# Annular Two-Phase Flow Regimen in Direct Steam Generation for a Low-Power Solar System

**Iván Martínez, Rafael Almanza, María Dolores Durán and Miriam Sánchez**

**Abstract** This study aims to quantify and to model the temperature profile around an absorber tube belonging to a parabolic trough concentrator when fluid is applied at low powers. This study was specifically developed for the Solar Power Plant of the Engineering Institute, National University of Mexico. This work presents experimental results under saturated conditions and low pressures (1.5–3 bars) using water as the thermal and working fluid for direct steam generation (DSG). The control variable was feed flow. Solar irradiance was used as the restriction variable because all experimental tests should be developed under very specific values of this variable (for example, $I > 700\ W/m^2$). The objective of this experiment was to study the thermal behavior of a temperature gradient around the absorber tube under steady-state conditions and with low flow. Additionally, a theoretical analysis was carried out by means of the homogeneous heat conduction equation in the cylindrical coordinate system using only two dimensions ($r$, $\varphi$). The finite-difference numerical method was used with the purpose of proposing a solution and obtaining a temperature profile. The aim of this theoretical analysis was to complement the experimental tests carried out for direct steam generation (DSG) with annular two-phase flow patterns for low powers in parabolic trough concentrators with carbon steel receivers.

## Nomenclature

| | |
|---|---|
| A | area in $m^2$ |
| Cp | heat capacity in $kJ/(kg \cdot K)$ |

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| | |
|---|---|
| d | pipe diameter in m |
| $G$ | irradiance in $W/m^2$ |
| h | convective heat transfer coefficient in $W/(m^2K)$ |
| $k$ | thermal conductivity in $W/(m \cdot K)$ |
| $\overline{Nu}$ | overall Nusselt number |
| P | pressure in bar |
| Pr | Prandtl number |
| $q''$ | heat flow in $kW/m^2$ |
| $\dot{q}$ | energy generation in $kW/m^2$ |
| Q | volumetric flow in L/s |
| r | radius in m; radial coordinate |
| Re | Reynolds number |
| t | time in seconds |
| T | temperature in K |
| u | velocity in m/s |

## Greeks

| | |
|---|---|
| $\alpha$ | absortance; void fraction |
| $\Delta$ | interval or difference |
| $\varepsilon$ | emittance |
| $\varphi$ | radial angle; azimuth coordinate |
| $\nu$ | kinematic viscosity in $m^2/s$ |
| $\rho$ | reflectance; density in $kg/m^3$ |
| $\sigma_{SB}$ | Stefan-Boltzman constant |

## Subscripts and Superscripts

| | |
|---|---|
| $a$ | air |
| $atm$ | atmospheric |
| $cov$ | convective |
| $Fe$ | Iron |
| $l$ | liquid |
| rad | radiation |
| sup | superficial |
| S | solar |
| T | total |
| $int$ | internal |
| $ext$ | external |

## 1 Introduction

Direct steam generation (DSG) in parabolic trough concentrators is a technique being increasingly developed by many researchers world-wide. The method has great potential to improve hybrid power systems and retain competitive energy prices. In this case, the main interest is to develop a low power system (Almanza et al. 2002). During the operation of the Solar Power Plant of the Engineering Institute, National University of Mexico (UNAM), it was observed that some problems for DSG occurred under specific flow and temperature conditions. These problems corresponded to the stratified two-phase flow regimen that occurs when boiling water is transferred into the receiver tube when the thermal gradient in its periphery is increased (Almanza et al. 2002). Such problems are related to the bending of tubes owing to thermal stress, causing their permanent deformation and breaking their glass covers.

In order to find a solution to this problem, a project was carried out that included experimental tests and the production of a mathematical model to predict stratified two-phase flow pattern under transient conditions of normal solar beam irradiance to low flows (Flores 2003). This study considered flows between 1 and 2.5 L/min, with solar beam irradiance that fell on the lower part of pipe or on one side of the pipe. It is essential to understand the thermal behavior of absorber pipes under annular two-phase flow conditions with low pressures and low flows, in order to contribute to the development of low power systems like the ones proposed by Almanza et al. (2002). In order to continue the project previously described, it was considered necessary to develop experimental tests and a mathematical model for annular flow pattern during DSG. Preliminary results were generated by Martínez and Almanza (2003) but, because of high uncertainty, experimental tests required more replication and better data acquisition techniques.

These works form part of the basis required to develop a proposal for direct commercial application of DSG using parabolic trough concentrators in Mexico. For example, it is possible that in the near future the Federal Commission for Electricity (CFE) of Mexico, along with the Engineering Institute (UNAM) and CIEMAT (Centro de Investigaciones Energéticas, Medioambientales y Tecnológicas) from Spain, might build an experimental hybrid installation in Baja California State, Mexico. However, it is known that there is a high probability that annular two-phase flow exists in most of their pipes. It is hence required to experimentally explore and evaluate this idea. It is necessary to increase the mass quality of steam, from the current mixture of 40 % steam and 60 % brine found in the geothermal wells of Cerro Prieto in the Northwest of Mexico. This brine has a mass concentration of different sorts of salts, mainly NaCl, of approximately 2 %. Therefore, it is not advisable to increase this parameter too much in order to avoid the buildup of scale in the receiver pipes, which is important in maintaining the annular flow pattern along the DSG with solar energy.

A geothermal system in Cerro Prieto works well with low pressures between 10 and 16 bars. Hence, it is possible that all tests made at the Engineering Institute's solar power plant could be extended for parameters like these, with the purpose of establishing better operational conditions with a steady-state regimen for the hybrid system proposed. Therefore, annular two-phase flow is a good regime to minimize thermal gradients around absorber pipes so to avoid thermal stress.

# 2 Experimental Development

The main intention of this experiment was to determine the temperature gradient around the receiver tube under steady-state conditions with an annular two-phase flow pattern. In order to achieve this, it was necessary to maintain control over the feed water (the control variable), which was done by choosing three volumetric flow rates; 4, 8, and 12 L/min. Each test was replicated at least five times per value.

The temperature and pressure were measured using a high performance acquisition system registered at the beginning and at the end of the solar field. The steam volumetric flow, which contains a small quantity of liquid drops because the steam trap cannot separate them completely, was also measured. The liquid from the steam trap was quantified and it was possible to establish its effectiveness in approximately 80 % of the trials. Temperatures, as detected by an RTD (Platinum Resistance Temperature Detector) fixed on the external surface around the end of the last absorber pipe, were captured by another acquisition system.

Additional data recorded by the weather station were also taken into account in order to determine the possible influence of these parameters on the values of the process variables and the superficial absorber temperature.

Steady-state conditions were reached when the fluid was under saturated conditions that depended on pressure. The mass quality of the generated steam was hence dependent on feed flow and the irradiance level. The experiments were carried out near 800 $W/m^2$ of normal solar beam irradiance. Each experiment was conducted maintaining the flow with constant feeding, as for a sub-cooled liquid, so that any change in the process could cause changes in the amount of heat absorbed by the working fluid and hence be automatically controlled to be within $\pm 0.5$ L/min variation. This sort of oscillation cannot be avoided and, on some occasions, the system oscillated beyond this control range. In these cases it was necessary to terminate the experiment. The process variables (pressure, flow and temperature) were registered every two seconds, and the absorber's surface temperature was measured every five seconds.

The data registered during the commencement and termination of each experimental test were not taken into account for this study, although it was noted that they were very useful for analyzing the thermal behavior of this system during transient conditions.

## *2.1 Description of Facilities and Equipment*

All tests were carried out in the solar field of parabolic trough concentrators located at UNAM in Mexico City (19° 19' 6.9" north latitude, 99° 11' 29.7" west longitude)

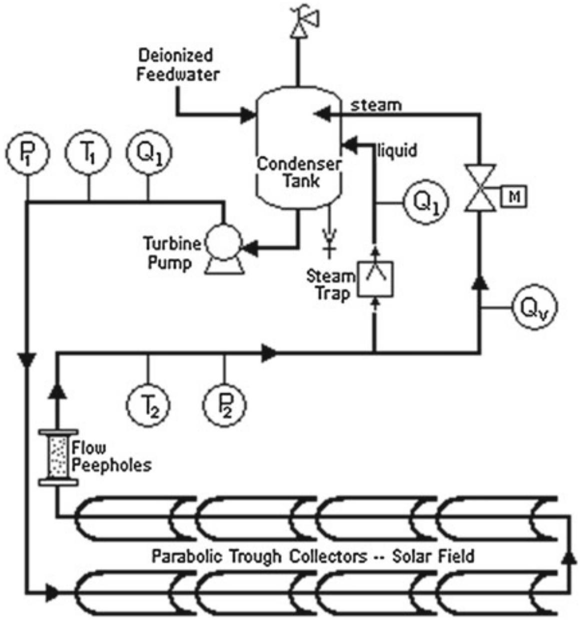


**Fig. 1** Solar power plant, Engineering Institute (UNAM)

and at an altitude of 2330 ± 20m meters above sea level (as measured with a GPS). Eight modules are connected in serial mode, each one with a length of 15 m and an aperture area of $34m^2$ (Fig. 1). The total aperture area of solar field is $272m^2$, with mirrors whose reflectance is approximately 0.85. The absorber pipes are 25.4 mm (1") nominal diameter and are covered with a black chrome selective film with an absorptance of approximately 0.89 and emittance of approximately 0.18 at 25 °C.

The solar plant has a water deionizer system that works with ionic interchange resins, so the water conductivity is reduced to 0.39 mS/cm at 20 °C. Although the main application of this study is for geothermal wells, as a first step it would be better to start with this water quality in order to avoid damage to the experimental equipment. This water is stored within a cistern, which serves as a reservoir for a feeding tank that is connected to one condenser, and this also serves as a feed water tank. The pump is a double pass regenerative turbine with a capacity ranging from 3 to 32 L/min. It is controlled by a 3 hp motor with a current frequency converter.

At the end of the 8th module concentrator, the two-phase flow achieved can be observed through two peepholes made of borosilicate glass of 24.5 mm diameter and 30 cm length. With these elements is was possible to visually determine the flow pattern that was obtained in the experimental tests. In order to show how a two-phase flow looks through a flow-peephole a photograph is shown at Fig. 2.

After the peepholes, a steam trap serves to separate the phases and allows the flow to be measured for each phase. Both currents reach the condenser tank and a

**Fig. 2** Flow peephole photograph with annular two-phase flow

**Table 1** Average values of feed water

| Experimental test (L/min) | Feed flow (L/min) | Velocity (m/s) |
|---|---|---|
| 4 | 4.3 | 0.47 |
| 8 | 8.1 | 0.87 |
| 12 | 12.6 | 1.37 |

saturated liquid is then re-circulated to the solar field. Measuring instruments used to make recordings were:

– 8 RTDs (platinum Resistant Temperature Detector) around the absorb er pipe.
– 2 RTDs at the beginning and the end of the system.
– 2 pressure transducers at the beginning and the end of the system.
– 1 'Headland' variable area transducer for measuring flow of liquid.
– 1 'Endress & Hauser' vortex sensor transducer for measuring flow of steam.
– 9 pressure dial indicators along the tube
– 8 temperature dial indicators along the tube

In addition, data were collected from a meteorological station that registers the values of global and diffuse horizontal irradiance (using a rotating shadow band pyranometer), dry bulb temperature, and speed and direction of wind every 5 min. Direct normal irradiance was also calculated.

## 2.2 Experimental Results and Discussion

The temperature distribution around the absorber pipe when the working fluid has an annular two-phase flow pattern is the most interesting result of this work. The goal of always working with a fluid in conditions of saturation with constant feeding flow was achieved (Table 1), and the maximum pressure reached on the system was 3.5 bars. The next step will be to repeat the experiment at 10 bars. The Goebel (1997) and Herbst et al. (1996) experiments were useful for this study in order to know what happens with DSG under higher values of process conditions.

Because the experimental tests were carried out in winter, the heating tended to be from below and to one side of the receiver, between $3/4\pi$ and $7/4\pi$ radians of the circumference. As a result, it was necessary to carry out tests by heating exclusively from below part of the absorber pipe.

Owing to small instabilities during the heating process, such as normal beam irradiance changes, pressure values oscillated throughout each test. It was thus necessary to calculate average values of pressure, temperature and velocity. This allowed the average value of each process property to be estimated from simply fixing the feed. At the beginning, the feed had a specific velocity which was constant until the boiling process started. When steam appeared we started to increase the feed's velocity in order to maintain a good mass balance. The liquid phase then changed its initial velocity according to the process conditions. For example, 4 L/min flow had 27 % reduction of its speed value, for 8 L/min the reduction was 7 % and for 12 L/min there was no change because variations were minimal with respect to feed. Hence the liquid velocity diminished for the first two tests at 4 and 8 L/min, whereas for the third test, at 12 L/min, it stayed relatively constant.

Temperature and pressure behaviors of feed flow were similar to a sub-cooled liquid (97 °C and 3.5 bars). These values are important in order to reach boiling point as soon as possible in the receiver pipes. It was observed that for greater flow, the system was more stable with these parameters. It was also observed that its variation did not have great repercussions with respect to low feed flow throughout heating in the solar concentrators. Its behavior is more stable when the evaporation stage begins.

In order to verify changes in temperature and pressure during the heating and evaporation process of water in the absorber tubes, dial indicators for temperature and pressure were installed in the bridges between each module of concentrators. At the end of the last module, the variables were measured with instruments connected to a data acquisition system.

Pressure is the main parameter affecting inlet and outlet currents. The other parameter that independently affects the process is solar beam irradiance, a factor that depends on the weather. The graph of outlet pressures (Fig. 3) shows a uniform behavior; nevertheless, small variations in pressure that occurred were translated into temperature changes. The velocities reached for the generated steam are shown in Fig. 4, and the typical patterns for different feed flows can be seen under similar conditions of irradiance, which oscillated around ±10 %. In order to calculate the steam quality, a phase separator with a steam trap was installed and steam flow was measured with a flow meter.

The mass balance was calculated under similar values of irradiance, and the results are presented in Fig. 5. Figure 6 shows a graph depicting the relationship between liquid velocities and steam velocities; this demonstrates that a smaller feed flow generates a greater amount of steam, and therefore the liquid film is thinner and the friction factor has more influence on its velocity.

In agreement with Hahne et al. (1997), the convective heat transfer coefficient increases if the liquid film thickness diminishes. The measurements confirm this effect because external temperature is higher at 12 L/min than at 4 L/min.

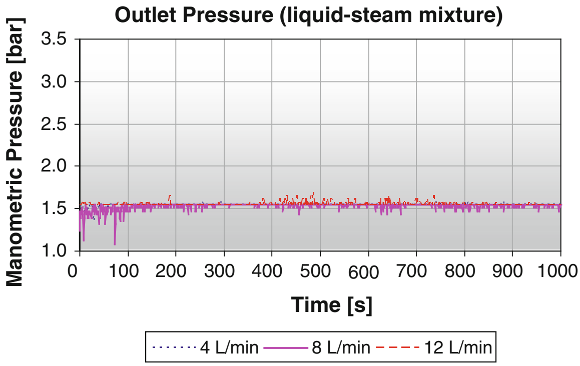


**Fig. 3** Liquid-steam mixture manometric pressure

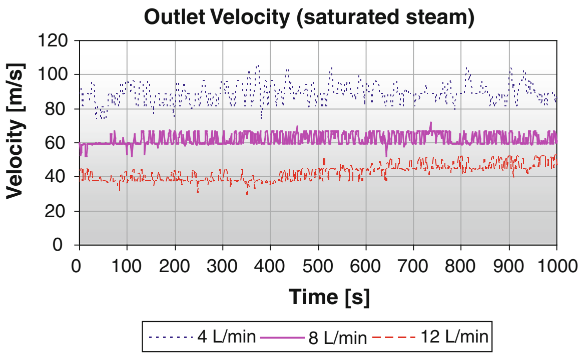


**Fig. 4** Outlet steam velocity

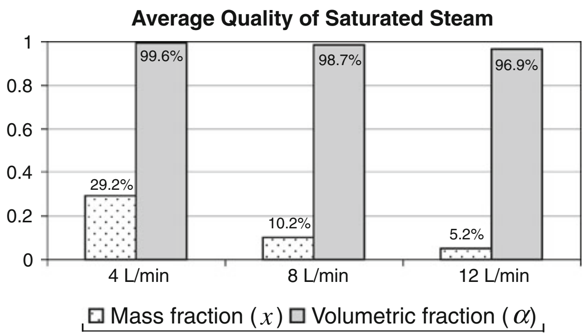


**Fig. 5** Mass and volumetric fraction for outlet steam

Nevertheless this condition is directly related to the amount of normal beam irradiance that is being received in each test. Table 2 shows the average values of normal beam irradiance that were recorded during these tests. The highest temperature dif-

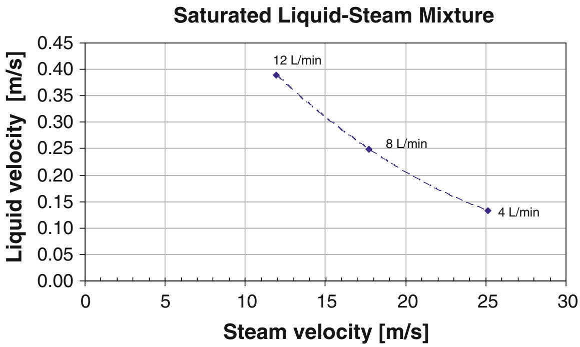


**Fig. 6** Relation of velocities between phases and the feed flow

**Table 2** Normal beam irradiance for experimental tests (average values)

| Irradiance (W/m$^2$) | 4 (L/min) | 8 (L/min) | 12 (L/min) |
|---|---|---|---|
| Global | 934 | 895 | 893 |
| Normal beam | 882 | 793 | 821 |
| Diffuse | 52 | 102 | 72 |

ferential value that was reached with annular flow was 41 K, which persisted for only a few minutes. If an average differential of all maximums is calculated, the resulting value is 31 K.

## 3 Theoretical Development

Equation 1 represents the general form of the heat diffusion equation in the cylindrical coordinate system; it determines the transference velocity of energy by conduction in a unitary volume, at any point within the work media. In addition, the volumetric generation velocity of thermal energy must be equal to the change velocity of the stored thermal energy within this volume (Incropera and DeWitt 2001). Specifically, the system under study takes the form of tubes which are warmed up by concentrated solar radiation falling on half of their external circumference. Therefore heat flux, as shown in Fig. 7, has been established for very clear conditions and to enable subsequent analysis of these conditions.

$$\frac{1}{r}\frac{\partial}{\partial r}\left(kr\frac{\partial T}{\partial r}\right)+\frac{1}{r^2}\frac{\partial}{\partial \varphi}\left(k\frac{\partial T}{\partial \varphi}\right)+\frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right)+\dot{q}=\rho Cp\frac{\partial T}{\partial t} \tag{1}$$

The following conditions are proposed in order to develop a simplified analysis: Steady state $\left(\frac{\partial T}{\partial t}=0\right)$; no variation along longitudinal axis $\left(\frac{\partial T}{\partial z}=0\right)$; it does not have internal heat generation ($\dot{q}=0$).

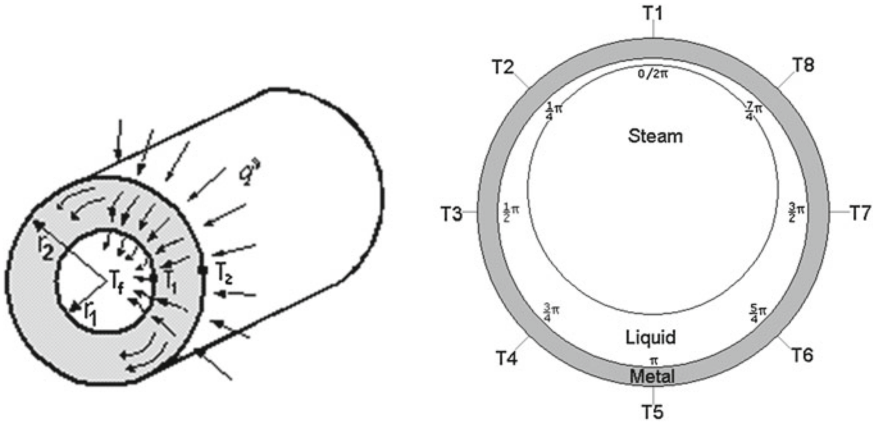


**Fig. 7** Heat flux and reference point in an absorber pipe

Because of the preceding conditions, the diffusion equation is reduced to Eq. 2.

$$\frac{1}{r}\frac{\partial}{\partial r}\left(kr\frac{\partial T}{\partial r}\right)+\frac{1}{r^2}\frac{\partial}{\partial \varphi}\left(k\frac{\partial T}{\partial \varphi}\right)=0 \tag{2}$$

Numerical methods are useful for solving heat transfer problems in order to obtain reliable results, when such problems can not be handled by exact analysis because of nonlinearities, complex geometries, and complicated boundary conditions. One major approach currently used in the numerical solution of partial differential equations of heat transport is the finite-difference method. The boundary conditions that correspond to this problem are of the third kind type, since they express the heat flux between the surface of the system under study and a moving fluid that is in contact with this surface, at a precisely-known temperature. This condition must demonstrate the relationship between the heat transmitted by conduction in the system and the heat transferred by convection.

## 3.1 Boundary Conditions

(1) If $r = r_{\text{int}} \quad \wedge \quad 0 \le \varphi < 2\pi$ then $-k\frac{\partial T}{\partial r} = h_l\,(T - T_l)$

(2) For concentrated irradiance on the lower part of the receiver tube:
If $r = r_{ext} \quad \wedge \quad 1/2\pi < \varphi < 3/2\pi$ then $-k\frac{\partial T}{\partial r} = q'' - h_a\,(T - T_a)$
If $r = r_{ext} \quad \wedge \quad 0 \le \varphi \le 1/2\pi \quad \wedge \quad 3/2\pi \le \varphi \le 2\pi$ then $-k\frac{\partial T}{\partial r} = h_a\,(T - T_a)$

(3) For concentrated irradiance on one side of the receiver tube:
If $r = r_{ext} \quad \wedge \quad 0 < \varphi < \pi$ then $-k\frac{\partial T}{\partial r} = q'' - h_a\,(T_a - T)$
If $r = r_{ext} \quad \wedge \quad \pi \le \varphi \le 2\pi$ then $-k\frac{\partial T}{\partial r} = h_a\,(T - T_a)$

### 3.2 Finite Differences

Different numerical methods allow the determination of the temperature only at discrete points, in contrast with an analytical solution that allows the determination of temperature at any point of interest in a specific environment. Therefore in order to obtain functions which describe the distribution of temperatures around a receiver tube, the first step is to select reference points, which usually are called "nodal points" or simply "nodes", so that a set of these points is known as a nodal grid Özişik 1993.

Each node represents a specific region whose value is a measurement of the average temperature in that region. The selection of these points depends generally on geometric convenience and the degree of precision that is desired. Figure 8 shows a proposed cross-sectional profile for this work and it displays the proposed nodal network in external circles that correspond to the walls of the absorber pipe. A third circle is also shown at the centre of Fig. 8, which has a slight displacement on the vertical axis, representing the steam-liquid interface. Therefore Fig. 8 exhibits a two-phase flow with an annular pattern. Such displacement is considered to be a phase velocity function as well as a function of liquid film thickness in the upper part of the pipe compared to the lower part.

Using this method, the heat equation is obtained by applying the law of energy conservation to a control volume around the nodal region, supposing that all the heat flow occurs towards the node ( Rohsenow and Hartnett 1973). Heidemann et al. (1992) used the finite differences method in order to propose a mathematical model for steady-state and transient conditions of temperature during DSG with stratified two-phase flow.

The bi-dimensional model considers the following elements: carbon steel pipe, liquid-steam flow with annular pattern, and air atmosphere around the external surface of the receiver tube. The reference angle is in radians, and begins in the upper part of the pipe in order to agree with Flores (2003). The general exposition of the energy balance equations for each node is based on the nodes that surround it. For this reason, two sorts of equations can be written according to the location of such nodes.

For the equation that represents the nodes from 1 to 60, a radiation term is added which can be greater than or equal to zero, depending on the boundary conditions that are being considered. The energy balance begins with Node 1, expressed in Eq. 3.

$$\frac{k_{Fe} \cdot (T_2 + T_{60} - 2T_1)(\Delta r/2)}{r_{ext}\Delta\varphi} + \frac{k_{Fe} \cdot (T_{61} - T_1) \cdot (r_{ext} - \Delta r/2) \cdot \Delta\varphi}{\Delta r} + q''_T = 0 \tag{3}$$

where,

$$q''_T = q''_{rad} + q''_{\text{cov}} - E_{\text{sup}} \Rightarrow \left\{ \begin{array}{l} q''_{rad} = \alpha_S G_S + \alpha_{atm} G_{atm} \\ q''_{\text{cov}} = h_a r_1 \Delta\varphi(T_a - T_1) \\ E_{\text{sup}} = \varepsilon_{\text{sup}} \sigma T^4_{\text{sup}} \end{array} \right\} \rightarrow G_{atm} = \sigma_{SB} T^4_{atm}$$

The heat contribution by radiation only applies for those nodes in which concentrated irradiance is being considered on half their circumference, whereas for nodes

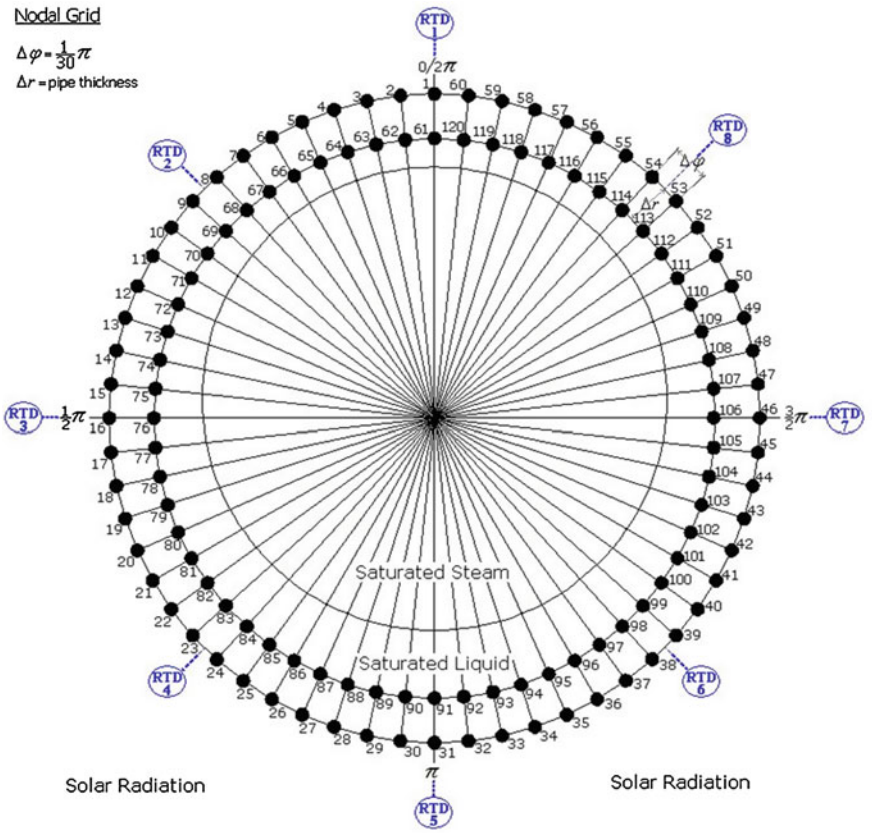


**Fig. 8** An $(r, \phi)$ nodal network for an absorber pipe with annular two-phase flow

that are not exposed to the concentrator effect, this term is near zero. After regrouping Eq. 3, each part can be modified in order to have an equation for each node. For example, node 1's equation is shown as Eq. 4.

$$-\left[\frac{\Delta r}{r_{ext}\Delta\varphi} + \frac{(2r_{ext} - \Delta r)\Delta\varphi}{2\Delta r} + \frac{h_a r_{ext}\Delta\varphi}{k_{Fe}}\right] T_1 + \frac{\Delta r}{2r_{ext}\Delta\varphi} T_2 + \frac{\Delta r}{2r_{ext}\Delta\varphi} T_{60} + \frac{(2r_{ext} - \Delta r)\Delta\varphi}{2\Delta r} T_{61} = -q''_T \tag{4}$$

where $q''_T = \frac{\varepsilon_{\sup}\sigma T^4_{\sup} - (\alpha_S G_S + \alpha_{atm}\sigma T^4_{atm} + h_a r_1 \Delta\varphi T_a)}{k_{Fe}}$

In order to enumerate Eq. 4, it is necessary to know the convective heat transference coefficient of the air next to the external surface of the receiver, which is determined by applying Eqs. 5, 6, and 7 (Bejan 1995).

$$\overline{Nu} = 0.3 + \frac{0.62 R_e^{1/2} P_r^{1/3}}{\left[1 + (0.4/\Pr)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{Re}{282000}\right)^{5/8}\right]^{4/5} \tag{5}$$

conditions:

$$\left\{ \begin{array}{l} Re \cdot \Pr > 0.2 \\ 7 \times 10^4 < Re < 4 \times 10^5 \end{array} \right\} \tag{6}$$

Annular two-phase flow regimen in direct steam generation.
where:

$$Re = \frac{u_a \cdot d_{ext}}{\nu_a} \text{ and } \quad h_a = \frac{\overline{Nu} k_a}{d_{ext}} \tag{7}$$

Correlations expressed in Eqs. 5 and 6 assume the following conditions:

- Single cylinder in cross-flow
- Speed of the air-flow is uniform
- Temperature of the air-flow is equal to room temperature

For nodes that are in the internal part of the steel tube, i.e. nodes from 61 to 120, the energy balance is expressed by Eq. 8.

$$\frac{k_{Fe}(T_1 - T_{61})(r_{int} + \Delta r/2)\Delta\varphi}{\Delta r} + \frac{k_{Fe}(T_{62} + T_{120} - 2T_{61})(\Delta r/2)}{r_{int}\Delta\varphi} + h_f r_{61} \Delta\varphi (T_f - T_{61}) = 0 \tag{8}$$

After rearranging Eq. 8, we obtain the expression (Eq. 9) that we will use to calculate the temperatures in the aforementioned nodes.

$$\frac{(2r_{int} + \Delta r)\Delta\varphi}{2\Delta r} T_1 - \left[\frac{\Delta r}{r_{int}\Delta\varphi} + \frac{(2r_{int} + \Delta r)\Delta\varphi}{2\Delta r} + \frac{h_f r_{int} \Delta\varphi}{k_{Fe}}\right] T_{61} + \frac{\Delta r}{2r_{int}\Delta\varphi} T_{62} + \frac{\Delta r}{2r_{int}\Delta\varphi} T_{120} = -\frac{h_f r_{int}\Delta\varphi}{k_{Fe}} T_f \tag{9}$$

The heat convective coefficient of heat transfer for two-phase flow was calculated by means of a self-developed algorithm that is described by Martinez (2005) and is supported from different proposed correlations by authors such as Goebel (1997), Gungor and Winterton (1986), and Kattan et al. (1998).

Once all equations for each node have been considered, it is possible to solve them by means of any algebraic method. For this work it is possible to assume the following values are constants: the pressure of the system for the two-phase mixture, the cross-sectional area of the receiver, and the mass flow of the mixture. One assumes that the convective coefficient of heat transfer is based on the liquid film thickness, and it can be calculated according to Luninski et al. (1983).

**Table 3** Data used for the simulation of temperature profile

| Symbol | Quantity | Unit | Description |
|---|---|---|---|
| Ta | 349.5 | (K) | Air film temperature |
| Tf | 406.15 | (K) | Temperature of the saturated fluid |
| ha | 8.05 | $(W/m^2K)$ | Air convective heat transfer coefficient |
| hf | 6,768.1 | $(W/m^2K)$ | Fluid convective heat transfer coefficient |
| kFe | 50.89 | $(W/m \cdot K)$ | Iron thermal conductivity |
| $r_{ext}$ | 0.0167 | (m) | External pipe radius |
| $r_{int}$ | 0.0131 | (m) | Internal pipe radius |
| $\Delta r$ | 0.0036 | (m) | Pipe thickness |
| $\Delta\phi$ | 0.1047 | (rad) | Angular separation between nodes |
| $G_{dir}$ | 550 | $(W/m^2)$ | Useful solar beam irradiance |
| $G_{tub}$ | 27,614 | $(W/m^2)$ | Concentrated solar beam irradiance |

## *3.3 Results and Discussion*

In order to obtain a theoretical temperature profile as function of $r$ and $\phi$, it is necessary to adjust some process parameters that are expressed in Table 3. We know that in annular flow conditions the liquid film thickness at the top of the pipe is thinner than at the bottom. It is possible, therefore, that the value of the two-phase heat transfer convective coefficient changes as a function of the film thickness.

It is a known fact that heat losses always exist, and it is therefore necessary to determine a thermal efficiency to quantify these losses. Nevertheless, it is possible to assume that heat losses only exist when the receiver temperature is lower than the working fluid. When the calculated values of temperature are compared with the experimental data, it is necessary to take this assumption into account.

In calculating the effect of heat transfer during the liquid's boiling process, we can assume the liquid-steam interface can be almost homogeneous and have a variable distance with respect to the centre of the tube, the reason being that the thickness of the annular liquid film will be based on slip ratio and the wet angle calculated from 0 radians (top part) to $\pi$ radians (bottom part) in such a way that the pattern is symmetrical in the interval from $\pi$ to $2\pi$ radians.

In order to know what happens when the convective heat transfer coefficient is considered constant (theoretical A) or variable with respect to the liquid film thickness (theoretical B), a relative error factor was calculated and results were compared with experimental data. Table 4 shows a maximum value of 18.7 % for option A and 5.3 % for option B. Figure 9 shows the comparison and data behaviour around the external surface of the absorber pipe. The calculated data of liquid film thickness were 1.06mm at the top and 2.66mm at the bottom. These values accorded with flow peephole observations, where it was possible to see a double thickness of liquid film at the lower part compared to the upper part, and the estimated thickness observed correlated approximately with the values already given above.

**Table 4** Relative error between theoretical model and experimental temperature

| RTD No. | Node No. | $\phi$ angle (rad) | Temperature (°C) | | | Relative error | |
|---|---|---|---|---|---|---|---|
| | | | Experimental | Theoretical A | Theoretical B | A (%) | B(%) |
| 1. | 1 | 0.00 | 107.0 | 108.0 | 104.5 | 1.0 | 2.4 |
| 2. | 8 | 0.79 | 117.6 | 126.6 | 120.1 | 7.1 | 2.1 |
| 3. | 16 | 1.57 | 118.8 | 131.6 | 123.6 | 9.7 | 3.9 |
| 4. | 23 | 2.36 | 125.4 | 131.6 | 123.7 | 4.8 | 1.4 |
| 5. | 31 | 3.14 | 114.2 | 129.1 | 122.0 | 11.6 | 6.4 |
| 6. | 39 | 3.93 | 107.2 | 111.1 | 107.0 | 3.5 | 0.2 |
| 7. | 46 | 4.71 | 99.3 | 104.7 | 102.4 | 5.2 | 3.0 |
| 8. | 54 | 5.50 | 97.0 | 104.1 | 102.0 | 6.8 | 4.8 |

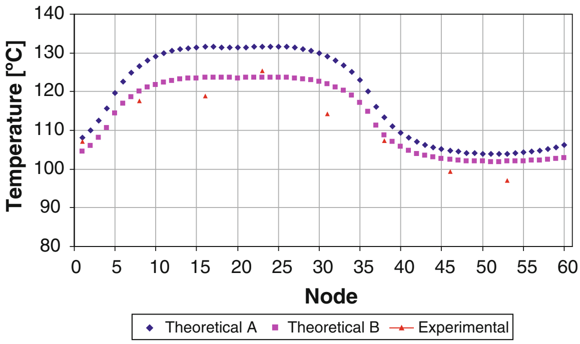


**Fig. 9** Temperature profile around absorber pipe

According to the graph in Fig. 9, it is possible to observe the tendency of data to demonstrate periodicity. This is because temperature analysis is done in radial and angular directions, and the angular direction should exhibit a periodic behavior. This aspect is very important because is possible to propose an analytical solution in trigonometrical terms that could be like Eq. 10. All constants (*A, B, C* and *D*) could be obtained from boundary conditions and specific experimental values, but its particular solution is a subject for other work.

$$T(r, \varphi) = \left(A \cdot r^{\nu} + B \cdot r^{-\nu}\right) \cdot (C \cdot \sin(\nu \cdot \varphi) + D \cdot \cos(\nu \cdot \varphi)) \tag{10}$$

In any sort of mathematical model, the convective heat transfer coefficient is a very important parameter that should be calculated by applying the best correlation that satisfies the specific process parameters required. To define an acceptable value for this coefficient, we have taken the model proposed by Goebel (1997), which is based on experimental data. Another theoretical analysis related to two-phase flow with DSG was developed by Zarza (2003); he emphasizes the importance of the convective coefficient.

Where this kind of analysis is applied to brine, for example, when analyzing a fluid in a working environment, it could be necessary to define a maximum salt concentration of 5 %, as in, for example, geothermal brine. This is because the heat transfer properties of the working fluid are not affected considerably, and therefore it is possible to consider it as a pure substance. Such an approximation should be acceptable in order to obtain preliminary results.

On the other hand, different types of methods can be applied to solve the homogeneous heat conduction equation in the cylindrical coordinate system for an absorber tube that is analyzed in two or more dimensions. For example, in the separation of variables in the cylindrical coordinate system method, where an exact will be required whose results will have to be very similar on order to separate the variables. However the choice of methods depends on the approaches that are needed to implement each one of them.

## 4 Conclusions

It has been shown that for annular two-phase flow within a pipe (i.e. an absorber tube) warmed by concentrated solar energy, the temperature differential registered between the hottest point and the coldest point over the external wall of the pipe will increase if the feeding flow increases too. This situation even happens when the internal wall of the pipe is completely wetted. This result verifies that the heat transference from the pipe to the liquid phase of the fluid is not constant and could depend on the liquid film thickness.

In order to be able to raise an experimental correlation for the change of heat transfer coefficient with respect to the film thickness, it is necessary to carry out further experiments with reliable and specialized devices.

The finite-differences method is useful in order to obtain a preliminary approximation for predicting the temperature values of an absorber tube with annular two-phase flow. Because the maximum relative error value was almost 12 % and the minimum was approximately 1 %, these results do not change significantly if the number of nodes in the network is increased. Nevertheless an improvement is obtained because of the error reduction.

In order to obtain preliminary comparative results to determine the viability of a hybrid power project, it is recommended to implement a theoretical analysis of this type. Results will depend on the boundary conditions and approximations that are considered in order to calculate thermal parameters such as the convective heat transfer coefficient. This is why it is very important to verify the theoretical values with experimental results.

To calculate a theoretical value for the convective heat transfer coefficient, some acceptable correlations may be used whose application depends on the operating conditions of the system. In the case of this work with low power and process conditions that are not extreme, is possible to obtain a very good approximation of the temperature profile around the receiver tube.

With respect to the comparison of experimental and theoretical data it was possible to prove that the convective heat transfer coefficient changes as a function of liquid film thickness. The theoretical temperature could correlate better with the experimental data if the convective heat transfer coefficient is considered a variable instead of a constant.

As a final suggestion, if low power solar systems with lower flows are necessary, it might be possible to operate them with an annular two-phase flow and a liquid-steam separator before the last module, in order to obtain steam of high mass quality without completely evaporating the feed fluid. This aspect will be tested in the near future.

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