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Performance Evaluation of Thermosiphon Based Solar Water Heater in India

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***Abstract*—**This paper aims to study performance of a thermosiphon solar water heating system with the help of the proposed analytical model. This proposed model predicts the temperature and mass flow rate in a thermosiphon solar water heating system depending on radiation intensity and ambient temperature. The performance of the thermosiphon solar water heating system is evaluated in the Indian context. For this, eight cities in India are selected considering radiation intensity and geographical positions. Predicted performance at various cities reveals the potential for thermosiphon solar water in India.

***Keywords*—**Collector outlet temperature, India, solar water heater, thermosiphon.

I. INTRODUCTION

WORLD energy consumption is increasing continuously. The rise in energy consumption has lead to increase in use of conventional energy resources. Considering the disadvantages of conventional energy resources, solar energy is accepted as a clean and prominent energy resource. India is blessed with an abundant amount of solar energy due to its geographical position [1]. In a wide range of applications in both urban and rural regions, there is tremendous potential for heating and cooling, as well as warming green houses for agricultural crops, etc. There is a requirement of hot water for both domestic as well as industrial purposes. The use of hot water in the domestic sector is more in India, and thus, there is huge potential for a solar water heating system as far as its domestic hot water requirements are concerned. The thermosiphon solar water heating system finds its popularity in the domestic hot water sector.

Performance of the thermosiphon solar water heater has been studied by many researchers in last decade. Zeghib and Chaker [2] carried out the modeling of a domestic solar water heater for analyzing the influence of thermosiphonic flow rate and degree of stratification in a storage tank. Results showed that the stratified tank of a domestic solar water heater has the advantage of obtaining higher heat energy output compared to a fully mixed tank. Karaghouli and Alnaser [3] carried out the same study in the region of Bahrain. The study has reported that the developed thermosiphon solar water heater system has good potential to convert solar energy into useful energy in Bahrain.

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Joudi [4] investigated the performance of a domestic thermosiphon solar water heater for the sunny and cloudy atmospheres of Basra city in Iraq. The authors noted that the temperature distribution of water in the tank and collector tube is linear. Maximum collector efficiency is found in the month of June and the minimum in the month of February. Hawas and Muneer [5] tested a thermosiphon solar water heater in Benghazi to check its year round performance. They observed that maximum water temperature of 66˚C is reached in the month of August, whereas it was a minimum of 23˚C in February. Chien et al. [6] studied a thermosiphon solar water system for different radiation intensity and angle of inclination. They have concluded that efficiency decreases by 5% when the tilt angle of the collector is less that 15˚. Huang et al. [7] carried out experimentation on a newly designed mantle heat exchanger incorporated with a thermosiphon solar water heater. They observed that mean daily efficiency is lower for systems using the new design mantle heat exchanger compared to the existing system.

Chang et al. [8] studied 12 different configurations of solar water heater during the energy collecting phase and energy losing phase. The study has proposed a modified efficiency coefficient which is representative of the overall performance of the solar water heating system. This coefficient is a measure of the energy provided to the user from solar heating system. Chen et al. [9] studied the hourly, daily and long term performance of two phase thermosiphon solar water heater. A long term performance test was carried out in Taipei, Taiwan, where the system has shown characteristic efficiency 18% higher than conventional solar water heater. Esen and Esen [10] investigated a two phase closed thermosiphon for different refrigerants like R134a, R410A, R407C. They observed that maximum collector efficiency of 50.84% is obtained for systems using refrigerant R410A as a working fluid. Kang et al. [11] carried out experimentation on a two phase closed loop thermosiphon solar water heater for different working fluids like water, ethanol, and a mixture of water and ethanol. The authors have concluded that the ethanol and water mixture has given the best system efficiency and filling ratio of 50% to 60% was found most appropriate for the given system.

A careful review of literature shows that there is limited study of the thermosiphon solar water heater in the Indian context. Hence, this study aims to develop an analytical model that predicts the operating parameters of a thermosiphon solar water heater in order to study its performance in the Indian context.

II. Analytical Model and Validation

An analytical model is developed based on the schematic of the thermosiphon solar water heater, as shown in Fig. 1. The analytical model is based on the assumptions that flow is one dimensional and temperature distribution in the collector tubes is linear.

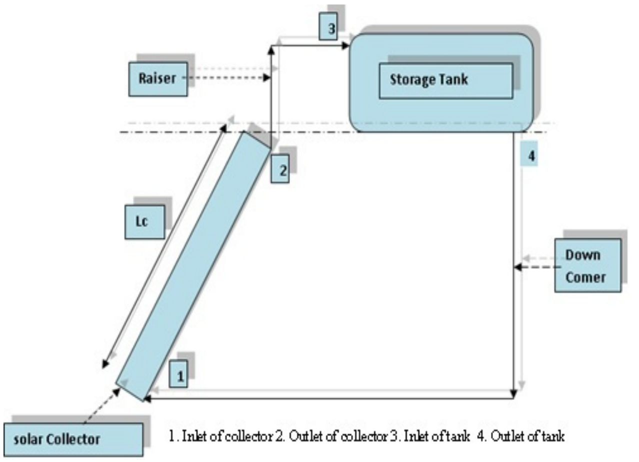

Fig. 1 Schematic of a thermosiphon solar water heater

A. Pressure Balance Equation

Buoyancy pressure drop across the entire loop is the sum of pressure drop in a collector and pressure drop across the raiser and down comer of system.

$$\Delta p_{Total} = \Delta p_c + \Delta p_r \tag{1}$$

where Δp_c is the pressure drop in a collector, Δp_r is the pressure drop across raiser and down comer.

$$\Delta p_{Total} = \rho g \beta \sin\theta \int_0^{Lc} (T_x - T) d\,x + g\,\rho \beta H (T_2 - T_1) \tag{2}$$

As temperature variation inside the collector tube is linear.

$$\frac{(T_x - T)}{x} = \frac{(T_2 - T_1)}{Lc} \tag{3}$$

Combining (2) and (3)

$$\Delta p_c = \rho g \beta \sin\theta \int_0^{Lc} \left(\frac{(T_2 - T_1)}{Lc}\right) x d\,x \tag{4}$$

$$\Delta p_c = \rho g \beta \sin\theta (T_2 - T_1) \left(\frac{Lc}{2}\right) \tag{5}$$

Substituting (5) in (1)

$$\Delta p_{Total} = \rho g \beta (T_2 - T_1) \left(\frac{Lc \sin\theta}{2} + H\right) \tag{6}$$

B. Frictional Pressure Drop

Total frictional loss in a loop is divided into two parts i.e. frictional loss in a collector tube and loss in other elements of the system such as the riser, down comer etc.

$$\Delta p_{Total} = \Delta p_{fc} + \Delta p_{other\ losses} \tag{7}$$

Δp_{fc} can be calculated as:

$$\Delta p_{fc} = f \times \frac{L_c}{d_c} \times \frac{1}{2} \times \rho \times V^2 \tag{8}$$

For laminar flow, friction factor is calculated as follows:

$$\therefore f = \frac{(64 \times \mu)}{(\rho \times V \times d_c)} = \frac{(64 \times \upsilon)}{(V \times d_c)} \tag{9}$$

Now, (8) becomes:

$$\therefore \Delta p_{fc} = 32 \times \upsilon \times \left(\frac{L_c}{{d_c}^2}\right) \times \rho \times V \tag{10}$$

Also

$$V = \frac{M}{(N \times \rho \times A)} = \frac{4 \times M}{(N \times \rho \times \pi \times {d_c}^2)} \tag{11}$$

substituting (11) into (10)

$$\Delta p_{fc} = 128 \times \left(\frac{\mu}{\rho}\right) \times \left(\frac{L_c}{\pi}\right) \times \frac{M}{(N * {d_c}^4)} \tag{12}$$

Therefore, (7) becomes:

$$\Delta p_{Total} = \left[128 \times \left(\frac{\mu}{\rho}\right) \times \left(\frac{L_c}{\pi}\right) \times \frac{M}{N \times {d_c}^4}\right] + \Delta p_{other\ losses} \tag{13}$$

The other frictional losses depend on the diameter and length of the raiser and down comer pipe. This also depends on the small bends or corners in the system. These losses are approximated as the multiple of frictional losses in a collector tube. This multiplying number is fixed for a given system of fixed dimensions. This number varies with change in the dimensions of the system.

Therefore, (13) can be modified as:

$$\Delta p_{Total} = \left[128 \times \left(\frac{\mu}{\rho}\right) \times \left(\frac{L_c}{\pi}\right) \times \frac{M}{N \times {d_c}^4}\right] + K_l \times \left[128 \times \left(\frac{\mu}{\rho}\right) \times \left(\frac{L_c}{\pi}\right) \times \frac{M}{N \times {d_c}^4}\right] \tag{14}$$

where K_l = multiplying number

$$\Delta p_{Total} = (1 + K_l) \left[128 \times \left(\frac{\mu}{\rho}\right) \times \left(\frac{L_c}{\pi}\right) \times \frac{M}{N \times {d_c}^4}\right] \tag{15}$$

Thermosiphon flow commences when the buoyancy pressure drop and frictional pressure drop in a given system becomes balanced. Therefore, combining (6) and (15) we get

$$\rho g \beta (T_2 - T_1) \left(\frac{Lc \sin\theta}{2} + H\right) - (1 + K_l) \times \left[128 \times \left(\frac{\mu}{\rho}\right) \left(\frac{L_c}{\pi}\right) \frac{M}{N \times {d_c}^4}\right] = 0 \tag{16}$$

In (16), the inlet temperature of the solar collector i.e. T_1, the outlet temperature of the solar collector i.e. T_2, mass flow rate i.e. M are three unknowns.

C. Heat Balance Equation

Heat absorbed by the solar collector is calculated as:

$$Q_c = I_T \times (\tau\alpha) \times A_c \left[FR(\tau\alpha) - FR(UL)\frac{(T_1 - T_a)}{I_T}\right] \quad (17)$$

Heat gain by water in a collector tube = Useful heat gain from collector.

$$Q_c = M \times C_p \times (T_2 - T_1) \quad (18)$$

Comparing (17) and (18)

$$M \times (T_2 - T_1) - \left(\frac{I_T \times (\tau\alpha) \times A_c}{C_p}\right) \times \left[FR(\tau\alpha) - FR(UL) \times \frac{(T_1 - T_a)}{I_T}\right] = 0 \quad (19)$$

D. Heat Loss in Raiser

As the diameter and length of the riser and heat loss coefficient for a riser pipe is not known prior, heat loss from the pipe is assumed to be as 20% of total heat carried by the water at 2, i.e. the outlet of the collector. If radiation intensity is less than the intensity at sunrise, then this heat loss is assumed to be 10% of total heat carried by the water at 2.

$$T_3 - 0.9\, T_2 = 0 \text{ for } I_i < I_T \quad (20)$$

$$T_3 - 0.8\, T_2 = 0 \text{ for } I_i > I_T \quad (21)$$

E. Heat Loss in Down Comer Pipe

As the diameter and length of down comer and heat loss coefficient for down comer is not known prior, heat loss from the pipe is assumed to be 15% of the total heat carried by the water at 4, i.e. the outlet of storage tank. If radiation intensity is less than the intensity at sunrise, then this heat loss is considered as 30% of the total heat carried by the water at 4.

$$T_1 - 0.85\, T_4 = 0 \text{ for } I_i < I_T \quad (22)$$

$$T_1 - 0.7\, T_4 = 0 \text{for } I_i > I_T \quad (23)$$

F. Ratio of Heat Loss

The heat loss balance in the raiser pipe can be written as:

$$M \times C_p \times (T_2 - T_3) = U_{ct} \times A_{23} \times \left(\frac{T_2 + T_3}{2}\right) \quad (24)$$

The heat loss balance in the down comer pipe can be written as:

$$M \times C_p \times (T_4 - T_1) = U_{ct} \times A_{41} \times \left(\frac{T_4 + T_1}{2}\right) \quad (25)$$

Taking the ratio of (24) and (25)

$$\frac{(T_2 - T_3)}{(T_4 - T_1)} = \frac{A_{23} \times \left(\frac{T_2 + T_3}{2}\right)}{A_{41} \times \left(\frac{T_4 + T_1}{2}\right)} \quad (26)$$

The diameter of the raiser and down comer is considered as the same.

$$\therefore \frac{(T_2 - T_3)}{(T_4 - T_1)} - \frac{L_r \times \left(\frac{T_2 + T_3}{2}\right)}{L_d \times \left(\frac{T_4 + T_1}{2}\right)} = 0 \quad (28)$$

After solving (16), (19)-(23) and (27), the temperature of the fluid at every location of the solar water heater and mass flow rate are obtained. Based on this value performance, study is further carried out.

III. Validation of Analytical Model

Based on the above methodology, a MATLAB code is developed. The developed analytical model is validated with the experimental study carried out by Koffi et al. [12]. The authors have carried out an experimental study at Cote d'Ivoire, in Yamoussoukro, an African country. Figs. 2 and 3 show a comparison of the mass flow rate from the present analytical model with the experimental results obtained by Koffi et al. [12].

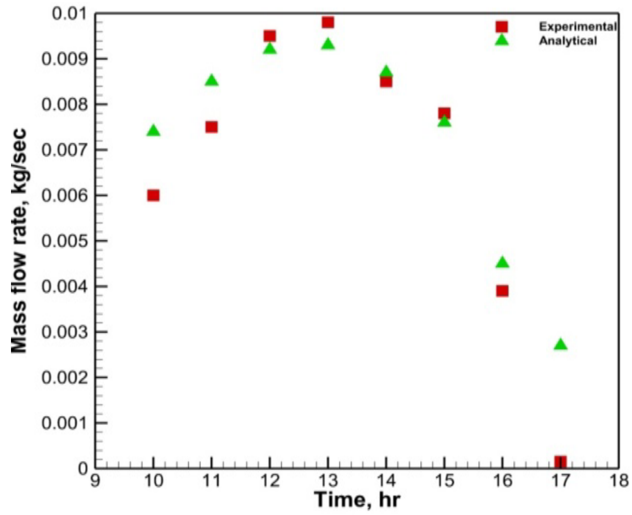

Fig. 2 Comparison of mass flow rate

It can be observed from Fig. 2 that the results from the present analytical model show good agreement with the experimental results from [12]. Some deviation can be seen between the results from the experimental and present study; however, this deviation is well within the specified limit.

IV. Performance Investigation in Indian Context

In the present study, performance of a thermosiphon solar water heater is checked for various locations in India. This study is carried out for the particular month of December when hot water requirement is at its peak.

A. Selection of Location

The selection of a particular location is needed in order to track the variation of geographical conditions. Thus, the location is selected based on radiation intensity over a region and climatic conditions. Radiation intensity is considered from annual global horizontal radiation over India [1]. A total of eight cities are selected based on low, medium and high radiation intensity over India. In the low radiation intensity region, Kolkata, Varanasi and Delhi are selected. In the medium radiation intensity region, Ahmadabad, Pune, Hyderabad and Bangalore are selected. In the high radiation intensity region, Jodhpur is selected. Apart from this, these selected eight cities are also representative of different states of India.

B. Radiation Intensity Over the Selected Cities

Radiation intensity data is taken from IMD publication [13]. Publication has provided the average data of radiation intensity and ambient temperature from 1986 to 2000. Mean monthly data is taken for one month of the winter season i.e. December. Fig. 3 shows the hourly global radiant exposure for the given cities in the month of December. Hyderabad has the highest radiation exposure whereas it is least for the city of Varanasi.

Variation in ambient temperature is shown in Fig. 4. In the month of December, Ahmadabad shows the highest ambient temperature of 28.3℃ whereas Delhi shows lowest ambient temperature of 14.4℃.

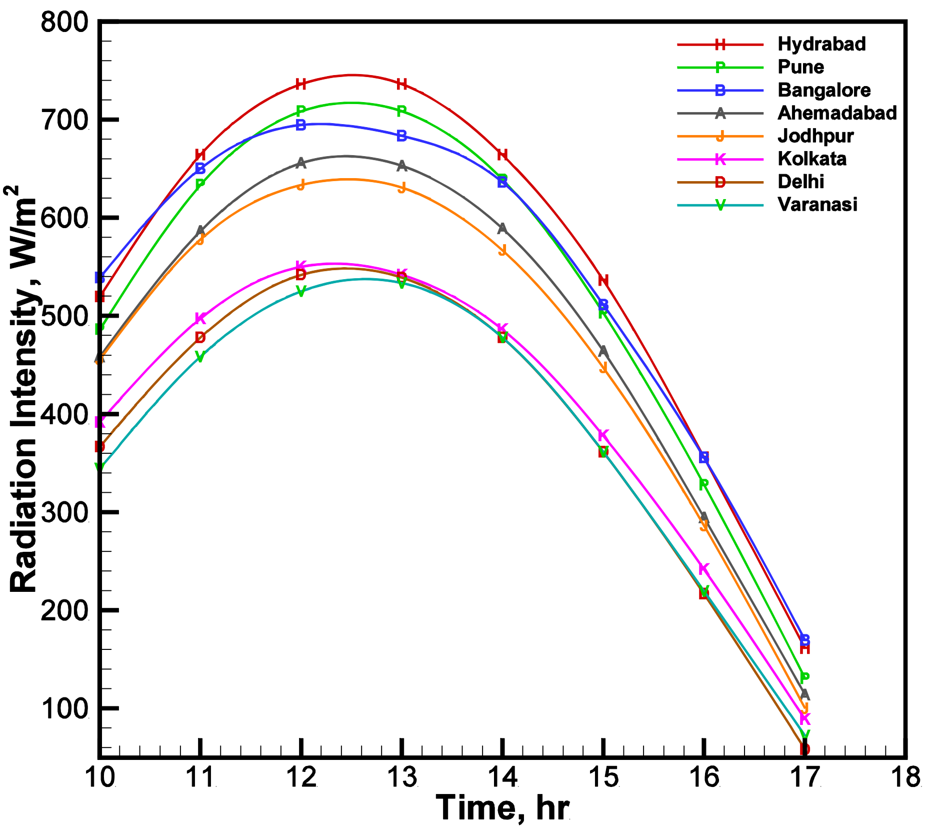

Fig. 3 Radiation intensity for different cities

B. Performance of Thermosiphon Solar Water Heater over India

Fig. 5 shows the variation of the predicted collector outlet temperature with time for different cities. The lowest collector outlet temperature is noted for Delhi. Lower radiation intensity over Delhi during the month of December justifies the decrease in the collector outlet temperature. Kolkata also lacks in gaining the collector outlet temperature along with the city of Varanasi. The highest gain in the collector outlet temperature is seen in the case of city of Hyderabad. The collector outlet temperature increases with radiation intensity. The inlet collector temperature varies between 37℃ to 45℃. Also, there is a steady rise in ambient temperature, while the radiation intensity is increasing. This steady rise in ambient temperature contributes to an increase in the collector outlet temperature. After 13.00 p.m., as radiation intensity decreases, a decrement is observed in the collector outlet temperature.

Lowest collector outlet temperature is noted for Delhi. Lower radiation intensity over Delhi during the month of December justifies the decrease in the collector outlet temperature.

The cumulative comparison of ambient temperature, radiation intensity, collector outlet and inlet temperature for Hyderabad is shown in Fig. 6. The collector outlet temperature increases, as there is a rise in radiation intensity i.e. up to 750 W/m^2. The inlet collector temperature varies between 37℃ to 45℃. Also, there is a steady rise in ambient temperature, which contributes to an increase in the collector outlet temperature. After 13.00 p.m. as radiation intensity decreases, there is a decrease in the collector outlet temperature.

Delhi has performed least in the group in terms of gain in the collector outlet temperature. Therefore, the cumulative comparison for Delhi is shown in Fig. 7. The collector outlet temperature is following the same pattern as in the case of the city of Hyderabad. But, gain in the collector outlet temperature is still less in the case of Delhi due to the low radiation intensity over the city.

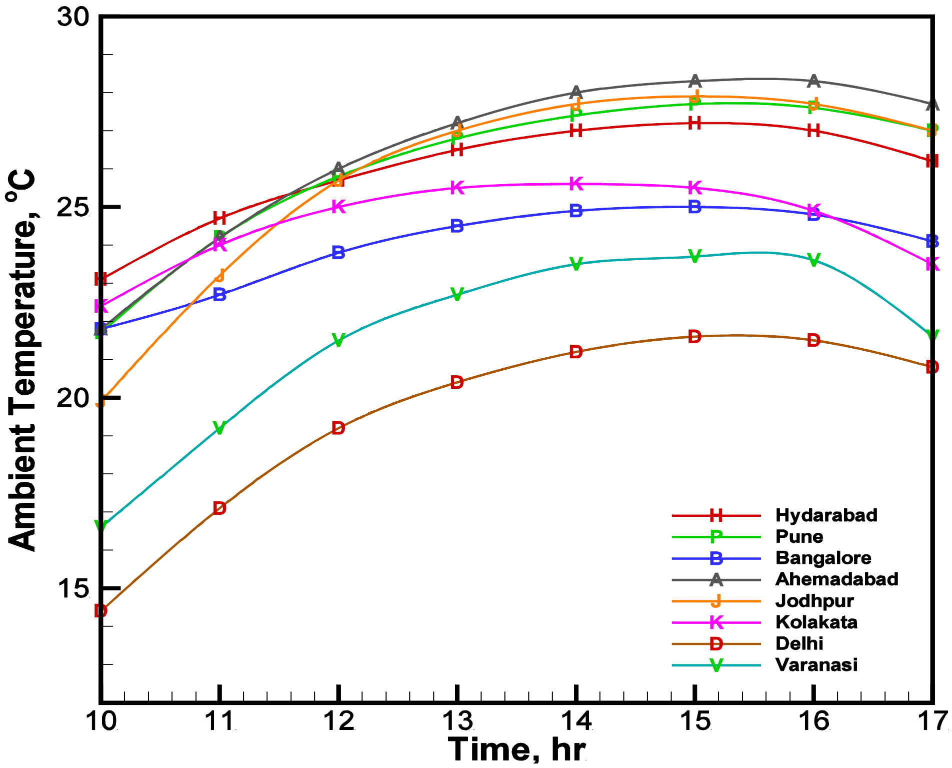

Fig. 4 Ambient temperature for different cities

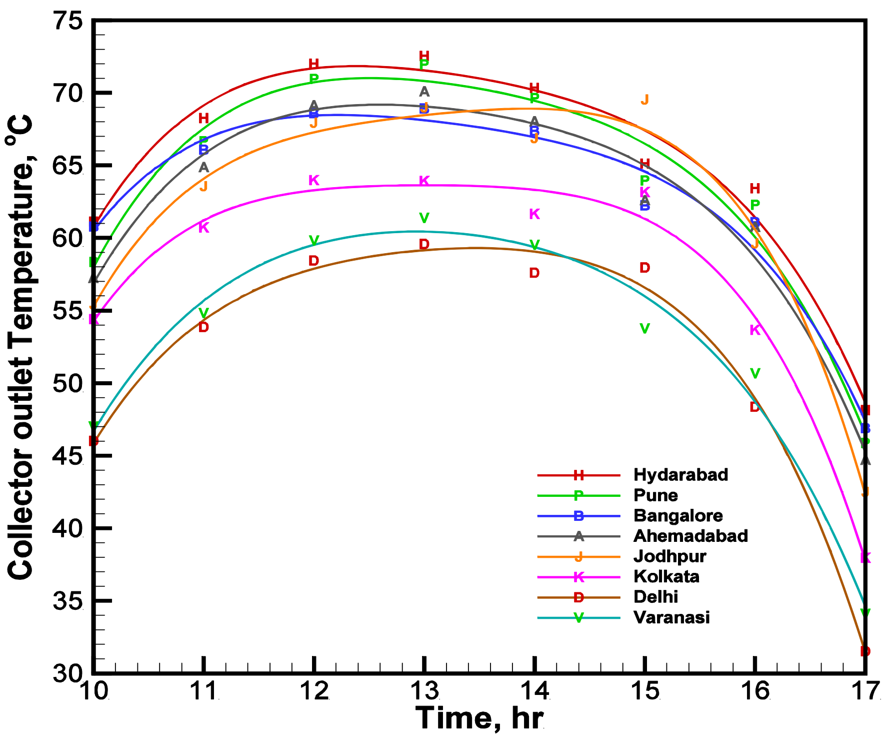

Fig. 5 Collector outlet temperature

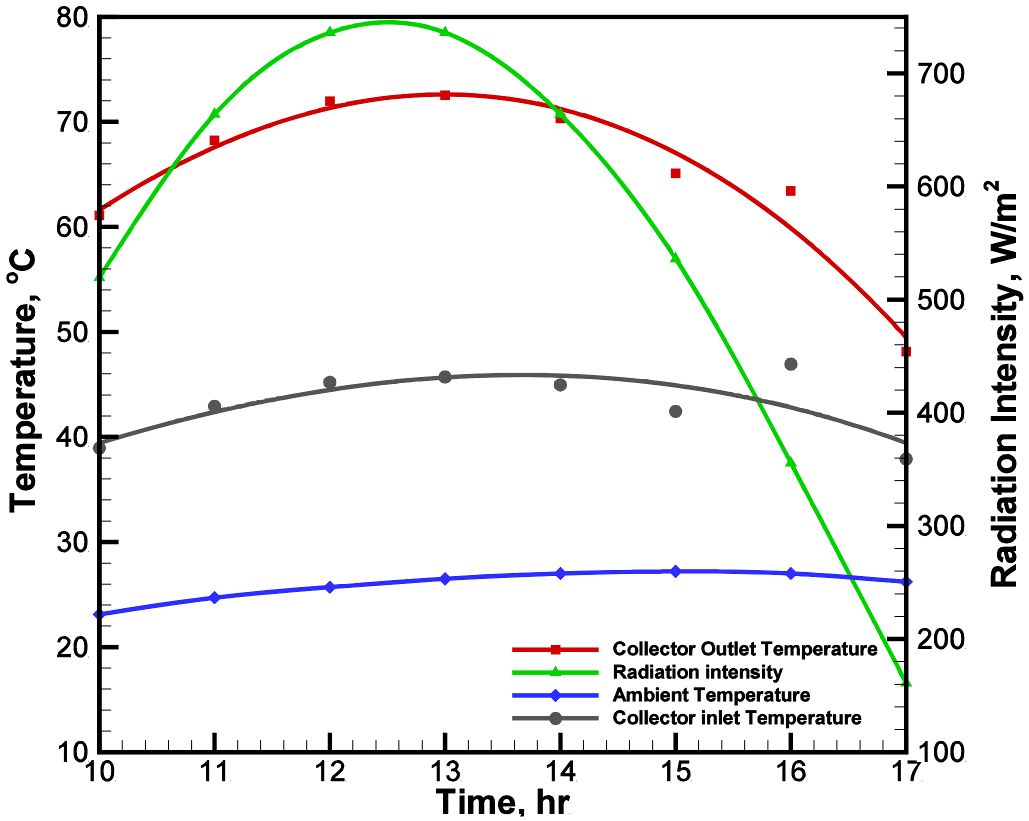

Fig. 6 Cumulative comparison for Hyderabad city

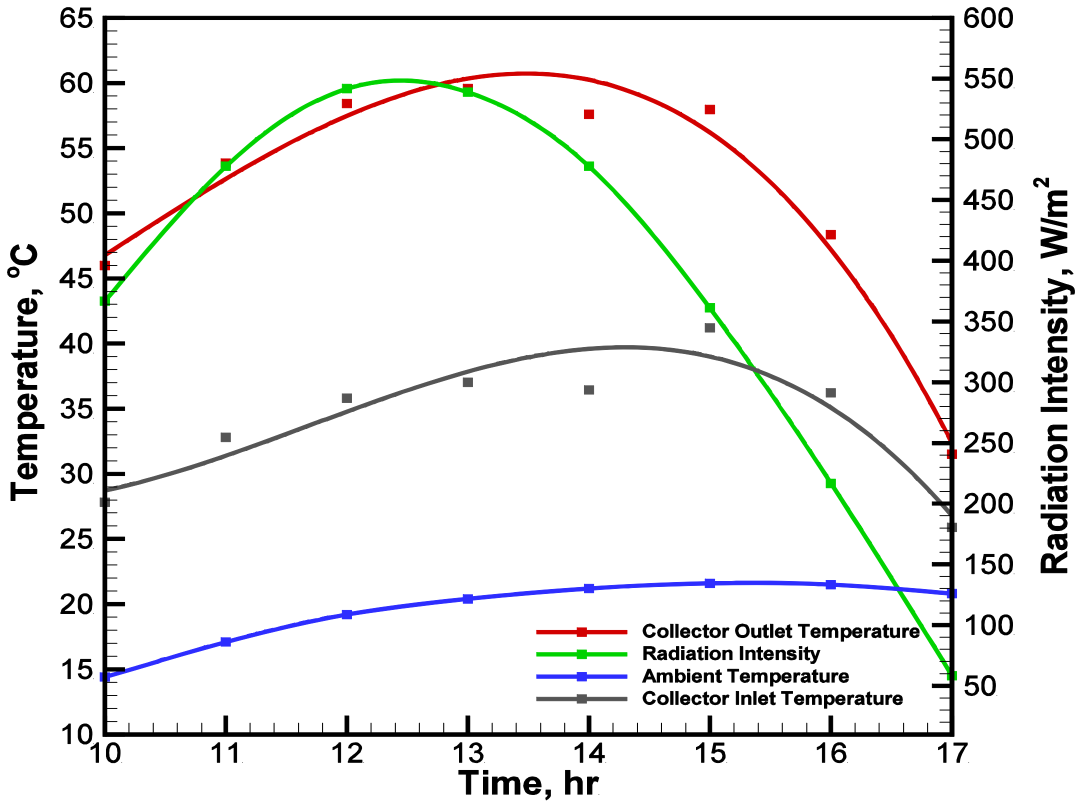

Fig. 7 Cumulative comparison for Delhi city

Also, lower ambient temperature variation is another reason. Radiation intensity reaches to its maximum value during 13.00 p.m. to 14.00 p.m. at around 550 W/m2, which is very low compared to other cities for the given time.

Overall efficiency is also an important parameter which is to be considered for the performance of the thermosiphon solar water heater. Thus, the overall efficiency for the thermosiphon solar water is calculated as suggested by [8].

$$\eta = \frac{M \times C_p \times (T_2 - T_1)}{A_c \times I} \tag{28}$$

Using (28), the overall efficiency is calculated for different cities and is shown in Fig. 8. Hyderabad has shown the highest overall efficiency of 54%. Increase in radiation intensity and collector outlet temperature has lead to an increase in the overall efficiency of Hyderabad city. The performance of other cities like Pune, Ahmadabad, and Bangalore is seen as satisfactory with the collector efficiency reaching an average value of overall efficiency of 49%. Jodhpur and Varanasi have shown an average overall efficiency of 47%. Kolkata and Delhi has least performance in a group with an average overall efficiency of 46%.

Fig. 9 shows the average values of the collector outlet temperature over India. The background of the map shows the annual global horizontal radiation over India [1]. From Fig. 9, it is clear that thermosiphon solar water heater can produce an average temperature around 60℃ at collector outlet, except is the region between the cities Delhi and Varanasi. When hot water is used for domestic purposes, especially in the winter season, the required temperature of the water is around 50℃. Thus, there is difference of 10℃ between the average collector outlet temperature and the required temperature for domestic use. Hence, with proper storage media at the collector outlet, the temperature of the hot water can be retained to the required temperature even after sunshine hours. Therefore, a thermosiphon solar water heater with the given specifications of dimensions shows good performance over India in the winter season, i.e. month of December.

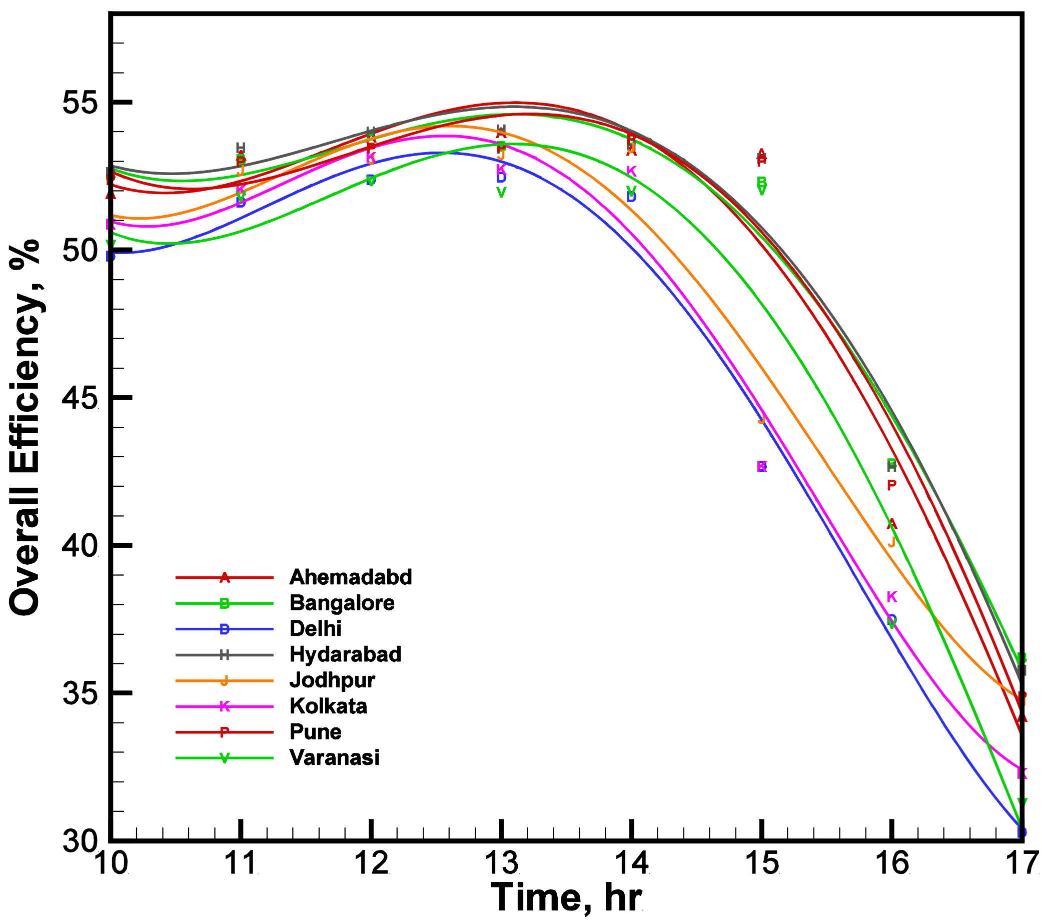

Fig. 8 Overall efficiency for different cities

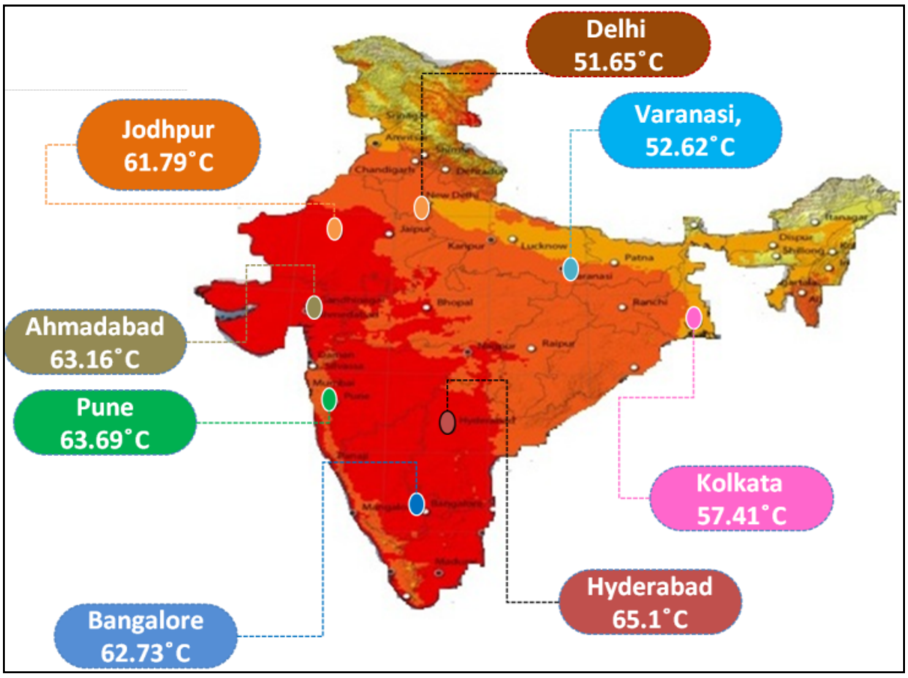

Fig. 9 Average Collector outlet Temperature over India

V. Conclusion

Performance of a thermosiphon solar water heater at various Indian locations is predicted using the proposed analytical model. Various locations are selected across India depending on radiation intensity and geographical positions. A total of eight cities are selected viz. Delhi, Varanasi, Kolkata, Hyderabad, Bangalore, Pune, Ahmadabad, and Jodhpur. Performance prediction at various Indian locations is carried out for a particular month of the winter season, i.e. December. The thermosiphon solar water heater has shown best performance for the city of Hyderabad. It showed highest collector outlet temperature of 65.1℃. Cities like Bangalore, Pune, Ahmadabad, and Jodhpur showed good performance giving an average collector outlet temperature of around 62℃. Performance is poor for cities like Delhi, Varanasi, and Kolkata, in which the average collector outlet temperature is around 55℃. Thus, a thermosiphon solar water heater can provide hot water at an average temperature of around 60℃ at various locations in India in the winter season. With effective storage, the media temperature of the hot water can be maintained even after the sunshine hours. Hence, the present analytical model may provide its importance in studying the performance of thermosiphon solar water heaters. With the help of this study, it is concluded that the thermosiphon solar water heater has good potential in India, especially in the winter season.

Nomenclature

β = coefficient of thermal expansion [1/K],
g = gravitational constant [m^2/s],
ɵ = angle of inclination of a collector,
Lc = Length of collector tubes [m],
H = Height of a storage tank [m],
Δp_{fc} =Frictional Losses in a collector tubes,
$\Delta p_{other\ losses}$ = Losses in other elements of system,
f =friction factor,
$\frac{L_c}{d_c}$ =ratio of collector length to diameter,
V =velocity of fluid [m/s],
υ = Dynamic viscosity [N-s/m2],
µ =Kinematic viscosity [m^2/s],
N = number of collector tubes,
I_T =Incident solar radiation [W/m^2],
τα = Fraction of solar radiation absorbed,
FR = Collector heat removal factor,
UL = Overall loss coefficient [W/m^2K],
T_1 =Inlet temperature of collector [K],
A_c =Area of collector [m^2],
T_a =Ambient temperature, [K],
T_3 =Temperature at oullet of riser pipe [K],
T_4 =Temperature at inlet of down comer pipe [K],
T_1 =Temperature at oullet of down comer pipe [K],

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