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Performance of Single Pass Down Stream Solar Air Collector with Inclined Multiple V-Ribs

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Abstract**—**Solar air heater is a type of heat exchanger which transforms solar radiation into heat energy. The thermal performance of conventional solar air heater has been found to be poor because of the low convective heat transfer coefficient from the absorber plate to the air. It is attributed to the formation of a very thin boundary layer at the absorber plate surface commonly known as viscous sub-layer. Thermal efficiency of solar air heater can be improved by providing the artificial roughness on absorber plate is the most efficient technique. In this paper an attempt is made to provide artificial roughness by incorporating inclined multiple V-ribs in the underside of the absorber plate. 60˚V – ribs are arranged inclined to the direction of air flow. Performance of collector estimated theoretically and experimentally. Results of the investigation reveal that thermal efficiency of collector with multiple V-ribs increased by 14%.

Keywords**—**Artificial roughness, inclined multiple V-ribs, performance, Solar air collector.

I. Introduction

SOLAR energy is the ultimate renewable energy. Clean, plentiful and thanks to today's technologies, easy to harvest. Every single day enough solar energy strikes the planet to meet the world's energy needs for four to five years. In an era where global warming and CO_2 buildup are of critical concern, solar energy can become an incredibly valuable solution for helping to protect our planet. Solar air heaters are being used for many applications at low and moderate temperatures. Some of these are crop drying, timber seasoning, space heating, cooking etc. The thermal efficiency of solar air heater has been found to be low due low thermal capacity of air and because of low convective heat transfer coefficient between absorber plate and flowing air in the duct [16]. The thermal efficiency of a single pass solar air collector a function of mass flow rate it is higher with an increasing the flow rate. Increasing the absorber area or fluid flow heat-transfer area will increase the heat transfer to the flowing air, on the other hand, will increase the pressure drop in the collector, thereby increasing the required power consumption to pump the air flow crossing the collector [2], [11]. Several researcher tried different configurations of absorber plates to improve the heat transfer coefficient. Artificial roughness obstacles and baffles in various shapes and arrangements were employed to increase the area of the absorber plate. As a result, the heat transfer coefficient between the absorber plate and the air pass is improved [14].

Bhagoria J.L. et al. [3] performed to collect heat transfer and friction data for forced convection flow of air in solar air heater rectangular duct with one broad wall roughened by wedge shaped transverse integral ribs. The experiment encompassed the Reynolds number range from 3000 to 18000; relative roughness height 0.015 to 0.033; the relative roughness pitch 60.17$\phi^{-.0264}$ (p/e12.12) and rib wedge angle (f) of 8, 10, 12 and 15. The effect of parameters on the heat transfer coefficient and friction factor are compared with the result of smooth duct under similar flow conditions. Jaurker A.R et al [9] described the experimental investigation on the heat transfer and friction characteristics of rib-grooved artificial roughness on one broad heated wall of a large aspect ratio duct shows that Nusselt number can be further enhanced beyond that of ribbed duct while keeping the friction factor enhancement low. Investigation clearly demonstrates that the heat transfer coefficient for rib-grooved arrangement is higher than that for the transverse ribs, whereas the friction factor is slightly higher for rib-grooved arrangement as compared to that of rectangular transverse ribs of similar rib height and rib spacing. Hikmet Esen [7] analyzed a novel flat plate solar air heater (SAH) with several obstacles and without obstacles. The results show that the largest irreversibility is occurring at the flat plate (without obstacles) collector in which collector efficiency is smallest. Kamali R. and Binesh A.R. [10] described the computer code to study the turbulent heat transfer and friction in a square duct with various- shaped ribs mounted on one wall. The results show that features of the inter-rib distribution of the heat transfer coefficient are strongly affected by the rib shape and trapezoidal ribs with decreasing height in the flow direction provide height heat transfer enhancement pressure drop than other shapes. Ebru Kavak Akpinar and Fatih Kocyigit [4] investigated the performance for a new flat plate solar air heater (SAH) with 3 several types of obstacles and a without obstacles and concluded that the collector supplied with obstacles appears significantly better than that without obstacles.

Varun et al. [19] described the experimental study has been carried out to the effective efficiency of a solar air heater duct provided with transverse and inclined ribs as artificial roughness elements on the absorber plate. The range of parameters considered for the present investigation; Reynolds number (Re) 2000-14000, relative roughness pitch (p/e) 3-8 and a fixed value of relative roughness height (e/D) of 0.030. Results show that the roughened collector with absorber plate having relative roughness pitch (p/e) of 8 gives the best

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performance. Hans V.S, et al [8] investigated the effect of multiple v-rib roughness on heat transfer coefficient and friction factor in an artificially roughened solar air heater duct. The experiment encompassed Reynolds number (Re) from 2000 to 20000, relative roughness height (e/D) values of 0.019–0.043, relative roughness pitch (P/e) range of 6–12, angle of attack (a) range of 30–75 and relative roughness width (W/w) range of 1–10. Ei-swai A.M. [5] studied theoretical comparison between flat plate, v-grooved and chevron pattern absorbers and found that chevron pattern is found to have higher performance, reaching up to 20% improvement in thermal efficiency and an increase of 10˚C in outlet temperature at some ranges of mass flow rates. El-Sebaii A. A. et al. [6] investigated theoretically and experimentally the double pass flat and v-corrugated plate solar air heaters. The results showed that the double pass v-corrugated plate solar air heater is 11 to 14% more efficient compared to the double pass flat plate solar air heater.

Lanjewar A. M. et al. [12] performed of roughened absorber plate in solar air heater by using W-shape rib roughness, the roughened wall being heated while the remaining three walls insulated. The roughened wall has relative roughness height (e/Dh) 0.018, relative roughness pitch (p/e) 10, rib height 0.8 mm, angle of attack in the range of 30^{o}-60^{o} and duct aspect ratio (W/H) 8. Performance comparison of ribs with different angle of attack show that W-shape ribs with angle of attack of 60^{o} gives best thermo-hydraulic performance. Rajendra Karwa and Girish Chitoshiya [15] experimented thermo-hydraulic performance of a solar air heater with 60˚ v-down discrete rib roughness on the airflow side of the absorber plate along with that for a smooth duct air heater. The enhancement in the thermal efficiency due to the roughness on the absorber plate is found to be 12.5 to 20%. Suman Saurav et al. [16] reviewed the experimental investigations on appropriate roughness geometries shows that the enhancement in heat transfer is accompanied by considerable rise in pumping power. Concluded that in future these V rib arrays could be arranged inclined to the direction of flow and subsequently arrays arranged in V type fashion could be tested in the quest of higher heat transfer rates.

In the present investigation, theoretical and experimental analysis of performance of absorber plate of the solar air collector with inclined multiple V – ribs were carried out.

II. Theoretical Analysis

The efficiency of the solar air collector may depend upon the amount of solar energy captured, area of the collector, irradiation, mass flow rate of air and ambient conditions. A traditional solar air collector has been taken and some of the heat transfer enhancement techniques adopted based on previous study to carry out this work. The effect of geometrical parameters of V-shaped ribs on heat transfer and fluid flow characteristics of rectangular duct of solar air collector with absorber plate having V-shaped ribs are analyzed. The thermal performance of flat plate solar air heater could be observed by considering the energy balance between solar energy absorbed by absorber plate and useful thermal energy output of the system accompanied by some losses. Definition sketch of energy balance of solar air heater is shown in Fig. 1.

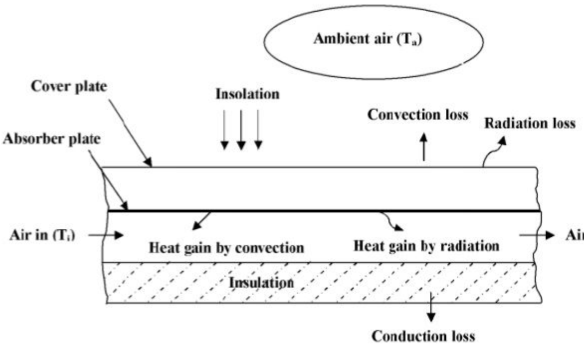

Fig. 1 Energy balance of solar air heater [16]

The energy balance equation can be written as follows,

$$Q_a = A_p\left[IR(\tau\alpha)_e\right] = Q_u + Q_L \tag{1}$$

where Q_a is the energy absorbed by the absorber plate (W), A_p is the area of the absorber plate (m^2), I is the intensity of insolation (W/ m^2), R is the conversion factor to convert radiation on horizontal surface to that on the absorber plane, $(\tau\alpha)_e$ is the effective transmittance absorptance product of the glass cover-absorber plate combination, Q_u is the useful energy gain and Q_L is energy loss from the collector (W).

The useful energy gain can be expressed in terms of inlet air temperature Ti and other system and operating parameters as,

$$Q_u = A_p F_R\left[IR(\tau\alpha)_e - U_L(T_i - T_a)\right] \tag{2}$$

where F_R is given by,

$$F_R = \frac{mC_p}{A_p U_L}\left[1 - \exp\left(\frac{F' U_L A_p}{mC_p}\right)\right] \tag{3}$$

where F_R is the collector heat removal factor which indicates the thermal resistance meet by the absorbed solar energy in reaching to the flowing air. U_L is the overall loss coefficient and T_i and Ta are the inlet air and ambient temperatures respectively. F' is termed as collector efficiency factor which provides the relative measurement of thermal resistance between absorber plate and ambient air to that of thermal resistance between the air flowing through collector and the ambient air. Collector efficiency factor (F') is expressed as,

$$F' = \frac{1}{\left(1 + \frac{U_L}{h_e}\right)} \tag{4}$$

where, h_e is the effective heat transfer coefficient (W/m^2-K) between the absorber plate and flowing Air, The thermal efficiency of the collector is the ratio of useful heat gain to the incident solar energy falling on the collector. Therefore,

$$\eta_{th} = \frac{Q_u}{IA_p} = F_R\left[\frac{IR(\tau\alpha)_e - U_L(T_i - T_a)}{I}\right] \tag{5}$$

Reynold's number is defined as the ratio of inertia force and viscous force.

$$\mathrm{Re} = \frac{m * D_h}{\mu * A_f} \tag{6}$$

where, m - mass of air (kg/sec), μ - Absolute viscosity of air (kg/ms), A_f - Fluid flow area (m^2), and D_h– hydraulic diameter (m)

$$D_h = \frac{2H * W}{H + W} \tag{7}$$

where, H - Height of the duct (m), W - width of the duct (m)
Nusselt number is calculated using following relation [13]

$$Nu = 0.067\,\mathrm{Re}^{0.888}\left(\frac{e}{D_h}\right)^{0.424}\left(\frac{\alpha}{60}\right)^{-0.077}\exp\left[-0.782\left(\ln\left(\frac{\alpha}{60}\right)\right)^2\right] \tag{8}$$

where, Re - Reynolds number, e - Height of rib and α - absorbance of collector

The Radiation heat transfer coefficient (h_r) is given by

$$h_r = \left(\frac{4\sigma(T)^3}{\frac{2}{\varepsilon_p} - 1}\right) \tag{9}$$

where σ - Stefan-boltzman constant, ε_p - emittance of absorber plate

The Convective heat transfer coefficient (h) is given by

$$h = \left(\frac{Nu * K}{D_h}\right) \tag{10}$$

where, N_u - Nusselt Number, K - Thermal Conductivity of air (W/m-K)

The top loss coefficient (U_t) is calculated using

$$U_t = \left[\frac{N}{\left(\frac{249}{T_p}\right)\left(\frac{(T_p - T_a)}{(N+\delta)}\right)^{0.22}} + \frac{1}{h_w}\right]^{-1} + \left[\frac{\sigma(T_p - T_a)(T_a^2 + T_p^2)}{[\varepsilon_p + 0.05N(1-\varepsilon_p)]^{-1} + \left[\frac{2N + \delta - 1}{\varepsilon_p}\right]^{-1} - N}\right] \tag{11}$$

where, N - number of glass cover, T_a - ambient temperature (K), T_p - plate temperature (K)

$$\delta = (1 - 0.04h_w + 0.005h_w^2)(1 + 0.091N)$$
$$h_w = 5.7 + 3.8V$$

V - wind velocity (m/s)

The loss through the bottom of the collector (U_b) is calculated using

$$\mathrm{Ub} = \frac{K_i}{X_i} \tag{12}$$

where, K_i - insulation thermal conductivity, X_i - insulation thickness (m)

The overall loss coefficient (U_L) is given by

$$U_L = U_b + U_t \tag{13}$$

Edge losses for large collector are usually negligible but for small collector the edge losses may be significant.

Fig. 2 Experimental setup of solar air collector system

1. Blower, 2. Control valve, 3.Orifice meter, 4. Inlet duct, 5. Solar air heater, 6. Outlet duct, 7. Digital thermocouple Multi meter

III. Experimental Analysis

A. Experimental Setup

An experimental test facility for testing inclined multiple V – ribs roughened absorber plate has been designed and fabricated as shown in Fig. 2. The experimental setup was designed as per ASHRAE standard 93-77 [1]. The solar collector dimension is of 1000 x 500 x 300 mm. The upper side is made from transparent glass cover. The thickness of transparent cover is window type Plexiglas of 3mm was used as glassing. Inlet and exit ducts are located in lengthwise. The atmosphere air is sent to the collector by single pass under an absorber plate and in the inclined multiple V – ribs which located under an absorber plate. The absorber was made of galvanized iron sheet with block coating. The thickness of absorber was 1 mm. Thermal losses through the back side are insulated by thermacol 5mm thick. The output from the thermocouple was recorded in degree Celsius by using digital thermocouple multi meter. Mass flow rate of air can be measured by an orifice plate. A standard Pitot tube was used for its calibration. The flow was changed by control valve.

Temperature measurements were done with calibrated thermocouples of T-type consisting of 0.3 mm diameter copper constantan wires. The inlet, outlet and middle portion of the collector temperature was measured. A digital micro-manometer of least count 0.1 Pa was used for measurement of test section pressure drop.

Inclined V – ribs are arranged in staggered array at one side of absorber plate. It is also made up of galvanized iron sheet. Angle between the two faces of V is 60º. Angle between the absorber plate to V face is 60ºand thickness of V rib is 1mm. Ribs are fixed with absorber plate by using silicon paste which having the thermal conductivity property. Totally 39 number of ribs are fixed. The relative roughness pitch and relative roughness height was made as per calculated dimension. Smooth and roughened absorber plate is shown in Fig. 3.

Fig. 3 Smooth and roughened absorber plate

B. Experimental Procedure

The experimental data was collected for inclined multiple V – ribs roughened absorber plate. For each experimental run, initially all the instruments were checked for their correctness and all joints were checked to avoid any air leakage. Data was recorded under steady state conditions for the air temperature at inlet to test section. The experimental setup was considered to be in steady state, when the temperatures of air after mixing section and absorber plate did not change appreciably for 10 min [17], [18]. The experiments begin at 10:00 a.m. and ends at 04:00 p.m. Every half an hour intervals inlet and outlet air temperature was measured and noted.

IV. Results and Discussion

A. Theoretical Evaluation

1. Thermal Efficiency with respect to Irradiation

Fig. 4 represents the variation of thermal efficiency of with and without multi V ribs roughened air collector with respect to irradiation. For a solar irradiation of 1000 W/m^2, the result shows that the rough collector has the 14% more thermal efficiency than smooth collector. This may be due to increase in the heat transfer area and turbulent flow created by inclined V ribs.

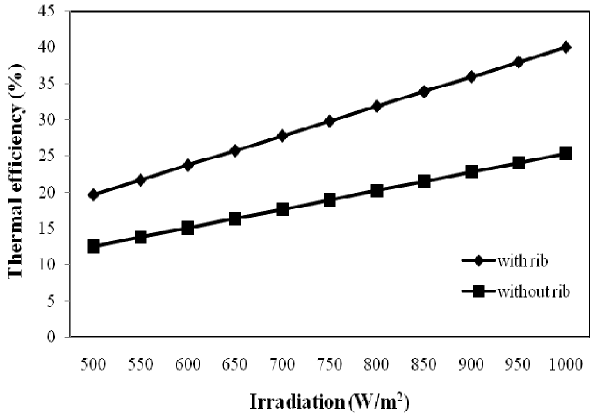

Fig. 4 Variation of thermal efficiency with respect to irradiation

2. Thermal Efficiency with respect to Mass Flow Rate

The variation of thermal efficiency for with and without ribbed solar collector with respect to mass flow rate of air shown in Fig. 5. The thermal efficiency of smooth collector gradually increases with increase in mass flow rate. At a mass flow rate of 0.045 kg/sec, the thermal efficiency of rough collector has 15% more than smooth collector.

3. Heat Removal Factor with respect to Mass Flow Rate

Fig. 6 shows the variation of heat removal factor for with and without ribbed solar air collector with respect to mass flow rate. The heat removal factor of the smooth solar collector gradually increases with increase in mass flow rate. The heat removal factor of roughened solar collector increases by 0.2 with the mass flow rate of 0.045 kg/s, when compared to the smooth collector.

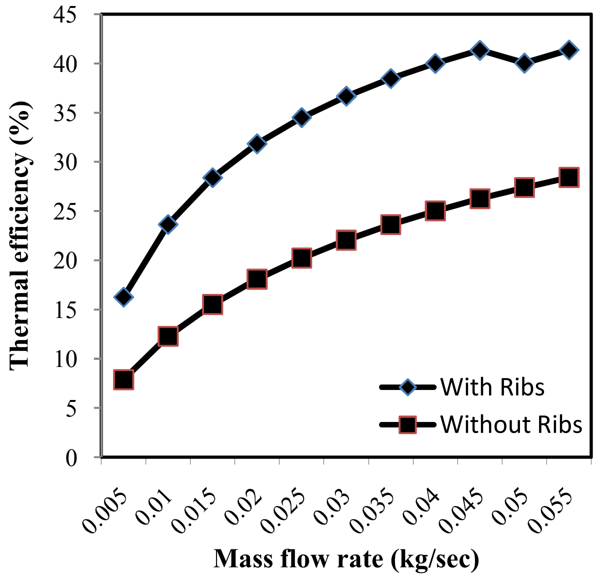

Fig. 5 Variation of thermal efficiency with respect to mass flow rate of air

B. Experimental Evaluation

1. Variation of Thermal Efficiency with respect to Irradiation

The variations in the thermal efficiency of smooth and roughened solar air collector with respect to irradiation were found experimentally (Fig. 7). The experimental investigation was carried for 5 day and the average values are plotted. When compared to the smooth collector thermal efficiency of rough collector is much more impressive. The maximum thermal efficiency of smooth collector efficiency found to be 33% and that of roughened is 47.22%. The percentage improvement in thermal efficiency of the roughened collector is 43%.

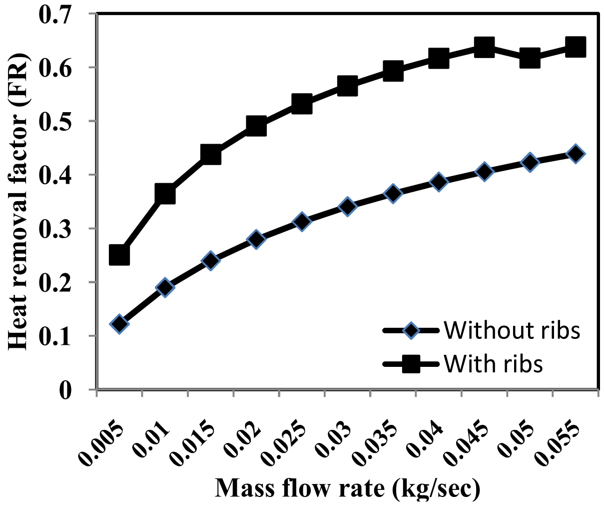

Fig. 6 Variation of heat removal factor with respect to mass flow rate of air

2. Temperature Difference with respect to Time

The temperature difference between the inlet and outlet is an evident that how effectively the system transfer energy to the air. The temperature difference between the inlet and outlet was noted for 5 days for smooth and roughened collector

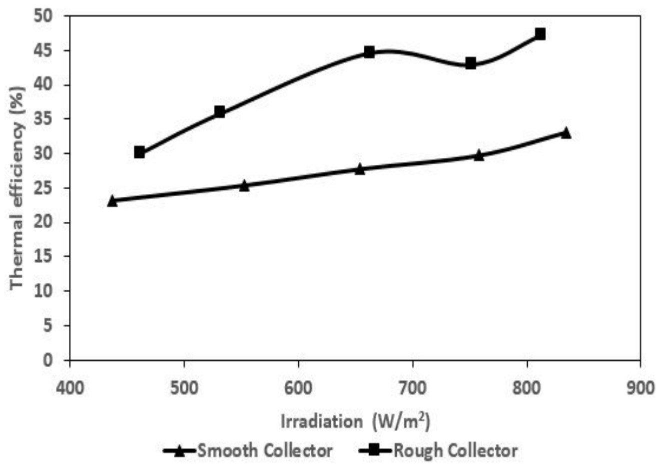

Fig. 7 Variation in thermal efficiency with respect to irradiation of smooth and rough solar collector

Fig. 8 shows the temperature difference between the inlet and outlet of smooth solar collector with respect to time interval of a day. At 10.00 AM the temperature difference is lesser, after half an hour interval gradually increases up to 12.30 PM and then decreases in all the above mentioned days. The temperature difference varies from 7 to 25°C. Maximum temperature difference is 25°C from the inlet to outlet of solar collector duct.

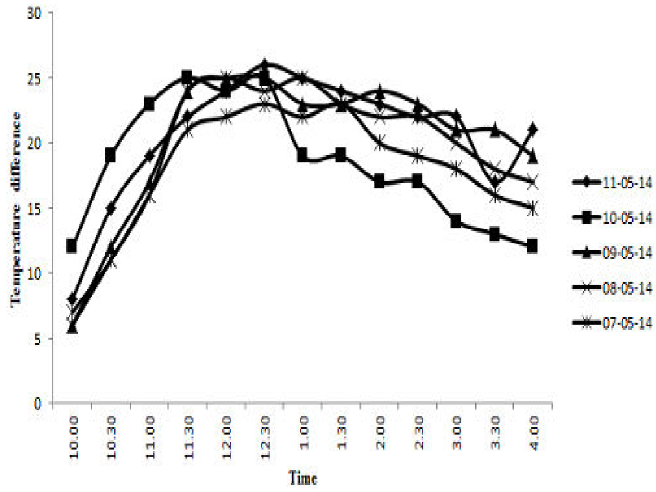

Fig. 8 Variation of temperature in smooth solar collector with respect to time

Fig. 9 shows the temperature difference between the inlet and outlet of rough solar collector. The difference in temperature varies from 7 to 34°C. Maximum temperature difference is 34°C from the inlet to outlet of solar collector duct. When compared to the smooth collector, the rough collector has 9°C increases in temperature difference. This is obtained due to artificial roughness of underside absorber plate in the form of inclined V – ribs arranged in staggered arrays.

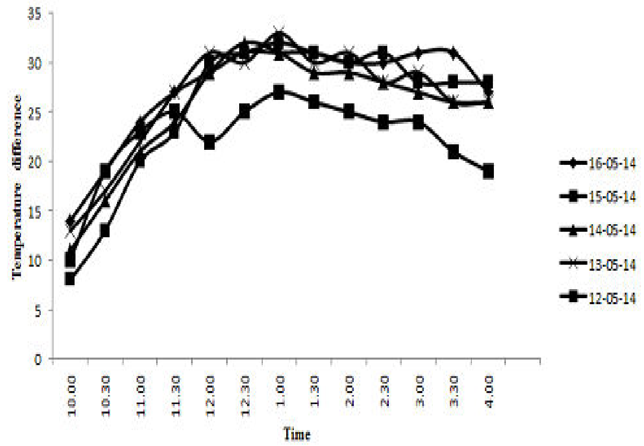

Fig. 9 Variation of temperature in rough solar collector with respect to time

V. CONCLUSIONS

A detailed theoretical and experimental investigation were carried out to evaluate the performance of solar air collector with inclined multi V ribs and the smooth air collector Based on the results obtained, the following conclusions are drawn:

- The variation between theoretical result and experimental results are in the range of 1-8%. The reasons for the deviations have been identified as the uncertainties in the estimates of the wind heat transfer coefficient, heat transfer coefficients and sky temperature from the available correlations.
- At maximum point, enhancement of thermal efficiency due to artificial roughness around 14% both in theoretical and experimental estimation.
- Maximum temperature difference between the inlet and outlet smooth solar collector is 25^{o}C and that of rough collector is 34^{o}C. This increase in temperature difference is a clearly indication for improved thermal efficiency of roughened duct solar air heater compared to the smooth duct air heater.

NOMENCLATURE

A_P	Collector plate area	m^2
A_f	Fluid flow area	m^2
Cp	Specific heat of air	J/kg K
C_{ps}	Specific heat	J/kg K
D_h	Hydraulic diameter	m
e	Height of rib	m
F_R	Collector heat removal factor	W/m^2-K
F’	Collector efficiency factor	W/m^2-K
H	Height of the duct	m
h	Convective heat transfer coefficient	W/m^2-K
h_r	Radiation heat transfer coefficient	W/m^2-K
h_e	Effective heat transfer coefficient	W/m^2-K
I	Intensity of insolation	W/ m^2
K	Thermal conductivity of air	W/m-K
K_i	Insulation thermal conductivity	W/m-K
m	Mass flow rate of air	kg/sec
N	Number of glass cover	
Nu	Nusselt number	
Q_u	Useful energy gain	W
Q_a	Energy absorbed by the absorber plate	W
Q_l	Energy loss from the collector	W
R	Conversion factor to convert radiation on horizontal surface to that on the absorber plane	
Re	Reynolds number	
T_a	Ambient temperature	K
T_i	Inlet air temperature	K
T_o	Outlet air temperature	K
T_p	Plate temperature	K
U_L	Over all loss co-efficient	W/m^2-K
U_t	Top loss coefficient	W/m^2-K
U_b	Back loss coefficient	W/m^2-K
V	Wind velocity	m/s
W	Width of the duct	m
x_i	Insulation thickness	m
α	Absorbance of collector	
η_{th}	Thermal efficiency of the collector	%
ρ	Density of air	kg/m^3
ε_p	Emissivity of absorber plate	
ε_g	Emissivity of glass plate	
σ	Stefan-Boltzmann constant	W/m^2-K^4
τ	Transmittance of collector	
μ	Viscosity of air	kg/ms
$(\tau\alpha)_e$	Effective transmittance absorptance product of the glass cover-absorber plate combination	

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