

PERFORMANCE CURVE GENERATION OF AN UNGLAZED TRANSPIRED COLLECTOR FOR SOLAR DRYING APPLICATIONS

Biona M. and Culaba A.

Department of Mechanical Engineering, Don Bosco Technical College
Gen. Kalentong, Mandaluyong City, Philippines

Serafica E. and del Mundo R.

Energy Engineering Program, University of the Philippines Diliman
1101 Diliman, Quezon City, Philippines

ABSTRACT: Cost and efficiency of collectors are the main hindrance to the wide scale adaptation of solar driers. Most designs present trade-offs between these two factors. In 1991, a low cost and efficient solar collector was developed. Unglazed transpired solar collectors (UTCs) have since then gained popularity in building heating applications. It is now currently being explored for small scale applications in the Philippines. This study focused on the development of the performance curve a system designed for drying applications in Palawan. Energy equations of the system have been set-up and an iterative program developed generating the predicted temperature rise and collector efficiency for different hole diameter and pitch distance combinations at face suction velocities between 0.2 to 0.6 m/s. Use of the graphs generated was established and demonstrated.

Keywords: Unglazed transpired collectors, UTC, solar drying

I. INTRODUCTION

Solar collectors for air heating are usually made of a duct with the top cover acting as the absorber (see Figure 1). The low cost solar collector with GI sheet absorbers developed by Mendoza et al. [1] had collector efficiencies ranging only between 4 and 49% and reached temperature rise of only about 14 °C. Glass absorbers on the other hand results to efficiencies as high as 70% and maximum temperature rise of 24 °C as shown by Jensa et al. (2001). The cost of the latter however has been the main reason for it's under-utilization in small scale crop/fish drying.

An unglazed transpired collector (UTC) on the other hand is consists of a perforated plate positioned in front of a wall forming a plenum (as shown in Figure 2). Air is drawn into the plenum through the holes in the plate, and finally into the building. Unlike most solar heaters, UTCs are not covered by glazings thus significantly reducing its cost but provides collector efficiencies between 50 to 80% [3]. Kustcher et al. [4] recorded temperature rise between 12 to 36°C. Meer et al. [5] concluded that it would be the most appropriate system for drying applications.

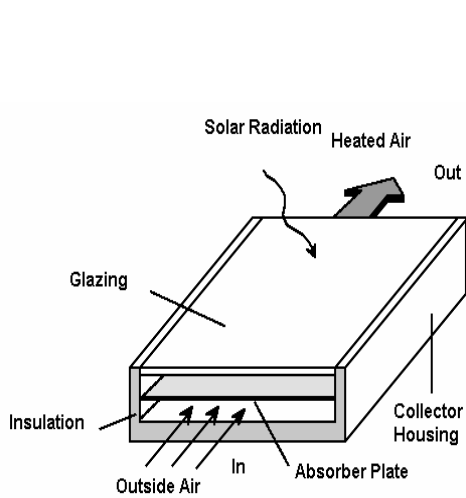


Fig 1. Typical Solar Collector for Air Heating

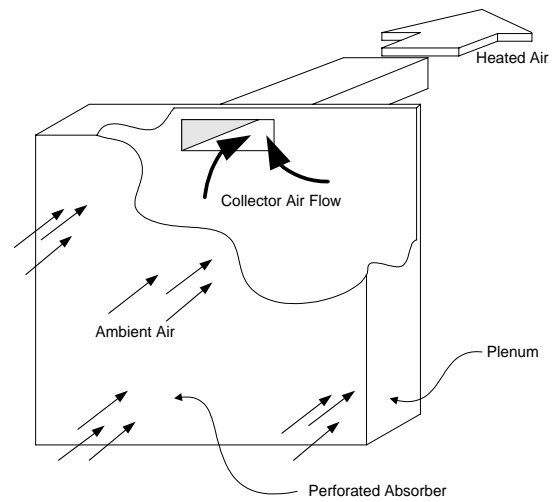


Fig 2. A Complete View of a UTC
(Adapted from the University of Wisconsin, 1999)

II. HEAT EXCHANGE EFFECTIVENESS

The heat exchange effectiveness of an Unglazed Transpired Collector is defined as the ratio of the amount of air temperature change occurring and the temperature difference between the collector plate and ambient air.

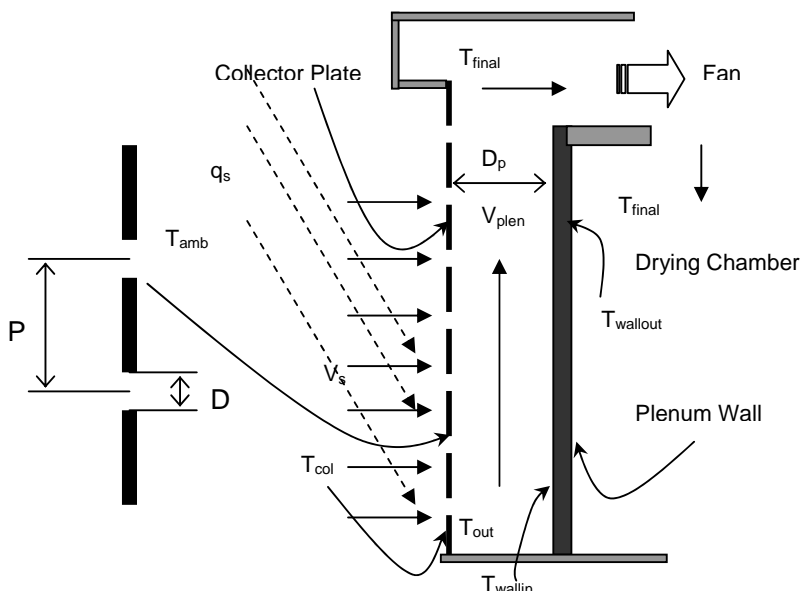


Fig.3 UTC state points and nomenclature

Kutscher [6] developed an expression for heat exchange effectiveness as a function of the hole diameter (D), pitch distances (P) and suction velocity (Vs) shown in equation (1).

$$\mathcal{E} = \frac{T_{out} - T_{amb}}{T_{col} - T_{amb}} = \frac{hA}{mC_p} = 1 - e^{-\frac{UA}{mC_p}} \quad [1]$$

$$Nu_D = 2.75 \left(\frac{P}{D}\right)^{-1.2} Re_D^{0.43} \quad [2]$$

The model developed neglected the effect of plenum depth which was found to have very minimal effects for the usual plenum depths ranging between 8 in to 12 in. The plenum depth of the small scale crop drier being evaluated however is limited only to a maximum of 130 mm. Experimental studies done by Biona and Culaba [7] showed that at depths between 130mm to 50mm, heat exchange effectiveness decreases with plenum depth. A regression equation relating heat exchange effectiveness with plenum depth was developed as shown in equation (3).

$$\varepsilon_{mod} = \varepsilon_{Kutscher} (0.9589 + 0.0004D_p + 0.000000032D_p^3); D_p \text{ in mm} \quad (3)$$

Equation (3) shows that heat exchange effectiveness could vary by as much as 5% with the Kutscher model within the range mentioned.

III. ENERGY ANALYSIS OF UTCS

Predicting the thermal performance of the UTC system requires the simultaneous computation of the energy balances of the collector plate, the air passing through it and plenum wall.

The collector plate absorbs part of the solar insolation hitting it and convects the bulk of it to the air passing through its holes. It radiates some to the surroundings and plenum walls. This phenomena is graphically shown in figure 3. Setting up the energy balance of the plate results to Equation (4). Defining each term of equation (4) and rearranging the result leads to equation (5).

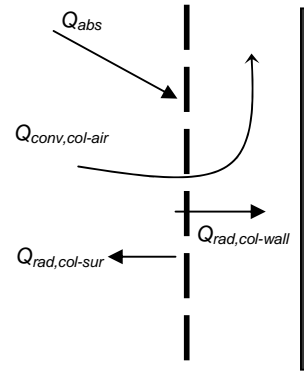


Fig.4 Collector Plate Energy Interaction

$$Q_{abs} + Q_{rad,wall-col} = Q_{conv,col-air} + Q_{rad,col-sur} \quad [4]$$

$$Q_{abs} + Q_{rad,wall-col} = m_{out}C_p\mathcal{E}(T_{col} - T_{amb}) + Q_{rad,col-sur}$$

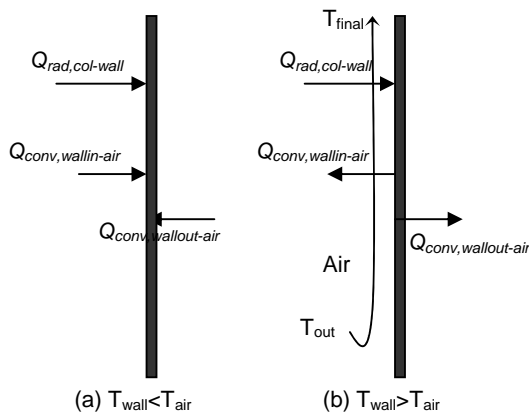
$$T_{col} = \frac{Q_{abs} + Q_{rad,wall-col} - Q_{rad,col-sur} + m_{out}C_p\mathcal{E}T_{amb}}{m_{out}C_p\mathcal{E}}$$

$$T_{col} = \frac{\alpha_{col}q_s A_s + \sigma_{sb} A_s \frac{(T_{wallin}^4 - T_{col}^4)}{(\frac{1}{\mathcal{E}_{plen}} + \frac{1}{\mathcal{E}_{col}} - 1)} - \mathcal{E}_{col} \sigma_{sb} A_s (T_{col}^4 - T_{sur}^4) + m_{out}C_p \mathcal{E} T_{amb}}{m_{out}C_p \mathcal{E}} \quad [5]$$

Energy analysis of the plenum wall shows that it receives radiated heat from the collector. Initially, the heated air flowing through the plenum and inside the chamber also contributes to the build-up of heat in the wall as defined by equation (6) and (7).

$$Q_{rad,wall-col} + Q_{conv,wall-in-air} + Q_{conv,wall-out-air} = m_{wall} C_{pwall} (T_{wall,ave} - T_{wall,init}) \quad [6]$$

$$\sigma_{sb} A_s \frac{(T_{wall,in}^4 - T_{col}^4)}{(\frac{1}{\epsilon_{plen}} + \frac{1}{\epsilon_{col}} - 1)} + h_{in} A (T_{out} - T_{wall}) + h_{in} A (T_{out} - T_{wall}) = m_{wall} C_{pwall} (T_{wall} - T_{wall,init}) \quad [7]$$



This phenomenon continues until the wall becomes hotter than the air from which point on the direction of convection heat transfer reverses. The rate at which heat is being absorbed however is faster than it is being released causing wall temperature to continue to increase. As the wall temperature increases, the rate of convection heat transfer to the air also increases while radiation heat from the collector decreases. Eventually the rate at which heat is received and given off equalizes leading to the attainment of a steady flow condition defined by equation (10).

Fig.5. Plenum Wall Energy Interaction

$$Q_{rad,wall-col} = Q_{conv,wall-in-air} + Q_{conv,wall-out-air} \quad [8]$$

$$\sigma_{sb} A_s \frac{(T_{wall,in}^4 - T_{col}^4)}{(\frac{1}{\epsilon_{plen}} + \frac{1}{\epsilon_{col}} - 1)} = h_{in} A (T_{wall,in} - T_{out}) + h_{out} A (T_{wall,out} - T_{final}) \quad [9]$$

$$T_{wall,in} = \frac{\sigma_{sb} A_s \frac{(T_{wall,in}^4 - T_{col}^4)}{(\frac{1}{\epsilon_{plen}} + \frac{1}{\epsilon_{col}} - 1)} - h_{out} A (T_{wall,out} - T_{final})}{h_{in} A} + T_{out} \quad [10]$$

Energy flow analysis of the air as it flows through the plenum shows heat being absorbed from the plenum walls (see equation (11)).

$$h_{out} A (T_{wall,in} - T_{out}) = \dot{m} c_p (T_{final} - T_{out}) \quad [11]$$

Equating the amount of heat conducted through the wall and that being convected away by the air on the drying chamber forms another expression for $T_{wall,out}$ (see equation (14)).

$$Q_{cond,wallin-wallout} = Q_{conv,wallat-air} \quad [12]$$

$$\frac{K_{wall}A(T_{wallin} - T_{wallout})}{x_{wall}} = h_{out}A(T_{wallout} - T_{final}) \quad [13]$$

$$T_{wallout} = \frac{\frac{K_{wall}T_{wallin} + h_{out}T_{final}}{x_{wall}}}{\frac{K_{wall}}{x_{wall}} + h_{out}} \quad [14]$$

IV. PRESSURE DROP THROUGH THE COLLECTOR

The total pressure drop of the UTC system is consists of losses through the plenum and across the collector plate. Kutcher [6] showed that the pressure drop through the collector plate may be computed using equation (15).

$$\Delta P_{col} = \xi \frac{1}{2} \rho V_s^2 \quad [15]$$

where :

$$\xi = 6.82 \left(\frac{1 - \sigma}{\sigma} \right)^2 \text{Re}_D^2$$

Plenum flow pressure drop on the other hand is consists of three terms. Frictional losses are developed as the air flows through it (see equation (16)).

$$\Delta P_f = f \rho \left(\frac{L}{D_H} \right) \left(\frac{V^2}{2} \right) \quad [16]$$

where :

f = Friction factor as determined from the moody's diagram

$$D_H = \text{Hydraulic Diameter} = \frac{4A_c}{P_c}$$

V = Average plenum velocity

With the air in the plenum being warmer and lighter than ambient air, buoyancy forces are developed aiding the flow of the air (see equation (17)).

$$\Delta P_b = \Delta \rho g H \quad [17]$$

The quantity of air increases as it flows through the plenum thus also increasing the bulk flow velocity. Applying the Bernoulli's equation leads to the development of the acceleration pressure drop term shown in equation (18).

$$\Delta P_{acc} = \rho \left(\frac{V_{\max}}{2} \right)^2 \quad [18]$$

where :

ρ = Air Density

V_{\max} = Maximum plenum velocity

Summing up all the terms mentioned leads to the attainment of the total pressure drop across an unglazed transpired collector. The required fan power is determined by multiplying the total pressure drop with the volume flow rate.

$$\Delta P_{tot} = \Delta P_{col} + \Delta P_{acc} - \Delta P_b + \Delta P_f \quad [19]$$

$$\Delta P_{tot} = \xi \frac{1}{2} \rho V_s^2 + \rho_{ave} \left(\frac{V_{\max}}{2} \right)^2 - \Delta \rho g h + f \rho \left(\frac{L}{DH} \right) \left(\frac{V}{2} \right)^2 \quad [20]$$

V. PERFORMANCE CURVE DEVELOPMENT

Collector performances are usually defined in terms of collector efficiency which represents the amount of useful heat collected per total solar insolation hitting the plate as shown in equation (21). This expression however failed to consider other auxiliary energy requirement of the system such as the fan power required to propel air through it which in some cases could be significant. Thus it was decided to adapt the proposed effective collector efficiency (η_{eff}) expression of Gupta et al. [8] detailed in equation (22).

$$\eta_{eff} = (Q_c - P_m / c) / Q_s \quad [22]$$

where :

Q_C = Collected Solar Energy

P_m = Pump required to propel the air through the system

c = Conversion factor to account for the conversion
of mechanical pump power to thermal energy

Q_s = Solar Insolation hitting the collector area

A computer program was developed to determine the average effective collector efficiency of the UTC system of the small scale drier being designed for the various hole diameter, pitch distance, plenum depth and suction velocity combinations considered.

The heat exchanged effectiveness was computed using equation (3). An iterative process was used to simultaneously solve equations (5), (10), (11) and (11) to yield the collector (T_{col}), plenum wall surfaces (T_{wallin} and $T_{wallout}$), plenum air (T_{out}) and exiting air temperatures (T_{final}). The total amount of solar heat absorbed was determined by summing up the $Q_{conv.col-air}$, $Q_{conv.wallin-air}$ and $Q_{conv.wallout-air}$. The corresponding fan power requirement was determined using equation (20).

Preliminary test computations showed that the heat collected by a UTC collector vary linearly with the amount of solar insolation hitting it. Figure 6 plots the air collected versus the solar insolation for a UTC collector system having the following dimensions: Hole Diameter=1.5mm; Pitch Distance=15mm; Plenum Depth=130mm; and Suction Velocity=0.03m/s.

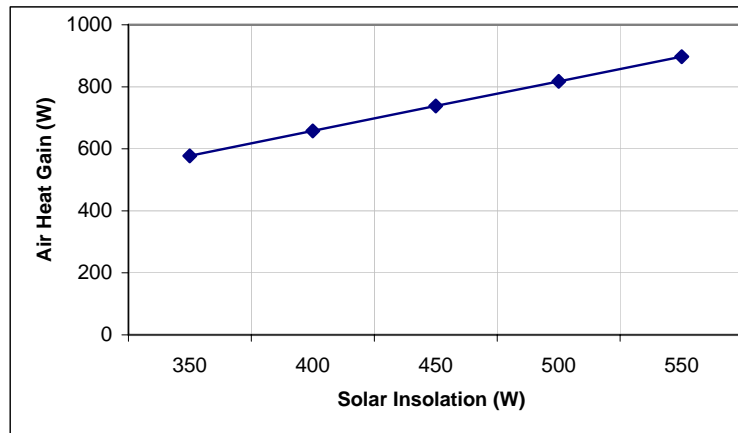


Fig.6. Air Heat Gain versus Solar Insolation

It follows then that solar insolation would have a very limited effect on the effective collector efficiency and would have a linear relationship with the temperature rise. These findings allowed us to base the performance curves on the annual average solar insolation instead of getting the average of the monthly collector efficiencies based on the monthly solar insolation. Also, predicted temperature rise may also be based on the annual average solar insolation. A linear regression equation relating the rate of change of temperature rise with variations in solar insolation could then be formulated to obtain necessary corrections. The average annual solar insolation in Palawan according to the New and Renewable Energy Atlas of the Philippines in the year 2000 is 452.98W/m^2 .

The program developed covered the parametric range given in Table 1.

Table 1. parametric range covered for the present study on UTC

Plate Dimensions	1.838 m x 1.2192 m
D	1.0 to 3.0 mm
P	5 to 25 mm
D_p	50 to 130 mm
V_s	0.02 to 0.06 m/s

The resulting collector efficiency and temperature rise for each set of UTC dimensions covered were plotted against the suction velocity.

VI. PERFORMANCE CURVES

Performance graphs showed that collector efficiency and temperature rise varied linearly with plenum depth showing a directly proportional relationship at average rates of 0.025% per mm and 0.0125°C per mm respectively. With this not so significant effect of plenum depth and to limit the number of graphs to be formed, it was decided to plot only results for plenum depths of 130 mm. We may however apply the rate changes mentioned earlier to obtain values for the other sizes in case greater precision is desired.

Graphs shown below indicate that collector efficiency tends to decrease with increasing hole diameter for plates with small (10mm and below) pitch distances. It produced very minimal effects for moderate pitch distances and tends to become directly proportional for bigger pitch distances. Face suction velocity on the other hand is directly proportional with efficiency except for plates with 1mm holes and pitch distances of 20 mm and higher. For plates with bigger holes, pitch distances tend to have a positive effect on collector efficiency contrary to those with smaller holes. This could be attributed to the following factors: bigger holes tend to have smaller net collection area which could be translated to lesser absorbed solar radiation. Bigger holes also results to smaller heat exchange effectiveness. For wider pitch distances however, effect of fan power requirements become more significant. Also smaller holes result in higher pressure drop across the UTC plate. The combination of these two factors tends to negate whatever positive effect of smaller holes and bigger pitch distances resulting to smaller effective collector efficiency for wider pitch distances.

It is also evident that suction velocity tends to increase collector efficiency. Although higher suction velocities lead to lower heat exchange effectiveness, the total heat is higher as greater quantity of air is heated. It could be recalled however that increasing volume flow rate results in bigger fan power consumption. This effect is amplified in very low porosity plates thus the decreasing efficiency in the 1mm hole curves of plates with pitch distances 25 and 20 mm (see figure 7a and 7b).

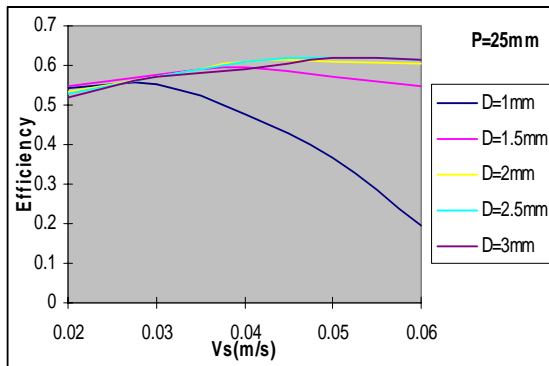


Fig.7a Collector Efficiency for P=25mm

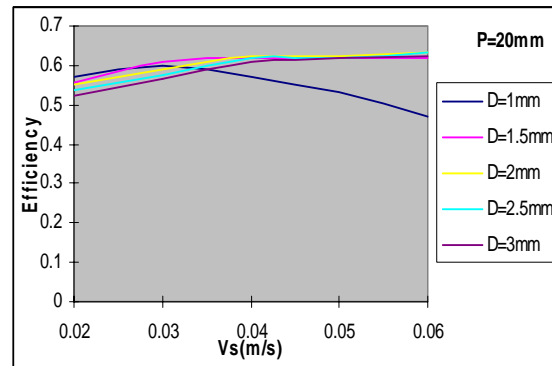


Fig.7b Collector Efficiency for P=20mm

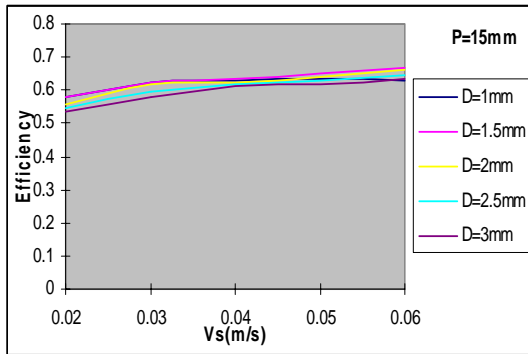


Fig.7c Collector Efficiency for P=15mm

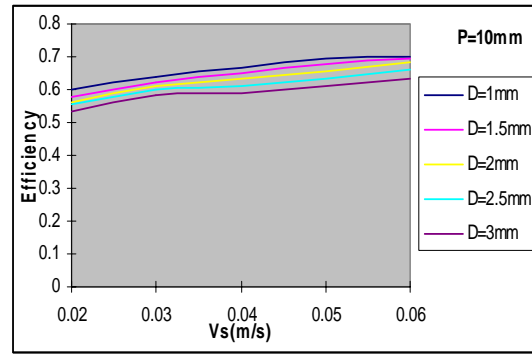


Fig.7d Collector Efficiency for P=10mm

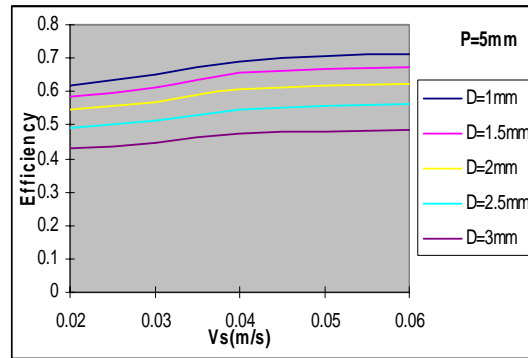


Fig.7e Collector Efficiency for P=5mm

Fig.7 Collector Efficiency Curves for Various Pitch Distances

Temperature rise tends to decrease with face suction velocity (V_s) and hole diameter (D). Greater suction velocities result in more air being heated leading to lower temperature rise. Smaller hole provides greater collection area causing higher temperature rise. Although wider pitch distance leads to greater absorbed radiation, it also translates to smaller heat exchange effectiveness. The latter tends to have a more significant contribution thus its negative effect on temperature rise.

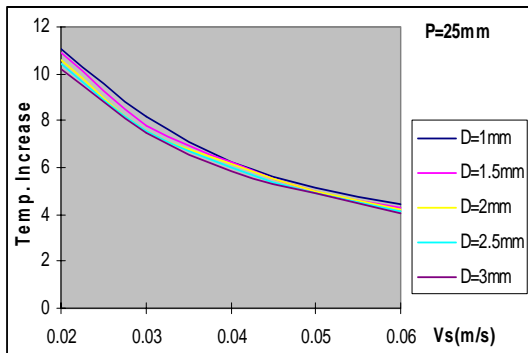


Fig.8a Average Temperature Rise for P=25mm

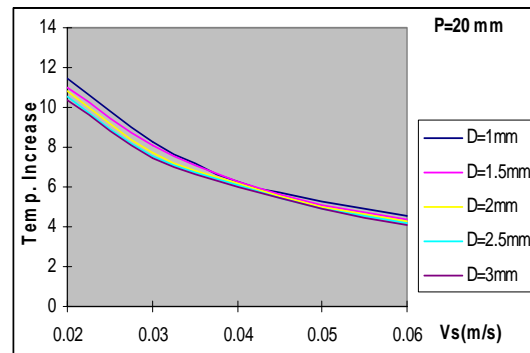


Fig.8b Average Temperature Rise for P=20mm

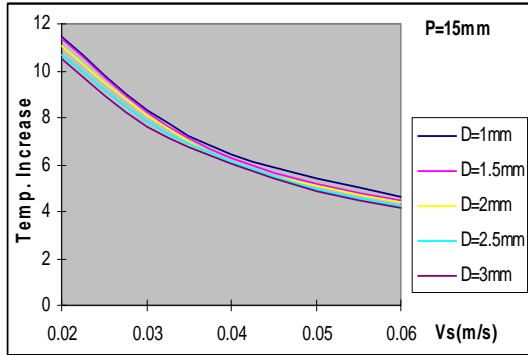


Fig.8c Average Temperature Rise for P=15mm

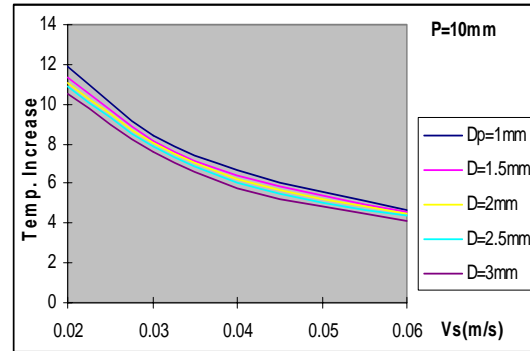


Fig.8d Average Temperature Rise for P=10mm

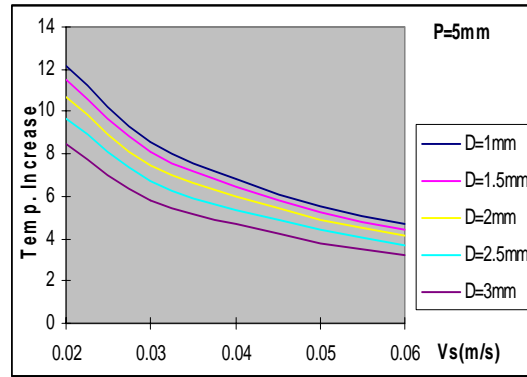


Fig.8 Average Temperature Rise Curves for Various Pitch Distances

The temperature rise graphs shown were based on the average annual solar insolation in Palawan of 452.98 W/m^2 . Simulation shows that the rate at which temperature rise varies with solar insolation changes linearly with the distance. A linear regression equation was formulated relating pitch distance with rate of temperature rise change with solar insolation. Equation (23) maybe applied to attain corrected values of temperature rise for solar insolutions not equal to the average annual value.

$$\frac{(T_{risecor} - T_{rise})}{(Q_{actual} - 452.98)} = 0.011747 + 0.00017P, \text{ P in mm. and } Q_{actual} \text{ in } \frac{\text{W}}{\text{m}^2} \quad [23]$$

It must be noted also that the curves developed coincided with the generalization of Kutscher [5] that both temperature and collector efficiency tends to become constant at suction velocities of 0.06 m/s and higher.

VII. USE OF THE PERFORMANCE CURVES

The performance curves obtained have been generated specifically based on the design parameters of the drying system having an area of 1.838m by 1.2192m . The systems being

developed would be used for different drying applications. A common chamber design would be used for the different systems with just the collector plate and plenum depth being varied to attain the desired heating effect. The curves generated would greatly aid the designers in identifying the optimum plate and plenum setting for a given situation. Once the required flow rate and drying air temperature is known for a given drying application, they could be plotted on the charts to the possible options that could deliver them. Efficiency curves may now then be used to determine which among the options is the most efficient. This process is demonstrated in the example discussed below. The example below demonstrates this process.

Assuming that the UTC solar dryer is required to supply 0.001344 Kg/s of drying air at 40 °C with the ambient temperature measured at 30 °C and average solar insolation at 452.98W/m². The UTC plate to be used could be easily identified using the charts available. The corresponding face suction velocity for the required mass flow rate is 0.025m/s. Referring to figure 8, we can identify candidate UTC plates for this application and these are as follows:

Plate 1: [P=20mm, D=1mm]; Plate 2: [P=10mm, D=1.5mm]; [Plate 3: P=5 mm, D=1.75mm]

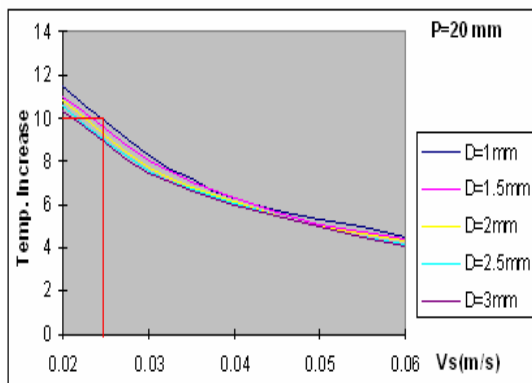


Fig.9a Plate 1

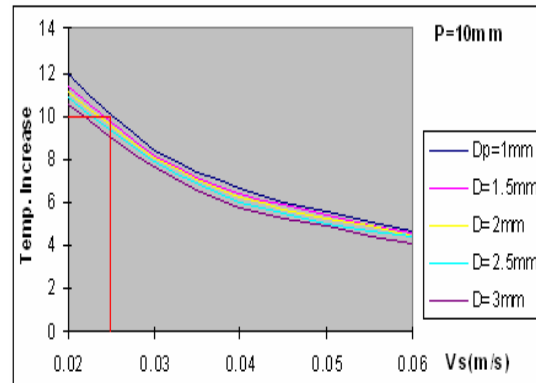


Fig.9b Plate 2

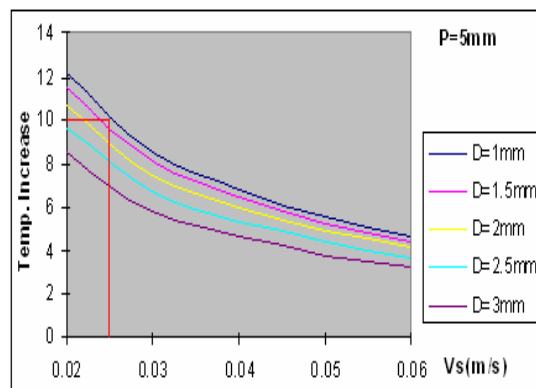


Fig.9c Plate 2

Fig.9 Temperature Rise Plots of the Candidate Plates

We can determine their corresponding efficiencies using the charts in figure 7. Plate 1 could be determined to have a collector efficiency of 59% while plate 3 have 60%. Plate 2 is shown to have a higher efficiency of 62%. Therefore in this case we choose candidate 2.

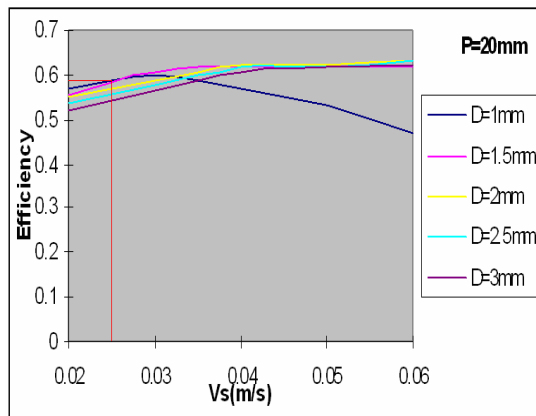


Fig.10a Plate 1

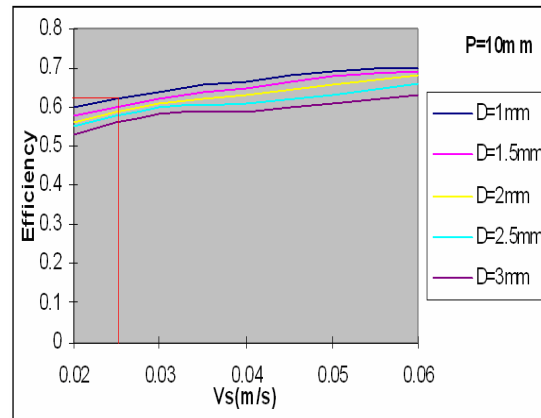


Fig.10b Plate 2

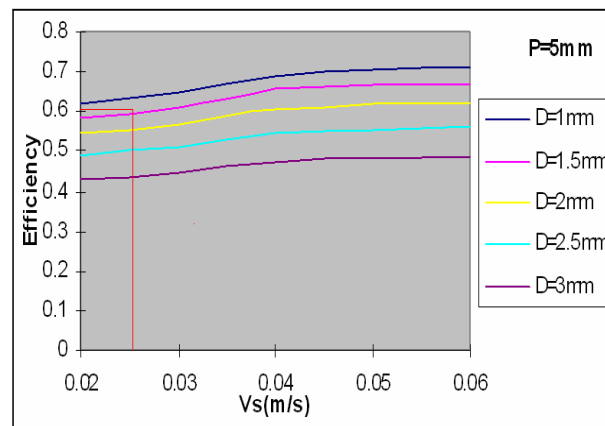


Fig.10c Plate 2

Fig.9 Effective Collector Efficiency Plots of the Candidate Plates

It must be noted however that this does not mean that plate 2 will collect more heat. Its higher efficiency instead could be traced to the lower fan power it would require. The plots used were based on a 130 mm plenum depth.

Decreasing the depth to 90 mm would bring down the output temperature to 39.5 °C and the efficiency to 61%. In case solar insolation fluctuates to 550W/m², the plate selected would provide a similar efficiency but increased output temperature of 41.63 °C computed using equation (23).

Nomenclature

A = UTC plate area including holes

A_s = UTC plate area excluding holes

C_p = Air specific heat

h = Convection coefficient

P = Pitch distance

D = Hole diameter

q_s = Incident solar radiation on the collector

Q_{abs} = Absorbed solar radiation

$Q_{conv,col-air}$ = Convection heat transfer from UTC plate to flowing air

$Q_{conv,wall-air}$ = Convection heat transfer from wall to flowing air

$Q_{rad,col-sur}$ = Radiation heat transfer from UTC plate to surrounding

$Q_{rad,wall-col}$ = Radiation heat transfer from UTC plate to surrounding

m_{out} = Suction mass flow rate

NTU = Number of transfer units

Nu_x = Nusselt number based on dimension x

Pr = Prandtl number

ΔP_{tot} = Total pressure drop

ΔP_{col} = Pressure drop across the UTC plate

ΔP_{acc} = Acceleration pressure drop

ΔP_b = Bouyancy pressure drop

ΔP_f = Frictional pressure drop inside the plenum

Re_D = Reynolds number based on dimension D

T_{amb} = Ambient temperature

T_{col} = UTC plate temperature

T_{out} = Suction air temperature

T_{surr} = Surrounding temperature

V_s = Face suction velocity

\mathcal{E} = Heat Exchange Effectiveness

$\mathcal{E}_{exp.}$ = Experimental Heat Exchange Effectiveness

$\mathcal{E}_{Kutscher}$ = Heat exchange effectiveness based on the Kutscher Model

$\mathcal{E}_{exp.mean}$ = Mean heat exchange effectiveness of an experimental run

$\mathcal{E}_{mod.}$ = Modified heat exchange effectiveness

\mathcal{E}_{wall} = Plenum wall emissivity

ϵ_{col} = Collector plate emissivity

E_{xratio} = Heat exchange effectiveness ratio = $\frac{\epsilon_{exp.}}{\epsilon_{Kutscher}}$

ξ = Nondimensional pressure drop

α_{col} = Collector plate absorptivity

σ_{sb} = Stefan Boltzman constant

σ = Plate porosity

ρ = Experimental air density

μ = Air viscosity

ν = Air kinematic viscosity

ACKNOWLEDGEMENT

The financial support by the Swedish International Development Co-operation Agency (Sida) for this study in the framework of the project “Renewable Energy Technologies in Asia - A Regional Research and Dissemination Programme” is gratefully acknowledged. We are also grateful to AIT for their assistance in finalizing this paper.

REFERENCES

- [1] PDMOE, 1978. Philippine Ministry of Energy (1978) Summary Report, Regional Workshop on Solar Drying.
- [2] Jensa S.O., Forma T. and Miljo A. (2001) Test of a Solar Crop Dryer 1st Ed., pp.4-23, Danish Institute of Agricultural Technology, Denmark.
- [3] Kutscher C. F., Christensen C., and Barker, G. (1991) Unglazed Transpired Solar Collectors: An Analytic Model and Test Results. Proceedings of ISES Solar World Congress 1991, Pergamon Press, Vol. 2, Part 1, pp. 1245-1250.
- [4] Kutscher C. F., Christensen C., and Barker, G. (1993) Unglazed Transpired Solar Collectors: Heat Loss Theory. ASME J. of Solar Eng., Vol. 115, No. 3, pp. 182-188.
- [5] Meer H.R. and Machlin A. (2001) Potential Applications Of Thermal Solar Energy In The Egyptian Food Processing Industry, USAID Project Report 263-0264, 37-51.
- [6] Kutscher C. F. (1994) Heat Exchanger Effectiveness and Pressure Drop for Air Flow Through Perforated Plates With and Without Crosswind, J. of Heat Transfer, 116, 391-399.
- [7] Biona J.B.M. and Culaba A.B. (2002) Performance Curve Generation of an Unglazed Transpired Collector System for Solar Crop/Fish Drying. M.Sc. Thesis, De La Salle University, Manila. pp. 46-51.
- [8] Gupta D., Solanki S.C. and J.S. Saini (1997) Thermohydraulic Performance of Solar Air Heaters With Roughened Absorber Plates. Solar Energy . 61(1), 33-42.