

Performance of a Water-in-Glass Evacuated Tube Solar Water Heater

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Abstract

There has been wide scale adoption of evacuated tubular solar water heaters in China and Europe. The performance of the water-in-glass evacuated tube solar water heater that has been commercialised in China has been evaluated using the ISO9459-2 test procedure. Factors limiting the performance of this type of product have been evaluated. A numerical model of the heat transfer and fluid flow inside a water-in-glass tube has been developed to investigate the heat-extraction efficiency and the flow structure inside the tube. Initial investigations have shown that the natural convection flow-rate in the tube is high and that the storage tank is forced into a fully mixed state by the high flow rate produced by the 21 tube solar collector.

1. INTRODUCTION

The world market for solar water heaters has expanded significantly in the last decade. As a result, there have been large-scale developments of new-technology and improved-quality products. The evacuated tube solar collectors perform better in comparison to flat plate solar collectors, in particular for high temperature operations. However, previously, it provided no real competition for flat plate solar collectors, because of difficulties in manufacturing and maintenance of the metal-to-glass vacuum seal. One of the most significant developments is the use of double-glass evacuated tubular solar water heaters, which now comprise 65% of 6 million m² / year solar collector market in China. The mechanism of this type of solar water heater is driven by natural circulation of the fluid in the collector and the storage tank. It consists of all-glass vacuum tubes, inserted directly into a storage tank, with water in direct contact with the absorber surface. The limitation of this concept is that it can only be used for a low-pressure system, as the tubes can only withstand a few metres of water head.

Morrison et al. (2001) presented daily test results of a commercial type of water-in-glass system and potential difficulties in heat extraction from the long and thin absorber tubes. In this paper, the performance of a commercial water-in-glass solar water heater was evaluated. The experimental results from the individual components were combined in a computer simulation program, TRNSYS, to obtain long-term predictions of system performance under a wide range of meteorological conditions, loads, load profiles and operating set points. Numerical simulations have been performed to investigate the flow structure and the mass transfer in a system of a tube connected into a section of a reservoir.

2. PERFORMANCE EVALUATION OF ALL-GLASS EVACUATED TUBE SOLAR WATER HEATING SYSTEM



Figure 1. Water-in-glass evacuated tube solar water heater under test at Solar Thermal Energy Laboratory UNSW

2.1 Daily tests

The performance of the all-glass evacuated tube solar water heater that has been commercialised in China (Figure 1) has been evaluated using the ISO9459-2 test procedure. A series of daily outdoor tests have been performed and the results were presented earlier (Morrison et al., 2001). In the test, the tank was used as a calorimeter to measure the net energy gain for the whole day without draw-off. The useful energy accumulated in the tank during the day was then correlated with the daily irradiation and a temperature difference factor.

2.2 Tank heat-loss measurement

A cool-down test was done to determine the heat loss from the storage tank (Figure 2). The water was pre-heated to 80 °C by using an electric-heater element. At the start of the test, the

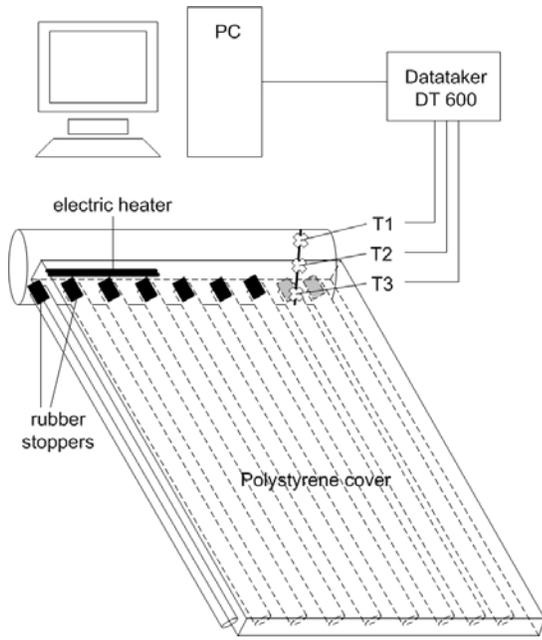


Figure 2. Experimental setup: Tank heat loss

heater was turned off and the temperature decrease over a few days was recorded. The water temperature is determined by averaging the readings of three thermocouples placed at different positions inside the tank. The ambient temperature was also monitored. To determine the heat loss coefficient from the tank only, rubber stoppers were used to seal the inlets to the absorber tubes. The collector was covered with a 100-mm-thick polystyrene to prevent the collector from transferring heat into the tank. The heat loss coefficient for the tank was then calculated over one-day test periods based on the following correlation (ISO9459-2, 1995).

$$U_s^* = \frac{\rho_w c_{pw} V_s}{\Delta t} \ln \left[\frac{t_i - t_{as(av)}}{t_f - t_{as(av)}} \right] \quad (1)$$

where ρ_w is water density at that particular temperature range (kg / m^3), c_{pw} is the specific heat of water ($\text{J} / \text{kg K}$), V_s is the volume of the storage tank (m^3), Δt is the period of averaging (s), t_i and t_f are water temperature at the start and at the end of the test respectively, $t_{as(av)}$ is the average ambient temperature over the period of measurement. The average heat loss coefficient of the tank is 1.5 W/K, with $\pm 5\%$ variation due to fluctuation in ambient temperature and wind velocity (Table 1).

Another heat loss experiment was undertaken for a complete system, where the collector is in connection with the storage tank. Air circulation over the collector was allowed by leaving a 20 mm gap between the collector and the polystyrene cover. To monitor water temperature inside the tubes, a fourth thermocouple was placed inside one of the tubes, 200 mm from the point where the tube is connected to the tank. As the test was performed outdoor, the ambient conditions varied significantly. As a result, the heat loss coefficient values obtained from this configuration scatter in the range of 1.6 – 1.8 W/K. These results indicated that there is some reverse circulation into the tubes at night, and hence an increase in tank heat loss.

Table 1. Heat loss coefficient of the storage tank

Test no.	Duration (hours)	T _a range (°C)	T _a ave (°C)	U _s * (W/K)
1	11	21.0 - 23.0	21.95	1.50
2	26	18.7 - 26.6	21.6	1.56
3	26	19.1 - 31.0	23.58	1.41
4	27	21.1 - 31.0	24.14	1.49
5	23	20.5 - 25.7	22.28	1.55

2.3 Tube heat-loss measurement

The measurement of the tube heat-loss was undertaken in a temperature-controlled room without direct solar radiation. Figure 3 shows the experimental set-up. The tube is held vertical, its open end insulated with a 100-mm thick polystyrene cover. The getter (sealed end of the tube) has a higher heat loss value, and is also insulated with a polystyrene cover. Five T-type thermocouples were placed at different axial positions along the tube. The tube was filled with hot water, and was monitored as it cooled down. The starting hot water temperature is 85-90 °C (corresponds to a temperature difference with the ambient of ~ 65 °C). The experiments were done on 8 tubes from two production batches, and two of the tubes were tested twice to ensure repeatability of the measurement. The heat loss coefficient of the absorber tube was calculated based on the following correlation:

$$U_t = \frac{\dot{m} c_p (t_i - t_f)}{A_a (t_m - t_a)_{av}} \quad (2)$$

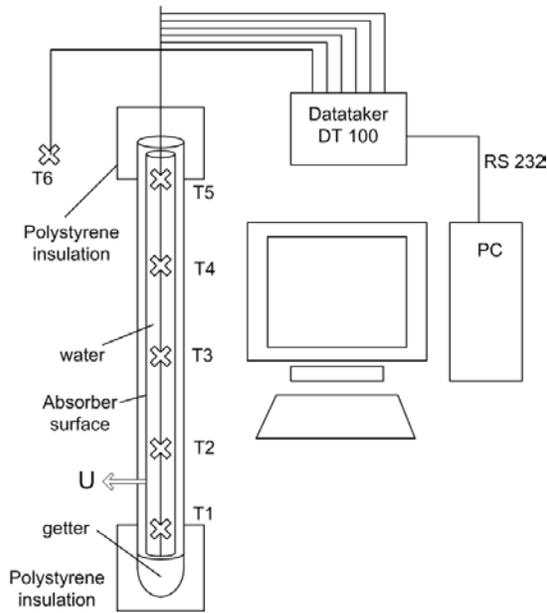


Figure 3. Indoor experimental setup: Tube heat loss measurement



Figure 4. Experimental setup: Optical efficiency measurement

where \dot{m} is the mass of water inside the tube (1.28 kg) divided by the period of averaging (3600s), c_p is the specific heat of water (J / kg K), t_i and t_f are the average water temperature inside the tube, obtained by averaging 5 thermocouple readings, at the start and at the end of the test, A_a is the surface area of the absorber and $(t_m - t_a)_{av}$ is the average difference between the mean water temperature and the ambient temperature over the one hour period. The test results were plotted in Figure 5. The coefficient varies slightly with temperature difference. It also shows that there are variations of tube quality. For good quality tubes, the heat loss coefficient varies from 0.5 W/m²K (at $T_m - T_a = 25$ K) to 0.65 W/m²K (at $T_m - T_a = 65$ K). For lower quality tubes, the coefficient varies from 0.7 W/m²K to 0.9 W/m²K for the same temperature range.

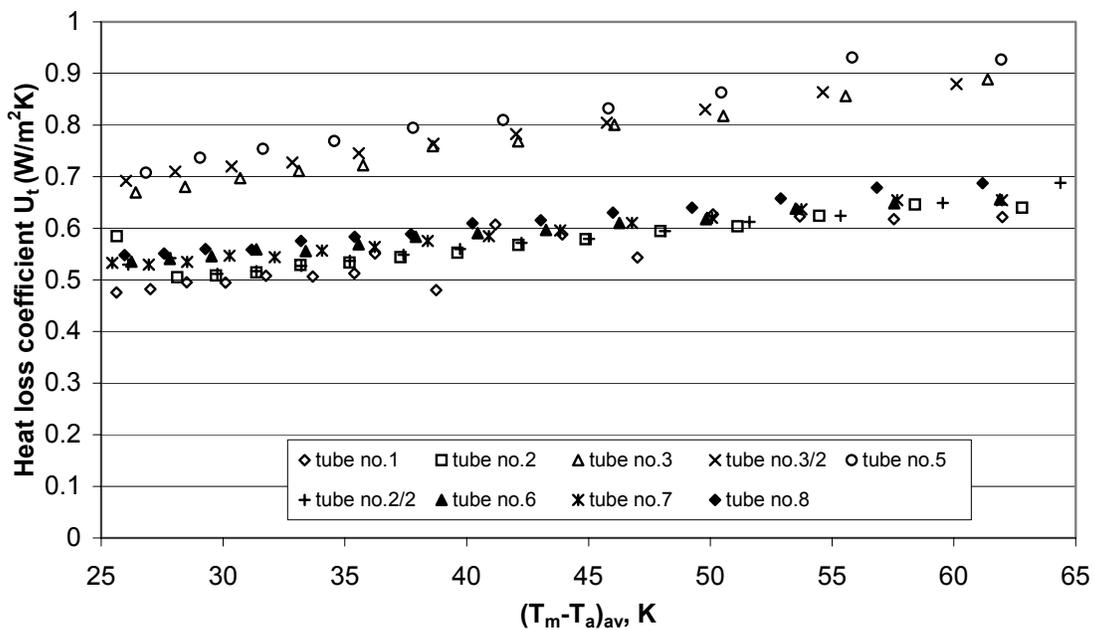


Figure 5. Heat loss coefficient of the tubes as a function of temperature difference

2.4 Collector optical efficiency test

The optical efficiency of the collector array has been determined experimentally. The test was done using a panel of ten water-in-glass evacuated tubes connected by a manifold and positioned on a diffuse reflector (Figure 4). The absorber diameter was 37 mm and the tube centre-line spacing was 70 mm. The panel was mounted on a sun-tracking rig. Cold water from the mains is pumped into the manifold at a constant flow-rate of 2 litres / minute, the inlet and outlet water temperatures are monitored. Tilting the collector east-west will effect the thermosyphon flow structure and flow-rate inside the tube and hence the measurement was limited to a short period across solar noon. An optical efficiency of 0.78 was obtained by averaging the values over a period of one hour across solar noon.

3. LONG-TERM PERFORMANCE PREDICTION USING TRNSYS SIMULATION

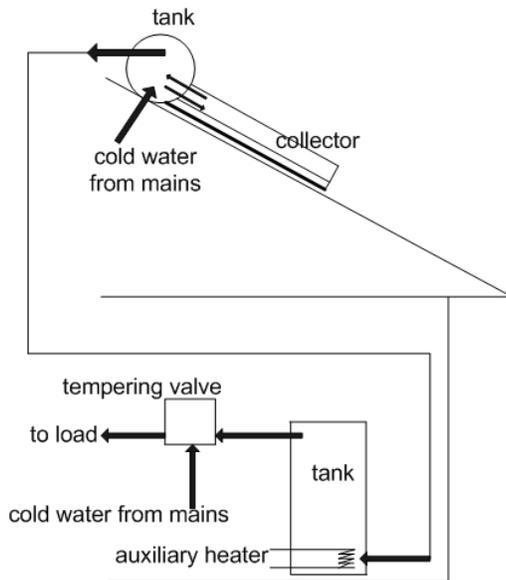


Figure 6. Schematic of pre-heater system

The performance of the above solar water heater was simulated in the TRNSYS modelling program for various applications and locations around the world. The result presented in this paper is for the water-in-glass system used as a solar pre-heater located in Sydney (Figure 6). The system is installed facing north on a 20° slope (typical roof inclination in Sydney). The collector is connected to a 150-litre horizontal tank, which stores the pre-heated water as an input to a larger 250-litre vertical tank with an internal electric heater.

The natural circulation rate of the fluid inside the evacuated tubes is a function of solar irradiation, water temperature and the degree of stratification inside the horizontal tank. Both numerical simulation and flow measurement are currently in progress to determine the mass-transfer correlation. Preliminary results suggest that the flow rate is linearly proportional to the amount of absorbed radiation, and is approximately 20 kg/h per tube at 1000 W/m². The variation of collector array efficiency as a function of incidence angle is taken into account by using bi-axial incidence angle modifier. The values for the East-West modifiers were taken from the previous work by Chow et al. (1984) on a similar collector configuration. A cosine function was assigned for the North-South modifiers (Figure 7). The heat loss coefficient of the collector tubes was found to be temperature-dependent, hence a second-order collector efficiency mode is used. Each tank is divided into 10 fully-mixed equal-volume segments. A tempering valve is used to mix the heated fluid from the tank with colder fluid so that the load flow is no hotter than necessary.

The rating factor that is commonly used to describe the monthly performance of the system is the solar contribution relative to a conventional energy source, f_R , which is how much the user's auxiliary energy consumption will be reduced by changing to a solar water heating system. The solar contribution factor for AS4234 (1994) load conditions is presented in Figure 8 for the assumed tube-to-tank mass-transfer correlation. Varying the coefficient in the flow-rate correlation by $\pm 50\%$ has a relatively small effect on the amount of energy savings.

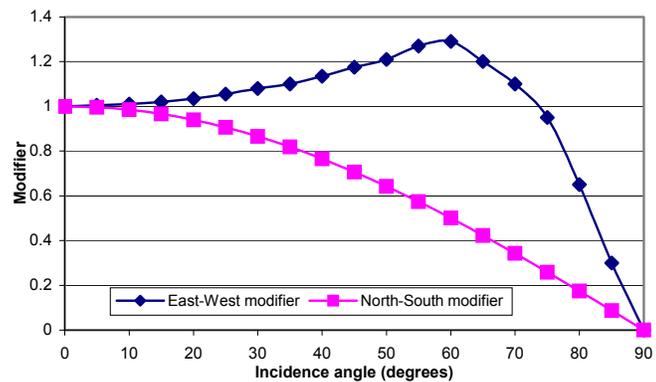


Figure 7. Bi-axial incidence angle modifier

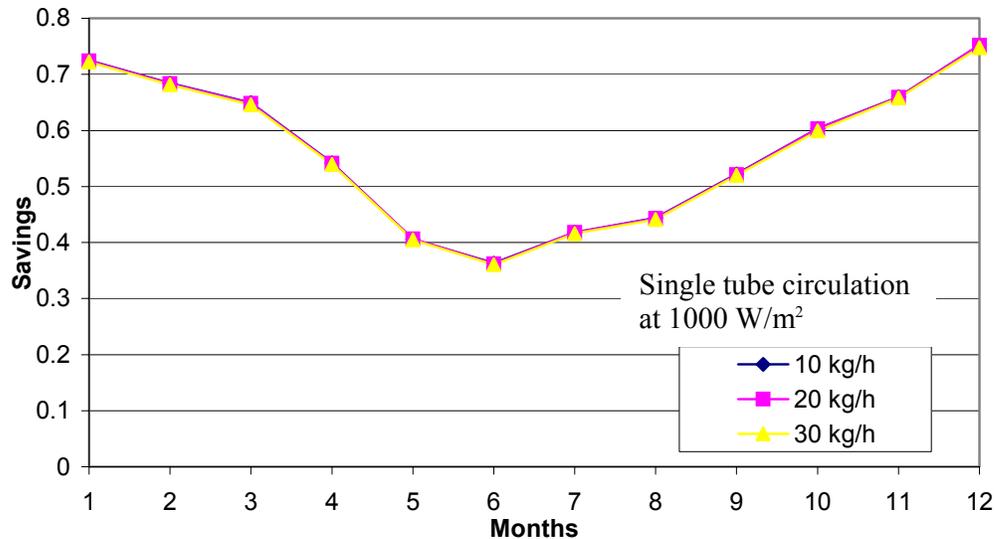


Figure 8. Monthly energy savings in Sydney for AS4234 load conditions; variations with collector mass flow-rate

4. INTERACTIONS BETWEEN THE TUBES AND THE TANK

The flow structure in an open thermosyphon has been investigated previously, identifying a problem in heat extraction for a tube under constant temperature wall conditions (Morrison et al., 2001). To investigate this, the mass transfer of the fluid into the tubes and the degree of stratification inside the storage tank are evaluated using a computational fluid dynamics package FLUENT. The thermosyphon flow-rate depends on the temperature of the fluid inside the tank, the degree of stratification as well as the inlet geometry. Simulating the tube alone assuming constant pressure at the inlet neglects the coupling between the tubes and the tank, and as a result, might lead to discrepancies in the predicted mass flow rate. However, due to limitation in the computing power, it is not possible to simulate the complete system. The numerical model consists of a thermosyphon with a length-to-radius ratio of 20, subjected to a uniform heat flux (100 W/m^2) round its sidewall, connected to a section of cylindrical tank (100 mm wide, 360 mm diameter). Rectangular mesh is used for the tube and most part of the tank, with hybrid mesh employed in the transition region between the two. Three sets of boundary conditions have been investigated. In the first simulation, the tank was made flow-through, having an inlet at the bottom and an outlet at the top (Figure 9a). The effect of varying flow-rate, and consequently varying the degree of mixing inside the tank, on the mass transfer from the thermosyphon was observed. In the second simulation, the tank was assumed to have a uniform heat loss round its sidewall (Figure 9b). The amount of heat loss was calculated based on the heat flux on the tube and the ratio of the surface area of the tube and the tank. The third simulation was done similarly to the previous one, except that the heat loss from the tank is distributed from 0 at the bottom, to 55 W/m^2 at the top. Comparisons of the mass transfer and flow structure were then made with those from the open thermosyphon model of the same dimensions (Figure 9c).

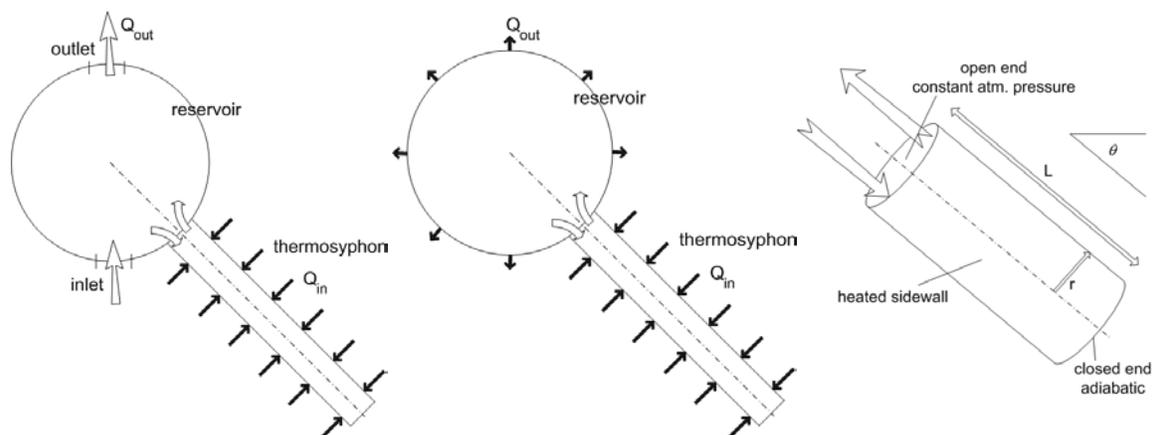


Figure 9a. Tube coupled to a flow-through reservoir b. Tube coupled to a closed tank c. Open thermosyphon

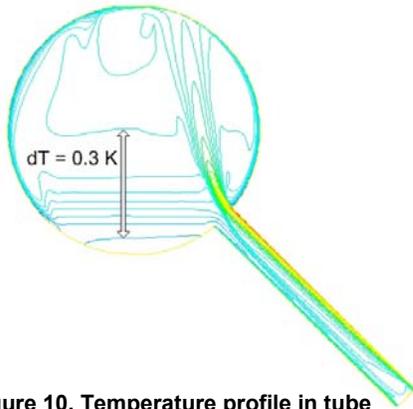


Figure 10. Temperature profile in tube and tank showing uniform temperature tank

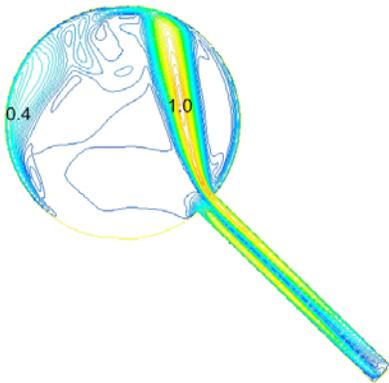


Figure 11. Velocity contour showing a hot stream rising towards the top of the tank

Table 2. Thermosyphon flow-rate for different numerical models

Configurations	mass flow-rate (kg/hr)	% difference
flow-through reservoir:		
a. 0.9 V-change / hour	5.61	0.9
b. 1.8 V-change / hour	5.65	1.62
closed reservoir with heat loss:		
a. uniform heat loss	5.45	-1.98
b. variable heat loss	5.56	-
thermosyphon only:	4.65	-16.37

Figure 10 and Figure 11 show the temperature and velocity profiles at the symmetry plane respectively. Table 2 shows the mass flow-rate of the fluid exiting the thermosyphon for different configurations. The effect of varying tank configurations and heat loss conditions on the mass transfer are relatively small. However, simplifying the simulation by replacing the inlet condition by a constant pressure open boundary results in a variation of up to 16% from the actual condition. The assumption of a constant pressure across the inlet of the tube is one reason for the discrepancy. Also with an open boundary, the inlet and outlet flows are assumed to be normal to the boundary. In reality, particularly for a small flow-rate, the hot fluid plume is almost vertical, which means that there is a radial velocity component of the outgoing flow. It is currently investigated how the assumed open boundary affects the mass transfer in the tube-and-tank system, if the tube is 1420 mm long.

5. CONCLUDING REMARKS

The performance of water-in-glass evacuated tube solar water heater that has been commercialised in China has been evaluated using the ISO 9459-2 test procedure. The long-term performance of the system used as a pre-heater in Sydney has been simulated using TRNSYS 15. The results suggest that for the AS4234 (1994) load conditions, energy savings of up to 36 % can be achieved during the winter period and 55 % over a year. A numerical model of the heat transfer and fluid flow has been developed to investigate the interaction between the fluid inside the tube and the reservoir. Initial investigations have shown that the natural circulation in the absorber tubes is high and the storage tank is forced into a fully mixed state.

6. ACKNOWLEDGMENTS

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7. REFERENCES

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