Heat Transfer Correlation for the V-Groove Solar Collector

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Abstract:- The performance of solar collectors of conventional design can be base on absorber shape factor by supposing fully developed turbulent flow. A mathematical model has been developed from the energy balance equations of the solar collector system to predict the value of heat transfer coefficient. The Charters correlation has been modified by adding a constant, *K*. This value is determined by comparing the theoretical efficiency with the efficiency from the experimental data. Since the heat transfer coefficients are temperature dependent, an initial values of the mean temperatures of the glass cover, the plate and the backplate have been approximated which allowed the heat transfer coefficients to be evaluated as a first guess. An iterative procedure have been developed that enabled the absorber's mean temperatures of the collector to be calculated. The newly calculated mean temperatures were then compared with the initially guessed temperatures. The iterative procedure was repeated until consecutive mean temperature values differed by less than 0.01°C. This mathematical model can also be apply to other solar collector with different kind of absorber plate in order to get the value of *K*.

Key-Words: Heat transfer correlation, V-Groove collector, heat transfer coefficient, heat balance, thermal performance

1 Introduction

Solar collectors are key components in many engineering applications. It is can be used for many applications in drying of agricultural products, space heating, solar desalination, etc. Improving their performance is essential for commercial acceptance of their use in such applications. It is important to note that the most crucial parameter of solar air collectors design is the forced convective heat transfer coefficient between the air and absorber plate.

Conventional solar air collectors have inherent disadvantages is lower thermal efficiency. One way to achieve considerable improvement in collector efficiency is to use an extended heat transfer area by using fins or corrugated surfaces [1,2], and many studies have been carried out on this topic. Hence, different modifications are suggested and applied to improve the heat transfer coefficient between the absorber plate and the air. One of the modifications is the use of V-grooved absorbers [1,2]. The V-groove absorber with the corrugation running horizontally improves the radiative characteristics of the absorber plate. This configuration also approximate the shape of some concentrating collectors.

In the study of solar air collectors, it is necessary to know the forced convection heat transfer coefficient between two parallel plates. Hence, many heat transfer correlations have been developed for sizing of the solar collectors and systems. Most correlations based on fully developed turbulent flow with one side heated and the other side insulated [3-5]. In this paper heat transfer coefficient for the V-groove solar collector has been developed by modifying existing correlations for the two parallel plates with fully developed turbulent flow with one side heated and the other side insulated.

2 Material and Methods

A typical V-groove collector is shown in Fig.1. An absorber plat in an aluminium sheets SWG 22 with 14 grooves, the dimension of the groove where a = 8.67 cm, b = 7.9 cm, c = 7.14 cm and $\theta = 48.6^o$. This V-groove absorber plate has wider surface that will provide higher efficiency. Fig. 2 shows the cross section of the single back-pass solar collector with V-groove absorber.

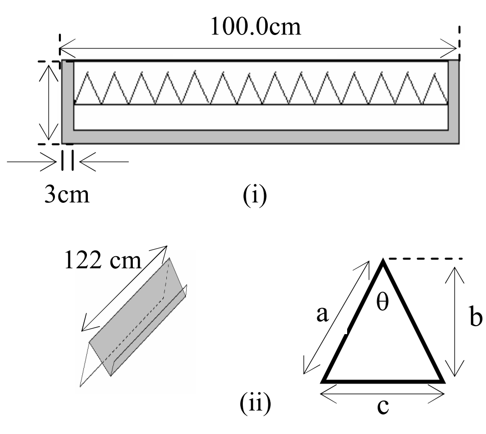

Fig. 1: (i) Cross-section of the V-groove solar collector; (ii) V-groove absorber design

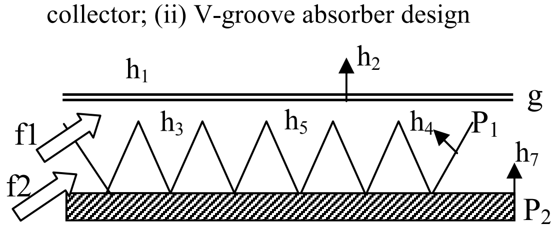

Fig. 2: Schematic of V-groove flat plate solar air collector showing energy balance

The fan is of centrifugal type, rated 1 hp, suitable to work under 2.5 kPa gauge pressure delivering approximately 250 - 300 cfm. The amount of air delivered by the fan is controlled by rheostat. The air velocity in the passage is measured by turbine flow meter. All temperatures are measured by type K thermocouples using a *Hybrid Recorders YOKOGAWA HR 1300*. The solar radiation incident on the collector surface is measured using a pyranometer. A solar simulator has been used to simulate controlled solar radiation. The simulator uses 45 halogen lamps, each with a rated power of 300 W. The maximum average radiation of 642 W/m^2 can be reached. Dimmers are use to control the amount of radiation that the test collector received. The dimmers are divided into six scales for producing different amount of radiation values. These values have been previously measured using the pyranometer. The measurement errors are about of 3.16 % for radiation value of 277.8 W/m^2 and 4.05 % for radiation value of 642 W/m^2. A heater is placed at the inlet of the collected undergoing test to vary the inlet temperature. The lighting control of the simulator is adjusted to obtain the required radiation levels. The solar collector is operated at varying inlet temperature, airflow rate, and radiation conditions. Air is circulated for thirty minutes prior to the period in which data are taken.

3 Theoretical Analysis

The schematic of the physical situation of interest is shown in Fig. 2. The energy balance for each part of the flat plate collector under steady-state heat flow are as follows:

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For the glass cover, g

$$h_1(T_g - T_a) + h_2(T_g - T_s) + h_3(T_g - T_{f1}) + h_4(T_g - T_{p1}) = \alpha_g I \quad (1)$$

For the static fluid, f_1

$$h_3(T_g - T_{f1}) = h_5(T_{f1} - T_{p1}) \quad (2)$$

For the absorber plate, p_1

$$\tau_g \alpha_{p1} I = h_4(T_g - T_{p1}) + h_5(T_{f1} - T_{p1}) = h_6(T_{p1} - T_{f2}) + h_7(T_{p1} - T_{p2}) \quad (3)$$

For the flowing air, f_2

$$h_6(T_{p1} - T_{f2}) = \frac{mC_{f2}}{B}\frac{dT_{f2}}{dx} + h_8(T_{f2} - T_{p2}) \quad (4)$$

For the back plate, p_2

$$h_8(T_{f2} - T_{p2}) + h_7(T_{p1} - T_{p2}) = U_r(T_{p2} - T_a) \quad (5)$$

Equation (1) to (5) have been combined to get this differential equation:

$$\frac{dT_{f2}}{dx} + \alpha T_{f2} = \beta \quad (6)$$

where the general solution is:

$$T_{f2} = \beta/\alpha + \gamma \exp(-\alpha x) \quad (7)$$

with boundary conditions.

$$T_{f2}(x = 0) = T_a = T_i \quad (8)$$

$$\gamma = T_i - \beta/\alpha \quad (9)$$

The final solution can written as:

$$T_{f2} = \beta/\alpha + (T_i - \beta/\alpha)\exp(-\alpha x) \quad (10)$$

where α and β are values are shown in the Appendix A. The output temperature can be obtained by replacing x with L (x = L). The Charters correlation for fully developed flow is used to obtain the heat transfer convection constant, h.

$$Nu = 0.0158\,\mathrm{Re}^{0.8} \quad (11)$$

$$h = NuD_h/k \quad (12)$$

where D_h= hydraulic diameter, m and k = fluid thermal conductivity, $Wm^{-1}\,^oC^{-1}$ The Reynolds Number can be obtained by equation:

$$\mathrm{Re} = mD_h/A\mu \quad (13)$$

Where ρ = fluid density, kgm^{-1}, v = airflow velocity, ms^{-1}, m = airflow rate, kgs^{-1}, A = cross section area of input air, m^2 and μ = dynamic viscosity, $kgm^{-1}s^{-1}$

A constant **K** has been added to the Charter Correlation to differentiate the Nusselt Number for different kind of absorber, which is:

$$Nu = 0.0158K\,\mathrm{Re}^{0.8} \quad (14)$$

The collector's efficiency, η can be attained by this equation:

$$\eta = \frac{mC(T_o - T_i)}{A_C I} \quad (15)$$

(8)

Where: C = air specific heat capacity, $kJkg^{-1}\,^oC^{-1}$, T_o = output temperature, ^{o} C. T_i = input temperature, ^{o}C, A_c = collector's opening area, m^2 and I = solar intensity, Wm^{-2}

4 Results And Discussion

Equations (1) to (5) have been combined to obtain a mathematical modeling where equations A_1 to A_{23}, β and α can be referred in the Appendix. Assuming the following parameters' value are; τ_g=0.9, α_{p1}=0.9, α_g=0.06 and U= 0.692 W/m^2k. In equation (1), h_1 and h_2 can be combined to form the overall heat transfer constant from the glass cover to the surrounding. Here, we assumed the value is 19 W/m^2K and others h constant are 6 W/m^2K. The values are fixed by presumptuous the sun shone at the collector uniformly. From equation (14), the collector's efficiency with V-groove absorber plate theoretically, for different K values. Fig. 3 shows the relation between η and I for different K values. In this calculation, two values of flow rate; 6.8 m/s and 4.25 m/s have been used as in the experiment. Based from this graphs, the efficiency from the experimental data is close to the mathematical modeling that has been built with K=2.3. From this value, the collector's efficiency with the V-groove absorber is 7.4% higher than the collector with the flat plate absorber. This efficiency value depends on **h**, which has been obtained by **K**.

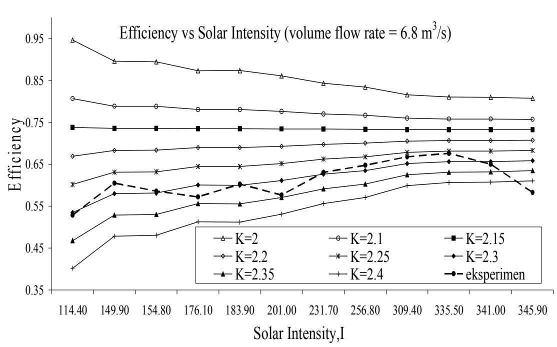

Fig. 3(a): Relationship between collector's efficiency and solar intensity with different K values (volume flow rate = 6.8 m^3/s)

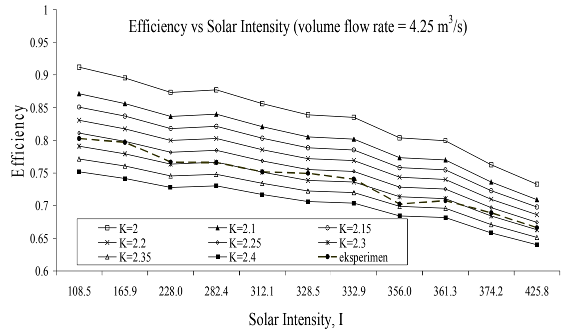

Fig. 3(b): Relationship between collector's efficiency and solar intensity with different K values (volume flow rate = 4.25 m^3/s)

5 Conclusion

This study shown that V-groove collector is 7.4% more efficient than flat plate collectors, which are widely used nowadays. Analytical modeling has been done to obtain the efficiency of the collector by changing the **K** value (equation (14). This value can assist to get the convection heat transfer coefficient for different kind of absorber. From this mathematical modeling, the output temperature not only depends on the h value but also other parameters such as collector's length, depth, flow rate, etc.

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APPENDIX

$\alpha = A_{11} - A_{12}A_{21} - A_{13}A_{23}$
$\beta = A_{12}A_{22} + A_{13}A_{24}$
$A_1 = h_3/a$
$A_2 = h_4 / a$
$A_3 = (\alpha_g I + h_1 T_a) / a$
$A_4 = h_3 / (h_3 + h_5)$
$A_5 = h_5 / (h_3 + h_5)$
$A_6 = \tau_g \alpha_{p1} I / b$
$A_7 = h_4 / b$
$A_8 = h_5 / bA_9 = h_6 / b$
$A_{10}= h_7 / b$
$A_{11}= (h_8 + h_6) / c$
$A_{12}= h_6 / c$
$A_{13}= h_8 / c$
$A_{14}= h_8 / d$
$A_{15}= h_7 / d$
$A_{16}= U_r / d$
$A_{17}= (A_4 A_2 + A_5) / (1- A_4 A_1)$
$A_{18}= A_4 A_3 / (1- A_4 A_1)$
$A_{19}= A_2 + A_1A_{17}$
$A_{20}= A_3 + A_1A_{18}$
$A_{21}= (A_9 + A_{10}A_{14}) / e$
$A_{22}= (A_6 + A_7A_{20} + A_8A_{18} + A_{10}A_{16}) / e$
$A_{23}= A_{14} + A_{15}A_{21}$
$A_{24}= A_{16} + A_{15}A_{22}$
$a = h_1+h_2+h_3$
$b = h_4 + h_5 + h_6 + h_7$
$c = mC_{f2} / B$
$d = U_r + h_8 + h_7$
$e = 1- (A_7A_{19} + A_8A_{17} + A_{10}A_{15})$

NOMENCLATURE

$h_1 = h_{cgw}$= convection heat transfer coefficient between the glass cover and surrounding ($Wm^{-2}K^{-1}$)
$h_2 = h_{rgs}$= radiation heat transfer coefficient between the glass cover and surrounding ($Wm^{-2}K^{-1}$)
$h_3 = h_{cgf1}$= convection heat transfer coefficient between the glass cover and static air ($Wm^{-2}K^{-1}$)

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h_4 = h_{rgp1}= radiation heat transfer coefficient between the glass cover and absorber plate ($Wm^{-2}K^{-1}$)

h_5 = h_{cf1p1}= convection heat transfer coefficient between the static air and absorber plate ($Wm^{-2}K^{-1}$)

h_6 = h_{cp1f2} = convection heat transfer coefficient between the absorber plate and working fluid ($Wm^{-2}K^{-1}$)

h_7 = h_{rp1p2}= radiation heat transfer coefficient between the absorber plate and back plate ($Wm^{-2}K^{-1}$)

h_8 = h_{cf2p2} = convection heat transfer coefficient between the working fluid and back plate ($Wm^{-2}K^{-1}$)