

Numerical Analysis on the Absorber Plate of a Flat Plate Solar Collector for Two-Dimensional Transient Heat Conduction

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Abstract Flat plate solar collectors remain one of the simplest and most widely used devices for converting incident solar radiation into low-grade thermal energy. This paper presents a numerical study of two-dimensional transient heat conduction in the absorber plate of a flat plate solar collector, based on a finite difference formulation of the governing energy balance equation. A symmetric region of the plate bounded by two adjacent fluid-carrying tubes is modelled, with convective losses at the top and bottom faces, an insulated (symmetry) condition at the centreline, and a temperature boundary condition coupling the plate to the working fluid at the tube side. The governing equation is non-dimensionalised and discretised using a forward-time, central-space (FTCS) explicit scheme, including separate treatments for edge and corner nodes. A FORTRAN code was developed to generate the transient temperature field for copper as the plate material and water as the working fluid. Results are presented as temperature contours at successive non-dimensional time steps, temperature profiles along the plate length and width, mean plate temperature, fluid exit temperature, and collector efficiency, each examined against varying solar irradiance and mass flow rate. The results show that the absorber plate approaches steady state at a non-dimensional time of approximately unity, that mean plate and fluid exit temperatures rise with irradiance, and that both quantities fall—while reaching steady state sooner—as the mass flow rate increases. Collector efficiency is found to increase moderately with irradiance but to fall sharply with

increasing mass flow rate.

Keywords— Flat plate solar collector; absorber plate; transient heat conduction; finite difference method; two-dimensional analysis; collector efficiency.

I. INTRODUCTION

Renewable energy sources are gaining increasing importance as concerns over fossil fuel depletion, pollution, and global warming continue to grow. Among these, solar energy stands out for being abundant, sustainable, and freely available, with applications ranging from water and space heating to distillation, drying, and electricity generation.

A solar thermal collector absorbs incident solar radiation and converts it into usable heat, which is then transferred to a working fluid such as water or air. Flat plate collectors are the most common non-concentrating type, consisting of an absorber plate, a transparent cover, insulation, a supporting frame, and fluid passages through which the heating fluid flows. Sunlight passing through the transparent cover is absorbed by a dark, thermally conductive plate—typically copper or aluminum and the resulting heat is conducted to tubes carrying the working fluid, raising its temperature from inlet to outlet.

Since a collector rarely operates under steady conditions solar irradiance, ambient temperature, and fluid demand all vary with time understanding its transient thermal response is essential for realistic performance prediction and design optimization. This paper focuses specifically on the two-dimensional transient conduction problem within the absorber plate itself, using a finite difference approach to obtain the temperature field, mean plate temperature, fluid exit temperature, and efficiency of the collector under varying irradiance and mass flow rate.

II. LITERATURE REVIEW

The transient behaviour of flat plate solar collectors has been studied extensively since Duffie and Beckman [1] compared zero-, one-, and multi-node thermal capacitance models and concluded that a one-node model is generally adequate when only hourly meteorological data are available. Tiwari and Srivastava [2] examined the effect of thin, reflective insulation on plate temperature, showing it can perform comparably to thick conventional insulation while also protecting against fluid-drainage damage. Mather [3] developed a transient response model linking fluid flow rate, thermal mass, and heat transfer coefficients, validated against evacuated-tube collector test data.

Multi-dimensional and node-based transient models were advanced by Onyegegbu and Morhenne [4], who incorporated component thermal masses and exergetic optimisation under varying insolation, and by Duarte and Zapparoli [5], who compared one- and two-dimensional conjugate heat transfer treatments and highlighted the discrepancies introduced by neglecting the second dimension. Dhariwal and Mirdha [6] obtained closed-form analytical expressions for transient collector behavior without the simplifying assumptions of constant initial and mean fluid temperature used in earlier work.

Finite-difference and finite-element treatments specific to two-dimensional transient conduction in the absorber plate were reported by Zima and Dziwiewa [7] and by Vettrivel and Mathiazhagan [8], both using implicit schemes solved for tube-side collectors, with reasonable agreement against experimental data. Roy and Kundu [9] applied a thermal-wave (non-Fourier) model with separation of variables to study collector start-up and shutdown transients, while Al-Tabbakh [10] presented an explicit finite-difference transient model similar in spirit to the one adopted in this work, examining top heat loss coefficient and efficiency variation with irradiance and mass flow rate. The present study builds on this body of work by carrying out a fully explicit FTCS finite-difference solution—including detailed treatment of all edge and corner boundary nodes—for a two-dimensional absorber plate segment, using irradiance and mass flow rate as the principal operating parameters.

III. MATHEMATICAL FORMULATION

A. Physical Model and Assumptions

A symmetric region of the absorber plate lying between two adjacent fluid-carrying tubes is considered, of length W (along the direction of fluid flow) and width L (between tube centrelines), with plate thickness t_p . The following assumptions are made: (i) the plate has constant thermal conductivity, thickness, and overall heat loss coefficient; (ii) the temperature distribution over the plate is two-dimensional; (iii) the fluid temperature inside the tube varies as a second-order (quadratic) function of position along the tube; and (iv) the absorbed solar flux is constant and uniform over the plate.

B. Governing Equation and Boundary Conditions

The two-dimensional transient energy balance for the absorber plate, including conduction, absorbed irradiance S , and convective loss characterised by an overall heat loss coefficient UL , is written as

$$\partial^2 T / \partial x^2 + \partial^2 T / \partial y^2 + S / (K t_p) - UL(T - T_a) / (K t_p) = (\rho c_p / K) \partial T / \partial t$$

The plate is initially at a uniform temperature T_0 . At the lower and upper edges ($y = 0$ and $y = L$) heat exchanges with the ambient at heat transfer coefficients h_1 and h_2 , respectively. Symmetry gives an insulated condition at the left edge ($x = 0$), while at the right edge ($x = W$)—coinciding with the tube—heat is transferred to the working fluid, whose temperature is assumed to vary quadratically along y .

C. Non-dimensionalisation

Introducing the non-dimensional variables $\theta = (T - T_a)/(T_{in} - T_a)$, $X = x/W$, $Y = y/L$, $Fo = \alpha t/W^2$, $\delta = W/L$, $M^2 = ULW^2/(K tp)$, and $N = SW^2/[K tp(T_{in} - T_a)]$, the governing equation reduces to

$$\partial^2 \theta / \partial X^2 + \delta^2 \partial^2 \theta / \partial Y^2 + N - M^2 \theta = \partial \theta / \partial Fo$$

with corresponding non-dimensional initial and boundary conditions expressed in terms of Biot numbers Bi_1 , Bi_2 (top and bottom faces) and Bi (tube side), and the quadratic fluid-temperature profile $\theta_f = A + BY + CY^2$.

D. Finite Difference Discretisation

The governing equation is discretised using central differences in space and a forward difference in time (FTCS scheme). For an interior node (i, j) with equal grid spacing $\Delta X = \Delta Y = h$ and $r = \Delta Fo/h^2$, the update equation takes the form

$$\theta(p+1)_{i,j} = [1 - 2r(1+\delta^2) - M^2 \Delta Fo] \theta(p)_{i,j} + r(\theta(p)_{i+1,j} + \theta(p)_{i-1,j}) + r\delta^2(\theta(p)_{i,j+1} + \theta(p)_{i,j-1}) + \Delta Fo \cdot N$$

Analogous expressions were derived for nodes on each of the four edges—using one-sided differences and the corresponding Biot-number or Dirichlet condition—and separately for each of the four corner nodes, where two boundary conditions apply simultaneously. This complete set of update equations was implemented in a FORTRAN program and solved explicitly, marching forward in non-dimensional time from the uniform initial condition.

IV. RESULTS AND DISCUSSION

Simulations were carried out in FORTRAN using the plate dimensions and thermophysical properties summarised in Table I, taken from representative values reported in prior collector studies. The absorber plate was modelled with copper as the plate material and water as the working fluid.

TABLE I. DIMENSIONS AND PARAMETERS USED IN THE ANALYSIS

Parameter	Value
Absorber plate length, L	2 m
Absorber plate width, W	1 m
Plate thickness, tp	0.02 m
Tube inner diameter, Di	0.05 m
Inlet fluid temperature, Tin	30 °C
Ambient temperature, Ta	30 °C
Initial plate temperature, T0	35 °C
Overall heat loss coeff., UL	3.5 W/m ² ·K
Irradiance, S (base case)	800 W/m ²
Plate material (thermal cond.)	Copper, 398 W/m·K
Working fluid	Water
Mass flow rate, m (base case)	0.001 kg/s
Bi1, Bi2 (top / bottom faces)	0.125
Bif (tube-side surface)	1.000

A. Temperature Contours over the Absorber Plate

Temperature contours were generated at non-dimensional times $Fo = 0, 0.2, 0.4, 0.6, 0.8$, and 1.0 for $S = 800$ W/m² and $m = 0.001$ kg/s. The plate temperature rises steadily from the uniform initial value, with the steepest gradients appearing near the tube-side boundary ($X = 1$), where heat is continuously withdrawn by the flowing fluid. The contour spacing between $Fo = 0.8$ and 1.0 is markedly smaller than between $Fo = 0.2$ and 0.4 , indicating that the plate is approaching steady state by around $Fo \approx 1.0$.

B. Temperature Distribution along Plate Length and Width

At steady state ($Fo = 1.0$), the temperature is highest along the insulated left boundary ($X = 0$) for every value of Y , consistent with the assumed symmetry condition, and falls monotonically toward the tube-side boundary ($X = 1$), where heat is being withdrawn by the fluid. Along the tube-side boundary itself, the temperature is lowest at the fluid inlet ($Y = 0$) — where the temperature difference between plate and fluid, and hence the local heat transfer rate, is greatest — and rises toward the fluid outlet ($Y = 1$) as the fluid warms and the local driving temperature difference diminishes.

C. Mean Plate Temperature

Table II lists the mean absorber-plate temperature against non-dimensional time for three levels of irradiance at a fixed mass flow rate of 0.001 kg/s. In every case the mean temperature rises rapidly during the early transient period and levels off close to $Fo = 1.0$, reaching approximately 76 °C, 96 °C, and 116 °C at steady state for irradiance values of 500, 750, and 1000 W/m², respectively — confirming that a higher irradiance produces a proportionately higher steady-state plate temperature.

TABLE II. MEAN PLATE TEMPERATURE (°C) vs. Fo FOR VARYING IRRADIANCE ($\dot{m} = 0.001$ kg/s)

Fo	S=500	S=750	S=1000
0.2	54.75	63.02	71.29
0.4	65.67	80.07	94.47
0.6	72.45	90.65	108.86
0.8	75.71	95.74	115.77
1.0	77.04	97.82	118.61
2.0	77.51	98.55	119.60

Table III shows the corresponding effect of mass flow rate, at a fixed irradiance of 800 W/m². Increasing the mass flow rate lowers the steady-state mean plate temperature — from about 102 °C at 0.001 kg/s to roughly 72 °C at 0.007 kg/s — because more heat is continuously carried away by the faster-moving fluid. A higher mass flow rate also causes the plate to approach steady state sooner, as reflected in the flatter, earlier-levelling curves at higher $\dot{m}$.

TABLE III. MEAN PLATE TEMPERATURE (°C) vs. Fo FOR VARYING MASS FLOW RATE ($S = 800$ W/m²)

Fo	0.001 kg/s	0.003 kg/s	0.005 kg/s	0.007 kg/s
0.2	64.67	66.88	65.12	61.01
0.4	82.95	82.76	76.75	68.91
0.6	94.29	89.04	80.44	71.11
0.8	99.74	91.23	81.60	71.77
1.0	101.98	91.98	81.98	71.98
2.0	102.76	92.20	82.09	72.04

D. Fluid Exit Temperature

The fluid exit temperature follows the same qualitative trend as the mean plate temperature: it increases with irradiance and reaches steady state at $Fo \approx 1.0$, while it decreases and stabilises sooner as the mass flow rate is increased. Physically, a larger mass flow rate reduces the residence time of the fluid in the tube, so each fluid element absorbs less heat before leaving the collector, even though the total heat extracted from the plate is higher.

E. Collector Efficiency

Collector efficiency at steady state ($Fo = 1.0$) was evaluated over a range of irradiance values (500–1000 W/m², at $\dot{m} = 0.004$ kg/s) and mass flow rates (0.001–0.008 kg/s, at $S = 1000$ W/m²), summarised in Tables IV and V.

TABLE IV. COLLECTOR EFFICIENCY vs. IRRADIANCE ($\dot{m} = 0.004 \text{ kg/s}$, $F_o = 1.0$)

Irradiance (W/m^2)	Efficiency (%)
500	40.63
600	43.23
700	45.09
800	46.48
900	47.57
1000	48.43

TABLE V. COLLECTOR EFFICIENCY vs. MASS FLOW RATE ($S = 1000 \text{ W/m}^2$, $F_o = 1.0$)

Mass Flow Rate (kg/s)	Efficiency (%)
0.001	60.21
0.002	56.29
0.003	52.36
0.004	48.43
0.005	44.51
0.006	40.58
0.007	36.65
0.008	32.73

Efficiency rises only moderately with irradiance — from about 41% to 48% over the 500–1000 W/m^2 range — since both useful heat gain and the elevated plate temperature (and hence loss) increase together. In contrast, efficiency falls sharply with increasing mass flow rate, from about 60% at 0.001 kg/s to 33% at 0.008 kg/s , in the specific configuration analysed here where a single tube serves two adjacent absorber-plate segments; this indicates that mass flow rate is the stronger lever for controlling steady-state collector performance in this configuration, though the relationship would need to be re-examined for a different tube-to-plate arrangement.

V. CONCLUSION

A finite difference formulation of two-dimensional transient heat conduction has been developed and solved for the absorber plate of a flat plate solar collector, using an explicit FTCS scheme with dedicated treatment of edge and corner boundary nodes. The model was used to obtain the transient temperature field, mean plate temperature, fluid exit temperature, and collector efficiency under varying irradiance and mass flow rate. The absorber plate was found to approach steady state at a non-dimensional time of approximately unity for the parameters considered. Mean plate and fluid exit temperatures increase with irradiance and decrease with mass flow rate, while the plate also reaches steady state sooner at higher flow rates. Collector efficiency was found to be a comparatively weak function of irradiance but a strong function of mass flow rate for the geometry and parameters examined. These findings reinforce the importance of accounting for genuinely two-dimensional, transient behaviour when predicting or optimising flat plate collector performance, rather than relying solely on one-dimensional or steady-state approximations.

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