Modeling of Natural Convection Non Reversible Single Pass Pressurized and Non Pressurized Solar Hot Water Systems for Domestic Applications

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Abstract

This paper mainly deals with the problem of reverse flow and explicit expressions have been presented which shows that the placement of hot water storage tank above the collector helps to eliminates reverse flow and it was observed that if tank is kept very high above the collector then there is always some flow of liquid through the collector especially if the outlet of the collector is connected near the middle of the storage tank. This is due to considerable heat losses resulting poor thermal efficiency of the system. The new design based on single and multi pass systems for domestic hot water systems which completely eliminate reverse flow problems have been forwarded and studies in details.

The single pass systems besides avoiding reverse flow has two distinct features:

(i) Allows the place the main water storage tank at any desirable place (even below the collector)
(ii) Water is obtained at pre-determined temperature irrespective of insolation levels.

The numerical calculations for single pass natural convection pressurized solar hot water systems meander tube fluid flow channel absorber, serpentine tube collectors and for non pressurized thermosyphonic solar hot water systems using parallel tube absorber were performed. The experimental measurements on such modified systems were performed manually by noting thermosyphonic flow rate and temperatures at every half hours time interval. The theoretical results matches well with experimental results. It was found that the non pressurized solar hot water systems using parallel tube absorber have average thermal efficiency of 41 % in the single pass and 27. 3 % in the multi-pass modes. The system efficiency of pressurized solar hot water systems using meander tube absorber are 30 % in single pass mode and 20 % in the multi pass mode due to higher temperature of absorber and higher heat losses from absorber in pressurized solar hot water systems.

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, geo-thermal, 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. The average daily hot water requirement for eastern countries is varying from 25 liters to 50 liters per person while in the western countries is about 60 liters to 85 liters per person. The temperature of hot water for domestic use is about 40 oC. The pattern of consumption very from place to place. The sizing of various components of solar energy system depends upon various physical parameters such as daily hot water demand of hot water, local weather conditions etc. Due to time variations of incident solar energy, ambient temperature, the solar energy system should preferably designed to have adequate storage system. The design and type of thermal energy storage in turn depends upon the application for which a solar energy system is generally be designed.

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.

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 required 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 discussed

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thoroughly in this paper. 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 adequate antifreeze solution in the collector loop becomes unavoidable, while in the later case, it is advisable to use diminearlized water in the collector loop for achieving long operating life of the systems

2. Phenomenon of Reverse Flow in the Thermoshphonic Solar Water Heating Systems And Modified System Designs

Reverse flow can occurs in thermosyphonic solar water heating systems during off sunshine hours, especially when the bottom of the hot water storage tank is placed below the top of the collector. During the day time, water in the hot water storage tank gets heated and hence the top portion of the hot water storage tank remains hot and the hot water comes in to the collector from hot water storage tank which gets cooled much faster rate than the water in the pipe due to larger surface area of the collector exposed to sky. Therefore left side of the tube gets cooler and heavier while the right side remains hot. The flow of liquid starts in reverse direction due to the density difference. This brings down the temperature of the water in the hot water storage tank drastically.

It was observed that this effect was not considered by the many manufactures of the hot water systems commercially available in the Indian market which has an average thermal efficiency of 50% or below should specially checked for reverse flow anticipating that the components are well designed (i.e. Collector, storage tank, piping and insulation is concerned)

The phenomenon of the back flow can be explained in details by considering the solar hot water systems as U tube. During the day time, the head of water in the hot water storage tank plus that of the water in the returned pipe constitutes the falling column of the water while head of the water in the solar collector and in the flow pipe constitutes the rising column. The water in the return and also in the flow pipe varies with the height and therefore it is very useful to introduce the concept of mean density and mean temperature in the following manner

$$T(t) = 1/h\int_0^h f(h)dh \qquad (1)$$

Where f(h) is the function describing the temperature at any given height (h). Let $\rho_1(T)$ and $T_1(t)$ be the mean density and mean temperature of water in the return pipe. $\rho_2(T)$ and $T_2(t)$ be the mean density and mean temperature of the left leg of the tube consisting of solar energy collector absorber and the flow pipe. Normally $T_1(t)$ is less than $T_2(t)$ and hence $\rho_1(T)$ is greater than $\rho_2(T)$, under these conditions, the left leg of water is no longer able to balance the right leg, which falls continuous displacing the water from left leg to the storage tank. The pressure causing the flow in the direction down the return pipe and up the flow pipe is $Hg(\rho_1(T) - \rho_2(T))$ and is termed as total circulation pressure.

If $T_2(t)$ is less than $T_1(t)$ i.e. $\rho_2(T)$ is greater than $\rho_1(T)$, then the circulation in the opposite direction called reverse (back) flow. During off sunshine hours, the mean temperature $T_2(t)$ could be reduced below $T_1(t)$ in two ways viz

(i) Insufficient or complete lack of insulation on the flow pipe
(ii) Heat losses from solar energy absorber itself.

In well constructed thermosyphonic hot water systems, the heat losses from the absorber would be predominant effect and on a cold night, the temperature of water in the absorber could be reduced to such a extent that the mean temperature of the water in the left leg would be less than the mean temperature of water in the right leg

$$T_1(t) = (T_r(t)^* H_1 + T_t(t)^* (t)\, h_1) / (H_1 + h_1) \ldots\ldots\ldots (1)$$

Where $T_1(t)$ is the weighted mean temperature over height H in meters,

$T_t(t)$ is the weighted mean temperature over height h_1 meters of the water in the hot water storage tank, $T_r(t)$ is the weighted mean temperature over height H_1 meters of the water in the returned pipe.

Similarly

$$T_2(t) = ((H_1 + h_1) * T_f(t) + h_2 (T_a(t) - T_f(t)) / (H_1 + h_1) \quad (2)$$

To prevent the condition of reverse flow, we must satisfy the condition that

$$T_1(t) = T_2(t) \qquad \ldots\ldots. (3)$$

Equating, therefore expressions (2) and (3) and rearranging, one gets

$$H_1 = h_2 (T_f(t) - T_a(t) + h_1 (T_t(t) - T_f(t)) / (T_f(t) + T_r(t)) \qquad \ldots\ldots\ldots (4)$$

Let h_3 is the difference between the top of the absorber and bottom of the hot water storage tank in meters i.e. $h_3 = H_1 - h_2$

Where h_3 can be expressed in the following expression

$$h_3 = h_2 (T_r(t) - T_a(t)) + h_1 (T_t(t) - T_f(t)) / (T_f(t) - T_r(t) \qquad \ldots\ldots (5)$$

From Eqn. (5), one can argue that it is unlikely for the flow to be stopped for prolonged periods.. From Eqn(2) and (3) one obtains

$$(T_1(t) - T_2(t)) = - [(h2 (T_f(t) - T_a(t)) + H_1 (T_r(t) - T_f(t)) + h_1 (T_t(t) - T_f(t))] / (H_1 + h_1). \qquad (6)$$

Keeping the temperatures and the heights h_1 and h2 of solar hot water systems are constant and varying the height H_1, we get following expression.

$$(d (T_1(t) - T_2(t)) / d H_1) = - (T_f(t) - T_a(t) * (h_2 / (H_1 + h_1)^2) - T_2(t) * ((H_1 + h_1) * T_f(t) + h_2 (T_a(t) - T_f(t)) / (H_1 + h_1 h_1(T_t(t) - T_f(t)) / ((H_1 + h_1)^2)) - ((T_f(t) - T_r(t)) / (H_1 + h_1)) + (H_1 / (H_1 + h_1))^* ((T_f(t) - T_r(t)) / (H_1 + h_1) \qquad \ldots(7).$$

Under off sunshine hours/ night time conditions, and with the tank full of the hot water, it is reasonable to expect that $T_f(t)$ is greater than $T_a(t)$, $T_t(t)$ is greather than $T_f(t)$, and also $T_f(t)$ is greater than $T_r(t)$. With the result that R.H.S. of Eqn. (6). is negative and the temperature difference $T_1(t) - T_2(t)$ decreases with increasing H1. If H_1 is greater than h_1, then one can gets

$$T_2(t) = T_f(t) - ((h_2 / H_1) * (T_f(t) - T_a(t)) .$$

For the limit $T_1(t) = T_2(t)$.

Thus by raising the height of storage tank relative the solar collector absorber, the temperature differences causing reverse flow is decrease if the height increases. By careful design and installation, the reverse flow can be partially avoided by placing the tank as possible above collector. The minimum height of the bottom of the storage tank from the top of the collector should be 200mm to 300 mm depending upon the collector design & configurations. Mishra observed that if the tank is kept very high above the collector, then there is always some flow of water/liquid through the collector especially if the outlet of the solar energy collector is connected near the middle of storage tank. This also results in considerable heat losses resulting in poor thermal performances of the natural convection solar hot water systems. Some modified single and multi pass natural convection solar hot water systems in which reverse flow has been completely eliminated was developed by Mishra (1992) shown in Figs(2-3). The measurements were performed for several days on the various modified non reversible natural convection solar hot water systems which were few variations. The numerical computations for water temperatures and thermal efficiency have been carried out and shown in Table (1) - (5) respectively. Some designs of thermosyphonic solar water heating systems which completely eliminated the reverse flow developed by Mishra (1993) the placement of hot water storage tank was allowed at an acceptable level have been shown in Figs(3-4) respectively.. In these modified single and multi pass natural convection hot water systems, the back flow completely absent and storage tank should be placed as high as possible above the collector, however one has to consider the extra piping cost and insulation cost associated with the thermal losses also. These modified non reversible solar hot water systems are working on single and multi pass modes with a height (h) of the outlet pipe of the collector adjusted in such a way to get the desire hot water temperature. In these systems, the outlet of the solar energy collector should be at the same level as water level in the cold water storage tank. Depending on the desire temperature, the solar collector configurations can be different. The various modified collector configurations are studied by Mishra ( ) in details. The performance parameters and thermal efficiency curves of various modified non reversible hot water systems are show in Figs ( ) respectively.

3. Modeling of Natural Convection Solar Hot Water Pressurized Systems

The schematic of the proposed modified systems for single pass mode have been shown in Fig (1) respectively. The buoyancy force responsible for the fluid flow in the whole system can be written by following expression.

$$\Delta P_t = \rho_{fi}(T)\, g\, h_1 - \sum \rho_n(y)\, g\, \Delta h_n \sin\theta - \rho_{fo}(T)\, g\,(h_1 + h_2) \quad \ldots\ldots (1)\ 2$$

Where $\rho_{fi}(T)$ is the density of the fluid at collector inlet at a height h_1 and $\rho_{fo}(T)$ is the density of hot water at collector outlet over a height $(h_1 + h_2)$. For a small temperature changes, the variations in density with temperature can be assumed to be given by following relationship.

$$\rho(T) = \rho_0 \{1- \beta\, T(t)\} \quad \ldots\ldots (2)$$

Where β is the coefficient of the volume thermal expansion and it was obtained by linear density variation with temperature. In the present system design analysis, the value of β was obtained by curve fitting will becomes approximately 0.000330 for the temperature range of 5 **^oC** to 100 **^oC**. $\rho_n(y)$ is the density variation of the water at any location y from the inlet of the collector. Rearranging Eq. (1) one gets

$$\Delta P_t = g\, h_1\, \rho_{fi}(T) - g\,(h_1 + h_2)\, \rho_{fo}(T) + g \sin\theta \int_0^L \{\rho_{fi}(T) - \rho_n(y)\}\, dy \quad \ldots(3)$$

Substituting temperature variation of densities from Eq. (2) to Eq (3), one gets

$$\Delta P_t = g\, \rho_0 \{[L/2\,(\sin\theta) + (h_1 + h_2)] * \beta\, [T_{fo}(t) - T_{fi}(t)] - h_2 \{1- \beta\, T_{fi}(t)\}\} \quad \ldots (4)$$

The value of exit temperature corresponding to P = 0 is the minimum temperature of the hot water obtained from the solar hot water systems known as balance point temperature T (t) by equating Eq(4) equal to zero, one gets

$$T_b(t) = T_{fi}(t) + h_2 \{1- \beta\, T_{fi}(t)\} / \{L/2\,(\sin\theta) + (h_1 + h_2)\} \quad \ldots. (5)$$

In the stationary condition of flow, the buoyancy force is responsible for the thermosyphonic mass flow rate is equal to the total pressure losses in the whole system (i.e. pressure losses in the collector plus pressure losses in the connecting tubes plus pressure losses in the hot water storage tank). In the present investigation, we assumed that no pressure losses in the hot water storage tank. Although the pressure losses in the hot water storage tank is very small as compared to the pressure losses in the whole system (i.e. collector plus connecting tubes), one can write

$$\Delta P_L = \Delta P_{L\,collector} + \Delta P_{L\,tubes} + \Delta P_{L\,tank} \approx \Delta P_{L\,collector}\, [1+ r_p] \quad \ldots (6)$$

where (r_P) gives the relationship between flow resistances in the outer connecting tubes to the flow resistance in the solar collector. In the present experimental set up, the fluid channels are of cylindrical geometries, one can therefore write pressure losses in the collector for laminar flow conditions as

$$\Delta P_L = \frac{32 \rho\, \nu\, v}{D^2}$$

The fluid velocity in the fluid flow channel is related to the mass flow rate inside the collector by following expression

$$V = \frac{4\, \dot{m}_c\,(t)}{\pi\, D^2 \rho}$$

and hence

$$\Delta P_L = \frac{128 \nu\, \dot{m}_c(t)}{\pi D^4} \left(L_n + 0.3N\right)$$

For the thermosyphonic mass flow rate to occurs, the total pressure drop through collector loop is equal to the buoyancy pressure in the whole system. Hence mass flow rate for the meander fluid channel absorber type solar sterilizer is

$$\dot{m}_c(t)=\left[\frac{g\rho_o\pi D^4}{128v(L_n+0.3N)}\right]\left\{\left[h_1-(h_1+h_2)-\sum\Delta h_n Sin\theta\right]-\left[\beta\left(h_1 T_{fi}-\Delta h_n\Delta T_n(t)Sin\theta\right)\right]-\beta(h_1+h_2)T_{fo}\right\}$$

4. Results and Discussions

Experiment was performed on the various systems for several days and data of experimental measurements for few days are presented in Table (2-3) respectively. It was observed that the meander fluid flow channel absorber type thermosyphonic solar sterilizer using copper fins with copper tube of seven turns (using 16.7 meters length) gives fifty liters of sterilized hot water for drinking and irrigation of sensible vegetable crops) within six hours. The computation was carried out to test the validity of proposed transient thermal model for thermosyphonic mass flow rate of sterilized water with the help of design thermal parameters shown in table-1 respectively.

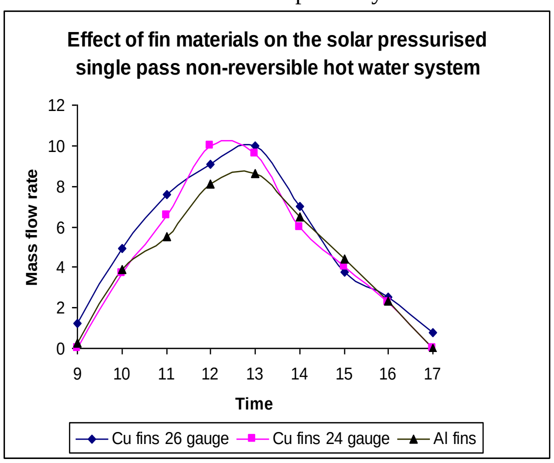

Fig: 1. Effect of fin materials on the pressurized solar water Heating Systems

Figs-1 shows the effect of various fins such as copper and aluminium fins of 24 & 26 gauges on three solar collectors, each of absorber area of 1.89 m2 and thermosyphonic mass flow rate was measured during half hour time (1800 seconds) intervals, and have been plotted . It was observed that copper fins gives better thermal performance than aluminium fins in the meander fluid flow channel absorber type solar pressurized systems. The performance of another collector of same absorber areas is also tested by using seven meander turn channel using tube diameter of 13.5 mm diameter of copper tube and copper fins of 26 gauge for two days and results have been presented in terms of half hourly collector mass flow rates, collector inlet and outlet temperatures, solar intensity and useful energy flux, thermal efficiency. It was observed that meander tube solar sterilizers are most suitable for rural applications in the remote areas where water quality is bad for drinking purpose. The effect of modification head variations is also plotted for keeping the water level is constant for meander and serpentine fluid flow channel absorber type solar sterilizers. Fig(2) show collector thermal performance of same absorber area of 1.89 m2 for inner tube diameter of 12 mm using copper fins and seven copper tube turns. It is clearly stated that meander shape fluid flow channel pressurized natural convection solar sterlizers are most useful.

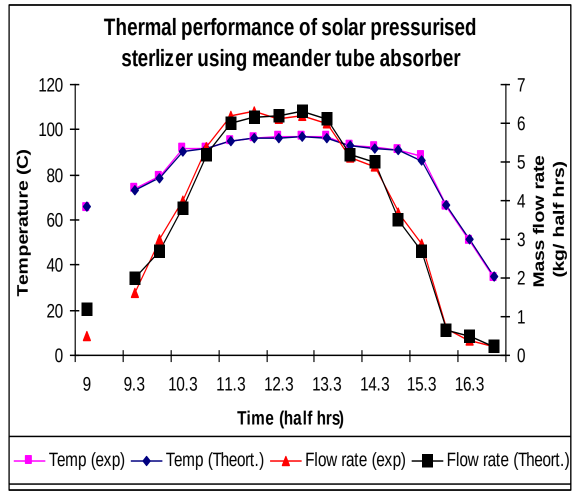

Fig: 2. Performance of Pressurized solar hot water Systems

The main drawback of these systems are lesser thermal collector efficiency due to less mass flow rate and absorber fluid at higher temperature resulting higher top loss from the collector absorber plate to ambient than non pressurized thermosyphonic solar hot water systems using parallel tube absorber/ parallel plate absorber of absorber area of 1.89 m2 but our purpose is to provide hot water at more than 80 ^oC can not be possible using non pressurized solar hot water systems using parallel tube / parallel plate absorber in the rural remote areas. Therefore pressurized solar sterilizers have been recommended. The problem of reducing top heat losses from the solar sterilizers was solved by inserting low cost one millimeter thick acrylic honeycomb structures of size 2100 mm X 900mm X 105 mm X 14 mm using seven aspect ratio between top of absorber and bottom of glazing surfaces and results in terms of thermal efficiency parameters are shown in Table 1. It was observed that the use of honeycomb structures are not techno- economically feasible for developing countries (such as Indian market conditions) but it reduces top heat losses from solar sterilizer surface of around 70%. The closed agreement has been observed which proves the validity of proposed transient thermal model.

Nomenclature:

A_t = Area of tank (m^2)
A_c = Area of solar energy collector (m^2)
B = Width of collector (m)
C_b=Bond Conductance (W/m ^oC)
d = Diameter of tube (m)
D = Diameter of header (m)
F_1=Fin efficiency factor for upper portion of absorber meander & serpentine type.
F_2=Fin efficiency factor for lower portion of absorber meander & serpentine type.
F_1' = Collector efficiency factor of upper portion of absorber in meander type.

F_2' = Collector efficiency factor of lower portion of absorber in meander type.
F' = Collector efficiency factor of parallel tube absorber
g = Acceleration due to gravity (m/sec^2)
H= Height of hot water storage tank from collector inlet (mm)
h_1= Height of tank up to water level (mm)
H_1= Height of tank bottom from collector outlet (mm).
H_2= Height of tank water level from collector outlet (mm)
h_2= Height of collector top (mm).
h_3= Height of tank bottom from collector outlet (mm)
h_e= Height of collector outlet from top absorber water level (mm).
h_4= Height of tank bottom from ground (mm).
h_5 = Height of collector inlet from ground (mm)
h_6= header diameter (mm)
h_7 = Distance between two header (mm)
h_8= Height of control valve (or tank top) from ground (mm).
h_{fi} = Heat transfer coefficient between tube and fluid (W/ m^2 ^{o}C)
I_t (t) = Solar intensity (or Insolation) W/m^2
K = Thermal conductivity of absorber (W/m ^{o}C)
K_b= Thermal conductivity of bond material in the absorber (W/m ^{o}C)
L = Length of solar energy absorber (m)
L_1 = Length of fluid flow channel (m)
m = fin factor
m_c(t)= mass flow rate of in the solar energy collector (Kg/sec)
M_w =Mass of water in the hot water storage tank (Kg)
N= Number of bends.
n = number of tubes
ΔP_c = Pressure due to buoyancy forces in the solar energy collector (N/ m^2)
ΔP_t = Pressure due to buoyancy forces in the storage tank(N/ m^2)
ΔP_{tube} = Pressure due to buoyancy forces in the storage tank (N/ m^2)
ΔP_{total} = Pressure due to buoyancy forces in the slar hot water system (N/ m^2)
ΔP_{Lc} = Pressure losses due to friction in the solar energy collector (N/ m^2)
P_{Lst} = Pressure losses due to friction in the storage tank (N/ m^2)
ΔP_{Lctube} = Pressure losses due to friction in the storage tank (N/ m^2)
ΔP_{Ltotal} = Pressure losses due to friction in the solar energy system (N/ m^2)
Q_u (t) = Useful energy heat flux (W/m^2)
r = Radius of tube in the meander & serpentine fluid flow absorber (mm)
R= Radius of headers in the parallel tube fluid flow absorber (mm)
R_e = Reynold number
t = time interval (hr or 1800 seconds).
T (t) = Temperature (^{o}C)
T_{ci} (t) = Mean temperature of water in the collector inlet (^{o}C)
T_{co}(t) = Mean temperature of water in the collector outlet (^{o}C)
T_a(t) = Ambient temperature (^{o}C)
T_b(t) = Local base temperature (^{o}C)
T_e(t) = hot water collector system outlet temperature (^{o}C)
T_{f1i}(t) = Inlet fluid temperature in the first tube of meander & serpentine type collector (^{o}C) T_{f20}(t)= Inlet fluid temperature in the first tube of meander & serpentine type collector (^{o}C)
T_c (t) = Variation of collector fluid temperature (^{o}C)
T_m (t) = Variation of hot water storage tank fluid temperature (^{o}C)
W = Distance between two absorber tubes (mm)
ρ(T) = Density (Kg/m^3)
ρ_{ci}(T) = Density of water at collector inlet (Kg/m^3)
ρ_{co} (T) = Density of water at collector outlet (Kg/m^3)
ρ_e (T) =Density of water at solar energy system outlet (Kg/m^3)
ρ(y) =Density of water in collector at distance y from inlet (Kg/m^3)
ρ_o = Density of water at 0 (^{o}C) (Kg/m^3)
$(\alpha \tau)_e$ = Effective absorptivity-and transmittivity products.
β = Coefficient of thermal expansion (^{o}C) -1
μ = Dynamic viscosity of water (kg-m/ sec)
ΰ = Kinematic viscosity of water (m^2/sec)
δ = Thickness of fins (mm)
θ = solar collector angle (Degree/ radians)

Table: 1a. Specifications of Modified Thermosyphonic Solar Hot Water Systems

Type of system	**Non-pressurized**	
Collector type	**System1: Parallel tube collecto**	**System2: Parallel tube collector**
Absorber plate	Aluminium	Copper
Total tube length/number Of fluid channels	10	10
Thickness of absorber plate	0.8 mm	0.5 mm
Paint on absorber plate	Black	Black
Transmissivity	0.88	0.88
Absorptivity	0.88	0.88
Absorber Area (m2)	2.1	2.1
Internal tube diameter (mm)	12.5	12.5
External tube diameter (mm)	12.7	12.7
Internal header diameter (mm)	25.4	25.4
External header diameter (mm)	25.9	25.9
Collector tilt angle (Degree)	40	40
Tube materials	Copper	Copper

Table: 2a. Specifications of Modified Thermosyphonic Solar Hot Water Systems

Type of system	**Pressurized**	
Collector type	**System3: Meander tube collector**	**System4: Meander tube collector**
Absorber plate	Al	Copper
Total tube length/ number of fluid channels	17.6 m	17.6 m
Thickness of absorber plate	0.5mm	0.8mm
Paint on absorber plate	Black	Black

Transmissivity	0.88	0.88
Absorptivity	0.88	0.88
Absorber Area (m2)	2.1	2.1
Internal tube diameter (mm)	12.5	12.5
External tube diameter (mm)	12.7	12.7
Internal header diameter (mm)	-	-
External header diameter (mm)	-	-
Collector tilt angle(Degree)	40	40
Tube materials	Copper	Copper

Table: 2b. Thermo symphonic modified multi pass reversible solar hot water systems using aluminium fins and copper tubes/ copper tubes and GI fins parallel tube fluid flow channel absorbers

Time	T i(t)	Ta(t)	It (t)	Exp. Tank Temp. System 1	Exp. Tank Temp .system 2
9 A.M	21.0	21.7	431	22.0	21.0
10	24	23.4	535	29.0	27.0
11	28	26	675	33.0	30.0
12	32	28.4	758	38.0	33.5
13	32.5	29.3	764	42.0	36.0
14	34.5	29.7	693	46.0	40.0
15.0	35.5	29.8	560	48.0	42.0
16.0	35.0	29.9	387	48.5	43.0
17.0	33.5	28.8	182	48.5	42.0

Table: 2c. Thermo symphonic modified multi pass reversible solar hot water systems using aluminium fins and copper tubes/ copper tubes and GI fins parallel tube fluid flow channel absorbers

Time	T i(t)	Ta(t)	It (t)	T c(t) Exp System 1	T (t) Exp System 2
9 AM	21.0	21.7	431	23.0	0.5
10	24	23.4	535	36.0	16.5
11	28	26	675	38.0	35.0
12	32	28.4	758	38.0	24.5
13	32.5	29.3	764	35.0	29.0
14	34.5	29.7	693	28.0	21.5
15.0	35.5	29.8	560	19.0	13.0
16.0	35.0	29.9	387	6.5	0.0
17.0	33.5	28.8	182	0.0	0.0

Table: 2d. Thermo symphonic modified multi pass reversible solar hot water systems using aluminium fins and copper tubes/ copper tubes and GI fins parallel tube fluid flow channel absorbers

Time	T i(t)	Ta(t)	It (t)	Exp. Syste m1 T fo(t)	Exp. System 2
9	21.0	21.7	431	45	21.5
10	24	23.4	535	65	43.5
11	28	26	675	71	65
12	32	28.4	758	76	68
13	32.5	29.3	764	77	65
14	34.5	29.7	693	74	61.5
15.0	35.5	29.8	560	67	55
16.0	35.0	29.9	387	55	35
17.0	33.5	28.8	182	48	33

Table: 3 Performance parameters of experimentally tested modified thermosyphonic solar hot water systems (Mishra[2])

S.N.	Modified collector's specifications Absorber size (2.1mx1.0m) =2.1sq m	Design parameters F'(τα)e	Design parameters F'UL
1.	Meander Collector (using copper fins and copper tubes) of ten turns	0.730	6.12
2.	Meander Collector (using copper fins and copper tubes) of eight turns	0.725	6.5835
3.	Meander Collector (using aluminum fins and copper tubes) of ten turns	0.725	6.5625
4.	Meander Collector (using copper fins and copper tubes) of seven turns	0.720	8.370
5	Meander Collector (using copper fins and copper tubes) of six turns	.0.720	8.720
6.	Meander Collector (using aluminum fins and copper tubes) of five turns of 8.8m tube length	0.720	11.23
7.	Meander Collector (using aluminum fins and copper tubes) of seven turns of 13.3m tube length	0.720	8.0
8.	Meander Collector (using aluminum fins and copper tubes) of seven turns of 15.6m	o.720	6.3436
9.	Meander Collector (using copper fins and copper tubes) of ten turns with honey comb	0.60	2.98

10.	Meander Collector (using copper fins and copper tubes) of ten turns with honey comb	0.6	3.20
11	Meander Collector (using copper fins and copper tubes) of ten turns with honey comb	0.6	3.8
12.	Meander Collector (using copper fins and copper tubes) of ten turns with honey comb	0.645	2.8667
13.	Meander tube Collector (using copper fins and copper tubes) of ten tubes with honey comb	0.645	2.8667
14	Meander tube Collector (using aluminum fins and copper tubes) of ten tubes with honey comb	6.45	3.0353
15.	Serpentine tube Collector (using aluminum fins and copper tubes) of ten tubes with honey comb	0.645	4.5
16	Meander tube Collector (using copper fins and copper tubes) of ten tubes	0.725	6.4
17	Meander tube Collector (using copper fins and copper tubes) of ten tubes	0.725	7.20
18	Meander tube Collector (using copper fins and copper tubes) of ten tubes with honey comb	0.725	6.5825
19	Meander tube Collector (using copper fins and copper tubes) of ten tubes with honey comb	0.75	7.02
20	Meander tube Collector (using copper fins and copper tubes) of ten tubes	0.725	6.6732
21	Meander tube Collector (using copper fins and copper tubes) of ten tubes with honey comb	0.645	2.8667
22	Meander tube Collector (using copper fins and copper tubes) of ten tubes	0.725	6.5
23	Meander tube Collector(using copper fins and copper tubes) of ten tubes	0.725	7.25
24	Meander tube Collector(using copper fins and copper tubes) of seven turns	0.725	8.2857
25	Meander tube Collector(using copper fins and copper tubes) of five turns of tube length 8.9 m	0.725	10.6
26	Meander tube Collector(using copper fins and copper tubes) of six turns of tube length 8.8	0.125	11.6
27	Meander tube Collector(using copper fins and copper tubes) of ten tubes with honey comb	0.6939	4.4658
28	Meander tube Collector(using copper fins and copper tubes) of ten tubes with honey comb	0.6724	4.39
29	Meander tube Collector(using aluminum fins and copper tubes) of ten tubes with honey comb	0.6720	4.95
30	Meander tube Collector(using aluminum fins and copper tubes) of ten tubes	0.7696	7.3632
31	Meander tube Collector(using copper fins and copper tubes) of ten tubes	0.7744	6.77955

Table: 1b. Specifications of Modified Thermosyphonic Solar Hot Water Systems

Type of system	**Pressurized**	
Collector type	**System5:Serpen-tine tube collector**	**System6: Serpen-tine Tube collector**
Absorber plate	Copper	Al
Total tube length/number of fluid channels	17.6 m	17.6 m
Thickness of absorber plate	0.8mm	0.8mm
Paint on absorber plate	Black	Black
Transmissivity	0.88	0.88
Absorptivity	0.88	0.88
Absorber Area (m2)	2.1	2.1
Internal tube diameter (mm)	12.5	12.5
External tube diameter (mm)	12.7	12.7
Internal header diameter (mm)	-	-
External header diameter (mm)	-	-
Collector tilt angle(Degree)	40	40
Tube materials	Copper	Copper

Table: 1b. Specifications of Modified Thermosyphonic Solar Hot Water Systems

Type of System	**Pressurized**			
Collector type	**System5: Serpen-tine tube collector**	**System6: Serpen-tine Tube collector**	**System7: Serpen-tine Tube collector**	**System8: Serpen-tine Tube collector**
Absorber plate	Copper	Al	Copper	Al.
Total tube length/number of fluid channels	17.6 m	17.6 m	8.8 m	8.8 m
Thickness of absorber plate	0.8mm	0.8mm	0.8mm	0.8mm
Paint on absorber plate	Black	Black	Black	Black
Transmissivity	0.88	0.88	0.88	0.88
Absorptivity	0.88	0.88	0.88	0.88
Absorber Area (m2)	2.1	2.1	2.1	2.1
Internal tube diameter (mm)	12.5	12.5	12.5	12.5
External tube diameter (mm)	12.7	12.7	12.7	12.7
Internal header diameter (mm)	-	-	-	-
External header diameter (mm)	-	-	-	-
Collector tilt angle(Degree)	40	40	40	40
Tube materials	Copper	Copper	Copper	Copper

References

[1] R. S. Mishra, Thermal Performance of Meander Tube Collector for Solar Hot Water Systems, National Conference of Mechanical Engineers, I.I.T. Kanpur, 1993

[2] R.S. Mishra, Thermal Analysis of Novel Natural Convection Modified Solar Hot Water Systems, National Conference of Agricultural Engineers, Junagarh, 1994.