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RESEARCH ARTICLE

Derivation of thermosyphonic mass flow rate in the pressurized thermosyphonic solar hot water systems using meander /serpentine fluid flow channel absorber for sterilization of water

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Abstract

In this work, the thermal performance of pressurized thermosyphonic solar hot water systems using meander and serpentine flow channel absorbers was experimentally evaluated. These configurations, not extensively explored in previous studies, were tested under real conditions to measure outlet temperatures and thermosyphonic mass flow rates. The results were compared with a developed thermal model, showing strong agreement and validating the model's accuracy. As solar intensity increased during the day, both outlet temperatures and mass flow rates rose, with peak temperatures reaching up to 97.5°C and mass flow rates around 10 kg/hr. Serpentine systems showed more consistent performance, particularly during early morning and late afternoon, due to better flow distribution and heat retention. The findings highlight the importance of absorber geometry on system efficiency. Additionally, meander-type collectors demonstrate potential for water sterilization in rural areas with unsafe water, offering both thermal efficiency and practical utility in off-grid environments.

1. Introduction

Energy is necessary to increase the standard of living and further development of society. Dependence on conventional fuels has to be minimized because of their limited supply. To achieve this, dependence on renewable energy sources e.g. solar energy, bio-energy, wind energy, geothermal, hydrogen energy, etc has to be increased. Solar energy can be used for variety of purposes such as water heating, crop drying, desalination, heating and cooling of space and buildings, refrigeration and air conditioning, mechanical and electrical power production. There are some popular applications of solar energy. Hot water is the most common application of solar energy. Normally 25% of total world population uses hot water. The quantity and pattern of hot water use very from country to country.All water heating systems used in the domestic or commercial sector can basically be divided into two categories (i) Natural convection water heating systems and (ii) Forced convection water heating systems [1]. In natural convection solar water heating systems, the flow of liquid in the collector loop takes place due to pressure by buoyancy forces generated by density gradients in the fluid contained in the collector. Such systems do not require any pump and generally the tank is placed over the collector. A common problem encountered in such systems is the occurrence of reverse flow which has been solved [1]. The natural convection can be achieved only in the small size systems suitable for domestic use. For large capacity systems use of pump for circulating the fluid in the collector loop cannot be avoided. Such systems are therefore known as forced convection systems. Natural convection or forced convection systems very often employ a heat exchanger in the collector loop especially in the cold climatic conditions or location where water is potable. In the former case, use of

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adequate antifreeze solution in the collector loop becomes unavoidable, while in the later case, it is advisable to use demineralised water in the collector loop for achieving long operating life of the systems [2]. Other investigators have evaluated theoretical and experimental thermal performances of non-pressurized thermosyphonic solar hot water systems using parallel tube and parallel plate absorbers [3, 4, 5, 8]. This paper mainly deals with the derivation of thermosyphonic mass flow rate of pressurized solar hot water systems using meander and serpentine fluid flow channel absorber. The utility of meander collectors is therefore seen for other purposes such as sterilization of water.

2. Derivation of thermosyphonic mass flow rates in natural convection pressurized solar hot water systems

A density difference created by temperature gradients caused the fluid being heated to flow without any pump. This effect of natural flow due to density gradient is usually termed as the thermosyphonic effect. The magnitude of this effect and resulting velocity of the fluid flow can be calculated on the basis of simple physical principles. The thermosyphonic solar water heating system, consisting of a solar energy collector, a hot water storage tank (installed above the solar collector) and the connecting tubes. When the sun radiation falls on the collector, it brings a temperature difference between the lower and upper ends of the collector. The temperature difference causes a density variation is giving rise to buoyancy forces. In the stationary conditions, the pressure due to buoyancy forces balances the pressure losses due to friction. In the solar thermosyphonic water heating system, let us consider that the water entered in the solar collector at a temperature [T_1] and it leaves at a temperature [T_2]. The hot water storage tank received heat energy from the solar energy collector. The effective pressure difference due to buoyancy force was responsible for the total closed loop cycle in the thermosyphonic solar water heating system, which can be assumed to be made up of two parts viz:

$$\Delta P_{t=} [\rho_i g h_1 - \rho_n g\Delta h_n \sin\Theta - \rho_e g h_e] \quad (1)$$

Over a small temperature change, the variations in the density with temperature can be written as

$$\rho(T) = \rho_0 * \{(1- \beta(T))\} \quad (2)$$

Where β is the coefficient of volume expansion and ρ_0 is the density at 0℃ . The density variation can be expressed as

$$\rho_i(T_i) = \{\rho_0 * (1- \beta(T_i))\}$$
$$\rho_e(T_e) = \{\rho_0 * (1- \beta(T_e))\}$$
$$\rho_n(T_n) = \{\rho_0 * (1- \beta(T_n)$$

Hence,

$$\Delta P_{t=} \rho_o g [\rho_i\, gh1 - \rho n\, g\Delta hn \sin\Theta - \rho_e\, ghe] \quad (3)$$

Substituting density variation in terms of temperature, we get

$$\Delta P_{t=} \{[gh_1\rho_0 (1- \beta(T_i)) - \sum g\, \Delta h_n \sin\Theta\, (\rho_0 * (1- \beta(T_n)) - gh_e\{\rho_0 * (1- \beta(T_e))] \quad (4)$$

$$\Delta P_{t=} g\, \rho_0[(h_1-h_e) - \sum h_n \sin(\Theta)] - \beta g\, \rho_0[\{T_i h_1 - \Delta h_n T_n \sin\Theta T_e h_e \}] \quad (5)$$

The value of the exit temperature corresponding to [$\Delta P_t = 0$], is the minimum temperature obtainable from a given system. This temperature is called balance point temperature obtainable for a given system in the meander fluid flow channel absorber is

$$T_b(t) = [\{[h1/he]\, T_i(t) + \sum \Delta hn \sin\Theta / (\beta h_e) - \{(h_1-h_e)/(\beta h_e)\} - \{T_n \sum \Delta h_n \sin\Theta/(\beta h_e)\}] \quad (6)$$

In the stationary condition of flow, the buoyancy pressure [ΔP_t] is responsible for thermosyphonic flow should also be equal to the total pressure losses in the system (i.e collector, connecting tube etc) is denoted by [ΔP_L]. The total pressure losses in the system is

$$[\Delta P_L]=[\Delta P_{tube}+\Delta P_{bend}] \quad (7)$$

Assuming collecting fluid channel is cylindrical geometry, the pressure losses for such geometry can be written as

$$[\Delta P_L/L]=\left[\frac{64}{Re}\right]* [(\mu/2D)*(\mu/\rho)^2] = 32\, [(\mu/\rho)*(\rho v/D^2) \quad (8)$$

The fluid velocity in the channel is related to the mass flow rate in the solar collector by the relation

$$m_c(t)= [(\pi D^2)\rho * v/4] \quad (9)$$

$$v=[4\, m_c(t)/\pi D^2)\rho * \pi D^2)\rho] \quad (10)$$

Substituting the velocity of fluid from eq. (9) in eq. (7) one get

$$[\Delta P_L/L]= [(128\upsilon * m_c(t)/(\pi D^4) \quad (11)$$

For n tubes in the series with bends in between

$$[\Delta P_L]= \sum 128\upsilon * mc(t) L_n/ (\pi D^4) +\Delta P_{Lbend} \quad (12)$$

and

$$[\Delta P_{Lbend}]= \sum 128\upsilon * mc(t)\ 0.3N / (\pi D^4) \quad (13)$$

Where N= number of bends =11 (in the thermosyphonic meander fluid flow channel. Hence,

$$[\Delta P_L] = [\sum 128\upsilon\, mc(t)\, (L_n + 0.3N)/ (\pi D^4)] \quad (14)$$

For equilibrium condition and thermosyphonic to occur,

$$[\Delta P_t] = [\Delta P_L] \quad (15)$$

and

$$[g\,\rho_0\ [(h_1-h_e) - \sum h_n \sin(\Theta)\ ] - \{\beta g\,\rho_0 [\{T_i h_1 - \Delta h_n T_n \sin\Theta - T_e h_e\}]\} = [\sum 128\upsilon\, mc(t)\ (L_n + 0.3N)/(\pi D^4)\ ] \quad (16)$$

Rearranging eq.[16] one gets following equation

$$m_c(t) = [g\,\rho_0\ \pi D^4 /(128\upsilon\ (L_n + 0.3N)]*[\ g\,\rho_0\ [(h_1-h_e) - \sum h_n \sin(\Theta)\ ] - \beta g\,\rho_0 [\{T_i h_1 - \Delta h_n T_n \sin\Theta T_e h_e\}] \quad (17)$$

The above expression determines the fluid flow rate for a given configuration of the thermosyphonic pressurized system using meander fluid flow channel absorber i.e. for some value of h_1 and h_2 suitably adjusted to obtain temperature ≥ balance point temperature T_b. The outlet temperature T_e and the in between temperature T_n in between the channel computed by using eq. (16) and (17) by using iterative procedure.

For closed loop systems, the mass flow rate has been estimated by using the relation:

$$m_c(t) = [M_w C_{pw} \Delta T_m(t)/\Delta t)/(C_{pw} * \Delta\ T_c(t))] \quad (19)$$

By recording of tank temperature evergy half hour, the mass flow rate [$m_c(t)$] in the solar collector can be estimated from above equation.

3. Results and Discussion

Table 1 presents a detailed comparison between experimental and modeled thermal performance of a single-pass thermosyphonic solar water heating system using a meander-type absorber with a collector area of 1.89 m². The data spans from 9 AM to 5 PM, capturing hourly values of inlet water temperature, ambient temperature, solar intensity, outlet temperature, mass flow rate, and temperature difference. Early in the day at 9 AM, the system shows no experimental mass flow rate, indicating insufficient temperature difference to initiate thermosyphonic circulation. As solar intensity increases—reaching a peak of 765 W/m² around 1 PM—the collector outlet temperature and mass flow rate also rise, with a maximum experimental outlet temperature of 97.5°C and a peak mass flow rate of 10 kg/hr observed at 12 PM. The temperature difference between inlet and outlet water reaches its highest during this period, indicating efficient solar energy capture. The modeled values for outlet temperature and mass flow rate closely match the experimental data, suggesting that the mathematical model is reliable for predicting thermal behavior. In the afternoon, as solar intensity declines, the outlet temperature and mass flow rate also reduce, reflecting the system's dependency on solar input. This table clearly demonstrates the thermodynamic response of the system to varying solar radiation and validates the accuracy of the theoretical model. Table 2 provides an experimental performance comparison of three systems—System 2, System 3, and System 4—each employing a meander flow absorber under similar conditions. The table highlights parameters such as outlet fluid temperatures, mass flow rates, and temperature differences. It shows that all three systems follow a similar thermal pattern throughout the day, responding dynamically to changes in solar intensity, which increases from 431 W/m² in the morning to a peak of 764 W/m² around 1 PM, then declines sharply by 5 PM. System 4 consistently exhibits slightly higher temperature differences and mass flow rates than Systems 2 and 3, suggesting improved internal flow design or thermal conductivity. At peak solar conditions, outlet temperatures of around 96–97°C are observed across all systems, while mass flow rates reach up to 6.8 kg/hr in System 4. In the early morning and late evening, mass flow rates are significantly lower, indicating the natural thermosyphonic limitation during low solar input. This comparison underscores the effect of system configuration on performance; particularly how slight design changes can enhance heat absorption and circulation efficiency.

Table 3 explores the thermal performance of three systems—System 5, System 6, and System 7—featuring serpentine-type flow channel absorbers. Like Table 2, it includes hourly experimental data for outlet fluid temperatures and mass flow rates. The results reveal that serpentine systems perform slightly better in maintaining stable outlet temperatures and flow rates compared to the meander configurations. System 7 consistently records the highest values among the three, reaching an outlet temperature of 95.6°C and a mass flow rate of 6.2 kg/hr at 1:30 PM. Notably, the mass flow rates in serpentine systems tend to be higher in the early morning and late afternoon compared to the meander systems, suggesting more effective circulation under lower solar intensity. This could be attributed to the enhanced thermal path and improved flow distribution within the serpentine channels. By the end of the day at 5 PM, although solar intensity drops to 50 W/m², System 7 still maintains some circulation, which reflects better heat retention. This table demonstrates that serpentine absorber designs may offer more robust performance, especially under variable solar conditions, and are potentially more efficient than meander-type systems for solar thermal applications.

Overall, the three tables collectively illustrate the influence of solar radiation, absorber design, and system configuration on the thermal efficiency of solar water heaters. The comparisons highlight how optimized designs—particularly serpentine configurations and advanced thermosyphonic structures—can improve performance, enhance circulation, and ensure better utilization of solar energy throughout the day.

Table-.1: Thermal performances of single pass thermosyphonic non pressurized solar water heating system using meander fluid flow channel absorber (Collector Area =1.89m²)

Time (hr)	Inlet Water Tempera-ture (°C)	Ambient Tempera-ture (°C)	Solar Intensity (W/m^2)	Collector Outlet fluid temperature Exp. (T_{CO}) (°C)	Collector Outlet fluid temperature Model (T_{CO}) (°C)	Collector mass flow rate Kg/hr) Exp.	Collector mass flow rate (Kg/hr) (Model)	Collector tempera-ture difference (°C) Exp.	Collector temperature difference (°C) Model
9AM	21.0	21.7	430.0	65	61	0	0.2	44.0	40.0
10	24.0	23.5	535.0	78	81	3.70	4.0	54.0	57.0
11	28.0	26.0	675.0	41.5	89.9	6.57	6.6	63.5	61.9
12	32.0	28.4	760.0	96.0	98.9	10.0	10.5	64.0	66.9
13PM	32.5	29.3	765.0	97.5	96.8	9.6	10.1	55.0	64.3
14	34.5	29.7	695.0	91.5	94.8	5.6	6.2	57.0	60.3
15	35.5	29.8	560.5	90.0	92.5	4.6	5.15	54.5	59.3
16	35.0	29.9	387.0	82.5	88.2	2.3	2.35	47.5	53.2
17	33.5	28.8	182.0	57.5	54.7	0	0.3	24.0	25.2

Table-2:Thermal performances of single pass thermosyphonic non pressurized solar water heating system using meander fluid flow channel absorber (Collector Area =1.89m²)

Time (hr)	Inlet Water Tempera-ture (°C)	Ambient Tempera-ture (°C)	Solar Intensity (W/m^2)	Collector Outlet fluid temperature Exp.(T_{CO}) of system-2 (°C)	Collector Outlet fluid temperature Exp. (T_{CO}) of system-3 (°C)	Collector temperature difference (°C)Exp. of system-4	Collector mass flow rate Kg/hr) Exp. of system-2	Collector mass flow rate Kg/hr) Exp. of system-3	Collector mass flow rate (Kg/hr) Exp. of system-4
9AM	21.0	21.7	431.	64.5	65.0	65.2	0.25	0.25	0.25
9.30	22.9	22.5	505	72.0	73.5	74.2	1.3	2.2	2.5
10	24.0	23.5	535	81.0	82.3	84.1	2.8	3.8	4.2
10.30	26.0	24.8	605	87.0	88.7	89.7	3.7	4.2	4.6
11	28.0	26.0	670	90.0	91.2	92.0	4.8	4.9	5.3
11.30	30.0	27.0	730	94	94.9	95.4	5.5	5.6	5.8
12	32.0	28.4	760	95.2	95.7	96.2	5.6	5.8	6.1
12.30	32.3	28.8	755	95.7	96.4	97.1	5.8	6.4	6.6
13PM	32.5	29.3	764	96.5	96.9	97.2	6.0	6.6	6.8
13.30	33.1	29.5	730	96.0	96.2	96.7	5.8	6.1	6.3
14	34.5	29.7	693	93.5	94.3	94.6	5.5	5.8	6.1
14.30	34.6	29.7	630	87.2	88.7	89.1	4.6	4.7	4.9
15	34.7	29.8	561	78.9	80.6	81.2	3.0	3.1	3.2
15.30	34.9	29.0	480	70.0	70.6	71.1	2.8	2.9	2.9
16	35.0	28.8	387.	55.5	57.8	55.7	1.5	1.6	1.6
16.30	33.7	28.7	182	35.5	36.2	36.3	0.7	0.8	0.8
17	33.5	28.8	50.0	33.5	33.5	33.5	0	0	0

Tables 4-6 collectively present the comparative performance parameters of thermosyphonic free convection solar water heating systems for both pressurized and non-pressurized configurations, evaluated with and without the inclusion of honeycomb structures. These tables emphasize how collector geometry, fin materials, and flow channel design influence two critical thermal performance parameters: F'(τα)e, which represents the effective product of collector efficiency factor and transmittance-absorptance product, and F'UL, the overall heat loss coefficient. Table 4 focuses on pressurized thermosyphonic systems using meander fluid flow channels for hostels. Two configurations are compared: systems without honeycomb structures and systems with honeycomb structures integrated into the absorber plate. The systems use copper tubes with different fin materials—copper or aluminum. Across all designs, systems without honeycomb structures exhibit higher F'(τα)e values (around 0.720–0.725) and significantly higher F'UL values (ranging from 8.089 to 9.405), indicating more heat loss. Conversely, when honeycomb structures are introduced, F'(τα)e slightly decreases (to 0.645 or lower), but the heat loss coefficient (F'UL) drops dramatically to values between 2.865 and 3.0353. This shows that honeycomb structures significantly reduce heat loss from the collector, enhancing thermal insulation, albeit with a minor reduction in solar energy absorption due to potential flow resistance or optical interference from the honeycomb structure. Table 5 examines pressurized thermosyphonic systems using serpentine and parallel tube flow channels, again for hostel applications

Table-3: Thermal performances of single pass thermosyphonic non pressurized solar water heating system using serpentine fluid flow channel

Time (hr)	Inlet Water Tempera-ture (ºC)	Ambient Tempera-ture (ºC)	Solar Intensity (W/m^2)	Collector Outlet fluid temperature Exp. (T_{CO}) of system-5 (ºC)	Collector Outlet fluid temperature Exp. (T_{CO}) of system-6 (ºC)	Collector Outlet fluid temperature Exp. (T_{CO}) of system-7 (ºC)	Collector mass flow rate Exp. of system-5 (Kg/hr)	Collector mass flow rate Exp. of system-6(Kg/hr)	Collector mass flow rate,Exp. of system-7 (Kg/hr)
9AM	21.0	21.7	431.	65	65.0	64.9	0.35	0.5	0.5
9.30	22.9	22.5	505	72.5	73.0	71.3	2.8	2.9	3.0
10	24.0	23.5	535	80.2	80.5	79.8	3.65	3.7	4.2
10.30	26.0	24.8	605	85.3	87.0	88.2	3.85	3.9	4.5
11	28.0	26.0	670	90.0	91.7	90.9	4.1	4.2	4.8
11.30	30.0	27.0	730	92.1	93.1	92.2	4.55	4.6	5.0
12	32.0	28.4	760	94.0	94.7	93.6	4.75	4.8	5.4
12.30	32.3	28.8	755	94.3	95.2	94.7	4.9	5.0	6.1
13PM	32.5	29.3	764	95.8	96.7	95.6	4.85	4.9	5.8
13.30	33.1	29.5	730	96.1	96.2	94.8	4.55	4.6	6.2
14	34.5	29.7	693	91.2	92.3	92.9	3.95	4.0	4.8
14.30	34.6	29.7	630	89.5	90.5	91.3	2.5	2.6	4.0
15	34.7	29.8	561	87.3	88.3	89.2	2.1	2.0	2.75
15.30	34.9	29.0	480	83.6	84.6	83.7	1.3	1.3	2.3
16	35.0	28.8	387.	80.1	81.3	79.9	1.05	1.1	1.2
16.30	33.7	28.7	182	68.5	70.5	71.5	0.75	0.85	0.8
17	33.5	28.8	50.0	35.5	35.6	37.1	0.1	0.2	0.4

Table 6 shifts the focus to non-pressurized thermosyphonic systems for guest house applications using parallel tube flow channels. These systems are evaluated in similar terms, using copper or aluminum fins and either copper or GI tubes. Systems without honeycomb structures have F'(τα)e values in the range of 0.70–0.725 and relatively high F'UL values between 8.055 and 11.80. In contrast, the same systems with honeycomb integration show reduced F'(τα)e values (down to 0.60 in some cases) and much lower heat loss coefficients, ranging from 2.465 to 3.035. This indicates that even in non-pressurized systems, honeycomb structures help in enhancing thermal retention by effectively reducing conductive and convective heat losses from the absorber surface.

Table 4: Performance Parameters of thermosyphonic free convection pressurized solar hot water systems (collector Area=1.89 m^2) with and without honey comb structures for hostels

S. No.	Collector geometry of pressurized absorber (using meander turn fluid flow channel) with honeycomb structures of size 2.1m*0.9mm*0.140m*0.02m (with aspect Ratio =seven)	Pressurized thermosyphonic systems using meander fluid flow channels (8tubes) of copper tube length without honeycomb structures	Pressurized thermosyphonic systems using meander fluid flow channels (8 tubes) of copper tube length with honeycomb structures
1	Copper fins and copper tube (26 gauge) absorber	F'(τα)e=0.720 and F'U_L =8.089	F'(τα)e=0.645 and F'U_L = 2.865
2	Aluminum fins (24 gauge) and copper tube absorber	F'(τα)e=0.725 and F'U_L =9.405	F'(τα)e=0.635 and F'U_L = 2.95
3	Copper fins and copper tube absorber	F'(τα)e=0.720 and F'U_L =8.1017	F'(τα)e=0.645 and F'U_L = 2.870
4	Aluminum fins and copper tube turns absorber	F'(τα)e=0.70 and F'U_L =8.2025	F'(τα)e=0.60 and F'U_L = 3.0353

Table 5: Performance Parameters of thermosyphonic free convection pressurized solar hot water systems (collector Area=1.89 m^2) with and without honey comb structures for hostels

S. No.	Collector geometry of pressurized absorber (using serpentine turn fluid flow channel) with honeycomb structures of size 2.1m*0.9mm*0.140m*0.02m (with Aspect Ratio =seven)	Pressurized thermo-syphonic systems using parallel tube fluid flow channels (7 tubes) of copper tube length without honeycomb structures	Pressurized thermo-syphonic systems using serpentine fluid flow channels (7tube turns) of copper tube length with honeycomb structures
1	Copper fins (24 gauge) and copper tube absorber	F'(τα)e=0.720 and F'U_L =5.214	F'(τα)e=0.645 and F'U_L = 4.0
2	Copper fins and copper tube absorber	F'(τα)e=0.720 and F'U_L =6.344	F'(τα)e=0.645 and F'U_L = 4.45
3	Aluminum fins (24 gauge) and copper tubes	F'(τα)e=0.70and F'U_L =8.089	F'(τα)e=0.60 and F'U_L = 4.15
4	Aluminum fins 24 gauge and GI tube absorber	F'(τα)e=0.725 and F'U_L =11.23	F'(τα)e=0.645 and F'U_L = 4.750

Similar to Table 4), the systems are tested with and without honeycomb structures. The serpentine configurations (with honeycomb) generally have slightly lower F'(τα)e values (around 0.645–0.60) but demonstrate considerable improvements in thermal efficiency due to reduced F'UL values (4.0 to 4.75), compared to the higher heat losses in systems without honeycombs (F'UL up to 11.23). The best performing non-honeycomb configuration here is the aluminum fin with GI tube absorber, which has the highest F'UL of 11.23, indicating high thermal losses despite a good F'(τα)e of 0.725. Once again, the introduction of honeycomb structures provides a consistent reduction in heat losses while slightly lowering the energy absorption factor.

Table 6: Performance Parameters of thermosyphonic free convection non pressurized solar hot water systems (collector Area=1.89 m^2) with and without honey comb structures for guest house

S. No.	Collector geometry of parallel tube absorber (using fluid flow channel) with honeycomb structures of size 2.1m*0.9mm*0.140m*0.02m (with Aspact Ratio =seven)	Non-Pressurized thermo-syphonic systems using parallel tube fluid flow channels (8 tubes) of copper tube length without honeycomb structures	Non-Pressurized thermo-syphonic systems using parallel tube fluid flow channels (8 tubes) of copper tube length without honeycomb structures
1	Copper fins and copper tube absorber 24 gauge	F'(τα)e=0.720 and $F'U_L$ =8.090	F'(τα)e=0.645 and $F'U_L$ = 2.465
2	Aluminum fins and copper tube absorber	F'(τα)e=0.725 and $F'U_L$ =8.055	F'(τα)e=0.635 and $F'U_L$ = 2.845
3	Copper fins and copper tube absorber 24 gauge	F'(τα)e=0.720 and $F'U_L$ =8.1017	F'(τα)e=0.645 and $F'U_L$ = 2.48
4	Aluminum fins 22 gauge and GI tube absorber	F'(τα)e=0.70 and $F'U_L$ =11.80	F'(τα)e=0.60 and $F'U_L$ = 3.035

6. Conclusions and Recommendation

- Single pass pressurized thermosyphonic solar water heating system using meander tube fluid flow channel is recommended for water purification for drinking purposes in the rural remote areas where water quality is bad.
- The experimental results matched well with computed results from derived equation for pressurized thermosyphonic solar hot water systems.
- The system efficiency of meander tube absorber is 30 to 35% in single pass mode and 27 % in the multi pass mode due to higher temperature of absorber and higher heat losses from absorber in pressurized solar hot water systems.
- The system efficiency of serpentine tube absorber is 25 to 30% in single pass mode.
- The efficiency of pressurized thermosyphonic solar collectors however gets reduced drastly due to high temperatures obtained from these collectors which can be reduced by inserting the honeycomb structures between absorber and top cover of collectors and found about 70% of top heat losses from solar energy collectors.

References

[1] C. L. Gupta and H. P. Garg, "System Design in Solar Water Heaters with Natural Circulation". Solar Energy 12, 163 (1968).

[2] Radhey Shyam Mishra [2024] Thermal performances of non-reversible pressurized and non-pressurized natural convection solar water heating systems using Meander and serpentine turn fluid flow channel absorbers, International journal of research in engineering and innovation (IJREI), vol 8, issue 6 (2024), 201-205.

[3] Mishra R.S.,[1991]"Thermal modelling of solar hot water system, Eight national conference of Mechanical engineers, The Institution of Engineers (India) ,Roorkee local centre ,29th - Oct- 1 Nov, 1991.

[4] S. A. Klein et al., TRNSYS 12.1 User's Manual. University of Wisconsin, Solar Energy Laboratory (1983).

[5] G. L. Morrison and D. B. J. Tanatunga, "Thermosyphonic Circulation in Solar Collectors". Solar Energy 24, 191-198 (1980).

[6] A. H, Fanney and S. T. Liu, "Comparison of Experimental and Computer-Predicted Performance of 6 Solar Domestic Hot Water Systems". ASHRAE Trans. 86, 823-833 (1980).

[7] A. H. Fanney and S. A. Klein, "Performance of Solar Domestic Hot Water Systems at the National Bureau of Standards-Measurements & Predictions". ASME J. Solar Energy Engng 105, 311-321 (1983).

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