Validation of an Alternative Methodology for Direct Steam Generation Modelling in Parabolic Collectors

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Abstract

The analysis of the two-phase flow in linear concentrators remains a fundamental part of implementing the concept known as direct steam generation in concentrated solar thermal plants. The models developed to date require, on the one hand, an iterative analysis to estimate the coefficient h, and, on the other hand, high demand for computational fluid dynamics calculations. In the present work, with the help of Adiutori's methodology, a one dimensional thermo-hydraulic model of a parabolic trough collector in the direct steam generation is described. The difference between the present model, which stands out from the rest, is that for the analysis of convective heat transfer, a functional of the temperature is evaluated, instead of dimensionless groups to estimate the h coefficient, a similar approach is used for the pressure drop instead of the fluid friction factor. This allows to solve directly, and without the need for iterative processes, the model to predict the temperature rise and the pressure drop along the receiver, especially in the two-phase flow zone. The results obtained are similar to the experimental data published by different researchers, which validates the developed model, and will allow simulations to be carried out in a more agile way and reduce calculation errors. This ensures the usefulness of the model for further analysis.

Keywords: Thermo-hydraulic model, two-phase flow, heat transfer, boiling, parabolic collector

1. Introduction

Solar plants for conversion of thermal energy into electrical energy through power cycles (CSP) have been oriented to solar fields with parabolic trough collectors (PTC), using mainly thermal oil as heat transfer fluid (HTF), so two different circuits are operating with two different fluids, the solar field operating with thermal oil and the feed block operating with water/steam (Moya, 2021).

The next evolution in linear concentrating systems has been to implement HTFs that operate at higher temperatures, where an alternative is a process called direct steam generation (DSG), where steam is generated directly in the solar field (Giglio et al., 2017; Hakkarainen and Kannari, 2015; Zarza Moya, 2017). Establishing the behavior of CSP systems is important for this technology to begin to be introduced into the energy matrix because although there are currently approximately 120 plants of this type, the installed capacity is of the order of 6387 MW (IRENA, 2022a), which corresponds to less than 2% of the installed capacity of renewable energy sources (IRENA, 2022b).

The main issue with DSG modeling is the two-phase flow zone, due to the inherent complexity of the boiling process in a stream of water flowing inside a tube (Alobaid, 2018; Hewitt, 1998). To date, there is considerable progress in DSG analysis (Giglio et al., 2017; Islam et al., 2018; Sandá et al., 2019), however, they all resort to the concept of classical heat transfer, with the calculation of the convective coefficient h, in combination with the friction factor f, to understand the thermo-hydraulic behavior of the installation, even in CFD models (Lobón et al., 2014; Pal and K, 2021).

Adiutori (2017) has proposed a generalized methodology to eliminate the convective heat transfer coefficient h and the friction factor f to model more simply the behavior of thermo-hydraulic systems, known as "The New Engineering". This methodology requires transforming the Nusselt number relationships and the Moody diagram

doi:10.18086/eurosun.2022.12.06 Available at http://proceedings.ises.org

to obtain functional relationships dependent on measurable variables such as pressure and temperature. Thus, it is possible to solve a thermohydraulic model without the need of iterative processes, reducing the calculation time and the error in the solution. In previous work (González-Mora and Duran García, 2022), we proposed a methodology, summarized in Section 2. However, its validation was pending. In this current work, we aim to validate this methodology by comparing it with experimental data reported by other researchers, as described in Section 3. This validation will establish the reliability and applicability of our proposed thermo-hydraulic model for future analyses.

2. Thermo-hydraulic model

The developed model takes as reference those described by other authors (Forristall, 2003; Hachicha et al., 2018; Montes Pita, 2008; Sun et al., 2015; Vasquez Padilla, 2011). The general one dimension (1D) modeling approach consists of subdividing the receiver into smaller HCE (heat collector element), averaging the physical quantities ψ over the cross-sectional area A perpendicular to the flow direction x_1 by:

$$\int_V \psi \, \mathrm{d}V = \int_{\Delta x_1} \left[\int_A \psi \, \mathrm{d}A \right] \mathrm{d}x_1 \qquad \text{(eq. 1)}$$

where Δx_1 is the increment in length. The purpose of the 1D models is to understand the behavior of the systems using the balance equations, complemented with the constitutive equations (El Hefni and Bouskela, 2019). The difference in the model to be developed lies in the fact that we will opt for a methodology that does not require iteration to solve the heat flows and associated temperatures in the thermal model, as proposed by Adiutori (2017).

2.1. Assumptions

Temperature, heat fluxes, and thermodynamic properties are assumed to be uniform around the circumference of each HCE. The optical properties of all materials involved are considered as constant isotropic quantities and independent of temperature, except for the emittance of the receiver. Radiation fluxes in opaque (such as absorber) and semitransparent (glass cover) materials are approximated to surface phenomena (Howell et al., 2015; Özışık, 1973).

The Eurotrough parabolic collector is 12,27 m long, and the supports are placed at 4,06 m from each other. Here it will be assumed that each module has 3 supports, so the effective length of each module will be 12,12 m, however, it is initially assumed that the remaining 12 cm corresponds to protective shields and joints that allow connecting the modules to form a loop and that in this small section, there are no heat losses, nor considerable pressure drops.

2.2. Model description

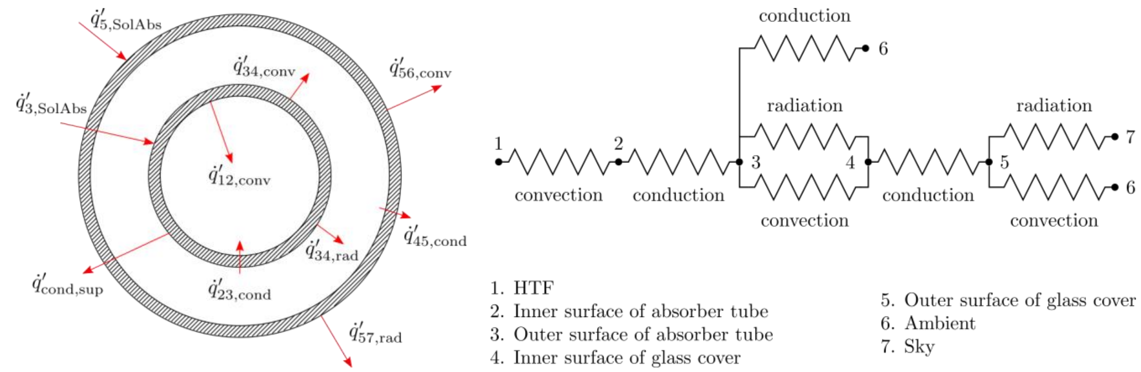
As the HTF absorbs energy, the HCE increases in temperature. Due to the difference between the average fluid temperature at each cross-section and the ambient temperature, heat losses will exist. Fig. 1(a) allows for identifying the heat flows in the HCE, where the heat losses through the supports are also identified; and Fig. 1(b) shows the thermal resistance model specifying the surfaces involved.

With these schematics, it is possible to establish the balance equations applying the first law of thermodynamics, considering flows per unit length, we can define the equations for the thermal resistance model shown in Fig. 1(b).

$$\dot{q}'_{12,conv} = \dot{q}'_{23,cond} \qquad \text{(eq. 2)}$$

$$\dot{q}'_{3,SolAbs} = \dot{q}'_{34,conv} + \dot{q}'_{34,rad} + \dot{q}'_{23,cond} + \dot{q}'_{38,cond} \qquad \text{(eq. 3)}$$

$$\dot{q}'_{45,cond} = \dot{q}'_{34,conv} + \dot{q}'_{34,rad} \qquad \text{(eq. 4)}$$



a) Heat fluxes on the HCE **b) Thermal resistance model**

Fig. 1: Thermal model

$$\dot{q}'_{45,cond} + \dot{q}'_{5,SolAbs} = \dot{q}'_{56,conv} + \dot{q}'_{57,rad} \qquad \text{(eq. 5)}$$

$$\dot{q}'_{12,conv} = \dot{m}(h_{in} - h_{out}) \qquad \text{(eq. 6)}$$

where each of the heat fluxes can be expressed as:

$$\dot{q}'_{i,SolAbs} = IAM\rho_{PTC}(\alpha\tau)_i W G_{bn} \qquad \text{(eq. 7)}$$

$$\dot{q}'_{ij,cond} = \frac{2\pi k_{ij}(T_i - T_j)}{\ln D_j/D_i} \qquad \text{(eq. 8)}$$

$$\dot{q}'_{ij,rad} = \frac{\sigma\pi D_i(T_i^4 - T_j^4)}{\frac{1}{\varepsilon_i} + \frac{(1-\varepsilon_j)D_i}{\varepsilon_j D_j}} \qquad \text{(eq. 9)}$$

$$\dot{q}'_{ij,conv} = h_{ij} D_j \pi (T_j - T_i) = f(\Delta T) \qquad \text{(eq. 10)}$$

where IAM is the incidence modifier angle, ρ_{PTC} is the reflectance of the mirrors, $(\alpha\tau)_i$ is the product of the absorptance and transmittance of surface i, W is the trough width, G_{bn} is the incident solar radiation in the normal direction, k is the surface conductivity, T is the temperature, D is the diameter, σ is the Stefan-Boltzmann constant, ε is the emittance and h is the convective coefficient. The subscripts i and j allow us to identify each of the surfaces, according to Fig. (1b). Note that Eq. (10) initially involves the coefficient h, which will be replaced in its entirety by a functional of the temperatures, as explained below. In the present work, it is worth mentioning that only Eq. (10) is discussed, as it presents the novelty of the thermal model, while Eqs. (2) to (9) are not ignored since they are used to solve the system of equations.

The annulus formed between the absorber tube and the glass envelope is modeled as free convection between two cylinders because the receiver is evacuated. Between the glass cover and the environment the convection can be forced or natural. In the first case the Žhukauskas equation is used, while the Churchill and Chu equation is adequate for the second case (Bergman et al., 2011). For convection between the absorber tube and the heat transfer fluid, there are two cases: the Pethukhov or Gnielisnki equation (Bergman et al., 2011) for single-phase fluid, and the Gungor and Winterton correlation for two-phase flow (Gungor and Winterton, 1986).

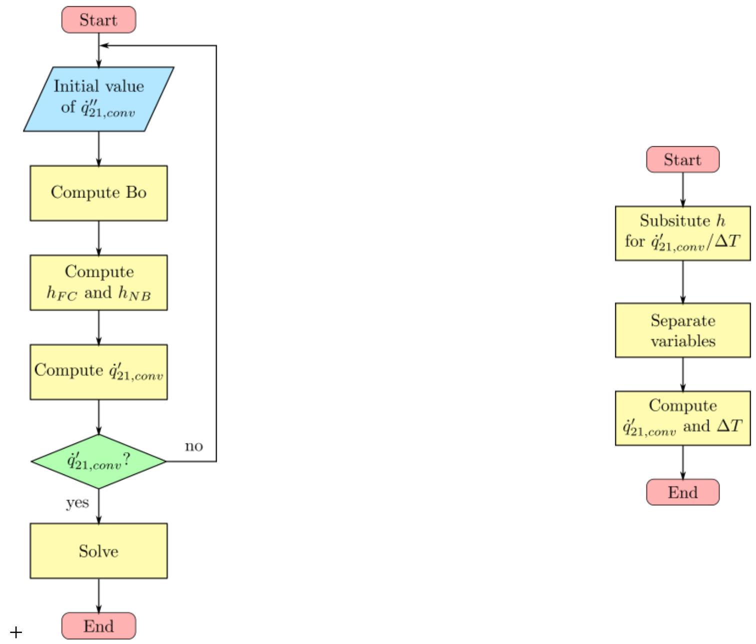
2.3. Transformation of heat transfer equations according to Adiutori's methodology

Traditionally, when temperatures and heat fluxes are unknown in a heat transfer problem, it is necessary to resort to iterative processes to get to know these unknowns. The problem becomes more complex when the three transfer phenomena are coupled since the equations are highly nonlinear. Special attention is given to convection due to the nature of the Nusselt relations that allow determining the convective coefficient h, generating great interest in fluids with phase change. Adiutori (2017) has proposed a methodology that allows giving a more agile solution to heat transfer problems where temperatures and heat fluxes are unknowns, without the need for iterative processes.

In this methodology, the aim is to obtain functionals of separable variables that allow the solution of any heat transfer problem. Conventional heat transfer coefficient correlations are easily transformed in the new engineering by replacing h with $q/\Delta T$, then separating q and ΔT, which allows obtaining correlations of the type $q = f(\Delta T)$ or $\Delta T = f^{-1}(q)$. Thus, dimensionless group parameters are replaced by individual parameters, and those that are temperature-dependent are replaced by temperature functions.

In this study, we present an approach that allows for the evaluation of parameters without relying on dimensionless groups as stated by Adiutori (2017). This reveals the direct relationships between individual parameters, enabling correlations to be developed based solely on experimentally measurable quantities. This approach facilitates the computation of current phenomena under specific operating conditions.

For comparison, Fig. 2 illustrates the flowcharts used to obtain the convective heat flow in the two-phase zone. In the traditional methodology (Fig. 2(a)), one needs to consider an initial value of $\dot{q}'_{21,conv}$, determine h_{FC} and h_{NB} using the Bo number, compute $\dot{q}'_{21,conv}$ and iterate until convergence is achieved. However, Fig. 2(b) shows that by substituting $\dot{q}'_{21,conv}/\Delta T$, one can obtain a functional that can be separated and solved more easily. This transformation, according to Adiutori's methodology, proves to be more appropriate, resulting in lower solution times. Overall, our proposed approach offers a more efficient and effective way to evaluate parameters, and the results demonstrate its superiority over the traditional methodology in terms of computational efficiency.



a) Calculation with traditional methodology **b) Calculation with the proposed methodology**

Fig. 2: Comparison of methodologies for heat flow calculation

2.4. Transformation of the two-phase flow equation and friction factor

The Gungor and Winterton correlation involves two convective coefficients, one for forced convection and the other for nucleated boiling; where a boiling factor (F) dependent on the Martinelli parameter (X) and the boiling number by (Bo) is involved:

$$h_{12} = h_{FC} + h_{NB} \quad \text{(eq. 11)}$$

$$h_{FC} = 0{,}023\mathrm{Re}_l \mathrm{Pr}_l^{0{,}4}(k_l/D_2)F \qquad \text{(eq. 12)}$$

$$h_{NB} = 55p_r^{0{,}12-\log\varepsilon} - \log^{-0{,}55} p_r\, PM^{-0{,}5}\left(\dot{q}'_{12,conv}\right)^{0{,}67} S \qquad \text{(eq. 13)}$$

$$F = 1 + 2{,}4 \times 10^4 \mathrm{Bo}^{1{,}16} + 1{,}37X_{tt}^{-0{,}86} \qquad \text{(eq. 14)}$$

$$X = \left(\frac{x}{1-x}\right)^{-0{,}9} \left(\frac{\rho_l}{\rho_g}\right)^{-0{,}5} \left(\frac{\mu_g}{\mu_l}\right)^{-0{,}1} \qquad \text{(eq. 15)}$$

$$\mathrm{Bo} = \frac{\dot{q}'_{12,conv}\pi D_2^2}{4\dot{m}h_{fg}} \qquad \text{(eq. 16)}$$

$$S = \left(1 + 1{,}15 \times 10^{-6}F^2\mathrm{Re}_l^{1{,}17}\right)^{-1} \qquad \text{(eq. 17)}$$

$$\mathrm{Re}_l = \frac{4\dot{m}^2(1-x)}{\pi\mu_l D_2} \qquad \text{(eq. 18)}$$

where Re is the Reynolds number, Pr is the Prandtl number, k is the conductivity, p_r is the reduced pressure, ε is the pipe roughness, PM is the molecular weight, x is the vapor quality, ρ is the density, μ is the viscosity and h_{fg} is the enthalpy of vaporization; with subscript l for the liquid phase and subscript g for the vapor phase.

Incorporating the methodology outlined in the previous section, we can rewrite Eqs. (11) to (13) to establish a relationship between the two-phase convection heat flow ($\dot{q}'_{12,conv}$) and the temperature difference (ΔT) between the fluid and the receiver surface. This relationship can be expressed in the following form:

$$\Delta T = C_1\dot{q}'_{12,conv}\left[C_2\left(C_3 + C_4\dot{q}'_{12,conv}{}^{1{,}16}\right)^2 + C_5 - \frac{C_6\dot{q}'_{12,conv}{}^{0{,}67}}{C_7\left(C_8+C_9\dot{q}'_{12,conv}{}^{1{,}16}\right)^2 - 1}\right]^{-1} \qquad \text{(eq. 19)}$$

Each C_i in the functional parameter of temperature is clearly positive ($C_i > 0$), and can be obtained from any thermodynamic properties database. The general behavior of the function is illustrated in Fig. 3(a), revealing a continuous function with a unique correspondence between ΔT and $\dot{q}'_{12,conv}$. This implies that there is no need for iterations in the solution of the heat transfer in each HCE. In Fig. 3(b), the reported behavior of the boiling curve as a function of temperature is presented. By comparing the results with the reported boiling curves, it can be confirmed that the transformation is indeed adequate, further validating the proposed methodology.

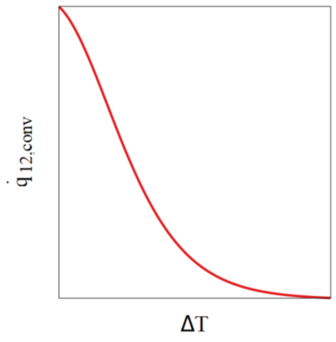
The friction factor is obtained in an iterative manner from

$$\frac{1}{\sqrt{f}} = -2\log\left(\frac{\varepsilon/D_2}{3{,}7} + \frac{2{,}51}{\mathrm{Re}_D\sqrt{f}}\right) \qquad \text{(eq. 20)}$$

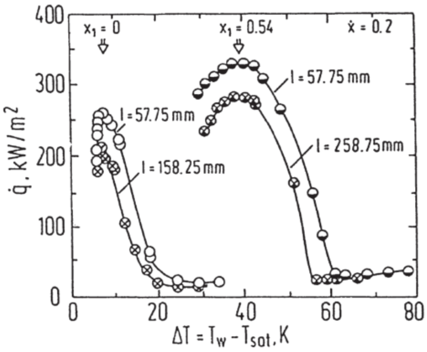
because it represents the Moody diagram (Bergman et al., 2011), with the variables:

$$f = \frac{\mathrm{d}P}{\mathrm{d}x}\frac{2D}{\rho u_m^2} \cong \frac{\Delta P}{L}\frac{2D}{\rho u_m^2} \qquad \text{(eq. 21)}$$

$$\mathrm{Re}_D = \frac{\rho u_m D}{\mu} \qquad \text{(eq. 22)}$$



a) Boiling curve for the proposed model



b) Boiling curves for pure substances x=0 and mixtures with a concentration x=0.54 (Auracher and Marroquin, 2019)

Fig. 3: Convection of the two-phase flow inside the tube

and the ratio ε/D. Multiplying the friction factor by $0{,}5\mathrm{Re}^2$ yields a set of parameters $(D^3 g\rho/L\mu^3)\Delta P$, which is independent of $\dot{m}$. Thus, by plotting the parameter $(D/\mu)\dot{m}$ on the abscissae, and the parameter $(D^3 g\rho/L\mu^3)\Delta P$ on the ordinates, the resulting plot will have the form of pressure drop behavior concerning mass flow, i.e., it will have the form $\Delta P = \Delta P(\dot{m})$, i.e., since they are independent parameters, iterative processes will not be necessary to estimate the pressure drop. Fig. 4(a) shows the conventional Moody diagram, while Fig. 4(b) shows the modified Moody diagram.

3. Model evaluation and validation

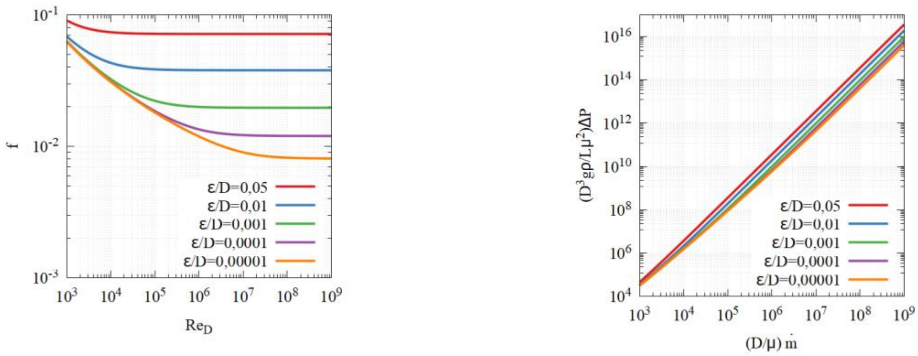
To validate the thermo-hydraulic model, a two-part approach was employed to ensure the reliability of the results. This unique methodology eliminates both the convective heat transfer coefficients (h) and, the friction factor (f). The first part of the validation focused on verifying the two-phase flow pattern, while the second part involved analyzing the pressure drop and temperature increase along the loop. This separation was necessary as the model employs an alternative approach compared to traditional models, and it was crucial to ensure its full applicability and accuracy in both aspects of the validation process.

3.1. Two-phase flow regime

When two-phase flow circulates within a pipe, different flow patterns can occur (see Fig. 5), each with very special thermal and hydrodynamic characteristics (Tomei et al., 2015), therefore, characterizing the type of flow is of utmost importance. It has been described in the literature that an annular flow should be used in DSG plants since this minimizes the deformations of the absorber, and maximizes its useful life (Almanza et al., 1997; Cui et al., 2013; Khanna et al., 2014; Li et al., 2017), in addition, to enhance the heat transfer (Eck and Steinmann, 2005; Montes Pita, 2008; Zarza Moya, 2003).

To identify the type of flow, at least two maps have been proposed. The first of these maps was developed by Taitel and Dukler (1976), involving various dimensionless groups (such as the Froud number), or the more recently developed map of Barnea (1987) relating the velocity of stream and water. It has been decided to use the Taitel and Dukler map due to its ease of use, and its solution is quite large with well-defined flow intervals.

Fig. 6 shows a comparison of the results obtained in the DUKE loop (Feldhoff et al., 2016) plotted using a dashed line with black marker, and those obtained using the proposed model (solid orange line), for a mass flow of 1,3 kg s^{-1}, an irradiance of 900 Wm^{-2}, and a pressure of 80 bar. One can see that the black and orange lines are superimposed, thus the solution obtained is adequate. The bibliography mentions that caution must be taken in the use of this diagram, especially in the area near the beginning or end of evaporation, since some type of stratification may occur (Hirsch et al., 2014). When analyzing the results shown, it can be seen that at the beginning of the evaporation (Martinelli numbers X higher than 1.6), there is an intermittent flow. The presence of intermittent flow in the initial stages of boiling process in a horizontal pipe carrying water-steam mixture, as explained by Feldhoff et al. (2016), may not be a cause for concern if it eventually evolves into an annular flow. The transition to annular flow results in a more stable and predictable two-phase mixture, and the pressure drop across the pipe becomes more consistent, as stated in the same reference.



a) Moddy's diagram **b) Modified Moddy's diagram**

Fig. 4: Pressure drop model

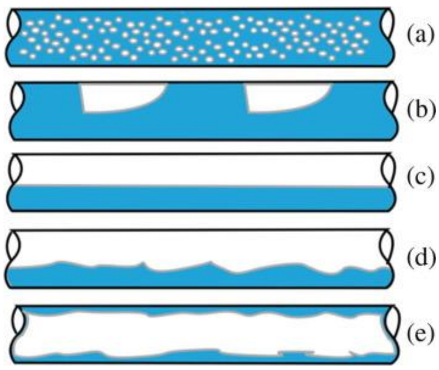


Fig. 5: Flow pattern for horizontal pipes. (a) Disperse bubble. (b) Intermittent. (c) Smooth stratified. (d) Wavy stratified. (e) Annular (López et al., 2016)

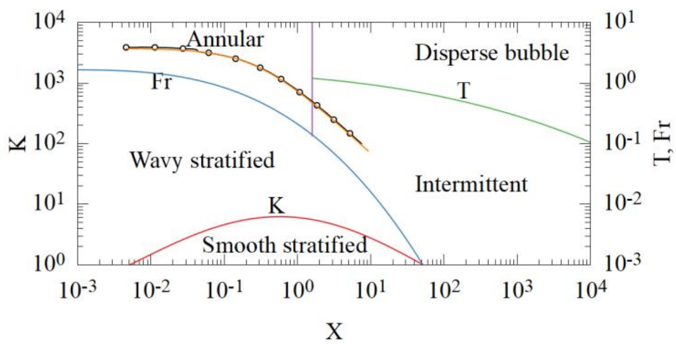


Fig. 6: Superposition of the simulation flow regime on the generalized flow regime map for two-phase flow in horizontal pipes. Adapted from (Feldhoff et al., 2016)

Once the flow pattern in the two-phase zone has been verified, it is possible to compare the pressure drop in this zone with Eck's model (2005), emphasizing that Eck's model employs a "traditional" model, in addition to being an approximation of a 2D model to 1D, by considering the wet angle inside the pipe; while the model developed is 1D. In this case, a flow of 1 kg s^{-1} has been simulated for a pipe of 50 mm inner diameter.

The results of both simulations are presented in Fig. 7, with solid lines representing the simulation results of the developed model, and dashed lines with markers representing Eck's results (2005) . It is evident that the profiles are quite similar, with only slight deviations observed in the developed model, where the values are slightly higher by no more than 15% for ΔP, and less of 1% for the total pressure along the loop, for each case (30, 60 and 100 bar). . This outcome aligns with the findings of Montes (2008), which suggested that the values should be slightly higher in comparison to 2D or 3D models. However, it is worth noting that a potential limitation is the need to generate a new heat transfer function when changing the operating conditions, such as pressure.

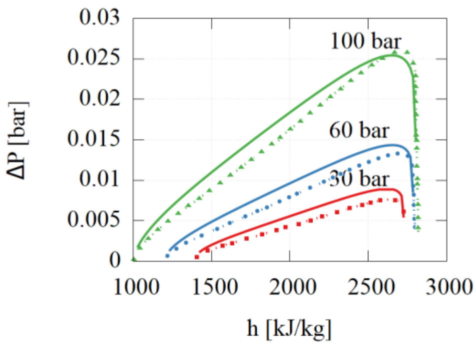


Fig. 7: Pressure drop in the two-phase flow region. Adapted from (Eck and Steinmann, 2005)

3.2. Temperature rise and pressure drop evolution through the loop

The last stage of the validation consisted of analyzing the entire loop to determine the evolution of the water temperature and pressure drop along the loop as a whole. For this, the operating parameters and the experimental results of the DISS loop were taken (Lobón et al., 2014): a mass flow of 0,59 kg s^{-1}, an irradiance of 807 W $m^{-2,}$ and a pressure of 101,9 bar at the loop inlet. Fig. 8 shows the experimental results (markers) and the continuous line shows the simulation results, where the convergence of the results of the developed model can be seen. In this case, the error between temperatures is less than 1%, so the model is validated.

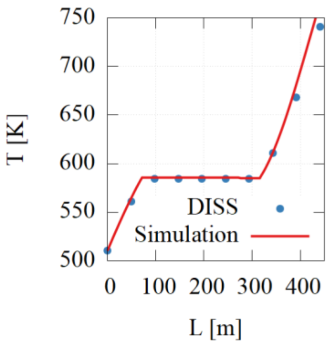


Fig. 8: Pressure drop in the two-phase flow region. Adapted from (Eck and Steinmann, 2005)

4. Conclusions

The energy model developed allows evaluating without complications the evolution of temperature and pressure drop along the parabolic trough loop. It has been identified that the flow pattern can be predicted according to the Taitel and Dukler diagram (see Fig. 6) and that when simulating the DUKE loop conditions obtained experimentally [8], a mostly annular flow pattern is verified, except for the beginning of the boiling zone, which is intermittent. The transition from intermittent to annular flow during the boiling process in a horizontal pipe carrying water-steam mixture is not a concern if it evolves to an annular flow.

The thermohydraulic model was fully validated by estimating the evolution of temperature and pressure drop along the loop. In both cases, the difference between the results obtained by the developed model and the experimental results in the DISS loop (Eck and Steinmann, 2005) is less than 1%, as shown in Fig. 8, which ensures the convergence of the analysis.

With the aforementioned, the proposed methodology can be confidently applied to various future works, such as the modeling of a conceptual DSG plant in the Northwest region of Mexico. This methodology can effectively establish the operating parameters and design conditions for concentrating solar power systems that utilize direct steam generation. The robustness and reliability of this methodology make it a valuable tool for advancing research and development in the field of solar energy technologies.

5. Acknowledgments

Authors acknowledge CONACyT for a Ph.D. grant for EGM.

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