

Dynamic Modeling and Experimental Study of Forced Convection Evacuated Tube Solar Collector Used for Grape Dryer



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Abstract The two dimensional performance model, analyzing evacuated tube solar collector system, used for grape drying is developed. A Solar collector is designed and fabricated to dry 10 kg of Thompson seedless grapes efficiently, under the forced convection heat transfer. But this article is restricted to discussion on solar collector and its thermal performance only. The collector setup consists of ten evacuated tubes of outer diameter 58 mm and length 1800 mm arranged parallel. This solar system's performance is tested experimentally and an analytical thermal model is developed by solving coupled linear differential equations using Rung-Kutta's fourth order method. The experimental data of the solar collector is compared with the analytical data of thermal model. It is observed that the developed model predicts the behavior of the actual system accurately with average percentage error in the range of 2.45–4.6% with coefficient of determination $R^2 = 0.957–0.98$. The experimental thermal efficiency of the system is observed as high as 31.2% which is higher than the thermal efficiency of the flat plate collector solar system (15%). Theoretically calculated and experimentally measured results of air outlet temperature from solar collector shown a good agreement.

Keywords Evacuated tube solar collector · Forced convection · Runge–kutta method

Nomenclature

C_f	Sp. Heat of working fluid (J/kgK)
H_{con}^{r-f}	Heat transfer coeff. betn absorber and working fluid (W/m ² k)
T_r	Absorber surface temperature (k)

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T_f	Working fluid temperature (k)
R_e	Renold's number
D	Diameter of air flow pipe (m)
T_g	Outer glass temperature (k)
H_{con}^{g-a}	Heat transfer coeff. betn outer tube and atmosphere (W/m ² k)
T_a	Atmospheric temperature (k)
μ	Viscosity of working fluid
I_G	Global solar radiation (W/m ²)
m	Mass flow rate of working fluid (kg/s)

1 Introduction

For minimizing the convective heat loss to the surrounding from the solar collector now a day evacuated tube solar collector is being used widely. A twin-glass evacuated tube is made up of strong material like; transparent borosilicate glass with the outer surface of inner tube is coated with a special selective coating (Al-Ni/Al). This coat is having excellent solar heat absorption and minimal heat reflection features. The top of the two tubes is fused together and evacuated. The sunlight enters through the outer glass tube; the heat generated is absorbed by the selective coatings and subsequently transferred to the circulating working fluid. Since evacuated tube collectors can operate under cloudy conditions, they are more efficient throughout the year and can achieve higher temperature with higher efficiency as compared to flat plate collectors.

However, evacuated tube collector is more expensive than flat plate collector [17–7]. Yadav Kumar [18] studied experimentally the performance of an evacuated tube solar air collector for different air mass flow rates. Air was used as a working fluid in the several experiments and tested in Indian climatic conditions. Sabiha [1] has thoroughly reviewed comprehensive literature on, why evacuated tube collector is preferable, its types, structures, applications and challenges. Applications of evacuated tube solar collectors in water heating, air conditioning, heat engines, solar cooker, swimming pool heating, steam generation and solar drying for industrial as well as residential sectors have been summarized and presented. Philippe [6] have developed a dynamic model of a solar evacuated tube collector under variable weather conditions to predict the performance of the system theoretically. They have validated this mathematical model with experimental results and parametric sensitivity analysis was applied to the model to improve the accuracy. Sharma [14] has studied the thermal analysis of a novel solar collector with mini channel with the help of numerical method. The novel solar collector was fabricated by using a U-shaped flat-tube absorber with the selective coating on its external surface. The working fluid was circulated inside the array of mini channels which were located at the cross-section of the absorber and along the length of evacuated tube. Two dimensional mathematical models were developed to compare the performance of the system

with experimental results. Rushi [12] has investigated experimentally an air heating system using one ended evacuated tubes powered by solar energy. The solar collector system containing forty evacuated tubes was used for air heating purpose. Naik [9] presented the analytical model of a U type evacuated tube solar collector for heating Aqueous lithium chloride solution (LiCl-H₂O), water and air used as working fluid. The developed mathematical model predicted the outlet temperature and net heat gain by the working fluid with high accuracy with that of experimental results.

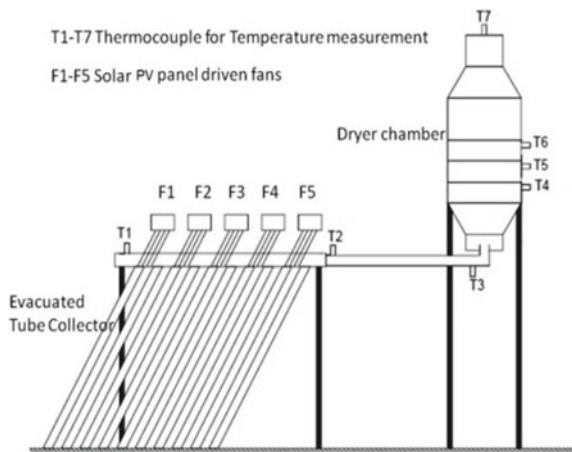
Two dimensional thermal model of the system is developed under transient conditions. Air is used as a working fluid so as to avoid freezing, overheating of working fluid. Another advantage of air as working fluid is that it can be used in an open loop cycle and risk of getting contaminated is eliminated. Variations of ambient temperature, solar radiation, mass flow rate of working fluid, and wind speed are considered in this model for the analysis of collector outlet temperature. The developed thermal mathematical model depends on the equations of energy conservation to a small control volume along the longitudinal axis of the evacuated tube. The first order differential equations for each control volume are obtained and solved by using a fourth order Runge–Kutta algorithm. In particular, dynamic modelling predicts the behaviour of collectors significantly. The experimental setup has been validated to assess the predictions given by the thermal model.

2 Materials and Methods

2.1 Experimental Setup

Evacuated tubes are made up of two coaxial borosilicate glass tubes, joined at the top and sealed at the bottom which contains a vacuum as shown in Fig. 1. The outer

Fig. 1 Solar ETC system with forced circulation (Schematic)



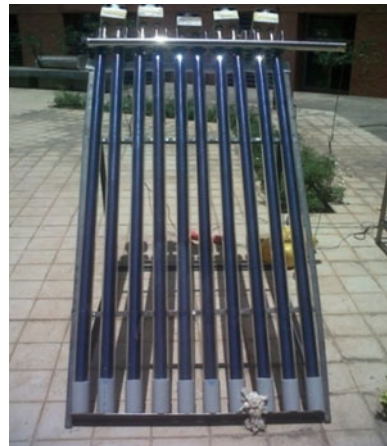
tube is a transparent tube of 58 mm in diameter and 1800 mm in length, called as cover tube. Whereas, the inner tube is of 47 mm in diameter and 1750 mm in length, which is called as Absorber Tube. The thickness of the inner tube and the outer tube is 1.6 and 2.00 mm respectively. The working fluid is circulated through the inner tube which absorbs the solar energy. The outer part of the inner tube is coated with a selective coating material (Aluminium Nitrite) for absorbing the incident solar radiation further it is transmitting to working fluid flowing through it. As the space between the outer and the inner tube is evacuated, hence it works as a thermal insulator which prevents heat losses primarily due to conduction and convection.

Thus the trapped solar energy absorbed and transmitted to a working fluid, gets prevented from escaping back to the environment (green house phenomena). Based on the design the required area of solar collector is 1.51 m^2 to collect the solar energy to dry 10 kg of Thompson seedless grapes. According to the selected size of the evacuated tube, it requires 9.28 tubes to make 1.51 m^2 exposed area of the collector.

Hence ten evacuated tubes are arranged parallel to each other and to collect the output of the individual tube, a perpendicular tube is arranged at the top of all evacuated tube. This central tube is of stainless steel material of 60 mm diameter. It is further connected to the bottom of the dryer to give the output hot air to dryer chamber.

From the literature, it is found that the angle of inclination of the solar collector is kept at a longitudinal value of location $+15^\circ$. In this setup array of tubes is inclined 45° to horizontal on a steel frame to gain maximum solar beam radiation and absorb diffused radiation as well. All the time the collector area is measured which is exposed to the direct sunlight and it is found as 1.61 m^2 . The contacting area between the supportive frame and the evacuated tube is insulated with polyethylene foam sheet with the thermal conductivity of 0.04 W/m-k , to minimize the heat losses (not shown in above Fig. 2). Forced air circulation is provided with the help of 12 V/1A five fans which run on the 15 W solar panel.

Fig. 2 Actual solar ETC system



2.2 Thermal Model of Solar ETC System

The air mass flow rate is the very important parameter which acts on the defined performance indicators. The higher is the airflow, better is the thermal efficiency but lower is the outlet working fluid temperature.

Figure 3 shows the thermal model of the evacuated tube exposed to the solar radiation. At the bottom of evacuated tubes large pressure drop is observed due to 180° change in the direction of the air flow [3–7].

The following assumptions are made to solve the set of equations,

- Heat transfer in the system is considered in one-dimension which is along the radial coordinate and axis symmetric.
- Diffused radiation gained by the actual system is neglected.
- Finite differential equation is solved for single evacuated tube and uniformity is considered to all tubes with constant mass flow rate.
- For any axial position on the tube and the cover, heat distribution is equal in azimuth direction.
- As the walls are thin, the resistances due to conduction in the glass walls are assumed negligible.
- There is only one temperature that characterizes the inner and outer surface of glass tubes and the temperature varies in the axial direction which is essential because of the heat gained by the fluid.
- The temperature dependent data for variable thermal properties with air temperature is readily available.

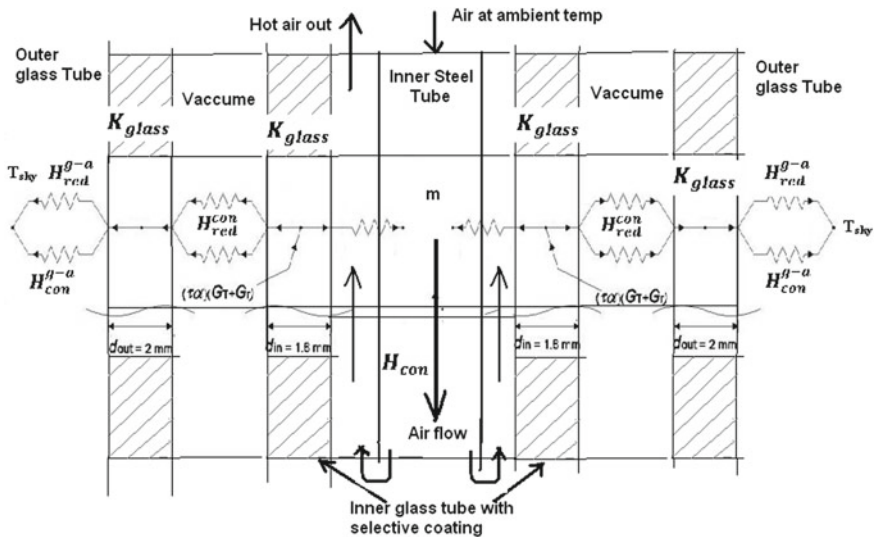


Fig. 3 Thermal model of the ETC exposed to solar radiation

Because the flow in an evacuated tube collector is completely enclosed, energy balance equation can be applied to determine the variation in mean temperature of working fluid. Also the temperature at different positions along this tube and the total convection heat transfer is related to the difference in tube inlet and outlet temperatures. For the constant mass flow rate heat gained by the working fluid is given by following equation, [8, 10, 11, 13–16]

$$C_f \left(\frac{\partial T_f}{\partial t} + u \frac{\partial T_f}{\partial x} \right) = \{H_{con}^{r-f} (T_r - T_f)\} \quad (1)$$

The evacuated tube is divided in to ‘n’ (180) segments along its longitudinal direction, the velocity of the working fluid is more as compared to the width of each segment, and hence change in fluid temperature is negligible. The velocity of the working fluid, ‘u’ is assumed to be constant, thus equation above reduces to,

$$C_f \left(\frac{\partial T_f}{\partial t} \right) = \{H_{con}^{r-f} (T_r - T_f)\} \quad (2)$$

It is considered that the heat transfer coefficient is zero due to convection, in between the inner to outer glass, as the vacuum is present. Heat transfer coefficient due to convection between working fluid and receiver is calculated by Reynolds number and Nusselt number (Nu) as follows,

$$Re = \frac{4m}{\pi D \mu} \quad (3)$$

For the turbulent flow region ($Re > 6000$), it recommended to use the correlation obtained from the relationship of Reynold number and Prandtl number (Pr), [3],

$$Nu = \frac{\frac{f}{8} (Re - 1000) Pr}{1 + 12.7 \sqrt{\frac{f}{8}} (Pr^{2/3} - 1)} \quad (4)$$

The Darcy friction factor (f) for the setup can be obtained from:

$$f = (0.79 \log Re 1.64)^{-2} \quad (5)$$

and

$$H_{con}^{r-f} = \frac{Nu k}{D} \quad (6)$$

Heat transfer to the receiver is given by the equation,

$$C_r \left(\frac{\partial T_r}{\partial t} \right) = \{ \tau \alpha I_G + \varepsilon_g \sigma (T_g^4 - T_r^4) + H_{con}^{r-f} (T_f - T_r) \} \quad (7)$$

where the product ‘ $\tau\alpha$ ’ is the transmittance absorptivity of the evacuated tube collector, ϵ_g is Emmisivity of the receiver and σ is the Stefan Boltzman constant.

Temperature of the receiver tube at the end is calculated as;

$$T_r = \frac{q^{con}}{H_{con}^{r-f}} + T_f \tag{8}$$

The outer glass tube temperature is calculated as;

$$C_g \left(\frac{\partial T_g}{\partial t} \right) = \{ \epsilon_g \sigma (T_a^4 - T_g^4) + \epsilon_g \sigma (T_r^4 - T_g^4) + H_{con}^{g-a} (T_g - T_a) \} \tag{9}$$

Equation (9) states the change in cover temperature which is proportional to the difference between net heat gain rate by the receiver to the heat losses to the environment by radiation and convection.

The convective heat transfer coefficient between the atmosphere and the outer glass is given by the equation,

$$H_{con}^{g-a} = 5.678 + 3.8v \tag{10}$$

where, ‘ v ’ is ambient air velocity in m/s.

Finite difference method is used to solve this system. In this system, the collector is defined as a single fluid channel and it is divided into ‘ n ’ segments ($n = 180$). The single order differential equations are solved for each segment in the time domain using a 4th order Runge–Kutta method. A MATLAB-14B program is developed to solve this set of equations simultaneously. At first iteration boundary conditions like the glass temperature T_g and air temperature T_a is considered as the same.

Equations 1, 7 and 9 are solved simultaneously for the first segment (that is. $n = 0$ node). The output of the first segment is passed as an input for the next segment and so on. The last segment gives the output temperature of the outer glass tube, inner glass tube and the outlet temperature of the working fluid. Total tube length is divided in 180 segments.

Table 1 Solar ETC data for day one

Time Hrs	Solar radiation W/m ²	Ambient		Collector outlet Temp	Velocity of air m/s	Mass of air kg/s	Heat gain J/s	Thermal efficiency
		Temp	% RH					
9.00	409	23.4	32	31.2	2.3	0.01092	85.68442	13.09
10.00	649	30.8	25	51.8	2.4	0.011394	240.7188	23.18
11.00	810	31.6	22	61.2	2.5	0.011869	353.4363	27.27
12.00	847	32.5	21	67.5	2.4	0.011394	401.198	29.60
13.00	960	34.5	21	73.1	2.6	0.012344	479.336	31.21
14.00	850	35.8	19	73.5	2.3	0.01092	414.1414	30.45
15.00	780	34.8	22	68.1	2.3	0.01092	365.8066	29.31
16.00	610	32.8	25	61.3	2.2	0.010445	299.4656	30.68
17.00	490	29.8	28	51.2	2.2	0.010445	224.8619	28.68
18.00	320	25.6	31	41.2	2.1	0.00997	156.4672	30.56

Table 2 Solar ETC data for day two

Time Hrs	Solar radiation W/m ²	Ambient		Collector outlet Temp	Velocity of air m/s	Mass of air kg/s	Heat gain J/s	Thermal efficiency
		Temp	% RH					
9.00	411	23.5	29	32.1	2.4	0.010852	93.8858	14.28
10.00	651	29.7	24	52.1	2.4	0.010852	244.540	23.48
11.00	809	31.8	21	62.1	2.5	0.011304	344.566	26.62
12.00	845	32.8	20	68.2	2.7	0.012208	434.768	32.16
13.00	915	34.6	19	73.5	2.6	0.011756	460.059	31.42
14.00	880	35.7	19	73.6	2.4	0.010852	413.752	29.39
15.00	720	35.1	20	67.9	2.3	0.0104	343.156	29.79
16.00	610	33.1	23	61.2	2.3	0.0104	293.984	30.12
17.00	490	29.8	26	51	2.4	0.009948	212.153	27.06
18.00	324	25.4	31	41.3	2.4	0.009948	159.115	30.69

3 Results and Discussion

3.1 Experimental Results

Tables 1 and 2 shows the sample experimental data for a particular test. The experiments are conducted under the uncontrolled conditions of the environment. The forced air is generated by DC fans run on the solar 15 W panel. The fan speed is the function of the solar radiation, as more is the beam radiation more will be a PV panel current and voltage and hence more air is circulated through the system. The

air velocity at the exit to the solar collector is measured from 2.1 m/s to 2.7 m/s. Based on the discharge the instantaneous mass flow rate is calculated and tabulated.

The temperature of air at the inlet and outlet of the collector is measured and recorded for every hour during the test. The diurnal variation of the solar radiation ambient air temperature (T_{sky}) and relative humidity, collector outlet temperature, mass flow rate and instantaneous thermal efficiency of the ETC system is shown in Tables 1 and 2.

During the test time, solar radiation varies from 380 to 925 W/m², with the average solar radiation value 730 W/m² in the month of April where as it rises to the average value of 830 W/m² with highest solar radiation of 1050 W/m² in the month of May during the year 2013–2015. The ambient air temperature varies in the range of 23–35 °C in the month of April and it goes up to the maximum ranges from 35.8 °C to 38.3 °C in the month of May.

The minimum average temperature range is 24.3–25.9 °C during the test time. The maximum gain in the temperature is observed to 42 °C, which is quite high as compared to other type of solar collector. The maximum collector outlet temperature reached to 80.5 °C to a mass flow rate of air 0.0113Kg/s with average out let temperature of collector is 55.1 °C. The mass flow rate is directly proportional to the voltage generated by the solar PV panel. This voltage is a function of the solar radiation. Hence the mass flow rate in the afternoon is observed higher during 12 noon to 2 pm.

3.2 Simulated Collector Output

Below Tables 1 and 2 gives the detail about the thermal properties of the solar ETC system along with their properties. The program out is calculated for a single evacuated tube and it considered as same for all the tubes.

As all the ten tubes are arranged as parallel to each other, together the mass flow rate will be increased and temperature output of all tubes will remain the same as no considerable changes are observed in input parameters to each tube.

The Tables 3, 4 and Figs. 4, 5 illustrates the software calculated values of collector outlet temperature.

Figure 4 is a plot of the single day collector outlet working fluid temperature variation. This shows simultaneously the outlet temperature values measured using accurate measuring instruments and analytically calculated values at the same point. Last column in the Tables 3 and 4 demonstrates the percentage error in individual reading, which is observed in the range of 1.07–6.98% with the total average percentage error of 2.46% which is well within the acceptable limit.

Similar kinds of results are observed for all experimental data when analyzed theoretical as explained in the Fig. 5 with the percentage error between experimental and analytical collector outlet temperature in the range of 1.5–7.06%. The total average percentage error is 3.55% with $R^2 = 0.985$, which is also well within the acceptable limit. The Fig. 5 shows the plot of actual/experimental versus predicted

Table 3 Experimental and theoretical collector outlet temperature

Time in Hrs	Solar radiation	Ambient temp	Collector outlet Temp. (Actual)	Collector outlet Temp (Analytical)	% error
9.00	411	23.5	32.3	31.8	1.57232
10.00	651	29.7	52.3	54.8	4.56204
11.00	809	31.8	62.3	65.4	4.74006
12.00	845	32.8	68.3	71.3	4.20757
13.00	915	34.6	73.8	77.3	4.52781
14.00	880	35.7	73.4	76.1	3.54796
15.00	720	35.1	67.6	71.4	5.32213
16.00	610	33.1	63.4	62.2	1.92926
17.00	490	29.8	54.3	57.2	5.06993
18.00	324	25.4	40.8	43.9	7.0615
Average % error					3.5537

Table 4 Experimental and Theoretical collector outlet temperature

Time in Hrs	Solar radiation	Ambient temp	Collector outlet Temp (Actual)	Collector outlet Temp (Analytical)	% error
9.00	409	23.4	32.1	30.6	4.90196
10.00	649	30.8	52.1	54.5	4.40367
11.00	810	31.6	62.1	65.4	5.04587
12.00	847	32.5	68.2	71.4	4.48179
13.00	920	34.5	73.5	74.6	1.47453
14.00	850	35.8	73.6	74.4	1.07527
15.00	713	34.8	67.9	70.3	3.41394
16.00	530	32.8	63.2	61.9	2.10016
17.00	464	29.8	54.1	57	5.08772
18.00	320	25.6	41.3	44.4	6.98198
Average % error					2.496

Fig. 4 Diurnal Collector outlet temperature experimentally and theoretically

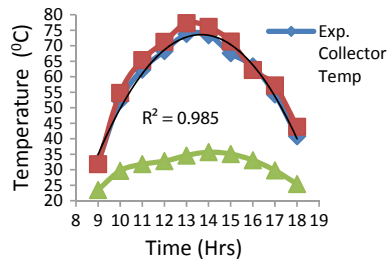
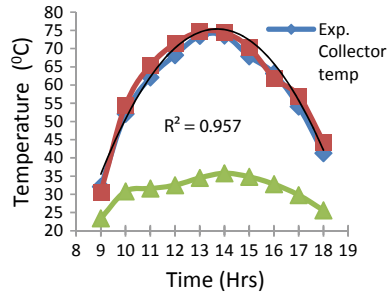


Fig. 5 Diurnal Collector outlet temperature experimentally and theoretically



collector outlet working fluid temperature for a single day. It is observed that the predicted and experimental values are in good agreement with $R^2 = 0.984$. Similar kinds of results are observed in all experimental and predicted values. It indicated that the simulated mathematical model is accurate.

4 Conclusion

The performance of solar evacuated tube collector is studied analytically and experimentally. A set of single order differential equations are formed which simulates the performance of the system. These equations are solved numerically by 4th order Runge–Kutta method with the help of MATLAB program. To simulate the flow analysis certain assumptions are made and collector outlet fluid temperature is calculated for a single tube, by considering there will not be much variation in physical data like mass flow rate and solar radiations, among all evacuated tubes. The validation of the simulated results is done with the experimental values. It is observed that the average percentage error in the actual collector outlet temperature and simulated or analytically calculated value is in the range of 2.45–4.6% with average coefficient of determination $R^2 = 0.973$ to for all experiments. Hence it is concluded that the actual collector outlet working fluid temperature has good agreement with calculated collector outlet working fluid temperature with developed thermal model.

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