SIMILARITY THEORY OF SOLAR WATER HEATER WITH NATURAL CIRCULATION

B. J. HUANG

Department of Mechanical Engineering, National Taiwan University, Taipei, Taiwan, Republic of China

(Received 27 June 1979; revision accepted 11 December 1979)

Abstract—The similarity theory of solar thermosyphon collector is developed in the present paper. Ten dimensionless groups or system characteristic parameters which uniquely determine the performance of the collector are derived. The solution shows that the mean efficiency generally increases with increasing incident solar radiation and relative height of the tank. For the frictional parameters N_c and N_f higher than 10^5 , the efficiency appears to be independent of the incident radiation and the relative height of the tank. Therefore, for parallel plate absorber, the tank may be designed to sit on the floor without sacrificing the efficiency since the values of N_c and N_f are usually larger than 10^5 in most designs.

INTRODUCTION

A solar hot water heater with natural circulation is now one of the most economic devices for collecting solar energy for domestic or industrial uses. Many theoretical and experimental work have been carried out during the past decades to study the thermal performance of the collector. A common conclusion deduced from a great deal of experiments performed by many investigators indicate that for better performance the relative height of the storage tank above the absorber generally must be higher than thirty centimeters for the ordinary collector sizes being tested [1]. For different collector designs or large-scale collectors which might be used in large-scale applications, it is believed that this design criteria would change. However, little has been known about this.

A mathematical model which introduces a mean system temperature and assumes linear temperature distributions in absorber plate and storage tank was first proposed by Close [2] to predict the collector performance. Further improvements are made by Gupta and Garg [3] later and more recently by Ong [4, 5]. Gupta and Garg [3] improved Close's analysis by incorporating a plate efficiency factor to account for the thermal efficiency of the plate and considering the variations of ambient temperature and radiation intensity. Ong [4] further carefully considered the plate efficiency factor and also assumed linear temperature distributions in absorber plate and storage tank. A finite difference method was then employed to solve the governing equations. Another modification was again made by Ong [5]. Instead of assuming linear temperature distribution, Ong further considered the entire system to be broken up into a finite number of sections each individual section having a uniform mean temperature, then solved the finite difference equations. The solutions were found in very good agreement with experimental results. The theoretical analyses mentioned above however can only predict the performance for a specified collector with known dimensions. No general design rules can be drawn from the above analyses to tell what is the effect of the relative height of the storage tank above the absorber plate on the performance for different sizes of collector.

Only little evidence was noticed by Ong [5] in his experiment which indicates that the mean collector efficiency increases with increasing relative height of tank as well as solar radiation intensity.

Grossman *et al.* [6] attempted to normalize the governing equations of the collector to obtain similarity solutions for a parallel-plate type absorber itself. For simplicity, they assumed that the performance is quasi-steady with a known function of net heat flux distribution reaching the fluid through the upper plate and ignored heat transfer and frictional loss in the connecting pipes and the storage tank. Therefore, the analytical results only give a very rough picture of the collector performance. The effect of the relative height of the storage tank above the absorber plate as well as the frictional loss on the system performance was still not quite clear.

The present work attempts to develop a more general similarity theory by a model which divides the entire collector into three components: absorber plate, storage tank and connecting pipes. Two energy equations for absorber and storage tank and a momentum equation for the entire flow system are derived. The governing equations are then normalized to come up with ten dimensionless groups which characterize the performance of thermosyphon collector. The similarity solutions thus give a very clear picture on the collector performance for any size of collector and result in some important design rules.

MATHEMATICAL MODEL

The thermosyphon solar collector studied in the present paper basically consists of three major components: namely, absorber plate, storage tank and connecting pipes (see Fig. 1). The incident solar energy radiation is absorbed by the blackened surface of the absorber plate and transmitted through the plate material to the fluid. The temperature rise of the fluid inside the absorber then causes to a density decrease and induces a driving force between the absorber side (hot side) and the storage tank side (cold side) due to the density difference. By this driving force, called thermosyphon head, the fluid circulates upward from the absorber

Fig. 1. Schematic diagram of thermosyphon solar collector and temperature distribution curve.

Fig. 2. Schematic diagram of parallel-plate absorber.

through the storage tank and returns to the absorber. For simplicity, we assume that no water is drawn off from the storage tank during daytime although hot water drawn-off would significantly affect the performance of the collector. Moreover, the absorber design is assumed to be of parallel plate type as shown in Fig. 2 in the present analysis.

To facilitate the analysis, some assumptions are made:

(1) The velocity and temperature distributions of the fluid in the storage tank are approximately one-dimensional [7, 8].

(2) The velocity and temperature distributions inside the absorber plate are one-dimensional.

(3) The transient behavior of the heat transfer inside the connecting pipes is ignored.

(4) The storage tank and the connecting pipes are well-insulated therefore the heat loss from them are negligible.

(5) The effect of the heat capacity due to the constructing material of the whole system can be ignored.

(6) The flows in the tank, the connecting pipes and the absorber are laminar since the Reynolds number is usually smaller than 3000 in most applications.

(7) The heat conduction along the flow direction in the absorber is negligible since the Peclet number is usually large.

(8) No water drawn-off during daytime is allowed.

(9) The incoming solar energy received and actually converted into heat energy at the black surface of the absorber can be approximately represented by a sine function, i.e. $S = G_m \sin(\pi t/\omega)$ where ω is the total sunshine duration, G_m is the radiation intensity at solar noon.

Energy balance

For parallel-plate absorber considered in the present paper (Fig. 2), taking an energy balance on an element of fluid inside the absorber yields the energy equation:

$$\frac{\partial T}{\partial t} + \left(\frac{qL_c}{\delta} \right) \frac{\partial T}{\partial x} = \frac{F'}{\rho C_p \delta} \left[G_m \sin(\pi t/\omega) - U_L(T - T_a) \right] \quad (1)$$

where

$$F' \equiv \frac{h_i}{U_L(1 + B_i) + h_i},$$

$$B_i \equiv h_i e / k_c,$$

$$q \equiv v_c \delta / L_c.$$

Here, F' is the absorber efficiency factor which depends on the collector design, v_c is the mean velocity inside the absorber, B_i is the Biot number which represents the ratio of heat convection to the heat conduction between the absorber plate and the fluid, and q represents the volumetric flowrate per unit area of collector. U_L is the overall heat loss coefficient from the black surface of the absorber to the ambient, T_a is the ambient temperature which is assumed constant in the present analysis, T is the local fluid bulk temperature. The above equation can be normalized to the form:

$$N_c \frac{\partial T^*}{\partial t^*} + q^* \frac{\partial T^*}{\partial x^*} = F' N_r \sin \pi t^* - F' N_L T^*, \quad (2)$$

where

$$T^* \equiv \frac{T - T_a}{S_o/U_o},$$

$$x^* \equiv x/L_c$$

$$q^* \equiv q/(U_o/\rho C_p)$$

$$t^* \equiv t/\omega$$

$$N_c \equiv \rho C_p \delta / U_o \omega$$

$$N_r \equiv G_m/S_o$$

$$N_L \equiv U_L/U_o.$$

Here, S_o is the solar constant, 1.353 kW/m^2 , U_o is the reference heat loss coefficient and is usually taken as the value for collector with single glazing, and L_c is the length of the absorber plate. It can be seen that normalization of eqn (1) results in four dimensionless groups which characterize the thermal performance of the absorber: namely, N_c , N_r , F' and N_L . N_c has the physical meaning of the rate of heat storage by the fluid inside the absorber plate and which controls the thermal transient behavior. N_r represents the radiation intensity incident upon the collector. N_L represents the quantity of heat loss from the absorber plate.

Similarly, applying energy balance to the storage tank for the condition of no water drawn-off we obtain:

$$\frac{\partial \theta}{\partial t} + \frac{A_c H_T}{V_T} q \frac{\partial \theta}{\partial y} = \frac{k_f}{\rho C_p} \frac{\partial^2 \theta}{\partial y^2}, \quad (3)$$

where A_c is the absorber surface area, V_T is the water volume in tank. H_T is the water height in the tank. Equation (3) can be normalized to

$$\frac{\partial \theta^*}{\partial t^*} + \frac{1}{N_T} q^* \frac{\partial \theta^*}{\partial y^*} = \frac{1}{P_e} \frac{\partial^2 \theta^*}{\partial y^{*2}}, \quad (3a)$$

where

$$\theta^* \equiv \frac{\theta - T_a}{S_o/U_o}$$

$$y^* \equiv y/H_T$$

$$t^* \equiv t/\omega$$

$$N_T \equiv \frac{\rho C_p V_T}{A_c U_o \omega}$$

$$P_e \equiv \frac{\rho C_p H_T^2}{k_f \omega}.$$

It is seen that the normalization of eqn (3a) results in two dimensionless groups which govern the performance of the storage tank: N_T and Pe . N_T is a number physically representing the ratio of the water quantity in the tank to the absorber surface area and has long been shown very important in describing the collector performance. Thus N_T can be called the "tank number". Pe is a Peclet number which illustrates the ratio of convective to conductive heat transfer along the fluid flow inside the

storage tank. Mathematically, the last term of eqn (3a) can be ignored with negligible errors if the value of Pe is larger (larger than 50). In general, the values of Pe in most designs are much larger than 50, therefore eqn (3a) can be simplified to

$$\frac{\partial \theta^*}{\partial t^*} + \frac{1}{N_T} q^* \frac{\partial \theta^*}{\partial y^*} = 0. \quad (4)$$

Momentum balance

The thermosyphon head induced in the collector can be evaluated by the integration of the density curve as shown in Fig. 1. The result can be written as

$$H_t = \int_0^{H_1} [SG(T_1) - SG(T)] dz + H_2 [SG(T_1) - SG(T_2)] + \int_{H_1+H_2}^H [SG(\theta) - SG(T_2)] dz, \quad (5)$$

where for water $SG(T) = -4.05 \times 10^{-6} T^2 - 3.906 \times 10^{-5} T + 1.0002556$, T in $^{\circ}\text{C}$. The above equation can be normalized to the form:

$$h_t^* = \int_0^{h_1^*} [SG(T^*) - SG(T^*)] dz^* + h_2^* [SG(T_1^*) - SG(T_2^*)] + \int_{h_1^*+h_2^*}^1 [SG(\theta^*) - SG(T_2^*)] dz^*, \quad (6)$$

where $h_t^* \equiv H_t/H$, $z^* \equiv z/H$, $h_1^* \equiv H_1/H$, $h_2^* \equiv H_2/H$.

The frictional loss due to fluid motion in the collector can be written as:

$$H_f = f_c \frac{L_c}{D_c} \frac{v_c^2}{2g} + f_p \frac{L_p}{D_p} \frac{v_p^2}{2g} + E_T \frac{v_p^2}{2g}, \quad (7)$$

where D_c is the hydraulic diameter of the parallel-plate absorber and equal to $2\delta W/(\delta + W)$, v_p is the velocity in the connecting pipe, W is the width of the absorber. f_c and f_p are frictional coefficients in the absorber plate and in the connecting pipe respectively, E_T is the loss factor due to valves, fittings and bends etc. Since the flow is assumed laminar in the collector, eqn (7) can be rewritten in the following normalized form after introducing the Fanning friction factor.

$$h_f^* = 3\nu^* q^*/N_f + q^{*2}/(2N_e), \quad (8)$$

where

$$h_f^* \equiv H_f/H$$

$$\nu^* \equiv \nu/\nu_o$$

$$1/N_f \equiv \frac{\nu_o}{gH} \left[\frac{L_c^2}{\delta^2} + \frac{8\pi}{3} \left(\frac{\delta L_p}{A_p} \right) \left(\frac{A_c}{A_p} \right) \right] \frac{U_o}{\rho C_p \delta}$$

$$1/N_e \equiv \frac{E_T}{gH} \left(\frac{A_c}{A_p} \right)^2 \left(\frac{U_o}{\rho C_p} \right)^2.$$

ν_o is the kinematic viscosity and usually taken at the ambient temperature. It is seen from eqn (8) that the

magnitude of N_f and N_e represent the flow resistance of the collector. Since the thermosyphon head is balanced by the frictional head, equating eqns (6) and (8), we obtain the momentum equation:

$$3\nu^* q^*/N_f + q^{*2}/2N_e = \int_0^{h_1^*} [SG(T_1^*) - SG(T^*)] dz^* + h_2^* [SG(T_1^*) - SG(T_2^*)] + \int_{h_1^*+h_2^*}^1 [SG(\theta^*) - SG(T_2^*)] dz^*. \quad (9)$$

In order to take into account the influence of temperature on the kinematic viscosity, the viscosity ν^* in eqn (9) is taken at the mean temperature of the absorber, $(T_1^* + T_2^*)/2$.

Descriptions of collector performance

The collector performance can be described in a number of ways. The solutions to the above equations could be used to calculate the specific instantaneous efficiency η which is defined as

$$\eta = \frac{mC_p(T_2 - T_1)}{A_c S} = \frac{q^*(T_2^* - T_1^*)}{N_r \sin \pi t^*}, \quad (10)$$

where m is the mass flowrate through the collector. It should be noted here that the specific collector efficiency is not the actual overall collector efficiency as defined in the presentation of the collector performance tests. Instead, the specific efficiency is defined based on the amount of solar incident radiation converted into the thermal energy at the black surface of the absorber. For convenience, we may define the specific mean efficiency as

$$\bar{\eta} = \frac{\int_0^\omega mC_p(T_2 - T_1) dt}{\int_0^\omega A_c S dt} = \frac{\int_0^1 q^*(T_2^* - T_1^*) dt^*}{\int_0^1 N_r \sin \pi t^* dt^*}, \quad (11)$$

and define the mean flowrate as

$$\bar{q}^* = \int_0^\omega q^* dt / \omega = \int_0^1 q^* dt^*. \quad (12)$$

The final mean tank temperature is defined as

$$\bar{\theta}_f^* = \frac{\bar{\theta}_f - T_a}{S_o/U_o}. \quad (13)$$

From the above analysis, it is seen that the collector performance can be described by the three energy and momentum equations, eqns (2), (4) and (9). In addition, another relation concerning the collector geometry also should be satisfied, i.e.

$$h_1^* + h_2^* + h_3^* = 1. \quad (14)$$

Thus the present analysis comes out with ten dimensionless groups or system characteristic parameters which

will uniquely determine the system performance of the collector: namely, N_c , F' , N_T , Pe , N_L , N_r , N_f , N_e , h_1^* and h_2^* . In other words, the specific mean collector efficiency or flowrate is uniquely a function of these ten parameters. This development forms the basis of the similarity theory of the solar collector with natural circulation. Through these parameters, the collector performance can be clearly understood and a general design criteria can be drawn.

NUMERICAL SOLUTION

Equations (2), (4) and (9) can only be solved numerically with the initial condition that the water temperature is uniform and assumed to be the same as the ambient temperature for simplicity. Therefore, the final mean tank temperature can be evaluated by the summation of the temperature rise due to the total amount of energy flowing into the tank and the initial ambient temperature, i.e.

$$\bar{\theta}_f = \frac{\int_0^\omega \rho C_p A_c q (T_2 - T_1) dt}{\rho C_p V_T} + T_a, \quad (15)$$

where T_a is assumed constant and equal to 20°C in the present analysis. Substituting eqn (15) and combining with eqn (13) we obtain

$$\bar{\theta}_f^* = (\pi/2) (N_r/N_T) \bar{\eta}. \quad (16)$$

Equation (16) shows that the final mean water temperature in the storage tank is linearly proportional to N_r/N_T and $\bar{\eta}$. The smaller the tank number, the higher the final mean temperature; and the higher the incident radiation the higher the final mean temperature.

Rewriting the governing equations, eqn (2), (4) and (9), into the finite difference form and applying the finite difference technique, the numerical solutions can be found when proper values of the ten parameters are given. Since the connecting pipes are assumed well-insulated, at any instant the inlet temperature of the storage tank will always equal to the exit temperature of the absorber plate, and the exit temperature of the storage tank will always equal to the inlet temperature of the absorber. Since the transient behavior of the heat transfer inside the connecting pipe is ignored, the temperatures at the exits of the absorber and the storage tank will immediately transmit to the inlets of the tank and the absorber respectively. This is the boundary conditions to be obeyed in numerical calculation. The numerical computations are carried out by dividing each of the absorber and the tank into sixteen segments and using the time step of 0.008. From the results obtained, it is shown that twelve segments are satisfactory for the present computation. A CDC 6000 Cyber 72 digital computer was used in the computational work. The calculation starts with the initial condition: $\theta^* = 0$ and $T^* = 0$ at $t^* = 0$, and stops at $t^* = 1$ or when $q^* = 0$.

Fig. 3. Effect of solar radiation on collector mean efficiency. ($F' = 0.98$, $N_L = 1.0$, $N_c = 0.20$, $N_T = 1.35$, $h_1^* = h_2^*$.)

Fig. 4. Variation of mean flowrate with solar radiation. ($F' = 0.98$, $N_L = 1.0$, $N_c = 0.20$, $N_T = 1.35$, $h_1^* = h_2^*$.)

Fig. 5. Effect of relative height of tank on mean efficiency. ($F^* = 0.98$, $N_r = 600/1353$, $N_c = 0.20$, $N_L = 1.0$, $h_1^* = h_2^*$.)

RESULTS AND DISCUSSIONS

For simplicity, the present analysis assumes that $h_1^* = h_2^*$ and Pe can be ignored since the values of Pe are usually larger than 50 in most designs (it has been numerically shown in the present computation that the results do not make any difference when Pe is larger than 50). Therefore there are only eight system characteristic parameters to be considered.

Figure 3 shows that the mean efficiency increases with increasing incident solar radiation and increasing relative height of the tank. This is consistent with Ong's experiments [5]. However, it is very interesting to note that, as the values of N_c and N_f become very high (greater than 10^5), i.e. the resistance to the fluid motion is very small, the efficiency tends to reach an equilibrium value and be independent of both the incident solar radiation and the relative height of the tank. Even at the case of $h_2^* = -1.0$, which means that the storage tank is sitting on the ground, the mean efficiencies are still independent of incident solar radiation and very close to each other (within 5 per cent deviation) for different relative heights of the tank. Therefore we can conclude that in order to reduce the influence of solar radiation due to the variations of seasons or weather conditions and obtain the highest efficiency, it is preferable to design a thermosyphon collector with N_c and N_f larger than 10^5 and under this circumstance we may also let the storage tank sit on the ground to reduce the construction cost without sacrificing efficiency. This saturation phenomenon on efficiency is mainly attributed to the balance between flowrate and heat loss. It is seen that higher incident radiation and higher relative height of the storage tank causes to high thermosyphon head as well as flowrate, as shown in Fig. 4; higher flowrate then induces higher moving speed of the temperature wave front in the

tank and consequently raises the inlet temperature of the absorber and relatively lowers the increase of efficiency. An equilibrium is finally reached and the efficiency approaches constant as shown in Fig. 3.

It is important to see how the relative height of the tank influences the collector performance. Figure 5 has clearly shown that the relative height of the storage tank has very little effect on the mean efficiency for collectors with N_c and N_f larger than 10^5 . Also, the influence on the mean efficiency is relatively small for smaller tank number. These results indicate that the relative height of the storage tank is mainly determined by the flow resistance of the collector. For ordinary size of parallel-plate absorber such as 3 ft $\times$ 6 ft $\times$ 1/10 in. in which the frictional loss is very low, N_c and N_f are usually larger than 10^5 , therefore we may lower the height of the storage tank without sacrificing efficiency.

Since the tank number reflects the amount of water to be heated and stored in the tank with respect to the absorber surface area, it controls the water temperature and the efficiency. High tank number would result in lower water temperature as well as heat loss and therefore increases the efficiency as shown in Fig. 6. The rate of increase of efficiency with respect to the tank number is higher for higher values of N_c and N_f . From Fig. 7 it also can be seen that the lower heat loss of the collector will increase the efficiency.

It is known that the amount of water contained in the absorber has some effect on the performance as shown in Fig. 8. The efficiency is found to decrease with increasing N_c because higher N_c means that more non-applicable hot water is kept in the absorber.

As discussed previously, the flow resistance of the collector affects the flowrate as well as the efficiency.

Fig. 6. Effect of tank size on mean efficiency. ($F' = 0.98$, $N_r = 600/1353$, $N_c = 0.20$, $N_L = 1.0$, $h_1^* = h_2^*$)

Fig. 7. Variation of mean efficiency with collector heat loss.

When both N_e and N_f are larger than 10^5 , the resulting high flowrate will cause to a relative increase of the inlet temperature at the absorber due to fast moving of the temperature front in the tank. An equilibrium condition will be eventually reached as shown clearly in Fig. 9, and the efficiency will no longer increase with N_e and N_f . Figure 10 describes the instantaneous motion of the temperature wave front in the tank. For the case of low flow resistance (high N_e and N_f), the temperature wave front moves very fast. But at the end of the performance near sunset hours, the temperature distribution in the tank tends to form a peak near the top of the tank, especially for the case of low flowrate. This temperature peak would destroy the condition of thermal stratification in the tank as assumed previously and mixing process would take place at the top of the tank. Under this condition, the thermosyphon head would change and the performance prediction would result in some errors. However, as this phenomenon usually occurs at the end of the performance, the resulting error would not be too serious. The occurrence of the mixing phenomenon also can

Fig. 8. Variation of mean efficiency with absorber size (N_c).

be seen from Fig. 11. In the present analysis the mixing effect is ignored, so the computation is terminated at the time when the negative thermosyphon head resulting from the peak temperature in the tank is equal to the positive thermosyphon head as shown in Fig. 11, i.e. when $q^* = 0$. More work should be carried out to investigate the effect of mixing.

CONCLUSIONS

The similarity theory of solar water heater with natural circulation is developed in the present paper. Ten dimensionless groups or system characteristic parameters which uniquely determine the system performance are derived. The numerical solutions to the governing equations has shown that the mean efficiency

Fig. 9. Effect of flow resistance on mean efficiency. ($F' = 0.98$, $N_c = 0.20$, $N_r = 600/1353$, $N_L = 1.0$.)

Fig. 10. Motion of temperature wave front in storage tank. ($F' = 0.98$, $N_c = 0.20$, $N_L = 1.0$, $N_r = 600/1353$, $h_2^* = 0.2$.) (a) $N_e = N_f = 20,000$, (b) $N_e = N_f = 1500$.

increases with increasing incident solar radiation and relative height of the tank. However, for high N_e and N_f (larger than 10^5 which are of the values for ordinary parallel-plate absorber), the efficiency appears to be independent of the incident radiation and the relative height of the tank. This points out a very important design criteria which differs from other investigators' in that, the tank may be designed to sit on the floor without sacrificing the efficiency for parallel-plate absorber since its N_e and N_f are usually larger than 10^5 . This also indicates that the resistance to flow is the most important factor which should be carefully considered in the design of a solar collector with thermosyphon cycle. Low flow resistance would largely favor the collector performance in both the mean efficiency and the structural design of

the relative height of the tank. The analysis also shows that higher tank number and lower N_c and N_L increase the mean efficiency. From the motion of the temperature wave front in the tank, we also note that the mixing process near the top of the tank will take place and affect the performance as the flowrate is low. Since the present analysis ignores the effects of water heating load (i.e. hot water drawn-off) during daytime, mixing in the tank, heat loss from the tank and piping and transient behavior in the pipes, etc. some analytical errors are inevitable and further theoretical and experimental investigations are required.

Acknowledgement—The author is grateful to Mr. H. T. Hsu for his assistance in the numerical calculation. He also thanks Aitech Engineering Co. for providing financial support.

Fig. 11. Instantaneous temperature distributions of collector system. ($F' = 0.98$, $N_c = 0.20$, $N_L = 0.6$, $N_r = 600/1353$, $N_T = 1.20$.) (a) $N_e = N_f = 20,000$, (b) $N_e = N_f = 5000$.

NOMENCLATURE

A_c absorber surface area, m^2
 A_p cross-sectional area of connecting pipe, m^2
 B_i Biot number ($= h_i e / k_c$)
 C_p heat capacity of fluid, $cal/g \cdot C$
 θ temperature distribution in tank, C
 θ_f final mean tank temperature, C
 θ_f^* dimensionless final mean temperature
 D_p diameter of connecting pipe, m
 δ spacing of parallel-plate absorber, m
 E_T loss factor due to valves, fittings, and bends etc.
 η instantaneous efficiency
 $\bar{\eta}$ mean efficiency
 e thickness of absorber plate material, m
 F' absorber geometry efficiency factor
 f_c Fanning frictional factor in connecting pipe
 f_p Fanning frictional factor in absorber
 G_m converted solar energy at solar noon, kW/m^2
 H_T height of water in tank, m
 H_i thermosyphon head, m
 H_1 absorber height, m
 H_2 relative height of tank above absorber, m

h_f^* dimensionless water height in tank
 h_i heat convection coefficient inside the absorber, $kW/m^2 C$
 h_f^* dimensionless frictional head
 h_i^* dimensionless thermosyphon head
 h_1^* dimensionless height of absorber
 h_2^* dimensionless relative height of tank above absorber
 k_c thermal conductivity of absorber plate material, $kW/m^2 C/m$
 k_f thermal conductivity of fluid, $kW/m^2 C/m$
 L_c absorber length, m
 L_p connecting pipe length, m
 m total mass flowrate, kg/hr
 $N_c = \rho C_p \delta / U_o \omega$ (absorber number)
 N_e loss number
 N_f friction number
 N_L heat loss number
 N_r insolation number ($= G_m / S_o$)
 N_T tank number ($= \rho C_p V_T / A_c U_o \omega$)
 ν_o reference kinematic viscosity, m^2/hr
 ν^* dimensionless kinematic viscosity
 ω sunshine duration, hr

- Pe Peclet number in tank ($= \rho C_p H_T^2 / k^w$)
 q volumetric flowrate per unit area of absorber, kg/hr- m^2
 q^* dimensionless volumetric flowrate
 S solar energy actually converted into heat energy at the absorber surface, kW/ m^2
 SG specific gravity of fluid
 S_o solar constant, 1.353 kW/ m^2
 T local fluid bulk temperature in absorber, C
 T_a ambient temperature, C
 t time, hr
 U_L heat loss coefficient, kW/ m^2 C
 U_o reference heat loss coefficient, kW/ m^2 C
 V_T total water volume in tank, m^3
 v_c fluid velocity in absorber, m/hr
 x, y, z coordinates, m
 x^*, y^*, z^* dimensionless coordinates

2. D. J. Close, The performance of solar water heaters with natural circulation. *Solar Energy* 6, 33 (1962).
3. C. L. Gupta and H. P. Garg, System design in solar water heater with natural circulation. *Solar Energy* 12, 163 (1968).
4. K. S. Ong, A finite-difference method to evaluate the thermal performance of a solar water heater. *Solar Energy* 16, 137 (1974).
5. K. S. Ong, An improved computer program for the thermal performance of a solar water heater. *Solar Energy* 18, 183 (1976).
6. G. Grossman, A. Shitzer and Y. Zvirin, Heat transfer analysis of a flat-plate solar energy collector. *Solar Energy* 19, 493 (1977).
7. A. Cabelli, Storage tanks—a numerical experiment. *Solar Energy* 19, 45 (1977).
8. Z. Lavan and J. Thompson, Experimental study of thermally stratified hot water storage tanks. *Solar Energy* 19, 519 (1977).

REFERENCES

1. J. C. McVeigh, *Sun Power*, Chap. 9. Pergamon Press, Oxford (1977).