THEORETICAL INVESTIGATION OF INTEGRATED MANIFOLD SOLAR WATER HEATER

INVESTIGAȚIE TEORETICĂ A ÎNCĂLZITORULUI SOLAR DE APĂ CU COLECTOR INTEGRAT

Pradeep PATANWAR[1], T V ARJUNAN[2], Manikant PASWAN[3], Anulal MAHTO[4]

Abstract: *A more elaborated mathematical model is developed in the current research to examine the thermal efficiency of a solar water heater with an evacuated tube collector (ETC). The configuration investigated is that of six copper tubes that are positioned evenly in a single evacuated tube whereby one tube is placed centrally and the other five positioned around the central tube in equal radial directions. This multi tube design aims at maximising the heat transfer and improving the effective heat-exchange surface area and improving an even temperature field in the collector. It derives energy balance equations of each tube considering conductive, convective and radiative heat transfer mechanisms and thermal interaction between neighbouring tubes. Correlations of heat-transfers are formulated and used to assess the temperature distributions and thermal characteristics of system on different operating conditions. The mathematical model developed is solved to give forecasts of outlet water temperature, useful heat gain as well as the overall thermal efficiency of the solar water-heating system. The measurements show that there is a great enhancement in thermal performance, when compared with traditional single-tube systems, and the overall system efficiency is seen to be improved by about 41 per cent. The suggested design and method of analysis offer invaluable information on the optimization of the evacuated tube solar water heaters and lead to the creation of solar thermal systems with high efficiency to be used both at home and at work.*

[1] Assistant Professor, Department of Mechanical Engineering, Guru Ghasidas Vishwavidyalaya, Bilaspur, Chhattisgarh, e-mail: pradeep.patanwar@gmail.com
[2] Professor, Department of Mechanical Engineering, Guru Ghasidas Vishwavidyalaya, Bilaspur, Chhattisgarh, email:arjun_nivi@yahoo.com
[3] Director, Sant Longowal Institute of Engineering and Technology, Punjab, email: manikant.nit@gmail.com
[4] Associate Professor, Workshop Superintendent, Department of Mechanical Engineering, Bilaspur, Chhattisgarh, email: anulalmahto@gmail.com

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Rezumat: *Un model matematic mai elaborat este dezvoltat în cercetarea curentă pentru a analiza eficiența termică a unui încălzitor solar de apă cu colector cu tuburi vidate (ETC). Configurația investigată constă din șase tuburi din cupru poziționate uniform într-un singur tub vidat, dintre care un tub este amplasat central, iar celelalte cinci sunt dispuse în jurul acestuia pe direcții radiale egale. Acest design cu mai multe tuburi are ca scop maximizarea transferului de căldură, mărirea suprafeței efective de schimb termic și obținerea unui câmp de temperatură mai uniform în colector.Sunt derivate ecuațiile de bilanț energetic pentru fiecare tub, luând în considerare mecanismele de transfer termic prin conducție, convecție și radiație, precum și interacțiunea termică dintre tuburile învecinate. Corelațiile de transfer de căldură sunt formulate și utilizate pentru a evalua distribuțiile de temperatură și caracteristicile termice ale sistemului în diferite condiții de funcționare. Modelul matematic dezvoltat este rezolvat pentru a furniza predicții ale temperaturii apei la ieșire, ale câștigului util de căldură și ale eficienței termice globale a sistemului de încălzire solară a apei. Măsurătorile indică o îmbunătățire semnificativă a performanței termice în comparație cu sistemele tradiționale cu un singur tub, eficiența globală a sistemului fiind crescută cu aproximativ 41%. Designul și metodologia de analiză propuse oferă informații valoroase pentru optimizarea încălzitoarelor solare de apă cu tuburi vidate și contribuie la dezvoltarea unor sisteme solare termice cu eficiență ridicată, utilizabile atât în aplicații rezidențiale, cât și industriale.*

Cuvinte cheie: Model matematic, Încălzitor solar de apă, Colector cu tuburi vidate, Tub de cupru, Eficiență termică

1. Introduction

The growing energy demands in the world, alongside the exhaustion of fossil-energy sources and the growing environmental activism has heightened the necessity of sustainable and renewable energy technologies. Solar energy is one of the renewable opportunities; it is a clean, free and has a wide availability to fulfill the thermal energy needs, especially in domestic and industrial water-heating purposes. The solar water heaters (SWHs) can be considered one of the oldest and most affordable solar-thermal technologies that play an important role in energy preservation and decreasing greenhouse-gas emissions. One of the major industries to harness solar energy is domestic and industrial water heating system which is mainly through the adoption of solar water heaters. Such systems are very popular because they need minimum maintenance, they are easy to design, manufacture, and operate [7]. The individual type of collector to have gained significant interest is the evacuated-tube collectors (ETCs) due to its high thermal performance when compared to the flat-plate collectors, particularly in low ambient temperature and diffuse radiation conditions. The vacuum envelope that surrounds the absorber is of much essence in decreasing convective and conductive losses and thus, allowing efficiencies of over 70% in actual applications [3]. Therefore, solar water heaters that are based on ETCs are being widely used in the residential, business, and institutional market. Concepts such as water-in-glass thermo-syphon tubes, U-tube metal inserts, nano-fluid in solar collector, roughened absorber surface and heat-pipe configurations have been commercialized and each has been demonstrated to have unique strengths and weaknesses in terms of pressure resistance, freeze capability, manufacturability, cost [1,6,7,8,9,10,11,12,14,]. Recent studies have been directed on enhancing the thermal contact between the absorber and the heat removal channel. Traditional U-tube ETCs also use copper fins and air gaps, which come with a great deal of thermal resistance. To overcome this, filled type evacuated tube with U-tube (UFET) concept was suggested in which a conductive filling layer is used to get rid of fin-absorber gap and increase heat transfer to working fluid [2]. The preeminent losses and the effect of flow rate and tube geometry have been further explained using detailed resistance network based analytical models [5]. Budihardjo and Morrison have carried out extensive experimental studies on glass filled evacuated tubes and have made models using Transient System Simulation Tool (TRNSYS) to predict long term performance in a wide range of climatic conditions. According to their results, despite high optical efficiency, natural-circulation flow rate, tank losses and daily draw-off patterns, overall system performance is modulated [4]. However, standard ETC designs are usually limited by a low area of heat-transfer surface and non-uniform temperature distributions in the absorber, so this does not permit a subsequent increase in thermal efficiency. To overcome these issues, a number of studies have examined design changes and internal flow improvements such as the addition of heat pipes, U -tube absorbers, and multi-channel flow configurations. Such studies have revealed that the effective heat-transfer area and optimization of the mechanism of internal heat exchange could considerably increase the performance of the collector. The current designs however tend to have complicated geometries, high manufacturing cost or even operational limitations; hence restricting their feasibility. In addition, the application of multi-tube geometry to a single evacuated tube with comprehensive mathematical modeling is also limited in literature. The current research paper in this regard suggests a new internal absorber pattern of an evacuated-tube solar water heater, which consists of six copper tubes symmetrical in one evacuated tube. A central tube is placed in the middle of the tube, with the rest of the five tubes placed at an equal distance in all directions. The aim of this setup is to maximize the absorption of heat, even thermal contact between the tubes, and minimization of temperature difference in the collector. An elaborate mathematical model is established to derive the equations of energy-balance in each of the tubes with consideration of inter-tube thermal resistances and-heat loss to the external environment. The suitable heat-transfer correlations are used to study the system performance at the realistic operating conditions. The main aim of the work is to assess the thermal characteristics and efficiency improvement of the proposed multi-tube construction with the help of the rigorous analytical method. The findings that were made with the developed model are that the overall efficiency of the solar water-heating system has increased significantly, and the improvement is about 41 per cent as compared to traditional systems. These results offer useful information about optimization of the evacuated-tube collector design and offer a good perspective of the future of high performance solar water-heating device.

2. Mathematical Modeling

In the current study, a composite consisting of six copper conduits is used in which a single tube is placed in the centre and the other 5 tubes are placed symmetrically around it with an equal distance in between. All the tubes are positioned parallel to each other in the same longitudinal axis, and thus maintain a constant interstitial distance amongst them. Fluid is added through the centre conduit and flowing through the surrounding five conduits, each is attached at its lower end of the conduit to a common manifold. Following assumptions are considered during the theoretical analysis-

1) The flow is steady one-dimensional and single-phase;
2) The solar radiation is uniform;
3) Edge losses are negligible;
4) Conductive cooling of glass is insignificant;
5) The convective heat transfer coefficient of copper tube is constant;
6) The convective heat transfer coefficient and radiative heat transfer coefficient of the ETC tube is constant;

7) Properties of fluid and materials are constant;

8) The fluid flow inside the copper tube is water, which is considered as incompressible.

Based on the energy balance equation, the amount of heat that the water will acquire is equal to the net heat input which is the amount of heat absorbed by the coated surface of the solar evacuated tube less the quantity of heat lost to the environment. The equation is given by

$$Q_a = Q_T - Q_L = A_L [I_T (\tau\alpha)_e - U_L(T_u - T_a)] = \dot{m} c_p (T_o - T_i) \qquad (1)$$

where A_L represents the luminance area, I_T denotes the incident solar insolation, 'U_L' is the heat-loss coefficient of the solar evacuated tube collector. The term $(\tau\alpha)_e$ refers to the effective transmittance–absorbance product. The symbol '$\dot{m}$' indicates the mass flow rate of water, while 'c_p' is the specific heat of water at constant pressure. 'T_u' is the outer surface temperature of absorber tube, and 'T_i' and 'T_o' correspond to the inlet and outlet temperatures of the U-tube, respectively.

Since edge losses are assumed to be negligible, the overall heat-loss coefficient 'U_L' can be expressed as follows.

$$U_L = \frac{1}{\frac{1}{h_{ga}} + \frac{1}{h_{ugr}}} \qquad (2)$$

where, h_{ga} is the convective heat transfer coefficient from outer glass tube to the surrounding and h_{ugr} is the heat transfer coefficient due to radiation between inner glass and outer glass.

The h_{ugr} changes with the temperatures of the outer surface of absorber tube T_u and the outer glass tube T_g, which can be expressed as

$$h_{ugr} = \sigma\, \varepsilon_u\, (T_u^2 + T_g^2)\, (T_u + T_g) \qquad (3)$$

where, σ is the Stefan-Boltzmann constant having the value of 5.670374 x 10^{-8} W/m^2K^4 and ε_u is the emissivity of the coating present on the absorber's tube

The efficiency of the system can be defined by-

$$\eta = \frac{Q_a}{A_p I_T} \qquad (4)$$

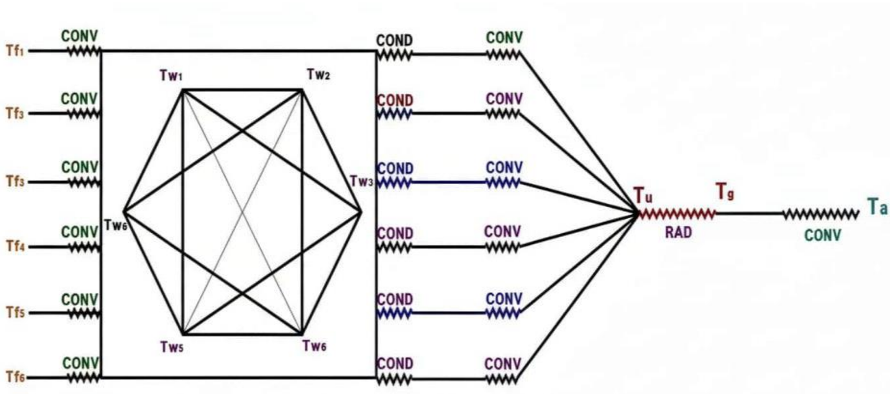

Fig 1. Thermal Network Diagram

The temperatures of all the pipes along with the temperatures of the liquid water flowing inside each pipe are being shown in fig.5. The energy balance equation for the liquid water flowing inside each pipe can be defined as-

$$Q_a = - \dot{m} c_p \frac{dT_{f6}}{dY} = \frac{T_{f6} - T_u}{R_6} + \frac{T_{f6} - T_{f1}}{R_{16}} + \frac{T_{f6} - T_{f2}}{R_{26}} + \frac{T_{f6} - T_{f3}}{R_{36}} + \frac{T_{f6} - T_{f4}}{R_{46}} + \frac{T_{f6} - T_{f5}}{R_{56}} \quad (5)$$

$$Q_a = \dot{m} c_p \frac{dT_{f1}}{dY} = \frac{T_{f1} - T_u}{R_1} + \frac{T_{f1} - T_{f2}}{R_{12}} + \frac{T_{f1} - T_{f3}}{R_{13}} + \frac{T_{f1} - T_{f4}}{R_{14}} + \frac{T_{f1} - T_{f5}}{R_{15}} + \frac{T_{f1} - T_{f6}}{R_{16}} \quad (6)$$

$$Q_a = \dot{m} c_p \frac{dT_{f2}}{dY} = \frac{T_{f2} - T_u}{R_2} + \frac{T_{f2} - T_{f1}}{R_{21}} + \frac{T_{f2} - T_{f3}}{R_{23}} + \frac{T_{f2} - T_{f4}}{R_{24}} + \frac{T_{f2} - T_{f5}}{R_{25}} + \frac{T_{f2} - T_{f6}}{R_{26}} \quad (7)$$

$$Q_a = \dot{m} c_p \frac{dT_{f3}}{dY} = \frac{T_{f3} - T_u}{R_3} + \frac{T_{f3} - T_{f1}}{R_{31}} + \frac{T_{f3} - T_{f2}}{R_{32}} + \frac{T_{f3} - T_{f4}}{R_{34}} + \frac{T_{f3} - T_{f5}}{R_{35}} + \frac{T_{f3} - T_{f6}}{R_{36}} \quad (8)$$

$$Q_a = \dot{m} c_p \frac{dT_{f4}}{dY} = \frac{T_{f4} - T_u}{R_4} + \frac{T_{f4} - T_{f1}}{R_{41}} + \frac{T_{f4} - T_{f2}}{R_{42}} + \frac{T_{f4} - T_{f3}}{R_{43}} + \frac{T_{f4} - T_{f5}}{R_{45}} + \frac{T_{f4} - T_{f6}}{R_{46}} \quad (9)$$

$$Q_a = \dot{m} c_p \frac{dT_{f5}}{dY} = \frac{T_{f5} - T_u}{R_5} + \frac{T_{f5} - T_{f1}}{R_{51}} + \frac{T_{f5} - T_{f2}}{R_{52}} + \frac{T_{f5} - T_{f3}}{R_{53}} + \frac{T_{f5} - T_{f4}}{R_{54}} + \frac{T_{f5} - T_{f6}}{R_{56}} \quad (10)$$

Now, equation (5) to equation (10), should satisfy the following boundary conditions.

At Y = 0, T_{f6} = Ti (11)

At Y=L, $T_{f1} = T_{f2} = T_{f3} = T_{f4} = T_{f5} = T_{f6}$ (12)

where, L is total length of tube

The transformation of equation (5) to equation (10) is carried out by expressing it in a non-dimensional form. Let us assume reference temperature difference as ΔT defined by-

$$\Delta T = T_{f,initial} - T_u \quad (13)$$

Dimensionless axial coordinates as x defined by-

$$x = \frac{Y}{L} \quad (14)$$

Dimensionless temperature, $\boldsymbol{\theta}_i$ is given by-

$$\boldsymbol{\theta}_i(x) = \frac{T_{fi}(Y) - T_u}{\Delta T} \qquad \text{for } i = 1,2,3,4,5,6 \quad (15)$$

Also, dimensionless conductance parameters k_i and k_{ij} is defined by-

$$k_i = \frac{L}{mc_pR_i}$$

$$k_{ij} = \frac{L}{mc_pR_{ij}}; \text{ where } i \neq j \quad (16)$$

Thereby, the overall dimensional term equation can be defined as-

$$\frac{dT_{fi}}{dx} = \frac{L}{mc_p}\left(\frac{T_{fi} - T_u}{R_i} + \sum\nolimits_{i \neq j} \frac{T_{fi} - T_{fj}}{R_{ij}}\right) \quad (17)$$

$$\text{Or, } \frac{d\theta_i}{dx} = k_i\boldsymbol{\theta}_i + \sum\nolimits_{i \neq j} k_{ij}(\theta_i - \theta_j) \quad \text{for } i = 1,2,3,4,5,6 \quad (18)$$

Alternatively, since, $R_1=R_2= R_3= R_4=R_5$ are the thermal resistance between absorber tube surface and respective pipe due to symmetry. Similarly, due to symmetry, $R_{12}= R_{13}= R_{14}= R_{15}=R_{21}= R_{23}= R_{24}= R_{25}=R_{31}= R_{32}= R_{34}= R_{35}=R_{41}= R_{42}= R_{43}= R_{45}= R_{51}= R_{52}= R_{53}= R_{54}$ are the thermal resistance between copper pipes placed inside evacuated tube collector and $R_{16}= R_{26}= R_{36}= R_{46}=R_{56}$ are the thermal resistance between steel manifold and copper pipes. Thereby, equation (5) to equation (10) may be rearranged as-

$$Q_a = - m^{\cdot} c_p \frac{dT_{f6}}{dY} = \frac{T_{f6}-T_u}{R_6} + \frac{T_{f6}-T_{f1}}{R_{16}} + \frac{T_{f6}-T_{f2}}{R_{16}} + \frac{T_{f6}-T_{f3}}{R_{16}} + \frac{T_{f6}-T_{f4}}{R_{16}} + \frac{T_{f6}-T_{f5}}{R_{16}} \quad (19)$$

$$Q_a = m^{\cdot} c_p \frac{dT_{f1}}{dY} = \frac{T_{f1}-T_u}{R_1} + \frac{T_{f1}-T_{f2}}{R_{12}} + \frac{T_{f1}-T_{f3}}{R_{12}} + \frac{T_{f1}-T_{f4}}{R_{12}} + \frac{T_{f1}-T_{f5}}{R_{12}} + \frac{T_{f1}-T_{f6}}{R_{16}} \quad (20)$$

$$Q_a = m^{\cdot} c_p \frac{dT_{f2}}{dY} = \frac{T_{f2}-T_u}{R_1} + \frac{T_{f2}-T_{f1}}{R_{12}} + \frac{T_{f2}-T_{f3}}{R_{12}} + \frac{T_{f2}-T_{f4}}{R_{12}} + \frac{T_{f2}-T_{f5}}{R_{12}} + \frac{T_{f2}-T_{f6}}{R_{16}} \quad (21)$$

$$Q_a = m^{\cdot} c_p \frac{dT_{f3}}{dY} = \frac{T_{f3}-T_u}{R_1} + \frac{T_{f3}-T_{f1}}{R_{12}} + \frac{T_{f3}-T_{f2}}{R_{12}} + \frac{T_{f3}-T_{f4}}{R_{12}} + \frac{T_{f3}-T_{f5}}{R_{12}} + \frac{T_{f3}-T_{f6}}{R_{16}} \quad (22)$$

$$Q_a = m^{\cdot} c_p \frac{dT_{f4}}{dY} = \frac{T_{f4}-T_u}{R_1} + \frac{T_{f4}-T_{f1}}{R_{12}} + \frac{T_{f4}-T_{f2}}{R_{12}} + \frac{T_{f4}-T_{f3}}{R_{12}} + \frac{T_{f4}-T_{f5}}{R_{12}} + \frac{T_{f4}-T_{f6}}{R_{16}} \quad (23)$$

$$Q_a = m^{\cdot} c_p \frac{dT_{f5}}{dY} = \frac{T_{f5}-T_u}{R_1} + \frac{T_{f5}-T_{f1}}{R_{12}} + \frac{T_{f5}-T_{f2}}{R_{12}} + \frac{T_{f5}-T_{f3}}{R_{12}} + \frac{T_{f5}-T_{f4}}{R_{12}} + \frac{T_{f5}-T_{f6}}{R_{16}} \quad (24)$$

Equation (19) to equation (24) may be rewritten as-

For Pipe 6 (Reverse flow direction)

$$-\beta \frac{dT_{f6}}{dY} = \frac{T_{f6}-T_u}{R_6} + \sum_{j=1}^{5} \frac{T_{f6}-T_{fj}}{R_{16}} \quad (25)$$

For Pipe 1 to 5 (Forward flows direction)

$$\beta \frac{dT_{fi}}{dY} = \frac{T_{fi}-T_u}{R_1} + \sum_{j=1}^{5} \frac{T_{fi}-T_{fj}}{R_{12}} + \frac{T_{fi}-T_{f6}}{R_{16}};\ i = 1, 2,3,4,5 \quad (26)$$

where, $\beta = \frac{m^{\cdot} C_p}{Q_a}$ (27)

To solve the above differential equation, let us define the temperature 'T' in Vector-matrix form. It is defined as

$$T = [T_{f_1} \quad T_{f_2} \quad T_{f_3} \quad T_{f_4} \quad T_{f_5} \quad T_{f_6}]^T \tag{28}$$

Then,

$$\frac{dT}{dY} = -BT + C \tag{29}$$

where,

$$B = \begin{bmatrix} b_1 + 4b_{12} + b_{16} & -b_{12} & -b_{12} & -b_{12} & -b_{12} & -b_{16} \\ -b_{12} & b_1 + 4b_{12} + b_{16} & -b_{12} & -b_{12} & -b_{12} & -b_{16} \\ -b_{12} & -b_{12} & b_1 + 4b_{12} + b_{16} & -b_{12} & -b_{12} & -b_{16} \\ -b_{12} & -b_{12} & -b_{12} & b_1 + 4b_{12} + b_{16} & -b_{12} & -b_{16} \\ -b_{12} & -b_{12} & -b_{12} & -b_{12} & b_1 + 4b_{12} + b_{16} & -b_{16} \\ -b_{16} & -b_{16} & -b_{16} & -b_{16} & -b_{16} & b_6 + 5b_{16} \end{bmatrix} \tag{30}$$

And,

$$C = \begin{bmatrix} b_1T_u \\ b_1T_u \\ b_1T_u \\ b_1T_u \\ b_1T_u \\ b_6T_u \end{bmatrix} \tag{31}$$

$$b_1 = \frac{1}{\beta R_1} \; ; b_6 = \frac{1}{\beta R_6} \; ; \; b_{12} = \frac{1}{\beta R_{12}} \; ; \; b_{16} = \frac{1}{\beta R_{16}} \tag{32}$$

Thereby, the temperature 'T' can be solved as-

$$T(Y) = T_s + \sum_{i=1}^{i} C_i V_i e^{\lambda_i Y} \tag{33}$$

where,

- The **eigen values** of matrix K are represented as λ_i.
- V_i represents the respective **eigenvectors**
- C_i are constants which are found by the inlet boundary conditions
- T_s is the **steady-state solution**

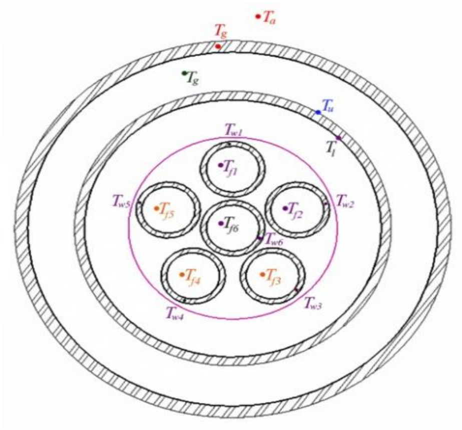

Fig. 2. Indication of Temperature at each interface

3. Results and Discussion

Measurements of solar irradiance were taken at Bilaspur (22.133698 ^{o}N, 82.142211 ^{o}E) on January 28, 2026 alongside measurements of ambient temperature. Two different flow configurations were studied through comparative analytical measurements, (i) single inlet, five outlets and (ii) five inlet, one outlet with both outlet/inlet combinations analyzed under the same environmental conditions especially the solar irradiance to provide a consistent analytical point of reference.

Table 1. **Variation of Solar intensity, ambient temperature and outlet water temperature along with time**

For 5 Inlet and 1 Outlet			
Time (IST)	**Solar Intensity (W/m^2)**	**Ambient Temperature (Degree Centigrade)**	**Outlet Water Temperature (Degree Centigrade)**
8:00 AM	250	28.6	27.5
9:00 AM	500	33.1	38.9
10:00 AM	955	36.7	47.8
11:00 AM	1000	38.8	57.6
12:00 PM	1035	41.8	63.9
1:00 PM	1100	42	68.3
2:00 PM	957	42.9	71.2
3:00 PM	940	43.1	70.2
4:00 PM	810	43.8	69.2
5:00 PM	630	42.2	66
6:00 PM	200	35.5	63

Table 2. **Variation of Solar intensity, ambient temperature and outlet water temperature along with time**

For 1 Inlet and 5 Outlet			
Time (IST)	**Solar Intensity (W/m^2)**	**Ambient Temperature (Degree Centigrade)**	**Outlet Water Temperature (Degree Centigrade)**
8:00 AM	250	28.6	27.5
9:00 AM	500	33.1	40.2
10:00 AM	955	36.7	50.1
11:00 AM	1000	38.8	60.2
12:00 PM	1035	41.8	67.3
1:00 PM	1100	42	72.3
2:00 PM	957	42.9	75.4
3:00 PM	940	43.1	76.1
4:00 PM	810	43.8	75
5:00 PM	630	42.2	73.1
6:00 PM	200	35.5	71

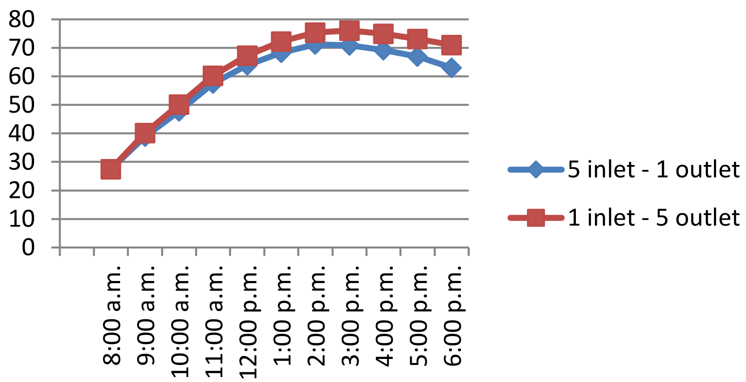

Fig. 3. Variation of Outlet Water Temperature with respect to time

The effect of inlet outlet configuration on thermal performance is clearly shown by the temporal variation in outlet water temperature that is observed in the two flow configurations. In both designs, the outlet temperature increases gradually throughout the early morning, peaking in the mid-afternoon due to the increase in the solar insolation, and then decreases in the evening. However, one inlet five outlet configuration has always given higher outlet temperatures than the five inlet one outlet set-up throughout the diurnal cycle. The maximum outlet temperature of the one inlet five outlets case is seen to reach about 75-76 ^{0}C whilst the maximum outlet temperature of the five inlet one outlet case is seen to reach about 70-71 ^{0}C. This enhancement can be explained by better flow distribution, increased heat removal, and reduced mixing losses of the several outlets that will lead to a more effective use of the absorbed thermal energy. On the other hand, multi-inlet single-outlet could enhance the flow disruptions and reduce the residence time leading to lower outlet temperatures. All in all, the findings indicate that the one-inlet-five-outlet design is thermally better and therefore more effective design in achieving high water outlet temperatures in a solar water heating system.

4. Conclusion

In the current paper, the application of inlet-outlet flow pattern on the outlet water temperature of solar water warmer was examined analytically. Two setups were tested, that is 1 inlet -5 outlets and 5 inlet -1 outlets, under the same operating condition and environmental condition. The findings reveal that the 1 inlet-5 outlet design is always effective compared to the 5 inlet-1 outlet design during the day resulting in better outlet water temperatures between the morning and evening hours. The highest temperature recorded at the outlet of the 1 inlet 5 outlet setup was about 75-76 ^{0}C, significantly higher than the 70-71 ^{0}C of the alternative setup. This is because it improves flow uniformity, residence time of water in the collector, and more efficient removal of heat since there is a multiplicity of outlet paths, which reduce thermal and mixing losses together. Both setups have a common time pattern and the highest temperature of the outlet is recorded until mid-afternoon and after that, this temperature drops. On the whole, the results validate the importance of streamlining the inlet–outlet system in enhancing thermal efficiency and the 1 inlet 5 outlet system may be considered a more efficient and practical design to apply to the process of solar water heating. These findings are a good foundation to optimize the design and scale of high-performance solar thermal systems in the future.

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Authors' biographies

Mr. Pradeep Patanwar , holds M. Tech degree in Mechanical Engineering from Indian Institute of Technology - BHU, Varanasi , India in 2012.Currently he is working as Assistant Professor in Dept. Mechanical Engineering at Guru Ghasidas Vishwavidyalaya (A central University) Bilaspur, India. His area of research is in Thermodynamics, Fluid Mechanics, Refrigeration and Air conditioning.

Email: pradeep.patanwar@gmail.com

Dr. T. V. Arjunan is a Professor in the Department of Mechanical Engineering at Guru Ghasidas Vishwavidyalaya (Central University), Bilaspur, Chhattisgarh, with over 26 years of teaching and research experience. His core areas of specialization include solar thermal systems, renewable and sustainable energy, heat transfer, solar desalination and drying technologies, and composite and advanced materials. He has played a significant role in advancing applied research and innovation-driven academic practices, with a strong focus on industry relevance, sustainability, and entrepreneurship in higher education. Dr. Arjunan has led and contributed to several nationally funded projects, notably the Capacity Building on Design and Entrepreneurship (CBDE) project funded by the Ministry of Education, Government of India (₹169 lakhs) as Co-Principal Investigator, which is currently ongoing, and the DST-funded project on Solar Evacuated Tube Collectors for Industrial Air Heating (₹162.63 lakhs), which he completed successfully as Principal Investigator. His academic and research contributions are substantial, having guided 15 Ph.D. scholars to completion, with many scholars currently pursuing doctoral research under his supervision. He has published over 140 research papers in reputed SCI and Scopus-indexed journals and conferences, holds six granted national and international patents, and has authored one international book along with seven book chapters.In leadership and academic administration, Dr. Arjunan has served as Head of the Department of Mechanical Engineering and Dean, School of Engineering and Technology. He was the Convener of the International Conference on Advances in Renewable and Green Energy Technology (ICARGET-2023). His excellence has been recognized through several honors, including the prestigious K. F. Antia Memorial Award from the Institution of Engineers (India), and a Certificate of Recognition from ZF Wind Power, Coimbatore. He has also fostered international academic collaboration through an MoU with Riga Technical University, Latvia. Dr. Arjunan is a Life Member of ISTE, the

Solar Energy Society of India, SAE, and the Bio Energy Forum, and continues to contribute actively to research, academic leadership, and national capacity-building initiatives.

Email: arjun_nivi@yahoo.com

Dr. Manikant Paswan
Director SLIET Punjab (Professor in Deptt, of Mechanical Engg., ex Dean Faculty Welfare), NIT Jamshedpur

Prof. Mani Kant Paswan has been a Professor in the Department of Mechanical Engineering since May 2004 and has served as Dean (Faculty Welfare) since August 2021. Prior to joining as Dean (F/W), he was Head of the Mechanical Engineering Department at National Institute of Technology (NIT) Jamshedpur, India, for 2.5 years (2018-2021). He is serving as President Nominee at all NIT faculty recruitments since 2023, as well as at Central University Guru Ghasidas Bilaspur, Chhattisgarh, in faculty recruitments since September 2022. He also served as Senate member of NIT Meghalaya from 2019-2022. He is working as academic council and Board of Studies member of various universities such as SarlaBirla University Ranchi, Amity University Kolkata, Amity University Ranchi, Arka-Jain University Jamshedpur etc. He is working as Nominated Moderator by AICTE of 2 Universities in Jharkhand Ram Govind Singh Engineering College Koderma and Chandravanshi Engineering College Gadhwa. He is senate member of NIT Jamshedpur for last 18 years, Chanclor of bihar nominated as chancller nominee for promoting of university faculty in all stream like arts,commerce,science and engineering. He is working as an Accessor in NAAC and NBA since 3 years and 4 years respectively. During his nearly three-decade teaching and research career, he has been an investigator on several funded research projects in MHRD (R&D), TEQIP, and various consultancy work. Renewable energy, composite materials, power plant engineering, and industrial applications are among his current research interests. He has over a hundred papers published in prestigious journals and has attended over 25 conferences; four patents also awarded. He was supervisor and research supervisor associate of thirteen PhDs to date, with another six currently in progress. He has also supervised 63+ M.Tech scholars and 31+ B.Tech project groups to date. Prof. Paswan is a leading proponent of solar thermal heating systems and natural fibre composite materials.

Email: manikant.nit@gmail.com

Dr. Anulal Mahto is an Associate Professor (Workshop) in the Department of Mechanical Engineering at the School of Studies in Engineering and Technology, Guru Ghasidas Vishwavidyalaya. He holds a Ph.D. in Production Engineering and possesses an extensive academic and professional career spanning more than 28 years. His areas of specialization include Ergonomics, Manufacturing Processes, Welding Engineering, and Industrial Mental Stress. His academic and research work covers diverse topics such as anthropometric studies, statistical analysis of body flexibility, and heat recovery systems. He has made significant research contributions through several peer-reviewed publications in reputed journals, including Materials Today and the International Journal of Ambient

Computing and Intelligence. Dr. Mahto is also an innovator and holds an Australian Patent for his workstation furniture design (Patent Number: 2021103041). In recognition of his outstanding research contributions, he received the Best Research Award in Ergonomics. Apart from teaching and research, he has played important roles in university administration by serving on various committees related to examinations, sports, and document verification. He is a life member of professional bodies such as Indian Society for Technical Education, Institution of Engineers (India), and Indian Institution of Industrial Engineering. Dr. Mahto actively participates in workshops and Faculty Development Programs (FDPs), with expertise in advanced areas such as 3D Printing, Lean Six Sigma, and Sustainable Manufacturing. His continued contributions have made a meaningful impact on academia, research, and industrial innovation.

Email: anulalmahto@gmail.com