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Investigating two configurations of a heat exchanger in an Indirect Heating Integrated Collector Storage Solar Water Heating System (IHICSSWHS)

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Abstract. Water heating makes up 14% of domestic energy usage. 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 integrated collector storage solar water heater system (IHISSWHS) is one of the most economical systems. The objective of this study was to investigate ways to enhance the performance of this system to encourage many households using it. Two configurations of the system were studied; 16.2 m long double row heat exchanger (HX) and 8.1m and 10.8 m long single row HX. Inside pipe diameters were 10.7mm and 17.1 mm; using two flow rates 500 L/h and 650 L/h. The steady state continuity, momentum and energy equations were numerically solved, using FLUENT software. A standard k-ω turbulent model and surface to surface radiation was used. The system with 10.8 m long single row HX provided higher outlet temperature than the system with 16.2 m double row. Therefore, a significant reduction in cost and power usage can be achieved by using a single row HX. For both flow rates investigated, the larger diameter pipe need about five folds less power to run, with very little sacrifice of outlet temperature.

Key words

Heat exchanger, solar heating, indirect heating, integrated collector.

1. Introduction

The increase in the price of fossil fuel and its negative environmental impact led to increased research involving cheap and clean sources of energy such as solar energy. The use of solar energy has been growing in electricity generation, air conditioning and water heating. An important and costly use, environmentally and economically, occurs in the production of domestic hot water. Water heater systems consume fourteen per cent of the domestic energy consumption [1]. An economic and efficient system is required to encourage households to use solar water heating.

Solar flat plate collectors are used for producing domestic hot water. Integrated collectors are a type of flat plate collector characterized by incorporating the collection of the solar energy part and the storage of hot water in one unit [2]. This reduces the cost of the system as there are no connection pipes and only a small area is needed for installation [3, 4]. There are two types of the integrated systems: 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) [5]. Construction of the storage tank in the direct heating type does not come cheaply. Since the storage tank is connected directly to town water pressure, the pressure inside the tank is relatively high. Therefore, the storage tank in this type needs to be manufactured from a high corrosion resistance material that is able to withstand high pressures. This leads to increased construction costs since the cost of constructing the tank is more than 50 percent of the total system price [5].

Enhancing the heat tranfer in the direct and indirect integrated collector system has been investigated quite widely. Kumar and Rosen [2] used a corrugated absorber surface to enhance the heat transfer between the absorber and the storage water in the direct heating system. Gertzos, Caouris and Pnevmatikakis [5] used a pump to circulate the storage water, in order to enhance the heat transfer rate in the indirect heating system. Gertzos, Caouris and Panidis [4] optimized the heat exchanger (HX) positions relative to the tank wall, the service water tube length and diameter for the indirect heating system with circulating pump.

Reducing the heat loss leads to improved performance of the system. Kumar and Rosen [6] investigated five strategies for reducing top heat loss. The 100L tank capacity model and 1 m^2 absorber area was used to assess five cases: (1) single glass cover without night insulation;

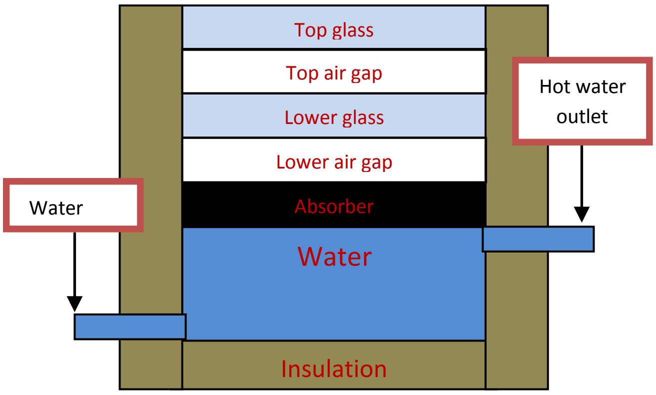

Fig. 1. Direct heating integrated collector system

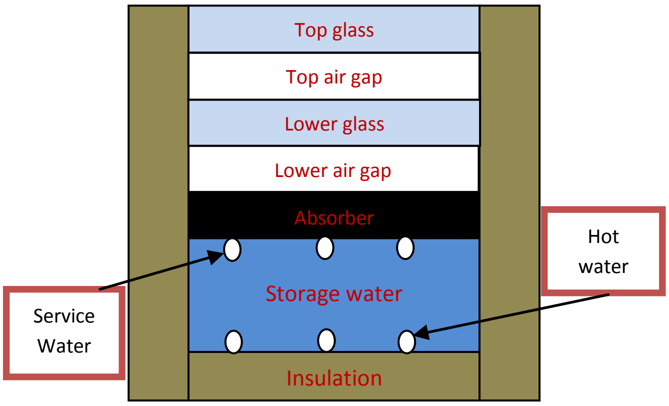

Fig. 2. Indirect heating integrated collector system

(2) single glass cover with night insulation cover; (3) double glass cover without night insulation cover; (4) transparent insulation with single glass cover and (5) insulating baffle plate with single glass cover. They found that case three provided the greatest thermal performance while case 5 has the lowest thermal efficiency. AL-Khaffajy and Mossad [7], have studied the optimum top and lower air gap spacing in the integrated collector system with double glass covers They found that a combination of 15 mm for the lower gap and 35 mm for the top gap gave the minimum radiation and convective heat losses.

The previous study in the indirect heating system [4,5] investigated the heat transfer between the storage and service water assuming that the initial condition of the storage water is 80^{o} C. The present study will investigate the indirect heating system using different assumption. This study will investigate the collector using the assumption that the temperature of the absorber surface is constant at 60^{o} C. This temerpture is based on an energy balance at the absorber surface assuming a solar intensity of 650W/m^{2}. Two designs of the heat exchanger (HX) have been investigated; double row HX with service water tube length of 16.2 m and single row HX with tube length of 8.1 and 10.8 m. Two flow rates of the service 500 L/h and 650 L/h and two tube inside diameters 10.7 mm and 17.1 mm have been investigated. The goal of this work was to identify the size of the pipe that gives the highest outlet temperature with the least energy usage.

2. MATHEMATICAL MODEL

In regards to the heat balance in the indirect heating integrated collector storage solar water heater system, the absorber surface is heated from the solar radiation which has been transmitted through the glass covers. During the daytime, the heat flows from the absorber to the storage and service water. When there is no solar radiation, the energy in the storage water flows to the absorber and service water. In both cases, the absorber loses some of the heat due to convection to the air in the lower gap, radiation to the lower-side walls (the side-wall surrounding the lower air gap spacing) and radiation to the lower glass cover. The lower glass cover loses heat due to convection to the air in the top gap and through radiation to the upper-side walls and to the top glass cover. The top glass cover loses heat due to convection to the ambient air and due to radiation to the sky.

To predict the heat gained by the service water, the flow governing equations should be solved. According to Raisee and Hejazi [8], the continuity, momentum and energy equation for steady, incompressible and turbulent flow can be written in abbreviated form as:

Continuity equation

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

Momentum equation

$$\frac{\partial U_j U_i}{\partial x_j} = -\frac{1}{\rho}\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\upsilon \frac{\partial U_i}{\partial x_j} - \overline{u_i u_j}\right) + f_i \qquad (2.2)$$

Energy equation

$$\frac{\partial (U_j T)}{\partial x_j} = \frac{\partial}{\partial x_j}\left(\frac{\upsilon}{Pr}\frac{\partial T}{\partial x_j} - \overline{u_j \theta}\right) \qquad (2.3)$$

Where:
f_i: Body force
p: Fluid pressure
U_i: Fluid mean velocity component (u, v, w)
$\overline{u_i u_j}$: Reynolds stress tensor
x_i: Cartesian coordinates (x, y, z)
$\overline{u_j \theta}$: Turbulent heat flux tensor
Pr: Turbulent Prandtl number
υ: Fluid Kinematic viscosity

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 that uses finite volume approach to solve the continuity, momentum and energy equation in steady state condition. The pressure-based type solver was used. The effect of gravity was included considering full buoyancy effect. The variation of the properties of air with temperature has been included by using incompressible ideal gas equation for estimating the density and kinetic theory equations for specific heat, thermal conductivity and viscosity. The variation of the properties of water such as density, ρ, viscosity, μ, specific heat, c_p, and thermal conductivity, k, with temperature, T, was also included using equations 2.4a, 2.4b, 2.4c and 2.4d respectively, which were recommended by Gertzos, Pnevmatikakis & Caouris [9].

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

$$\mu = 3.533 * 10^{-11} T^4 - 4.8141 * 10^{-8} T^3 + 2.4637 * 10^{-5}\, T^2 - 0.0056188T + 0.48281 \qquad (2.4b)$$

$$C_p = 3.321729 * 10^{-6} T^4 - 4.459811 * 10^{-3} T^3 + 2.248733T^2 - 5.041488 * 10^2\, T + 4.654524 * 10^4 \qquad (2.4c)$$

$$k = 6.2068 * 10^{-10} T^4 - 8.0897 * 10^{-7} T^3 + 3.8437 * 10^{-4} T^2 - 7.7569 * 10^{-2} T + 6.1019 \qquad (2.4d)$$

The standard k-ω turbulence model has been used because the model has been tested by Gertzos and Caouris [5] and there was a good agreement between their experimental results and their numerical ones. The velocity-pressure coupling was treated by using the SIMPLE algorithm and a first order upwind scheme for Momentum, Turbulent Kinetic Energy and Turbulence Dissipation. For the residual, 10^{-4} was used for all terms: x-velocity, y-velocity, z-velocity, kinetic energy, epsilon and continuity, except for the energy, which was taken as 10^{-8}.

3. CFD SIMULATION

Steady 3D CFD models for two different configurations of the indirect heating integrated collector system (fig. 3a and b) have been developed to evaluate the heat gained by the service water. One has a double row heat exchanger (HX); the other has a single row HX. For both models, two types of copper tube were used for the service water; type A DN 15 (1/2"), inside diameter D_i=10.7 mm, wall thickness 1 mm and type B DN 20 (3/4"), inside diameter D_i=17.1 mm, wall thickness 1 mm. The effective length (tube inside the collector) for the tube in the double raw HX was 16.2 m, in a single raw HX was chosen as 8.1 m and 10.8 m.

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, Ta, of 290 K and radiation to the sky at a temperature which was taken according to Akhtar and Mullick [10] as $0.0552\ Ta^{1.5}$. A constant temperature of 333 K was assumed for the absorber surface in all cases. A "velocity-inlet" with 290 K temperature for the service water at inlet and "pressure-outlet" at the service water outlet. The collector angle was chosen to be at 45^o. For both models, the inner dimensions of the storage tank were as 81cm x 135 cm x 10 cm containing about 109L of water; the side and lower walls of the collector was assumed to be adiabatic; the absorber was made of metallic-nickel chrome (M-N-C) with 10 mm thickness; the thickness of the lower and top glass cover was 3 mm; the top and lower air gap spacing was taken as 35 and 15 mm, since it was found to be the most efficient combination [7]. The physical properties adopted in the simulation of these materials are given in table I.

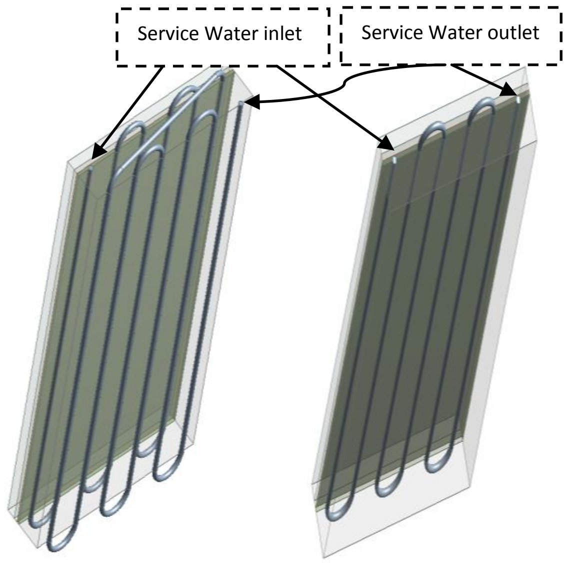

Fig. 3a Indirect heating system with double row HX

Fig. 3b Indirect heating system with single row HX

Table I. Physical properties of the materials used

Material	ρ kg/m^3	C_p J/ (kg. K)	K W/(m^2K)	ϵ emissivity
M-N-C	7865	460	19	0.94
Glass	2800	800	0.81	0.93
Copper	8978	381	387.6	Not included

The geometry and computational grid were generated using ANSYS 13.0 software. To validate the grid dependency, three computational grids have been developed for the model with double row HX; 2.5 million, 2.74 million and 3.125 million elements. The results of all three models were almost the same, but the model of 2.74 million elements was converged faster than the others because the mesh quality was better. The researchers adopted the same way to generate the computational grid in the model of 2.74 million for all other models.

Radiation between surfaces (absorber, lower glass and top glass) has been considered, using a surface to surface radiation model, ignoring the effect of the medium between the surfaces. The radiation process was started by estimating the view factors between the surfaces. The radiosity of the surfaces have been updated every ten iterations throughout the solution, based on the new surfaces' temperature. This is to include the heat transfer due to radiation from these surfaces more accurately.

4. RESULTS and DISCUSSION

For each collector configuration, tube types A DN 15 and type B DN 20 was used. For each case, two service water flow rates were examined; 500 L/h and 650 L/h. Service water temperature at inlet was taken as 17^o C. Flow with four values of Reynolds number (R_e) were investigated, for tube A (D_i=10.7 mm); R_e= 2.28 x 10^4 and 2.96 x 10^4

and for tube B (D_i=17.1 mm); R_e = 1.42 x 10^4 and 1.85 x 10^4. The length of the tube was 16.2 m for the double row HX, 8.1 m and 10.8 m for the single row HX.

The outlet temperatures, heat gained by the service water (q) and the power required to run the system of all cases are presented in table II. The higher flow rate enhanced the heat exchanged since more heat transfer took place; however the outlet temperature was less than that for the lower flow rate cases, which is expected. As expected, the longer length for the single row pipe produced higher outlet temperature. The power required to run the system with 8.1 m tube length for both tube A and B was half of the power required for the system with 16.2 m, while the outlet temperature was the same. Therefore, the back row of the tubes was not effective.

Table II Outlet temperature, heat gained by the service water (q) and power required to run the system for all studied cases

	Outlet Temperature and q for 500 L/h	Outlet Temperature and q for 650 L/h	Power required (W)	
			500 L/h	650 L/h
Double row HX tube type **A**	56 °C	54 °C	8.2	16.95
	q= 22.6 W	q= 27.9 W		
Single row HX tube type **A** 8.1m	56 °C	54 °C	4.1	8.4
	q=22.6 W	q= 27.9 W		
Single row HX tube type **A** 10.8 m	58 °C	56.9 °C	5.4	11.3
	q=23.7 W	q=30 W		
Double row HX tube type **B**	54 °C	52.3 °C	1.3	2.4
	q=21.45 W	q=26.6 W		
Single row HX tube type **B** 8.1m	54 °C	52 °C	0.86	1.4
	q=21.45 W	q=26.3 W		
Single row HX tube type **B** 10.8 m	57 °C	55.75 °C	1	1.82
	q=23.2 W	q=29.2 W		

The temperature contours of the service water for the double row HX tube type A are presented in figures 4a. The temperature of the service water increased only in the front row of the HX and there was not much increase in the temperature in the back row. This behaviour was the same when type B of the tube was investigated, (figure 4b). This was the case for both mass flow rates. This can confirm that the back row of the tube is not very effective.

Figure 5a and 5b present the temperature contours of the service water for the single row HX, tube type A, with flow rate of 500 L/h and for tube length 8.1m and 10.8 m, respectively. Most of the service water tube was effective because there was an increase in the temperature along the tube.

The temperature contours for the single tube HX, tube type B, with 650 L/h flow rate and length 8.1 and 10.8 m are presented in figure 6a and 6b, respectively. It is obvious that the outlet temperature increased to 57^o C versus 54^o C in the 8.1 m. The temperature difference between the absorber and the outlet service water was around 6 degrees at exit. This indicates that there is no need for further increase in the tube length according to the recommendations in reference [11] in regards to the heat exchanger design.

For the single row HX it was obvious that the longer the pipe the higher heat gain and hence higher outlet temperature. Increasing the flow rate, which increased the Reynolds number, increased the heat gain but due to the higher flow rate the outlet temperature was not as high. The system with type A, 10.8 m long tube, gave the highest outlet temperature which was 58^o C while tube type B with the same length, gave 57^o C. However, the power required to run the system with tube B was much lower than with tube A for the same length, which is expected since the power depended on the square of the average velocity in the tubes, which is lower for tube B. This will have an advantage of reducing the running cost of the system with very little sacrifice of the final outlet temperature.

The outlet temperature of the service water, the required flow rate, the power needed to run the system as well as the cost of the system should be considered in choosing the optimum HX configuration for a household user. However, this research highlighted that the double row HX proved not to be a good design for both size tubes considered in this work, because of the high cost and high pumping power required with no additional benefits than the single row HX.

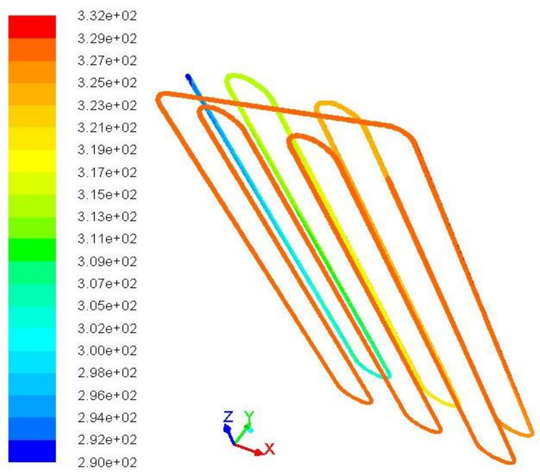

Fig. 4a temperature contours of service water for double row HX, with tube A and flow rate 500L/h

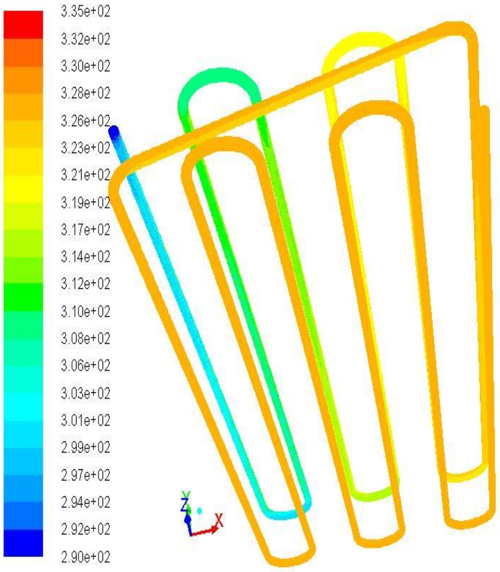

Fig. 4b temperature contours of service water for double row HX, with tube B and flow rate 500L/h

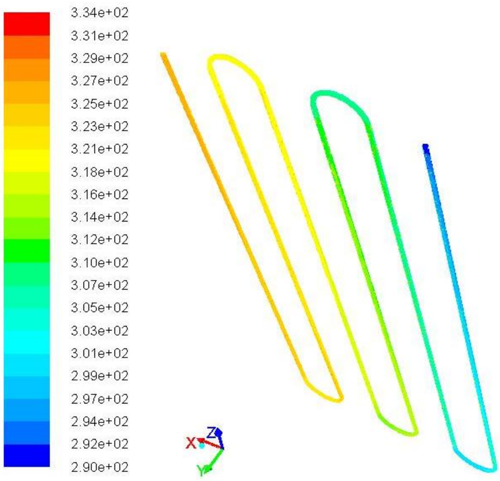

Fig. 5a temperature contours of service water for single row HX, tube A, length 8.1 m and flow rate 500 L/h

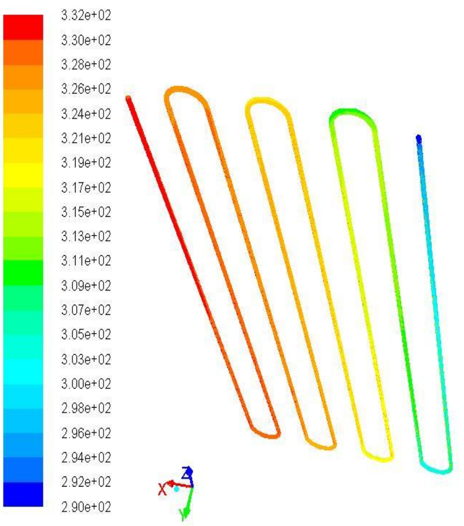

Fig. 5b temperature contours of service water for single row HX, tube A, length 10.8 m and flow rate 500 L/h

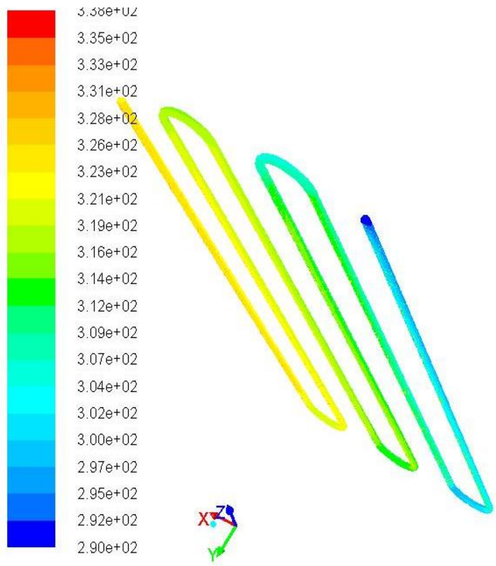

Fig. 6a temperature contours of service water for single row HX, tube B, length 8.1 m and flow rate 500 L/h

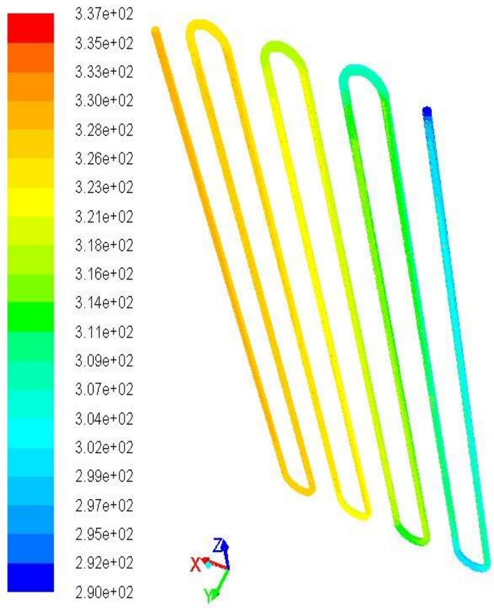

Fig. 6b temperature contours of service water for single row HX, tube B, length 10.8 m and flow rate 500 L/h

5. CONCLUSION

The power required to run the system and the initial cost can be reduced by using a single row heat exchanger (HX) rather than a double HX. For the single row HX, the larger pipe type B needed power almost 5 times lower than the smaller pipe type A. Although the initial cost of type B pipe may be slightly higher than type A, the savings in the running cost is justified with very little reduction in the final outlet temperature. After all, the main objective of using the solar heater is to reduce our impact on the environment, so reducing the energy needed to run the solar system is justified.

6. References

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