

Modeling of a Single Pass Solar Air Heater

¹S. Shanmuga Priya*, ²Stefan Bauregger, and ³I. Thirunavukkarasu

Abstract: We have fabricated and analysed three different types of front pass natural circulation solar air heating collectors in test conditions of Manipal, Karnataka. Based upon the models constructed and the experiments conducted, analysis of data has been done which helps us to conclude about the most efficiently erected solar air heater. Different parameters were considered to reach to these results, namely ambient air temperature and weight loss. This paper gives the basic knowledge and processes behind the analysis and modeling of a single pass solar air heater. The general idea behind the processes will be shown with MATLAB results that have been used to code the model.

Keywords: natural circulation; solar air heaters; modeling

1. INTRODUCTION

In this study, three different types of solar air heating collectors were designed and fabricated for comparison of their thermal performances as stated by reference [9]. They include single plastic glazing of black painted hardboard absorber, flat plate absorber and zigzag plate absorber. Each unit consist of a frame made up of hardboard, vent holes, insulation, absorbing surface was made up of black coated aluminium sheet and clear plastic glazing. They were tested simultaneously for its performance. The best one was found out according to the results obtained.

When the air passes through the air heater, two forms of heat transfer take place simultaneously. Fig. 1 shows the estimated heat flow processes for the whole system. For the calculation this system will be divided into K parts and then processed step by step.

At the entry point air temperature and wall temperatures will be regarded as unique with the ambient air temperature. Starting from there the different processes will take part on the side walls, the aluminum sheet and the polyethylene cover. Since the air heater is tilted by the angle of 30 degrees the convection heat flow is divided into natural and forced. The relevant part of the heat will still be transferred through forced convection (Nusselt around 127 compared to only 4 for natural). Polyethylene transmits solar radiation very good (alpha of 0.95)

Since the cover in this case consists out of polyethylene and is very thin, the heat absorption and the heat emission (h_{rs}) will be considered 0. Calculations have been done with both of the parameters unequal to zero and no difference in the result has been shown. The bottom in our case is seen as not isolated hence the bottom loss coefficient will be set equal to the heat transfer coefficient for the wind (h_w). Solar radiation will be transmitted by almost 100 % through the polyethylene. Still the reflection, transmission and absorption values for blackened aluminum will be guessed. The heat transfer coefficients are the

^{1,3} Associate Professor, ²IAESTE Student Intern

¹ Department of Chemical Engineering

³ Department of ICE, Manipal Institute of Technology, Manipal University, Manipal, Karnataka, India

* email : shanpriyanitt@yahoo.co.in, shan.priya@manipal.edu

essential values for the outcome. They rely on the materials, the air velocity, the flowing liquids and the temperature difference.

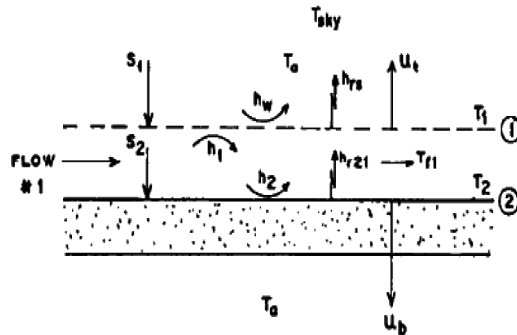


Figure 1: Heat transfer in air heater

Reynolds Number

This parameter indicates whether the fluid follows a steady pattern, a partially turbulent or a fully turbulent flow. It is calculated by the formulae

$$Re = \frac{c \cdot d}{\nu} \quad (1)$$

with c as air velocity, d as characteristic length and ν as kinematic viscosity of the fluid.

Prandtl Number

The Prandtl number shows the ratio of kinematic viscosity to thermal diffusivity.

$$Pr = \frac{\nu}{a} \quad (2)$$

with ν as kinematic viscosity – a as thermal diffusivity.

Nusselt Number

In heat transfer the Nusselt number is the ratio of convective to conductive heat transfer normal to the boundary. It depends on the Prandtl and Reynolds number. Since the Reynolds number differentiates between laminar, partial laminar and turbulent flow the equations for the Nusselt number vary too.

Nusselt laminar with $1 < Re < 10^3$:

$$Nu_{L',lam} = 0.664 \cdot \sqrt[3]{Pr} \cdot \sqrt{Re_{L'}} \quad (3)$$

with L' as characteristic length (here: vertical plate $L' = d$ with d as distance into heater to the calculated infinite part).

Nusselt turbulent with $10^5 < Re < 10^7$:

$$Nu_{L',turb} = 0.037 \cdot Re_{L'}^{0.8} \cdot Pr^{0.48} \cdot (T/T_w)^{0.121} \quad (4)$$

with L' as characteristic length – T as air temperature – T_w as wall temperature.

Grashoff Number

The Grashoff Number is a dimensionless parameter i.e. the ratio of buoyance to viscous force acting on a fluid.

$$Gr = \frac{g \cdot l^3 \cdot \beta \cdot (\vartheta_w - \vartheta_0)}{\nu^2} \quad (5)$$

with g as gravitation constant, l as characteristic length, ϑ as temperature and

$$\beta = \frac{1}{T_0} \quad (6)$$

with T_0 as temperature of the fluid in a distance, where no heat transfer with the wall takes place. This equation is only valid for ideal gases, which is reasonable in this case because of the small temperature changes during the process.

Rayleigh Number

The Rayleigh number can be compared to the Reynolds number since it also indicates the flow type. The transition area between laminar and turbulent flow lies within $10^8 < Ra < 10^9$.

$$Ra = Gr \cdot Pr \quad (7)$$

For this model the limit $Ra \leq 10^5$ has been estimated.

Heat Transfer Coefficient

All the shown parameters can now be formed into the needed heat transfer coefficients. The equation

$$Nu = \frac{\alpha \cdot d}{\lambda} \quad (8)$$

shows the connection between the Nusselt number, the characteristic length (d), the thermal conductivity (λ) and the heat transfer coefficient (α). The used Nusselt number varies, depending on the local circumstances. In case of mixed natural and forced convection one more equation has to be taken into account.

$$Nu_l = \sqrt[3]{Nu_{l,f}^3 + Nu_{l,n}^3} \quad (9)$$

with $Nu_{l,f}$ as Nusselt due to forced convection and $Nu_{l,n}$ number as Nusselt number due to natural convection. This connection only exists in the laminar state.

Apart from heat transfer through convection, radiation was mentioned. The intensity of the sun in [W/m^2] can be easily taken into account by simply multiplying the value with the transmission constant of polyethylene.

Constants

A set of values in this calculation have been seen as constant. Either due to the laws of physics or because of the process allowing to make restrictions. For example the density depends on the temperature of the gas. Still the difference between λ at 30 °C and 60 °C is negligible. Fig 2 gives the transmission coefficient with respect to the wavelength.

air constants

R = 287;

viscosity = 1.9230e-05;

specific heat capacity = 1006.4;

thermal conductivity = 0.2624;

Prandtl = 0.71008;

density air = 1.078;

polyethylene constants

reflection coefficient = 0.05;

transmission coefficient = 0.94;

absorption coefficient = 0.01;

aluminum constants

reflection coefficient = 0.55;

transmission coefficient = 0.05;

absorption coefficient = 0.4;

specific heat capacity = 900;

general constants

Boltzmann constant = $5.67 \cdot 10^{-6}$;

According to the setup, the aluminum sheet was black colored hence the absorption (α_{AL}), transmission (τ_{AL}) and reflection (ρ_{AL}) values could only be estimated. τ_{AL} could not be set to 0 since the calculation would end up with an error message (multiplied by 0 = NaN in MATLAB). Due to the small value chosen, the effect is not relevant. The following picture shows the transmission coefficient in relation to the wavelength of polyethylene.

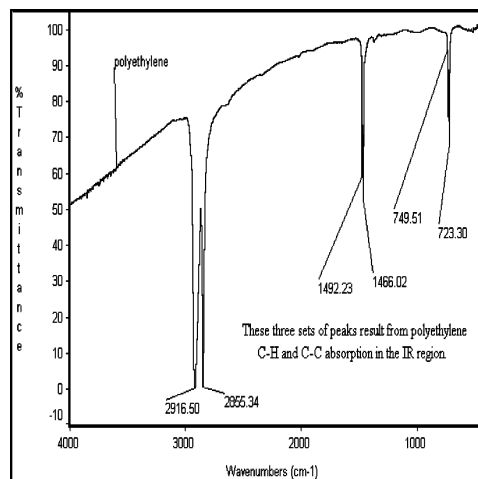


Figure 2: Transmission coefficient with respect to wavelength

2. MATLAB MODEL

The model has been built in three different ways. First approach was programmed with the amount of steps being 1 ($K = 1$). As a result, the model was not exactly precise. The general equations for Nusselt, Prandtl, etc. were adapted for the other models but the program was abandoned. The second model included iteration already but was not considered good enough, since the results varied too much after changing the entry parameters. The third approach changed the transfer equations.

Program Flow

The program calls the configuration file where the user can set up the amount of steps (K) and the xls-file, where data like radiation intensity will be read from. After that the constants will be read into the workspace for further calculation. The data m-file will actually handle the predefined xls-file and save the data vectors. The next step contains the calculation and preparation of some parameters through the params.m file. Not only basic equations are done at this point but also the initiations of the temperature matrixes take place. By now, all the necessary values and parameters for starting the calculation are available in the workspace.

The model assumes that the temperatures of the walls surrounding the air streams are uniform and the temperatures of the air streams vary linearly along the collector. A long collector can be assumed to be equally divided into a finite number of short collectors, or sections. The wall and mean air temperatures of the first section are initially guessed and specified and in all cases they are set equal to the inlet temperature. Now the heat transfer coefficients are evaluated, according to the initially-guessed temperatures of air, cover and aluminum sheet. An iterative process is then created and the mean temperatures for the section is calculated using the equations derived by employing a standard package matrix-inversion subroutine. The newly-calculated temperature values are then compared with the previously assumed ones. The iterative process is repeated until all consecutive mean temperatures differ by less than 0.01°C .

Here another section of the collector, with length equal to the previous section, was added to the end of the first section. The mean wall and air temperatures of the second section of collector are then set equal to the mean wall and air temperatures of the section before it. The inlet air temperature of the second section is set equal to the outlet air temperature of the first section. The iterative procedure is repeated until all the sections of the given collector are considered. By this procedure, wall and mean air temperatures can be predicted for the complete length of collector.

The start of the program considers the first section. An initial guess of the mean wall temperatures and mean air streams temperatures are made. For the first section, these temperatures are specified equal to ambient temperature. (For subsequent sections, the initial temperatures are specified equal to the respective temperature values of the respective previous sections.) An iterative process is then initiated. The program calculates all the required heat transfer and heat loss coefficients based on the initially-guessed temperatures. The matrices $[A]$, $[T]$, and $[B]$ are then set up. A standard matrix inversion subroutine is then called to invert $[A]'$ to determine a new set of temperature matrix $[T']$. Each new temperature value in the matrix $[T']$ is then compared with the corresponding initially-guessed $[T]$.

If the difference between any corresponding new and old values exceeds 0.01° , the iteration is stopped and the old temperatures are then replaced with the newly-calculated ones and taken to be the required temperatures at the section concerned. At the end of the iteration, the program calculates the outlet temperatures of the two air streams and also the efficiencies at the end of the current section. It then sets the air temperatures at the inlet of the next section equal to the outlet air temperatures of the current section. The program then proceeds to look at the next section. The iterative solution procedure to determine the

new mean temperatures of this next section is then repeated. By this repetitive and iterative process, the required temperatures along the entire length of the solar heater could be obtained, section by section. Normally, the number of iterations required is not more than two or three. For the calculation the equation

$$\begin{bmatrix} (h_1 + h_{r21} + U_i) & -h_1 & h_{r21} \\ h_1 & -(h_1 + h_2 + \Gamma_i) & h_2 \\ -h_{r21} & -h_2 & (h_2 + h_{r21} + U_b) \end{bmatrix} \begin{bmatrix} T_1 \\ T_{f1} \\ T_2 \end{bmatrix} = \begin{bmatrix} U_i T_a + S_1 \\ -\Gamma_i T_{f1,i} \\ S_2 + U_b T_a \end{bmatrix}$$

has been used. Corresponding to the abstract, this would be the expanded form of:

$$[A] [T] = [B]$$

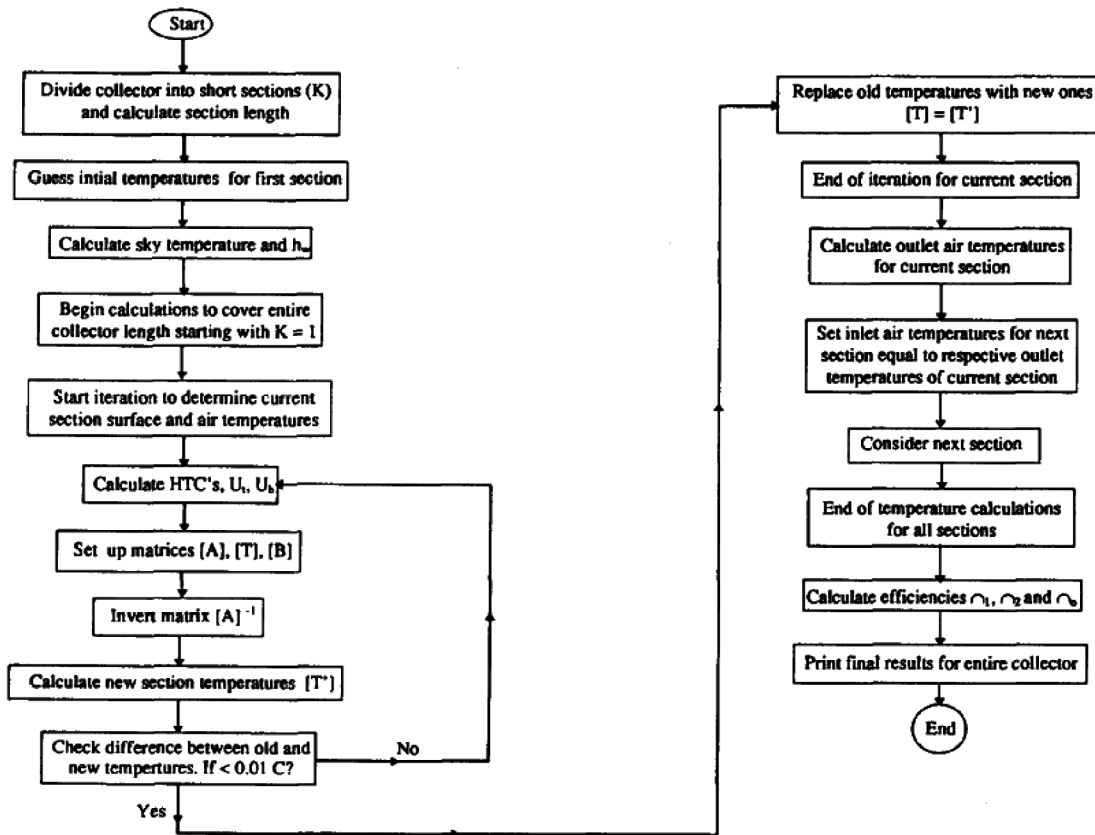


Figure 3: Flowchart

The flowchart shown in Fig 3 shows the logical steps to be done. This concept was adapted and reformed for the purpose of calculating the air outlet temperature of the solar air heater. In comparison with the experimental data the offset of the result lies within 2°C.

3. RESULTS AND DISCUSSION

The iterative calculation of the whole system hands back a set of data with the interval according to the steps. The following figure 4 shows the Nusselt number and its unsteady flow in the beginning of the graph. This is a result of the calculation where many of the beginning parameters are guessed and sometimes also equations deactivated due to derivation by 0.

These Fig 5 show the outcome on the temperature flow. T_1 being the cover temperature, $T_{-f,1}$ can be seen as the air temperature and T_2 expresses the collector temperature (compare beginning of the report where the system is drawn). h_1 is the heat transfer value between air flow and cover. We can see the big difference between the first values and the following values. The same image can be seen in the next plot – the h_2 coefficient. It concludes the transfer value between air and collector. The h_{21} value shows a different outcome. It can be compared to the temperature flows, which is not surprising since it correlates only with the temperatures. The next step is going to ease down the first few values to moderate the temperature differences and after that compare the newly calculated values to the empirical data.

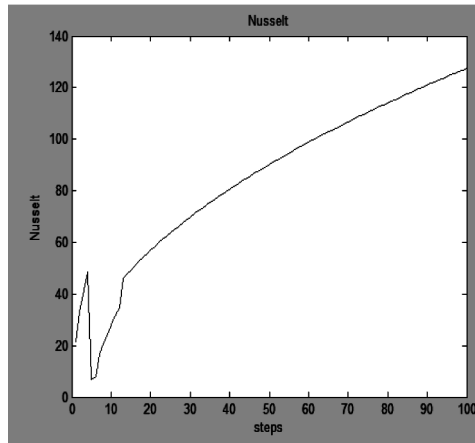


Figure 4: Variation in Nusselt number

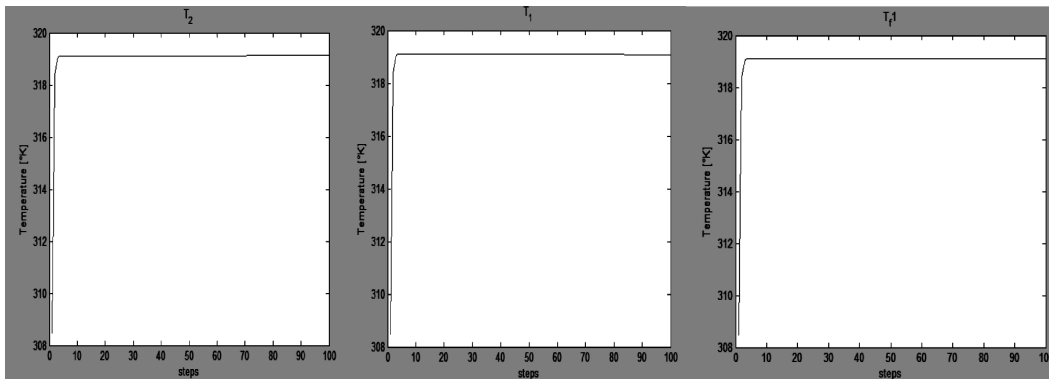


Figure 5: Variation in temperature

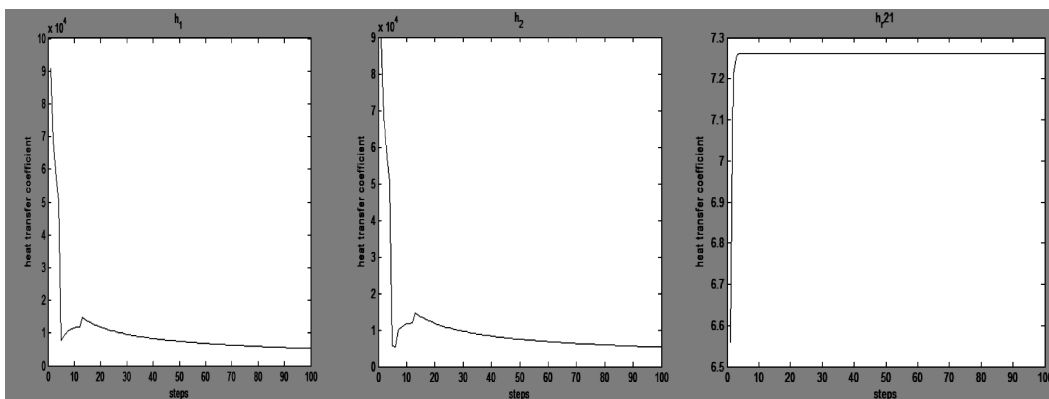


Figure 6: Variation in Heat Transfer coefficient

Mean Values

The easiest approach would be calculating the system with the mean values of the arrays set in the previous step. Since one mean value would lead to the problem of having only one iterative step for the calculation where an inaccurate outcome is inevitable, taking one tenth of the values seems more like a pursuable idea. Additionally the first tenth will be disregarded in the calculation and an eleventh element will be put into account. This value will be estimated according to the mean gradient of the values.

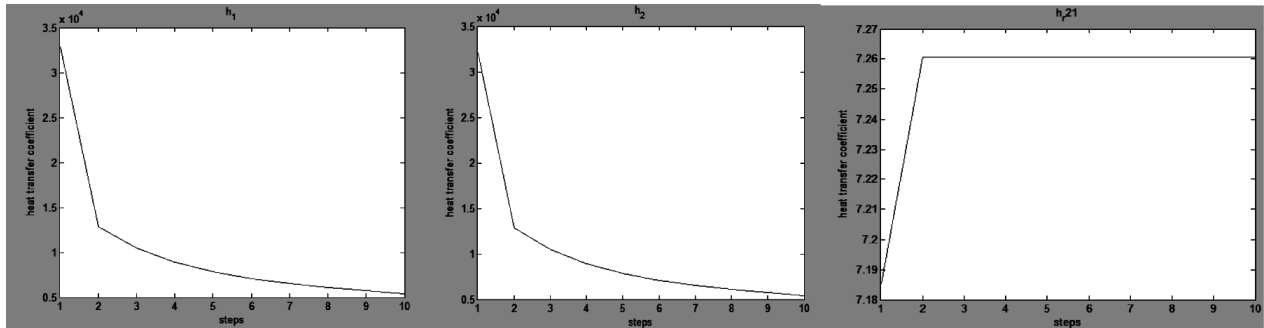


Figure 7a: Variation in Heat transfer coefficient

These plots make the source of the error very obvious. The first iteration step will now be cut and replaced by an estimated step in the end. h_{r21} will then pretty constant, which is reasonable since small temperature differences (T_{cover} and $T_{collector}$) are present.

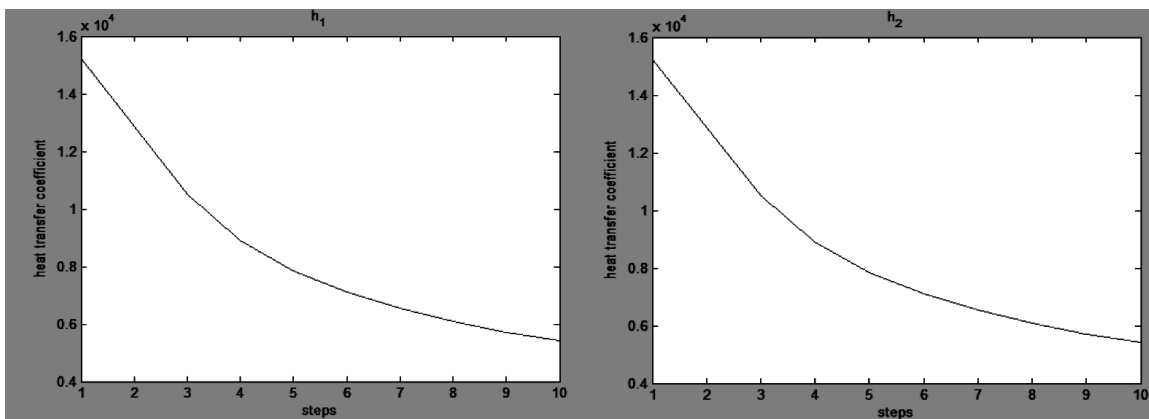


Figure 7b: Variation in Heat transfer coefficient

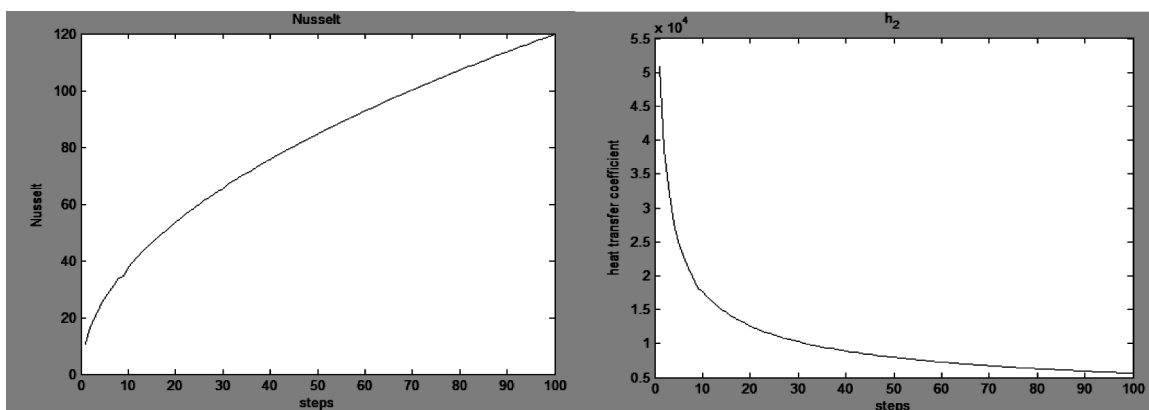


Figure 8: Variation in Nusselt number

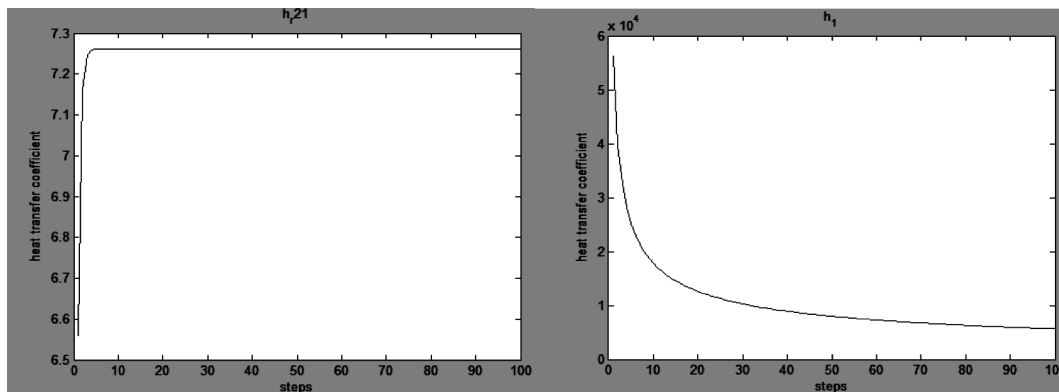


Figure 9: Variation in heat transfer coefficient

These assumptions were validated after finding the error in the calculation model. The Reynolds number has been calculated with a change to turbulent flow at $Re > 6 \cdot 10^3$. These values were reached after a few steps and the result were the irregularities in the beginning. The next page shows the corrected graphs.

4. CONCLUSION

No matter which values was chosen now the calculation of the outlet temperature will prove the validity of the model. With the calculations shown the temperature has been calculated and compared to the result. The temperature differences that have exceeded specific parameters haven't been taken into account. Especially the first few steps of the temperature rise seen in the shown graphs of T_1 , T_2 , T_{fl} , have been eliminated according to the iterative process mentioned in the flowchart.

The temperature calculated lies within 2 degrees of the empirical value. The difference can be explained by the transmission, absorption and reflection coefficients of blackened aluminum, the mean values of empirical data that it has been calculated and also the assumed rectangular entry instead of the circular one.

ACKNOWLEDGEMENT

The first author would like to thank Manipal University, Manipal for providing the IAESTE student exchange programme for collaborating the ideas with the Universities located worldwide.

References

- [1] William H. Klein, Calculation of Solar Radiation and the Solar Heat Load on Man *J. Meteor.*, Vol: 5, pp. 121–130, 1948.
- [2] A.F. Mills, *Heat Transfer*, Second Edition, Prentice-Hall, New Jersey, 1999.
- [3] J.R. Welty, G.L. Rorrer, and D.G. Foster, *Fundamentals of Momentum, Heat, and Mass Transfer*, Sixth Edition, Wiley, New Jersey, 2014.
- [4] C.O. Bennett and J.E. Myers, *Momentum, Heat, and Mass Transfer*, Third Edition, McGraw-Hill, New York, 1982.
- [5] K. S. Ong, *Thermal Performance of Solar Air Heaters: Mathematical Model and Solution Procedure* Vol: 55(2):93-109, 1995.
- [6] J. Buzás, R. Kicsiny, Transfer functions of solar collectors for dynamical analysis and control design. *Renewable Energy*, vol. 68, Issue C, pp. 146-15, 2015.

- [7] Natalia C. Rosca, Alin V. Rosca, Ioan Pop Stagnation point flow and heat transfer over a non-linearly moving flat plate in a parallel free stream with slip Communications in *Nonlinear Science and Numerical Simulation*, vol. 19, no. 6, pp. 1822–1835, 2014.
- [8] Alejandro L. Hernández, José E. Quiñonez Analytical models of thermal performance of solar air heaters of double-parallel flow and double-pass counter flow. *Renewable Energy* 55, pp. 380 – 391, 2013.
- [9] Koyuncu, T.. “Performance of various design of solar air heaters for crop drying applications”, *Renewable Energy*, Vol:31, No: 7, pp. 1073-1088, 2006.