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"Research Note"

THERMAL PERFORMANCE EVALUATION OF A PROPOSED POINT-FOCUS SOLAR COLLECTOR FOR LOW POWER APPLICATIONS*

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Abstract– In this study, thermal performance of a proposed point-focus solar collector for low power applications was estimated under different operating variables. For this purpose, theoretical analysis was employed with varying relevant parameters, using a set of thermodynamics and energy equations, i.e., ambient temperature, beam solar insolation, wind speed, wind incidence angle and wall temperature of the absorber. The results show decreasing trend of the wind incidence angle along with increasing the convective heat loss coefficient as the highest related values obtained under head-on wind flow, but the wall temperature of the absorber exerts negligible influence. The maximum thermal efficiency of 79.68% was obtained in August with the side-on wind flow of $4.9\,m/s$ and an ambient temperature of 29.2℃ when the absorber wall temperature has a minimum value of 150℃.

Keywords– Solar energy, point-focus solar collector, low power application, thermal performance

1. INTRODUCTION

Solar energy is the most interesting and promising source that plays a vital role in meeting the increasing energy demand and saving the depleting fossil fuel resources. The rapid consumption of fossil fuels to meet the increasing energy demand leads to environmental pollution and lack of fossil fuels is prompting the search for alternative energy resources to achieve sustainable development [1]. Concentrating Solar Power (CSP) systems namely parabolic trough, linear Fresnel reflector, power tower and parabolic dish can be used effectively to convert solar energy into heat [2]. Solar dish systems can provide an economical source of power and become a key source of renewable energies in the coming years [3]. In general, there are two different designs of receivers in solar dish systems; external and cavity designs. External receivers which are usually spherical absorbing radiation coming from different directions while cavity receivers have an aperture through which the radiation passes [4]. The external receiver could be interesting for low power applications. In a cavity receiver, on the other hand, a large part of the emitted radiation remains inside the cavity and is absorbed again so that the total radiative heat loss is lower [5].

Since very few studies have considered simulations using actual meteorological data, the focus of this study is to conduct monthly performance of a proposed steam generating point-focus solar collector for use in low power applications under typical weather conditions of Tehran, the capital city of Iran. The monthly thermal performance of the system is estimated under varying different operating parameters such as, wind angle incidence and wall temperature of the absorber (receiver). For this purpose, thermodynamics and energy balance equations are employed. At the end, the best month with the most advantageous of parameters is presented.

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2. DESCRIPTION OF THE PROPOSED POINT-FOCUS SOLAR COLLECTOR

Figure 1 presents the schematic of the proposed point-focus solar collector consisting of a parabolic dish reflector, a cylindrical absorber and a tracking system. The absorber located at the focal point receives the maximum amount of concentrated solar energy and transfers it to thermal energy. The bottom side of the absorber is covered with black chrome to increase the absorptivity. The reflector surface is covered with rectangular high reflective mirror sheet segments with 1mm thickness.

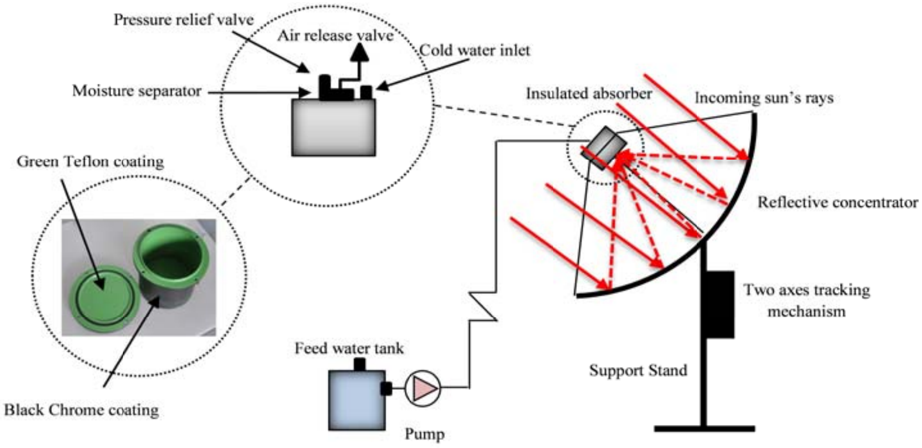

Fig. 1. The schematic of the point-focus collector

a) Thermal efficiency of the collector η_{th}

The thermal efficiency of the collector (dish/absorber) is defined as the ratio of the useful energy delivered to the energy incident on the concentrator aperture [6]:

$$\eta_{th} = \dot{Q}_u / \dot{Q}_s \quad (1)$$

Consider that the concentrator has an aperture area A_{ap} and receives solar radiation at the rate $\dot{Q}_s$ from the sun, as shown in Fig. 2. The net solar heat transferred $\dot{Q}_s$ is proportional to A_{ap}, and the direct normal insolation per unit of collector area I_b [7]:

$$\dot{Q}_s = I_b A_{ap} \quad (2)$$

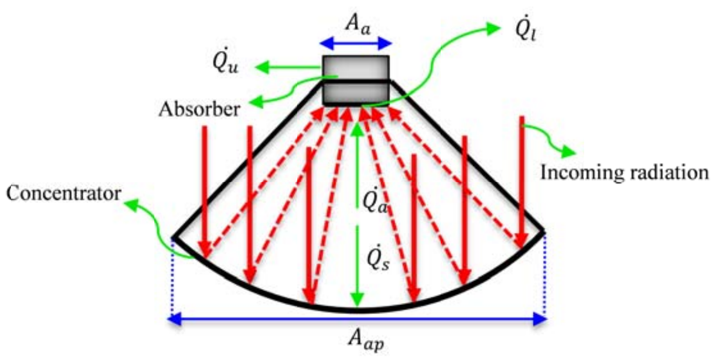

Fig. 2. Schematic of the collector thermal modeling

Under steady state conditions, the useful heat delivered by a solar collector system is equal to the energy absorbed by the heat transfer fluid, which is determined by the radiant solar energy falling on the absorber, minus the direct or indirect heat losses from the absorber to the surroundings [7]:

$$\dot{Q}_u = \dot{Q}_a - \dot{Q}_l \tag{3}$$

The optical efficiency could be obtained using Eq. (5), where $\dot{Q}_a$ is the concentrated solar radiation intercepted by the absorber [8–10]:

$$\eta_o = {\dot{Q}_a}/{\dot{Q}_s} \tag{4}$$

The absorber efficiency η_a is also defined as the ratio of the useful energy delivered to the energy falling on the absorber [6]:

$$\eta_a = {\dot{Q}_u}/{\dot{Q}_a} \tag{5}$$

b) Optical efficiency of the collector η_o

The optical efficiency depends on the optical properties of the materials involved, the geometry of the collector, and the various imperfections arising from the construction of the collector. The following equation can be used to approximate the optical efficiency [6]:

$$\eta_o = \lambda\rho\tau\alpha\gamma\cos(\theta) \tag{6}$$

Where λ is the un-shading factor, ρ is dish reflectance, $\alpha\tau$ is transmittance-absorptance product, γ is the intercept factor of the absorber, which is defined as the ratio of the energy intercepted by the absorber to the energy reflected by the parabolic dish and θ is the angle of incidence.

c) Total heat loss rate of the absorber $\dot{Q}_l$

The total heat loss rate of the absorber, $\dot{Q}_l$, includes three contributions: (i) conductive heat loss from $\dot{Q}_{lk}$, (ii) convection heat loss, $\dot{Q}_{lc}$, and (iii) radiation heat loss, $\dot{Q}_{lr}$. The total heat loss rate $\dot{Q}_l$ can be expressed as [6]:

$$\dot{Q}_l = \dot{Q}_{lk} + \dot{Q}_{lc} + \dot{Q}_{lr} \tag{7}$$

Therefore, heat transfer coefficient, h_c, for laminar fluid flow on a circular flat plate with diameter of D is expressed as follows:

$$h_c = \left(0.664 Re_D^{0.5} Pr^{\frac{1}{3}}\right).k_{air}/D \qquad Re_D < 5 \times 10^5 \tag{8}$$

Kendoush [11] proposed an equation to predict the convection heat loss coefficient in the case of the head-on and oblique fluid flow over a flat plate with specific temperature:

$$h_c = 0.848 k_{air}(cos\theta V_w Pr/\nu)^{0.5}\,(D/2)^{-0.5} \tag{9}$$

All thermo-physical properties of air are calculated in film temperature, $T_f = (T_s + T_{air})/2$. The following equation is used to evaluate the radiation loss from the absorber [12-13]:

$$\dot{Q}_r = h_r A_a (T_s - T_{sky}) \tag{10}$$

where

$$h_r = \varepsilon\sigma(T_s + T_{sky})(T_s^2 + T_{sky}^2) \tag{11}$$

where

$$T_{sky} = 0.0552 T_{air}^{1.5} \tag{12}$$

Therefore the useful energy gained, $\dot{Q}_u$, can be expressed in mathematical form as:

$$\dot{Q}_u = \eta_o I_b A_{ap} - U_t A_a (T_s - T_{air}) \tag{13}$$

where

$$U_t = h_c + h_r \tag{14}$$

3. RESULTS AND DISCUSSIONS

a) Variation of forced convection and radiation heat loss with related parameters

In order to evaluate the thermal performance of the system, heat loss mechanism associated with the absorber must be determined with sufficient accuracy. Convection and radiation heat loss of the absorber are estimated under windy environments. Figures 3-5 present the variation of h_c, with α and T_s for different values of T_{air} and V_w. Head-on and side-on wind flows of each month in conjunction with receiver tilt angles of 0°, 30°, 60°, and 90° are considered. As evident from the figures, h_c increases with increasing V_w but investigating the effect of wind direction requires further deliberation. As can be seen from Fig. 3, in the case of $T_s = 150$°C, the highest value of $h_c = 22.79\ W/m^2.K$ is obtained under *head-on* wind flow ($\alpha = 0^\circ$) in May which has the highest value of $V_w = 7.2\, m/s$ and the lowest value of $h_c = 15.88\, W/m^2K$ is obtained under *side-on* wind flow ($\alpha = 90^\circ$) in January with the lowest value of $V_w = 3.5\, m/s$. In addition, the estimated values of h_c at $\alpha = 30^\circ$ are larger than the values of $\alpha = 60^\circ$ and smaller than values obtained under head-on wind flow ($\alpha = 90^\circ$). Figures 4 and 5 show similar trends in the case of $T_s = 200$°C and 250°C respectively. As a result, decreasing the incidence angle of the wind causes the convection heat loss coefficients to increase. Therefore it is concluded that no significant relevance exists between h_c and T_s.

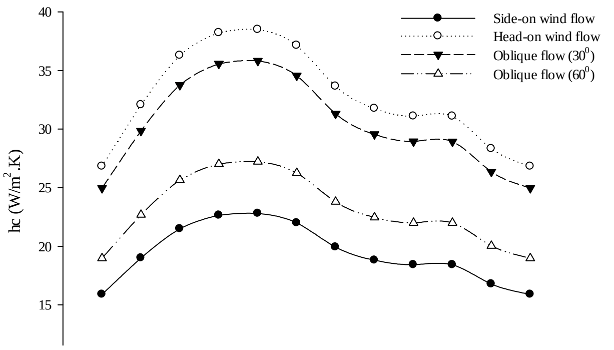

Fig. 3. Variation of h_c with different α for the case of $T_s = 150$°C

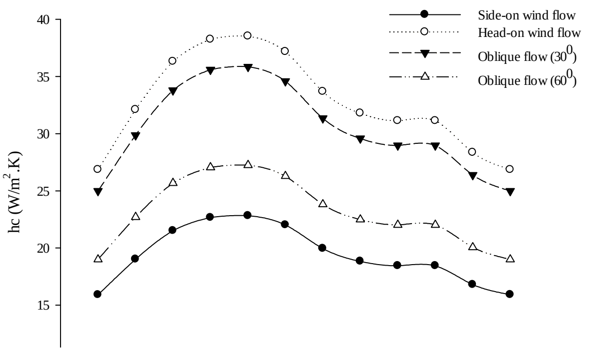

Fig. 4. Variation of h_c with different α for the case of $T_s = 200$°C

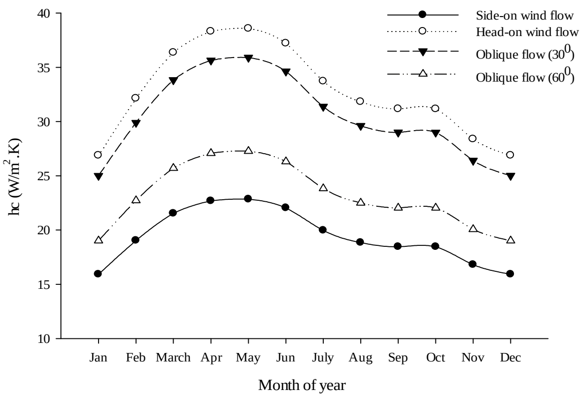

Fig. 5. Variation of h_c with different α for the case of $T_s = 250$°C

b) Evaluation of thermal efficiency of the collector under different operating conditions

In order to demonstrate the effect of varying parameters on the total efficiency of the point-focus solar collector, the values of η_{th} are plotted as a function of T_s (Fig. 6 and 7). As expected, apart from the effect of wind direction, the largest values of η_{th} are obtained in the smallest values of T_s. As shown in Fig. 6, η_{th} reaches the highest average value of 76.33% when $T_s = 150$°C and the lowest average value of 70.82% when $T_s = 250$°C under side-on wind flow.

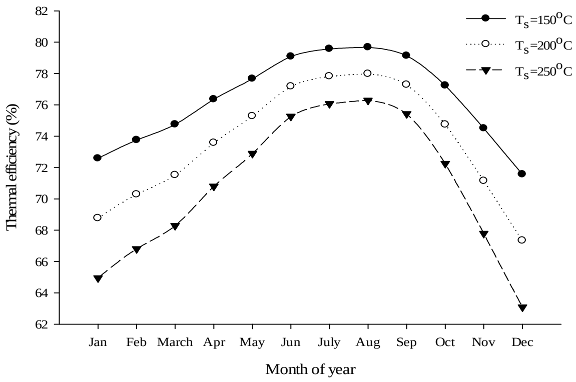

Fig. 6. Variation of η_{th} with different T_s under side-on wind flow

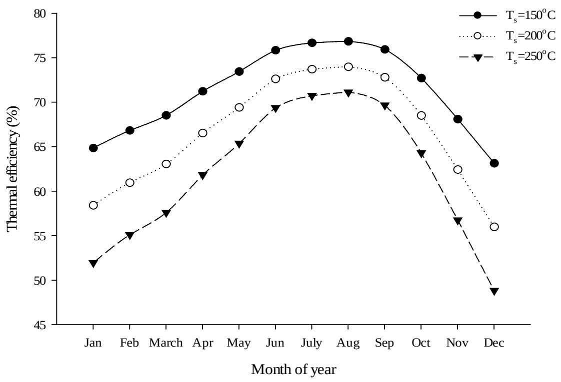

Fig. 7. Variation of η_{th} with different T_s under head-on wind flow

Based on the estimated results, the influence of α on η_{th} becomes more considerable along with increasing of T_s.

4. CONCLUSION

The monthly thermal performance of the proposed point-focus steam generating solar collector is estimated under weather condition of Tehran. In the case of $T_s = 150$°C, the highest value of $h_c = 22.79\ W/m^2.K$ is obtained in May under head-on wind flow and consequently the lowest value of $h_c = 15.88\,W/m^2.K$ is obtained in January under side-on wind flow ($\alpha = 90^{\circ}$). It is evident from the obtained results that η_{th} increases with increasing α. The lowest $\eta_{th} = 48.87\%$ is obtained when $\alpha = 0°$, while the η_{th} reaches the highest value of 79.68% when $\alpha = 90°$. It is found that η_{th} also increases evidently with decreasing T_s and reaches the largest average value of 76.33% when $T_s = 150$°C and the smallest average of 70.82% when $T_s = 250$°C under side-on wind flow. The use of typical meteorological data is essential for comparative studies of the performance of thermal devices.

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