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CFD Simulation of Weather Condition Effect on The Performance of Dual Purpose Solar Collector

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Abstract. In this research, study the performance of dual purpose solar thermal collector numerically simulation was done. The study included studying a new type of absorber plate and water pipes distribution. The simulation part of this research is done by using COMSOL Multiphysics program version 5.5. The new type was corrugated absorber plate with (2*1) m dimensions and zig-zag water pipes with (0.01) m diameter passage through the absorber plate. The effect of solar irradiance, ambient temperature, wind speed, water volumetric flow and inlet water temperature on the performance of the dual purpose solar thermal collector was studied. The numerical simulation results that the maximum error between present work and the previous experimental work was 2.4%., and also that when the solar irradiance, inlet water temperature and ambient temperature increases the outlet temperature increases, but it is decreases with the wind speed.

Keywords: dual purpose solar collector, COMSOL Multiphysics, Absorber plate.

1. INTRODUCTION

Nowadays, the world is very much relying on fossil fuels, gas, petrol and Charcoal, meeting the energy needs to reach about 80 per cent of global energy demand, whereas wind, hydro, biomass and geothermal are renewable energies. Energy is a major ingredient driving economic growth and socio-economic development. Power is an important component in all contemporary aspects and is therefore the live wire in conventional thermal power plants for industrial and agricultural production, transport fuel and power generation. The alternative use of fossil fuels is the exploitation of renewable energy [1][2].with thermal collectors solar energy, it is known to be more renewable energy. These collectors are most common form of application for solar energy [3]. The most popular solar collector for solar water heating systems in homes and solar space heating is the flat-plate collectors [4]. Several studies, experimental and numerical, have studied this type of solar collector to enhancement the thermal efficiency[5]. The CFD simulation have been a considerable benefit to researchers. CFD's potential to shorten lead times, test device under unsafe conditions and research regulated device tests that are difficult or impractical to carry out in practice are some of the main advantages [6]. O. Mahfoud et al. [7] study the heat transfer and air flow in solar collector with obstacle numerically. ANSYS program was used to building and analyse the model. This paper discuss the study of hydrodynamic and thermal air lows around artificial ruggedness and the inlet mass flow rates affect the thermal transfer flow structure and heat transfer degree. The study

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concluded that when the Reynold number increase (between 100 and 4500), pressure drop very important.. M. A. Amraoui and K. Aliane [8] presented a numerical simulation to study the heat transfer and fluid flow in solar air flat plate collector by using computational fluid dynamics (CFD) by ANSYS program. They add modelling baffles to find structure of a meandering form. There is a good compromise between the experimental and the simulated outlet air temperature results. Beate Vetter et al. [9] presented a numerical study to development and optimization method for flat plate collectors is provided based on CFD simulations and simulated thermal efficiency tests. The simulation is done by using STAR CCM+ program. The results showed that the peak collector output of the simulated thermal performance test matches very well with the findings from the experimental thermal performance test. Tzu-Chen Hung et al. [10] Experimentally and numerically investigated thermal performance of flat plate solar air collector. Three kinds of glasses and a few aspect ratios flow channels were compared to the conclusion of optimal configuration. The result showed that the combination of a 50 mm aspect ratio is dominating the optimal design: 10 mm, and suitable mass flow rate to device height.

M.S. Manjunath et al. [11] numerically investigated the effect of sinusoidal profiled absorber plate on the performance of solar air heaters. The effect on thermo-hydraulic performance of variation of design parameters of the corrugated absorber plate. The aspect ratio ranges from 1.5 to 4.0. The results indicated that the design with an aspect ratio and non-dimensional wavelength of 2.0 indicates better thermo-hydraulic output at lower mass flow rate conditions and has an acceptable maximum efficiency of about 64%. A. A. Hawwash et al. [11] Carry out an experimental and numerical analyses to study thermal performance of solar water collector with different types of working fluid. The numeric model is designed by using the software ANSYS 17. The numerical results showed that increasing the percentage of the Alumina Nano fluid improves the thermal efficiency of the FPCS up to a volume fraction of 0.5%, and any further increase has a negative effect on the thermal performance. Liqun Zhou et al. [12] Presented a numerical study to investigated analyses effect of flow rate, ambient temperature, tilt angle and transparent insulation materials (TIM) on collector performance. Collector efficiency with TIM is 6.2 % higher than traditional collector efficiency. With the traditional flat plate solar collector, the tilt angle effect is greater significant relative to that on a TIM collector.

I. N. Unar et al. [13] carry out a numerical study of the effect of geometric and operating parameters on thermal efficiency by using CFD simulation by ANSYS FLUENT 14.0. Two fluids were exchanged. The findings of the experiment showed that water and air temperatures were risen by approximately 79 ° C and 73 ° C respectively. Z. Badiei et al. [14] performance of flat plate solar collector enhancement by using phase change materials was investigated numerically. To investigate a solar flat plate collector integrated with a PCM layer, three dimensional transient CFD model is created. The fins are also integrated into the PCM and the resulting temperature distribution over two separate summer and winter days is analysed. Results showed that although the PCM system has lower morning output temperatures, during discharge, hot water can be supplied in the evening at a longer duration. The aim of this study is to design dual purpose solar collector with new type of the absorber plate and pipes distribution and simulated by using CFD analyses with COMSOL Multiphysics 5.5 program.

2. MATHEMATICAL MODELING

2.1. Problem definition

Figure.1 illustrates the cross section of dual purpose solar collector with novelty designed of the absorber plate and Pipe distribution. The system is set in Al-Muthanna city, Iraq with latitude 31° 26'

29.4561" N and longitude 45° 26' 50.6714" E, toward the south with 20° tilt angle. The working fluid is water and air. The absorber plate is made of aluminum with black paint cover. Table 1 showed the specification of material and dimensions that used.

Table 1. Dual Purpose Solar Collector Geometric Specification

Collector component	Material	Quantity	Value
Absorber plate corrugated with the angle of 60°	Aluminium	High	0.94m
		Width	1.94m
		Thick	0.001m
Water pipes	copper	Diameter	10mm
Glazing	Float glass	Thick	4mm
Bottom insulation	Glass wool	Thick	5cm
Sid insulation	Silicon rubber	Thick	2cm
Main frame	wood	Thick	2 cm

The design of the absorber plate and distribution water pipe is a new type, the absorber plate is corrugated plate and the water pipe distribution with zig zag as shown in figure 1 below.

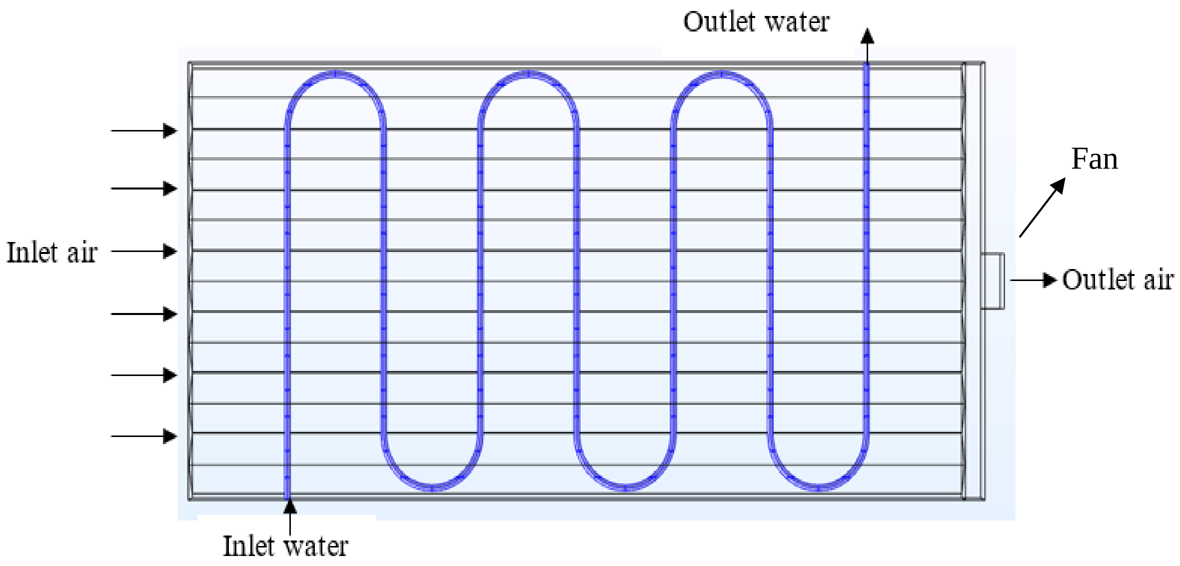

Figure 1. Cross Section of DPSC

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The primary stage of the built dual purpose solar collector (DPSC) model is the type of mesh which built automatically. Figure 2. Shows the mesh of (DPSC) and close to the absorber plate, the absorber plate is small sizes to the accuracy of the solution.

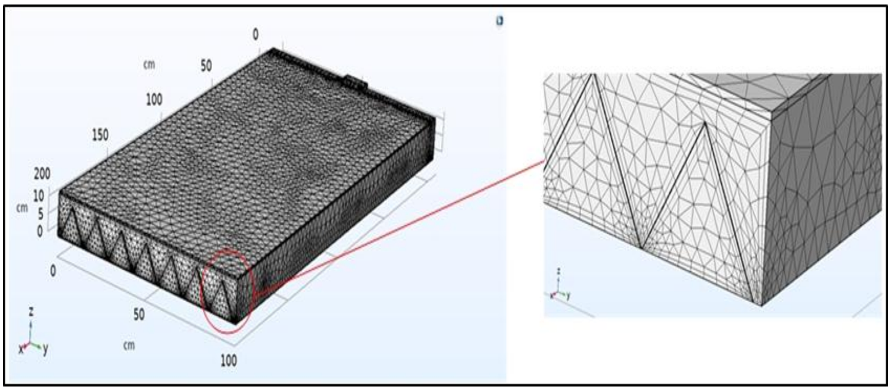

Figure 2. The Mesh Type for DPSC Model

2.2 Assumption of model

The following assumptions are adopted in the proposed model simulation process:

1- The dual purpose solar thermal collector (DPSC) is 3-D unsteady flow.
2- There is no heat dissipation in the back and side surface.
3- Water flow in the pipes is laminar.
4- Air flow in the channels is laminar.
5- Radiation and conduction in the water pipes and air channels assume negligible.

2.3. Governing equations

The equations using in the fluid flow is incompressible Navier-Stokes equations. The governing equations of modelled [16]:

1- Mass Conversation Equation

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

Where ρ is fluid density (kgm^{-3}), **u** is velocity vector (ms^{-1}).

2- Momentum conservation equation

a- X- direction momentum equation

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

b- Y- direction momentum equation

$$\frac{\partial u}{\partial t}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}= -\frac{1}{\rho}\frac{\partial P}{\partial y}+ \mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right) \quad (3)$$

c- Z- direction momentum equation

$$\frac{\partial u}{\partial t}+u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}= -\frac{1}{\rho}\frac{\partial P}{\partial z}+ \mu\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right) \quad (4)$$

3- Energy conservation equation

$$\rho . C_p\left(\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=k\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \quad (5)$$

4- The efficiency of DPSC[2]

$$q_u = \varepsilon_f \dot{m_f} C_{P.f}(T_{pm}-T_{fi}) \quad (6)$$

$$\varepsilon_f=\frac{\dot{m_f}\ C_{p.f}(T_{f2}-T_{f1})}{\dot{m_f}\ C_{p.f}(T_{pm}-T_{f1})}=\frac{(T_{f2}-T_{f1})}{(T_{pm}-T_{f1})} \quad (7)$$

$$\eta=\frac{q_u}{A_p I_T} \quad (8)$$

2.4 Initial conditions

The above equations have been solved by the following initial conditions. At first the initial temperature of the initial air, water inlet and absorber plate equal the ambient air temperature at the starting time (which t=0).

$$T_{air,w,p}(x,y,z,t=0)=T_{amb}(t=0) \quad (9)$$

While, the initial air and water velocity field (in 3D) are equal to zero:

$$u_{air,w}(x,y,z,t=0)=v_{air,w}(x,y,z,t=0)=w_{air,w}(x,y,z,t=0)=0 \quad (10)$$

2.5 Boundary conditions

The constant inlet water temperature, however the inlet air temperature is equal to ambient temperature, while the inlet pressure equals the measured atmospheric pressure:

$$T_w(x=0,y,z,t)=20℃ \quad (11)$$
$$T_{air}(x=0,y,z,t)=T_{amb}(t) \quad (12)$$
$$P(x=0,y,z,t)=P_{atm}(t) \quad (13)$$

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The air channel walls which are not in contact with the absorber plate or there is a good thermal insulation, also there are No-slip condition at the air channel walls, therefore:

$$-k_{air}\nabla T = 0 \quad (14)$$
$$u_{air}(x,y,z,t) = v_{air}(x,y,z,t) = w_{air}(x,y,z,t) = 0 \quad (15)$$

3. NUMERICAL SOLUTION PROCEDURE

Numerical solution is done by two parts; the first one: - by using CFD equations, eq. (1) to eq. (5). These equations are solved by using COMSOL Multiphysics 5.5 to calculate the absorber plate temperature (T_{pm}), water outlet temperature (Tw_o), air outlet temperature (Ta_o). These calculations are depend on solar radiation (G), water inlet temperature (Tw_i), ambient temperature (T_{amb}) and wind velocity (V).

The second one: - after found calculations of (T_{pm}, Tw_o and Ta_o) from numerical solution, now using equations (6), (7) and (8) to fined efficiency of collector mathematically. Figure 3 below shows producer of solution by COMSOL program.

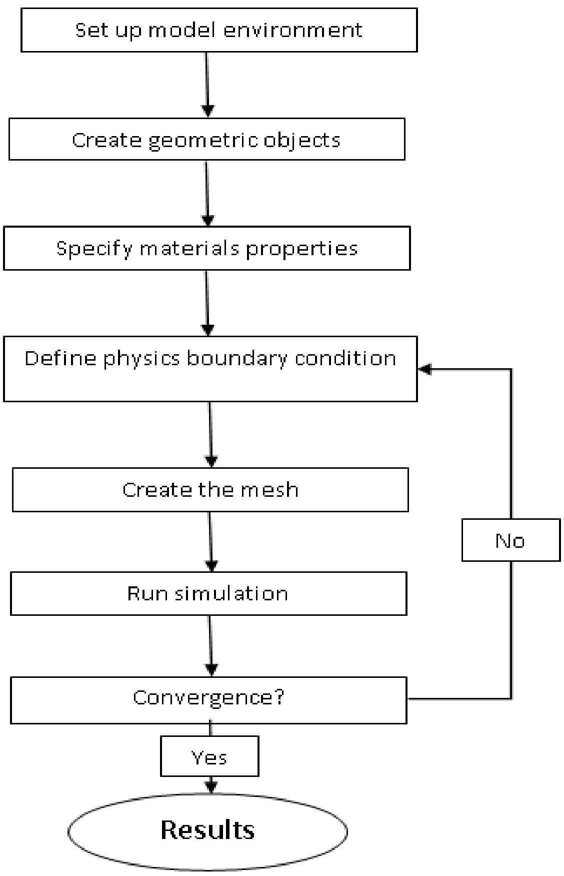

Figure 3. Simulation Methodology

4. VALIDATION

In order to verify accuracy of the numerical results that are achieved by using COMSOL program, the results were compared with previous study and present experimental results. A comparison between the experimental work of Omid et al. [17] with the present numerical work. The input data of the previous experimental work were introduced in the present simulation work to validated it. Figure 4 shows the experimental results of the efficiency for Omid [17] and the results of present work. Noticed the maximum error between the results of present work and previous experimental work was 2.4%.

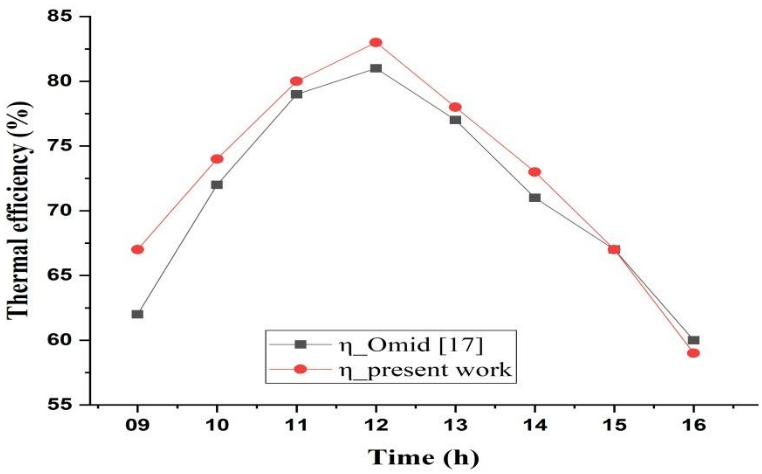

Figure 3. Comparison Between Experimental Results of Omid [17] with Numerical Results of Present Work of Efficiency

5. RESULTS AND DISCUSSION

A. Effect of ambient temperature

The study of effect of variation ambient temperature on DPSC performance was carry out with the other parameters remain same at in each cases to find out real effect of ambient temperature only. The values of solar irradiance, inlet water temperature and outlet air velocity was(1100 W/m^2, 35^{o}C and 2 m/s) respectively. The study includes five cases (15, 25, 35, 45 and 55)^{o}C. The results show that the outlet water and air temperature increases with the ambient temperature increases as shown on figure 5. The outlet water temperature was (64.44, 65.85, 67.38, 68.80 and 70.18)^{o}C for cases with ambient temperature at (15, 25, 35, 45 and 55)^{o}C respectively with increment rate of outlet water temperature of (1.4)^{o}C for every (10)^{o}C ambient temperature increases with the other parameters remains constant. The outlet air temperature was (73, 79.6, 83.2, 88.5 and 90.1)^{o}C for (15, 25, 35, 45 and 55)^{o}C respectively.

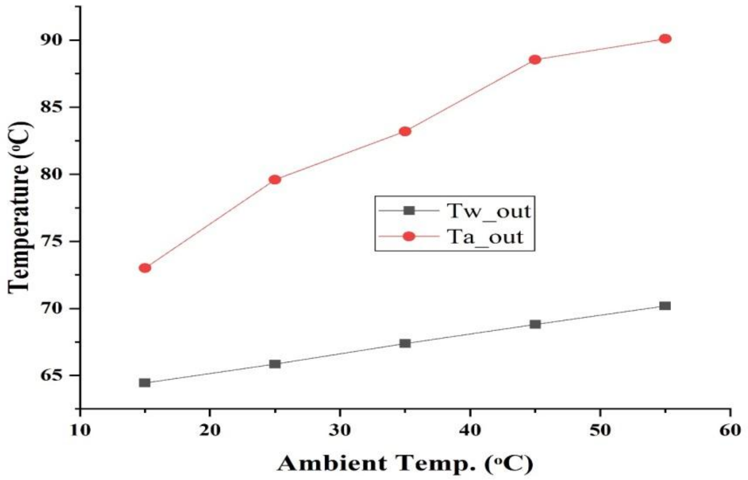

Figure 4. Variation of Outlet Temperature with Ambient Temperature Variation

B. Effect of solar irradiance

Solar irradiance is a source of heat in solar collectors systems, the performance of this system depend mainly on it, so, the effect of variation of solar radiation was studied in this section. The effect of variation studied at (300, 500, 800 and 1100) W/m^2 with the other parameters constant, as shown in figure 6. The outlet temperature increases with solar irradiance increases, it was (52.9, 56.64, 62.4 and 68.61) ^oC for (300, 500, 800 and 1100) W/m^2 respectively for water and (57.6, 65.89, 78.36 and 83.2) ^oC for (300, 500, 800 and 1100) W/m^2 respectively for air.

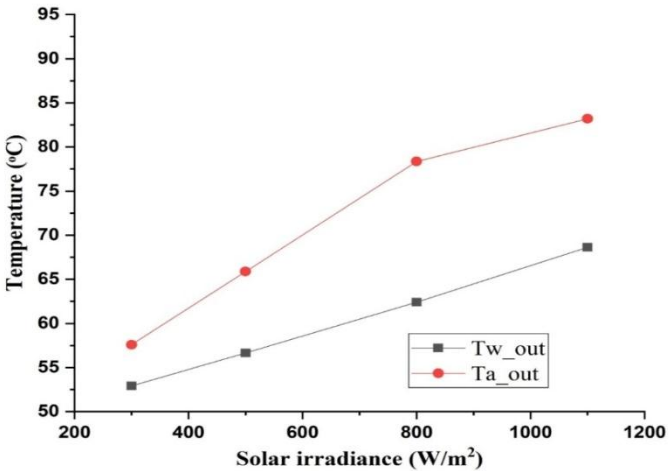

Figure 5. Variation of Ambient Temperature with Variation of Solar Radiation

C. Effect of outlet air velocity

The air was pulled from upper collector by fan air to pull the air through inside the collector. The air velocity effect on rate of heat transfer between working fluid and absorbation part. In this section, the variation of outlet air velocity was conducted at (0.5, 1, 1.5 and 2) m/s. Figure 7 clears that the outlet temperature decreases with outlet air velocity increases. The results of outlet water temperature was (68.8, 65.55, 63.35 and 61.75) ^{o}C for (0.5, 1, 1.5 and 2) m/s respectively, and outlet air temperature was (84.4, 78.5, 73.5 and 69.04) ^{o}C for (0.5, 1, 1.5 and 2) m/s respectively.

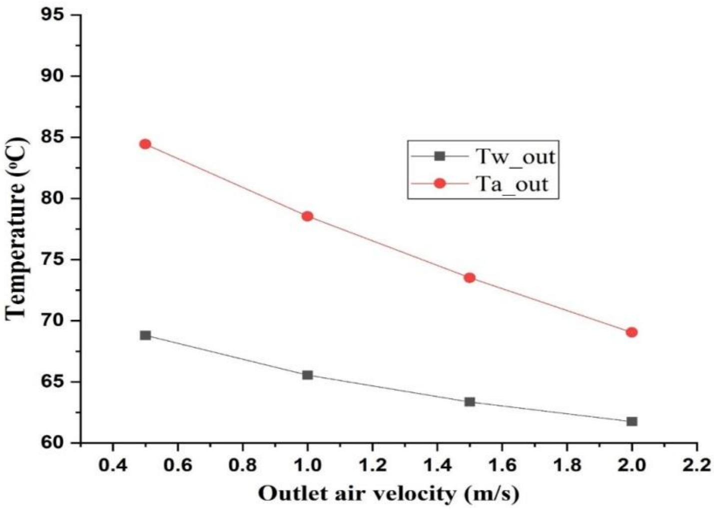

Figure 6. Variation of Outlet Temperature with Variation of Outlet Air Velocity

D. Effect of inlet water temperature

In closed solar collector systems, as this present work, the inlet water temperature increased gradually. In this study of effect illustrated how variation inlet temperature effected on performance of dual-purpose solar collector for four cases. Figure 8 shown that the outlet water temperature increases as inlet water temperature increases with high rate of solar radiation while the outlet water temperature was increased with solar radiation increase until a certain limit, then it will decreases due to low solar radiation rate, therefore, absorber plate temperature will decreases gradually and the heat will transfer from water to absorber surface and air flowing. The values of outlet water temperature were (46.93, 55.01, 65.55 and 71.29) ^{o}C for water, and (73.97, 76.78, 80.4 and 82.34) ^{o}C for air, for (25, 35, 45 and 55) ^{o}C inlet water temperature.

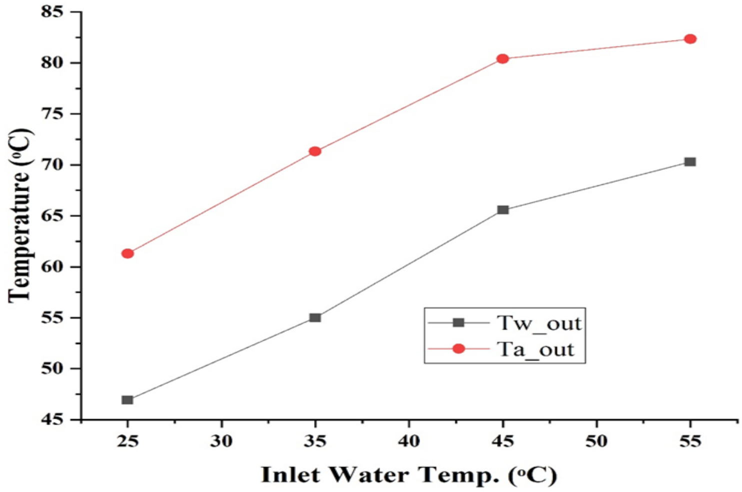

Figure 7. Variation of Outlet Temperature with Variation of Inlet Water Temperature

E. Effect of water volume flow rate

In this section, the effect of four different water volumes flow rate was studied. The ranges were (40, 60, 80 and 100) L/h. Figure 9 shows that when the water volume flow rate increases the outlet temperature decreases. The values of outlet temperature were (76.62, 68.61, 64.211, 64) ^{o}C in water part and (88.1, 86.9, 84.8 and 84.7) ^{o}C in air part for (40, 60, 80, and 100) L/h respectively.

From above, the water volume flow rate is effecting directly on the heat transfer between water flowing and absorption part. Also, it was illustrated that the variation in water part is higher than variation in air part because the air does not effected mainly by volume flow rate change as water part. The temperature difference between maximum and minimum values of outlet temperature was (12.4 and 5.9) ^{o}C for water and air parts respectively.

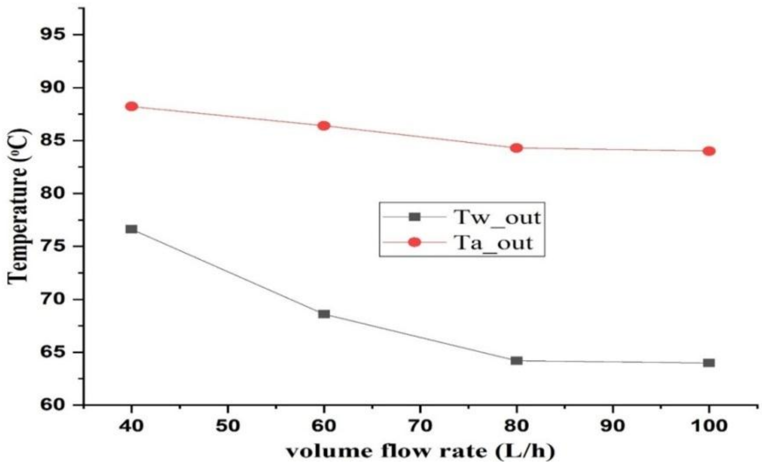

Figure 8. Variation of Outlet Temperature with Variation Water Volume Flow Rate

6. CONCLUSIONS

The following topics were conducted from this study:

1- The outlet temperature (water and air) increases as the ambient temperature rise. The outlet water and air temperature have been recorded (64.44, 65.85, 67.38, 68.80 and 70.18) ^oC and (73, 79.60, 83.2, 88.5 and 90.1) ^oC for ambient temperature (15, 25, 35, 45 and 55) ^oC respectively.

2- The DPSC evaluation of the effects of air velocity on the collector efficiency was included; they were tested at four variable air velocity (0.5, 1, 1.5 and 2) m/s.

3- The outlet temperature decreases with outlet air velocity increases. The results of outlet water temperature was (68.8, 65.55, 63.35 and 61.75) ^oC for (0.5, 1, 1.5 and 2) m/s respectively, and outlet air temperature was (84.4, 78.5, 73.5 and 69.04) ^oC for (0.5, 1, 1.5 and 2) m/s respectively.

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