TRANSIENT ANALYSIS OF TWO-PHASE FLOWS IN A CSP PLANT WITH DIRECT STEAM GENERATION

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Abstract. *Parabolic trough solar concentrators for electric power generation plants are currently being designed all over the world. Concentrating solar power plants generally use synthetic oil as the collector working fluid. The vapor generation is carried out by a heat exchanger. The vapor can be directly generated inside the concentrator by substituting oil with water to avoid the need for a huge quantity of oil. Nevertheless, controlling the convective boiling of the water inside the concentrator represents a considerable challenge due to the strong increase in the heat transfer coefficient and the instability of the flow during steam production.*
In this study, a theoretical transient model to simulate direct steam generation in a solar concentrator is presented. This model computes the quantity and quality of steam produced, its energy (pressure, temperature), the thermal loss and the efficiency of the collector. A parametrical study is described herein and the results show that optimal operation conditions can be found and used as a strategy for the sizing of real installations. Depending of the length of the collector and the solar irradiation, an ideal mass flow is found that minimizes the heat loss and optimizes the evaporation of the fluid. Also, in this case, cloud shading simulation is done to see the transient comportment of the vapor quality.

Keywords: *Direct Steam Generation, Solar Energy, Convective Boiling, Renewable Energy, Heat Transfer.*

1. INTRODUCTION

Electric power generation plants based on parabolic trough solar concentrators are currently in operation. The largest of these is in Kramer Junction, Southern California, and it uses synthetic oil in the collector loop to transfer thermal energy to a Rankin cycle turbine via a heat exchanger (Odeh et al., 2003; Price et al., 2002; Frier and Cable, 1999). To improve the efficiency and reduce the production costs, direct steam generation (DSG) inside the collector field has been extensively studied. The technology is similar to that of the oil-based collector, but water is used instead of oil. When the temperature of the water reaches the saturation temperature a two-phase flow appears inside the collector. The main difficulty associated with this technology is related to control this two-phase flow.

Preliminary experimental studies on the flow patterns and pressure drop in a DSG collector have been reported (Muller, 1991). The feasibility of applying the DSG process in horizontal parabolic trough concentrators has been verified in the DISS project (Zarza et al., 2004). Three different operating concepts have been studied, namely the once-through, recirculation and injection modes of operation (**Fig. 1**). Experiments using this technology carried out at the Plataforma Solar de Almeria (PSA) (Zarza et al., 2004) showed that a stratified flow phase can damage the absorber tube. (J. Zhou et al., 2015) study the instability inside DSG collector and show the importance of the flow pattern on the stability of the system. (M. Elsafi, 2015) describes a steady state where he shows the influence of the inlet pressure to keep the optimal annular flow pattern in the two phase flow part of the collector. Theoretical studies on the heat transfer between the collector wall and the fluid (Odeh et al., 1996) have shown that the efficiency increases with the irradiation up to the point where dry steam is generated. At this point, the heat transfer coefficient strongly decreases along with the efficiency. Thus, due to the transient aspect of direct solar irradiation, it is very important to predict the behavior of the two-phase flow in order to adjust the feed mass flow and maintain the optimal conditions.

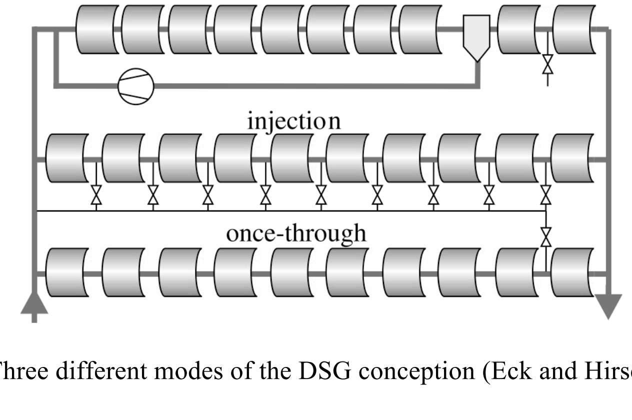


Fig. 1. Three different modes of the DSG conception (Eck and Hirsch, 2007).

In this regard, in (Eck and Hirsch, 2007) the authors present a dynamic non-linear simulation model that allows the study of the interaction between the collector loop and the feed mass flow and shown how controlling the feed mass flow can optimize the system. Recently, (Arousseau et al., 2016) describes the several kind of control method system for DSG. Some of them are currently tested on real prototype.

This paper describes a transient thermo-hydrodynamic model for a DSG system. This model will be experimentally validated, in a near future, on a real DSG collector which is under construction. After a description of the physical model, we will show how the model can be applied to size a real installation.

2. Model description

The ratio between the length and the diameter of the collector is sufficiently large to allow a one space dimension discretization. A finite volume discretization is used. The different equations are all expressed as a function of two state variables: the pressure and the enthalpy (p,h). All state laws and the physical properties of a fluid are deduced from the IAPWS IF97 standard formulations through interpolation. These formulations provide very accurate data for water, steam and mixture properties from 0 – 1000 bar and 0 – 2000 °C. The relative error of the interpolation function is less than 1% for all physical properties. The external heat flux, that corresponds to the solar irradiation, is a variable temporal function created from real DNI measurements or artificially computed.

2.1 Flow conservation equations

Along the collector row and over time (sun intensity), different flow conditions, such as sub-cooled water, two-phase flow and super-heated steam, can be found. For all of these conditions, a common set of conservation equations is used along the length, Δz, of the volume control.

The following energy equation is used to predict the enthalpy value:

$$\frac{\partial h}{\partial t} = \frac{h}{A\rho}\frac{\partial \dot{m}}{\partial z} - \frac{1}{A\rho}\frac{\partial (h\dot{m})}{\partial z} + \frac{Q}{V\rho} \qquad (1)$$

The pressure is computed from the pressure drop correlation, Darcy's equation is used for mono-phasic flow and for two-phase flow a specific correlation is used (Müller-Steinhagen and Heck, 1986), which gives the pressure drop as a function of the vapor title $x(h,p)$ (deduced by the following equation), the mass flow and the pressure.

$$x = \frac{h-h'}{h''-h'}\ ;\ \frac{\partial p}{\partial z} = f(\dot{m}, h(z), p(z)) \qquad (2)$$

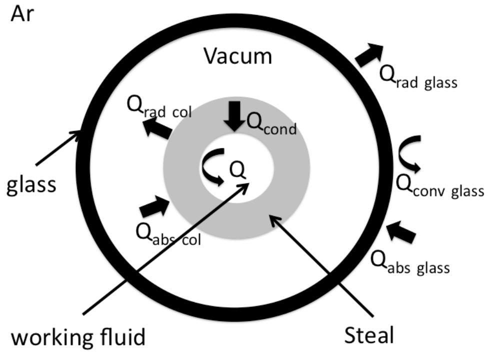
Where h' is the saturated liquid enthalpy which is a function of the pressure and $h"$ is the saturated vapor enthalpy which is a function of the pressure. These functions are interpolated functions computed from the IAPWS IF97. In the one-phase flow area, the density is deduced as a function of the pressure and the enthalpy from the interpolation function found in IAPWS97; $\rho = \rho(p,h)$. In the two-phase flow area, the average density is computed as follows:

$$\rho = \left(\frac{1}{\rho(p,h'(p))} + x\frac{1}{\rho(p,h"(p))} \right)^{-1} \qquad (3)$$

The computation of the thermal flux between the wall and the liquid is described in the following subsection.

2.2 Thermal conservation equation

The heat flux from the wall collector to the flow, Q, is computed using the following set of equations:



$$Q = Q_{cond} = \alpha_H \pi d \Delta z \left(T_{wi} - T\right) \qquad (4)$$

$$A_w \rho_w c_w \frac{\partial T_w}{\partial t} = \underbrace{Q_{col\,abs} - Q_{col\,rad}}_{Q_{ext}} - Q \qquad (5)$$

$$Q_{col\,rad} = Q_{glass\,conv} + Q_{glass\,rad} - Q_{glass\,abs} \qquad (6)$$

$$Q_{ext} = Q_{col\,abs} - Q_{glass\,conv} - Q_{glass\,rad} + Q_{glass\,abs} \qquad (7)$$

Fig. 2. Energy balance for solar collector.

The variables are describes in the nomenclature.

The irradiation fluxes $Q_{col\,abs}$ and $Q_{glass\,abs}$ are dependent on the direct normal irradiation (DNI), the radiation properties (transmissivity τ, reflectivity ρ, and absorptivity α) of the glass, mirror and collector, the concentrator factor C and the modified coefficient IAM, which is dependent on the geographical coordinate and the day of the year. Information to compute this coefficient can be found in (Rabl, 1985; Montes Pita, 2008). The shape factor between the sun irradiation and the collector is considered equal to one and the absorber is considered as a selective surface, obtained by selective paint. This means that the absorptivity of the collector is considered to one and the emissivity is low, defined by a temperature dependent equation. The radiation fluxes $Q_{rad\,col}$ and $Q_{rad\,glass}$ are computed classically, on a Δz element, with the appropriate properties by the following equations (details of the demonstration can be found in (Montes Pita, 2008; Velázquez et al., 2010)):

$$Q_{rad\,col} = \frac{\sigma\pi\Delta z D_{col}(T_{col}^4 - T_{glass}^4)}{\dfrac{1}{\varepsilon_{col}} + \dfrac{\left(1-\varepsilon_{glass}\right)D_{col}}{\varepsilon_{col}D_{glass}}} \tag{8}$$

$$Q_{rad\,glass} = \varepsilon_{glass}\sigma\pi\Delta z D_{glass}\left(T_{glass}^4 - T_{sky}^4\right) \tag{9}$$

T_{sky} is computed as follows (Duffie and Beckman, 1980): $T_{sky} = 0.0552 T_{ext}^{1.5}$. Q_{cond} is computed normally as a cylindrical thermal resistance and gives a relation between T_{win} (wall temperature of the part of the collector in contact with the fluid) and T_{col}. T_w corresponds to the average temperature of the collector tube (between T_{win} and T_{col}) for the element considered. $Q_{conv\,glass}$ is computed using a classical convection correlation depending of the wind velocity (Incropera et al., 2011).

The heat transfer coefficient, α_H, is computed from correlations. For single-phase flow, the classical Dittus – Boetler correlation is used:

$$\alpha_H = 0.0235\frac{\lambda}{D_h}Re^{0.8}Pr^{0.4} \tag{10}$$

For two-phase flow, a correlation which is dependent on the vapor quality and the heat flux is used: $\alpha_H = f(\dot{m}, x(h,p), h, p, Q)$ (Gungor and Winterton, 1986).

2.3 Boundary conditions

The enthalpy is imposed at the entrance and the pressure can be imposed at the inlet or the outlet. The mass flow is imposed at the entrance and can be time dependent. In this version there is no equation for the mass flow and therefore the mass flow is considered to be spatially constant. For the thermal model, an ambient temperature is considered as known, a wind velocity is imposed and the DNI is known.

3. MODEL VALIDATION

To validate the model, the results are compared with predicted results for one row of collectors at a DSG thermosolar plant of 20 MWe (Montes Pita, 2008). Table 1 describes the parameters used for the computation.

Table 1. Parameters for one row of collectors.

Inner diameter of steel absorber pipe (m)	0.055
Outer diameter of steel absorber pipe (m)	0.07
Parabola width (m)	5.76
Equivalent peak optical efficiency	0.74
Emissivity collector	0.04795 + 0.0002331 T(°C)
DNI (W/m^2)	850

Figure 3 shows a schematic representation of a typical solar field considered in this study. Our results are compared with the predicted results of the DSG thermosolar plant (Montes Pita, 2008). The errors for the outlet quality vapor, pressure and enthalpy are less than 2%. The configuration consists of a water preheating and biphasic part of 588 (m) and a super heated dry vapor part of 245 (m). This configuration represents the recirculation configuration mode in which the water remaining at the end of the preheating and biphasic part is re-injected at the inlet. The vapor quality at the end of this part is 0.77, which corresponds to a recirculation rate of 0.3.

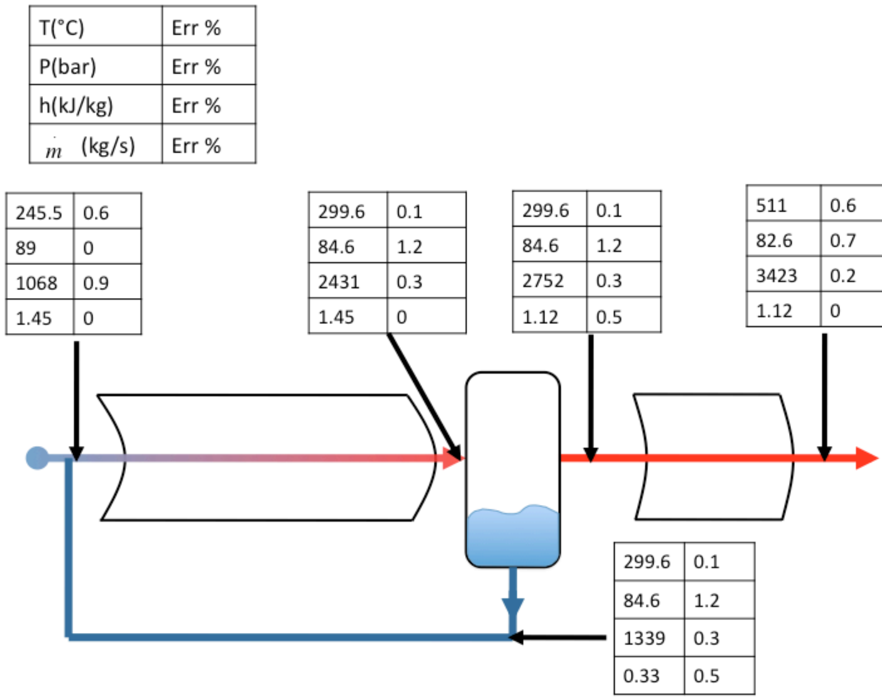


Fig. 3 Comparison of model results with those reported by (Montes Pita, 2008) for a typical row of collectors.

4. MODEL ANALYSIS

4.1 Parametric studies

In this section, the model is used to get a better understanding of the behavior of the system for several values of length of the collector and the mass flow. At the end of the collector, if the vapor quality differs from 1, the rest of the water is re-injected at the inlet at its saturation temperature. This corresponds to the recirculation mode. A vapor quality equal to one indicates the one-though mode. The same conditions used for the validation are employed. The mass flow and the length of the collector are changed and the ratio between the thermal loss and the power used for evaporation Q_{vap} is computed, as shown in figure 4. Q_{vap} is computed using the following equation :

$$Q_{vap} = x\dot{m}\left(h_{out} - h_{in}\right) \tag{11}$$

Figure 4 shows that there is a mass flow value that minimizes the thermal loss for each length. This mass flow corresponds to an outlet vapor quality of 0.95 in this configuration, but this may be dependent on the inlet temperature. Consequently, for each length, the left part of the curve corresponds to the one-though mode and the right part to the evaporator portion of the recirculation mode.

If the outlet vapor quality is equal to 1 (super heated part of the collector), when the mass flow decreases the thermal loss strongly increases. This is due to the increases in the length of the super heater part. In this part, the heat transfer coefficient strongly decreases and the fluid and wall temperature increase, leading to a high thermal loss.

This large difference between the wall temperatures of the super heater part and the evaporator part can be a problem if the junction between these two parts is moving. This is the main drawback associated with the one-though mode. The fluctuation of the DNI can move the location of the junction and generate high thermal stress (Eck et al., 2003) which can damage the collector. Thus, the recirculation mode with an evaporator part (with outlet vapor quality < 1) and a second separate super heater part is preferable. Figure 5 shows the ratio between Q_{vap} and the solar power that arrives at the outer diameter of the tube $Q_{sol\ abs} = IAM.\eta_{optic}.DNI$. These curves show that the same value observed in figure 4 for the mass flow optimizes this ratio in figure 5. Therefore, it appears that an outlet vapor quality value of close to 0.95 is required to optimize the evaporator part of the system and with a lower vapor quality the efficiency of the system will decrease.

A similar study was carried out with a collector length of 800 m and the variable parameter was the DNI (between 650 W/m^2 and 1050 W/m^2). Figure 6 shows the ratio between Q_{vap} and $Q_{sol\ abs}$ as a function of the mass flow for several DNI values. The same profile is observed, which indicates that a certain mass flow value will optimize the system depending on the DNI. This mass flow value also corresponds to a vapor quality equal to 0.95.

In summary, for a particular length of one row of the collector, the model described represents an important aid for the optimization of the mass flow in the evaporator part. However, the optimized value for the mass flow leads to an outlet vapor quality of 0.95, which is very close to one. If this value increases slightly, the system could generate super heated vapor in the evaporator part, which would lead to high thermal stress and damage the collector. Thus, a transient study is fundamental to understand the dynamic behavior of the system.

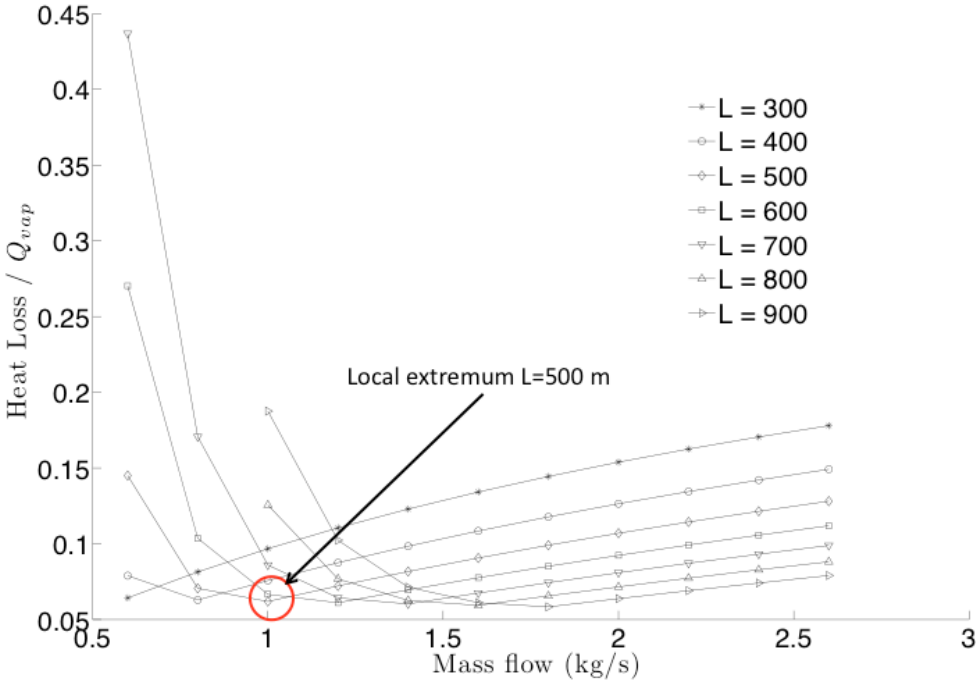


Fig. 4. Ratio between thermal loss and power gain by the vapor (Q_{vap}).

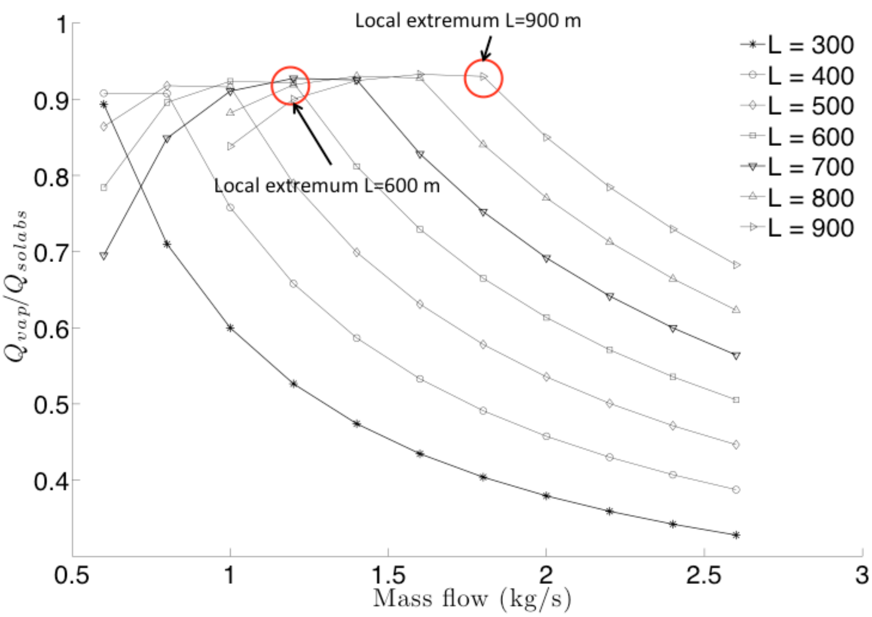


Fig. 5. Ratio between power gain by the vapor (Q_{vap}) and the power from the sun.

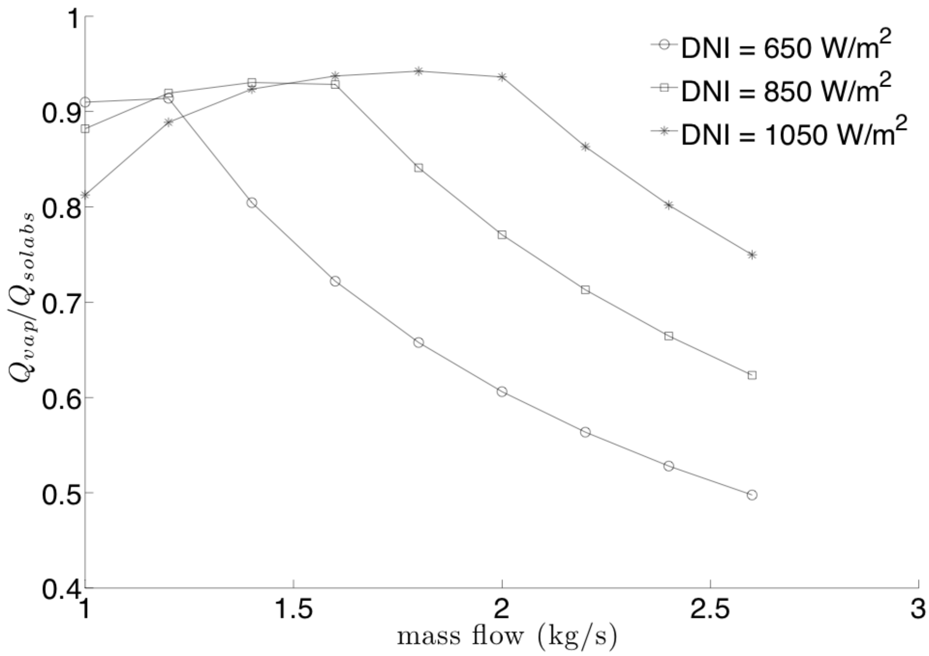


Fig. 6. Ratio between Q_{vap} and $Q_{sol\,abs}$ for several DNI values.

4.2 Transient study of the system

Under real conditions, the DNI can vary considerably, for example, when clouds form. As previously mentioned, a high vapor quality value can optimize the system. Therefore, when a perturbation of the DNI arises, it is of interest to regulate the mass flow in order to maintain optimal conditions. Nevertheless, the recirculation mode has the particularity of increasing the parasite load of the system. If the DNI decreases suddenly due to the passing of a cloud, the vapor quality will also decrease leading to an increase in the amount of recirculating water. Thus, a higher quantity of quasi-saturated water will enter on the inlet of the collector leading to an increase in the vapor quality. Consequently, it is very important to study the characteristic time of the system in order to obtain a system with good mass flow regulation.

Figure 7 shows the evolution of the vapor quality after a 50% decrease in the DNI. A period of oscillation can be observed before the stabilization of the system. In this case, the characteristic time is more or less 15 min. This means that 15 min after the perturbation the system has stabilized. Under the optimal conditions, this value is not strongly dependent on the length of the collector. A more complete study of the transient response of the system can be carried with the model and the procedure is straightforward. The aim is to understand how to gain good control of the mass flow in order to maintain the optimal conditions to avoid generating a high level of thermal stress.

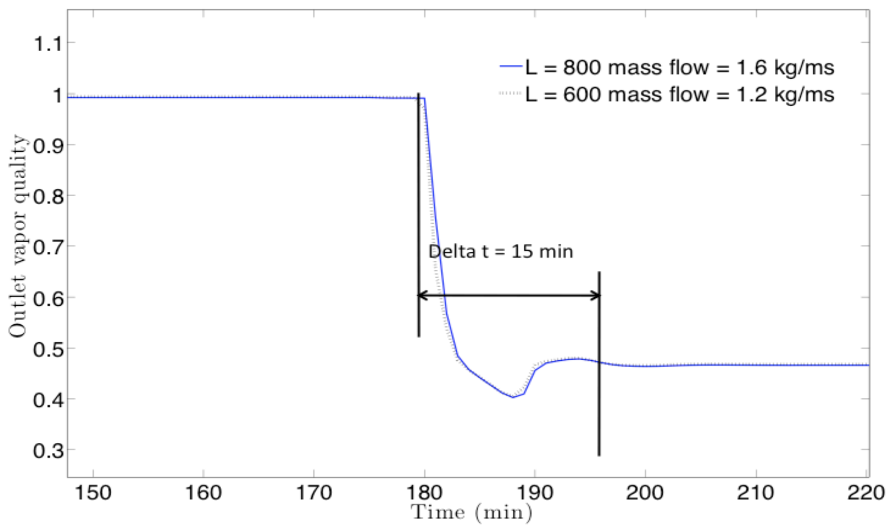


Fig. 7. Response in terms of the vapor quality to a decrease of 50% in the DNI.

5. CONCLUSIONS

In conclusion, in this paper the initial stage of a numerical study on direct steam generation is presented. The transient bi-phasic model is described and validated. The model shows that optimum operation parameters as the mass flow for the sizing of a row of collectors can be found. The model can also be used to study the dynamic response of the system under transient conditions.

The gathering of further results for the transient analysis is a straightforward task and these will be used to develop an inlet mass flow control methodology. A post-treatment algorithm for the local weather data has already been developed in order to obtain real DNI values for several places in Brazil. As an end result, these studies will provide a powerful sizing tool.

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NOMENCLATURE

p (pa)	Pressure	**Subscript**:	
h (J/kg)	enthalpy	cond	Conduction inside the steel part of the collector
$\dot{m}$ (kg/s)	Mass flow	col	Flux on the outer diameter of the steel part of the collector
A (m^2)	Cross section	glass	Flux on the glass part of the collector
ρ (kg/m^3)	Density	conv	Convective flux
Q (W)	Heat flux	rad	Radiation flux
x	Quality vapor	abs	Absorption flux
c (j/kgK)	Thermal capacity	Q_{vap} (W)	Power gain of the vapor
α_H (W/m^2K)	Heat transfer coefficient	in	Inlet properties
DNI (W/m^2)	Direct normal irradiation	out	Outlet properties

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