EXPERIMENTAL INVESTIGATION OF HEAT REMOVAL FACTOR IN SOLAR FLAT PLATE COLLECTOR FOR VARIOUS FLOW CONFIGURATIONS

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Abstract – This paper reports the performance of a solar flat plate collector for various flow configurations and presents a comparative evaluation. Three cases have been discussed: (i) Serpentine Flow, (ii) Parallel-Wall-to-wall flow and (iii) Spiral Flow. The heat removal factor (F_R) is the ratio of the actual heat transfer to the maximum possible heat transfer through the collector plate. A special case showing the effect of phase change material (PCM) on F_R in the case of serpentine is discussed.

A sheet-and-tube type collector where the tubes are bonded to the absorber plate by a suitable adhesive is considered. Equations showing the dependence of F_R on fin factor were studied and indoor experiments were conducted to determine the value of F_R. Finally a numerical investigation was done and comparisons were evaluated.

Results indicate that when the mass flow rate ($\dot{m}$) of water is 1, the value of F_R for serpentine and parallel wall-to-wall flow is 0.44 and 0.88 respectively. It was also observed that for the case of PCM, F_R is a variable quantity and its value ranges from 0 to 0.28. In this case, F_R is a function of the PCM melting and solidifying phases rather than mass flow and is proportional to the melting temperature of the PCM. Also, it is observed that in a conventional water heater with serpentine flow or parallel wall-to-wall flow, the value of F_R does not increase significantly beyond a mass flow rate ($\dot{m}$) of 4 kg/hr.

Keywords: Solar flat plate collector, Heat removal factor, Flow, Phase change material, fin factor.

1. INTRODUCTION

Solar water heating is one of the most attractive solar thermal applications from an economic standpoint. Solar water heaters are the devices that convert solar radiation into heat energy. A typical system consists of a solar flat plate collector and a storage tank. Flat plate collectors are usually permanently fixed in position and require no tracking of the sun. The tank is located above the collector and in passive water heater system the water circulates by natural convection or thermo siphoning, whereas in a forced circulation system, a pump is required (Sukhatme, 2004).

The useful energy gained by the collector per unit time can be calculated by the equation

$$Q_{use} = A_{col}\left[S - U_{los}\left(T_{p,m} - T_{amb}\right)\right].$$

However, the problem with this equation is the quantity $T_{P,m}$ which is neither easily known, nor can be found with ease. In order to solve this problem, the heat removal factor (F_R) was introduced such that the values of $T_{P,m}$ can be related to the fluid inlet temperature, which is a known quantity (Duffie and Beckman, 1980). The maximum possible energy gain occurs when the whole collector is at the fluid inlet temperature. This maximum possible heat gain when multiplied with F_R gives the actual useful energy gain $Q_{use} = A_{col}.F_R[S - U_{los}(T_{in} - T_{amb})]$. This is a very convenient expression known as the Hottel and Whillier equation for calculating Q_{use} because the fluid inlet temperature is usually known.

1.1 Heat removal factor (F_R)

All variables influencing the efficiency of a flat-plate solar heat collector as a heat exchanger can be combined into a single "efficiency factor." These efficiency factors are more or less design constants of the particular collector design, and are only slightly influenced by operating conditions (Bliss, 1959). This efficiency factor is known as the heat removal factor which can be defined as the quantity that relates the actual useful energy gain of a collector to the useful energy gain if the whole collector surface were at the fluid inlet temperature (Duffie and Beckman, 1980). F_R can be written, as;

$$F_R = \frac{\dot{m}.c_p}{A_{col}.U_{los}}\left[1 - \exp\left(-\frac{A_{col}.U_{los}.F'}{m.c_p}\right)\right] \qquad (1)$$

F_R is an important design parameter as it is a measure of thermal resistance encountered by the absorbed solar radiation in reaching the collector fluid (Sukhatme, 2004). This factor also depends on the collector efficiency factor (F') which further depends on the fin efficiency (F), the function of geometry and area of the absorber and the type of flow. The flow arrangement can be parallel, spiral or serpentine (Malvi, 2012).

1.2 Phase Change Material (PCM)

A phase-change material (PCM) is a substance with a high heat of fusion which, when melting and solidifying at a certain temperature, is capable of storing and releasing large amounts of energy (Sharma, 2008; Kishore et al., 1984). Heat is absorbed or released when the material changes its phase from solid to liquid and vice versa. Thus, PCMs are classified as latent heat storage (LHS) units. The thermal energy transfer occurs when a material changes from a solid to a liquid or from a liquid to a solid, this is called a change in state, or "phase". It absorbs a large amount of heat in the process of changing from solid to liquid phase and this results in cooling of the surroundings. The PCM functions both as an energy storage material for the stabilization, theoretically, of the water temperature and as an insulation material owing to its low thermal conductivity value (Malvi et al., 2011), (Kürklü et al., 2002).

1.3 Objective

The objective of this investigation is to analyze the performance of solar flat plate collectors for various flow configurations and to show the effect of PCM on F_R. For this purpose, indoor experiments were conducted and numerical investigations were done. This paper also presents a qualitative comparison between the performances of collectors with different flow configurations for the same mass flow rate.

2. METHODOLOGY

Indoor experiments and flow calculations corresponding to all the cases are conducted separately for the purpose of finding out the value of F_R. A sheet-and-tube type collector where the tubes are bonded to the absorber plate by a suitable adhesive is considered. Equations showing dependence of F_R on fin factor were studied and experiments were conducted for determining the value of F_R. Finally, a numerical investigation was done and comparisons were evaluated.

2.1 Case (i) Serpentine Flow: In serpentine flow, the fluid flows in a zigzag or a serpentine fashion, and is constructed from a single, integral length of tubing. Many formulations and solutions have been presented for serpentine-flow (Kurt O'Ferrall, 1989; Abdel-Khalik, 1976; Mehmet, 1988; Zhang and Lavan, 1985). Kurt O'Ferrall developed an effectiveness-NTU (number of transfer units) relationship to directly compare with parallel-flow. The results indicate superior performance for the serpentine collector for the same number of heat transfer units. All these studies are 20 years old and these methods have been accepted to calculate the heat removal factor and evaluate a general thermal analysis of serpentine-flow. However, the serpentine flow equations have been modified several times and the latest modified equation (Mehmet, 1988) is used for further heat removal factor calculations (here F_1 is used instead of F' to avoid confusion with F_2. The photograph of the tested and fabricated serpentine type collector is shown in Figure 3 a.

$$F_R = \frac{\dot{m}.c_p}{A_{col}.U_{los}}\left[1 - exp\left(\frac{A_{col}.U_{los}.F_1(F_2-1)}{\dot{m}.c_p}\right)\right] \tag{2}$$

Where;

$$F_1 = \left(\frac{N.K.L}{F_2.A_{col}.U_{los}}\right)\frac{1}{[K.R\,(1+y)-1]^2 - (K.R)^2},$$

$$F_2 = \frac{1}{K.R\,(1+y)^2 - 1 - y - K.R},$$

$$K = \frac{k.\delta.n}{(W-D)Sinhn},$$

$$n = (W-D)\sqrt{\left[\frac{U_{los}}{k.\delta}\right]}$$

Where n is a non-dimensional quantity,

$$y = -2\cosh n - \left(\frac{D.U_{los}}{K}\right),$$

$$R = \frac{1}{\pi.h_{fin}.D_{in}}.$$

2.2 Case (ii) Parallel Wall-to-Wall Flow: In this experiment, parallel wall- to-wall flow was observed and studied. These experiments were performed on channels having rectangular cross section (Schematic view in figure 1). Since these channels have no sharp turns, they provide a smooth passage to the flow of water. There are apparently no pipe losses because of no bends in the passage of flow and hence they function as heat removal conduits with the *fin efficiency* '$1'$(Hottel and Whillier, 1958). The *collector efficiency factor*(F') can be defined as (Sandnes and Rekstad, 2002). F' = h_{fin} / (h_{fin} + U_{los}); where h_{fin} is heat transfer coefficient and U_{los} is overall heat loss coefficient. A three-wall polycarbonate sheet was chosen with a channel width of 15 mm, however, the market also offers twin-wall but that only provides a 4 to 10 mm channel width which was found to be too narrow to allow adequate water flow (Malvi, 2012). These channels were filled with polystyrene insulating material at both the ends to protect the sides' heat loss.

2.3 Case (iii)-Spiral Flow: Mehmet developed a spiral tube model in which fluid flow was toward the center of the plate and it was assumed that heat conduction in the plate was away from the center everywhere except in the region between the plate boundary and the outermost segment of the spiral (Akgün, 1988). As the liquid moves progressively towards the centre, its temperature gradually rises and in the process it absorbs a part of the heat transferred radially from the centre of the collector, thus reducing heat losses (Pillai and Agarwal, 1980).

Spiral flow in tube-in-tube was investigated under this research. Tube-in-tube was developed for which a black polymer tube was filled with PCM and water tubes were passed into it. This was preferred because the serpentine shape has sharper turnings and caused the polymer tubes to burst.

2.4 Case (iv) - Heat Removal Factor with PCM Effect in Serpentine Flow: Figure 2 shows a schematic cross-sectional view of the PCM filled flat plate collector. A polymer tube was chosen for water flow because it was flexible enough for serpentine-flow (Kurt O'Ferrall, 1989). The distance between two adjacent tubes from center to center was kept at 60 mm. The serpentines ends remained extended for inlet and outlet water supply at one side. For filling with PCM, six galvanized iron tubes of 12 mm diameter and 450 mm length were cut and water tubes were passed through the iron tubes in a manner that only the turning 'U' shape remains outside (Figure 3 (b)).The close-folding ends were sealed by rubber and the iron tubes with silicone. The PCM was filled from the other end which was later sealed in the same way (Malvi, 2012). Step by step fabrication of a PCM filled flat plate collector has been shown in Figure 3.

For the purpose of this experiment, OM35 PCM was used which has a melting temperature of 35 ^{o}C and constituted of the right mix of various salts, additives and nucleating agents allowing equilibrium between solid and liquid phases to be attained at the melting point (Pluss Polymers, India). The effect of the PCM in case of serpentine flow was investigated experimentally as shown in Figure 7.

3. Results and discussions

3.1 Case (i) Serpentine Flow: From equation (2) and Table (1), collector efficiency factor F_1 is found to be 4.66, fin efficiency F_2 i.e. a factor depending upon the geometry of collector and type of flow is 0.874, and heat removal factor F_R for this arrangement is 0.44. This low value of F_R is the result of sharp turns in the serpentine tube and corresponding losses. By increasing the number of serpentine turns the value of F_R can be increased. Comparison with other configurations is presented in section 4.

3.2 Case (ii) Parallel Wall-to-Wall Flow: Referring From Table (1), the collector efficiency factor F_1 is 0.99, fin efficiency F_2 is '1' and from equation (1), the value of F_R is found to be 0.88. This shows that parallel wall-to-wall flow can be a better alternative to the traditional design of solar collector and should be implemented after further scientific investigations.

3.3 Case (iii) Spiral Flow: During the experiments in the lab, the spiral shape did not work well owing to the bursting and leakage problems. Therefore a metallic tube can be used for this purpose, however corrosion on inner linings of tubes can deteriorate the performance of the collector and hence proper polishing of the tube is highly recommended. Also, because of the sharp turns in the spiral tube, the pipe losses prevail. Hence, it cannot be a superior alternative to parallel wall-to-wall flow.

3.4 Case (iv) Heat Removal Factor with PCM: Figure 4 shows the heat removal factor considering the PCM effect. It is observed that for the case of the PCM, F_R is a variable quantity. In this case F_R is the function of the PCM melting phases rather than mass flow; and, it is proportional to the melting temperature of the PCM. The factor decreases during phase change because in the charging stage the PCM itself absorbs heat. The factor increases during the sensible heat stage of the PCM. Variable values of F_R from 0 to 0.28 are obtained.

Figure 5 shows the temperature profile of water, and heat gained where thermal efficiency was calculated by (T_{out}-T_{in})/T_{out}. The effect of the PCM's heat of fusion is observed in the first hours of charging, which is encircled. However, it shows a very slow change in the outlet temperature with respect to time. The phase change is also evident from photographs shown in Figure 2.

Figure 6 shows the variation in melt fraction of PCM as it melts layer-by-layer. Vertical arrows A and B show the beginning of the charging and discharging respectively. The melting fraction (solid line, left axis) and the PCM temperature (dashed line, right axis) are shown. Arrow A shows the temperature where the PCM is completely melted (35^0C) and therefore from this stage the melting fraction is '1'. Similarly, arrow B shows that the PCM began to solidify and therefore there is a drop in melting fraction. However, in solidification the melting fraction does not reach zero quickly, which means the PCM remains charged. This state is advantageous as it facilitates the pre-heating of inlet water. Figure 7 shows the various stages of phase-change with OM35(PCM).

In the charging process, the average heat transfer coefficient increases sharply with increasing the molten layer thickness, as the natural convection grows stronger. In the discharge process, the useful heat gain was found to be increasing as the water mass flow rate increased (Mettawee et al., 2006).

3.5 Analysis of results

The results indicate superior thermal performance of parallel wall-to-wall flow relative to serpentine flow.

S. I. Abdel-Khalik (1976) calculated the value of F_R to be 0.915, which cannot be achieved practically. The value of F_R mentioned in section 3.1 is 0.44, which is a much lower value than that calculated by Abdel- Khalik in his paper. This is because of the various losses viz. pipe losses, leakages and other manufacturing losses.

One more striking result of this analysis is that the F_R increases as the mass flow rate ($\dot{m}$) increases, but there are some limits beyond which the increment in mass flow rate does not lead to an appreciable rise in the value of F_R. A graph (Figure 8) was plotted by varying the value of $\dot{m}$ for serpentine and wall-to-wall parallel flow. The Figure depicts the following results:

- For serpentine flow 4 kg/hr is the maximum mass flow rate where F_R is 0.47, since the F_R does not change significantly above that.
- For parallel wall-to-wall flow 4 kg/hr is the maximum mass flow rate where F_R is 0.96.
- The arrows indicate the value from where the value of F_R approximately remains unchanged.

Figure 9 shows a generalized chart for F_R for serpentine flow of arbitrary geometry. F_1 and F_2 are parameters which are functions of plate thickness and conductivity, tube spacing, and other physical design parameters. F_R/F_1 is known as collector flow factor and it is a function of single variable, dimensionless collector capacitance rate $\dot{m}.C_P/F_1A_{col}U_{los}$. This chart is plotted by varying the value of $\dot{m}$ along the abscissa. This chart is very useful, as the designer can determine the value of any design parameter if the value of other variables are predetermined or known.

4. Conclusions

It can be concluded that parallel wall-to-wall flow has superior thermal performance relative to the serpentine flow. It also shows the dependence of the heat removal factor on the fin efficiency of the system. As it is higher for parallel wall-to-wall flow, the heat removal factor of the system is greater. Fin efficiency of the parallel wall-to-wall flow is 1 as it provides smooth passage for the flow and there are no pipe losses.

The usage of phase change material decreases the value of heat removal factor as it is observed to be in the range 0–0.28 which is less than the value of heat removal factor when PCM is not used. However, the usage of PCM facilitates latent heat storage and provides heating even in the absence of sunlight. Therefore the usage of PCM is justified as well as fruitful when heating in the absence of sunlight is a required feature of the solar water heater. Additionally, the value of the heat removal factor does not increase significantly for the mass flow rate beyond 4 kg/hr.

NOMENCLATURE

A_{col}	area of collector (m^2)
c_p	heat capacity of water (4190 J/kg °C)
D	outside diameter of tube (m)
D_{in}	inside diameter of tube (m)
F_1	collector efficiency factor for serpentine flow
F_2	fin efficiency for serpentine flow
F'	fin efficiency for wall-to-wall-parallel flow
F_R	heat removal factor
G	radiation incident on collector (W)
h_{fin}	convective heat transfer coefficient for water (W/m^2 . °C)
K	parameter defined in equation (5)
L	length of collector (m)
$\dot{m}$	mass flow rate of circulating water (kg/hr)
N	number of serpentine turns
n	non dimensional parameter defined in equation (5)
Q_{use}	useful energy gained by the collector per unit time (W)
R	thermal resistance between the absorbing plate and circulating water (°C.m/W)
S	absorbed solar energy per unit time per unit area of the collector (W/m^2)
T_{amb}	ambient air temperature (°C)

T_{in}	inlet water temperature (°C)
$T_{p,m}$	mean temperature of the absorber plate (°C)
U_{los}	overall heat loss coefficient (W/m^2.°C)
W	spacing between serpentine tubes (m)
W_c	width of collector (m)
W_t	space between serpentine curve to edge (m)
y	non dimensional parameter defined in (5)

Greek symbols

k (kappa)	thermal conductivity of flat plate (W/m^2.°C)
δ (delta)	thickness of the absorber plate (m)

Subscripts

Amb	ambient
Col	collector
In	inlet
los	loss
out	outlet
p,m	mean temperature
th	thermal
use	use

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