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Critical heat flux prediction for water boiling in vertical tubes of a steam generator

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Abstract

This paper presents a methodology for the prediction of the critical heat flux (CHF) for the boiling of water in vertical tubes operating under typical conditions found in steam generators. At the furnace, the water flows through long vertical tubes under an axially non-uniform heat flux and with relatively low mass fluxes. This fact causes that the recent theories and correlations, which have been developed for conditions typically found in nuclear reactors, cannot be directly applied for the prediction of the CHF in the furnace tubes. In this context, the mechanistic theories focused into the CHF prediction have proved their usefulness to predict CHF avoiding the use of correlations and experimental constants. Hence, in order to assist the CHF problem in steam generators, the sublayer dryout theory, initially formulated for CHF in vertical tubes uniformly heated, is extended by combining it with the shape factor method (F-factor), to account for the effects of the axially non-uniform heat flux distribution. The critical wall temperature (CWT) of the tubes is calculated from CHF data. The reliability of the modified theory for the CHF prediction is tested by comparing CWT results against measured data from a steam generator of a power plant. Good consistency and approximation is found between predicted and measured data.

Keywords: Critical heat flux prediction; Steam generator; Boiling in vertical tubes

1. Introduction

During the boiling of water inside the tubes in steam generator, the critical conditions in the heat transfer, sometimes, provoking instability in the process. The most common instability is the boiling crisis which appears, with undesirable conditions for the system operation, as a combined phenomenon of hydrodynamics and thermodynamics. When the boiling crisis occurs in a system where the heat flux is controlling, its effect is an abrupt temperature rise in the heated surface with the consequent detriment of the material. In those systems where the surface temperature is controlling, the possible consequences are deficiencies in the process due to insufficient heat transfer. It has been experimentally observed that boiling crisis occurs after reaching a high heat flux into the system, which is commonly called *Critical Heat Flux* (CHF) [1].

A CHF condition sets a limit in the design of boiling heat transfer equipment. Therefore, many researchers have studied the phenomenon with the aim to generate information to improve the design methodologies [2–7]. However in the past years, the focus of almost all this scientific activity has been centered to the study of nuclear reactors. Essentially, nuclear reactors work with cooling systems configured by subcooled water flowing through short tubes with small diameters operating under high heat flux uniformly distributed along their axis. In contrast to nuclear reactors, in a furnace, water flows through long vertical tubes exposed to an axially non-uniform heat flux. In addition, the mass flux in steam

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Nomenclature

C_p	specific heat	$J\cdot kg^{-1}\cdot K^{-1}$
D	tube diameter	m
F	shape factor (F-factor)	
f	friction factor	
G	mass flux	$kg\cdot m^{-2}\cdot s^{-1}$
h	heat transfer coefficient	$W\cdot m^{-2}\cdot K^{-1}$
h_{fg}	latent heat of vaporisation	$J\cdot kg^{-1}$
i	enthalpy	$J\cdot kg^{-1}$
L	length	m
n	experimental exponent	
P	pressure	Pa
P_h	heated perimeter	m
Pr	Prandtl number	
q	heat flux	$W\cdot m^{-2}$
Q_n	parameter defined in Eq. (8)	K
q_{CHF}	critical heat flux	$W\cdot m^{-2}$
R	tube radius	m
S	total surface of the tube	m^2
S_{ht}	available surface of the tube for heat transfer	m^2
T	temperature	°C
t	passage time	s
U	velocity	$m\cdot s^{-1}$
U_τ	friction velocity	$m\cdot s^{-1}$
x	steam quality	
y	radial length measured from the tube wall	m
y^*	superheated layer thickness	m
y^+	dimensionless radial length measured from the tube wall	
y^{+R}	dimensionless tube radius	
z	axial position measured from the tube inlet	m

Greek symbols

Δ	difference of a variable	
δ	initial thickness of the liquid sublayer	m
μ	dynamic viscosity of saturated liquid	$kg\cdot m^{-1}\cdot s^{-1}$
Ω	factor defined in Eq. (18)	
Φ	ratio between S_{ht} and S	
Ψ	experimental coefficient	
ρ	density	$kg\cdot m^{-3}$

Subscripts

B	vapor blanket
b	average property of the subcooled flow core
crit	boiling crisis
f	property of saturated liquid
in	property evaluated at the inlet of the tube
i	any axial position inside of the subcooled flow boiling region
m	average property of the flow
sat	saturation state
sl	average property of the superheated layer
w	wall of the tube

generators is lower than in nuclear reactors because of the big diameter of their tubes.

The CHF prediction in steam generators could be useful to know the real causes of the failures presented, like the burnout of tubes or leaks that appear as consequence of an accelerated process of corrosion caused by the high temperature reached in material of the tubes. The boiling of water inside the tubes of a steam generator could propitiate two kinds of critical heat flux, one of them at high quality region and another one at low quality region. At high qualities the CHF causes a gradual increase in the wall temperature of the tubes and its consequences are not so harmful. In fact, the CHF at high qualities represents the transition from boiling of water into superheated steam. The CHF at low qualities involves a suddenly increase in the temperature of the tube wall that could burn the material causing leaks and other damages.

2. The CHF problem inside the furnace of a steam generator

The need to analyze the CHF under conditions of the furnace of a steam generator arises from studies published in [8], where failures in tubes within furnaces in mexican steam power plants have been reported. The results of these studies showed that tubes of some steam generators were exposed to high temperatures over the permissible value for its material burning the tubes until the internal pressure could not be withstood. This fact caused leaking of water in many tubes. Since the leaks were located in tubes at the furnace level, where the steam quality is around 18% [9], a deviation from nucleate boiling to film boling was expected, i.e., it is possible that the heat absorption rate in the furnace reached the CHF value. It means that the heat transfer coefficient inside the tubes diminished and the temperature of the wall increased causing the burning of the metal. On the other hand, it is possible that the permissible temperature surpassed by a few degrees without reaching the values for the appeareance of CHF, in this case, an accelerated corrosion is activated. The acelerated corrosion provokes damages more slowly than those caused by CHF. On this basis, any analysis of the problem of water boiling in tubes must consider both cases.

In order to analyze this problem a review of the recent CHF prediction techniques was made. It was found that mechanistic theories have had a great acceptance due to the good approximation achieved compared to experimental CHF data and because the experimental constants and correlations are avoided in the mathematical model formulated, so there are no a limiting range of validity [2,4,5]. Among the mechanistic theories that were studied, the one known as *sublayer dryout theory*, initially postulated by Lee and Mudawar [2], was considered useful for the CHF prediction under nucleate boiling. But this theory is formulated for vertical tubes uniformly heated and it has proved for the thermo-hydraulic conditions that prevail in a nuclear reactors. However, making some modifications, this model can be used to make a mechanistic analysis of the CHF for the boiling of water in vertical tubes within a furnace. For this purpose, this study considers the sublayer dryout theory combining with the shape factor method (F-factor), in order to take account of the effects of the axial non-uniform heat flux distribution. Also, to prevent failures in the tubes before reaching the CHF condition, a mathematical model to calculate the temperature distribution along the furnace tubes was developed. In this way, the calculated temperature in the tubes can be checked against the permissible value to ensure the feasible furnace operation.

Also, the *Critical Wall Temperature* (CWT) of the tubes is calculated from the CHF predictions. CWT data are useful to test the reliability of the modified theory for the CHF prediction by comparing them against temperature measurements reported in Ref. [10]. These temperature measurements were carried out in the furnace of a steam generator of a power plant where a CHF problem was presumable.

For the CHF results, good approximation between theoretical and measured data was obtained. And using actual design information of the steam generator analyzed, the model to determine the temperature distribution was validated.

3. Analysis

3.1. Reference conditions

The furnace tubes are assumed to be axially non-uniformly heated and they are installed in the furnace as shown in Fig. 1. It can be observed that a furnace tube is partially insulated and only one half of its surface is available for heat transfer. This means that the heat flux distribution is non-uniform axial and circumferentially. The axial heat flux distribution can be expressed as a function of the position measured from the inlet of the tube, e.g., $q = q(z)$. In turn, the circumferential heat flux distribution is considered into the energy balances defining the ratio between the surface available for heat transfer and the total surface of the tube, this is written as:

$$\Phi = \frac{S_{\mathrm{ht}}}{S} \tag{1}$$

evidently, in this case, Φ takes a value of 0.5.

At the inlet of the tube slightly subcooled liquid conditions are assumed. This assumption is in order to eliminate the effects of a large subcooled flow boiling region, since it does not represent an important stage in the steam generation process.

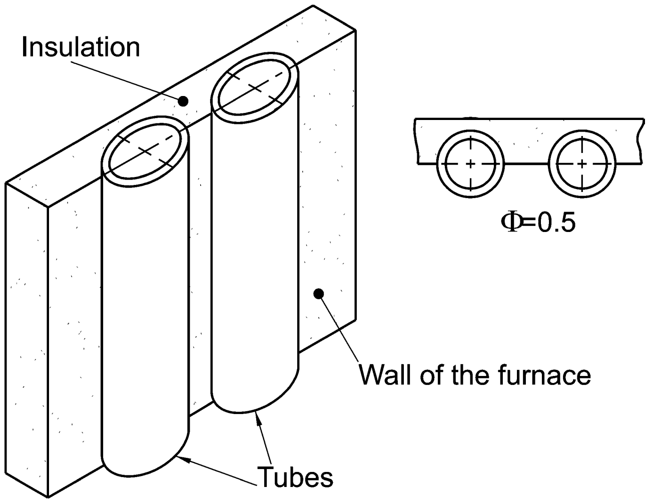

Fig. 1. Installation of the furnace tubes.

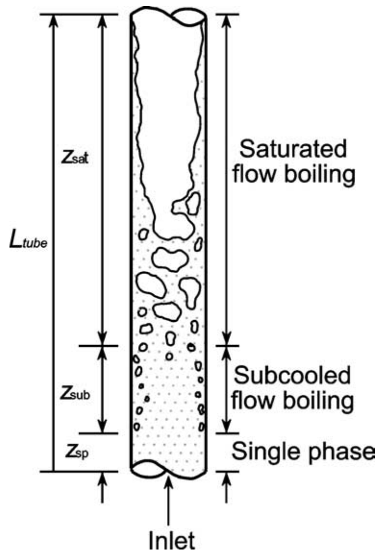

Fig. 2. Water flow boiling patterns.

In saturated boiling the flow pattern is defined by the regime of boiling. The first regime is the nucleate boiling which prevails up to a steam quality around 40%. In nucleate boiling, generally, the flow pattern varies from bubbly to slug, as shown in Fig. 2. The next regime is the convective boiling together with the annular flow pattern. The CHF has different mechanisms according to the boiling regimes. The CHF associated with nucleate boiling, generally occurs in the low steam quality region. This critical point appears at high heat flux so intense nucleate boiling is favored, therefore, it is expected that the critical condition be similar to those that occur in pool boiling, i.e., the bubbles crowd about the heated wall impeding the coolant access. The bubbles crowd form vapor blankets which move on the wall along the tube due to drag and buoyancy forces. Between the blanket and the tube wall prevails a thin liquid layer which provides cooling to the portion of the wall underneath. When the CHF is reached the liquid layer is evaporated and the vapor blanket remains in contact with the tube wall, then film boiling starts. The CHF associated with convective boiling ocurrs at the anular flow (see Fig. 2), the expected mechanism is the dryout of the liquid film due to evaporation [3].

3.2. Temperature distribution

The temperature distribution at the wall of a furnace tube is evaluated depending on the boiling region. Because of the inlet flow is slightly subcooled it is necessary to determine the position along the tube where the saturated condition is achieved. It is known that the saturation state begins when the steam quality is equal to zero, so this is the physical boundary between the saturated and subcooled flow boiling regions. The steam quality is calculated from an energy balance considering the enthalpy increase due to the axial non-uniform heat flux and the Φ factor defined above. Then steam quality can be expressed by:

$$x = \Phi \frac{4}{GDh_{\mathrm{fg}}} \int_0^z q(z)\,\mathrm{d}z + \frac{\Delta i_{\mathrm{in}}}{h_{\mathrm{fg}}} \tag{2}$$

where Δi_{in} is the inlet subcooling, and it is defined as the difference between the saturated liquid enthalpy and the inlet enthalpy of the flow. Now, from Eq. (2) and for $x = 0$, the length, z_{sat}, where the saturation regime is reached can be obtained from:

$$\int_0^{z_{\mathrm{sat}}} q(z)\,\mathrm{d}z = -\frac{\Delta i_{\mathrm{in}} GD}{2} \tag{3}$$

On this basis, the temperature distribution along the tube length, before z_{sat}, is calculated by using the Martinelli's temperature profile for turbulent flow in a tube [11]

$$T_{\mathrm{w}} - T(y^+) = Q_n Pr\, y^+, \quad 0 \leqslant y^+ < 5 \tag{4}$$

$$T_{\mathrm{w}} - T(y^+) = 5Q_n \left\{ Pr + \ln\left[1 + Pr\left(\frac{y^+}{5} - 1\right)\right]\right\} \quad 5 \leqslant y^+ < 30 \tag{5}$$

$$T_{\mathrm{w}} - T(y^+) = 5Q_n \left[Pr + \ln(1 + 5Pr) + 0.5\ln\left(\frac{y^+}{30}\right)\right] \quad y^+ \geqslant 30 \tag{6}$$

where T_{w} is the wall temperature, Pr is the liquid Prandtl number, $T(y^+)$ is the radial temperature profile, y^+ is the dimensionless value of radial distance, y, measured from the inner wall of the tube, it is defined by:

$$y^+ = y \frac{U_\tau}{\mu_{\mathrm{f}}} \rho_{\mathrm{f}} \tag{7}$$

Q_n is group defined as a function of the local heat flux,

$$Q_n = \frac{\Phi \int_0^{z_{\mathrm{i}}} q(z)\,\mathrm{d}z}{\rho_{\mathrm{f}} C_{\mathrm{pf}} U_\tau} \tag{8}$$

where z_{i} is the axial distance measured from the inlet of the tube to any point in the subcooled flow boiling region.

The term U_τ in Eqs. (7) and (8) is known as friction velocity, according to Ref. [12], it is expressed by:

$$U_\tau = \frac{G}{\rho_{\mathrm{f}}} \sqrt{\frac{f}{8}} \tag{9}$$

The wall temperature is calculated from the Martinelli's equations at any point of the tube wall. The exact position is determined by the average fluid temperature, T_{m}, which depends on the axial position, z, along the tube and it is calculated from an energy balance,

$$T_{\mathrm{m}} = \Phi \frac{4}{GC_{\mathrm{pf}}D} \int_0^{z_i} q(z)\,\mathrm{d}z + T_{\mathrm{in}} \tag{10}$$

but, from Martinelli's equations, T_{m} also can be calculated by integration of the radial temperature profile $T(y^+)$, therefore,

$$T_{\mathrm{m}} = \frac{1}{y^{+R}} \left(\int_0^5 T(y^+)\,\mathrm{d}y^+ + \int_5^{30} T(y^+)\,\mathrm{d}y^+ + \int_{30}^{y^{+R}} T(y^+)\,\mathrm{d}y^+ \right) \tag{11}$$

where y^{+R} is the dimensionless tube radius, it is obtained evaluating Eq. (7) in $y = R$:

$$y^R = y^+|_{y=R} \tag{12}$$

Combining Eqs. (10) and (11), the only unknown is the wall temperature, T_{w}, at the position where T_{m} is evaluated; the resultant equation can be readily solved.

For the saturated boiling region, the fluid temperature is the saturation temperature corresponding to the system pressure, which can be considered constant, therefore, the wall temperature depends only on the local heat flux and is calculated from the necessary superheat to maintain the boiling at this regime. This is given by:

$$\Delta T_{\mathrm{sat}} = T_{\mathrm{w}} - T_{\mathrm{sat}} = \psi q(z)^n \tag{13}$$

where $\psi = 25\mathrm{e}^{-P/62}$ and $n = 0.25$ [13].

3.3. CHF prediction

The CHF prediction is achieved by means of the *modified sublayer dryout theory*. Initially, Lee and Mudawar [2] proposed the sublayer dryout theory as a mechanistic form to predict the CHF under conditions of subcooled water boiling at high heat and mass fluxes, however their model can be combined with the shape factor method. According to the experimental observations reported by Lee and Mudawar, during the subcooled boiling process a vapor blanket is formed close to the tube wall due to the coalescence of small bubbles. Between the tube surface and vapor blankets, a thin

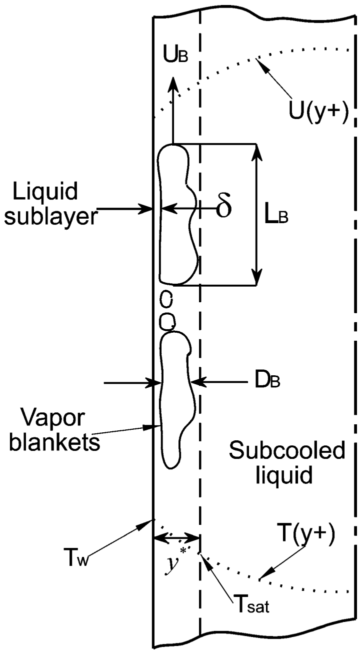

Fig. 3. Sublayer dryout model

liquid layer prevails providing cooling to the tube wall. The CHF is reached when the liquid layer is evaporated by the high heat flux. In Fig. 3 a descriptive scheme is observed, where the parameters involved in the sublayer dryout theory are shown. This parameters are the initial thickness of the layer underneath the blanket, δ, the equivalent diameter of the vapor blanket, D_{B} and its length, L_{B}. It is assumed that the vapor blanket moves inside the superheated layer with a velocity U_{B}. The superheated layer is limited by the temperature of the wall and the position where the radial profile of fluid temperature reaches the saturation value. The distance between this boundaries is denoted as y^* and it is referred as thickness of the superheated layer.

The initial thickness of the liquid layer, δ, is determined from the difference between y^* and D_{B}, since a radial temperature profile exists in the subcooled fluid, the bubbles only subsist in those regions where the temperature is equal or greater than the saturation value, and considering the radial profile of the fluid velocity, the bubbles are dragged into the fluid core, then, the bubbles or vapor blankets that subsist, prevail in the boundary of the superheated layer. The relationship between the CHF and the above mentioned parameters is expressed as:

$$q_{\mathrm{CHF}} = \frac{\rho_{\mathrm{f}}\delta h_{\mathrm{fg}}}{t} \tag{14}$$

where, t, is the passage time, defined as the required time to evaporate the liquid layer of thickness δ,

$$t = \frac{L_{\mathrm{B}}}{U_{\mathrm{B}}} \tag{15}$$

The rest of the parameters to evaluate in Eq. (14) can be calculated from equations proposed by the original authors of the theory or by those proposed by Katto [4], who made an extension of this theory with the aim to apply it to a wide range of pressure. However, recently, Celata et al. [5], improved on the mechanistic concept by developing a mathematical model based on the sublayer dryout theory, where they avoid all experimental constants and correlations that first authors had used.

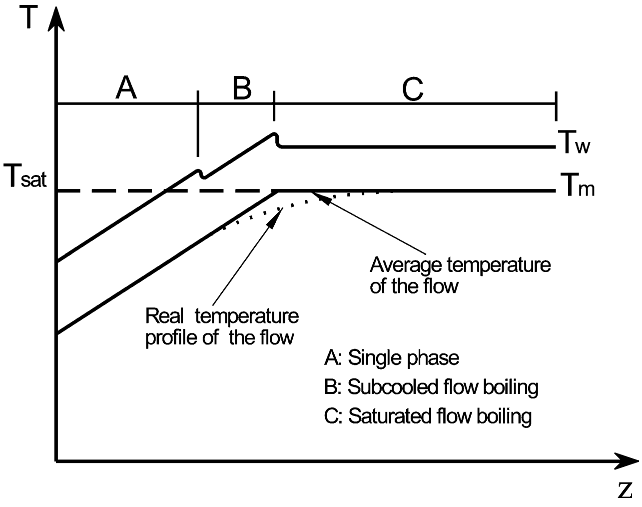

Fig. 4. Wall and flow temperatures in a fluid boiling process [13].

It has been observed that the sublayer dryout theory only can be applied to predict the CHF in the subcooled flow boiling regime, since, while the boiling process develops, the steam quality increases gradually and at any moment the radial temperature profile leaves the subcooling and will varies between the wall and saturation temperatures. In this case, the two-phase flow patterns developed (bubbly, slug, churn), propitiate the mixing between both phases, and the probability of the CHF, increases again until the annular flow pattern develops, where the boiling crisis is carried out by dryout of the annular liquid film.

Thermodynamically, the saturated boiling begins when $x = 0$, however, subcooled fluid still exits even when the steam quality is greater than zero [13]. Fig. 4 shows the typical wall and fluid temperature distribution, where it can be seen the real variation of the average fluid temperature through the transition from subcooled to saturated boiling. It is observed that the average fluid temperature continues under the saturation value after this thermodynamic stage is reached. This means that mechanism of the boiling crisis must be similar to that of subcooled boiling.

In the furnace of a steam generator, the steam quality inside the tubes at the level immediately above of the set of burners is around 15% for its maximum production capacity. This is due to the fact that the main purpose of the furnace tubes is to provide cooling to the furnace wall, therefore, nucleate boiling at low quality flow is preffered in this zone of the steam generator [10]. For this reason, the use of the sublayer dryout theory under saturated flow boiling was considered, provided the low steam quality stays in the water flow. The low steam quality is considered for those values under 30% [14].

Now, a modification to the sublayer dryout theory is required in order to take account of the non-uniform heat flux into the furnace tubes. Tong et al. [6] developed an empirical method to treat the CHF under axial non-uniform heat flux, which is based on an energy balance inside the superheated layer. The superheated layer subsists while the core of the flow remains subcooled. The axial length developed by the superheated layer depends on the magnitude of heat flux and its axial distribution.

The energy balance proposed by Tong is derived in terms of the average conditions across the superheated layer and it is based on the diagram of Fig. 5, where the liquid that flows in the tube core provides cooling to the tube wall by mixing it with the flow inside the superheated layer. This mix reduces the liquid subcooling and, also, it is useful to maintain a bubble population balance close to the wall, since, as a consequence of the incoming subcooling in the superheated layer, some bubbles collapse due to condensation, leaving free space to incipient bubbles. Along the tube, the average enthalpy of the superheated layer increases due to the heating, therefore, the liquid subcooling diminishes and the bubble condensation is lower in the flow direction. In a critical case, when the heat flux be very intense, the rate of formation of bubbles surpass the rate of condensation, so the bubbles crowd close to the tube wall impeding the cooling effect by the liquid. Then, the temperature of the wall increases suddenly and the average enthalpy of the superheated layer increases too. At this point, the bubbles form vapor blankets, which without sufficient cooling, are directly exposed to the tube wall starting the local film boiling.

In according to Tong, the critical case above described is predicted by monitoring the average enthalpy of the superheated layer which is a measure of the closeness of the incipience of the boiling crisis. The energy balance can be written as:

$$\frac{\mathrm{d}(\rho_{\mathrm{sl}} U_{\mathrm{sl}} P_{\mathrm{h}} y^* i_{\mathrm{sl}})}{\mathrm{d}z} + h P_{\mathrm{h}}(T_{\mathrm{sl}} - T_{\mathrm{b}}) = q P_{\mathrm{h}} \tag{16}$$

In order to evaluate Eq. (16) at the point where the critical heat flux first appears, Tong et al. made some simplifications:

- constant bubble population inside the superheated layer just before the critical boiling occurs,
- an average enthalpy, i_{sl}, is considered inside the superheated layer, i.e., the average conditions of the flow across it are considered in a state of superheating,
- the thickness y^* and the superheated layer velocity, U_{sl}, are constant,
- the average temperature, T_{sl}, inside the superheated layer is constant and uniform, and
- the heat transfer coefficient, h, between the superheated layer and liquid core remains constant,

applying these assumptions, Eq. (16) is simplified to

$$\frac{\mathrm{d}(i_{\mathrm{sl}} - i_{\mathrm{b}})}{\mathrm{d}z} + \Omega(i_{\mathrm{sl}} - i_{\mathrm{b}}) = \Omega \frac{C_{\mathrm{p\,sl}} q}{h} \tag{17}$$

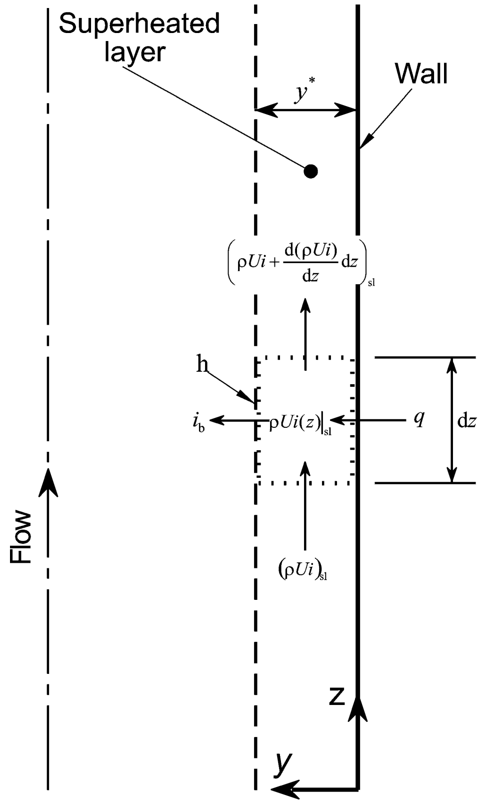

Fig. 5. Energy balance inside the superheated layer [6].

where Ω is defined as:

$$\Omega = \frac{h}{\rho_{\mathrm{sl}} U y^* C_{\mathrm{p\,sl}}} \tag{18}$$

According to the simplifications made, Ω is a constant independent of the axial position, and it is calculated empirically from the correlation [15]:

$$\Omega = \frac{185.6[1 - x(z_{\mathrm{crit}})]}{G^{10.478}} \tag{19}$$

In the case of uniform heat flux, i.e., for $q = constant$ and for the initial condition $i_{\mathrm{sl}}(0) - i_{\mathrm{b}} = 0$ at the tube inlet, the solution of Eq. (17) is

$$i_{\mathrm{sl}}(z) - i_{\mathrm{b}} = q\left(\frac{C_{\mathrm{p\,sl}}}{h}\right)\left[1 - \mathrm{e}^{-\Omega z}\right] \tag{20}$$

and for the case of non-uniform heat flux, i.e., $q = q(z)$, the solution is given by:

$$i_{\mathrm{sl}}(z) - i_{\mathrm{b}} = \Omega\left(\frac{C_{\mathrm{p}}}{h}\right)\int_0^z q(z)\mathrm{e}^{-\Omega(z-z')}\,\mathrm{d}z' \tag{21}$$

where z' is the position where the non-uniform heat flux profile is evaluated.

As it previously mentioned, the critical boiling is determined by a critical value of superheated layer enthalpy, therefore,

$$\left[i_{\mathrm{sl}}(z) - i_{\mathrm{b}}\right]_{\mathrm{crit,U}} = \left[i_{\mathrm{sl}}(z) - i_{\mathrm{b}}\right]_{\mathrm{crit,NU}} \tag{22}$$

where, the uniform critical heat flux equivalent for a system under an axially non-uniform heat flux is obtained as:

$$q_{\mathrm{CHF,U}} = \frac{\Omega}{1 - \mathrm{e}^{-\Omega z_{\mathrm{crit}}}} \int_0^{z_{\mathrm{crit}}} q(z)\mathrm{e}^{-\Omega(z_{\mathrm{crit}}-z)}\,\mathrm{d}z \qquad (23)$$

where z_{crit}, is the axial length measured from the tube inlet to the position where the CHF first appears. Nevertheless, z_{crit}, could be different for the cases where $q = constant$ or $q = q(z)$. Therefore, a correction factor is defined as

$$F = \frac{q_{\mathrm{CHF,U}}}{q_{\mathrm{CHF,NU}}} \qquad (24)$$

explicitly,

$$F = \frac{\Omega}{1 - \mathrm{e}^{-\Omega z_{\mathrm{crit}}}} \int_0^{z_{\mathrm{crit}}} \frac{q(z)}{q(z_{\mathrm{crit}})}\mathrm{e}^{-\Omega(z_{\mathrm{crit}}-z)}\,\mathrm{d}z \qquad (25)$$

where, F is called *shape factor* (F-factor). On this basis, the sublayer dryout theory can be extended by adding the F-factor to the CHF expression,

$$q_{\mathrm{CHF,NU}} = \left(\frac{1}{F}\right)\frac{\rho_{\mathrm{f}}\delta h_{\mathrm{fg}}}{t} \qquad (26)$$

4. Case study

According to Refs. [8,10] a steam power plant in Mexico recently reported troubles in the furnace of a steam generator. The problem was caused by failures in the furnace tubes; corrosion, burnout and tube leaks were detected on many of them. These failures were located in tubes at the level of the middle furnace, this is about 12 m from the furnace bottom. The metallurgy analysis of the damaged tubes showed that they were operating at very high temperatures, above 400 °C which is the permissible value for the material of construction. A partial solution, based on pressure variations was proposed with the aim to diminish the saturation temperature and increase the gradient temperature between the wall and internal flow of the tubes. It was assumed that heat absorbed by the flow could be magnified so the high temperatures will be avoided. However, in addition to the high operating costs that the variable pressure technique originated, the failures in the furnace tubes continued to appear, although, not as frequently as before.

Considering this, in the first place is required to determine the real causes of the tube failures, i.e., it is necessary to know whether the high tube wall temperature is the consequence of the CHF or it is caused by the heat absorption rate in the furnace, which does not reach critical values yet. One this is done it will be possible to make the correct changes in furnace operation in order to control the causes of the problem. In addition, any change made for the furnace conditions could be tested with the mathematical model before they are put into practice.

Table 1

Operating conditions of the furnace	
General conditions:	
Unit capacity	158 MW
Water circulation system	natural
Tube material	A-210-A1
Permissible temperature	400 °C
Geometrical characteristics:	
Inner tube diameter	53.5 mm
External tube diameter	63.5 mm
Average tube length	22 m
Thermo-hydraulic conditions:	
Pressure	14.2 MPa
Inlet flow temperature	333 °C
Average mass flux per tube	950 $\mathrm{kg \cdot m^{-2} \cdot s^{-1}}$

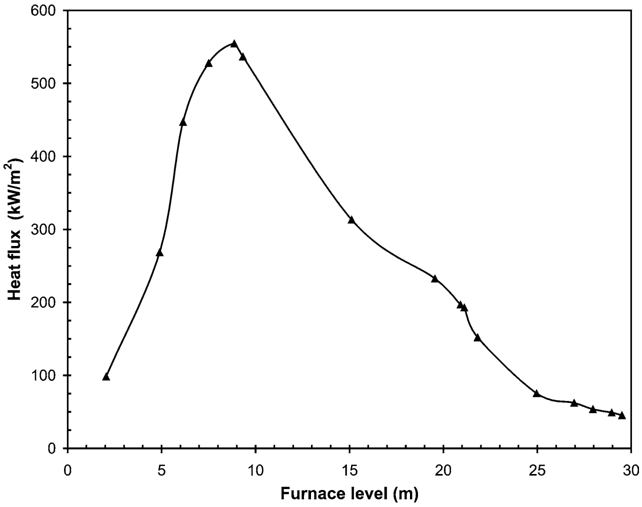

Fig. 6. Heat absorption rate for design conditions.

4.1. Furnace operating conditions

The steam generator analyzed is in operation from March 1996. The specific information about design operating conditions in the furnace tubes, are taken from Ref. [10] and are reproduce in Table 1.

Fig. 6, taken from the same reference above, shows the heat absorption rate per unit of area, corresponding to design conditions. In Fig. 7 the temperature distribution along of the tube wall, for design conditions is presented. The steam quality, in turn, is shown in Fig. 8.

This information will be useful to solve the temperature distribution model and for comparing the results with the design information. Specifically, the thermo-hydraulic conditions listed in Table 1 and the heat absorption rate shown in Fig. 6 are required for solving the CHF prediction model.

5. Results and discussion

The furnace tube configuration inside the steam generator could vary from one commercial model to another. For convenience and in agreement with the reference conditions

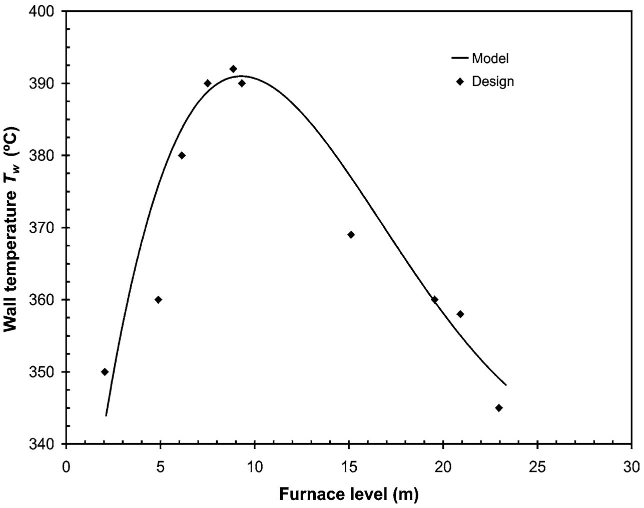

Fig. 7. Tube wall temperature of the furnace tubes.

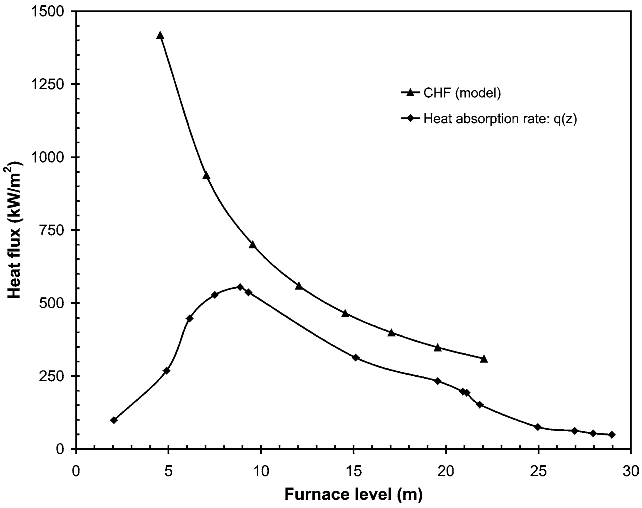

Fig. 9. Predicted CHF for the furnace tubes.

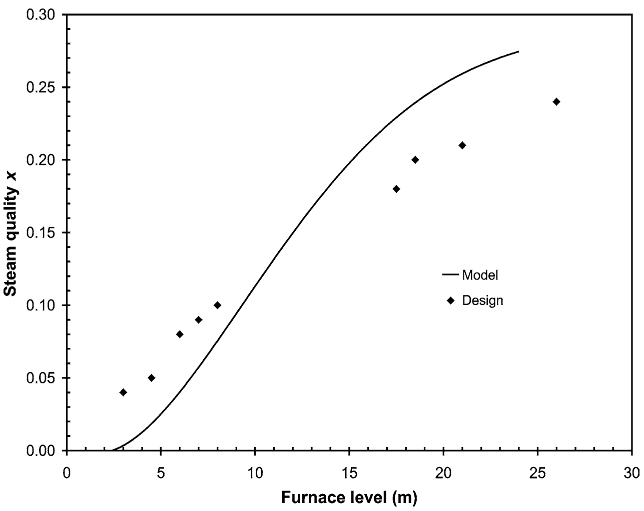

Fig. 8. Steam quality along the furnace tubes.

established for the proposed models, straight tubes vertically oriented inside the furnace are assumed.

The result for the temperature distribution is shown in Fig. 7, where it is compared versus design temperature distribution. The difference between the theoretical and design data is about 4.39% at corresponding points, which is a very acceptable value considering that, for this kind of studies, differences around 20% are commonly accepted [9]. Although, these results are useful just to verify the model reliability, if they are compared with real temperature measurements, under the same thermo-hydraulic conditions, a general idea of furnace heat transfer behavior could be obtained.

Another related result is the steam quality under design conditions, and this is shown in Fig. 8, where the main differences are observed in the zone close to the tube outlet. The theoretical steam quality is above the design steam data by 19%. This difference are around the acceptable error margin and could be attributed to some simplifications that were made in the analysis.

The CHF results are presented in Fig. 9, and they are compared against the heat absorption rate. According to the predicted CHF made by the proposed model, the minimal difference between the CHF and the heat absorption rate curves is 10.5%, which is a minimal margin of safety. In order to validate the CHF results the critical wall temperature was calculated from an energy balance and compared versus corresponding temperature measurements reported in Ref. [10]. According to this reference, the thermo-hydraulic conditions in the steam generator during the measurements were as close as possible to those listed in Table 1. This fact enables the comparison between results and measuremntes for the validation of the model.

In Fig. 10 the measurements made in the wall of the furnace tubes are presented together with the CWT and design temperature curves. Because of the difficulty to make this kind of measurements inside the furnace, only three temperature values were obtained from Ref. [10]. The position of the measurements were in the zone of frequent failures, between 10 and 15 m from the furnace bottom. It is observed that the temperature measured surpasses the permisible value for the material of the tubes (400 °C) at the levels of 12.5 and 15.5 m. In addition, these critical points are detected by the CWT curve with a difference of 5.7 and 1.4%, respectively. This fact confirms that the high temperatures are consequence of the CHF occurrence.

The predicted CHF shows that during the real operation of the steam generator, the axial heat flux profile must be different from that expected for design operating conditions. The peak of thermal load shown in Fig. 6 at a level of 9 m, was displaced 6 m, since in Fig. 10, the highest temperature measurement is at 15.4 m from the bottom of the furnace. At this point, it is assumed that the thermal load peak appears. It also shows, that the burners flame deviates upwards to the furnace zone where the tubes are failing. Considering this, different actions to improve on the furnace operation could

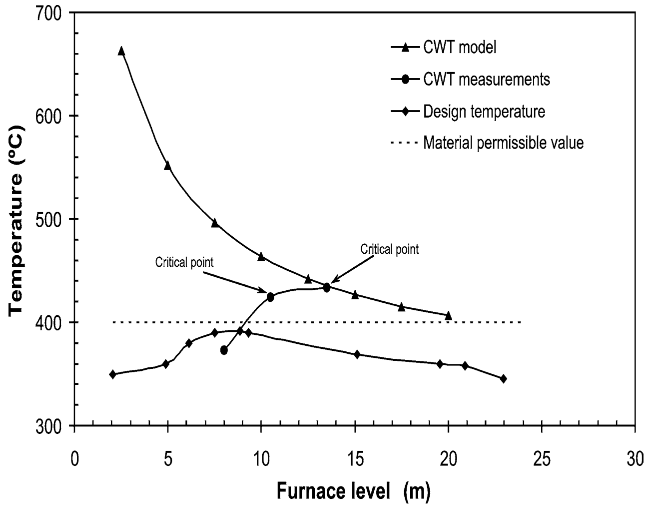

Fig. 10. Critical wall temperature.

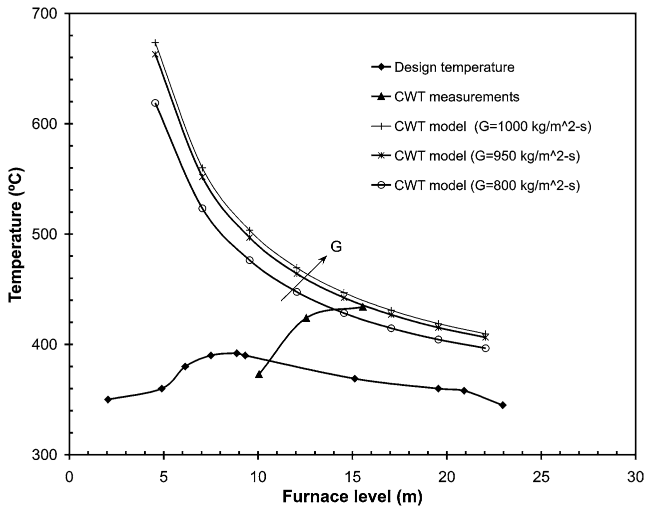

Fig. 11. CHF-mass flux dependence.

be taken. Evidently, the burners must be realigned to avoid the flame deviation, but, according to the parametric trends of the CHF showed in Ref. [16], a quicker solution could be to manipulate the thermo-hydraulic parameters. For instance, if the mass flux is increased, the CHF increases too, therefore the heat absorption rate in the furnace stays lower than the critical values; this is presented in Fig. 11. However, it will be necessary to determine the temperature distribution in the tube wall to ensure that a mechanism of corrosion has not been activated and that the heat flux is still under the CHF.

6. Conclusions

A mathematical model for the prediction of the CHF under real operating conditions inside a steam generator has been proposed on the basis of the sublayer dryout theory. The original sublayer dryout theory was modified, by combining it with the shape factor method, to take account for the effects of an axially non-uniform heat flux distribution on a vertical tube, which is a typical case inside the furnace of a steam generator. The concept of the sublayer dryout theory limits the application of the developed model to those cases where is expected that the CHF occurs at low qualities. It means that a bubbly boiling regime must prevails to apply the proposed model.

Furthermore, a mathematical model to determine the temperature distribution along a vertical tube with internal water boiling has been presented. Particularly, this model can be applied to predict the temperature distribution along all boiling regimes since a set of equations are proposed for every one of them.

A case study based on failures reported by a mexican steam power plant was solved by using both models. For this aim was assumed a straight vertical tube operating under the average conditions in the furnace of the steam generator specified by the designer. The temperature distribution along the tube wall was calculated and compared against designer data and a good approximation was found. The CHF was predicted for the same conditions, and the curve of critical wall temperature for the tubes was determined from this results. The critical temperature values were compared against measurements made in plant under conditions similar to those specified by the designer and the predicted critical temperatures coincide with the measurements.

The good approximation between the critical temperature points obtained using the CHF model and the measured data confirms the reliability of the modified sublayer dryout theory. Considering the parametric trends of the CHF reported in the literature, a solution for the CHF problem based on the manipulation of the thermo-hydraulic parameters is proposed and its preliminary results are also shown.

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