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Classifications Design Parameters and Development of Solar Air Heating

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Abstract— *With the need to tap renewable energy sources increasing rapidly, solar energy has shown remarkable promise, but has its share of drawbacks as well. Although costs of solar panels has decreased in the last few years, the low flux density of solar energy has made its use commercially less viable. In this paper, we have carried out detailed thermal analysis to study the absorption of heat by a 1.5*2.5 feet aluminium panel, and confirmed the results via experimental studies on a self-fabricated double pass double glazed solar air heater. The different kinds of solar air collectors have also been studied. The effects of various parameters such as inlet fluid temperatures, mass flow rate, and depth of air channel on the thermal performance of the double pass double glazed solar air heater have also been studied, and a graph for outlet air temperature VS time of day has been plotted via MATLAB. Moreover, it has been concluded that although the amount of heat absorbed on the absorber plates increases to a large extent in a double pass double glazed heater, the introduction of cylindrical fins will further increase the efficiency.*

Keywords— *Renewables, Solar Air Heating, Renewable Energy, Solar Energy, Solar Air Heater (SAH), Solar Air Collector, Solar Radiation, Designing SAH.*

I. INTRODUCTION

With estimates that oil will be gone in about a hundred years, and coal in about five hundred years, exhaustion of our fuel resources is a matter of concern for the entire civilisation. Global energy consumption in the last half-century has rapidly increased and is expected to continue to grow over the next 50 years, with India and China fuelling majority of the growth. On the positive side, the renewable energy (RE) technologies of wind, biofuels, solar thermal, and photovoltaics (PV) are finally showing maturity and the ultimate promise of cost competitiveness. A review of the present energy resources and their availability (Kreith and Goswami 2007; WBGU 2003) shows that as much as 50% of the global energy use in 2050 will have to come from RE sources, a vast majority being from solar energy and wind.

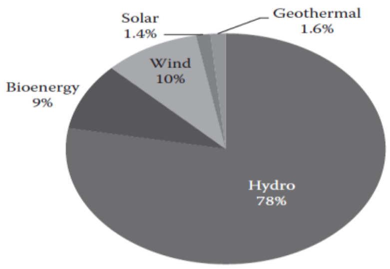

Fig. 1. 2011 Renewable resource shares in world electricity capacity.

The total share of all renewables for electricity production in 2011 was approximately 20.1%, a vast majority (78%) of it being from hydroelectric power (Figure 1). Even though worldwide solar power capacity represented only 1.4% of the total electricity capacity, it was growing at an average annual rate of approximately 50%.

Solar Energy

The amount of sunlight striking the earth's atmosphere continuously is 1.75 × 105 TW. Considering a 60% transmittance through the atmospheric cloud cover, 1.05 × 105 TW reaches the earth's surface continuously. If the irradiance on only 1% of the earth's surface could be converted into electric energy with a 10% efficiency, it would provide a resource base of 105 TW, while the total global energy needs for 2050 are projected to be approximately 25–30 TW. The present state of solar energy technologies is such that commercial solar panel efficiencies have reached more than 20%, laboratory multi junction solar cell efficiencies under concentrating sun have exceeded 40%, and solar thermal systems provide efficiencies of 40%–60%.Solar PV panels have come down in cost from approximately $30/W to approximately $0.50/W in the last three decades. At $0.50/W panel cost, the overall system cost is around $2/W.

However, solar energy has its share of limitations as well, and some of them are:

- *Low flux density*: For large scale utilization, it is necessary to have a large surface to collect solar energy. Naturally, the larger the surfaces, the more expensive the delivered energy becomes.
- *Transmission:* Most of the solar energy falls on remote areas and would therefore require some means of transmission to be useful to the industrialized nations. The mean amount of energy available on a horizontal plane is greatest in the continental desert areas around latitudes 25°N and 25°S of the equator and falls off toward both the equator and the poles.
- *Intermittency:* Solar energy has a regular daily cycle owing to the turning of the earth around its axis and a regular annual cycle owing to the inclination of the earth axis with the plane of the ecliptic and to the motion of the earth around the sun, and is also unavailable during periods of bad weather. This introduces the need for storage, thereby increasing the cost further.

In this paper, the focus is primarily on the first limitation i.e. low flux density. In order to make solar energy viable for mass usage, efficient utilization of the energy is very important.

II. Literature Review

Conventional SAHs consist of insulated hot air ducts, panels, and air blowers in active systems. The thermal performance of an SAH depends on several factors such as the dimensions of the collector, the absorber's type and shape, glass cover, inlet temperature, wind speed, etc. Many researches have been carried out for studying in detail the effect of those parameters on the thermal performance of SAHs. Venegas et al. (2011) studied the effect of wind velocity, ambient temperature, and relative humidity on the performance of the SAH.

Several configurations were suggested by researchers in designing the absorber plates of SAHs in order to improve the heat transfer coefficient between the absorber plate and the working fluid. In several studies, the areas of the absorber plates were increased by employing artificial roughness, obstacles, and baffles in different shapes and arrangements (Romdhane and Slama 2007). Esen (2008) experimentally investigated the effect of fixing obstacles on the absorber plate while Varun et al. (2009) employed transverse and inclined ribs as elements of artificial roughness on the absorber.

Esen (2008) and Varun et al. (2009) concluded that the obstacles are playing a crucial role for the improvement of the performance of the collector. Mittal and Varshney (2006) worked with different types of roughness elements, Youcef (2005) fixed offset rectangular plate fins, and Ozgen, Esen, and Esen (2009) inserted aluminium cans fixed on the absorbing plate, and all reported that there were considerable enhancements in the effective efficiency of SAHs having roughened ducts. They argued that this configuration improved the collector efficiency substantially through the increase of the fluid velocity and the enhancement of the heat transfer coefficient between the absorber plate and flowing air. Akpınar and Koçyi˘git (2010) experimentally investigated a flat-plate SAH with several obstacles and found that the performance of the collector without obstacles was considerably less compared to the SAH with obstacles. Alvarez et al. (2010) experimentally showed that using serpentine geometry enhanced the performance of SAH. Gupta and Kaushic (2009) investigated different types of artificial roughness to conclude that the roughness in the absorber plate of SAH increases the efficiency.

Experimental and analytical studies on SAHs have been carried out by Gao et al. (2006). They designed and constructed a cross-corrugated shape (wave-like) absorbing plate which was placed in between the bottom plate and the glass cover. Gao et al. (2006) in their study indicated that the maximum achievable efficiency reached was 60.3% for the wave-like shape absorber plate. El-Sebaii et al. (2011) found that the SAH with v-corrugated absorber plate is more efficient by 9.3–11.9% than the one with fins attached. Alta et al. (2010) studied three different types of SAHs and reported that the SAH with obstacles fixed on the absorber plate has better performance compared to the SAHs without obstacles.

Romdhane and Slama (2007) reported that introducing suitable baffles in air passage increases both the efficiency and the outlet temperature. Improvements in the thermal performance of SAHs were also achieved by replacing the flat absorber plates with porous materials such as wire mesh. The porous media increases the turbulence of flowing air; thus, the heat transfer coefficient between packing porous medium and air increases. Another advantage of using porous medium is the increase in surface area per unit volume (Varun et al. 2009). On the other hand, the porous medium increases the pressure drop across the channel. In order to decrease the pressure drop and reduce power consumption, Kolb, winter, and Viskanta (1999) suggested a new bed design where two parallel wire mesh layers were attached to the absorber plate. In their design, the pressure drop across the channel of SAHs is much less than for those having obstacles or fins fixed on the absorber plates.

They reported that the thermal efficiencies varied from 58% to 81% for air mass flow rates between 0.01 kg⁄s and 0.0367 kg⁄s and the maximum pressure drop across the channel was 19 Pa. El-Sebaii et al. (2007) designed two different SAH configurations and used gravel and limestone as packed bed materials. They achieved 65% thermal efficiency at air mass flow rate of 0.05 kg⁄s, while the pressure drop across the channel was 400 Pa. Prasad et al. (2009) presented an experimental investigation of the SAH having wire mesh as an absorber. They reported 28.5% improvement in the performance of the SAH packed with wire mesh (with porosity of 0.599) compared to the conventional flat-plate absorber SAH. Furthermore, they concluded that the efficiency increased by decreasing the porosity of the bed. Paisarn (2005) studied numerically the characteristics of the heat transfer and the performance of the double-passed flat-plate SAH with and without porous media for mass flow rates between 0.03 and 0.07 kg⁄s.

Paisarn, reported that the efficiency varied from 38% to 59% for the bed without porous medium, while for the bed with porous medium the efficiency varied from 42% to 70%. Aldabbagh, Egelioglu, and Ilkan (2010) investigated an SAH with 10 wire mesh layers as absorber plate for single and double pass, Omojaro and Aldabbagh (2010) used seven wire mesh layers with longitudinal fins for single and double pass SAH. El-khawajah, Aldabbagh, and Egelioglu (2011) investigated the performance of an SAH with12 wire mesh layers used as an absorber and investigated the effects of using transverse fins on a double-pass flow SAH. In order to reduce the pressure drop across the SAHs, the wire mesh used was arranged in layers with spaces between them.

Solar Air Collectors

Solar air collectors can be divided into two categories:

- Unglazed Air Collectors or Transpired Solar Collector (used primarily to heat ambient air in commercial, industrial, agriculture and process applications)
- Glazed Solar Collectors (recirculating types that are usually used for space heating)

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Collector Types

Solar collectors for air heat are classified according to their air distribution paths or by their materials, such as glazed or unglazed. For example:

- through-pass collectors
- front-pass
- back pass
- combination front and back pass collectors
- unglazed
- glazed

Unglazed solar air collector: These kind of solar air collectors show promise for applications such as ventilation air heating or crop drying. It was shown by Kutscher (1993) and Pesaran (1994) via a heat loss theory that in an unglazed transpired collector, almost all the heat conducted into the thermal boundary layer is removed by the suction air so that the radiation loss is the dominant part of the top heat loss of the collector.

D. Njomo (2000) in his paper, Unglazed selective absorber solar air heater, developed a mathematical model to analyse the heat exchanges in an unglazed non porous selective absorber solar air heater, and showed the effects of various parameters such as inlet fluid temperatures, mass flow rate, and depth of air channel on the thermal performance of the unglazed selective absorber solar air heater.

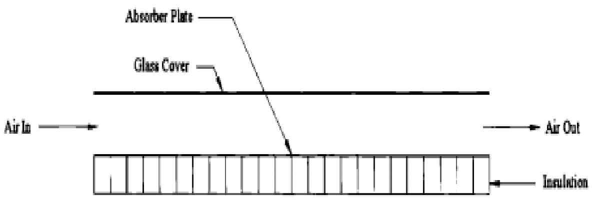
Glazed solar air heater: Functioning in a similar manner as a conventional forced air furnace, glazed air systems provide heat by recirculating conditioned building air through solar collectors. Through the use of an energy collecting surface to absorb the sun's thermal energy, and ducting air to come in contact with it, a simple and effective collector can be made for a variety of air conditioning and process applications.

III. SOLAR AIR HEATER CLASSIFICATION

1) *Non-Porous Type Solar Air Heater*

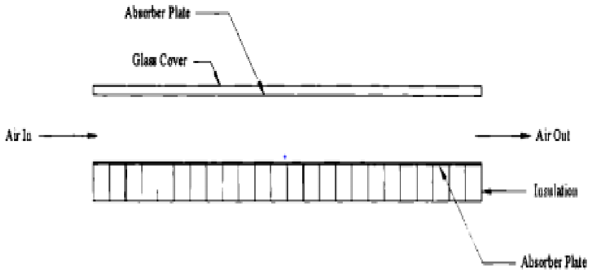
In non-porous type, air stream does not flow through below the absorber plate but air may flow above and/or behind the plate.

(a) Type I — The air steam does not flow through the absorber plate in this system, however it may flow above or below it. There is no requirement of a separate passage and air flows between transparent cover system and absorber plate. This type of air heater has a disadvantage. Due to the flow of hot air above the absorber, cover receives much of the heat and in tum losses it to the ambient.



Type I Non-porous solar air heater (flow above the absorber)

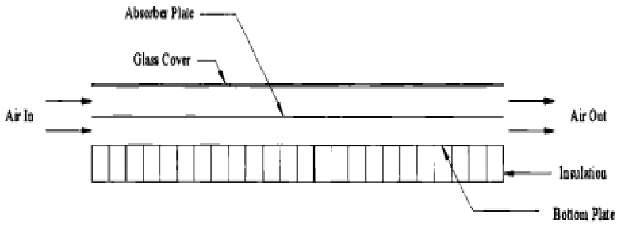
b) Type II – Here, air passage is below absorber plate. A plate parallel to the absorber plate is provided in between absorber and insulation, forming the passage. Also loss of heat is minimal in this type.



Type II Non-porous solar air heater (flow under the absorber)

(c) Type III

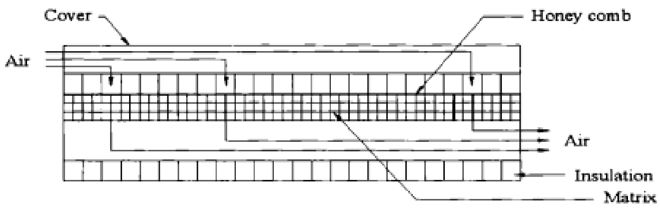
In this type, absorber plate is cooled by air stream flowing on both sides of the plate. It may be noted that heat transfer between the absorber plate and the airflow is low. This is because of low efficiency of this type of solar air heaters. Roughening the absorber surface or using corrugated plate as the absorber, improves performance. Turbulence induced in the airflow increases convective heat transfer. Unless selective coatings are used, radiative losses from absorber plate of nonporous type are significant. Hence collection is poor. Also use of fins may result in prohibitive pressure drop, limiting the applicability of nonporous type.



Type III Non-porous solar air heater (flow both above and beneath the absorber)

2) *Porous Type Solar Air Heater*

The second type of air heaters has porous absorber which may include slit and expanded metal, overlapped glass plat absorber and transpired honeycomb.



Honeycomb porous bed air heater

The solar heater with porous type of absorber has the following advantages:

Advantages of porous solar air heater

Solar radiation penetrates to a great depth and is absorbed along its path. Thus the radiation loss decreases. Air stream heats up as it passes through the matrix.

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The pressure drop is usually lower than the non-porous type.

It may be noted however, that an improper choice of matrix porosity and thickness may cause reduction in efficiencies as beyond an optimum thickness, matrix may not be hot enough to transfer the heat to air stream.

IV. Design Methodology

Parameters Associated with the Construction of an Air Heater

1) Heater Configuration - Heater configuration is the aspect ratio of duct and length through which air passes.

2) Airflow - Air must be pumped through the heater. Increasing the air velocity results in higher collection efficiencies.

3) Transmittance Properties of the Cover - The type and number of layers of cover material must be considered and spectral transmittance properties must be examined. In general, as temperature requirement is high, more number of covers is required. Principle underlying the use of multicovers is that each air layer between two successive covers provides a barrier against heat loss from the absorbing surface to the surroundings. However, with a large number of covers, reflective losses increase. Covers of high transmissivity and low reflectivity are desired to keep the amount of reflected and absorbed radiation low.

4) Absorber Plate Material Selective surfaces can improve performance of solar air heaters by increasing collector efficiency. Absorber is coated black to absorb maximum amount of incident radiation.

5) Natural Convection Barriers - Stagnant air interposes high impedance to convective heat flow between the absorber plate and the ambient air. Losses are reduced by the use of multiple covers or honeycombs.

6) Plate-to-air Heat Transfer Coefficient - Absorber can be roughened and coated to increase coefficient of heat transfer between the air and the plate. Roughness ensures high level of turbulence in the boundary layer of the flowing stream. For this reason, crumpled or corrugated sheets and wire screens are attractive as absorbing materials.

7) Insulation - Insulation is required at the absorber base to minimize heat losses through the underside of the heater.

8) Solar Radiation Data - Solar radiation data corresponding to the site are needed to evaluate heater performance

Heater Description

1. Length, L: 750 mm or 2.5 feet
2. Breath, b: 500 mm or 1.64 feet
3. Height, H: 14 cm
4. Thickness of wood: 0.75 inch or 1.9 cm
5. Thickness of glass: 0.3 mm
6. Thickness of Al sheet: 0.2 mm
7. Thickness of Styrofoam: 1 cm
8. Distance of middle Al sheet: 4 cm from bottom glass
9. Length of middle sheet: 655 mm
10. Height of each channel, h: 4.5 cm

V. Thermal Analysis

1. Top Glass Cover

$\alpha_g IA_g + A_g(h_{rgg}+h_{cg1})(T_{g1}-T_{gu}) = A_g h_w(T_{gu} - T_a) + A_g h_{rgus}(T_{gu} - T_s)$

2. Second Glass Cover

$\alpha_g IA_g T_g + A_g h_{g1}(T_{fu} - T_{gl}) + A_g h_{rpg1}(T_p - T_{g1}) = A_g(h_{rgg} + h_{cg1})(T_{g1} - T_{gu})$

3. Air through the upper channel

$\dot{m}_{fu} C_p \frac{dTf1}{dx}$.dx = $h_{f1ge}(T_{g1} - T_{f1})$.bdx + $h_{f1p}(T_p - T_{f1})$*.bdx + bUs(Tf1-Ta)*

4. Absorber Plate

$\alpha_p IA_p Tg^2 = A_p h_{flp}(T_p - T_{f1}) + A_p h_{f2p}(T_p - T_{f2}) + A_p h_{rcp}(T_p - T_{c2}) +$
$h_{rpb} A_p (T_p - T_b)$

5. Air through the Lower channel

$\dot{m}_{f2} C_p \frac{dTa}{dx}$.dx = $h_{f2p}(T_p - T_{f2})$.bdx + $h_{f2b}(T_b - T_{f2})$.bdx – bUs (Tf2-Ta)

6. Bottom plate

$0 = A_b h_{f2b}(T_b - T_{f2}) + A_b h_{rpb}(T_b - T_p) + A_b U_b (T_b - T_a)$

7. Outlet Air Temperature

$T_o = T_1 + \frac{L}{mc_p}[b h_{f1g2}(T_p - T_{f1})$ + b$h_{f1p}(T_p - T_{f1})$ + b$h_{f2p}(T_p - T_{f2})$+ b$h_{f2p}(T_b - T_{f2})$-bU_s(Tf1-Ta)- bUs (Tf2-Ta)]

VI. Mathematical Relations

- In order to solve the model, the convective heat transfer coefficients for air flowing over the outside surface of the top glass cover and inside the channel are needed. The following correlation proposed by McAdams for air flowing over the outside surface of the glass cover is used to predict the convective heat transfer coefficient

$$h_a = 5.7 + 3.8V$$

Where ha is the convective heat transfer coefficient, and V is the wind velocity.

Heaton et al. proposed the convective heat transfer coefficient correlation between the channel for laminar flow region (Re<2300) as follows:

$$Nu = \frac{h_i D_e}{k} = Nu_\infty + \frac{0.00190\left(Re\,Pr\left(\frac{D_e}{L}\right)\right)^{1.71}}{1 + 0.00563\left(Re\,Pr\left(\frac{D_e}{L}\right)\right)^{1.17}}$$

where Nu is the Nusselt number, Re is the Reynolds number, Pr is the Prandtl number, Nu∞=5.4 and De is the equivalence diameter of the channel as follows:

$$D_e = \frac{4 \times \text{free flow area}}{\text{wetted perimeter}}$$

- For the transition flow region (2300<Re<6000), the correlation proposed by Hausen is used to predict the heat transfer coefficient

$$Nu = 0.116(Re^{2/3} - 125)Pr^{1/3}\left(1 + \left(\frac{D_e}{L}\right)^{2/3}\right)\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

- The correlation proposed by Tan and Charters is used to predict the heat transfer coefficient for turbulent flow region follows:

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$$Nu = 0.018\, Re^{0.8} Pr^{0.4}$$

VII. CALCULATIONS

1. Coefficient of thermal conductance of glass

$K_{glass} = 0.78\ Wm^{-1} K^{-1}$

2. Average flow velocity (v), v= 1.196 m/s
3. Average Kinematic Viscosity of air

$\nu_{avg} = 1.79 \times 10^{-5}\ m^2/s$

4. Reynolds Number (Re)

$Re = vL/\nu_{avg} = 1.196 \times 0.655/1.79 \times 10^{-5} = 43764.25$

(suggests Turbulent flow)

5. Average Thermal Diffusivity of air $\alpha_{avg} = 0.0158\ m^2/s$
6. Prandtl Number (Pe) of air $Pr = \nu_{avg}/\alpha_{avg} = 0.7089$
7. Nusselt Number (Nu) of flow (Using equation 5)

Nu= 0.018 X (Re)^0.8 x Pr ^(0.4) = 0.018 X (43764.25) ^(0.8) X (0.71)^(0.4) =81.0348

8. Equivalent Diameter (De)

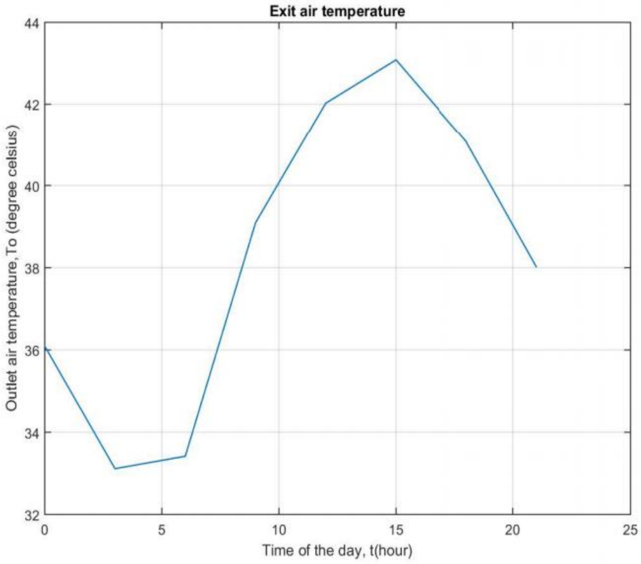
De= 4 X b X h/ 2(b+h) =0.8256881 m

9. Heat transfer coefficient (h)

h= Nu X K/De = 2.343

Fabricated solar air heater

Graph for the designed heater.



Matlab Code

```
b=0.5;
Lp=0.655;
L=0.75;
mf=0.0317;
Cp=1005;
h=2.343;
Us=0.3;
t=[0,3,6,9,12,15,18,21];
Ti=[35,32,32,38,41,42,40,37];
Ta=[40,42,41,45,45,44,43,42];
Tg=[41,38,39,44,46,48,45,43];
Tf1=[51,52,47,57,60,64,61,59];
Tp=[65,64,67,69,70,72,70,73];
Tf2=[48,49,42,55,56,59,57,58];
Tb=[38,37,30,41,42,44,43,44];
To=Ti+b/(mf*Cp)*[L*h.*(Tf1-Tg)+Lp*h.*(Tp-Tf1)-L*Us.*(Tf1-Ta)+L*h.*(Tp-Tf2)+L*h.*(Tf2-Tb)-
L*Us.*(Tf2-Ta)];
plot(t,To,'Linewidth',1);
xlabel('Time of the day, t(hour)');
ylabel('Outlet air temperature,To (degree celsius)');
title('Exit air temperature');
grid on;
```

VIII. CONCLUSIONS

The graph for the exit air temperature vs time of day has been shown above.

The outlet air temperature achieved experimentally coincides with that of the graph plotted i.e. around 3 pm.

Further, it has been seen that double glazing significantly increases the amount of heat absorbed by the absorber plates, as is shown in the thermal analysis.

Moreover, because of the large number of operating parameters associated, further study needs to be conducted, and the effects of introducing cylindrical fins into the double glazed heater will be shown in the upcoming studies.

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NOMENCLATURE

a = Extinction coefficient (m- ~)
C = Specific heat (kJ/kg-°C)
G = Mass flow rate (kg/m2-h)
h = Heat transfer coefficient (W/m2-°C)
l,h = Solar insolation (W/m 2)
k = Thermal conductivity (W/m-°C) /,
x = Length of air heater (m)
M = Mass (kg)
m = Number of harmonics
Re = Reynolds number
Pr = Prandtl number
T = Temperature (°C)
t = Time (h)
UB = Heat loss coefficient from rear plate to ambient air (W/m2-°C)
v = Wind velocity (m/s)
x = Coordinate (m)
y = Matrix thickness
p = Density of iron filing (kg/m 3)