THEORETICAL ANALYSIS OF COLLECTION CHARACTERISTICS OF A SOLAR AIR HEATER WITH A BUILT-IN NET

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The collection characteristics of a solar air heater with a built-in net, in which a metallic net was inserted parallel in the space between the transparent cover plate and the absorber, were theoretically analyzed. The working fluid (air), which entered the solar air heater at a uniform inlet velocity and a constant inlet temperature, was divided into two layers (upper and lower layer) by the metallic net. Solar radiation impinges uniformly from above, passing through the transparent cover plate and being absorbed partly by the net and partly by the absorber. For the present solar air heater, the continuity, momentum and energy equations were derived under several assumptions and numerically solved. The calculations were performed for the cases of eight kinds of metallic nets which had different transmissivities, under different combinations of mass flow rate of air and incident solar radiation. Also, the cases of a conventional solar air heater and a solar air heater without a net were theoretically analyzed. The calculated results were compared with the experimental data reported in a previous paper, and the effects of the net transmissivity, net setting position and solar air heater length on temperature distributions within the air layer and the collection efficiency were examined.

The calculated results show fairly well the tendencies of the experimental data, and the several assumptions made and the calculation method used in the theoretical analysis are found to be valid.

Introduction

In order to improve the collection characteristics of the conventional type solar air heaters, we have previously proposed a solar air heater with a built-in net, which consisted of a top transparent plate, a transparent working fluid layer (air), an absorber, and a metallic net inserted parallel in the space between the transparent plate and absorber. This solar air heater is considered to be able to change the portion of the radiant-energy absorption within the air layer and to enhance the heat transfer to the air. The collection characteristics of the solar air heater with a built-in net, the solar air heater without a net which we called VHT (Volume Heat Trap) and the conventional solar air heater which we called SHT (Surface Heat Trap) were experimentally studied in the indoor experiments (Hido *et al.*, 1995). From the experimental results, it was clarified that when the net was inserted inside the solar air heater, the temperature distribution within the air layer was quite different from those of VHT and SHT types solar air heaters. The collection efficiency of the solar air heater with a built-in net was 16 to 35 % higher than those of VHT and SHT types solar air heaters. In addition, the collection efficiency had a maximum when the net transmissivity was about 0.60.

There have been several studies on theoretical analysis of one-pass conventional solar air heaters (Close, 1963, Gupta and Garg, 1967, Parker, 1981 and Bhargava *et al.*, 1982) or two-pass conventional solar air heaters (Wijeyesundera *et al.*, 1982 and Bhargava *et al.*, 1983). All

these studies have been restricted to calculations of the temperature distribution in the flow direction, the outlet temperature of air and the collection efficiency, while the temperature distributions in the depth direction have not been analyzed.

In this paper, two dimensional velocity and temperature fields in the present solar air heater were considered, and the basic equations were derived and numerically solved for the cases of nets with various transmissivities, under different combinations of the mass flow rate of air and the incident radiant heat flux. The cases of the solar air heater without a net VHT and the conventional solar air heater SHT were also theoretically analyzed. The calculated results were compared with the experimental data (Hido *et al.*, 1995), and the effects of the net transmissivity, net setting position and solar air heater length on the temperature distributions within the air layer and the collection efficiency were discussed.

1. Structures of Solar Air Heaters Studied

The structures of the solar air heaters discussed in this paper are shown in **Fig. 1**. They are different from each other in the position of the absorber and number of air layers. In the conventional type solar air heater SHT, the absorber rear surface of which is blackened is set at the top surface of the air layer and there is only one air layer below the absorber. In the present type of solar air heater, a transparent plate is set at the top surface of the air layer, the absorber is set at the bottom of the air layer, a metallic net is inserted parallel in the space between the transparent

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Conventional type

Present type

Fig. 1 Structures of solar air heaters studied

plate and absorber, and there are two air layers, one above and another below the net. In the solar air heater without a net VHT, the structure is the same as in the present type, but there is no net inserted inside it.

2. Theoretical Analysis

2.1 Basic equations

The physical model for theoretical analysis of the solar air heater with a built-in net is shown in **Fig. 2**. The present solar air heater consists a top transparent plate, a transparent working fluid layer (air), a metallic net, an absorber, and an insulating layer, and is uniformly irradiated from above by a constant solar radiation ($I_o = \int_0^\infty I_{o\lambda} d\lambda$). The fluid layer is divided into two layers (upper and lower layer) by the net, and the air is flowing at a uniform inlet velocity in the x direction in both layers.

For the theoretical analysis we assume that (1) the conductive heat transfer is two-dimensional, (2) the incident solar radiation is collimate and the radiative heat transfer due to solar radiation is one-dimensional, (3) the momentum transfer is two-dimensional, (4) the physical properties are constant, (5) the flow is a steady laminar flow and natural convection is neglected, (6) the net is a very thin plate which is not permeable to air (confirmed as shown in the previous paper) and transmits uniformly the radiation according to the transmissivity, and (7) the transparent plate, net and absorber are black bodies for the long wavelength radiation.

In this model, the velocity distributions are not affected by the temperature distributions and are obtained independently by solving the well-known two-dimensional continuity and momentum equations, which are not described here. For the solar air heater with a built-in net, the energy equations for its each component are expressed as follows :

Fig. 2 Physical model

Transparent plate:

$$k_T (\partial^2 T_T / \partial x^2 + \partial^2 T_T / \partial y^2) - \partial q_s(y) / \partial y = 0 \quad (1)$$

Upper fluid layer:

$$C_{pF} \rho_F \{ u_1 (\partial T_{F1} / \partial x) + v_1 (\partial T_{F1} / \partial y) \} = k_F (\partial^2 T_{F1} / \partial x^2 + \partial^2 T_{F1} / \partial y^2) \quad (2)$$

Lower fluid layer:

$$C_{pF} \rho_F \{ u_2 (\partial T_{F2} / \partial x) + v_2 (\partial T_{F2} / \partial y) \} = k_F (\partial^2 T_{F2} / \partial x^2 + \partial^2 T_{F2} / \partial y^2) \quad (3)$$

The inlet and outlet conditions are given as follows:

$$x = 0, 0 \leq y \leq D_T; \quad \partial T_T / \partial x = 0 \quad (4)$$

$$x = 0, D_T \leq y \leq D_T + D_{F1}; \quad T_{F1} = T_{in} \quad (5)$$

$$x = 0, D_T + D_{F1} \leq y \leq D_T + D_{F1} + D_{F2}; \quad T_{F2} = T_{in} \quad (6)$$

$$x = L, 0 \leq y \leq D_T; \quad \partial T_T / \partial x = 0 \quad (7)$$

$$x = L, D_T \leq y \leq D_T + D_{F1}; \quad \partial^2 T_{F1} / \partial x^2 = 0 \quad (8)$$

$$x = L, D_T + D_{F1} \leq y \leq D_T + D_{F1} + D_{F2}; \quad \partial^2 T_{F2} / \partial x^2 = 0 \quad (9)$$

Neglecting the radiation from the side walls, and inlet and outlet cross sections the boundary conditions are given as follows:

$$y = 0, 0 \leq x \leq L; \quad -h_c (T_T - T_a) - h_r (T_T - T_a) = -k_T (\partial T_T / \partial y) \quad (10)$$

$$y = D_T, 0 \leq x \leq L; \quad -k_T (\partial T_T / \partial y) - k_F (\partial T_{F1} / \partial y) + q_{dT-N} + q_{dT-A} \quad (11)$$

$$y = D_T + D_{F1}, 0 \leq x \leq L; \quad -k_F (\partial T_{F1} / \partial y) + (1 - Tr) q_s(D_T) - q_{dN-T} = -k_F (\partial T_{F2} / \partial y) + q_{dN-A} \quad (12)$$

$$y = D_T + D_{F1} + D_{F2}, 0 \leq x \leq L; \quad -k_F (\partial T_{F2} / \partial y) + Tr q_s(D_T) - q_{dA-N} - q_{dA-T} = h_L (T_A - T_a) \quad (13)$$

Here, h_r is the radiative heat transfer coefficient on the solar air heater surface, $h_r = \sigma [T_T^2 + T_a^2] [T_T + T_a]$, h_c and h_L are the convective heat transfer coefficient on the solar air

Table 1 Multibanded values of radiation properties

$\lambda_i - \lambda_{i+1}$ [mm]	η_i [-]	α_i [m ⁻¹]
0.3 - 0.4	0.002	683.277
0.4 - 0.5	0.045	4.382
0.5 - 0.6	0.117	2.744
0.6 - 0.7	0.182	1.122
0.7 - 0.8	0.136	0.042
0.8 - 0.9	0.106	2.208
0.9 - 1.0	0.083	6.027
1.0 - 1.1	0.047	6.027
1.1 - 1.2	0.045	48.807
1.2 - 1.3	0.043	14.284
1.3 - 1.4	0.017	68.873
1.4 - 1.5	0.034	66.394
1.5 - 1.6	0.037	26.099
1.6 - 1.7	0.023	223.322
1.7 - 1.8	0.011	342.276
1.8 - 1.9	0.009	177.267
1.9 - 2.0	0.013	239.684
2.0 - 2.1	0.017	179.602
2.1 - 2.2	0.016	544.790
2.2 - 2.3	0.010	1677.280

heater surface and overall heat transfer coefficient for the heat loss from the absorber to the ambient air, respectively.

$q_s(y)$ in Eq. (1) is the radiant heat flux within the transparent plate based on the solar radiation. q_{dT-N} in Eq. (11) means the net radiation interchange between a differential area of the transparent plate and a finite area of the net. The other q 's with suffix in Eqs. (11) to (13) are defined in the same way. These net radiation interchanges are expressed as shown in **Appendix**.

For the solar air heater without a net VHT, there is only one fluid layer (upper fluid layer), and so Eqs. (1) and (2) are the same, while Eq. (3) is not applicable. Also the inlet and the outlet conditions are the same, while the boundary conditions are given as follows:

$$y = D_T, 0 \leq x \leq L; \\ -k_T(\partial T_T / \partial y) = -k_F(\partial T_{F1} / \partial y) + q_{dT-A} \quad (14)$$

$$y = D_T + D_{F1}, 0 \leq x \leq L; \\ -k_F(\partial T_{F1} / \partial y) + q_s(D_T) - q_{dA-T} = h_L(T_A - T_a) \quad (15)$$

For the conventional solar air heater SHT, the same equations and inlet and outlet conditions as for the solar air heater without a net are applicable. There are some changes in the boundary conditions, which are given as follows:

$$y = D_T, 0 \leq x \leq L; \\ -k_T(\partial T_T / \partial y) = -k_F(\partial T_{F1} / \partial y) + q_s(D_T) + q_{dT-A} \quad (16)$$

$$y = D_T + D_{F1}, 0 \leq x \leq L; \\ -k_F(\partial T_{F1} / \partial y) - q_{dA-T} = h_L(T_A - T_a) \quad (17)$$

The collection efficiency is defined as the ratio of the fluid enthalpy increment inside the solar air heater to the solar radiation, and it is calculated by the equation:

$$Ef = C_{pF} [G_1(T_{M1} - T_{in}) + G_2(T_{M2} - T_{in})] / (I_o A) \quad (18)$$

Here, G_1 and G_2 are the mass flow rates of the air in the upper and lower fluid layers respectively. T_{M1} and T_{M2} are the outlet mixing cup temperatures of the air in the upper

Table 2 Physical properties and parameters used for calculations

C_{pF} [J·kg ⁻¹ ·K ⁻¹]	1007	L [m]	0.525	1.050
k_T [W·m ⁻¹ ·K ⁻¹]	0.2	D_T [m]		0.003
k_F [W·m ⁻¹ ·K ⁻¹]	0.027	D_{F1} [m]		0.010
ρ_F [kg·m ⁻³]	1.128	D_{F2} [m]		0.010
μ_F [kg·m ⁻¹ ·s ⁻¹]	18.9×10^{-6}	D [m]		0.010
I_o [W·m ⁻²]	440 630 880	Δy_T [m]		0.0006
G_1 [kg·s ⁻¹]	0.35 1.05 1.8×10^{-3}	Δy_{F1} [m]		0.0005
G_2 [kg·s ⁻¹]	0.35 1.0 1.8×10^{-3}	Δy_{F2} [m]		0.0005
T_{in} [K]	298	Δx [m]		0.005
T_r [-]	0.00 0.21 0.34 0.48			
	0.60 0.71 0.78 0.86			

and lower fluid layers respectively. T_{in} is the inlet temperature of the air in both fluid layers.

2.2 Numerical method

The continuity and momentum equations were first transformed by using of the vorticity and stream function, and then were numerically solved together with their boundary conditions by the finite-difference method and the iterative numerical procedure to obtain the velocity distribution within the fluid layer. In the calculations, a solar air heater with the length of the test section longer than the real length L , was assumed, and the assumed length was long enough for the velocity to become developed. From the obtained results, only the data within the real length of the test section L , were used in the energy equations (2) and (3). Equations (1) to (3) were numerically solved together with their boundary conditions by the finite-difference method to obtain the temperature distributions within the transparent plate, the upper fluid layer, and the lower fluid layer, respectively. In the calculations the iterative numerical procedure was used.

3. Results and Discussion

3.1 Characteristic values used for calculation

Numerical analysis was performed for the same conditions as the previous paper (Hido *et al.*, 1995). The transparent plate, the metallic nets, the absorber and the insulating layer were consisted of an acrylic resin plate, stainless steel nets, a copper plate and a polystyrene foam layer, respectively. The nets and the absorber were coated with black paint. As the solar simulator were used twelve halogen lamps (Philips Ltd., quartz halogen dichroic mirror lamp, 17 V - 150 W). The characteristic values, which were used in the calculations, were estimated in the following manner:

$q_s(y)$ was calculated by applying a multiband model for the radiation properties (Viskanta and Anderson, 1975). The spectral power distribution of the halogen lamp on the solar air heater surface and the absorption coefficient of the acrylic resin plate were multibanded with the band interval of $0.1 \mu\text{m}$ at the effective range of wavelength of $0.3 - 2.3 \mu\text{m}$ (Bando *et al.*, 1986, 1989). In the wavelength band $\lambda_i - \lambda_{i+1}$, the ratio of radiation energy to total energy and the mean absorption coefficient are noted by η_i and α_i , respectively, as listed in **Table 1**. The radiant heat flux within the transparent plate, $q_s(y)$ was expressed by using

Fig. 3 Temperature distributions of flow direction for three types of solar air heaters ($G = 0.7 \times 10^{-3} \text{ kg} \cdot \text{s}^{-1}$, $I_o = 880 \text{ W} \cdot \text{m}^{-2}$)

Fig. 4 Outlet temperature distributions at various mass flow rates for three types of solar air heaters ($x/L = 1.0$, $I_o = 880 \text{ W} \cdot \text{m}^{-2}$)

an assumption of “Cold Semitransparent Media” (the internal emission can be neglected) (Bando *et al.*, 1989).

The values of the physical properties and several parameters used in the calculations are shown in **Table 2** (for the case of setting the net at the middle horizontal plane of the air layer depth). The calculations were performed for the cases of eight kinds of stainless steel nets with various transmissivities Tr , whose values are also listed in Table 2. The values of the ambient and inlet temperatures T_a and T_{in} were taken as 298 K on the basis of the experimental conditions of the previous paper.

In the calculation, the convective heat transfer coefficient on the solar air heater surface was estimated from the equation: $h_c = 2.86 [T_T(x) - T_a]^{1/4}$, which was derived from the correlation for the heat transfer by natural convection from horizontal surfaces facing upward (Al-Arabi and El-Riedy, 1976). The value of the overall heat transfer coefficient for the heat loss from absorber to the ambient h_L was taken as $0.7 \text{ W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$ on the basis of the structure of insulation.

Fig. 5 Comparison of outlet temperature distributions for all cases

The transparent plate and the air layers were divided into equal subdivisions. The number of subdivisions in the x directions was taken as 105 ($L = 0.525 \text{ m}$, $\Delta x = 0.005 \text{ m}$) or 210 ($L = 1.05 \text{ m}$, $\Delta x = 0.005 \text{ m}$) for all the cases. The number of subdivisions in the y direction of the transparent plate was taken as 5 ($\Delta y_T = 0.0006 \text{ m}$) for all the cases, and that of each air layer was taken as 40 ($\Delta y_{F1} = 0.0005 \text{ m}$) for the cases of SHT and VHT and 20 ($\Delta y_{F1} = \Delta y_{F2} = 0.0005 \text{ m}$) for the case of solar air heater with a built-in net.

In most of calculations, the net setting position was the middle horizontal plane of the air layer depth, i. e. $(y - D_T) / (D_{F1} + D_{F2}) = 0.50$, and in this case $D_{F1} = D_{F2} = 0.010 \text{ m}$. In a part of calculations, the net setting position was changed to a quarter and three quarters horizontal plane of the air layer depth, i. e. $(y - D_T) / (D_{F1} + D_{F2}) = 0.25$ and $(y - D_T) / (D_{F1} + D_{F2}) = 0.75$, respectively. In the former case, $D_{F1} = 0.005 \text{ m}$ ($\Delta y_{F1} = 0.00025 \text{ m}$) and $D_{F2} = 0.015 \text{ m}$ ($\Delta y_{F2} = 0.0005 \text{ m}$), while in the later case it is vice versa.

In the iterations, the starting value for the velocity was the inlet velocity, and the starting value for the temperature was an initial temperature which was decided from the experimental data. The iterations were terminated when the following conditions were satisfied: $|u^{n+1} - u^n| / u^n \leq \epsilon_u$, $|v^{n+1} - v^n| / v^n \leq \epsilon_v$, $|T^{n+1} - T^n| / T^n \leq \epsilon_T$ where, ϵ_u , ϵ_v , and ϵ_T are the predetermined error values, and $\epsilon_u = \epsilon_v = \epsilon_T = 10^{-5}$.

3.2 Temperature distribution

In **Figs. 3 to 5** are shown comparisons of the calculated and experimental temperature distributions.

Figures 3 and 4 show a comparison of the temperature distributions within the air layer in the depth direction in the case of the conventional solar air heater (SHT), the solar air heater without a net (VHT), and the solar air heater with a built-in net ($Tr = 0.60$) at three different points along the solar air heater and at three different mass flow rates

Fig. 6 Effect of net setting position on outlet temperature distribution at various mass flow rates ($x/L = 1.0$, $I_o = 880 \text{ W} \cdot \text{m}^{-2}$)

of the air, respectively. A comparison of the outlet temperature distributions within the air layer for all the cases (SHT, VHT, $Tr = 0.0$ to 0.86) is shown in Fig. 5. The calculated results show fairly well the tendencies of the experimental data* shown by symbols. However, there are some discrepancies between both results which could be considered to be attributed mainly to the heat transfer enhancement by the flow disturbances on the net surfaces, the natural convection, the reflections from the net surfaces, and the temperature dependency of the fluid physical properties. The effects of these factors were neglected in the present model (assumptions (4) to (7)). In the case of VHT, a discrepancy between both results, especially at low mass flow rate of the air, may be considered to be mainly due to the natural convection. Comparing the results of the solar air heaters with a built-in net in Fig. 5, it is seen that the discrepancy between both results is the smallest in the case of $Tr = 0.0$ (blackened plate). Since there is no flow disturbances in this case, the discrepancy is mainly attributed to the reflections. In the cases of $Tr = 0.48$ to 0.86 , the discrepancies between both results are larger because of the stronger flow disturbances on the net surfaces (judging from the effect of the net transmissivity on the pressure drop as shown in the previous paper). The flow disturbances are considered to decrease with decreasing of the mass flow rate of the air while the reflections are independent of it. When the ratio of bulk to wall temperature is small, for example in the case of high mass flow rate in VHT, the temperature dependency of the fluid physical properties is considered to become significant.

On the basis of the calculated and experimental results, the temperature distributions of three types of solar air heaters are examined below. From Fig. 3, it is seen that

* The temperatures within the solar air heaters were measured by using $200 \mu\text{m} \phi$ K-type thermocouples inserted sideways, whose positions were shown in detail in the previous paper. The effect of the incident solar radiation on the accuracy of the temperature measurements was clarified to be not significant (about 0.2 K) by observing the output response of the thermocouples when the halogen lamps were suddenly turned off.

Fig. 7 Effect of net transmissivity on collection efficiency

in each type of solar air heater the temperature level becomes higher with increasing of distance from the inlet. In the case of SHT, the incident radiation is almost absorbed at the absorber (the absorptivity of the blackened rear surface = 0.95) which is set at the top surface of the air layer, and as a result the surface temperature $((y - D_T)/(D_{F1} + D_{F2}) = 0)$ is the highest, and the temperature level of the air layer is the lowest among three types of solar air heaters. In the case of VHT, a part of the incident radiation is absorbed by the transparent plate (reflectivity = 0.07 and absorptivity = 0.09 for solar radiation) and the greater part is absorbed at the absorber, and so the bottom temperature $((y - D_T)/(D_{F1} + D_{F2}) = 1)$ is the highest of the three types of solar air heaters. In the solar air heater with a built-in net, the radiation passing through the transparent plate is partly absorbed by the net and the other part reaches the absorber. Therefore, the air is also heated by convective heat transfer from the net surfaces. In the case of $Tr = 0.60$, the temperature distribution within the air layer is different from those of the other two types and the temperature level is higher.

In Fig. 4, it is seen that the pattern of temperature distribution is scarcely affected by the mass flow rate of the air in each type of solar air heater.

It is also seen in Fig. 5 that as the net transmissivity decreases, the net temperature increases, while the absorber temperature decreases.

Figure 6 shows the effect of the net setting position on the outlet temperature distributions within the air layer in the case of $Tr = 0.60$. In these cases, since the resistance to flow in each air layer is difficult to estimate, the calculations were performed with an approximation of the same inlet velocity in both air layers. In spite of this approximation, the tendency of the calculated results is similar to that of the experimental data.

3.3 Collection efficiency

The plots of the collection efficiency against the net transmissivity, Tr are shown in **Fig. 7**. The net setting position was the middle horizontal plane of the air layer depth, i. e. $(y - D_T)/(D_{F1} + D_{F2}) = 0.50$. The calculated results show the tendencies of the experimental data fairly well. The calculation results show that the collection effi-

Fig. 8 Effect of net setting position on collection efficiency ($I_o = 880 \text{ W} \cdot \text{m}^{-2}$)

ciency increases with decreasing the incident radiant heat flux and with increasing the mass flow rate of the air. In both results, the collection efficiency has a maximum when the net transmissivity is about 0.60.

The effect of the net setting position on the collection efficiency for the cases of $Tr = 0.48$ and $Tr = 0.60$ is shown in Fig. 8. Though the collection efficiency is hardly affected by the net setting position, it is seen that the collection efficiency of the case of $(y - D_T) / (D_{F1} + D_{F2}) = 0.50$ is a little higher than those of the cases of $(y - D_T) / (D_{F1} + D_{F2}) = 0.25$ and 0.75 .

Figure 9 shows the effect of the solar air heater length on the collection efficiency for the cases of VHT, $Tr = 0.0$ to 0.86 . From the calculation results, it is seen that the longer solar air heater has a lower collection efficiency than the shorter one, because of larger heat loss from the transparent plate to the ambient.

The above results show that the present physical model predicted well the tendencies of the experimental data, especially the collection efficiency. However, we think that in the future it is necessary to analyze the characteristics of the proposed solar air heater taking into consideration of the flow disturbances on the net surface.

Conclusion

The collection characteristics of the solar air heater with a built-in net, in which a metallic net was inserted parallel in the space between the transparent cover plate and absorber, were theoretically analyzed. To analyze this system, the governing equations for velocity and temperature distributions were derived under several assumptions and numerically solved by the finite-difference method. The calculations were performed for the cases of eight kinds of

Fig. 9 Effect of solar air heater length on collection efficiency

metallic nets which had different transmissivities. Also, the cases of the conventional solar air heater and the solar air heater without a net were theoretically analyzed. The calculated results were compared with the experimental data. The effects of the net transmissivity, net setting position and solar air heater length on the temperature distributions within the air layer and the collection efficiency were examined.

The calculated results match the tendencies of the experimental data fairly well. The several assumptions and the calculation method in the theoretical analysis are thought to be valid.

Appendix

Since the ratio of width to depth in the present solar air heater was sufficiently large, the equation of the configuration factor between a differential area and a frontal finite area for a system of two parallel plates is applicable (Siegel and Howell, 1981). A frontal finite length $2Lc$ (symmetric to the differential area) is taken, and by using it, the configuration factor, $\zeta(X_i)$ is expressed by the following equation: $\zeta(X_i) = X_i / \sqrt{1 + X_i^2}$, $X_i = Lc/D_{Fi}$ ($i = 1, 2$). In the calculation, the values of Lc were taken as 0.03, 0.045, and 0.06 m for the case of $D_{F1} = 0.01, 0.005$ (0.015) and 0.02 (VHT and SHT), respectively. In all these cases, the configuration factors were larger than 0.95 and so were considered to be approximately unity. In the zones of $0 \leq x \leq Lc$ and $L-Lc \leq x \leq L$, the net radiation interchanges are different from the remainder zone $Lc \leq x \leq L-Lc$, because there is no net in $x \leq 0$ and $x \geq L$ and there are some direct radiation interchanges between the transparent plate and absorber, without passing through the net. Considering this situation, the net radiation interchanges for the zone $0 \leq x \leq Lc$ are given as follows:

$$\begin{aligned} q_{dT-N} &= 0.5(1-Tr) \sigma [T_T^4 - (T_N^4)_{av}] + \zeta(X_1')(1-Tr) \sigma [T_T^4 - (T_N^4)_{av}] \\ q_{dT-A} &= 0.5 Tr \sigma [T_T^4 - (T_A^4)_{av}] + \zeta(X_1') Tr \sigma [T_T^4 - (T_A^4)_{av}] \\ &\quad + [0.5 - \zeta(X_1')] \sigma [T_T^4 - T_A^4(0)] \\ q_{dN-T} &= 0.5(1-Tr) \sigma [T_N^4 - (T_T^4)_{av}] + \zeta(X_1')(1-Tr) \sigma [T_N^4 - (T_T^4)_{av}] \\ &\quad + [0.5 - \zeta(X_1')] (1-Tr) \sigma [T_N^4 - T_T^4(0)] \end{aligned} \quad (A-1)$$

$$q_{dN-A} = 0.5(1-Tr) \sigma [T_N^4 - (T_A^4)_{av}] + \zeta(X_2')(1-Tr) \sigma [T_N^4 - (T_A^4)'_{av}] \\ + [0.5 - \zeta(X_2')](1-Tr) \sigma [T_N^4 - T_A^4(0)] \\ q_{dA-T} = 0.5 Tr \sigma [T_A^4 - (T_T^4)_{av}] + \zeta(X_2') Tr \sigma [T_A^4 - (T_T^4)'_{av}] \\ + [0.5 - \zeta(X_2')](1-Tr) \sigma [T_A^4 - T_T^4(0)] \\ q_{dA-N} = 0.5(1-Tr) \sigma [T_A^4 - (T_N^4)_{av}] + \zeta(X_2')(1-Tr) \sigma [T_A^4 - (T_N^4)'_{av}]$$

Here, $(T_T^4)_{av}$, $(T_N^4)_{av}$ and $(T_A^4)_{av}$ are the average fourth power of temperatures from x to $x + Lc$ of the transparent plate, metallic net, and absorber, respectively. $(T_T^4)'_{av}$, $(T_N^4)'_{av}$ and $(T_A^4)'_{av}$ are the average fourth power of temperatures from 0 to x of the transparent plate, metallic net, and absorber, respectively. They are expressed by the equations:

$$(T^4)_{av} = \int_x^{x+Lc} T^4(x) dx / Lc \quad (A-2)$$

$$(T^4)'_{av} = \int_0^x T^4(x) dx / x \quad (A-3)$$

$T_T(0)$ and $T_A(0)$ are the temperatures of the transparent plate and the absorber at the inlet of solar air heater. $\zeta(X_i')$ is the configuration factor for this zone, which is expressed by the following equation: $\zeta(X_i') = K_i X_i' / \sqrt{1 + X_i'^2}$, $X_i' = x/D_{Fi}$, ($i = 1, 2$), K_i ($i = 1, 2$), is a modification factor, the values of which are 1.05 and 1.005 for the case of $D_{F1} = D_{F2} = 0.01$ m and $D_{F1} = 0.005$ m ($D_{F2} = 0.015$ m), respectively. For the zone $Lc \leq x \leq L - Lc$, the net radiation interchanges are given as follows:

$$q_{dT-N} = (1-Tr) \sigma [T_T^4 - (T_N^4)_{av}], q_{dT-A} = Tr \sigma [T_T^4 - (T_A^4)_{av}], \\ q_{dN-T} = (1-Tr) \sigma [T_N^4 - (T_T^4)_{av}], q_{dN-A} = (1-Tr) \sigma [T_N^4 - (T_A^4)_{av}], \\ q_{dA-T} = Tr \sigma [T_A^4 - (T_T^4)_{av}], q_{dA-N} = (1-Tr) \sigma [T_A^4 - (T_N^4)_{av}] \quad (A-4)$$

Here, $(T_T^4)_{av}$, $(T_N^4)_{av}$ and $(T_A^4)_{av}$ are the average fourth power of temperatures within the length $2Lc$ of the transparent plate, metallic net, and absorber, respectively and are expressed by the equation:

$$(T^4)_{av} = \int_{x-Lc}^{x+Lc} T^4(x) dx / 2Lc \quad (A-5)$$

For the zone $L-Lc \leq x \leq L$, the net radiation interchanges are expressed in the same way as for the zone $0 \leq x \leq Lc$.

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Nomenclature

A	= irradiated surface area of test section	[m ²]
C_p	= specific heat of air	[J·kg ⁻¹ ·K ⁻¹]
D	= depth	[m]
E_f	= collection efficiency	[-]
G	= mass flow rate of air	[kg·s ⁻¹]
h_c	= convective heat transfer coefficient on solar air heater surface	[W·m ⁻² ·K ⁻¹]
h_L	= overall heat transfer coefficient for heat loss from absorber to ambient	[W·m ⁻² ·K ⁻¹]
h_r	= radiative heat transfer coefficient on solar air heater surface	[W·m ⁻² ·K ⁻¹]
I_o	= incident radiant heat flux	[W·m ⁻²]
k	= thermal conductivity	[W·m ⁻¹ ·K ⁻¹]
L	= length of test section	[m]
$q_s(y)$	= radiant heat flux	[W·m ⁻²]
q	= net radiation interchange between a differential area and a finite area	[W·m ⁻²]
T_a	= ambient temperature	[K]
T	= temperature	[K]
T_{in}	= inlet temperature of air	[K]
T_M	= outlet mixing cup temperature of air	[K]

T_r	= net transmissivity	[-]
u	= velocity vector in x direction	[m·s ⁻¹]
v	= velocity vector in y direction	[m·s ⁻¹]
x	= distance from inlet of solar air heater	[m]
y	= distance from upper surface of transparent plate	[m]
α_i	= mean absorption coefficient in a given wavelength of transparent plate	[m ⁻¹]
ε	= predetermined error value	[-]
η_i	= ratio of radiation energy in a given wavelength band to total energy	[-]
λ	= wavelength	[μm]
$\zeta(X)$	= configuration factor	[-]
ρ	= density	[kg·m ⁻³]
σ	= Stefan-Boltzmann constant	[W·m ⁻² ·K ⁻⁴]
μ	= viscosity	[kg·m ⁻¹ ·s ⁻¹]

<Subscripts>

A	= absorber
F	= working fluid
N	= metallic net
T	= transparent plate
1, 2	= upper and lower fluid layers

Literature Cited

- Al-Arabi, M. and M. K. El-Riedy; "Natural Convection Heat Transfer from Isothermal Horizontal Plates of Different Shapes," *Int. J. Heat Mass Transfer*, **19**, 1399-1404 (1976)
- Bando, Y., M. Nishimura, M. Kuraishi, T. Kasuga and M. Hasatani; "Effect of Optical Depth on Outdoor Performance of "Volume Heat-Trap" Type Solar Collector," *Heat Transfer Japanese Research*, **15**, 57-71 (1986)
- Bando, Y., M. Nishimura, M. Kuraishi, M. Masuda and H. Tajima; "Effect of Thermal and Optical Properties of Transparent Plates on the Collection Characteristics of "Volume Heat-Trap" Solar Collectors," *Heat Transfer Japanese Research*, **18**, 54-67 (1989)
- Bando, Y., M. Nishimura, M. Kuraishi and M. Masuda; "Collection Characteristics of a "Volume Heat-Trap" Type Multipass Solar Collector-Effect of Flow Mode and Partitioning Plate," *Heat Transfer Japanese Research*, **18**, 88-100 (1989)
- Bando, Y., M. Nishimura, K. Toyoda and M. Kuraishi; "Collection Characteristics of Composite Solar Collector with Volume Heat-Trap and Surface Heat Trap," *J. Chem. Eng. Japan*, **22**, 665-669 (1989)
- Bhargava, A. K., H. P. Garg and V. K. Sharma; "Evaluation of the performance of Air Heaters of Conventional Designs," *Solar Energy*, **29**, 523-533 (1982)
- Bhargava, A. K., H. P. Garg and V. K. Sharma; "A Two-Pass Solar Air Heater," *Energy*, **8**, 267-276 (1983)
- Close, D. J.; "Solar Air Heaters for Low and Moderate Temperature Applications," *Solar Energy*, **7**, 117-124 (1963)
- Gupta, C. L. and H. P. Garg; "Performance Studies on Solar Air Heaters," *Solar Energy*, **11**, 25-31 (1967)
- Hido, E. M., Y. Bando and M. Nishimura; "Collection Characteristics of a Solar Air Heater with Built-In Net," *J. Chem. Eng. Japan*, **28**, 340-345 (1995)
- Parker, B. F.; "Derivation of Efficiency and Heat Loss Factors for Solar Air Heaters," *Solar Energy*, **26**, 27-32 (1981)
- Siegel, R. and J. R. Howell; *Thermal Radiation Heat Transfer*, 822-831, Hemisphere Pub. Co. (1981)
- Viskanta, R. and E. E. Anderson; "Heat Transfer in Semitransparent Solids," *Advances in Heat Transfer*, **11**, 317-441, Academic Press (1975)
- Wijesundera, N. E., L. L. Ah and L. E. Tijoe; "Thermal Performance Study of Two-Pass Solar Air Heaters," *Solar Energy*, **28**, 363-370 (1982)