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RESEARCH ARTICLE

Experimental and theoretical investigation in the non-pressurized thermosyphonic domestic solar hot water systems

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Abstract

This paper presents a dynamical model for calculating the thermosyphonic mass flow rate in natural convection solar domestic water heating systems, specifically for systems with a vertical hot water storage tank. The numerical results from the developed model exhibit excellent agreement with experimental measurements. The model effectively predicts the performance of various domestic thermosyphonic non-pressurized solar hot water systems, including those with parallel plate absorbers. Additionally, the performance of sixteen thermosyphonic solar hot water systems using different fin materials is discussed. The computed results for the thermosyphonic mass flow rate from the developed model for a single-pass thermosyphonic solar water heating system with a parallel tube fluid flow channel match experimental measurement well. Furthermore, a multi-pass non-pressurized solar hot water system, designed to eliminate reverse flow, is introduced. The non-pressurized thermosyphonic systems with parallel tube absorbers show an average thermal efficiency of 41% in the single-pass mode and 27.3% in the multi-pass mode using aluminum-finned absorbers. To further enhance efficiency, honeycomb structures are proposed to reduce top heat losses from the solar flat plate collector by approximately 70%.

1. Introduction

Energy is a critical input in the national development process. In fact, it is the basic requirement for human life, agriculture, industry, transportation, communication and many other economic activities of the present civilization. In the present day, the depleting fossil fuels in the various countries, the terms of energy crises underline the need of paying serious attention to the effective /efficient utilization of existing conventional and non-conventional energy sources in terms of energy conservation through effective management for maximum agricultural production. Solar water heater is one of the most successful solar technologies. Nowadays, world's demand of energy has dramatically increased; furthermore, process to collect hot water by solar radiation is yet expensive. Most solar water heater designs used for single family are the closed and opened solar water heating systems. These two systems are categorized into two groups: forced circulation and natural convection. The advantages of thermosyphon systems are that they do not rely on pumps and controllers, are more reliable, and have a longer life than forced circulation systems.

1.1 Analysis of natural convection water heating systems

Various theoretical and experimental studies which have been performed on thermosyphonic solar water heating systems starting from Closed et.al, [2], The studies of Metrol [3] and other investigators [4,5], attempted to write down very detailed

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equations for each component. The thermal analysis, if employed in practice always required the use of bigger expensive computer time. The studies of Ong [5], Sodha and Tiwari [6] Uhelmann & Bansal et.al [7] based on the simple assumption that whole solar hot water system is at some average temperature, lack in sense of comparison with detailed experimental investigations. Also, the basis for estimating thermodynamic mass flow rate has not been established from the first principles. Mishra [8,9,] has developed thermal model for predicting thermal performances of thermosyphonic solar hot water systems for without withdrawal, with withdrawal cases for a given typical hot water demand which vary from place to place, region to region, country to country and found that theoretical values obtained from developed thermal model matched very well with the experimental measurements on natural convection solar hot water systems [11-15].

1.2 Modified system design for free convection solar water heating systems

In natural convection solar water heating systems, buoyant forces created by density gradients in the fluid inside the collector create pressure, which causes the liquid to flow in the collector loop. These systems don't need a pump, and the tank is typically positioned above the collector. Reverse flow is a frequent issue in these systems, which causes significant heat losses during the off sun shining hours.

In thermosyphonic free convection solar water heating systems, the phenomenon of reverse flow causes significant heat losses from solar energy collectors during the hours when the sun is not shining. Only a tiny amount of literature has addressed in the literature consequently. Reverse flow can occurs in the thermosyphonic solar water heating systems during periods of no sunshine hours, especially when the bottom of storage tank is placed below the top of the solar collector. During the day time, the water in the storage tank gets heated and the top portion remains hot as compared to bottom portion due to radiation falling on a solar collector which absorb solar radiation to heat water in the solar collector. From the storage tank, during off sunshine hours the hot water comes in to solar collector due to density difference and gets cooled much faster than the pipe due to large surface area of the collector is exposed to sky. The results is that the left-hand side of the tube gets cooler and becomes heavier while right hand side of the tube remains hot and flow of liquid starts in the reverse direction. This brings down the temperature of water in the storage tank drastically. Measurements and discussion of reverse flow is rare in literature. Recently few investigators have observed the phenomenon of reverse flow in the system and carried out experimental studies on pressurized and non-pressurized systems and found that if the tank is kept very high above the collector, then there is always some flow of liquid through the collector especially during night time, if the outlet of collector is connected near the middle of the tank. This also results in considerable losses resulting poor thermal performance of the system. It was observed that by raising the height of the storage tank relative to the top of absorber, the temperature difference causing the reverse flow is decreased. By careful design and installation, the reverse flow can ne completely avoided. For reverse flow to be completely absent, the tank should be placed as high as possible above the collector (i.e. 200mm) between 150 mm to 250 mm depending upon collector fluid flow geometry. However, one has to consider the extra piping associated with the cost and also thermal losses. Some novel designs of solar water heating systems which can completely eliminate reverse flow and also allow the placement of hot water storage tank at an acceptable level are described [16].

2. Derivation of thermosyphonic mass flow rates in free convection 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:

$$\Delta P_{t=}\ \Delta P_1\ + \Delta P_2 \tag{1}$$

The pressure difference ΔP_1 is due to the buoyancy force in the solar energy collector and another part ΔP_2 is due to the density variation in the connecting tubes. The buoyancy pressure in the solar energy collector was calculated through an integration over the length of the collector i.e.

$$\Delta P_1\ = \mathrm{gsin}(\Theta)\times[\rho_1 - \rho\,(\mathrm{y})]\mathrm{dy} \tag{2}$$

Where ρ (y) is the density of water at the any location from the inlet of the solar energy collector and the ρ_1 is the density of water corresponding to T_1, the temperature of water at the inlet of the solar collector. Similarly, by considering the density variations over the height (H to be constant, the buoyancy pressure ΔP_2 can be written as

$$\Delta P_2 = [\rho_1\ \text{-}\rho_2]\mathrm{gH} \tag{3}$$

Substituting equation (2) and equation (3) in equation (1), we gets.

$$\Delta P_t = [g\sin(\Theta)\times[\rho_1 - \rho(y)]dy] + [\rho_1 - \rho_2]gH \quad (4)$$

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

$$\rho(t) = \rho_0 \times (1-\beta(T)) \quad (5)$$

where β is the coefficient of volume expansion and ρ_0 is the density at 0℃. Now substituting of equation of equation [5] in equation (4) yields

$$\Delta P_t = g\,\beta\rho_0\,[g\sin(\Theta)\times(T(y)-T_1)dy + H(T_2-T_1)] \quad (6)$$

In the stationary conditions of the flow, the total buoyancy pressure [$\Delta P_{t=}\Delta P_{Ls}$], in the system (i.e. collector, connecting tubes and the storage water tank). Denoting the pressure losses in the solar collector as ΔP_c and in the connecting tubes as [ΔP_z] and we assumed the pressure losses in the storage tank being negligible, one can write

$$\Delta P_{Ls} = \Delta P_c + \Delta P_z = (\Delta P_c\,(1+(\Delta P_z/\Delta P_c)) = (\Delta P_c\,(1+r_p) \quad (7)$$

Where $r_p=(\Delta P_z/\Delta P_c)$ gives the relationship between the flow resistance in the outer connecting tubes to the flow resistance in the solar collector. Equating ΔP_t *to* ΔP_{Ls} given by equation (7), one gets

$$g\,\beta\rho_0\,[\sin(\Theta)\times(T(y)-T_1)dy + H(T_2-T_1)] = (\Delta P_c\,(1+r_p) \quad (8)$$

we assumed that the temperature distribution in the collector to be linear then one can write

$$T(y)-T_1=(T_2-T_1)\times(y/L) \quad (9)$$

Hence

$$[T(y)-T_1=(T_2-T_1)\times(L/2)] \quad (10)$$

Substituting equation (10) in equation (8), one gets

$$([g\,\beta\rho_0\times(T_2-T_1)\times((L/2)\sin(\Theta)+H) = (\Delta P_c\,(1+r_p)] \quad (11)$$

One should now get a relationship between the temperature and mass flow rate(m_c). For this one can use the energy balance equation in the solar collector

$$Q_u = m_c Cpw\,(T_2-T_1) = (F'Ac(I_a-U_L(T_m-T_a)) \quad (12)$$

Where Qu is the useful energy collected from solar collector, m_c is the mass flow rate (kg/sec) in the closed loop cycle, A_c is the solar collector area (m^2),[I_a] is the effective absorbed solar radiation(insolation), [U_L] is the total heat loss coefficient (W/m^2K) from solar collector and [T_m] is the mean solar collector temperature and [T_a] is the ambient temperature. Normally we assumed that

$$[T_m=(T_2-T_1)/2)] \quad (13)$$

Substituting Eq.[13] into eq.[12] and adjusting, one gets

$$(T_2-T_1)=[(F'Ac)/(m_c Cpw)]\times[\,(I_a-U_L(T_m-T_a))] \quad (14)$$

Eliminating (T_2-T_1) from Eq.(11) and eq. (14) and substituting A_c=(B×L) in which [B =width of the solar collector and L is the length of solar the solar collector), one gets

$$[(B\times L/m_c)\times[g\,\beta\rho_0\,F'/(1+r_p)Cpw]\times[I_a-U_L(T_m-T_a)] = (\Delta P_c/L_1)L_1] \quad (15)$$

Where [L_1] is the length of the fluid flow channel in the solar collector. The pressure losses ($\Delta P_c/L_1$) per unit length of the solar collector depends on the mass flow rate of the fluid in the solar collector. If this dependence is known, Eq.[15] can be solved for finding mass fow rate in the solar collector. Assuming the fluid channel is to be of cylindrical geometry, the pressure losses can be written as

$$(\Delta P_c/L_1) = (64/Re)\times(\rho V^2/2D) = [32\,(\mu/\rho)\,\rho\,V/D^2] \quad (16)$$

Where (μ/ρ) is known as kinematic viscosity (m^2/sec) and [μ] is dynamic viscosity in (poise), [V] is the velocity of fluid (m/sec) and [D] is the diameter of the tube. For two parallel plates at a distance of [w] from each other, one has

$$[\Delta P_c/L_1] = [12\,(\mu/\rho)\,\rho\,V/w^2] \quad (17)$$

Assuming for an absorber with [N] tubes one can get following expression for parallel tube absorber

$$[m_c] = [3.14\times(D^2/4)\times(N\,\rho V)] \quad (18)$$

and for parallel plate absorber $[m_c] = \rho VBW$(19)

substituting for [V] in equation (16) and equation (17) one gets for parallel tube absorber

$$(\Delta P_c/L_1) = [128B/(3.14\times ND^4)((\mu/\rho)\times(m_c/B) \quad (20)$$

and for parallel plate absorber

$$(\Delta P_c/L_1) = [12/(3.14\times W^3)((\mu/\rho)\times(m_c/B) \quad (21)$$

Substituting equation [19] in equation [15], one gets

$$(m_c/B) = [3.14\times(L/128)\,(ND^4/(BL_1)\times((g\,\beta\rho_0\,F'/(1+r_p)\,\mu\,Cpw)\times[\,I_a-U_L(T_m-T_a))\times(L/2)\sin(\Theta)+H)]^{0.5} \quad (22)$$

For parallel tube absorber and for parallel plate absorber, the below formula allow the calculation of the mass flow rate of parallel plate absorber of thermosyphonic solar water heating systems

$$(m_c/B)= [W/12]\times(g\,\beta\rho_0\ F'/(1+r_p)\ \mu\ Cpw)\times[\ I_a-U_L(T_m-T_a))\times(L/2)\sin(\Theta)+H)]^{0.5} \quad (23)$$

The volumetric expansion coefficient can be calculated from the temperature dependence of the density by using following expression

$$(\Delta\,\rho/\,\rho)= -\,\beta\,\Delta T \quad (24)$$

In the temperature range from 20°C to 50°C, the value of β will be ($3.3\times1^{-4}/\,^{o}C$).

3. Results and Discussion

The results obtained from the developed thermal model for mass flow rate were validated against experimental measurements, and the comparison of these results for thermosyphonic solar hot water systems is shown in Table 1. The experimental and theoretical results exhibited a good agreement, confirming the reliability and accuracy of the developed model. The study also focused on the thermal efficiency of non-pressurized solar hot water systems with parallel tube absorbers. The average thermal efficiency was determined to be 41% in single-pass mode and 27.3% in multi-pass mode, as shown in Tables 1. These results highlight the effectiveness of thermosyphonic solar water heating systems in converting solar energy into usable heat. The thermal performance of non-pressurized thermosyphonic solar water heating systems using parallel copper tubes and copper fin absorbers was compared with systems using parallel copper tubes and aluminum fin absorbers. As indicated in Table 1, systems with copper fins demonstrated better thermal performance than those with aluminum fins. The enhanced performance of copper fins can be attributed to their superior thermal conductivity, which promotes better heat transfer and overall system efficiency. In contrast, the use of aluminum fins, while still effective, resulted in slightly lower thermal performance.

Table 1: Thermal performances of single pass thermosyphonic non pressurized solar water heating system using parallel tube fluid flow channel absorber (Collector Area =2.1m²)

Time (hr)	Inlet Water Tempera-ture (°C)	Ambient Tempera-ture (°C)	Solar Intensity (W/m²)	Collector Outlet fluid tempera-ture Exp. (T_{CO}) (°C)	Collector Outlet fluid tempera-ture Model (T_{CO})(°C)	Collector mass flow rate Exp. (Kg/hr)	Collector mass flow rate (Model) (Kg/hr)	Collector tempera-ture differ-rence (°C) Exp.	Collector tempera-ture differ-rence (°C) Model	Useful energy from solar water heating collector ($Q_u(t)$) W	Thermal efficiency of solar water heating collector
9AM	21.0	21.7	430.0	66.0	65.5	3.0	2.96	45.0	44.5	157.5	0.174
10	24.0	23.425	535.0	78.0	79.8	9.0	8.97	54.0	55.8	262.5	0.234
11	28.0	26.05	675.0	83.0	84.2	11.0	11.0	55.0	56.2	567.0	0.40
12	32.0	28.4	760.0	84.0	84.8	14.0	14.05	52.0	52.8	705.8	0.443
13PM	32.5	29.32	765.0	83.0	84.5	12.0	13.55	50.5	52.0	849.3	0.5295
14	34.5	29.67	695.0	81.5	80.8	11.55	12.0	47.0	46.3	707.0	0.4860
15	35.5	29.8	560.5	77.0	75.9	7.0	7.24	41.5	40.5	498.6	0.4236
16	35.0	29.9	387.0	65.0	66.1	6.0	5.96	30.05	31.1	338.9	0.417
17	33.5	28.8	182.0	41.0	40.5	3.55	2.97	5.5	7.0	210.0	0.5495

Table 2: Thermal performances of single pass thermosyphonic non pressurized solar water heating system using parallel tube fluid flow channel absorber (Collector Area =2.1m²)

Time (hr)	Inlet Water Temperature (°C)	Ambient Temperature (°C)	Solar Intensity (W/m²)	Collector Outlet fluid tempera-ture Exp. (T_{CO}) (°C)	Collector Outlet fluid tempera-ture Model (T_{CO}) (°C)	Collector mass flow rate Exp. (Kg/hr)	Collector mass flow rate (Model) (Kg/hr)	Collector tempera-ture differ-rence (°C) Exp.	Collector tempera-ture differ-rence (°C) Model
9AM	21.0	21.7	430.0	66.6	65.5	3.0	2.96	45.0	44.5
10	24.0	23.5	535.0	78.0	79.8	9.0	8.97	54.0	55.8
11	28.0	26.0	675.0	83.0	84.2	11.0	11.0	55.0	56.2
12	32.0	28.4	760.0	84.0	84.8	14.0	14.05	52.0	52.8
13PM	32.5	29.3	765.0	83.0	84.5	12.0	13.55	50.5	52.0
14	34.5	29.7	695.0	81.5	80.8	11.55	12.0	47.0	46.3
15	35.5	29.8	560.5	77.0	75.9	7.0	7.25	41.5	40.5
16	35.0	29.9	387.0	65.0	66.1	6.0	5.96	30.5	31.1
17	33.5	28.8	182.0	41.0	40.5	3.55	2.97	5.5	7.0

Tables 1 and Table 2 present the thermal performance data for a single-pass thermosyphonic non-pressurized solar water heating system using parallel tube fluid flow channel absorbers with a collector area of 2.1 m². The data is provided for different time intervals during the day, typically from 9 AM to 5 PM, showing various system parameters affecting performance. The tables include information such as inlet water temperature, ambient temperature, solar intensity, and the collector outlet fluid temperature. Experimental and modeled data are compared, with the experimental data showing the actual outlet temperature, mass flow rate, and temperature differences, while the model predicts these values. The data shows good agreement between the experimental and modeled results, confirming the accuracy of the thermal model. The tables also present the useful energy harvested from the collector and the thermal efficiency of the solar water heating system at each time interval. The efficiency is highest during midday when solar intensity is greatest (around 0.44 to 0.49), and it decreases in the early morning and late afternoon (around 0.17 to 0.42). This indicates that the system performs optimally during peak sunlight hours. Overall, the data provides a comprehensive understanding of the system's performance, demonstrating the relationship between solar intensity, thermal efficiency, and time of day. Similarly, in the multipass mode. system efficiency of meander tube absorbers is ranging from 24% to 30.6% as shown in Table 2 to Table 3 respectively.

Table 3: Thermal performances of single pass thermosyphonic non pressurized solar water heating system using meander fluid flow channel absorber (Collector Area =2.1m^2) (Eff=(2.5604/10.47)= 0.2445

Time (hr)	Inlet Water Temperature (°C)	Ambient Temperature (°C)	Solar Intensity (W/m^2)	Collector Outlet fluid temperature Exp. (T_{CO}) (°C)	Storage tank inlet fluid temperature Model ($T_w(t)$ (°C)	Collector tempera-ture difference (°C) Exp.	Storage tank fluid temp difference Model ($T_w(t)$)	Useful energy from solar water heating collector ($Q_u(t)$) W	Thermal efficiency of solar water heating collector
9AM	21.0	21.7	430.0	21.5	21.0	0.5	0	0	0
10	24.0	23.5	535.0	43.5	27.0	16.5	5	58.2	0.06423
11	28.0	26.0	675.0	65.0	30.0	35.0	3	698.33	0.6213
12	32.0	28.4	760.0	68.0	33.5	34.5	3.5	349.17	0.24620
13PM	32.5	29.3	765.0	65.0	36.0	29.0	2.5	349.2	0.2462
14	34.5	29.7	695.0	61.5	40.0	21.5	4.0	290.9	0.2459
15	35.5	29.8	560.5	55.0	42.0	13.0	2.0	465.56	0.32844
16	35.0	29.9	387.0	35.0	43.0	0	1.0	232.7	0.1642
17	33.5	28.8	182.0	33.0	42.0	0	0	116.39	0.08216
17.30	31.5	27.5	50.0	30.0	40.0	0	0	0	0

Table 3 presents the thermal performance data for a single-pass thermosyphonic non-pressurized solar water heating system using a meander fluid flow channel absorber with a collector area of 2.1 m². The data covers various time intervals throughout the day, from 9 AM to 5:30 PM, and includes parameters like inlet water temperature, ambient temperature, solar intensity, and the collector outlet fluid temperature. Additionally, the table compares experimental and modeled values for the storage tank inlet fluid temperature and temperature differences between the collector and storage tank. The thermal efficiency of the system is calculated as 0.2445, as shown at the top of the table, based on the ratio of useful energy extracted to the available solar energy. At 9 AM, the inlet water temperature and ambient temperature are relatively low, resulting in negligible useful energy and thermal efficiency. As the day progresses and solar intensity increases, the collector outlet temperature rises, leading to higher thermal efficiency, peaking at 11 AM with a thermal efficiency of 0.6213, corresponding to a significant temperature difference between the collector and the storage tank. However, the thermal efficiency decreases as solar intensity begins to decline in the afternoon, dropping to 0.08216 at 5 PM. Thermal performance parameters of eight pressurized solar hot water systems and another eight non pressurized solar hot water systems using parallel tube absorbers by using HWB equation are shown in Table 4.

Table 4: Performance Parameters of thermosyphonic free convection solar hot water systems (collector Area=2.1 m^2)

S.No	Non-Pressurized thermo-syphonic systems using parallel tube fluid flow channels (10 tubes) of copper tube length
1	$F'(\tau\alpha)e$=0.720 and $F'U_L$ =8.089
2	$F'(\tau\alpha)e$=0.725 and $F'U_L$ =8.055
3	$F'(\tau\alpha)e$=0.720 and $F'U_L$ =6.1017
4	$F'(\tau\alpha)e$=0.725 and $F'U_L$ =5.80
5	$F'(\tau\alpha)e$=0.720 and $F'U_L$ =5.214
6	$F'(\tau\alpha)e$=0.720 and $F'U_L$ =6.344
7	$F'(\tau\alpha)e$=0.725 and $F'U_L$ =8.089
8	$F'(\tau\alpha)e$=0.725 and $F'U_L$ =6.12
9	$F'(\tau\alpha)e$=0.720 and $F'U_L$ =6.344
10	$F'(\tau\alpha)e$=0.725 and $F'U_L$ =6.12
11	$F'(\tau\alpha)e$=0.720 and $F'U_L$ =8.055
12	$F'(\tau\alpha)e$=0.725 and $F'U_L$ =7.20
13	$F'(\tau\alpha)e$=0.725 and $F'U_L$ =5.2174
14	$F'(\tau\alpha)e$=0.725 and $F'U_L$ =5.80
15	$F'(\tau\alpha)e$=0.725 and $F'U_L$ =6.1017
16	$F'(\tau\alpha)e$=0.725 and $F'U_L$ =7.20

4. Conclusions

Following conclusions were made from present investigations.

- Computed results for thermosyphonic mass flow rate from developed thermal model using single pass thermosyphonic solar water heating system using parallel tube fluid flow channel matches very well with experimental measured values.
- The multi pass solar hot water non pressurized systems eliminating reverse flow have been developed, which have two unique characteristics. (i.e.Water is obtained at a predetermined temperature regardless of insolation levels, and also permits the main water storage tank to be placed anywhere desired (below the solar collector).
- The non-pressurized solar hot water thermosyphonic systems with parallel tube absorber have average thermal efficiency of 41 % in the single pass and 27. 3 % in the multi-pass modes using aluminium finned absorbers.
- To reduce top losses from solar flat plate collector, the honeycomb structures have been proposed which can reduce around 70% of top heat losses from solar collector.

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