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Technical Features and Thermal Efficiencies of Various Flat Plate Solar Collectors

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Abstract. Flat plate solar water collectors are widely used for supplying hot water or hot air for domestic or industrial applications in the world. These collectors provide a pretty big profit when compared with other energy resources. For instance, it was determined by a study that these collectors are approximately 2.0, 3.5, 4.0, 6.0, 7.0 and 12.0 times more profitable than wood, coal, natural gas, oil, LPG, and electricity, respectively for heating water. Therefore, technical features and thermal efficiencies of different type flat plate solar air and water collector (heater) systems such as collector panels (arrays), heat exchangers and hot water storage tanks were presented in this research. Besides, calculation of ideal collector panel tilt angle and different theoretical efficiency analysis methods such as energy balance method and complicated method was discussed in this work. As a result, the high thermal efficiencies were found as 46%, 88% and 95% for an air collector, for a natural circulation corrugated 304 chromium water collector, and for a vertical hot water tank, respectively.

Keywords: Solar Water Collector, Solar Air Collector, Solar Heater, Flat Plate Collector.

1. Introduction

Water is used in households predominantly for drinking, cooking and hygiene purposes such as bathing and the washing of dishes. Each person in a household requires at least 20 L of potable water per day, half of which is for personal hygiene. Water is also needed to be heated to meet hot water necessity for cooking and hygiene. The use of hot water during the day is approximately the same in the morning as it is in the evening, but less during the afternoon in households [1].

Different heat sources are employed for heating water. However, in most developing countries, supplies of non-renewable sources of energy are either unavailable, unreliable or too expensive. In renewable energy sources, solar energy is the most appropriate for heating water. This energy allows independent systems to be constructed and possesses a thermal conversion mode which necessitates a simple technology [2].

Solar energy received on the ground is abundant and inexhaustible. In addition to its inexhaustible nature, solar energy has the advantage of being a source of nonpolluting energy. This energy could be harnessed in several ways. The most promising energy forms are solar collectors with thermal conversion, which can be used to heat water for domestic and industrial applications. These applications are developing most rapidly and are the basis of small but growing industry [3].

Collectors are the heart of the solar processes. In solar collector, energy transfer from a distant source of radiant energy to a fluid or store as latent heat energy by using PCM [4-10]. The flux of incident radiation is, at best, approximately 1100 W/m2 (without optical concentration), and it is variable. The wavelength range is from 0.3 to 3 μm, which is considerably shorter than that of the emitted radiation from most energy absorbing surfaces. Thus, the analysis of solar collectors presents unique problems of low and variable energy fluxes and the relatively large importance of radiation. Flat plate collectors can be designed for

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applications requiring energy delivery at moderate temperatures, up to perhaps 100 ℃ above ambient temperature. They use both beam and diffuse solar radiation, do not require tracking of the sun, and require little maintenance. The major applications of these units are in solar water heating, building heating, air conditioning, and industrial process heat [11, 12]. The importance of flat plate collectors in thermal processes is such that their thermal performance is treated in considerable detail. This is done to develop an understanding of how the component functions [13, 14].

Many types of conventional solar collectors with metal absorber plates and glass covers are widely used to transforms solar energy into heat for heating water for domestic applications and industrial processes in Turkey and world [15, 16]. These collectors have high efficiency and they are approximately, 2.0, 3.5, 4.0, 6.0, 7.0 and 12.0 times more profitable when compared with other energy sources such as wood, coal, natural gas, oil, LPG and electricity, respectively for heating water for domestic applications in Turkey [17].

2. Theoretical Analysis

The optimum or ideal collector tilt angle (collector panel slope) for any location can be computed by using Equations 1 and 2. The ideal collector array slope is changing depends on latitude of the location, declination angle and the number of the day [17]. All symbols used in equations are described in the Nomenclature.

$$\beta = (\phi - \delta) \tag{1}$$

$$\delta = 23{,}45 \sin\left(360 \frac{284+n}{365}\right) \tag{2}$$

Basically, the flat plate solar collectors operate under quasi steady-state conditions (Fig. 1). In these conditions, the thermal performance or efficiency of a solar collector is described by an *main energy balance* that indicates the distribution of incident solar energy into useful energy gain, thermal losses, and optical losses (Fig. 2 and 3) [13, 15, 18, 19].

The useful heat gain by a collector can be expressed as:

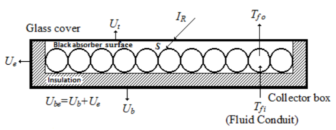


Fig. 1: Description of the flat plate solar water collector (main parts and general heat transfer exchanges).

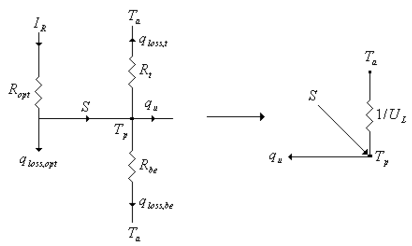
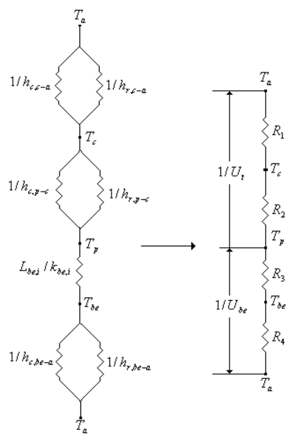


Fig. 2: Equivalent thermal network for flat plate solar water collector.



(b)

Fig. 3: Thermal network for the flat plate solar water collector in terms of conduction, convection and radiation (a), and in terms of resistances between plates (b)

$$q_s = I_R A_c = q_{loss} + q_u \tag{3}$$

$$q_u = q_s - q_{loss} = I_R A_c - q_{loss} \tag{4}$$

$$q_{loss} = q_{loss,opt} + q_{loss,t} + q_{loss,be} \tag{5}$$

$$q_u = I_R A_c - q_{loss,opt} - q_{loss,t} - q_{loss,be} \tag{6}$$

$$SA_c = I_R A_c - q_{loss,opt} = I_R A_c (\tau_c \alpha_p) \tag{7}$$

$$q_u = SA_c - q_{loss,t} - q_{loss,be} \tag{8}$$

$$q_u = I_R A_c (\tau_c \alpha_p) - q_{loss,t} - q_{loss,be} \tag{9}$$

$$q_u = \dot{m} c_{p,f} (T_{f,o} - T_{f,i}) \tag{10}$$

A measure of collector performance is the collector efficiency, defined as the ratio of useful heat gain over any time period to the incident solar radiation over the same period we can, thus, define efficiency as,

$$\eta = \frac{q_u}{q_s} \tag{11}$$

$$\eta = \frac{\sum q_u}{\sum q_s} \tag{12}$$

From Equations (3, 10, 11 and 12),

$$\eta = \frac{\dot{m} c_{p,f} (T_{f,o} - T_{f,i})}{I_R A_c} \tag{13}$$

In addition, the thermal efficiency of a flat plate solar collector can be determined by using some *complicated equations* as discussed below.

It is convenient to define a quantity that relates the actual useful energy gain of a collector to the useful gain if the whole collector surface were at the fluid inlet temperature. This is quantity is called the collector heat removal factor F_R. The actual useful energy gain and the collector heat removal factor can be expressed as

$$q_u = A_c F_R \left[F_{D,S} S - U_L (T_{f,o} - T_{f,i}) \right] \tag{14}$$

$$F_R = \frac{\dot{m} c_{p,f} (T_{f,o} - T_{f,i})}{A_c \left[S - U_L (T_{f,i} - T_a) \right]} \tag{15}$$

$$F_R = \frac{\dot{m} c_{p,f}}{A_c U_L} \left[\frac{T_{f,o} - T_{f,i}}{\frac{S}{U_L} - (T_{f,i} - T_a)} \right] \tag{16}$$

$$F_R = \frac{\dot{m} c_{p,f}}{A_c U_L} \left[\frac{\left(T_{f,o} - T_a - \frac{S}{U_L} \right) - \left(T_{f,i} - T_a - \frac{S}{U_L} \right)}{\frac{S}{U_L} - (T_{f,i} - T_a)} \right] \tag{17}$$

$$F_R = \frac{\dot{m} c_{p,f}}{A_c U_L} \left[1 - \frac{\frac{S}{U_L} - (T_{f,o} - T_a)}{\frac{S}{U_L} - (T_{f,i} - T_a)} \right] \tag{18}$$

U_L is the collector overall heat loss coefficient. The thermal energy lost from the collector to the surroundings by conduction, convection and infrared radiation. U_L is equal to the sum of energy loss through the top (U_t), bottom (U_b) and edge (U_e) of the collectors given below (Fig. 2 and 3) [15] :

$$U_L = U_t + U_b + U_e = U_t + U_{be} \tag{19}$$

The energy loss through the top is the result of convection and radiation between parallel plates. The top loss coefficient from the collector plate to the ambient is

$$\frac{1}{R_1} = \frac{1}{\frac{1}{h_{c,c-a}}} + \frac{1}{\frac{1}{h_{r,c-a}}} = \frac{h_{c,c-a} + h_{r,c-a}}{1} \tag{20}$$

$$R_1 = \frac{1}{h_{c,c-a} + h_{r,c-a}} \tag{21}$$

$$\frac{1}{R_2} = \frac{1}{\frac{1}{h_{c,p-c}}} + \frac{1}{\frac{1}{h_{r,p-c}}} = \frac{h_{c,p-c} + h_{r,p-c}}{1} \tag{22}$$

$$R_2 = \frac{1}{h_{c,p-c} + h_{r,p-c}} \tag{23}$$

$$\frac{1}{U_t} = R_1 + R_2 = \frac{1}{h_{c,c-a} + h_{r,c-a}} + \frac{1}{h_{c,p-c} + h_{r,p-c}} \tag{24}$$

$$U_t = \left[\frac{1}{h_{c,c-a} + h_{r,c-a}} + \frac{1}{h_{c,p-c} + h_{r,p-c}} \right]^{-1} \tag{25}$$

Besides, h_w, U_b and U_e must be obtained from the equations as follows [20]:

$$h_{c,c-a} = h_w = 2.8 + 3\upsilon \tag{26}$$

$$U_t = \left[\frac{1}{h_w + h_{r,c-a}} + \frac{1}{h_{c,p-c} + h_{r,p-c}} \right]^{-1} \tag{27}$$

$$R_3 = \frac{L_{be,i}}{k_{be,i}} + \frac{A_c}{\left(\frac{k_{be,i}}{L_{be,i}} \right) A_{c,e}} \tag{28}$$

$$\frac{1}{R_4} = \frac{1}{\frac{1}{h_{c,be-a}}} + \frac{1}{\frac{1}{h_{r,be-a}}} = \frac{h_{c,be-a} + h_{r,be-a}}{1} \tag{29}$$

$$R_4 = \frac{1}{h_{c,be-a} + h_{r,be-a}} \tag{30}$$

$$\frac{1}{U_{be}} = R_3 + R_4 \tag{31}$$

$R_4 \cong 0$ is very small and negligible, so

$$\frac{1}{U_{be}} = R_3 = \frac{L_{be,i}}{k_{be,i}} + \frac{A_c}{\left(\frac{k_{be,i}}{L_{be,i}} \right) A_{c,e}} \tag{32}$$

$$U_{be} = \frac{k_{be,i}}{L_{be,i}} + \frac{\left(\frac{k_{be,i}}{L_{be,i}} \right) A_{c,e}}{A_c} \tag{33}$$

$$U_t = \left(\frac{1}{h_{c,p-c} + h_{r,p-c}} + \frac{1}{h_w + h_{r,c-a}} \right)^{-1} \tag{34}$$

In order to find $h_{c,p-c}$, $h_{r,p-c}$ and $h_{r,c-a}$ for these solar water collectors, a series of equations seen below must be used [15].

$$R_a = \frac{g\beta_{p-c}\Delta T_{p-c}L_{p-c}^{\ 3}}{\nu_{a,p-c}\lambda_{a,p-c}} \tag{35}$$

$$\beta = \frac{1}{T_m} \tag{36}$$

$$T_m = \frac{T_c + T_p}{2} \tag{37}$$

$$P_r = \frac{\nu_{a,p-c}}{\lambda_{a,p-c}} \tag{38}$$

$$R_a = \frac{g\Delta T_{p-c}L_{p-c}^{\ 3}P_r}{T_m\nu_{a,p-c}^{\ 2}} \tag{39}$$

$$R_a = \frac{g\beta_{p-c}\Delta T_{p-c}L_{p-c}^{\ 3}}{\nu_{a,p-c}^{\ 2}} \tag{40}$$

$$N_u = 1 + 1{,}44\left[1 - \frac{1708(\sin 1{,}8\beta_{p-c})^{1{,}6}}{R_a\cos\beta_{p-c}}\right]\left[1 - \frac{1708}{R_a\cos\beta_{p-c}}\right] + \left[\left(\frac{R_a\cos\beta_{p-c}}{5830}\right)^{1/3} - 1\right] \tag{41}$$

$$h_{c,p-c} = \frac{N_u k_{a,p-c}}{L_{a,p-c}} \tag{42}$$

$$h_{r,p-c} = \frac{\sigma(T_p^{\ 2} + T_c^{\ 2})(T_p + T_c)}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_c} - 1} \tag{43}$$

$$h_{r,c-a} = \varepsilon_c\sigma(T_c^{\ 2} + T_a^{\ 2})(T_c + T_a) \tag{44}$$

$$T_c = T_p - \frac{U_t(T_p - T_a)}{h_{c,p-c} + h_{r,p-c}} \tag{45}$$

Another equation to find U_t was developed by Klein. U_t was also calculated by using this equation (46). It was seen that it gives the same results with very small and negligible differences [13].

$$U_t = \left[\frac{n}{\frac{C}{T_p}\left[\frac{(T_p - T_a)}{(n+f)}\right]^e} + \frac{1}{h_w}\right]^{-1} + \left[\frac{\sigma(T_p + T_a)(T_p^{\ 2} + T_a^{\ 2})}{(\varepsilon_p + 0{,}00591nh_w)^{-1} + \frac{2n + f - 1 + 0{,}133\varepsilon_p}{\varepsilon_c} - n}\right] \tag{46}$$

In Equation (46),

$$f = (1 + 0{,}089h_w - 0{,}1166h_w\varepsilon_p)(1 + 0{,}07866n) \tag{47}$$

$$C = 520(1 - 0{,}000051\beta_{a,p-c}^{\ 2}) \tag{48}$$

$$e = 0{,}430\left(1 - \frac{100}{T_p}\right) \tag{49}$$

As a result of this theoretical analysis, it should be noted that calculated thermal efficiency of any flat plate solar collector by using main energy balance equations (Eq. 3…13) or complicated equations (Eq. 13…49) give the same results with very small differences. In general, the results of main energy balance equations are approximately %2 higher than the results of complicated equations.

3. Results of Various Flat Plate Solar Collectors

Six different solar air heaters have designed, manufactured and tested [21]. The highest thermal efficiency was determines as 46%.

Four identical flat plate solar air collector were experimented at 25 ° (ideal tilt angle from Eq. 1 and 2), 30 °, 35 ° and 40 ° tilt angles. It was seen from the result that the maximum and minimum efficiencies were obtained as % 40 and % 36 for 25 and 40 ° tilt angels, respectively [17].

The effect of some components on thermal efficiencies of flat plate solar water collectors were investigated [22]. It was seen that the efficiencies of aluminum and copper collector panels are changing between 56%...68%. In addition, it was found that the black copper selective absorber surface increase thermal efficiency as 7.13%, the prismatic glass raise the efficiency as 9.37%, the radiation reflector located between absorber and insulater grow the efficiency as 1.12% and the normal tempered glass cover develop the efficiency as 9.63.

A black copper selective surface panel solar water collector system that can be operated as natural circulation closed thermosyphon system with a chromium 304 fluid-to-fluid heat exchanger was designed, manufactured and tested [23, 24]. The efficiency of the collector array and heat exchangers were separately investigated. The results showed that the efficiency of the collector array, horizontal type and vertical type heat exchangers are 63%, 29% and 33%, respectively. The efficiency of combination of both collector array and heat exchangers were also defined as 18% and 21% for horizontal and vertical heat exchangers, respectively.

Three type chromium 304 solar water collectors such as dry type, wet type and storage type was designed, manufactured and tested [25]. The results showed that thermal efficiencies of dry type, storage type and wet type are 22%, 33% and 43%, respectively. The highest efficiency was found for wet type collector when there is fluid (mixture of 60% water and 40% glikol) between pipes.

A domestic type of a 304 stainless steel chromium flat plate solar water collector that can be directly connected to the pressurized city water line system and keep hot water temperature with PCM (phase change material) placed inside the collector during night was designed, constructed and tested [26]. Test results showed that, thermal efficiency of the collector was defined as 12.5% and 62.0% while fluid stored and flows, respectively.

Three different collectors which made of 304 matte chromium that could be directly connected to the city's main waterworks were manufactured for test of experiments [27]. Pipe diameters of collectors were 20mm, 25mm and 32mm. In first step, the collector panels were experimented for their efficiency. In second step, a pressurized and insulated vertical hot water tank was connected with collector panels and tested. The results showed that the efficiencies of collector panels are 67%, 74% and 77% for 20mm, 25mm and 32mm pipe diameters, respectively. Vertical hot water tank's standalone efficiency was found as 70%. The average efficiency of combination of both collector panel with 32 mm pipe diameter and hot water tank was determined as 54%.

A flat plate solar water collector manufactured from matt type stainless steel chromium 304 were designed, constructed and tested [28]. As a result, the thermal efficiency of the collector was defined as 50 % and 64 % when gap between pipes full of air and mixture of water-antifreeze, respectively. The thermal efficiency of natural circulaton and insulated vertical hot water tank was also determined as 95%. The average efficiency of combination of both collector panel (while gap full of water-antifreeze) and hot water tank was determined as 61%. A corrugated solar water collector panel which is natural circulation and made up of 304 matte chromium plate was investigated. The efficiency of the collector panel was determined as 88% [29].

4. Conclusions

1) Flat plate collector panels should be located by considering ideal collector tilt angle.

2) High efficient flat plate solar air collector should be used for air heating.

3) Corrugated and made up of 304 chromium solar water collector panel should be preferred.

4) Natural circulation open thermosiphon collector systems should be used.

5) Natural circulaton and insulated vertical hot water tank should be preferred.

6) It is avoid to use flat plate closed thermosiphon collector systems and heat excahngers.

5. Nomenclature

A_c : Collector area, m^2

$A_{c,e}$: Collector edge surface area, m^2

$c_{p,f}$: Fluid specific heat at constant pressure, J/(kg.K)

$F_{D,S}$: Collector dust and shading factor

F_R : Collector heat removal factor

g : Gravitational acceleration, m/s^2

$h_{c,c-a}$: Convection heat transfer coefficient between cover and ambient air, $W/(m^2.K)$

$h_{c,be-a}$: Convection heat transfer coefficient between bottom-edge and ambient air, $W/(m^2.K)$

$h_{c,p-c}$: Convection heat transfer coefficient between plate and cover, $W/(m^2.K)$

$h_{r,be-a}$: Radiation heat transfer coefficient between bottom-edge and ambient air, $W/(m^2.K)$

$h_{r,c-a}$: Radiation heat transfer coefficient between cover and ambient air, $W/(m^2.K)$

$h_{r,p-c}$: Radiation heat transfer coefficient between plate and cover, $W/(m^2.K)$

h_w : Wind heat transfer coefficient, $W/(m^2.K)$

I_R : Incident solar radiation, W/m^2

$k_{a,p-c}$: Air thermal conductivity between plate and cover, W/(m.K)

$k_{be,i}$: Bottom-edge insulation thermal conductivity, W/(m.K)

$L_{be,i}$: Bottom-edge insulation thickness, m

$L_{a,p-c}$: Air length between plate and cover, m

$\dot{m}$: Fluid mass flow rate, kg/s

n : Number of cover; the day of the year ($1 \leq n \leq 365$)

N_u : Nusselt Number

P_r : Prandtl number

q_{loss} : Thermal energy losses, W

$q_{loss,be}$: Thermal energy losses through the bottom-edge, W

$q_{loss,opt}$: Optical energy losses, W

$q_{loss,t}$: Thermal energy losses through the top, W

q_s : Incident solar power, W

q_u : Useful thermal power gain, W

R_a : Rayleigh number

R_{be} : Bottom-edge resistance, $(m^2.K)/W$

R_{opt} : Optical resistance, $(m^2.K)/W$

R_t : Top resistance, $(m^2.K)/W$

R_1, R_2, R_3, R_4 : Resistances, $(m^2.K)/W$

S : Absorbed solar radiation, W/m^2

T_a : Ambient air temperature, K

T_{be} : Bottom-edge surface temperature, K

T_c : Cover temperature, K

$T_{f,i}$: Fluid inlet temperature, K

$T_{f,b-1}$: Fluid temperature at point b-1, K

$T_{f,b-2}$: Fluid temperature at point b-2, K

$T_{f,t-1}$: Fluid temperature at point t-1, K

$T_{f,t-2}$: Fluid temperature at point t-2, K

$T_{s-1...s-10}$: Black painted pipe surface temperature at point s-1…s-10, K

$T_{f,o}$: Fluid outlet temperature, K

T_p : Plate surface temperature, K

T_m : Mean temperature, K

U_{be} : Bottom-edge heat loss coefficient, $W/(m^2.K)$

U_L : Collector overall heat loss coefficient, $W/(m^2.K)$

U_t : Top heat loss coefficient, $W/(m^2.K)$

υ : Wind speed, m/s

Greek

$\lambda_{a,p-c}$: Thermal diffusivity of air between plate and cover

α_p : Plate absorptance coefficient

β : Ideal (optimum) collector panel tilt angle (collector panel slope), ° ($0° \leq \beta \leq 180°$), ($\beta > 90°$ means that the surface has a downward facing component)

$\beta_{a,p-c}$: Volumetric expansion coefficient of air between plate and cover (for an ideal gas β = 1/Tm),

ϕ : Latitude angle of location, °(the angular location north or south of the equator, north positive; $-90° \leq \phi \leq 90°$)

δ : Declination angle, °($-23.45° \leq \delta \leq 23.45°$)

ΔT_{p-c} : Temperature differences between plate and cover, K

ε_c : Cover emittance

ε_p : Plate emittance

η : Collector thermal efficiency

$\nu_{a,p-c}$: Kinematic viscosity of air between plate and cover, m^2/s

σ : Stefan-Boltzmann constant ($5.67x10^{-8}$), $W/(m^2.K^4)$

τ_c : Cover transmittance

$(\tau_c \alpha_p)$: Transmittance-absorptance product

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