.

3.3 Boundary condition and operating parameters

The boundary condition at the inlet was taken as "velocity-inlet". The velocity was 1.768 m/s (500 L/h flow rate), hydraulic diameter 0.01 m, turbulence intensity, I, 4.6% which is a function of the Reynolds number, R_e, and was computed from $I = 0.16(R_e)^{-1/8}$ [8] and temperature 16.9^o C which was the average of the experimental inlet temperatures over the entire period. The boundary condition at the outlet was taken as "pressure-outlet" with atmospheric pressure. The interfaces between the tube wall and the water were defined as wall with coupled condition to allow the heat to transfer from the storage water to the tube wall and from the tube wall to the service water. The outer part of the service water tube was assumed to be fully insulated. The boundary condition at the outer wall of the storage tank was taken as radiation to the laboratory walls with temperature of 20^o C and convection to the surrounding air with a heat transfer coefficient, h_c=5 W/m^{2}K, and 20^o C temperature. The convective heat transfer coefficient was estimated using Kreith's equations, (5.13) and (5.14) [19].

3.4 Validation results

Two transient CFD simulations (for a one hour period) were studied. The Realizable k-ϵ and the standard k-ω turbulence models were used. The initial temperature of the storage water was taken as 80^o C as in the experiment. Every five seconds of the solution, the average temperature at the outlet, T_out, average temperature of the storage water, T_ta, and the temperature of the points T_1, T_2, T_3 and T_4 were recorded.

Figures 6 and 7 present a comparison of the experimental value obtained by Gertzos, Pnevmatikakis and Caouris [17] for the service water outlet temperature, T_out, and the average storage water temperature, T_ta, with the numerical results obtained by using the k-ϵ and k-ω turbulent models, respectively. Figures 8, 9, 10 and 11 show the experimental and numerical temperatures of the points T_1, T_2, T_3 and T_4, using both turbulence models. There was good agreement between the experimental and the numerical results using both models. However, the percentage difference between the CFD simulation and the experiment for the k-ϵ model was higher than for the k-ω model. The difference varied between zero and 15 per cent for k-ϵ and zero to 8.5 per cent for k-ω model. For both models, the maximum difference was in the outlet temperature. This finding agrees with the finding of Gertzos and Caouris [9] who also have tested different turbulence models and found that the k-ω turbulence model gave better agreement than k-ϵ turbulence model.

It is important to note that the difference between the experiment and the CFD model was expected due to some sources of error. These sources can be summarized as follows: (i) thermocouple accuracy was ±1^o C and flow meter accuracy was ± 0.35%, (ii) in the experiment, the inlet service water temperature was in the range of 16.5-17.8^o C. In the numerical simulation, the inlet service water temperature was assumed constant and estimated by taking the average over the whole period as 16.9^o C, (iii) the heat transfer coefficient was assumed to be constant at 5 W/m^2K for the outer wall of the tank during the whole simulation period. Due to the decrease in the storage water temperature, the temperature of the wall would have decreased as the time progressed which may slightly reduce the heat transfer coefficient and (iv) the laboratory air and walls temperature was assumed to be 20^o C in the CFD simulation, while it was not provided by the experimental study.

The good agreement between the experimental and the numerical results, using the k-ω model, confirms that the k- ω model can be used in this study with more confidence. The next step is to investigate different configurations of the heat exchanger to identify the optimum design.

4. Optimization of the heat exchanger

Two configurations of the HX were investigated: double and single row HX. The effective length (tube inside the collector) for the tube in the double row HX was 16.2 m, in a single row HX was changed to 8.1 m and 10.8 m (Figure 12). Two circular pipe sizes were modelled; a copper tube type A DN 15 (1/2”) with an inside diameter of 10.7 mm and wall thickness of 1 mm, type B DN 20 (3/4”) with an inside diameter of 17.1 mm and wall thickness 1 mm. An elliptical cross-section copper tube was also studied. The cross sectional area of the elliptic tube was equivalent to the area of the circular tube of 17.1 mm inside diameter with an aspect ratio of 2:1; i.e. the major radius is 12.092 mm, the minor radius is 6.046 mm, and the hydraulic diameter is 15.295 mm. The hypothesis for using an elliptic tube is to increase the area that is in contact with the heat source i.e. the absorber surface. The tube position was chosen as half of the tube was placed in the absorber surface and half in the storage water to enhance the heat transfer between the absorber and the service water (Figure 13). The flow rate was taken as 500 and 650 L/h.

4.1 CFD simulation

To validate the grid dependency, three computational grids were developed for the model with type A tube of 16.2 m length; 2.5, 2.74 and 3.125 million elements. The results of all three models were almost the same, but the model of 2.74 million elements converged faster than the others because the mesh quality was better. Therefore; the same method adopted to generate the computational grid in the model of 2.74 million, was applied for all others models.

ANSYS-13.0-FLUENT software was used to solve the continuity, momentum and energy equation in a steady state condition. Since the standard k-ω model gave a good agreement with the experiment, it was used in all of these cases. A surface-to-surface radiation model was also included, ignoring the effect of the medium between surfaces. The radiation process was started by estimating the view factors between the surfaces. The radiosity of the surfaces was updated every ten iterations based on the new surfaces' temperature, through another iterative process, to include the heat transfer due to radiation more accurately.

The collector angle was varied to investigate its effect on the heat exchangers performance. Angles of 45^o, 30^o and 60^o were chosen. The inner dimensions of the storage tank were 81 x 135 x 10 cm containing about 109 L of water. The absorber was made of metallic-nickel chrome (M-N-Chrome) with 10 mm thickness. The thickness of the lower and upper glass cover was 3 mm. The upper and lower air gap spacing was taken as 35 and 15 mm, as this was found to be the most efficient combination [11]. The physical properties of the materials adopted in the simulation are given in table I.

4.2 Boundary condition and operating parameters

The boundary conditions on the top-glass cover was taken as convection with a heat transfer coefficient of 10 W/m^2 K to an ambient temperature, T_a=17^o C, and radiation to the sky at a temperature which was calculated as T_s= 0.0552 $Ta^{1.5}$ [20]. A constant temperature of 60^o C was assumed for the absorber surface in all cases. The velocity of the service water at the inlet was varied to be 0.604, 0.786, 1.544 and 2.00 m/s, as the mass flow rate and tube diameter varied and the inlet temperature was constant at 17^o C. A constant pressure was assumed at the outlet and was taken as atmospheric pressure. The side and lower walls of the collector were assumed to be adiabatic. The no-slip condition was applied on all walls with roughness constant of 0.5. In order to allow the heat to transfer through the walls, the interfaces were defined as walls with coupled condition.

4.3 Results and discussion

Tables II and III show the outlet temperature, heat gained and pumping power required as the flow rate changed from 500 L/h to 650 L/h for the different type tubes (circular A& B and elliptic) and tube length. It was observed that the heat gained increased as the mass flow rate increased, while the outlet temperatures decreased. This means that the increase in the flow rate enhanced the heat gained by the service water from the solar radiation. Moreover, there was a small difference in the outlet temperature (1-2^o C) between type A and B tubes, but there was up to a 70% reduction in the power required when type B tube was used. The elliptical tube gave a one degree higher in the outlet temperature than the circular one for the same length, flow rate and the same cross sectional area.

To choose the optimum HX configuration, many factors must be taken into consideration. These are: (a) the outlet service water temperature (b) power required to run the system as well as (c) the initial cost of the system. The tube cost and power required running the system with a single row HX and 8.1 m tube length for types A and B tube was investigated and found to be almost half of the cost for a system with a double row HX, while the outlet temperature was the same (Table IV). It was noticed that the temperature of the service water increased only in the front row of the double row HX and there was no increase in the temperature in the back row. This can be seen in the temperature contours of the service water for the double row HX type A tube which is presented in Figures 14. Similar behaviour was noticed when type B tube was used. Thus, the double row HX is not considered to be a good design because of the high initial and running cost with little thermal gain.

To increase the outlet temperature and hence reach an acceptable temperature for domestic use, 2.7 m was added to the tube in the front row (i.e. in the absorber side). Figures 15, 16 and 17 present the temperature contours of the service water for the tube length 10.8 m and flow rate 500 L/h for type A tube, type B tube and tube with elliptical cross-section respectively. The temperature increased by 2-3 degree for type A and B tubes (Table IV). Therefore, adding 2.7 m to the front row (i.e. single row HX) was more effective than adding 8.1 m in the back row, i.e. double row HX, with only small increase in the pumping power and the initial cost.

The increase in the power required was lower for type B tube (0.4 W) than for type A tube (3 W), as type B tube has bigger diameter. For both mass flow rates and for a tube length of 10.8 m, the outlet temperature for type B tube was one degree lower than type A tube. The running cost of type A tube was five times higher than for type B tube, but the initial cost of type B tube was slightly higher than A (Table IV). The advantages of lower initial cost for the type A tube systems is expected to be quickly overtaken by the high operating cost of these systems. As a result, type B tube systems are more economical than type A tube systems.

For the elliptical tube, the outlet temperature was higher than the circular one by about 1-3 degrees for the same length and same-cross sectional area (Table IV). This can be attributed to the area in contact with the absorber surface being bigger. The power required was similar to type B tube, because the average service water velocity was similar in both tubes. However, the initial cost of the elliptic tube is not available and it needs further investigation.

In heat exchanger systems, when the temperature difference between the cold and hot fluids is reduced to lower than 6 degrees, the heat exchanger configurations are acceptable [8]. In this study, the difference between the absorber temperature (the heat source) and the service water outlet temperature (cold fluid) was reduced to a range of 2.1^{o} and 6^{o}. It is to be noted that a circulating pump was not used to circulate the storage water. As the pump requires electricity and maintenance, eliminating the circulating pump, reduce both the initial and the running costs.

The initial and operating costs were also minimized by reducing the service water tube length. According to Gertzos, Caouris and Panidis [8], the optimum tube length was 21.68 m for the system with circulating pump. However, the present study showed that the service water tube length can be reduced to 10.8 m, and the heat exchanger produced an acceptable temperature difference between the absorber surface and the outlet service water temperature. The absorber area and the volume of the storage water tank was the same for both the present study and previous studies [8, 15]. Thus, the initial cost and the running cost of the system is reduced.

To study the effect of the inclination angle of the system, five more CFD models have been developed for different collector angles. The angle was changed to 30^o and 60^o in the system of type A and B tube with 10.8 m length and flow rates 500L/h and 650 L/h. The results, (Figure 17) showed that there was very small increase (0.02^o C) in the outlet temperature when the collector angle increased from 30^o to 60^o, while the increase in the storage water average temperature was 0.4^o C. This resulted from the increase in the natural convection heat transfer between the storage water and the absorber. Previous study [17] reported that as the indirect heating integrated flat plate collector slop increased, the outlet temperature increased. The increase in the outlet temperature in [17] was less than 4^o C when the collector angle was changed from 0^o to 90^o. The difference between the present study and the previous study is that the present study assumed the absorber temperature to be constant (the heat source is the absorber), while in [17] the storage water initial temperature was assumed to be 80^o C (the heat source was the storage water). The increase in the outlet temperature in [17] was also due to the increase in the convective heat transfer coefficient between the storage water and service water.

5. Conclusion

Different designs of the heat exchanger in the indirect heating integrated collector storage solar water heater system were studied. The increase in the outlet service water temperature was used as a measure of the thermal performance and the power required to operate the system was used as a measure of the running cost. ANSYS 13.0-FLUENT software was used to identify the optimum configuration of the indirect heating integrated collector storage solar water heating system. The results for a particular system using the realizable k-ϵ and standard k-ω turbulence models were compared to available experimental results to determine the best turbulence model to use in the heat exchanger investigation. The difference between the experimental and the numerical results using the k-ϵ model was higher than using the k-ω model. Therefore, the standard k-ω model was used in the heat exchanger investigation. This study concluded that the single row HX of 10.8 m length for both the elliptical and type B tube gave high service water outlet temperature (that is acceptable for heat exchanger design) and with lower power needed in comparison to the other cases studied. An increase in thermal efficiency and a significant reduction in the initial and running costs of the system have been achieved. This is due to the increase in outlet temperature, the reduction in the length of service water tube and the circulating pump elimination. This study has also found that the angle of the collector has negligible effect on the heat exchanger design, as the change in outlet temperature was very small when the collector angle was varied.

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