A theoretical model of a natural convection solar air heater

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ABSTRACT The performance of a flat plate solar air heater operating under natural convection was considered. A theoretical model was proposed and validated experimentally. The 2 m long x 0.5 m wide x 0.16m deep experimental model was fabricated using aluminium sheets and painted matt black and provided with a top glass sheet. Sides and rear insulation was 50 mm thick rock wool. Tests were conducted at varying tilt angles. Surface temperatures of the heat absorbing wall and glass cover and the air temperature and induced air velocity in the flow channel were measured and compared with predicted values. Satisfactory agreement was obtained between predicted and experimental results.

1. INTRODUCTION

Various theoretical models have been presented on solar air heaters for crop drying under forced circulation. For example, Vijeysendera et al. (1982), Biondi et al. (1988), Parker et al. (1993), Bala and Woods (1994), Ong (1995). However, there are very few reports on natural convection. The present mathematical model would attempt to predict mean surface temperatures of the heat absorbing wall and glass cover and temperature and velocity of the induced air flow in the natural convection solar air heater.

2. MATHEMATICAL MODEL

The air heater shown in Figure 1 consists of a blackened collector plate insulated at the bottom and covered by a transparent glass sheet on top. The channel between the absorber plate and glass cover forms an open-ended duct through which air could flow in between. Solar energy (H) absorbed by the collector plate (S_2) heats up the air in the channel and the hot air rises to the top. Inlet air temperature (T_{fi}), assumed equal to ambient temperature (T_a) exits at the outlet temperature (T_{fo}). The glass cover is assumed thin and with high thermal conductivity such that its temperature (T_g) is assumed uniform throughout its thickness. The solar collector is assumed thin at a temperature (T_{wi}). The plate is insulated and the bottom temperature of the casing is at a temperature (T_{wo}). One-dimensional heat flow (along the length of the collector) is assumed and all temperatures are assumed uniform across the width of the solar air heater. Heat capacity effects are assumed negligible and glass and collector temperatures are assumed uniform throughout.

Following the solar chimney model by Ong and Chow (2003), the heat balance from the thermal network shown in Figure 2 results in the following mean temperature matrix:

$$T_g: \quad (h_{gi} + h_{rwg} + h_{go} + h_{rs})T_g - h_{gi}T_f - h_{rwg}T_{wi} = S_1 + (h_{go} + h_{rs})T_a \tag{1}$$

$$T_f: \quad h_{gi}T_g - (h_{gi} + h_{wi} + \frac{mc_f}{\gamma WL})T_f + h_{wi}T_{wi} = -\frac{mc_f}{\gamma WL}T_{fi} \tag{2}$$

$$T_{wi}: -h_{rwg}T_g - h_{wi}T_f + \left(h_{wi} + h_{rwg} + \frac{h_{wo}k_{ins}}{k_{ins} + h_{wo}\Delta x_{ins}}\right)T_{wi} = S_2 + \left(\frac{h_{wo}k_{ins}}{k_{ins} + h_{wo}\Delta x_{ins}}\right)T_a \tag{3}$$

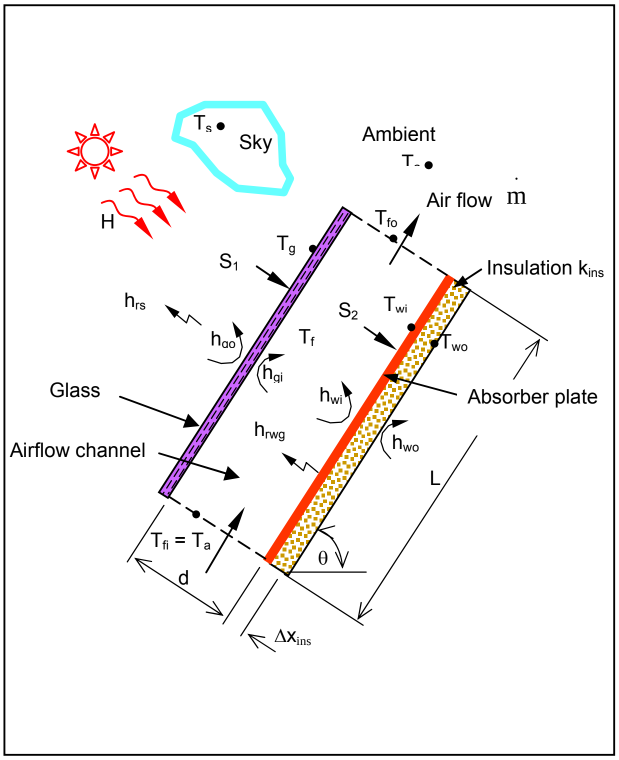

Figure 1. Longitudinal section of solar air heater.

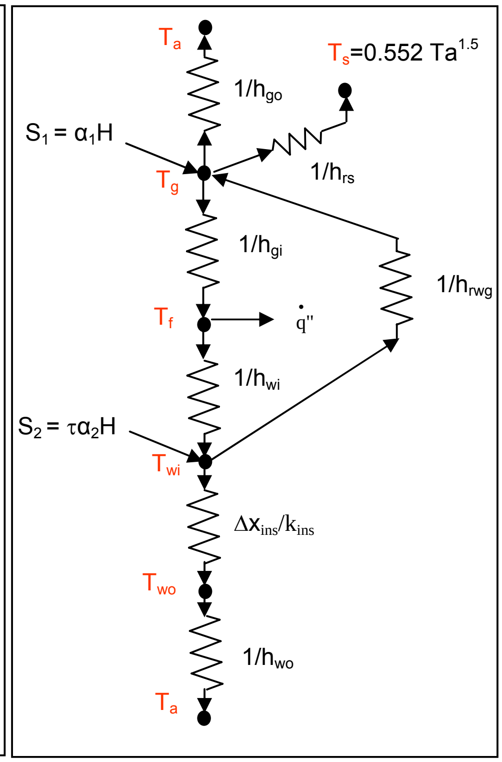

Figure 2. Thermal network for heat flow.

The axial mean air temperature was experimentally determined to follow the non-linear form:

$$T_f = \gamma T_{fo} + (1-\gamma) T_{fi} \tag{4}$$

with $\gamma = 0.8$.

Considering free convection and equating the buoyancy force (driving force due to the temperature increase between inlet and outlet of the solar air heater) to the sum of all flow pressure losses between inlet and outlet, the air mass flow rate was obtained from

$$m = \rho_i\, A \sqrt{\frac{2}{f_e}} \sqrt{g\, L \sin\theta \frac{(T_{fo} - T_{fi})}{T_f}} \tag{5}$$

The equivalent friction factor is given by

$$f_e = K_i + \left(\frac{\rho_i}{\rho_o}\right)^2 K_o + f \frac{L}{D} \left(\frac{\rho_i}{\rho_o}\right)^2 \tag{6}$$

where for laminar flow

$$f = \frac{64}{\mathrm{Re}} \tag{7}$$

The radiation heat transfer coefficient from the top glass surface to the sky referred to the ambient temperature may be obtained from

$$h_{rs} = \sigma\varepsilon_g (T_g + T_s)(T_g^2 + T_s^2)(T_g - T_s)/(T_g - T_a) \tag{8}$$

The sky temperature is given by Swinbank (1963) as

$$T_s = 0.0552 T_a^{1.5} \tag{9}$$

The convection heat transfer due to wind is given by McAdams (1994) as

$$h_{go} = h_{wo} = 5.7 + 3.8\ u_{wind} \tag{10}$$

The radiation heat transfer coefficient between plate and glass cover may be obtained from

$$h_{rwg} = \sigma(T_g^2 + T_{wi}^2)(T_g + T_{wi})/(1/\varepsilon_g + 1/\varepsilon_w - 1) \tag{11}$$

Hollands (1976) recommended for inclined plates the average heat transfer coefficient

$$\overline{Nu} = 1 + 1.44\left[1 - \frac{1708\,(\sin 1.8\,\theta)^{1.6}}{Ra\,\cos\theta}\right]\left[1 - \frac{1708}{Ra\,\cos\theta}\right]^{+} + \left[\left(\frac{Ra\,\cos\theta}{5830}\right)^{1/3} - 1\right]^{+} \tag{12}$$

for tilt angles of 0° to 75° (θ is set to 75° for tilt angle of 90°). The terms denoted by the exponent "+" in the square brackets are taken as zero when negative. The coefficients h_{gi} and h_{wi} are assumed equal and calculated from the above equation. All property values are evaluated at average surface – air temperatures $0.5(T_{wi} + T_f)$ and $0.5(T_g + T_f)$, respectively.

The solar radiation heat flux absorbed by the glass cover is given by

$$S_1 = \alpha_1\ H \tag{13}$$

and the solar radiation heat flux absorbed by the blackened plate by

$$S_2 = \tau\ \alpha_2\ H \tag{14}$$

3. EXPERIMENTAL INVESTIGATION

The south facing 2.05 m long x 0.5 m wide x 0.16 m deep experimental model was fabricated from 1.2 mm thick aluminium sheet painted matt black and covered with a 5 mm thick transparent glass sheet leaving an air gap 0.4 m wide x 0.11 m deep between the absorber and the glass. Insulation was 50 mm thick rock wool. Copper-constantan thermocouples were employed to measure temperatures at 4 points along the surface of the heat absorbing wall, 3 points on the glass surface and 7 points along and in the middle of the air gap. A hot wire anemometer was used to measure the induced air velocity in the air gap near the outlet section. Instantaneous solar radiation was measured by a Kipp and Zonen solarimeter placed on the glass surface Experiments were conducted at various tilt angles from 20° to 90°.

In the tropics, because of the highly fluctuating nature of solar radiation, it is very difficult to obtain near-steady radiation. Hence instantaneous solar radiation and temperatures were recorded at one-minute intervals and plotted continuously over a period of 2 hours. "Quasi-steady" conditions were assumed when solar radiation did not fluctuate by more than 50 W m^{-2} over a five-minute duration. Once this condition was established, all the data recorded over the five-minute interval were averaged to represent the "steady" value of each parameter concerned.

Preliminary observations showed that the temperature distributions across the width of the glass surface and of the absorber plate were uniform to within $\pm$ 1°C. The temperature distribution of the air in the duct between the glass and the absorber surfaces was uniform to within $\pm$ 1.5°C. The air flow velocity fluctuated over a wide range, by as much as $\pm$ 30% owing to wind factor.

A typical set of axial temperature distribution for the glass and wall surfaces and the air flow is shown in Figure 3 for a tilt angle of 60° and at a radiation level of 800 W m^{-2}. All the temperatures were observed to increase non-linearly as we progressed along the air heater. Also, they exhibited a tendency to decrease near to the outlet of the air heater. This tendency was attributed to end cooling effects and mixing of the outgoing hot air with the ambient air due to recirculation. The glass surface exhibited quite a uniform temperature profile and a uniform mean glass temperature could be assumed and calculated using the arithmetic average of the three temperature probes. The variation was found to be about ± 2.5°C from inlet to outlet. Quadratic equations were fitted to the data obtained for the wall and air temperatures. Bulk mean temperature values for wall and air were obtained by integrating the equations obtained over the length of the air heater. A correction was applied in the case of the last air temperature data. The temperature value here was taken to be equal to the previous probe temperature. From Figure 4 it could be seen that mean temperatures were about 69° ± 6°C for wall, 44° ± 5°C for both air for glass. It was generally observed that the temperature difference between air and glass was very small and that the wall temperature was always higher than the air or glass temperature throughout the length of the air heater.

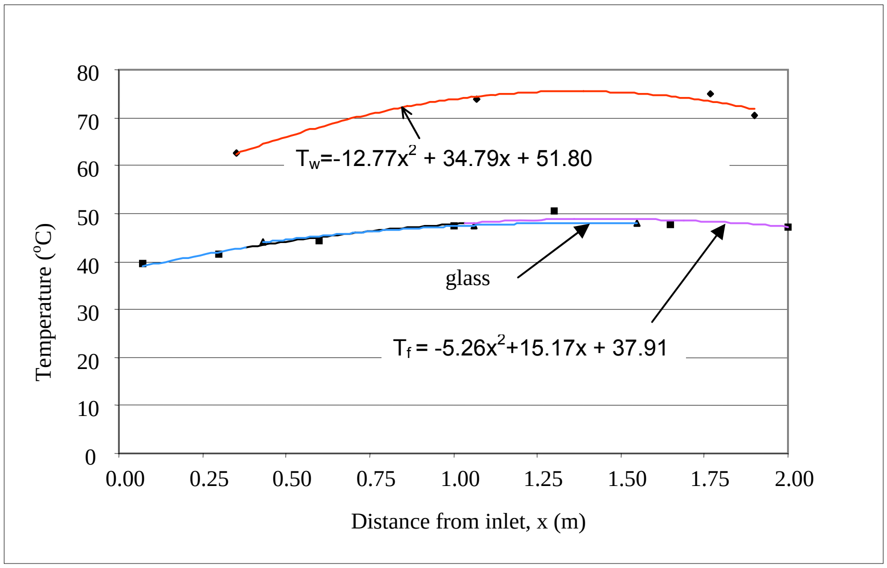

Figure 3. Air, glass and wall temperatures along solar air heater (60° tilt, H = 800 W m^{-2}).

A typical plot of mean glass, wall and air temperatures, outlet air temperature, outlet air flow velocity and instantaneous efficiency against solar radiation is shown in Figure 4 for a 20° tilt angle. Results were non-dimensionalised with the ambient temperature in order to minimize the effects of varying inlet air temperature. Absolute temperatures in °C were employed. Linear regression was adopted to obtain the best lines to represent the variation of temperature versus solar radiation as shown. It was observed that all the variables plotted increased with incident solar radiation. Wall temperature was higher than glass or air temperatures. Glass temperature was observed to be lower than air temperature initially and at low radiation levels. At higher radiation glass temperature was higher than air temperature. At 90° tilt, glass temperature was always higher than air. This indicated that the air flow in the duct was non-symmetrical and that the air was initially heated up by the heat absorbing wall and then transferred some heat to the glass as it flowed upwards.

Figure 5 shows the predicted wall, glass and air temperatures as well as the outlet velocity at 550 W m^{-2} for the various tilt angles. An ambient temperature of 30°C was employed to obtain absolute rather than temperature ratio for comparison. The results showed that the inclination of the heater did not have much of an effect on the performance. The effects could also have been clouded by the small differences in temperatures experienced and the widely fluctuating nature of the flow due to solar intensity and wind conditions. We would expect a higher velocity with larger tilt angles because of the longer vertical distance between the inlet and outlet of the collector giving rise to a greater driving buoyancy force.

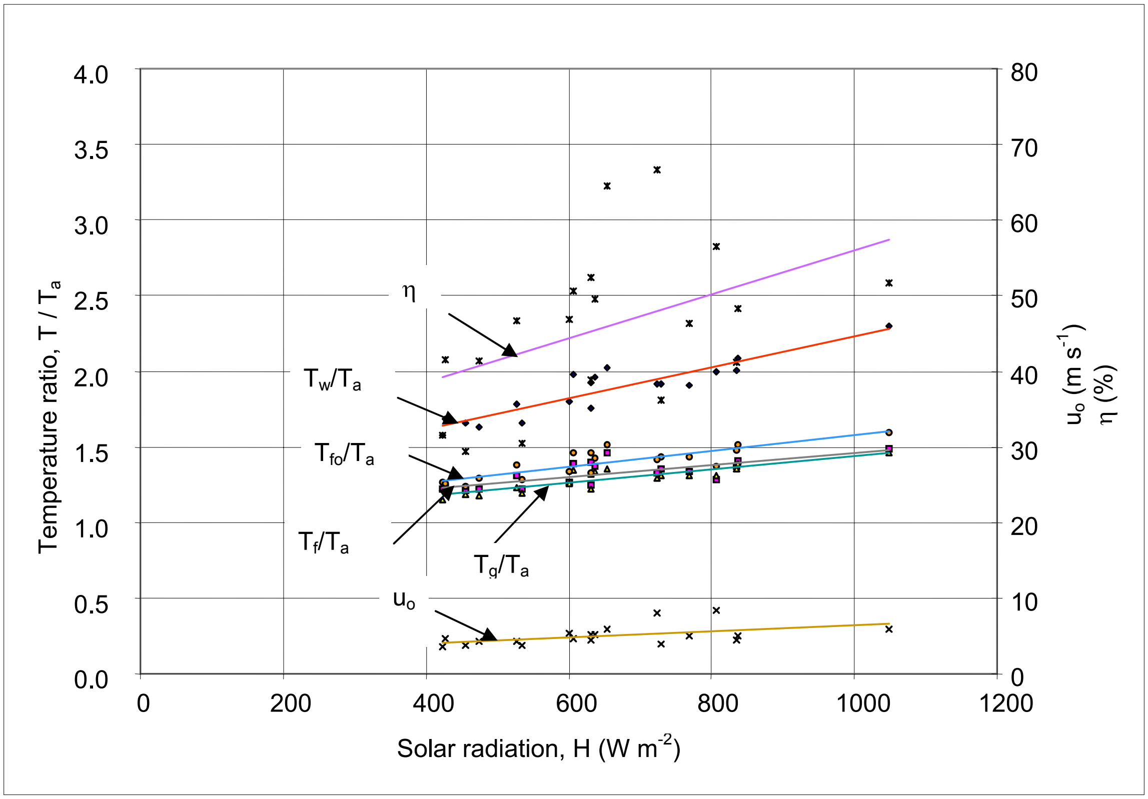

Figure 4. Wall, glass and air temperatures and outlet air flow velocity vs solar radiation (20^o tilt angle).

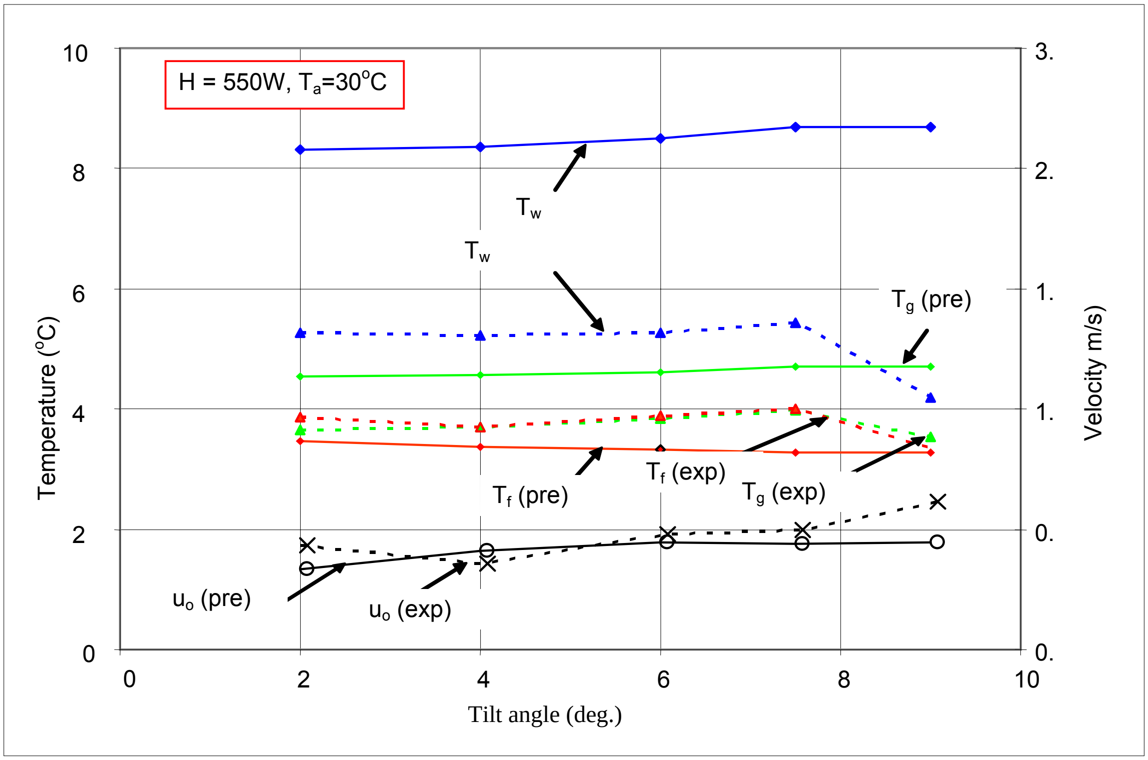

Figure 5. Experimental and theoretical results compared.

Comparison of the experimental and predicted results in Figure 6 for the various tilt angles at a solar radiation level of 550 W m^{-2} showed that:

(i) Glass temperatures were over-predicted by about 10^{o}C.
(ii) Wall temperatures were over-predicted by about 30^{o}C.
(iii) Mean air temperature was quite well-predicted to within 2 - 5^{o}C.
(iv) Air flow velocity was quite well-predicted to about $\pm$ 0.17 m s^{-1}.

Considering that the measured air flow rate fluctuated by as much as $\pm$ 30%, the difference between experimental and predicted values of about $\pm$ 10% could be considered satisfactory. Satisfactory comparison for mean air flow temperature was also obtained. However, the comparisons for glass and wall temperatures were not good. The over-prediction for these temperatures could be partially due to the heat transfer correlations employed.

Figure 6 compares the experimental results with predicted results using 5 times the normal heat transfer coefficient values between the solid surfaces and the air stream. It could be seen that the predicted wall surface temperature was lowered by as much as 20^{o}C, the glass temperature lowered by 6^{o}C and the air temperature increased by 3^{o}C. The velocity increased by about 0.1 m s^{-1} All these point to a closer agreement between experimental data and prediction.

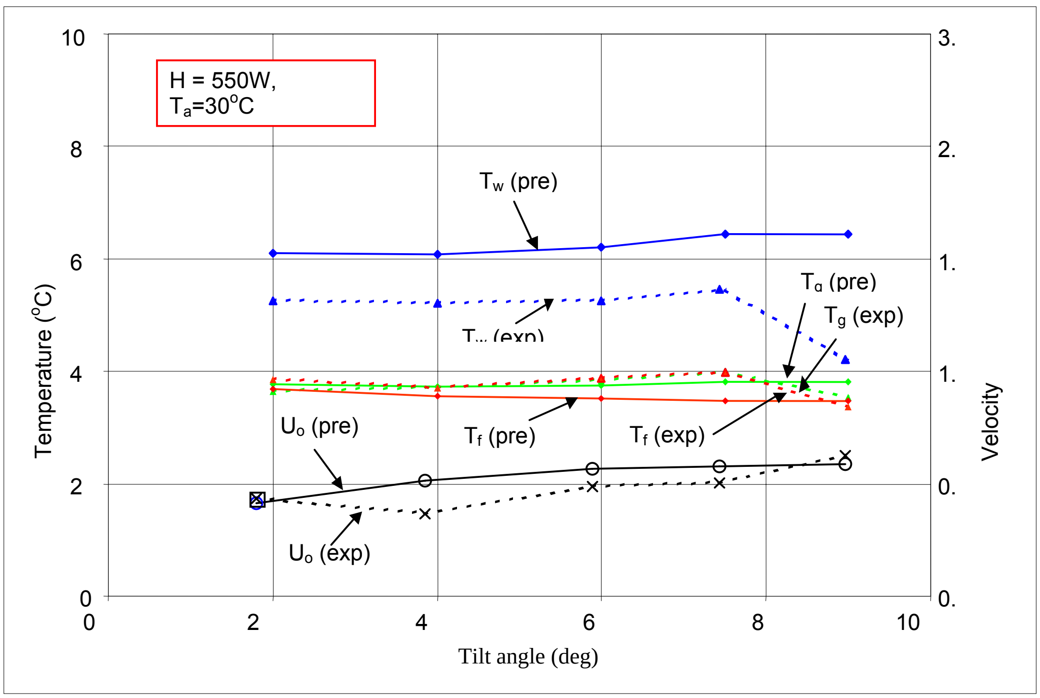

Figure 6. Experimental and theoretical results with 5 x Nusselt number.

4. CONCLUSIONS

A theoretical model of a natural convection solar air heater has been proposed. An experimental investigation was also conducted to compare with predicted results. Satisfactory agreement was obtained between predicted and experimental results for the air flow temperature and velocity. Wall and glass temperatures were over predicted.

ACKNOWLEDGEMENTS

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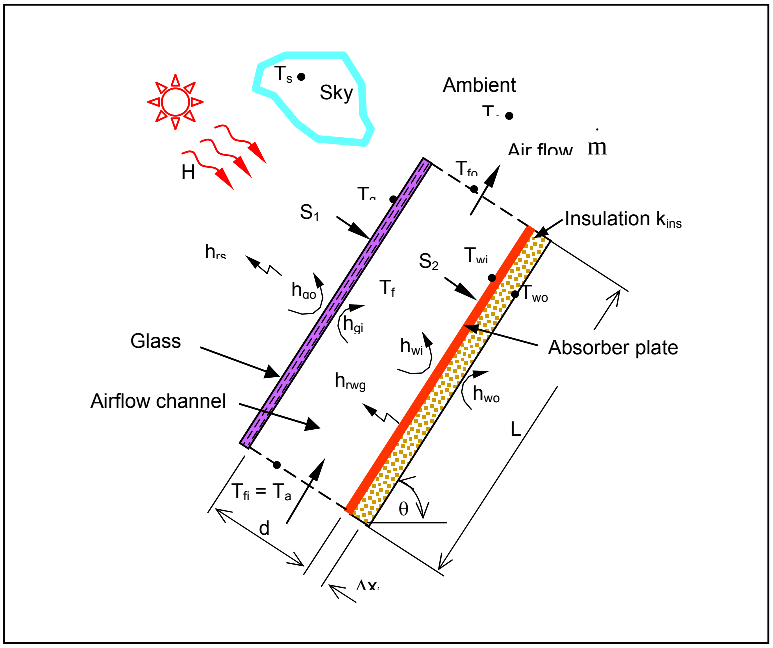

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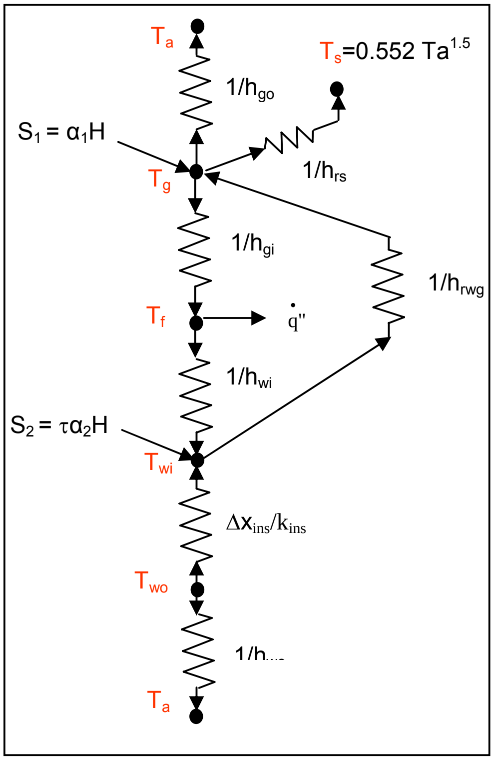

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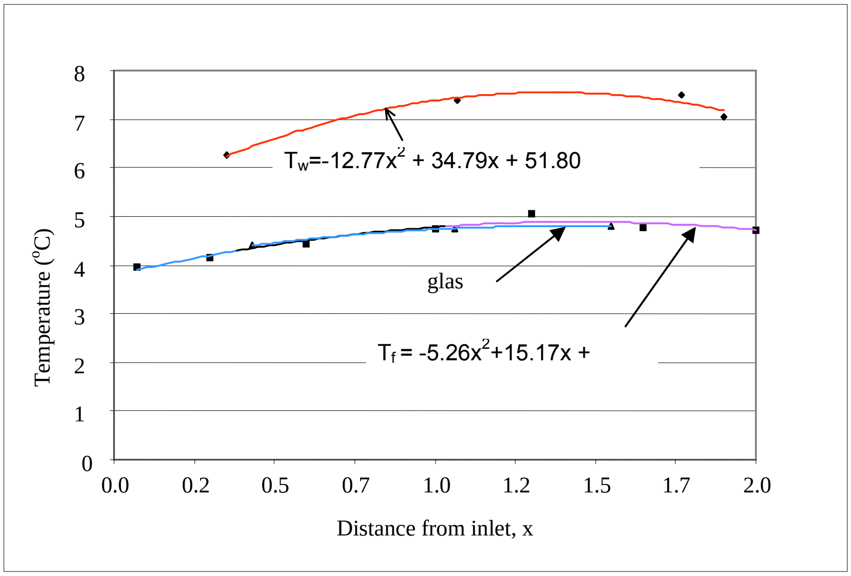

Figure 3. Air, glass and wall temperatures along solar air heater (60^o tilt, H = 800 W m^{-2})

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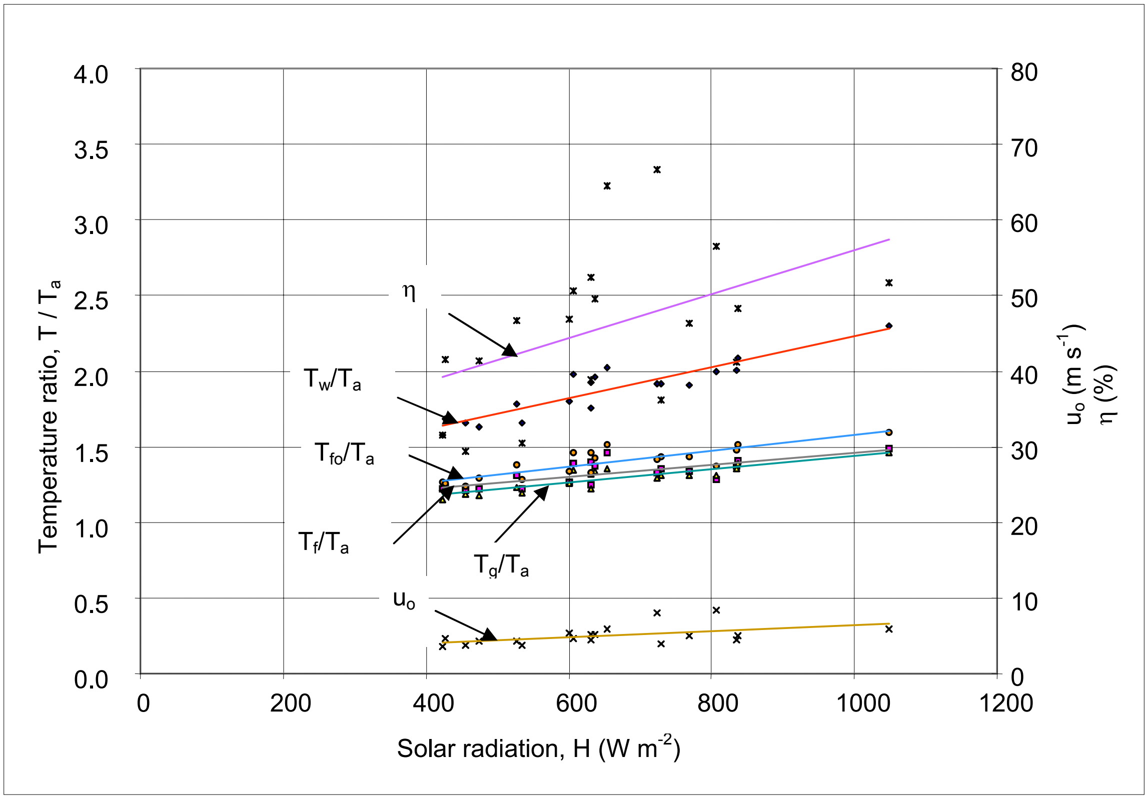
Tf



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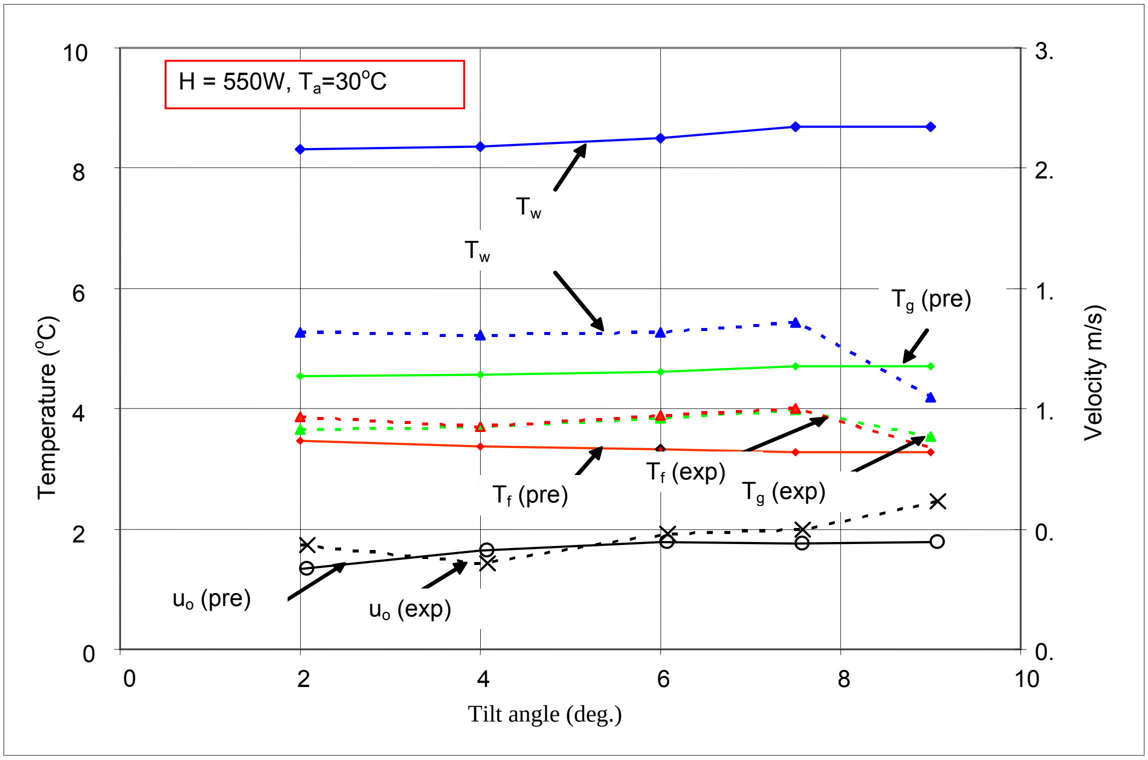

Figure 5. Experimental and theoretical results compared (H = 550 W m^{-2}, T_a = 30^{o}C).

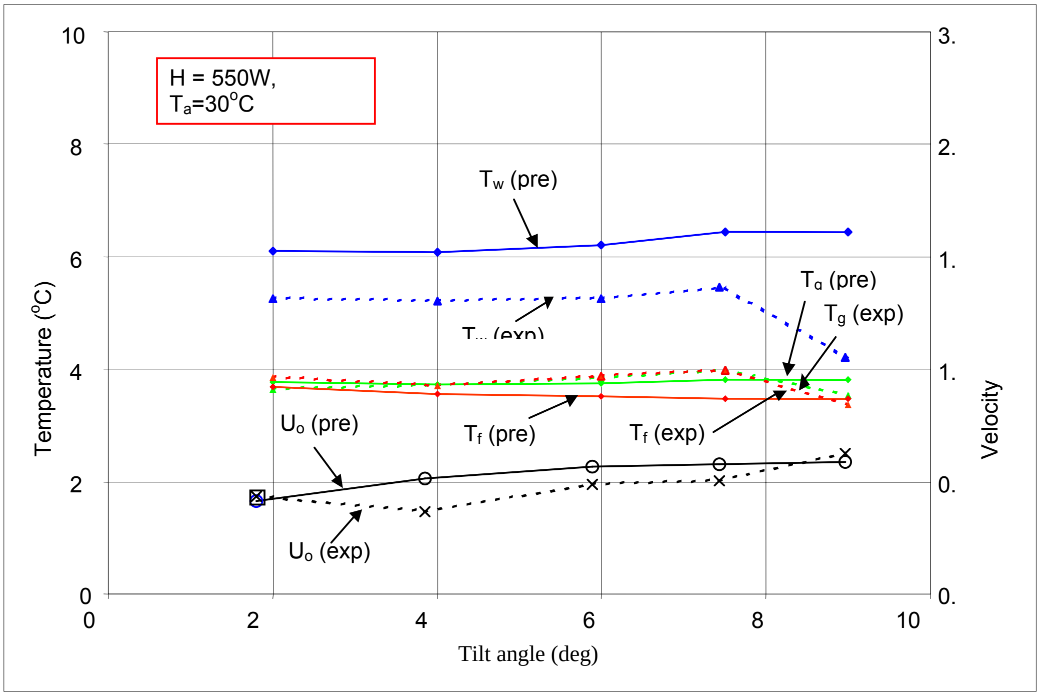

Figure 6. Experimental and theoretical results with 5 x Nusselt number (H = 550 W m^{-2}, T_a = 30°C).