# Modelling of a cross flow evaporator for CSP application: analysis of the use of different two phase heat transfer and pressure drop correlations

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**Abstract**

Heat exchangers consisting of bundles of horizontal plain tubes with boiling on the shell side are widely used in industrial and energy systems applications. A recent particular specific interest for the use of this special heat exchanger is in connection with Concentrated Solar Power (CSP) applications. Heat transfer and pressure drop prediction methods are an important tool for design and modelling of diabatic, two-phase, shell-side flow over a horizontal plain tubes bundle for a vertical up-flow evaporator. With the objective of developing a model for a specific type of cross flow evaporator for a coil type steam generator specifically designed for solar applications, this paper analyzes the use of several heat transfer, void fraction and pressure drop correlations for the modelling the operation of such a type of steam generator.

The paper after a brief review of the literature about the available correlations for the definition of two-phase flow heat transfer, void fraction and pressure drop in connection with the operation of steam generators, focuses attention on a comparison of the results obtained using several different models resulting by different combination of correlations. The influence on the analysis of the performance of the evaporator, their impact on significant design variables and the effective lifetime of critical components in different operating conditions, simulating the daily start-up procedures of the steam generator is evaluated. The importance of a good calibration of the model based on the comparison with some experimental data is recognised.

**Keywords**: Concentrated Solar Power, Coil Steam Generators, Tube bundle, Evaporator, Heat transfer coefficient, Load variations.

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## 1. Introduction

In recent years the power generation industry has been affected by a significant growth of small capacity intermittent renewable on-site power plants. In such a new distributed energy scenario, concentrated solar power (CSP) plants can surely play an important role, in the near future.

Concentrating solar is a promising technology, but it still has open operational issues that are a challenge compared with other rival technologies. Considering the discontinuous nature of the power source [1], the analysis of solar plant operation has to consider not only the design of the components, but also the analysis of the performance of the system under different operating conditions. Very fast transients may occur and all the operation units may have shorter life cycles for strong dynamics and frequent changes of operating conditions.

For this reason, from the process viewpoint, it is necessary to go through a dynamic modelling simulation phase. This is particularly important for systems involving strong dynamics and for particular components, like the steam generator.

The steam cycle involves components similar to those operating in a conventional power plant. With focus on the dynamic performance of these, the steam generator is the most important one. The operation of these components will be constrained by the allowable temperature and pressure gradients during the start-up and shut-down phase causing pressure and thermal stresses.

In this case the simulation of the various operating condition is important not only in order to evaluate the performance of the system, but also for evaluating the expected lifetime of the components operating under strong transient conditions. In recent times several dynamic simulations were performed to improve the knowledge and to assess the effectiveness of the various components of solar plants under design and to monitor, predict and control the operation of them [2-4].

Solar thermal power plants undergo lengthy start-up and shut-down operations due to the variation of solar radiation during the day. Therefore, valid modelling of their performance must address those transient conditions in order to accurately model the daily performance of a solar thermal power plant including start-up and shut-down operations and for this reason the various components involved, must be studied. [5-6].

Unfortunately there are limits on how fast a thermal power plant can be started-up. The importance of a quick start-up (producing power some minutes earlier) has a specific value in CSP plants. In general if it is not possible to have a quick start-up of the plant, as the sun rises, the energy will be lost. A faster start-up means therefore higher daily energy production.

A steam generator of the coil type or cross flow evaporator working in transient conditions are interesting for a large range of engineering applications; in particular for CSP applications.

For the design of a cross-flow evaporator working in transient conditions, a dynamic study is of primary importance. This is true both in applications referred to conventional thermal power plants, because in recent times they are subject to frequent load changes in order to be regulated according to the electricity grid requirements and also for systems like CSP applications, in which the steam generation system works in highly transient conditions, both during the aforementioned daily start-up and shut-down phase and due to the possible variation of the input heat source, mainly due to cloud passing, [7]. In this case the flexibility of the evaporator is extremely important in order to be able to operate the power plant with maximum efficiency in the limited daily amount of sun hours. To design and optimize a CSP it is essential to know the performances of the different subsystems and components. Dynamic modelling tools have recently been used to assess the performance of solar systems under transient conditions but obviously each specific component requires a specific tool.

A mathematical model to simulate the operation of the steam generator and of the evaporator has to consider the heat transfer between the liquid-vapour side and on the oil side. Several studies have been developed on the dynamic aspects of two-phase flow on the shell side of staggered and in-line horizontal plain tube bundles, such as the evaluation of void fraction, two-phase flow behaviour, two-phase pressure gradients and heat transfer prediction methods on tube bundles in cross-flow. These prediction methods are generally based on the calibration of an analytical model with experimental results. In general, noticeable discrepancies between the prediction and the experimental results are exhibited [8-9]. But an accurate idea about the differences that can be associated to the use of different modelling approaches is surely a preliminary step. Several mathematical models of key unit operations are developed and available in the literature to study the dynamic behaviour of solar plants to characterize their operations and to improve the reliability of the plant as well as to identify the main critical aspects in operating the plant. But even if the topic is well covered in reduced attention is given to specific elements like the definition of the heat transfer coefficients and the friction factors. A lot of work is generally needed to adapt the available codes for calculation to specific features and specific needs of each project.

The difficulty of getting proper insight into the boiling over tube bundles in industrial heat exchangers is due mainly to the heat transfer model; on the other hand industrial scale experiments are difficult to carry out. [8].

In a recent paper, the authors have analyzed a new type of evaporator in which the coil-type evaporator does not have thick tube plates and in which the hot oil flows are distributed to the heat transfer tube bank via a circular manifold, [6]. According to previous experience on CSP plants the cross flow evaporator represent the critical component and the great advantage of the innovative Coil Steam Generator, is represented by the new design of the oil collectors as cylindrical headers in the cross flow evaporator in order to lower the thermal stresses that seriously affects the lifetime of the component.

The objective of this paper is to define criteria and guidelines for a semi-dynamic fluid and thermo-mechanical model for the cross flow evaporator for CSP applications. The structure of the model has already been discussed in [5-6] but in this paper the aim is to present the impact, of implementing different correlations in the model, on the results of the simulations over a wide range of operating conditions. The main objective of this study is the particular analysis and modelling of the evaporator section. This, as already seen in [6], requires a preliminary analysis concerning the correct selection of the correlation for two phase heat transfer coefficient void fraction and pressure drop in boiling conditions.

This objective is to investigate in detail how the cross flow evaporator of the Coil Steam Generator, specially designed for CSP applications, can be modelled in order to be able to evaluate the thermal performance and the fatigue life and damage in connection with daily start-up routines and what could be the uncertainties connected to the modelling.

The paper is structured in two parts: in the first part a preliminary analysis about steam generators for utilization in CSP in the different load conditions and a discussion of the available correlations that can be used in the model is presented. In the second part a sensitivity analysis connected with the simulation of various different operating conditions of the steam generator will be shown and finally examples of the application of the model in order to predict the life of the evaporator under highly transient operating conditions is presented.

## 2. System Description and general elements for modelling

The specific object of the analysis is a cross flow heat exchanger, included in the two shells of a Steam Generator of the coil type, specially designed for solar applications by the company Aalborg CSP A/S [10] and schematically represented in Fig. 1. From this point it will be referred to as Coil Steam Generator (CSG). The coil steam generator consists of two evaporators in parallel that are cross flow heat exchangers with horizontal plain tube bundle and one separate steam drum. Evaporation occurs on the shell side of the tube bundle. The steam drum is connected to the evaporators with external downcomers and risers. The tube bank has three passes on the oil side which results in a thermally flexible tube bank relative to the shell. These three passes guarantee low temperature difference between oil and water/steam flows, resulting in a small temperature difference across the wall thickness of the outlet header. This means lower thermal stress that is the critical limit for faster start-ups. In the coil-type evaporator the steam drum is connected to the evaporators with external downcomers and risers. The circulation in this type of evaporator is natural driven by the difference in density between the saturated liquid in the downcomer and vapour/liquid mixture in the evaporator and riser. In steam generators for CSP plants, the operating pressure is of the order of 100 bar.

Steam generators for solar application requires start-up and shut-down every day and they can be subject also to frequent and rapid load changes during the day, due to the passage of clouds and variable meteorological conditions. This causes variations in temperature of the heating medium and thermal stress in the tube plates. High material thickness is undesirable with respect to thermal stress and in such a case fatigue cracking is a risk. The coil-type evaporator developed does not have the typical thick plates of shell and tube steam generators. The oil manifolds are cylindrical and have moderate thickness. The cylindrical shape which allows more reduced thicknesses, involves less thermal stress.

With this particular design, the evaporator is less sensitive to fast temperature gradients and therefore is more flexible and is of specific interest for solar applications that require faster start-ups.

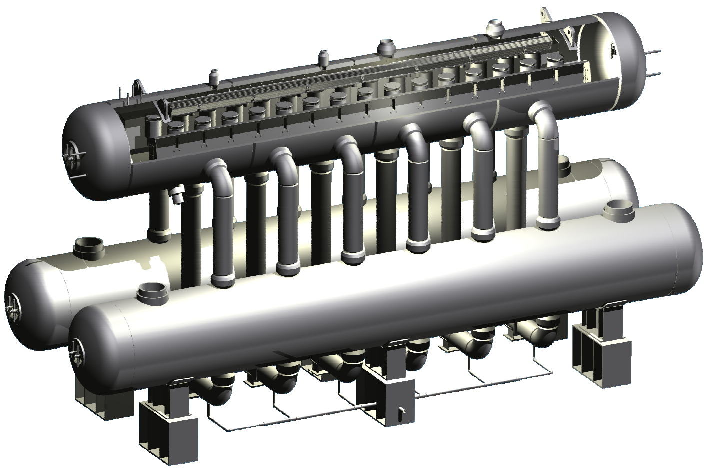

**Fig. 1. Steam generation system of the CSP plant [10]**

Fig. 2 provides a typical schematization of the coil type steam generator in a CSP power plant, operating with oil at temperatures in the range between 300 and 400 °C and steam at pressure between 20 and 100 bar. The steam generator can be considered basically as a steam boiler of water-tube concept, in which high steam capacity and high steam pressure are required. This is due to the frequent start and stop cycles (at minimum one for day) and load changes. Evaporation occurs on the shell side of the tube bundle. The hot oil, at a temperature of 380-400 °C is distributed to the tube banks by means of a circular manifold.

The round shape of the header results in a relatively small thickness; this means quite low thermal stress, because this is proportional to the square of the thickness. For this reason, such a kind of evaporator is less sensitive to fast temperature gradients. Considering the particular application of the concentrated solar power; from [6] it can be understood that the variations of the operating conditions can have significant effects on the local flow conditions, which in turn will affect the local heat transfer coefficient and pressure drop and consequently the overall heat transfer effectiveness. During the design phase the engineer has to face difficult choices on which predictive method are used in order to estimate the overall heat transfer coefficient of the bundle, especially in the case of transient operating conditions. For this reason, it is necessary to consider different load conditions for the evaporator and this means different ranges of temperature/pressure and heat flux at which the evaporator operates.

In order to develop a transient model for simulating the operation of the steam generators (with specific attention to the evaporator), considering the different operating conditions, a relevant number of correlations for calculating void fraction, two-phase pressure drop, heat transfer coefficient in evaporation and condensation and in the single phase section, can be considered.

While many papers give most importance to a system analysis, like the number of dimensions considered, the details of definition domain and the accuracy of the calculation, most of them do not consider in a more accurate way some details concerning the definition of heat transfer coefficients and the pressure drop.

The purpose of this study is to analyze the impact from applying different correlations and to establish a basis for verifying these. Two different fluids are considered in the analysis: water and oil respectively.

The objective of the authors is to analyze and test the different correlations developed in the literature for two-phase flow in the particular steam generator system under analysis, in order to develop a model the prediction of the transient operation of this system under different load conditions, and to furnish criteria and guidelines for the development of an analytical model of steam generators for CSP systems.

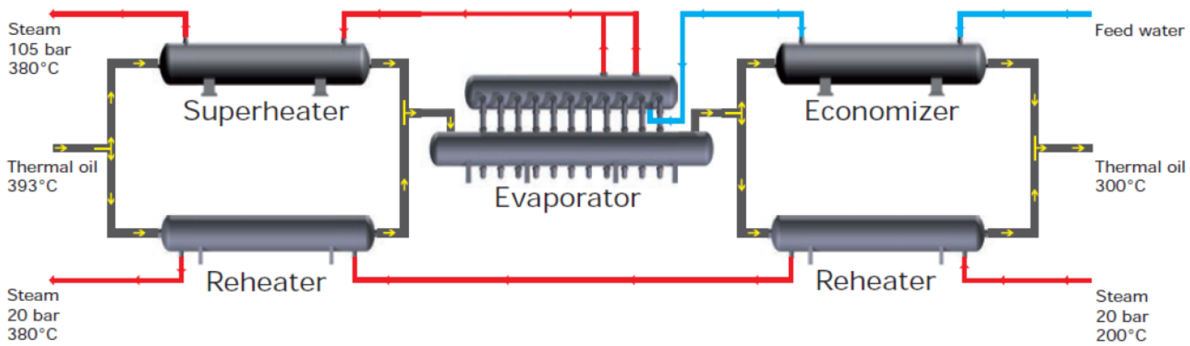


**Fig. 2. Typical operating condition of the particular evaporator**

### 3. Dynamic modelling of the CSG: principles

The design of a solar steam generator is a transient and 3D problem. The temporal nature of the problem is due to the solar heat flux change with time and also at start-up, before the system reaches a steady operating condition. The main objective of this study is to develop a semi-dynamic 3D thermodynamic model for the study of the start-up phase of a cross-flow evaporator operating in conditions typical of CSP operation. The thermodynamic model developed for the 3D cross flow evaporator in steady state conditions is made quasi-dynamic and able to model some variables, like temperature and pressure. These variables are calculated at every time step with the steady state model and the dynamic profile in the start-up time phase has been reconstructed using some interpolating functions. In particular the tubes are modeled by means of a 1D model and the tube bundle by means of a 3D model. The evaporator schematically shown in Fig. 1, represents the reference basis for this study; major details about the tube bundle position in the system are conceptually shown in Fig. 3. The thermodynamic model, based on some classical simplified assumptions typical of heat exchangers lumped capacity model, includes energy balances and the use of heat transfer correlations, necessary to describe the cross flow evaporation process on the evaporator tube bundle. The analysis of the heat exchanger is based on the classic ε-NTU method, available in any textbook about heat exchangers, like [11]. The purpose of the model is to calculate the temperature distributions in the tube bundle of the evaporator, at each point of the 3-D space in the shell and thus determine the temperature in the oil headers. The model is based on the following assumptions: quasi-static conditions, no heat loss, uniform wall thermal resistance and negligible longitudinal heat transfer both in wall and fluid. The model solves an analytical system of partial-ordinary differential equations and uses several MATLAB routines. The complete structure of the model, including the input and output variables is provided in a recent publication [5].

The conceptual scheme of the model of CSG evaporator is shown in Fig. 4. The set of equations giving the maximum heat flux exchanged in the evaporator will be compared with the set giving the lowest value to see the influence on the required area of exchange in the evaporator to guarantee the same heat flux. The influence of several different variables, like temperature T, pressure drop ΔP, heat flux Q, heat transfer coefficient h, steam quality of vapour, x, and flow velocity V, can be analyzed.

The use of the correct correlations has a strong influence on the results of the dynamic model. In the literature there is not yet an agreed heat transfer correlation to be the one to use for describing a boiling phenomenon. Instead there are several that are quite good for specific conditions, mainly correlated to operating temperature, pressures and heat fluxes. It is clear therefore that when dealing with start-up, where the phenomenon of evaporation is highly transient, there is an important problem as to which correlation is used, since the range of pressure, temperature and heat flux vary in time.

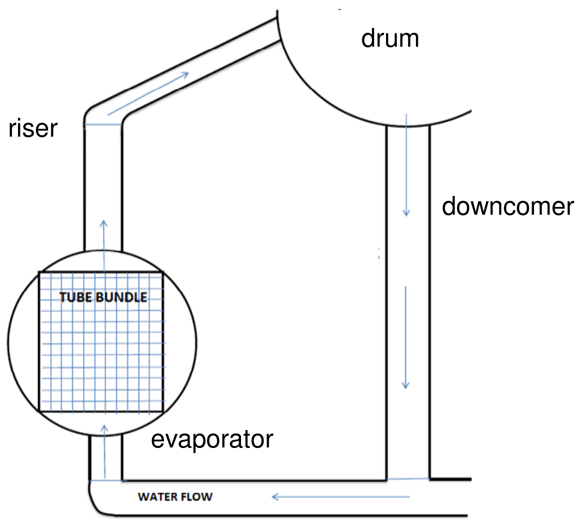


**Fig. 3. Cross sectional volume discretization principle of the entire tube bundle and CSG evaporator circuit.**

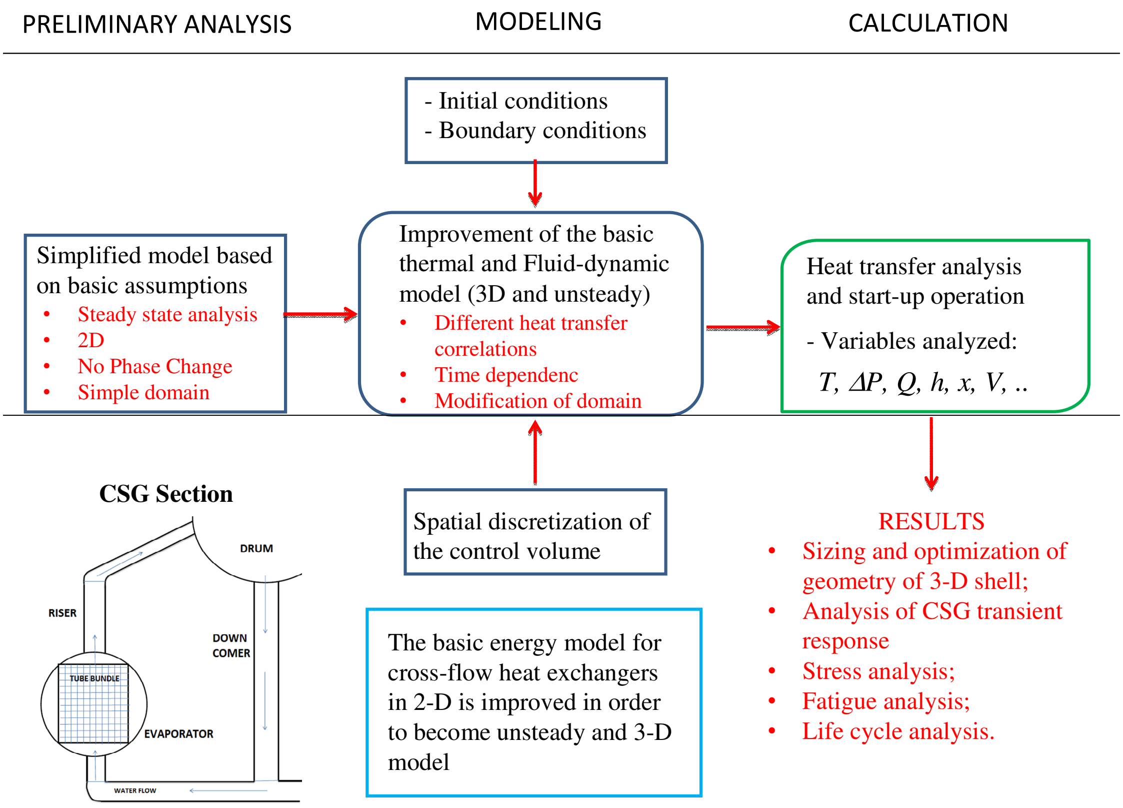


**Fig. 4. Flow chart of the model and solution procedure**

A solution is to use different correlations during the different phases, mainly accordingly to the range of temperature, pressure and heat fluxes involved. In order to do that the dynamic model used needs to be highly detailed and smart enough to be able to select the right implemented correlation for the right conditions.

The experimental data that are used for the calibration of the model are based on heat balances and plant measurements provided by Aalborg CSP. The purpose of this model is to calculate the temperature distributions in the tube bundle of the evaporator, at each point of the shell. When the temperature of the two fluids is known, it is also possible to derive the temperature in some component inside the shell of the evaporator. For instance it is possible to define the inlet and outlet oil headers, since these two components are considered to be critical from the point of view of the thermal-stress. The objective of the model is to be useful for simulating different operating conditions. For this reason it was decided to analyze the suitability for nine different load conditions. The nine different conditions represents all the loads occurring during a start-up procedure for the steam generator from warm conditions until reaching the design load and stationary conditions and these were divided into three main regimes of operation:

- 70-100% of the design load (the load cases are the number 1, 2, 3, 4 and 100% design load): the conditions are approaching the design load at the end of the transient phase of the start-up.
- 50-70% of the design load (the load cases are identified with the numbers 5 and 6).
- 20-50% of the design load (the load cases are the number 7 and 8): conditions describing the first part of the start-up from warm start-up conditions.

Table 1 summarizes with major details the parameters of the nine different operating conditions (loads regimes) that must be considered in the simulation of a CSP plants. The geometry and configuration is the one of the cross flow evaporator in the CSG [10]. Several sets of correlations have been used in the model obtaining six different configurations of the model and the results analyzed and discussed.

**Table 1. Values of pressure, temperature and heat flux in the evaporator at the various operating conditions**

| Load condition | Shell pressure [bar] | Shell temperature [°C] |
|---|---|---|
| Load 0: 100% design load | 105.4 | 314.8 |
| Load 1 | 102 | 312.4 |
| Load 2 | 98.5 | 310 |
| Load 3 | 92.7 | 305.4 |
| Load 4 | 89.8 | 303 |
| Load 5 | 68.5 | 284 |
| Load 6 | 46 | 260 |
| Load 7 | 35 | 243 |
| Load 8 | 34 | 241 |

### 3.1 Correlations for modelling the different section of the steam generator

There are many correlations that can be suitable for defining heat transfer coefficient, void fraction and pressure drop in two phase systems, even if there is no well-recognized correlations that can be applied as an agreed design tool for cross flow evaporator with tube bundle, as stated in [12] and for this reason a careful analysis of the various parts in necessary. The two-phase pressure drop is an important element in the design of tube bundle evaporation due to the pumping power requirement and the close temperature approaches in certain large evaporators where the change in saturation temperature could have a significant effect on heat transfer. The calculation of the two-phase pressure drop in a vertical flow requires the preliminary definition of the void fraction; therefore the definition of the pressure drop is sensitive to modelling assumptions related to void fraction. Tables 2 shows the most relevant available correlations that can be used for calculating void fraction and two-phase pressure drop. Considering the void fraction, the most widely recognized correlation for the prediction is the one defined by Feenstra et al. [15].

**Table 2. Authors of correlations for definition of void fraction and pressure drop in two phase flow**

| Void fraction | Two phase pressure drop |
|---|---|
| Ishihara et al., 1980 [13] | Ishihara et al., 1980 [13] |
| Cornwell et al., 1980 [14] | Xu et al., 1998 [18] |
| Fair and Klip, 1998 [15] | Consolini et al., 2006 [19] |
| Shrage et al., 1988 [16] | |
| Feenstra et al., 2000 [17] | |

For the definition of the pressure drop the most recent and widely used is the correlation proposed by Consolini et al. in [19]. However, it is possible to affirm that no particularly meaningful differences can be connected to the use of different correlations. An important element connected to the development of a model for the analysis of a steam generator of the coil type is surely the definition of the heat transfer coefficient. This has been extensively studied and correlations have been proposed to predict both

convection and pool boiling heat transfer coefficient. Commonly-used nucleate boiling correlations fall into two categories: correlations based on reduced pressure and correlations based on the thermo-physical properties of the fluids [20-28]. Concerning the correlations for the water/steam two phase flow over the tube bundle, pool boiling heat transfer coefficient and mean tube bundle coefficient in evaporation section, a relevant number of correlations can be selected from the literature, as reported in Table 3.
Regarding the definition of the heat transfer over the tube bundle ($h_{bundle}$), both asymptotic and linear structures can be implemented. All the analyses propose the use of a combination of a nucleate boiling heat transfer coefficient ($h_{ncb}$), defined, as discussed below with a specific correlation, and a pure convective heat transfer coefficient, ($h_c$), defined with conventional correlations.

$$h_{bundle} = f\left(h_{ncb}, h_c\right) \tag{1}$$

The different structure for the definition of the tube bundle heat transfer coefficients are discussed in some papers, [29-33]. Palen and Yang in [29] proposed for example a form of the first order, like the one shown below, with $F_b$ and $F_c$ coefficients:

$$h_{bundle} = F_b \cdot F_c \cdot h_{ncb} + h_c \tag{2}$$

Thome and Robinson in [32] proposed a completely different structure:

$$h_{bundle} = \sqrt{h_{ncb}^2 + h_c^2} \tag{3}$$

Shah in [33], also proposed a generalized correlation for covering different conditions of boiling in the tube bundle using several fluids, based on different boiling regimes for saturated boiling conditions. In particular this author studied and analyzed test data from many sources. As a result, three regimes of heat transfer were identified and separate equations were developed for heat transfer in each regime.
A specific summary of the most useful correlations developed in the literature in the last sixty-five years for defining two phase heat transfer coefficients, containing also the details of the variables involved is provided in Table 4. In this heat exchanger, as a result of the small temperature differences between the two operating fluids, the heat flux is very far from the critical heat flux (CHF), so it is possible to assume that the heat transfer regime on the water side will surely be pool boiling.

**Table 3. Correlations for pool boiling and Mean Tube Bundle heat transfer coefficient**

| Pool boiling heat transfer coefficient | heat transfer coefficient correlation for mean tube bundle |
|---|---|
| Rohsenow, 1951 [20] | Palen and Yang, 1983 [29] |
| Foster and Zuber, 1955 [21] | Cornwell et al., 1986 [30] |
| Mostinski, 1963 [22] | Rebrov et al., 1989 [31] |
| Stephan and Adbelsalam, 1980 [23] | Thome and Robinson, 2006, [32] |
| Cooper, 1984 [24] | Shah, 2007, [33] |
| Gorenflo, 1993 [25] | |
| Cornwell and Huston, 1994 [26] | |
| Bhaumik et al., 2004 [27] | |
| Ribatski et al., 2008 [28] | |

**Table 4. Details of the correlations for definition of pool boiling heat transfer coefficient ($h_{ncb}$)**

| Correlation | Details |
|---|---|
| Bhaumik et al. | $h = C_2 \cdot q^{0.7} \cdot P^{0.32}$<br>$C_2$: constant that depends on the heating surface characteristics and boiling liquid (e.g. $C_2$ = 0.477 for distilled water); q: heat flux (W/m$^2$); P: pressure (Pa) |
| Cooper | $h = C \cdot 55 \cdot P_r^{0.12-0.343\ln R_p} \cdot \left(-0.4343 \ln P_r\right)^{-0.55} \cdot M^{-0.5} \cdot q^{0.67}$<br>C: constant (e.g. C=1 for flat surfaces and C=1.7 for cylindrical surfaces)<br>$P_r = \frac{p}{P_{cr}}$ reduced pressure<br>M: molecular weight of fluid; $R_p$: surface roughness in µm (e.g. $R_p$ = 1 µm for unknown surfaces) |
| Cornwell and Huston | $h = \frac{Nu_b \cdot k}{D}$<br>$Nu_b = A \cdot F_p \cdot Re_b^{0.67} \cdot Pr^{0.4}$ $\quad Re_b = \frac{q \cdot D}{\mu \cdot h_{fg}}$ $\quad A = 9.7 \cdot P_{cr}^{0.5}$ |
| Foster and Zuber | $q = 0.00122 \cdot \left[ \frac{k_l^{0.79} \cdot C_{pl}^{0.45} \cdot \rho_l^{0.49}}{\sigma^{0.5} \cdot \mu_l^{0.29} \cdot h_{fg}^{0.24} \cdot \rho_v^{0.24}} \right] \cdot \left(T_w - T_{sat}\right)^{1.24} \cdot \Delta P_{sat}^{0.75}$<br>$k_l$: thermal conductivity; $C_{pl}$: heat capacity; $\rho_l$, $\rho_v$: liquid and vapour density; $T_w$ and $T_{sat}$: wall and fluid saturation temperature; σ: surface tension; $h_{fg}$: latent heat of vaporization; $\mu_l$: liquid viscosity; $\Delta P_{sat}$: variation of saturation pressure connected to temperature difference. |
| Gorenflo | $h = h_0 \cdot F_{pf} \cdot \left(\frac{q}{q_0}\right)^{nf} \cdot \left(\frac{R_p}{R_{p0}}\right)^{0.133}$<br>$nf = 0.9 - 0.3 \cdot P_r^{0.15}$<br>$F_{pf} = 1.73 \cdot \left(P_r^{0.27}\right) + \left(6.1 + \frac{0.68}{1 - P_r}\right) \cdot P_r^2$<br>Reference values: $h_0 = 5600 \ W/m^2K$ $\quad q_0 = 20000 \ W/m^2$ $\quad R_{p0} = 0.4 \ \mu m$ |
| Mostinski | $h = 0.00417 \cdot Fp \cdot q^{0.7} \cdot P_{cr}^{0.69}$<br>$Fp = 1.8 \cdot P_r^{0.17} + 4P_r^{1.2} + 10P_r^{10}$ |
| Ribatski et al. | $h = B \cdot \left(q^{0.9-0.3P_r^{0.2}} \cdot P_r^{0.45} \cdot \left(-\log_{10}(P_r)\right)^{-0.8} \cdot R_p^{0.2} \cdot M^{-0.5}\right)$<br>M: the molecular weight for the fluid expressed in g/mol; $R_p$: surface roughness in µm. |
| Rohsenow | $h = \frac{1}{C_{sf}} \cdot \frac{C_{p,l}}{\left(h_{fg}\right)^{0.67}} \cdot \left[ \frac{1}{\mu_l} \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \right]^{-0.33} \cdot (q)^{0.67} \cdot \left(Pr_l\right)^{-1.7}$<br>$C_{sf}$: a constant; g: specific gravity |
| Stephan and Abdelsalam | $\frac{h \cdot D_b}{k_f} = 207 \cdot \left(\frac{q \cdot D_b}{k_l \cdot T_{sat}}\right)^{0.745} \cdot \left(\frac{\rho_v}{\rho_l}\right)^{0.581} \cdot Pr_l^{0.533}$<br>$D_b = 0.146 \cdot \beta \cdot \left[2\sigma / g \cdot (\rho_l - \rho_v)\right]^{0.5}$<br>β: contact angle (35 °); Pr: Prandtl number |

All the correlations shown in Table 4 have been largely used in the literature for modelling heat transfer in the boiling regime and in principle can all be suitable for use in a two-phase heat transfer model of a steam generator even if some structural difference can be seen amongst them.

Regarding the heat transfer coefficient on the oil side, in general two correlations can be used: the Dittus-Boelter correlation, [34] and the Gnielinski one [35].

Joining the various available correlations, for the two different fluid side (oil and water/steam) and for the two main components, six different version of the model can be obtained as summarized in Table 5.

**Table 5. Combination of correlations for calculating the heat transfer coefficients in six different cases**

| Case | Oil side | Mean tube bundle | Evaporator shell side |
|---|---|---|---|
| 1 | Dittus-Boelter, [34] | Thome and Robinson, [32] | Cooper, [24] |
| 2 | Dittus-Boelter, [34] | Palen and Yang, [29] | Stephan and Abdelsalam, [23] |
| 3 | Dittus-Boelter, [34] | Thome and Robinson, [32] | Shah, [33] |
| 4 | Gnielinski, [35] | Palen and Yang, [29] | Cooper, [24] |
| 5 | Gnielinski, [35] | Thome and Robinson, [32] | Gorenflo, [25] |
| 6 | Gnielinski, [35] | Thome and Robinson, [32] | Ribatski et al., [28] |

## 4. Use of the model for simulating the different operating conditions: analysis and results

A preliminary test analysis has been carried out in order to understand which could the best set of correlations that can be used for simulating the CSG for the different load conditions.

In this section the influence of using different predictive methods on the results of the simulating model is presented and explained. In particular, considering different combination of methods for predicting heat transfer coefficients, six different combinations have been obtained: the results of the simulations are summarized in Figures 5 to 8. In particular the six cases discussed require the use of different sets of correlations for oil heat transfer coefficient, nucleate boiling heat transfer coefficient and mean tube bundle coefficient of evaporation. The combination of the various correlations for the definition of the heat transfer coefficient is shown in Table 4: in this way six different structures of the model can be examined and tested. Regarding the correlation for defining the two phase flow frictional pressure drop, both the Ishihara et al., [13] and the Consolini et al., [19] ones give similar results.

The same conclusions cannot be said for the heat transfer coefficient. In this case the model appears to be really sensitive to the different correlations used. The models including the six different combinations of heat transfer coefficients have been tested at the different load conditions. At first the analysis was relative to the thermal power and has been used to assess the behaviour of the existing surface area in case of relevant load changes. Finally, the results were used to estimate the lifetime of the most critical component of the evaporator: the oil header. The results of the various analyses are shown in the next figures.

Fig. 5 shows the oil flow temperature at the outlet evaporator oil header for the nine loads considered and the six cases studied. As can be seen the three cases using the Dittus-Boelter correlation for the oil heat transfer coefficient give higher values of the oil temperature than the three cases using the Gnielinski correlation. Anyway the differences in the temperature are always below 10 K, and this could be acceptable.

Fig. 6 represents the mean tube bundle heat transfer coefficient on shell side for all the load conditions. As it can be observed there are significant differences between the six cases as well as between the nine loads. At higher loads the evaporator shows a higher mean tube bundle heat transfer coefficient and among the cases it is always case 6 that gives the highest results, while case 2 gives the lowest. For the medium loads the situation changes and for low loads case 6 gives lower results and case 3 gives the highest. The analysis of the data provided in Fig. 6 exhibits remarkable differences among the values of the heat transfer coefficients using the various correlations tested.

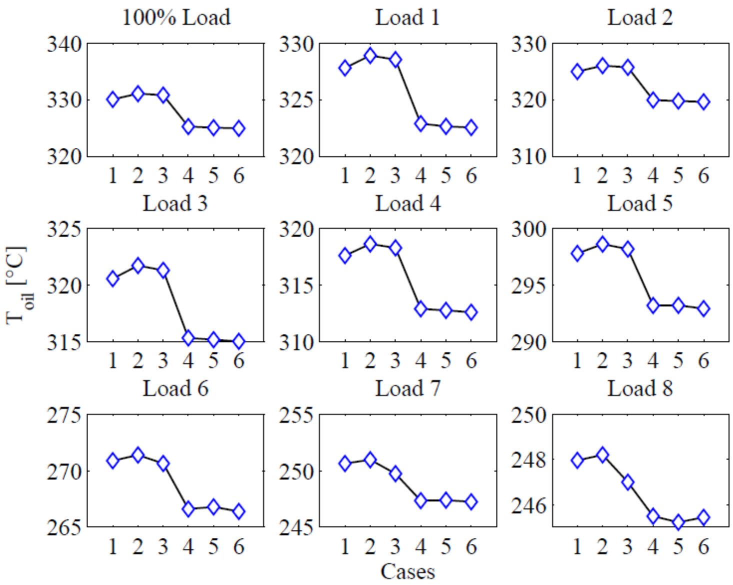


**Fig. 5. Oil temperature at the outlet oil header for all the loads and the six compared cases**

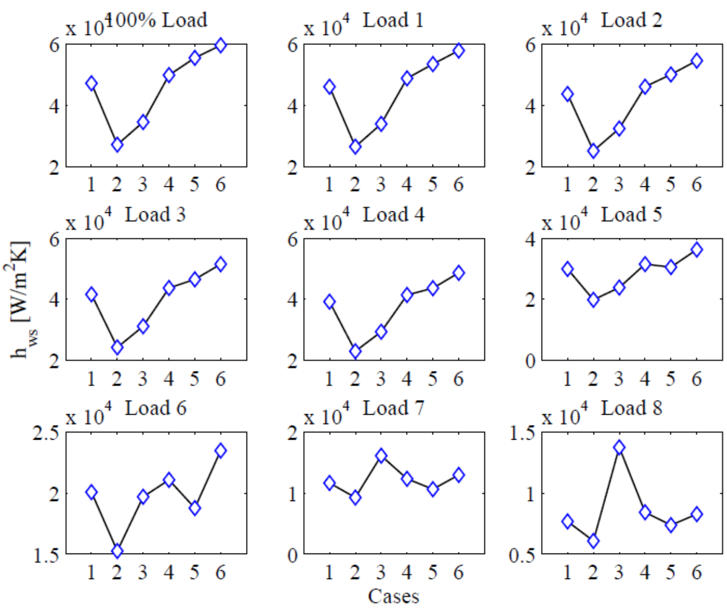


**Fig. 6. Mean tube bundle heat transfer coefficient on shell side (water) for all the loads and the six compared cases**

Fig. 7 shows the predicted values of the heat transfer coefficient on the oil side of the evaporator. The differences shown are qualitatively the same as those seen in Fig. 5 and can be compared to the different results that can be obtained by means of the Gnielinski or Dittus Boelter combination, but the differences in quantitative terms are striking.

Finally, Fig. 8 compares the total heat transfer rate in the evaporator tube bundle for the different load conditions. The qualitative trend of the data is similar to the one shown in Fig. 7. Meaningful differences in terms of the total heat transfer rate can be clearly seen, showing the importance of a correct selection of the correlation for defining the heat transfer coefficient both on the oil side and on the water side. If the data concerning the heat transfer coefficient can be really influenced by the use of the various correlations, the same considerations are not evident for the pressure drop. In particular the oil frictional pressure drop from the inlet to the outlet oil header, relevant to determining the required pumping power shows little variation between the cases for all the loads, with a maximum difference at high loads of about 1%. The calculated two phase frictional pressure drop across the tube bundle of the evaporator, using either the Ishihara et al., [13] or the Consolini et al., [19], correlations, shows relatively small differences between the results obtained (about 10-20 Pa), and therefore either the first or the last can be used, giving the same results for the aim of this study with no relevant impact on the result.

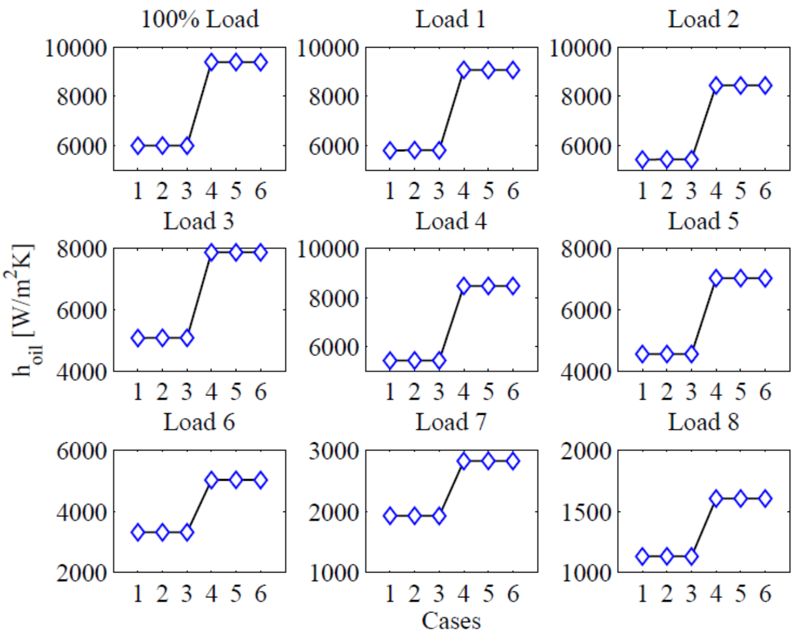


**Fig. 7. Oil heat transfer coefficient in the tube bundle tubes for all the loads and the six compared cases for the various operating conditions**

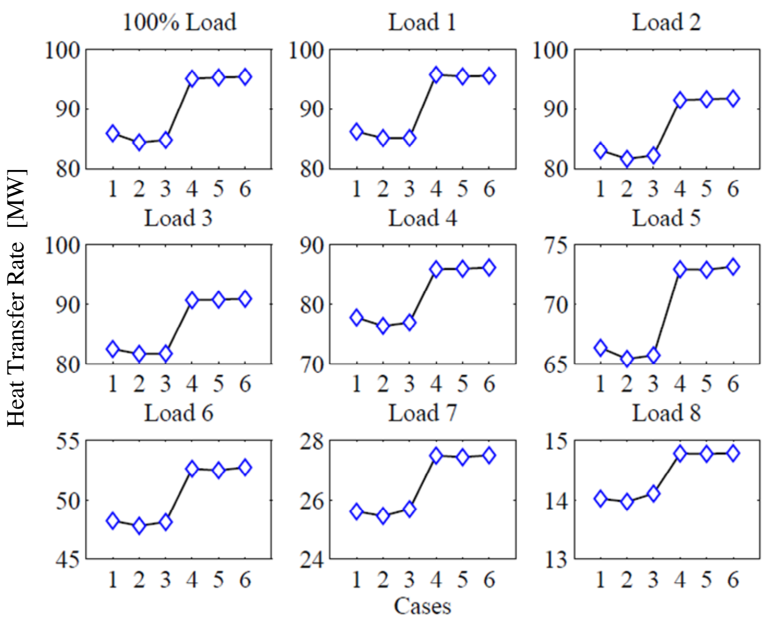


**Fig. 8. Heat transfer rate in the evaporator for all the loads and for the six compared cases**

Considering the results of the analysis presented above, it can be concluded that the simulations at lower loads requires the use of a model basically different with respect to the model that can be useful to simulate high loads conditions. This is because different heat transfer correlations are best suited for certain load conditions where the right operating ranges of pressure (p), temperature (T) and heat flux (q) for that specific correlation are present, whilst other correlations appears to be less adequate. Table 6 summarizes the suitable ranges of pressure, temperature and heat flux for the use of the different heat transfer correlations that can be used in the model. In Table 6, the various load conditions can be grouped according to the different value of the heat flux in the evaporator and the typical operating pressures: in particular it can be concluded that low load conditions can be at level until 40 kW/m$^2$, medium load from 40 to 60 kW/m$^2$, and high load over 60 kW/m$^2$.

**Table 6. Ranges used for deducting the correlations**

| Correlation | Acronym | Pressure | Heat Flux | Fluid |
|---|---|---|---|---|
| Stephan and Abdelsalam, [23] | SA | $10^{-4}\ p_{cr} \leq p \leq 0.9\ p_{cr}$ | No limitations | General |
| Cooper, [24] | C | $P \leq 0.9\ p_{cr}$ | $q \leq 200$ kW/m$^2$ | General |
| Gorenflo, [25] | G | $P \leq 0.9\ p_{cr}$ | $q \leq 200$ kW/m$^2$ | General |
| Cornwell and Huston, [26] | CH | No restrictions | $2.2 \leq q \leq 62$ kW/m$^2$ | General |
| Bhaumik, Agarwal, Gupta, [27] | BAG | No restrictions | $20 \leq q \leq 60$ kW/m$^2$ | General |
| Ribatski, [28] | R | $P \leq 0.65\ p_{cr}$ | $q \leq 40$ kW/m$^2$ | General |
| Mostinski, [22] | M | $p \leq p_{cr}$ | $q \leq 200$ kW/m$^2$ | General |
| Shah, [33] | S | $0.005\ p_{cr} \leq p \leq 0.189\ p_{cr}$ | $1 \leq q \leq 1000$ kW/m$^2$ | General |

Considering this possible criterion, the correlations that should better describe the boiling phenomenon for the evaporator operating at high loads are the one referred by Cooper (C) and those of Gorenflo (G) and Shah (S). At quite low loads other correlations can be used, as reported in Table 6, for example the Ribatski (R) one.

Both Fig. 9 and Fig. 10 show a suitable combination of heat transfer correlation for different operation ranges. In particular Fig. 9 provides a comparison between the heat transfer coefficients calculated with different heat transfer correlations for nucleate boiling. As it can be seen the shape of the curves follows a similar trend for different correlations but the value changes for each correlation: remarkable differences can be observed for the various load conditions in connection with the use of the different correlations.

Fig. 10 shows how a combination of different correlations for defining heat transfer coefficients used in the different operating ranges can produce better results if compared to the single heat transfer coefficient in the whole load range. It shows that there is always a difference in using a single correlation compared to dividing the whole load range into separate loads regimes and being able to follow the load with the most suitable correlation. In Fig. 10 the correlations that are capable of predicting accurately in the load range considered (according to Table 5), or correlations that can follow all the range (Cooper and Gorenflo), have been applied. As it can be seen, even if there are differences, the correlations used show, in general, good agreement with the results. In general, while the Gorenflo equations seems to be qualitatively good for the whole range of operation, the combined use of the three different correlations of Ribatski et al. [28] of Shah, [33] and of Copper [24] seems to be the optimal solution for the simulation of the three different load conditions (low, medium and high). The authors would like to stress the fact that different correlation for heat transfer coefficients for two-phase flow and evaporation, have significant limitations in their use in a large range of temperature, pressure and heat flux at which the evaporation occurs. Therefore it is clear that when dynamic conditions, that involve significant variation of temperature, pressure and heat flux, a realistic model requires the use of different correlations for different operating conditions.

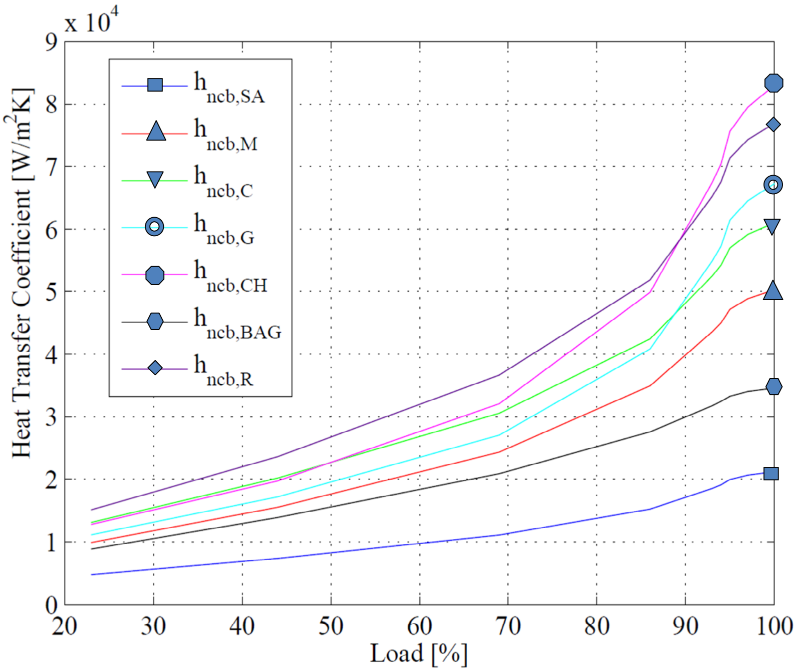


**Fig 9. Nucleate boiling heat transfer coefficient for the various heat transfer correlations**

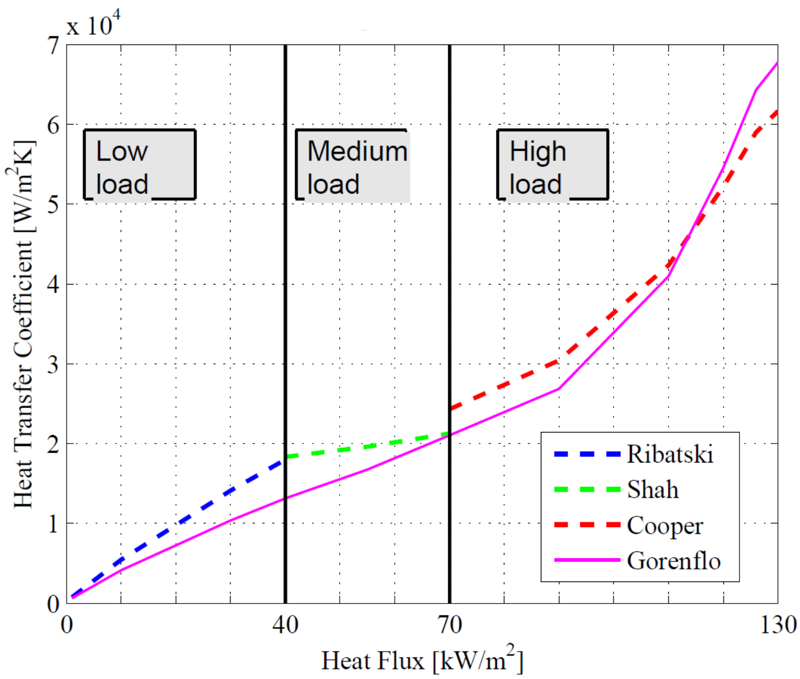


**Fig. 10. Comparison between the use of a set of three correlations or a single correlation for the different loads**

## 5. Comparison among the different tested correlations and discussion

Comparing the analysis of the various results of the previous section, and in particular the results of Fig. 8 containing the results of the total heat transfer rate obtained with the various correlations with Fig. 7 and Fig. 6 it can be seen that in the particular component under analysis, the oil heat transfer correlation has an higher influence with respect to the mean bundle heat transfer. Thus the great difference in the results can be exhibited by using the Dittus-Boelter correlation or the Gnielinski correlation in order to define the oil side heat transfer coefficient. Moreover, based on the results obtained and considering Fig. 10, it can be clearly seen which should be the best correlation to be used in the different operating ranges of the two-phase fluid. Table 7 summarizes which one of the six cases discussed with reference to the combination of correlations analyzed in Table 5 that can be proposed in order to better simulate the different load conditions.

The cases 4, 5 and 6 are the only available cases and the combination described as case 4 of Table 5 seems surely to be the most appropriate in order to analyze the case under analysis.

Considering Fig. 8 that shows the different heat transfer rate in the evaporator for the different cases, it is interesting to investigate how the use of different correlations would affect the definition of the required surface area for the heat transfer in the evaporator in the design phase.

In particular the two cases, the one identified with number 2, that determines the lowest heat amount and the one identified with case number 6, in which the highest amount is obtained, are compared.

A comparison of the two situations is considered; Table 8 summarizes the results, showing the differences between the two cases; in particular the difference between the highest (case 6) and lowest result (case 2) in terms of heat transfer rate is approximately of 13%. This basically means that there is a lower predicted overall heat transfer coefficient, for the tube bundle, in case 2 and for compensating this lower overall heat transfer coefficient an increased surface heat transfer area would be required. It is clear that a validation of the model could be based on experimental analysis, but this would require a lot of time in order to simulate the various load conditions.

**Table 7. Best cases to use for the various load conditions**

| Load | Most suitable predictive case |
|---|---|
| 100 % | CASE 4 |
| Load 1 | CASE 4 |
| Load 2 | CASE 4 |
| Load 3 | CASE 4 |
| Load 4 | CASE 4 |
| Load 5 | CASE 4 |
| Load 6 | CASE 4 |
| Load 7 | CASE 4, 5, 6 |
| Load 8 | CASE 4, 5, 6 |

**Table 8. Comparison of the results obtained with two of the models tested**

| Variable | Value |
|---|---|
| $Q_6$ | $Q_{max}$ |
| $Q_2$ | $Q_{min}$ = 88.5% $Q_{max}$ |
| $Q_6/Q_2$ | ~ 1.13 |
| $A_2/A_6$ | ~ 1.13 |
| $(h_6/h_2)_{oil}$ | ~ 1.57 |
| $(h_6/h_2)_{bundle}$ | ~ 2.27 |

**6. Application and use of the CSG model to predict the evaporator lifetime**

The model developed can be used not only for defining the main design variables of the steam generator and to simulate its operation at the various load conditions, but to obtain other important information too.

Using the model developed, it is possible to conduct a thermo-mechanical analysis of the system. An interesting additional information is, for example, the predicted lifetime of the component.

For example an analysis of the stresses acting on the oil headers of the evaporator has been carried out and from that analysis the fatigue life evaluation has been estimated using the theory from [36-38]. The analysis consists of calculating the thermal stresses, due to temperature gradients acting on the headers, with the equations of Plane Strain and adding linearly the pressure stresses, caused by the pressure gradients between oil pressure and steam pressure, with the modified Lame equations. Calculating the overall state of stress acting on the headers for all the loads during a start-up phase it is possible to reconstruct an alternate fatigue stress cycle for the evaporator during the plant lifetime. The S-N curves (also known as Wohler curves) specific for the oil headers can be used to estimate the number of cycles, before rupture, for the headers. The results are summarized in Fig. 11 for the six cases tested.

The estimation of the lifetime of the evaporator is extremely important because, the replacement of such a components requires a stop of the power plant (no energy is produced) and high additional costs. While in the case of the geometrical design the differences between the various cases are not so relevant, in this case the differences are more remarkable.

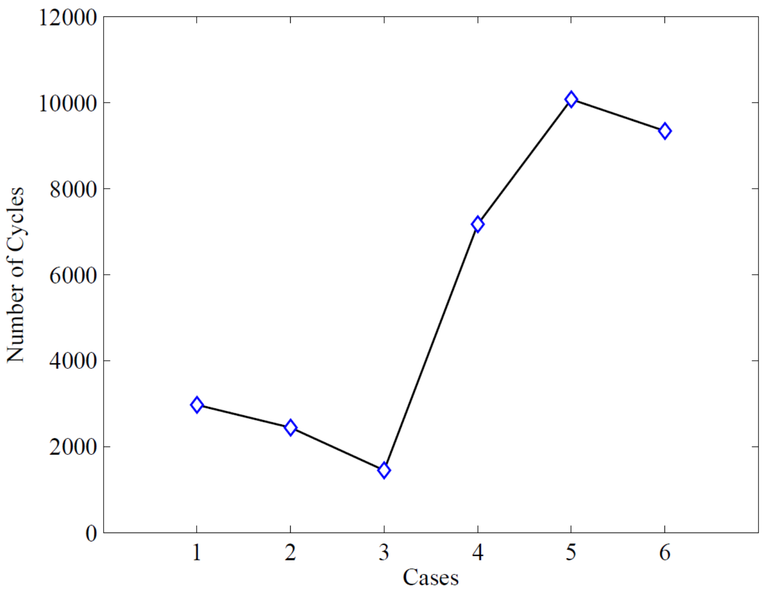


**Fig. 11. Oil header lifetime results for the six compared cases**

The results of the fatigue analysis can be graphically divided into two groups: in the first three cases (1-2-3) in the model, in which the Dittus-Boelter correlation for the heat transfer coefficient the oil side is used, while in the last three cases (4-5-6) the Gnielinski correlation is used. The Dittus-Boelter correlation gives lower heat transfer coefficient inside of the tube and therefore the outlet temperature of the oil flow at the outlet header would be higher when calculated with the one calculated using the Dittus-Boelter correlation.

A higher temperature difference across the thickness of the outlet header results in higher thermal stresses and fatigue damage. So the result is a reduced lifetime of the components in the first three cases compared with the lifetime obtained in the last three cases. Table 9 summarizes the various expected life of the components at the different load regimes. After the fatigue analysis some design changes can be proposed in order to increase the effective lifetime.

**Table 9. Component life for different start-up scenarios**

| Start-up scenario | Expected life (years) |
|---|---|
| Warm start-up (20 minutes) | 13 |
| Warm start-up (28 minutes) | 16 |
| Warm start-up (47 minutes) | 19 |
| Combination of cold and warm start-up with radiation changes | 15 |
| Warm start-up (20 minutes) with oil pressure at 100 bar | Unlimited (> 50) |
| Warm start-up (20 minutes) optimized header thickness and oil pressure at 10 bar | 30 |
| Warm start-up (20 minutes) optimized header thickness and oil pressure at 100 bar | Unlimited (> 50) |
| Warm start-up (20 minutes), optimized header thickness and oil pressure at drum pressure | Unlimited (> 50) |
| Fast start-up (10 minutes) with overall optimization | 25 |

## 7. Conclusions

This study analyzes the problem of modelling a particular type of steam generators, designed for solar application with particular attention to the dynamic behaviour during the start-up phase.

The analysis of the literature shows the uncertainty in the modelling of two-phase phenomenon in transient conditions. Due to the fact that there is an important impact on the model of the evaporator, a combination of different correlations can be used for determining heat transfer coefficient correlations in the different load ranges and this could be mainly suggested if a structural analysis of the components, including the predicted lifetime of the components were to be carried out.

The analysis of using different correlations for defining the two phase heat transfer coefficient, the pressure drop and the void fractions aimed at the simulations of a particular steam generator for application in CSP plants during the transient conditions of a start-up procedure, has given interesting informations about the use of the various correlations. In particular six different combinations of them obtaining six kind of simulation model analyzed and tested referring to nine different operating conditions of the steam generator, corresponding to different load.

Analyzing the results of the various simulations it is possible to conclude that the most critical element is the definition of the correct value of the oil side heat transfer coefficient. The definition of the value of two-phase heat transfer coefficient on the water side appears to be a less critical aspect.

In the case under analysis, the use of the Dittus Boelter correlations, instead of the Gnielinski correlation, gives substantial discrepancies in the value of the fluid temperatures (up to 10 K difference at full load), heat load (up to 13% of difference) and expected lifetime of the evaporator (up to 5 years).

Considering the analysis of the different operating conditions, corresponding to different load, it is also shown that the higher differences among the results occurs at the higher loads; moreover for each load condition there is a most suitable set of correlations that can be used but some of the available correlations, can be used in wide range of operating conditions with a low influence on the ~~a~~ good accuracy of the thermo-fluid dynamic results.

Using the model developed, it is possible to conduct a thermo-mechanical analysis of the system obtaining interesting information on the predicted lifetime of the component connected to fatigue analysis.

Even if the indication from the paper can be considered meaningful from a methodological point of view because the range of variation of the output results connected to the use of different correlation for the definition of the heat transfer coefficients, void fractions and pressure drop, it is necessary to underline the fact that they can be considered only at a preliminary step and a definitive validation could be possible only by comparing the results obtained by the modelling with experimental results obtained analyzing the operation of the Steam Generator. This topic will be object of future investigation by the authors of the present paper.

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