



Design, Fabrication and Testing of Shell and Tube Heat Exchanger Integrated with Vacuum Tubes Solar Water Heater

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Authors' contributions

This work was carried out in collaboration among all authors. Author KSA designed the study, performed the analysis, wrote the protocol and reviewed the work of the co-authors. Authors AA and NA wrote the first draft of the manuscript, managed the analyses of the study, perform the calculations and drawings. Authors MA and MA managed the literature survey and searches and prepared the raw materials. Author AA carried out the experimental work and set up. All authors read and approved the final manuscript.

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ABSTRACT

Heat exchanger is a device used to accomplish the transfer of heat from one fluid to another. There are a wide variety of applications regarding shell and tube heat exchangers in the fields of petroleum and industrial applications, due to its enhanced heat transfer characteristics. This project was designed to establish an insight of detailed design and performance of the shell and tube heat exchanger based on energy and mass conservation laws. Solar water heating system techniques were used to provide the system with necessary hot water. One of these techniques was to evacuate tube solar heating system which can be considered as a more efficient way to supply this system with hot water. To enhance the system performance, proper material selection for shell and tubes structure and flow pipe network based on their availability in the local markets was brought into consideration as well. Furthermore, the implemented design was examined under Medina

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climatic conditions for its cost-effectiveness, simplicity, execution and sustainability. It was found that the heat exchanger efficacy, performance and the vacuum tube efficiency were in highly acceptable ranges and cost effective. In addition, the vacuum tube solar water heating was found to be a clean and safe source of renewable energy. Finally, a comprehensive analysis of the system effectiveness was conducted and the outlet temperature determined for the system varied between 44 to 50°C for the heat exchanger whereas the vacuum tube exit temperature was elevated up to 84 to 90°C. The efficiency of the solar collector was found to be 61.84%.

Keywords: Performance; heat exchanger; vacuum tube; renewable energy; effectiveness.

Table 1. List of symbols and abbreviations

Symbol	Parameters	Unit	Symbol	Parameters	Unit
\dot{m}_h	Mass flow rate of hot fluid	Kg/s	Pr	Prandtl's Number	---
\dot{m}_c	Mass flow rate of cold fluid	Kg/s	μ	Dynamic Viscosity	N.s/m ²
T_{hi}	Hot water temperature at shell inlet	°C	A	Area	m ²
d_i	Inside tube diameter	mm	d_o	Outside tube diameter	mm
t	Tube thickness	mm	t	Tube thickness	mm
T_{ho}	Hot water temperature at shell outlet	°C	c_p	Specific heat	J/kg.K
T_{ci}	cold water temperature at tube inlet	°C	k	Thermal conductivity	W/mK
T_{co}	cold water temperature at tube outlet	°C	U	Overall heat transfer coefficient	W/m ² K
T_f	Mean film temperature	°C	LMTD	Log mean temperature difference	°C
\dot{V}	Volume flowrate	m ³ /s	P	Number of passes	---
\dot{Q}	Rate of heat transfer	W	n	Number of tubes	---
Re	Reynolds number	---	L	Tube Length	mm
Nu	Nusselt number	---	CF	Correction factor	---
ρ	Density	Kg/m ³	NTU	Number of transfer unit	--

1. INTRODUCTION

Heat Exchanger is a system that provides a heat flow between two mediums at different temperatures. In a wide range of engineering applications such as power generation, waste heat recovery, air conditioning, refrigeration and petrochemical industries, heat exchangers are used [1]. Heat exchangers are classified according to a number of criteria, including flow structure, direct and indirect touch. One of the most common types of heat exchangers is the shell and tube heat exchanger because there is a large heat transfer area, a wide temperature range and high pressure can be used. Typical applications include the heating or cooling of a concerned fluid stream and the evaporation or condensation of fluid streams with single or multi-components [2]. Fig. 1 shows the main parts of shell and tube heat exchanger which represents

one of the most common shapes and types of the heat exchanger where the fluids are separated by a heat transfer surface, so preferably they do not mix or leak [3].

As mentioned previously the function of the heat exchanger is to transfer heat between two mediums as a result of temperature differences. It is essential to supply the system with one fluid at high temperature. In order to provide heat to water up to the desired temperature, a large amount of energy is required, which is achieved by burning of fossil fuel and this is a cost and has an anti-environmental impact. Therefore, it is essential to look for alternative energy resource which is economical, sustainable and renewable for example solar energy.

The Kingdom of Saudi Arabia (KSA) extends over a large area and blessed with abundant

renewable energy resources where solar energy is the one that is the most easily accessible throughout the year. A study conducted in the past shows that the average annual solar radiation rate in Saudi Arabia is around 2300 kW/m² [4]. Evacuated tube technology is one of the promising technologies for collecting solar radiation because this collector has greater efficiencies compared with traditional flat plate solar collector, especially at low temperatures and insulations. Fig. 2 shows the schematic diagram of the evacuated tube solar collector. For selective coating, the central inner tube is painted while the outer tube is translucent. The solar radiation passes through the conduit and is absorbed by the inner tube. All internal and external tubes have limited properties for reflection of electromagnetic waves. Typically, the system is made up of a series of vacuum sealed glass tubes. These all have an outer glass shell, an inner absorber plate and a fluid-containing heat transfer pipe. In addition, the glass envelope is two glass tubes, one inside the other, and joined to create a vacuum tube around the central absorber and heat pipe at the exposed end. After processing, the glass tubes are evacuated, which eliminate the convective and conductive heat loss from the hot copper tubes to the cold ambient air at the end of their use. The sun's radiations penetrate the vacuum into the heat pipe of the copper and heats up the internal fluid to boil, causing it to rise to the bulb at the top of the tube. Therefore, the most efficient solar thermal collector is the evacuated tube due to its state of the art design, performance and efficacy [4]. Compared to a flat plate collector in high temperature applications, the output of evacuated tube solar collectors is higher. Thus, in thermal application, the evacuated tube is considered as an important component, particularly in solar water heating systems [5]. There are three common types of evacuated tube solar collectors, which are (a) water-in glass evacuated tube solar collector, (b) U-type evacuated tube solar collector and (c) evacuated tube heat pipe solar collector an evacuated tube solar collector absorbs heat from the sun through water flow inside vacuum tubes. Commonly, heat pipes use copper or aluminium tubes that are filled with porous media called wick inside the pipe, covering the inner wall of the tube at a specific thickness. The evacuated tube ones can collect the sun's energy from multiple angles. This is possible because of its uniquely crafted 360° tubular design. When the sun is at lower angles, it is almost impossible for typical solar collectors to trap the solar energy.

This is not the case with evacuated solar collectors. They are built with a broader collector area so as to allow maximum utilization of the sun's energy [6].

This technology is the vacuum tube solar water heating system which can be considered as the best source of energy because renewable energy is a clean, safe and eco-friendly. Moreover, this technology is coherent to the KSA vision 2030 which explains that the focus is on the renewable energy as an alternative future energy resources to replace the dependency on the fossil fuels. Heat exchangers design and application and the evacuated tube solar water heating system has been studied and conducted by many researchers. Lunsford, 1998 [7] investigated the enhancement of heat exchanger performance in the logical and acceptable steps: The first step is initially operating of the heat exchanger. The second step considers increasing pressure drop if available in exchangers with single-phase heat transfer. Durgesh Bhatt, Priyanka M Javhar, 2012 [8] conducted a shell and tube heat exchanger performance analysis and found that by changing the value of one variable and keeping the other variables as constant, new results are obtained. They optimized the design of shell and tube type heat exchanger based on these results. When the thermal conductivity of the tube metallurgy is higher, the greater heat transfer rate will be achieved. Less is the baffle spacing, more is the shell side passes, higher the heat transfers but at the cost of the pressure drop. Dawit Bogale, 2014 [9] conducted an experiment on shell and tube heat exchangers for both mechanical and thermal designs and the simulation for the heat transfer between the two fluids. Vindhya and Raj, 2014 [10] conducted a performance analysis of shell and tube type heat exchanger under the effect of varied operating conditions. They concluded that the insulation is a good tool to increase the rate of heat transfer if used properly. The cotton wool and the tape have been given the best values of effectiveness. Moreover, the effectiveness of the heat exchanger also depends upon the value of turbulence provided. The ambient conditions for which the heat exchanger was tested do not show any significant effect over the heat exchanger's performance. Gaddis and Gnielinski, 1997 [11] carried out an experimental and theoretical study for evaluating the shell side pressure drop in shell-and-tube heat exchangers with segmental baffles. The obtained results for shell side pressure drop were compared and calculated

with the help of computational procedure. Baghban et al. [12] studied the thermal behaviour of the shell side flow of a shell and tube heat exchanger. The experimental method provided the effect of the major parameters of the shell side flow on thermal energy exchange and in the numerical method, in addition to the effect of the major parameters, the effect of different geometric parameters such as Reynolds number (Re) on thermal energy exchange in shell side flow has been considered. Hosseini et al. [13] performed an experimental analysis of heat transfer coefficient and pressure drop on the shell side of a shell and tube heat exchanger. Three different types of copper tubes (smooth, corrugated and with micro-fins) our analysis and compared theoretical data available. It was found that higher Nusselt number and pressure drops with respect to theoretical correlations. Durgesh Rai [14] analysed the effect of tube bundle geometry in shell and tube heat exchanger. It was found that the orientation of the tube layout had a significant influence on tube side pressure drop and heat transfer of the heat exchangers. Also, the increase in heat exchanged by 11.04% can be seen, decrease in pressure drop in the tube side found to be by 29.59%. Decrease in heat transfer coefficient in tube side by 11.2%. Kiran et al. [15] carried out a finite element analysis on 125KVA diesel generator which loses the exhaust gases at a temperature of 350°C. Waste heat was recovered by replacing the silencer of the power plant by a heat exchanger. Fluid flow in the heat exchanger was considered as a fluid dynamic problem and was modelled

using finite element methods, the reported result show that about 72% of heat is recovered and 28% heat is lost to the atmospheres with the help of finite element analysis (FEA). Tengyue et al. [16] conducted a design of two types of solar air collectors, called, a conventional-tube collector and a transparent-tube collector, the design based the flat micro-heat pipe arrays and they are compared well with previous work. The reported results can be utilized as a reference for the selection and design of solar air collectors for heating applications and drying process like crop drying. Teng. Wang. Y. [17] carried out and experimental and numerical investigation to design a solar air collector with transparent-vacuum glass tube by using selective absorption film and transparent-vacuum glass tube. The display results provide a theoretical guidance for designing solar air collector. Yuchen. Bai et al. [18] studied and developed a novel solar thermal storage heating system filled with phase-change material (PCM) and coupled with finned heat pipe to improve the heat dissipation process. Hussein M et al, 2020 [19] conducted the effect of the inclination angle (θ) of a shell and coiled tube heat exchanger on the performance of the heat exchanger using water as a working fluid. They proposed empirical correlations to estimate the coil Nusselt number (Nu_c). From the previous literature survey, it can be seen that very little work has been conducted the utilization of solar water heating system by evacuated tube in Saudi Arabia which can be considered as motivation for this work.

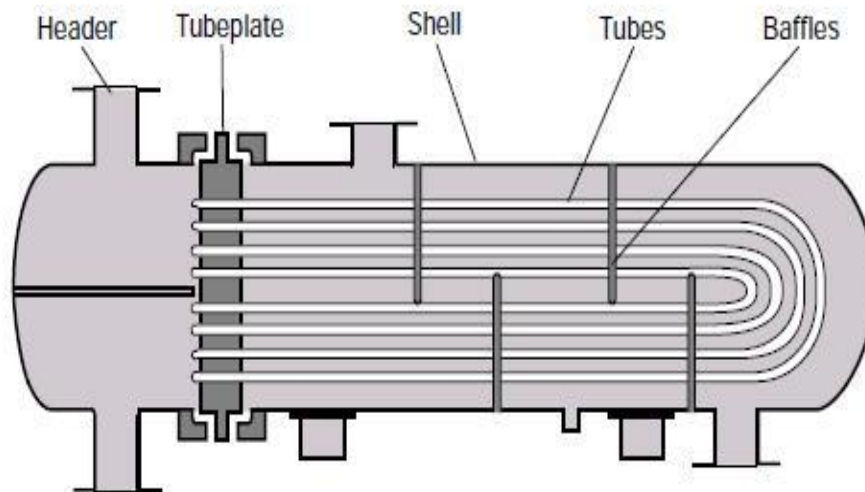


Fig. 1. Main components of a shell and tube heat exchanger

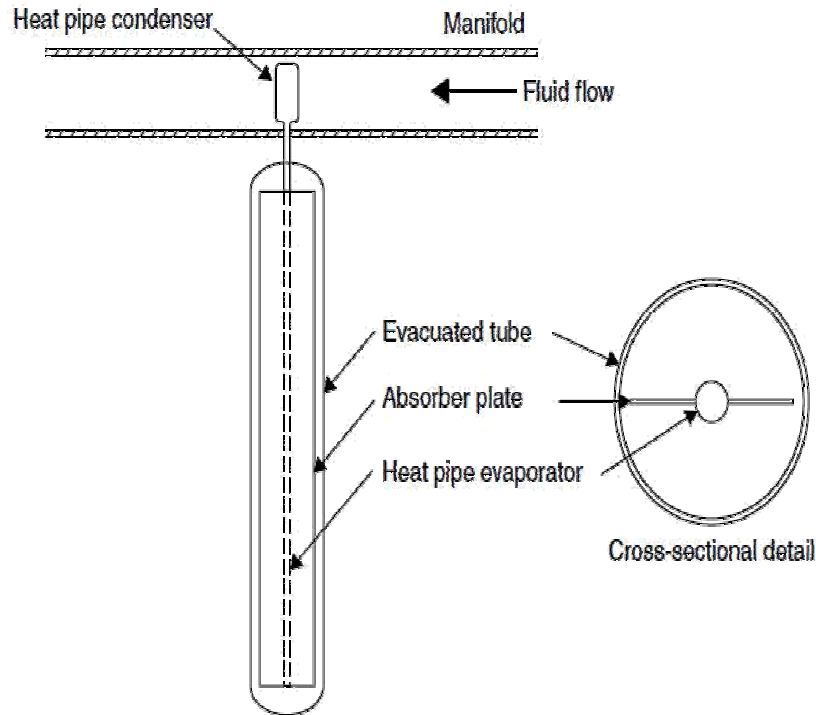


Fig. 2. Schematic diagram of vacuum tube solar collector [4]

2. MATERIALS AND METHODS

The optimal design of any engineering system requires the availability of a set of factors and conditions, the most important of which are safety, high efficiency, reliability, quiet performance, ease of movement and transportation, availability of basic energy sources in addition to environmental aspects, and that it is not in any way a source of pollution and this is what was taken into account in this work. Fig. 3 shows the proposed design for the shell and tube heat exchanger integrated with vacuum tube solar water heating system. The system consists of vacuum tubes solar collector including heat pipe which contains the phase change material, the storage tank to provide with necessary amount of hot water, shell and tubes coupled with hot and cold water sources, circulation pumps, flow control system like valves, flow meter and other necessary fittings for the network connections. Before the calculations and design, the first step is to identify the system capacity, heat source, cold

water source, volume flow rates and temperature of cold water, desired output temperature and the basic materials for each component. Based on the availability of the above specifications, the design and calculations will be performed in the coming section.

The suggested and recommended materials for the shell is the galvanized steel because galvanized steel will its reliability and availability and ability to resist corrosion. The tube was made of copper because it has a high thermal conductivity than other metals as well as light weight and easy to be fabricated in addition to its fouling resistance. The rock wool was used as thermal insulation material to maintain the amount of heat transmitted into the shell, also to ensure no heat gain or loss from outside. The gasket was made of rubber to prevent flanges leakage. For easy movement and maintenance issues, the shell sides welded with steel flanges and can be connected to the water sources by using steel bolts and nuts.

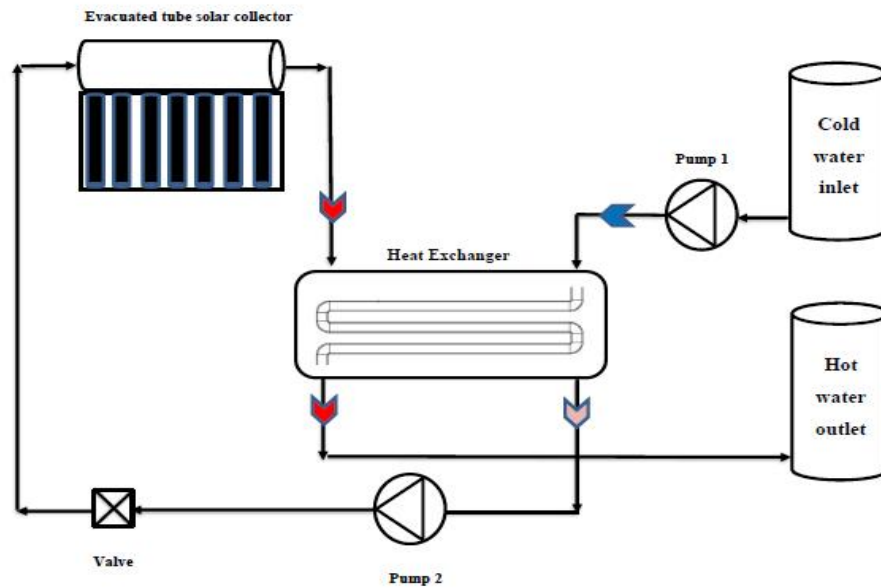


Fig. 3. Suggested design of shell and tube heat exchanger integrated with vacuum tube solar collector

3. CALCULATIONS AND DESIGN

3.1 Heat Exchanger Calculations and Design

Based on the available spaces and needs, the mass flowrate for hot and cold water assumed to be as follow: The volume flow rates for hot water is 15 L/min and for cold water 20 L/min.

$$\dot{V}_h = 15 \frac{\text{L}}{\text{min}} \text{ for hot water} \quad , \quad \dot{V}_c = 20 \frac{\text{L}}{\text{min}} \text{ for cold water} \quad , \quad \rho = 1000 \frac{\text{kg}}{\text{m}^3}$$

$$\dot{m}_h = \dot{V}_h \rho = \frac{15}{60} \times 10^{-3} \times 1000 = 0.25 \frac{\text{kg}}{\text{s}}$$

$$\dot{m}_c = 20 \frac{\text{L}}{\text{min}} = \frac{20}{60} \times 10^{-3} \times 1000 = 0.33 \frac{\text{kg}}{\text{s}}$$

The inlet and outlet temperature for hot fluid and the inlet cold fluid assumed to be $T_{hi} = 80 \text{ }^\circ\text{C}$, $T_{ho} = 50 \text{ }^\circ\text{C}$, $T_{ci} = 25 \text{ }^\circ\text{C}$

$$Q_h = \dot{m}_h C_p h [T_{hi} - T_{ho}] \tag{1}$$

$$Q_h = 0.25 \times 4178 \times [80 - 50] = 31335 \text{ W. Performing energy balance for both hot and cold fluids:}$$

$$Q_h = Q_c$$

$$Q_c = \dot{m}_c C_p c [T_{co} - T_{ci}] \tag{2}$$

$$31335 = 0.33 \times 4178 \times [T_{co} - 25] \quad , \quad T_{co} = 47.727 \text{ }^\circ\text{C} = 48 \text{ }^\circ\text{C}$$

$$T_f = \frac{T_{co} + T_{ci}}{2} = \frac{48 + 25}{2} = 36.5 \text{ }^\circ\text{C} = 309.5 \text{ K}$$

Properties of water from Thermodynamics table at $T_f = 36.5\text{ }^\circ\text{C} = 309.5\text{K}$ [20]

$$C_p = 4.18 \text{ kJ/kgK}, \quad k_{fluid} = 0.628 \frac{W}{mK}$$

$$Pr = 4.62, \quad \mu = 0.000695 \text{ N.s/m}^2 \text{ [20]}$$

The material selection for the tube depending on the availability of them on the market, the selected material for the tube is a copper, therefore,

$d_o = 27.5 \text{ mm}, d_i = 25 \text{ mm}, t = 2.5 \text{ mm}$ and the Thermal Conductivity of the copper

$$k_{copper} = 401 \text{ W/mk}$$

$$\text{Then, } A = \pi \times d_i \times n \times P \times L = \pi \times 0.025 \times 1 \times 3 \times L$$

Where, P= number of passes and n= number of tubes

LMTD: is a logarithmic average of the temperature difference between the hot and cold streams at each end of the exchanger. The larger the LMTD, the more heat is transferred. The use of the LMTD arises, straight forwardly, from the analysis of a heat exchanger with constant flow rate and fluid thermal properties [20].

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}} \text{ [20]}$$

The LMTD for Shell & tube heat exchanger can be estimated by using the following formula

$$LMTD_{shell} = LMTD \times CF \tag{3}$$

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}}, \quad \text{where } \Delta T_1 = Th_i - Tco, \quad \Delta T_2 = Th_o - Tci$$

$$LMTD = 28.35\text{ }^\circ\text{C}$$

To find the shell and tube correction factors as shown in Fig. 4, The shell and tube can be modelled as a counter flow heat exchanger with correction factor as shown in Fig. 4 [20], the correction factors found from the following relations P and R displayed in Fig. 4.

$$P = \frac{t_2 - t_1}{T_1 - t_1} \quad \text{and} \quad R = \frac{T_1 - T_2}{t_2 - t_1}$$

$$P = \frac{48 - 25}{80 - 25} = 0.41, \quad R = \frac{80 - 50}{48 - 25} = 1.3$$

Then the correction factor = 0.86

$$\text{Then the LMTD} = 28.35 \times 0.86 = 24.09\text{ }^\circ\text{C}$$

The Reynolds numbers (Re) can be estimated using the following formula:

$$Re = \frac{\rho \times D \times \dot{m}}{\mu \times \rho \times \frac{\pi}{4} D^2} = \frac{4 \times \dot{m}_C}{\pi \times D \times \mu} \tag{4}$$

$$Re = \frac{4 \times 0.33}{\pi \times 0.025 \times 0.000695}$$

$Re = 24182.4$ so, the flow is turbulent

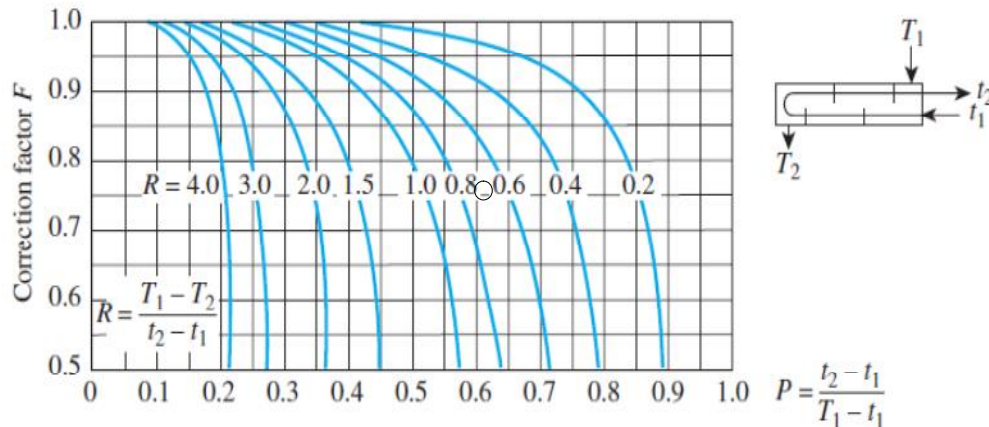


Fig. 4. Correction factors for shell and tube heat exchanger [20]

By using Dittus Boelter equation [20].

$$Nu_D = 0.023 Re^{0.8} Pr^n \tag{5}$$

for heating $0.4 = n$

$$Nu_D = 0.023 \times Re^{0.8} \times Pr^{0.4} = 0.023 \times 24182.4^{0.8} \times 4.62^{0.4} = 136.26$$

and heat transfer coefficient from $Nu_D = \frac{h_i \times D}{K_f}$

$$so, h_i = \frac{136.26 \times 0.025}{0.025} = 3422.97 \frac{W}{m^2 K}$$

Take $h_o = 4000 w/m^2 k$ for hot fluid

$$R_{conv 1} = \frac{d_o}{d_i \times h_i}$$

$$R_{cond} = \frac{d_i}{2 \times K} \times \ln \frac{d_o}{d_i}$$

$$R_{conv 2} = \frac{1}{h_o}$$

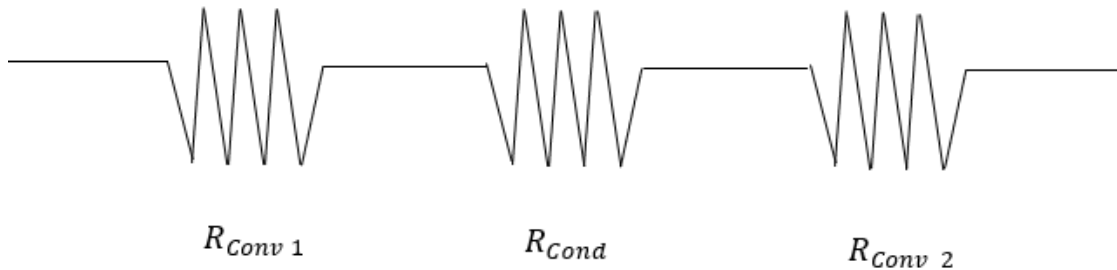


Fig. 5. Thermal resistance for inside tubes

The overall heat transfer coefficient can be estimated as it clears in Fig. 5 above

$$U = \frac{1}{\frac{d_o}{d_i \times h_i} + \frac{d_i}{2 \times K} \times \ln \frac{d_o}{d_i} + \frac{1}{h_o}} = \frac{1}{\frac{0.027}{0.025 \times 3422.97} + \frac{0.025}{2 \times 401} \times \ln \frac{0.027}{0.025} + \frac{1}{400}}$$

$$U = 1760.28 \frac{W}{m^2K}$$

$$Q = U \times A \times \text{LMTD} \tag{6}$$

$$31335 = 1760.28 \times \pi \times 0.025 \times L \times 1 \times 3 \times 24.09$$

L = 3.136 m = 3136.17 mm which represents the total tube length.

The total length for the inside tube found to be 3.136 m and can be divided to 3 passes based on the available area inside the shell. So, we need one tube of length 3.136 m with 3 passes each pass 1.04 m.

To estimate the performance of the heat exchanger, the effectiveness of heat exchanger can be found from the following equation

$$\varepsilon = \frac{Q_{actual}}{Q_{max}} = \frac{C_h(T_{hi}-T_{ho})}{C_{min}(T_{hi}-T_{co})} \tag{7}$$

$$C_h = \dot{m}_h c p_h = (0.25) (4178) = 1044.5 \text{ W/K}$$

$$C_c = \dot{m}_c c p_c = (0.33) (4178) = 1378.74 \text{ W/K}$$

Then $C_{min} = 1044.5 \text{ W/K}$, then the effectiveness

$$\varepsilon = \frac{Q_{actual}}{Q_{max}} = \frac{C_h(T_{hi}-T_{ho})}{C_{min}(T_{hi}-T_{co})} = \frac{1044.5(80-50)}{1044.5(80-48)} = 0.9375$$

The second heat exchanger performance indicator is the number of transfer unit

(NTU)

$$NTU = \frac{UA}{C_{min}} = \frac{(U)(\pi \times D \times N \times L \times P)}{C_{min}} \tag{8}$$

$$NTU = \frac{(1760.28)(\pi \times 0.025 \times 1 \times 3.136 \times 3)}{1044.5} = 1.245$$

The shell and tube heat exchanger perform its function by circulating a hot water around tubes that contain a cold water. The hot water circulates in an enclosed area called the shell. Tubes containing the cold water are looped through the shell. Heat transfer from the hot water in the shell to the cold water inside the tubes, as a result of heat transfer from the hot to the cold water as a results of temperature difference Fig. 6 shows the final shape of the copper tubes arrangement in shell and tube exchanger. A gasket must be inserted between the shell and the cover to prevent the leakage. Also, the Thermocouples were used to measure both hot and cold water inlet and outlet temperatures.

The shell outside also must be insulated to prevent heat transfer to the environment. The selected outside dimension for the shell is 480 mm inside 380 mm and the total length 1040 mm. While the inner tube includes three passes and two elbows with inner diameter of 40 mm and total length of 920 mm. The fillet connects between shell and tube surfaces has been properly welded to withstand the pressure differences and other loads Although, the whole system must be insulated from the outside to reduce temperature drop through night and cold days.

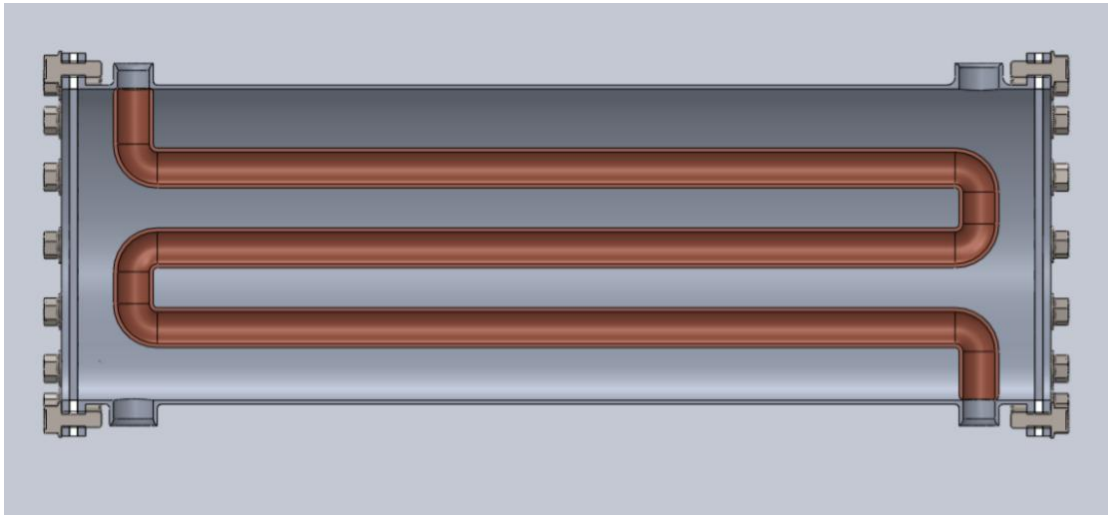


Fig. 6. Shell and tube heat exchanger arrangement with 3 passes

3.2 Solar Radiation Calculations

Using the basic solar thermal radiation equations the following parameters were calculated for Medina where the Latitude angle = $24.55^\circ N$ and Longitude angle = $39.70^\circ E$ based on the selected days and months [21]:

$$\text{Declination angle } \delta = 23.45 \sin\left(\frac{360 \times (248 + 106)}{365}\right) = 23.0387^\circ \text{ where } n = 106, \text{ at 15 February}$$

$$\text{Hour angle } \omega = 0^\circ \text{ at 12:00 AM}$$

$$\text{Inclination angle } \alpha = \sin^{-1}(\cos(24.55) \times \cos(23.0387) \times \cos(0) + \sin(24.55) \sin(23.0387)) = 88.488^\circ$$

$$\text{Zenith angle, } \theta_z = \frac{\pi}{2} - 88.488 = -86.917^\circ$$

$$\text{Incident angle, } \theta_i$$

$$\cos \theta_i = \cos(24.55 - 24.5) \times \cos(23.0387) \times \cos(0) + \sin(23.0387) \times \sin(24.55 - 24.5)$$

$$\theta_i = 22.988^\circ$$

$$\text{Tilt angle } \beta = 1 + (0.95 \times 24.55) = 24.5^\circ \text{ [21]}$$

$$\text{Tilt beam factor, } r_b \text{ [21]}$$

$$r_b = \frac{\sin(23.0387) \sin(24.55 - 24.5) + \cos(23.0387) \cos(0) \cos(24.55 - 24.5)}{\sin(23.0387) \sin(24.55) + \cos(23.0387) \cos(0) \cos(24.55)} = 0.921$$

$$\text{Tilt diffuse factor } r_d = \frac{1 + \cos(24.5)}{2} = 0.9549$$

$$\text{Tilt reflected factor, } r_r = 0.6 \left(\frac{1 - \cos(24.5)}{2} \right) = 0.0270$$

$$\text{Direct radiation from the sun } I_{DN} = 1215 * e^{-\frac{0.144}{\sin 88.488}}, \quad I_{DN} = 1052 \text{ w/m}^2$$

$$\text{Diffuse radiation from the sky } I_d = 0.06 \times 1052 \times 0.9549, \quad I_d = 60.273 \text{ w/m}^2$$

$$\text{Solar flux incident } I_T = I_{DN} \cos \theta_i + I_d r_d + (I_{DN} + I_d) r_r \tag{9}$$

$$I_T = 1052 \cos(22.988) + 60.273(0.9549) + (1052 + 60.273)(0.0270)$$

$$I_T = 1056.0432 \text{ w/m}^2$$

To calculate the incident of solar flux, S absorbed by the absorber plate

$$S = I_{DN} r_b (\tau\alpha)_b + [I_d r_d + (I_{DN} + I_d) r_r] (\tau\alpha)_a \tag{10}$$

$$S = 1052 * 0.921 * 0.6422 + [60.273 * 0.9549 + (1052 + 60.2730) * 0.0270] * 0.54$$

$$S = 669.52 \text{ W/m}^2$$

3.3 Energy Balance on Vacuum Tube Solar Collector

To evaluate the efficiency of solar collector, it is necessary to estimate the system input and output and the main losses in the system by

performing energy balance and indicate the main solar collector losses. Fig. 7 shows the heat transfer process through the solar collector and the main sources of energy losses. Measurement of the total intensity of soalr radiation strike the collector will assist the evaluation process.

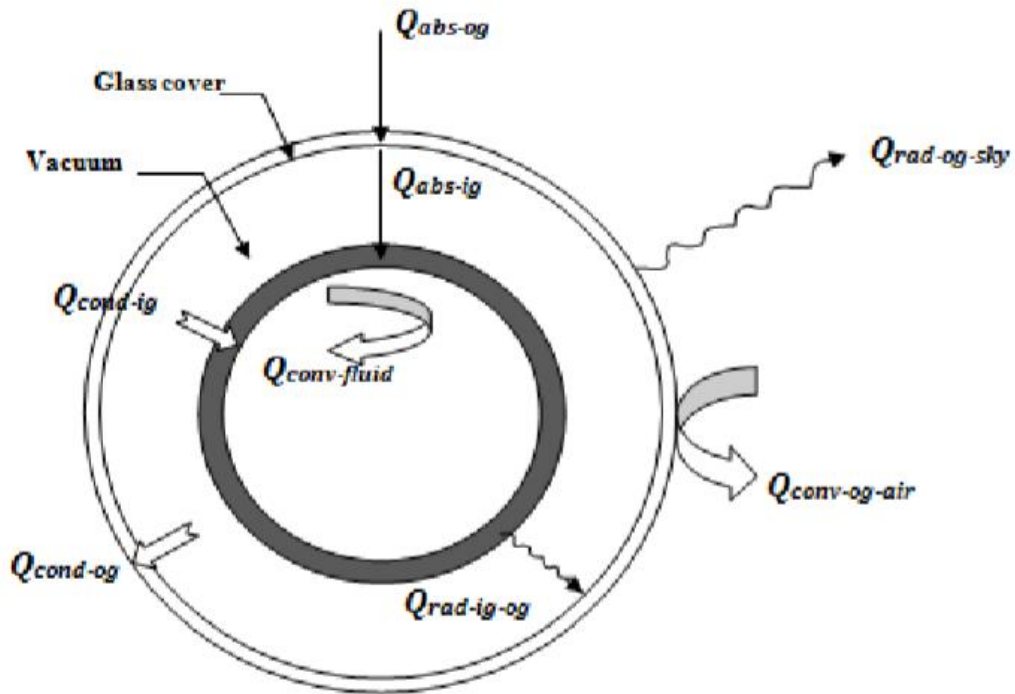


Fig. 7. Heat transfer process in vacuum tube solar collector [21]

Total beam radiation from the previous section can be calculated using the main equations [21]

$$I_{total} = 1056.0432 \frac{W}{m^2}$$

$$Q_{loss} = U_L \times A_P \times (T_{avg} - T_{ambient}) \tag{11}$$

$$Q_{loss} = 0.6 \times 2.2 \times (52.5 - 25)$$

$$Q_{loss} = 36.3 \text{ w}$$

$$Q_{useful} = S * A_p - Q_{loss} \tag{12}$$

$$= 669.52 \times 2.2 - 36.3 = 1436.64 \text{ w}$$

$$\eta_{solar \ collector} = \frac{Q_{useful}}{I_{total} * A_p} * 100\%. \tag{13}$$

$$\eta_{solar \ collector} = \frac{1436.64}{1056.0432 * 2.2} * 100\% = 61.84\%$$

The energy balance on the solar collector and the losses are shown in Fig. 7

$$E_{in} = E_{out}$$

$$I_{total} = Q_{useful} - Q_{loss}$$

$$I_{total} = 1056.0432 \frac{W}{m^2} = 3801.7556 \text{ kJ/m}^2$$

$$Q_{useful} = \dot{m} \times C_p \times \Delta T = 4.8 \times 10^{-3} \times 4178 \times [80 - 25]$$

$$Q_{useful} = 1122.5 \frac{W}{m^2} = 4041.6$$

$$Q_{loss} = U_L \times A_p \times (T_{avg} - T_{ambient}) \tag{14}$$

Fig. 7 shows the energy balance performed on the vacuum tube

$$Q_{loss} = 1.1 \times 2.2 \times (52.5 - 25) = 66.5 \frac{W}{m^2} = 239.85 \text{ kJ/m}^2$$

The overall heat transfer coefficient for the vacuum tube can be estimated using the following relation [20]

$$U_L \frac{1}{\left(\frac{1}{h_{rad,og} + h_{wb}}\right) + \frac{1}{h_{rad,ig}}} \tag{15}$$

Where, $h_{rad,og}$: is the radiation heat transfer coefficient of the outer glass tube.

h_{wb} : is the convection heat transfer coefficient of the outer glass tube.

$h_{rad,ig}$ is the radiation heat transfer coefficient of the inner glass tube?

$$h_{rad,og} = \varepsilon \times \sigma \times \frac{T_{og}^4 - T_a^4}{T_{og} - T_a} \tag{16}$$

Where, ε : is the emissivity of the outer glass.

σ : is the Stefan –Boltzmann Constant (5.6697×10^{-8}) $\frac{W}{m^2.K^4}$ [20]

T_{og} : is the temperature of the outer glass tube in K.

T_a : is the ambient temperature, K

T_s : is the equivalent sky temperature as a function of the ambient air temperature, K.

$$T_s = 0.0522 \times T_a^{1.5} \quad (17)$$

$$T_s = 0.0522 \times 298^{1.5} = 268.5 \text{ K}$$

$$h_{rad,og} = 0.92 \times 5.6697 \times 10^{-8} \times \frac{353^4 - 268^4}{353 - 268} = 9.796 \frac{W}{m^2}$$

$$h_w = 5.7 + 3.8 \times v_\infty \quad (18)$$

where, v_∞ is the wind speed m/s [18]

$$h_w = 5.7 + 3.8(2.77778) = 16.255555 \frac{W}{m^2}, \text{ where, } v_\infty = 10 \text{ km/hr at February}$$

$$h_{wb} = \frac{A_{og}}{A_{ig}} \times 0.6 \times h_w \quad (19)$$

$$h_{wb} = \frac{0.002642}{0.002498} \times 0.6 \times 16.25555 = 10.3155 \frac{W}{m^2}$$

$$h_{rad,ig} = \varepsilon_{ig-og} \times \sigma \times \frac{T_{ig}^4 - T_{og}^4}{T_{ig} - T_{og}} \quad (20)$$

$$\varepsilon_{ig-og} = \frac{1}{\frac{1}{\varepsilon} + \frac{A_{ig}}{A_{og}} \left(\frac{1}{A_{og}} - 1 \right)} = 0.00279$$

$$h_{rad,ig} = 0.00279 \times 5.6697 \times 10^{-8} \times \frac{343^4 - 353^4}{343 - 353} = 0.026672 \frac{W}{m^2}$$

$$U_L = \frac{1}{\left(\frac{1}{h_{rad,og} + h_{wb}} \right) + \frac{1}{h_{rad,ig}}} = \frac{1}{\left(\frac{1}{9.796 + 10.3155} \right) + \frac{1}{0.026672}}$$

$$U_L = 0.0266 \text{ W/m}^2\text{K}$$

3.4 System Performance

The measurement of the intensity of solar radiation falling on the surface of the solar collector over a period of 24 hours and as it evident in Fig. 8 which confirmed that the solar energy is a promising energy source in this region. Also, it can be effectively exploited for power generation and other applications. This figure shows that in spite of the presence of many losses in the solar system, the amount of radiation contributed to achieve the desired temperature through the heat exchanger. From the Fig. 8 it appears that the amount of radiation exceeded 860 W/m^2 at the mid-day which highlights the importance of focusing on solar

energy as an important source of renewable energy in the region, in addition to that it is clean, safe energy and has no impacts on the environment.

Fig. 9 shows the temperature distribution along the shell and tube heat exchanger by considering the shell and tube heat exchanger as counter flow with correction factor and the vacuum tube output as input heat source to the system. The exit water temperature from the vacuum tube is hot water inlet to the heat exchanger varied between 90 to 84°C and the cold water leaving the heat exchanger at about 44°C in one of the recorded results.

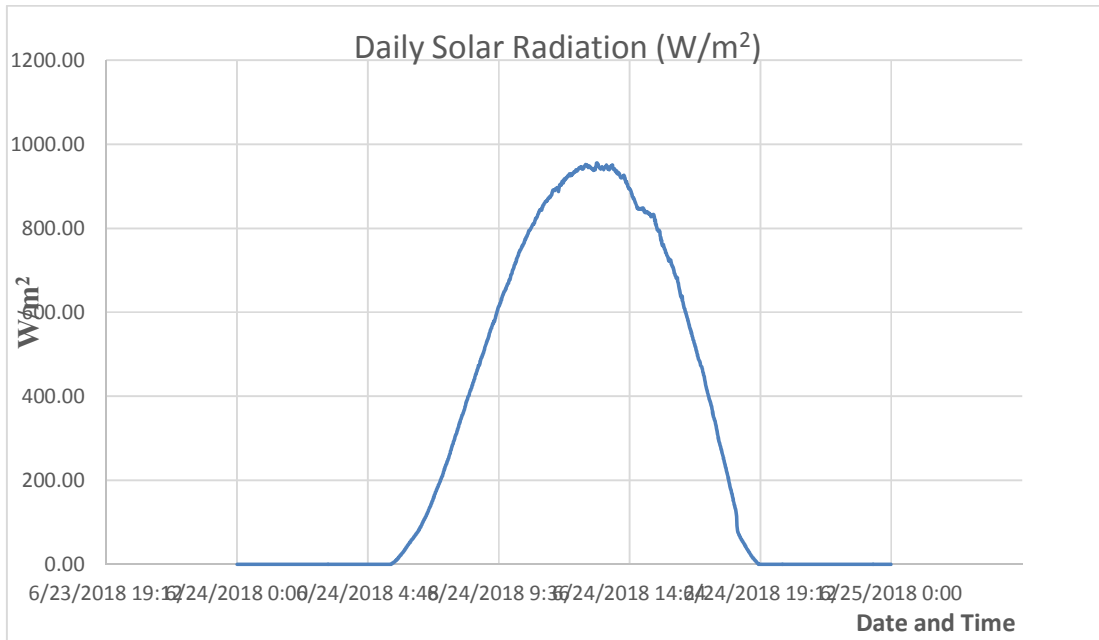


Fig. 8. Variation of solar radiation falls on the solar collector along 24 hour

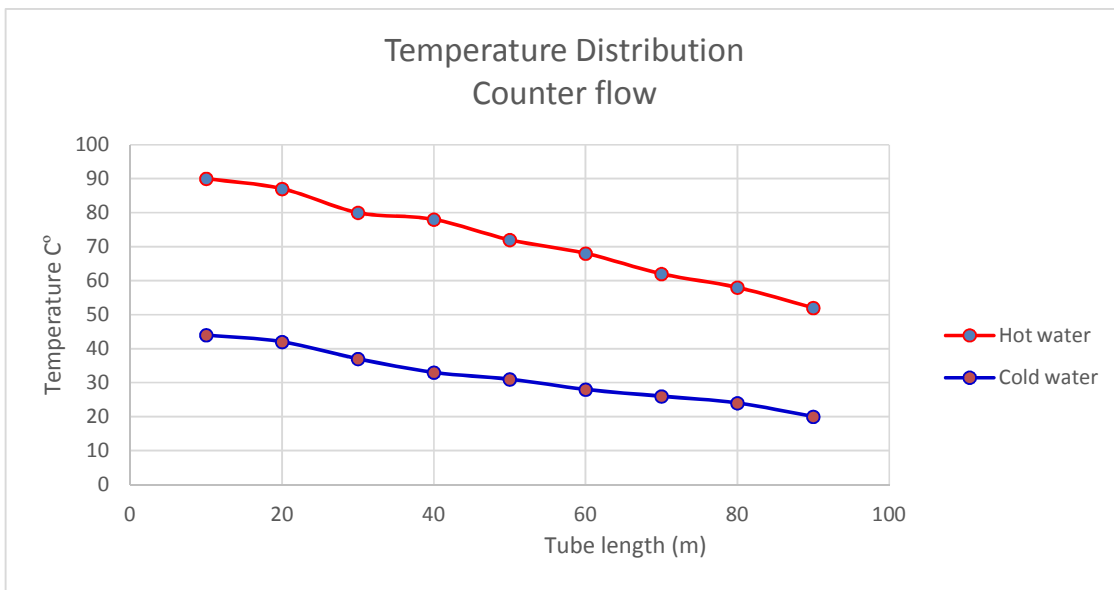


Fig. 9. Temperature distribution along the shell and tube heat exchanger

The same trend can also be seen in Fig. 10, where the variation in temperature along the heat exchanger ranged from 86°C to 52 °C degrees at the shell exit. This variation can also be observed in the tube side where the temperature ranged from 20°C at the tube inlet entry to 44°C at the tube exit, and

this proves the well performance for the heat exchanger. It was found that a good agreement between the system performance compared with the available previous works reported the literature like Mani et al and Dileep et al under same flow arrangement and materials.

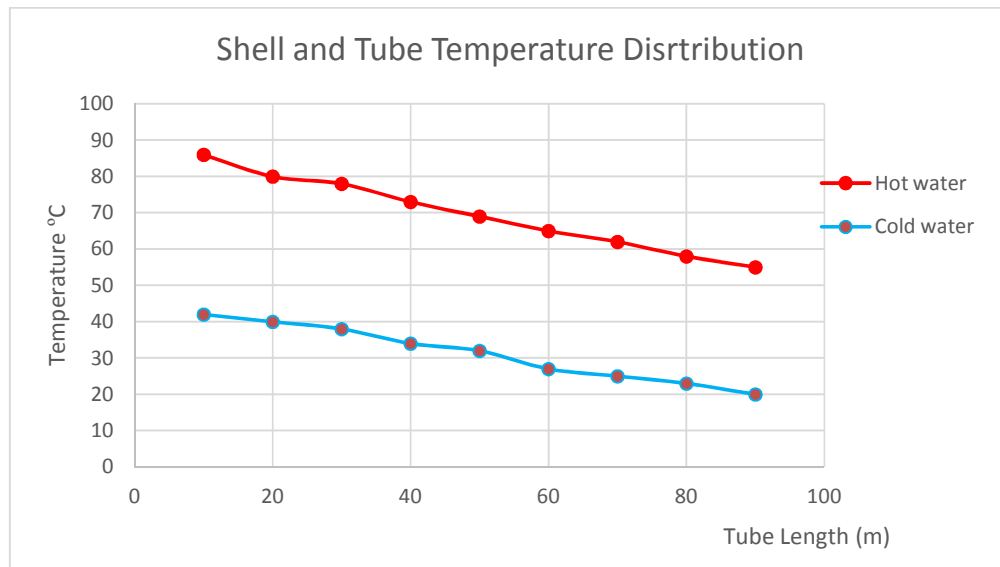


Fig. 10. Temperature distribution along the shell and tube heat exchanger

It can be seen that the overall heat transfer coefficient calculated for the vacuum tube is too small compared with other solar collector because the function of vacuum is to eliminate the effect of convection and conduction heat transfer and only radiation heat transfers prevailing in this type of solar collector which makes it more efficient to supply hot water at high temperatures. It was recommending to use helical coil tube instead of U-Tube to enhance the system performance because this option leads to an increase in the surface area.

4. CONCLUSION

Heat exchangers are widely used in industries both for cooling and heating large scale industrial processes by facilitate the exchange of heat between two fluids that are at different temperatures. In this work the design and fabrication of shell and tube heat exchanger has been conducted based on the heat transfer basic knowledge and energy balance equations. SolidWorks software has been used to design all the components. The evacuated tube solar water heating system was being utilized to provide the system with the necessary amount of hot water because using a renewable energy source is cheap, clean, safe and environment friendly. The comprehensive analysis of the integrated system has been conducted as well as effectiveness and the system efficiency. The outlet temperature for the system varied between 44 to 50°C for the heat exchanger where the vacuum tube exit

temperature about 84 to 90°C. The efficiency of the solar collector found to be 81.84% and the total calculated intensity of solar radiation is 1056.0432 W/m² whereas the measured value is 860 W/m² and heat exchanger effectiveness is 0.93 and the number of transfer unit 1.245.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

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