

Analysis of solar flat plate collector with straight and helical flow path heat tube using mathematical modeling and java based simulation

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Abstract. In this study, a Mathematical Model for solar flat plate collector having plain and helical flow path heat tubes is developed for a Java based simulator software to study the performance of SFPC. A software simulation is a proven tool for efficient and economic prediction of heat transfer process results. The mathematical model of a SFPC is developed to investigate the fluid mean temperatures, heat gain and collector recovery factor, edge loss, bottom loss and collector efficiency. The simulation results were verified with experimental results and found that the simulator has the ability to predict the performance of the solar flat plate collector accurately as proven by the comparison of experimental data with simulation. The difference between the predicted and experimental results is, at maximum, approximately 4% which is within the acceptable limit considering some uncertainties in the input parameter values. A parametric study was performed between plain heat tubes and helical strip inserted heat tubes and it was found that the outlet fluid temperature and the energy efficiency of the system are improved in the helical strip inserted heat tubes and also the mass flow rate has a significant effect on the efficiency of the system.

1. Introduction

In the past few decades solar power is getting more importance as a strong natural source of energy. The easiest and direct application of the solar energy is the conversion of solar radiation into heat. Solar radiation shall be widely used for water heating in hot water systems, for central heating installations in hospitals and community buildings and swimming pools also. During the past century, Solar water heating technology has improved to a great extent. Approximately around 30-32 million square meters of solar energy collectors are installed worldwide. Commonly used solar heating systems in the commercial/domestic buildings have two main components namely a storage tank and a solar collector of which the solar collectors are the main component of the system. The solar energy is transformed into heat and then the heat is transferred to a working fluid. Gravity based passive systems and active systems with pumps are the two main categories of a Solar water heating system. The objective of this project is to develop a Java based simulator for a solar flat plate collector and compare the simulator results with experimental results for heat transfer rates, collector heat recovery factor, edge loss and bottom loss and energy efficiency of the system.



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2. Literature survey

Ranjith P *et al* have done “*A comparative study on the experimental and computational analysis of solar flat plate collector using an alternate working fluid*” [1] and found that the numerically computed and the experimentally measured results match closely and satisfactorily. The proposed model is suitable for the validation of overall system efficiency and can be used for any number of working fluids in order to find the outlet temperature. An updated review of models like “Lumped-capacitance model and Discretized model for solar flat plate collectors with classification and description of performance characteristics” is done by Luca A Tagliafico, *et al* [2]. Ahmad fudholi *et al* have given a “Mathematical model of double pass solar air collector:” [3]. In this [3] study a numerical model is evolved to predict the efficiency of a double pass solar collector. The solution arrived by them solve simultaneously the temperature equations for energy balance conditions in steady state. The simulated results go in match with the experimental results obtained. Also it was observed that the mass flow rate highly influenced the efficiency of the solar collector.

Ranga Babu J *et al* [5] studied the heat transfer phenomena inside the risers of SFPC and analysed numerically by developing a 3D model using commercial CFD software ANSYS FLUENT 14. It is observed that the proposed model is predicting the heat loss coefficients and fluid temperatures with good accuracy. The thermal efficiency obtained with this numerical model is validated against the measured data. A simulation procedure using HFE 7000 as working fluid was developed and the fluid outlet temperature of simulation and experiment results are compared by [6] Helvacı H U & Kham Z A and found that both measured and simulated results are closely matching. Marroquin-De Jesus Angel *et al* have done “Analysis of Flow and Heat Transfer in a Flat Solar Collector with rectangular and cylindrical Geometry using CFD” [4]. The study consists of an experiment with solar collectors of one with circular and another with rectangular heat tubes and also numerical analysis of the experimental setup. For both the solar collectors with varying geometry of heat tubes, fluid outlet temperature was calculated and results are compared.

As an outcome of the above survey, in this study, a Java simulator based on numerical model of SFPC is developed for performance analysis and predict the influence of mass flow rate and insertion of helical strip inside the heat tubes on the outlet temperature of the working fluid, absorber plate temperature and energy efficiency of the system.

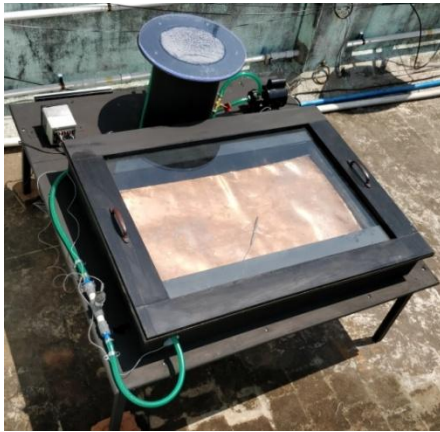
3. Experimental Specifications

In this experiment, flat plate collector with parallel copper tubes, glass cover, copper absorber plate and insulation at the back and edges of the collector was used and also for the simulation the same specifications are used. Glass cover with 4 mm thickness is used for the reduction of convection and radiation heat loss from the system to the surroundings.

The absorber plate is a copper sheet of 0.3 mm thickness and 0.8m x 0.4m of size. The working fluid is circulated through the copper tubes by a small pump of 5 LPM capacity to give a forced circulation.

Insulation is used to reduce the conduction loss from the edges and backside of the collector. The heat tubes are placed at a distance of 0.084m from each other. The outer diameter of the heat tube is 0.0127m and inner diameter is 0.0115m. The experimental setup is shown below in figure 1.

Instrumentation provision is given to measure the fluid inlet and outlet temperature and absorber plate temperature with a selector switch in a digital display unit. The thermocouples are mounted in thermowell in the fluid inlet and outlet path and one is directly fixed on the absorber plate to measure respective temperatures.

**Figure 1.** Experimental setup**Figure 2.** Copper with twisted stripes inserted**Table 1.** Material Properties and constants

Thermal conductivity of the bottom insulation, K_b	0.04 (W/mK)
Thickness of the bottom insulation, X_b	0.025 m
Edge area	0.390 m ²
Collector area A_c	0.320 m ²
No of glass covers	1
τ	0.85
α	0.96
σ	5.67x10 ⁻⁸ W/m ² K ⁴
ε_p	0.12
ε_g	0.88
Specific heat of water C_p	4180 J/Kg K
Outer diameter of tube	0.0127 m
Inner diameter of tube	0.0115 m
Center distance between tubes	0.084 m
Bond Resistance C_b	58 w/mK
Absorber plate thickness, δ	0.0003 m
Thermal conductivity of copper plate	390 w/mK

4. Mathematical Modelling

Analytical model of the SFPC with parallel heat tube is developed with the following assumptions.

- Steady-state conditions.
- The thermal and radiation properties of the Absorber and glass cover are assumed constant and not dependent on the temperature.
- Heat flux conditions are considered uniform.
- Negligible effect of entry regions.
- One dimensional heat transfer in system layers
- The Properties of the insulation material and the glass are not dependent on the temperature.

4.1 Heat loss coefficient

In a flat plate collector there are two types of losses occur. One is the optical losses and the other is the thermal losses. Thermal losses are further divided into edge losses, bottom losses and top losses. Other losses occur due to radiation loss from top of the solar collector and convection heat loss from the top glass to the atmosphere and from the absorber plate to the cover. Klein's [9] formula for the top heat loss coefficient is shown in Eq. (1)

Klein's Correlation for Top loss co-efficient, U_t

$$U_t = \left[\frac{N}{\left(\frac{C}{T_{pm}} \right) * \left(\frac{T_{pm} + T_a}{N+F} \right)^e} + \frac{1}{h_w} \right]^{-1} + \left[\frac{\sigma(T_{pm} + T_a)(T_{pm}^2 + T_a^2)}{(\epsilon_p + 0.0059Nh_w) + \left(\frac{2N+f-1+0.133\epsilon_p}{\epsilon_g} \right) - N} \right] \quad (1)$$

Where

σ = Stephan Boltzmann Constant

$F = (1 + 0.089 h_w - 0.1166 h_w \epsilon_p) * (1 + 0.786N)$

N = number of glass covers

$C = 520 * (1 - 0.000051\beta^2)$; $0 < \beta < 70$; for $70 < \beta < 90$ take $\beta = 70$;

$e = 0.43 * (1 - 100/T_{pm})$

Heat loss coefficient from bottom is

$$U_b = \frac{K_b}{X_b} \quad (2)$$

Where K_b = Thermal conductivity of the bottom insulation in W/m^0c

X_b = Thickness of the bottom insulation in m

Heat loss coefficient from Edge is given by

$$U_e = U_b \left(\frac{A_e}{A_c} \right) \quad (3)$$

Where U_b = Bottom loss coefficient in W/m^0c

A_e = Area of edge in m^2

A_c = Area of collector in m^2

$$\text{Hence Overall Heat Loss Coefficient } U_L = U_t + U_e + U_b \quad (4)$$

Collector Fin Efficiency is given by

$$f = \frac{\tanh m \left(\frac{W-D}{2} \right)}{m \left(\frac{W-D}{2} \right)} \quad (5)$$

$$\text{Where } m = \sqrt{\frac{U_L}{K\delta}} \quad (6)$$

W = centre distance of the absorber tubes

D = diameter of absorber tubes in m

Collector Efficiency factor

$$F' = \frac{\left(\frac{1}{U_L} \right)}{W * \left[\left(\frac{1}{U_L(D+(W-D)F)} \right) + \left(\frac{1}{C_b} \right) + \left(\frac{1}{\pi D_i h_{fi}} \right) \right]} \quad (7)$$

Where,

C_b = Bond resistance in W/m^0K

D_i = Inner Diameter of absorber tube in m

h_{fi} = Heat transfer coefficient of inner tube surface to water in $W/m^2 K$

$$= \frac{Nu - K_{water}}{D}$$

Nu = Nusselt Number

K_{water} = Thermal Conductivity of water in W/m^0K

D = Diameter of absorber tube in m

Collector Heat Removal Factor F_R is determined by

$$F_R = m \cdot c_p \cdot \left[1 - e^{-\left(\frac{F^I \cdot A_c \cdot U_L}{m \cdot c_p} \right)} \right] \quad (8)$$

where m = mass flow rate of water in Kg/sec

c_p = Specific conductivity of water in J/Kg^0K

A_c = Collector area in m^2

U_L = Overall loss coefficient in $W/m^2 K$

F^I = Collector efficiency factor

The useful Energy Gained by the system is expressed as

$$Q_u = A_c \cdot F_R [I (\tau\alpha) - U_L (T_{fi} - T_a)] \quad (9)$$

$$\text{Total Input Energy} = \text{Total radiation (I)} + \text{pump work} \quad (10)$$

The energy efficiency is

$$\eta = \frac{\text{Useful heat energy gained by system}}{(\text{solar radiation} \cdot \text{collector area}) + \text{pump work}} \quad (11)$$

For $Re < 2300$ with constant heat flux and fully developed single-phase laminar flow in a circular tube, Nusselt (Nu) number is constant and independent from Reynolds (Re) and Prandtl (Pr) numbers.

$$Nu = \frac{hD}{k} = 4.36 \quad (12)$$

For fully developed turbulent flow where $0.5 < Pr < 2000$ and $3 \cdot 10^3 < Re < 5 \cdot 10^6$ in a circular tube with helical strip, Nusselt number is

$$Nu = \left(\frac{\omega D^2}{g} \right)^{0.5} \quad (13)$$

Since, forced circulation is made by a pump, nucleate boiling is avoided.

4.2 Procedure of Iteration

This model relies on calculating the working fluid outlet temperature, absorber plate temperature iteratively, the bottom heat loss coefficient, the edge heat loss coefficient and the useful heat energy. For each iteration the working fluid outlet temperature is set equal to the inlet conditions of the next iteration. In the beginning, the simulation model calculates the top heat loss coefficient by using Eq. (1). Since the absorber plate temperature (T_p) is necessary to determine the top heat loss coefficient (U_t) and T_p is unknown, an arbitrary value of 5^0C higher than fluid inlet temperature is assumed. Then, the useful heat gain (Q_u) is calculated using the above calculated U_t using Eq. (9). The Re is calculated to check the flow for laminar or turbulent so as to determine the heat transfer coefficient. Based on the flow type whether laminar or turbulent, the Nusselt (Nu) number is found out and the corresponding heat transfer

coefficient is calculated by using Eqs.(12) and (13). Other properties like thermal conductivity (k), density (q), specific heat (Cp) kinematic viscosity are calculated for the respective temperatures. Since the fluid outlet temperature is unknown, the absorber plate temperature is used to fix the initial fluid mean temperature and it is given by

$$T_m = \frac{T_p + T_{in}}{2} \quad (14)$$

Once, the heat gain is calculated, the fluid outlet temperature is calculated by

$$T_{out} = T_{in} + \frac{Q}{mCp} \quad (15)$$

Above fluid outlet temperature T_{out} is used in Eq. (16) to determine the new fluid mean temperature as

$$T_m = \frac{T_{in} + T_{out}}{2} \quad (16)$$

The above mean temperature determined by Eq. (14), is used to calculate the new Q_u . After recalculating the new Q_u , a new plate temperature is calculated as below.

$$T_p = T_{in} + \frac{Q_u / A_c}{F_R * U_t} (1 - F_R) \quad (17)$$

The iteration procedure re-calculates U_t and Q_u by using the new absorber plate temperature and the new fluid mean temperature. The iteration procedure is repeated in loop until the difference between T_p and its value in the previous iteration and T_{out} and its value in the previous iteration is lower than 0.01°C .

4.3 Software Technical Stack

The simulator software is developed in java platform which includes simulation of the mathematical model with iteration procedures for predicting the performance in varying conditions and a performance calculator to calculate the results based on experimental values. It includes the equipment SFPC with plain absorber plate and plain heat tubes and twisted tape inserted heat tubes. There is a possibility of plugging in open source analysis libraries like 'OPENFOAM' with this software package. The technical stack used for the software development are

JDK 1.8,
Spring 4.3.2 framework,
JSF 2.2,
Primefaces 6.0,
maven 3.5.2,
sqlite db.

5. Results

Simulation results were compared with the experimental results and the variation of fluid outlet temperature vs energy efficiency, Time vs efficiency, Time vs fluid outlet temperature, Time vs Radiation and Time vs wind speed are plotted as below.

In case of helical strip, the temperature distribution and velocity along the heat tube are disturbed. It makes the velocity to decrease because of the mixture of the water close to the top and bottom half of the tube more intensively, destroying the flow which generates relatively more eddy, leading to

mechanical energy dissipation. Apart from this, the insertion of twisted tape increases the contact surface area for the working fluid, thereby strengthening the frictional resistance also. Also, the hot and cold water mixture is getting mixed uniformly and hence the fluid temperature is uniform.

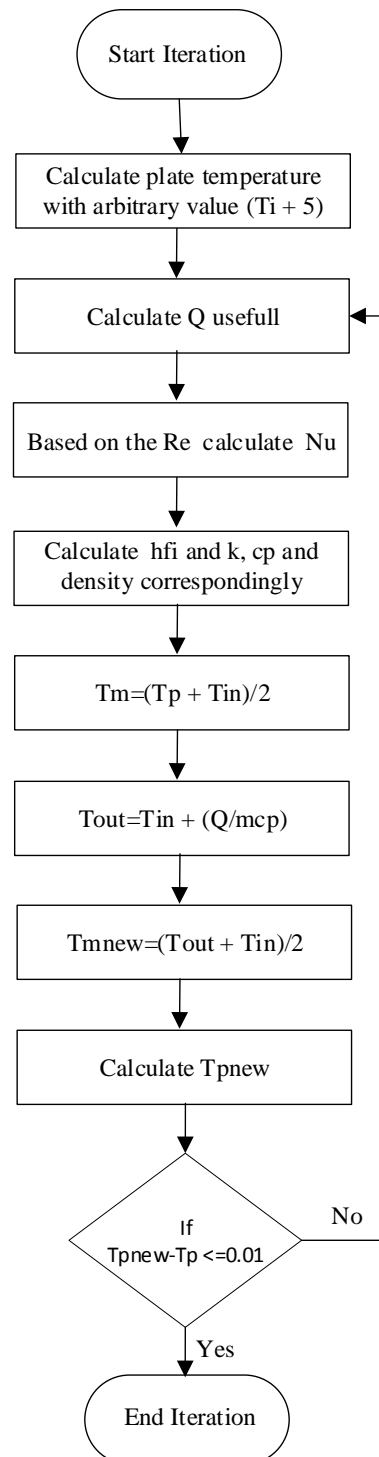


Figure 3. Iteration Flow chart

SFPC with Plain Heat Tube:

It is observed that the efficiency of the system decreases with increase in fluid outlet temperature. The simulator data and the experimental data variations are very close and the simulator data is validated and found to be within acceptable limits.

Also the variation in Fluid outlet temperature and efficiency in Day time hrs for both simulator and experiment are plotted as below.

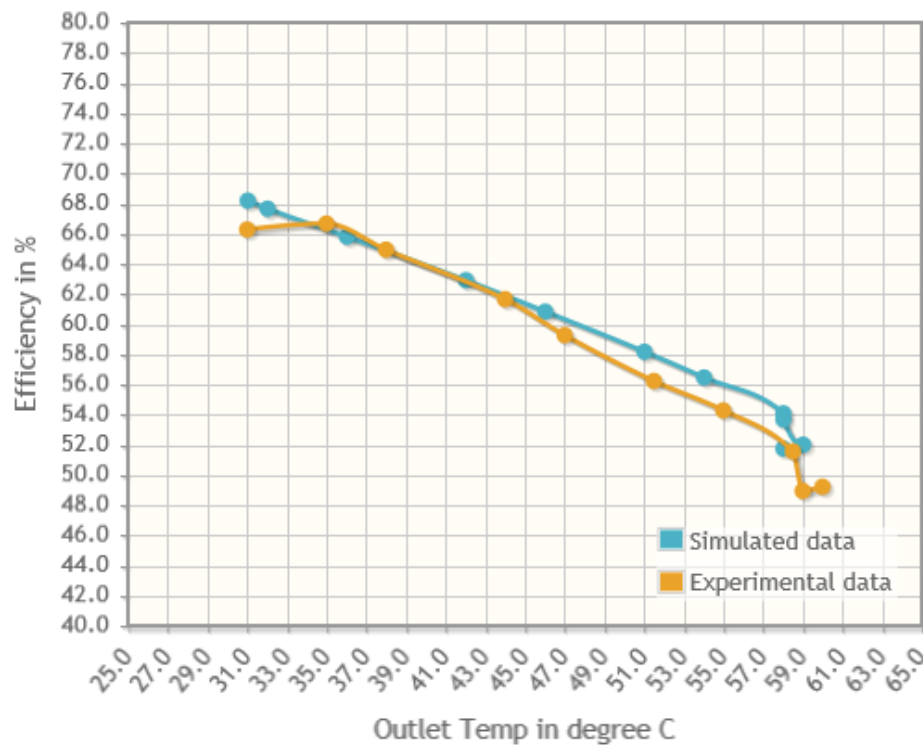


Figure 4. Efficiency vs Fluid outlet temperature

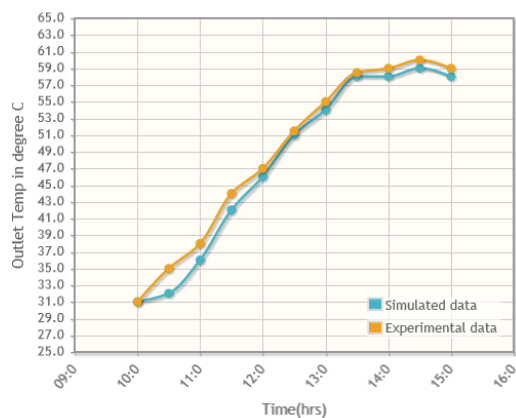


Figure 5. Outlet temperature vs Time (hrs)

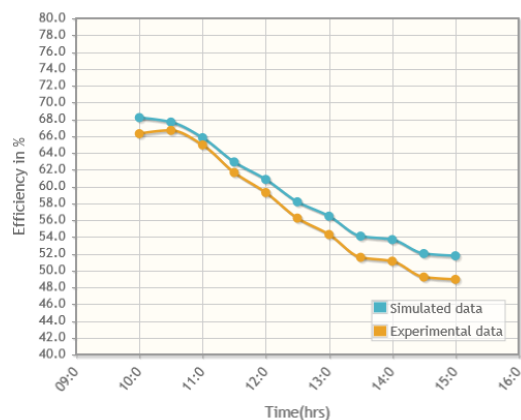


Figure 6. Efficiency vs Time (hrs)

Radiation level and wind speed are plotted for Day time variations. The change in radiation influences the efficiency and outlet temperature. Also the wind speed considerably influences the heat losses and thus the efficiency of the system.

In the helical strip inserted heat tube also, the efficiency of the system decreases with raise in the fluid outlet temperature the simulator and experimental data are very close and the deviations are within acceptable limits.

The efficiency and Fluid outlet temperature, radiation and wind speed variations in Day time hrs are as below.

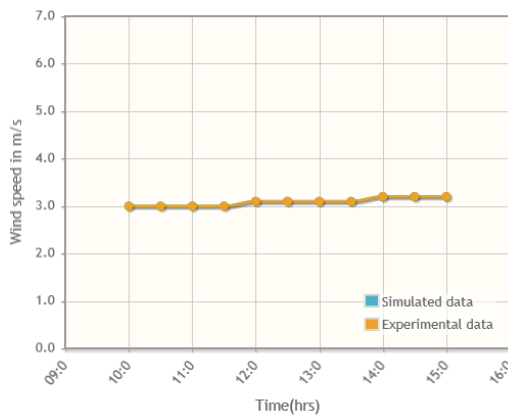


Figure 7. Wind speed vs Time (hrs)

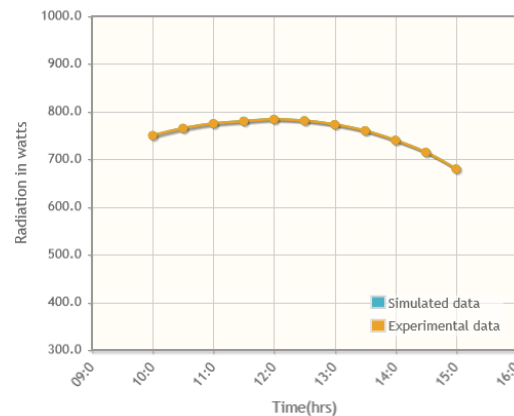


Figure 8. Radiation vs Time (hrs)

SFPC with Helical stripe inserted Heat Tube:

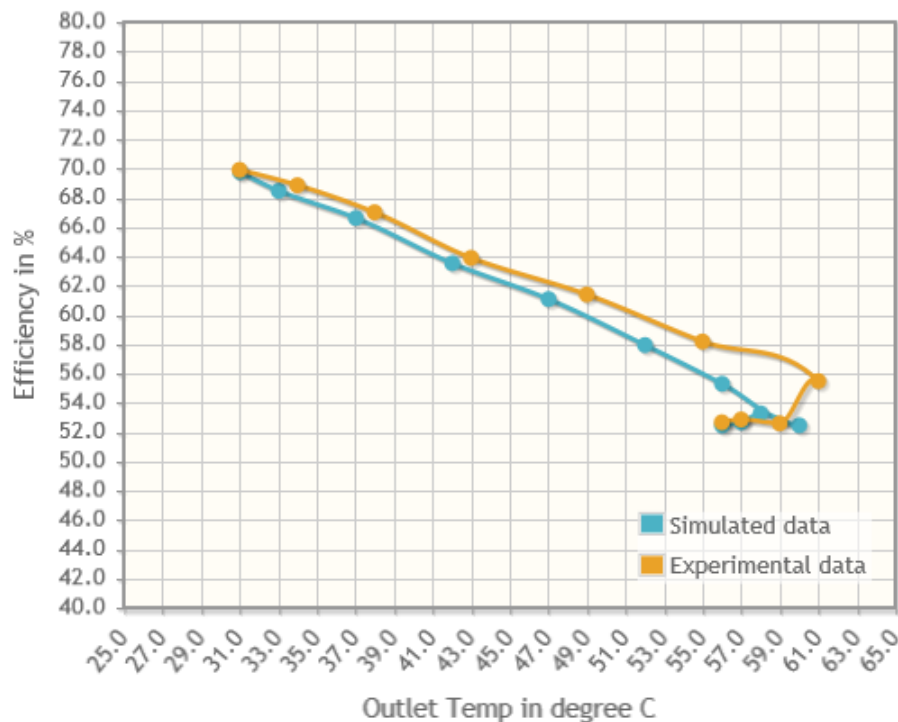
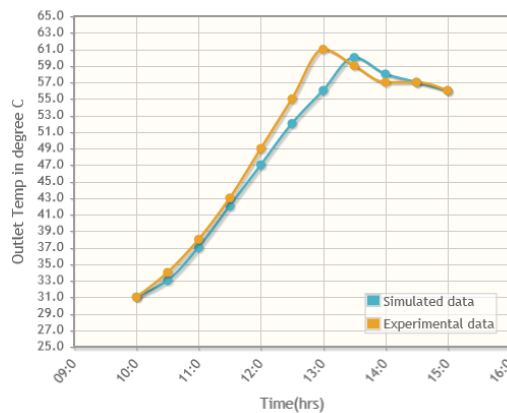
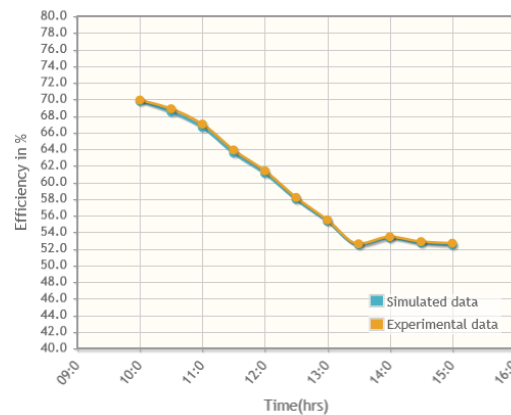
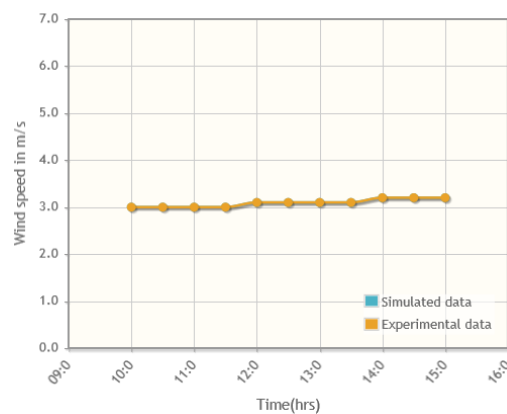
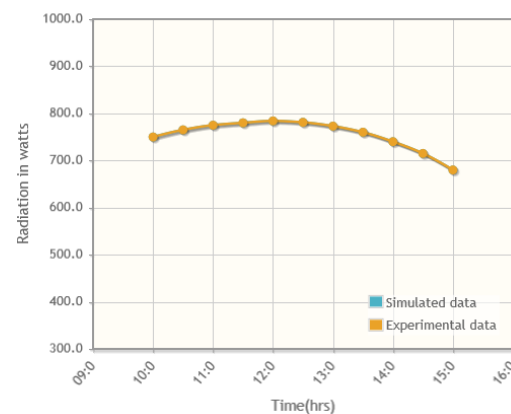


Figure 9. Efficiency vs Fluid outlet temperature

**Figure 10.** Outlet temperature vs Time (hrs)**Figure 11.** Efficiency vs Time (hrs)**Figure 12.** Wind speed vs Time (hrs)**Figure 13.** Radiation vs Time(hrs)

6. Conclusion

The mathematical model is developed for the solar flat plate collector to predict the performance in various operating conditions and geometrical change in heat tube by insertion of twisted strip. The simulator result data and the experimental result data are compared and found to be closely matching. The deviations are within 4% and are in an acceptable limit. The simulator is developed in Java using JDK 1.8, Spring 4.3.2 framework, JSF 2.2, Primefaces 6.0, maven 3.5.2 and sqlite db. The variations in system efficiency and fluid outlet temperature are observed for both simulator and experiment and found that the efficiency decreases with raise in fluid outlet temperature.

7. References

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